



School of Engineering

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Application of a
Vane-Recessed Tubular-Passage
Casing Treatment
to a Multistage Axial-Flow Compressor

Ph.D. Thesis

November 2001

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Supervisor: Professor Robin Lee Elder

November 2001

Dedications

In the Name of Allah, Most Gracious, Most Merciful

The day of union of friends remember!
Those times, remember, remember!
Although free of recollection of me, are friends
Them, on my part a thousand times, remember!

I dedicate the following dissertation to the pious souls of those, who are not with me, but their memory will never depart me. This collection of research is presented to the souls of the beloved ones who have lost their lives.

Thirty-three years ago, when I was only 10 years old, my father, Abolghasem Akhlaghi, left eternally and separated from his first child in great sadness and pain.

I would also not forget the period of my childhood which was spent beside the blessed Ahmad Akhlaghi, my grandfather, who although departed from this world twenty-six years ago. The sweet memories of his companionship and the advice given by him are always fresh and would never become old memories for me.

Fatemeh Maddahzadeh, my beloved mother, who had faced the difficulties of this life, left me in this world when I had hardly spent two months working on my PhD in England. I could not see her before she closed her eyes and left for her eternal journey.

Also, the late Ali Havakeshian (BSc), my best and kind friend. Him I had named as brother and was my teacher during my adolescence. He who parted to eternity just months before completion of this thesis in all disbelieves.

However, the passage of times has taught me that when I lose the beloved ones I should face it with forbearance and I know that one-day I also will be following their footsteps.

It is my wish that I too leave a remarkable thing before my departure for I don't see this place as eternal.

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ABSTRACT: The current study investigates a range of issues relating to the use of a vane-recessed tubular-passage casing treatment as a passive stall control technique in a multistage axial-flow compressor. The focus of the research was to determine whether such a treatment could delay the initiation of stall at lower mass flow rates as well as providing the most beneficial improvement in flow characteristics without sacrificing compressor efficiencies. Specific objectives of this study were to examine possible improvements or deterioration in the flow characteristics including stall margin, peak pressure rise coefficients and maximum efficiency in a multistage axial flow compressor.

A casing treatment in addition to several spacer rings was developed from two initial designs and tested on the first stage of a low speed three-stage axial-flow compressor with a (0.7) hub to tip diameter ratio. The treatment configuration consisted of three parts: an outer casing ring, with a tubular shaped passage on the inside diameter, a set of 120 evenly spaced curved vanes, and a shroud or inner ring. The casing treatment was positioned following the inlet guide vanes upstream and partly covering the tip of the rotor blades. The main parts of the casing treatment including the recessed vanes in addition to some of the spacer rings were manufactured from high quality acrylic.

Eight additional spacer rings of various shapes and geometry were added. The first ring held and partly covered the IGVs, in front of the casing treatment. The rotor tip exposure ratio was thought to have a significant impact on the effectiveness of the casing treatment. Therefore the other seven rings were used to provide the desired uncovered region of the rotor tip axial chord of about 10% in order to provide a range of exposures of (23.2%, 33.3%, 43.4%, 53.5%, 63.6%, 73.7%, and 83.8%).

The results showed significant improvements in stall margin in all treated casing configurations along with insignificant efficiency sacrifices in some compressor builds. About (28.56%) of stall margin improvement in terms of corrected mass flow rate was achieved using a casing treatment with a (33.3%) rotor tip exposure.

The compressor build with (0.535) rotor exposure ratios was the best configuration in terms of efficiency gain and loss characteristics. This build was able to provide the highest values of the maximum efficiencies in comparison with the performance achieved from the solid casing. An improvement of (1.81%) in the maximum efficiency in terms of the overall total-total pressure ratio, in association with a (22.54%) stall margin improvement in terms of the corrected mass flow rates were achieved by the application of this treatment configuration. The improvement in the peak pressure rise coefficients in terms of the overall total-total pressure ratio, obtained from this build was (2.33%).

The compressor configuration using a casing treatment with a (0.636) rotor exposure ratio was the best build in terms of the pressure rise coefficients. This configuration was able to provide highest value of the peak pressure rise in comparison with the characteristics achieved from the datum build. An improvement of (2.65%) in the peak pressure rise coefficient in terms of the overall total-total pressure ratio, in association with a (22.49%) improvement in stall margin in terms of the corrected mass flow rates was achieved from this casing treatment build. The improvement in maximum efficiency in terms of the overall total-total pressure ratio, obtained from this build was (1.03%).

The results suggest that the vane-recessed tubular-passage casing treatment designed as part of this investigation achieved the objectives, which were established for the research. In the majority of instances it not only produced gains in flow range, pressure rise coefficients and efficiencies, but also enabled the rotating stall, which developed at much lower mass flow rates in the compressor, to become progressive rather than abrupt.

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Nomenclatures

A_C	Compressor flow through area	p_{VFM}	Static pressure at venturi
A_{VFM}	Area cross section of venturi	PR	Pressure ratio
B	B factor	Re	Reynolds number
C	Blade chord	S/C	Space to chord ratio
C_a	Blade axial chord	T	Temperature
C_D	Discharge coefficient	TCF	Torque calibration factor
C_P	Specific heat at constant pressure	T/C	Thickness to chord ratio
C_S	Speed of the sound	T_{Amb}	Temperature of the ambient
C_V	Specific heat at constant volume	t_C	Tip clearance
D_{Ann}	Annulus passage diameter	t_{Vane}	Thickness of the recessed vane
D_{Com}	Diameter of compressor	T_{VFM}	Temperature of flow in the venturi
D_{Hub}	Hub diameter	U	Circumferential velocity
D_{IR}	Diameter of inner ring (shroud)	V_a	Axial velocity
D_{Mean}	Mean (reference) diameter	V_P	Plenum volume
D_{Tip}	Tip diameter	V_{Tan}	Tangential velocity
D_{TC}	Diameter of casing treatment	Z	Number of blades or guide vanes
D_{VFM}	Diameter of venturi flow meter	Z_{Vane}	Number of the recessed vane
DR	Diameter ratio	α	Absolute velocity angle
e_{IR}	Shroud (inner ring) thickness	β	Relative velocity angle
g	Acceleration due to the gravity	η	Efficiency
h	Blade height	ΔH	Total energy transfer to airflow
h/C	Aspect ratio	ΔP	Pressure difference
h_{Vane}	Height of the recessed guide vane	ΔT_s	Isentropic temperature rise
L_{Arm}	Torque Arm	$\Delta \Phi$	Difference in mass flow rates
L_C	Effective length of the duct	ε	Expansion coefficient
L_{TC}	Length of the casing treatment	χ	The specific heats ratio
L_{Vane}	Length of the recessed guide vane	μ	Dynamic viscosity of air
Mach	Mach number	ψ	Pressure rise coefficient
N	Running speed	φ	Flow coefficient
P	Total (stagnation) pressure	Φ	Mass flow rate
p	Static pressure	$\Phi \varepsilon$	Corrected Mass flow rate
P_{Amb}	Ambient pressure		

ω	Angular velocity of rotor blades	τ_{Net}	Torque (transducer output)
Ω	Power	ρ	Density of flow
$\Omega\epsilon$	Corrected applied power	θ	Camber angle
τ	Torque	ζ	Stagger angle
$(C_R)_{\text{Axial-Tip}}$	Rotor blade axial chord at the tip diameter		
$(C_R)_{\text{AT/TC}}$	Rotor axial chord at tip diameter exposed to treatment recessed chamber		
$(DR)_{\text{Hub/Tip}}$	Hub to tip diameter ratio		
$(d_{\text{TC}})_{\text{Axial}}$	Axial distance between the guide vane's leading edge and end of shroud		
$(D_{\text{Vane}})_{\text{Mid-Height}}$	Diameter of recessed guide vanes corresponding to the vane mid height		
$(G_{\text{TC}})_{\text{Axial}}$	Axial length of the gap between the shroud and rotor blade leading edge		
$(L_{\text{TC}})_{\text{Axial}}$	Axial length of the vane-recessed treated casing		
$(L_{\text{TC}})_{\text{Inlet}}$	Axial length of the casing treatment inlet passage		
$(L_{\text{TC}})_{\text{Shroud}}$	Axial length of the inner ring (shroud)		
$(L_{\text{TC}})_{\text{Outlet}}$	Axial length of the outlet passage of the treated casing		
$(L_{\text{Vane}})_{\text{Axial}}$	Axial length of the vanes inside the recessed chamber of casing treatment		
$R_{\text{Universal}}$	The universal gas constant		
$(S_{\text{Vane}})_{\text{Mid-Height}}$	Space between two adjacent vanes corresponding to the vane mid height		
$(S_{\text{Vane}})/(L_{\text{Vane}})$	Space to chord ratio of the recessed guide vane		
$(t_{\text{TC}})_{\text{Shroud}}$	Thickness of the shroud (inner ring) of the casing treatment		
$W_{\text{Effective}}$	Effective load required for balancing the stator of the electrical motor		
λ_{RTAC}	Rotor blade tip axial chord exposure ratio		
ΔH	Enthalpy rise across the compressor		
$(\Delta T_s)_{\text{Overall}}$	Overall isentropic temperature rise		
ΔV_w	Change of the whirl velocity across the rotor blades		
$(\Delta \eta)_{\text{ME}}$	Improvements or losses in terms of the maximum efficiencies		
$(\Delta \Phi \epsilon)_{\text{SP/(CMFR)}}$	Improvements in stall margin in terms of the corrected mass flow rates at near the stall conditions		
$(\Delta \psi)_{\text{PPRC}}$	Improvements or losses in terms of the peak pressure rise coefficients		
$\Delta[(\Phi \epsilon)_{\text{SP}} * \psi_{\text{SP}}]$	Stall margin improvement in terms of the pressure rise coefficients and the corrected mass flow rates both at close to the stall points		

CHAPTER ONE

1)- General Introduction, Objectives, Backgrounds and Structure of the Current Investigation:

1.1)- General Introduction:

In modern, technical societies the aero-engine compressor market is becoming increasingly competitive globally, and more sensitive to the need for high performance particularly high operational efficiency, broad operating range and also low manufacturing cost. The stable operating of an axial flow compressor is limited by the onset of non-linear aerodynamic instabilities, which leads to the fluctuation of the pressure and mass flow rate.

An operational constraint on the flow range of a compressor that often limits the pressure rise capability of the compressor is the occurrence of stall and/or surge. The occurrence of the surge or stall however, may cause plant shutdown. Their effects may also lead to a loss in production and possibly costly damage. It is important that the compressor can be operated in such laborious conditions free from the risk of stall or surge.

The present investigation has the objective of extending the stable operating range of machines and is applicable to both aero-engine and industrial multistage axial flow compressors.

1.2)- Aim of Project and the Current Work Objectives:

It is well known that the stable working range in a compressor is very restricted due to the appearance of two phenomena, which are called stall and surge. When one of them or both occur, the operation of compressor usually becomes unstable. The major problem in a multistage axial-flow compressor is how to delay, prevent, or suppress the onset of stall and surge. One way to tackle this problem is to move the stable operating lines further away from the instabilities. This of course provides a decrease in the stage loading.

On the other hand, casing treatments have the effect of increasing the level of stage loading at which the inception of stall occurs. By using these devices it is possible to increase the stall/surge margin of a compressor without moving the operating lines. In order to achieve this, a proper understanding of stall inception or surge onset in either low or high speed multistage axial-flow compressors is crucial.

Previous researchers have shown that, at lower speeds, pressure rises in a given stage may cause the density of flow to be lower than designed in the rear stages. As a result, the relative velocities are changed and a negative incidence angle occurs. This situation leads to the choking of these rear stages and the throughflow is limited. Subsequently, the flow in the front stages is reduced and these stages tend to stall.

Considerable efforts have been made in the past to understand firstly the stall phenomenon and secondly to apply different methods of delaying or suppressing stall by some means of control. A large number of conventional stall control techniques were used in the basic concept of restricting the operation of the compressors during stall time occurrence.

The emphasis in the current study is on the avoidance of these aerodynamic instabilities by allowing a sufficient margin between the operating region of compressor and its stability limit. It is acknowledged that in the majority of cases the different types of casing treatments which have been applied to axial-flow fans or compressors have been reasonably effective in improving the stall margin by displacing the rotating stall point to a lower mass flow region.

Of the stall onset delaying techniques that have been applied so far, it seems that an anti-stall passive technique known as, vane-recessed casing treatment could provide the desired potential stall margin improvement with the least sacrifice in efficiency. The potential benefits of this type of casing treatment to multistage axial-flow compressors could possibly be significant and the investigation of this hypothesis forms the basis of the current study.

Additionally, there are a number of applications, which would benefit from the installation of a casing treatment on a multistage axial flow compressor, especially industrial and jet engine compressors. In other words, the objective of present study is to investigate possible improvement or deterioration of the flow characteristics including stall margin increase and peak pressure rise as well as maximum efficiency losses in a multistage axial flow compressor. The ability of a stepped passive casing treatment, which is fitted to some part of the compressor's conventional casing, can do this.

In order to achieve the specified objective of the current study it is necessary to obtain an optimum casing treatment configuration, by means of an experimental investigation of the performance of a three-stage low-speed axial-flow research compressor. To obtain this, the aerodynamic behaviour of the flow needs to be modified by changing the shape of the conventional casing outer annulus wall. This latter procedure is referred to as a vane-recessed casing treatment application.

The vane-recessed casing treatment in fact involves some modification to the inner surface (partly above the compressor moving blades and partly upstream of the rotor blade cascade) of the conventional outer casing. In this process, the compressor geometry and performance characteristic limitations imposed by the compressor have to be considered quite carefully.

It is obvious that other restrictions such as the design considerations, existing geometry features, in addition to the manufacturing cost, also need to be taken into account. It is worth noting that in order to overcome or prevent rotating stall due to the reduction of mass flow rate inside the compressor, an attempt has to be made to obtain the followings.

- 1)- To address the problem of rotating stall inception in a low speed three-stage axial-flow research compressor of (0.7) hub to tip diameter ratio test rig. The experiments were carried out, using an existing three-stage axial-flow compressor with 406.4 [mm] annulus outlet diameter that is hereafter called (Peregrine Compressor)¹.
- 2)- The application of a vane-recessed tubular chamber casing treatment in a compressor test rig, without major losses occurrence in flow characteristics, from the datum build performance.

¹ This compressor test rig has been designed to a specification of Professor A. G. Smith by Peregrine and partner consulting engineers, Ryosan, and manufactured by David Bedworth Ltd, Harwich. The Peregrine research compressor has also many features in common with high-speed machines as well as convenient ways of varying the normally fixed angles and spacings. The performance of this three-stage compressor was also considered in more detail in the chapter four of the current dissertation.

1.3)- Background of Vane-Recessed Casing Treatment Applications:

An increasingly common demand for improved compressor performance focuses on the need for a wider stall margin range. The anti-stall design prevents stalling, extends the range of stable operation, and eliminates previous restrictions in the selection of axial compressors for systems with dynamic and varying duties.

It is quite easy to realise from the majority of papers which have been published on the subjects of stall and/or surge phenomenon in compressors or fans so far that the designers always preferred to use a stall control technique in order to restrict the operation of machines.

These of course have concentrated on using conventional stall control techniques and running the compressors far from the stall limitation points through a predetermined safety margin. It is obvious that this interrupts and limits the compressor's operating range at the maximum efficiency, which based on the different designed types of the compressors may lie at or quite close to the stall line.

Over the past nearly forty years, the compressor designer has tried to use alternative approaches such as active or passive control devices to delay stall initiation in order to allow the compressor to operate at normally stalled flow rates.

An active stall control system normally works through the introduction of further dynamic activities into the compression process of the compressor in order to dissipate any tendency for instability to develop. The applications of fast acting bleed valves or variable guide vanes also could consider among stall phenomenon active control techniques, since they introduce some additional dynamic activities to delay or suppress stall of the compressors.

The passive stall control system normally involves some modification of the design features of the compressor. Based on the type of stall occurrence, the modified parts can be used either in the top of the rotor blades or beneath the fixed blading. It was found that passive stall control devices normally provide continuously extended flow in the compressor which finally increases the machine pressure rise characteristic that peaks at a much lower mass flow rate associated with a comparatively small sacrificed efficiency.

The passive anti-stall devices such as vane-recessed casing treatment have further advantages and fewer drawbacks in comparison to an active stall control system. It seems that it can reduce the operational and maintenance costs by considerable amounts owing to the reduction of the cumulative fatigue damage than compressor blades normally suffer.

The current research suggests that in the majority of cases, after using vane-recessed casing treatment on the test rig (Peregrine Compressor), the stall status changed from the (abrupt stall) to the (progressive stall). This will be considered in detail in chapter eight.

Although an additional manufacturing cost may be involved, it seems that the extra costs can be recovered by the potential gains in compressor performance. It is quite obvious that the manufacturing cost of many passive anti-stall devices is much less than active anti-stall systems such as bleed valves or variable guide vanes.

The applications of different types of casing treatment to compressors or fans in order to control or delay the stall phenomenon, started in the early 1960s. The groove or slot types passive stall treated casing techniques used by some of the previous investigators² with

² (Fujita et al., 1977), (Takata et al., 1977) and (Smith et al., 1984).

different compressors, since 1970's, showed a limited but valuable stall margin improvement along with significant pressure rise associated with considerable efficiency losses. In these casing treatments, about (10-20)% stall margin improvements were achieved, whereas the corresponding stage efficiency penalties were around (2-4)%.

It seems that the prototype vane-recessed casing treatment, which was undertaken on a two-stage axial flow ventilation fan, first invented by Ivanov et al. (1984 & 1989)³ was an alternative and more successful type. Compared to the previous work, it seems that this type of casing treatment has shown remarkable stall margin and peak pressure rise improvements in relation to industrial fans.

Prior to this work, the idea of applying a casing treatment to mixed flow blowers and suction fans using some vanes inside the treated casing had also been investigated by others (Russell et al., 1982). This resulted in a patent that has many similarities to that produced two years later by Ivanov et al.

Before the advent of passive stall control, in order to avoid the risk of stalling, usual practice was to select an axial flow fan with a working range restricted to a maximum of (90%) of the stall line pressure. This obviously restricted the selection of the optimum fan size and type, as the normal operating range often is far below the peak conditions selected with a certain safety margin below the stall line. However, it does not provide an absolute guarantee against stalling.

Bard (1984) used a unique ring-shaped stabilising device, which was incorporated into an industrial axial flow fan casing. The applied stabiliser did not contain any moveable elements and it was located outside the main flow stream and it did not interfere with flow in the normally stable range of operation. The applied stabilising anti-stall device on a 3.4 [m] rotor tip diameter, axial-flow fan, provided fully stable performance characteristics without stall line. Using the stabilisation ring eliminated the unstable performance of the fan due to stalling without affecting the peak efficiency.

The anti-stall devices casing treatment arrangements that were used by Ivanov, et al. and Bard have a lot of common features. However the prototype vane-recessed treated casing which used by Ivanov et al., was located partly upstream and partly over the tip rotor blades, whereas that one used by Bard, was placed just upstream of and not over the tips of the fan's rotor blades.

Further experimental work was also carried out by Miyake et al. (1985), who used an air separator as an alternative means of stabilising the characteristics in the low flow rate region of an axial flow fan. The mode of operation of the air separator anti-stall device is very similar to the vane-recessed treated casing. The air separator bleeds a part of the stalled flow near the rotor blade tip and returns it to the mainstream flow via the air separator passage.

The results also showed that for usual rotors of reasonable load with small aspect ratio blades, an air-separator was fairly effective in removing the stalled flow and delaying the stall initiation. It was found that the air separator treatment was useful, even in severe conditions in which the blade separator treated casing had no significant effect.

³ The same patent has published through (UK) and (US) patent offices on (1984) and (1989), respectively. Ivanov S. K., Dudkin V. E., Peredery V. P. & Molchanov V. N., (1984 & 1989). ("Axial-flow ventilation fan", United Kingdom Patent Application, GB2, 124, 303,A, February 1984) and ("Axial-flow fan", United State Patent Number: 4,871,294, October 1989).

The effect of an air-separator to improve the low flow rate performance of an axial flow fan in comparison with the effect of a blade separator was investigated in more detail through the experimental work conducted by Miyake et al. (1986). The flow fields around the rotor of the test fan showed that both the blade and air separators are effective in preventing rotating stall, but that the air separator was more effective in removing the fluid with low relative velocity.

It is also suggested that the passive anti-stall air separator is useful even for a rotor, which originally accompanies a heavy rotating stall, and it can also be applied to an arbitrary stage of a multistage compressor, (Miyake et al., 1987).

Azimian et al. (1987) investigated the effects of vane-recessed casing treatment on a single stage and an isolated rotor axial flow fan with a typical aeroengine blade loading. It should be noted that an aero-engine axial flow fan has a much higher blade loading in terms of work coefficient ($\Delta H/U^2$) in comparison with a more or less similar industrial fan.

In this research the test rig consisted of a compressor with a single stage low speed axial flow fan with a rotor tip speed of 39 [m/sec], a tip diameter of 508 [mm], hub to tip diameter ratio of 0.5, and blade loading [$\Delta H/U^2$] near unity at the hub. The rotor and stator were equipped with (28) and (27) blades, both with (C4) circular arc profiles, respectively. The rig was tested using casing treatment on the stage build. However the best results obtained with the complete stage were achieved with (35.5%) rotor blade tip axial chord exposure ratio.

Test results also showed that the treated builds with more than (50%) of the rotor blade axial chord exposed to the recessed casing were favourable. This result was achieved with an isolated rotor. It seems that owing to the premature onset of stator hub stall, the obtained improvements from complete stage builds were not as good as that found with the isolated rotor builds.

It is worth noting that the amount of rotor blade tip axial chord exposure ratio is at odds with the conclusions of Bard, whose optimum exposure ratio was considerably less. It is worth noting that no data was provided for the anti-stall unique stabilisation ring casing treatment and the operating speed of the corresponding applied fan. However from the published drawing it appears that the rear wall of the treated casing chamber had been extended a little forward of the rotor blade leading edge. Although it has not been confirmed yet, it is possible to suggest that the compressor blade loading may have some influence on the effects of the rotor tip axial chord exposed to the recessed chamber.

It should be noted that one of the ways to provide a more energy efficient compressor is to increase the blade loading and obtain a high pressure rise per unit length, and this enables a reduction in the overall size and weight of the unit. Increased blade loading in the compressor or fan induces a higher axial pressure gradient across the unit. This adverse pressure gradient then has an impact on the basic loss mechanism.

It was found that the best stall margin improvement amount was achieved with an optimum build which consisted of the testing fan with an isolated rotor, with (67%) rotor blade tip axial chord exposure ratio, when using an "aerofoil section". An aerofoil section was used in order to reduce the area from the rear portion to forward of the vanes inside the recess chamber of the casing treatment. For this optimum arrangement, about (60.8%) stall margin improvement for flow rate without incurring a significant penalty in efficiency had been noted compared to solid casing.

The obtained improvement in terms of the flow range for the isolated rotor build was slightly better than those of groove or slot type treated casings, which have been used in

previous investigations. It is worth noting that with the test build of the isolated rotor compressor, about (60.5) and (56.7%) stall margin improvement for flow rate has been achieved for (7.5%) and (-4.2%) tip axial chord exposure ratio, respectively. The tests with zero axial chord exposure ratios were not considered neither for isolated rotor nor for the stage builds, (Azimian et al., 1990).

The application of vane-recessed casing treatment with three different (0.5, 0.7 & 0.9), hub to tip diameter ratios were conducted by Ziabasharhagh et al. (1992), on the same test rig fan as used by Azimian et al., together with another rig that had the same annulus diameter. The former rig was also used in order to investigate the effect of the flow distortion on the casing treatment. The results showed that with the inlet flow distortion, an improved stall margin was achieved.

The running speed of both rigs was 1500 [rpm], but the hub to tip diameter ratio could be adjusted on the second rig for values of (0.7 & 0.9). The higher hub to tip diameter ratios investigations were undertaken on a single stage low speed axial flow fan with 39.9 [m/sec] and tip diameter of 508 [mm], where the same casing treatment could be tested with either hub configuration.

It is worth noting that the applied vane-recessed casing treatments related to the (0.7 & 0.9), hub to tip ratios had different geometry and number of vanes. But the casing treatment with (0.5) hub to tip diameter ratio was the same treated casing, which had already been used by previous investigators. Three different casing treatments with the same vane-recessed profile and general configuration, but different dimensions, had been applied. Those casing treatments were classified as large, medium and small.

The (0.5) hub to tip diameter ratio design was based on the Ivanov et al. patents, whereas the dimensions of the others two treated casings were scaled in proportion to the rotor blade heights of the (0.7 & 0.9) hub to tip diameter ratio stages. It means that the heights of the two latter casing treatments were scaled to (60% & 20%) of the casing treatment developed originally by Azimian et al.

The optimum build results from the three treatments showed that for (0.7) hub to tip diameter ratio configuration, the large size treatment provided the largest stall margin gains and the smallest efficiency penalty. For the (0.9) hub to tip diameter ratio, the best results were obtained with the medium size treatment, (Ziabasharhagh et al., 1992).

One conclusion that can be drawn from both previous research studies is that the rotor blade axial chord exposure ratio is an important determinant of the efficacy of the casing treatment in relation to compressor performance. A second conclusion based on the later study is that the size of the recessed chamber also has a vital role on the effectiveness of the treated casing.

Bard (1993) used an anti-stall device in axial-flow industrial fans for ventilation in tunnels. As it was reported, the unstable performance of the fan due to the stalling was completely eliminated due to the introduction of the unique stabilisation ring, which was integrated into the fan casing close to the impeller. He also explained that, since the stable working range of the traditional axial flow fan is restricted by stall line, when the system resistance is higher than the stall line, the duty point becomes unstable and simultaneously a shortfall in fan performance results.

The application of both single and double anti-stall devices provided significant improvement on the unstable performance of tunnel fans, because stalling was completely removed by the introduction of the unique stabilisation rings, (Bard, 1993).

It seems that the mechanism of the “stabilisation ring anti-stall” is very similar to that of vane-recessed casing treatment. The radial flow components, which become more significant as the flow is throttled, are removed into the anti-stall stabilisation ring. The ring eliminates, by means of turning vanes the swirling element and returns the flow into the main flow in a completely axial sense, thereby creating a stable, continuously rising fan performance characteristic. It is also worth noting that the performance of fan in the working range below the stall line is not affected since the entire anti-stall ring is located outside the main air flow, and functions on when the fan is throttled into the previously unstable range.

Hall et al. (1994) have applied a time dependent treatment/rotor aerodynamic interaction analysis with a (3-D Navier-Stokes Solver) to a modern fan rotor with a vane-recessed casing treatment. The results were then correlated with the corresponding steady state analytical data. A time dependent solution of the aerodynamic interaction between the test fan rotor and an embedded vane casing treatment with (1:2) rotor blade to treatment vane airfoil ratio was performed at the untreated rotor near stall condition.

It was reported that the time averaged results from the unsteady analysis showed favourable agreement with steady state results, using end-wall treatment time-average, boundary condition. The time-dependent analysis indicated a time-dependent periodic vortical flow structure near the rotor tip. However, this was a rather extreme geometric departure from the original design, (Hall et al., 1994).

Roy et al. (1994) applied a particular vane-recessed type treated casing in a low speed twin-rotor contra-rotating axial flow fan unit with low aspect ratio blades. The treatment was located above the two rotor sets. The injection portion and re-inject area of the casing treatment covered part of the second and first fan rotors, respectively. The casing treatment which worked on a principle of automatic suction and injection was intended to remove the casing boundary layer from near the second fan tip and inject the flow back near the first fan tip.

It was also reported that the flow circulation through the vane was about (13% to 18%) of the main mass flow. It was felt that this greater flow circulation is essential for efficient boundary layer control for a contra-rotating fan, where strong axial adverse pressure gradient is imposed across the rotor on the casing boundary layer.

It seems that the vane-recessed treatment applied to the twin-rotor fan provided a reasonable increment in the stall margin levels of the test contra-rotating axial flow fan unit, (Roy et al., 1994). It should be noted that the advantage of such an arrangement is to return the un-stalled flow which has already been removed, at stall conditions from near the tip of the second fan rotor leading edge into the main flow near the trailing edge of the first rotor fan.

Kang et al. (1995) carried out the application of vane-recessed treated casing by using two slightly different size and shroud shape casing treatments in addition to that one which already used by Azimian and Ziabasharhagh, respectively. He also conducted his investigations on the same test rig, which had been used by the lately noted previous researchers in Cranfield University, but just on an isolated rotor.

All three of the slightly different treated casings that were used had just (24) vanes inside the recessed chambers. The applied casing treatments have not had significant effects on the radial pressure profile, whereas the tangential velocity had been changed notably. Due to the examination of the three different geometrical designs of vane-recessed casing treatment on isolated test fan rotor blades more than (65%) of stall margin improvement

and over twice its pressure rise at stall were obtained.

The best configuration resulted in a stall margin improvement of (67%) associated with (120%) pressure rise in stall region and insignificant change in peak efficiency. With using the treated casing, the work inputs also were increased which as a matter of fact this led to higher pressure rises. It was found that the applied treatments have not significantly altered the efficiency curve, neither the peak efficiency and nor the efficiency in the extended unstalled range, (Kang et al., 1995).

Based on the obtained analysis of vane-recessed casing treatment and its effectiveness, it appeared that an isolated rotor blade compressor with higher blade loading angles showed rather abrupt characteristics. From this, it was suggested that the generally higher blade loading of aero-engine fans, in comparison to industrial compressors, might result in a more uneven flow characteristic, where vane-recessed casing treatment is applied.

However the present study will show that the effect of vane-recessed casing treatment on low blade loading compressors reveals similar problems. Therefore it is impossible to conclude at this stage, that it is simply the blade loading which is the cause of the uneven flow characteristics.

Hobbs (1995) applied an active vaned passage casing treatment to an advanced ducted rotor only fan driven by an aircraft gas turbine engine in order to afford a large stall margin. It has been acknowledged that as higher stall margin is built into the treated casing the higher the likelihood that the efficiency of the engine will be adversely affected.

It was reported that the control system was particularly able to solve the problem of enhancing the stall line so as to avoid the manifestation of an incipient stall while assuring that the engine does not incur an efficiency deficit at its normal operating condition. However, it was explained that under certain conditions it has been found that the stall margin has an adverse effect on the performance of the engine.

The problem addressed by the control system was to increase the stall margin in certain modes of operation and decrease the stall margin in other modes of operation. According to the later mentioned invention, the case passage duct or the vaned passage casing treatment is rendered inoperative or partially so, during certain regimes of the aircraft engine operation, (Hobbs, 1995).

The author suggest that under certain conditions the vaned passage casing treatment used in the control system, was operating as a passive stall control system. The term active denotes that the recirculating flow passage is at certain times flowing or partially flowing bleed or recirculating air and at other times it is blocked of or partially blocked off.

A simulation experiment was designed in order to verify the unsteady characteristics of the casing treatment with a recirculation area⁴. In this investigation a three dimensional compressible flow stability model which could be used to predict the stall inception condition of a single stage or multistage compressors had been applied by Sun (1996).

Nonetheless, based on the analysis of the unsteady phenomenon caused by the treated casing, the function of the casing treatments had been modelled by a wall impedance condition which was involved in the stability model through the eigenvalues and the corresponding eigenfunctions of the system. The results showed that the casing treatment was responsible for a remarkable change to the wall impedance condition.

⁴ The majority of the casing treatments, which have already mentioned above, due to their affect on the flow through the flow passages, could be considered as a recirculating area casing configuration.

Hill et al. (1996) conducted their study to investigate the influence of a vane-recessed passive stall control treatment on the performance of an industrial single rotor axial flow fan with a hub to tip diameter ratio of (0.4), and a tip speed of 76 [m/sec]. The treatment has been tested on a fan that was tip critical⁵.

The experiments had been carried out in a variety of different amounts of fan rotor blade tip axial chord exposure to the treatment. Two sets of blading were used, both equipped with (6) blades whose setting angle could be adjusted and one of which was of the fully reversible blade profile. The first set was a standard airfoil type blade with chord length of 104 [mm], and the second was a fully reversible blade profile with chord length 110 [mm].

Two types of blades, unidirectional and fully reversible in association with two vane-recessed types casing treatments, single and double (back to back) were examined with three values of blade setting angles (19° , 27° & 32°). Fans with both sets of blading showed reasonable improvements in flow range compared with the untreated build setting angles. With the exception of blade setting angle (27°), a small reduction in efficiency was noted in the normal operating range.

The results at the lower setting angle of (11°), showed a loss in efficiency for no beneficial flow range improvement, (Hill et al., 1997). The best improvements however, have been achieved in the region of (60%) rotor blade tip axial chord exposure ratio to the recess chamber of the casing treatment. The actual rotor blade tip axial chord exposures were dependent upon the blade setting angle and blade profile used.

As it had been reported, during the performed works, two different test rigs were used. The bulk of the study was carried out on an industrial fan. The rest of the investigation was conducted on the same fan, which had already been used by three previous researchers at Cranfield University⁶.

With a casing treatment applied, however, there were improvements in stall margin, up to (15%) for the unidirectional blade and about (21%) for the reversible blade dependent upon the amounts of rotor tip axial chord exposure to the treatments, (Hill et al., 1998).

The obtained results from the investigations, utilising vane-recessed casing treatment on the aero-engine compressors, have shown that this type of treated casings is capable of providing much greater benefits in stall margin. It also offers less performance losses than most other types of treatments reported in the literature, such as slotted type casing treatments.

It seems that an additional manufacturing cost and the geometry restrictions are among the problems, which normally restricts the application of vane-recessed casing treatment in aero-engines. The application of casing treatment of this type to axial flow compressors has also shown considerable improvements in compressor operating and noise reduction. These are in addition to the encouraging results, which have already been obtained in relation to flow characteristic improvements.

It is worth noting that the noise of the under test Peregrine Compressor has also been reduced due to the application of the designed vane-recessed treated casing. This will be discussed in more detail later in this work.

⁵ The rotating stall, occur first at the tip of the rotor blades and such machine are reasonably suitable for casing treatment application investigations.

⁶ (Azimian et al., 1987), (Ziabasharhagh et al., 1992) and (Kang et al., 1996).

1.4)- Structure of the Present Dissertation:

A general introductory description and the main objectives of the current investigation in addition to brief background activities corresponding to the some similar types of the casing treatments, were presented in this chapter.

A literature review of the local and global flow instabilities namely stall and surge phenomenon in addition to the different types of the stalls in axial flow compressors is presented in chapter two. Chapter two also discusses the differences between and the influences of, rotating stall and surge on axial-flow compressors, particularly in multistage machines. This literature review covers the most important studies and observations in the field of rotating stall effects on compressor performance.

Chapter three attempts to present a brief literature review about the different types of passive control techniques, for the delay and prevention of rotating stall, including casing treatment applications. The effects of the different types of the casing treatments on the isolated rotor as well as the single and multistage axial-flow compressors were also described. To the author's knowledge, the most important previous works and observations in the field of the casing treatment application in axial-flow compressors have been reviewed in this chapter.

Chapter four provides an accurate description of the test rig. The general geometry and parametric specifications of the applied three-stage research compressor including the corresponding measurement techniques and instrumentation are also presented in this chapter. The detail principals that were used to calibrate the test rig torque measuring device and associated transducer (load cell), are also described in chapter four.

The three-stage compressor geometry specifications under nominal attitude, in addition to the overall and individual characteristics, are presented in chapter five. The effects of the total pressure probe traversing in addition to the influence of the honeycomb flow straighteners on the compressor performance are also surveyed in this chapter. The determination procedures of some of the flow characteristics in addition to some significant equations, which were used in order to provide an appropriate calculation program, are also described in chapter five. It is worth noting that the applied calculation program is provided with the objective, which with some modification can also be used with high-speed compressors.

The general discussion regarding the mechanism and applications of vane-recessed casing treatments in addition to the parametric influence of the geometrical effects on this type of treatment as an anti stall passive device including the preliminary consideration in the design procedures are discussed in more detail, in chapter six. The effects of the most significant compressor related parameters on the design of the casing treatment are also described in this chapter. The influences of the casing treatment geometrical parameters in addition to the some significant achievements, resulting from the application of the vane-recessed type casing treatments are also described in chapter six.

Some significant geometry limitation factors such as the restriction of the available axial length of the compressor, which imposed significant challenges to the design of the casing treatment, are discussed in chapter seven. A set of equations including particular coefficients, which provide an appropriate procedure for the design of a vane-recessed tubular-passage casing configuration are also described in this chapter.

The flow measuring and flow visualisation devices along with a unique table, which consists of the significant geometry of the designed casing treatment, are also presented in

chapter seven. In order to facilitate a comparison of procedures, the relevant dimensions and data achieved from similar casing configurations, that have already been used by the previous researchers at Cranfield University, are also discussed in this chapter.

The compressor geometry specifications under the one-stage configurations with the solid and treated casings in addition to the first stage rotor only builds, all in the presence of the inlet guide vanes are described in the chapter eight. The influences of the honeycomb air straighteners on the performance of the rotor only build and one-stage configuration are also discussed.

The detailed flow characteristics and compressor performance obtained from the one-stage and three-stage test results are presented in chapter eight. This chapter ends by looking at the one-stage compressor builds with the solid casing (datum configuration) as well as the seven casing treatment builds of (0.232, 0.333, 0.434, 0.535, 0.636, 0.737 and 0.838) rotor blade tip axial chord exposure ratios.

The improvements or deterioration of the one-stage compressor builds with different rotor tip exposures are described in chapter nine. The improvements of the stall margin in terms of flow rates are also illustrated. The associated losses or performance gains, achieved with the casing configurations in terms of the work coefficients, overall and stage maximum efficiencies and peak pressure coefficients are also described in this chapter.

The influences of the treated casings on the features of gained or sacrificed performance have been presented in terms of improvements in stall margin, as well as the gains or losses of maximum efficiencies and peak pressure rise coefficients in chapter nine. This chapter ends with some conclusions, which have been drawn from the test results, and presents some proposals and suggestions for future work in the area of the current investigation.

This dissertation concludes with the list of references to previous studies, which the author has used in the current research. A number of appendices are also included for information purposes.

CHAPTER TWO

2)- Aerodynamic Instabilities in Axial-Flow Compressors:

2.1)- Introduction:

The performance of a compressor is commonly characterised by pressure ratio, efficiency, mass flow and work input. The stability of compressor could also be considered as a performance characteristic. From another point of view the compressor stability is related to the response of the machine to a disturbance, which perturbs compressor operation from a steady operating point. When the disturbance is transient, and the system returns to the original point of operational equilibrium, the performance is considered stable. But when the compressor operates away from the original point, the machine performance is unstable.

Consequently, there are two areas of compressor performance that relate to stability. One deals with operational stability, whereas the other is associated with aerodynamic stability. The operational stability is concerned with the matching of performance characteristics of the compressor with a downstream flow device such as a throttle, turbine, etc. But the aerodynamic stability is associated with a limitation of steady state operation due to stall and surge.

The compressor aerodynamic stability is the ability of the entire system such as inlet guide vanes, rotor, stationary blades and diffusers to maintain or increase the delivery flow associated with appropriate pressure to compressor downstream, when for any reason the compressor operation has been perturbed to a lower flow.

In other words, when the mass flow rate through a compressor is changed, the performance of a compressor is stable, if steady state operation can be established at new mass flow. Meanwhile, if steady state operation is not possible, the compressor performance becomes unstable.

Since, the main objectives of the present investigation is to understand the onset of the non-linear aerodynamic instabilities, which leads to fluctuation of the pressure and mass flow rate, in multistage axial-flow compressors, in order to improve the stall margin by applying a vane-recessed tubular chamber casing treatment.

The subject of compressor surge and stall consist of the study of stability requirements and of the physical aspects of the aerodynamic process. Therefore, it is necessary to identify the effectiveness of the different flow instabilities in axial-flow compressor. The current chapter however, attempts to outline briefly the development in two aforementioned flow instabilities, which normally lead to an undesirable phenomenon in compressors that is called rotating-stall.

Realising the effects of the two global and local flow instabilities in compressors namely surge and stall can do this. Surge is a term that has been applied to the audible instability of compressors operation. It is a catastrophic experience that results in compressor instability. The stalling of some subsystem element precedes the occurrence of the surge.

Stall can be regarded as the cessation of a continued rise in lift on an airfoil or in static pressure recovery in a diffuser or cascade. In a compressor, a blade row element can be

completely or partially stalled, but the compressor system can still be stable. Surge is the result of the stalling of some system component or combination of components such that continued reduction in mass flow makes the entire system unstable and causes audible flow fluctuations.

The prediction of surge or stall is much less tractable. The requirement for a stable range of flow is normally dictated by the performance characteristics of a downstream component such as a gas turbine in an aero-engine system.

2.2)- Surge Phenomenon:

The surge is a violent instability of the whole compressor system during which the flow reverses at a frequency of a few cycles per second. As far as the overall characteristic of the compressor is considered, the instability due to the stall is referred to as localised flow instability. The global system instability moreover can occur, which finally leading to surge phenomenon.

The compressor operating life in the unstable area of stall and surge is not favourable from both the compressor performance and also mechanical point of view. Therefore, if a compressor surges, it results in large fluctuation of the annulus flow with time, which subsequently may lead to the mechanical damage of the compressors.

The transient flow behaviour not only makes it difficult to predict surge but it also prevents accurate assessment of structural integrity because of vibratory loads that occur during the transient aerodynamic behaviour. If the problem of stall can be avoided, it is likely to suppress surge.

To ensure the compressor against the dangerous effect of surge, it is required to delay or suppress the stall, which is a requirement for surge and may be ultimately lead to surge. The rotating stall in severe cases may lead to surge, which is marked by a complete breakdown of the annulus averaged compressor flow field.

2.3)- Stall Phenomenon:

The term "stall" is used by analogy with an aeroplane wing where, as the angle of incidence of airflow is increased, the lift force on the wing at first increases, but falls sharply when critical angle is exceeded. The similar effects occur in a compressor blade row, and in terms of the pressure rise there is a fall off beyond the stalling incidence and below the critical flow coefficient, (McKenzie, 1997).

In the annulus of a compressor, a blade may be only partially stalled along the span and moreover, not all blades may be stalled even though all blades should be at the same incidence angle. The stall cells normally have total pressure depressions and contain little or no flow extending upstream and downstream of the blade row.

The stall means one thing to an external aerodynamic and something else to an internal aerodynamic. This phenomenon also can be regarded as the result of boundary layer separation. In a compressor the ability to provide a pressure rise is limited by blading and diffuser stalls. The stalling of the compressor blade row element precedes the surge or complete system collapse.

There are two types of stall that have been experimentally identified for compressors. One is individual blade stall and the other is rotating stall. Both of them however, may lead the compressor to flow instability conditions.

2.3.1)- Blade Stall:

At a constant speed, a compressor produces a total pressure increase in a particular corrected mass flow rate and speed. Then the operating range of the compressor is bounded on the high-flow end by blade passage choking and on the low-flow is ended by blade or end-wall stalling.

The low flow effects occur, because mass flow of a compressor flows from a region of the lower pressure to a one of the higher pressure. This means that the pressure gradient induced acts against the direction of flow and provides suitable conditions for a rapid boundary layer growth.

A type of stall in which all the blades around the compressor annulus without the occurrence of a stall propagation mechanism stall simultaneously is called the "individual blade stall". This type of the blade stall in fact is characterised by the development of large separated flow regions or zones of the low flow, in the wake of each blade.

Even still, the prediction of surge or stall rests with the transient flow conditions leading up to and occurring at blade row stall.

2.3.2)- Stall Zone:

The stall initiated by either wall or blade stalls in the majority of cases creates a stall cell, which causes a flow blockage in the compressor annulus. This blockage that has a significant net effect through the flow and areas of the reverse flow is also known as the "stall zone".

The stall blockage can range from covering only part of the span either at the tip or in rare cases at the root, and also being only a few blades in the annular width to covering the full span and extending over more than (180°) of the compressor annulus. This latter situation commonly occurs near the design speed of multistage compressors.

Typically, when a blade row stalls, all the airfoils do not stall uniformly. Instead, groups of airfoils stall to form a zone or zones of stalled flow around the annulus. When the zones are established, mass flow ceases to change and stable operation is restored but at a reduced level of mass flow relative to unstalled operation.

In addition the stall zones (cells) rotate relative to the rotor around the cascade annulus in the opposite of rotor direction. The speed of stall cells rotation is from (10% to 90%) of the rotor speed and in the direction of rotor blade rotation as viewed from the absolute frame of a stationary observer, (Pampreen, 1993).

It is worth noting that the compressor can still operate under certain conditions with some stall zones. However this is quite risky, because the rotating stall zone sets up vibrating stresses in the compressor. The number of possible stall cells can be as many as nine or as few as one. The number of stall zones is associated with the type of stall. There are two types of stall, namely progressive stall and abrupt stall.

2.3.3)- Different Types of Stalls:

The axial compressor tests so far have shown two types of stall phenomena, which are normally known as progressive (continuous) stall and abrupt (discontinuous) stall. As it was stated in progressive stall, there is a continuous drop in pressure rise characteristic as the flow field passes from unstalled to stalled conditions. In abrupt stall, there is a discontinuity in the pressure rise characteristic as compressor operation transitions to stall.

The stall phenomena (either progressive or abrupt) in compressors can be associated by the

initiation of several types of aerodynamic instabilities, which normally limit the performance of the compressors.

The classification of stall by type of stage performance characteristic is probably the best way to denote the stall type. The difference between progressive and abrupt stalls however, can be illustrated as follows.

2.3.4)- Progressive Stall:

A smooth continuous stage performance characteristic in the stall region shows a gradual increase in blocked annulus area due to the stall and is therefore referred to as progressive stall. It means that the progressive stall has a gradual reduction in total pressure ratio after stall begins. The progressive stall contains multiple cells.

The characteristics of progressive stall in the compressor are shown in [Figure (2-1)]. When progressive stall occurs, the drop in the overall performance of the compressor is quite small and often the presence of stall is indicated only due to the noise changing.

2.3.5)- Abrupt Stall:

A discontinuous stage performance characteristic apart from the progressive stall, which is known as the "abrupt stall", is also possible. It has been observed that in this type of stall, a single stall zone covering as much as half the annulus area and extending over the entire blade span.

The abrupt stall has a sharp discontinuity in the pressure ratio characteristic with a sudden drop in pressure. The abrupt stall appears to always occur with a single stall cell. It seems however, that due to the occurrence of discontinuity in the pressure curve, this type of stalling phenomena is called abrupt stall.

The abrupt stall characteristic in a compressor is shown in [Figure (2-2)]. It is worth noting that there is a discontinuity on the entering stall, as the operating point falls along a constant throttle line from the point of maximum pressure rise onto the stalled branch of the characteristic.

Once the rotating-stall is countered, it may not be possible to return to the unstalled position merely by opening the throttle due to the system hysteresis. For the stages exhibiting abrupt stall a hysteresis can occur when entering and leaving stall.

2.3.6)- Progressive or Abrupt Stalls Criteria:

For both cases of the progressive or discontinuous stalls, the flow is no longer axisymmetric, but instead has a circumferential non-uniform pattern rotating around the compressor's annulus.

It has been shown that the total to static pressure rise coefficient of a stage (ψ_{TS}) is approximately constant at a value of $[(\psi_{TS})_{\text{stall-point}}=0.11]$ from the stalling flow coefficient $[(\phi)_{\text{Stall}}]$ almost to zero flow coefficient $[\phi=(V_a/U)]$, (Day et al., 1977). This is the case whether the stage exhibits abrupt stall or progressive stall and appears to be true for a wide variety of stage designs.

Moreover, if a stage develops a value of the total to static pressure rise coefficient (ψ_{TS}), greater than $[(\psi_{TS})_{\text{stall-point}}>0.11]$ before stall, it will have an abrupt stall and the operating point will fall along a throttle line as indicated in figure (2-2). If on the other hand the stage stalls at the total to static pressure rise coefficient less than $[(\psi_{TS})_{\text{stall-point}}<0.11]$, the total to static pressure rise coefficient will fall off progressively towards zero flow coefficient, as it has shown in figure (2-1).

The total to static pressure rise coefficient of a stage has been defined as follows:

$$\psi_{TS} = \frac{\Delta P}{\rho U^2} = \frac{P_2 - P_1}{\rho U^2} \quad [2-1]$$

Where:

(P_2): The stage outlet static pressure, [Pa]. (P_1): The stage inlet static pressure, [Pa].

(ρ): The mean density, [kg/m^3]. (U): The mid span rotor blade speed, [m/sec].

2.4)- Rotating Stall Phenomenon:

Rotating stall may be defined as a flow non-uniformity that is moved circumferentially relative to the blade row. The presence of rotating-stall is usually obvious from the circumferential traces of axial velocity especially after overall stall. The rotating stall is the most commonly observed type of stall in compressors and its effects are stronger upstream of the rotor with a rapid decay occurring on the downstream side. Therefore it could be regarded as a local instability of the compressor.

The rotating stall is a non-axi-symmetric flow instability, which has a circumferentially non-uniform pattern and rotates normally around the compressor annulus. The rotation of the stall cells can be explained by viewing the blade row from the top of the annulus as shown in [Figure (2-3)].

It is worth noting that the stall is normally initiated on some blades, rather than all of them. It is still not known completely for sure, why all the blades do not stall simultaneously. It was pointed out that the dimensional tolerances could be one reason, (Schulze et al., 1950). It means that one or more blades could be manufactured with a slightly higher stagger angle, or their profiles may be slightly different.

Nevertheless, when the stall begins on a blade as it can be seen on figure (2-3), the flow is deflected toward passages on either side of the affected blade. The stalling zone then intensifies and as a consequence the channel becomes blocked. The streamline deflection at the blade row inlet decreases the incidence angle of the forward blade and increases incidence on the rearward blade, opposite to the rotor rotation. The rearward blade then becomes the next stalled blade.

The stall cell is induced to propagate opposite rotor rotation in the relative frame. As it does so, the original blade comes under the influence of the low incidence side of the cell and becomes unstalled. As it can be observed from the absolute frame, the cell appears to be moving in the direction of the rotor rotation, but at much reduced speed.

In a rotating-stall, the annulus contains regions of stalled flow, usually referred to as "stall cell" and regions of the unstalled flow. The overall or time-mean mass flow remains constant, but the local mass flow varies as the rotating cell passes the circumferential points within the compressor. These stall cells induce large vibrating stresses in the compressor blading that can cause premature fatigue of the blades.

It is worth noting that in some cases the rotating stall phenomenon may only be apparent through signal spectral analysis, (Mathioudakis et al., 1985).

The rotating stall is in fact a disturbance of the compressor rotor blade region, which may lead to the occurrence of surge. Because of the huge pressure fluctuation involved, the phenomena may cause a large fall in pressure rise with aerodynamic instability, blade vibration, noise and sometimes, mechanical failure as a result of the periodic loading and

unloading of the blades.

In certain compressors rotating stall seriously restricts the flow and may lead to shutdown. The restarting of the stalled machine in some cases is not often possible until the stall cell clears and normal flow conditions are restored. This situation normally requires an almost complete shutdown of the engine.

Where the mass flow is too small, rotating stall, which is sometimes known as propagating stall, allows the compressor to adapt the reduced amount of mass flow. Instead of trying to share the limited flow over the whole annulus, the flow is shared unequally, so that some blade passages have a quite large flow and some very little.

The rotating stall in fact can be expressed as a flow breakdown into sectors of stalled and unstalled flows. The stalled sectors may have a very low axial velocity, or even a small negative one, whereas the unstalled sectors operate at a level of axial velocity consistent with the unstalled flow.

The stall cells rotate in the direction of the blade speed, but at a lower velocity. The number of cells and their speed of rotation may vary as the flow coefficient varies. The rotating stall in a compressor can be associated with the initiation of several types of aerodynamic instabilities that limit the compressor's performance.

As the rotating-stall consists of one or more low flow zones that rotate in the compressor annulus, the blade row will experience periodic aerodynamic forces as the low and high flow zones pass by the blades. The fluctuating forces due to the rotating-stall can be a source of resonant blade vibration, where the frequency of stall relative to the blades excites the blades.

Although several theories of rotating stall in compressors appear in the literatures, none of them can be used to predict propagation rates with any appreciable degree of the reliability. These theories however, are useful in evaluating the significance of the parameters pertinent to the rotating stall mechanism.

2.4.1)- Rotating Stall Cells:

Many researchers have suggested that it is possible to divide the flow into a stall cell and a clean flow region based on certain criteria. It should be noted that the boundaries of the stall cell could be defined as those regions in which, the axial velocity is negative or zero. However there is an alternative view which claims that such a stall cell or clean flow distinction does not actually exist, (Yu et al., 1990).

Although the axial velocity inside the rotating stall cells is less than the unstalled flow, the circumferential component can be very large. This type of stall can also be viewed as a successive stalling and unstalling of blades in a particular cascade.

The radial extent of the stall zone along a blade is another convenient characteristic for classifying the rotating stall. The stall cell covers part or the entire blade span in addition to extending circumferentially over several blade passages. It is worth noting that the stalled blade channels offer a high resistance to the flow through the cascade and the approaching flow tends to be diverted to each side of the stalled region.

The stall usually starts with multiple cells, and then as the flow reduces, the number of cells may increase. However, as the flow is further reduced, the stall cell number may diminish until only a few cells are present. Moreover, it is so much difficult to determine the precise stall cell count, when the flow through the compressor is extremely unstable.

It was reported that after a small number of rotor blade revolutions, the affected stall cells

can further break up and then again rejoin temporarily to form the same number or fewer cells, as existed at the point of breakdown, (Pampreen, 1993). The number of rotating stall cells depends upon flow conditions and the compressor geometry. These types of stall cell normally propagate between (20% to 70%) of the rotor speed.

Stall cells have a variety of geometry and are seldom stable. The stall cell can occupy only part of the blade span, or it can cover the entire blade span. The term partial span refers to stall covering part of the blade span, whereas the term total or full span stall is used to refer to the stall covering the whole blade length. However, the full span stall can be progressive or abrupt.

A relationship between the radial extent of stall cells and the type of the overall total to static characteristic had been found. When the stall cell is full span, normally there is only one cell, but when the stall is part span, there can be multiple stall cells. The characteristics of an axial flow compressor under progressive stall (partial span stall) in addition to which under abrupt stall (full span stall) conditions are shown in [Figure (2-4)] and [Figure (2-5)], respectively.

The progressive stall type characteristics which produce small or moderate drops in pressure rise indicates a part span stall, whereas, the abrupt stall type characteristic that provides a large drop in the pressure rise is accompanied with the full span stall.

A compressor may also have an intermittent stall pattern at a particular operating point. In this case one stall pattern exists for an instant before changing to a new stall pattern. It was also found that the stall pattern begins at peak pressure rise and is initiated by random flow separation from the blading, (Pampreen, 1993).

It has been noted that the number of stages has a significant effect on the speed of the full span rotating stall cells. The results also show that the speed of the cells was increasing as the number of stages was increased, (Day, 1976).

2.4.2)- Axisymmetric Stall:

The axisymmetric stall can be described as reverse or retarded flow area, which is, located axi-symmetrically in the annulus and over parts of the blade span. Axi-symmetric stall normally is preferred to rotating stall, because it gives continuous pressure rise characteristics. This type of stall is also favourable from a mechanical perspective since it induces less blade vibrations compared with rotating-stall.

An experimental investigation of the stalled flow behaviour in a single-stage axial-flow compressor and subsequent calculation aimed at obtaining the axisymmetric compressor characteristic was carried out by Zhang et al. (1991). Experiment results showed that in deep stall, both forward and backward flow might occur in the stalled blade passage.

It was indicated that a stall cell is an intense flow-interaction zone with both cross flow and rapid variations of flow velocity and pressure at its boundaries. It was also concluded that a stall cell in axial compressor rows could not be axially extended though it may well be so in multistage axial compressors on a stage-to-stage basis.

2.4.3)- A Brief Historical Development of Stall in Axial Flow Compressors:

The feature of the blade row and stage stall that has received a great deal of research attention in the last fifty years is called rotating stall. The phenomenon was first reported in a centrifugal flow compressor, (Cheshire, 1945).

It is worth noting that the most of the information on rotating stall comes from axial flow compressor testing. When the rotating stall was the main cause of the compressor blade

failures and the major problem in the development of an axial flow turbojet engine, it was rediscovered again and reported. It had also been stated that the experimental investigation by means of wool tufts led to a detection of an asymmetric stall flow pattern occurrence in a test compressor, (Schulze et al., 1950).

In a study of the stalling characteristics in an axial-flow induced centrifugal impeller system, rotating stall was identified for the first time as propagating stall, (Grant, 1951).

A preliminary experimental investigation on a single stage axial flow compressor with (0.5) hub to tip diameter ratio showed that the rotational speed of the stall regions increased with compressor speed and the number of stall cells, (Huppert et al., 1952).

At this stage the main features of rotating stall were more or less an unknown. An experimental and theoretical investigation of the rotating stall characteristics of a single stage axial flow compressor with (0.76) hub to tip diameter ratio compressor was carried out by Graham et al. (1953).

The obtained results by Graham et al., stated that a wide range of stall data was achieved from (60% to 100%) of design speed. It was also noted that the full-span and part-span stall patterns showed that at the fixed compressor speed, the stall frequency is almost independent of cell number. This is however, in contradiction to the results of Huppert et al. (1952) in which stall frequency was shown to increase with cell number at fixed compressor speed.

Subsequently, Marble (1953), Huppert et al. (1953) and Iura et al. (1954) were also amongst the first researchers who reported their observations relating to the propagating stall phenomena through experimental investigations.

The results from an experimental investigation of a transonic flow compressor rotor with double-circular-arc airfoil blade sections, showed that the number of stall zones diminished as part-span stall developed into full-span stall, (Lewis et al., 1954). The experimental study that was conducted by Tysl et al. (1955) on a transonic compressor rotor illustrated that the performed uniform tip stall was immediately followed by surge.

After nearly a decade of stall phenomena research in centrifugal compressors, a research group then made the first detailed study by using hot-wire technique to investigate the flow conditions in an axial flow compressor, (Emmons et al., 1955). Two principal types of rotating stall were observed in this investigation, one with the two or more stall cells over part of the blade height and other with a single stall cell over the full range of blade height.

An experimental investigation which was conducted on a compressor rotor with (0.9) hub to tip diameter ratio and without inlet guide vane showed that as flow is reduced, the two-cell stall pattern changes to a three-cell stall pattern, (Costilow et al., 1955).

To more clearly identify stall characteristics between rotor and stage, an axial flow compressor, which consists of an inlet guide vane, a rotor and a stator, had been tested by using hot-wire anemometers in order to record flow fluctuation and determine stall patterns. As it was noted with the stator removed, there was a significant change in the stall characteristics, (Montgomery et al., 1957). This study made no reference to either heavy blade stall before rotating stall or between part-span and full-span stall regions, but it did discover that one-cell stall zone had been appeared on the inlet guide vane.

A report that was released by Continental Aviation and Engineering Corporation (CAE) (1966) identified a further influence on the abruptness of the stall characteristics, that is the level of single stage pressure ratio. As it was stated in this report, the stable range from choke to surge tends to diminish as the single stage pressure ratio increases and as the rotor

tip relative Mach number increases.

An investigation relating to the influence of aspect ratio on rotating stall was conducted by Fahmi, (1971). Two single stage compressors were used. All three blade rows in each compressor were designed with the same chord length. Both compressors consisted of an inlet guide vane, a rotor and a stator, and had identical blade characteristics.

The hub to tip diameter ratio of both compressor was (0.75), whereas, the aspect ratios were (1.0) and (2.0), respectively. The result showed that the compressor with the lower aspect ratio had a higher pressure rise coefficient and a wider range of the stall free flow with a positive slope to the characteristic. The compressor with the higher aspect ratio showed no positive characteristic at all. The performance curves of both compressors also exhibited abrupt stall and stall recovery hysteresis.

The influence of the rotor-stator coupling on stalling was investigated by Ludwig et al., (1976). A summary of the results showed that rotating stall inception was delayed to higher values of inlet relative flow angle or lower values of the flow coefficient. Some time later more detailed measurements were carried out on low speed compressor assemblies with one to four stages over a range of three-design flow coefficient. The test results showed that the stall cells extended the full length of the compressor from the inlet of the first rotor to the exit of the last stator, (Day et al., 1978).

Ludwig et al. (1979) carried out a further study of stall criteria in relation to rotor-stator coupling as well as the isolated rotor. It was found that the rotor total pressure coefficient was higher in the coupled rotor stator situation than in the isolated rotor test.

The measurements, which have been done on an axial flow compressor, indicated that the flow-field in the rotating stall was strongly three-dimensional, (Breugelmans et al., 1983).

Another correlative approach, which determines the rotating stall angle and the stalled flow coefficient rather than the stalled pressure rise, was proposed by Zika, (1985). Two-dimensional compressible modelling has been used in order to control rotating stall in high-speed multistage compressors. This analytical model has been manipulated into an input/output form.

As it was noted the applied modelling and evaluation procedures in this theoretical investigation can be used as a first step toward a rigorous approach in order to design an active stall control system for high-speed axial compressor, (Feulner et al., 1996).

A fundamental study of compressibility effects on dynamic stall of fixed and adaptive airfoils was investigated in order to understand some of the key mechanisms of compressible dynamic stall including the effects of Reynolds number and flow behaviour over the airfoil at transonic conditions, (Chandrasekhara, 1997).

2.4.4)- Rotating Stall Inceptions:

The process by which a compressor moves from a nominally axisymmetric flow regime to a rotating stall regime may be termed rotating stall inception. The rotating stall onset is defined more specifically as the first noticeable change in the structure of the flow leading to the formation of a stable cell pattern.

Although rotating stall has been researched for nearly fifty years, it is only in the last twenty years that the study of the inception process itself has commenced. In order to predict rotating stall inception or delay for appropriate stall margin improvement, considerable efforts have been made over the years since the rotating stall phenomenon was first discovered and reported.

The inception of the rotating stall phenomena is remarkably complex and both theoretical and the experimental studies have necessarily involved large degrees of the simplification. It was stated that the rotating stall inception sequence can be originated as a disturbance of the tip clearance vortex, (Freeman, 1985).

It was reported that if part span cells at the blade tip constitute the stable condition after rotating stall inception, it would then be concluded that rotating stall inception has occurred at the tip, (Jackson, 1986).

There is however evidence that when the rotating stall cell is fully developed, it begins as a passive blockage at inception and then transitions to the active mode during the period of development. This was observed in an experimental investigation on a single stage low speed axial flow compressor by Jackson, (1987).

From the observation of the pre-stall waves in multistage axial flow compressors, it has been reported that the rotating stall inception sequence can differ considerably between compressors, (Day, 1989).

Investigations have also been carried out on a variety of compressors. In some of them the pre-stall waves have been observed to grow through a continuous process into rotating stall, whereas, in some others, the pre-stall waves were apparently absent, (Day, 1989) and (McDougall et al., 1990).

The experimental knowledge of the stall inception process has largely been based on correlative studies of the conventional work state parameters such as the casing boundary layer thickness and the blade row static pressure rise. A greater understanding of the rotating stall inception process should lead to better prediction and control of the undesirable effects of stall and finally could contribute to more rational design methods for casing treatments as an anti-stall passive control technique. This will be discussed in more detail in the following chapters.

A measurements using hot wires on two laboratory test axial flow compressor led to the conclusion that the occurrence of modal perturbations and the formation of a finite amplitude stall cell, are two separate phenomena, both occurring under roughly the same conditions at the peak pressure rise characteristics, (Day, 1993). The measurements also underlined the hitherto unsuspected importance of short length scale disturbances in the process of stall inception.

The stall inception and post stall transients have been studied through an experimental investigation in an aero-engine axial flow compressor by Freeman et al., (1993). The obtained results from the undertaken tests, showed that the varying nature of the stall process was initiated as engine speed was increased. It was found that the initial stall subsequently led to deep rotating stall, surge and flame out.

While most of the research work in the subject of compressor flow instabilities was done on isolated compressors on test rigs or by numerical simulation, Hoss et al. (2000) carried out their investigations on the compressor system stall inception measurements of a two-spool turbofan engine at various power settings.

Using several analysis techniques, three different types of stall inception processes were observed in the low-pressure compressor, depending on the rotor speed. Meanwhile at high speed, rotor shaft imbalance dominates the stall inception process as an external forcing function.

It seems that blade or wall stall features point to the important role that may be played by the components of the flow structure. The wall stall approach would suggest that rotating

stall inception could be associated with a thick annulus boundary layer, even when the stream tubes are predicted to be stable.

2.4.5)- Rotating Stall in Relation to Hub to Tip Diameter Ratios:

As the mass flow through the compressor is reduced, the tip of the rotor blade in a tip critical compressor, as well as the hub of the rotor blade in a hub critical compressor will reach the peak of their characteristics. The operation at or near the flow characteristic peak is linked by both experiment and theory to the rotating stall inception, particularly for those compressors with high hub to tip diameter ratio.

For rotors stall begins at the blade tips, whereas for stator vanes, it initiates at the hub. For high hub to tip diameter ratio rotors, it will probably cover the entire blade span. As flow is reduced further, number of stall cells and the type of stall can change, (Pampreen, 1993).

The existing experimental data of stall cell patterns in low hub to tip diameter ratio compressors indicates that both part span and full span stall cells may occur, (Tanaka et al., 1975). The available experimental data on rotating stall in high hub to tip diameter ratio types of compressor mainly describe the form of the fully developed cells. It is worth noting that there is very little data on the rotating stall process in this class of compressor. Hence it is not yet known at which radial location the stall inception takes place and whether an incipient cell may rapidly move out to the tip.

In the low hub to tip diameter ratio stages, stall cells usually appear at the outer diameter first as flow is reduced, and spread toward the hub with further flow reduction. But in the high hub to tip diameter ratio stages, the stall cells often occupy the whole blade span immediately on stall initiation, and increase in the extent of the circumference, which they occupy, as the flow is further reduced, (Pampreen, 1993).

Some types of the stall cells in a low-speed four-stage research compressor with (0.8) hub to tip diameter ratio were observed by Day et al. (1977) and these are shown in figure (2-4) and figure (2-5), respectively. It was found that in the case of high hub to tip diameter ratio stages, the static pressure exhibits a discontinuity and remains approximately constant on the stalled branch of the characteristic. This has led to a suggestion that the unstalled segments of the annulus operate on the normal unstalled branch of the characteristic at the same pressure rise as the stalled areas which operate at the approximately zero axial velocity.

For the low hub to tip diameter ratio stages that exhibit progressive stall, it can be seen that the outlet static pressure is approximately constant at its maximum value from the stall point to zero flow. The part span rotating stall is only initiated below the flow coefficient at which the outer section of the blading reaches a maximum total pressure rise.

For a stage, which stalls progressively, the operation is similar but the unstalled flow is at the point where the static pressure first reaches the maximum value as flow is reduced. As the throttle is closed the mass flow is reduced by a reduction of the annulus portion occupied by unstalled flow while the stalled part expands.

The majority of detailed measurements on rotating stall have been carried out on the high hub to tip rotor diameter ratio compressors. The understandable relationship between the radial extent of the rotating stall cells and the overall total to static characteristic may be described such that the abrupt stall type characteristic promotes a full span stall, whereas, the progressive stall type characteristic provides a part span stall pattern.

Recent investigations of high hub to tip diameter ratio compressors have revealed that forward spillage of the tip clearance vortex can play a dominant role in the rotating stall

inception process. In the case of the low hub to tip diameter ratio compressors, much less is known.

Sharma et al. (1986) carried out an experimental investigation on three axial-flow test compressors with three different hub-to-tip diameter ratios namely, a single stage of (0.66), a two-stage machine of (0.71) and an isolated single rotor of (0.4) hub to tip diameter ratios. The study was conducted in order to illustrate the effects of the various parameters such as rotor stagger, aspect ratio, hub to tip ratio in addition to the exit boundary condition on the level of pressure rise to which the compressor operation deteriorates at occurrence of rotating stall.

As it was reported, the hysteresis and overlap between the stalled and unstalled characteristics are both dependent on rotor outlet angle, hub to tip ratio and aspect ratio. The compressors with higher hub-tip ratio and high aspect ratio exhibit large hysteresis but no overlap. Additionally, it was noted that the stall blockage prior to recovery from the full span has also been found to vary with aspect ratio and hub to tip ratio.

The effects of stall inception related to the rotor blade tip clearance of a single stage low speed axial flow compressor with high (0.8) hub to tip diameter ratio was investigated by McDougall et al., (1990). It was however, concluded that with high reaction stages the rotor is much more likely to be the cause of stall. The results also showed that for the same overall geometry a compressor could stall at the rotor hub for small tip clearance and at the rotor tip for larger tip clearance.

The low speed stall occurs in the low hub to tip diameter ratio inlet stage or stages, while high-speed stall occurs in the high hub to tip ratio rear stages. Test results from low hub to tip ratio axial compressors have shown progressive stall, while data from high hub to tip diameter ratio compressors show abrupt stall, (Pampreen, 1993).

While streamtube contraction considerably is modified elementary ideas about stall inception in low hub to tip diameter ratio compressors, real fluid effects appear to play an even more important role in the relative stability of the root and tip regions. Therefore It seems that there is a need for a clear knowledge of the actual events taking place before and during stall inception.

The evidence from an investigation which has been carried out in order to define the process of rotating stall inception in a low-speed, low hub to tip diameter ratio fan suggests that the root has a very strong tendency to remain stable, (Soundranayagam et al., 1993). As it was stated this was largely due to the centrifuging of the boundary layer flows and reduction of axial velocity skew caused by stream tube contraction. It is worth noting that the authors were convinced that the phenomena described above were relatively independent of hub to tip diameter ratio.

An experimental investigation which was carried out on the various low hub to tip diameter ratio axial flow fans has been shown that there were two types of overall fan pressure characteristics for low hub to tip diameter ratio industrial fans; continuous type and abrupt type. It was also found that rotating stall had much stronger influence on the flow upstream of the rotor than the flow downstream, (Yu, 1993). It is worth noting that these criteria are significant in multistage axial flow compressors.

2.4.6)- Rotating Stall Corresponding to Number of Stages:

The inlet stages of a multistage compressor are liable to be stalled when the compressor is operating at the rotational speed, which is somewhat below the design speed. As Howell et al. (1951) have stated the stalling of the inlet stages may persist up to (70%-

80%) of the compressor design speed.

Single stage compressors exhibit a wider range of stall propagation rates than multistage compressors. Most multistage compressors have stall propagation rates of about half rotor speed. It has reported that, single stage compressors have stall propagation rates between (10%-85%) of the rotor blade speed, (Huppert et al., 1956). It has also stated that the rotational speed of the stall regions and consequently their frequency increase almost linearly with the rotational speed of the compressor rotor.

The experimental work carried out by Yershov (1971) has shown that changing blade geometry and increasing the blade setting angle has little effect on stall cell velocity, but this would affect solidity. He also showed experimentally that, when the solidity was halved, the rotating speed of the stall cells decreased from the (30%-20%) of the rotor speed. The previous mentioned experiments also have shown that the obtained information and data for the propagation rates or number of the stall cells in single stage compressors does not apply to the rotating stall characteristics of multistage compressors.

The compressor designer must always be aware of the possibility of blade failure due to the resonance between the blade frequency and the stall frequency. Many multistage compressor failures are not attributable to forced vibrations but are associated with the stall. The experimental findings on the pre-stall waves have shown that the rotating stall inception process can be fundamentally different even for similar multistage compressor geometry, indicating the possible existence of more than one rotating stall inception mechanism must be considered.

The experimental investigation of rotating stall in a mismatched three stage axial flow compressor clearly showed that stages can operate in an axisymmetric fashion even when heavily stalled, since rotating stall inception requires the stall of more than one stage, (Giannissis et al., 1989).

It also reported that for rotating stall to develop in a multistage machine, a simple blade or stage stall is not necessarily followed by rotating stall as other factors, such as stage interaction and overall compressor stability, influence the point of rotating stall inception. It was noted that during the experimental investigation the test compressor could be operated with rotating stall restricted to only the hub region of the span.

The stability of flow through a research stage operating in isolation and as part of a multistage axial flow compressor has been considered by Longley et al. (1990). It was reported that the unstalled flow range of a stage when operating as part of a multistage compressor could be considerably greater than when that stage is operating in isolation. It also found that the stall point and the stall performance of a stage, and by inference that of a compressor, are a property of the system in which it operates rather than of the stage or compressor alone.

Generally, rotating stall has the following undesirable effects, which can influence a multistage axial flow compressor performance.

- 1)- The fluctuation in the blades load.
- 2)- The marked drop in the pressure rise, mass flow and also overall efficiency.
- 3)- The triggering of surge, depending on the system's configuration.
- 4)- The restrictions to the lower end of the range of operating mass flow.
- 5)- The movement of the operating point away from the best efficiency due to the stall margin requirements.

2.5)- Difference between Rotating Stall and Surge:

Two types of local and global aerodynamic flow instabilities are common in axial flow compressors. Firstly, local flow instability¹, which is known as “rotating stall”, and secondly the global flow instability, which is called “surge”.

The useful extent of an axial flow compressor operating range is therefore greatly influenced by its stalling characteristics. Over the entire range of possible compressor speeds there exists a minimum flow point for each speed, where the compressor will either surge or exhibit rotating stall. Investigations into axial compressors stall phenomenon have concentrated on understanding the nature of stall in terms of stall initiation, the geometry of the stall cell, the aerodynamics of the stall cell, and the behaviour of the stall zones.

As the flow in an axial compressor is reduced, there is a transition from random separation to an established stall event. This can be a rotating or non-rotating stall patterns or surge. For rotors, the stall pattern can envelop part or the entire rotor span, (Pampreen, 1993).

The rotating-stall is normally associated with a sudden loss of performance, whereas, the surge is marked by a complete breakdown of the flow through the compressor and result in a large fluctuation of the whole annulus flow with time. The characteristics of rotating stall and surge are shown in [Figure (2-6)] and [Figure (2-7)], respectively.

The consequences associated with rotating stall can be quite severe with the flow rate and delivery pressure greatly reduced and therefore a loss in efficiency will take place, after which the compressor face with stall.

There can also be extra blade vibration and noise, the former of which may lead to mechanical failure as well. The process of the surge may be so violent that the mass flow can reverse and previously compressed mass then emerges out of the inlet. This is some times called "deep-surge".

In order to protect the compressor against surge, it is essential to delay or suppress the stall, which is a requirement for surge, and may also lead to it. It is also important to know which of these modes of instability will occur, since the consequences can be different for the each case.

2.5.1)- How to Distinguish Rotating Stall from Surge:

Whether a particular compressor will exhibit surge or rotating stall depends upon a non-dimensional parameter known as the “B Factor”. The (B) factor is a ratio of the pressure to inertial forces that was defined by Greitzer (1976). In a compressor if the calculated value of the (B) factor is higher than the critical value for that system, the compressor exhibits surge. But if the aforementioned factor is less than the critical one, the compressor exhibits rotating stall. The (B) factor is not only highly dependent on the compressor but also on the system in which it is operating, such as the duct, the plenum volume and throttle as well. The results obtained from these experiments largely concur with the predicted performance of the (B) factor, which can be calculated using the following formula.

$$B = \frac{U}{2C_s} \sqrt{\frac{V_p}{A_c L_c}} \quad [2-2]$$

¹ This type of the flow-instability can occur in one or more blades, and is called blade row instability.

Where: (U): The compressor's rotor blade speed

(V_p): The plenum volume.

(A_c): The compressor flow through area.

(C_s): The speed of the sound.

(L_c): the effective length of the duct.

An analysis of the applicability of surge/stall criterion or (B) factor related to the practical engineering problems of the compressors had been conducted by Tang et al., (1985). The authors realised however that this criterion, while being of great theoretical value, appears to be somewhat uncertain when applied to engineering compressor systems with the difficulties arising from changeable compressor characteristics and the diversity in the geometry and time lag in different systems.

Nonetheless, it was concluded from the numerical results that there exists a single critical value of parameter (B) for a given compressor working in possibly different systems while no identical critical value could be employed to different compressor in different systems.

2.6)- Stall and Surge in Multistage Axial Flow Compressors:

When in a multistage axial flow compressor one or more stages stall progressively, they may not initiate surge immediately, whereas a single stage entering abrupt stall is more likely to initiate surge, as was indicated by Huppert et al. (1953).

The progressive stall is generally associated with stages of low hub to tip diameter ratios and the abrupt stall is more common in stages of higher hub to tip diameter ratios. There is also a tendency of progressive stall to occur in stages with low design work coefficient and flow coefficient of about $[(V_a/U) \leq 0.35]$, even in high hub to tip ratio compressors.

Since the hub to tip diameter ratios from the first to last stages almost necessarily increases in those compressors in which annulus diameters are not constant, it is common for early stages to exhibit progressive stall and rear stages to exhibit abrupt stall.

At low speeds, when the mass flow is reduced the front stages stall first, but they do not immediately cause surge. In fact approximately half of the total number of stages may have to reach stall before surge occurs at low speeds. But on the other hand, at higher speeds the rear stage approaches stall first, and because it is likely to stall abruptly, surge may occur immediately.

It is normal in axial flow compressors that as the mass flow is reduced, the pressures rise increases. Generally a point is reached at which further reduction in the mass flow leads to non-linear flow instabilities within the compressor. Beyond this point the compressor enters an unstable region, where the dynamics of the compressor flow are not defined. The term conventionally used to describe the onset of axisymmetric global instability within the compressor flow is the "surge point".

The line marking the locus of these points for the different rotational speeds is known as the "surge line". The surge line divides the graph into two regions with the flow no longer steady to the left of the line where rotating stall or surge or a combination of both may occur. As a consequence, the surge line represents the limit of the safe operation of multistage axial flow compressors.

In multistage compressors, the stall of one or two stages does not mean the stall of the whole compressor. A multistage compressor may operate in a quite stable manner even though one or more of the stages are stalled. As it was described by Emmons et al. (1959) this type of the instability (surge) is different from the non-axisymmetric local instability

usually at the lower rotational speed, which is, referred to as “rotating stall”.

The measurements which carried out on a low-speed three-stage axial-flow compressor² indicated that the stable part span hub stall cells can in fact occur in some multistage compressors, (Giannisis, 1987).

A number of investigators have reported approximately similar results in association with some rotating stall control devices, for example Cargill et al. (1990). This experimental study was conducted on a high-speed simple jet compressor. The authors suggested that the results obtained from the research on the low speed rigs might not be directly applicable to high-speed compressor conditions that are commonly used in aero-engines. The high-speed surge of a multistage axial flow compressor is determined by the stall characteristic of the rear stage. The rear stages normally show an abrupt stall characteristic due to the high hub to tip diameter ratio, which is typical of a rear stage.

The phenomenon of stall and surge in a eight stage axial flow aero-engine compressor was studied using fast response static pressure measurements, as reported by Wilson et al. (1994). The results indicated the growth of flow instability at various speeds, from a small zone of stalled fluid involving only a few blades into the violent surge motion of the entire machine.

As it was noted the point of stall inception was moved rearward as compressor speed was increased. The authors also suggested that the final unstable operation of the under test axial flow compressor could be divided into rotating stall at low speed and surge or multiple surge at high speed. The experiment also found that stall/surge always started as a circumferentially small flow disturbance, rotating around the annulus at some fraction of rotor speed. However the stall inception process was independent of whether the final unstable operation was rotating stall or surge.

The majority of papers published so far in relation to the local and global instabilities in the axial flow compressor, suggest that our understanding about stall and surge phenomenon, have generally derived from investigations on low-speed axial flow compressors. As a general rule however, because of the effects of compressibility, the performance results achieved from low-speed machines are not always applicable to high-speed compressors. In view of this, any attempts to use results obtained from low speed compressors in modelling the performance of high-speed machines must be regarded as having doubtful validity.

Direct comparisons have been made between an eight-stage fixed geometry high-speed engine compressor and a four-stage low-speed laboratory axial flow compressor by Day et al., (1994). The results however showed noteworthy similarities in both the stalling and surging behaviour of the two machines, particularly when the engine compressor ran at intermediate speeds. It was noted that the engine results also indicated that, as in the laboratory tests, surge is precipitated by the onset of rotating stall.

It was also observed that when surge was initiated in a high-speed compressor, the initial pressure perturbation did not appear at the rear stages of the machine, as might have been expected, but more towards the middle of the compressor. This investigation also confirmed that laboratory testing of low-speed compressors is an effective and inexpensive way of gaining insight into many aspects of the behaviour of aero-engine compressors.

²

The same three-stage axial flow compressor, which was also used in the current investigation in order to study the effects of a vane-recessed tubular-passage casing treatment.

2.7)- Off-Design Performance of Axial Flow Compressors:

In multistage axial flow compressors, each stage operates in an aerodynamic environment determined to some extent by all the stages. The steady-state performance of a compressor can be expressed by providing the stage and overall characteristics. Normally when a compressor is throttled at a lower speed, low velocity regions appear, and these travel around the annulus in the same rotational direction of the rotor blades. Rotating stall that occurs in the compressor annulus produces a huge amount of losses and therefore affects the compressor performance.

The typical overall steady-state characteristic map of a multistage axial flow compressor has been illustrated in [Figure (2-8)]. This characteristic map which was described by Huppert (1965) shows different steady-state mass flow and rotational speeds that can be derived by testing a compressor on a test rig. In general, the flow in an axial compressor is axisymmetric and steady.

Both theoretically and experimentally it has been shown that the compressor stall point is not unique. A multistage axial flow compressor must be able to operate over a wide range of operating conditions, which may at times vary from the design point.

Each stage of a multistage axial flow compressor has certain characteristics that might vary between neighbouring stages. Therefore in order to have an efficient compressor, the characteristics of each stage have to be carefully matched. This is a relatively simple process to implement for one set of the conditions such as the design mass flow, the pressure ratio and the rotational speed, but it is much more difficult when reasonable matching is to be achieved over a wide range of conditions.

The operating conditions that fall either side of the design point is called the off-design performance (i.e. one or more than one of stages is stalled or a stage operates in the choked condition), and this normally causes inefficient compressor operation.

It is important to gain a better understanding of multistage axial flow compressor performance in the part-speed region of the running speed especially at a rotational speed less than the design speed. To do this it is necessary to consider, how the operating points alternate relative to the design point at the compressor's front, middle and rear stages.

The pressure rise (ΔP) of a stage versus corresponding mass flow coefficient (V_a/U) which presents the characteristics of each stage at different conditions has been illustrated in [Figure (2-9)]. In the first stage, if the compressor rotational speed is reduced to point (b), the density at the outlet from the stage is very close to the design value.

The reduction in the mass flow is entirely due to a reduction in axial velocity. This changing value of (V_a) is much greater than the altering of the blade linear speed (U), so that the part speed flow coefficient (V_a/U), is less than the design value, therefore the stage moves towards stall.

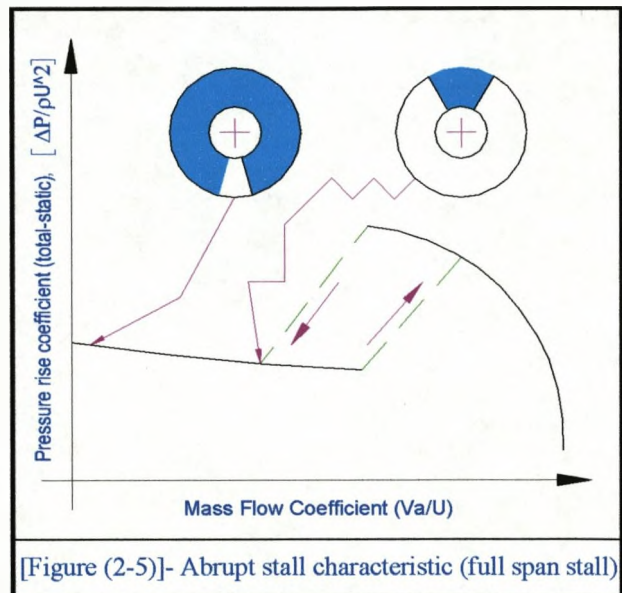
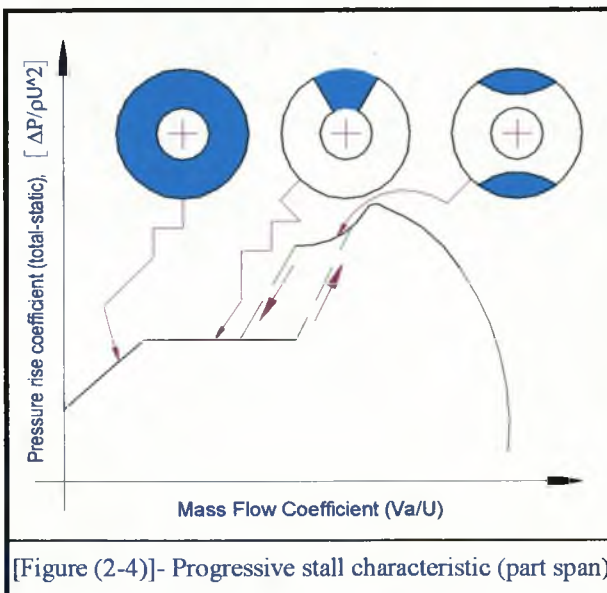
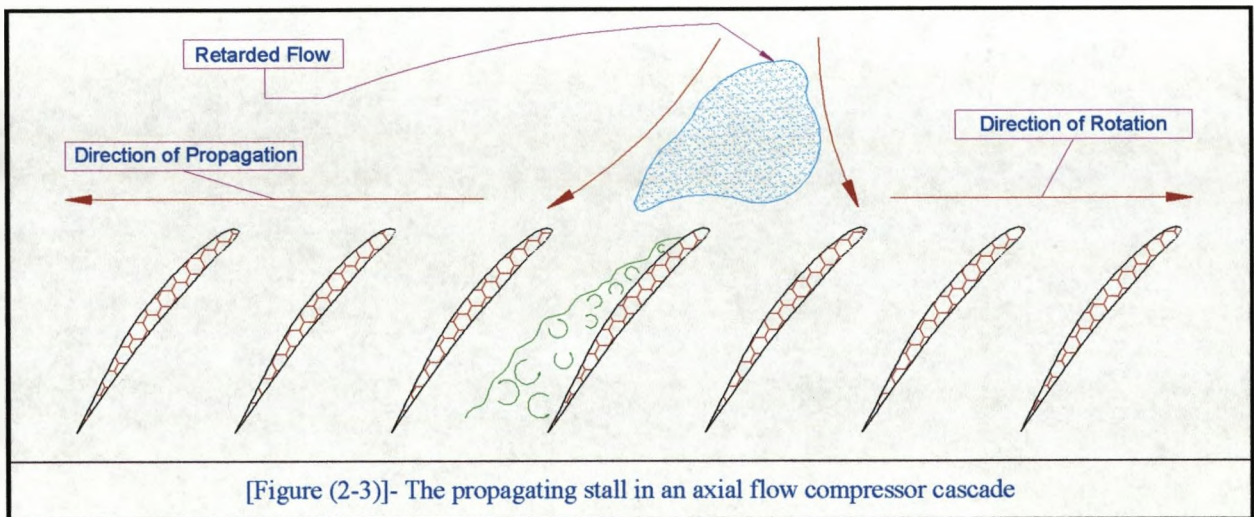
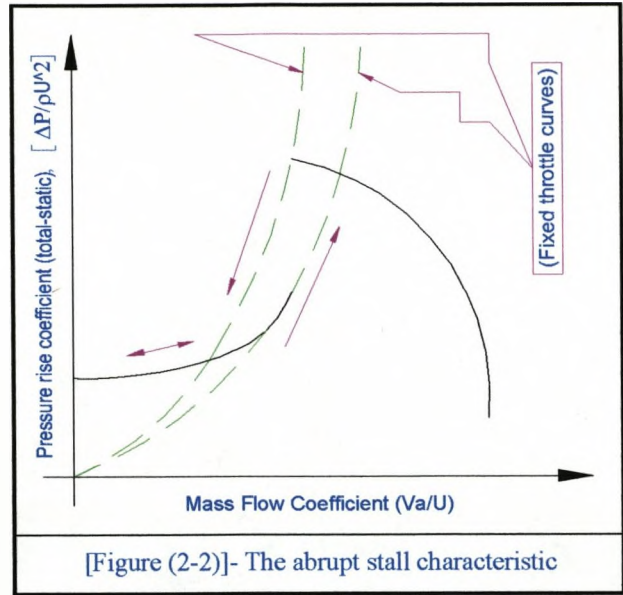
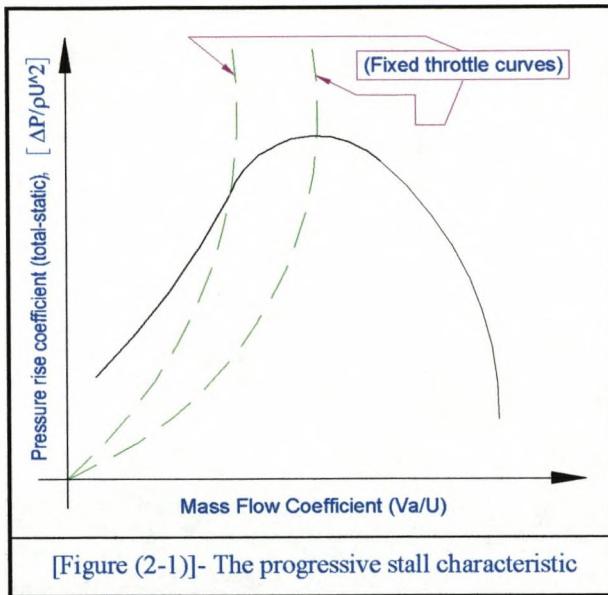
In the last stages, however, the density at the exit from the stage is much lower than the design value, so that the change in the axial velocity will be smaller than the first stage exit. Since the change of the blade speed is then the same as the first stage, the flow coefficient at part speed will be greater than the design value and the stage moves towards choke condition.

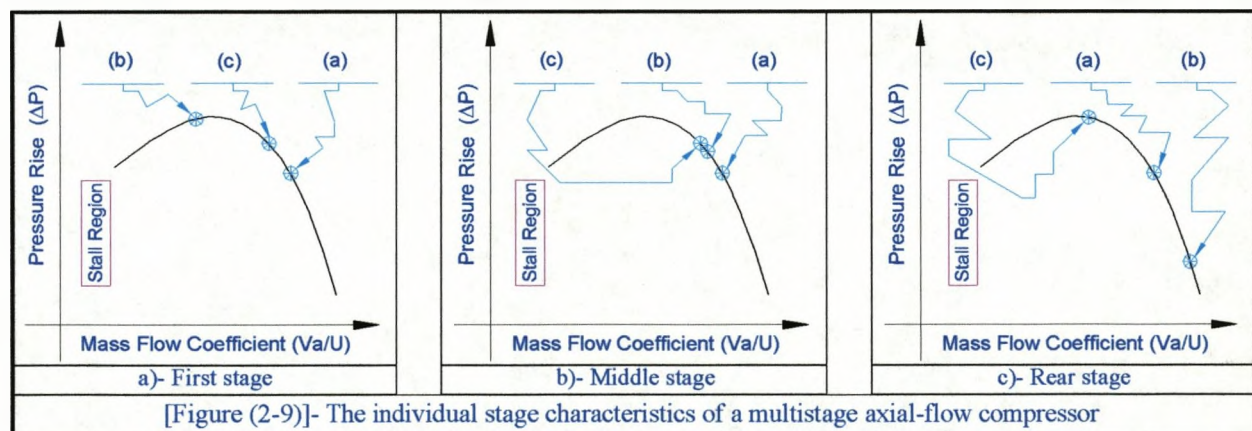
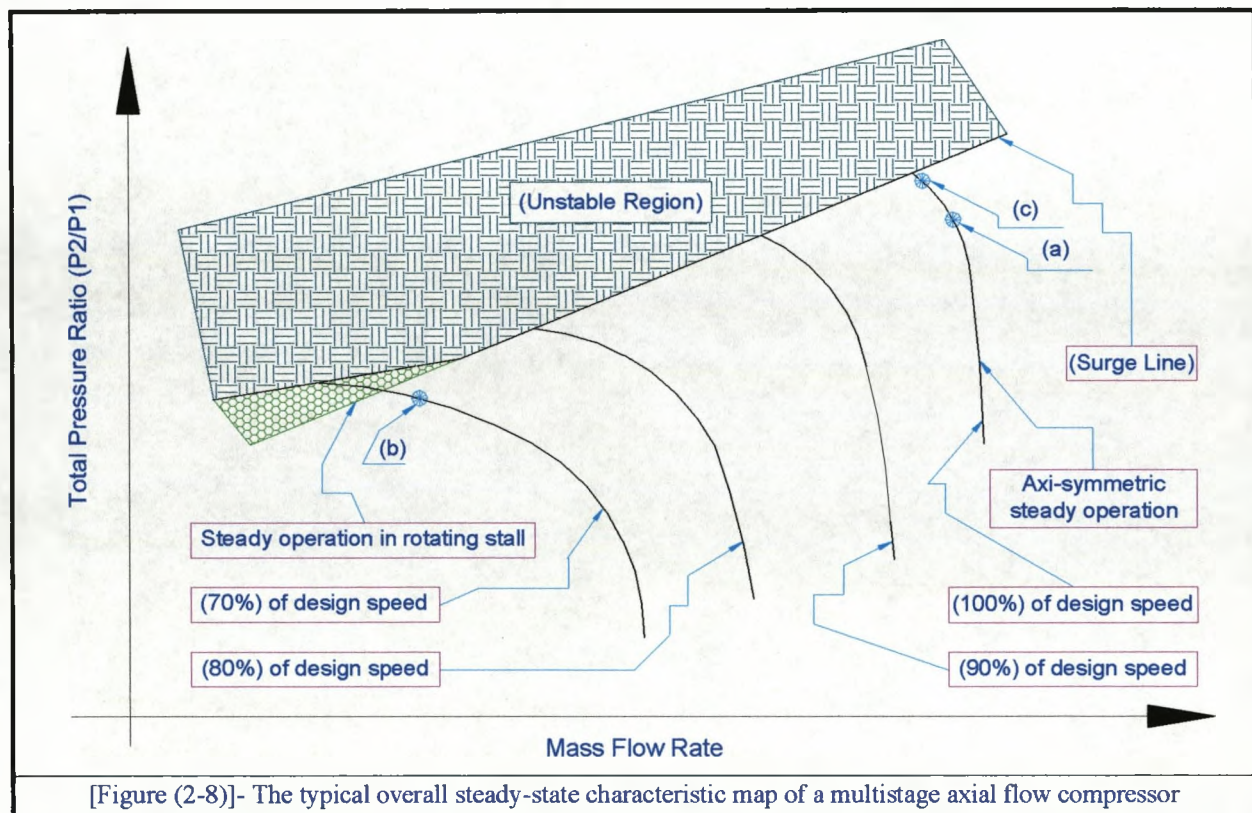
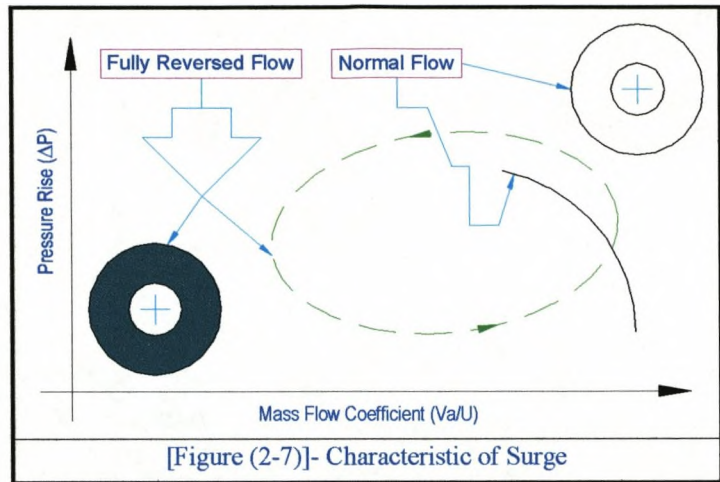
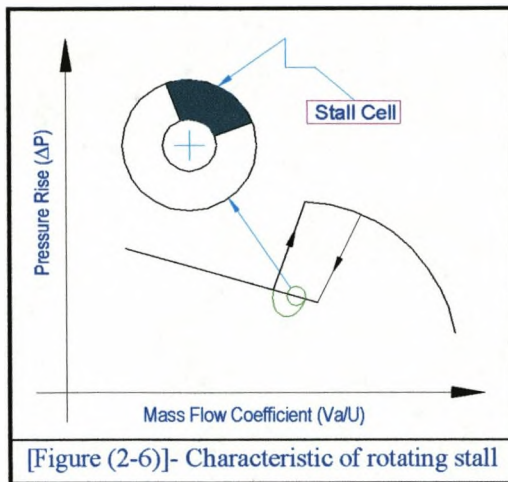
In the middle stage, the combined density and axial velocity changing produce only small changes in the operating point. Therefore in this condition, it is said that the rear stages do less work, and it means that the compressor loading is shifted to the front stages, as was implied by Kerrebrock (1992).

At the design speed however, as the pressure ratio is increased, to point (c), the front stage operates very close to its design point. The density then at the rear stage is much higher than the design value, so that the axial velocity (V_a) is much lower. In this case, the rear stages are highly loaded, and are therefore, prone to stall.

As a consequence, as the rotational speed is reduced, the flow restriction of the rear stages causes the front stage to operate close to the stalling incidence. This problem, referred to as off-design mismatch between the compressor stages, becomes greater as the number of stages in the compressor increases.

The overall characteristic of the compressor that was shown previously illustrates the reduction of running speed corresponding to movement from the design point (a) to part-speed operating point (b).





CHAPTER THREE

3)- Rotating Stall Prevention and Casing Treatment Applications:

3.1)- Introduction:

In axial flow compressors, particularly the multistage ones, a frequent requirement is to operate the compressor at flow rates less than its design value. These conditions however, cause the flow become unstable which in turn limits the stable operation of the compressor. Operating the compressor at flow rates less than the design value will in the majority of cases lead to the onset of stall.

Stall initiation commonly creates stall cells, which cause flow blockage in the compressor annulus. Experiments have demonstrated that normally, the stall cells start to rotate in the same direction as the moving blades. The rotating stall is often associated with a sudden decrease in compressor performance. This will also produce more flow instability and an increase in the vibration mode, which finally leads to fatigue fractures in the compressor blades. It has also been shown that rotating stall initiation can alter the stable operation of the compressor and thus greatly reduces pressure delivery, which leads to a corresponding decrease in efficiency.

As far as the overall characteristic of the compressor is concerned, the instability due to the stall is referred to as localised flow instability. Furthermore global system instability can occur, which ultimately causes the compressor to surge. In severe cases this can result in a complete breakdown of the flow in the compressor annulus. With the compressor working in an unstable condition, its operating life will suffer both in terms of performance and mechanical integrity. Therefore, if a compressor surges, it results in large fluctuations of the annulus flow which in time, may subsequently lead to mechanical damage of the compressor.

To protect a compressor against the dangerous effect of surge, it is necessary to delay or suppress the stall, which is a requirement for surge and may be ultimately lead to surge. If the problem of stall can be avoided, it is likely to suppress the surge. There are a lot of the compressor manufacturing companies that provide operational guide notebooks together with their products in order to describe the details of the operating limitations which are often rotating stall related. It is possible to select a compressor with a conservative instability margin but this usually involves the drawback of an enhanced cost penalty. Another approach to stall prevention is to tackle the instability problem by internally modifying the aerodynamic behaviour of the flow.

Many attempts have been made so far to implement various theories aimed at providing a better understanding of the rotating stall phenomena in order to find ways of delaying or suppressing stall. To tackle the problem of rotating stall, several different approaches either active or passive anti-stall control techniques have been used, which can be categorised as follows.

3.2)- Active and Passive Control Systems of Rotating Stall:

Experimental and theoretical investigations have shown that the limitation in

compressor operating range, caused by both aerodynamic local and global instabilities (surge and stall), can be overcome by means of some auxiliary mechanical devices. A variety of external control devices can be applied in compressor control systems in order to limit the operational range of the machine and to ensure that the compressor does not stall during its normal operating activities.

In order to achieve a greater stall free range, rotating stall, as one of the major instabilities in the compressor has to be avoided if possible. In the last forty years, considerable efforts have been made to devise means of sensing the initiation of rotating stall and actively or passively controlling the onset of it. These generally had been done by the application of pressure transducers close to the stage where the rotating stall commences.

Previous researchers however, have developed a number of active or passive stall-delaying devices. The most important and relevant of these to the present study are described below. Active control devices are normally used during transient operating of the compressor such as in starting or acceleration. For example to avoid stall below (70%) of the compressor's design speed, it is possible to use bleed valve or variable geometry techniques, which can be considered as active control systems.

There are also different types of static devices, which can be used to shift the surge line, and these can be described as passive control systems. These are commonly used in applications that require a wide range of stall-free performance in steady state operation.

3.3)- Bleed Valves:

Bleed valves are one method of delaying the onset of stall. Since the early days of the jet engine, this has been the most frequently used approach for avoiding rotating stall, which may lead to surge during start up and compressor acceleration.

Air bleeding can take place through valves positioned at the compressor exit or at the inter-stage locations. The mechanism of the bleed valve can be described by using the compressor pressure rise versus mass flow coefficients characteristic. As can be determined from this characteristic, when the mass flow is increased, the operating point of the front stages of the compressor will move to the right. This results in a delay in rotating stall inception in the front stages.

When a multistage axial flow compressor is running at lower speeds, insufficient pressure rises in the stage cause the density level of the fluid to be too low in the rear stages. This situation results in high relative velocities and also low incidence angles, which cause the rear stages to choke and limits the flow in the front stages and therefore causes them to stall. By damping some portion of the mass flow from the middle stages it is possible to avoid the onset of stall in the front stages, due to the mass flow increasing through the front stages of a multistage axial flow compressor.

3.4)- Variable Inlet Guide Vanes (VIGVs):

The variable inlet guide vanes control technique which, is normally used in the early stages of low speed multistage axial flow compressors, is another common stall delaying method. This technique works by reducing the mass flow inside the compressor, so that the angle of attack on the blades is increased. This in effect increases the flow separation from the suction side of the blades. The flow separation from the suction side of the blades will eventually lead to blade stalling. By applying this device to the compressor, the angle of the attack can be decreased as the mass flow is reduced, and hence this delays stall inception.

The (VIGVs) produce a shift of the surge line by lowering the incidence angle at which stall occurs at a lower flow coefficient. With the application of (VIGVs), the same value of inlet relative flow angle occurs at a lower value of axial velocity, and therefore the stalling incidence angle will occur at a lower mass flow rate and thus can provide a wider range of stable operation. The need for variable inlet guide vanes in axial flow compressors stems from aircraft engine requirements for engine starting, for supersonic flight and for optimising multistage performance in combination with variable stators.

3.5)- Casing Treatments:

Although a number of the different approaches have been used in order to delay stall initiation, such as variable inlet guide vanes, bleed valves, rotor blade re-staggering and variable stators, under certain conditions all of these fail to provide satisfactory results. This has led to the search for alternatives. One of the more heavily investigated and most effective methods for delaying the onset of stall, is the application of casing treatments. It was in fact the use of casing treatment in a series of experiments with boundary layer blowing and bleeding, that showed that by inhibiting rotor tip stall it was possible to delay the incipience of rotating stall, and thereby increase operating range.

Previous research on the effectiveness of casing treatment, has shown that it is possible to suppress and control the incipience of rotating stall in the most tip critical compressors by applying some modification devices to the outer casing near the tip of the rotor blades. Casing treatment research has tended to focus on tip critical compressors, because in these types of machines, stall is normally initiated near the tip regions. In addition the modification of the hub region near the tip of the stator blades can also be used to delay stall inception in situations where stall is initiated in the hub region. These are described as hub critical compressors.

The use of casing treatments as alternative passive stall control techniques was proposed initially by NASA in the early 1960s, and over the last 40 years, these techniques have been the subject of regular examination and research. It was in fact the desire to find ways of removing boundary layers from compressor blading, which led to the development of casing treatments.

As the name implies, casing treatments involve some modification to the inner surface of the solid casing above the rotor blades. The application of casing treatment has demonstrated that the flow conditions close to the stall can be stabilised and thereby produce a substantial improvement of the stall margin. Normally the treated casing partially covers the axial chord of the rotor blade tip. In most casing treatment configurations, improvements in stall margin are obtained with a corresponding reduction in performance efficiency.

The various types of casing treatments that had been applied over the past thirty years have shown that most types of these anti-stall devices can provide significant stall margin improvement in addition to increasing the pressure ratios. The following section presents a summary of the previously published investigations on casing treatment, which are relevant to the present study. Some of these researchers have adopted an experimental focus, while others have attempted to present a theoretical approach to the application of casing treatment to stall prevention.

These earlier studies looked at the effects of casing treatment on the performance of single and multistage axial flow compressors. The investigations have examined a variety of different casing treatment configurations, including slots, honeycombs, and grooves. Some

of these approaches have been shown to be more effective than others in delaying stall inception.

A study of the published literature reveals that few papers attempt to describe the theoretical principles of casing treatment. The theories that have been produced commonly suggest that airflow with a high whirl velocity at the rotor tip be forced into the treated casing. The treatment then removes the whirl energy, and the air is discharged into annulus near the leading edge of the rotor blades. The flow leaving the casing treatment to rejoin the mainstream has some remaining kinetic energy. This appears to prevent the establishment of rotating stall cells until a lower flow coefficient.

In summary it is clear that the tip casing treatment can be used in order to extend the stall-free range of axial compressor rotors. The treated casing applicability seems to depend on the reference aerodynamic conditions of the rotor tip section as well as the solid casing. The applicability of stage performance is limited to that part of the compressor map, which is dictated by rotor stall. It should be noted that some of the stall events are immediately followed by surge. The surge margin extension could also be expected from the tip casing treatment applications.

To date a variety of casing treatments types have been successfully applied to a range of industrial and aero-engine compressors. These treatments can be categorised as follows.

3.5.1)- Blowing and Bleeding Configurations Casing Treatments:

It is common in low hub to tip diameter ratio axial flow compressors to have annular rotating stall at the tip. It is thought that the tip stall is affected by the boundary layer of the casing, as well as by the loading of the blade tip in terms of diffusion factor.

In order to overcome to the undesirable tip stall, the methods of blowing and bleeding treated casings were explored in the hope of energising or removing the casing boundary layer and by these means to effect an increase in stall free range, (Koch, 1970).

The blowing casing configuration described by Koch consisted of three rows of converging-area round channels. Two of the rows discharged air ahead of the leading edge, whereas the third row discharged air over the rotor tip. These holes were oriented at an angle of (20°) to the circumferential direction and aligned to impart (30°) of counter swirl. The design conditions of the jet were such that the air would pass the rotor at a relative air angle that would approximately yield design incidence angle. The blowing configuration treated casing is shown in [Figure (3-1)]. The performance of the rotor with the blowing treatment inserted in the casing produces a shift of the stall line towards lower flows.

The bleeding casing configuration consists of a collection of honeycomb channels slanted (20°) to circumferential path in the direction of the rotor blade rotation. The honeycomb were blocked forward of the leading edge. The open cells extended from just behinds the leading edge to just beyond the trailing edge. The flow area was sized to pass (4%) of the rotor mass flow with the honeycomb choked. The bleeding configuration treated casing is shown in [Figure (3-2)]. The performance of the rotor with the bleeding treatment inserted in the casing also shows a significant shift of stall line with and without bleed flow.

A consequence of applying these methods has been the exploration of a variety of casing cavity configurations to find an optimum technique that provides wide stall free range with no or minimal impact on compressor efficiency.

An experimental investigation was carried out to examine the effects of end-wall suction and blowing on the compressor stability enhancement and stall margin of flow injection into and flow removal out of, the end-wall region of an axial flow compressor blade row

by Lee et al. (1990). A primary objective of the investigation was to clarify the mechanism by which a casing treatment could use both removal and injection of air to suppress stall in the test rig.

To simulate the relative motion between blade and treatment, the injection and removal took place through a slotted hub rotating beneath a cantilevered stator row. The overall performance data and detailed flow field measurements showed that both the flow injection and flow removal increased the stalling pressure rise, but neither was as effective as the wall treatment. The results also indicated that the removal of high blockage flow was not the sole reason for the observed stall margin improvement in casing or hub treatments, but injection could contribute significantly to stall suppression.

In summary however, there were noteworthy shifts of the stall line to lower flows by both blowing and bleeding treated casings. Meanwhile under uniform inlet conditions, the blowing casing configuration showed more effectiveness in comparison with the bleeding casing types.

3.5.2)- Suction Ring Type Casing Treatments:

Suction ring casing treatment, which is similar in some respects to vane-recessed casing treatment, has been studied by a number of previous researchers. This is a quite simple casing treatment configuration, which does not lead to an increase in noise.

The apparent stabilising effect of a steady reverse flow during operations beyond the stability limit of a gas turbine axial flow compressor resulted in the use of a suction ring casing configuration as reported by Ponomareff et al. (1948). The applied suction ring on the suction side of the axial flow compressor stage prevented the re-circulation of the reversed flow air across the rotating blade row. The results of the study also showed that the suction ring was able to eliminate pulsation of the pressure and produced a slowly rising pressure –capacity curve throughout the zone of instability.

A suction ring casing configuration has also been applied by Tanaka et al. (1975) to a single stage compressor with three different types of the rotor blades. Note that owing to the influence of rotor and stator blade interaction on rotating stall conditions, the tests were carried out without the stator and with just three different types of isolated rotors. The configuration of the applied suction ring, which was fitted to the hub by three stays bolts, is illustrated in [Figure (3-3)].

The obtained results showed that for a lightly loaded rotor, a suction ring casing treatment could improve the performance to a considerable extent. However in the case of a heavily loaded rotor blade, the casing treatment did not produce much of a performance improvement. Moreover the dimensions of the suction ring also influence the likelihood of a heavier stall, which has an undesirable effect on compressor performance.

As a conclusion, it seems that a suction ring casing technique can be used upstream of the rotor blades in order to suck the low energy flow from the tip region of a lightly loaded compressor and this will provide some improvement in stall margin.

3.5.3)- Porous Type Configurations Casing Treatments:

The casing treatments that were initially developed in the 1970s consisted of porous or slotted type treatments. These treatments are normally located over tip of the rotor, covering approximately (70%-100%) of the blade chord in axial compressors.

It is worth noting that the concept of the porous type casing treatment, which goes back to the use of outer casing blowing and bleeding techniques, has been the type most commonly

used in multistage axial compressors.

Some examination of the effects of porous casing treatment on a single axial flow compressor rotor was investigated in a series of NASA studies by Bailey et al. (1970), Ball et al. (1971) and Osborn et al. (1971), respectively. Six types of porous treated casing namely perforated sheet, honeycomb, circumferentially grooved, axially slotted, skewed slotted and blade angle slotted were fixed over the rotor blade tip in order to carry out the investigation.

Drawing of the perforated, honeycomb, circumferentially grooved, axially slotted, skewed slotted and blade angle slotted casing treatment configurations are shown in [Figure (3-4)], [Figure (3-5)], [Figure (3-6)], [Figure (3-7)], [Figure (3-8)], [Figure (3-9)], respectively. A summary of the results relating to stall margin and maximum efficiency showed that the simple perforation of the casing provided little or no change in stall margin. Meanwhile the honey comb casing provided a little increase in stall margin, but there was a significant negative impact on efficiency.

The results also indicated that the circumferential groove casing treatments produced an increase in stall margin as well as an increase in maximum efficiency. The open skewed slot treated casing provided the largest increase in stall margin, whereas, the closing slot treatments diminished the gains made in stall margin. The long blade angle casing treatment did not provide significant improvement in the stall margin, but it seemed that the reduction of efficiency was relatively high in comparison to the solid casing.

Although the objective of the early tip treatments was to find a geometry that would retard rotor tip stall and thus extend compressor stall free range, it should be realised that the range extension would occur only on that part of compressor map where the rotor initiates stall. In principle, the rotor should initiate stall at low speeds, and the stator should initiate in near design speed and above.

The effects of closed skewed slot tip casing treatment have been studied by Harley et al., (1972). The obtained results showed much poorer performance than those, which had been reported earlier by Osborn et al. (1971). It seems that two causes for the difference may be due to the influence of the stator on the stage configuration and the rotor design point approach.

Subsequently, three configurations of the porous types treated casings namely axial skewed slots circumferential grooves, and blade angle slot which, were covered approximately (70%) of the rotor blade axial chord, were applied to an axial flow compressor by Prince et al. (1974). According to the investigator's reports, the beneficial effects of casing treatment, as stall inhibitors are not dependent on high Mach number or cavity resonance.

They also added that the treated casing firstly increased annulus wall pressure gradients in the trailing edge region, as if wall stall effects are being suppressed and secondly increased also both the rotor and stage pressure rises near stall as if incipient separation is being suppressed. The obtained results had also shown that the circumferential grooves provided an increase of (5.5%) improvement in stage stall margin, whereas the axial skewed and blade angle slots yielded to (15.3%) and (15%) stall margin improvements, respectively.

The effects of the casing treatment application on a low-speed multistage axial flow compressor was studied by Wenzel et al., (1975). The blade angle slots and the circumferentially grooved casing treatments with four various inlet configurations were inserted over the rotor tip of a six stage compressor. At low speeds multistage compressors seemed to perform quite differently in comparison to single stage or isolated rotor machines, because the casing treatments were not effective in producing either stall margin

improvement or an extension to the flow range. They also observed that the casing treatments did not appreciably effect the individual stage pressure and temperature coefficients.

A study conducted by Takata et al. (1975) investigated the mechanism of stall margin improvement due to using casing treatment. This study is very similar to the one carried out earlier by Osborn et al. (1971), except for the fact that the skewed slot treated casing was tested with the slots skewed in the opposite direction. However the performance characteristics of the test which took place on a low speed axial compressor rig showed similar results to the NASA one, because the largest gains in stall margin were achieved with the skewed and axial slots casing treatments. It is also worth noting that in contrast to the previous results reported by NASA, both the circumferential grooves and short blade angle slots casing configuration in the later investigation did not produce appropriate stall margin improvements.

The effects of seven varieties of porous type casing treatment configurations on the performance of a low-speed axial flow single-rotor compressor were examined by Takata, et al., (1977). The varieties included in the study were as follows; deep skewed slots, shallow skewed slots, axial skewed slots, short blade angle, circumferential grooves, reversed deep slots and reversed shallow slots.

As it was stated from the obtained results, the effective casing treatments had strong effects on the stabilising flow near the rotor blade tip of the compressor and therefore prevented the occurrence of the rotating stall at the tip regions. In addition the investigators realised that the casing treatment influenced the flow through the blade row across almost the entire span. It was reported that the effective casing treatments displaced the rotating stall point to that where the whole blade span begins to stall simultaneously.

A teardrop insert treated casing in addition to eight variations of the skewed slot casing treatments and a solid casing were examined on an axial flow transonic fan stage with pressure ratio of (1.75) and tip solidity of (1.5) by Osborn et al., (1977). The skewed slot casing types included in the study were as follows; skewed short mid opened, skewed short mid closed, skewed short forward opened, skewed short forward closed, skewed medium forward opened, skewed medium forward closed, skewed long forward opened, and skewed long forward closed.

The results showed that the applied skewed slot casing treatments with long opened slots, ahead of and over the front portion of the rotor blade displaced the stall line to the lowest flow at the all speeds. It was also pointed out that the skewed slot treated casing with the short opened slots over mid portion of the rotor blade was equally as effective at the design speed.

The result obtained from the application of the skewed slot casing treatments also indicated that the peak efficiency of the long forward slots was (1%) and (3%) less than the peak efficiencies of the short opened slots over mid portion and the solid casing, respectively. The results also showed that for each skewed slot configurations, the improvement of the stall margin for the opened slots was better than the closed slots ones. It was also reported that although the all skewed slots casing treatment introduce reasonable stall margin improvements, in all cases the efficiency was less than that of the solid casing ones.

The teardrop treated casing in comparison with the solid casing configurations were examined on the same rig. The obtained results indicated that the pressure ratio and the efficiency of the under test compressor were greatly reduced at the design speed.

The researchers however, concluded that the simpler forms of treatments, such as

circumferential grooves, blade angle or axial slots casing treatments, could produce better results than the complex treatments like the teardrop treated casing.

The effectiveness of a skewed slot type casing treatment was investigated in a single rotor in a low speed rig by Greitzer et al. (1979). The results showed that the reduction of blade stall had little impact on stall margin gain, whereas, the gains in stall margin can be uniquely attributed to a reduction of wall stall. It was pointed out that the blade stall can be related to diffusion factor, and wall stall to the dimensionless static pressure rise over the rotor tip.

In a study by Gao et al. (1982) a circumferential groove type casing treatment was used over the tip region of the last three rotor blades of a nine-stage engine compressor. The study was carried out in order to investigate the effects of a tip circumferentially grooved casing in relation to stall margin improvement and efficiency losses in the rear stages of a high-speed multistage axial flow compressor.

Another series of tests was also conducted on a modified compressor with twisted rotor blades in the rear stages. The results showed that the casing treatment of rear stages in a multistage axial flow compressor could effectively improve the stall margin of the compressor at high speeds. The tested casing treatment had almost no effect on the efficiency of the compressor with blades untwisted, while at the inlet stage it mainly improved the performance of the compressor at low and medium speeds.

An axial skewed slot casing treatment has been tested over the tips of an isolated low-speed rotor with a hub to tip diameter ratio of (0.4), by Smith et al., (1984). The results showed an improvement in stall margin of (22%) along with an efficiency reduction at nominal clearance.

The effects of three porous type casing treatment configurations have been examined on the performance of an axial flow compressor with just isolated rotor blades and without any corresponding stator application, by Fujita et al. (1984). The study investigated the application of three types; the axial skewed slots, axial radial slots, and circumferential grooves casing treatments on both isolated rotor and single-stage compressor, including inlet guide vanes in both cases. This investigation was carried out with the emphasis on the relationship between stall margin improvement and efficiency penalties.

The results showed that in order to obtain the desired amount of stall margin improvement, a particular efficiency loss is inevitable. The research showed that for all the tested treatments, the stall margin improvement and also the compressor efficiency fell on a smooth curved line, and thus one treatment configuration was not superior to the others. The research suggested that when the tip clearance was rather large, casing treatments could effectively improve the stall margin, with a comparatively small efficiency reduction.

The stall margin improvements produced by most of the slot type treated casings suggest that in the majority of cases the improvements were accompanied by efficiency losses. They predicted that about (10%) stall margin improvement normally provided (1%) efficiency losses. It was also reported that among the three casing treatments which were used, the circumferential grooves casing configuration, offered the least adverse effects on efficiency, but improved the stall margin only to a limited extent.

The application of the baffles in the circumferential grooves treated casing has also shown a better improvement than the case without the baffles. It appears that the baffles could reduce the circumferential velocity of the flow inside the grooves and therefore increase the improvement of the treatments. It seems that this type of treatment could provide

produce some stall margin improvement without significant efficiency sacrifices.

An isolated rotor of a single-stage axial-flow compressor with and without a skewed slots type casing treatments were tested by Zhuang et al., (1985). This study also examines the initiation of rotating stall and stall margin improvement. It found that the incipience of rotating stall in the compressor was closely related to the complicated three-dimensional flow in the region near the end-wall. In addition the flow was separated from the surfaces of several blades, when the mass flow in the compressor was decreased and the blade incidences were increased to a certain extent.

The study also showed that at the off-design operating conditions, the secondary flows such as the cross channel flow, the passage vortex, the entrained flow and also the vortex induced by the entrained flow became much more serious and stronger than those under the design operating condition. It illustrated that the flow in the blade passage was effected by intense secondary flows, when it was turned towards the suction surface. Meanwhile the separation blockage regions appeared in the blade passage, and therefore the blades did not produce sufficient pressure rise to maintain the flow around them.

It was also reported that the propagation phenomenon of the separated cells in the reverse direction of rotation appeared to be the rotating stall phenomenon. In addition the periodic interaction of the flow between the blade passage and skewed slots casing treatment, suppressed the cross channel flow, broke the vortices down, blew the separated boundary layer off and also enhanced the energy of the boundary layer on the end-wall region.

The assessment of the seven various designs of axial slot casing treatment has been investigated on a low speed compressor rotor of (0.7) hub to tip diameter ratio in order to consider their effects on stall margin, pressure rise and torque penalty, (Knox-Davies et al., 1985). The results showed low levels of stall margin improvement for a range of different axial skew slot casings. It was also found that the stall margin improvements were increased with skewed angle over the (0°) to ($+60^\circ$) with a slight drop in (60°). For negative angles of skew, a stall margin improvement was still recorded, but it was found to be largely independent of slot angle. Meanwhile, all used treated casings provided some higher pressure ratios at their respective stall than the value recorded for the solid wall build.

In order to find out the efficiency penalty associated with the fitting of axial skewed slot type casing treatments, Adams et al. (1987) applied seven treatment geometry with varied slot angles from ($+60^\circ$) to (-60°), above the rotor tip of a single stage axial flow compressor test rig. Note that this experiment was conducted on the same test rig compressor, which had already been used by Knox-Davies et al. (1985).

The results showed a torque penalty, which increased linearly as the slot skew angle varied from negative through to positive angles. The authors also noted that the stall margin improvement, torque penalty and pressure rise improvement were found to vary in a near linear manner and to increase as the angle of skew of the casing treatment was increased from (-60°) through the radial to ($+60^\circ$). It was also found that as the angle of skew was reduced, the velocity of the jet leaving the leading edge of the slot was decreased and the area of re-energised flow in the blade passage moved progressively to the mid-pitch region. However, a skewed angle of ($+45^\circ$) was found to offer better stall margin improvement and less efficiency trade-off than the more commonly used ($+60^\circ$) casing insert.

A numerical investigation of an axial flow compressor associated with a slot type casing treatment has been conducted using an inviscid model with inlet total pressure gradient, by Zhuang, (1991). The applied mode provided an inlet total pressure gradient to calculate

the 3-D flow of slotted casing treatments.

A three-dimensional viscous numerical calculation was applied by Du et al. (1994) in order to explain the mechanism of extending stability of circumferential grooved casing treatment. The study focussed on the influences of the geometric parameters on improving the stall margin. The results of the study corroborated the findings of other experimental work.

An investigation of some factors affecting the effectiveness of three types of casing treatments namely axial slots, circumferential grooves and axial skewed slots has been carried out in a contra-rotating axial flow compressor stage with hub to tip diameter ratio of (0.667) and tip diameter of 0.486 [m]. The applied casing treatments have been examined for different contra-speed ratios and for two settings of axial spacing between contra-rotors by Pundhir, (1994).

The results showed that the treated casing improved the aerodynamic performance of the test contra-rotating compressor stage by providing a significant improvement in stall margin and contra-stage pressure rise. It also found that the axial slot type treated casing configuration had an adverse effect on the stage pressure rises and stalls margins for rotor-stator stage as well as contra-stage.

It was reported that the variation of total to total blade element efficiency for contra-rotors and contra-stage was improved over a wide operating range including the off-design operation when a treated casing was used. In previous research on axial flow compressors, the slot type casing treatment has been used most commonly on an isolated rotor, in order to avoid the interaction effect of the rotor blade and its corresponding stator.

3.5.4)- Separator Type Casing Treatments:

The blade-separator and air separator casing treatments are among the devices that can improve unstable characteristics of axial flow compressors. The term separator is derived from the fact that it divides the reverse flow from the normal flow at the inlet portion of the rotor blades when the flow rate of the machine is relatively small.

Both blade separator and air separator casing techniques can be inserted upstream of the rotating blades of axial flow compressors in order to improve the off-design performance. The blade-separator is installed in an ordinary casing and has been found to be useful to moderately loaded rotor blades. The air-separator device is installed in a partly enlarged casing and has proven quite effective for the improvement of unstable characteristics of axial flow compressors with relatively highly loaded rotors.

Since the stall conditions of rotors are affected by the influence of the interaction between rotor and stator, the blade separator was used upstream of three different rotor blades in a single stage axial flow compressor, in research undertaken by Tanaka et al. (1975).

The basic function of this type of treated casing is to firstly separate the reverse flow from the regular inlet flow and re-circulates it, and secondly to re-inject it to the blade-separator. In other words, as soon as the rotor blade falls into the stall regime, the casing treatment makes all blades stall and then induce a centrifugal effect.

The blade-separator treated casing has shown quite large improving effects. The most important geometrical dimension was the outlet angle of the fins. The results also showed that the blade-separator insert configuration could change the stall conditions of the rotor by reducing the rotating stall regions.

The blade-separator device had not been effective enough to serve properly the heavily

loaded of under test only rotor blades of an axial flow compressor, because it was failed of suppressing completely the rotating stall in the lower flow rate regions.

The performance of the blade-separator casing in partial flow rate region was tested on two axial compressors with rotor blades of different aspect ratios and hub to tip diameter ratios by Miyake et al. (1985). The blade-separator ring is installed in the suction duct just upstream of the rotor. The blade separator casing insert configuration consists of a ring and fins, bladed in the gap between the casing and ring as is shown in [Figure (3-10)].

The experimental investigation indicated that the blade-separator was effective for rotors of reasonable load with small aspect ratio blades, but was useless for extremely highly loaded rotors. It was also appears that the blade-separator was quite ineffective for rotors of high hub to tip diameter ratios. It was also reported that with the application of the blade-separator device, the function of the discharge was not enough and the effectiveness was reduced for a rotor with wide blades of small aspect ratio, (Miyake et al., 1986).

The second separator-type casing treatment is normally referred to as an air-separator. This is installed inside a recessed casing. The prototype of the air-separator casing configuration was proposed and patented by Ivanov, (1965). This type of casing treatment which, is installed inside a recessed casing is somewhat similar to the treatment that was originally patented by Russell et al. (1982) and Ivanov et al. (1984) and was later used by Bard (1984) under name of the unique-ring shaped stabilising anti-stall device casing treatment.

An air-separator casing treatment is located partly over and mainly upstream of the rotor blades to provide stall-free performance if possible, or to delay rotating stall initiation in the compressors. The air-separator casing treatment was composed of the casing and a separator ring, the diameter of which was the same as that of the normal part of the casing. The ring was installed within the enlarged part of the casing. Both ends of the ring make clearances with the casing which constitute suction and exhaust port of a reverse flow from the rotor.

On the ring of an air-separator casing treatment generally fins are planted on the side of the flow passage between the ring and the casing. The air-separator device is installed co-axially in the casing just ahead of the rotor. It is also possible to use either bent or straight fins to serve the requirement of the flow turning inside the air-separators.

The schematic drawing of an air-separator casing with its corresponding separator ring including the location of turning fins is shown in [Figure (3-11)]. The behaviour of flow around the rotor blade with the effect of an air-separator treated casing as a new means for stabilising the characteristics in the low flow rate region of an axial flow fan has been studied experimentally by Miyake et al. (1985).

Completely straight vanes without any camber over the rotor blades were also applied. The results indicated that at the low flow coefficients, the straight shape type vanes had prevented a smooth flow into the recessed passage and caused an overflow back into the main flow with a resulting stall inception.

Additionally, an increase in the number of the recessed guide vanes improved the treated casing performance at low flow coefficient less than (0.18). It was also reported that the stall margin improvement was decreased due to the flow blockage increasing in the recessed chamber, when the axial length of the casing treatment outlet passage was halved.

By applying an air-separator casing treatment however, the state of stall was changed. The obtained results suggested that after using the air separator casing, the full span stall cells,

which were already observed with the normal casing, disappear and a mild axi-symmetric stall was established.

The obtained results however indicated that for usual rotors of reasonable load with small aspect ratio blades, an air-separator treated casing was sufficiently effective for the purpose of improving the under test axial flow fan characteristics. But an air-separator was well effective for highly loaded rotors, if auxiliary devices such as large fins are used.

It was also noted that for the rotor of high hub to tip diameter ratio generating a nearly full span stall in small flow rate region, an air-separator insert was equally useful. But for the rotors having blades of high aspect ratio, the application of an air-separator casing was also useful.

Stall margin improvement and the various geometrical effects of an air-separator casing treatment have been studied by Miyake et al. (1986) on both highly and lightly loaded industrial axial flow fans. The air separator successfully improved the stall margin.

In the normal casing, a reverse flow from the rotor blade occurs with the onset of rotating stall. The reverse flow then, causes a strong pre-rotation in the inlet flow of the rotor and therefore, decreases the incidence angle to the corresponding blade element. The strong pre-rotation makes the static pressure high toward the casing which, finally leads to a shift of the inlet flow toward the hub. Nevertheless, when the air-separator casing is applied, the separator fins eliminate the swirl component of the reverse flow and consequently the pre-rotation flow disappears.

It was also noted that the measurement of the outward discharge flow through the blade tip to the air-separator device revealed the effectiveness of the removal of the fluid of low relative velocity which accumulates toward the blade tip as the flow rate decreases.

The performance characteristic including the geometrical effects of an air-separator casing on an axial flow fan was studied by Miyake et al. (1987). The influence of several parameters such as the axial chord of the rotor blade which was exposed to the inlet part of the air-separator, in addition to a number of the fins, height of the separator and some other parameters have been examined.

The moving blade cascade of the test machine was a free-vortex type rotor with (12) blades of a constant length. The stator vanes were removed during the experimental investigation, since they hardly affected the performance of the applied air-separator casing. The results showed that the applied air-separator casing was useful even for a rotor, which originally was associated with a heavy rotating stall. It was also noted that an air-separator treated casing device with shorter length than that of a set of rotor and stator can be applied to an arbitrary stage of multistage compressors.

The effects of the both blade-separator and air-separators casing treatments have been examined on the performance of both highly and lightly loaded rotors of a single stage axial flow industrial fan by Miyake et al. (1985 & 1986). The two casing configurations including their effectiveness associated with the hub to tip diameter ratios and the aspect ratios of the used rotor blades were compared for each case.

It was found however, that both the blade-separator and the air-separator devices were effective in preventing rotating stall and therefore, to improve the total characteristics when the reverse flow near the blade tip was fully absorbed in the separator passage. The results also showed that in fairly heavily loaded rotors with small aspect ratio, an air-separator casing was as effective as a blade-separator casing treatment. But in the more heavily loaded rotor blade fans, an air-separator was effective only, if an auxiliary device

such as large fins was used, whereas, a blade-separator treatment in the heavily loaded fans had no benefit.

Moreover, an air-separator treatment in high hub to tip diameter ratio rotor blades with a full span stall at lower mass flow was effective while a blade-separator device for the same application was useless. Meanwhile, an air separator was equally beneficial in both rotors with high and low aspect ratios.

It is worth noting that the most works mentioned above were done on the relatively light loaded fans in comparing to the heavy loading axial flow compressors which are commonly used in the aero-engines.

3.5.5)- Unique-Ring Shaped Stabilising Device Type Casing Treatments:

It is well known that the stable working ranges of both adjustable and variable pitch type axial fans are restricted by stall line. On the other hand when the system resistance is higher than the stall line, the fan's working point becomes unstable and simultaneously a shortfall in fan performance appears.

In order to avoid the risk of stalling, and to derive benefit from the complete working range of a machine, a unique-ring shaped stabilising anti-stall device was inserted into the fan casing entry of an industrial fan by Bard, (1984). The schematic drawing of the position and function of the stabilising ring anti-stall device including the positions of the turning vanes is shown in [Figure (3-12)].

The unique-ring stabiliser, containing no moveable elements, was located outside the main flow stream as shown in the previous figure. It seems that this type of treated casing does function quite similar to the other categories of the recirculation area casing treatments. It does not interface with the flow in the normal stable range of operation and it also works automatically within the early occurrence of the unstable range of the machine.

The unique ring stabilising device was integrated into entry of the fan casing. Its introduction did not affect the minimum tip clearance between the blade tip and impeller housing. The unstable performance of the fan due to stalling was also completely eliminated after inserting the unique stabilisation ring. It was realised however that with the stabilisation ring anti-stall casing, the entire impeller annular area performed under fully stable flow conditions irrespective of the chosen working point.

This was indeed made possible by the fact that radial flow vectors which become more prominent as the flow is throttled, are removed from the main flow annulus and enter the stabilising peripheral ring shaped duct just upstream of the impeller blade tips. The rotational element from the flow can be removed by means of turning vanes of the annular chamber and would be returned again into the main stream in a completely axial sense.

The result then being that the blade tip region still receives the necessary high axial flow vector and therefore continues to support pressure generation. In these circumstances the back flow does not create pre-swirl in the fan inlet.

It is worth noting that the fan performance in the working range, which was earlier stable, was not affected, since the entire stabilising ring device was located outside the main flow stream. It means that the annular chamber of the unique-ring stabilisation casing also functions only when the fan is throttled into the unstable range.

The unique-ring stabilisation anti-stall feature had initially been used in an industrial ventilation axial flow fan. A 3.4 [m] diameter axial flow fan used in an automotive wind tunnel was among the early machines which had been upgraded by the fitting of a new

inlet section incorporating the anti-stall ring.

Since the introduction of the unique-ring stabilisation casing treatment, more than (4000) anti-stall fans have been installed in variable air volume plants. They also have served quite successfully in eight normal and supplementary tunnels and industrial process ventilation fans with impeller diameters and design flow rates of {2 [m] & 80 [m³/sec]} and {4 [m] & 300 [m³/sec]}, respectively, (Bard, 1993).

The channel tunnel fans introduced double stabilisation anti-stall rings, one on either side of the impeller, providing stable flow conditions and continuously rising fan performance characteristics in both flow directions and for all blade pitches. It was found that by applying the double stabilising rings, the fan duty points occurred throughout the full extent of the fan performance range for different ventilation scenarios.

It should be noted that the casing treatment devised by Bard (1993) has some features in common with the vane-recessed configurations that will be described below. The unique-ring shaped stabilising anti-stall device did produce stall margin improvements despite the fact that it was placed exactly upstream of the rotor blades and therefore did not have any rotor blade tip axial chord exposure ratio to the recessed area.

3.5.6)- Recirculating Cavity Type Casing Treatments:

A detailed experimental investigation on stall inception delaying techniques, including the effects of varied corrected speeds under various conditions, of three types of casing treatments namely smooth casing, circumferential groove and recirculating cavity treated casing configurations were studied by Gorrell et al. (1994).

A recirculating cavity casing treatment is a combination of an axial skewed slot with a recirculation area above the slots. The schematic drawing of the applied recirculating cavity casing treatment in comparison with the circumferential groove configuration treated casing is shown in [Figure (3-13)].

The experiments were carried out on a two stage, high-speed, low aspect ratio fan that is representative of current operational commercial and military fan technology. The experimental results obtained from analysing the stage performance and the static pressure data showed that the stall inception was characterised by the formation of a stall inception cell, which grew to fully developed rotating stall.

The results suggest that for highly loaded transonic rotors the flow conditions favouring stall inception may happen on the negative, neutral or positive slope of the stage characteristic. If conditions favouring stall occur at a higher mass flow, the characteristic may be negative or neutral sloped. If these conditions occur at a lower mass flow, the characteristic may be negative or positive sloped.

It was found that the casing treatments did have an effect on the shape of the pressure characteristic at the near stall operating condition stall. It also has been noted that stall did not immediately follow once the peak of characteristic was attained.

The results also indicated that recirculating cavity treated casing similar to the smooth case and the circumferential groove treatments had a significant effect on the rotor tip flow and changed the stall inception path of the fan.

It is also worth noting that as part of the augmented of low aspect ratio fans program at the compressor research facility of Wright-Patterson Laboratory, a recirculating cavity treated casing which was made of aluminium alloy was tested in operation. Recirculating cavity casing treatment failure has been described in more detail by Kerney (1994).

3.5.7)- Active Vaned Passage Type Casing Treatments, (AVPCT):

It appears that if a casing treatment is designed in order to accommodate a higher stall margin in a compressor or fan of an engine, there is a higher likelihood of the efficiency of the engine being adversely affected.

An active vaned passage casing treatment used in an aircraft engine turbofan has been described by Hobbs (1995). This addresses the problem of providing especially large range of stall margin improvement and is primarily concerned with the degradation of efficiency of the engine in these situations. Under certain operating conditions and in different engines, it has been found that the stall margin has an adverse effect on the performance. The problem has to be solved by increasing the stall margin in certain modes of operation and decreasing it in other operation related modes.

A technique which is related to the application of an active vaned passage treated casing to a compressor, is the insertion of a bypass passage in the casing. The vaned passage is applied in order to straighten the swirling flow which was admitted into the passage adjacent to the tips of the rotor blades, and recirculated the treated flow to the mainstream in a discretely contoured outlet passage.

Ideally the aim is to be able to increase stall margin and at the same time obtain a corresponding increase in engine performance. However the inventor of the active vane passage casing has claimed that this invention can solve the problem of enhancing the stall line so as to avoid the development of an incipient stall while assuring that the engine does not incur an efficiency deficit at its normal operating condition. The partial views of an active vaned passage casing treatment illustrating a preferred embodiment of this invention when in the deployed positions are shown in [Figure (3-14)].

It was noted that active vaned passage casing treatment could eliminate or reduce the recirculating flow at engine conditions at which higher rotor efficiency and a lower stall margin are required. The reduction of recirculating flow is typically essential at aircraft climb or cruise conditions, and allowance of the full recirculating flow is necessary where additional stall margin is required, which typically is at aircraft takeoff conditions.

The active vaned passage casing treatment is a system that completely or partially eliminates the recirculating flow and modulates this flow in the passage. According to this invention the casing treatment passage duct is rendered inoperative or partially so, during certain regimes of the aircraft engine operation.

It was also claimed that the modulation of the recirculation flow of the active vaned passage casing treatment would result in increased rotor efficiency and engine thrust while maintaining stall margin at the required safety level. In other words the system of this invention would be fail safe open to insure that a safe stall margin level is always present at the aircraft's takeoff condition. The times that an aircraft engine is most susceptible to stall is when the aircraft is in takeoff mode and as it assumes a climb attitude.

The inventor claimed, this type of treated casing could be used in industrial, power plant and marine gas turbine engines, particularly during start-up or in re-ignition conditions after the engine components have been operating and changed geometry due to temperature or mechanical growth. It was found however, that such a situation could change the clearances with a consequential propensity to stall.

3.5.8)- Flow Aligned Honeycomb Type Casing Treatments, (FAHCT):

It was noted that when the compressor stall margin exhibits a high sensitivity to tip clearance or tip radial distortion, casing treatment could be considered as a potential fix.

The flow aligned honeycomb casing treatment consists of a casing insert comprised of compound angled honeycomb cells which are inclined to both the casing normal (60°) and the tangential directions as shown in the [Figure (3-15)].

Due to the particular orientation of the cells selected for favourable cell flow/tip flow interaction, this casing treatment has been given the acronym (FAHCT). The cells are oriented such that the inclination of the cell axis makes the cell opening face downstream. The cells are blocked off at the outer end by the casing wall. The cell diameter is larger than the blade tip thickness. Khalid (1997) reported that the guidance on the selection of cell diameter relative to the blade thickness was obtained from the radial honeycomb configurations.

The length of each cell as tested in a three-stage research compressor was four times the cell diameter. The cell inclination in addition to imparting a degree of flow alignment results in sufficient cell depth in the limited radial space of multistage configuration. The experimental results achieved from testing both flow aligned honeycomb and circumferential groove casing treatments on the same test rig have shown (19%) and (9%) stall margin improvements, respectively. It means that the test incorporating the casing insert comprising the compound flow aligned honeycomb demonstrated up to (10%) higher stall margin than the circumferential grooves casing treatment.

3.5.9)- Circumferential Spaced-Apart Grooves Type Casing Treatments:

This type of the casing treatment is applied to turbine engines and particularly to both axial flow and centrifugal compressors, and was invented by LeBlance et al. (2000). A compressor with this casing configuration has an inner surface defining a plurality of circumferentially spaced-apart grooves adjacent the tip of a rotor blade cascade. With the application of such a casing treatment the compressor has a shroud which surrounds the radial extending blades of the rotor assembly.

Each blade comprises leading and trailing edges, a tip and the shroud that has an inner surface defining a plurality of circumferentially distributed parallel groove adjacent tips for allowing air to flow locally over the blade tips from high pressure to low pressure sides. Each groove has a depth in a radial direction, a length and a width. The depth is less than half width along all length.

It worth noting that each groove has opposed leading and trailing ends. The leading end being disposed substantially downstream of the leading edges of the blades, whereas the trailing end is disposed substantially upstream of the trailing edges of the blades. As the inventor has noted, an aim of the invention was to provide a compressor which could be adapted in order to improve stall/surge margins under normal operating conditions and therefore, to improve airflow characteristics of the compressors.

The schematic drawings of the geometry and function of the circumferentially spaced-apart groove type casing treatment including the position and form of the grooves is shown in [Figure (3-16)]. It was noted however, that this low aspect casing treatment had been applied successfully to an axial flow compressor and a centrifugal flow machine as well.

3.5.10)- Vane-recessed Type Casing Treatments:

The vane-recessed casing treatment originates from a patent by Ivanov et al. (1984). It was initially applied to an industrial two-stage axial flow fan in order to improve the fan performance by delaying the initiation of rotating stall. The patent contains few guidelines regarding the casing treatment geometry.

In this device a stepped casing was used to replace the conventional outer solid casing of the machine. The replacement casing was extended partly over and mainly upstream of the rotor blades. Some fully extended curved vanes were also used in the recessed portion of the casing treatment.

The present author has yet to find any evidence in the open literature of the application of a vane-recessed casing treatment to a multistage axial flow compressor. It should be noted that the work done by Ivanov et al. (1984 & 1989) was carried out on a double stage axial flow ventilation fan. It seems that the treated casing applied by Ivanov, et al. was somewhat similar to that of Russell et al. (1982), but significantly differed in using radial rotating extended vanes at the inlet of the casing treatment.

The overall configuration in addition to the general pressure rise versus mass flow characteristic and curved vane geometry proposed in the casing treatment by Ivanov et al. is shown in [Figure (3-17)]. The stepped casing however, consists of a recessed casing with a larger diameter than the original casing. A shroud is located co-axially in order to divide the inlet flow passage and the re-injection flow portion. The inlet diameter of the recessed casing and the outlet diameter of the inner ring form an annular chamber to bypass the stalled flow.

The annular chamber has an inlet and an outlet passage. The inlet passage is the space between the shroud and casing treatment rear wall, whereas, the outlet passage is the space between the forward wall of the casing and the inner ring. A numbers of guide vanes are positioned inside the annular chamber and these extend from near the rearward shroud end-wall to the forward wall of the casing treatment.

The operating principle of a vane-recessed casing treatment that could be applied to either a single stage or multistage axial flow compressors seems to be as follows. As the compressor operation moves toward the stall line, some unbalanced radial flow cells or in the more severe cases some unbalanced radial reverse flow is transported towards the blade tip where, the recessed casing can collect it. The collected flow is then turned by means of the vanes, which are located inside the recessed casing to the desired direction.

The flow inside the vanes is then re-injected in a nearly axial direction into the mainstream at the upstream side of the rotor blades. It is worth noting that in order to increase the fan pressure during the stall conditions, it is essential that the stalled flow can leave the annular chamber in a direction counter to the direction of the rotor blade rotation.

It appears that the radial flow components, which become more prominent as the flow is throttled, are removed from the annulus and immediately enter the recessed area of the casing treatment and then returns by means of the cambered vanes to the mainstream. The vanes remove the swirl from the stalled flow and then reintroduce it in a nearly axial direction to the main flow on the upstream region of the rotor blades. As a result the compressor rotor blades at the tip region still receive the essential high axial flow and also the reverse flow does not create pre-swirl flow in the compressor's inlet.

Ivanov et al. (1984) claim that this type of casing configuration could be used in either single-stage or multistage fans and also in axial flow ventilation fans with stable pressure characteristics over a wide range of air or gas delivery.

Of all stall-delaying techniques it would appear that the vane-recessed casing treatment could offer the best potential for stall margin improvement along with an acceptable level of performance efficiency sacrifice. Another advantage of the vane-recessed casing treatment suggests that in the majority of cases, when the compressor is working at off design conditions, the blade tip region still receives the necessary high axial flow vector

and therefore continues to support pressure generation.

The influence of a vane-recessed treated casing on the stall margin of a single stage axial flow compressor with hub to tip diameter ratio of (0.5) and blade loading of near unity at the rotor hub, has been investigated by Azimian et al. (1987). This examines the effects of an applied vane-recessed casing in association with different axial positions related to the rotor blade axial chord.

The schematic drawing of the applied vane recessed casing insert including the geometry of two partially curved and partially straight adjacent vanes are shown in [Figure (3-18)]. The results indicated a significant increase in the flow range. The study also found that the characteristics of the compressor with the application of the vane-recessed casing at different axial positions were better than the performance characteristics obtained from the solid wall casing.

Moreover, the best result in terms of stall margin in stage configuration was achieved from the treated build with (35.5%) rotor tip axial chord exposure ratio. The improvement of stall margin in this case was (32.2%) with (10.7%) drop in peak efficiency. It was however found that when the applied casing treatment was successful in delaying the onset of rotating stall in the rotor, the stall instead developed in the stator. From this it can be concluded that a significant efficiency penalty will be incurred in terms of compressor performance.

A further study carried out by Azimian et al. (1990) on the same axial fan stage test rig indicated that the stator hub precipitated the stall. To avoid this constraint the applied vane-recessed casing with (48) and (24) number of blades was inserted on a build consisting of the rotor but without the stator blades.

By applying the vane-recessed casing treatment to an isolated rotor configuration, it was found that the areas of the flow blockage near the rotor tip, which were observed in the solid wall, were removed. It was also noted that in this case the regions of low axial velocity at the rotor hub were increased as mass flow was reduced. It was found that without stationary blades, there was also some downstream hub stall as well.

The results showed that the applied treated casing on the isolated rotor provided considerable improvement in stall margin. In addition among various casing related parameters, the effect of the axial exposure over the rotor chord was found to be very significant. It was found that treated builds with more than (50%) of the axial chord exposed to the recess provided the best results. In comparison to the solid casing, no penalty in efficiency was noted for the optimum build arrangement. Halving the number of vanes inside the casing chamber also provided a further modest improvement.

Over the various modified configurations examined, it was found however, that the build with (67%) rotor exposure ratio and using an aerofoil-section in order to reduce the diffusion inside the recessed chamber provided an optimum build with (60.8%) improvement in stall margin.

An experimental investigation has been carried out on the influence of a vane-recessed casing treatment on the stall margin improvement of a single stage axial flow fan with three different hub-to-tip diameter ratios of (0.5, 0.7 & 0.9) by Ziabasharhagh et al. (1992). The compressor rig of (0.5) hub to tip diameter ratio that was used in this present research, was the same as that used by Azimian et al. (1990).

The investigation on the higher hub to tip diameter ratio was undertaken on another single stage low speed axial flow fan with the same tip speed and tip diameter. The compressor

could also be fitted with different blades in order to conduct tests with either (0.7) or (0.9) hub to tip ratios.

Three different sizes of vane-recessed casing treatments, namely large, medium and small were applied to the two fan builds with three different hub/tip ratios. The large casing was the same, which had already been used by Azimian et al. (1990). The dimensions of the medium and small treated casings were scaled in proportion to their rotor blade heights.

These three different treated casings however, had the same recess vane profile and general configuration but had different overall dimensions. It was noted that the large casing treatment design was broadly based on the ideas of Ivanov et al. (1984). The dimensions of the medium and the small casing treatments were scaled to (60%) and (20%) of the large one, respectively.

The results showed that the scaling of the medium and small treatments in proportion to the large casing was not particularly effective. It was suggested that scaling in proportion to the blade chord might have produced better results. However since this was not tried in practice it is not possible to be certain whether this would have been the case. Moreover it is not certain that Azimian's original designed casing treatment provides an optimum configuration.

The effects of the vane-recessed casing treatments developed by Azimian, et al. (1990) and Ziabasharhagh, et al. (1992) has been discussed in more detail by McKenzie, (1993). He suggests that the axial length of the gap between the shroud and the rotor blade leading edge, $[(G_{TC})_{Axial}]$ should be about (10%) of the rotor chord.

McKenzie (1993) also stated that the number of vanes inside the recessed chamber did not appear to be critical but should provide an overall space to chord ratio in the region of (0.5). He also highlighted the benefits of including an aerofoil or teardrop section in the inner vane shroud.

Roy et al. (1994) investigated the use of a vane-recessed type casing treatment in a low speed contra-rotating axial flow fan. This was intended to remove the casing boundary layer from near the second fan tip and inject the flow back into the mainstream near the trailing edge of the first fan tip. The test rig was a unit fan with low aspect ratio blades. The applied treated casing performed on a principle of automatic suction and injection, due to the static pressure gradient existing between the two axial sections.

The schematic drawing of the contra-rotating fan unit including the geometry position of the casing treatment with its turning vanes is shown in [Figure (3-19)]. The results showed that the applied treated casing provided an increase in the performance of the contra-rotating axial flow fan unit. It also affected the radial profile of the stagnation pressure. The flow circulation in the vane-recessed chamber was found to be between (13% to 18%) of the mainstream mass flow.

It was suggested that this amount of flow circulation was essential for efficient boundary layer control for the applied contra-rotating fan. In the end they concluded that the amount of the mass flow which was circulated to recessed chamber was too high.

In order to examine the influence of the vane-recessed casing treatment on stall margin, operating efficiency and flow field, an experimental investigation was carried out by Kang, et al., (1995)] on a low speed axial flow fan with aerospace type blade loading. The applied fan test rig was the same as that used earlier by Azimian et al. (1990), although a new computerised data acquisition system was added. Three different geometrical designs of the vane passage treatments were examined. The sketch of three different vane-recessed

casing configurations is shown in [Figure (3-20)]. The first configuration comprising (24) vanes was the same treated insert, which had been designed and used by Azimian et al. (1990) and subsequently applied by Ziabasharhagh et al. (1992).

The second casing configuration involved two major modifications. Firstly the airfoil section inner ring which had been used by Azimian et al. (1990) was simplified to a (30°) wedge crowned by a 10 [mm] radius circular arc. Secondly new vanes were applied to the whole recessed chamber length. The applied vanes consisted of a radially placed straight part and a radially cambered portion. The axial length of the recess was approximately (7%) shorter than that of the first configuration.

The third configuration consisted of an outer casing with (24) slightly curved vanes and an inner ring of substantial cross section. The curvature radius of both inlet and outlet regions of the vanes was increased from those used in second configuration. The inlet part of the vanes (section A-A) was cambered to (55°) with respect to the circumferential direction in order to reduce the flow incidence to the vanes. The leading edge of the inlet portion was cut back by (18%) of the recess height to avoid additional noise. The outlet part of the vane had a (90°) outlet angle (section B-B), in order to re-inject the recirculated flow into the mainstream in the radial direction. Each vane was inclined at an angle of (25°) with respect to the radial direction.

The results showed that more than (65%) of stall margin improvements and over twice pressure rises at stall conditions were obtained by applying the three different geometrical designs of the vaned passages. Applying all three treated casing configurations significantly altered neither the peak efficiency nor the efficiency in the extended unstalled range.

It was reported that the different vanes used in the recessed chambers had only a marginal effect on the overall performance of the fan and the different vane passages made a major difference to the stall margin but had little affect on pressure rise and efficiency characteristics. It was also noted that reducing the incidence or increasing the total pressure in the upstream tip region did not significantly effect the stall margin. The best result however was achieved from the first casing configuration with a (67%) improvement in the stall margin.

An experimental investigation had been carried out into the influence of a vaned recessed casing treatment on the performance of an industrial type axial flow fan with a hub to tip diameter ratio of (0.4) by Hill et al., (1998). The fan tested consisted of a single axial flow rotor with a tip speed of 76 [m/sec], tip diameter of 0.5 [m] and fixed nominal speed of 2910 [rpm]. Two rotors were used, both equipped with six blades whose setting angle could be adjusted. The first rotor set was a standard airfoil-type blades with 104 [mm] chord, whereas, the second was a set of fully reversible blades of chord 110 [mm].

A vane-recessed casing treatment with a set of (24) vanes and similar geometry in vane height, shroud length, and outlet passage axial length to that which had already been used by Azimian, et al. (1990) have been applied to the both rotor blade sets described above. Note that this treatment differed from that used by Azimian, et al. in the axial length of the inlet passage.

Fans with both sets of blading showed good improvements in flow range, compared with the untreated build for where rotating stall was normally presented on the characteristics of (19° , 27° & 32°) blade setting angles. The results showed that with the exception of (27°), a small reduction in efficiencies were noted in the characteristics of the normal operating range. The test results obtained from both sets of blading at various rotor blade axial chord

exposures indicated that the optimum benefits were achieved with an exposure of around (60) percent. It was also reported that an increase in power was required in the extended flow range and hence rises in work inputs, for the treated casing builds.

3.6)- Some Facts about Recirculating Area Type Casing Treatments:

The majority of the casing treatment configurations, which have already been discussed above, could also be described as recirculating area type casing treatments. This refers to those treatments, which operate on the basis of stalled flow being injected into the recess chamber, and then re-circulate the flow in the space between the vanes. Here the swirl is removed from the flow, so that the static energy is increased, and the flow is then re-injected back into the mainstream.

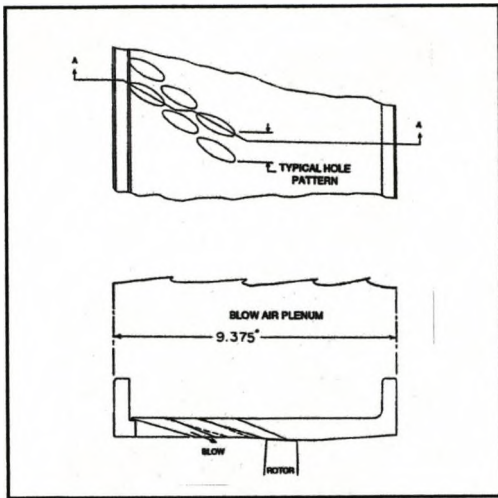
The present author has coined the term, "recirculating area type", in order to describe this mode of operation. The following section highlights some additional research, which influenced the design features of the casing treatment developed as part of this investigation. The first of these phenomena to be considered are breakaway instabilities.

These instabilities are less important in radial fans, because in low-speed, low-pressure fans the characteristic curve is steady up to zero volume flow. In the case of high-pressure radial fans with a pressure factor greater than (0.9) and circumferential speeds greater than about 150 [m/sec], there is a clear limit for the breakaway of flow in the characteristic curve. It is at about half the design volume flow. The pressure jump at the stall point is about (10%) of the total pressure increase.

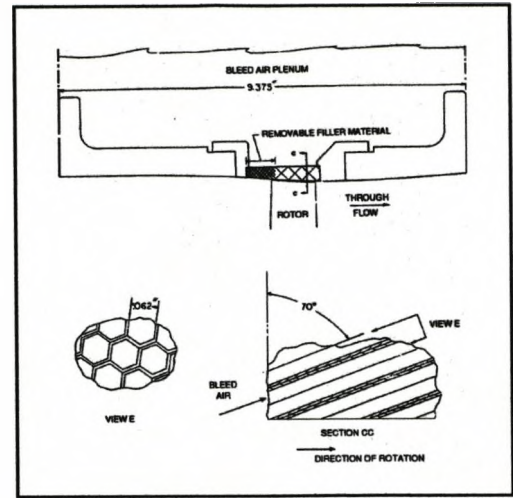
According to Henssler (1977), breakaway instability is much greater in axial flow fans, the stall point lies at about three-quarters of the design volume flow. The higher the load of the blading, the greater is the relative pressure drop to the left of the stall point. Henssler's application of a conical inlet part and bypass duct, which were used in order to locally stabilise the back-flow and also suppress the rotating stall in an axial flow fan could be considered as one of the earliest works regarding the application of a stepped recessed casing insert configuration. The stabilising device, which was used to provide stable flow characteristics without unstable operating zones, performed the same function as a recirculating area casing treatment. The stabilising device used by [Henssler] is shown in the schematic drawing contained in [Figure (3-21)].

Another of the earliest documents describing a recirculating area type casing configuration is the patent of Russell et al., (1982). They used a radial rotating extended vane at the inlet of the applied recessed casing treatment. This device is particularly suitable for use with axial or mixed flow blowers or suction fans in applications where a more stable operating performance is required. The anti-stall casing treatment device which had been invented by [Russell, et al.], in order to provide a stall-free axial flow fan had an impeller with extending blades enclosed in a tubular casing as is illustrated in [Figure (3-22)].

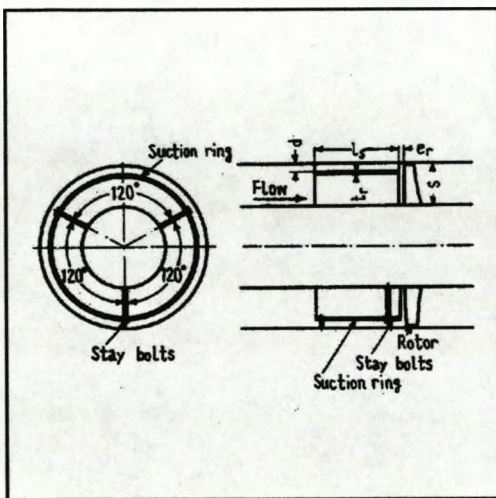
As can be seen from the drawing, the tubular casing is divided into an inlet portion, an intermediate portion of larger diameter than the minimum diameter of the inlet portion, and an outlet portion of a diameter no longer than the minimum diameter of the inlet portion. The impeller is also positioned with the leading edge of each blade protruding into the intermediate portion and the trailing edge protruding into the outlet portion. The inlet portion however, may contain radially extending vanes rotating about a spindle and extending between the stator and the inner wall of the inlet portion. The intermediate portion also may contain a fixed vanes arranged radially of the stator and axially of the casing.



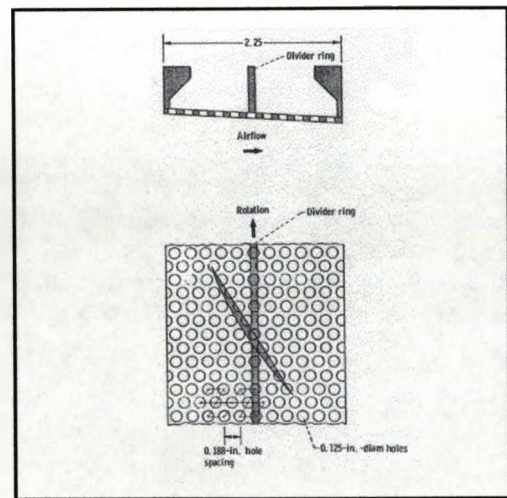
[Figure (3-1)]- The blowing configuration casing treatment



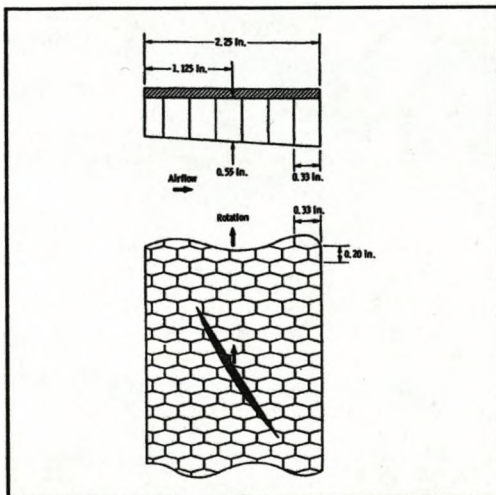
[Figure (3-2)]- The bleeding configuration casing treatment



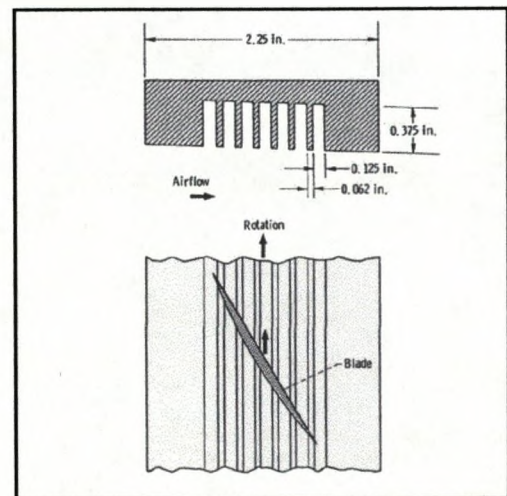
[Figure (3-3)]- The suction ring type casing treatment



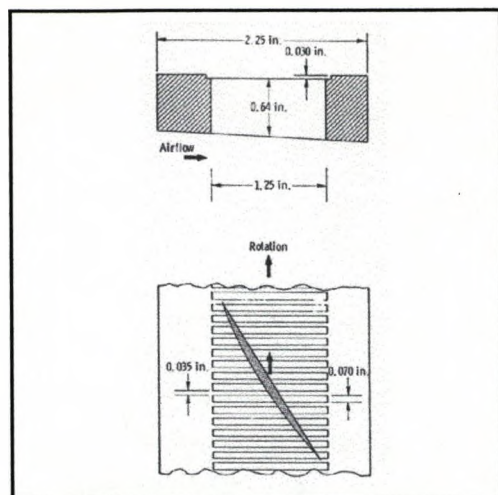
[Figure (3-4)]- The perforated type casing treatment



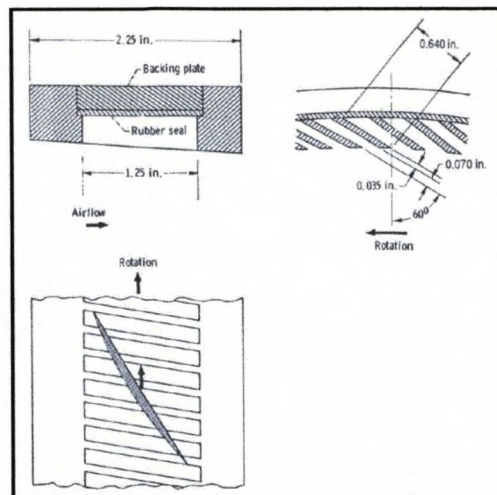
[Figure (3-5)]- The honeycomb type casing treatment



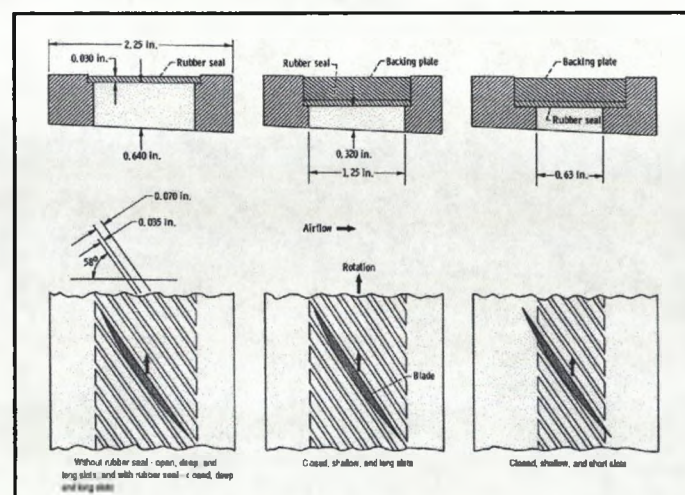
[Figure (3-6)]- The circumferentially grooved type casing treatment



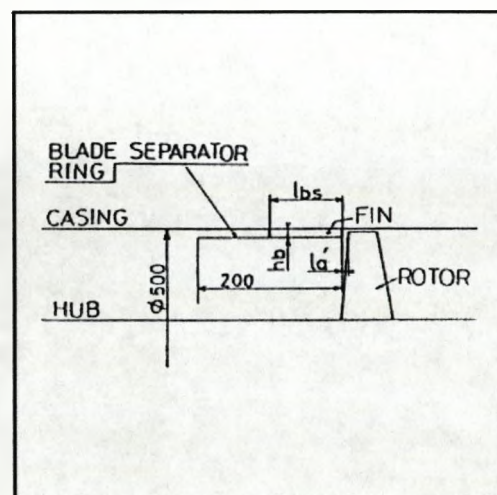
[Figure (3-7)]- The axially slotted casing treatment



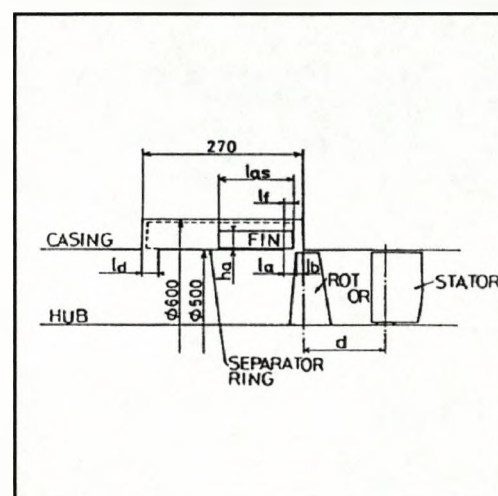
[Figure (3-8)]- The skewed slotted-type casing treatment



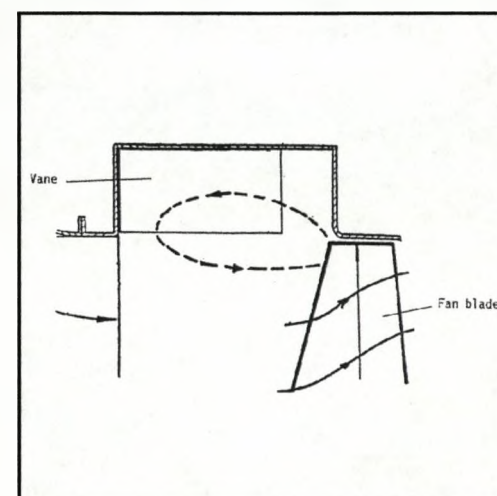
[Figure (3-9)]- Three configurations of the blade angle slotted type casing treatment



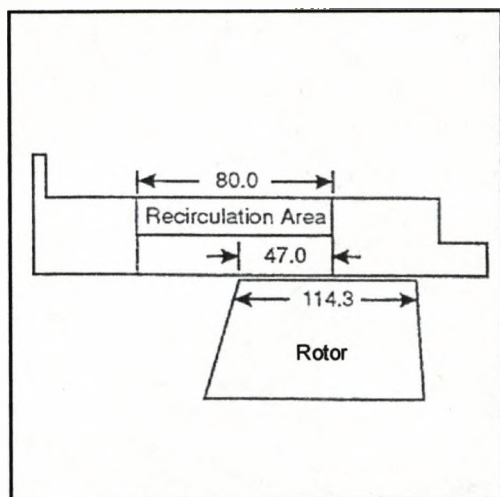
[Figure (3-10)]- The blade separator type casing treatment



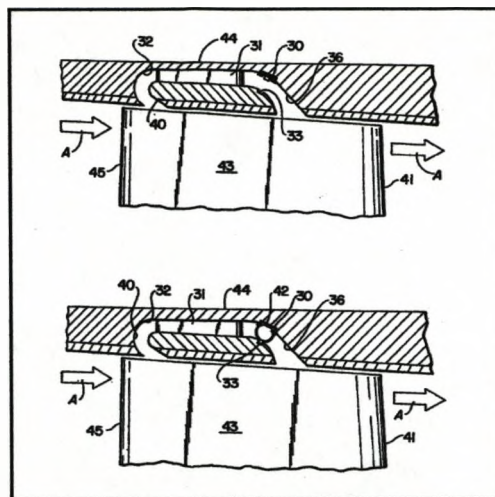
[Figure (3-11)]- The air separator type casing treatment



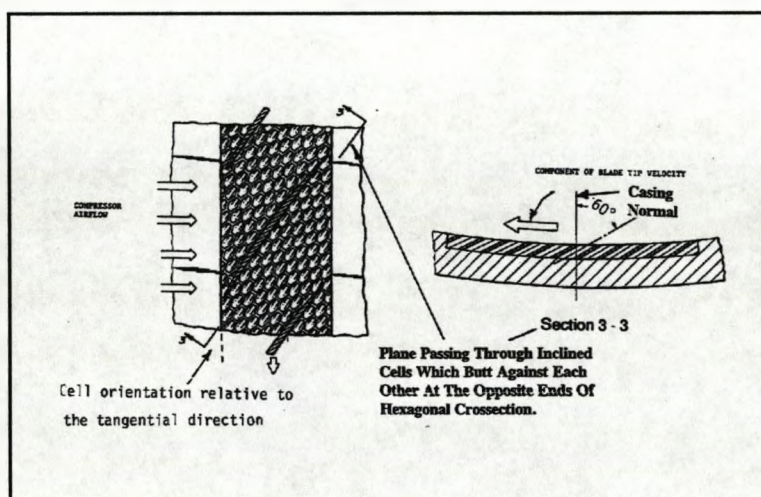
[Figure (3-12)]- The unique ring stabilising device casing treatment



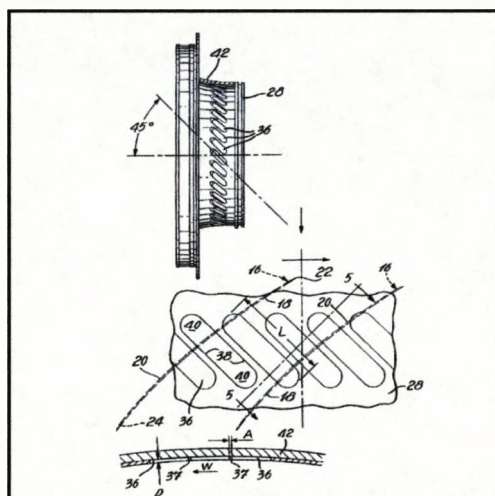
[Figure (3-13)]- The recirculating cavity type casing treatment



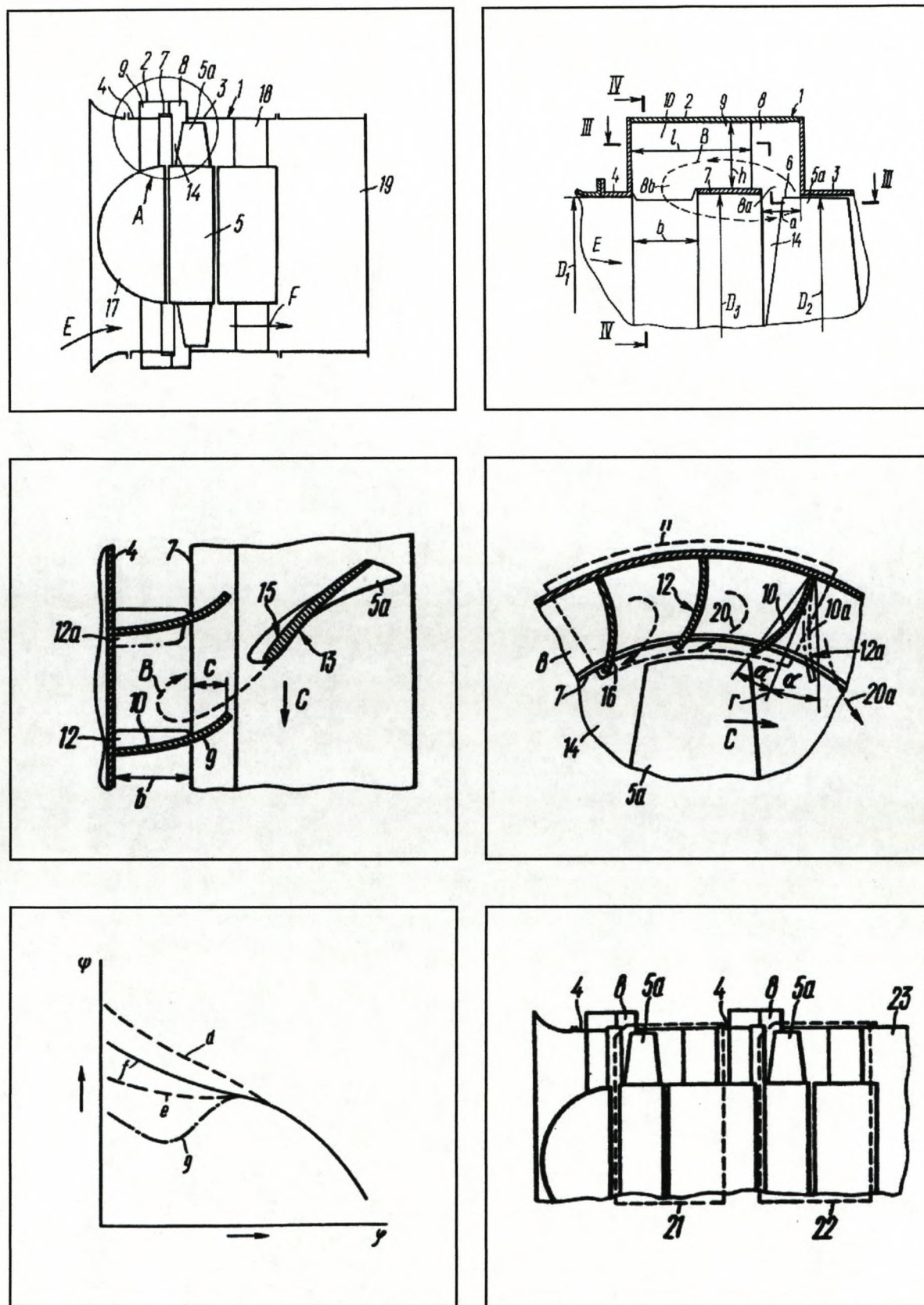
[Figure (3-14)]- The active vaned passage type casing treatment



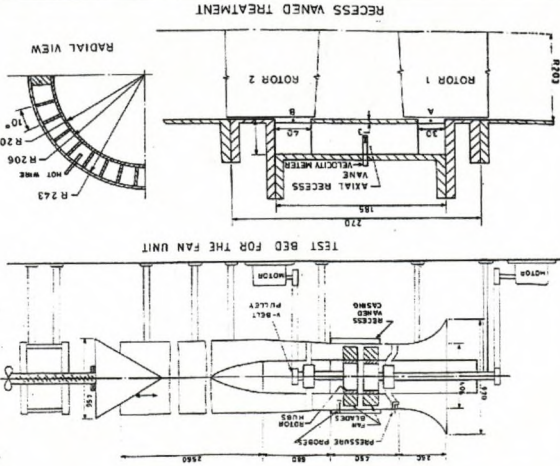
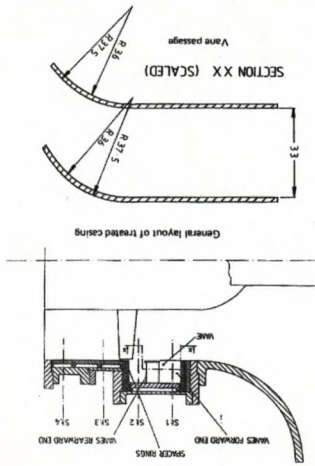
[Figure (3-15)]- The flow aligned honeycomb type casing treatment



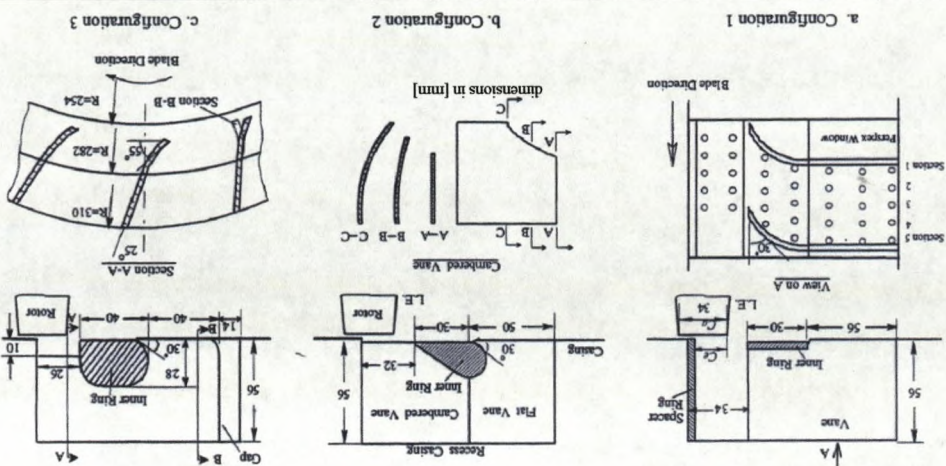
[Figure (3-16)]- The circumferentially spaced-apart groove-type casing treatment



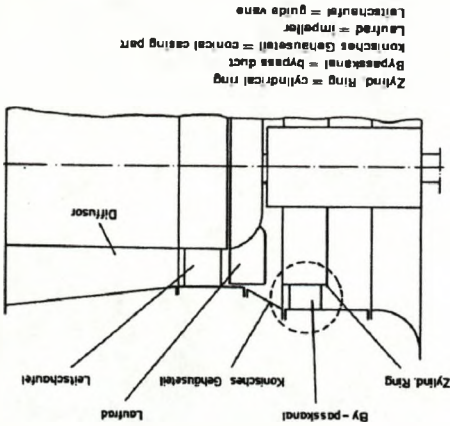
[Figure (3-17)]- The overall configuration and the general pressure rise versus flow characteristics including the curved vane geometry proposed by Ivanov et al. (1984)



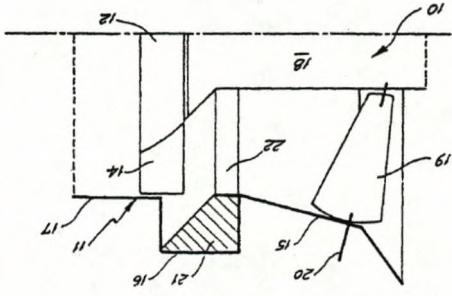
[Figure (3-18)]-vane-recessed casing treatment with partially curved vanes in a twin-rotor contra-rotating axial flow fan [Figure (3-19)]-The recess vaned treatment used



[Figure (3-20)] - The schematic drawings of three different vane-recessed type casing treatments



[Figure (3-21)]- The stabilising device configuration, (Henssler, 1977)



[Figure (3-22)]- The anti-stall casing treatment, (Russell et al, 1982)

CHAPTER FOUR

4)- Test Rig Facilities and Associated Instrumentation:

4.1)- Introduction:

As it was stated, the objective of the current research is to investigate the prevention or delay of the onset of non-linear aerodynamic instabilities, which lead to the fluctuation of the pressure and mass flow rate. This idea can be achieved via the application of an anti-stall passive technique into a compressor in order to improve the stall margin whilst ensuring a minimum sacrifice in performance efficiency.

These have to be considered in a multistage axial-flow compressor through applying a stepped casing passive stall control device as a stall onset delaying technique, which is called vane-recessed tubular passage casing treatment. Additionally the purpose of the present study is to evaluate any effects on multistage compressor performance associated with the design of a stepped casing that would be replaced by some part of the test compressor rig's conventional casing.

This implies that the rig needs to be capable of initiating the rotating stall phenomenon in off design conditions, particularly when the mass flow rates are low, in order to invoke such situations, which will lead to improvements of the compressor performance such as stall margin.

To attain the aforementioned objectives, the preliminary design and manufacturing of an appropriate vane recessed casing treatment was essential. The treatment design features will be considered in the next chapters, but the test rig, instrumentation and measurement techniques applied in this investigation will be described below.

The following introductory notes concern the operation, design features and construction of a low speed compressor, which has many features in common with high-speed machines as well as affording convenient ways of varying the normally fixed angles and spacing.

4.2)- Test Rig and Experimental Work Apparatus Facilities:

The experimental research test rig that was used for the investigation consisted of a three-stage compressor along with associated components and instruments, which are hereafter, called the (Peregrine Compressor Test Rig). This rig was supported on a special table and legs and when fully set up combined the followings:

- 1)- An electric motor.
- 2)- A bell-mouth inlet.
- 3)- A multistage (three stages plus inlet guide vanes) compressor.
- 4)- An outlet ducting and two flow straighteners, including honeycombs.
- 5)- A venturi flow meter, incorporating a (5°) angle diverging diffuser.
- 6)- An orifice (electrically operated).
- 7)- A rectifier/transformer

- 8)- A bank manometer.
- 9)- A rotational speed measuring device.
- 10)- A torque measurement transducer (load cell) device.
- 11)- Four temperature measuring (thermocouples) devices.
- 12)- A Cobra type total pressure probe.

The last five measuring devices in addition to a computer augmented the existing apparatus, in order to provide an appropriate data acquisition system. The general layout of the complete research facility rig which was employed for the current investigation is shown in [Figure (4-1)]. The entire rig was designed by Messrs Peregrine and Partners, Consulting Engineers, Royston, Herts, England, and manufactured by David Budworth Ltd. of Harwich, England.

4.3)- Electrical Motor:

The test rig's direct current (DC) motor had been designed and manufactured by English Electric Ltd. During operation, the 230 [volt] motor, fed from a Westinghouse silicon rectifier/transformer, developed 15 [hp] at 3000 [rpm]. The shaft of motor was directly connected through a flexible coupling to the compressor shaft protruding through the compressor bell mouth.

The starting of the motor in addition to the speed regulation was controlled from a steel cabinet incorporating voltmeter, ammeter, faceplate starter and four hand-wheel operated variable resistors for speed regulations. Two resistors in series with separately excited shunt field are used for coarse speed adjustment and two other resistors for the fine and extra fine adjustments.

The motor was provided with trunnion bearing (self-aligning) on pedestal supports, to which air bearings were fitted. The two bearing pedestals were aligned with the compressor shaft and were also equipped with self-aligning ball races whose function was to ensure alignment of each bearing. This construction permits the stator of the motor to swing freely. Hence, any applied torque can be measured with a reasonable degree of accuracy.

The electric motor however must run, when the air is passing through the fitted bearing. Air is controlled by a needle valve after passing through a filter and is usually supplied at 760 [kpa] into air bearings.

4.3.1)- Rotational Speed Measuring Device:

A 15 [hp] variable speed, direct current (DC) electrical motor was driven with a series armature regulation, which could be adjusted to vary the speed between 100 to 3600 [rpm]. For measuring the motor speed a stroboscopic device was employed. This consisted of a thin disc fitted with (60) equal spaced holes drilled around the circumference on a common diameter mounted onto the motor's shaft. A small electric bulb shine intermittently as the disc holes sweep past it, thus generating impulses, which are picked up by a photocell connected to a simple digital counter.

The impulse generator was fitted with an alternating current (AC) supply, thus enabling the accurate determination of the running speed to within one [rpm] at any time. Given that the rig rotational speed was subject to small but detectable fluctuations about the nominal speed in a typical range of ± 5 [rpm] it was found that the stroboscopic method suffered a time lag which prevented an accurate real time measurement of the speed.

To overcome this limitation a fast response hand held tachometer was employed. An added advantage of obtaining a more accurate running speed lay in the fact that the accuracy of the torque magnitude calculation was directly related to the exact evaluations of the speed as is detailed in the following chapter.

4.3.2)- Torque Measuring Device and the Relevant Calibrations:

Since the rotor turns clockwise, when looking towards the compressor, the torque reaction tends to impart anticlockwise rotation to the freely mounted stator. The torque was measured by finding the load required for balancing the electrical motor's stator. A torque arm, which was fixed onto the motor's stator from one side, was connected to a load cell from another side. During the operation, the effective load on the stator was digitally displayed in [millivolts].

The calibration of the load-cell was essential to adjust the digital indicator in order to display the correct amounts carried out by suspending known weights from the load-cell. The torque magnitude however can be determined by multiplying the length of the torque arm by the applied load. This will be discussed in more details in next chapter.

A state of balance can be achieved by adding the correct weight increments to the scale pan. Removable weights were suspended from the front side of the motor at a radius of 254 [mm] from the axis. The fixed weight on the back then gives initial balance with the load carrier in place.

The calibrations of the load cell have been done both on and also apart from test rig. In order to find out the applicable effective load and torque to the electrical motor, the calibration of the load cell was carried out, in both cases, without air and also when the air was fitted to the air-bearing pedestal. The final calibration status of the load-cell indicator however, was achieved through consideration of the following aspects.

- 1)- The transducer was connected to the motor's stator.
- 2)- The transducer was located on the test rig.
- 3)- The air was fitted to the rig's air bearing.
- 4)- The operating domain of transducer output had been fixed at appropriate magnitudes.

The load cell calibration principal including the corresponding results of the final torque calibration factor with the average of 1.24241 [millivolt/kg], is presented in [Table (4-1)]. Moreover, the effective load corresponding to the net amounts of torque transducer output, related to the final load-transducer calibration, is also shown in [Graph (4-1)].

It is worth noting that the effective load ($W_{\text{Effective}}$) required for balancing the stator of the electrical motor in relation to the applied torque then can be calculated for different magnitudes of the net torque transducer output, by using the following equation.

$$W_{\text{Effective}} = 0.8076 * \tau_{\text{Net}} - 0.0091; \quad [\text{kg}] \quad [4-1]$$

Where: (τ_{Net}) is the torque transducer output in [millivolt].

The Applied torque calibration factor, under each individual status of compressor running then can be determined from the following formula in [millivolt/kg].

$$(\text{TCF})_{\text{Applied}} = \frac{\tau_{\text{Net}}}{W_{\text{Effective}}}; \quad [\text{millivolt/kg}] \quad [4-2]$$

As every attempt was made to carry out data measurement accurately, the load transducer

was calibrated under different configurations. Hence some of them did not match the real compressor performance. Therefore, it was decided to re-calibrate the load cell in a variety of different modes based on the configuration of the test rig.

Two different approaches for load transducer calibration including their corresponding plots under different related conditions such as connecting to or releasing from the motor's stator and with or without air fitting to the air bearing are presented in [Appendix (A)].

4.4)- Air Flow Inlet and Outlet Ducts and the Flow Straighteners:

Air was introduced straight from the atmosphere into the bell-mouth of the compressor, which was directly driven through the inlet by an electrical direct current (DC) variable speed swinging electrical motor. The flow then passed down from the compressor into the outlet ducting through the two flow-straightener boxes, and was passed into the venturi. After passing the outlet ducting and relevant orifice the flow was discharged into the atmosphere.

The three stage compressor and venturi were connected in-line through a flexible joint by a fairly short sheet-steel duct containing two completely separate flow straightener honeycombs. The hexagonal type 6.5 [mm] across the flats (a.f.) holes and 35.75 [mm] depth, honeycombs were used inside the flow straighteners in order to straighten the flow before it entered into the measuring venturi.

The flow throughput was controlled by using an electrically operated rectangular cross-section orifice, which enabled the variation of the outlet flow at the end of a large light alloy cast venturi, where the mass flow was measured. The electrically operated orifice controlled the mass flow by rotating an electrical control switch, which was located at the test house. The orifice had five rectangular vanes, which were located at the end part of the outlet duct. Some part of the test rig outlet ducts and orifice were positioned outside the test area.

The orifice's rectangular vanes could be easily rotated about their axis in order to vary the flow of air by changing their positions from fully open to completely closed. The flow passing through the orifice was then discharged into the atmosphere. The compressor had no outlet guide vanes, and therefore the outlet flow from the compressor's third stage stator blades cascade entered the venturi after passing the two air flow straighteners.

A flow filter, which surrounds the bell-mouth inlet, may be used in order to protect the compressor's bladings from foreign objects that could be sucked into the compressor during its operation. However, all the experimental works were carried out without using the flow filter, because it was agreed that in the reality there was very little risk of foreign object damage.

4.5)- Low Speed Three Stage Axial Flow Compressor:

The experiments have been carried out, using an existing three stage¹ axial flow Peregrine compressor. This compressor has an annulus flow passage with 406.4 [mm] diameter (D_{Ann}) and (0.7) hub to tip diameter ratio. Given that the present research was restricted to the use of this three-stage machine, a valid area for further research could

¹ The ultimate steady state is normally reached in four stages. The application of a vane recessed tubular passage casing treatment to a multistage axial flow compressor with or more than four stages also can be proposed in the future investigations in order to be confidence of the achieved results and performances.

address the use of a compressor with four or more stages.

The four rows of stationary blades including inlet guide vanes were held through spigots and stud roots with nuts and washers into half-rings and clamped between spacer rings by eight external tie bolts. A cross-section view of the low speed three stage axial flow Peregrine research compressor in its second configuration, which was predominantly constructed of light alloy, is shown in [Figure (4-2)].

The three rows of rotating blades were similarly mounted, into their related whole rings, which were clamped via distance rings on the drum by a nut on the end of the shaft. This composite construction give latitude for a wide range of alternative builds, with variations in blade spacing, stagger angles and profiles.

Four screw actuators were provided in order to limit the rotation of each stationary blade ring. A movement of around (1.5) pitches is possible. The movement of the stator rings relative to the traverse probe facilitated the circumferential traversing. The relative position of the probe to the centre line of the blade pitch was obtained by an indicator plate set against the actuator link on the body of the compressor.

4.5.1)- Peregrine Compressor Dismantling and Reassembling Procedures:

The aims of the current investigation and the programme of work required the compressor to be dismantled and reassembled again in a number of different builds. Therefore it was necessary to have a procedure in place for dismantling the compressor.

During the experimental phase of the study the test rig was dismantled (34) times and was run more than (51) times on different day and in a variety of conditions. The procedure for dismantling the compressor was as follows:

- 1)- After disconnecting the measuring devices in addition to all pipe connections, the polythene joint ring in the ducting must be removed firstly and then the driven coupling bolts can easily be released.
- 2)- Four fitted bolts which fix the compressor body to the support stools can be removed at this step.
- 3)- The compressor unit then must be lifted from the mounting table and then up-end on the inlet end casting. The drive shaft must be allowed to protrude below by using an appropriate stool.
- 4)- The four stator adjusting screw assemblies in addition to eight tie bolts can then be removed from the outside of the compressor body.
- 5)- In order to disconnect from the ducting, the nuts of the fixing studs from the end-ducting flange have to be released.
- 6)- It is then possible to remove the inner nose cowl in order to access the outer ring of six set screws about the rotor bearings which when removed allows the outlet casting to be lifted off. The oiling connections can then be released.
- 7)- The stator distance rings except for the group at both ends, which they are made up in sets held together by four set screws could be then lifted off. It is worth noting that the stator blade rings must be lifted a little out of the register, then both halves can be removed laterally from between the rotor blades.
- 8)- By releasing the nut on the top of the rotor assembly, it is then possible to lift off the complete rotor drum with blades and distance pieces. In order to prevent the blades from falling off, it is necessary to temporarily lash them together.

This procedure has to be repeated in reverse in order to reassemble the compressor after the required changing of blade attitudes, spacer rings, stagger angles and other configuration adjustments.

4.5.2)- Layout of the Blades and the Spacer rings at the Nominal Attitude:

The three-stage compressor used in the present research enabled three different axial blade gap arrangements by using the normal blades setting and the design stagger angles. These arrangements provided three axial blade gaps of 22.2 [mm], 19.1 [mm] and 12.7 [mm], respectively, as measured at the reference diameter of 335.3^2 [mm].

However, when the blades were set at design stagger angles, the axial spacing between the blade rows could be (0.729, 0.625 and 0.417) of the blade chord length, depending on the use of different spacers. The 22.2 [mm] axial blade gap setting of the compressor's bladings and spacer rings geometry will be hereafter referred as the nominal attitude.

Flow was introduced into the compressor via a bell-mouth intake (No. 44460). The outer spacer ring (No. 44462) was fitted to the left adjuster ring, before the inlet guide vanes. The (34) inlet guide vanes were held in place by two half-rings, each of which held (17) vanes. The two half rings were fixed inside two spacer ring segments [(No. 44462) and (No. 44463)]

The axial distance from the end portion of the inlet spacer ring (No. 44460) to the centre line of the inlet guide vanes was 39.7 [mm]. The spacer ring (No. 44463) was also located after ring (No. 44462) and covered the rest of the inlet guide vanes' two half-ring holders. Two spacers rings [(No. 44479) & (No. 44477)] with an outside diameter equal to the compressor's hub diameter (D_{hub}) were used before and after the first stage rotor blades respectively not only covered the rotor blades' ring holder, but also formed part of the inside diameter of the compressor's annulus.

The rotor blades holding rings have provisions for (38) blades in each stage. The normal rotating blades were of free-vortex design. The entire three stage rotor blades were fitted onto their corresponding holes in the ring holders. This arrangement gave an axial distance of 48 [mm] from the centreline of the inlet guide vanes to the centreline of the first stage rotor blades cascade.

Each stator set was located inside two unequal, half-rings, which together have provision for holding (37) stationary blades. Two other spacer rings [(No. 44461) & (No. 44464)] including the spacer ring (No. 44466) were used to provide an axial distance of 95.1 [mm] between the inlet guide vanes and the first stage stator blades centre lines.

Two spacer rings [(No. 44474) & (No. 44478)] were located between two identical spacer rings (No. 44477) in order to provide an adequate axial distance of 94.4 [mm] between the centrelines of the first and second stage rotor blades. The arrangement of the outer spacer rings provided an axial distance of 94.4 [mm] between the centre lines of one stator cascade and the next. The spacer rings [(No. 44466) & (No. 44467)] which were fitted before and after the first stage stator blade sets respectively, covered the two half-rings with (37) blade retaining holes of the first stage stator blade holder.

The spacer rings [(No. 44462) & (No. 44467)] covered the two half-rings of the second stage stator blade holder which are similar to the first stage ones. Two sets of spacer rings

² The mean diameter of 13.2 [inch] is adopted as a datum for aerodynamic expediency and identification in giving rebuild references.

[(No. 44461) & (No. 44464)] were used exactly before and after the two previously mentioned spacers. Two other spacer rings [(No. 44466) & (No. 44467)] were used right before and after the third stage stationary blades respectively. These spacers formed part of compressor's annulus and were also used to cover the two half-rings of the third stage stator blade holder.

The axial distance between the centrelines of the inlet guide vanes and the first stage stator blades was 95.1 [mm]. Meanwhile, the axial distances between the centreline of the first stage stator and the centrelines of the second and third stage stator blades were 94.4 [mm] and 188.8 [mm], respectively. The inside diameter of all the outside spacer rings was 406.4 [mm] and this was the same as the compressor annulus inside diameter (D_{Ann}).

The rotating blades holders of the second stage were located between the two spacers which were identical to the ring (No. 44477), while the third stage rotating blade set holder was located between two spacer rings with similar geometry to (No. 44477) and (No. 44475), respectively.

By using two sets of spacers with the same geometry as rings [(No. 44474) & (No. 44478)] in between two spacers identical to ring [(No. 44477), an axial distance of 94.4 [mm] was provided between second stage and third stage rotor blade centrelines. With the fitting of the last spacer ring (No. 44473), the drum of the compressor test rig was complete. The axial distance between the centreline of the third stage rotor and the later spacer ring in nominal attitude was 27 [mm].

The outside diameter of the compressor drum spacer rings was 284.4 [mm], the same as the compressor hub diameter (D_{Hub}).

4.5.3)- Normal Blades Settings and Relevant Design Parameters:

The Peregrine compressor has a rotating blade tip diameter (D_{Tip}) of 405.2 [mm], tip clearance (t_c) of 0.6 [mm] and a hub to tip diameter ratio of ($D_{Hub}/D_{Tip}=0.7$). The three sets of rotating blade cascades with ($Z_{Rotor}=38$) blades in each row were held into the rings and together with the distance rings form the drum which is the rotating hub of the compressor.

Each stage stator blades with the ($Z_{Stator}=37$) blades fit onto the two half-rings that are located in distance rings which build together, to form the casing of the compressor. The inlet guide vanes with ($Z_{IGVs}=34$) blades also fit into the two half-rings and the complete holder ring was located between two distance rings which together form the casing portion of the inlet guide vanes.

In order that the blade rows in the compressor should be fully interchangeable, the width of each rotor and stator blade rings in which the blades were mounted was identical. It means that the axial distance between the centre lines of each blade row was fixed. Four moving screw actuators were mounted onto the casing to facilitate the circumferential movement of the three stationary blade rows and the inlet guide vanes in their rings.

Both the rotor and the stator blades were held onto their corresponding rings, which have been recessed to accept each blade's root, by a nut and a bolt arrangement. In this way the blades may be set for any desired stagger angles (ξ). However, when the stagger angles are altered the distance between two adjacent blade rows can be considered as a function of the stagger settings.

The compressor can be built for three different axial blade gaps by rearranging the hub and casing distance rings. The blading design was specified at a reference (mean) diameter (D_{Mean}) of 335.3 [mm].

At the mean diameter, the design stagger setting angles are given as ($\xi_{IGVs}=18.45^\circ$) for the inlet guide vanes, ($\xi_{Rotor}=36.3^\circ$) for the rotor blades and ($\xi_{Stator}=36^\circ$) for the stator blades. These however, provide a stage design work coefficient of $[(\Delta H/U^2)=0.347]$.

The design parameters of the rotating and stationary blades related to the radial distributions, such as space, space to chord ratio (S/C), stagger angles (ξ), camber angles (θ), axial chord (Ca) and flow velocities with both absolute velocity angles (α) and relative velocity angles (β) have been shown in [Table (4-2)]. By changing the annulus diameters, the stationary and rotating blades stagger angles are altered from the hub to tip diameters. The details of the radial distribution of the inlet guide vanes, rotor and stator blades stagger angles versus the annulus span diameters alteration are shown in [Graph (4-2)].

The blade chord of the entire fixed and moving blades (C_{IGVs}), (C_{Stator}) and (C_{Rotor}) have been kept constant and equal to 30.5 [mm], in spite of the blade height changing. Therefore, the magnitudes of the space to chord ratio of the inlet guide vanes $[(S/C)_{IGVs}]$, rotor blades $[(S/C)_{Rotor}]$, and the stator blade cascades $[(S/C)_{Stator}]$ are changed with the radial alteration of the annulus span diameters. The details of the radial distribution of the inlet guide vanes, rotor and stator blades space to chord ratio corresponding to alterations of the annulus span diameters is illustrated in [Graph (4-3)].

The axial chord of the inlet guide vanes, rotating blades and the stators corresponding to the designed stagger angles at the reference diameter (D_{Mean}), and the blades chord (C) are $(Ca)_{IGVs}=28.93$ [mm], $(Ca)_{Rotor}=24.58$ [mm], and $(Ca)_{Stator}=24.68$ [mm], respectively. The axial chords of the bladings are changed as the diameter of the compressor's annuls altered. The axial chord alteration based on the radial distribution has been shown in [Graph (4-4)].

It is worth noting that the distance between the blades at the mean diameter in the nominal configuration is 22.2 [mm], Therefore any modification of the stagger angles, except the normal ones, will alter the gaps between the blades from the nominal attitude. The rotor blades should effectively be tightened when individually located with a fixture offered to the chord that are points corresponding to the mean diameter. When the applied fixture is horizontal, the angle of the inclined plane corresponds to the desired stagger angles.

4.5.4)- Rotor Blade Reaction and other Related Design Parameters:

The existing normal blading for the Peregrine compressor is of free vortex design with the stage reactions varying from around (31%) at the inner diameter (D_{Hub}) to about (66%) at the outer diameter (D_{Tip}). At the reference diameter (D_{Mean}), the blades were designed for (50%) stage reaction with the absolute inlet and outlet air velocity angles of (50°) and (30°), respectively. These give a design flow coefficient of ($\varphi=0.565$).

The radial distribution details of the inlet and outlet absolute velocity angles versus radial geometrical alteration (annulus span diameters) of the fixed and the moving blades have been shown in [Graph (4-5)].

The blades have a (C4) profile thickness distribution on the circular arc camber line with a maximum thickness to chord ratio of $[(T/C)=0.1]$. In order to provide an aspect ratio of $[(h/C)_{Rotor}=2.0]$, the blade chord (C_{Rotor}) has been fixed constant along the span at 30.5 [mm]. Rotor blades operate with a tip clearance (t_c) about (0.15%) of the annulus inlet diameter (D_{ann}) to give nearly 0.6 [mm] clearance. These blades also turns clockwise when looking towards the compressor. The inlet and outlet absolute velocity angles at reference diameter have been considered as ($\alpha_1=\alpha_3=50^\circ$ & $\alpha_0=\alpha_2=\alpha_4=30^\circ$).

In order to meet the desired configuration described above, the designer calculated the

blade geometrical angles by using recommendations of Cox (1946). The inlet and outlet absolute velocities corresponding to the radial distribution of the different annulus diameters have also been drawn in [Graph (4-6)]. The velocity triangles of the absolute and relative velocities at the mean diameter, based on the normal blades setting and related design stagger angles is shown in [Figure (4-3)].

4.6)- Venturi Flow Meter:

The venturi of the Peregrine compressor test rig is a non-standard venturi flow meter, which has been manufactured as a long sheet steel tube with different inlet and outlet diameters. It has an inlet diameter $[(D_{VFM})_{Inlet}]$ and throat diameter $[(D_{VFM})_{Throat}]$ of 406.4 [mm] and 254 [mm], respectively.

The profile of the applied venturi is a compromise between standard, (BS1042), conical and nozzle types, (Farmakalides, 1992). The mass flow rate through the test rig was calculated by measuring the static pressure of the flow at the inlet cross-section and the throat of the venturi. Due to the application of two upstream honeycombs in the flow straighteners, it can be assumed that the flow through the venturi would be uniform.

The Peregrine rig non-standard venturi flow meter incorporating a (5°) angle diverging diffuser is also shown in [Figure (4-1)]. The venturi comprises eight static pressure tapings at 45 [degree] radial intervals, in both the inlet and the throat diameters. With eight tapings on both sides which the inlet and throat cross sections were connected to circular pipe collectors.

The average static pressures of the inlet $[(p_{VFM})_{Inlet}]$ and throat $[(p_{VFM})_{Throat}]$ cross section areas could then be experimentally found by connecting the two static pressure collectors into the manometers by using two pipes of identical length and diameter.

4.7)- Pressure Measurements and Relevant Instrumentation:

One of the main objectives of the present investigation was to find out the performance of the compressor under different builds and attitudes. Therefore accurate measurements of the different parameters, which were used to calculate the compressor's characteristics such as total or static pressures, were very crucial.

The current investigation was similar to other turbomachinery research in that there were quite large variations in flow conditions, particularly when the casing treatments with different rotor blade tip axial chord exposure were applied.

To accurately determine the flow characteristics, it was necessary to take a large number of pressure values at different stations and various conditions. This requirement for high accuracy meant that testing took a considerable amount time. The applied measuring devices including the general principal that were used in this study in order to measure both total and static pressures are described below.

4.7.1)- Total Pressure Measuring Device:

It is a frequent requirement to measure total pressure of air at a given point in a fluid field of a compressor in order to assess the machine performance. Total pressure measurement was an area of interest for the current investigation. The objective was to find an accurate relationship between the actual total pressure and that, which was indicated by the applied devices.

The traverse gear instrument stations were located between each row of blades and at one

point in the discharge annulus. To fit the traverse gears to any of the aforementioned stations, a cover plate has to be removed and the traverse head can then be fitted in its corresponding place. A movement of 1.5 pitches of the stator blades is also possible in order to facilitate the circumferential traversing, if required.

The main difficulty in pressure measurements, results from the fact that in the majority of cases and particularly in stalled conditions, the swirling flow phenomena may provide some undesirable effects at the upstream, downstream or at both positions of the probe that can affect the pressure measurement.

A combined wedge total-static yaw meter traverse-gear had originally been applied in order to measure the total pressure in the Peregrine test rig. This however was damaged during one of the early investigations of the rig. The previous researchers³ that used the same test rig to serve their investigations therefore used different types of total pressure probe such as cylindrical three holes probe.

It was thought that the use of a cylindrical yaw-meter that normally offers minimum disturbance to the flow could be utilised as a convenient total pressure measurement device by measuring total and static pressures in association with flow angles when required. This type of pressure probe is probably one of the most common forms of combined pressure and yaw meter, which has been applied, in many different flow-measuring investigations.

The present research subsequently adopted a 4 [mm] diameter three hole pneumatic cylindrical total pressure probe in order to measure the total pressure in the third stage stator blades downstream at the outlet of the compressor. This type of probe is normally used to measure just the axial and the tangential aspects of the flow, and therefore, it can be considered as a two-dimensional probe.

The use of this probe proved very time consuming and frustrating, as it failed to provide the desired results. Although normally reliable one of the major concerns with cylindrical total pressure probe measuring devices seems to relate to the positions of the two holes, which were on the same plane as the centre hole.

It is worth noting that the pneumatic pressure probe in general needs to be carefully calibrated to read static pressure. This type of probe was not able to provide acceptable pressure measurement results, since considerable errors occurred when it was used in order to measure the pressure near the wall. The reason for inaccuracy of the static pressure measurements near the compressor's casing or hub is normally related to the wall effects phenomena.

The three-hole pneumatic probe was designed by Das et al. (1984), specifically for measuring total pressure in rotating stall conditions. This suitability was supported by Giannissis, (1987). It was decided therefore, to use a cobra type pressure probe instead. The traverse probe used throughout the present investigation was a steady state pneumatic cobra probe. This consisted of three holes of 0.6 [mm] inside diameter, and is mainly suitable for two-dimensional pressure measurement.

The Cobra probe is one of the more widely used types of combination probe which is designed to measure total and static pressures in addition to the yaw angle, using three air-filled pressure lines. Total pressure is measured by centre tapping when facing directly into the prevailing flow. This is achieved by rotating the probe until the pressure read from

³ (Giannissis, 1987) and (Farmakalides, 1992).

the two outer tappings are equal, whereupon the probe is said that to be “nulled”. Static pressure is derived from the side tappings via an aerodynamic calibration. The yaw angle is taken to be the probe setting angle relative to the rig axis.

As it was reported by Smout et al. (1998) the performance of a cobra-type pressure probe even in steady flow is compromised in two ways. Firstly, this probe fails to sense the correct static pressure when operating in close proximity to a wall through which the probe is introduced, yet well outside the boundary layer. It was also noted that the static pressure, indicated by the cobra probe, was less than that of the free stream static pressure.

The applied total pressure probe comprised three hypodermic tubes running through the stem of probe and bent through (90°) in a swan-naked shape. The side ports are chamfered back at (45°) to the plane of the central pitot port and the head of probe lies within the envelope of the stem.

The radial traverse mechanism enabled the probe to be traversed at its location radially between the rotor hub and inside casing wall. The cobra-type pressure probe was located on the third stage stator blades downstream at the same circumferential span of which the four static tapping were used in order to measure the total pressure static pressure at the downstream of the third stage stator blades.

The gauge pressure obtained from the probe was measured using a manometer. Both side port tubes in addition to the centre hole tube were connected by three identical lengths of high quality flow measuring pipe into a (36) channel manometer filled with water. Each connection consisted of a tube with 1.5 [mm] inside diameter and 350 [cm] length.

4.7.2)- Static Pressure Measuring Device:

The compressor test rig was equipped with conventional slow response wall static tappings at the inlet and outlet of the compressor. It also had provision for inter-row wall static measurements at inter rows gaps.

In order to average out wake effects, at each measuring station four wall static tappings were equally located at (90°) intervals around the circumference of the distance rings used for housing the rotating and fixed blades. To measure the required static pressures, the 1.4 meter height manometer which filled with water had been set at (30°) inclination from the ground line. In such condition the static pressure could then be measured in [mm-H₂O].

It is worth noting that the static pressure tappings were located before and after each blade row. Every two adjacent wall static pressure tappings had been connected via a 30 [cm] pipe to two sides of a (T) junction. The third tapping of the T-junction was then connected to the manometer by using a 350 [cm] flow pipe connection.

To reduce the averaging error, it is recommended that the static pressure tappings be connected in pairs by two equal lengths of the connecting tube from the tappings to the bank manometer. In the current investigation every two static pressure tappings were also connected to one of the manometer's channels.

The value of static (gauge) pressure of each blade cascade can then be determined by averaging the measured gauge pressure from two corresponding manometer's channels to each inter blades row. The static (gauge) pressures at the inlet, outlet or inter-row upstream or downstream of each blade cascades can then be derived in [Pascal] as follows.

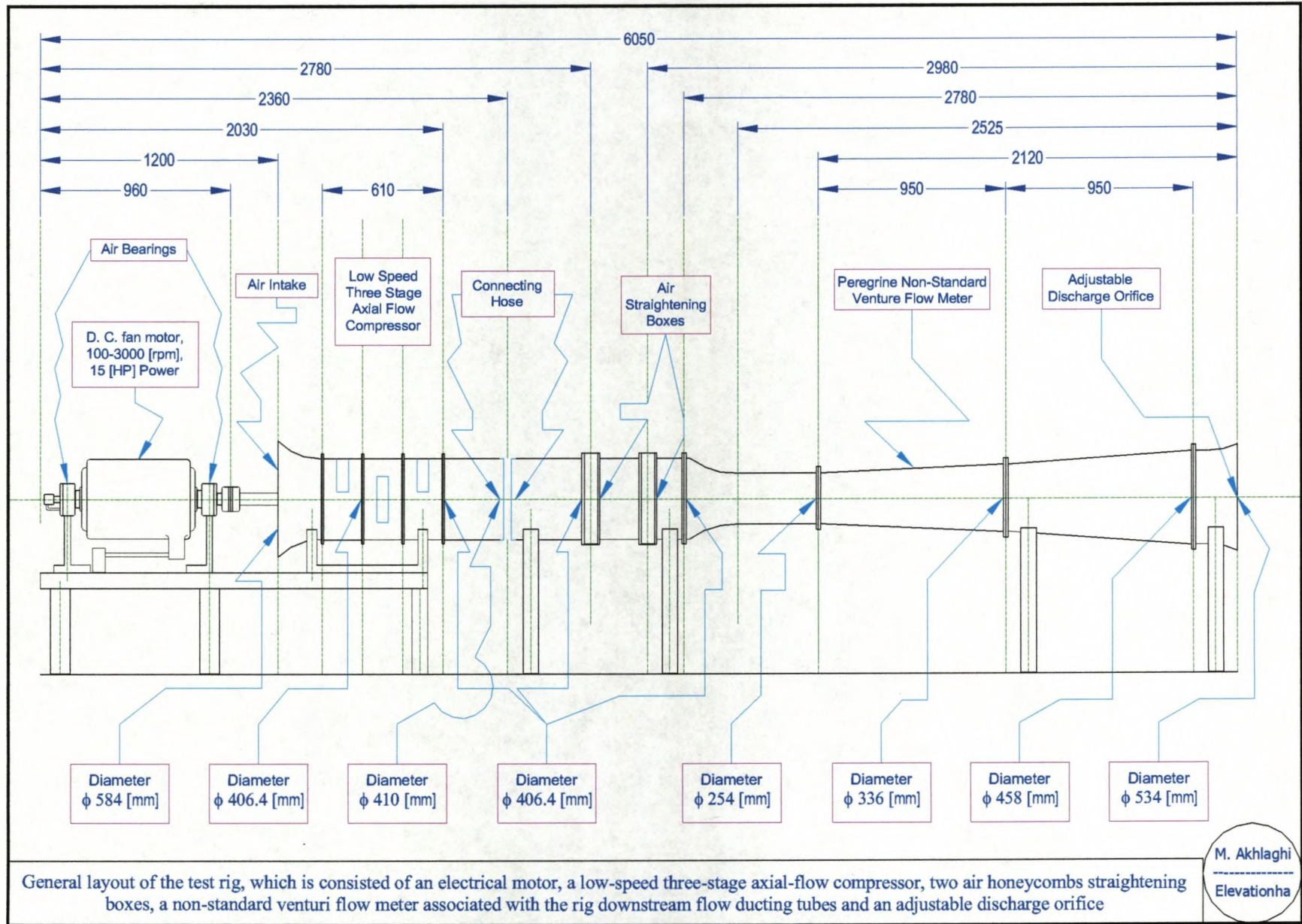
$$p_{S\text{-gauge}} = 98.1302 * \sin(30^\circ) * [p_{S\text{-Datum}} - (\frac{p_{S\text{-Tappings (1\&2)}} + p_{S\text{-Tappings (3\&4)}}}{2})] ; \quad [\text{Pa}] \quad [4-3]$$

It is worth noting that the required final static (absolute) pressures for each blade cascade or venturi flow meter up or down stream can then be calculated by adding the ambient pressure to the measured inter rows blades and venturi inlet or throat gauge pressures.

$$P_{S-Absolute} = P_{S-Gauge} + P_{Ambient} ; \quad [Pa] \quad [4-4]$$

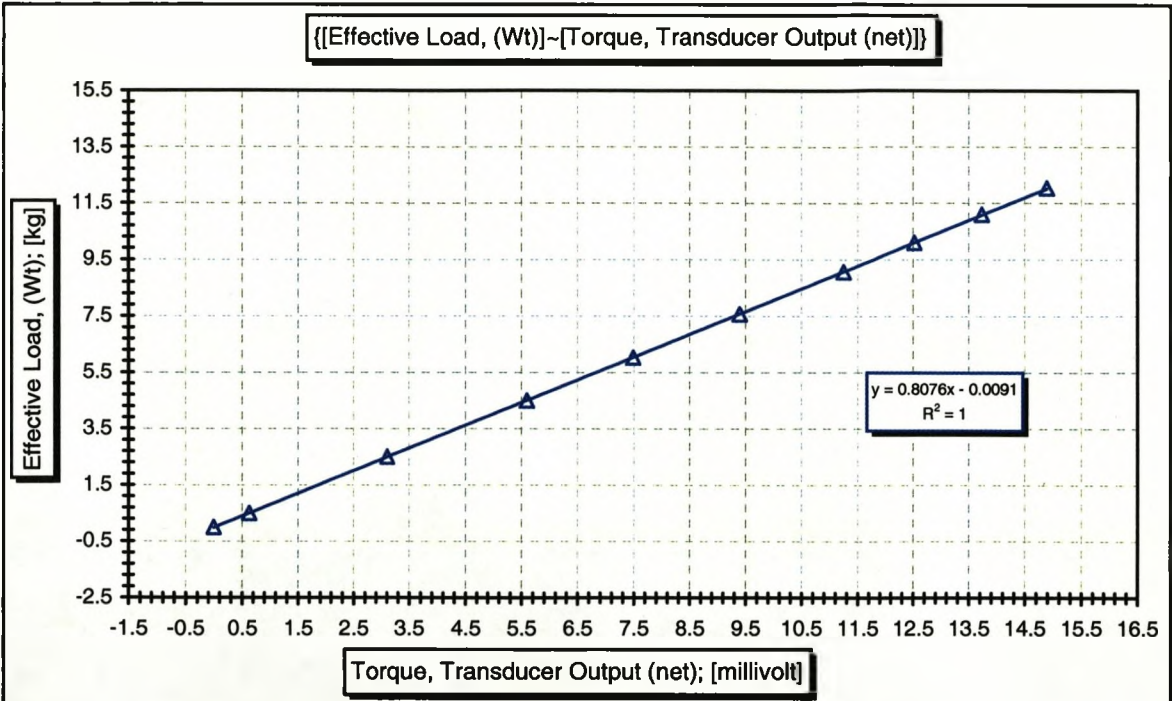
The locations of the entire static pressure tapings in the outer casing of the Peregrine compressor corresponding to the inter-row upstream or downstream of both stationary and moving blades cascades have been presented in [Table (4-3)].

The data acquired from the wall static tapings were used in order to obtain the steady state overall and stage characteristics of the compressor. These data were used in calculation of the blade row characteristics as well, where it was required.

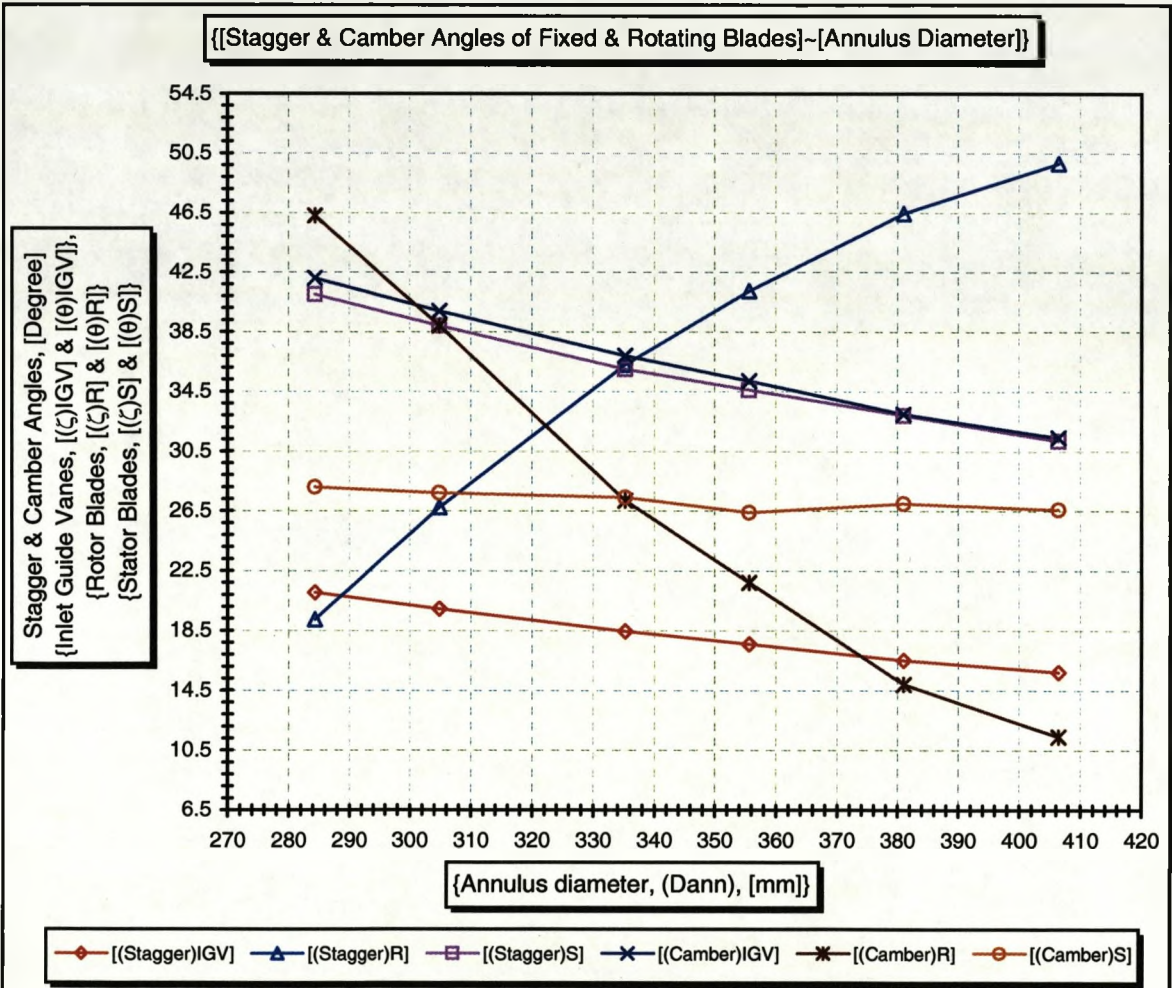


[Figure (4-1)]- General layout of the complete rig facilities, employed in the current investigation

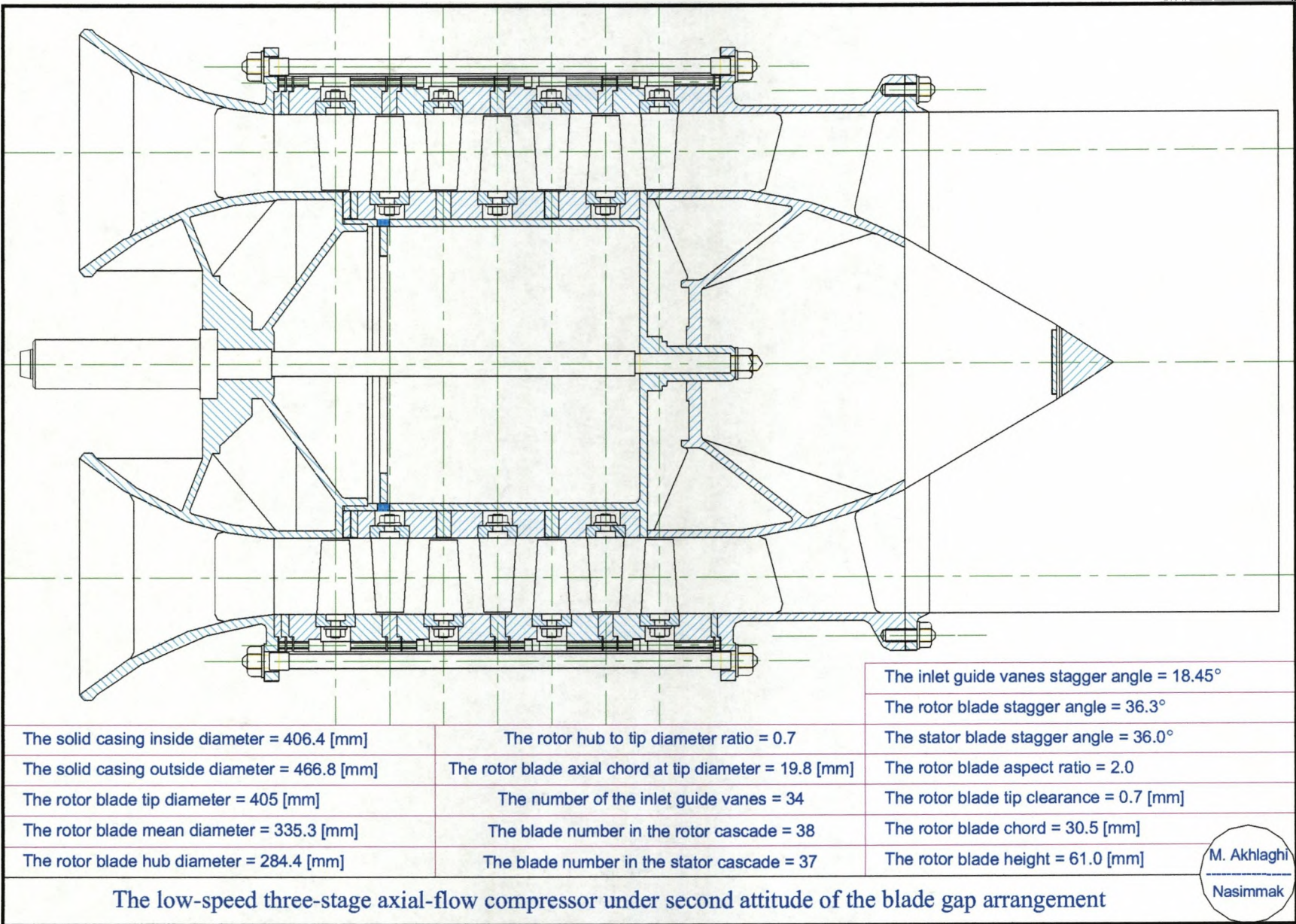
Torque, transducer output (raw); [millivolt]											
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[Average]
0.000	-0.020	0.010	0.010	0.010	0.010	0.000	0.000	0.010	0.010	0.000	0.004
0.500	0.610	0.630	0.620	0.640	0.650	0.640	0.640	0.640	0.630	0.640	0.634
2.505	3.100	3.120	3.100	3.130	3.100	3.120	3.090	3.140	3.110	3.090	3.110
4.505	5.580	5.630	5.570	5.650	5.580	5.640	5.560	5.630	5.570	5.620	5.603
6.035	7.460	7.520	7.460	7.550	7.440	7.530	7.460	7.530	7.460	7.540	7.495
7.570	9.350	9.430	9.340	9.430	9.400	9.450	9.350	9.420	9.370	9.440	9.398
9.065	11.250	11.270	11.260	11.270	11.240	11.280	11.210	11.250	11.250	11.260	11.254
10.100	12.520	12.540	12.520	12.510	12.490	12.540	12.530	12.540	12.510	12.550	12.525
11.090	13.710	13.760	13.700	13.760	13.750	13.750	13.720	13.750	13.740	13.760	13.740
12.030	14.850	14.850	14.880	14.880	14.930	14.930	14.880	14.880	14.910	14.910	14.890
Torque, transducer output (net); [millivolt]											
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	Net [mean]
0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
0.500	0.630	0.620	0.610	0.630	0.640	0.640	0.640	0.630	0.620	0.640	0.630
2.505	3.120	3.110	3.090	3.120	3.090	3.120	3.090	3.130	3.100	3.090	3.106
4.505	5.600	5.620	5.560	5.640	5.570	5.640	5.560	5.620	5.560	5.620	5.599
6.035	7.480	7.510	7.450	7.540	7.430	7.530	7.460	7.520	7.450	7.540	7.491
7.570	9.370	9.420	9.330	9.420	9.390	9.450	9.350	9.410	9.360	9.440	9.394
9.065	11.270	11.260	11.250	11.260	11.230	11.280	11.210	11.240	11.240	11.260	11.250
10.100	12.540	12.530	12.510	12.500	12.480	12.540	12.530	12.530	12.500	12.550	12.521
11.090	13.730	13.750	13.690	13.750	13.740	13.750	13.720	13.740	13.730	13.760	13.736
12.030	14.870	14.840	14.870	14.870	14.920	14.930	14.880	14.870	14.900	14.910	14.886
Torque Calibration Factor (Individual); [millivolt/kg]											
Load [kg]	C.F. [1]	C.F. [2]	C.F. [3]	C.F. [4]	C.F. [5]	C.F. [6]	C.F. [7]	C.F. [8]	C.F. [9]	C.F. [10]	C.F. [mean]
0.500	1.260	1.240	1.220	1.260	1.280	1.280	1.280	1.260	1.240	1.280	1.260
2.505	1.246	1.242	1.234	1.246	1.234	1.246	1.234	1.250	1.238	1.234	1.240
4.505	1.243	1.248	1.234	1.252	1.236	1.252	1.234	1.248	1.234	1.248	1.243
6.035	1.239	1.244	1.234	1.249	1.231	1.248	1.236	1.246	1.234	1.249	1.241
7.570	1.238	1.244	1.232	1.244	1.240	1.248	1.235	1.243	1.236	1.247	1.241
9.065	1.243	1.242	1.241	1.242	1.239	1.244	1.237	1.240	1.240	1.242	1.241
10.100	1.242	1.241	1.239	1.238	1.236	1.242	1.241	1.241	1.238	1.243	1.240
11.090	1.238	1.240	1.234	1.240	1.239	1.240	1.237	1.239	1.238	1.241	1.239
12.030	1.236	1.234	1.236	1.236	1.240	1.241	1.237	1.236	1.239	1.239	1.237
Torque Calibration Factor (Average), [(TCF)-ave]; [millivolt/kg]										1.24241	
[Table (4-1)]- The final applied torque calibration factor calculation											



[Graph (4-1)]- The effective load correspondence to the torque calibration factor



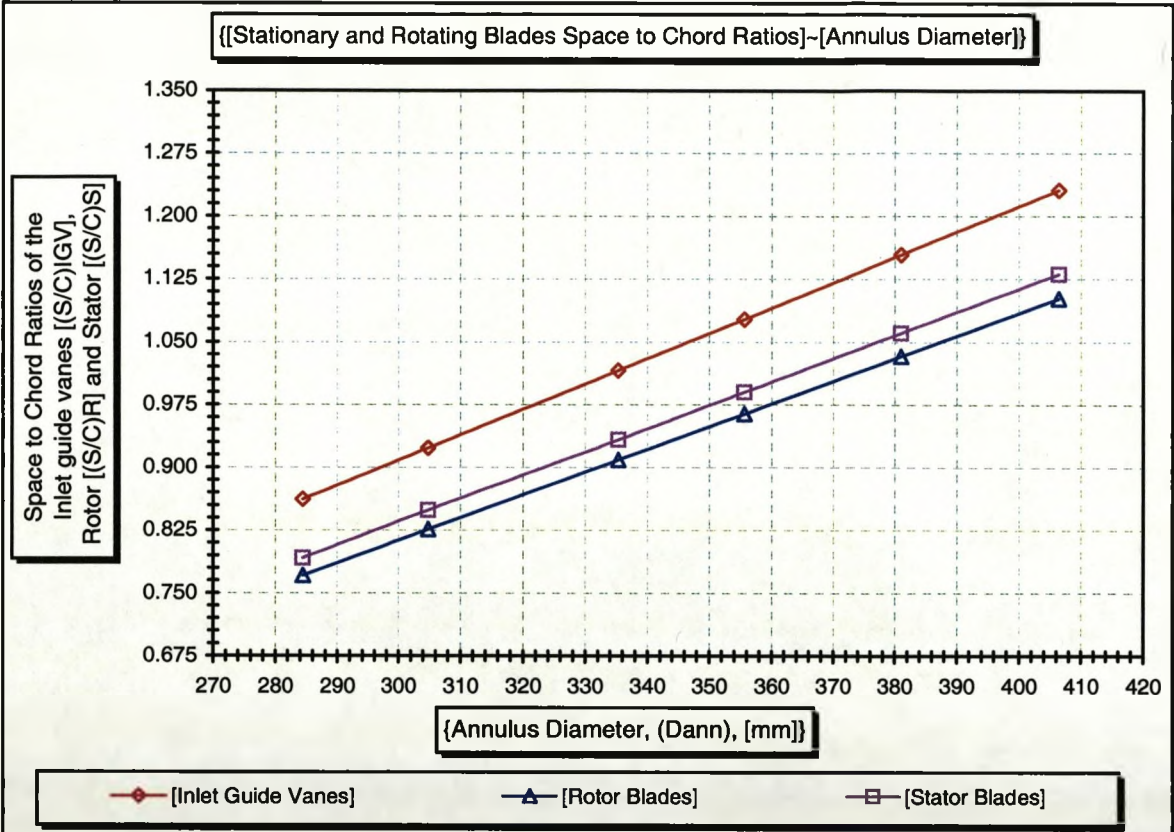
[Graph (4-2)]- Radial distribution of the IGVs, rotor and stator stagger and camber angles



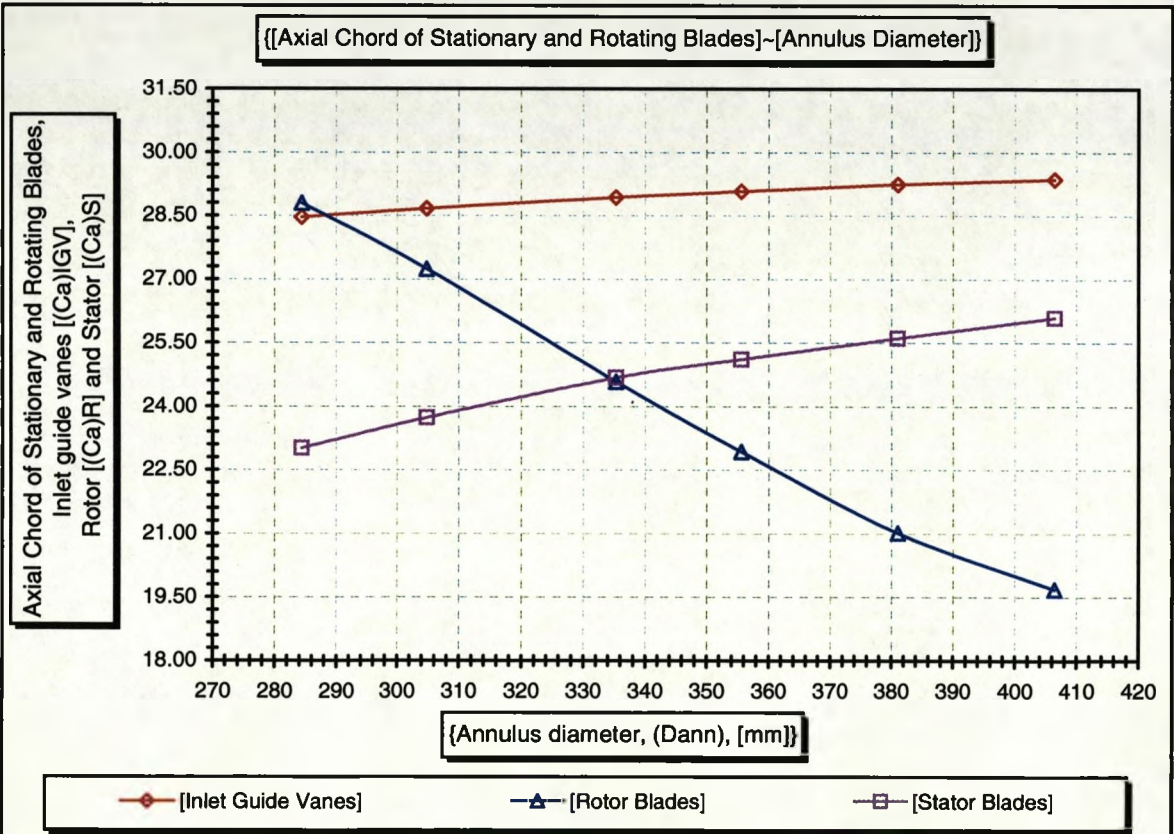
[Figure (4-2)]- A cross-section view of the low-speed three-stage axial-flow Peregrine research compressor in the configuration of the second attitude

Inlet guide vanes, rotor and stator blades design parameters corresponding to annulus diameters						
Number of the inlet guide vanes	34					
Number of the rotating blades in each rotor blade cascade	38					
Number of the stationary blades in each stator blade cascade	37					
Annulus diameter, (D_{ann}); [mm]	284.40	304.80	335.30	355.60	381.00	406.40
Inlet guide vanes space, (S_{IGV}); [mm]	26.278	28.163	30.982	32.857	35.204	37.551
Inlet guide vanes space to chord ratio, [$(S/C)_{IGV}$]	0.862	0.923	1.016	1.077	1.154	1.231
Inlet guide vanes stagger angle, [$(\zeta)_{IGV}$]; [degree]	21.05	19.95	18.45	17.60	16.50	15.70
Inlet guide vanes camber angle, [$(\theta)_{IGV}$]; [degree]	42.10	39.90	36.90	35.20	33.00	31.40
Inlet guide vanes axial chord, [$(Ca)_{IGV}$]; [mm]	28.465	28.670	28.932	29.072	29.244	29.362
Inlet guide vane outlet velocity angle, [$(\alpha_0)_{IGV}$]; [degree]	34.20	32.40	30.00	28.60	26.80	25.50
Inlet guide vane outlet velocity, [$(V_0)_{IGV}$]; [m/sec]	35.99	35.26	34.38	33.91	33.35	32.98
Rotor blades cascade space, (S_R); [mm]	23.512	25.199	27.720	29.399	31.499	33.599
Rotor blades cascade space to chord ratio, [$(S/C)_R$]	0.771	0.826	0.909	0.964	1.033	1.102
Rotor blade stagger angle, [$(\zeta)_R$]; [degree]	19.25	26.75	36.30	41.25	46.45	49.80
Rotor blade camber angle, [$(\theta)_R$]; [degree]	46.30	38.90	27.20	21.70	14.90	11.40
Rotor blade axial chord, [$(Ca)_R$]; [mm]	28.795	27.236	24.581	22.931	21.014	19.686
Rotor inlet absolute velocity angle, [$(\alpha_1)_R$]; [degree]	39.40	44.20	50.00	53.10	56.40	59.00
Rotor blades inlet absolute velocity, [$(V_1)_R$]; [m/sec]	38.53	41.53	46.31	49.58	53.80	57.80
Rotor outlet absolute velocity angle, [$(\alpha_2)_R$]; [degree]	5.80	16.50	30.00	37.00	43.80	48.10
Rotor blades outlet absolute velocity, [$(V_2)_R$]; [m/sec]	29.92	31.05	34.38	37.28	41.25	44.58
Stator blades cascade space, (S_S); [mm]	24.148	25.880	28.470	30.193	32.350	34.507
Stator blades cascade space to chord ratio, [$(S/C)_S$]	0.792	0.849	0.933	0.990	1.061	1.131
Stator blades cascade stagger angle, [$(\zeta)_S$]; [degree]	41.00	38.90	36.00	34.60	32.90	31.20
Stator blades cascade camber angle, [$(\theta)_S$]; [degree]	28.10	27.70	27.40	26.40	27.00	26.60
Stator blade axial chord, [$(Ca)_S$]; [mm]	23.019	23.736	24.675	25.106	25.608	26.089
Stator inlet absolute velocity angle, [$(\alpha_3)_S$]; [degree]	54.80	52.70	50.00	48.30	46.40	44.50
Stator blade inlet absolute velocity, [$(V_3)_S$]; [m/sec]	51.65	49.13	46.31	44.75	43.17	41.74
Stator outlet absolute velocity angle, [$(\alpha_4)_S$]; [degree]	34.20	32.40	30.00	28.60	26.80	25.50
Stator blades outlet absolute velocity, [$(V_4)_S$]; [m/sec]	35.99	35.26	34.38	33.91	33.35	32.98

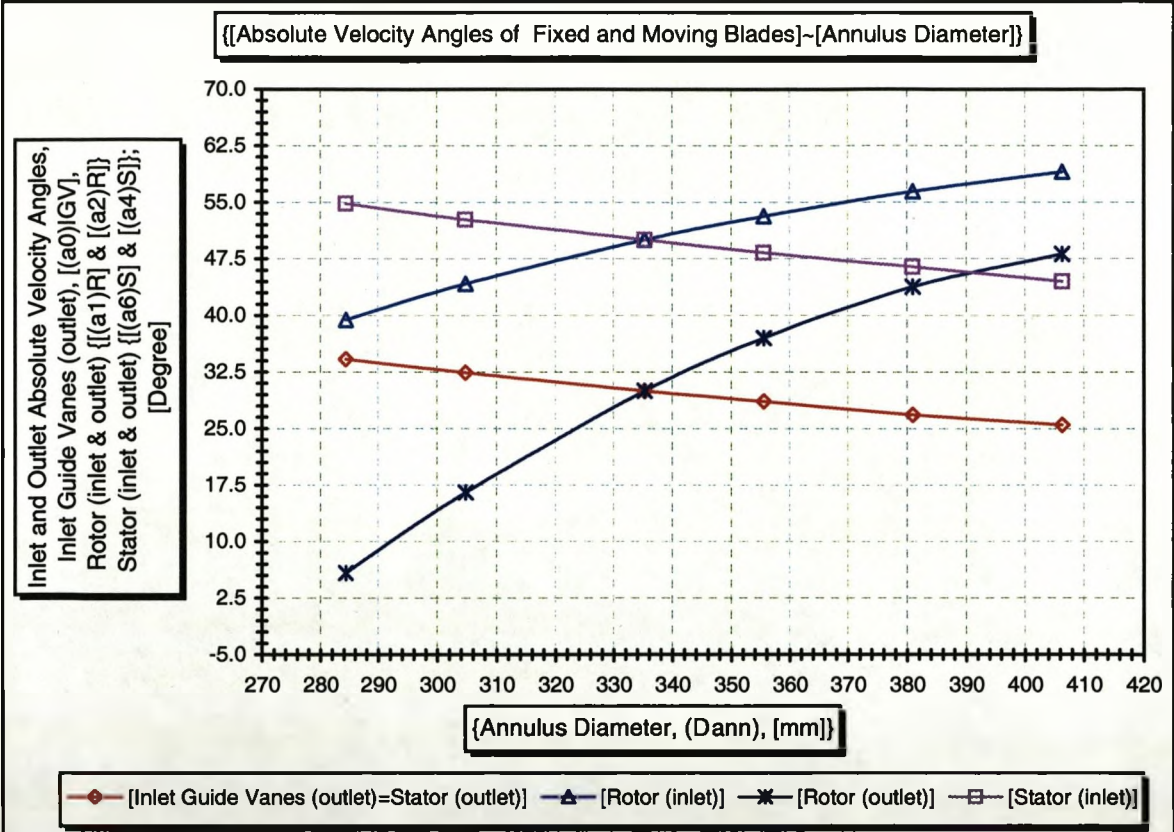
[Table (4-2)]- The rotor and stationary blades design parameters of the test rig compressor



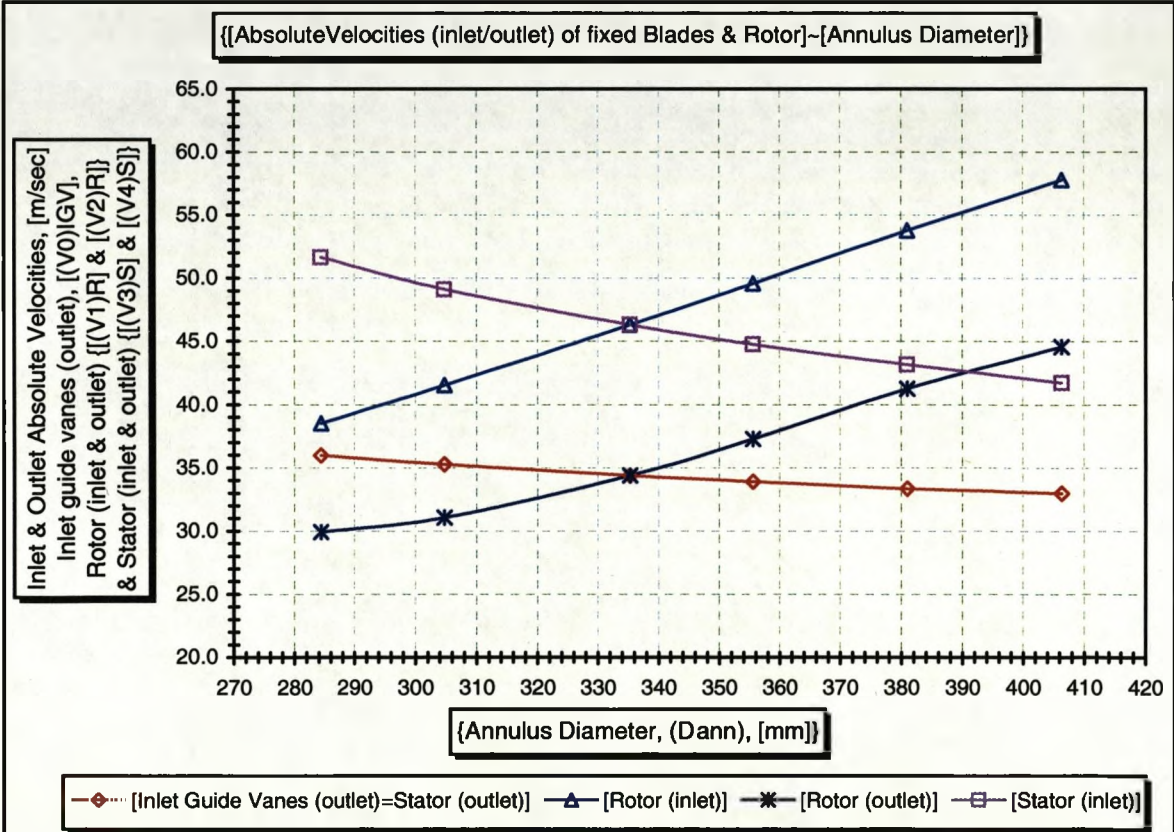
[Graph (4-3)]- The radial distribution of the space to chord ratios of fixed and rotor blades



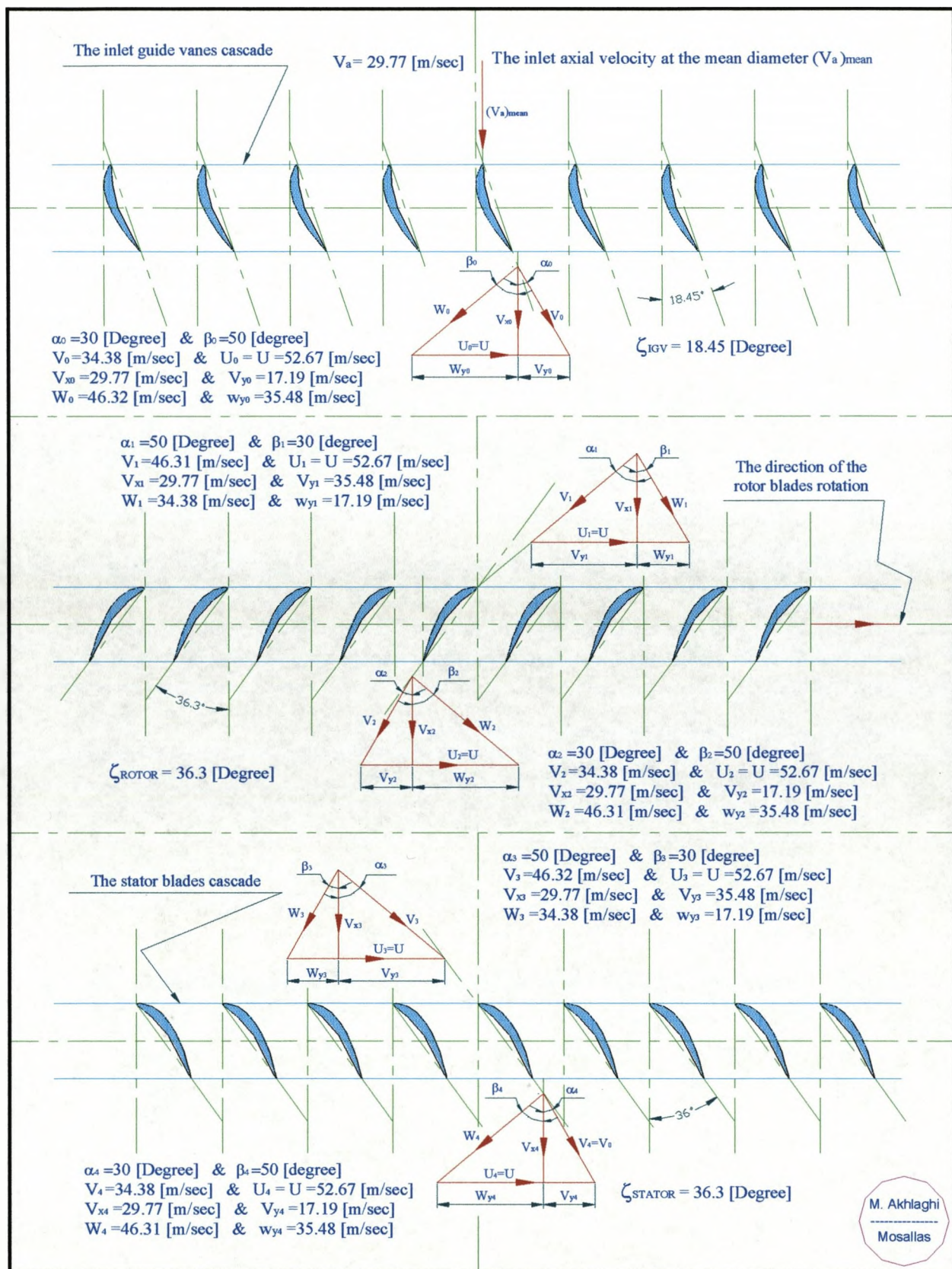
[Graph (4-4)]- The axial chord radial distribution of fixed and rotating blades



[Graph (4-5)]- The radial distributions of the inlet and outlet absolute velocity angles



[Graph (4-6)]- The radial distributions of the inlet and the outlet absolute velocities



[Figure (4-3)]- The velocity triangles of the absolute and relative velocities at the mean diameter based on the normal blades setting and related design stagger angles

Static Pressure Tappings Location on the Peregrine Compressor Outer Casing	Angle of Location [degree]
First static pressure tapping, upstream of the inlet guide vanes cascade, [P _{S1} -(IGV-US)]	22.5
Second static pressure tapping, upstream of the inlet guide vanes cascade, [P _{S2} -(IGV-US)]	112.5
Third static pressure tapping, upstream of the inlet guide vanes cascade, [P _{S3} -(IGV-US)]	202.5
Fourth static pressure tapping, upstream of the inlet guide vanes cascade, [P _{S4} -(IGV-US)]	292.5
First static pressure tapping, downstream of the inlet guide vanes cascade, [P _{S1} -(IGV-DS)]	67.5
First static pressure tapping, upstream of first stage rotor blades cascade, [P _{S1} -(1R-US)]	157.5
Second static pressure tapping, downstream of the inlet guide vanes cascade, [P _{S2} -(IGV-DS)]	
Second static pressure tapping, upstream of first stage rotor blades cascade, [P _{S2} -(1R-US)]	247.5
Third static pressure tapping, downstream of the inlet guide vanes cascade, [P _{S3} -(IGV-DS)]	
Third static pressure tapping, upstream of the first stage rotor blades cascade, [P _{S3} -(1R-US)]	337.5
Fourth static pressure tapping, downstream of the inlet guide vanes cascade, [P _{S4} -(IGV-DS)]	
Fourth static pressure tapping, upstream of the first stage rotor blades cascade, [P _{S4} -(1R-US)]	22.5
First static pressure tapping, downstream of first stage rotor blades cascade, [P _{S1} -(1R-DS)]	
First static pressure tapping, upstream of first stage stator blades cascade, [P _{S1} -(1S-US)]	112.5
Second static pressure tapping, downstream of first stage rotor blades cascade, [P _{S2} -(1R-DS)]	
Second static pressure tapping, upstream of first stage stator blades cascade, [P _{S2} -(1S-US)]	202.5
Third static pressure tapping, downstream of first stage rotor blades cascade, [P _{S3} -(1R-DS)]	
Third static pressure tapping, upstream of the first stage stator blades cascade, [P _{S3} -(1S-US)]	292.5
Fourth static pressure tapping, downstream of first stage rotor blades cascade, [P _{S4} -(1R-DS)]	
Fourth static pressure tapping, upstream of first stage stator blades cascade, [P _{S4} -(1S-US)]	67.5
First static pressure tapping, downstream of first stage stator blades cascade, [P _{S1} -(1S-DS)]	
First static pressure tapping, upstream of second stage rotor blades cascade, [P _{S1} -(2R-US)]	157.5
Second static pressure tapping, downstream of first stage stator blade cascade, [P _{S2} -(1S-DS)]	
Second static pressure tapping, upstream of second stage rotor blades cascade, [P _{S2} -(2R-US)]	247.5
Third static pressure tapping, downstream of first stage stator blades cascade, [P _{S3} -(1S-DS)]	
Third static pressure tapping, upstream of second stage rotor blades cascade, [P _{S3} -(2R-US)]	337.5
Fourth static pressure tapping, downstream of first stage stator blades cascade, [P _{S4} -(1S-DS)]	
Fourth static pressure tapping, upstream of second stage rotor blades cascade, [P _{S4} -(2R-US)]	22.5
First static pressure tapping, downstream of second stage rotor blade cascade, [P _{S1} -(2R-DS)]	
First static pressure tapping, upstream of second stage stator blades cascade, [P _{S1} -(2S-US)]	112.5
Second static pressure tapping, downstream of second stage rotating blades, [P _{S2} -(2R-DS)]	
Second static pressure tapping, upstream of second stage stator blade cascade [P _{S2} -(2S-US)]	202.5
Third static pressure tapping, downstream of second stage rotor blade cascade, [P _{S3} -(2R-DS)]	
Third static pressure tapping, upstream of second stage stator blades cascade, [P _{S3} -(2S-US)]	292.5
Fourth static pressure tapping, downstream of second stage rotating blades, [P _{S4} -(2R-DS)]	
Fourth static pressure tapping, upstream of second stage stator blades, [P _{S4} -(2S-US)]	67.5
First static pressure tapping, downstream of second stage stator blade cascade, [P _{S1} -(2S-DS)]	
First static pressure tapping, upstream of third stage rotor blades cascade, [P _{S1} -(3R-US)]	157.5
Second static pressure tapping, downstream of the second stage stator blades, [P _{S2} -(2S-DS)]	
Second static pressure tapping, upstream of third stage rotor blades cascade, [P _{S2} -(3R-US)]	247.5
Third static pressure tapping, downstream of the second stage stator blades, [P _{S3} -(2S-DS)]	
Third static pressure tapping, upstream of the third stage rotor blades cascade, [P _{S3} -(3R-US)]	337.5
Fourth static pressure tapping, downstream of the second stage stator blades, [P _{S4} -(2S-DS)]	
Fourth static pressure tapping, upstream of third stage rotor blades cascade, [P _{S4} -(3R-US)]	22.5
First static pressure tapping, downstream of third stage stator blades cascade, [P _{S1} -(3S-DS)]	
Second static pressure tapping, downstream of the third stage stator blades, [P _{S2} -(3S-DS)]	112.5
Third static pressure tapping, downstream of third stage stator blades cascade, [P _{S3} -(3S-DS)]	202.5
Fourth static pressure tapping, downstream of third stage stator blades cascade, [P _{S4} -(3S-DS)]	292.5
The rotating blades turn clockwise, when looking towards the compressor. The angular datum (zero degree) for the specification of the angles of the static pressure tappings locations is positioned at the rightmost point of the compressor diameter, when this is taken horizontally with anticlockwise increments.	

[Table (4-3)]- The locations of the static pressure tappings in the compressor's outer casing

CHAPTER FIVE

5)- Characteristics of Three-Stage Compressor Configurations:

5.1)- Introduction:

The preliminary running of the Peregrine axial-flow compressor using the three stage build confirmed that the stable working range of this compressor is similar to other types of low-speed axial-flow compressors in that it is limited due to the appearance of the stall phenomenon at lower flow rates.

In order to provide sufficient data and information, which were essential for designing an appropriate vane-recessed tubular passage casing treatment as a stepped casing to fit into some part of the conventional casing of the test compressor, the following activities were essential.

- 1)- The compressor geometry specifications.
- 2)- The compressor characteristics specifications.

5.2)- Geometry Specifications of Three-Stage Compressor Builds:

The preliminary step of the experimental work had to be carried out using the Peregrine compressor with normal setting angles of rotating and fixed blades. To serve this requirement, the compressor was dismantled and the inlet guide vanes including the rotor and stator blades were removed from their related housing rings

The outer spacer rings in addition to the compressor hub rings were then measured to a high level of accuracy. The accurate schematic geometry of each spacer ring then had been drawn in order to provide individual parts of the outer casing wall and inner compressor's hub. For example the schematic drawing of the outer ring (No. 44462) that can be fitted before ring (No. 44463), in order to cover together the two half-rings which facilitate the (34) hole holder ring of the inlet guide vanes is shown in [Figure (5-1)].

As it was noted in the previous chapter the compressor was designed to provide three different axial blade gap arrangements. Since it was realised that the appropriate axial length is a vital parameter in design feature of a vane-recessed tubular passage treated casing, it was decided to consider the geometry, characteristics and performance of the Peregrine compressor under the highest available axial blade gaps.

This importance of this requirement led the author to the decision of using a three-stage build with an axial blade gap of 22.2 [mm]. It is worth noting that this arrangement was capable of facilitating an axial space ratio of (0.729) relative to the rotor blade chord length.

The schematic drawing of the compressor in the configuration described above was then produced, using all of the available individual drawings of the hub and casing spacer rings. The normal blade setting data corresponding to the nominal attitude and the designed stagger angles were also taken into consideration in the production of the schematic drawing. A cross section view of the compressor under nominal attitude conditions, covering the view of the rotating shaft centreline to the top of the outer casing including

the geometry of the spacer rings and position of the fixed and rotor blades settings is shown in [Figure (5-2)].

Even though the highest available axial blade gaps were used, the resulting axial length proved insufficient to design an appropriate casing treatment. This will be described in more detail in chapter seven. The present author believes this to have been one of the most significant constraints imposed on the design of the proposed casing treatment.

The preliminary examinations of the compressor were carried out after the calibrated measuring devices were fixed to the test rig as well as the compressor. It is worth noting that owing to a variety of factors, the early compressor performance characteristics were disappointing. In order to overcome these problems, different possible causes had to be considered in some detail before a proper diagnosis could be made, and appropriate solutions devised. Some of the actions taken at this stage of the study were as follows.

- 1)- Use of a cobra-type total pressure probe, instead of the three hole-pneumatic cylindrical pressure probe at the third stage stator blade cascade downstream.
- 2)- Repairing the load cell, changing the location of the transducer in the rig, re-calibrating in the connected conditions to the electrical motor's stator in association with air fitted to the air bearings.
- 3)- Applying two additional running speed devices, a low response stroboscope and a fast response hand held tachometer.
- 4)- Changing the previously used manometer.
- 5)- Re-fixing the electrical throttling.
- 6)- Re-arranging the speed regulation devices and the variable resistors.
- 7)- Re-fixing the outlet ducts and changing the paper-type honeycombs to aluminium-type ones in the two flow straighteners.
- 8)- Providing flow collectors at the venturi flow meter up and down streams, in addition to using identical length connecting flow measuring pipes to the manometers.
- 9)- Using the (T) form flow junction, and changing the entire flow measuring pipes to which with identical length and diameters.
- 10)- Using the computational calculation program during the period of rig running in order to take data in reasonably similar amounts of mass flow rate when the flow was reduced through the compressor, by closing of the orifice via electrical throttling.

To reassemble the compressor in the desired configuration, as presented in the figure (5-2), three blades stagger angles setting jigs were designed and manufactured. These jigs were used to set the inlet guide vanes, rotor and stator blades stagger angles by ($\xi_{IGVs}=18.45^\circ$), ($\xi_{Rotor}=36.3^\circ$), and ($\xi_{Stator}=36^\circ$), respectively in their related complete or two-half rings holder.

The compressor assemblies which were built by applying all the existing spacer rings followed by the inlet guide vanes, three stage rotors and stator blades in association with or without use of the honeycomb in the flow straighteners, are hereafter called builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)±(H)].

5.3)- Performance Specifications of Three-Stage Compressor Builds:

The problem of predicting the overall or stage characteristics of a multistage axial

flow compressor to a reasonable degree of accuracy is complex and in the majority of cases is time consuming. The need for a compressor performance specification therefore led to the idea of providing a calculation program to serve this requirement.

In order to find out the characteristics of the compressor in above-mentioned category, a calculation program had to be provided to serve the preliminary step of the experimental investigation. The written calculation program used for determining the performance characteristics of the Peregrine compressor, with minor amendments, also can be used with similar low-speed, as well as high-speed machines.

The overall or stage aerodynamic characteristics of a compressor can be achieved by running the calculation program using the available data, which were measured during the rig tests. Some of the characteristics including their related data measurements are described as follows.

5.4)- Calculations of the Mass Flow Rates:

The venturi flow meter of the Peregrine compressor test rig is a non-standard venturi, which has been manufactured as a long sheet steel tube with different inlet diameter of $[(D_{VFM})_{Inlet}]=406.4$ [mm] and outlet diameter of $[(D_{VFM})_{Throat}]=254$ [mm]. This non-standard venturi flow meter owing to its individual geometry had an impact on the flow discharge coefficient and therefore on the mass flow calculation process.

Since the venturi flow meter has an individual geometry, which is different from the standard ones, different relations including the calibration factor have been proposed by previous researchers and also the compressor's designer, to calculate the mass flow rate through the venturi. In order to find out accurate values of the mass flow rate, four similar proposals were considered in detail as follows. The test results were then compared by running the compressor rig in four different builds of the three-stage and the one-stage $[(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)\pm(H)]$ and $[(IGV)+(1R)+(1S)\pm(H)]$, respectively.

5.4.1)- Calculation of the Mass Flow Rates (Akhlaghi Procedure):

Since a venturi tube is a combination of a diffuser associated with a nozzle, the purpose of a venturi flow meter is to create a region of low static pressure at the venturi throat. The low static pressure then can be used to generate a pressure differential between the throat and the inlet static pressures.

For a compressible subsonic flow¹, the following relation can be used in order to calculate the mass flow rate (Φ) through a venturi.

$$\Phi = C_D * \epsilon * A_{VFM-Throat} * \left[\frac{1}{1 - (A_{VFM-Throat} / A_{VFM-Inlet})^2} \right]^{1/2} * [2 * \rho_{VFM-Inlet} * (p_{VFM-Inlet} - p_{VFM-Throat})]^{1/2} ; \quad [kg/sec] \quad [5-1]$$

Where: (C_D): The discharge coefficient².

¹ Although the changing of the corresponding Mach number during the current experimental study did not exceed than (0.25), but every attempt was made to measure data and calculate the relationship with the flow characteristics quite accurately.

² The discharge coefficient is the ratio of the actual flow rate to the ideal flow rate for a given static pressure differential.

(ϵ): The expansion coefficient.

($A_{\text{VFM-Throat}}$): The venturi flow meter throat cross section area, [m^2].

($A_{\text{VFM-Inlet}}$): The inlet cross section area of the venturi flow meter, [m^2].

($\rho_{\text{VFM-Inlet}}$): The density of the flow at the inlet of the venturi flow meter, [kg/m^3].

($p_{\text{VFM-Inlet}}$): The static pressure of the air at the inlet of the venturi flow meter, [Pa].

($p_{\text{VFM-Throat}}$): The static pressure of the flow at the venturi throat cross section, [Pa].

The venturi inlet and throat area cross sections [($A_{\text{VFM-inlet}}$) & ($A_{\text{VFM-Throat}}$)], then can be replaced by their corresponding diameters [($D_{\text{VFM-inlet}}$) & ($D_{\text{VFM-Throat}}$)], respectively.

Due to the application of the perfect gas relation, the density of the air at the inlet of the venturi flow meter ($\rho_{\text{VFM-Inlet}}$) then may be replaced as follows.

$$\rho_{\text{VFM-Inlet}} = \frac{p_{\text{VFM-Inlet}}}{R_{\text{Universal}} * T_{\text{VFM-Inlet}}}; \quad [\text{kg/m}^3] \quad [5-2]$$

Where: ($T_{\text{VFM-Inlet}}$): The temperature at the entrance of venturi flow meter, [K].

The temperatures of the flow at the entrance to ($T_{\text{VFM-Inlet}}$)³ and outlet of ($T_{\text{VFM-Throat}}$) the venturi have also been measured in [Celsius] by applying thermocouple devices. These temperatures however have to be converted to [Kelvin].

$R_{\text{Universal}}$: The universal gas constant, [Joul/kg*K]⁴.

The mass flow rate then can be calculated by using measuring results of the inlet pressure ($p_{\text{VFM-Inlet}}$) and inlet temperature ($t_{\text{VFM-Inlet}}$) of the venturi flow meter via following equation.

$$\Phi = C_D * \epsilon * \frac{\pi}{4} * (D_{\text{VFM-Throat}})^2 * \left[\frac{1}{1 - (D_{\text{VFM-Throat}} / D_{\text{VFM-Inlet}})^4} \right]^{\frac{1}{2}} * \left[\left(\frac{2 * p_{\text{VFM-Inlet}}}{R_{\text{Universal}} * t_{\text{VFM-Inlet}}} \right) * (p_{\text{VFM-Inlet}} - p_{\text{VFM-Throat}}) \right]^{\frac{1}{2}}; \quad [\text{kg/m}^3] \quad [5-3]$$

The venturi has a non-standard profile which is a compromise between [British Standard B.S.1042 (I.S.A.) conical and nozzle type dimensions]⁵, therefore the discharge coefficient was obtained about ($C_D=0.978$) by calibrating the venturi flow meter, (Ballal, 1968). In order to provide similar conditions, the discharge coefficient had been kept the same in all mass flow rate calculation compromising procedures.

The expansion coefficient (ϵ), can be determined by using the following relation. Similar equation in terms of the venturi inlet to throat cross section areas has also been given by Blevins (1984).

³ The static temperature will be within (2%) of the stagnation temperature for air at Mach numbers less than Mach=0.3, (Blevins, 1984).

⁴ The value of the universal gas constant which is the difference of the specific heat at constant pressure (C_P) and that at constant volume (C_V) for the air has been considered $R_{\text{Universal}} = C_P - C_V = 287.052$ [Joul/kg*K] at the entire part of the calculation program.

⁵ Flow measurements, British Standard Code, part one, 1042; (1964).

$$\varepsilon = \left\{ \left(\frac{\chi}{\chi - 1} \right) * (PR_{VFM})^{2/\chi} * \left[\frac{1 - (PR_{VFM})^{\chi-1/\chi}}{1 - (PR_{VFM})} \right] * \left[\frac{1 - (DR_{VFM})^4}{1 - (DR_{VFM})^4 * (PR_{VFM})^{2/\chi}} \right] \right\}^{1/2} \quad [5-4]$$

Where: (χ): The specific heats ratio⁶.

(PR_{VFM}): The venturi flow meter static pressures ratio.

This ratio can be determined by dividing the static pressure at the throat ($p_{VFM-Throat}$) to that one at the inlet ($p_{VFM-Inlet}$) of the venturi flow meter.

(DR_{VFM}): The venturi diameter ratio.

This is the ratio of the venturi throat diameter ($D_{VFM-Throat}$) to the inlet diameter ($D_{VFM-Inlet}$) of venturi flow meter.

5.4.2)- Calculation of the Mass Flow Rate (Peregrine Procedure):

The air mass flow rate in this procedure (Φ_1) can be determined as described in Peregrine booklets (1967).

$$\Phi_1 = 8.85 \sqrt{\frac{\Delta h(P_0 + \frac{h}{13.6})}{T_v}}; \quad [\text{lb/sec}] \quad [5-5]$$

Where; P_0 : Atmospheric pressure, [Inch-Hg].

Δh : Drop in static pressure between inlet and throat of venturi air meter, [Inch-H₂O].

h : static pressure of air entering venturi, [Inch-H₂O].

T_v : Temperature of air entering venturi, [K]

5.4.3)- Calculation of the Mass Flow Rate (Ballal Procedure):

The following relation was used in order to calculate the air mass flow rate.

$$\Phi_2 = 8.95 \sqrt{\frac{\Delta h(P_0 + \frac{h}{13.6})}{T_v}}; \quad [\text{lb/sec}] \quad [5-6]$$

By comparing the last two mass flow calculation procedures, it can be seen that the only difference between two applied formulas is the use of coefficients, (8.95), instead (8.85).

A description of the calibration of the non-standard venturi has been provided by Ballal (1968). It seems that the discharge coefficient of (0.978), which had been calculated from the calibration of the venturi, resulted in a different applied coefficient of (8.95).

In both above-mentioned procedures the obtained air flow rate was multiplied by (0.4536) to convert from [lb/sec] to [kg/sec].

5.4.4)- Calculation of the Mass Flow Rate (Giannissis Procedure):

Both Giannissis (1987) and Farmakalides (1992) which conducted their research

⁶ The specific heats ratio is the ratio of the specific heat at constant pressure (C_p) to that at constant volume (C_v). This ratio has been considered ($\chi=1.40015$) in all calculation program. It is worth noting that the value of (C_p) rises with increasing of temperature and it is usual to specify the data by the value of (χ), which falls from 1.4 at 250 [K] to 1.35 at 850 [K].

using the same test rig applied the following equation⁷ in order to calculate the air mass flow rate.

$$\Phi_3 = (4.33 * 10^{-2}) * \sqrt{(\Delta P_v) * \left(\frac{P_a + 98.02 P_{v1}}{T_{v1}} \right)}; \quad [\text{kg/sec}] \quad [5-7]$$

Where; ΔP_v : Drop in mean static pressure of air through the venturi in [Cm-H₂O].

P_a : Atmospheric pressure in [N/m²].

P_{v1} : Mean wall static pressure of air entering the venturi in [Cm-H₂O].

T_{v1} : Temperature of air entering the venturi flow meter.

5.4.5)- Corrected Mass Flow Rate:

In view of the non-negligible effects of varying inlet temperature and pressure on the compressor performance, and also the necessity to minimise uncertain effects from the obtained results in order to provide the best comparison of the compressor characteristics, the corrected mass flow rate is quite a helpful parameter.

The majority of the obtained experimental results however showed that the corrected mass flow rate provided the most accurate results in comparison to both the flow coefficient and the uncorrected mass flow rate.

By applying the standard pressure and the standard temperature⁸, the corrected mass flow rate (Φ_E) was then calculated by using the following relation.

$$\Phi_E = \left(\frac{P_{\text{Standard}} * \Phi}{P_{\text{Total-Inlet}}} \right) * \left(\frac{T_{\text{Inlet}}}{T_{\text{Standard}}} \right)^{1/2}; \quad [\text{kg/sec}] \quad [5-8]$$

Four different values of mass flow rate were calculated using the above-noted alternatives and based on the obtained data from the three stage compressor builds and in the presence of the honeycombs in the flow straighteners is shown in [Graph (5-1)].

Three different magnitudes of the mass flow rate difference percentage $[(\Delta\Phi)\%]$, which were determined relative to (Akhlaghi Procedure) by using the experimental results of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], is illustrated in [Graph (5-2)].

5.5)- Nominal and Corrected Running Speeds:

Although the instruction leaflet of the low-speed three-stage compressor, (Peregrine booklets, 1967) presents an overall specification data for the machine, it does not express the design point related features in more details. The design speed, which is hereafter called the nominal running-speed, was given as about $N=3000$ [rpm]. Desirable magnitude of the speed however is provided by the rig's direct current electrical motor, which could facilitate a running speed between 100 to 3600 [rpm].

The corrected running-speed then can be determined by using inlet total temperature (T_{Inlet}) and standard total temperature (T_{Standard}) as follows.

⁷ See the references, (Peregrine booklets, 1967). This equation had been supplied on the booklet, "Characteristics of a model low-speed axial-flow compressor, SI Units supplement part".

⁸ The standard pressure and standard temperature have been considered 101.325 [kpa] and 288.15 [K], respectively.

$$N_{\text{Corrected}} = N * \left(\frac{T_{\text{Standard}}}{T_{\text{Inlet}}} \right)^{1/2}; \quad [\text{rpm}] \quad [5-9]$$

It is worth noting that due to the fact that the variation of the running-speed; even though small in value could possibly affect the compressor's characteristics under different circumstances. In order to avoid any undesirable effects of variation in rotor blade tangential velocity during the operation of the compressor, both with and without the casing treatment, it was decided to keep the running speed as the nominal one during the data collection process for determining the flow characteristics.

5.5.1)- Rotating Blades Circumferential Velocity:

The rotor blade height was 61 [mm]. By taking into account the compressor's hub diameter of 284.5 [mm], the rotor mid diameter was of 315 [mm]. The normal blade settings and axial pitching of the test compressor was adopted at a mean diameter of 335.3 [mm] as datum for aerodynamic expediency.

Therefore it is essential to consider the entire blade configuration in relation to this reference which is hereafter called mean diameter (D_{Mean}).

The circumferential velocity of the moving blades at reference diameter (U_{Mean}) was calculated by using the following formula.

$$U_{\text{Mean}} = \left(\frac{\pi * D_{\text{Mean}} * N}{60} \right); \quad [\text{m/sec}] \quad [5-10]$$

5.6)- Flow Coefficient in the Compressor Annulus:

It is well known that the work input to an axial flow compressor is related to the ratio of axial velocity (V_a) to the blade tangential velocity (U) which is normally called flow coefficient (ϕ). Most axial flow compressor are designed to have a flow coefficient in the range (0.3~0.9), but nowadays compressors which are at the lower end of this range are more common, (Cumpsty, 1989).

The Peregrine compressor under three-stage builds and normal running conditions has shown a flow coefficient range of (0.4~0.7). The compressor was operated with and without a honeycomb in the flow straighteners that caused the flow coefficient to slightly change. Without application of the honeycombs, the flow coefficient showed a small increment at both ends.

Due to the application of the designed vane-recessed tubular passage casing treatment, the ranges of flow coefficients were significantly altered in the all seven different rotor blade axial chord exposure ratios.

Although the lower axial chord exposures showed a wider range in flow coefficient magnitudes, the increments of this compressor crucial flow parameter with the higher exposures were also quite significant.

Flow coefficient at compressor annulus was calculated as follows.

$$\phi = \frac{V_{\text{Axial}}}{U_{\text{Mean}}} = \frac{60 * \Phi}{\pi * \rho_{\text{Mean}} * A_{\text{Annulus}} * D_{\text{Mean}} * N} \quad [5-11]$$

Where; (V_{Axial}) is the axial flow velocity of air at the compressor annulus cross section area (A_{Annulus}) and can be determined through the following relation.

$$V_{\text{Axial}} = \frac{\Phi}{\rho_{\text{Mean}} * A_{\text{Annulus}}}; \quad [\text{m/sec}] \quad [5-12]$$

While: (ρ_{Mean}) was defined as air mean-density at the compressor constant annulus and can be find out via following formula.

$$\rho_{\text{Mean}} = \frac{(P_{\text{Total}})_{\text{Mean}}}{R_{\text{Universal}} * T_{\text{Mean}}}; \quad [\text{kg/m}^3] \quad [5-13]$$

Mean total pressure in the compressor annulus $[(P_{\text{Total}})_{\text{Mean}}]$, was calculated from the total pressures obtaining at the inlet to and outlet of compressor as follows.

$$(P_{\text{Total}})_{\text{Mean}} = 0.5 * [(P_{\text{Total}})_{\text{Inlet}} + (P_{\text{Total}})_{\text{Outlet}}]; \quad [\text{Pa}] \quad [5-14]$$

The ambient pressure (P_{Ambient}) considered as inlet total pressure $[(P_{\text{Total}})_{\text{Inlet}}]$, whereas the outlet total pressure $[(P_{\text{Total}})_{\text{Outlet}}]$ was measured by applying a three-hole cobra-type total pressure probe at a plane in the downstream of the third stage stator blade cascade.

The mean total temperature $[(T_{\text{Total}})_{\text{Mean}}]$ was also determined by averaging the inlet total temperature $[(T_{\text{Total}})_{\text{Inlet}}]$ and the outlet total temperature $[(T_{\text{Total}})_{\text{Outlet}}]$ which were measured by using two thermocouple temperature probes in the compressor intake plane and at the exit of the compressor, respectively.

5.6.1)- Axial Velocity of Air Flow at the Design Point:

The blade geometry of an axial flow compressor including the rotating and stationary blade radial distribution and setting angles are normally designed in order to provide the peak efficiency at its corresponding design point. By using the rotor blade's inlet and outlet absolute velocity angles of $(\alpha_1=50^\circ)$ and $(\alpha_2=30^\circ)$ at mean diameter, the flow axial velocity magnitude at the design point is determined as follows.

$$(V_a)_{\text{Design-Point}} = \frac{\pi * D_{\text{Mean}} * N}{60 * (\tan \alpha_1 + \tan \alpha_2)}; \quad [\text{m/sec}] \quad [5-15]$$

5.7)- Isentropic Temperature Rises (Overall):

The results which were obtained from running the different configurations of the three-stage compressor with and without the effects of the total pressure probe traversing (TPPT), builds $[(\text{IGV})+(1\text{R})+(1\text{S})+(2\text{R})+(2\text{S})+(3\text{R})+(3\text{S})\pm(\text{H})\pm(\text{TPPT})]$ confirmed that the overall isentropic temperature rise was in the range of (1.1~2.7).

These ranges however were achieved when the throttle was fully opened and at near stall conditions, respectively. It means that the test compressor was designed in order to produce about 0.9 [Celsius] increase in the temperature values per each stage.

The overall isentropic temperature rise $[(\Delta T_s)_{\text{Overall}}]$ was calculated by the application of the total inlet and total outlet pressures in addition to the inlet total temperature through the following calculation.

$$(\Delta T_s)_{\text{Overall}} = [(T_{\text{Total}})_{\text{Inlet}}] * \left\{ \left[\frac{(P_{\text{Total}})_{\text{Outlet}}}{(P_{\text{Total}})_{\text{Inlet}}} \right]^{\frac{(\chi-1)}{\chi}} - 1 \right\}; \quad [\text{K}] \quad [5-16]$$

While; (χ) is the specific heat ratio, which had been considered (1.40015) in the provided calculation program.

5.8)- Mach Number Under Compressor Operating Conditions:

As the Peregrine compressor was a low-speed three-stage compressor, the flows through the machine as well as the inlet and outlet flows were under subsonic conditions, during the current experimental investigation. It means that the effect of compressibility is not significant, due to the small variations in the air densities.

Having said that, since the idea of using a tubular passage vane-recessed treated casing have been considered to serve in high-speed industrial machines as well as aero-engine compressor. Therefore it was decided to write the compressor performance calculation program to cover above-noted opinion.

The Mach number, which is the ratio of the local velocity to the local velocity of sound, was determined as follows.

$$\text{Mach} = \frac{\Phi}{\rho_{\text{Mean}} * A_{\text{Annulus}} * (\chi * R_{\text{Universal}} * T_{\text{Mean}})^{1/2}} \quad [5-17]$$

Where: ($R_{\text{Universal}}$) is the universal gas constant, and for the air has been considered of 287.052 [Joul/kg*K].

McKenzie (1997) stated that while industrial fans generally operate at low enough velocities for Mach number effects to be negligible, the gas turbine and industrial compressor operate at high subsonic or even lower subsonic velocities and are designed with attention to the influence of the Mach numbers involved.

It is worth noting that even the upstream Mach number in a compressor is as low as (0.5), still local velocities within a blade passage may reach the local speed of sound and modify the flow. Mach number under operating condition of three stage compressor which was built using the honeycombs in the flow straighteners is shown in [Graph (5-3)].

5.9)- Reynolds Number Based on the Rotor Blade Chord:

Both industrial and aero-engine types compressors are very sensitive to flow shear stresses. It means that due to the flow conditions which can be laminar or turbulent, the test compressor could possibly be affected by a fairly significant characteristic variations.

Under conditions in which the flow is laminar, the shear stresses are low, and therefore the corresponding losses are small. On the other hand, when the flow is turbulent, an extensive region of separated flow restricts the performance of the compressor to a lower pressure rise and a smaller efficiency achievement.

The values of Reynolds numbers, which in the turbomachinery field area is defined, based on the machine's length and related flow characteristics are usually large. The Reynolds number is a non-dimensional parameter, which is defined as a ratio of the inertia forces of the flow to the related viscous forces.

The Reynolds number of the test compressor was considered through the following relation, by using the rotor blade chord as compressor's characteristic length

$$\text{Re} = \frac{\rho_{\text{Mean}} * C_{\text{Rotor}} * V_{\text{Axial}}}{\mu} = \frac{C_{\text{Rotor}} * \Phi}{\mu * A_{\text{Annulus}}} \quad [5-18]$$

Where, [(C)_{Rotor}]: Chord of the rotor blade in [m].

μ : Dynamic viscosity of air, in $\left[\frac{\text{N} * \text{sec}}{\text{m}^2} \right]$

Experimental results obtained from using the three-stage compressor configurations between the choke and stall conditions confirmed that in the majority of cases except the builds [(IGV)+(1R)±(H)] the flow in the compressor was laminar. Reynolds number of three stage compressor in the presence of the honeycombs in the flow straighteners is shown in [Graph (5-4)].

5.10)- Work Coefficient (Overall):

The pressure and shear stresses in a compressor blade provide momentum about the machine's axis. Under real conditions, it is not possible to evaluate those stresses. To overcome this problem it is usual in turbomachinery to consider the moment of momentum for the flow enters and leaves the compressor. The electrical motor which is connected to the compressor's shaft, produces a circumferential force on the blades and the work done by such force moving is transmitted to the airflow, (McKenzie, 1997).

In accurate analysis and design work it is necessary to take into account, the effects of the radial changes across the blade row, but it is sufficient to present the total energy transfer to the airflow (ΔH) as follows.

$$\Delta H = U * \Delta V_w = \frac{\tau * \omega}{\Phi}; \quad [\text{Joule/kg}] \quad [5-19]$$

Where, ($U = r * \omega$) is the blade tangential velocity, (ΔV_w) is the change of whirl velocity across the rotor blades, (τ) is the torque and (ω) is the angular velocity of blades.

The validation of the Euler equation is not restricted by the matter of how the moment of momentum is produced, and even viscous drag on the blade can produce a positive work input or not. In the majority of cases the tangential stresses at the compressor casing and the hub are usually very small. Therefore, the total work of the blade row can usually be determined from the measurements of the velocity components in the upstream and downstream of the blade cascade. It is worth noting that in particular cases, such as the condition of casing treatment application in an axial flow compressor, there may be large effective stresses at the walls, (Cumpsty, 1989).

Overall work coefficient (Ψ) of the test compressor however was calculated through application of the following equation.

$$\Psi = [(\omega * \tau_{\text{Applied}}) / (\Phi * U^2)] \quad [5-20]$$

Angular velocity (ω) had been considered of 314.159265 [rad/sec], by using the following equation during entire parts of the current experimental investigation due to the fact that the running speed of the compressor was kept constant of 3000 [rpm].

$$\omega = \frac{2 * \pi * N}{60}; \quad [\text{rad/sec}] \quad [5-21]$$

The multiples of the applied torque (τ_{Applied}), and the rotor angular velocity facilitate the magnitude of the applied power (Ω_{Applied}). The characteristic of the overall work coefficient using results obtained from the compressor testing under circumstance of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)] is presented in [Graph (5-5)].

5.10.1)- Corrected Applied Power:

As it was stated in previous chapter, the effective load ($W_{\text{Effective}}$) that is necessary to balance the electrical motor's stator can be determined through following equation.

$$W_{\text{Effective}} = 0.8076 * \tau_{\text{Net}} - 0.0091; \quad [\text{kg}] \quad [5-22]$$

(τ_{Net}) is the torque transducer output in [millivolt], which corresponds to the load magnitude. The obtained effective load is related to the applied torque, which is essential to rotate the compressor's shaft. During the test rig examinations the applied torque (τ_{Applied}) however, was found via following equation.

$$\tau_{\text{Applied}} = L_{\text{Arm}} * g * W_{\text{Effective}}; \quad [\text{N*m}] \quad [5-23]$$

(L_{Arm}) is the torque arm of 254 [mm] and was connected to the motor's stator from one side and to torque transducer indicator from the other side. The acceleration due to the gravity at the surface of the Cranfield University Aerodynamic Laboratory Test Area was considered of 9.81302 [m/sec²].

The applied power (Ω_{Applied}) then can be calculated as follows.

$$\Omega_{\text{Applied}} = \frac{\omega * \tau_{\text{Applied}}}{1000}; \quad [\text{Kwatt}] \quad [5-24]$$

The corrected applied power (Ω_{E}) then has been intended by using the following relation.

$$\Omega_{\text{E}} = (P_{\text{Standard}} / P_{\text{Total-Inlet}}) * (T_{\text{Standard}} / T_{\text{Inlet}})^{1/2} * \Omega_{\text{Applied}}; \quad [\text{Kwatt}] \quad [5-25]$$

Corrected applied power characteristic of three-stage compressor build in the present of the honeycombs in the flow straighteners is shown in [Graph (5-6)].

5.11)- Pressure Rise Coefficients (Overalls and Individual Stages):

The performance of an axial flow compressor can be expressed, using the pressure rise coefficient (ψ) versus flow coefficient (ϕ) or mass flow rate (Φ), characteristics. The experimental results obtained from the different builds of the Peregrine compressor also confirmed the idea of the previous researcher that it is better to use the corrected mass flow rate (Φ_{E}) to present the compressor performance.

Pressure rise coefficient is generally considered in non-dimensional form by following two more or less similar procedures for complete stage or all compressor stages.

5.11.1)- Pressure Rise Coefficients in terms of the Pressure Rises:

The overall and individual pressure rise coefficients of a multistage compressor can be found in terms of pressure rises, which occur through entire stages or just take place in an individual stage. It means that by applying the overall pressure rise obtained from a multistage compressor, the overall pressure rise coefficient can be calculated whereas, the usage of each individual stage pressure rise just provides the corresponding stage pressure rise coefficient.

Non-dimensional pressure rise coefficient in terms of pressure rise ($\psi_{\Delta P}$) can be determined as follows.

$$\psi_{\Delta P} = \frac{P_{\text{Outlet}} - P_{\text{Inlet}}}{\rho * U^2} = \frac{\Delta P}{\rho * U^2} \quad [5-26]$$

Where: (P_{Outlet}) and (P_{Inlet}) are the exit and intake pressures which exit from or entering into the compressor or individual stage, respectively.

(U), is the rotating blades tangential velocity.

5.11.2)- Pressure Rise Coefficients in terms of the Pressure Ratios:

Non-dimensional pressure rise coefficient in terms of pressure ratio (ψ_{PR}) obtained from outlet and inlet pressures [$PR = (P_{\text{Outlet}}/P_{\text{Inlet}})$] can also be calculated through the following equation.

$$\psi_{PR} = \frac{C_P * T_{\text{Inlet}}}{U^2} * \{(PR)^{[(\chi-1)/\chi]} - 1\} \quad [5-27]$$

(C_P) is the heat coefficient at the constant pressure attitude and had been considered as 1004.410 [Joul/kg*K] in the whole of the calculation program.

5.11.3)- Pressure Rise Coefficients in terms of Total and Static Pressures:

By measuring the casing static pressure between the blade rows of an individual stage, the characteristics can be derived from multistage tests and provide a useful analysis criterion for performance specifications, (McKenzie, 1997).

In order to provide a comparison between the experimental results obtained from different build configurations it was decided to calculate the non-dimensional pressure rise coefficients, using three different pressure ratios in the following four forms:

1)- Pressure rise coefficient (overall total-total), $[(\psi_{T-T})_{\text{Overall}}]$:

$$(\psi_{T-T})_{\text{Overall}} = \frac{C_P * T_{\text{Inlet}}}{(U_{\text{Mean}})^2} * \{[(PR_{T-T})_{\text{Overall}}]^{[(\chi-1)/\chi]} - 1\} \quad [5-28]$$

$[(PR)_{T-T}]$ is the pressure ratio which could be obtained from the ratio of total pressure at the machine exit $[(P_T)_{\text{Outlet}}]$ to the total pressure entering the compressor $[(P_T)_{\text{Inlet}}]$. In the majority of cases the total pressure corresponding to the flow under stagnation enthalpy condition is considered as inlet total pressure.

In the current research however, the ambient pressure (P_{Ambient}) was taken as inlet total pressure. The atmospheric (ambient) pressure was measured by means of a direct reading column-type mercury barometer located in the top of the aerodynamic laboratory. The outlet total pressure was measured via a two-dimensional cobra-type total pressure probe which was located in the third stage stator blades downstream over the whole of the present investigation.

2)- Pressure rise coefficient (overall static-static), $[(\psi_{S-S})_{\text{Overall}}]$:

$$(\psi_{S-S})_{\text{Overall}} = \frac{C_P * T_{\text{Inlet}}}{(U_{\text{Mean}})^2} * \{[(PR_{S-S})_{\text{Overall}}]^{[(\chi-1)/\chi]} - 1\} \quad [5-29]$$

$[(PR_{S-S})_{\text{Overall}}]$ is the pressure ratio that can be determined from the ratio of the static pressure in the third stage stator blade downstream plane $[(P_S)_{\text{Outlet}}=(P_S)_{3S-DS}]$ to the static pressure entering the compressor in the upstream plane of the inlet guide vanes $[(P_S)_{\text{Inlet}}=(P_S)_{\text{IGV-US}}]$.

3)- Pressure rise coefficient (overall total-static), $[(\psi_{T-S})_{\text{Overall}}]$:

$$(\psi_{T-S})_{\text{Overall}} = \frac{C_P * T_{\text{Inlet}}}{(U_{\text{Mean}})^2} * \{[(PR_{T-S})_{\text{Overall}}]^{[(\chi-1)/\chi]} - 1\} \quad [5-30]$$

$[(PR)_{T-S}]$ is the pressure ratio which is calculated from the ratio of static pressure in the plane of the third stage stator blade downstream $[(P_S)_{\text{Outlet}}=(P_S)_{3S-DS}]$ to the ambient pressure entering the compressor $[(P_T)_{\text{Inlet}}=(P_T)_{\text{Ambient}}]$.

Three non-dimensional pressure rise coefficient characteristics in terms of overall (total-total), (static-static) and (total-static) pressure ratios versus corresponding flow coefficients (ϕ), of builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], is shown in [Graph (5-7)].

4)- Pressure rise coefficient (stage static-static), $[(\psi_{S-S})_{\text{Stage}}]$:

$$(\psi_{S-S})_{\text{Stage-No.}} = \frac{C_P * T_{\text{Inlet}}}{(U_{\text{Mean}})^2} * \{[(PR_{S-S})_{\text{Stage-No.}}]^{[(\chi-1)/\chi]} - 1\} \quad [5-31]$$

(No.) is related to specification number of stage.

As described in the previous chapter, the three-stage compressor was configured for inter row wall static pressure measurements. This enables the measurement of the static pressure in the upstream and downstream planes of each individual stage.

Stage static-static pressure ratio of an individual stage can be determined from the ratio of stage downstream static pressure $[(P_S)_{\text{No.S-DS}}]$ to the same stage upstream static pressure $[(P_S)_{\text{No.R-US}}]$. The situation of the inter-row wall static tappings was described in more detail in the previous chapter.

The application of differential thermocouples for measuring the disturbance of both compressor inlet and outlet temperature confirmed that the temperature increment through the three-stage Peregrine compressor was small, about less than 3.0 [C]. This temperature rise agreed with the figure, which had already been reported in Peregrine booklets (1967) of a 0.882 [C] temperature rise per each stage. Therefore the use of compressor inlet temperature (T_{Inlet}) was considered as each individual stage inlet temperature.

Pressure rise coefficients of each of the three individual stages in terms of (static-static) pressure ratio $[(\psi_{S-S})_{\text{Stage-No.}}]$ versus flow coefficient (ϕ), based on the experimental results of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], is shown in [Graph (5-8)].

It is worth noting that for compressor performance analysis purposes the pressure rise coefficient of each rotating and fixed blade rows have been calculated and their related characteristics were plotted against their corresponding corrected mass flow rates or flow coefficients.

5.12)- Efficiencies (Overall and Individual Stage):

It is well known that the term efficiency has a very widespread application in turbomachinery as well in specification of axial flow compressors. The efficiency is defined as ratio of work input into the ideal compressor to that of actual one. Work input into a compressor under an adiabatic circumstance is equal to the increment of stagnation enthalpy.

Although the efficiency is normally used as a significant characteristic in a complete stage or whole compressor, it is also convenient to apply it for cascade or blade rows to relate the static pressure rise to the drop in stagnation pressure.

It can be convenient however to consider efficiency as the ratio of the actual pressure rise $[(\Delta P)_{\text{Actual}}]$ to the ideal work input to the compressor, i.e. without any associated losses, as indicated as follows.

$$\eta = \frac{(\Delta P)_{\text{Actual}}}{(\Delta P)_{\text{Ideal}}} = \frac{(\Delta P)_{\text{Actual}}}{\rho * (\Delta H)} \quad [5-32]$$

Where; (ΔH) is the enthalpy rise across the compressor.

5.12.1)- Efficiencies in terms of the Pressure Rises:

By using the pressure rise across the machine $[(\Delta P) = (P_{\text{Outlet}}) - (P_{\text{Inlet}})]$, the efficiency in terms of pressure rise ($\eta_{\Delta P}$) is calculated as follows.

$$\eta_{\Delta P} = \frac{\Phi * \Delta P}{\rho * \omega * \tau_{\text{Applied}}} \quad [5-33]$$

The above equation can be applied to the whole compressor or just to an individual stage, corresponding to the inlet and outlet pressure circumstances.

5.12.2)- Efficiencies in terms of the Pressure Ratios:

Efficiency in terms of pressure ratio (η_{PR}) can be determined through the following equation by using outlet and inlet pressures $[PR = (P_{\text{Outlet}} / P_{\text{Inlet}})]$ from and to either individual compressor stage or the whole machine.

$$\eta_{PR} = \frac{\Phi * C_p * T_{\text{Inlet}} * \{(PR)^{[(\gamma-1)/\gamma]} - 1\}}{\omega * \tau_{\text{Applied}}} \quad [5-34]$$

5.12.3)- Efficiencies in terms of the Total and Static Pressures:

To calculate the overall or the individual stage efficiencies, it is usual to consider the inlet flow in the stagnation enthalpy condition. But the circumstances of the outlet flow can be taken into account either under stagnation or static enthalpy conditions.

The efficiencies which are calculated by using both inlet and outlet parameters based on the stagnation enthalpy conditions are called total-to-total efficiency, (η_{T-T}). When the flow at the compressor exit is considered to be under static enthalpy conditions and the inlet one can be maintained in stagnation, then the calculated efficiency is called total-to-static efficiency (η_{T-S}).

To answer the question, as to which one of these is more appropriate for evaluating compressor efficiency, is related to what significance is attributed to the corresponding pressure ratio on the performance of the compressor. It seems however that when the outlet dynamic pressure is beneficial in terms of total energy, then the stagnation outlet conditions are more relevant, but when the dynamic pressure will be wasted for any reason then the static outlet condition can be far more appropriate.

According to Cumpsty (1989), when the outlet flow is uniform and the Mach number known, the stagnation and static conditions can be interchanged very easily for the reversible process, when assuming air as an ideal gas. If conditions are not uniform, there is no simple procedure for converting between total-total and total-static efficiencies.

The total-total efficiency is always higher than the total-static and there is an incentive to use it when the aim is to stress the quality of a compressor. For compressors with very high-pressure ratios the difference between the outlet stagnation and static pressures becomes a small fraction of the overall pressure rise and the distinction between the two efficiencies ceases to be significant.

Although both the overall and individual stage static-static efficiencies in view of the uncertainties related to accurate pressure measuring is not widely defined in the majority of the theoretical literature in the field of axial flow compressors. It was experimentally found that owing to the inter-row static pressure measurement facility, it is also possible to calculate the overall and stage efficiencies in order to provide more characteristics in

compromising the compressor performance under different builds circumstances.

Efficiency was found to be one of the most significant characteristics in the performance assessments of the test compressor builds. Effects of efficiencies however were even more crucial in the area of comparing the compressor with or without casing treatment. Six different magnitudes of efficiencies have been calculated during the present research under each build of [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)±(H)].

Three overalls and three stages related efficiencies were calculated in respect to the overall [(total-total), (static-static), (total-static)] and stage static-static pressure ratios. By using four different pressure ratios the efficiencies were determined in the four following forms.

5)- Efficiency (overall total-total), $[(\eta_{T-T})_{Overall}]$:

$$(\eta_{T-T})_{Overall} = \frac{\Phi * C_p * T_{Inlet}}{\omega * \tau_{Applied}} * \{[(PR_{T-T})_{Overall}]^{[(\chi-1)/\chi]} - 1\} \quad [5-35]$$

The overall total-total pressure ratio $[(P_{S-S})_{Overall}]$ can be find by using following equation.

$$(PR_{T-T})_{Overall} = [(P_T)_{3S-DS} / (P_T)_{Inlet}] \quad [5-36]$$

6)- Efficiency (overall static-static), $[(\eta_{S-S})_{Overall}]$:

$$(\eta_{S-S})_{Overall} = \frac{\Phi * C_p * T_{Inlet}}{\omega * \tau_{Applied}} * \{[(PR_{S-S})_{Overall}]^{[(\chi-1)/\chi]} - 1\} \quad [5-37]$$

The overall static-static pressure ratio $[(PR_{S-S})_{Overall}]$ can be calculated as follows.

$$(PR_{S-S})_{Overall} = [(P_S)_{3S-DS} / (P_S)_{IGV-US}] \quad [5-38]$$

7)- Efficiency (overall total-static), $[(\eta_{T-S})_{Overall}]$:

$$(\eta_{T-S})_{Overall} = \frac{\Phi * C_p * T_{Inlet}}{\omega * \tau_{Applied}} * \{[(PR_{T-S})_{Overall}]^{[(\chi-1)/\chi]} - 1\} \quad [5-39]$$

The overall total-static pressure ratio $[(PR_{T-S})_{Overall}]$ can be calculated as follows.

$$(PR_{T-S})_{Overall} = [(P_S)_{3S-DS} / (P_T)_{Ambient}] \quad [5-40]$$

The overall total-total, static-static and total-static efficiency characteristics of Peregrine compressor under build of [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)] against corrected mass flow rate ($\Phi\epsilon$) is shown in [Graph (5-9)].

8)- Efficiency (stage static-static), $[(\eta_{S-S})_{Stage}]$:

$$(\eta_{S-S})_{No.-of-Stage} = \frac{\Phi * C_p * T_{Inlet}}{\omega * \tau_{Applied}} * \{[(PR_{S-S})_{No.-of-Stage}]^{[(\chi-1)/\chi]} - 1\} \quad [5-41]$$

Each stage static-static pressure ratio $[(PR_{S-S})_{Stage-No.}]$, then can be determined by the ratio of corresponding outlet static at stator blade downstream over the related inlet static pressure at the rotor blade cascade upstream, as follows.

$$(PR_{S-S})_{Stage-No.} = [(P_S)_{Stage-No.-(S-DS)} / (P_S)_{Stage-No.-(R-US)}] \quad [5-42]$$

The efficiencies of each three individual stage $[(\eta_{S-S})_{Stage-No.}]$ in term of (static-static) pressure ratios versus the related corrected mass flow rate ($\Phi\epsilon$), based on the results of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], is illustrated in [Graph (5-10)].

5.13)- Effects of the Cobra-Type Total Pressure Probe Traversing:

By definition total pressure in each plane of an axial flow compressor is the pressure attained if the air is brought to rest isentropically and adiabatically. In practice by introducing an obstacle into the moving air it is possible to create a stagnation point. The pressure at such point is referred to as stagnation pressure, which is not necessarily equal to the local total pressure of the undisturbed flow.

The cobra-type traverse pressure probe used in the current investigation was a steady-state multipurpose instrument, consisted of three holes, each with 0.8 [mm] inside diameter. Such pressure-measuring device is normally suitable in two-dimensional flow measurement.

It is worth noting that the results from using a 4 [mm] diameter three hole pneumatic cylindrical total pressure probe did not produce satisfactory total-total characteristics. Therefore, it was thought that the cobra-type probe might offer less flow-disturbance to the air, due to its smaller size. Although the cobra-type probe construction was not as simple as the pneumatic cylindrical-type one, it was felt that it would be more sensitive in terms of rapid response in comparison with the pneumatic probe.

On the other hand most mounting devices for pressure probes are arranged in order to provide a means for successively positioning the probe at various points in the flow. Provision of a single linear traverse with rotation about the axis of traverse offers few mechanical problems.

Traversing and orientation gear generally includes devices, for which the majority of mechanical parts can be placed outside the flow, leaving more of the probe and its support immersed in the stream. This arrangement places few restrictions on the size and type of the mechanism, which can be used, but is likely to be applicable to compressors with low and high speeds, (Bryer et al., 1971).

It was found that still with cobra-type probe, the measured total pressure in the plane of the third stage stator downstream was a little bit higher than the value which was expected, based on the same plane corresponding wall static-pressure. This however led to an idea of boundary layer effectiveness and a requirement of a total pressure traversing in the compressor annuls cross section.

In order to find out the effects of the total pressure probe traversing (TPPT), on the performance of the test compressor, the cobra-probe was located in the sixteen additional locations in the third stage stator blade downstream. Under normal testing conditions the pressure probe was positioned in the same axis of the rotor blade mid diameter $[(D_R)_{Mid}]$.

During the traversing activities, the total pressure in addition to the manometer datum pressure were measured in double eight locations from rotor mid diameter axis up to the casing and down towards the compressor hub. Due to the effects of boundary layer near annulus hub and casing the probe location distances had been reduced when it was approaching the compressor walls.

Total pressures in addition to their corresponding datum pressures were measured at the mid diameter axis and in the eight different positions from each side at distances of 7,13,19,25,26,27,28 and 29 [mm], respectively.

The total pressure traversing results of the compressor at ten different flow coefficients related to the circumstances of the throttling positions from fully opening to near stall conditions under builds $[(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)+(TPPT)]$, is shown in [Table (5-1)].

The characteristics, which were obtained from the calculation results, showed that except for both the overall total-total efficiency and pressure rise coefficient, the changes of characteristics for build including total pressure probe traversing effects were not significant. The characteristics of the overall total-total efficiencies and pressure rise coefficients of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], with and without the effect of the cobra-type total pressure probe traversing are presented in [Graph (5-11)].

As it can be seen the total pressure probe traversing reduced the both values of efficiency and pressure rise coefficient. The reductions of efficiency were about (13.12%) in the situation which throttling was fully opened whereas, near the stall condition the efficiency showed less drop of (3.89%). It is worth noting that the overall total-total pressure rise coefficients $[(\psi_{T-T})_{\text{Overall}}]$ in the above graph have been determined by halving the results obtained from the equation [5-28]. This parameter was decreased about (3.57%) and (2.28%) in the choke and near stall conditions, respectively.

The calculation results of the overalls and stages efficiencies and pressure rise coefficients using (total-total), (static-static) and (total-static) pressure ratios, with and without effects of total pressure probe traversing of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)] are presented in [Table (5-2)].

The characteristics of the overall and stage efficiencies and pressure rise coefficients, obtained from the results of the compressor builds with and without effectiveness of the cobra-type total pressure probe traversing are shown in [Graph (5-12)], [Graph (5-13)], [Graph (5-14)] and [Graph (5-15)], respectively.

As it can be seen from already noted graphs, except the efficiency and pressure rise coefficient in term of total-total pressure ratios, the other overalls and stages characteristics have remained the same for both compressor builds with and without cobra-type total pressure traversing effects.

On the other hand, the overall efficiency and pressure rise coefficient characteristics have also shown that the pressure probe traversing caused a significant drop of overall total-total compressor performance. It seems that the difference between the total-total and static-static pressure ratios are going to be reduced significantly when the effects of boundary layer of the compressor casing and hub had been taken into account.

This however led to an advantage of being confident that the overall total-total characteristics were reasonably close to the same corresponding characteristics that were calculated in terms of static-static pressure ratios. Some of the overall and stage flow characteristic results including the maximum and near stall values obtained from testing the three-stage compressor, with and without the total pressure probe traversing, builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)±(TPPT)], are presented in [Table (5-3)].

However, it was concluded that the pressure measured with the Peregrine compressor wall static pressure tapping could be used with an acceptable degree of accuracy, in order to provide some characteristic for performance comparison.

It is worth noting that the measuring of the outlet static pressure in a plenum chamber was the best idea, if it was possible. But since the under test compressor rig had not such facility, therefore, the wall static pressure measurement based on the above-noted advantage were found to be quite acceptable.

It was decided to locate the total pressure probe at the third stage stator blades downstream during the whole of the current investigation and apply the obtained results in order to determine additional characteristics for compressor performance comparing purposes.

5.14)- Influence of Air Straighteners on the Compressor Performance:

From the atmosphere air is introduced straight through a bell-mouth into the three-stage compressor. The outlet flow passes from the compressor third stage downstream is then passed down through the two air straightening boxes, before entering into the venturi.

The venturi flow meter and the three stages of the compressor are connected in-line through a flexible joint by a fairly short sheet-steel duct containing two completely separate honeycomb flow straighteners. The honeycombs were hexagonal-shape and aluminium-type with an aspect ratio of (5.5). The hexagonal holes were of 6.5 [mm] across the flats and 35.75 [mm] depth. The idea of using the honeycomb flow straighteners was a necessary in order to straighten the flow before entering into the measuring venturi.

It is worth noting that the original flow straighteners which were significantly damaged from the previous research were the paper-type honeycombs with an across the flat of 0.5 [inch] in association with 3 [inch] length. It means that the aspect ratio of the applied honeycomb flow straightener was reduced by (0.5) in respect to the original one.

It is worth noting that since the compressor has not outlet guide vanes, the two honeycombs flow straighteners had crucial role in introducing the flow in a desirable direction into the venturi, when passing the compressor third stage downstream. The effects of honeycomb flow straighteners were also found to be very significant in the single stage and particularly with the isolated rotor builds. In both of the compressor build cases described below the inlet guide vanes were used.

By using the honeycombs inside the flow straighteners, small alterations in the some flow characteristics were found. The overall and stage characteristic results, obtained from the compressor builds of [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)±(H)] with and without influences of using the honeycombs inside the flow straightener boxes are presented in [Table (5-4)], and [Table (5-5)], respectively.

5.14.1)- Effects of Honeycomb Flow Straighteners on the Mass Flow Rates:

The characteristics of the mass flow rate versus the corrected mass flow rate showed that by applying the honeycombs inside the flow straightener boxes, there was a small alteration in the values of the mass flow rate. The results also indicated that with the application of the honeycombs the mass flow through the compressor increased by small percentages of (0.93%) and (0.02%) at choke and near stall conditions, respectively. The alterations of the corrected mass flow rates in the both cases, with and without the honeycomb flow straighteners were not significant.

It seems that the magnitudes of the range of the mass flow through the compressor are very similar during the compressor operations, but without honeycombs flow straightener, the reduction of losses, provides a little more flow rate at choke condition. By using the honeycombs, the corresponding flow coefficients were reduced by (1.46%) and (1.78%), when the throttling was fully opened and in near stall conditions, respectively.

Mass flow rate and its corresponding corrected characteristics, which were provided using the calculated results of the three stage compressor builds in the presence and absence of the honeycombs in the flow straighteners versus their related flow coefficients are shown in [Graph (5-16)] and [Graph (5-17)], respectively.

It can be deduced that at the choke condition and particularly, with higher mass flow, the system performed better when the honeycombs were absent. However as the mass flow was reduced through the compressor, more losses were created. In such circumstances the

use of honeycombs was found to be more desirable since in this condition the compressor was able to continue operating with lower flow coefficients.

Although the decision about the use of the honeycomb flow straighteners based on the above noted results is challenging, it seems that the application of the honeycombs had more advantage in comparison with putting nothing in the flow straightener boxes. In other build configurations the use of the honeycomb flow straighteners yielded better performance results and this will be described in more detail in chapter eight.

5.14.2)- Influence of Flow straighteners on the Corrected Applied Powers:

In the fully choke condition, the corrected applied power ($\Omega\varepsilon$) of the three-stage compressor build with the honeycombs in the flow straightener boxes was (0.59%) higher than without using the honeycombs. This result changed near stall conditions, because the obtained corrected power with the use of the honeycombs was decreased by (0.84%).

It means again that the honeycombs provided fewer losses in the higher mass flow rates in comparison to the number of occasions in which mass flow was reduced to a value at which stall occurred.

The maximum corrected applied power values and their corresponding corrected mass flow rates for the builds with and without honeycombs flow straightener were calculated as being (6.979) and (6.994) [kwatt] and (2.205) and (2.203) [kg/sec], respectively. These results however indicated that the flow, after passing the third stage stator blades, was in an axial direction on leaving the compressor.

Therefore it can be concluded that the application of the honeycombs flow straightener in respect of the input work is neutral. The characteristic of the corrected applied power with and without usage of the honeycombs inside the flow straightener boxes is illustrated in [Graph (5-18)].

5.14.3)- Effects of Honeycomb Flow Straighteners on the Work Coefficients:

Work coefficients (overall) at both ends of the characteristic obtained from the complete three-stage compressor build were higher with the honeycomb air straighteners fitted than without them. These increments were about (1.35%), when the orifice was fully opened and (0.75%) near stall conditions.

When the flow straightener honeycombs were used, the maximum work coefficient and its corresponding corrected mass flow found to be of (1.23) and 1.976 [kg/sec], respectively. Meanwhile, for the build without the air straighteners the highest value of the overall work coefficient was about (1.22) in association with 1.967 [kg/sec] its related corrected mass flow rate.

The overall work coefficients of the builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)±(H)] with and without the honeycomb flow straighteners is illustrated in [Graph (5-19)].

5.14.4)- Influence of the Honeycomb Flow Straighteners on the Efficiencies:

Overall (total-total) efficiency characteristics obtained from running the test rig with the inlet guide vanes and all three stages, with and without the honeycomb air straighteners showed that this type of efficiency was increased without the application of the honeycombs.

With the application of the honeycombs in the air straightener boxes, the overall total-total efficiency was increased by (7.96%), in the choke status and reduced by (2.52%) near the stall condition, respectively. Nonetheless, by removing the honeycombs from the flow

straightener boxes, the point of maximum efficiency was shifted to a lower flow coefficient, but in association with an increment of (1.45%). The maximum overall total-total efficiencies and their corresponding corrected mass flow rates of the compressor builds with and without the honeycomb air straighteners was found to be (0.836) & 2.522 [kg/sec] and (0.849) & 2.424 [kg/sec], respectively.

The overall (static-static) and (total-static) efficiency values obtained from testing the builds with and without the honeycomb flow straighteners, indicated that in both cases the higher efficiencies were obtained when the air straightener devices were used. These characteristics also showed that with the honeycomb air straighteners, the maximum efficiency point of (0.787) was shifted to a lower corrected mass flow of 2.287 [kg/sec].

With the application of the honeycomb flow straighteners the overall static-static efficiencies were increased by (3.08%) and (0.65%) at the choke and near stall conditions, respectively. Without the use of the honeycomb flow straightener devices, the overall static-static efficiency of (0.784) was occurred in correspondence to 2.322 [kg/sec] of corrected mass flow rate. The maximum efficiency also was decreased by (0.399%), when the honeycombs took apart from the air straighteners.

The characteristics of the overall total-static efficiencies were fairly similar to those obtained in terms of static-static pressure ratios. For the build with the honeycomb air straighteners, the maximum overall total-static efficiency was (0.666) with related corrected mass flow rate of 2.162 [kg/sec]. These were altered to (0.659) and 2.182 [kg/sec], respectively. When the honeycomb flow straighteners were taken out of the test rig the overall total-static efficiencies at choke and near stall conditions decreased by (37.12%) and (0.38%), respectively.

The overall total-static and static-static efficiency characteristics showed that in the higher flow rate, the compressor build without the honeycomb air straighteners performed a little better than the build with the honeycombs. These however were reversed in the lower mass flow rates and particularly near stall conditions.

The overall and stage efficiency characteristics versus corrected mass flow and flow coefficient for the builds with and without using the honeycombs inside the flow straightener boxes are shown in [Graph (5-20)] and [Graph (5-21)], respectively.

5.14.5)- Effects of the Air Straighteners on the Pressure Rise Coefficients:

The overalls pressure rise coefficient characteristics in terms of (total-total), (static-static) and (total-static) pressure ratios provided reasonably close results in the three-stage compressor configuration including the inlet guide vanes, with and without honeycombs flow straighteners. The overall pressure rise coefficient characteristics, similar to the efficiency ones, showed that in the all cases, the builds without the honeycomb air straighteners provided slightly lower pressure ratios, when the orifice was fully opened.

On the other hand when the honeycombs were removed from the air straightener boxes, the flow coefficients corresponding to the two ends of the pressure rise coefficient curve were increased by (1.46%) and (1.78%) in the choke and near stall conditions, respectively. Meanwhile in both cases the peak pressure rise coefficients took place at near stall points.

It is worth noting that both the overalls (total-static) and (static-static) peak pressure rise coefficients, were increased by (1.45%) and (1.43%), when the honeycombs air straighteners were used. Unlike the overall peak pressure rise coefficient which had shown a reduction of (2.06%) in this category.

The overall and stage pressure rise coefficient characteristics versus flow coefficient and

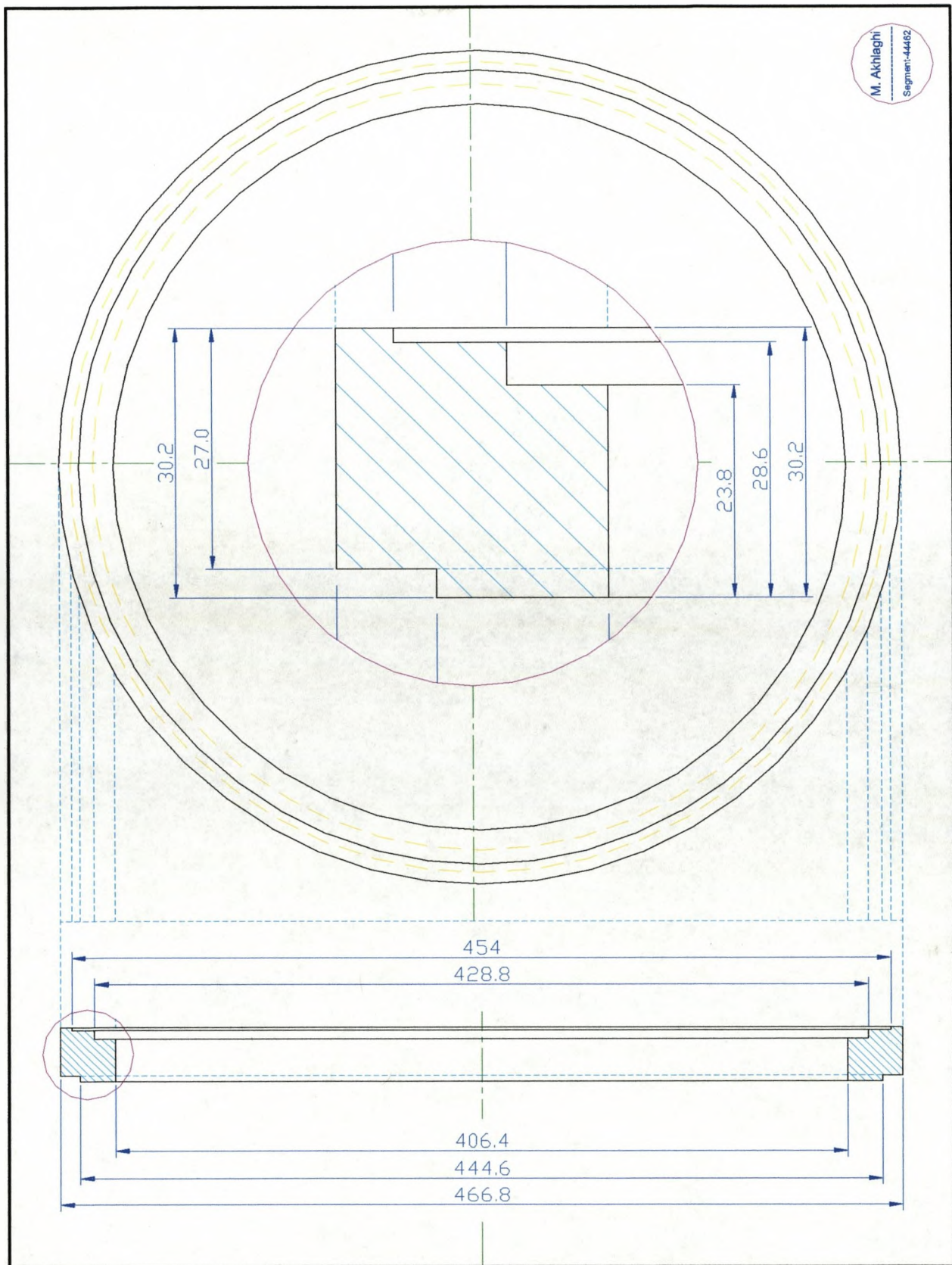
corrected mass flow for the builds with and without the honeycomb air straighteners are illustrated in [Graph (5-22)] and [Graph (5-23)], respectively.

By analysing the overall efficiency and pressure rise coefficient characteristics, it seems that the cobra-type total pressure probe had shown some sensitivity to the application of the honeycombs air straighteners as the mass flow rate was reduced through the compressor builds.

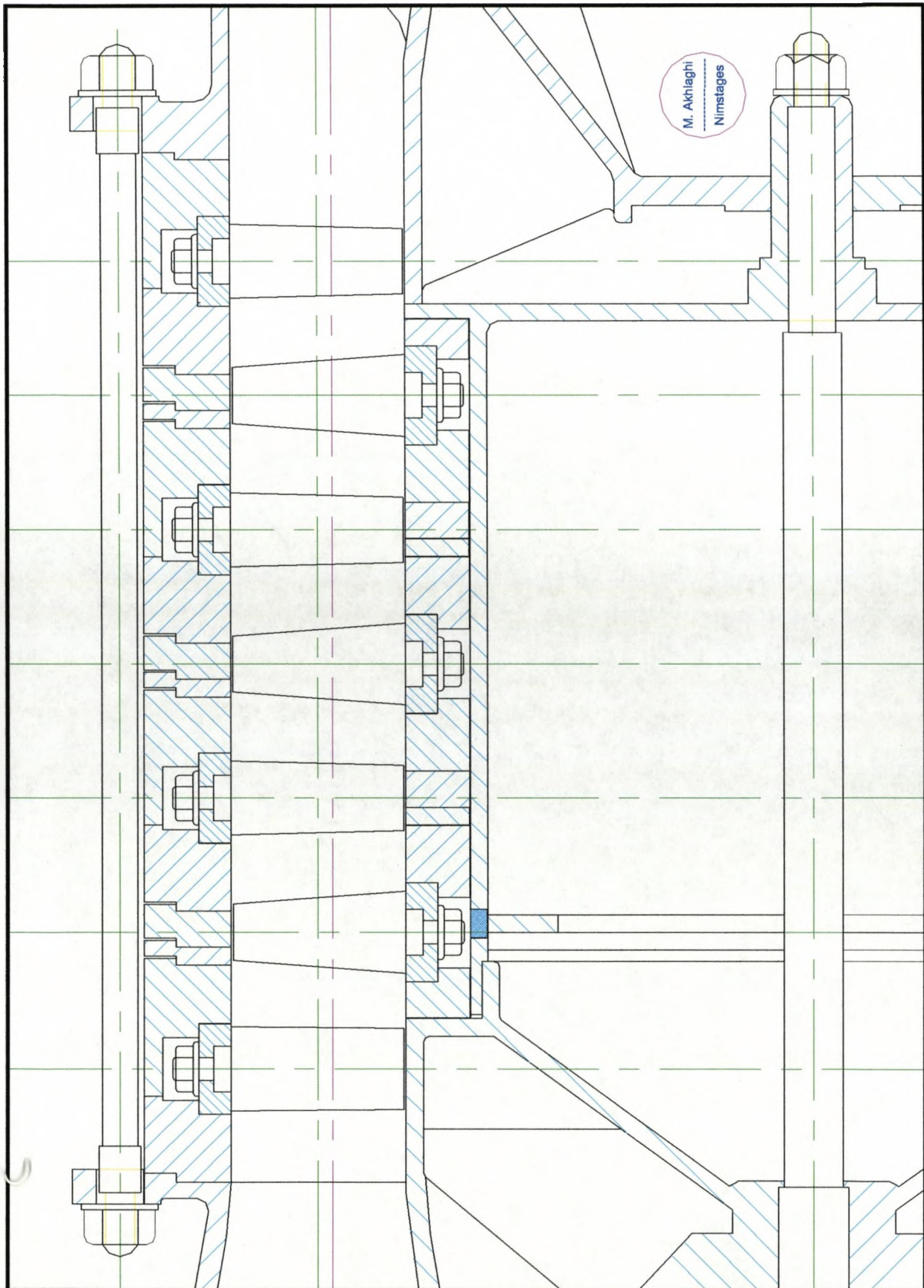
The analysis of the flow characteristics described above confirmed that the Peregrine compressor enabled the outlet flow to pass through the third stage stator blades downstream in an axial direction. Therefore the application of the honeycombs inside the air straightener boxes is arbitrary, when the compressor is working in the range of choke to near design point conditions.

Some of the overall and stage characteristic results, which were obtained from the three-stage compressor builds, with and without using the honeycomb inside the air straighteners are presented in [Table (5-6)].

Finally, it is possible to conclude that the use of the honeycomb flow straighteners is highly recommended, when the test compressor is operating at lower flow rates.

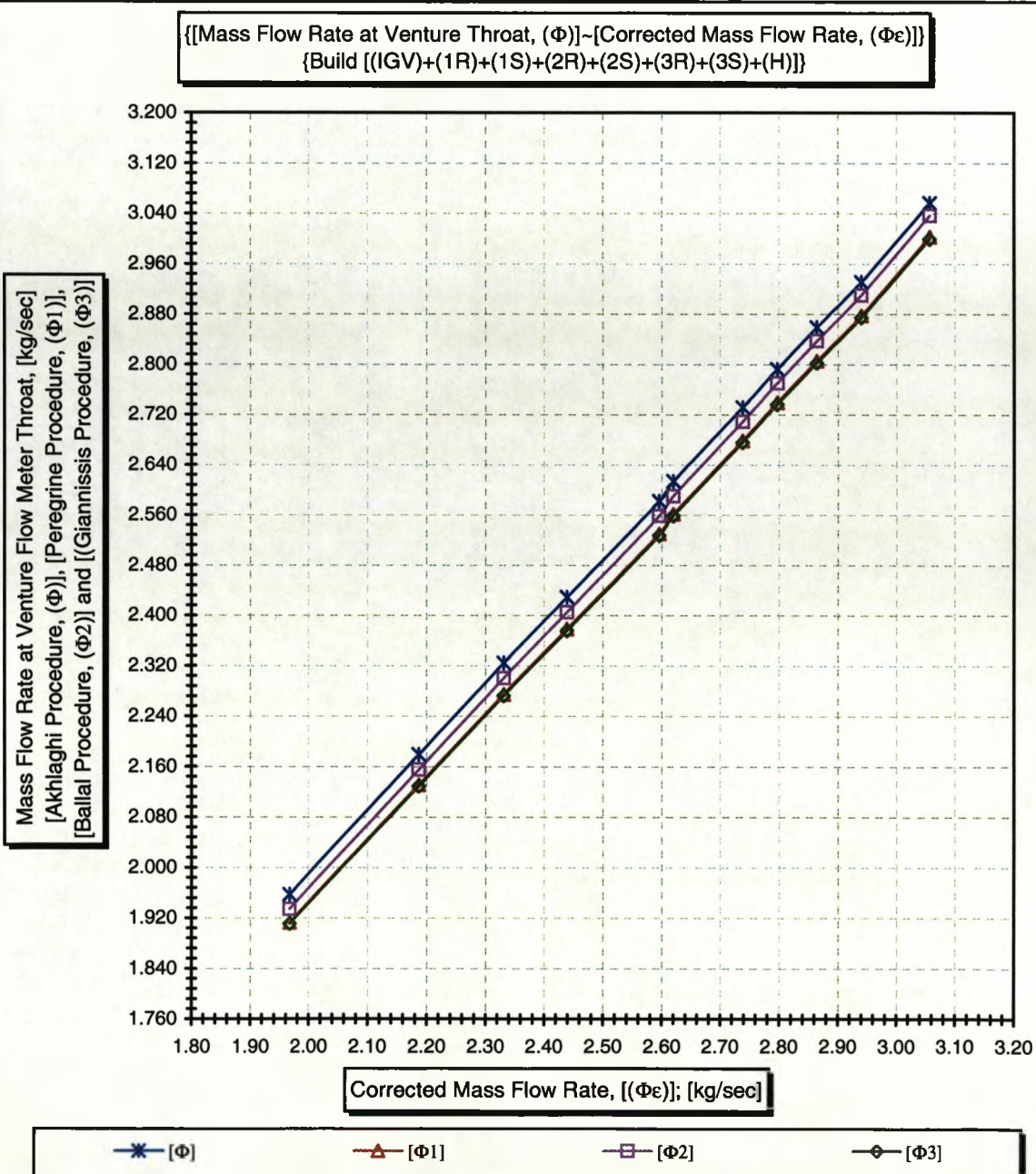


[Figure (5-1)]- The schematic drawing of the outer ring (No. 44462), which is fitted before ring (No. 44463), in order to cover together the two half-rings that facilitate the (34) hole holder ring of the inlet guide vanes



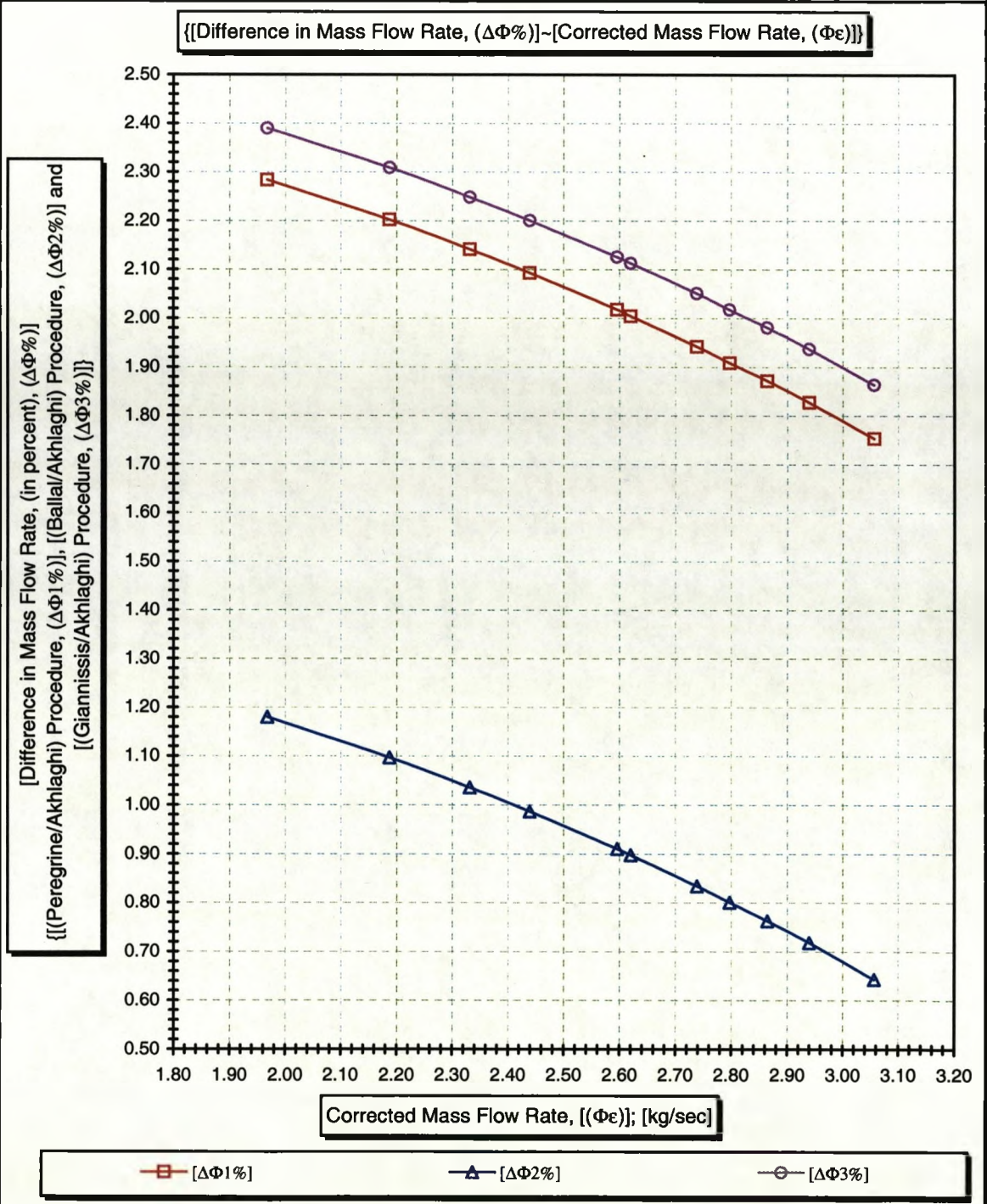
[Figure (5-2)]- A cross section view of the compressor under nominal configuration, covering the view of the rotating shaft centreline to the top of the outer casing including the geometry of the spacer rings and position of the fixed and rotor blades settings

$\{Build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]\}$										
Corrected mass flow rate, (Φ_e); [kg/sec]										
3.057	2.939	2.864	2.797	2.738	2.620	2.595	2.439	2.331	2.187	1.967
Mass flow rate at venturi flow meter throat, [Akhlaghi procedure], (Φ); [kg/sec]										
3.057	2.931	2.859	2.793	2.732	2.614	2.582	2.429	2.324	2.179	1.957
Mass flow rate [Peregrine procedure], (Φ_1); [kg/sec]										
3.003	2.878	2.806	2.739	2.679	2.561	2.530	2.378	2.275	2.131	1.913
Mass flow rate [Ballal procedure], (Φ_2); [kg/sec]										
3.037	2.910	2.837	2.770	2.709	2.590	2.559	2.405	2.300	2.155	1.934
Mass flow rate [(Giannissis procedure), (Φ_3); [kg/sec]										
3.000	2.875	2.803	2.736	2.676	2.559	2.527	2.376	2.272	2.129	1.911

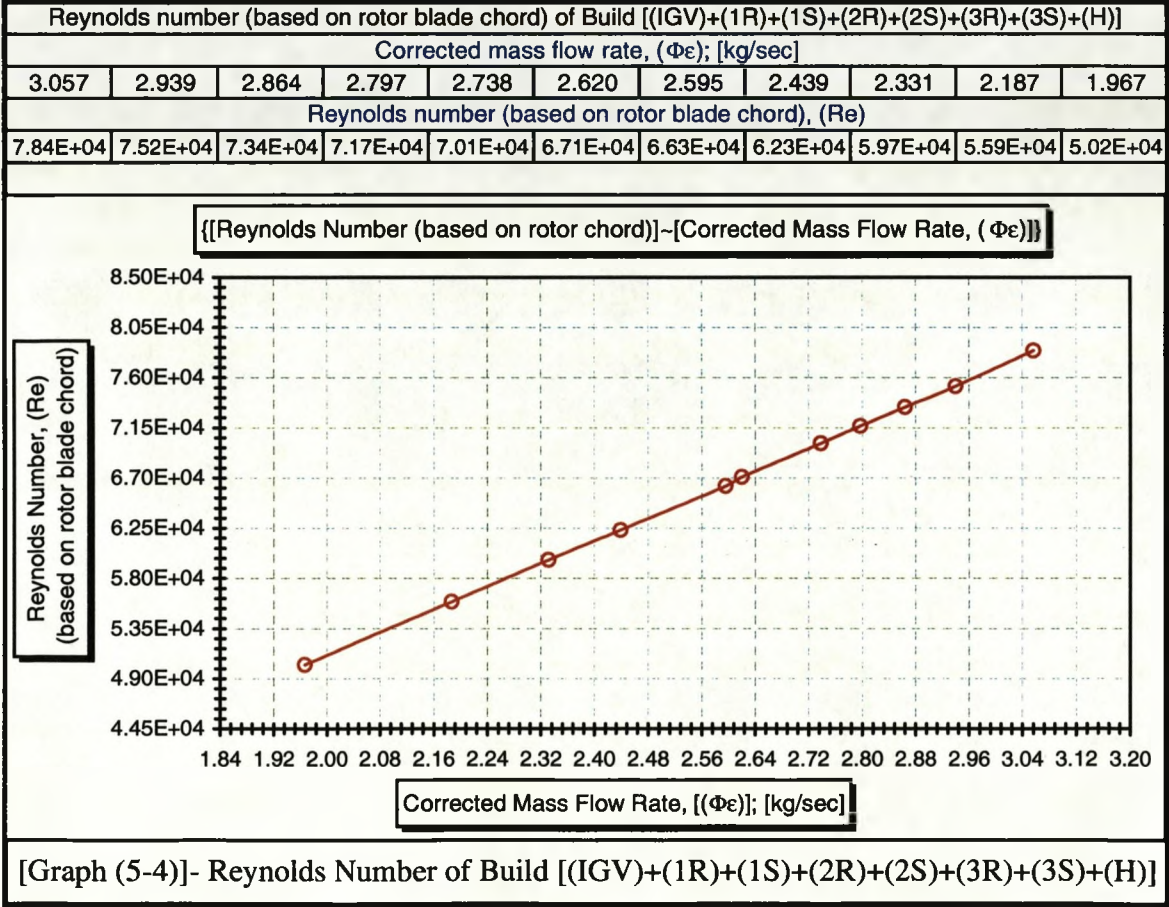
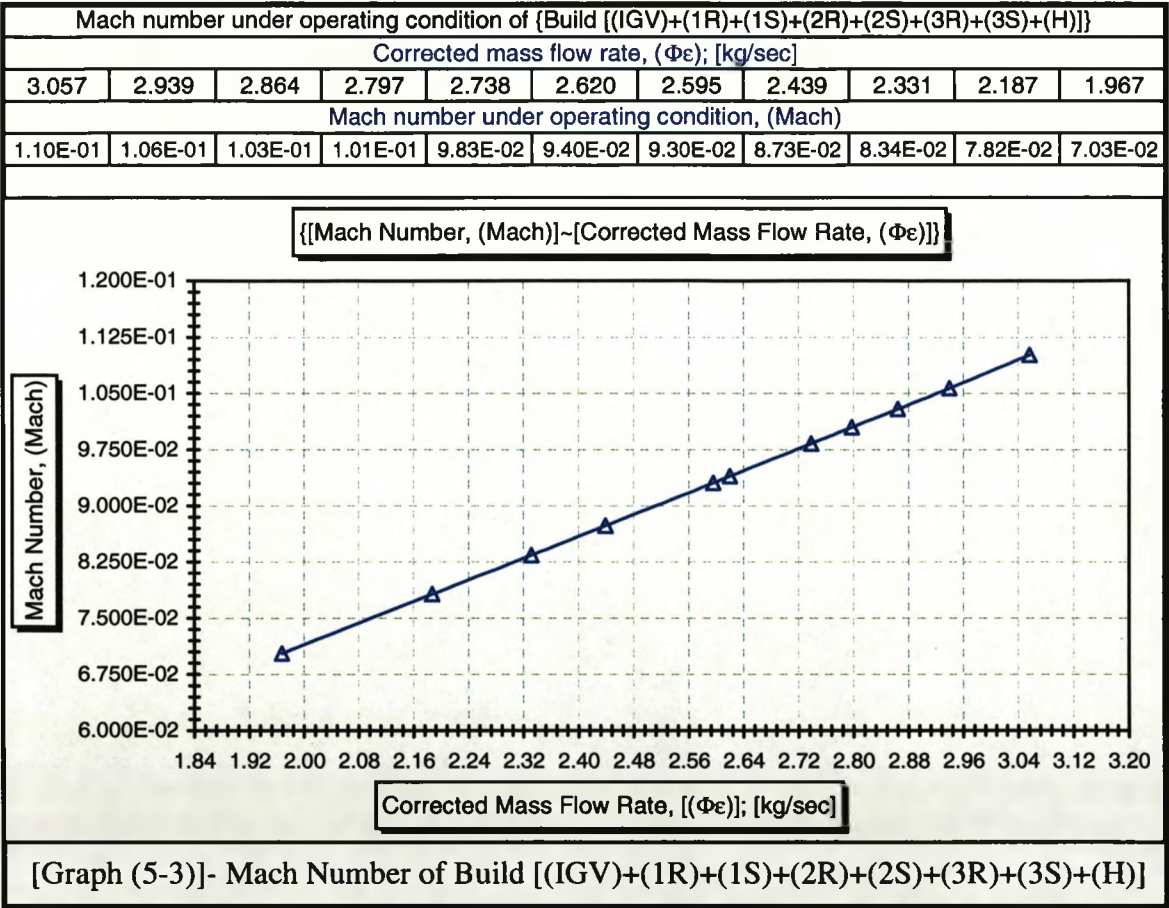


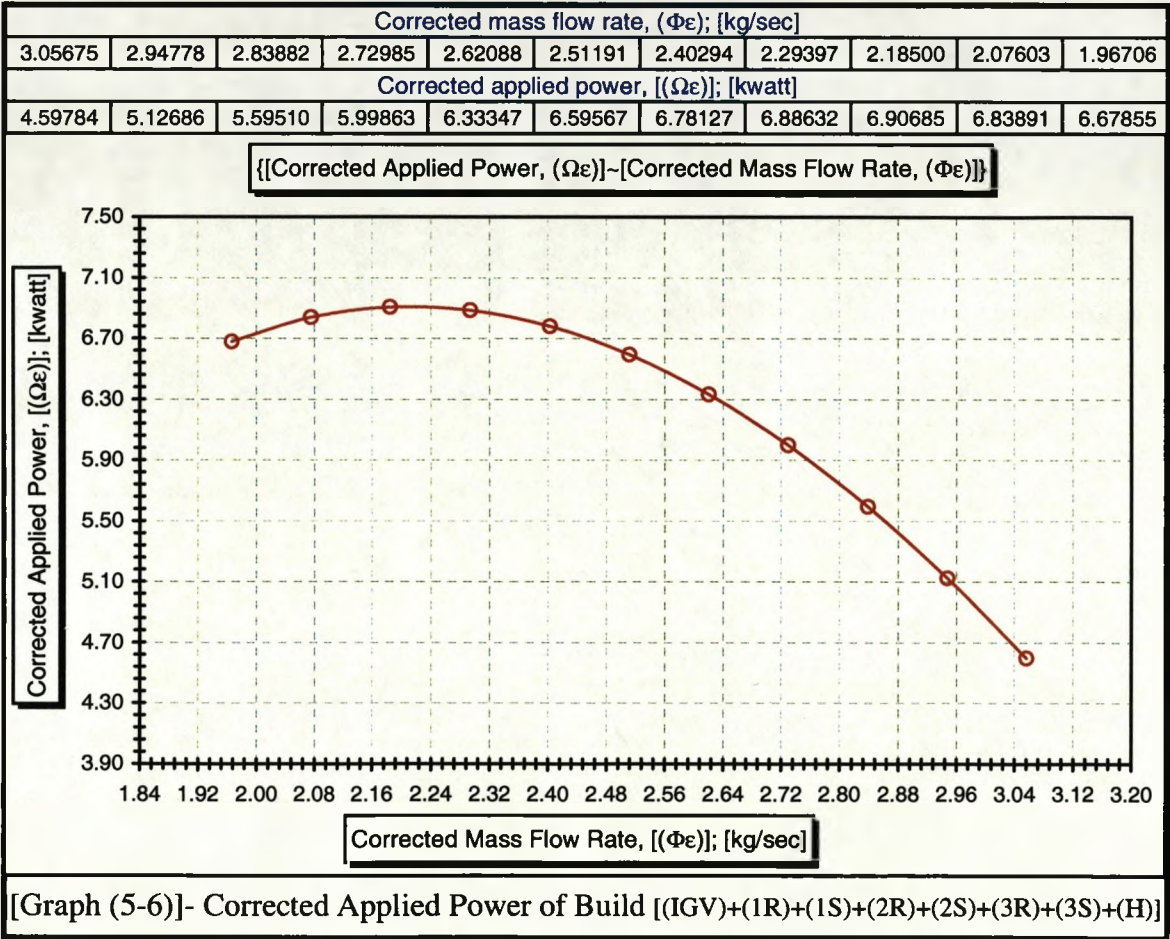
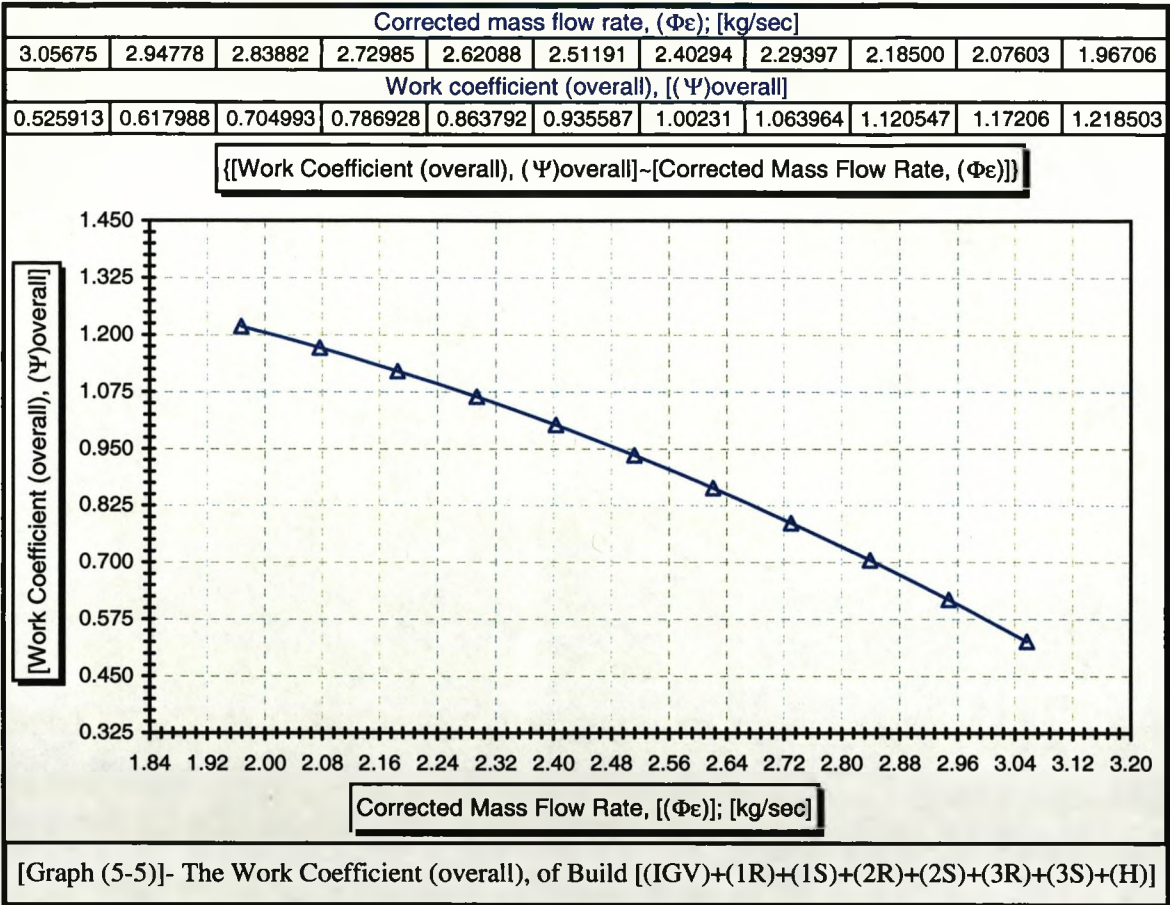
[Graph (5-1)]- Four different values of the mass flow rate through the three-stage compressor build in the present of the honeycombs in thr flow straighteners

Corrected mass flow rate, ($\Phi\epsilon$); [kg/sec]										
3.05675	2.93946	2.86399	2.79712	2.73830	2.61989	2.59540	2.43898	2.33134	2.18682	1.96706
[Difference in mass flow rate calculation (Peregrine/Akhlaghi) procedures (in percent), ($\Delta\Phi1\%$)]										
1.75329	1.82765	1.87182	1.90879	1.94233	2.00460	2.01770	2.09286	2.14143	2.20209	2.28373
[Difference in mass flow rate calculation (Ballal/Akhlaghi) procedures (in percent), ($\Delta\Phi2\%$)]										
0.64316	0.71835	0.76303	0.80042	0.83433	0.89731	0.91056	0.98656	1.03569	1.09703	1.17959
[Difference in mass flow rate calculation (Giannisis/Akhlaghi) procedures (in percent), ($\Delta\Phi3\%$)]										
1.86366	1.93736	1.98114	2.01778	2.05098	2.11278	2.12577	2.20031	2.24850	2.30875	2.39002

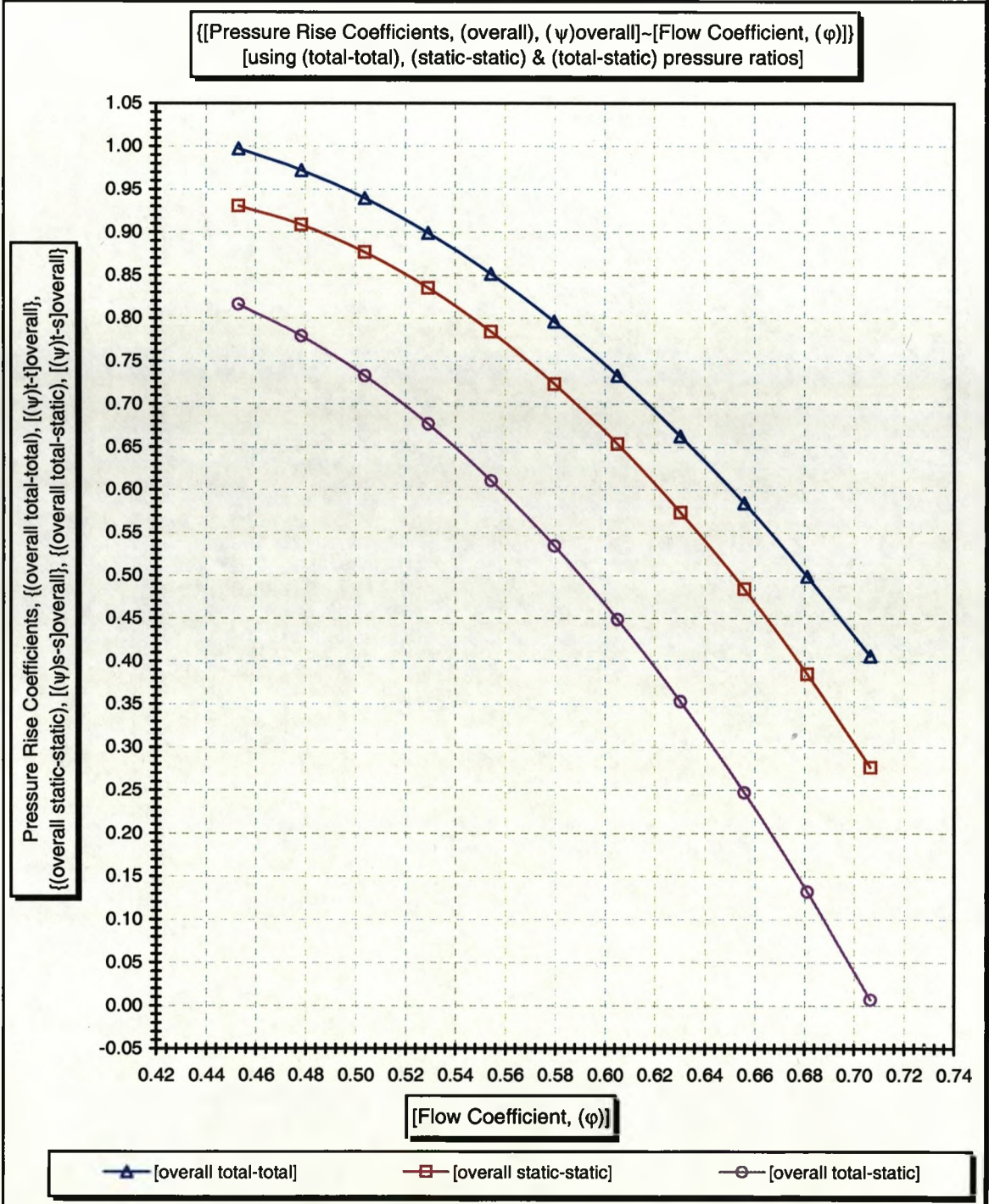


[Graph (5-2)]- Difference in mass flow rate calculation procedures (in percent), using experimental results obtained from Build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]



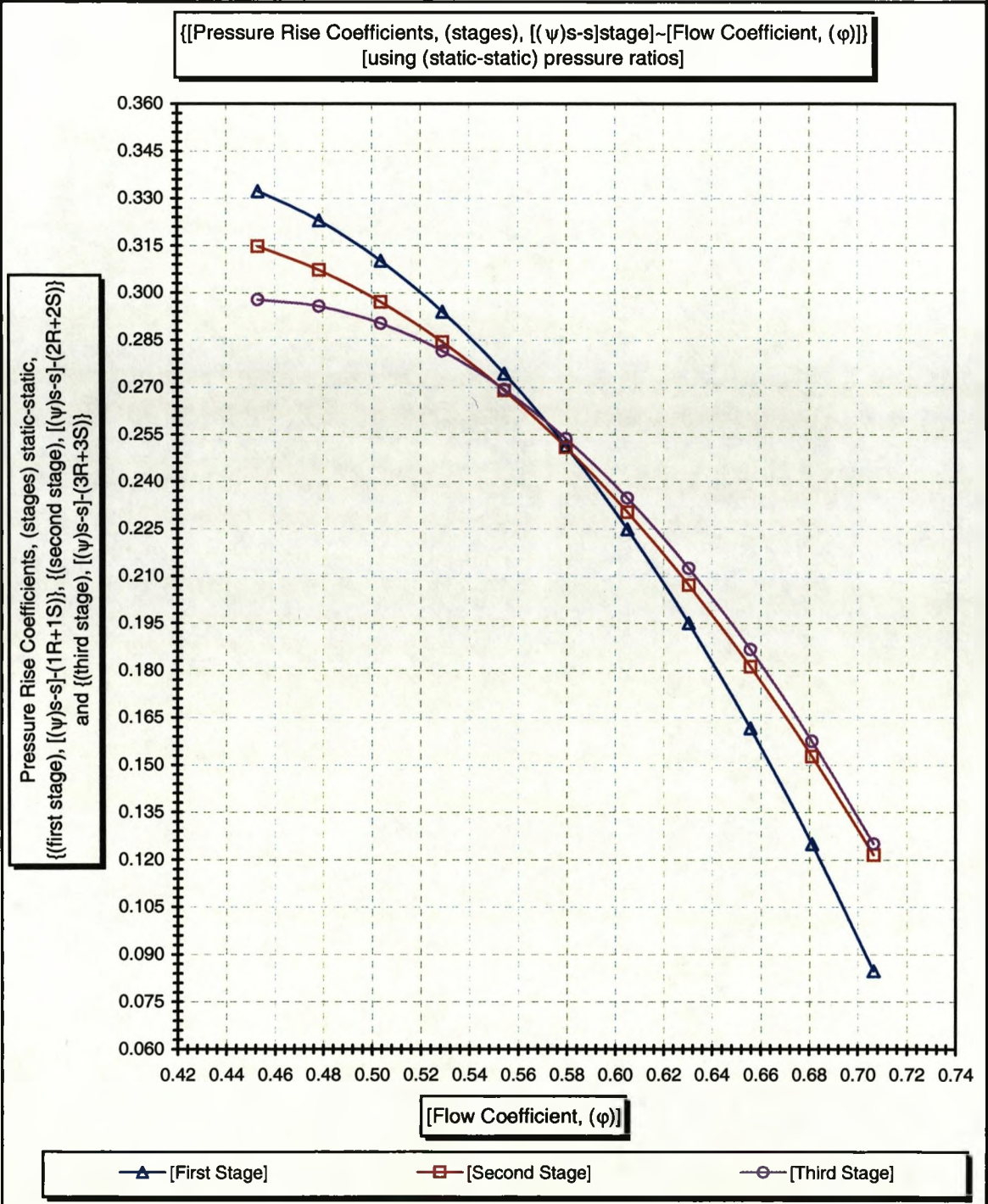


Flow coefficient at compressor annulus, (ϕ)										
0.70640	0.68106	0.65572	0.63039	0.60505	0.57972	0.55438	0.52905	0.50371	0.47838	0.45304
Pressure rise coefficient, (overall total-total), $\{[(\psi)t-t]\text{overall}\}$										
0.40584	0.49899	0.58459	0.66263	0.73311	0.79603	0.85140	0.89921	0.93946	0.97216	0.99730
Pressure rise coefficient, (overall static-static), $\{[(\psi)s-s]\text{overall}\}$										
0.27668	0.38521	0.48417	0.57354	0.65333	0.72353	0.78416	0.83520	0.87666	0.90853	0.93083
Pressure rise coefficient, (overall total-static), $\{[(\psi)t-s]\text{overall}\}$										
0.00675	0.13198	0.24737	0.35292	0.44861	0.53446	0.61047	0.67663	0.73294	0.77941	0.81603



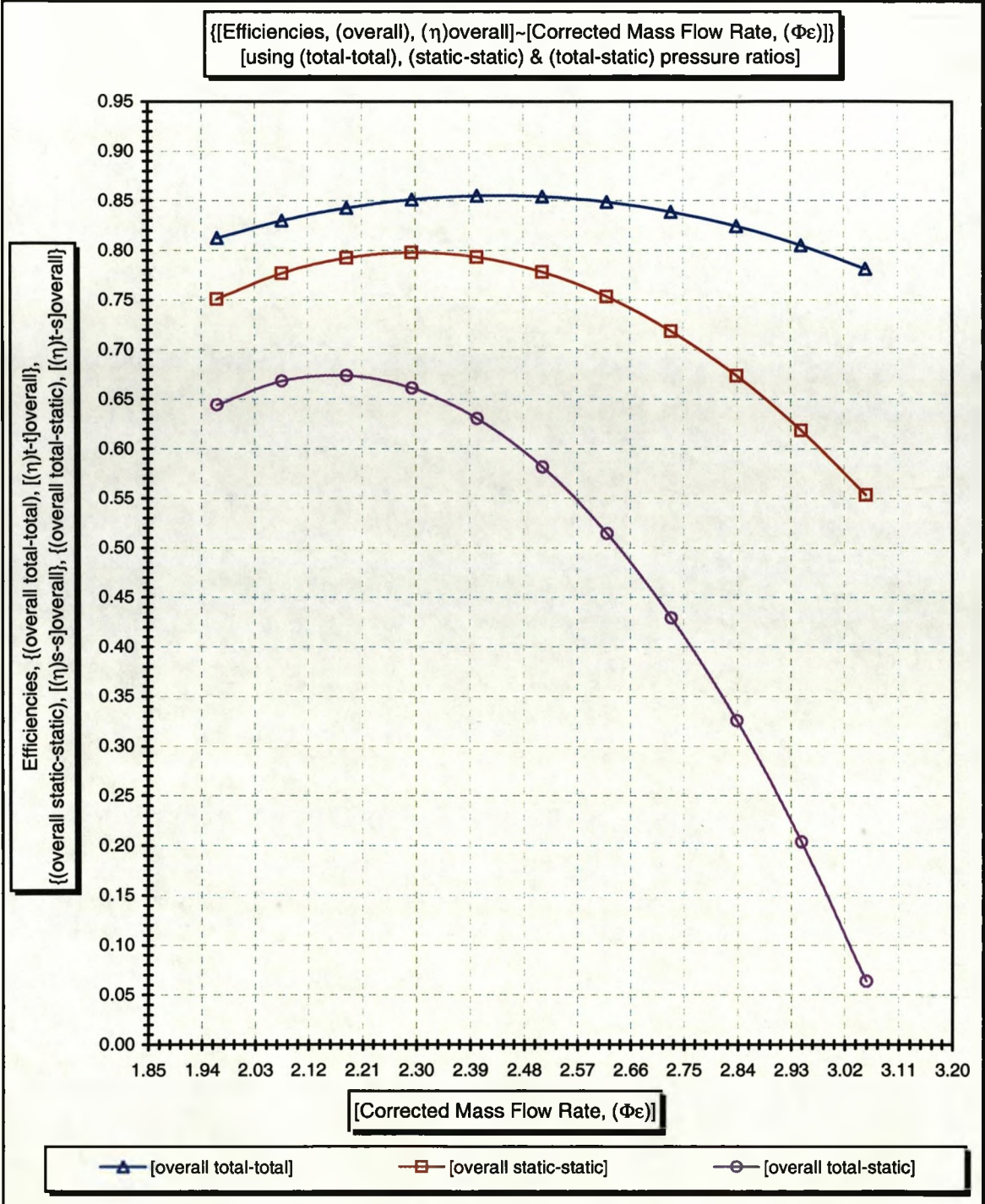
[Graph (5-7)]- Pressure rise coefficient characteristics, using experimental results obtained from build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]

Flow coefficient at compressor annulus, (ϕ)										
0.70640	0.68106	0.65572	0.63039	0.60505	0.57972	0.55438	0.52905	0.50371	0.47838	0.45304
Pressure rise coefficient, (first stage static-static), $\{[(\psi)s-s]-(1R+1S)\}$										
0.08474	0.12493	0.16168	0.19500	0.22490	0.25137	0.27440	0.29401	0.31019	0.32294	0.33226
Pressure rise coefficient, (second stage static-static), $\{[(\psi)s-s]-(2R+2S)\}$										
0.12155	0.15270	0.18121	0.20710	0.23035	0.25098	0.26898	0.28435	0.29709	0.30720	0.31468
Pressure rise coefficient, (third stage static-static), $\{[(\psi)s-s]-(3R+3S)\}$										
0.12505	0.15756	0.18668	0.21242	0.23477	0.25373	0.26931	0.28150	0.29031	0.29573	0.29776



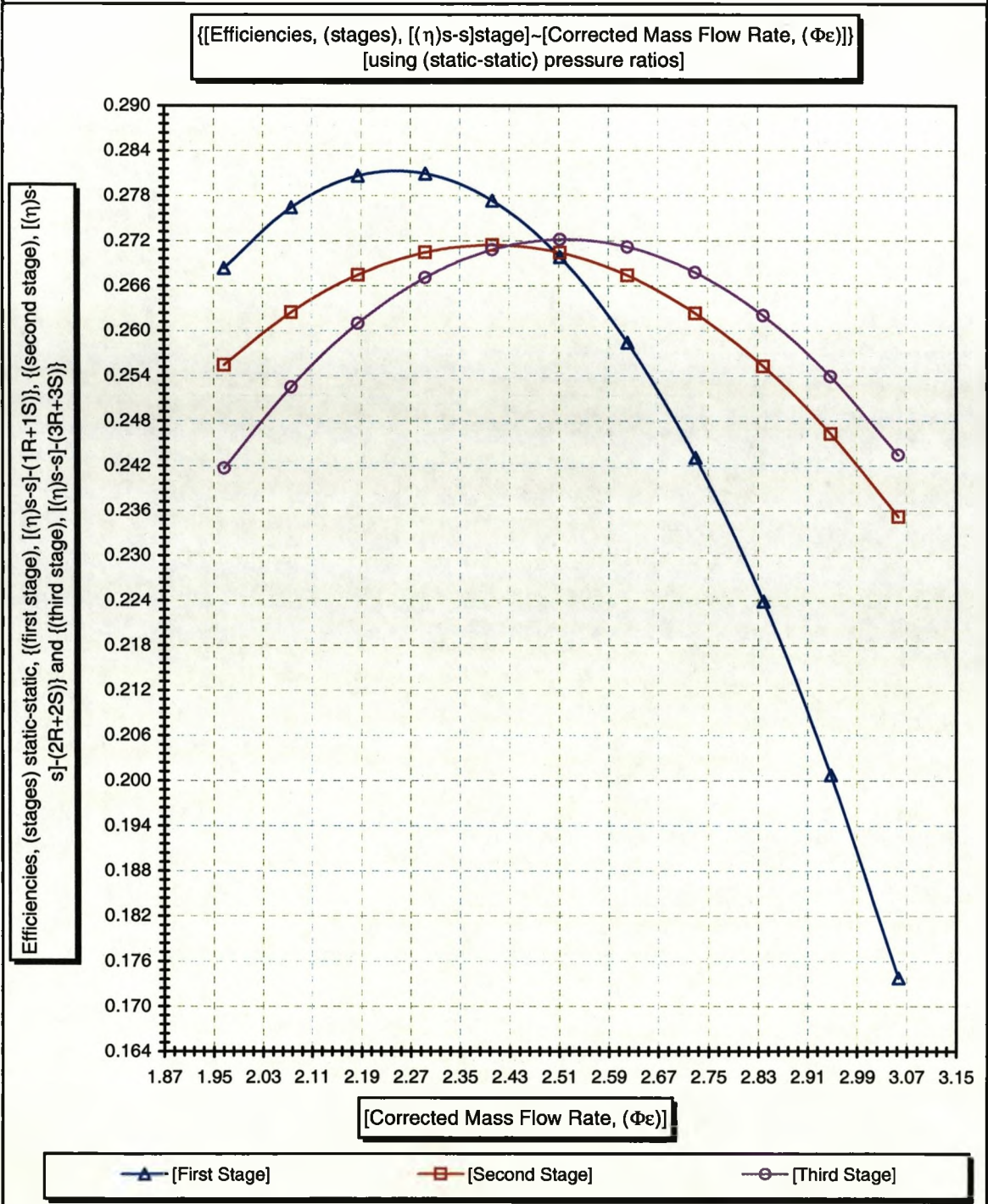
[Graph (5-8)]- Pressure rise coefficients of three repeating stages, based on results of Peregrine compressor under build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]

Corrected Mass Flow Rate, ($\Phi\epsilon$)										
3.05675	2.94778	2.83882	2.72985	2.62088	2.51191	2.40294	2.29397	2.18500	2.07603	1.96706
Efficiency, (overall total-total), $\{[(\eta)t-t]\text{overall}\}$										
0.78140	0.80507	0.82418	0.83872	0.84869	0.85409	0.85492	0.85118	0.84287	0.82999	0.81254
Efficiency, (overall static-static), $\{[(\eta)s-s]\text{overall}\}$										
0.55351	0.61866	0.67372	0.71870	0.75359	0.77840	0.79313	0.79777	0.79233	0.77680	0.75120
Efficiency, (overall total-static), $\{[(\eta)t-s]\text{overall}\}$										
0.06374	0.20369	0.32546	0.42902	0.51439	0.58156	0.63054	0.66132	0.67390	0.66829	0.64449



[Graph (5-9)]- The overall total-total, static-static and total-static efficiencies, using experimental results obtained from build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]

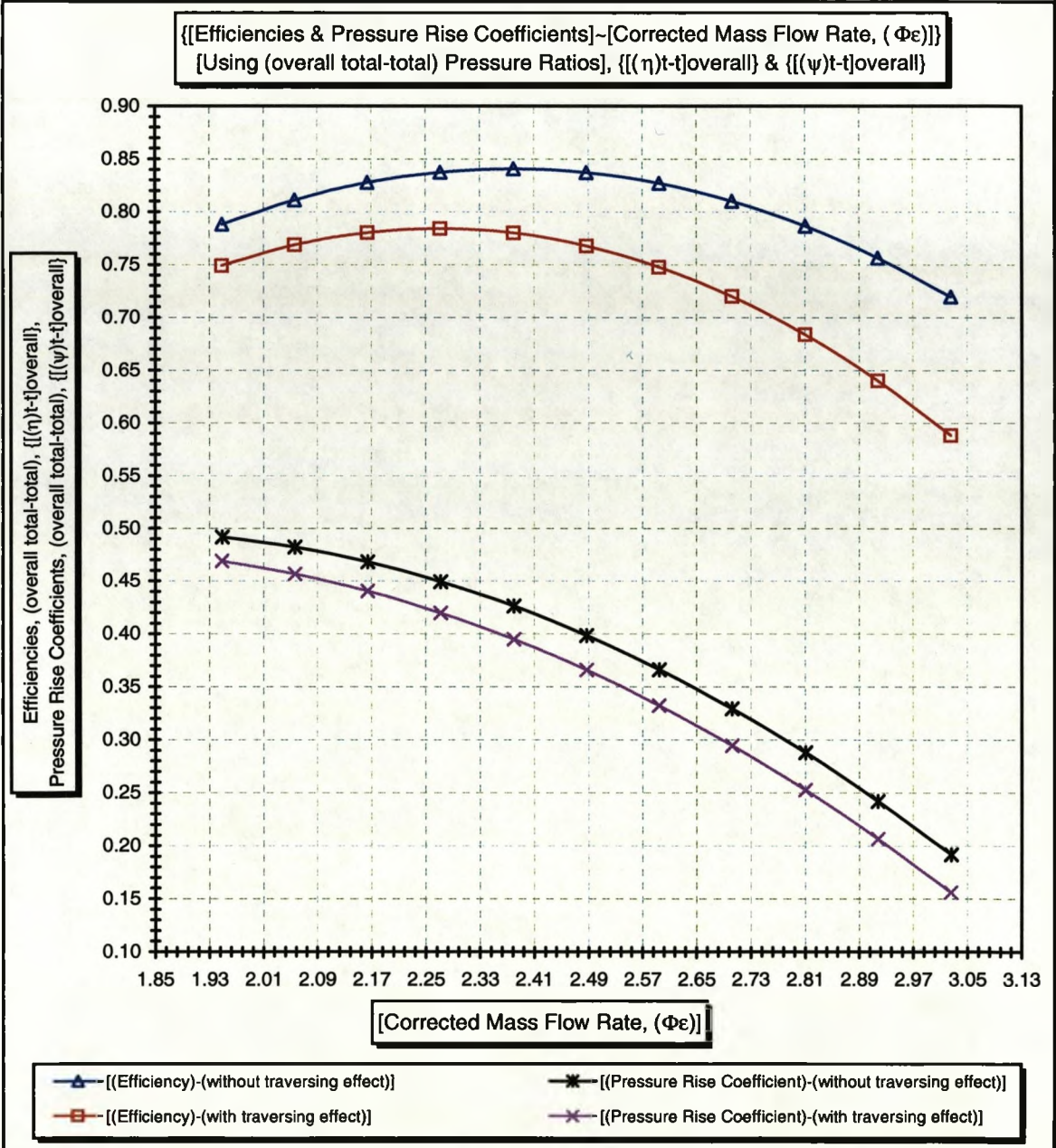
Corrected Mass Flow Rate, ($\Phi\epsilon$)										
3.05675	2.94778	2.83882	2.72985	2.62088	2.51191	2.40294	2.29397	2.18500	2.07603	1.96706
Efficiency, (first stage static-static), $\{[(\eta)s-s]-(1R+1S)\}$										
0.17374	0.20076	0.22388	0.24309	0.25840	0.26981	0.27732	0.28093	0.28064	0.27644	0.26834
Efficiency, (second stage static-static), $\{[(\eta)s-s]-(2R+2S)\}$										
0.23520	0.24624	0.25529	0.26233	0.26736	0.27039	0.27142	0.27044	0.26745	0.26247	0.25548
Efficiency, (third stage static-static), $\{[(\eta)s-s]-(3R+3S)\}$										
0.24345	0.25393	0.26204	0.26778	0.27115	0.27216	0.27080	0.26707	0.26097	0.25251	0.24168



[Graph (5-10)]- The individual stage static-static efficiencies of the Peregrine compressor three repeating stages, under build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]

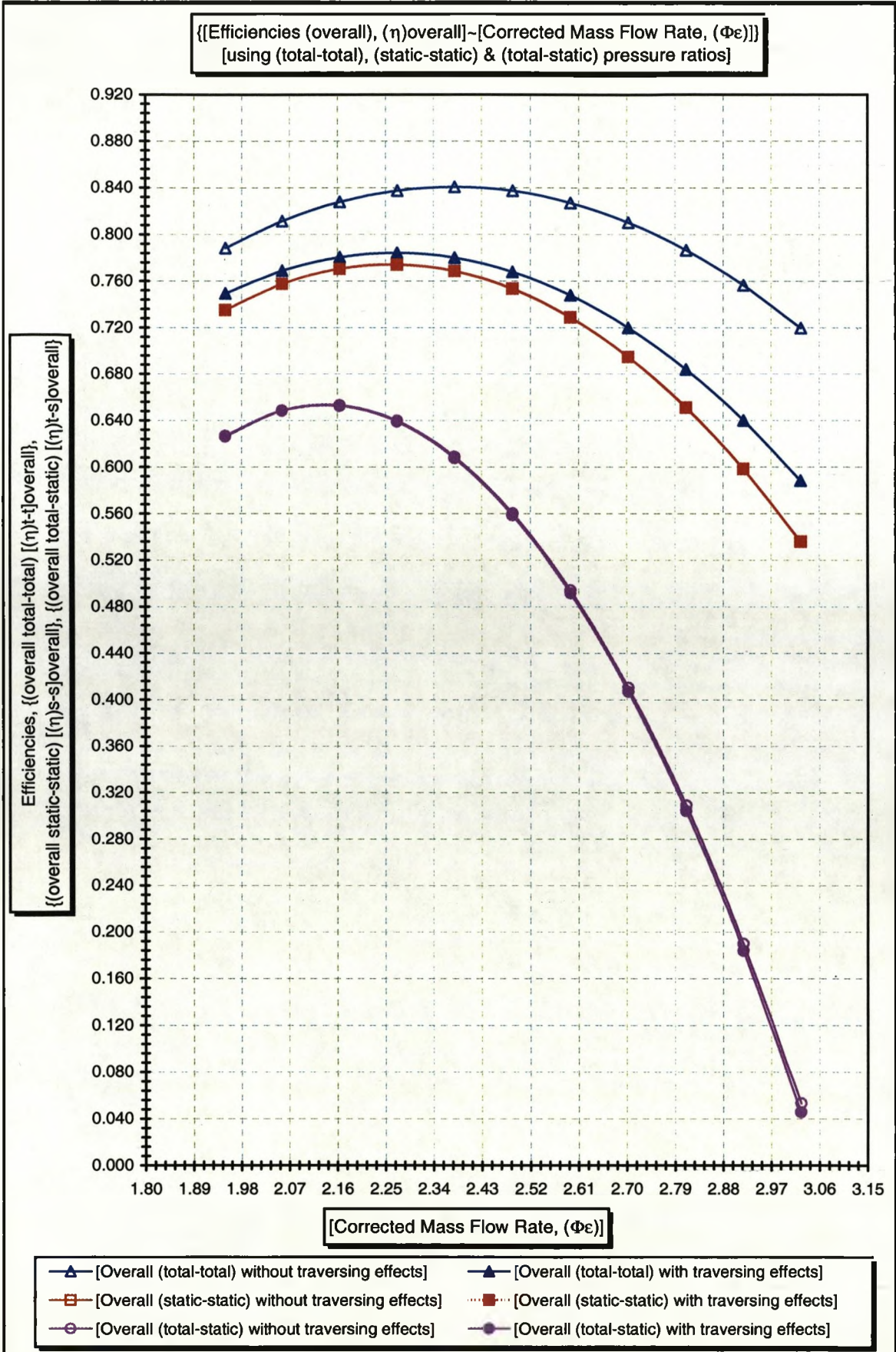
Results of total pressure traversing effect at flow coefficient ($\phi=0.704$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	41.5	38.1	32.7	30.6	30.2	28.8	29.3	31.4	31.5	33.0	36.5	39.7	40.4	41.5	42.2	43.7	45.9	
Pressure (datum), [cm-H ₂ O]	57.4	57.7	57.9	58.0	58.0	58.1	58.0	58.0	58.0	57.7	57.6	57.5	57.5	57.5	57.4	57.2		
Average of total pressure; [cm-H ₂ O]					36.29412					Average of datum pressure; [cm-H ₂ O]					57.74118			
Results of total pressure traversing effect at flow coefficient ($\phi=0.655$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	32.7	31.3	28.1	25.1	23.9	22.7	23.8	24.1	24.1	26.1	26.9	29.8	30.9	31.8	33.1	34.1	36.1	
Pressure (datum), [cm-H ₂ O]	63.1	63.3	63.5	63.6	63.7	63.7	63.7	63.7	63.7	63.6	63.6	63.5	63.4	63.3	63.3	63.2	63.1	
Average of total pressure; [cm-H ₂ O]					28.50588					Average of datum pressure; [cm-H ₂ O]					63.47059			
Results of total pressure traversing effect at flow coefficient ($\phi=0.597$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	27.4	26.6	25.6	23.4	21.3	18.1	18.3	18.0	17.7	19.4	20.3	22.6	23.1	23.7	24.4	25.8	28.3	
Pressure (datum), [cm-H ₂ O]	68.7	68.8	68.8	68.9	69.0	69.1	69.1	69.2	69.3	69.2	69.2	69.1	69.0	69.0	68.9	68.9	68.7	
Average of total pressure; [cm-H ₂ O]					22.58824					Average of datum pressure; [cm-H ₂ O]					68.99412			
Results of total pressure traversing effect at flow coefficient ($\phi=0.572$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	24.9	24.7	23.7	21.7	20.3	15.7	16.1	15.8	15.8	16.4	17.6	19.8	20.2	20.9	21.5	22.4	24.3	
Pressure (datum), [cm-H ₂ O]	70.8	70.9	71.0	71.1	71.1	71.3	71.3	71.4	71.4	71.3	71.3	71.2	71.2	71.1	71.1	71.1	71.0	
Average of total pressure; [cm-H ₂ O]					20.10588					Average of datum pressure; [cm-H ₂ O]					71.15294			
Results of total pressure traversing effect at flow coefficient ($\phi=0.565$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	23.9	23.3	22.5	21.0	19.5	15.0	14.7	14.4	15.0	14.3	15.6	18.8	19.4	19.9	20.5	21.7	23.3	
Pressure (datum), [cm-H ₂ O]	71.7	71.7	71.7	71.8	71.9	72.2	72.4	72.5	72.0	72.5	72.4	72.3	72.2	72.2	72.1	72.1	72.0	
Average of total pressure; [cm-H ₂ O]					18.98824					Average of datum pressure; [cm-H ₂ O]					72.10000			
Results of total pressure traversing effect at flow coefficient ($\phi=0.544$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	23.3	22.5	21.8	20.4	18.8	13.8	13.6	13.3	13.3	13.1	13.8	17.5	17.9	18.4	18.9	20.1	22.3	
Pressure (datum), [cm-H ₂ O]	72.9	72.9	72.9	73.0	73.0	73.3	73.3	73.3	73.3	73.4	73.3	73.2	73.2	73.1	73.1	73.0	72.9	
Average of total pressure; [cm-H ₂ O]					17.81176					Average of datum pressure; [cm-H ₂ O]					73.12353			
Results of total pressure traversing effect at flow coefficient ($\phi=0.520$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	23.0	22.5	21.4	20.3	18.5	13.8	12.1	11.5	11.6	12.3	12.3	13.4	13.7	13.9	14.3	15.8	18.4	
Pressure (datum), [cm-H ₂ O]	74.5	74.5	74.6	74.7	74.8	75.0	75.0	75.1	75.1	75.1	75.1	75.1	75.0	75.0	75.0	74.9	74.9	
Average of total pressure; [cm-H ₂ O]					15.81176					Average of datum pressure; [cm-H ₂ O]					74.90588			
Results of total pressure traversing effect at flow coefficient ($\phi=0.493$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	22.2	21.4	20.7	19.6	17.5	13.6	11.3	11.4	10.8	10.8	11.0	10.7	11.1	11.7	12.7	13.9	16.1	
Pressure (datum), [cm-H ₂ O]	76.3	76.3	76.3	76.4	76.4	76.7	76.8	76.8	76.3	76.8	76.9	76.9	76.8	76.9	76.8	76.7	76.7	
Average of total pressure; [cm-H ₂ O]					14.50000					Average of datum pressure; [cm-H ₂ O]					76.63529			
Results of total pressure traversing effect at flow coefficient ($\phi=0.467$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	22.0	21.4	20.2	19.7	17.4	13.3	11.2	11.1	11.2	10.5	10.2	10.1	10.4	11.1	11.9	13.4	15.4	
Pressure (datum), [cm-H ₂ O]	77.2	77.2	77.3	77.3	77.3	77.5	77.6	77.6	77.6	77.6	77.6	77.6	77.6	77.5	77.5	77.5	77.3	
Average of total pressure; [cm-H ₂ O]					14.14706					Average of datum pressure; [cm-H ₂ O]					77.45882			
Results of total pressure traversing effect at flow coefficient ($\phi=0.455$)																		
Position of Probe, [mm]	1.0	2.0	3.0	4.0	5.0	11.0	17.0	23.0	30.0	37.0	43.0	49.0	55.0	56.0	57.0	58.0	59.0	
Pressure (total), [cm-H ₂ O]	22.1	21.4	20.6	19.3	18.0	13.3	11.4	10.7	10.7	10.4	10.1	9.8	10.1	10.6	11.7	13.1	15.4	
Pressure (datum), [cm-H ₂ O]	77.5	77.5	77.5	77.6	77.7	77.9	78.0	78.1	78.1	78.1	78.1	78.1	78.1	78.1	78.0	77.9	77.8	
Average of total pressure; [cm-H ₂ O]					14.04118					Average of datum pressure; [cm-H ₂ O]					77.88824			
[Table (5-1)]- Total Pressure Traversing of Builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]																		

Corrected Mass Flow Rate, ($\Phi\epsilon$), (without total pressure probe traversing effect)										
3.02591	2.91820	2.81049	2.70278	2.59507	2.48736	2.37965	2.27194	2.16423	2.05652	1.94881
Efficiency, (overall total-total) $\{[(\eta)t-t]\text{overall}\}$, (without total pressure probe traversing effect)										
0.71953	0.75637	0.78655	0.81007	0.82693	0.83713	0.84066	0.83753	0.82775	0.81130	0.78818
Pressure Rise Coefficient (overall total-total), $\{[(\psi)t-t]\text{overall}\}$, (without total pressure probe traversing)										
0.19175	0.24219	0.28808	0.32943	0.36624	0.39850	0.42622	0.44940	0.46803	0.48212	0.49167
Corrected Mass Flow Rate, ($\Phi\epsilon$), (with total pressure probe traversing effect)										
3.02568	2.91799	2.81029	2.70259	2.59490	2.48720	2.37950	2.27180	2.16411	2.05641	1.94871
Efficiency, (overall total-total) $\{[(\eta)t-t]\text{overall}\}$, (with total pressure probe traversing effect)										
0.58834	0.64008	0.68390	0.71980	0.74777	0.76782	0.77994	0.78414	0.78041	0.76876	0.74919
Pressure Rise Coefficient (overall total-total), $\{[(\psi)t-t]\text{overall}\}$, (with total pressure probe traversing)										
0.15602	0.20652	0.25276	0.29472	0.33242	0.36584	0.39498	0.41986	0.44047	0.45680	0.46886

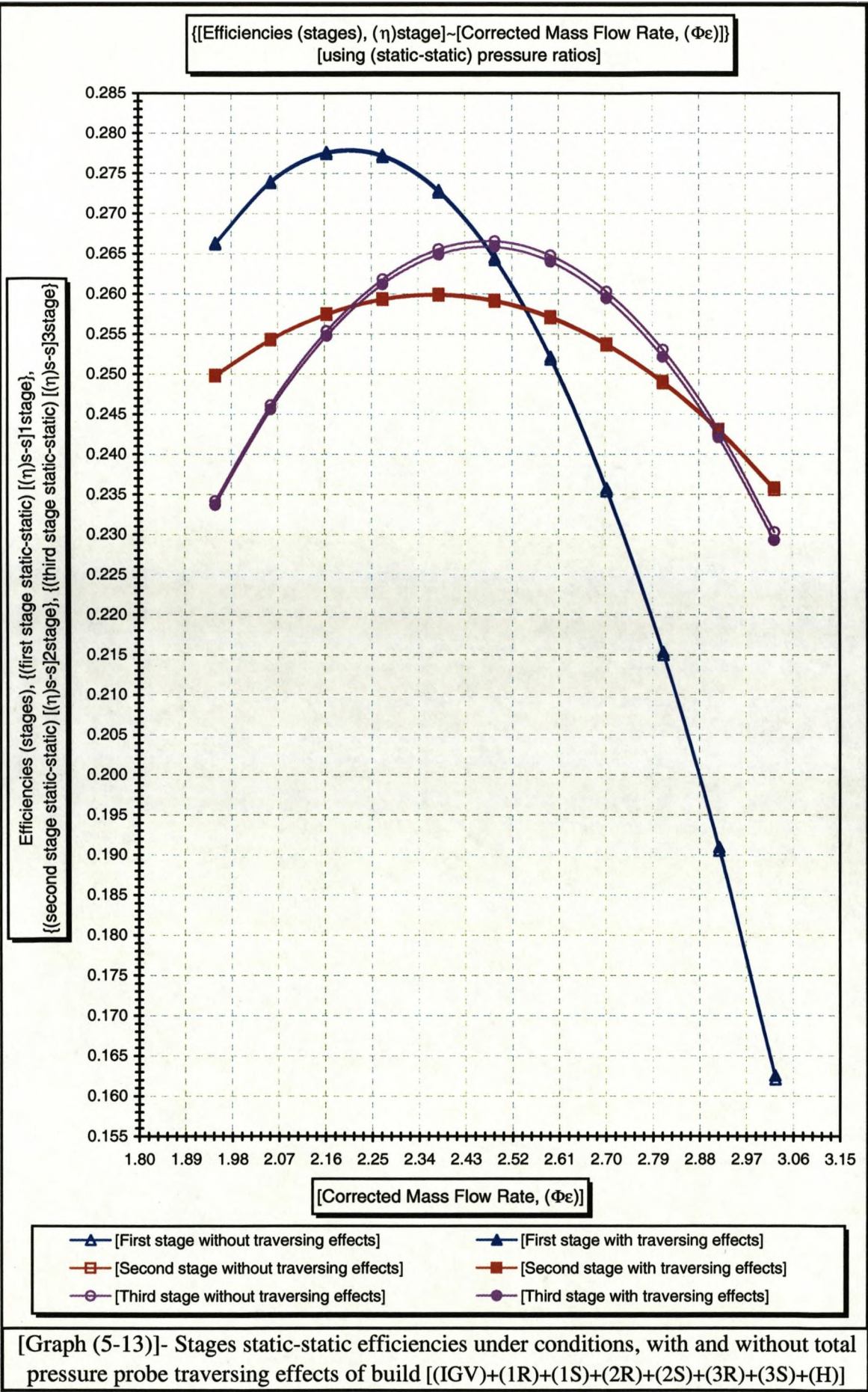


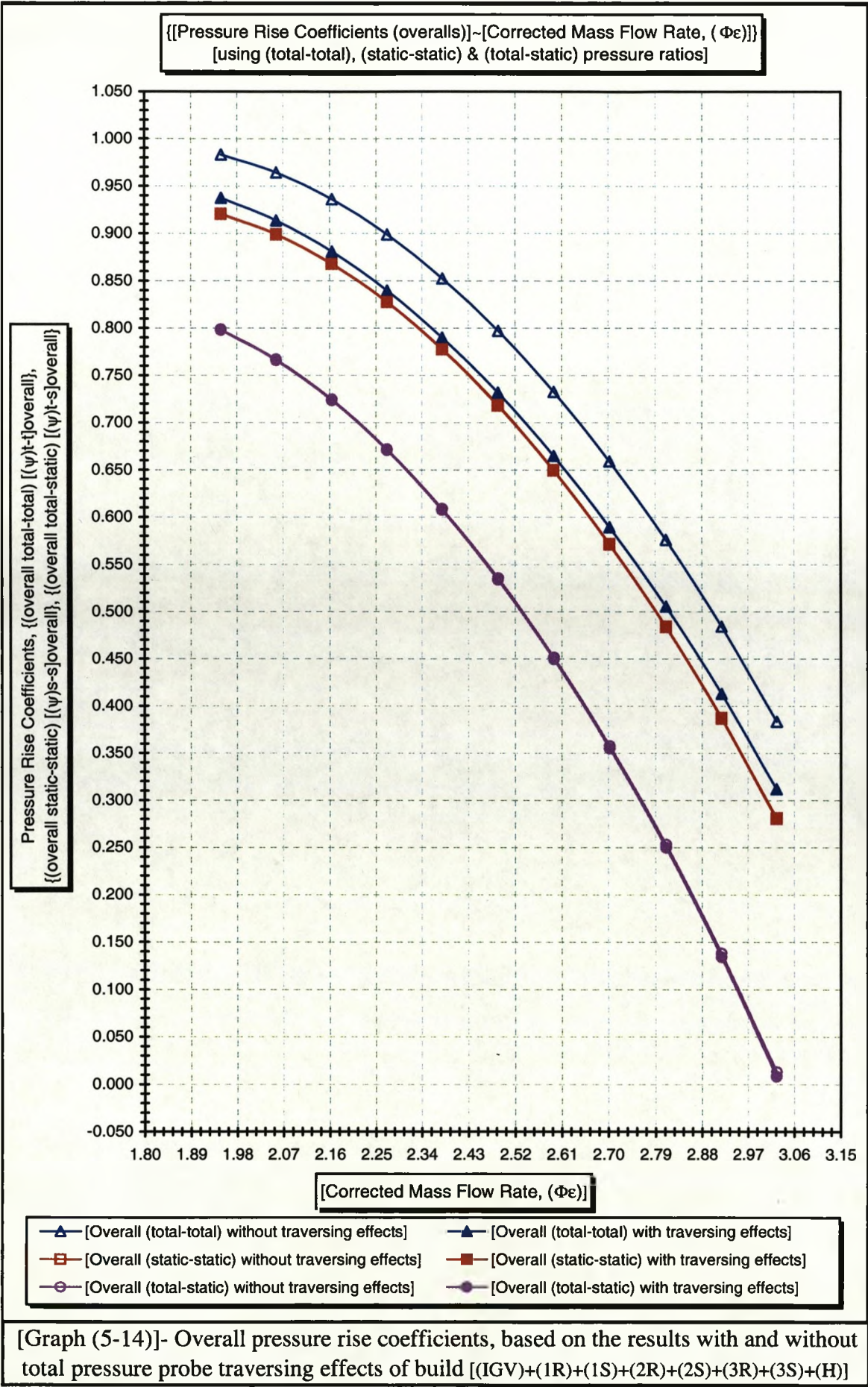
[Graph (5-11)]- Efficiencies and pressure rise coefficients (overall total-total) of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)], with and without pressure probe traversing

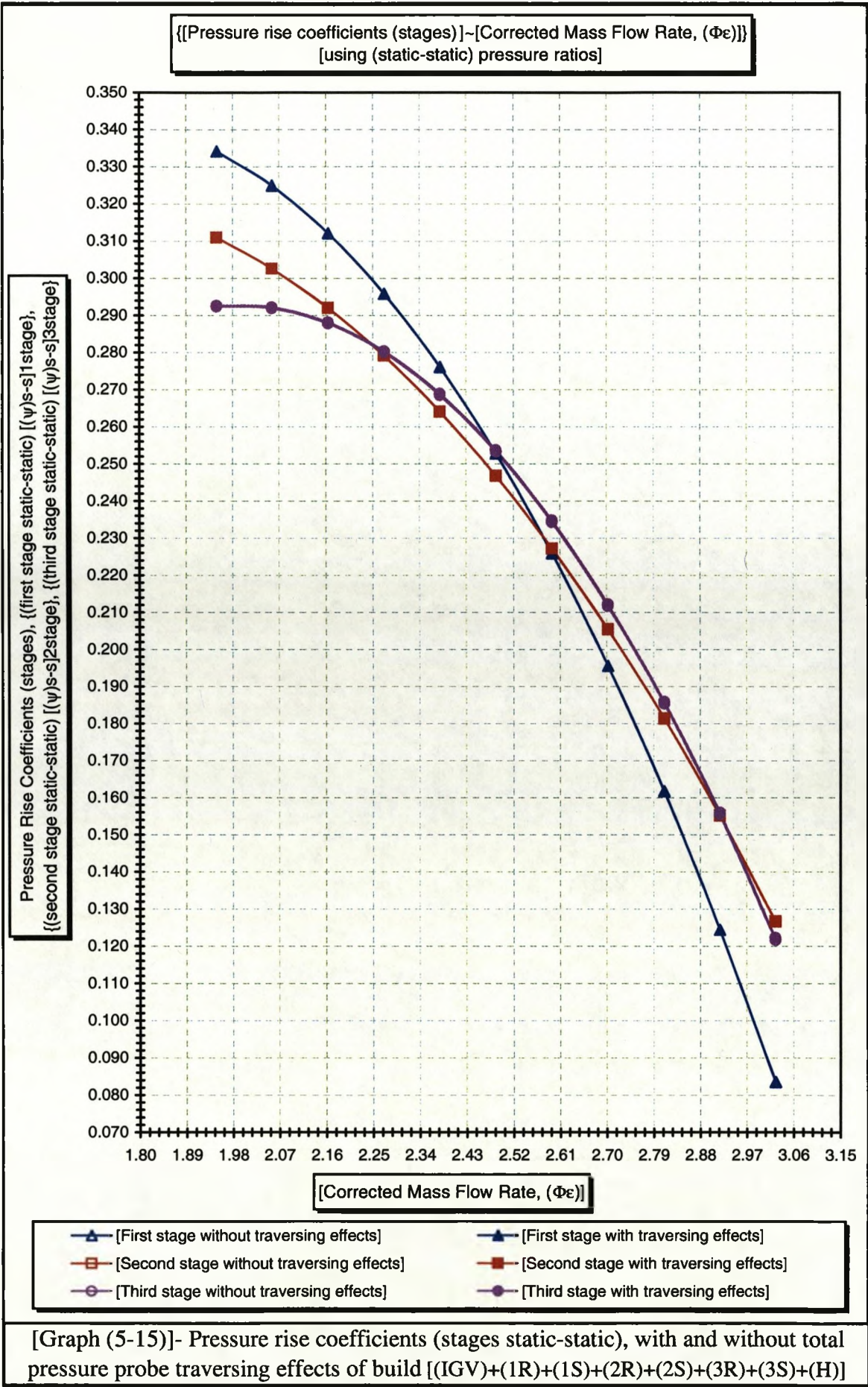
Characteristics of the compressor without total pressure probe traversing effects										
Corrected Mass Flow Rate, (Φ_e)										
3.02591	2.91820	2.81049	2.70278	2.59507	2.48736	2.37965	2.27194	2.16423	2.05652	1.94881
Efficiency (overall total-total), $\{[(\eta)t-t]\text{overall}\}$										
0.71953	0.75637	0.78655	0.81007	0.82693	0.83713	0.84066	0.83753	0.82775	0.81130	0.78818
Efficiency (overall static-static), $\{[(\eta)s-s]\text{overall}\}$										
0.53595	0.59828	0.65118	0.69465	0.72869	0.75330	0.76848	0.77423	0.77056	0.75745	0.73491
Efficiency (overall total-static), $\{[(\eta)t-s]\text{overall}\}$										
0.05322	0.18988	0.30891	0.41033	0.49413	0.56030	0.60886	0.63980	0.65311	0.64881	0.62688
Efficiency (first stage static-static), $\{[(\eta)s-s]-(1R+1S)\}$										
0.16209	0.19053	0.21497	0.23539	0.25181	0.26422	0.27263	0.27703	0.27742	0.27380	0.26618
Efficiency (second stage static-static), $\{[(\eta)s-s]-(2R+2S)\}$										
0.23552	0.24284	0.24885	0.25354	0.25693	0.25900	0.25976	0.25922	0.25736	0.25419	0.24971
Efficiency (third stage static-static), $\{[(\eta)s-s]-(3R+3S)\}$										
0.23028	0.24303	0.25302	0.26027	0.26478	0.26654	0.26556	0.26183	0.25535	0.24614	0.23417
Pressure Rise Coefficient (overall total-total), $\{[(\psi)t-t]\text{overall}\}$										
0.38351	0.48438	0.57616	0.65886	0.73248	0.79700	0.85244	0.89880	0.93607	0.96425	0.98334
Pressure Rise Coefficient (overall static-static), $\{[(\psi)s-s]\text{overall}\}$										
0.28128	0.38756	0.48444	0.57191	0.64998	0.71865	0.77791	0.82777	0.86822	0.89927	0.92091
Pressure Rise Coefficient (overall total-static), $\{[(\psi)t-s]\text{overall}\}$										
0.01292	0.13819	0.25308	0.35759	0.45173	0.53549	0.60888	0.67189	0.72452	0.76678	0.79866
Pressure Rise Coefficient (first stage static-static), $\{[(\psi)s-s]-(1R+1S)\}$										
0.08355	0.12441	0.16176	0.19560	0.22592	0.25274	0.27605	0.29584	0.31213	0.32490	0.33417
Pressure Rise Coefficient (second stage static-static), $\{[(\psi)s-s]-(2R+2S)\}$										
0.12665	0.15513	0.18138	0.20539	0.22718	0.24673	0.26405	0.27914	0.29199	0.30262	0.31101
Pressure Rise Coefficient (third stage static-static), $\{[(\psi)s-s]-(3R+3S)\}$										
0.12165	0.15537	0.18538	0.21169	0.23431	0.25323	0.26845	0.27997	0.28779	0.29192	0.29234
Characteristics of the compressor with total pressure probe traversing effects										
Corrected Mass Flow Rate, (Φ_e)										
3.02568	2.91799	2.81029	2.70259	2.59490	2.48720	2.37950	2.27180	2.16411	2.05641	1.94871
Efficiency (overall total-total), $\{[(\eta)t-t]\text{overall}\}$										
0.58834	0.64008	0.68390	0.71980	0.74777	0.76782	0.77994	0.78414	0.78041	0.76876	0.74919
Efficiency (overall static-static), $\{[(\eta)s-s]\text{overall}\}$										
0.53588	0.59821	0.65110	0.69456	0.72860	0.75321	0.76838	0.77413	0.77044	0.75733	0.73479
Efficiency (overall total-static), $\{[(\eta)t-s]\text{overall}\}$										
0.04562	0.18383	0.30423	0.40682	0.49159	0.55855	0.60770	0.63903	0.65255	0.64826	0.62615
Efficiency (first stage static-static), $\{[(\eta)s-s]-(1R+1S)\}$										
0.16256	0.19097	0.21537	0.23577	0.25215	0.26454	0.27291	0.27728	0.27765	0.27401	0.26636
Efficiency (second stage static-static), $\{[(\eta)s-s]-(2R+2S)\}$										
0.23584	0.24314	0.24914	0.25382	0.25719	0.25925	0.26000	0.25944	0.25757	0.25439	0.24990
Efficiency (third stage static-static), $\{[(\eta)s-s]-(3R+3S)\}$										
0.22930	0.24210	0.25215	0.25945	0.26401	0.26582	0.26489	0.26120	0.25478	0.24560	0.23368
Pressure Rise Coefficient (overall total-total), $\{[(\psi)t-t]\text{overall}\}$										
0.31203	0.41305	0.50552	0.58945	0.66483	0.73167	0.78997	0.83972	0.88093	0.91360	0.93772
Pressure Rise Coefficient (overall static-static), $\{[(\psi)s-s]\text{overall}\}$										
0.28028	0.38664	0.48359	0.57112	0.64925	0.71798	0.77729	0.82719	0.86769	0.89877	0.92045
Pressure Rise Coefficient (overall total-static), $\{[(\psi)t-s]\text{overall}\}$										
0.00820	0.13422	0.24979	0.35490	0.44955	0.53373	0.60746	0.67073	0.72354	0.76589	0.79778
Pressure Rise Coefficient (first stage static-static), $\{[(\psi)s-s]-(1R+1S)\}$										
0.08354	0.12440	0.16176	0.19560	0.22593	0.25275	0.27606	0.29586	0.31214	0.32491	0.33417
Pressure Rise Coefficient (second stage static-static), $\{[(\psi)s-s]-(2R+2S)\}$										
0.12681	0.15527	0.18150	0.20550	0.22727	0.24681	0.26411	0.27918	0.29202	0.30263	0.31101
Pressure Rise Coefficient (third stage static-static), $\{[(\psi)s-s]-(3R+3S)\}$										
0.12234	0.15600	0.18597	0.21225	0.23482	0.25370	0.26888	0.28036	0.28815	0.29223	0.29262
[Table (5-2)]- Efficiencies and pressure rise coefficients (overalls and stages) with and without total pressure probe traversing effects of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]										



[Graph (5-12)]- Efficiencies, overalls (total-total), (static-static) and (total-static) of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)] with and without pressure-probe traversing



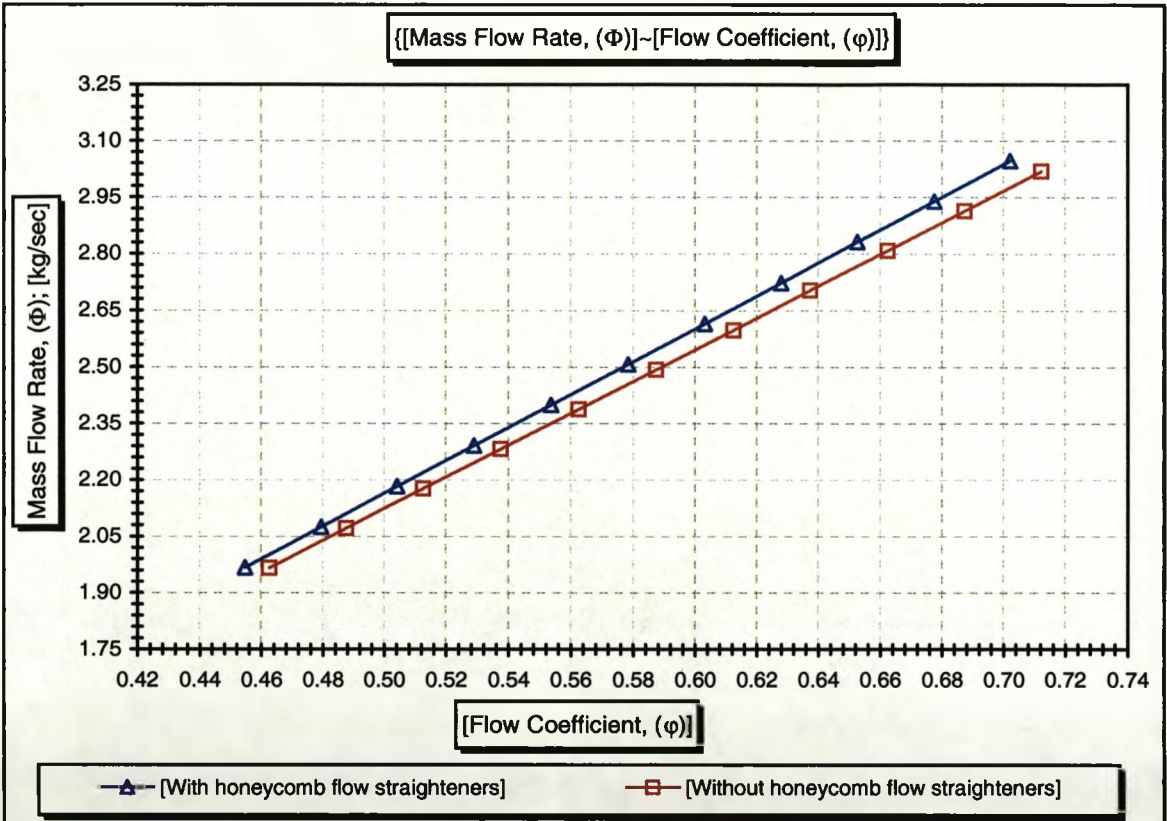




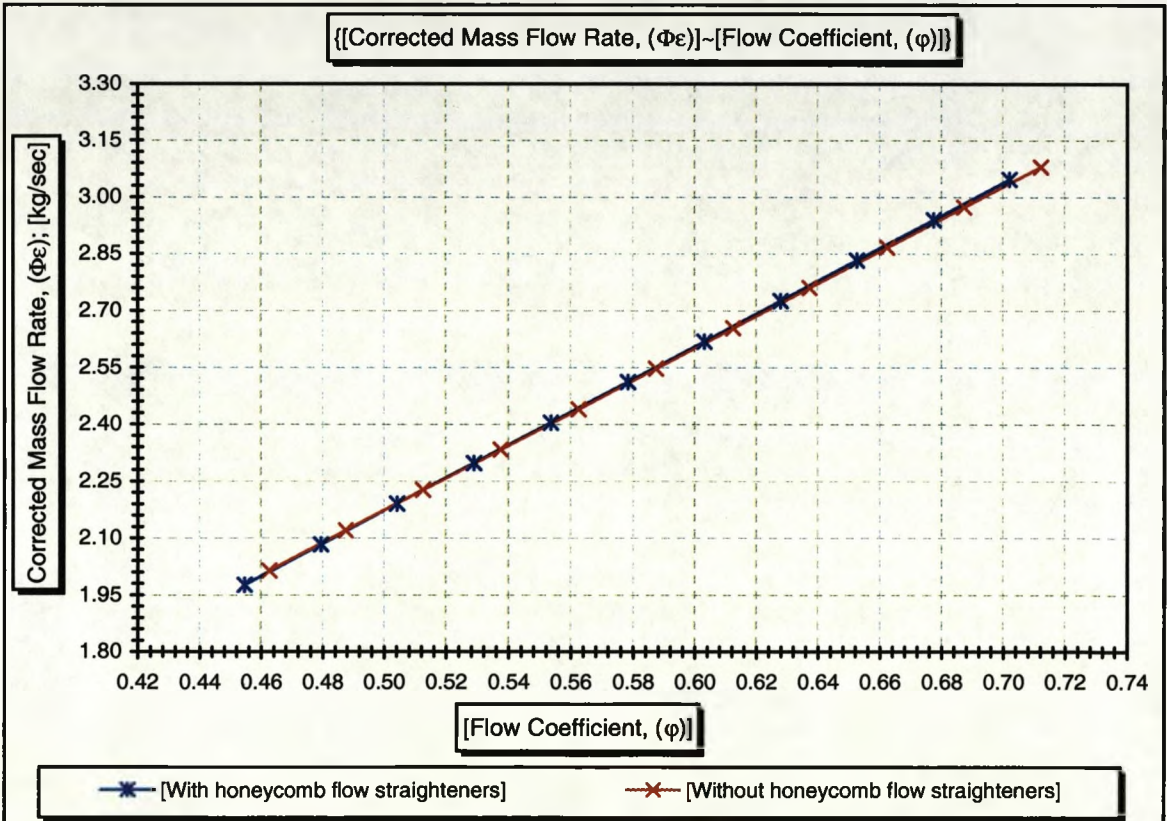
Flow (overalls and stages) results, without/with pressure-probe traversing effects	Without	With
Flow coefficient at stall point, $[(\phi)_{sp}]$	0.45345	0.45361
Mass flow rate at stall point, $[(\Phi)_{sp}]$; [kg/sec]	1.92066	1.92057
Corrected mass flow rate at stall point, $[(\Phi\epsilon)_{sp}]$; [kg/sec]	1.94881	1.94871
Maximum corrected applied power, $[(\Omega\epsilon)_{max}]$; [kwatt]	6.81167	6.81063
Maximum work coefficient (overall), $[(\Psi)_{max-overall}]$	1.23833	1.23818
Maximum efficiency (overall total-total), $\{[(\eta)_{max}]_{overall-(t-t)}\}$	0.84066	0.78414
Efficiency (overall total-total) at stall point, $\{[(\eta)_{sp}]_{overall-(t-t)}\}$	0.78818	0.74919
Maximum efficiency (overall static-static), $\{[(\eta)_{max}]_{overall-(s-s)}\}$	0.77429	0.77418
Efficiency (overall static-static) at stall point, $\{[(\eta)_{sp}]_{overall-(s-s)}\}$	0.73491	0.73479
Maximum efficiency (overall total-static), $\{[(\eta)_{max}]_{overall-(t-s)}\}$	0.65369	0.65315
Efficiency (overall total-static) at stall point, $\{[(\eta)_{sp}]_{overall-(t-s)}\}$	0.62688	0.62615
Maximum efficiency (first stage static-static), $\{[(\eta)_{max}]_{(1R+1S)-(s-s)}\}$	0.27774	0.27798
Efficiency (first stage static-static) at stall point, $\{[(\eta)_{sp}]_{(1R+1S)-(s-s)}\}$	0.26618	0.26636
Maximum efficiency (second stage static-static), $\{[(\eta)_{max}]_{(2R+2S)-(s-s)}\}$	0.25977	0.26001
Efficiency (second stage static-static) at stall point, $\{[(\eta)_{sp}]_{(2R+2S)-(s-s)}\}$	0.24971	0.24990
Maximum efficiency (third stage static-static), $\{[(\eta)_{max}]_{(3R+3S)-(s-s)}\}$	0.26657	0.26586
Efficiency (third stage static-static) at stall point, $\{[(\eta)_{sp}]_{(3R+3S)-(s-s)}\}$	0.23417	0.23368
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{peak}]_{overall-(t-t)}\}$	0.98334	0.93772
Pressure rise coefficient (overall total-total) at stall point, $\{[(\psi)_{sp}]_{overall-(t-t)}\}$	0.98334	0.93772
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{peak}]_{overall-(s-s)}\}$	0.92091	0.92045
Pressure rise coefficient (overall static-static) at stall point, $\{[(\psi)_{sp}]_{overall-(s-s)}\}$	0.92091	0.92045
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{peak}]_{overall-(t-s)}\}$	0.79866	0.79778
Pressure rise coefficient (overall total-static) at stall point, $\{[(\psi)_{sp}]_{overall-(t-s)}\}$	0.79866	0.79778
Peak pressure rise coefficient (first stage static-static), $\{[(\psi)_{peak}]_{(1R+1S)-(s-s)}\}$	0.33417	0.33417
Pressure rise coefficient (first stage static-static) at stall point, $\{[(\psi)_{sp}]_{(1R+1S)-(s-s)}\}$	0.33417	0.33417
Peak pressure rise coefficient (second stage static-static), $\{[(\psi)_{peak}]_{(2R+2S)-(s-s)}\}$	0.31101	0.31101
Pressure rise coefficient (second stage static-static) at stall point, $\{[(\psi)_{sp}]_{(2R+2S)-(s-s)}\}$	0.31101	0.31101
Peak pressure rise coefficient (third stage static-static), $\{[(\psi)_{peak}]_{(3R+3S)-(s-s)}\}$	0.29262	0.29291
Pressure rise coefficient (third stage static-static) at stall point, $\{[(\psi)_{sp}]_{(3R+3S)-(s-s)}\}$	0.29234	0.29262
[Table (5-3)]- Some of the comparison results obtained from builds with/without pressure probe traversing effects, under build of [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]		

Flow coefficient at compressor annulus, (ϕ)	0.70221	0.67747	0.65272	0.62797	0.60323	0.57848	0.55374	0.52899	0.50424	0.47950	0.45475
Mass flow rate at venturi throat, (Φ); [kg/sec]	3.04722	2.93922	2.83122	2.72322	2.61522	2.50722	2.39922	2.29122	2.18322	2.07522	1.96722
Corrected mass flow rate, [(Φ_e)]; [kg/sec]	3.04722	2.94014	2.83306	2.72598	2.61890	2.51182	2.40474	2.29766	2.19058	2.08350	1.97642
Corrected applied power, [(Ω_e)]; [kwatt]	4.64825	5.20450	5.68478	6.08922	6.41795	6.67111	6.84883	6.95124	6.97848	6.93067	6.80795
Work coefficient (overall), [(Ψ) overall]	0.53601	0.62631	0.71196	0.79296	0.86931	0.94100	1.00804	1.07042	1.12816	1.18124	1.22966
Efficiency, (overall total-total), [$(\eta)t-t$ overall]	0.78340	0.80280	0.81780	0.82839	0.83458	0.83637	0.83376	0.82674	0.81532	0.79949	0.77927
Efficiency, (overall static-static), [$(\eta)s-s$ overall]	0.53863	0.60362	0.65875	0.70405	0.73950	0.76510	0.78086	0.78678	0.78285	0.76907	0.74545
Efficiency, (overall total-static), [$(\eta)t-s$ overall]	0.05180	0.19147	0.31315	0.41685	0.50256	0.57028	0.62001	0.65176	0.66551	0.66128	0.63906
Efficiency, (first stage static-static), [$(\eta)s-s$ -(1R+1S)]	0.16878	0.19525	0.21807	0.23725	0.25278	0.26466	0.27290	0.27749	0.27844	0.27574	0.26939
Efficiency, (second stage static-static), [$(\eta)s-s$ -(2R+2S)]	0.23457	0.24393	0.25166	0.25776	0.26222	0.26506	0.26627	0.26584	0.26379	0.26010	0.25479
Efficiency, (third stage static-static), [$(\eta)s-s$ -(3R+3S)]	0.23171	0.24475	0.25501	0.26250	0.26722	0.26916	0.26832	0.26471	0.25833	0.24917	0.23723
Pressure rise coefficient, (overall total-total), [$(\psi)t-t$ overall]	0.41558	0.50509	0.58686	0.66091	0.72721	0.78578	0.83662	0.87972	0.91509	0.94272	0.96262
Pressure rise coefficient, (overall static-static), [$(\psi)s-s$ overall]	0.28006	0.38473	0.48057	0.56758	0.64575	0.71509	0.77559	0.82727	0.87011	0.90411	0.92928
Pressure rise coefficient, (overall total-static), [$(\psi)t-s$ overall]	0.01169	0.13318	0.24545	0.34848	0.44228	0.52686	0.60220	0.66831	0.72519	0.77285	0.81127
Pressure rise coefficient, (first stage static-static), [$(\psi)s-s$ -(1R+1S)]	0.08658	0.12480	0.16007	0.19238	0.22173	0.24812	0.27155	0.29203	0.30954	0.32410	0.33570
Pressure rise coefficient, (second stage static-static), [$(\psi)s-s$ -(2R+2S)]	0.12418	0.15353	0.18061	0.20540	0.22792	0.24816	0.26611	0.28179	0.29519	0.30631	0.31515
Pressure rise coefficient, (third stage static-static), [$(\psi)s-s$ -(3R+3S)]	0.12264	0.15538	0.18467	0.21053	0.23296	0.25194	0.26749	0.27960	0.28827	0.29350	0.29530
[Table (5-4)]- The overalls and stages characteristic results, with the honeycomb flow straighteners, of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+(H)]											

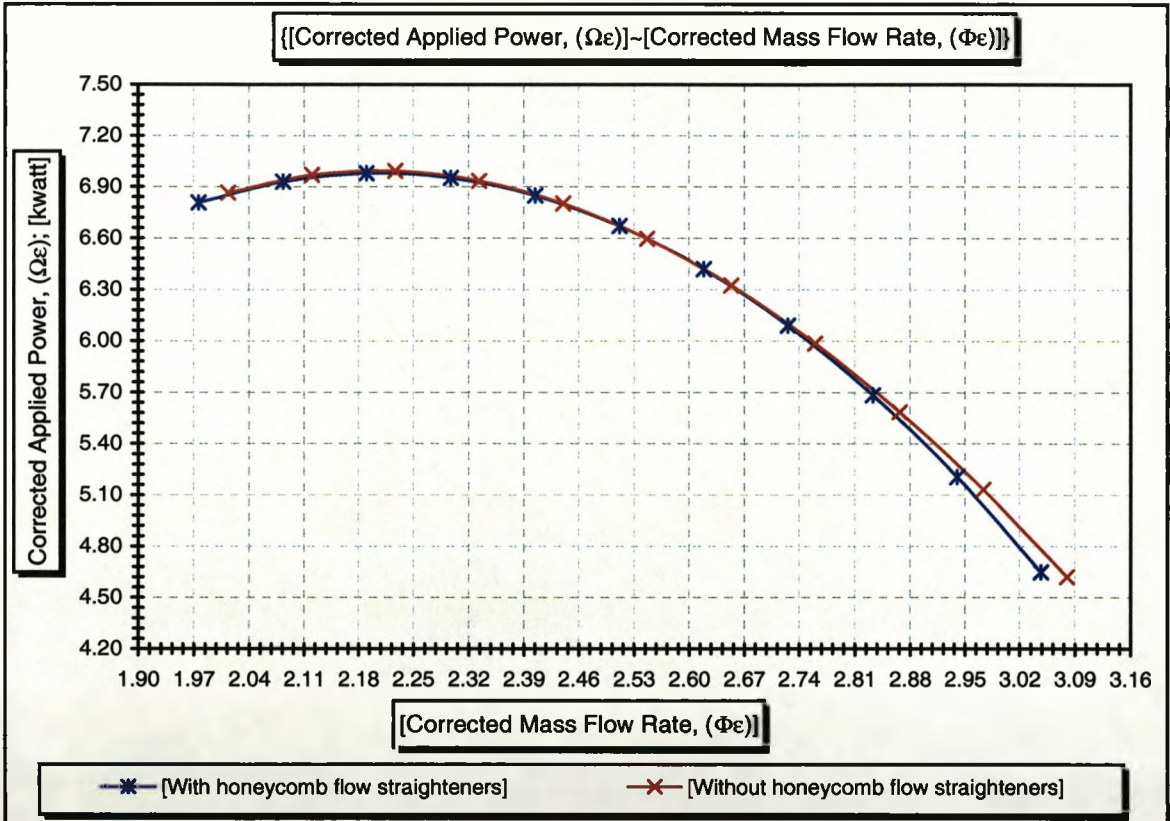
Flow coefficient at compressor annulus, (ϕ)	0.71249	0.68753	0.66256	0.63760	0.61264	0.58767	0.56271	0.53775	0.51278	0.48782	0.46286
Mass flow rate at venturi throat, (Φ); [kg/sec]	3.01888	2.91368	2.80847	2.70327	2.59806	2.49286	2.38765	2.28245	2.17724	2.07204	1.96683
Corrected mass flow rate, [$(\Phi\epsilon)$]; [kg/sec]	3.08026	2.97362	2.86699	2.76036	2.65372	2.54709	2.44045	2.33382	2.22719	2.12055	2.01392
Corrected applied power, [$(\Omega\epsilon)$]; [kwatt]	4.62084	5.13052	5.58663	5.98547	6.32337	6.59666	6.80164	6.93465	6.99200	6.97002	6.86502
Work coefficient (overall), [(Ψ) overall]	0.52878	0.61509	0.69760	0.77629	0.85116	0.92223	0.98949	1.05293	1.11256	1.16838	1.22039
Efficiency, (overall total-total), [$(\eta)t-t$ overall]	0.72104	0.75909	0.79041	0.81501	0.83289	0.84404	0.84847	0.84617	0.83715	0.82140	0.79893
Efficiency, (overall static-static), [$(\eta)s-s$ overall]	0.52206	0.59045	0.64851	0.69622	0.73358	0.76061	0.77729	0.78363	0.77962	0.76528	0.74059
Efficiency, (overall total-static), [$(\eta)t-s$ overall]	0.03257	0.17272	0.29514	0.39985	0.48684	0.55610	0.60765	0.64148	0.65758	0.65597	0.63663
Efficiency, (first stage static-static), [$(\eta)s-s$ -(1R+1S)]	0.17178	0.19869	0.22170	0.24081	0.25603	0.26734	0.27476	0.27828	0.27790	0.27363	0.26545
Efficiency, (second stage static-static), [$(\eta)s-s$ -(2R+2S)]	0.22792	0.23843	0.24714	0.25405	0.25916	0.26246	0.26397	0.26367	0.26157	0.25768	0.25198
Efficiency, (third stage static-static), [$(\eta)s-s$ -(3R+3S)]	0.22319	0.23875	0.25111	0.26026	0.26622	0.26898	0.26853	0.26489	0.25805	0.24801	0.23476
Pressure rise coefficient, (overall total-total), [$(\psi)t-t$ overall]	0.37650	0.47252	0.56066	0.64094	0.71335	0.77788	0.83455	0.88334	0.92426	0.95731	0.98249
Pressure rise coefficient, (overall static-static), [$(\psi)s-s$ overall]	0.26814	0.37210	0.46736	0.55390	0.63173	0.70085	0.76126	0.81296	0.85595	0.89023	0.91581
Pressure rise coefficient, (overall total-static), [$(\psi)t-s$ overall]	0.00352	0.12316	0.23391	0.33576	0.42872	0.51278	0.58795	0.65423	0.71161	0.76010	0.79969
Pressure rise coefficient, (first stage static-static), [$(\psi)s-s$ -(1R+1S)]	0.08693	0.12529	0.16048	0.19250	0.22134	0.24702	0.26953	0.28886	0.30503	0.31803	0.32785
Pressure rise coefficient, (second stage static-static), [$(\psi)s-s$ -(2R+2S)]	0.11957	0.14831	0.17489	0.19931	0.22156	0.24166	0.25960	0.27538	0.28899	0.30045	0.30975
Pressure rise coefficient, (third stage static-static), [$(\psi)s-s$ -(3R+3S)]	0.11641	0.14971	0.17947	0.20570	0.22840	0.24756	0.26319	0.27529	0.28385	0.28887	0.29037
[Table (5-5)]- The overalls and stages characteristic results, without the honeycomb flow straighteners, of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)-(H)]											



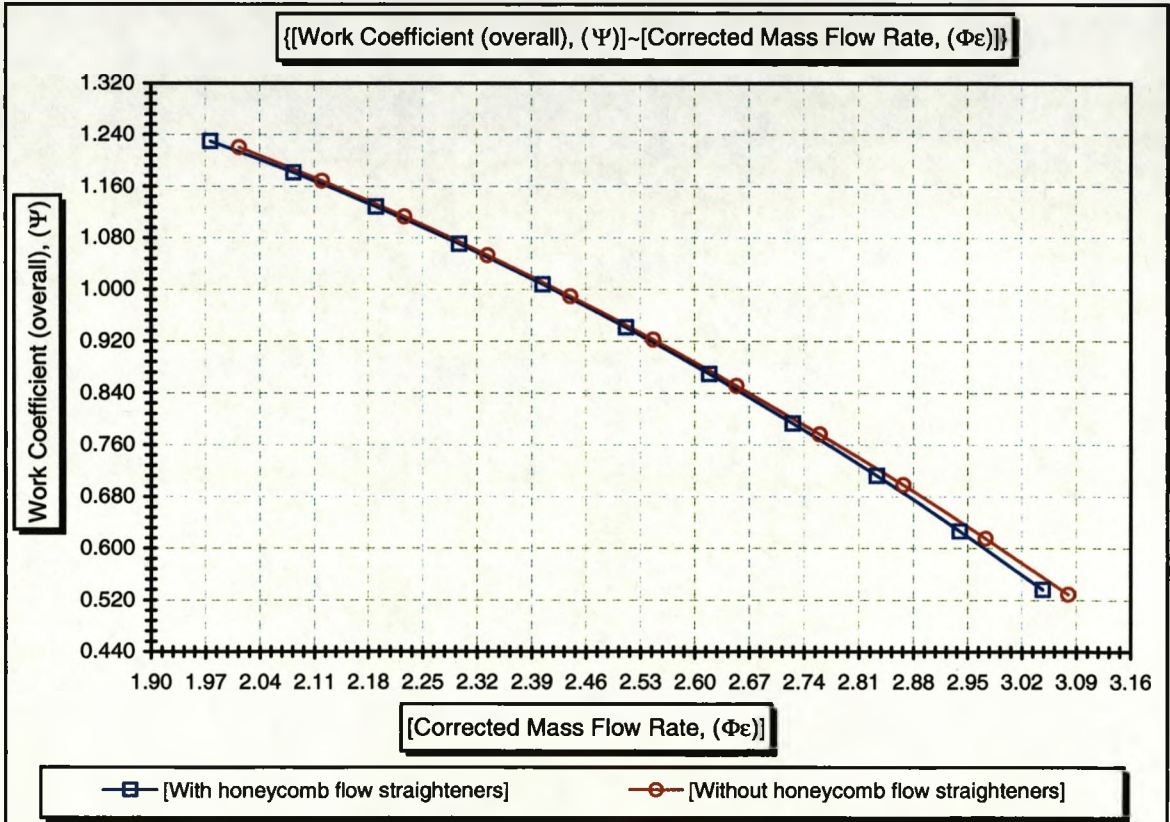
[Graph (5-16)]- Mass flow rate alteration in the present and absence of the honeycombs in the flow straighteners of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/-(H)]



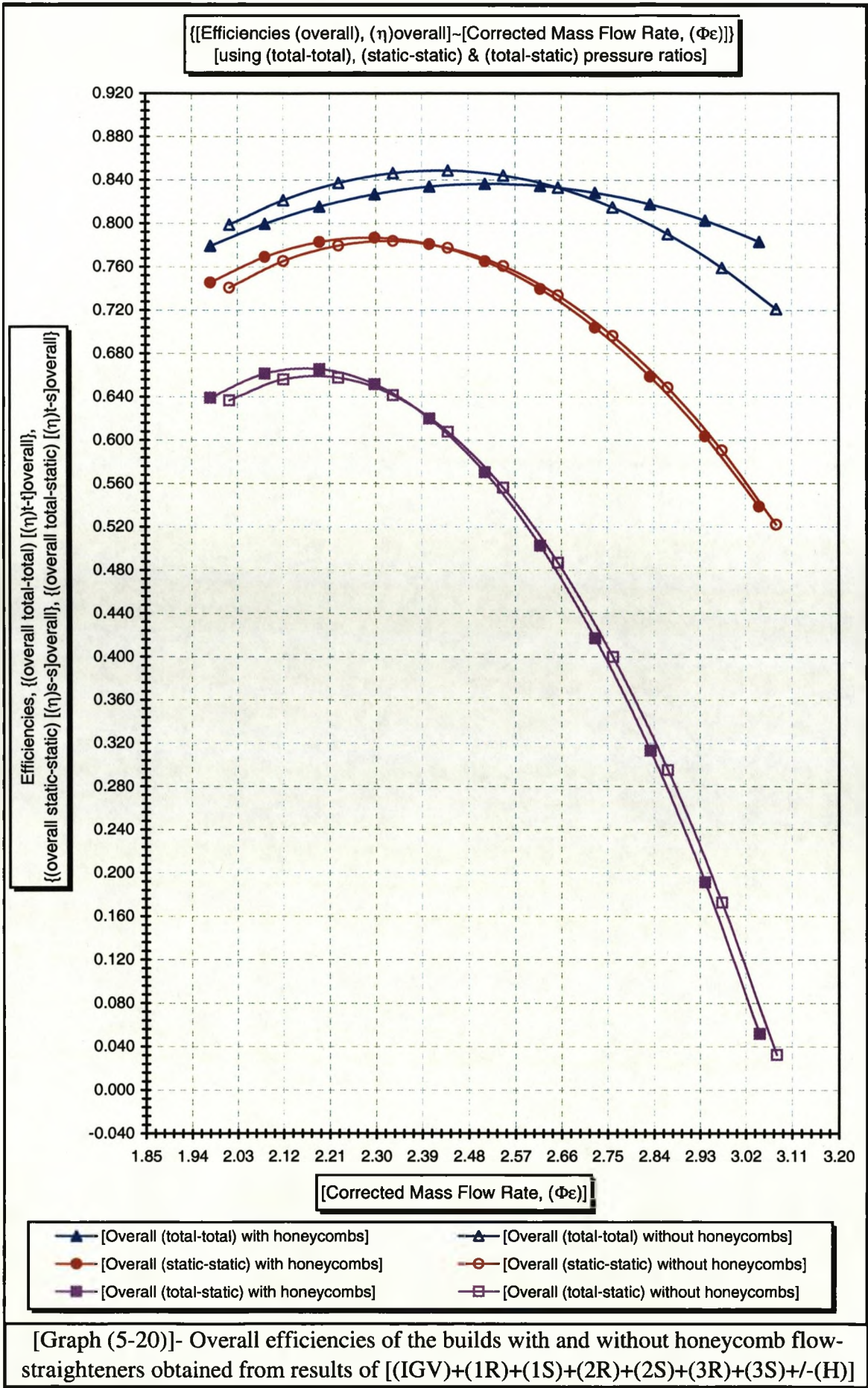
[Graph (5-17)]- Status of the corrected mass flow rate with and without honeycombs in the flow straighteners of build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/-(H)]

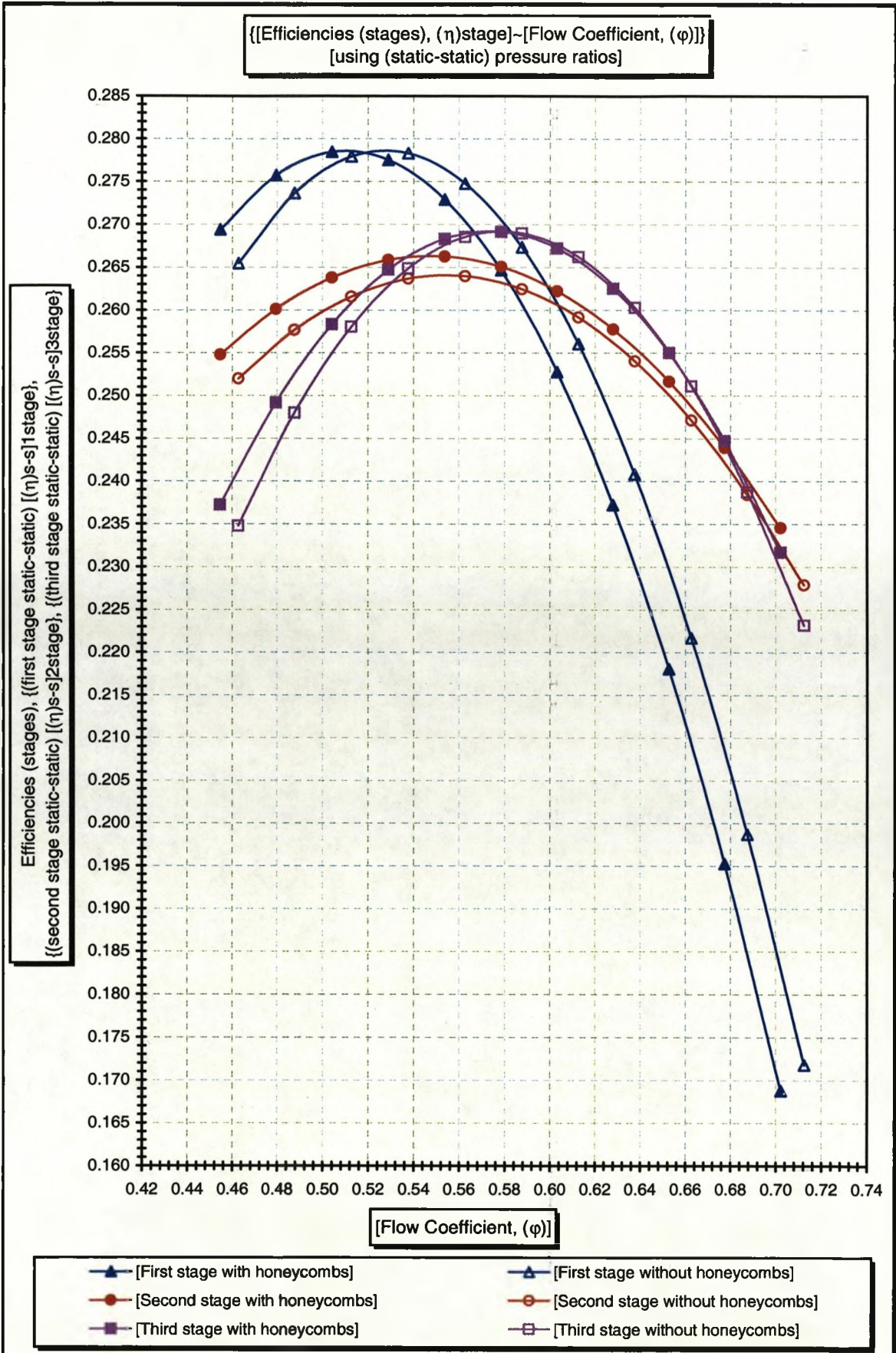


[Graph (5-18)]- Corrected applied power in conditions with and without the honeycomb flow straighteners of the build [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/(H)]

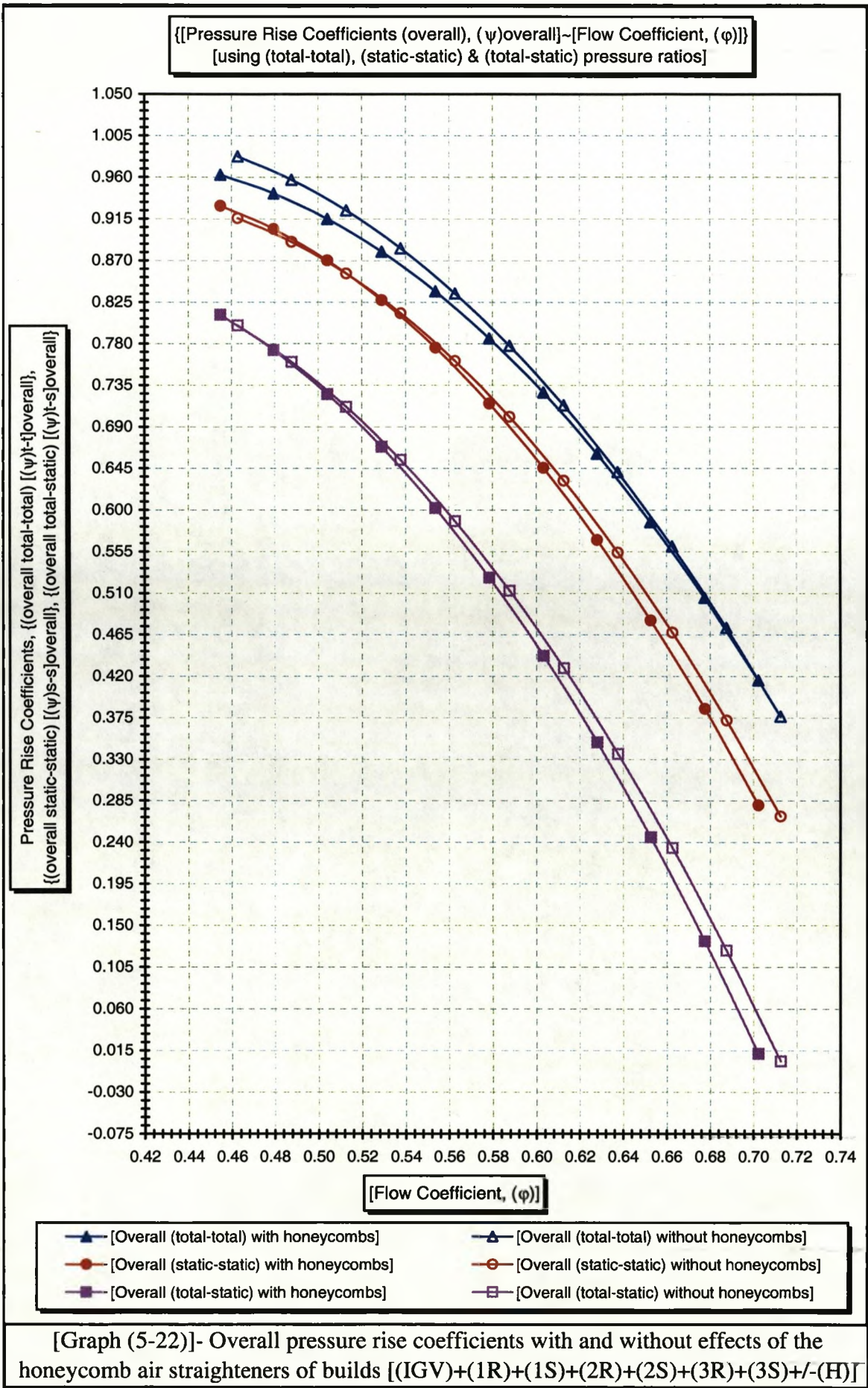


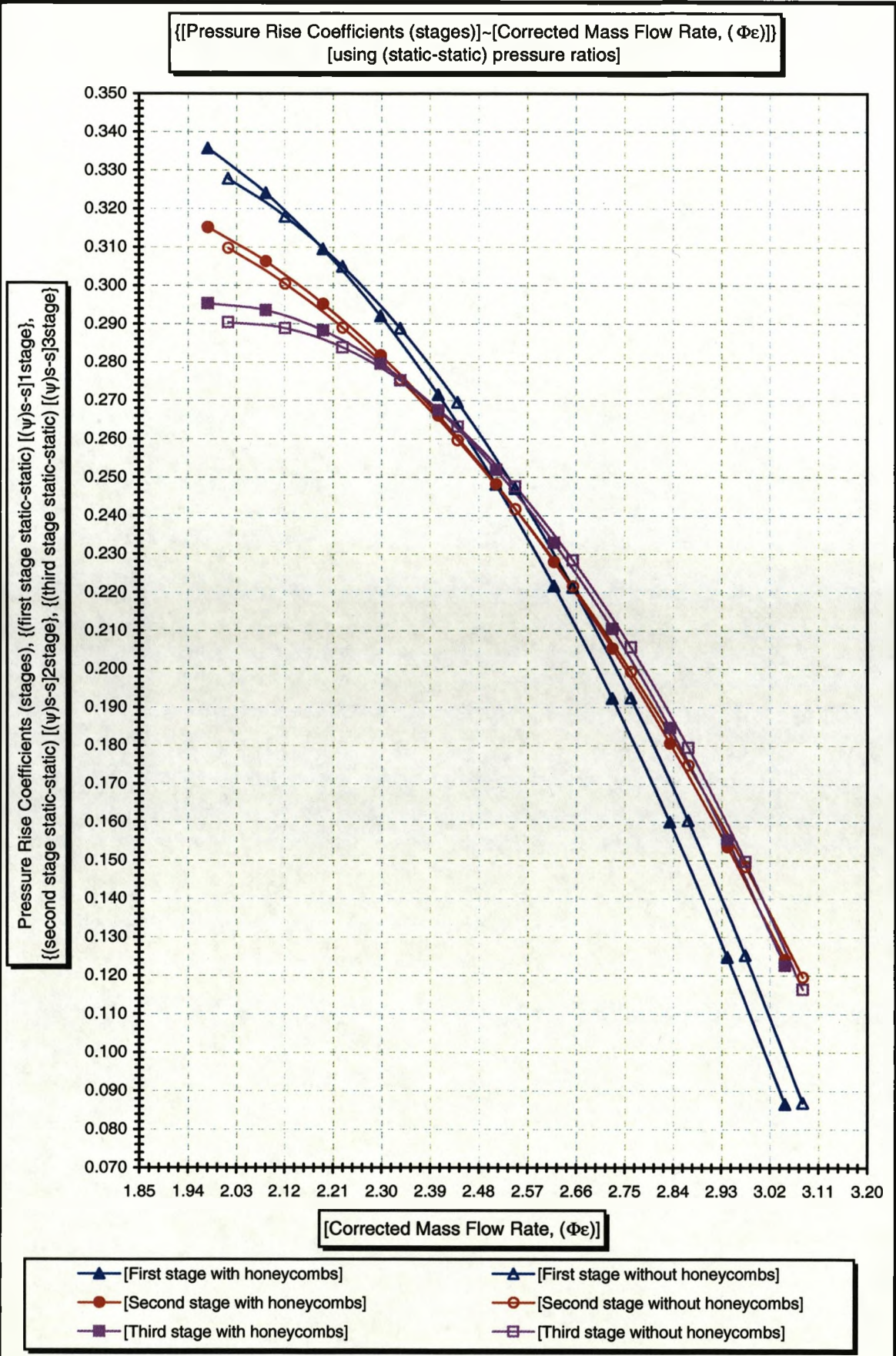
[Graph (5-19)]- Overall work coefficient in present and absence of the honeycomb flow straighteners, obtained from results of builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/(H)]





[Graph (5-21)]- Stages static-static efficiencies, with and without honeycomb flow straighteners obtained from results of the builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/-(H)]





[Graph (5-23)]- Pressure rise coefficients (stages static-static), with/ without honeycomb flow straighteners based on the results of builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/-(H)]

Flow (overalls/stages) results, without/with honeycomb flow straighteners effects	With	Without
Flow coefficient at stall point, $[(\phi)_{sp}]$	0.45475	0.46286
Mass flow rate at stall point, $[(\Phi)_{sp}]$; [kg/sec]	1.96722	1.96683
Corrected mass flow rate at stall point, $[(\Phi \epsilon)_{sp}]$; [kg/sec]	1.97642	2.01392
Maximum corrected applied power, $[(\Omega \epsilon)_{max}]$; [kwatt]	6.97918	6.99410
Maximum work coefficient (overall), $[(\Psi)_{max-overall}]$	1.22966	1.22039
Maximum efficiency (overall total-total), $\{[(\eta)_{max}]_{overall-(t-t)}\}$	0.83639	0.84855
Efficiency (overall total-total) at stall point, $\{[(\eta)_{sp}]_{overall-(t-t)}\}$	0.77927	0.79893
Maximum efficiency (overall static-static), $\{[(\eta)_{max}]_{overall-(s-s)}\}$	0.78683	0.78369
Efficiency (overall static-static) at stall point, $\{[(\eta)_{sp}]_{overall-(s-s)}\}$	0.74545	0.74059
Maximum efficiency (overall total-static), $\{[(\eta)_{max}]_{overall-(t-s)}\}$	0.66614	0.65906
Efficiency (overall total-static) at stall point, $\{[(\eta)_{sp}]_{overall-(t-s)}\}$	0.63906	0.63663
Maximum efficiency (first stage static-static), $\{[(\eta)_{max}]_{-(1R+1S)-(s-s)}\}$	0.27854	0.27860
Efficiency (first stage static-static) at stall point, $\{[(\eta)_{sp}]_{-(1R+1S)-(s-s)}\}$	0.26939	0.26545
Maximum efficiency (second stage static-static), $\{[(\eta)_{max}]_{-(2R+2S)-(s-s)}\}$	0.26632	0.26407
Efficiency (second stage static-static) at stall point, $\{[(\eta)_{sp}]_{-(2R+2S)-(s-s)}\}$	0.25479	0.25198
Maximum efficiency (third stage static-static), $\{[(\eta)_{max}]_{-(3R+3S)-(s-s)}\}$	0.26921	0.26919
Efficiency (third stage static-static) at stall point, $\{[(\eta)_{sp}]_{-(3R+3S)-(s-s)}\}$	0.23723	0.23476
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{peak}]_{overall-(t-t)}\}$	0.96262	0.98249
Pressure rise coefficient (overall total-total) at stall point, $\{[(\psi)_{sp}]_{overall-(t-t)}\}$	0.96262	0.98249
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{peak}]_{overall-(s-s)}\}$	0.92928	0.91581
Pressure rise coefficient (overall static-static) at stall point, $\{[(\psi)_{sp}]_{overall-(s-s)}\}$	0.92928	0.91581
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{peak}]_{overall-(t-s)}\}$	0.81127	0.79969
Pressure rise coefficient (overall total-static) at stall point, $\{[(\psi)_{sp}]_{overall-(t-s)}\}$	0.81127	0.79969
Peak pressure rise coefficient (first stage static-static), $\{[(\psi)_{peak}]_{-(1R+1S)-(s-s)}\}$	0.33570	0.32785
Pressure rise coefficient (first stage static-static) at stall point, $\{[(\psi)_{sp}]_{-(1R+1S)-(s-s)}\}$	0.33570	0.32785
Peak pressure rise coefficient (second stage static-static), $\{[(\psi)_{peak}]_{-(2R+2S)-(s-s)}\}$	0.31515	0.30975
Pressure rise coefficient (second stage static-static) at stall point, $\{[(\psi)_{sp}]_{-(2R+2S)-(s-s)}\}$	0.31515	0.30975
Peak pressure rise coefficient (third stage static-static), $\{[(\psi)_{peak}]_{-(3R+3S)-(s-s)}\}$	0.29530	0.29038
Pressure rise coefficient (third stage static-static) at stall point, $\{[(\psi)_{sp}]_{-(3R+3S)-(s-s)}\}$	0.29530	0.29037
[Table (5-6)]- Maximum values (overalls & stages) characteristics with and without effects of the honeycomb air straighteners of builds [(IGV)+(1R)+(1S)+(2R)+(2S)+(3R)+(3S)+/-(H)]		

CHAPTER SIX

6)- Parametric Influence of Geometrical Effects on Vane-Recessed Casing Treatments:

6.1)- Introduction:

Rotating stall is a highly complex phenomenon in an axial flow compressor particularly in a multistage one. Previous investigations have shown that simple models of casing treatment applications in multistage axial-flow compressors have not been very effective.

Removal of the inlet whirl (tangential) velocity is an essential part of the vane-recessed casing treatment operation. The majority of the previous studies in the area of casing treatment mechanisms and their corresponding application have revealed that without treated casings, there was undesirably high whirl velocity ahead of the rotor blade in the stalled regime of the flow conditions.

The following discussion is going to provide a review of the existing data obtained from previous research in order to arrive at an optimal design approach for a vane-recessed casing treatment, which can be applied to a three stage axial flow compressor. In order to identify the most important criteria for an optimum design, an attempt is made to present an overall view of the results obtained from previous work and then to explain how an improved design could be achieved.

6.2)- Vane-Recessed Casing Treatments:

A vane-recessed casing treatment as the name implies is a treated casing that replaces a part of the compressor's original solid casing. The new casing is to be installed co-axially upstream of the rotor with the vane-less region partially over the rotor blade tips. In fact the compressor has a stepped casing. In the majority of previous work in this area the inserted stepped casing extends some distance upstream and over the rotor blades. The new compressor casing consists of three regions as follows.

The first region is the part of the annulus passage diameter equal to (D_{Ann}) , which is located upstream of the rotor blade cascade. The second region is the most important section of the treated casing and is placed partly upstream and partly over the rotor blade tip with an inside diameter equal to $[(D_{TC})_{Inner}]$. The third region is normally considered to be the same as the annulus flow passage diameter, or a slightly different diameter, which is located down stream of the rotor blade cascade.

The configuration of a vane-recessed casing treatment with partly curved vanes located partly upstream and to some extent over the rotating blades is shown in [Figure (6-1)]. The recessed portion has a larger inside diameter $[(D_{TC})_{Inner}]$, than either the inlet or outlet portions, which joins them. The latter two usually are identical in inside diameter. It means that in the majority of cases $[(D_{Ann})_{Inlet}]$ is equal to $[(D_{Ann})_{Outlet}]$.

A portion of the rotor blade axial chord at the tip diameter $[(C_R)_{Axial-Tip}]$ is normally exposed to the recess. The recessed portion normally consists of two main parts; the vanes and the inner ring.

6.2.1)- Casing Treatment Vanes:

The purpose of applying the vanes inside the casing treatment is to remove the whirl components of the reversed flows, which pass from the rotor blade tip to the recess. It means that the idea of inserting a set of vanes into the recess is to remove as much of the tangential velocity (V_{Tan}) as possible from the stall induced separated flow, which is centrifuged to the recess. The elimination of the tangential velocity from the flow leaving the rotor blade tip is essential for stall delay.

Ivanov et al. (1984) reported that the vanes cause swirling of the stalled flow which produces an increase in fan pressure. Takata, et al. (1977) has also suggested, that reduction of the whirl velocity is an essential factor in stall delay. The vanes can be completely or partly curved in both radial and axial directions to remove the swirl flow component.

A number of researchers including Azimian (1987) and Hill (1998) have suggested that no stall margin improvement can be achieved using vane-less recessed casing treatments.

6.2.2)- Casing Treatment Inner Ring (Shroud):

The inner ring with the inside diameter $[(D_{IR})_{Inner}]$ is located co-axially with the compressor axis at a distance from the forward wall of the recess. The shroud is fitted to the bottom and inside diameter of the vanes.

There are two different reasons for applying the inner ring in a vane-recessed treated casing. Firstly, to separate the main flow passage from the recessed chamber, and secondly to divide the recessed chamber axial length into two dissimilar lengths namely, entry and exit sections.

6.3)- Preliminary Considerations in the Design of Casing Treatments:

In order to design the desired casing configuration to fit into the conventional casing of the Peregrine compressor, it was necessary to produce a conceptual overview, which would identify vital design features. The following discussion highlights the results obtained from previous research in order to identify the optimum design data and characteristics of a vane-recessed casing treatment, which could be applied to an axial flow multistage compressor.

There are a large number of parameters, which are related either to the compressor or casing treatment. These all have to be considered, because of their positive or negative effects on the design of the vane-recessed type treatment. In the author's experience, it seems that the effectiveness of the different parameters varies according to individual compressor geometry and conditions as well as to particular treatment design parameters.

Moreover, it was found that different geometrical and characteristic related parameters do not function identically. Some of the parameters have more impact on compressor performance than others. The same conclusions were achieved when the casing treatment corresponding parameters were studied in more detail. The few previous works, which have been carried out on the subject of vane-recessed casing treatment, show that this type of casing configuration has significant advantages for compressor performance.

The author believes there is a long way to go in order to achieve an appropriate scheme and optimum design factors. This conclusion arises from the fact that under some conditions the application of the optimal results for one machine has given a worse result for another. One example of this is the effect of the rotor blade tip axial chord exposure to

the recessed chamber (λ_{RTAC}).

Bard (1984) applied a unique ring-shaped stabilising device with zero recess exposed over the rotor blade tip axial chord, into an industrial axial flow fan casing. The stabilising anti-stall device that was applied on a rotor blade cascade with 3.4 [m] tip diameter provided fully stable performance characteristics without appearance of stall line.

It was also reported that the stabilisation ring casing eliminated the unstable performance of the fan due to stalling without affecting the peak efficiency. Note that the stabiliser device that was placed just upstream of and not over the tips of the fan's rotor blades did not include any moveable elements and was fixed outside the main flow stream.

Although the unique ring-shaped stabiliser was located over the rotor blade leading edge, significant stall margin improvement was reported. It is worth noting that the anti-stall device casing arrangements that were used by Bard (1984) and Ivanov et al. (1984) have many common features. The latter was located partly upstream and partly over the tip of the rotor blades.

Azimian (1987) applied a vane-recessed type casing treatment with zero rotor blade axial chord exposure on a single stage and an isolated rotor axial flow fan with a typical aero-engine blade loading. In this case there was no gain in stall margin improvement. This treated casing was fairly similar to the prototype treatment, which had been proposed by Ivanov et al. (1984).

In order to determine the relevant characteristics and dimensions for a possible optimum design configuration, some of the more important parameters of both the machine and the treatment have to be considered in more detail.

6.4)- Effects of Compressor Parameters in Casing Treatment Design:

In order to provide some desirable information on achieving optimum dimensions and design features of an effective vane-recessed treated casing, it is essential to consider the previous researcher's detail studies corresponding to the area of particular casing treatments which normally behave similar to that of current investigation interest.

This part of the study has attempted to afford some beneficial design guidelines through using the available literatures in the area of the present work in addition to analysing the effectiveness of the major compressor's related parameters, which might affect the casing treatment activities from both positive and negative viewpoints.

6.4.1)- Effects of Rotor Blade Tip and Hub Diameters:

One of the major problems, which are not desirable in compressors and normally are associated with the generation of stall cells, is the appearance of flow blockage zones near the tip or hub of the rotor blades. The appearance of rotating stall cells takes place near rotor tips in tip critical machines, whereas this phenomenon occurs near the rotor hub in hub critical compressors. In both cases rotating stall creates unbalanced forces on the rotor blades.

An increase or reduction of the rotor hub diameter will affect the rotor blade hub to tip diameter ratio, therefore in the majority of cases the effects of the hub diameter could be considered an influence on the hub to tip diameter ratio as well. The effects of the rotor blade tip diameter ($D_{R-Tip}=D_{Tip}$) and hub diameter ($D_{R-Hub}=D_{Hub}$) indeed have to be considered in relation to the rotor blade height and hub to tip diameter ratio as well as the tip clearance of the machine.

6.4.2)- Influence of Hub to Tip Diameter Ratios:

The single stage compressor, used by Azimian (1987) was of (0.5) rotor blade hub to tip diameters ratio, $(DR)_{H/T} = (D_{Hub}) / (D_{Tip})$.

An investigation was carried out by Ziabasharhagh (1992), in order to study the effects of vane-recessed treated casings on the fans with different hub to tip diameter ratios. Two single stage fans with hub to tip diameter ratios of (0.7) and (0.9) in addition to that one, which previously used by Azimian applied in this regards. The vane-recessed casing treatments that were used in the (0.7) and (0.9) hub to tip diameter ratio fans had the same general configurations and vane profiles, but their overall dimensions were different. According to this research, a high hub to tip diameter ratio results in more diffusion from the stator blades.

This is because as the hub to tip diameter ratio was reduced, the static pressure difference between the hub and the tip was increased and the stator hub was required to undertake more diffusion. It was also found that due to the occurrence of the stall at the hub of the stator, the rig was not able to produce enough pressure rises. This indeed could suggest that in low hub to tip ratio fans, the application of a hub treatment would reduce the stator root stall to a desired amount.

The results obtained from the examination of a (0.5) hub to tip diameter ratio stage showed that the stator hub was most often the first section to stall before the rotor tip at the lower flow, (Kang, 1996). He concluded that in such circumstance, both the rotor blades tips treated casing in addition to the stator hub treatment should be used in combination in order to gain the maximum effect.

Increasing the value of the hub to tip diameter ratio produces a flow rate reduction in the machine, which would necessitate a treated casing with the lower recessed chamber height. The lower hub to tip diameter ratio compressors, then need to pass larger mass flow rate which leads to the requirement for a higher vane-recessed height.

The work carried out by Hill (1996) was performed on a fan with 500 [mm] tip diameter and a hub to tip diameter ratio of (0.4). Each of the two sets of moving blades was equipped with six blades. The first set consisted of standard airfoil type and the second was reversible double circular arc profile blades.

6.4.3)- Effects of Rotor Blade Tip Clearances:

It was found by Kang (1996) that the applied vane-recessed casing configuration significantly modified the tip clearance flow and could remove the flow reversal into the recessed chamber. The results also indicated a strong radial flow from the rotor blade tip into the recessed chamber treated casing. He also implied that the modification of the tip clearance flow is a key factor to the stall margin improvement mechanism.

The best results indicated that the optimum rotor blade tip clearance (t_c) values were between zero and (2%) of the blade height, exclusive zero amounts. The higher values of the tip clearance had been shown to give lower values of the blade loading as well. The effect of the rotor tip clearance values differed according to the operating conditions. It was found that the more heavily loaded blades or the blades with the lowest flow coefficient were more sensitive to the tip clearance alteration, (Kang, 1996).

Likewise it was reported by Hill (1998) that the modification of tip-clearance flow and inward-radial flow from the recessed chamber could play a considerable role in the effectiveness of treated casing.

6.4.4)- Influence of Rotor Blade Rotational Speed:

The improvement of stall margin in a single stage compressor with the application of a vane-recessed type casing treatment has been investigated by Azimian, (1987). In this study a range of the different running speeds (N_R) were tried on a low speed single stage axial flow compressor, but the obtained results did not show significant improvements.

The design speed used by Ziabasharhagh and Kang for all types of the under test machines were 1500 [rpm], with the ± 5 [rpm] accuracy. The six rotor only fan which applied by Hill (1996) was driven with a fixed nominal 2910 [rpm] running speed.

6.4.5)- Effects of Rotor Blade Height:

Three vane-recessed type casing treatments with different sizes were applied on the (0.5), (0.7) and (0.9) hub to tip diameter ratios fans. The first treated casing was the treatment used by [Azimian], whereas, the geometric categories of the other two were achieved by scaling the related geometry in proportion to the rotor blade height (h_R), (Ziabasharhagh, 1992). Since the results were not satisfactory, the latter suggested that the approach might not be the best means of achieving the desired objective.

6.4.6)- Influence of Rotor Blade Chord:

The obtained results showed that the scaling of treated casing geometry in proportion to the rotor blade height similar to the procedure that had been done by Ziabasharhagh (1992) was not entirely effective. It was suggested that if the scaling of vane-recessed casing treatment geometry relative to a compressor related parameter such as rotor blade height become essential, it seems that the scaling in proportion to the rotor blade chord (C_R) might be another reasonable approach that possibly could provide more useful results. Although the above observation seems reasonable, there is not sufficient evidence yet to confirm such a conclusion.

As it was reported the (0.9) hub to tip diameter ratio blades had suffered appreciable damage at the trailing edge during its previous running, therefore investigators, decided to solve this problem by trimming away about (25%) of the original size from the rotor chord. Reducing the rotor blade chord had led to a lower pressure rise being available from (0.9) hub to tip diameter ratio blades comparing with that of (0.7) hub to tip diameter ratio, (Ziabasharhagh, 1992).

6.4.7)- Effects of Aspect Ratio of Rotor Blades:

The single stage fan with (0.5) hub to tip diameter ratio, which had been used in a complete stage or isolated rotor configurations by three previous investigators¹ in order to study the effects of vane-recessed casing treatments, had an aspect ratio of (2.63).

6.4.8)- Influence of Numbers of Moving and Stationary Blades:

With the exception of the current investigation, the rest of the research on the area of vane-recessed casing treatment applications in either the isolated rotor or the complete stage test rigs had been carried out at Cranfield University by using compressors without inlet guide vanes. The number of the inlet guide vanes in the Peregrine compressor was ($Z_{IGV}=34$). The (0.5) hub to tip diameter ratio fan was equipped with ($Z_R=27$) rotor followed by ($Z_S=28$) stator blades both with (C4) circular arc profiles.

¹ (Azimian, 1987), (Ziabasharhagh, 1992), and (Kang, 1996).

6.4.9)- Effects of Tip Speed of Rotating Blades:

The applied test rig with (0.5) hub to tip diameter ratio, was a single stage machine axial flow fan with a design tip speed of $V_R=39$ [m/sec]. It was stated that the vane-recessed treated casing has not had the function of reducing inlet flow incidence onto the rotor blades. On the contrary it also found that in the majority of cases the whirl reduction in the rotor blade tip region tended to increase the flow incidence and relative inlet velocity to the rotor blade, (Kang, 1996).

The 76 [m/sec] rotor blade tip speed fan with six adjustable setting angle blades was used in order to examine the effects of a vane-recessed type casing treatment on a fan which had six adjustable rotor blades by Hill (1996).

6.4.10)- Influence of Inside Diameter of Conventional Casings:

It was suggested by Ivanov et al. (1984) that the ratio between the inside diameter of the solid casing $[(D_{SC})_{Inner}]$ and the rotor blade tip diameter (D_{Tip}), preferably can be considered in the range of $[(1:1) \sim (1.01:1)]$. All previous related research conducted at Cranfield University on vane-recessed casing treatment has considered the advice suggested by the patent of Ivanov et al., (1984).

The single stage compressor that has been used by Azimian (1987) had an annulus flow passage of 508 [mm]. Ziabasharhagh (1992) carried out the first part of his experimental work on the same test rig. Whereas the second part had been conducted using another compressor again with 508 [mm] annulus passage diameter. Kang (1996) also conducted his research using the same machine but used only the isolated rotor configuration.

6.4.11)- Effects of Stagger Angles and Profiles of Rotor and Stator:

It is desirable to find out how the alteration of the blade angle effects the vane-recessed casing treatment performances. It was noted that the blade pitch or stagger angle (ξ) changes were causing a considerable alteration of the operating load in terms of the work coefficient ($\Delta H/U^2$). The type of bladings which were used in the fan of (0.5) hub to diameter ratio were of fixed stagger blade angles, therefore the aerodynamic loading was fixed as well, (Kang, 1996).

The two different sets of bladings, which were tested by Hill (1998), indicated that the stalling of the test fan was independent of blade profiles, because the performance characteristics followed a similar pattern for both sets. Four stagger blade setting angles (11° , 19° , 27° and 32°) were used. At the lower blade setting angles of (11° and 19°) a continuously pressure rising characteristic was observed. There was no effect on the rotating stall reported in either case.

At the higher blade setting angles (27° and 32°) a rotating stall was detected and also a large drop in the pressure rise characteristic observed as the flow was broken down. With the lower blade setting angle of (19°) the small value of stall margin improvement was obtained, but the results at a blade setting angle lower than (11°) showed a loss in fan performance.

6.4.12)- Influence of Rotor Blade Tip Axial Chord Exposure Ratio:

As it has been implied by the most of the previous researchers who have done their investigations on the effects of the vane-recessed casing treatment performances, the percentage ratio of the rotor blade tip axial chord exposure ratio (λ_{RTAC}) plays a vital role on the effectiveness of the casing treatment.

This ratio can be determined by dividing the value of the rotor axial chord at the tip of the blade which is exposed to the casing treatment recessed chamber $[(C_R)_{AT/TC}]$ over the value of the rotor axial chord at blade tip $[(C_R)_{Axial-Tip}]$. The effect of the rotor blade tip axial chord exposure ratio, which is hereafter called "Rotor Exposure Ratio", has been normally considered by percentage.

Most investigators, who have studied the effects of vane-recessed casing treatment, claimed that a minimum of the rotor exposure ratio is essential in order to achieve an effective stall delay. The majority of them also state that the treated casing by more than (50%) rotor exposure ratio percentage provided considerable performance improvements.

It may be then concluded when the percentage of rotor exposure ratio is more than (50%) the reverse flow in rotor tip region can be reasonably absorbed by the applied casing treatment and is turned back to the main flow with a significant reduction in the flow circumferential velocity component.

Due to the results obtained from the testing of single stage fan it was suggested that the applied casing treatment with approximately (0.6-0.7) rotor exposure ratio would provide the optimum results. It was reported that when this ratio was reduced to less than (0.5), although the casing treatment had still worked, but the amounts of obtained stall margin improvement started to decrease significantly, (Azimian, 1987).

He also reported that the single stage fan build with (67%) rotor blade exposure which had provided (46.3%) stall margin improvement for flow rate was an optimum arrangement. The best improvement of stall margin for the fan with both rotor and stator was (47.7%) for flow rate with a rotor blade axial chord exposure of (43.3%).

By applying the treated casing over an isolated rotor, it was stated again that the ratio of rotor tip exposure was a very significant parameter in the improvement of the stall margin. When the rotor exposure ratio was less than (0.5), the improvements were less than for greater exposures. The obtained results from the application of a vane-recessed treatment on an isolated rotor build showed that with an optimum arrangement of (0.67) rotor blade tip axial chord exposure ratio about (50%) improvement in stall margin for flow rate was achieved, (Azimian, 1987).

It was also reported that the stall margin improvement starts to deteriorate, as the percentage ratio of the rotor exposure was reduced to less than (50%). It was also found that as the amount of this ratio was increased from (70%), although the stall margin improvement was still reasonable, the peak pressure rise for the stage declined.

Three vane-recessed casing treatments with different sizes and number of vanes were tested by Ziabasharhagh, (1992). The results showed that among the various parameters, which influenced the operation of the casing treatment, the ratio of rotor blade tip axial chord exposure was quite significant.

The results obtained from the application of a vane-recessed casing treatment on a (0.5) hub to tip diameter ratio single stage compressor have shown that as the ratio of rotor tip axial chord exposure was decreased the pressure rise remained almost constant. Tests also showed that by further reducing the rotor exposure ratio to the recessed chamber, the pressure rise coefficient was significantly decreased.

At the rotor blade tip exposure of (35.9%), the pressure rise coefficient was dropped to a value approximately near that, which was achieved with the untreated casing. On the other hand, as the ratio of rotor exposure was decreased the stall margin improvement remained almost constant again until (0.359), after which the stall margin dropped sharply.

From the results achieved with the (0.5) hub to tip diameter ratio build it has been concluded that when the ratio of rotor tip axial chord exposure was less than (0.4), the applied casing treatment had not worked as well as for the higher values of exposure ratios. It has also been stated that when the percentage of the ratio of rotor blade tip axial chord exposure were less than (40% and 40%~50%) for (0.7 and 0.9) hub to tip diameter ratios machines, both the stall margin and pressure rise coefficients were reduced sharply. By using the three above-noted treated casings into (0.7) hub to tip diameters ratio fan, it was indicated that, when the rotor blade tip axial chord exposures were decreased to a value less than (30%), a sharp fall in stall margin improvement, had occurred.

It seems however when the rotor exposure ratio is less than a certain value, the amount of the reversed flow which can be absorbed by the casing recessed chamber is reduced.

An experimental study carried out on the (0.5) hub to tip diameter ratio isolated rotor fan showed that, when the percentage of the rotor blade tip axial chord exposure ratio was less than (50%), both the stall margin and pressure rise coefficient were dramatically altered, (Kang, 1996). The exposed value of the rotor blade tip axial chord had been found to be an extremely critical parameter, and one which has a close link with the mechanism of stall margin improvement.

It was also found that the low momentum and reversed flows were centred approximately on the two-thirds of the rotor blade tip axial chord exposure ratio position. Therefore it was noted that very low exposures might lead to a compressor performance that is worse than for the untreated casing. It is likely that the complicated flow near the end-wall of the solid casing can reduce the stability of the axial-flow compressor. It was found that after applying the casing treatment, the pressure field in the end-wall region had been significantly modified due to more than (50%) of the rotor blade tip axial chord being exposed.

It was appeared that by increasing the rotor exposure ratio, it is possible to significantly increase the quantity of the reverse flow to the vane-recessed chamber. It was also reported that the best experimental result was achieved at (0.67) of the rotor blade tip axial chord exposure ratio, (Kang, 1996). These results are not surprising due to the fact that the three previous mentioned researchers have used the same fan as their test rig. It is also interesting to note that the latter result was also gained by applying a slightly different configuration of the vane-recessed casing treatment.

From the above it can be concluded that in order to maximise the effects on the unfavourable flows, at least (50%) of the rotor blade tip axial chord exposure is essential. However this conclusion can not be totally true. Although in some compressor test rigs, the very low values of the rotor tip axial chord exposure led to a worse performance in comparison to that of the untreated casing, there is some evidence, which suggests an alternative conclusion.

Bard (1984) for example used a vane-recessed type casing treatment on an industrial 3.4 [m] diameter rotor blade fan in which the casing treatment was located entirely upstream of the rotating blades. Therefore, the value of the rotor exposure ratio was so small, that practically there was no rotor blade tip axial chord exposure applied at all. It was also reported that the results were quite encouraging.

A number of previous researchers have provided useful information on the impact of various related parameters, particularly the rotor blade axial chord exposure, on compressor performance. It was believed that the very low rotor exposure ratio might lead to a worse performance than that of the untreated casing.

Hill (1998) believed that the rotor blade tip axial chord exposure ratio is independent of the major design parameters such as the casing size, the vane number and their positioning on the recessed chamber, the hub to tip diameter ratio and the blade number.

As a result of this belief he ran his test rig with a casing treatment located entirely upstream of the moving blades, so that there was no rotor tip axial chord exposure applied to the top of the rotor blade. The performance characteristics showed that there was no benefit to having a casing treatment without exposure, due to a small reduction in performance with the loss in efficiency of about (5%) over the design operating range.

The stall margin improvement, which was achieved through the application of vane-recessed type casing treatment on an isolated rotor fan with six moving blades was quite significantly sensitive to the values of the rotor tip axial chord exposure ratio. It was also reported that exposures with more than (55%) of the rotor blade tip axial chord had produced a significant improvement, whereas less than (45%) exposure values gave a marginal improvement.

6.5)- Effects of Treated Casing Parameters in Treatment Design:

All of the above-noted parameters or characteristics are relevant to the machine geometry and behaviour, whereas there are still some other parameters, which are casing treatment relevant. These latter indeed have quite significant influences on the vane-recessed casing treatment performance.

It was found from a study of previous research that the physical size, geometry, different parameter dimensions including the shape of the vanes inside the casing treatment recessed chamber play a vital role in the improvement of compressor performance. It was noted that the application of a treated casing is beneficial only when rotating stall is known to occur, (Hill, 1998).

During the current investigation it was found that the use of the applied vane-recessed casing insertion was helpful during the entire range of the compressor operations. This idea leads to the design of a casing treatment, which would provide more advantages and fewer drawbacks. It seems this can be achieved by detail analysing the casing treatment related parameters and their effectiveness from different view aspects.

Some of the most important parameters, which may significantly influence the casing treatment performances, can be considered as follows.

6.5.1)- Influence of Inside Diameter of Casing Treatments:

It seems that the inside diameter of the vane-recessed treated casing [$(D_{TC})_{Inner}$] has a significant effect on the performance of the casing treatment. This parameter has a direct relationship with the amount of the mass flow rate, which enters into the recessed chamber. It is likely therefore that this will have a considerable effect on machine performance.

Previous researchers which have investigated the field of vane-recessed casing treatment have used a flat inside diameters, whereas the current study has been conducted with a design involving a semi-tubular passage casing which provides different inside diameter values across the axial length. The effects of the inside diameter can also be considered in relation to the height of the vane, which will be discussed later.

6.5.2)- Effects of Shroud Inner and Outer Diameters on Casing Treatments:

In the majority of the studies, which have been done on the application of vane-recessed casing treatment, at Cranfield University, the inner ring (shroud) was made by

cutting 3 [mm] thickness sheet metal to a required strip width and length.

The strip metal was rolled and welded according to the size of the inner ring inside diameter $[(D_{IR})_{Inner}]$. The inner diameter of the vanes was then spot welded to the outer diameter of the inner ring.

6.5.3)- Influence of Axial Length of Casing Treatments:

The axial length of the vane-recessed treated casing region $[(L_{TC})_{Axial}]$ is located partly over and partly upstream of the rotor blade tip. The behaviour of the flow inside the recessed tube of the casing treatment showed that the flow entered the vane-recessed passage from the rearward end near to the outer wall and exited the vane passage from the forward end before mixing with the mainstream flow.

The axial length of the casing treatment can be modified by inserting or taking away some spacer rings into or from the upstream and downstream area of the recessed chamber. With more spacer rings applied to the forward or rearward ends of the recessed chamber, it is possible to further extend the casing treatment over and/or upstream of the rotor blades.

By inserting or taking away the spacer rings with a diameter identical to that of treated casing inside diameter $[(D_{TC})_{Inner}]$ into or from the casing rearward or forward walls, it is possible to modify the treatment recessed tube axial length $[(L_{TC})_{Axial}]$. These activities however could increase or reduce the casing treatment length as well as the axial length of both the recessed tube inlet $[(L_{TC})_{Inlet}]$ and outlet $[(L_{TC})_{Outlet}]$ passages.

In addition the axial position of the casing treatment in respect to the fixed position of the rotor blade's leading edge could also be modified using the same procedure. The axial length of the recessed chamber is another significant parameter, which can significantly influence the effectiveness of a vane-recessed casing treatment performance.

It means that once the optimum value of the recessed axial length has been determined, when the hub to tip diameter ratio reduces, the size of the vane-recessed casing treatment can be appropriately increased. Therefore finding the optimum value of the recessed tube length is essential in order to have an effective casing treatment.

When the size of the casing treatment is not optimal, there is an increased possibility of flow interaction between the recessed tube flow and the mainstream flow. This may lead to blockage inside the vane-recessed treated casing and consequent undesirable efficiency losses. This conclusion supports a similar suggestion made by Ziabasharhagh (1992).

As he noted, when the size and position of the treated casing are optimal, the treatment would provide the minimum effective area for the flow exchange between the recessed tube and the main annular flows. It was also stated by him that by increasing the hub to tip diameters ratio (D_{Hub}/D_{Tip}) , the optimum value of the recessed chamber axial length, inevitably has to be decreased.

Experimental results obtained from the (0.7) hub to tip diameter ratio fan have shown that the applied casing treatment with larger scales had provided the largest stall margin for the smallest penalty in efficiency. However for the (0.9) hub to tip ratio fan, the best results were obtained with the application of medium sized treatment, (Ziabasharhagh, 1992).

6.5.4)- Effects of Inlet Passage Axial Length of Casing Treatments:

The geometry of the casing treatment inlet passage axial length $[(L_{TC})_{Inlet}]$ is closely associated with the rotor blade tip axial chord exposure ratio (λ_{RTAC}) . It seems that in order to remove the stalled flow from the machine, the inlet passage axial length of the recessed tube chamber has to be optimum.

Ziabasharhagh (1992) used the same guidelines which were presented in the axial flow fan patent by Ivanov et al. (1984) for determining appropriate dimensions for two inlet passages axial length treated casing configurations with (0.7 and 0.9) hub to tip diameter ratio test fans.

The effect of the casing treatment axial length inlet passage on the performance of a vane-recessed casing configuration was examined in a test fan with a (0.7) hub to tip diameter ratio. The axial length of the inlet vane-less portion was reduced by (7.7%). Both the stall margin improvement and pressure rise coefficient were higher when the larger value of the axial length inlet passage was used. The application of larger axial length inlet passage however, led to a significant loss in efficiency as well. When the axial length of the treatment inlet passage applied on the (0.9) hub to tip ratio fan was increased by (28%), the results showed that although the stall margin improvement was increased by (1.3%), the efficiency losses also increased.

Kang (1996) suggested that the most efficient inlet passage axial length for the casing treatment should be in a range between (80% to 90%) of the rotor axial chord.

6.5.5)- Influence of Gap Size between Shroud and Rotor Leading Edge:

It is recognised that the distance between the treated casing inner ring and the leading edge of the rotor at the plane of the rotating blade plays an important role in vane-recessed casing treatment performance.

In order for the casing treatment to be effective there must be a gap between the rotor blade leading edge and the end of the shroud. Evidence for this can be seen from the results obtained from the optimised builds of the treatment devised by Ziabasharhagh (1992) for example. In order to achieve an appropriate performance from the casing treatments, the gap between the inner ring and the rotor blade leading edge had to be about 10 [mm] for all three of the different size treatments, which were tested.

The optimum build of the (0.7) hub to tip diameter ratio test fan with an axial gap of 13 [mm] when used with the large scale casing treatment did not show any significant loss of the efficiency. With the medium and small size treatments, the length of the gap was reduced to almost zero. This produced a corresponding loss in performance efficiency.

In the case of the medium size casing treatment using the 8 [mm] gap with the (0.9) hub to tip diameters ratio test fan, no efficiency losses were reported. The application of both the large and small-scale treatments together with two different gap sizes of 4 [mm] and 6 [mm] led to significant corresponding losses in efficiency.

The results obtained from the test rig indicated that the gap size for achieving the best performance characteristics was between 8-12 [mm]. It was also stated that the reduction in the value of the gap between the rotor blade leading edge and the end portion of the inner ring produced more efficiency losses. These results suggest that this parameter is one, which is critical in regards to the casing treatment performance. One approach to reducing the overall axial length $[(L_{TC})_{Axial}]$ of the recessed chamber is to decrease the inlet passage axial length.

The effect of the gap between the shroud and the rotor blade leading edge has been examined on an isolated rotor fan with (0.5) hub to tip diameter ratio by Kang, (1996). The results showed that when the distance of the gap was less than (10%) of the rotor blade axial chord, there was a reduction in the performance of the casing treatment.

It was also reported that less than ten percent of the rotor blade axial chord value was used for the gap size and this produced some stall margin improvement (Hill, 1998). This latter

result conflicts with the conclusions reached by previous researchers about the appropriate value of the axial gap size between the rotor blade leading edge and shroud $[(G_{TC})_{Axial}]$. It is possible that this difference might be related to the different type of machine and the number of rotor blades in the rotating cascades.

6.5.6)- Effects of Inner Ring (Shroud) Axial Length of Casing Treatments:

A small improvement in stall margin can be obtained without the use of a shroud, but most previous studies have found the inner ring to be an important component of a vane-recessed casing treatment. The inner ring is not only important as a bridge between the inlet and outlet passages, but also has a significant affect on the performance of the vane-recessed casing treatment.

The axial length of the shroud $[(L_{TC})_{Shroud}]$ is dependent upon the area which is required by the recessed casing for the removal of the stalled flow from the rotor blade tips and its re-injection flow with enough static pressure back into the main stream, having taken out the swirl.

A vane-recessed treated casing without vanes and with just an inner ring was applied to an isolated rotating blade fan with (67%) percentage ratio of the rotor blade tip axial chord exposure to the recessed chamber, by Azimian (1987). The results showed that although the stall margin improvement was not significant, it was still better than the build without both the vanes and the shroud. These results showed that a small amount of flow is trapped inside the recessed casing and then re-circulated, without any improvement gain.

One of the major difference in design between the unique stabilising ring that had been used by Bard (1984) and the vane-recessed type casing treatment which are very similar in configuration and applied by other investigators, appears to be the lack of an inner ring in the former design.

Fan with six rotor blade was tested with a variety of blade tip axial chord exposure ratios (λ_{RTAC}) from zero to (0.57) by Hill (1998). The resulting pressure characteristics revealed that that application of the shroud was not influential, since there was no difference in performance between the treatment with and without an inner ring. Both configurations gave the same improvement in stall margin, and the removal of the shroud did cause a drop in efficiency. It was also noted that with a zero value for the rotor blade tip axial chord exposure, there was no improvement in stall margin using a vane-recessed casing treatment without an inner ring.

In view of this efficiency drop, it can be suggested that the lack of an inner ring might reduce the blade loading at the tip of the rotor blades causing a reduction of the design flow rate. Without an inner ring it is possible that the stall flow and re-injection flow might mix upstream of the rotor blades and thereby cause a reduction in efficiency.

6.5.7)- Influence of Shroud Thickness of Casing Treatments:

With the exception of Hill (1998) who does not specify any thickness for the inner ring all other studies in the area of vane-recessed casing treatment at Cranfield University have utilised a 3 [mm] thick shroud.

There is very little information available on the optimum thickness value of the inner ring $[(t_{TC})_{Shroud}]$ as part of a vane-recessed casing treatment. The shrouds of the casing treatments initially designed for use with the Peregrine compressor was of 2 [mm] thickness. However owing to manufacturing requirements, the inner ring thickness of the modified treatment had to be altered to 3 [mm].

6.5.8)- Effects of Outlet Passage Axial Length of Casing Treatments:

The outlet passage area of the vane-recessed treatment $[(L_{TC})_{\text{Outlet}}]$ seems to be very important in order to re-inject the recessed chamber flow in the desired direction of the main flow. Before the flow is re-injected the swirl has to be removed from the injected flow by means of vanes inside the recessed chamber.

The nature of the interchange between the recessed tube flow and the mainstream flow imposes a geometrical limitation on the vane-recessed outlet passage axial length value. Prior to the design phase, it was thought that probably there could be a physical and characteristic relationships between the amount of the mass flow rate and the requirement of the outlet passage axial length.

The patent of Ivanov et al. (1984) provided the basis for the design features of the outlet passage axial length for the all of the vane-recessed casing treatment investigations carried out at Cranfield University.

6.5.9)- Influence of the Recessed Chamber Height of Casing Treatments:

According to Azimian (1987) the reduction of the recessed chamber height (h_{TC}) failed to improve on the performance of the initial height. This might be due to the fact that reducing the height caused a flow rate alteration inside the chamber and an exchange of flow criteria between the main and the recessed portion flows. The result of this might also give rise to early onset of rotating stall.

When the rig was tested under the isolated rotor configuration, it was found that a height reduction of up to (50%) did produce some improvement in the vane-recessed casing treatment performance. Once the height was reduced to more than half of the design height, the performance of the treatment decreased significantly. It was also reported that as the height of the recessed chamber was reduced, the internal flow, which was re-circulating inside the recessed chamber, tended to disappear.

It was suggested that some of the reversed flow inside the recessed tube area began to re-circulate and as a result created turbulence which produced further internal flow blockage inside the casing treatment. The conclusion reached was that the preliminary design height of the recessed chamber was too high.

This indicates that a casing treatment with a lower recessed chamber height limits the recessed area, which in turn probably results in a lower mass flow through the treatment. Lower mass flow is normally associated with the higher hub to tip diameter ratio machines. It can be suggested that when the hub to tip diameter ratio $[(DR)_{\text{Hub/Tip}}]$ of the compressor is low, vane-recessed casing treatments with smaller recessed chamber heights might provide better stall margin improvements.

In order to achieve an optimal treatment operation, a minimum flow rate exchange through the vane-recessed treated casing is essential. This conclusion received further confirmation, when the effects of the hub to tip diameter ratios and the recessed tube height in relation to stall margin improvement was considered by Hill (1998).

6.5.10)- Effects of Vane Shape Inside Casing Treatments:

The experimental investigation provided evidence to suggest that when the mass flow rate was reduced to particular value, rotating stall would be initiated, and this would in turn produce a reverse flow from the rotor blade cascade. In this situation the reverse flow would cause a strong pre-rotation in the inlet flow to the rotor and therefore decrease the incidence angle of the flow to the corresponding blade.

In order to remove the swirl from the stalled flow in the casing treatment, it is possible to use a set of fully or partially curved vanes inside the recessed chamber. The vanes are positioned in the annular recessed area so that the stalled flow in the recessed chamber can be bypassed.

To increase the compressor pressure rise performance during the occurrence of rotating stall, the stalled flows have to be swirled in a direction counter to the direction of the rotor blade rotation. This will ensure that the retrieved flow leaves the recessed chamber outlet passage in the appropriate direction. A similar conclusion was reached by Ivanov et al. (1984).

With the application of the vane-recessed casing treatment the turning vane passages can eliminate the swirl components of the reverse flow, which re-circulates inside the recessed chamber. A set of partly curved and partly straight vanes with (60°) turning angle from the axial direction has been used by Azimian (1987). Those vanes were partly cambered by (30°) from the circumferential direction.

By using the wool tuft flow visualisation technique, it was confirmed that the application of a set of vanes positioned in the recessed chamber of the casing treatment proved highly effective in removing the swirl from the circulating flow. Using this technique, it was found that the sharp interaction between the flow leaving the recessed chamber and the main flow was not desirable.

As it was reported, when the test compressor stalled, the reverse flow was detected at an angle between $(100^\circ$ to $120^\circ)$ and mostly (110°) from the axial direction. From this it was concluded that it was necessary to turn the flow which leaves the recessed chamber from the radial direction to a near axial direction in order to reduce the sharp (90°) interaction angle between the two flows.

As it was stated by Ziabasharhagh (1992), at some positions inside the vane-recessed casing treatment, there was evidence of the flow separation on the suction surface of the applied vanes. This however indicates the importance of aerodynamic modification of the vane shape.

Another possible way of overcoming this problem is through the application of radial shaped vanes, as used in the present study. This suggestion is supported by the work of Ivanov et al. (1984).

Kang (1996) noted that the test results clearly revealed that the outflow having zero axial velocity left the vane-recessed casing treatment in a radial, rather than an axial, direction. This can cause a detrimental impact on performance of both the casing treatment and the compressor, owing to the undesirable interaction between the outflow and the mainstream. In terms of overall performance it was concluded that this flow interaction is more significant than increasing axial flow velocity in order to energise the low momentum flow in the tip region of the rotor blades.

This view contradicts some of the findings reported by previous experimental studies on vane-recessed treatments, including Azimian (1987).

Three different vane shape treatments were tested on an isolated rotor fan by Kang, (1996). The results showed only a marginal effect on overall fan performance. However since there was a small benefit in having cambered, rather than straight vanes over both the rotor blade and the full length of the recess, he concluded that it is always advisable to select a simple vane structure in order to avoid significant performance losses.

Hill (1998) also applied a vane set which was cambered (60°) from the axial directions.

6.5.11)- Influence of Recessed Vane Axial Length Inside Casing Treatments:

The axial length of the vanes inside the recessed chamber $[(L_{\text{Vane}})_{\text{Axial}}]$ has to be considered in relationship to the entire axial length of the casing treatment. This parameter is more relevant to two other casing treatment related parameters namely outlet passage axial length and inner ring axial length, respectively. This parameter will be discussed in more detail in the following chapter.

6.5.12)- Effects of the Vane Height Inside Casing Treatments:

The height of the recessed guide vane (h_{Vane}) can be considered in terms of the treated casing height. When the tubular-passage vane-recessed casing treatment is applied, the height of the vane varies according to the axial length of the treatment. When a straight inside casing treatment is used, the height of the vane is constant.

According to Azimian (1987), by reducing the vane height, including the recessed chamber height, by more than (50%) of the design value on an isolated rotor fan, it was found that the stall margin which had been improved under the full height configuration then decreased quite significantly.

It seems that in order to achieve an appropriate vane-recessed treated casing performance a minimum vane height is essential. The present author shares the view expressed by Azimian (1987) that the reduction in stall margin improvement, which occurred with the reduction in vane height, is possibly linked to the appearance of an internal flow blockage inside the vane-recessed passage.

Variation of the vane height in relation to the hub to tip diameter ratio was considered in proportion to the rotor blade height by Ziabasharhagh (1992). He first thought that it would be reasonable to relate the height of the vanes to the height of the rotor. Therefore, the heights of both sets of vanes for (0.7) and (0.9) hub to tip diameter ratio configurations were scaled to (60%) and (20%) of the treated casing which had been originally designed by Azimian (1987).

The results however showed that the proportional scaling to the rotor blade height does not constitute to a useful design criterion. He suggested instead that the scaling of the vane height in proportion to rotor blade chord might lead to more beneficial results.

6.5.13)- Influence of Vane Thickness:

In spite of searching the available literature, the author could not find any information on either the optimum or indeed appropriate values for vane thickness (t_{Vane}). In the majority of previous research conducted at Cranfield University, the sheet metal was cut initially as unfolded vanes. These were then shaped into the desired geometry using a male and female moulding system.

Sheet metal with 1.5 [mm] thickness was used to manufacture the set of vanes used by Azimian (1987), Kang (1996) and Hill (1998) in their experimental work. All (40) vanes, which had been applied inside the recessed chamber of the (0.7) hub to tip ratio casing treatment used by Ziabasharhagh (1992) were shaped from sheet metal of 1.5 [mm] thickness. However sheet metal of only one [mm] thickness was used in order to manufacture the (90) vanes for the (0.9) hub to tip diameter ratio fan treated casing.

6.5.14)- Effects of the Number of Vanes Inside Casing Treatments:

A set of cambered or straight vanes is equally spaced in a radial arrangement inside the recessed chamber of the casing treatment. The vanes are normally located from the

forward end of the recessed chamber to somewhere near the end part of the inner ring. It appears that the application of the vanes into the recessed chamber can reduce the circumferential component of the flow inside the casing treatment.

Most of the previous research in the area of vane-recessed treatment has indicated that the optimum number of recessed vanes (Z_{Vane}) is a vital component of the casing treatment. It also seems that without vanes inside the recessed chamber, it is not possible to provide significant improvements in stall margin.

There is evidence to suggest that a recessed type casing treatment without any vanes not only fails to produce an improvement in the stall margin of the tested machines but will also have adverse consequences on performance as well, (Azimian 1987). These unfavourable results may be due to the fact that reversed flow starts to rotate in a circumferential direction inside the greater space left in the recessed chamber by the absence of the vanes.

A recessed casing treatment without both vanes and an inner ring was applied to an isolated rotor fan. The optimum results were achieved with a rotor blade tip axial chord exposure ratio of (0.67). The results indicated that the peak pressure rise for this build was less, and the stall margin performance was worse, than the corresponding values for the untreated casing, (Azimian, 1987).

In this case, because of the lack of information regarding the optimum number of vanes, a space to chord ratio of (0.5) at the mid-height of the vane was assumed. The obtained result gave (41) as the number of vanes. However in order to simplify the manufacturing process so that a halving or doubling of the number of the vanes could be provided without too much extra work, there was a strong reason to choose a set of vanes with an even number.

A set of (48) vanes was therefore inserted into the vane-recessed casing treatment and this was applied in both the isolated rotor and single stage fan configurations. The effect of vane numbers was examined by halving the vanes of the original designed casing treatment on the test single stage fan with optimum value of (67%) rotor tip axial chord exposure. The results indicated that the (24) vanes gave a slightly better performance than that of the (48) vanes set.

The modifications which were carried out on the effectiveness of the van-recessed treated casing by Azimian (1987) not only involved halving the number of vanes, but also included the addition of a flow deflector, the introduction of an aerofoil section and the insertion of a recessed height reducer.

A comparison of the results is somewhat inconclusive, because the alterations to some of the geometrical parameters makes it difficult to attribute which of these are responsible for changes in the flow characteristics. It is therefore not possible to conclude that the results, which were achieved, were due simply to changing the number of vanes.

The first of the three vane-recessed type casing treatment configurations which were used by Kang (1996), comprised of (24) vanes with an overall pitch to chord ratio of approximately one. It was suggested that where the vane pitch was too narrow, the surface friction reduced the efficiency, but where it was too large, flow separation occurred and this caused high losses in performance.

According to Hill (1998) variations in the number and shape of the vanes, which determine the geometry and shape of the vane passage in the recessed chamber have only a marginal effect on the overall performance of the compressor. He also notes that the modification

produced a slight improvement in stall margin in terms of the flow coefficient, but made little or no difference to the pressure rise coefficient or efficiency. This research suggested that at high flow coefficients the performance of the casing treatment was not particularly sensitive to the number of the vanes.

It is the view of the present author that the number and shape of the vanes, which in turn determines the configuration of the vane passage, are all critical factors in the performance of the casing treatment. This is especially true in heavy rotating stall conditions when very large stall cells occur in the compressor. Without a sufficient number of vanes the casing treatment will be unable to break-up the stall cells in order to retrieve the flow which has to be re-injected back into the mainstream.

Unless the pitch between two adjacent vanes is optimised, the swirl cannot efficiently be removed from the stalled flow. The results obtained from applying casing treatments without vanes have shown that in order to achieve the desired stall margin improvement it is essential to install a set of vanes inside the recessed chamber.

The procedure for calculating the optimum number of vanes, based on the space to chord ratio of the vanes, at the mid height diameter inside the recessed chamber of the casing treatment, will be discussed in more detail in chapter seven. This provides a full description of the geometry and design features of the casing treatment.

6.5.15)- Influence of Applying Flow Deflectors in Casing Treatments:

The primary purpose of applying a vane-recessed casing treatment is to remove the tangential component of the injected flow and reintroduce it in an appropriate direction back into the mainstream. In order to achieve this aim the designed vanes and other components of the casing treatment must be able to prevent any flow separation in the recessed chamber.

The results of the wool tuft examination carried out by Azimian (1987) inside the recessed chamber passage showed some flow separation on the suction surface of the vanes, which was an indication of flow overturning. It was suggested that in order to improve the flow separation on the suction surface of the vanes, some modification to the vane-recessed passage of the casing treatment was required.

One of these modifications involved the application of a flow deflector inside the casing treatment. This was examined on an isolated rotor fan in association with spacer rings that provided an optimum value of (67%) rotor blade tip axial chord exposure. This part of the experiment was intended to change the re-injected flow from a radial to an axial direction. This however failed to produce any positive results, because it seems that the deflected streamlines near the tip of the rotor blades were too large.

In order to achieve an optimum result the re-injected flow must leave the casing treatment as far as possible in an axial direction. It is possible that in this experiment the flow deflector might have caused a flow separation inside the recessed chamber as well as upstream of the rotor blades. This might have led to the onset of a local rotating stall, which would have reduced the effectiveness of the treated casing.

On the other hand Azimian (1987) suggested that the near (90°) turning flow angle from radial to the near axial direction was too much and indeed he believed that this was the reason that the application of flow deflector had produced an adverse effect. The occurrence of the flow separation at lower flow conditions following the application of the flow deflector might also have been due to the fact that the flow suffered too much turning by the vanes and deflector.

The view of the present author is that the recovered flow direction has to be changed from the radial to axial. In order to reduce the flow separation on the suction surface of the vanes so that the flow behaviour inside the recessed chamber is improved, particularly when straight or partly curved vanes are used, some geometrical modification similar to the application of the flow deflector is unavoidable. An alternative solution to a flow deflector is to use fully curved vanes.

It appears that one of the undesirable effects of using a flow deflector was that it restricted the available outlet flow area as well as the re-injected flow. It is worth noting that as the direction of the re-injected flow angle moves to the axial direction, the recessed flow might enter the mainstream flow over the shroud at the tip region of the rotor blades.

It can be concluded that the application of the flow deflector at the downstream edge of the recessed chamber, which had shown more flow re-circulation inside the recessed treated casing was not an appropriate approach for tackling the problem of reduced efficiency. An increase of the recessed flow re-circulation might also cause a decrease in the stall margin improvement.

Most of the experimental work in the area of vane-recessed casing treatment to date suggest that the best results were achieved from those treatments which introduced the recessed flows back into the mainstream in an axial direction, but did not use a flow deflector for this purpose.

6.5.16)- Effects of Applying Aerofoils in Casing Treatments:

A previous examination of a vane-recessed treatment on both an isolated rotor and a single stage compressor has shown that a minimum flow exchange between the recessed and the mainstream flows is required in order to produce significant stall margin improvement effects, (Azimian, 1987). He also thought that by improving the vane-recessed passage design, it would be possible to inject the reverse flow back into the mainstream flow in order to achieve the desired performance enhancement

Using an aerofoil section in order to reduce the area from the rearward end to the forward end of the vane-recessed treatment was fairly beneficial. The application of an aerofoil section successfully reduced the internal flow blockage and resulted in an improvement in the casing treatment performance.

The use of an aerofoil not only can reduce the area of the recessed chamber but also is able to provide a fairly aerodynamic passage for the flow as well. The addition of the aerofoil shape also performs a barrier that prevents the reversed flow within the recessed chamber to return back to the mainstream from the treatment inlet passage. Although this may not have influenced Azimian's decision to apply the aerofoil section, it is the view of the present author that, at lower mass flow rates, the reverse flow tends to turn back and exit via the inlet passage of the casing treatment.

6.6)- Effects of Casing Treatments on Compressor Performances:

The removal of inlet whirl velocity is an essential requirement of vane-recessed casing treatments. This is because when rotating stall occurs in the solid casing, a high whirl component of velocity would be present upstream of the rotor.

With the application of a vane-recessed casing treatment, the re-circulating flow will centrifuge into the treated recessed chamber, where the vanes can remove the tangential component before the flow is reintroduced into the main annulus. These activities might increase the work input and enable an improved pressure rise without stall development

until the lower flow coefficients.

It is known that some machine characteristics such as the following can be affected by using the different type of casing treatments including the vane-recessed ones.

6.6.1)- Alterations in Flow Characteristics Resulting from Application of Casing Treatments:

It seems that the flow exchange between the compressor mainstream and the recessed chamber flows is of great importance. Azimian (1987) reported that quantitative flow measurements by means of hot-wire were not successful, due to the intensive level of flow fluctuations inside the recessed tube chamber.

The peak pressure rise in addition to the stalling points were the same for the both, a single stage and isolated rotor builds, whereas in the case of the single stage, there was an appearance of pressure drop between the peak pressure rise and the stall point. It was then concluded that the pressure drop appeared to be due to the stalling of the stator hub.

Using wool tufts for flow investigation inside the vane-recessed treatment showed that the flow entered the recessed passage from the rearward end near to the outer wall of the recessed chamber and left it from the forward end. It was then mixed with the mainstream flow, (Ziabasharhagh, 1992). The results obtained from the (0.5 and 0.9) hub to tip diameter ratio builds indicated that the flow was more uniform than the same configuration of the (0.7) hub to tip ratio fan.

Kang (1996) showed that a vane-recessed treatment had neither the function of increasing the total pressure upstream of the tip region, or the function of reducing the inlet flow incidence onto the rotor blade. It was also observed that the casing treatment had a strong influence on the end wall flow field. The effect of the treatment on the rotor blade inlet flow was much stronger than on the rotor blade outlet flow.

The area of the flow recirculation appeared to be more confined by increasing the flow rate within the vane-recessed passage. By using (CFD) modelling, it has also been reported that the low momentum energy and the reversal flows were centred on (2/3) of the rotor blade tip axial chord exposure ratio position.

The treatment was shown to reduce the whirl velocity component and also removed the flow blockage, which was previously found in the tip region of the solid casing. The magnitude of the flow vectors in the recessed chamber was increased while the flow rate decreased.

It was also reported that the effectiveness of vane-recessed casing treatment when operating under stall conditions can depend on several factors of which the most important is the elimination of the tangential or whirl velocity component of the flow in the casing treatment recessed chamber, (Kang, 1996).

Hill (1998) demonstrated that a vane-recessed casing treatment led to a continually rising performance characteristic with no evidence of rotating stall. It was also found that with a rotor blade tip axial chord exposure ratio of more than half, in some cases, a considerable drop in performance occurred.

As the maximum efficiency of the compressor usually occurs just before the stalling point, in order to avoid stall, the operating range of the compressor is normally selected such that the maximum efficiency is usually avoided. This undesirable phenomenon, which can largely be negated by using a vane-recessed casing treatment, could have a noticeable effect on the compressor's operating cost.

6.6.2)- Improvements in Stall Margins Resulting from Application of Casing Treatments:

Stall margin improvement can be used as a parameter for comparing the performance changes between the solid wall and the treated configuration casings. It can also be used to assess the performance of the compressor with different casing treatment configurations. Previous research has demonstrated that the application of a vane-recessed type casing treatment can provide significant improvements in the stall margin of the tip critical machines.

Azimian (1987) for example applied a vane-recessed casing treatment to a complete stage and isolated rotor builds. In this instance although the occurrence of the stall was delayed to a much lower flow rate, the treated builds did not show any significant change in the pattern of the stall. The best result showed about (47.7%) and (50%) improvement in stall margins in term of flow rate obtained from the complete stage and isolated rotor builds.

Those improvements however occurred within the application of (43.3%) and (67%) of rotor blade tip axial chord exposures respectively. The results obtained from the single stage and also the isolated rotor builds showed that the test fan was a tip critical machine and that stall margin improvement was achieved by reversing flow exchange inside the recessed chamber.

Ziabasharhagh (1992) carried out a comparison between the obtained pressure rise characteristics of the conventional solid casing and the casing treatment for a (0.5) hub to tip diameter ratio fan. The results indicated that the stall margin improvement for peak pressure rise was just (8%). However a (59%), improvement in stall margin in terms of flow rate was achieved with the same configuration.

It was also reported that in the case of casing treatments applied to both (0.7 and 0.9) hub to tip diameter ratios fans, the stall margin improvements for either flow range or peak pressure rise were the same. The general trend of the stall margin improvement and the pressure rise of (0.9) hub to tip diameter ratio build were found to be the same as those with (0.5) and (0.7) hub to tip diameter ratios.

Large increases in stall margin improvement were obtained with the application of a uniform, rather than distorted, inlet flow to the vane-recessed casing treatment. Moreover, it was noted that the stall margin improvements were substantially greater than those that had been reported in the literature for conventional casing treatments.

Kang (1996) described the application of three different vane geometrical designs on an isolated rotor fan of (0.5) hub to tip diameter ratio. The best stall margin improvements of (51.2%), (58.7%) and (67%) were achieved in terms of flow rate. The corresponding rotor blade tip axial chord exposure for the first and third configurations was (66%), whereas the second one was about (62.5%). It was reported that the stall margin improvements resulting from using the casing treatment had been achieved without reducing the flow incidence or by increasing the total pressure in the upstream tip regions.

The casing treatment, which had been used by Hill (1998), improved the stall margin of a tested fan by about (42%). As with other Cranfield University research on vane-recessed casing treatment, he concluded that the results were very dependent on the amount of rotor tip axial chord exposure ratio to the recessed chamber. At no or very low rotor exposure ratios there was no gain in stall margin.

The optimum result was obtained using about (60%) rotor blade tip axial chord exposure. The stall margin improvements were less than those that had been reported by previous

investigators despite using very similar casing treatment configurations. It was also concluded that the casing treatment performance was related to the increase in flow rate as the fan was throttled.

6.6.3)- Alterations in Efficiencies Resulting from Application of Casing Treatments:

By comparing the efficiency curves for the vane-recessed configuration and the solid wall casings, Azimian (1987) showed that for identical running points, the efficiencies were almost identical.

Ziabasharhagh (1992) reported a set of experimental results that indicated a small loss in peak efficiency with the application of several vane-recessed type casing treatments. He also found that the pressure rise improvement was insignificant in relation to some of the treatment configurations. The general trend of efficiencies for the (0.7) hub to tip diameter ratio treated casing were found to be the same as for that of (0.5) hub to tip diameters ratio, except that the level of efficiency penalty was higher for the former hub to tip ratio.

The optimum build of the latter treated configuration showed only a (2%) loss in peak efficiency. The general trend of the (0.9) hub to tip diameter ratio treated casing efficiency was the same as for the (0.5 and 0.7) ones, except that the level of efficiency penalty was higher. The optimum build of (0.9) hub to tip diameter ratio casing treatment showed a loss of just (1%) in the peak efficiency.

The results showed that the minimum loss in efficiency had occurred in the range of (0.5 to 0.7) rotor blade tip axial chord exposure ratio, (Ziabasharhagh, 1992). It was also noted that the applied vane-recessed casing did not significantly alter the peak efficiency or the efficiency in the extended unstalled range curves.

Kang (1996) suggested that the peak efficiency penalty depended to some extent on the proportion of the of the rotor blades tip axial chord exposure ratio to the recessed chamber. He also noted that the greater exposure ratio generally leads to the greater value of efficiency penalty.

Hill (1998) reported that when the axi-symmetric type stall occurred the treated casing did not significantly alter the efficiency curve. It did result however in an increase in the work input in the extended stall free range. He added that, because the best results were achieved at about (0.60) rotor exposure ratio. This exposure ratio also produced the largest stall margin improvement.

6.6.4)- Variations in Pressure Rise Coefficients Resulting from Application of Casing Treatments:

Azimian (1987) presented an examination of the pressure rise characteristics, and this indicates a positive slope between peak pressure rise and the inception of rotating stall.

A similar conclusion was suggested by Ziabasharhagh, (1992). He illustrated that there was no evidence of rotating stall between the peak pressure rise and stalling point. It was found that the peak pressure rise coefficient for the vane-recessed casing treatment build with distorted inlet flow was higher than that for the undistorted solid casing. The peak pressure rise coefficient was also much higher than that achieved for the solid wall build with the distorted inlet flow.

The results obtained from both of the solid casings with the (0.7 and 0.9) hub to tip diameter ratio fan builds showed that there were sharp falls in the pressure rise when the fan stalled. These decreases in performance were due to the occurrence of abrupt stall.

Kang (1996) reported that the efficiency in the extended non-stalled flow range for the treated builds was not significantly different, but that the pressure rise for the casing treatment was substantially higher which was an indication of more work input. The applied vane-recessed treatment hardly affected the flow field of the test fan until the fan was throttled to a condition near the stall point where pressure rise was normally higher.

Hill (1998) found that the application of a vane-recessed casing treatment at the higher values of blade setting angles (27° and 32°) led to a continuously increasing pressure rise characteristic with no pattern of rotating stall detection,

6.6.5)- Changes in Work Coefficients Resulting from Application of Casing Treatments:

Ziabasharhagh (1992) suggested that in order to maintain higher pressure rises through the casing treatment, more work input is required.

According to Kang (1996), it appears that with the solid casing, the work input was considerably reduced as the flow was decreased into the stall region. The application of the casing treatment however caused a progressive increase in work input and pressure rise in the extended unstalled range. Although the casing treatment prevented the onset of rotating stall, the work coefficient increased as the flow rate decreased. Therefore more work input is required in order to maintain the same pressure rise through the treatment.

The casing treatment which was used by Hill (1998) also did not significantly alter the efficiency curve but did increase the work input in the extended stall free flow range.

6.7)- Significant Achievements Resulting from Application of Casing Treatments:

There are some relevant points and parameters, which are critical in order to achieve an optimum vane-recessed treated casing design. These are as follows.

A recessed casing treatment with (48) vanes inside the recessed chamber was applied to a fan in both single stage and isolated rotor configurations. Azimian (1987) noted that with several significant exceptions, there were some similarities between the results obtained from the isolated rotor and those achieved with the single stage test fan.

With the application of the casing treatment to the single stage fan, an unforeseen problem was encountered in the form of a large flow blockage at the inner diameter of the stator blades. Using the vane-recessed treatment the initiation of the rotating stall was postponed, but the delay to the onset of stall at the rotor tip allowed the stator hub stall to develop to a greater extent.

In an attempt to improve the rotor pressure rise characteristics against mass flow rate rather than that of the stage, it was decided to run the test fan with the isolated rotor only and examine the performance of the treated casing. This was prompted by the tendency of the stator to stall. In spite of some similarities there were also significant differences between the results obtained from the single stage and isolated rotor fan builds. In the case of the isolated rotor tests, the peak pressure rise and the stalling point were similar.

However when the stator blades were introduced, a pressure drop appeared due to the inception of a blade hub stall. It seems that the designed treated casing was only effective in improving the rotor blade performance, because the stator hub stall caused a continuous fall in the stator pressure rise.

The same experience was also reported by Ziabasharhagh, (1992). As he noted a dip in the

pressure rise characteristic which had been seen between the peak pressure rise and the stalling point of the (0.5) hub to tip ratio fan. Typically the results showed that the (0.5) hub to tip diameter ratio fan was stalled abruptly at a flow coefficient after which the rotating stall occurred.

Moreover the wool tuft examination and pressure probe measuring results indicated that the regions of blockage were in front of the rotor blades. However no disappointing evidence of hub stall in both the (0.7 and 0.9) hub to tip diameters ratio builds was reported. It seems that the latter builds had provided a continuous pressure rise until the rotating stall inception took place.

It was noted that when the size of the casing treatment and also the axial position of the rotor blade exposed to the recessed chamber were optimised, the deficit of the maximum efficiency could be negligible, although such a configuration may not provide maximum stall margin improvement. A similar problem with hub stall was reported by both of the above-noted investigators². It is not perhaps surprising, since both carried out their research using the same test rig fan with a (0.5) hub to tip diameter ratio.

Kang (1996) showed that the vane-recessed casing treatment caused a large work input in order to increase the pressure rise in the extended stall range. It was also noted that the blade loading of the machine might have a significant effect on the performance of the vane-recessed treated casing.

It was also noted that the inward flow from the recessed chamber, which was injected at a variety of velocities back into the mainstream, played a very active role in delaying the onset of stall. It proved difficult to gain any further performance improvement from the isolated rotor test fan through the application of the vane-recessed casing treatment, because the hub stall or flow separation imposed a new restriction on the machine's operating range.

Since the rotor blade hub stall set a new limitation on the stall margin, any further optimisation of the casing treatment design could not significantly change the flow characteristics of the test fan. The researcher concluded this was the reason why the application of three different casing treatment configurations yielded very similar results.

The second of the applied casing configurations had radially flat and radially cambered vanes part of, which were located over the rotor blades. Two modification devices were introduced. These consisted of an airfoil section and a new vane, which covered the whole recessed chamber width. The third treated casing configuration had (24) slightly curved vanes and an inner ring of substantial cross section.

The first vane-recessed treated casing configuration produced a worthwhile stall margin improvement of (51%) and a pressure rise of (115%) with an efficiency penalty of about a half percent. In comparison, the second configuration provided a (59%) stall margin improvement, but without any advantage in pressure rise (114%) or efficiency (0.5%). The best results were obtained from the application of the third vane-recessed type casing treatment configuration. This produced up to (67%) stall margin improvement, an increase of more than twice in pressure rise (120%), and a slight peak efficiency increment.

The applied test fan proved to be tip critical with the original solid wall casing. However with the application of the casing treatments it was found to stall at the hub. Two previous researchers had already used the test rig fan employed by Kang (1996). He did however

² (Azimian, 1987) and (Ziabasharhagh, 1992).

include a new computerised data acquisition system.

Hill (1998) measured turbulence intensity using a single hot wire and this showed that the tested fan at lower blade setting angles of (11° and 19°) exhibited an axi-symmetric stall flow pattern. He also reported that the high values of the whirl velocity associated with the solid casing at the stalling flow rate, were minimised with the application of the treated casing. The application of the casing treatment led to a continually rising performance characteristic without any evidence of rotating stall. At lower mass flow rates, when stall did occur, it was of the axi-symmetric type. It was also concluded that the application of a vane-recessed casing treatment is beneficial only when rotating stall is known to occur.

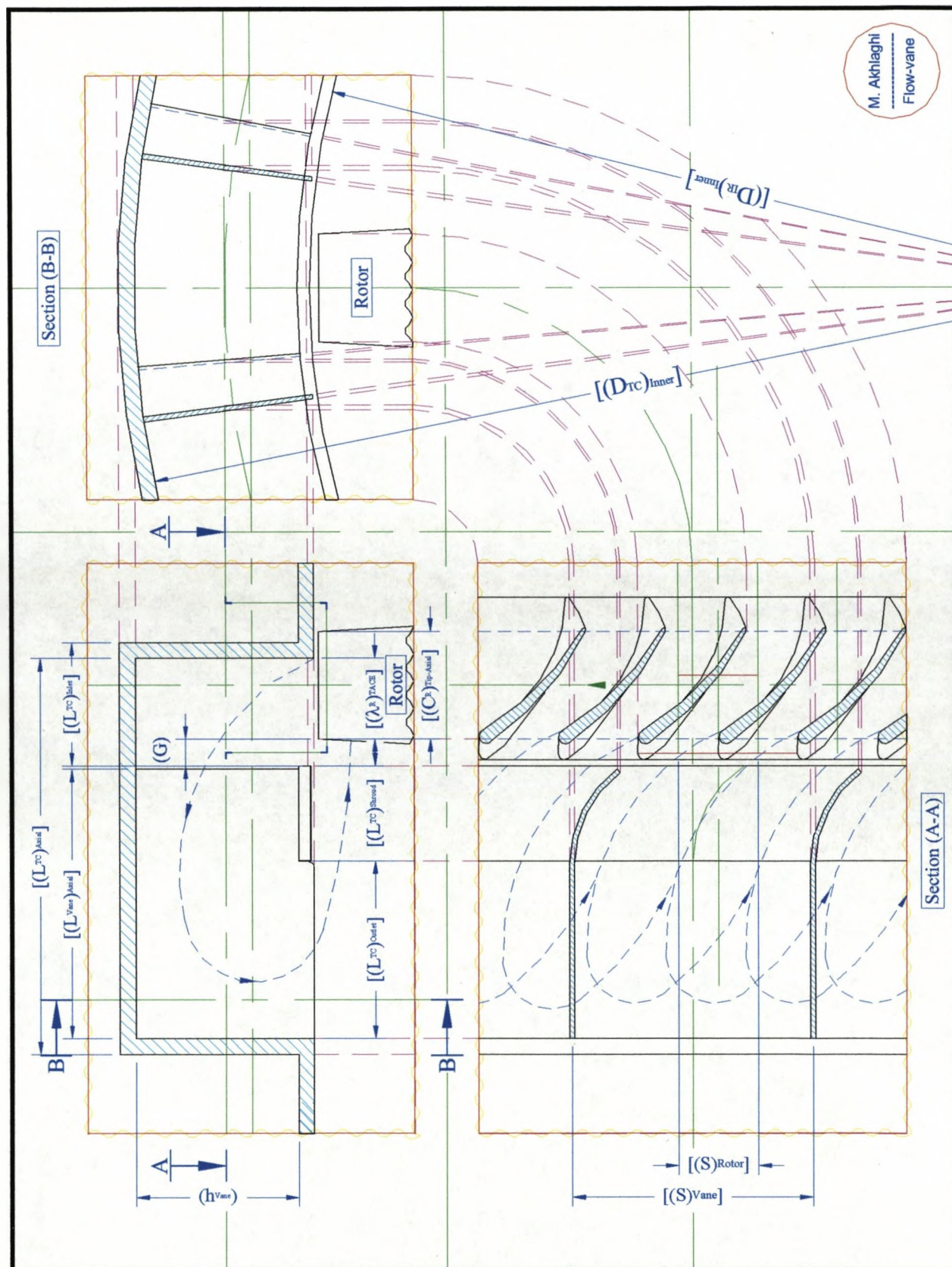
6.8)- Summary of Most Significant Factors:

The most significant factors and parameters, which should be considered in the design of an effective vane-recessed casing treatment, are summarised below.

- 1)- One of the major problems is the appearance of flow blockage zones near the tip or hub diameters of the rotor blades.
- 2)- The simple models of casing treatment applications in multistage axial-flow compressors have not been very effective.
- 3)- Removal of the inlet whirl (tangential) velocity is an essential part of the vane-recessed casing treatment operation and is crucial for stall delay.
- 4)- The purpose of applying the vanes inside the casing treatment is to remove the whirl components of the reversed flows. The removal of the tangential component of the flow before it is reintroduced into the mainstream might increase the work input and improve pressure rise without stall development, until lower flow coefficients.
- 5)- The vanes can be completely or partly curved in both radial and axial directions to remove the swirl flow component. No stall margin improvement can be achieved using vane-less recessed casing treatments.
- 6)- Using the curved vanes inside the casing treatment, the turning vane passages can eliminate the swirl components of the reverse flow.
- 7)- The effectiveness of different design parameters varies according to compressor geometry as well as to particular casing treatment features.
- 8)- A high hub to tip ratio results in more diffusion from the stator blades.
- 9)- Increasing the value of the hub to tip diameter ratio produces a flow rate reduction, which necessitates a treated casing with the lower recessed chamber dimensions. Lower ratio compressors pass larger mass flow rates requiring larger size vane-recessed casing treatments.
- 10)- Modification of the tip clearance flow is a key factor to stall margin improvement.
- 11)- The scaling of treated casing geometry in proportion to the rotor blade height was not entirely effective.
- 12)- The rotor stagger angle changes caused alteration of the operating load in terms of the work coefficient.
- 13)- At lower blade setting angles a continuously pressure rising characteristic without occurrence of rotating stall was observed. At higher blade setting angles, a rotating stall was detected with a large drop in the pressure rise characteristic.

- 14)- Rotor blade tip axial chord exposure ratio plays a vital role on the effectiveness of vane-recessed casing treatment. By increasing the rotor exposure ratio, it is possible to significantly increase the quantity of the reverse flow to the vane-recessed chamber. Very low rotor exposures are likely to lead to a worse performance than that of the untreated casing. In some cases, with a rotor exposure ratio of more than half, a considerable drop in performance occurred.
- 15)- Physical size and geometry of the casing treatment parameters including the shape of the vanes inside the recessed chamber play crucial roles in the improvement of the compressor performance.
- 16)- The current study has been conducted with a design using a semi-tubular passage casing which provides different inside diameter values across the axial length of the treated casing.
- 17)- Finding the optimum value of the recessed tube axial length is essential in order to have an effective casing treatment. When the size of the casing treatment is not optimal, there is an increased possibility of flow interaction between the recessed tube flow and the mainstream.
- 18)- When the size and position of the treated casing are optimal the treatment provides the minimum effective area for the flow exchange between the recessed tube and the main annular flows.
- 19)- An increase of hub to tip ratio means that the optimum value of the recessed chamber axial length has to be decreased.
- 20)- In order to remove the stalled flow from the machine, the inlet passage axial length of the recessed chamber has to be optimal.
- 21)- The Inner ring or shroud also has a significant affect on the performance of the vane-recessed casing treatment. Without an inner ring it is possible that the stall flow and re-injection flow might mix upstream of the rotor blades and thereby cause a reduction in efficiency.
- 22)- Axial length of the shroud is dependent upon the area, which is required by the recessed casing for the removal of the stalled flow from the rotor blade tips.
- 23)- The outlet passage area of the vane-recessed treatment is very important in order to re-inject the recessed flow in the desired direction of the mainstream.
- 24)- The nature of the interchange between the recessed flow and the mainstream imposes geometrical limitations on the axial length of the outlet passage.
- 25)- The reduction of the recessed chamber height failed to improve on the performance of the initial design height.
- 26)- In order to achieve an optimal treatment operation, a minimum flow rate exchange through the vane-recessed treated casing is essential.
- 27)- In order to achieve better performance, vane-recessed casing treatments with smaller recessed chamber heights must be used with higher hub to tip ratios compressors.
- 28)- Effective performance is dependent on the relationship between the axial length of the vanes and the axial length of the casing treatment. The relationship between the vane axial length and axial lengths of the outlet passage and inner ring is also critical.
- 29)- The application of vanes inside the recessed chamber can reduce the circumferential component of the flow inside the casing treatment.

- 30)- The optimum number of vanes is a vital component of the casing treatment. The pitch between two adjacent vanes must be optimised, so that the swirl can efficiently be removed from the stalled flow.
- 31)- The number and shape of the vanes are critical to the configuration of the vane passage. Without a sufficient number of vanes the casing treatment will be unable to break-up the stall cells in order to retrieve the flow which has to be re-injected back into the mainstream.
- 32)- In order to reduce the flow separation on the suction surface of the vanes, the reverse flow direction has to be changed from the radial to axial. The method used to achieve this was to apply a set of fully curved vanes.
- 33)- A minimum flow exchange between the recessed flow and the mainstream is required in order to produce significant stall margin improvements.
- 34)- The application of the vane-recessed casing treatment produced a continually rising performance characteristic with no evidence of rotating stall.
- 35)- The ratio of the rotor exposure is an important determinant of efficiency. Exposure ratios greater than the optimum generally lead to higher efficiency penalties.
- 36)- In order to maintain higher pressure rises through the casing treatment, more work input is required. The work coefficient increased as the flow rate decreased.
- 37)- Note under certain conditions some casing treatments may not significantly alter the efficiency characteristic but may increase the work input in the extended stall free flow range.



[Figure (6-1)]- The configuration of a vane-recessed casing treatment with partly curved vanes located partially upstream and to some extent over the rotating blades

CHAPTER SEVEN

7)- A Design System for Vane-Recessed Tubular-Passage Casing Treatments:

7.1)- Introduction:

This chapter will provide an in-depth analysis and description of the design features of a tubular-passage shaped vane-recessed casing treatment. The elimination of the whirl component of the inlet flow is an essential part of the stall delay mechanism, and therefore a primary objective of the vane-recessed tubular-passage treatment.

The results obtained from the present investigation indicate that the tubular-passage vane-recessed casing treatment was able to separate the re-circulating flow as well as removing the tangential velocity from the injected flow. The flow was reintroduced upstream of the rotor blades with a higher relative velocity.

The treated casing may increase the work input, but normally produces higher pressure rises and thereby prevents the occurrence of rotating stall cell development even in much lower flow rates. In the case of the present investigation, all of the casing treatment configurations produced increases in the maximum values for the overall work coefficient. This increase in the stability of the compressor at reduced flow ranges is highly beneficial both to compressor manufacturers and users. Compressor selection can be optimised to achieve higher efficiency in the normal duty range without the risk of stalling.

There are a large number of parameters, which can affect the design of a vane-recessed tubular-passage casing treatment. These parameters will be considered in detail in this chapter, as the basis for the casing treatment design. The discussion draws heavily on the information obtained from the existing literature that was used to formulate the conceptual analysis of vane-recessed treated configurations as described in the previous chapter.

7.2)- Available Space and Minimum Geometry for Treatment Design:

The objective of the current research is to investigate the provision of a stall free flow range on a typical tip critical multistage axial-flow compressor. This will be achieved through the application of a passive stall control device, which is fitted as a stepped casing in place of some part of the conventional solid casing.

A low-speed three-stage axial-flow (Peregrine) research compressor was selected as a typical tip critical compressor. The vane-recessed tubular-passage type casing treatment was chosen as the passive stall control device, which appeared to offer the most potential performance benefit.

The test compressor is a low-speed machine, but owing to its design category and construction, it has many features in common with high-speed compressors. It also offers a convenient way of varying the normal blade fixed angles and spacing. The design and construction of the test rig is discussed in detail in chapter four.

The series of casing treatment designs undertaken by previous researchers at Cranfield University were based on that of Ivanov et al. (1984) who designed a casing treatment for a

two stage axial fan which used an inner shroud with a set of fully cambered vanes. The Cranfield University adaptation of this design consisted of a flat inner ring, with vanes that were partly, rather than full cambered and positioned within the recessed chamber in the direction of flow.

The design employed in the present investigation builds on the work of Ivanov et al. (1984), the experience of earlier Cranfield University researchers and other studies that used casing treatments, which incorporated some relevant features. The present work however differs from previous Cranfield research in that, like Ivanov et al. (1989), it utilises fully cambered vanes.

The first step was to dismantle the three-stage compressor. The solid casing including the rotor blade drum had to be taken apart in order to measure both the inner and the outer spacer rings. The dimensions of each spacer rings were then measured with a high level of accuracy and individual ring drawings produced.

The individual drawings of each inner and outer spacer rings including the sketch of the inlet guide vanes, rotor and stator blades in association with the related fixing devices then were used to produce the schematic illustration of the test compressor. An example, schematic drawing of one spacer ring [segment (44463)] which is located after the inlet guide vanes is shown in [Figure (7-1)].

The inlet guide vanes holding ring, which retains (34) vanes, is located inside two solid casing spacer rings, namely segments (44462) and (44463), respectively. An assembly of the solid casing with moving and fixed blades including the spacer rings as well as their related locations at the nominal attitude geometry of the three stage axial flow Peregrine compressor is illustrated in chapter five, [Figure (5-2)].

In order to provide an appropriate position for the insertion of a vane-recessed tubular-passage casing treatment, a new segment casing spacer was designed. This was used to replace part of the corresponding segment (44463) of the conventional solid casing.

A new solid casing ring, which hereafter is referred to as segment (63) was designed and manufactured from steel with reference to the geometry restrictions that existed in the axial dimension of the compressor. This ring was designed to fit exactly after the spacer ring segment (44462) and before the main part of the stepped casing to avoid any undesirable axial movement of the inlet guide vane ring holder.

The initial idea was to insert and fix the new segments into part of the original solid casings, by using three pins located at the circumferential interval to join the newly designed segments to the existing adjacent spacer rings. However it was decided to adopt a different solution because there proved to be enough distance in the radial direction to allow for the design of a stepped joint either side of the spacers.

From an early stage of the casing treatment design, it was realised that there was a significant restriction in the available axial length that was required for the insertion of the stepped casing configuration. In fact the axial distance between the centreline of the inlet guide vanes and the first stage rotor blade cascade centreline at the nominal attitude of the compressor setting was restricted to just 48 [mm].

[Figure (7-2)] shows the cross section and side views of the new solid casing segment (63) which was designed to measure 28.5 [mm] wide at the outside diameter of the compressor and 3 [mm] wide at the inside diameter of the annulus. This 3.0 [mm] section of the new space ring was the minimum size for holding the complete inlet guide vane holder ring, which was designed to prevent any undesirable axial movement of the (IGVs).

With the insertion of the new segment only 17.5 [mm] remained between this new spacer ring and the adjacent original spacer ring segment (44461), which provided insufficient space to locate the intended casing treatment. The axial distance between the leading edge of the space ring segment (44461) and the leading edge of the rotor measured 1.6 [mm].

Even if the spacer (44461) were also removed, this would only have provided an axial distance of 18.6 [mm], but this would have resulted in a casing treatment with zero rotor blade tip axial chord exposure ratio. This was not desirable given that most previous research had demonstrated that the performance of the casing treatment, particularly in relation to stall margin improvement, was very sensitive to the amount of rotor blade exposure ratio.

In order to solve this problem of the axial length restriction, it was decided to take out the spacer ring, segment (44461) that was located largely over the rotor blade tip. This spacer ring measured 6.2 [mm] in width at the annulus outside diameter. The removal of spacer ring (44461) extended the available axial distance from 17.5 [mm] to 23.8 [mm].

However this was still less than the estimated axial length required for an effective casing treatment design. In fact this axial length was applied to the two preliminary casing treatment designs, and was then adopted for as part of the modified design for use on the test rig.

Within the configuration described above, the uncovered portion of the rotor tip in the axial direction measured 4.6 [mm], and this was divided by rotor blade tip axial chord of 19.8 [mm], to produce a (0.232) rotor tip axial chord exposure ratio.

By comparing the design guidelines and the geometry of the two preliminary as well as the final casing treatment configurations, it was apparent that the axial length of the vanes inside the recessed chamber had to be about (2/3) of the axial length available for the casing treatment. [Figure (7-3)] shows three views of the modified tubular-passage vane-recessed treated casing that was designed to be inserted partly over and partly upstream of the rotor blades after the inlet guide vanes with a rotor tip axial chord exposure of (23.2%).

7.3)- Design of the Vane-Recessed Tubular-Passage Treated Casing:

In order to design an appropriate casing treatment to fit over the existing space between the inlet guide vanes and first stage rotor blade cascade of the Peregrine compressor, the following procedures were followed.

It was decided to start by examining the first stage, including the inlet guide vanes, to find out how such a compressor configuration would behave when operating with, and without, the vane-recessed casing treatment.

The starting point for the design of an optimum casing configuration, was the patent of [Ivanov, et al.], as well as the conceptual analysis developed in the previous chapter. In fact two different stepped treated casing configurations were designed. A modified casing treatment was then designed by applying all the known geometrical and characteristic data, as well as the previously prepared design guidelines.

The results of a detailed geometric study were added to the above data, and from this the author selected a ratio of (1.4875) between the axial length of the treated casing and the axial length of the corresponding guide vane. A set of fully cambered vanes measuring 16 [mm] in width was designed to fit inside the recessed chamber.

The casing treatment design with some further modification to a number of the original spacer rings could have been applied to both the second and third stages of the test

compressor rig, although this was not part of the agreed research plan. The experimental tests on the first stage produced sufficiently encouraging results and confirmed that the applied stepped casing was reasonably effective and had achieved many of the research goals.

7.3.1)- Inner Diameter of the Recessed Chamber:

The inner diameter of the recessed area $[(D_{TC})_{Inner}]$ is one of the most significant parameters of the treated casing. An optimum dimension for this diameter also seems to be crucial and can easily affect the performance of the casing treatment. The modified design of the stepped casing in the current investigation consists of a partly curved recessed chamber with two, minimum and maximum inner diameters.

A good deal of time was spent in trying to devise some guidelines which could be used to determine suitable dimensions for the inner diameter as well as other geometry relating to the casing treatment. This proved to be very difficult to achieve.

However, it was realised that the few guidelines which had been presented by [Ivanov, et al.] relating to the treated casing vane height and inside diameter of the inner ring as well as the inlet and outlet passages, might be helpful.

By using the following simple equation, it was possible to suggest a value for the recessed chamber inside diameter, $[(D_{TC})_{Inner}]$.

$$[(D_{TC})_{Inner}] = [(D_{IR})_{Inner} + 2 * (e_{IR}) + 2 * (h_{Vane})]; \quad [mm] \quad [7-1]$$

Where;

$[(D_{IR})_{Inner}]$: the casing treatment inner ring (shroud) inside diameter, [mm].

(e_{IR}) : thickness of the shroud, [mm].

(h_{Vane}) : height of the vane inside the casing treatment, [mm].

The ratio between the inside diameter of the shroud and the rotor blade tip diameter had been proposed by [Ivanov, et al.] to be in the range of $[(1.01:1) \text{ to } (1.05:1)]$.

It is possible to present an equation based on this proposal as follows:

$$[(D_{IR})_{Inner}] = [(C_1) * (D_R)_{Tip}]; \quad [mm] \quad [7-2]$$

Where; $C_1 = 1.01 \sim 1.05$

It had been considered in the both preliminary design configurations to use a sheet metal with 2.0 [mm] thickness in order to manufacture the shroud of the treated casing. The subsequent decision to make the treatment from different types of materials such as high quality Perspex or Acrylic resulted in a manufacturing requirement for a minimum inner ring thickness of 3 [mm].

The ratio of the vane height in the radial direction to the corresponding axial length was also proposed by [Ivanov, et al.], in the range of $[(0.4:1) \text{ to } (0.65:1)]$. This ratio can also be presented as follows:

$$(h_{Vane}) = [(C_2) * (L_{Vane})_{Axial}]; \quad [mm] \quad [7-3]$$

Where; $C_2 = 0.4 \sim 0.65$

By taking into account the axial distance, which was available for the insertion of the treated casing, the following equation can be applied in order to calculate the axial length of the casing treatment, $[(L_{TC})_{Axial}]$.

$$[(L_{TC})_{Axial}] = [(L_{TC})_{Outlet} + (L_{TC})_{Shroud} + (L_{TC})_{Inlet}] = [(L_{Vane})_{Axial} + (d_{TC})_{Axial} + (L_{TC})_{Inlet}] \quad [7-4]$$

Where;

$[(L_{TC})_{Inlet}]$, $[(L_{TC})_{Shroud}]$ and $[(L_{TC})_{Outlet}]$ are the axial lengths of the casing treatment's inlet passage, inner ring and outlet portion, respectively.

$[(d_{TC})_{Axial}]$; the axial distance between the guide vane's leading edge and end of the shroud.

$[(L_{Vane})_{Axial}]$; the axial length of the guide vanes inside the casing treatment.

The effect of the axial distance from the vane leading edge to the end portion of the inner ring is reasonably short in comparison to the axial length of the casing treatment as well as the other aforementioned lengths. This effect can be ignored for time being, although it was completely neglected in previous research¹. The proper axial length of $[(d_{TC})_{Axial}]$ will be considered separately later on.

Therefore; the above equation can be written as follows.

$$[(L_{TC})_{Axial}] = [(L_{Vane})_{Axial} + (L_{TC})_{Inlet}]; \quad [mm] \quad [7-5]$$

It was also noted from the patent of [Ivanov, et al.], that the ratio of the vane axial length to the axial length of the treated casing inlet passage should preferably be in the range of [(2:1) to (3:1)].

The equation based on this proposal can be presented as follows:

$$[(L_{Vane})_{Axial}] = [(C_3) * (L_{TC})_{Inlet}]; \quad [mm] \quad [7-6]$$

Where; $C_3=2\sim3$

By using the two above noted equations, the casing treatment axial length can be determined through the following formula.

$$[(L_{TC})_{Axial}] = [(L_{Vane})_{Axial}] + \left[\left(\frac{1}{C_3}\right) * (L_{Vane})_{Axial}\right] = \left[\left(\frac{C_3 + 1}{C_3}\right) * (L_{Vane})_{Axial}\right]; \quad [mm] \quad [7-7]$$

Or;

$$[(L_{TC})_{Axial}] = [(A_1) * (L_{Vane})_{Axial}]; \quad [mm] \quad [7-8]$$

Where; $A_1=1.334\sim1.50$

It is also possible to present the vane axial length via following equation in terms of the recessed chamber axial length:

$$[(L_{Vane})_{Axial}] = \left[\left(\frac{C_3}{C_3 + 1}\right) * (L_{TC})_{Axial}\right] = [(A_2) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-9]$$

Where; $A_2=0.667\sim0.750$

By applying the above result into equation [7-3], the following formula in terms of the recessed vane height in the radial direction then can be presented as follows.

$$(h_{Vane}) = \left[\left(\frac{C_2 * C_3}{C_3 + 1}\right) * (L_{TC})_{Axial}\right] = [(A_3) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-10]$$

¹ (Hill, 1998).

Where; $A_3=0.267\sim0.488$

The recessed chamber inside diameter can then be found through the following equation.

$$[(D_{TC})_{Inner}] = [(C_1) * (D_R)_{Tip}] + [2 * (e_{IR})] + \left[\left(\frac{2 * C_2 * C_3}{C_3 + 1} \right) * (L_{TC})_{Axial} \right]; \quad [mm] \quad [7-11]$$

Or;

$$[(D_{TC})_{Inner}] = [(C_1) * (D_R)_{Tip}] + [2 * (e_{IR})] + [(A_4) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-12]$$

Where; $A_4=0.534\sim0.975$

By taking into account the available axial length of 23.8 [mm] for inserting the casing treatment, the 405 [mm] rotor blade tip diameter, as well as the 3.0 [mm] thickness of the shroud, the following two equations can be proposed. These are based on two of the most significant of the compressor and casing treatment parameters.

1)- Inner diameter of the recessed chamber in terms of the rotor blade tip diameter:

$$[(D_{TC})_{Inner}] = [(A_5) * (D_R)_{Tip}]; \quad [mm] \quad [7-13]$$

Where; $A_5=1.056\sim1.122$

2)- Inside diameter of the recessed chamber in terms of the treated casing axial length:

$$[(D_{TC})_{Inner}] = [(A_6) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-14]$$

Where; $A_6=17.972\sim19.095$

It can be concluded from the two above-noted equations, that it is possible to have a casing treatment with an inner diameter in the range of 427.74 [mm] to 454.45 [mm].

It should be emphasised that the current investigation is the first time a recessed chamber with a curved shaped inside diameter has been applied. The modified tubular-passage vane-recessed treated casing which was designed and manufactured from high quality acrylic had two minimum and maximum recessed inside diameters of 427.40 [mm] and 441.40 [mm], respectively.

The front and side views including the geometry of the outer casing which was designed using the different recessed chamber inside diameters is shown in [Figure (7-4)].

7.3.2)- Inner and Outer Diameters of the Shroud:

Ivanov et al. (1984) recommended a range of (1.01 to 1.05) for the ratio of the shroud inside diameter $[(D_{IR})_{Inner}]$ to the rotor tip diameter. The Peregrine compressor consisted of a rotor blade cascade with a tip diameter of 405 [mm], and therefore the inner diameter of the shroud has to be in a range of 409.05 [mm] to 425.25 [mm].

In order to provide sufficient space for the stalled flow to be injected easily into the recessed chamber, it was decided to move the casing treatment inner ring upwards in relation to the inner diameter plane of the original solid casing. This will be generally beneficial for casing treatments, which are applied to compressors that have insufficient space in the axial direction.

This has the added advantage of providing a wider inlet passage, which allows the unstable flow to move towards the rotor blade tip, where the mainstream flow can be energised in order to force the stalled flow up into the treatment.

It is recommended that the inner ring should be designed to have an aerodynamic shape. It

seems that inner rings that have this desirable geometry shape and configuration are more able to reduce the flow losses near the rotor blade tip, at both the inlet and outlet treatment passages, as well as inside the treatment chamber.

The inside diameter of the inner ring has to be an optimum value. A larger diameter can cause problems in that a portion of the mainstream flow may enter the casing treatment recessed chamber. This will lead to a reduction in both the performances of the treatment and of the compressor.

A larger diameter would also prevent the stalled flow from being injected into the recessed chamber. A shorter shroud inside diameter would reduce the area, which is essential for the stalled flow cells to move towards the rotor blade tip, and from there into the casing treatment. It appears that a less than optimum shroud inside diameter will have a more significant effect on machines, which are more restricted in space in the axial direction.

A shroud with an inner diameter $[(D_{IR})_{Inlet}]$ of 409.4 [mm] and an outer diameter $[(D_{IR})_{Outer}]$ of 413.4 [mm] was selected for the recessed treatment inner ring. However owing to difficulties in manufacturing the acrylic to the required dimensions it was decided to increase the shroud thickness by one [mm]. Therefore the shroud's inner diameter was decreased to (407.4) [mm].

In order to avoid undesirable flow interactions at both the inlet and the outlet passages, the inner ring was filleted in three corners. The front and side views of the treated casing shroud including the corresponding dimensions are shown in [Figure (7-5)].

It is also possible to suggest a convenient formula for the determination of the inside diameter of the shroud in relation to the axial length of the recessed chamber.

$$(D_{IR})_{Inner} = [(A_7) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-15]$$

Where; $A_7 = 17.187 \sim 17.868$

7.3.3)- Inlet Passage Axial Length:

[Ivanov, et al.], recommended that the ratio between the axial length of the guide vane measured along the compressor in the direction of the mainstream flow and the axial length of the inlet passage of the recessed chamber should be in the range of (2:1 to 3:1). Therefore the following equation can be used in this regard:

$$[(L_{TC})_{Inlet}] = [(1/C_3) * (L_{Vane})_{Axial}]; \quad [mm] \quad [7-16]$$

By applying the above equation [7-9] into this formula, the axial length of the inlet passage $[(L_{TC})_{Inlet}]$ can be determined in terms of the axial length of casing treatment, as follows.

$$[(L_{TC})_{Inlet}] = \left[\left(\frac{1}{C_3 + 1} \right) * (L_{TC})_{Axial} \right] = [(A_8) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-17]$$

Where; $A_8 = 0.250 \sim 0.333$

By applying this equation, it can be seen that the proper size of the inlet passage axial length should be in a range between 5.95 [mm] to 7.94 [mm]. For the (23.2%) rotor blade tip axial chord exposure an axial length of 6.8 [mm] for the inlet passage was selected.

An increment of 2 [mm], which corresponds to around a (10%) increase in the rotor tip axial chord exposure was chosen in order to provide six different inlet passage axial length settings. The axial length of the inlet passage, the ratio of this parameter to the axial length of the recessed chamber in relation to the corresponding rotor tip exposure ratios are shown

in the following table.

Rotor Tip Axial Chord Exposure Ratio (λ_{RTAC})	0.232	0.333	0.434	0.535	0.636	0.737	0.838
Inlet Passage Axial Length $[(L_{TC})_{Inlet}]$; [mm]	6.8	8.8	10.8	12.8	14.8	16.8	18.8
Ratio of the Inlet Passage to the Recessed Chamber Axial Length, $[(L_{TC})_{Inlet}]/[(L_{TC})_{Axial}]$;	0.286	0.370	0.454	0.538	0.622	0.706	0.790

7.3.4)- Outlet Passage Axial Length:

It appears that the flow, which leaves the recessed chamber and is re-injected back into the mainstream through the treatment outlet, plays a vital role in the performance of the machine as well as in producing improvements in the stall margin.

According to [Ivanov, et al.] the ratio between the axial length of the recessed chamber guide vane to the axial length of the outlet passage should preferably range between [(1.4:1) and (1.6:1)]. This recommendation can be expressed through the following equation.

$$[(L_{Vane})_{Axial}] = [(C_4) * (L_{TC})_{Outlet}]; \quad [mm] \quad [7-18]$$

Where; $C_4 = 1.4 \sim 1.6$

The axial length of the treatment outlet flow passage $[(L_{TC})_{Outlet}]$ the can be written as follows.

$$[(L_{TC})_{Outlet}] = [(1/C_4) * (L_{Vane})_{Axial}]; \quad [mm] \quad [7-19]$$

The axial length of the guide vane inside the recessed chamber was defined using the formula [7-9]. Therefore the above-noted equation can then be presented as follows.

$$[(L_{TC})_{Outlet}] = \left[\left(\frac{C_3/C_4}{C_3 + 1} \right) * (L_{TC})_{Axial} \right] = [(A_9) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-20]$$

Where; $A_9 = 0.469 \sim 0.476$

By taking into account the available space for inserting the casing treatment, the axial length of the recessed outlet passage based on the above formula can be considered in a range of 11.156 [mm] to 11.334 [mm]. Therefore this value in terms of the axial length of the guide vanes has to be considered in the range of 10.0 [mm] to 11.42 [mm].

However in view of the casing treatment axial length restriction, it was decided to choose the minimum boundary end for the re-injection flow passage, which was just 10 [mm]. In this configuration, the axial length of the recessed chamber outlet passage accounts for about (42%) of the axial length of the casing treatment.

7.3.5)- Inner Ring (Shroud) Axial Length:

There is no suggestion regarding suitable dimensions for the shroud either in the patent of [Ivanov, et al.] or in the previous work carried out at Cranfield University. In order to determine the appropriate axial length of the recessed chamber inner ring $[(L_{TC})_{Shroud}]$, the equation [7-4] above can be modified in the following form, by using the both inlet and outlet passage formulas which were presented through equations [7-17] and [7-20], respectively.

$$[(L_{TC})_{Axial}] = [(L_{TC})_{Inlet} + (L_{TC})_{Shroud} + (L_{TC})_{Outlet}]; \quad [mm] \quad [7-21]$$

Or;

$$[(L_{TC})_{Axial}] = \left[\left(\frac{1}{C_3 + 1} \right) * (L_{TC})_{Axial} \right] + [(L_{TC})_{Shroud}] + \left[\left(\frac{C_3/C_4}{C_3 + 1} \right) * (L_{TC})_{Axial} \right]; \quad [mm] \quad [7-22]$$

$$[(L_{TC})_{Shroud}] = \left\{ \left[\frac{C_3 * (C_4 - 1)}{C_4 * (C_3 + 1)} \right] * [(L_{TC})_{Axial}] \right\} = [(A_{10}) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-23]$$

Where; $A_{10} = 0.191 \sim 0.281$

According to the above coefficient, the proper size of the casing treatment inner ring should be in a range of 4.6 [mm] to 6.7 [mm]. However in order to be confident that the stalled flow can be separated from the flow which has to be re-injected back into the mainstream, an axial length of 7 [mm] has been proposed for the recessed chamber shroud in the current investigation.

7.3.6)- Axial Distance between the Vane Leading Edge and Shroud End:

Finding the optimum value for the axial distance between the end portion of the treatment inner ring and the leading edge of the guide vane $[(d_{TC})_{Axial}]$ is likely to play a significant role in the effectiveness of the casing treatment. This distance must provide enough space for the stalled flow to easily enter the recessed chamber, before being distributed into the passages between the two adjacent guide vanes.

No suggestions for an appropriate dimension for this axial distance were found in the existing literature. The following procedure is offered as a suitable solution to this problem. The axial length of the treated casing can be presented by writing the equation [7-4] in the following form.

$$[(L_{TC})_{Axial}] = [(A_9) * (L_{TC})_{Axial}] + [(A_{10}) * (L_{TC})_{Axial}] + [(A_8) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-24]$$

Or;

$$[(L_{TC})_{Axial}] = [(A_2) * (L_{TC})_{Axial}] + [(d_{TC})_{Axial}] + [(A_8) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-25]$$

By using the two above-noted equations, the following formulas can be produced.

$$[(d_{TC})_{Axial}] = [(A_9) * (L_{TC})_{Axial}] + [(A_{10}) * (L_{TC})_{Axial}] - [(A_2) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-26]$$

$$[(d_{TC})_{Axial}] = [(L_{TC})_{Axial}] - [(A_2) * (L_{TC})_{Axial}] - [(A_8) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-27]$$

However, each of the above is able to provide corresponding boundary values for coefficient (A_{11}). This coefficient can be expressed as follows.

$$[(d_{TC})_{Axial}] = [(A_{10}) * (L_{TC})_{Axial}]; \quad [mm] \quad [7-28]$$

Where; $A_{11} = 0.007 \sim 0.083$

By using the above coefficient in association with the dimension of the treated casing, the proper axial distance between the guide vane leading edge and the end of the inner ring can be found in a range of 0.167 [mm] to 1.975 [mm]. However, in order to provide sufficient room for the stalled flow to be injected without difficulty into the recessed chamber, an axial distance of one [mm] was selected.

7.3.7)- Recessed Guide Vane Axial Length:

By applying the equation [7-9], $[(L_{Vane})_{Axial}] = [(A_2) * (L_{TC})_{Axial}]$ together with its corresponding coefficient; $[A_2 = 0.667 \sim 0.750]$, the axial length of the recessed chamber

guide vanes $[(L_{\text{Vane}})_{\text{Axial}}]$ can be determined in the range of 15.875 [mm] to 17.850 [mm].

Both of the preliminary designed casing configurations had a vane axial length of 17 [mm], but due to the geometry restriction in the direction of the mainstream, it was decided to have a set of 120 vanes with an axial length of 16 [mm]. Each vane has a curve shape with a chord of 16.75 [mm], whereas the vane axial length just covered around (67.2%) of the recessed treated casing axial length, when the rotor blade tip exposure ratio was (0.232).

7.3.8)- Height of the Recessed Guide Vane:

The equation [7-10], $[(h_{\text{Vane}}) = (A_3) * (L_{\text{TC}})_{\text{Axial}}]$ in which the coefficient of $(A_3 = 0.267 \sim 0.488)$ can be used in order to find the appropriate height of the recessed guide vanes (h_{Vane}) , which are located inside the treatment recessed chamber.

The heights of the vane set could therefore range between 6.355 [mm] to 11.615 [mm]. Owing to the decision to use a tubular passage treatment, each guide vane, had three different heights of 13.4 [mm], 11.5 [mm], and 14 [mm]. These are the heights of the vanes at the leading and trailing edges as well as the maximum height.

Note in order to reduce possible interaction between the mainstream and the re-injected flow the height of the vanes were increased by one [mm] at the outlet passage of the recessed chamber in the mainstream flow region.

7.3.9)- Number of the Guide vanes Inside the Recessed Chamber:

Owing to the absence of information in the literature regarding the optimum number of vanes (Z_{Vane}) inside the recessed treatment chamber, an alternative approach was sought. It was decided to set the space to chord ratio of the vanes at the mid height diameter corresponding to the maximum height of the recessed chamber vanes equal to the hub to tip ratio of the compressor.

The aim of this was to minimise any undesirable interactions between the compressor rotor blades and recessed guide vanes. Additionally it was thought that this decision of setting the vane space to chord ratio equal to a significant compressor parameter such as hub to tip diameter ratio would also provide better matching between the casing treatment and the test compressor.

The number of vanes can be calculated through following equation.

$$Z_{\text{Vane}} = \frac{[\pi * (D_{\text{Vane}})_{\text{Mid-Height}}]}{[(S_{\text{Vane}})_{\text{Mid-Height}}]} = \frac{[\pi * (D_{\text{Vane}})_{\text{Mid-Height}}]}{[(L_{\text{Vane}}) * (S_{\text{Vane}})_{\text{Mid-Height}} / (L_{\text{Vane}})]} \quad [7-29]$$

Where, $[(D_{\text{Vane}})_{\text{Mid-Height}}]$ and $[(S_{\text{Vane}})_{\text{Mid-Height}}]$ are respectively the diameter and the space between two adjacent vanes corresponding to the recessed guide vane mid height. $[(S_{\text{Vane}})_{\text{Mid-Height}} / (L_{\text{Vane}})]$ is the space to chord ratio of the vane.

The variation of the vane space to chord ratio has a significant impact on the number of vanes inside the recessed area. When this ratio is near unity, the vane number will reduce, but by decreasing the ratio towards zero, the number of the vanes increases.

Assuming the space to chord ratio to be equal to (0.5, 0.75 and 1.0), the number of vanes inside the recessed chamber will be (168, 112 and 84), respectively. When the space to chord ratio of the vanes is assumed to be identical to the hub to tip diameter ratio (0.7), the number of vanes was found to be (120). The complete side view of (120) guide vanes inside the recessed chamber of the casing treatment along with (38) rotating blades is shown in [Figure (7-6)].

It is worth noting that the Peregrine compressor has (38 and 37) blades in each rotor and stator cascades, respectively. In order avoid any possible undesired interaction between the mainstream, and the re-injected flow from the recessed chamber outlet passage, it was considered more sensible to choose a prime number set of vanes, for instance (121).

However, clearly an even set of (120) vanes is simpler to manufacture. Moreover an even number of vanes, makes it possible to reduce the number of vanes inside the recessed chamber to half or one third of the original set. However, owing to the way in which the casing treatment was manufactured and the type of material used, it was not possible test the impact of various reductions in the number of vanes without causing significant damage to the treatment.

7.3.10)- Recessed Vane Shape and related Applied Manufacturing Material:

A set of fully curved vanes was applied to the recessed chamber of the treated casing in order to assist the passing of the stalled flow between the two adjacent vanes. The vanes, which were manufactured from a high quality Acrylic, were positioned between the shroud and the outer ring. They extended one [mm] from the inner ring end to the end of the recessed chamber in the direction of the reverse flow.

In order to manufacture the guide vanes, a master tool was produced and the vanes formed to the tool profile and trimmed to the appropriate size on a CNC Miller. All (120) recessed guide vanes have a thickness of one [mm] and were cambered about (30°) from a direction which was perpendicular to the compressor axis. The inner and outer diameters of each vane curvatures were 20.3 [mm] and 21.3 [mm], respectively.

Due to the curved shape of the recessed chamber inside diameter, the injected flow to the recessed portion will meet the vane at a height of (13.4) [mm]. The maximum height of the vane is located at a point corresponding to (25.62%) of the vane axial length from the leading edge side.

As the flow passes the curved part of the chamber, which corresponded, to the maximum point of the recessed chamber inside diameter through the two adjacent vanes, the vane height is reduced down to 11.5 [mm]. Front, top and side views of the final designed vane-recessed tubular-passage casing treatment including the complete geometry features and dimensions of the vanes are shown in [Figure (7-7)].

7.3.11)- Rotor Blade Tip Axial Chord Exposure Ratio:

As noted earlier, the rotor blade tip axial chord exposure ratio (λ_{RTAC}) is one of the most significant parameters in the performance of the casing treatment. This ratio is not only a compressor related parameter, but it can also be considered among the treatment related parameters as well. The majority of previous research in the area of vane-recessed casing treatment suggests that the specification of an optimum value for the rotor tip exposure ratio is essential in order to achieve an effective stall delay mechanism. This criterion is also associated with the compressor geometry restrictions.

The removal of the conventional solid casing, spacer ring segment (44461) had two objectives. Firstly to provide sufficient room for the casing recessed chamber and the inlet passage axial lengths and secondly to facilitate a minimum value for the rotor tip exposure, within the range previously identified. The modified configuration of the treatment with a recessed chamber axial length of 23.8 [mm], which was designed to replace two original spacer rings, segments (44463 and 44461), was able to provide a minimum rotor tip exposure ratio of (0.232).

Another possibility that was considered was the removal of original segment (44464) in addition to the above-noted spacer rings. The insertion of a suitably designed spacer ring instead, would have given a treated casing build with (87.4%) rotor blade tip axial chord exposure. However it was decided not to proceed with this modification because first test runs showed that the higher exposure ratios produced the least impressive results.

To reduce manufacturing costs it was decided to produce just two spacer rings which were intended to provide two rotor blade tip exposures of (43.4% and 53.5%). As with the treated casing outer ring, shroud and guide vanes, the two exposure spacer rings were manufactured from high quality Acrylic. Schematic views and the geometry of the two space rings that provided alternative uncovered rotor tip axial chords of 8.6 and 10.6 [mm] and their respective exposure ratios of (0.434) and (0.535), are shown in [Figure (7-8)].

The individual drawing including the dimensions of the 53.5 % exposure spacer rings with 15.1 and 6.7 [mm] axial lengths at the outer and the inner diameters, respectively, is also shown in the [Figure (7-9)]. The experimental results obtained from the builds associated with aforementioned exposures indicated that more exposure spacer rings were essential for further experiments.

As a result five more spacer rings with the 6.6, 12.6 14.6 and 18.6 [mm] of uncovered axial length corresponding to (33.3%, 63.6%, 73.7%, and 83.8%) rotor blade tip axial chord, respectively were designed and manufactured from aluminium alloy. The front schematic views of the eight different spacer rings corresponding to an increment of nearly (10%) and in a range of (23.2% to 83.8%) rotor blade tip exposures are shown in [Figure (7-10)].

It was also possible to remove the original spacer ring segment (44464) from the test compressor and design a new spacer ring with a shorter height to provide a rotor exposure ratio percentage of (87.4%). The results indicated that the best improvements were achieved with rotor exposures less than the above noted one. It was therefore decided not to manufacture any more new spacer rings. The configuration of the compressor with this ring is shown in [Figure (7-10)] as well.

7.4)- Casing Treatment Flow Diagnostics and Measurement Devices:

Some of the previous research has used the wool tuft flow visualisation technique inside the vane-recessed casing treatment to show flow separation on the suction surface of the vanes. It was hoped that the application of a set of fully curved vanes inside the recessed chamber would prevent these undesirable effects by removing the swirl from the circulating flow inside the vanes.

Further study of the flow direction in order to gain a better understanding of the flow behaviour between the two adjacent vanes was required. It was therefore decided to design an area containing (21) holes to allow examination of the flow inside the recessed zone of the casing treatment.

The locations of the holes in the body of the casing treatment outer ring, which covered the vane-less part of the treatment, as well as an area between the two adjacent recessed guide vanes, are illustrated in [Figure (7-11)]. As shown in the above figure, the vane-less area contained six holes with 0.5 [mm] diameter, whereas the area, which is, located between the two adjacent vanes comprised (15) holes with the same diameter size.

The compressor of the test rig was equipped with conventional slow response wall static pressure measurement tappings at the inlet and outlet of each blade cascades as well as in the gaps between the blade rows. At each measuring station four wall static pressure tappings were equally located at (90°) intervals around the circumference of the solid wall

spacer rings.

However, it was decided to apply four equal size wall static pressure tapings at (90°) intervals around the circumference of the recessed tube treated casing outer ring. The centrelines of these tapings were located on the maximum curvature point of the casing treatment outer ring. The distance between each pressure tapping and the leading edge of the recessed guide vanes in the mainstream flow direction was 4.1 [mm].

These tapings measured the static pressure downstream of the inlet guide vanes as well as upstream of the first stage rotor blades. In order to avoid any averaging errors, it is recommended that every two of the tapings be connected in pairs by two equal lengths of tube to the two measuring pipes of a manometer with 36 channels filled with water. Three views of the wall static tapings with 1.65 [mm] and 1.0 [mm] outside and inside diameters, corresponding to a depth of 7.65 [mm] and 5.0 [mm], along with their locations in the recessed treatment outer ring is shown in [Figure (7-12)].

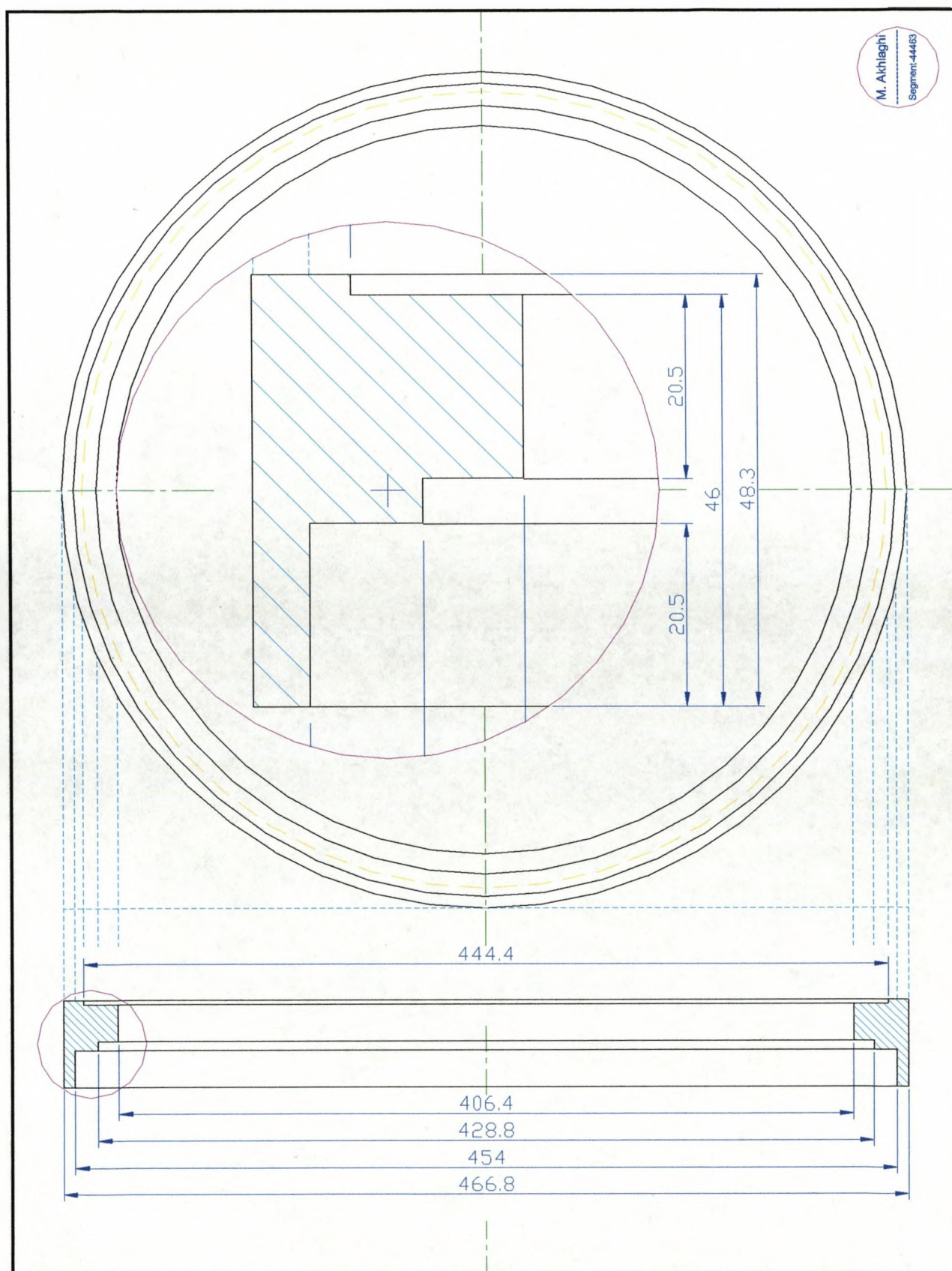
A hole of 4.0 [mm] diameter and 11.8 [mm] depth was designed at a point (112.7°) anticlockwise from the compressor axis, in order to provide a location for the total pressure probe. The centrelines for the four static tapings, one set of flow visualisation window holes, and the total pressure probe are shown in [Figure (7-6)].

7.5)- Summary of the Casing Treatment Design Results:

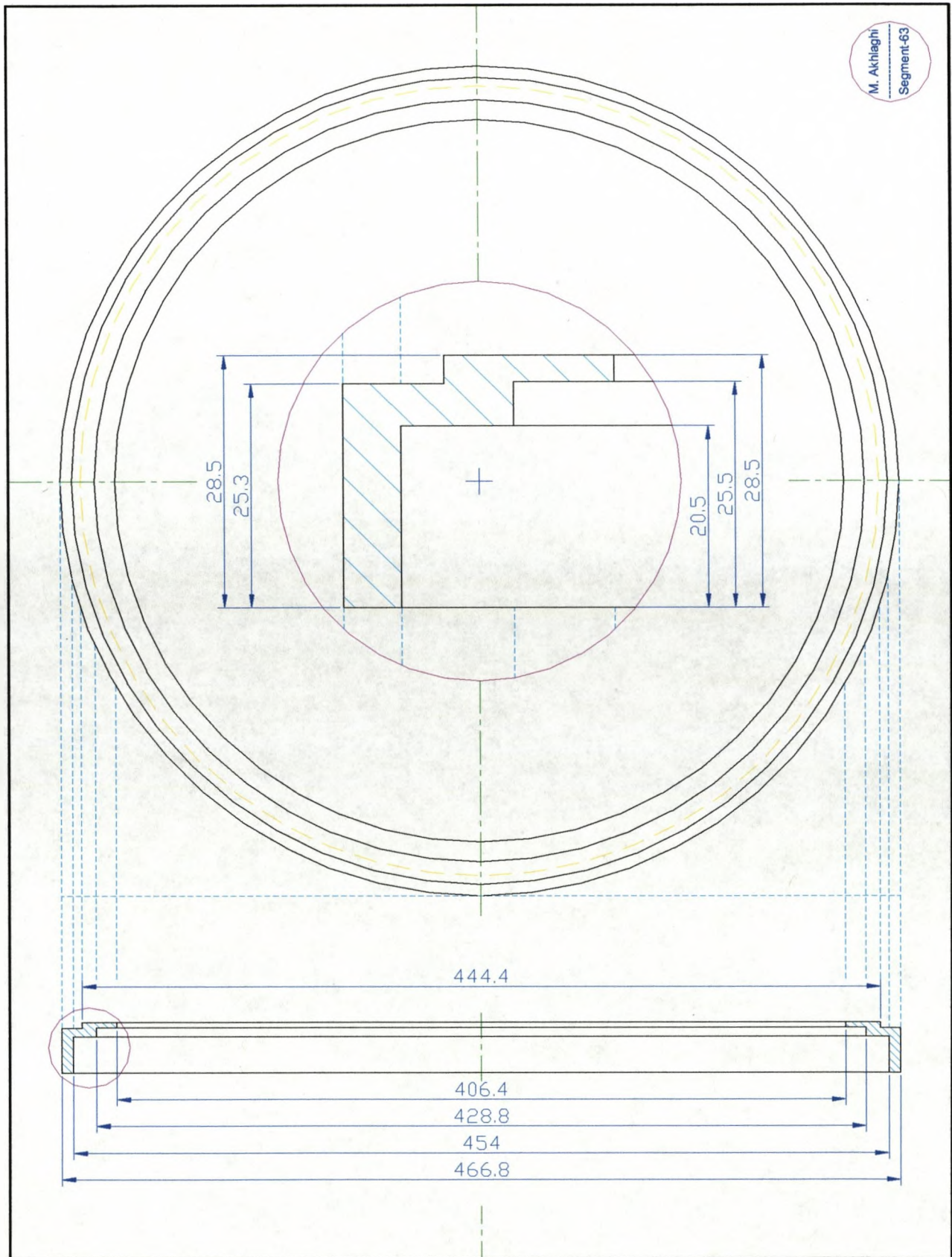
The available geometry and space for the design and insertion of a casing treatment proved to be quite challenging, in both the axial and radial directions. The geometry of the three-stage compressor used as part of the test rig in the current investigation was quite restricted in terms of space in the axial direction, but provided more than enough space in the radial direction.

Followed the analysis of the casing treatment design requirements, the maximum available length in the direction of the flow was determined. A set of the formulas was also suggested in relation to the most significant parametric influences of the compressor as well as the casing treatment. The flow measuring devices were also described.

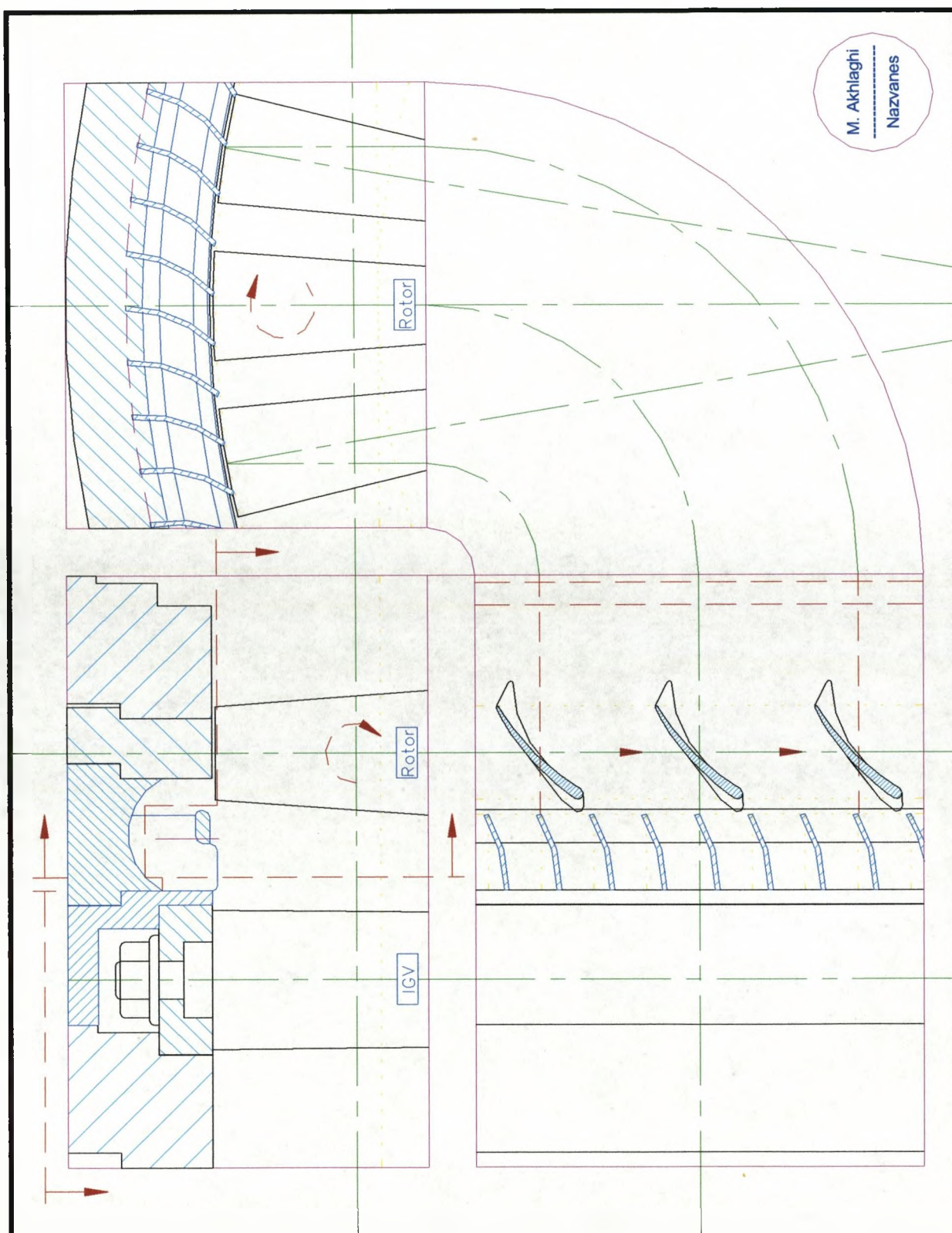
The most important of the compressor and casing treatment relevant parameters used in the current design, including all of the available data and parameters from the previous experimental research carried out at Cranfield University in this area has been tabulated in [Table (7-1)]. This table of the information in addition to the guideline data presented in this chapter has been included to provide a unique reference source for future designs, which utilise this type of casing treatment.



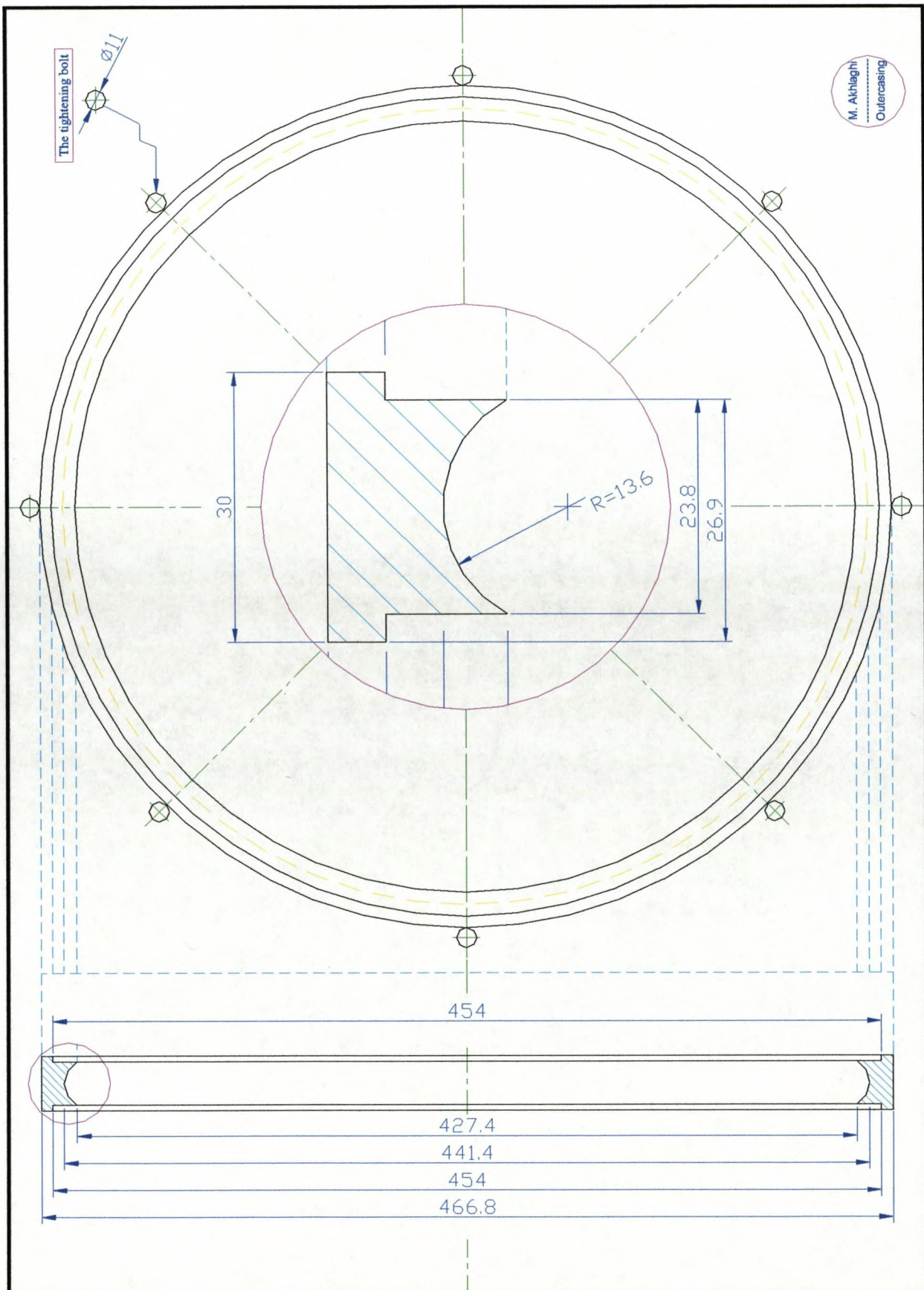
[Figure (7-1)]- The schematic drawing of spacer ring [segment (44463)] which is located after the inlet guide vanes



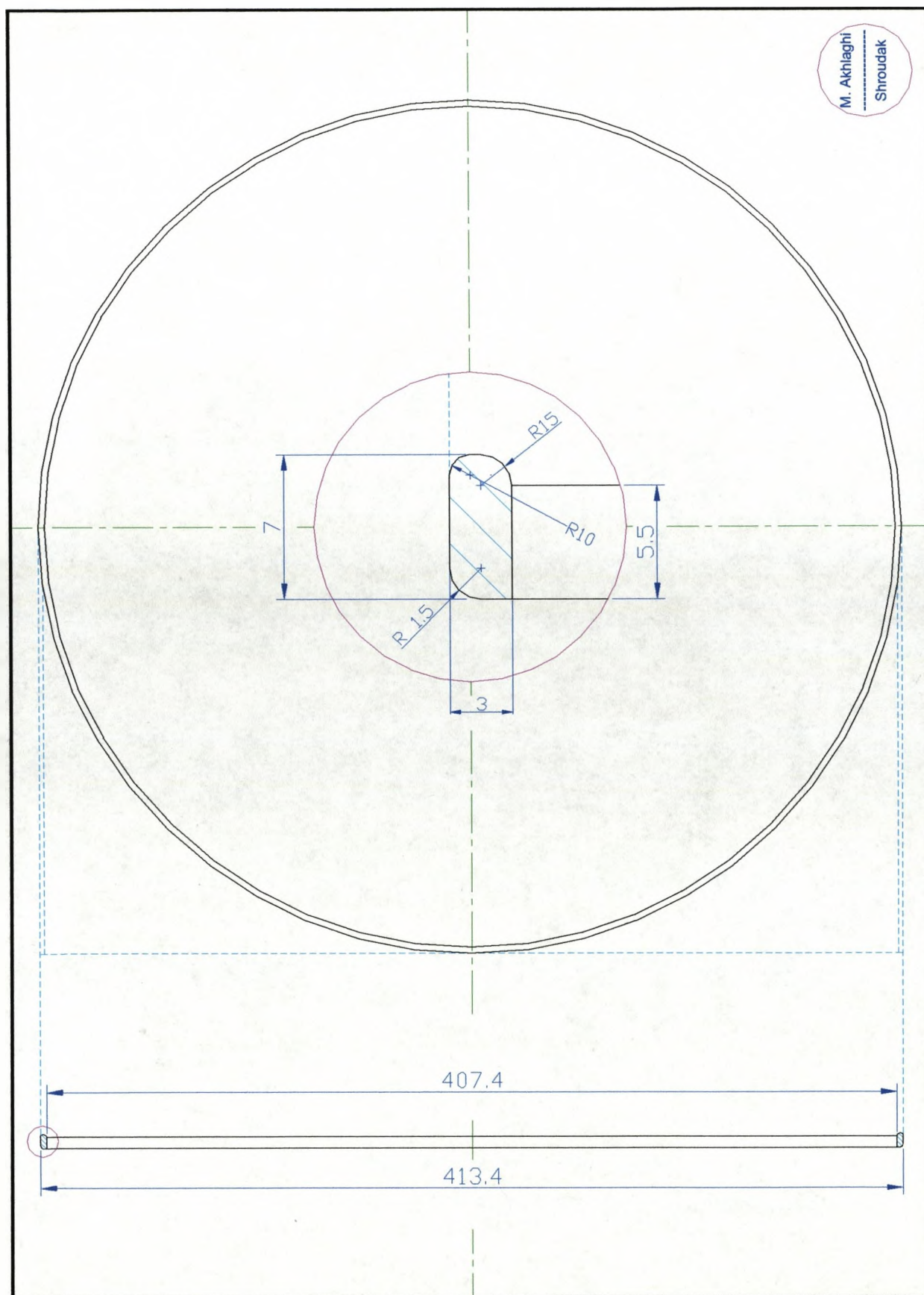
[Figure (7-2)]- The cross section and side views of the new solid casing segment (63) which was designed to measure 28.5 [mm] wide at the outside diameter of the compressor and 3 [mm] wide at the inside diameter of the annulus



[Figure (7-3)]- The three views of the modified tubular-passage vane-recessed casing treatment which was designed to be inserted partly over and partly upstream of the rotor blades after the inlet guide vanes with a rotor tip axial chord exposure ratio of (0.232)



[Figure (7-4)]- The front and side views of the recessed chamber outer casing that replaced some part of the (segment 44463) and all portion of the (segment 44461) axial lengths



[Figure (7-5)]- The front and side views of the treated casing inner ring (shroud)

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The centreline position of the first hole static pressure tapping with an outside diameter of 1.65 [mm] and 7.65 [mm] deep. The inside diameter of the hole is 1 [mm] with a depth of 6.2 [mm].

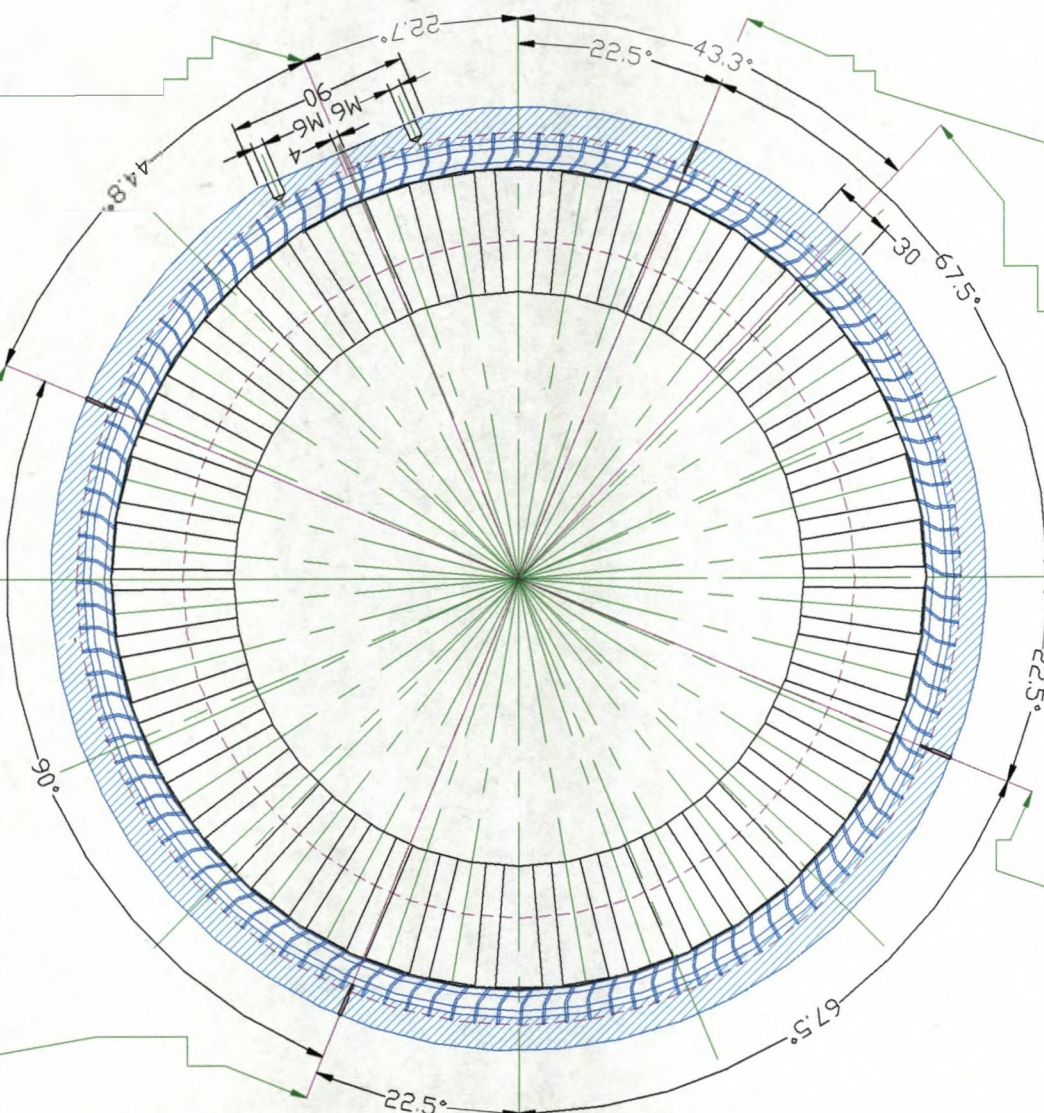
The centreline position of flow visualisation holes set. The inside diameter of each (21) holes are about (0.5) [mm], but the heights of them are three with 12.5 (mm), six with 13 (mm), six with 14 (mm) and six with 16 (mm), respectively.

The centre line position of the fourth hole static pressure tapping with an outside diameter of 1.65 [mm] and 7.65 [mm] deep. The inside diameter of the hole is 1 [mm] with a depth of 6.2 [mm].

The centreline position of the probe hole with a 4 [mm] diameter and a depth of 11.8 [mm]. The width of the 90 [mm] length is identical to the width of the treated casing which is 30 [mm].

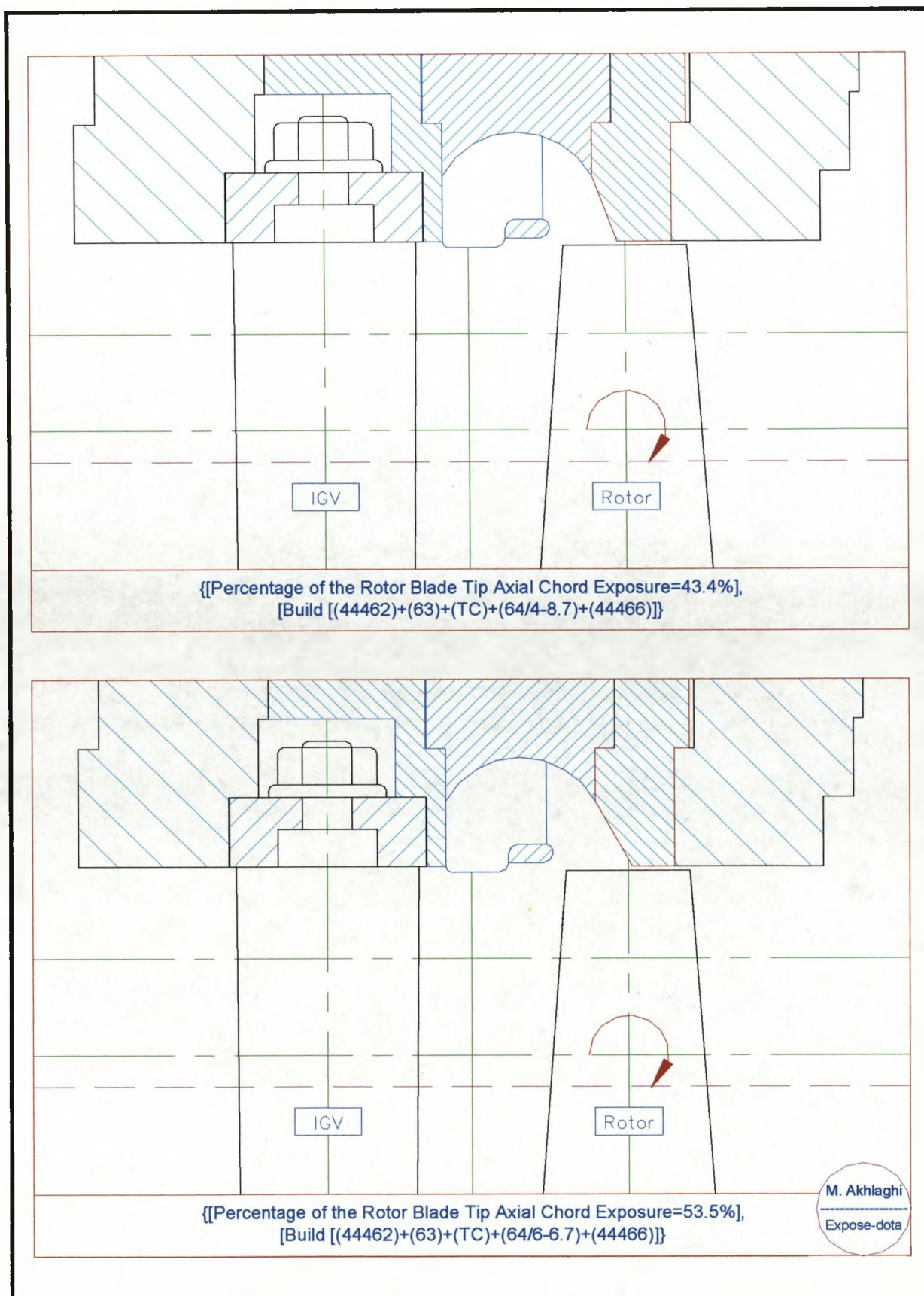
The centreline position of the second hole static pressure tapping with an outside diameter of 1.65 [mm] and 7.65 [mm] deep. The inside diameter of the hole is 1 [mm] with a depth of 6.2 [mm].

The centreline position of the third hole static pressure tapping with an outside diameter of 1.65 [mm] and 7.65 [mm] deep. The inside diameter of the hole is 1 [mm] with a depth of 6.2 [mm].

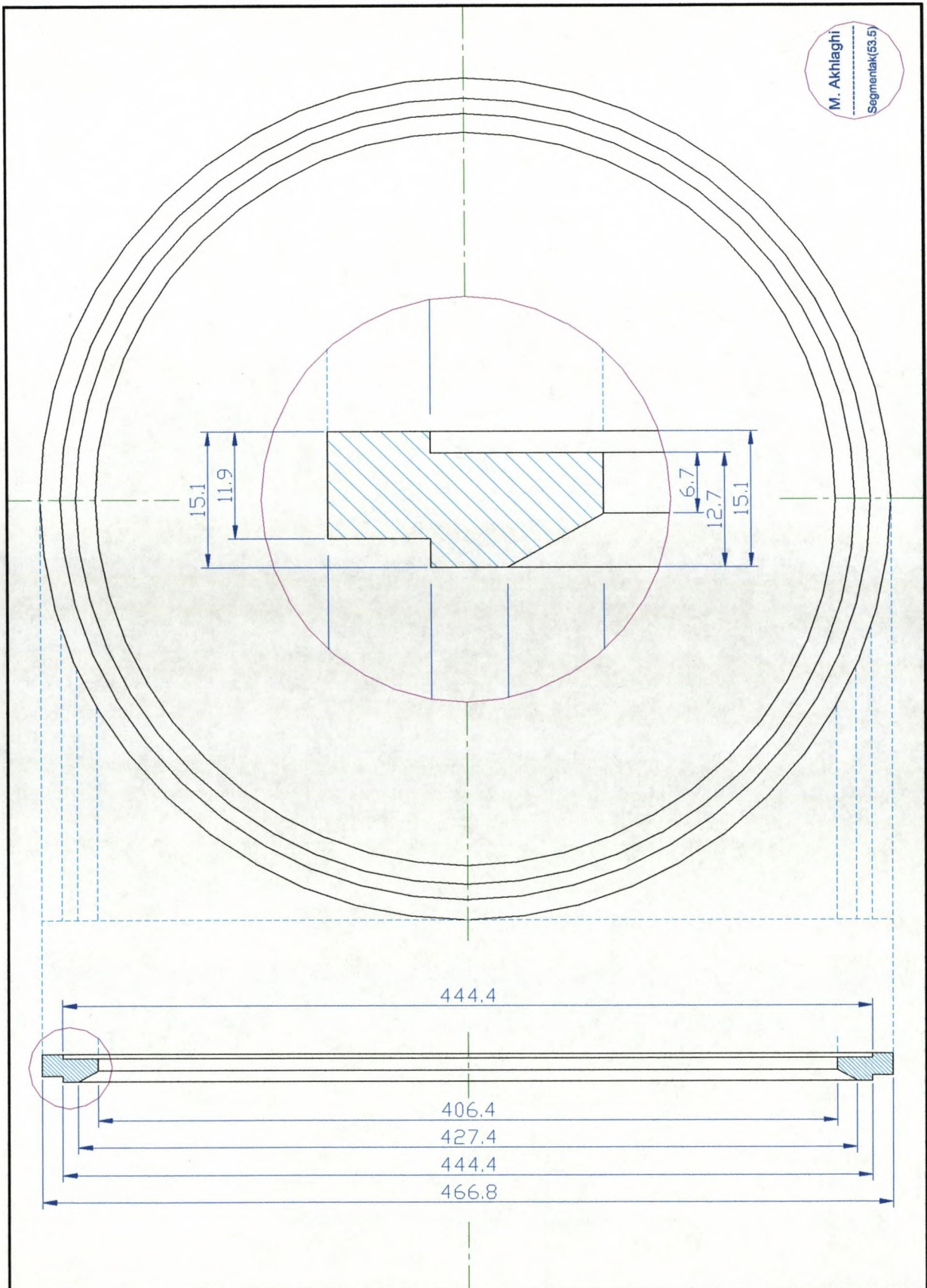


[Figure (7-6)]- The side view of 120 recessed guide vanes, 38 rotor blades, and the centreline positions of the four static pressure tapplings, a total pressure probe and the set of flow visualisation holes

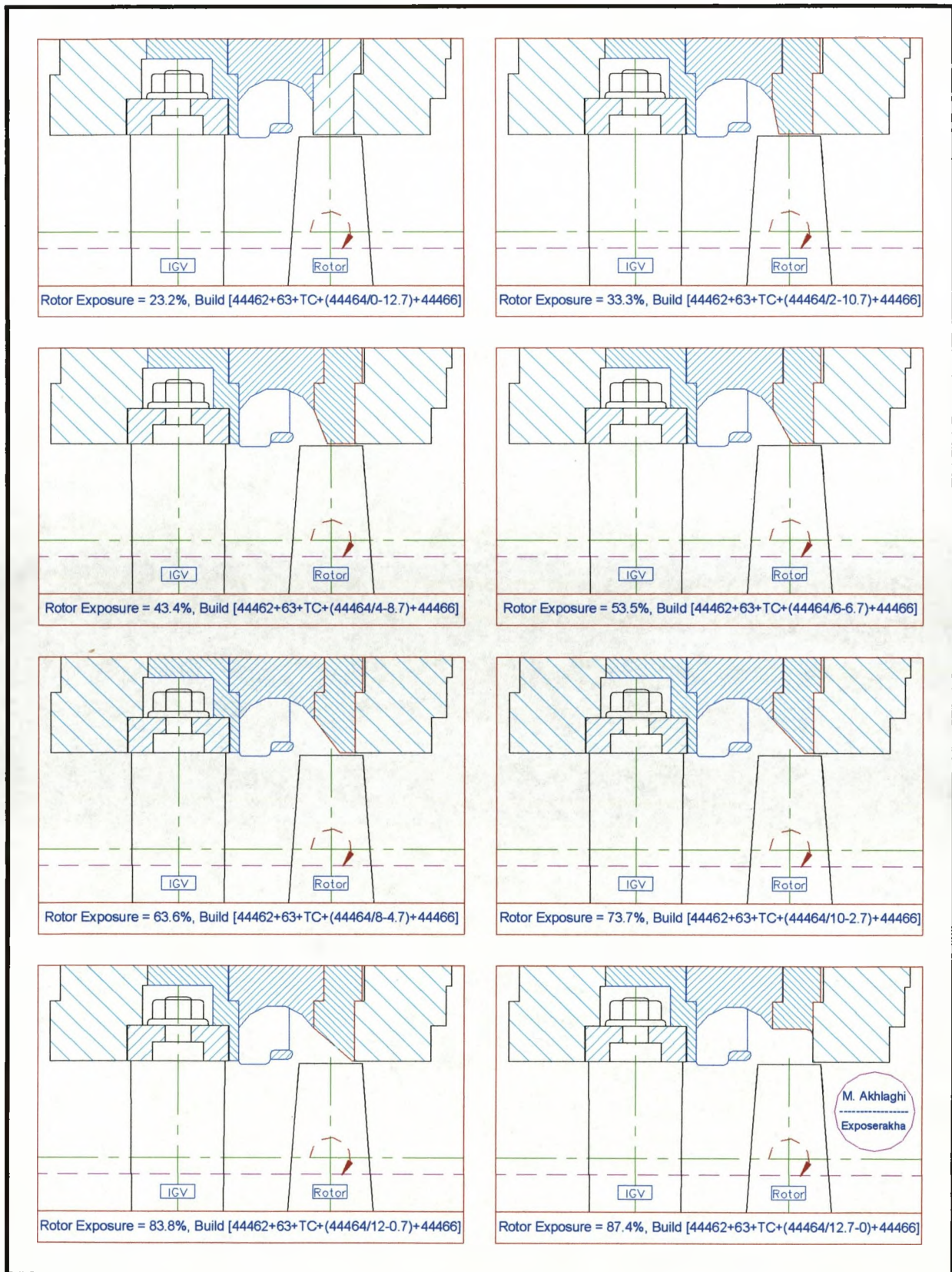




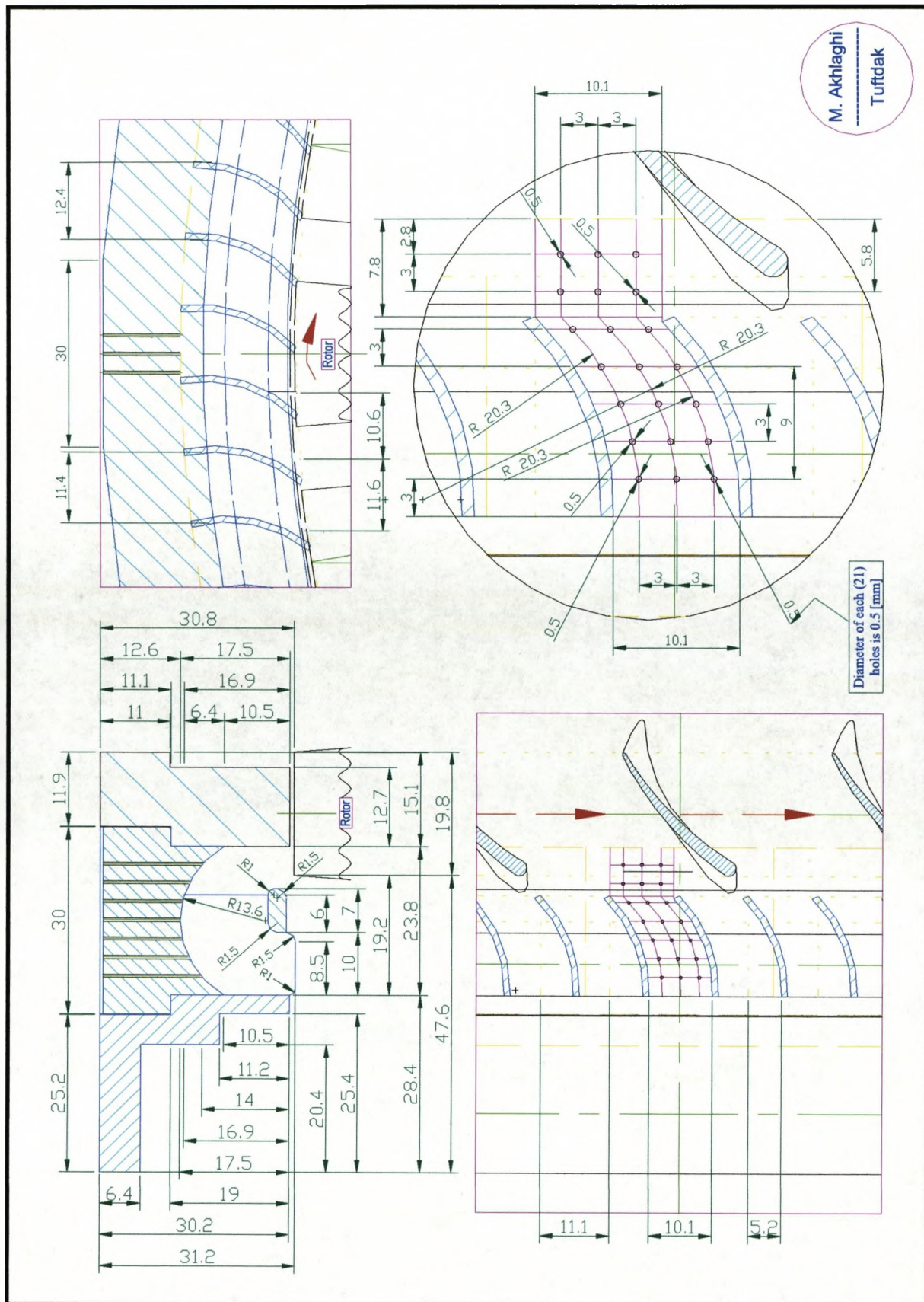
[Figure (7-8)]- The schematic views and geometry of two builds with different values of 8.6 [mm] and 10.6 [mm] of the rotor blade tip axial chord uncovering corresponding to exposure ratio of (0.434 and 0.535)



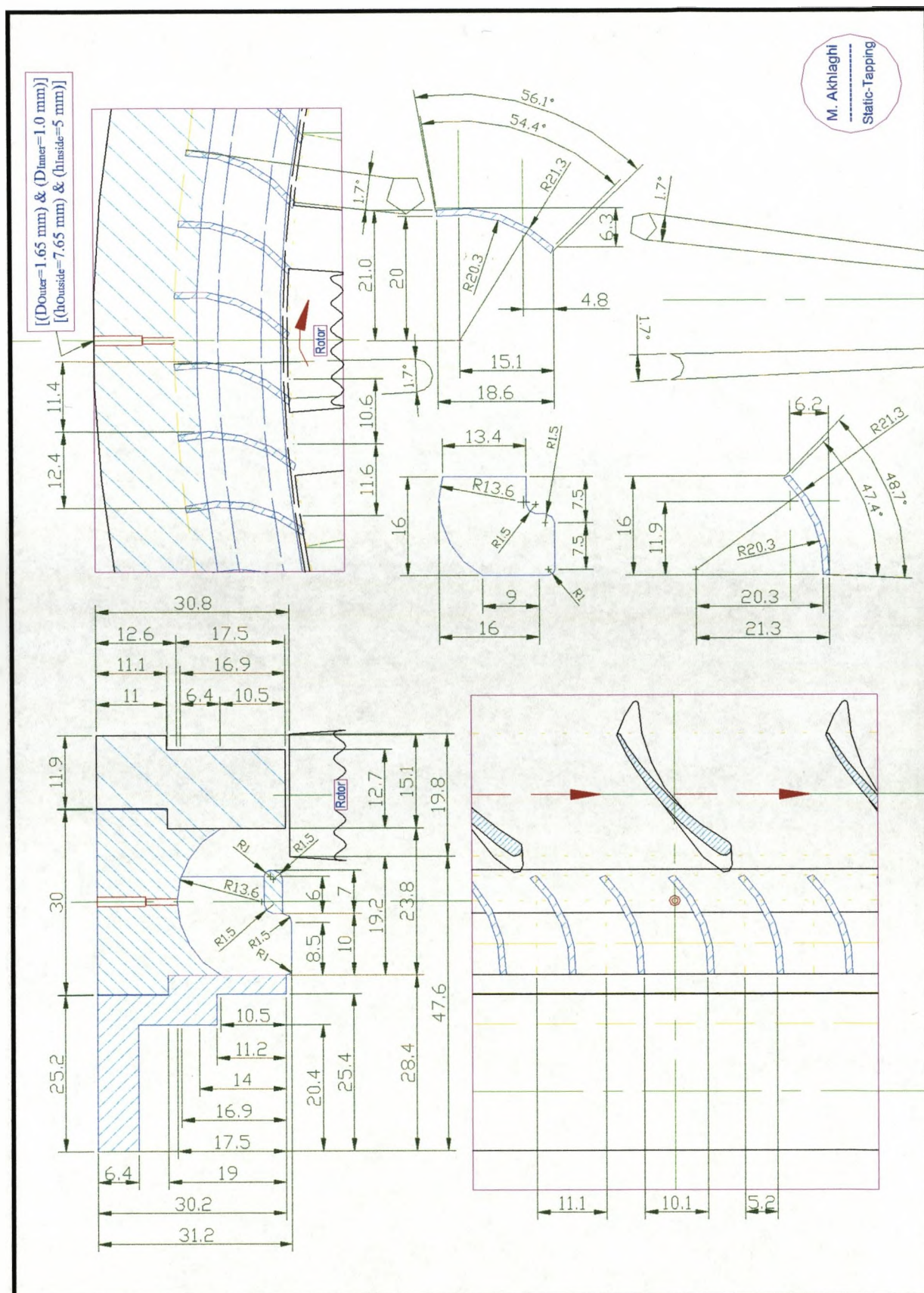
[Figure (7-9)]- The drawing of the applied spacer ring which was inserted after main part of the treated casing in order to provide a rotor blade tip axial chord exposure of (53.5%)



[Figure (7-10)]- The front views of the treated casing with different builds corresponding to the eight values of the rotor blade tip axial chord exposure ratio



[Figure (7-11)]- The locations of the flow visualisation holes that are covered the vane-less part including the area between two adjacent recessed vanes passage inside the treated casing chamber



[Figure (7-12)]- Three views of the wall static tapping in the treatment outer ring with (1.65 & 1.0) [mm] outside and inside diameters, corresponding to (7.65 & 5) [mm] depths

Names Parameters	(Azimian) [$D_H/D_T=0.5$]	(Ziabasharhagh)		(Kang) [$D_H/D_T=0.5$]	(Hill) [$D_H/D_T=0.4$]	[(Akhlaghi) [$D_H/D_T=0.7$]
		[$D_H/D_T=0.7$]	[$D_H/D_T=0.9$]			
Situation of the axial-flow machine	Single-stage & Isolated rotor fan ¹	Single-stage compressor ²	Single-stage compressor	Isolated-rotor fan ³	Rotor only fan ⁴	Three-stage compressor
Rotor blade tip diameter, $(D_R)_{Tip}$; [mm]	505.0	506.0	507.0	505.0	495.0	405.0
Rotor blade hub diameter, $(D_R)_{Hub}$; [mm]	254.0	356.0	457.0	254.0	200.0	284.4
Rotor hub to tip diameter ratio, $(DR)_{H/T}$	0.5	0.7 ⁵	0.9	0.5	0.4	0.7
Rotor blade running speed, (N_R) ; [rpm]	1500.0	1500.0	1500.0	1500.0	2910.0	3000.0
Rotor blade tip clearance, (t_c) ; [mm]	1.50	1.0	0.50	1.50	2.50 ⁶	0.70
Number of inlet guide vane, (Z_{IGV})	No IGV	No IGV	No IGV	No IGV	No IGV	34.0
Number of rotor blades, (Z_R)	27.0	?	?	27.0	6.0 ⁷	38.0
Number of stator blades, (Z_S)	28.0	51.0	59.0	rotor only	rotor only	37.0
Axial chord at rotor tip, $[(C_R)_{A-T}]$; [mm]	34.0	?	? ⁸	34.0	? ⁹	19.80
Circumferential velocity at the rotor tip diameter, $(U_R)_{Tip}$; [m/sec]	39.663	39.741	39.820	39.663	75.422	63.617
Rotor blade chord, (C_R) ; [mm]	47.80	50.0	35.0	48.0	variable	30.50
Rotor blade height, (h_R) ; [mm]	125.50	75.0	25.0	125.50	147.50	61.0
Rotor blade aspect ratio, (h_R/C_R)	2.625	1.50	0.715	2.615	variable	2.0
Annulus inside diameter of the solid casing, $[(D_{Ann})_{Inner}]$; [mm]	508.0	508.0	508.0	508.0	500.0	406.40
Ratio of the annulus inside diameter to the rotor tip diameter, $[(D_{Ann})_{Inner}/(D_R)_{Tip}]$	1.006	1.004	1.002	1.006	1.010	1.003
Inside diameter of the treated casing, $[(D_{TC})_{Inner}]$; [mm]	620.520	583.0	538.0	620.0	600.0	427.40 & 441.40
Ratio of the treatment inside diameter to the rotor tip diameter, $[(D_{TC})_{Inner}/(D_R)_{Tip}]$	1.229	1.152	1.061	1.228	1.212	1.055 & 1.090
Shroud Inner diameter, $[(D_{IR})_{Inner}]$; [mm]	513.080	515.0	512.0	512.0	508.0	407.40
Ratio of shroud inside diameter to the rotor tip diameter, $[(D_{IR})_{Inner}/(D_R)_{Tip}]$	1.016	1.0187	1.010	1.014	1.026	1.006
Axial length of the treated casing, $[(L_{TC})_{Axial}]$; [mm]	120.0	81.0 ¹⁰	39.50	120.00	175.0 ¹¹	23.80
Casing treatment height, (h_{TC}) ; [mm]	50.0 & 25.0 ¹²	31.0	10.0	51.0	43.0 ¹³	10.50, & 17.50
Ratio of the treatment axial length to the corresponding height, $[(L_{TC})_{Axial}]/(h_{TC})$	2.40	2.613	3.950	2.353	4.070	2.267 & 1.360
Ratio of treatment axial length to rotor tip diameter, $[(L_{TC})_{Axial}]/[(D_R)_{Tip}]$	0.238	0.160	0.078	0.238	0.353	0.059

Table (7-1)- Most important parameters of compressors and casing treatments-Continue

Names Parameters	(Azimian) [$D_H/D_I=0.5$]	(Ziabasharhagh)		(Kang) [$D_H/D_I=0.5$]	(Hill) [$D_H/D_I=0.4$]	[(Akhlaghi) [$D_H/D_I=0.7$]
		[$D_H/D_I=0.7$]	[$D_H/D_I=0.9$]			
Ratio of treatment axial length to hub to tip diameters ratio, [$(L_{TC})_{Axial}/[(D_R)_{H/T}]$]	240.0	115.714	43.889	240.0	437.50	34.0
Axial length of vane [$(L_{Vane})_{Axial}$]; [mm]	85.0	53.0	25.0	85.0	85.0	16.0
Ratio of the treatment axial length to the vane axial length [$(L_{TC})_{Axial}/(L_{Vane})_{Axial}$]	1.412	1.528	1.580	1.412	2.059	1.4875
Height of the recessed guide vane inside the casing treatment, (h_{Vane}), [mm]	56.260	37.50	15.0	56.0	50.0	13.40, 14.0 & 11.5 (10.5)
Ratio of the vane height to the vane axial length [$(h_{Vane})/(L_{Vane})_{Axial}$]	0.662	0.707	0.600	0.659	0.588	0.8375 & 0.875
Inlet passage axial length, before any modification, [$(L_{TC})_{Inlet}$]; [mm]	33.7	26.00	12.50	34.00	90.00	6.80
Ratio of inlet passage axial length to vane axial length, [$(L_{TC})_{Inlet}/(L_{Vane})_{Axial}$]	0.396	0.491	0.50	0.40	1.059	0.425
Shroud axial length, [$(L_{IR})_{Axial}$]; [mm]	30.0	21.0	10.0	30.0	30.0	7.0
Ratio of the inner ring axial length to the vane axial length [$(L_{IR})_{Axial}/(L_{Vane})_{Axial}$]	0.353	0.396	0.400	0.353	0.353	0.4375
Outlet passage axial length of treated casing chamber, [$(L_{TC})_{Outlet}$]; [mm]	56.30	34.0	17.0	56.0	55.0	10.0
Ratio of outlet passage axial length to vane axial length, [$(L_{TC})_{Outlet}/(L_{Vane})_{Axial}$]	0.662	0.641	0.680	0.659	0.647	0.625
Axial length of gap between shroud and rotor leading edge, [$(G_{TC})_{Axial}$]; [mm]	11.0 ¹⁴	13.0 ¹⁵	6.0 ¹⁶	1.0, 5.0, 7.0 & 12.0 ¹⁷	10.30, 10.20 & 20.0 ¹⁸	2.20
Ratio of axial length of gap between inner ring and rotor leading edge to vane axial length, [$(G_{TC})_{Axial}/(L_{Vane})_{Axial}$]	0.129	0.153 ¹⁹	0.240	0.012, 0.059, 0.082 & 0.141	0.121, 0.120 & 0.235	0.1375
Axial distance between vane leading edge and shroud end, [$(d_{TC})_{Axial}$]; [mm]	1.30	2.0	2.0	1.00	0.0	1.0
Ratio of axial distance between vane leading edge and end of inner ring to vane axial length [$(d_{TC})_{Axial}/(L_{Vane})_{Axial}$]	0.015	0.038	0.080	0.012	0.0	0.0625
Thickness of the shroud (e_{IR}); [mm]	3.0	3.0	3.0	3.0	3.0	3.0
Ratio of the inner ring thickness to the axial length of vane, [$(e_{IR})/(L_{Vane})_{Axial}$]	0.035	0.057	0.120	0.035	0.035	0.1875
Top view of vane inside the treatment	Partly curved	Partly curved	Partly curved	Partly curved	Partly curved	Fully curved
Recessed vane thickness, (t_{Vane}); [mm]	1.50	1.50	1.0	1.50	1.50	1.0
Diameter corresponding to the vane mid height, [$(D)_{Vane-Mid Height}$]; [mm]	564.26	545.50	523.00	564.00	550.00	427.40
Number of vanes, (Z_{Vane})	48.0 & 24.0	40.0	90.0 ²⁰	24.0	24.0	120.0

Table (7-1)- Most important parameters of compressors and casing treatments-Continue

Names Parameters	(Azimian) [$D_H/D_T=0.5$]	(Ziabasharhagh)		(Kang) [$D_H/D_T=0.5$]	(Hill) [$D_H/D_T=0.4$]	[(Akhlaghi) [$D_H/D_T=0.7$]
		[$D_H/D_T=0.7$]	[$D_H/D_T=0.9$]			
Space between two adjacent vanes at vane mid height, [(S) _{Vane-Mid Height}]; [mm]	36.931 & 73.861	42.843	18.256	73.827	71.995	11.189
Ratio of the space between two adjacent vanes at the vane mid height to the vane axial length [(S) _{Vane-Mid Height} /(L _{Vane}) _{Axial}]	0.435 & 0.869	0.808	0.730	0.868 ²¹	0.847	0.699

Table (7-1)- Most important parameters of compressors and casing treatments-Continue

- ¹ Azimian (1987) conducted his experiments on a high loading aero-engine type fan.
- ² The application of vane-recessed casing treatment with three different (0.5), (0.7) and (0.9), hub to tip diameter ratios were conducted by Ziabasharhagh (1992). The first part of the investigation was carried out on the same test rig fan, which previously was used by Azimian. The second was carried out on another compressor rig, which had the same annulus diameter.
- ³ Kang (1996) carried out his study on the same test fan, which had been used by Azimian (1987).
- ⁴ Hill (1996) was investigated the influence of a vane-recessed treatment on the performance of an industrial single rotor axial flow fan with a hub to tip diameter ratio of (0.4), and a tip speed of 76 [m/sec]. The treatment has been tested on a fan that was tip critical.
- ⁵ The variation of the recessed chamber height in relation to the hub to tip diameter ratio had been considered by Ziabasharhagh (1992) by designing two different height size treatments based on the scaling in proportion to the blade height of the machine that was used earlier by Azimian (1987). The results tended to show that scaling in proportion to the rotor chord was not an appropriate idea.
- ⁶ Due to different values of the rotor blades tip clearance for both solid and treated casings corresponding to two types of the unidirectional and reversible blades which was indicated by Hill (1998) as follows.

Casing Type	Rotor Blade Tip Clearance	
	Unidirectional Blades	Reversible Blades
Solid Casing	(1.7%-2.7%) of rotor height =2.5-4.0 [mm]	(0.8%-1.6%) of rotor height =1.2-2.4 [mm]
Treated Casing	(1.7%-2.0%) of rotor height =2.5-3.0 [mm]	(0.8%-1.1%) of rotor height =1.2-1.7 [mm]

The rotor blade tip clearance of 2.5 [mm] was chosen due to the application of the single vane-recessed casing treatment associated with the unidirectional rotating blades. It is worth noting that the rotor blade tip clearances of the other experimental works presented on the [Table (7-1)] have also been considered from their solid casing configurations.

- ⁷ Two types of the rotor blades were used. Each of them was equipped with (6) blades whose setting angles could be adjusted. The first set was standard airfoil type blades, and the second was fully reversible double circular arc profile rotor blades.
- ⁸ Since the (0.9) hub to tip diameter ratio rotor blades had suffered appreciable damage at the trailing edge during its previous running, therefore some part of the blades portion were removed by trimming away about (25%) of the chord. This action was led to a lower deflection and subsequently a lower available pressure rise from those blades than from the one of (0.7) hub to tip diameter ratio.
- ⁹ The same recessed treated casing by four different blade setting angles of (11°, 19°, 27°, and 32°) were applied over the under test rotor fan with (6) unidirectional blading. The obtained characteristic results were showed some improvement of three stagger angles of (19°, 27°, and 32°). However, it was not found any improvement in the flow range with use of casing treatment at (11°) rotor setting angle.
- ¹⁰ The recessed chamber axial length was calculated by summing up the inlet passage, the shroud and the outlet passage axial lengths.

- ¹¹ The above-noted recessed chamber axial length of 175 [mm], was calculated from the values of the inlet passage axial length of 90 [mm], the axial length of the inner ring of 30 [mm] and the outlet passage axial length of 55 [mm], as stated by Hill (1998). Nevertheless, if there was not any evidence of any mistake, he is the first researcher which used an inlet passage axial length, which was nearly (164%) greater than the axial length of the outlet passage. It is worth noting that in the majority of the previous investigations, which carried out on the area of the vane-recessed casing treatment, the axial length of the outlet passage definitely were greater than the axial length of the inlet passage.
- ¹² A further height reduction to value of 25.0 [mm], resulted in an internal flow blockage inside the vane-recessed passage. This caused a significant reducing of the flow exchange between the mainstream and the recessed flows association with a resulting tendency towards an early stalling regime.
- ¹³ As Hill (1998) stated it, the recessed chamber height was considered of 47.0 [mm]. By taking into account the specified recessed area and shroud inner ring diameters of 600 [mm] and 508 [mm], respectively as well as the inner ring thickness of 3.0 [mm], the correct height of the recessed chamber will be of 43.0 [mm]. Meanwhile the recessed height of 47.0 [mm] has to be related to a larger recessed portion inner diameter of 8.0 [mm].
- ¹⁴ The axial gap between end portion of the inner ring and rotor blade leading edge at the tip of the rotor did not clarified by Azimian (1987). Therefore it was decided to find this significant parameter by using the optimum build and its corresponding value of the rotor blade tip axial chord exposure ratio as follows.
- a)- The value of the rotor blade tip axial chord, which was exposed to the recessed chamber $[(C_R)_{AT/TC}]$, can be calculated by using the optimum value of the exposure ratio $(\lambda_{RTAC})=0.67$ and the axial chord of the rotor blade at the tip diameter of fan $[(C_R)_{Axial-Tip}]$, via following equation.

$$[(C_R)_{AT/TC}] = (\lambda_{RTAC}) * [(C_R)_{Axial-Tip}]$$

- b)- The axial length of the gap between the shroud and the rotor leading edge at the tip diameter $[(G_{TC})_{Axial}]$, then can be determined by way of subtracting the portion of the rotor blade tip axial chord exposure $[(C_R)_{AT/TC}]$ from the inlet passage axial length of the recessed casing treatment $[(L_{TC})_{Inlet}]$ as follows.

$$[(G_{TC})_{Axial}] = [(L_{TC})_{Inlet}] - [(C_R)_{AT/TC}]$$

- ¹⁵ The optimum result was achieved at about $[(G_{TC})_{Axial}]=13.0$ [mm], when the large scale casing treatment which firstly design by [Azimian] was applied into the (0.7) hub to tip diameter ratio fan. However, by application of the both medium and small sizes casing treatments and reducing the axial gap between end of the inner ring and the rotor leading edge at the tip of the rotating blades to almost zero, the losses in efficiencies were increased significantly.
- ¹⁶ With (0.9) hub to tip diameter ratio test fan and the applied medium size treated casing, the value of the axial gap between end of the inner ring and rotor blade leading edge was about $[(G_{TC})_{Axial}]=8.0$ [mm], while there was no loss in efficiency. Whereas with the application of the large scale casing configuration, the optimum value of aforementioned gap was about $[(G_{TC})_{Axial}]=4.0$ [mm].
- ¹⁷ It was reported that, when the axial gap between the shroud end and rotor leading edge at the tip diameter $[(G_{TC})_{Axial}]$, were 5.0 [mm], 7.0 [mm] and 12 [mm], the performance of the applied treated casing were almost similar. But, when the above noted gap, was reduced to 3.0 [mm] or 1.0 [mm], the early stall at the middle flow range was observed and a dip in the performance curves were occurred.
- ¹⁸ Three different values of the axial gap between the shroud end and the rotor blade leading edge at tip diameter of 10.30 [mm], 10.20 [mm] and 20.0 [mm] are corresponding to the three different values of the rotating blade setting angles of (32°, 27° and 19°), respectively.
- ¹⁹ The axial length of the vanes, which was applied inside the large scale treated casing, was of 85.0 [mm].
- ²⁰ In order to find out the desirable number of the recessed guide vane, Ziabasharhagh (1992) has used the space to chord ratio of the best Azimian's design.
- ²¹ Kang (1996) reported that the applied van-recessed casing treatment had an overall pitch to chord ratio approximately of unity.

CHAPTER EIGHT

8)- Performance of (One & Three) Stage Compressor Builds and One-Stage with and without Treated Casing Configurations:

8.1)- Introduction:

The present chapter describes all attempts that had been made to determine the performance and characteristics of the test machine configuration on a single-stage compressor builds with both the solid casing build and the casing treatment configurations.

One of the main and significant parts of the current experimental investigation is related to running of the test rig under single-stage compressor configurations with the conventional solid casing as well as the designed vane-recessed tubular-passage casing treatment.

Therefore the objective of the compressor testing was divided in two following main parts:

Firstly the tests had to be carried out in order to find the performance of the compressor using the conventional casing as a datum build, which is hereafter is called the solid casing builds. Secondly the works which was conducted in the aim of finding the characteristics of the compressor when the casing treatment was applied and henceforward is called the casing treatment builds.

The one-stage compressor builds which in all configurations included the inlet guide vanes and in the majority of the builds the two associated honeycomb flow straighteners, had been considered as solid casing or datum build in comparison with the similar builds with the treated casings.

As already mentioned, the Peregrine compressor was a low-speed machine with three repeating stages. Therefore it is quite accurate for determining the results of using the casing treatment in an individual stage. The obtained experimental results then can be considered in relation to other two stages in order to design a most appropriate casing treatment with an optimum rotor blade axial chord exposure ratio that can be applied into the compressor under three-stage builds.

Nonetheless, due to the above-noted explanation, to reduce both the casing treatment manufacturing costs and the experimental testing expenses, it was decided to test the designed treated casing only in the first stage builds with the different values of the rotor blades exposure ratios.

It is worth noting that the applied vane-recessed tubular-passage casing treatment was designed to correspond to the available room of the three-stage compressor build and therefore it was able to be used upstream of the rotor blades of each individual stage of the three-stage Peregrine axial flow compressor.

The reason behind the above-noted decision of using the treated casing downstream of the inlet guide vanes and upstream of the first stage rotor blades, was the fact that the first stage rotor blades were located in a fairly small axial distance after the inlet guide vanes.

This circumstance was the most difficult one, particularly due to the flow behaviour at lower mass flow rates which were affected by the inlet guide vanes. It appeared that although the designed casing treatment was completely able to postpone the initiation of

the rotating stall caused by the rotor blades, the inlet guide vanes lost its function in the lower flow rates.

8.2)- Compressor Geometry Specifications Under One-Stage Builds:

In the all categories of the one-stage builds, the entire outer and inner spacer rings of the compressor hub and conventional casing were used in order to build the compressor nominal attitude, but without the second and third stages moving and fixed blades.

The compressor datum configuration was built, using the original spacer rings, whereas the treated casing builds were provided by applying some of the compressor original spacers in addition to the main part of the casing treatment with the recessed guide vanes and some newly designed spacer rings. The lattes were located exactly after the inlet guide vanes and before the original spacer rings.

Except for the treated casing build with (0.232) rotor blade axial chord exposure ratio, which was built with using the spacer segment (44464), the other six similar builds with the rotor exposure ratios of (0.333, 0.434, 0.535, 0.636, 0.737 & 0.838) were constructed by applying six new manufactured spacers. The spacer rings relating to (43.4% & 53.5%) rotor exposures were manufactured from high quality acrylic, whereas the other four spacer ring were provided from aluminium alloy constructions.

The one-stage compressor assemblies with and without applications of the casing treatment configurations and the honeycomb flow straighteners can be defined with the specification of [(IGV)+(1R)+(1S)±(H)] and [(IGV)+(CT)+(1R)+(1S)±(H)], respectively.

8.2.1)- Calculation Program of the One-Stage Builds:

The calculation program that was written in order to determine the characteristics of the compressor under three-stages builds was modified to be used for finding the performance of the one-stage compressor builds with and without casing treatments.

There were small differences between these two programs, which were related to the temperature measurements. The ambient temperature (T_{Amb}) and the inlet temperature (T_{Inlet}) that the latter was measured in the front of the compressor bell-mouth were the same in both the data collections and the calculation programs.

The outlet temperature (T_{Outlet}) in three-stage compressor builds was measured downstream of the third stage stator blades, whereas for one-stage builds, it was taken in the downstream plane of the first stage stator blade cascade. The temperature upstream of the inlet guide vanes [T_{IGV-US}] was measured in all one-stage compressor builds, whereas it was omitted in three-stage builds.

Nonetheless, the Peregrine compressor was designed to provide a temperature rise of only around one Celsius per each stage.

The calculation program, which was applied to the all one-stage compressor builds with and without casing treatment configurations, is illustrated in [Appendix (B)]. The data including the results obtained from testing the compressor with (0.535) rotor tip exposure ratio casing treatment, at near the stall point corresponding to a flow coefficient of (0.352) is also presented in the above-noted appendix.

8.3)- Specification and Performance of Rotor Blade Compressor:

It was reported that the application of vane-recessed casing treatment was more beneficial for the compressor builds with isolated rotor rather than the builds which were

included stator blades as well, (Azimian, 1987).

Therefore as a part of the current study, it was essential to determine the characteristics of the rotor only compressor builds with and without using the treated casings as well as the flow-straightener honeycombs. The compressor with configuration of the isolated rotor were specified as [(IGV)+(1R)±(H)] and [(IGV)+(CT)+(1R)±(H)], respectively.

8.3.1)- Characteristics of the Rotor Only Build without Flow Straighteners:

In order to provide the rotor only compressor configuration without using the honeycomb flow straighteners known as build [(IGV)+(1R)-(H)], the compressor of three-stage build was taken apart and reassembled again with application of the inlet guide vanes and the first stage rotor blades.

The results obtained from above-noted build indicated that the compressor in this configuration was not able to work properly. This was due to the fact that the existing flow from the first stage rotor blades left the rotor in a non-axial direction. This caused the flow to approach the compressor walls and was increased the machine losses dramatically.

With very small throttle closures, the compressor was immediately dropped into the stall condition and there was no possibility to close the orifice and reduce the mass flow rate through this build. The corrected mass flow rate obtained and its corresponding flow coefficient in this condition were 1.786 [kg/sec] and 4.21, respectively.

8.3.2)- Performance of the Rotor Only Build with Air Straighteners:

In order to build the compressor under build [(IGV)+(1R)+(H)], the honeycomb flow straighteners were positioned again in the test rig ducting. In this condition the operation of the compressor showed a small improvement in respect to the build without flow straighteners. Because the machine was working relatively normally it was possible to close the throttle in order to reduce the mass flow rate through the compressor.

The data were collected, by putting the electrical switch of the end-ducting orifice in the four different positions. There was no possibility however to reduce the mass flow rates by means of throttle closing any more, since the compressor changed its status and instantaneously dropped to the stall condition. The obtained corrected mass flow rates as well as the corresponding flow coefficients in this circumstance did not alter, 1.914~1.915 [kg/sec] and 0.449, respectively.

8.3.3)- Rotor Only Builds and the Corresponding Final Decision:

The obtained flow characteristics results from both the rotor only compressor builds with and without flow straightener devices, indicated that due to the occurrence of high losses, the compressor was not able to benefit from appropriate operations.

It appeared that in the case in which honeycombs were not used, all test rig ducting in addition to the annulus downstream of the first stage rotor blades caused the loss to the flow leaving the rotor. But with the air straightener device, the flow was not able to travel in an axial direction in the annulus part of the compressor as well as in the ducting located upstream of the flow straighteners boxes. It seems the above description was the probable reason behind the observation that with the application of the flow straighteners, it was possible to close the compressor throttling.

Although it was very important to test the designed vane-recessed tubular-passage casing treatment in the Peregrine compressor under rotor only build, due to the above-noted behaviour of the compressor it was decided to continue the investigation with just stage

configurations. The results obtained from the rotor only compressor builds with and without the honeycombs flow straighteners is shown in [Table (8-1)].

It can also be concluded from the efficiencies and pressure rise coefficients presented in the above noted table that the flow had lost its energy and it was not able to discharge into the atmosphere passing through the test rig ducting. The results also indicated that due to the downstream losses, rotating stall occurred in the rotor, the instant the compressor entered into surge condition.

8.4)- Effects of Flow Straighteners on One-Stage Compressor Builds:

The performance and the function of the designed casing treatment can be found by means of a comparison procedure. This activity can be applied to the characteristics of the one-stage compressor builds, assembled using the datum build and the treated casing configurations. In order to provide the aforementioned facility, the characteristics of the datum build have to be specified as part of the preliminary activities.

As described in chapter five, the application of the honeycomb flow straighteners had fairly significant effect on the performance of the three-stage compressor builds. It was also observed that using the air straighteners in rotor only compressor builds facilitated machine operation slightly.

Therefore in order to make an appropriate decision on whether the honeycomb flow straighteners were to be used in the one-stage compressor builds or not, it was essential to test the compressor in this configuration once more with and without the application of air straightener devices. By using the honeycomb inside the flow straighteners, small alterations in some flow characteristics were found.

It is worthy to mention that due to testing area requirement, the Peregrine test rig including its corresponding equipment were moved from the "Test House 3" into the "Aerodynamic laboratory". Considerable efforts were made to set up the test rig and the relevant measuring devices, exactly as in the previous location.

The one-stage compressor builds with and without air straighteners had been tested more than eleven times in the different times. The overalls and individual stage characteristic results obtained from testing the compressor builds of [(IGV)+(1R)+(1S)±(H)] with and without the influence of using the honeycombs inside the flow straighteners boxes are presented in [Table (8-2)] and [Table (8-3)], respectively.

8.4.1)- Influence of Honeycomb Air Straighteners on the Flow Rates:

When the test was carried out without the air straightener honeycombs the flow rates decreased by (15.76%) and (3.91%) at the choke and near the stall conditions, respectively, relative to the three-stage compressor build. Moreover, in the case, in which the honeycombs were located in the test rig, the reduction of the flow rate of the one-stage build with respect to the three-stage build at the condition of full throttle opening was (16.89%), whereas at near stall circumstance it showed a (2.39%) drop off.

The characteristics of the corrected mass flow rates versus corresponding flow coefficients obtained from one-stage compressor builds that were tested with and without honeycomb air straighteners are shown in [Graph (8-1)]. The results also indicated that with the application of the honeycombs the mass flow through the one-stage compressor builds decreased by small percentages of (0.42%) at the choke conditions and increased of (1.6%) at near stall conditions.

Meanwhile, in the throttle fully open condition the corrected mass flow rates of the build with the honeycombs air straighteners showed a reduction of (1.66%) with respect to the case in which the compressor was run without the honeycomb flow straighteners.

However, near the stall condition the build with air straighteners was slightly better, since there was a small increment of (0.55%) in the corrected mass flow values. As can be observed from both the above graphs, the one-stage compressor build with the honeycomb flow straighteners was able to provide slightly better mass flow characteristics.

It can also be deduced that similarly to the three-stage build, the one-stage compressor build at the choke condition and particularly, with the higher mass flow, the system performed better when the honeycombs as absent. However as the mass flow through the compressor was reduced more losses were created. In such circumstances the use of honeycomb air straighteners was found to be more desirable since in this condition the compressor was able to continue operating with slightly higher flow rate.

8.4.2)- Effects of the Air Straighteners on Corrected Applied Powers:

The corrected applied power ($\Omega\epsilon$) of the one-stage compressor build at the fully choked condition with application of the honeycombs inside the flow straightener boxes was (0.98%) higher than without using the honeycombs. This result changed near the stall conditions, because the obtained corrected power with use of the honeycombs decreased by (0.75%). Similar behaviour had also been seen in the three-stage compressor builds.

However, the maximum values of the corrected applied power for the one-stage builds with and without air straightener devices were 2.846 and 2.844 [kwatt]. The corresponding corrected mass flow rates for the above values were 2.24 and 2.26 [kg/sec], respectively.

The characteristics of the corrected applied power in one-stage compressor builds with and without use of the honeycomb flow straighteners is illustrated in [Graph (8-2)]. As it can be seen from this graph there are not significant differences between characteristics obtained from builds with and without flow straightener devices.

8.4.3)- Influence of the Flow Straighteners on the Work Coefficients:

The characteristic of the overall work coefficients obtained from the one-stage compressor builds showed that at the choke condition the build with the honeycomb air straighteners fitted was able to provide (3.53%) higher work coefficient relative to the build without them. However at the near stall condition the build without the air straighteners performed slightly better and produced about (1.3%) higher work coefficient.

For the build without the honeycomb air straighteners the highest value of the overall work coefficient was about (0.509) with a 1.94 [kg/sec] corrected mass flow rate. Meanwhile, when the flow straightener devices were used, the maximum work coefficient and its corresponding corrected mass flow rate were found to be (0.502) and 1.95 [kg/sec], respectively.

8.4.4)- Effects of the Honeycomb Flow Straighteners on the Efficiencies:

The overall efficiency characteristics in terms of the total-total pressure ratio obtained from running the test rig with the inlet guide vanes and just first stage rotor and stator blades, showed fairly different characteristics when the honeycomb air straighteners were used and in the case which was examined without them.

With the application of the honeycombs inside the air straightener boxes, the overall total-total efficiency was decreased by (4.56%) in the choke status and increased by (1.67%)

near the stall condition. Moreover, by applying the honeycombs flow straighteners, the point of maximum efficiency was shifted to a lower flow rate, in association with a decrement of (1.82%).

When the honeycombs as applied the overall total-total efficiency and the corresponding corrected mass flow rate were (0.747) and 2.202 [kg/sec], respectively. Meanwhile without the air straighteners they increased to (0.761) and 2.348 [kg/sec], respectively. The efficiency characteristics in terms of the overall static-static pressure ratios indicated that the one stage compressor configuration with the air straighteners showed (0.65%) and (1.51%) higher static-static efficiency in choke and near stall conditions respectively.

The build without the honeycomb flow straighteners was able to provide a higher maximum static-static efficiency of (0.662) corresponding to 2.16 [kg/sec] corrected mass flow rate. However with the air straightener devices this efficiency was reduced by (1.14%) to (0.655) with an associated 2.08 [kg/sec] corrected mass flow rate.

The overall total-static efficiency characteristics showed that in the majority one-stage compressor cases, the build without the honeycomb air straighteners performed a little better than the build with the honeycombs. These results however were reversed at the choke and near the stall conditions.

It is worth noting that the build with the flow straighteners provided a total-static maximum efficiency of (0.434) in corresponding to 1.95 [kg/sec] corrected mass flow rate. Meanwhile, the configuration without the honeycombs was able to achieve a maximum total-static efficiency of (0.425), which was (2.07%) less than the build with the honeycombs with a similar correspondence corrected mass flow rate of 1.94 [kg/sec].

The first stage efficiency characteristics calculated in terms of the static-static pressure ratios were also similar to the overall static-static ones for both with and without air straightener devices.

Nonetheless, the configuration with the honeycombs provided (1.80%) and (2.08%) higher efficiency in terms of the stage static-static pressure ratio at the choke and near stall conditions, respectively. However, the maximum stage efficiency of the build without air straighteners was (2.04%) higher than which with the honeycombs. The stage maximum efficiencies for the builds with and without the honeycombs were (0.704) and (0.719), respectively. These efficiencies corresponded to the corrected mass flow rates of 2.262 [kg/sec] and 2.233 [kg/sec], respectively.

It is worth noting that the one-stage compressor build without the flow straighteners had a corrected mass flow rate range of 0.664 [kg/sec], whereas without the honeycombs, this range was reduced to 0.610 [kg/sec]. The characteristics of the overalls and stage efficiencies of the one-stage compressor configurations with and without the honeycomb flow straighteners, builds [(IGV)+(1R)+(1S)±(H)] are illustrated in [Graph (8-3)].

8.4.5)- Influence of the Air Straighteners on the Pressure Rise Coefficients:

In a similar manner to the characteristics of the pressure rise coefficient obtained from three-stage compressor builds, the overalls and stage characteristics in terms of total-total, static-static and total-static pressure ratios of the one-stage configurations that were examined with and without honeycomb flow straighteners also provided reasonably consistent results.

At the fully open throttle condition the all overall and stage pressure rise coefficients were higher for the compressor build without the honeycomb air straighteners except for the case calculated in terms of the overall total-total pressure ratio. Meanwhile, the peak

points of the overalls and stage pressure rise coefficient for both of the one-stage build, with and without the honeycomb flow straighteners occurred at near the stall conditions.

The overall pressure rise coefficient characteristics, as with the efficiency ones, showed that in the all cases, the builds without the honeycomb air straighteners provided slightly lower pressure ratios when the orifice was fully opened. But at the near stall condition the one-stage compressor build with the flow straighteners definitely performed better than that without use of honeycombs.

The overall total-total pressure rise coefficient for the build with air straightener devices was (1.52%) lower than the compressor build without honeycombs at the choke condition. This however was changed by reduction of the flow through the test rig. At near stall conditions the build with flow straighteners performed slightly better and gave a higher overall total-total pressure rise coefficient of (0.368) which was (0.65%) higher than that of the build without honeycombs.

The overall pressure rise coefficient in terms of static-static pressure ratio was slightly better for the compressor build with the air straighteners relative to the one without the honeycombs. The overall static-static pressure rise coefficients with the flow straightener devices were higher by (4.42%) and (0.52%) at the choke and near stall conditions, respectively.

The overall total-static pressure rise coefficient characteristics indicated that the one-stage compressor build including the honeycombs showed slightly better performance in comparison to that without them. The build with air straighteners also provided a higher (0.29%) peak pressure rise coefficient.

The stage pressure rise coefficient characteristics, determined in terms of the static-static pressure ratio indicated that at both ends of this characteristics (choke and near the stall conditions) the compressor build with the honeycomb flow straighteners was able to provide (5.44%) and (0.97%) higher pressure rise coefficient.

The overall and stage pressure rise coefficients of the one-stage compressor configurations with and without the honeycombs flow straighteners, builds [(IGV)+(1R)+(1S)±(H)] are shown in [Graph (8-4)].

8.4.6)- Application of the Honeycomb Flow Straighteners-Final Decision:

Previously described the applied honeycombs were hexagonal-shape and aluminium-type with an aspect ratio of (5.5). The hexagonal holes were of 6.5 [mm] across the flats and 35.75 [mm] depth. The use of the honeycomb flow straighteners was found to be necessary in order to straighten the flow before entering into the test rig downstream ducting at higher flow rates. However, it was felt that it was also beneficial at lower mass flow rates.

It is worth noting that in both cases (three-stage and one-stage compressor builds) the flow left the last stator blades in an axial direction. Therefore it was not easy to make an appropriate decision in the preliminary stages on whether the use of the honeycomb air straighteners would be beneficial in respect to system losses.

Meanwhile the testing of the compressor with first stage rotor blade and the inlet guide vanes was indicated that the application of the honeycomb flow straighteners can facilitate flow passage through the test rig ducting in an appropriate axial direction. This also was confirmed from the flow characteristics of the one-stage compressor builds, particularly at the lower mass flow rate and near the stall conditions.

On the other hand the main objective of the current investigation is to use a tubular-passage vane-recessed casing treatment in order to facilitate the compressor operating in the lower range of the mass flow rates. Nonetheless, based on the above information in addition to this conclusion, which was drawn in chapter five, it can be recommended that the applications of the air straightener devices are essential.

Therefore it was decided to use the honeycomb inside the air straighteners devices in all experimental investigations including the datum (solid casing) and comparing (treated casing) builds, that were going to be conducted in the Peregrine compressor test rig.

8.5)- Alteration in Performance of One-Stage and Three-Stage Builds:

The design features of the Peregrine compressor indicated that this compressor was originally designed as a multistage test compressor. As previously described, in order to reduce the manufacturing expense, it was decided that the designed casing treatment, would be used to the one-stage compressor build over the first stage rotor blades.

On the other hand in order to design an appropriate vane-recessed tubular-passage casing treatment the characteristics of the one-stage compressor build firstly had to be determined. Although alterations in the characteristics of the one-stage compressor build in respect to the three-stage one were expected, there was no any evidence of how large the variation would be. Therefore it was decided to run the compressor with two configurations; [(IGV+1R+1S+2R+2S+3R+3S)+(H)] and [(IGV+1R+1S)+(H)], respectively.

These parts of the investigation were also carried out when the rig was located inside the "Test House 3". The flow characteristics of the one-stage and the three-stage compressor builds before relocation of the test rig in the aerodynamic laboratory are illustrated in [Table (8-4)] and [Table (8-5)], respectively.

In order to provide better understanding of the above-noted compressor builds, some of the flow characteristics results related to particular points such as the stall and the peak points of the one-stage and the three-stage compressor configurations are additionally presented in [Table (8-6)] and [Table (8-7)], respectively.

8.5.1)- Difference in the Nominal and the Corrected Mass Flow Rates:

The mass flow rates through the one-stage build configurations were dramatically reduced in comparison with the three-stage compressor. The corrected mass flow rate also showed very similar characteristics to that of the uncorrected flow rates.

The maximum mass flow rate through the compressor increases when the throttle was completely opened. In this condition, both the nominal and the corrected mass flow rates of the one-stage build were reduced in comparison to those of three-stage compressor configuration by (15.85%) and (15.96%), respectively. Moreover the minimum values of the flow rates occurred near the stall conditions.

The nominal and corrected mass flow rates of the one-stage compressor configuration were less than (0.39%) and (0.77%) respectively, for those of the three-stage build. Meanwhile the three-stage compressor configuration was able to cover a flow coefficient range of (0.253), whereas this range showed a (43.07%) reduction and was about (0.144) for the one-stage build.

The alteration of the corrected mass flow rates versus corresponding flow coefficients, obtained from the flow characteristics of the three-stage and one-stage compressor builds are illustrated in [Graph (8-5)].

8.5.2)- Changes in the Corrected Applied Powers:

As was expected the corrected applied power that was necessary for running the test rig under one-stage compressor build was significantly lower than that needed for the three-stage configuration. When the throttle was fully opened, the corrected applied power of the three-stage compressor build was (43.35%) higher than that of one-stage build. This situation was maintained and near the stall condition was increased to about (59.6%).

The maximum values of the corrected applied powers and their corresponding corrected mass flow rates for one-stage and three-stage compressor builds were (6.91 & 2.83) [kwatt], and (2.214 & 2.207) [kg/sec], respectively which presents an increase of (59.04%) for the latter. As can be seen the corrected mass flow rates corresponding to the maximum points of the corrected applied powers for both one-stage and three-stage compressor configurations are very close to each other.

8.5.3)- Alteration in the Overall Work Coefficients:

The maximum overall work coefficients for both three-stage and one-stage compressor configurations occurred at near the stall conditions. The work coefficient of the one-stage compressor was always less than that of the three-stage build. At the choke condition the relevant work coefficient of one-stage compressor was about (29.87%) less than that of the three-stage build. At near stall conditions, the change was about (58.79%). The alterations in the overall work coefficients corresponding to the flow characteristics of the three-stage and the one-stage compressor configurations are shown in [Graph (8-6)].

8.5.4)- Differences in Overall Total-Total Efficiencies and Pressure Rises:

In all cases the efficiencies in terms of the overall total-total pressure ratios are higher for the three-stage compressor build. When the throttle was totally opened the one-stage compressor configuration provided (7.71%) less overall total-total efficiency. This however increased to (10.15%) near the stall condition.

Meanwhile, the maximum values of this efficiency and their corresponding corrected mass flow rates for the one-stage and three-stage compressor builds were determined to be about (0.755 & 0.855) and (2.237 & 2.438) [kg/sec], respectively. This meant that the three-stage compressor configuration was able to provide (11.65%) higher overall total-total efficiency in comparison with just one-stage. Changes in the efficiency characteristics in terms of the overall total-total pressure ratio corresponding to the flow characteristics of three-stage and one-stage compressor configurations are presented in [Graph (8-7)].

The peak values of the overall total-total pressure rise coefficients for both the three-stage and one-stage compressor builds occurred at near the stall conditions. In this circumstance the three-stage build was able to provide (63.05%) higher value of the peak pressure rise coefficient. However, in the fully choked condition the pressure rise coefficient of the one-stage compressor configuration was (34.85%) less than that of the three-stage build. Alteration of the overall total-total pressure rise coefficients that were achieved from the flow characteristics of the three-stage and one-stage compressor configurations are illustrated in [Graph (8-8)].

8.5.5)- Changes in Overall Static-Static Efficiencies and Pressure Rises:

The flow characteristics obtained from the examination of three-stage and one-stage compressor configurations showed that the latter one performed slightly better about (6.88%) in the fully choked condition. But near stall circumstances the three-stage build (14.46%) was more efficient than the other compressor build.

Meanwhile the maximum values of the overall static-static efficiencies and their corresponding corrected mass flow rate for the three-stage and one-stage compressor builds were (0.798 & 0.657) and (2.298 & 2.148) [kg/sec], respectively. This meant that the complete compressor with three stage builds was able to provide (17.67%) higher maximum efficiency in terms of the static-static pressure ratio. In all cases the overall static-static pressure rise coefficients of the three-stage compressor build were higher in comparison with the one-stage configuration.

When the throttle was completely opened, about (21.93%) higher overall static-static pressure rise coefficient was achieved with three-stage configuration. Near stall points this value increased to (65.17%). It is worth noting that in the both compressor configurations the peak pressure-rise coefficients occurred at near the stall conditions.

The alterations of the efficiencies and the pressure rise coefficients of the three-stage and the one-stage compressor configurations that were determined in terms of overall static-static pressure ratios are presented in [Graph (8-9)] and [Graph (8-10)], respectively.

8.5.6)- Alterations in Overall Total-Static Efficiencies and Pressure Rises:

The maximum overall total-total efficiency of the one-stage compressor build occurred at near stall condition, but that of three-stage configuration took place before the stall with a maximum of (0.674), corresponding to a corrected mass flow rate of 2.164 [kg/sec]. The corrected mass flow rates obtained at near stall conditions from the three-stage and the one-stage builds, were 1.967 and 1.952 [kg/sec], respectively. Moreover the three-stage configuration was able to provide (38.3%) higher value in the maximum overall total-static efficiency in comparison to that of the one-stage build.

On the other hand at the choke conditions the one-stage compressor build performed slightly better with (15.46%) higher efficiency, calculated in terms of the overall total-static pressure ratios. However, this changed at near stall condition, as the three-stage configuration was provided (35.454%) higher in the above-noted efficiency. The changes in the overall total-static efficiencies from the relevant flow characteristics of the three-stage and the one-stage compressor configurations are presented in [Graph (8-11)].

The peak pressure rise coefficients in terms of the overall total-static pressure ratio for both compressor configurations occurred at the near stall condition. Meanwhile the three-stage build was able to attain (74.12%) higher peak pressure rise coefficient in comparison with the one-stage build. Alterations of the overall total-static pressure rise coefficients obtained from the flow characteristics of the three-stage and the one-stage compressor configurations are presented in [Graph (8-12)].

8.5.7)- Difference in Stage Static-Static Efficiencies and Pressure Rises:

The first stage flow characteristics of the three-stage and one-stage compressor configurations were significantly different from those of the overall ones. In this case the one-stage build showed higher values in both efficiency and pressure rise coefficient in comparison with the three-stage configuration.

When the compressor throttle was fully opened, the one-stage compressor build provided (296.2%) higher efficiency in terms of the stage static-static pressure ratio, comparing to the similar result obtained from the first stage characteristic of the three-stage compressor build. This was nearly halved close to the stall conditions. The three-stage compressor in this circumstance produced (152.56%) less stage static-static efficiency in comparison with the one-stage build.

The maximum stage static-static efficiencies and their corresponding corrected mass flow rates for three-stage and the one-stage compressor configurations were (0.281 & 0.714) and (0.248 & 2.287) [kg/sec], respectively.

The peak point of the first stage static-static pressure rise coefficient for the three-stage compressor build occurred at near stall point, whereas, for the one-stage compressor configuration took place before the stalling point. The peak pressure rise coefficients of the three-stage and the one-stage builds in association with their corresponding corrected mass flow rates were (0.332 & 0.343) and (1.967 & 2.009) [kg/sec], respectively.

The first stage pressure rise coefficient of the three-stage compressor build was very small in the choked condition. Therefore the one-stage compressor configuration was able to provide more than a (197.31%) higher pressure-rise coefficient under this circumstance. Moreover, near the stall condition, the first stage pressure rise coefficient of the one-stage compressor build was just (2.9%) higher than that of three-stage configuration.

The alterations of the efficiencies and the pressure rise coefficients of the three-stage and the one-stage compressor configurations that were determined in terms of first stage static-static pressure ratios are illustrated in [Graph (8-13)] and [Graph (8-14)], respectively.

8.6)- Characteristics of the Solid Casing (Datum) Build:

As indicated earlier, due to the flow characteristics obtained from the three-stage, rotor-only as well as the one-stage compressor builds, it was decided to design a vane-recessed tubular-passage casing treatment in aim of improving the performance of the one-stage solid casing configuration.

To meet the investigation objective, it was essential to examine the Peregrine compressor in a one-stage configuration as a solid casing, build [(IGV)+(1R)+(1S)+(H)]. In order to achieve this the compressor was dismantled and reassembled again, using the blade setting angles and the spacer rings with the category of the compressor nominal attitude.

It is worth noting that in order to find the flow characteristics of the datum build, the compressor with the solid casing was dismantled and reassembled four different times. The test examinations were also conducted more than seven times and on different days, which the ambient temperature varied widely. Some of these runs were carried out when the test rig was located in the Test House 3, but the majority of examinations were conducted, when the test rig was relocated into the Aerodynamic Laboratory.

The characteristic results obtained were then compared to each other with a high degree of care. The problems encountered then were solved one by one. Finally, the results, which were obtained from the last test of the above build, were taken as the baseline results for the solid casing compressor configuration.

The flow characteristics of the solid casing, which has been used as datum configuration in the comparison procedures to the results obtained from the treated casing configurations are illustrated in [Table (8-8)] and [Table (8-9)], respectively. In this configuration the highest value of the mass flow rate, about 2.534 [kg/sec], which corresponded to a flow coefficient of (0.592) occurred when the throttle was fully opened.

The relevant mass flow rate, which was achieved near the stall condition, was 2.534 [kg/sec]. Meanwhile when the flow coefficient of the datum build was less than (0.457), the compressor experienced an abrupt stall. It is worth noting that the solid casing build was able to cover the corrected mass flow rate across a range of 0.579 [kg/sec] from choke to near stall conditions.

In these circumstances, it was not possible to continue for data collections; the occurrence of the rotating-stall caused a loud noise and immediate deterioration in the compressor performance. The behaviour of the solid casing also showed that without using the casing treatment, the datum build was not able to continue normal operation when the flow coefficient decreased to a value less than (0.46). This was confirmed throughout the entire examination, of the solid casing compressor configuration.

The minimum value of the corrected applied power that had been used at the choke condition was about 2.706 [kwatt]. This parameter increased as the flow coefficient reduced. The maximum corrected applied power of 2.884 [kwatt] occurred at a corrected mass flow rate of 2.173 [kg/sec]. This meant that the one-stage compressor builds in the absence of the rotor and stator blades of the other two stages was reasonably able to act as a single-stage compressor as well.

The characteristics of the corrected applied power and overall work coefficient, obtained from the testing of the one-stage solid casing compressor build versus the corrected mass flow rate, including their corresponding equations are presented in [Graph (8-15)] and [Graph (8-16)], respectively.

At the choke condition the efficiency in terms of the total-total pressure ratio showed a higher value of (0.715) in comparison to that of (0.679) obtained near the stall condition. The maximum overall total-total efficiency achieved was about (0.746) with a corrected mass flow rate of 2.297 [kg/sec].

It can be concluded from the characteristics of the overall static-static and total-static efficiencies that these values in some cases were different in comparison with the overall total-total ones. For example, except the latter efficiency, the minimum values of the other two above-noted overall efficiencies were achieved when the throttle was fully opened. The minimum values of both overall static-static and total-static efficiencies at the choke condition were (0.555) and (0.079). Meanwhile at the near stall condition corresponding to the corrected mass flow rate of 1.952 [kg/sec], these values were determined as (0.615) and (0.386), respectively.

Conversely, the maximum values of both overall static-static and total-static and their corresponding corrected mass flow rates were (0.639 & 0.386) and (2.154 & 1.964) [kg/sec], respectively. This meant however that the maximum overall total-static efficiency occurred fairly close to the stall point.

The stage static-static efficiency showed a characteristic similar to that one of the overall total-total one, but with slightly lower values. These magnitudes at the choke and near stall conditions were (0.672) and (0.655) respectively. The maximum value of the stage static-static efficiency and its related corrected mass flow rate were (0.706) and 2.271 [kg/sec], respectively.

The characteristics of the overall and stage efficiencies of the one-stage solid casing compressor build versus the corrected mass flow rate including the relevant trendlines are illustrated in [Graph (8-17)]. It can be concluded from the above-noted characteristics that the maximum points of all four efficiencies occurred across a fairly small range of corrected mass flow rates, about 1.964-2.297 [kg/sec].

The wall static pressure measurements and the location of the total-total pressure probe caused these small differences. It is worth noting that it was better to have a plenum in order to bring the flow to rest and measure the static pressure. This however was not possible due to the situation of the test rig.

From another point of view, since the calculated characteristics of the compressor configurations with and without casing treatment were going to be compared with each other, therefore such insignificant differences which were imposed by the test rig facilities would cancel out and be negligible. The characteristics of the overall and stage pressure rise coefficients are slightly different from those of the efficiency ones.

When the throttle was completely opened the overalls and stage pressure rise coefficients showed their corresponding minimum values. Meanwhile, the peak points of pressure-rise coefficients, both overall static-static and overall total-static occurred near the stall condition. These values were (0.325) and (0.205) and their corresponding corrected mass flow rate was 1.952 [kg/sec].

The peak points of the overall total-total and the stage static-static pressure rise coefficients and their corresponding corrected mass flow rates were (0.36 & 0.346) and (2.026 & 1.987) [kg/sec], respectively. The pressure rise coefficient characteristics of the one-stage datum build compressor versus the corrected mass flow rate including their corresponding equations are presented in [Graph (8-18)]. As it can be concluded from the latter graph the trendlines of all four types of the pressure rise coefficients were very similar.

8.7)- Performance of the Casing Treatment Configurations:

In order to fulfil the current investigation objective, it was necessary to run the one-stage compressor build with the inlet guide vanes, along with the designed casing treatment, the appropriate rotor blade exposure spacer ring in addition to the original solid rings and the honeycomb flow straighteners.

Due to the significant restriction of the available axial distance inside the test compressor it was not possible to examine the compressor with a stepped casing followed the spacer rings that facilitated a rotor blade tip axial chord percentage ratio less than (23.2%). This was explained in more detail in the previous chapter.

In order to reduce the expense necessary in the manufacturing of the new spacer rings for the test running, it was decided to use seven different rotor blade axial chord exposure ratios between (23%) and (84%) in approximately (10%) increments. The rotor tip exposure ratio should not only be considered as a compressor related parameter, but also has to be thought of as one of the most significant parameters affecting the overall performance of the casing treatment significantly.

The above decision however had also been based partially on the fact that the majority of researches, conducted in the field of the vane-recessed casing treatment implied that the best results were achieved in a range of (50% to 70%) of the rotor tip axial chord exposure. However, the test rig was examined by using the spacer rings which enabled seven different rotor tip exposures of 23.2%, 33.3%, 43.4%, 53.5%, 63.6%, 73.7% and 83.8%, respectively.

It is worth noting that the vane-recessed tubular-passage casing treatment including two new spacer rings, which were designed for the current investigation. These were manufactured from high quality Acrylic by EDM LIMITED (a world class providers of engineering solutions), Lancashire, England, UK.

Two aforementioned spacer rings enabled the rotor tip axial chord to be exposed at about 8.2 [mm] and 10.45 [mm], respectively. These values corresponded to casing treatments of (0.414) and (0.527), rotor tip exposure ratio, respectively. Moreover, it was decided that before trimming these spacer rings, the test rig would be run with the above exposure ratios as the preliminary stage of the compressor configuration with the casing treatments.

Therefore the test compressor was examined at least two times using each spacer rings on different days. Moreover, the obtained flow characteristics were analysed in detail, with improvements in the stall margin being quite significant in both cases. It was also found that at lower flow rates the abrupt stall, which occurred in both the datum build and the treated configuration with (23.2%) rotor exposure was changed to a progressive stall.

Nonetheless, the initial results indicated that it was necessary to test the compressor and the casing treatment with the application of more spacer rings that could provide additional values of the (43.4% & 53.5%) rotor axial chord exposures.

It is worth mentioning that during data accusation procedures and running of the one-stage compressor configurations for the datum solid casing and the treated casing configurations, different problems occurred. Some of these, lead to a compressor shutdown whereas some others just imposed some difficulties in obtaining accurate characteristics. These problems will subsequently be discussed in more detail.

8.7.1)- Characteristics of (0.232) Rotor Exposure Ratio Casing Treatment:

The one-stage compressor in this configuration was built with the inlet guide vanes, the casing treatment with rotor tip axial chord exposure of (23.2%), first stage rotor and stator blades followed by the honeycomb flow straighteners. This build that did not have any new spacer ring after the main part of the casing treatment, hereafter specified as [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)].

In order to configure the test rig in the above configuration, the compressor needed taken apart and reassembled. This activity was conducted five times; moreover, the compressor test rig was examined on more than six different times and days. After solving the all the problems encountered, the results obtained from the final run were found to be the most acceptable one. The flow characteristics including the results related to the peak or near stall points of the one-stage compressor in association with casing treatment configuration of (23.2%) rotor tip exposure, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)] are illustrated in [Table (8-10)] and [Table (8-11)], respectively.

At the choke condition this compressor configuration was able to sustain the highest mass flow rate of 2.529 [kg/sec]. The corresponding flow coefficient in this circumstance was found to be (0.593). It meant that when the throttle was fully opened, the one-stage compressor with casing treatment of (0.232) rotor exposure ratio was able to sustain just (0.2%) greater mass flow rate than the datum build.

Moreover near the stall condition, the configuration with casing treatment was able to postpone the initiation of stall to a flow coefficient of (0.358), which corresponds to a mass flow rate of 1.52 [kg/sec]. Therefore it can be concluded that the casing treatment with (23.2%) rotor tip exposure provided the possibility to run the test rig with (21.9%) lower mass flow rate than the datum build.

This configuration was able to cover a corrected mass flow rate range of 1.01 [kg/sec] from choke to near stall conditions. However, at near the stall condition the corrected mass flow rate lowered to a value of 1.534 [kg/sec]. There was no possibility in this circumstance to close the compressor throttle any further in order to reduce the flow rate through the test rig, due to occurrence of abrupt stall in association with heavy noise; the performance of the test compressor also dropped sharply.

The minimum value of the corrected applied power of 2.717 [kwatt] was determined when the test rig flow control throttle in the one-stage compressor with (23.2%) rotor exposure was completely opened. When the corrected mass flow through the compressor was

reduced to a value of 2.177 [kg/sec], the corrected applied power was increased to a maximum value of 2.916 [kwatt].

Moreover with further closing of the rig discharge orifice, both the corrected mass flow rate and the corrected applied power were reduced to values of 1.654 [kg/sec] and 2.781 [kwatt], respectively. But beyond this point, further mass flow reduction lead to an increase in corrected applied power.

It seems that at around this point the initiation of the stall cell occurred, but the compressor was able to continue normal operation. This situation continued near to the stall point where the corrected applied power was determined to be 2.817 [kwatt] corresponding to the corrected mass flow rate of 2.79 [kg/sec].

Nonetheless, it was not possible to have a flow coefficient less than (0.358) with the above-noted compressor build. This was due to the action of stall cells, which caused the compressor to enter an immediate rotating stall with a sharp drop in the machine performance in association with loud noise.

It is worth noting that the trendlines for the corrected applied power characteristic of the one-stage compressor with casing treatment of (23.2%) rotor exposure was significantly different in some areas in comparison with that of the solid casing one. The overall work coefficient of this build showed a normal characteristic. At the choke condition the minimum value of (0.380) was corresponded to a flow coefficient of (0.593). Meanwhile, the maximum value (0.658) of the work coefficient (overall) occurred near the stall condition with a corresponding corrected mass flow rate of 1.534 [kg/sec].

The characteristics of the corrected applied power and the overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment of (0.232) rotor tip exposure ratio, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)] are illustrated in [Graph (8-19)] and [Graph (8-20)], respectively. The corresponding equation fits are also presented in these graphs.

The overall and stage characteristics of the compressor with casing treatment of (23.2%) rotor tip exposure indicated that this configuration performed significantly better than that of the solid casing (datum build) one. Except for the overall total-static efficiency, the overalls and stage efficiencies were higher when the compressor throttle was fully opened in comparison with those achieved at the near stall condition. Meanwhile by reducing the mass flow inside the compressor, all characteristics of the overall and stage efficiency were increased to their corresponding maximum.

The overall total-total, overall static-static and stage static-static efficiencies at the choke condition were found to be (0.726, 0.559 & 0.674), respectively, corresponding to a flow coefficient of (0.593). The maximum overall total-total, static-static and total-static efficiencies and their corresponding corrected mass flow rates were (0.738, 0.628 & 0.406) and (2.365, 2.145 & 1.835) [kg/sec], respectively. Nonetheless the stage efficiency characteristic trendline was very similar to that of the overall total-total one.

The maximum value of this efficiency was determined to be about (0.628), corresponding to a corrected mass flow rate of 2.302 [kg/sec]. The minimum value of the overall total-static efficiency of (0.084) occurred at the choke condition, corresponding to a corrected mass flow rate of 2.548 [kg/sec]. The overall total-static efficiency at the near stall condition was determined to be (0.348) corresponding to a flow coefficient of (0.358).

The minimum magnitudes of the overall total-total, overall static-static and stage static-static occurred near stall condition with the values of (0.5, 0.469 & 0.495), respectively,

corresponding to a mass flow rate of 1.52 [kg/sec]. The overall and stage efficiency characteristics versus the corresponding corrected mass flow rate of the one-stage compressor with casing treatment of (0.232) rotor tip exposure ratio, configuration [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)] are shown in [Graph (8-21)].

The characteristics of the overall and the stage pressure rise coefficients showed that the minimum values of the coefficients occurred at the choke condition. Each of these characteristics increased to its peak points as the mass flow through the compressor reduced. But with further reduction of flow coefficient, it dropped again to near stall condition.

The overall total-total, static-static and total-static efficiencies including that of the stage static-static one at both choke and near stall conditions corresponding to their related mass flow rate of (2.529 & 1.52) [kg/sec] are (0.280, 0.214, 0.029 & 0.259) and (0.327, 0.307, 0.233, 0.323), respectively.

The peak pressure rise coefficients pertaining to the overall total-total, stage static-static, overall static-static and the overall total-static pressure ratios and their corresponding corrected mass flow rates were calculated as (0.359, 0.346, 0.326 & 0.234) and (1.927, 1.879, 1.829 & 1.616) [kg/sec], respectively. The overall and stage pressure rise coefficients characteristics against the corresponding corrected mass flow rate of the one-stage compressor with casing treatment of (0.232) rotor tip exposure ratio, configuration [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)] are presented in [Graph (8-22)].

8.7.2)- Characteristics of (0.333) Rotor Exposure Ratio Casing Treatment:

The one-stage compressor in association with the casing treatment configuration of the (0.333) rotor blade tip axial chord exposure ratio was built using the inlet guide vanes, the casing treatment followed by a new aluminium alloy spacer ring, first stage rotor and stator blades including the honeycomb flow straighteners. This build henceforward is identified by the specification of [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)].

The compressor was dismantled and reassembled again based on the required geometry and specifications of the above configuration. The investigation necessitated testing the one-stage compressor at five different times on different days. Therefore, in order to meet these requirements, the compressor test rig was taken apart and rebuilt on five different occasions.

The results was then calculated and difficulties were removed one by one. It was found that the final flow characteristics results obtained, were the most accurate and reasonable. Some results showing the flow characteristics of the one-stage compressor in association with casing treatment configuration of (33.3%) rotor tip axial chord exposure are illustrated in [Table (8-12)].

The other results relating to the peak points or the near stall condition of the build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)] are also presented in [Table (8-13)]. The above-noted compressor build was one of the best configurations in terms of providing a significant stall margin improvement. The mass flow rates at the choke and near the stall conditions were calculated as (2.498 & 1.373) [kg/sec], respectively. The corresponding flow coefficients in these circumstances were (0.593) and (0.325), respectively.

This configuration of casing treatment with (33.3%) rotor tip axial chord exposure ratio was capable of operating with (28.89%) lower mass flow rate in comparison with the datum build. It also covered a range of 1.146 [kg/sec] corrected mass flow rate. The minimum value of the corrected mass flow rate at the near stall condition was determined

to be 1.394 [kg/sec]. With very small further reduction of the flow through the machine, this compressor build experienced a stall phenomenon.

It is worth noting that the compressor builds with rotor tips axial chord exposure ratio of (0.333) performed quite better in comparison with the build of (0.232) exposure ratio ones. This was because, at lower mass flow rates the latter drooped to an abrupt stall rather than a progressive stall. For the configuration with the (33.3%) exposure, it was however possible to reduce the mass flow rate to lower values, but the performance of the test compressor was also deteriorated significantly.

The compressor in this category was able to continue operation at lower mass flow rate. However, due to the imposition of reasonably high losses, it was decided carry on operating to a point where the pressures upstream and downstream of the inlet guide vanes being identical. The above decision was also applied in the examinations of the one-stage compressor with casing treatment configurations, using higher values of the rotor tip exposure ratios.

It has to be added that with this configuration not only old nature of the stall altered from the abrupt to the progressive stall, but the loud noises due to the compressor operation were also diminished significantly during the testing procedures and particularly at lower mass flow rates. Both of these advantages were also observed in the other one-stage compressor with casing treatment configurations except for that of (0.232) rotor exposure ratio.

It was also found that the builds with exposures of (33.3%, 43.4%, 53.5% & 63.6%) were able to operate across a greater range of low mass flow rates, after the point at which pressures upstream and downstream of the inlet guide vanes became equal. This feature, however, can be considered as a drawback of the compressor builds with higher values, (0.737 & 0.838) rotor tip axial chord exposure ratios.

The compressor with (33.3%) rotor exposure casing treatment provided a minimum value of corrected applied power of about 2.702 [kwatt] at the choke condition. This parameter increased to a value of 2.878 [kwatt], corresponding to 2.196 [kg/sec] corrected mass flow rate with further reduction of the flow coefficient.

As the corrected mass flow through the compressor was reduced to a lower value of 1.691 [kg/sec], the corrected applied power was also decreased to a value of 2.811 [kwatt]. Moreover by further closing the test rig throttle, the flow coefficient was again reduced, but the corrected applied power increased to a maximum value of 2.881 [kwatt] at near the stall condition.

The characteristic of the corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment configuration of (0.333) rotor tip exposure ratio, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)] is shown in [Graph (8-23)]. As can be seen from this graph the trendline for the corrected applied power is significantly different in some parts to those for the solid casing configuration and casing treatment build of (0.232) rotor tip exposure ratio.

The overall work coefficient of this build also showed normal characteristic with minimum and peak values of (0.384) and (0.744) at choke and near stall conditions, respectively. The characteristic of the overall work coefficient against corrected mass flow rate of the one-stage compressor with casing treatment configuration of (0.333) rotor tip exposure ratio, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)] is presented in [Graph (8-24)].

Similarly to the previous casing treatment configuration of (0.232) rotor exposure ratio, the current build overall total-static efficiency value of (0.32) at the near stall condition was

higher than that (0.088) which calculated at under choke condition using the corresponding pressure ratio.

These similarities also were achieved through the other efficiency characteristics. The overall total-total, stage static-static and overall static-static efficiencies at the choke and near stall conditions were determined to be (0.722, 0.662 & 0.555) and (0.439, 0.417 & 0.410), respectively.

As the mass flow through the compressor was decreased, the overall total-total, stage static-static and both the overall static-static and overall total-static of the efficiency characteristics increased to their peak points. The maximum efficiencies were calculated to be (0.731, 0.693, 0.626 & 0.412) associated with corrected mass flow rate values of (2.363, 2.255, 2.122, & 1.793) [kg/sec], respectively.

The characteristics of the overall and stage efficiencies versus corresponding corrected mass flow rate of the one-stage compressor with casing treatment of (0.333) rotor tip exposure ratio, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)] are shown in [Graph (8-25)].

The pressure rise coefficient characteristics in terms of the overall total-total, stage static-static, overall static-static and overall total-static pressure ratios at the choke and close to the stall conditions were determined to be (0.282, 0.257, 0.215 & 0.032) and (0.327, 0.311, 0.306 & 0.243), respectively. The peak values of the above overall and stage pressure rise coefficients in association with corresponding corrected mass flow rates achieved were (0.360, 0.350, 0.331 & 0.247) and (1.846, 1.849, 1.756 & 1.520) [kg/sec], respectively.

The characteristics of the overall and the stage pressure rise coefficients versus the relevant corrected mass flow rate of the one-stage compressor with casing treatment build of (0.333) rotor tip exposure ratio, configuration [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)] are illustrated in [Graph (8-26)].

8.7.3)- Characteristics of (0.434) Rotor Exposure Ratio Casing Treatment:

In order to examine the one-stage compressor with vane-recessed casing treatment of (43.4%) rotor tip axial chord exposure, the compressor was taken apart and reassembled in three different times. For this part of the investigation the one stage machine was built by applying the inlet guide vanes, the casing treatment with two new steel and acrylic spacer rings located exactly before and after the vane-recessed chamber treatment, and honeycomb flow straighteners.

This configuration from now on is called, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]. The compressor test rig with the above-noted assembly was examined on five different days and times. The first two sets of results relating to the rotor exposure ratio of (41.4%) were considered in the intermediate stages of the current investigation, but were discarded in the final analysis.

The last test results obtained from the three examinations, has been included in the final part of the study. It is worth noting that although the obtained flow characteristic results were not used in the final analysis due to the specified increment of the rotor blade exposure, but they were quite encouraging.

The modified flow characteristic results obtained from the final testing of the one-stage compressor with casing treatment of (0.434) rotor exposure ratio, calculated after removing the different difficulties encountered, are presented in [Table (8-14)].

The other associated results relevant to the peak points or near the stall condition of the build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] are also given in [Table (8-15)].

This compressor configuration was able to operate in a flow coefficient range of (0.592) and (0.334) at the choke and near the stall conditions, respectively. It also covered a corrected mass flow rate range of 1.111 [kg/sec], whereas the mass flow rates achieved at the two ends of the flow characteristics were (2.546 & 1.432) [kg/sec].

In comparison with the datum build, this configuration performed completely well in the lower flow rates. It was able to delay stall initiation, while the compressor was working at (26.95%) lower mass flow rate than in the case of the solid casing.

As with the (0.333) rotor tip exposure ratio, this configuration had the capability to continue normal operation at the lower values of corrected mass flow rate of 1.439 [kg/sec]. But in such circumstance the compressor performance dropped sharply due to the positive pressure rise in the inlet guide vanes.

In this configuration, when the first stall cells occurred, the compressor was able to continue fairly normal operation under progressive stall conditions. With further reduction of the mass flow rate, the nature of stall changed to an abrupt one in association with the occurrence of loud noise. By using the build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] the compressor noises during normal operations were also reduced considerably.

The characteristic of the corrected applied power against the corrected mass flow rate of the compressor configuration using casing treatment with (43.4%) rotor tip exposures showed reasonably similar results in this regard in comparison with the build of (0.333) exposure ratio. The minimum value of the corrected applied power of 2.633 [kwatt] occurred at the choke condition, corresponding to the maximum corrected mass flow rate value of 2.549 [kg/sec].

By closing the compressor orifice, the corrected mass flow rate was reduced to a value of 2.212 [kg/sec] with a maximum corrected applied power of 2.824 [kwatt]. With further reduction of the compressor flow rate, the corrected applied power started to decline to a value of 2.770 [kwatt] corresponding to a corrected mass flow rate of 1.752 [kg/sec]. Moreover from this flow rate to the near stall condition, where the corrected mass flow rate was about 1.439 [kg/sec], the corrected applied power again increased to a maximum value of 2.856 [kwatt].

The corrected applied power characteristic versus the corrected mass flow rate of the one-stage compressor configuration with casing treatment of (0.434) rotor exposure ratio, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] is illustrated in [Graph (8-27)].

The corresponding minimum and peak values of the overall work coefficient were calculated as (0.371) and (0.744) at the choke and near the stall conditions, respectively. The overall work coefficient against the corrected mass flow rate of the one-stage compressor with casing treatment configuration of (43.4%) rotor blade tip axial chord exposure ratio, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] is presented in [Graph (8-28)].

The overall and the stage efficiency characteristics of this configuration had similar trendlines to those of the (0.232 and 0.333) rotor tip exposure ratios. However, the values of the both the overall total-total and the overall static-static including the stage static-static efficiencies at the choke condition were higher near the stall. These values were found to be (0.752, 0.568 & 0.679) and (0.488, 0.454 & 0.462) at the choke and near the stall conditions, respectively. However, the efficiency characteristic obtained in terms of the total-static pressure ratio, showed the values of (0.086) and (0.355) at the choke and near the stall conditions, respectively. This meant that the trendline of the overall total-static efficiency was different to the trendlines of the other above-noted efficiencies.

Nonetheless, by closing the rig orifice and reducing the mass flow through the compressor, both overalls and stage efficiencies were increased to their peak values. The maximum values of the overall total-total, overall and stage static-static, overall total-static and the corresponding corrected mass flow rate values were determined to be (0.761, 0.720, 0.647 & 0.435) and (2.378, 2.234, 2.116 & 1.799), respectively.

The overall and the stage efficiency characteristics against the relevant corrected mass flow rate of the one-stage compressor in association with (0.434) rotor tip exposure ratio casing treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] are shown in [Graph (8-29)]. The trendlines of the entire, the overall and the stage pressure rise coefficient characteristics of the present configuration showed many similarities. In all cases the minimum values of the pressure rise coefficients occurred when the orifice was fully opened.

By reducing the flow coefficient through the compressor, the above characteristics were increased to their peak points. The overall and the stage pressure rise coefficients, calculated in terms of their corresponding static-static pressure ratios, approached each other near the stall condition.

The values of the overall total-total, overall static-static, overall total-static including the stage static-static pressure rise coefficients at both the choke and near the stall conditions were determined to be (0.284, 0.213, 0.030 & 0.254) and (0.344, 0.321, 0.255 & 0.325), respectively. The peak values of the aforementioned pressure rise coefficients and their corresponding corrected mass flow rates were calculated to be (0.364, 0.335, 0.256 & 0.356) and (1.810, 1.720, 1.500 & 1.831), respectively.

The characteristics of the overall and the stage pressure rise coefficient versus the corrected mass flow rate of the one-stage compressor configuration with casing treatment of (0.434) rotor tip axial chord exposure ratio, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)] is illustrated in [Graph (8-30)].

8.7.4)- Characteristics of (0.535) Rotor Exposure Ratio Casing Treatment:

The test rig configuration with the one-stage compressor with (0.535) rotor exposure ratio casing treatment, designated [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)], was dismantled and reassembled three times during the current research.

The first instance was related to the application of the Acrylic-made spacer ring before trimming. This ring provided a rotor tip axial chord exposure ratio of (0.527). The compressor configuration was then examined on two different times and dates. The results obtained from this build were not used in the intermediate or final procedures of the investigation, but they were considered in the preliminary steps of the study.

The three sets of test results obtained from the second and third compressor builds with casing treatment of (53.5%) rotor tip exposure were used in the other parts of the investigation. It is worth noting that the only results from the last compressor build have been considered in the final analysis.

Some results obtained from the flow characteristics of the one-stage compressor with (0.535) rotor exposure ratio casing treatment configuration are illustrated in [Table (8-16)]. Other results relating to particular points such as highest or near the stall characteristics of the build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)] are provided in [Table (8-17)].

The compressor build with (0.535) rotor exposure ratio was one of the best configurations in terms of efficiency characteristics due to its low losses. The flow coefficients at the choke and near stall conditions were calculated as being (0.591) and (0.351), respectively. The corresponding mass flow rates in these conditions were (2.566 & 1.525) [kg/sec].

The current configuration of the compressor with the casing treatment of (53.5%) rotor exposure was able to operate with (23.25%) lower mass flow rate in comparison with the datum build. It was also able to cover a corrected mass flow rate range of 1.032 [kg/sec].

The minimum and maximum values of the corrected mass flow rate of (1.512 & 2.544) [kg/sec] were achieved at near the stall and choke conditions, respectively. In this configuration it was possible to close the compressor throttle with the aim of further reducing the flow through the compressor. However, due to the positive pressure rise through the inlet guide vanes, additional losses caused a significant reduction in both the compressor and the treated casing performances.

Although the rotating stall status of this configuration changed from the abrupt to progressive, it was not as good as the build with (0.333) rotor exposure ratio in terms of improving the stall margin. The latter compressor build was able to sustain a lower mass flow rate, albeit with significant performance deterioration.

The compressor build with casing treatment of (0.535) rotor exposure ratio provided a corrected applied power characteristic with a minimum value of 2.639 [kwatt] at the choke condition. This parameter increased to a maximum value of 2.832 [kwatt] at a 2.139 [kg/sec] corrected mass flow rate as the flow was reduced inside the test rig. With further reduction of the flow, the corrected applied power was decreased to a value of 2.756 [kwatt], corresponding to corrected mass flow rate of 1.717 [kg/sec].

The corrected applied power characteristic against the corrected mass flow rate of the one-stage compressor with casing treatment configuration of (53.5%) rotor tip exposure, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)] is shown in [Graph (8-31)]. The trendline of the corrected applied power is different to the datum build, but quite similar to those with the casing treatment configurations.

The overall work coefficient of this configuration, exhibited minimum and peak values of (0.336) and (0.670) at the choke and near stall conditions, respectively. The characteristic of the overall work coefficient versus the corresponding corrected mass flow rate of the one-stage compressor with casing configuration of (0.535) rotor tip exposure ratio, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)] is illustrated in [Graph (8-32)].

The efficiency characteristics of the present compressor configuration showed significant similarities in the stage and overall static-static results in addition to that of the overall total-total. In all above-noted cases the values of compressor efficiency at the choke condition were higher than those at the near stall circumstance. This was not achieved however from the characteristic of the overall efficiency that was calculated in terms of the total-static pressure ratio, since the value of this efficiency was higher at the near stall condition.

The values of the overall total-total, stage static-static, overall static-static and overall total-static efficiencies at the choke and near stall conditions were (0.751, 0.680, 0.570 & 0.083) and (0.532, 0.508, 0.498 & 0.388), respectively. The maximum values of these efficiencies together with their corrected mass flow rates also were calculated as (0.764, 0.725, 0.649 & 0.442) and (2.348, 2.222, 2.111 & 1.802) [kg/sec], respectively.

The overall and stage efficiency characteristics of the one-stage compressor with casing treatment of (0.535) exposure ratio, configuration [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)] versus the corresponding corrected mass flow rate are illustrated in [Graph (8-33)]. The fitted trendlines for the characteristics of overall and stage pressure rise coefficients were generally similar in the majority of cases except the maximum value of the overall total-static efficiency which occurred at the near stall condition.

Moreover, the efficiency characteristics, in terms of the overall total-total, stage and overall static-static and the overall total-static pressure ratios showed values of (0.281, 0.252, 0.212 & 0.029) and (0.353, 0.337, 0.331 & 0.262), at the choke and near stall conditions, respectively. The peak values of the above-noted overall and stage pressure rise coefficients together with their corrected mass flow rates were calculated as (0.369, 0.359, 0.338 & 0.262) and (1.816, 1.835, 1.705 & 1.512), respectively.

The characteristics of the overall and stage pressure rise coefficient of the one-stage compressor configuration with casing configuration of (0.535) rotor tip exposure ratio versus the corrected mass flow rate, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)] are shown in [Graph (8-34)].

8.7.5)- Characteristics of (0.636) Rotor Exposure Ratio Casing Treatment:

The one-stage compressor was taken apart and reassembled on two different occasions in order to provide the compressor build with rotor tip exposure ratio of (0.636). This configuration was built using the inlet guide vanes followed by casing treatment with a newly manufactured aluminium alloy spacer ring in addition to the first stage rotor and stator blades, resulting the build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)].

This test rig compressor assembly was examined across three different days. The first two sets of results were considered in the preliminary as well as the intermediate stages of the investigation, but they were discarded in the final analysis. Thus, the data obtained from only the last day of testing was considered in the final part of the study.

The final results obtained from the flow characteristic of the one-stage compressor with casing treatment of (0.636) rotor exposure ratio are shown in [Table (8-18)]. The characteristic of the other results relevant to the peak points or near the stall condition of the build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] are presented in [Table (8-19)].

This compressor configuration was able to cover a corrected mass flow rate range of 1.009 [kg/sec]. The mass flow rates achieved at either end of the flow characteristics were (2.498 & 1.487) [kg/sec] corresponding to flow coefficient ranges of (0.587) and (0.355) at the choke and near the stall conditions, respectively. In comparison to the solid casing build, this configuration performed reasonably well at lower flow rates. It was able to delay stall initiation and work with (22.46%) lower mass flow rate than the datum build.

Unlike the three previous treatment configurations with (0.333, 0.434 and 0.535) rotor exposure ratio, this configuration was only capable of continuing its normal operation in a relatively lower values of mass flow rate, where the static pressure upstream and downstream of the inlet guide vanes became identical. Although in such condition it was also possible to reduce somewhat the mass flow rate inside the compressor, but the closing angle of the orifice electrical switch was small and quite restricted.

The values of the corrected mass flow rate at the choke and near stall conditions were calculated of (2.521 & 1.513) [kg/sec], respectively. However it was found that with further reduction of the flow, the compressor performance deteriorated sharply due to the occurrence of the positive pressure rise in the inlet guide vanes. It is worth noting that in this configuration when the initiation of the early stall cells occurred, although the compressor was also able to continue fairly normal operation under progressive stall status, the obtained range of the flow coefficient in this circumstance was less than the three previous builds.

With further reduction of the mass flow rate, the nature of stall changed to the abrupt ones again in association with the occurrence a loud noise, as with the previous compressor

build with casing treatments. With the build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] the noise produced by compressor during normal operating was also reduced noticeably.

The corrected applied power characteristic against the corrected mass flow rate indicated that this build performed slightly different in the near stall condition in comparison to the three previous compressor configurations, as the maximum value of the corrected applied power did not occur at the near stall condition. The highest value of this parameter was about 2.198 [kwatt], corresponding to a corrected mass flow rate of 2.163 [kg/sec].

Nonetheless, the minimum value of the corrected applied power, 2.724 [kwatt] occurred at the choke condition corresponding to the maximum corrected mass flow rate through the compressor. By closing the compressor throttle, as the corrected mass flow rate reduced, the corrected applied power was increased to its maximum value. With further reduction of the flow inside the test rig, the magnitude of the corrected applied power reduced to a value of 2.752 [kwatt] with a corresponding corrected mass flow rate of 1.700 [kg/sec].

By further closure of the orifice the flow rate through the compressor diminished to its minimum value near the stall point, where the value of the corrected applied power was determined of 2.899 [kwatt]. The characteristic of the corrected applied power versus the corrected mass flow rate of the one-stage compressor configuration with casing treatment of (0.636) rotor tip exposure ratio, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] is illustrated in [Graph (8-35)].

The maximum value of the overall work coefficients characteristic of this configuration that occurred at near the stall points was not higher than the value of build with (43.4%) rotor exposure, but was greater than that belonging to configuration with (0.535) rotor exposure ratio. This might owe to the fact that in the latter case, the compressor build was able to operate at a lower mass flow in association with higher losses.

The trendline of the characteristic obtained from the maximum values of the overall works coefficients versus rotor tip exposure was similar to the trendline achieved from the characteristics of the stall margin improvements in terms of the flow coefficients. It is worth noting that the improvements in stall margin was calculated in terms of the only flow rate, as well as both the corrected flow rates and the pressure rise coefficients. However, this will be discussed in more detail in the subsequent chapter.

The minimum and the peak values of the overall work coefficient were determined as (0.381) and (0.687) corresponding to lowest and highest values of the flow coefficients, respectively. The characteristic of the overall work coefficient against the corrected mass flow rate of the one-stage compressor with casing treatment build of (0.636) rotor exposure ratio, configuration [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] is shown in [Graph (8-36)].

The characteristics of the overall and stage efficiency in this configuration indicated that the values of the overall total-total, overall static-static and stage static-static efficiencies at the choke condition were also higher than those achieved near the stall. These values were calculated as (0.722, 0.547 & 0.646) and (0.529, 0.496 & 0.498), respectively.

Moreover, the overall total-static efficiency was found to be (0.086) and (0.355) at the choke and near the stall conditions, respectively. The fitted trendlines also showed reasonable similarities between the present build efficiency characteristics and those of the previous configurations.

Due to the reduction of the mass flow rate from closing the rig orifice, all overall and stage efficiencies were increased to their corresponding peak values. The maximum values of the overall total-total, stage and overall static-static, and the overall total-static efficiencies

together with their corrected mass flow rates in this build were calculated of (0.735, 0.702, 0.628 & 0.433) and (2.317, 2.175, 2.077 & 1.791) [kg/sec], respectively.

The overall and stage efficiency characteristics of the one-stage compressor with casing treatment of (0.636) rotor tip exposure ratio versus the relevant corrected mass flow rates, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] are illustrated in [Graph (8-37)].

The trendlines that were fitted into the overall and the stage pressure rise coefficient characteristics of the present configuration showed reasonable similarities, except in the case of the overall total-static which the maximum value of this parameter occurred close to the stall condition. The values of the overall total-total, stage and overall static-static including the overall total-static pressure rise coefficients at both the choke and near the stall conditions were calculated as (0.283, 0.251, 0.214 & 0.031) and (0.357, 0.337, 0.336 & 0.263), respectively.

By reducing the mass flow through the compressor, the above-noted characteristics were increased to their maximums. Except the peak value of the overall total-static pressure rise coefficient, which occurred close to the stall condition, the other maximums and their corresponding corrected mass flow rates were determined as (0.370, 0.361 & 0.341) and (1.791, 1.837 & 1.679), respectively.

The compressor build with (0.636) exposure ratio was one of the best configurations in terms of improving the peak pressure rise coefficients, due to reasonably low losses. It is worth noting that the overall and the stage pressure rise coefficients, which were calculated in terms of their corresponding static-static pressure ratios, were becoming equal near the stall condition.

The characteristic of the overall and stage pressure rise coefficients against the corrected mass flow rates of the one-stage compressor with (0.636) rotor exposure ratio treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)] is presented in [Graph (8-38)].

8.7.6)- Characteristics of (0.737) Rotor Exposure Ratio Casing Treatment:

In order to assemble the one-stage compressor rig with (0.737) rotor exposure ratio, an aluminium alloy spacer ring was manufactured and used to replace the ring following the main part of the vane-recessed chamber of the casing treatment. This configuration, which is hereafter is called build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)], was dismantled and reassembled on two different occasions and was examined on three different times and dates during this part of the investigation.

The results obtained from the first two test sessions were utilised in the most parts of the current research with the exception of the ultimate analysis. The last set of results for this compressor build encountered no problems and was processed in the final analysis. The results obtained from the flow characteristics of the one-stage compressor build with casing treatment of (0.737) rotor exposure ratio are illustrated in [Table (8-20)]. Some of the other results relating to particular points such as maximum or near the stall condition of the build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)] are also presented in [Table (8-21)].

It is worth noting that this compressor configuration did not provide the ability to close the throttle across a reasonably wide range of flow coefficient reduction after the pressures upstream and downstream of the inlet guide vanes became identical. When the throttle was fully opened this configuration was able to attain a mass flow rate of 2.467 [kg/sec]. This reduced to a value of 1.551 [kg/sec], when the compressor was running close to the stall condition. The corresponding flow coefficients in these circumstances were (0.582) and (0.366), respectively.

The current compressor build with casing treatment of (0.737) rotor exposure ratio was able to operate with (19.88%) lower mass flow rate in comparison with the solid casing. This configuration was also able to cover a corrected mass flow rate range of 0.925 [kg/sec]. The maximum and the minimum of the corrected mass flow rate of (2.498 & 1.573) [kg/sec] were achieved at the choke and close to the stall conditions, respectively.

In this configuration it was also possible to close the test rig throttle in order to reduce the mass flow rate through the compressor after the pressures upstream and downstream of the inlet guide vanes became equal. But after above-noted point, in both cases, with the normal or under progressive stall conditions, the compressor could not sustain operation for long as with the lower rotor exposure ratios.

It is worth noting that although the nature of the rotating stall in this configuration was also altered from the abrupt to progressive, it did not behave as well as the build with lower rotor tip axial chord exposure ratio. Further reduction of the mass flow in this case not only led to an abrupt stall, but also was associated with a significant drop in performances.

The maximum value of the corrected applied power of the present configuration did not occur close to the stall condition as happened in the cases with the lower exposures. This value was determined as 2.900 [kwatt] in association with a corrected mass flow rate of 2.091 kg/sec]. The corrected applied power characteristic of the compressor build with casing treatment of (73.7%) rotor exposure showed a minimum value of 2.688 [kwatt] at the choke condition.

By reducing the mass flow through the rig, this parameter increased to its maximum value, but again reduced to a value of 2.820 [kwatt] in association with a corrected mass flow rate of 1.698 [kg/sec] as the flow diminished. As the orifice of the test rig was closed slightly to reduce the mass flow rate through the compressor, the corrected applied power again increased to a value of 2.858 [kwatt] which was slightly lower than the peak value near the stall condition.

The characteristics of the corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment build of (0.737) rotor exposure ratio, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)] is presented in [Graph (8-39)].

The trendline of the corrected applied power of this build was different in comparison to the equivalent characteristic of the solid casing. However, it is a nearer match than the characteristics obtained from the other configurations with lower rotor exposure ratios.

The maximum value of the overall work coefficient for this build was calculated as (0.654) at an operating point close to the stall condition. When the test rig orifice was fully opened the overall work coefficient characteristic dropped to a minimum value of (0.385).

The characteristics of the overall work coefficient of the one-stage compressor with casing treatment of (0.737) rotor tip axial chord exposure ratio, versus the relevant corrected mass flow rate is illustrated in [Graph (8-40)]. The overall and stage efficiency characteristics of the present compressor build did not show significant differences in comparison with the characteristics obtained from the configurations with the lower rotor exposure ratios.

Nevertheless, the degree of the similarity in the some cases was not high. In this configuration except for the characteristic of the overall total-static efficiency, the values of the other overall and stage efficiencies at the choke condition were higher in comparison to those calculated close to the stall condition.

However, the corresponding values of the overall (total-total, static-static and total-static) and stage static-static efficiencies at the choke and near stall conditions were determined as

(0.721, 0.546, 0.092 & 0.649) and (0.552, 0.519, 0.393 & 0.520), respectively.

The maximum values of the overall total-total, stage and overall static-static and the overall total-static efficiencies together with their relevant corrected mass flow rates were calculated as (0.728, 0.697, 0.624 & 0.425) and (2.346, 2.180, 2.069 & 1.789) [kg/sec], respectively. The overall and stage efficiency characteristics against the corresponding corrected mass flow rate of the one-stage compressor with treated casing of (0.737) rotor exposure ratio, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)] are shown in [Graph (8-41)].

The characteristics of the pressure rise coefficients showed that the trendlines of the stage and the overall efficiencies, which were calculated in terms of the static-static pressure ratio, coincided at a point close to the stall condition. Except for the peak value of the overall total-static pressure rise coefficient, which occurred at near the stall condition, the maximum values of the other overall and stage characteristics were achieved before the stall point.

However, the maximum values of the pressure rise coefficient characteristics in terms of the overall total-total, stage and overall static-static and the overall total-static were calculated as (0.366, 0.359, 0.340 & 0.260) corresponding to corrected mass flow rates of (1.787, 1.851, 1.688, & 1.573), respectively. The values of the overall (total-total, static-static and total-static) pressure rise coefficients including the stage static-static were determined as (0.282, 0.212, 0.034 & 0.252) and (0.359, 0.338, 0.260 & 0.339) at the choke and near stall conditions, respectively.

The characteristics of the overalls and stage pressure rise coefficient of the one-stage compressor with casing treatment of (0.737) rotor exposure ratio versus the corrected mass flow rate, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)] are shown in [Graph (8-42)].

8.7.7)- Characteristics of (0.838) Rotor Exposure Ratio Casing Treatment:

In order to assemble the single-stage compressor with the casing treatment configuration of (0.838) rotor exposure ratio, the compressor test rig was again taken apart and reassembled on two different occasions.

The compressor was built using the inlet guide vanes followed by the main part of the casing treatment with newly manufactured aluminium alloy spacer rings in order to achieve the configuration [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]. The first assembly with above-noted configuration was tested on two different days. The results obtained were used in the both the preliminary and the intermediate parts of the present study. These results due their associated uncertainties were ignored in the final analysis.

However, the data obtained from the testing of the second assembly was used in the calculation program in order to find out the flow characteristics. This set of results was also utilised in the final analysis procedure of the current investigation. The final flow characteristic results obtained from the one-stage compressor build with (0.838) rotor exposure ratio casing treatment are shown in [Table (8-22)]. Some other associated results corresponding to the peak points or relevant to the near stall condition of build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)] are also illustrated in [Table (8-23)].

In the majority of cases the flow characteristic behaviour of this arrangement performed fairly well at the lower flow rates in comparison to the datum build configuration. Mass flow rates of (2.478 & 1.605) [kg/sec] were achieved in association with a flow coefficient range of (0.585) and (0.379) at the choke and near the stall conditions, respectively.

In comparison with the other casing treatment configurations, a corrected mass flow rate range of just 0.883 [kg/sec], was covered by use of the current build. On the other hand in

comparison with the solid casing, this configuration performed reasonably well at lower flow rates, because it was able to delay the onset of stall with the compressor working at (17.15%) lower mass flow rate than the datum build.

The values of the corrected mass flow rate achieved at the choke and close to the stall conditions were (2.518 & 1.635) [kg/sec], respectively. However this build did not provide a substantial margin for further reduction of the flow when the wall static pressure at the upstream and downstream of the inlet guide vanes became equal. By small reduction of the mass flow rate after approaching to the above-noted point, the compressor performance deteriorated straight away due to the occurrence of the positive pressure rise at the inlet guide vanes.

It is worth noting that the present build was able to continue operation under progressive stall conditions, subsequent to the early stall cell formation, despite the low flow coefficient range achieved in comparison with the other casing treatment configurations. By further reducing the mass flow rate, the progressive stall behaviour altered to an abrupt stall. Again in this circumstance loud noise immediately occurred. As with the previous casing treatment configurations (except the one with 23.2% exposure) this build was able to operate with noticeably lower noise levels.

The characteristic of the corrected applied power versus the corrected mass flow rate of the casing treatment configuration with (0.838) rotor exposure ratio showed that this build performed somewhat differently in comparison to the previous compressor configurations.

In this case the minimum value of the corrected applied power, which was about 2.655 [kwatt] occurred at the choke condition. As the compressor rig throttle was closed, this parameter started to increase to a maximum value of 2.895 [kwatt], corresponding to a corrected mass flow rate value of 2.131 [kg/sec]. With further reduction of the mass flow, the corrected applied power reduced to a value of 2.860 [kwatt] relevant to 1.797 [kg/sec] corrected mass flow rate. From this particular point to near stall condition, the corrected applied power increased again to a value of 2.894 [kwatt], as the mass flow decreased.

The corrected applied power characteristic of the one-stage compressor configuration with casing treatment of (0.838) rotor exposure ratio versus the corrected mass flow rate, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)] is illustrated in [Graph (8-43)].

The maximum value of the overall work coefficient characteristic of the (0.838) rotor exposure ratio casing treatment build was less than the other treatment configuration. This might be due to the fact that in this particular case, the amount of the rotor tip axial chord, exposed to the recessed chamber of the casing treatment was higher than optimum.

The peak and the minimum magnitudes of above coefficient were calculated as (0.630) and (0.374), corresponding to highest and lowest values of the mass flow rates, respectively. The characteristic of the overall work coefficient of the one-stage compressor with casing treatment configuration of (0.838) rotor exposure ratio against the relevant corrected mass flow rate, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)] is presented in [Graph (8-44)].

The overall and stage efficiency characteristics of the current treated casing configuration indicated that the values of the overall total-total, overall static-static and the stage static-static efficiencies at the choke conditions were higher than those, at near the stall circumstances. The values were calculated as (0.710, 0.560 & 0.667) and (0.562, 0.528 & 0.530), corresponding to the highest and the lowest values of the relevant flow coefficients, respectively.

The efficiency in terms of the overall total-static pressure ratio was found to be (0.073) at

the choke and (0.390) and near the stall conditions. The efficiency characteristic trendlines indicated some similarities in comparison to those with different rotor tip exposure ratios.

By reducing of the mass flow rate of the test rig, all overall and stage efficiencies increased to their related peak values. The maximum values of the overall total-total, stage and overall static-static and the overall total-static efficiencies together with their corrected mass flow rates were calculated as (0.717, 0.702, 0.627 & 0.409) and (2.365, 2.244, 2.119 & 1.804) [kg/sec], respectively.

It is worth noting that the overall and stage efficiency characteristics, which were calculated in terms of their corresponding static-static pressure ratio, tended to coincide at a point close to the stall condition. The overall and stage efficiency characteristics of the one-stage compressor with casing treatment of (0.838) rotor exposure ratio against the corresponding corrected mass flow rates, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)] are illustrated in [Graph (8-45)].

The characteristics of the overall total-total, overall static-static and the stage static-static pressure rise coefficients of the present casing treatment configuration showed acceptable similarities, as their peak values did not occur close to the stall point. However, the maximum value of the overall total-static pressure rise coefficient had been achieved at a point close to the stall condition.

The values of the overall total-total, stage and overall static-static and the overall total-static pressure rise coefficients were calculated as (0.270, 0.252, 0.212 & 0.026) and (0.352, 0.332, 0.331 & 0.248), at the choke and near the stall conditions, respectively.

By reducing the mass flow through the compressor, the above characteristics increased to their corresponding maximum points. Except for the peak value of the overall total-static pressure rise coefficient, which occurred close to the stall condition, the maximum values of these characteristics including their corresponding corrected mass flow rates were determined as (0.356, 0.350 & 0.334) and (1.797, 1.898 & 1.738) [kg/sec], respectively.

The trendlines of the overall and the stage pressure rise coefficient characteristics, which were calculated in terms of their associated static-static pressure ratios, converged at a point close to the stall condition. The characteristics of the overall and stage pressure rise coefficients against the corrected mass flow rate of the one-stage build with treated casing of (0.838) rotor exposure ratio, configuration [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)] are shown in [Graph (8-46)].

8.8)- Summary of the Results:

This chapter has presented the flow characteristic results obtained from testing the compressor under the inlet guide vanes and rotor-only configuration, both with and without the honeycomb air straightener devices. These indicated the occurrence of huge losses inside the compressor as well as in the downstream ducting. The losses were due to the undesired axial direction of the exit flow from the rotor. As a result the compressor was unable to operate without stalling.

Although the rotor-only compressor build, which incorporated the honeycomb air straighteners performed slightly better than the build without the honeycombs, even this did not provide the hoped for benefits.

The chapter also described the testing of the one-stage compressor with the inlet guide vanes, as well as the first stage rotor and stator blades. The results indicated that the application of the honeycomb flow straighteners was able to assist the flow to pass more

easily through the compressor rig and the downstream ducting in the desired axial direction. It was noted that similar flow characteristics results, particularly at the lower mass flow rates and the near stall conditions, had also been achieved from the three-stage compressor builds. The overall and stage characteristics of the one-stage compressor indicated that all of configurations, which incorporated the casing treatment performed significantly better than the original solid casing or datum build.

Results were presented from the one-stage compressor with both the solid and treated casings. The treated configurations incorporated seven different values of rotor tip exposure ratios. The results obtained from the casing treatment builds showed that the nature of the stall changed from the abrupt to progressive, except in the case of the build with the (23.2%) rotor tip exposure.

All of the treated builds, except the (0.232) rotor exposure ratio, were able to operate at much lower mass flow rates. Although six of the treatment configurations resulted in progressive rather than abrupt stall, four of these (0.333, 0.434, 0.535 and 0.636 rotor exposure ratios), permitted the test rig throttle to be closed more than the other two (73.7% and 83.8%) rotor blade exposures. These six treated configurations were tested by reducing the mass flow rate until the static pressures, upstream and downstream of the inlet guide vane became equal. This point was actually before the onset of stall for these builds.

It should be noted that the study did not pursue this line of investigation because of the low compressor performance achieved beyond this point. Further reduction of the mass flow rate caused the occurrence of heavy rotating stall and associated high noise levels at the inlet guide vanes.

The (0.737 and 0.838) rotor blade exposure ratios configurations performed less well than the other four, when the static pressures upstream and downstream of the IGVs were equalised. While six of the casing treatment configurations produced benefits in terms of progressive instead of abrupt stall, as well as reduced noise levels, especially near the stall condition, none of these advantages were observed with the (0.232) rotor exposure ratio.

In terms of flow rates the results show that the builds with (0.333 and 0.434) rotor exposure ratios produced most improvements in performance. As far as overall and stage maximum efficiencies are concerned the configurations with (33.3% and 43.4%) rotor tip exposures produced the best results. The compressor builds with (0.535 and 0.636) rotor tip exposure ratios were the configurations that provided the best results in terms of the peak pressure rise coefficients.

A full analysis of the test results will be presented in more detail in the following chapter.

Builds with and without the honeycombs flow straighteners	[(IGV)+(1R)+(H)]				[(IGV)+(1R)-(H)]
Corrected mass flow rate, $[(\Phi \epsilon)]$; [kg/sec]	1.91417	1.91388	1.91438	1.91483	1.78613
Flow coefficient at the compressor annulus, (ϕ)	0.44875	0.44884	0.44887	0.44883	0.42053
Mass flow rate at venturi throat, (Φ) ; [kg/sec]	1.90178	1.90141	1.90197	1.90196	1.76556
Mach number under operating condition, (Mach)	0.06903	0.06904	0.06908	0.06904	0.06443
Reynolds number (based on rotor blade cord), (Re)	4.880E+04	4.880E+04	4.881E+04	4.881E+04	4.531E+04
Pressure rise (Inlet guide vanes static-static), $\{[(\Delta P)_{s-s}]_{IGV}\}$; [Pa]	-1391.225	-1386.318	-1383.865	-1378.958	-1283.281
Pressure ratio (Inlet guide vanes static-static), $\{[(PR)_{s-s}]_{IGV}\}$	0.9863783	0.9864258	0.9864387	0.9864997	0.9874273
Pressure rise (rotor static-static), $\{[(\Delta P)_{s-s}]_{1R}\}$; [Pa]	807.121	802.214	799.761	797.308	613.314
Pressure ratio (rotor static-static), $\{[(PR)_{s-s}]_{1R}\}$	1.0080118	1.0079630	1.0079451	1.0079126	1.0060853
Pressure rise (overall static-static), $\{[(\Delta P)_{s-s}]_{Overall}\}$; [Pa]	-596.370	-591.463	-591.463	-586.557	-584.104
Pressure ratio (overall static-static), $\{[(PR)_{s-s}]_{Overall}\}$	0.9941608	0.9942087	0.9942039	0.9942575	0.9942774
Pressure rise (overall total-total), $\{[(\Delta P)_{t-t}]_{Overall}\}$; [Pa]	1295.319	1295.319	1300.23	1295.32	1187.38
Pressure ratio (overall total-total), $\{[(PR)_{t-t}]_{Overall}\}$	1.0128095	1.0128100	1.0128692	1.0128082	1.0117495
Corrected applied power, $[(\Omega \epsilon)]$; [kwatt]	2.77479	2.78752	2.78592	2.77357	2.64482
Work coefficient (overall), $[(\Psi)_{Overall}]$	0.52728	0.52978	0.52843	0.52723	0.54343
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Overall}\}$	0.72758	0.72418	0.72812	0.72808	0.65358
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{Overall}\}$	-0.33468	-0.33048	-0.33126	-0.32927	-0.32097
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{Overall}\}$	0.23360	0.23526	0.23565	0.23929	0.23559
Efficiency (rotor only static-static), $\{[(\eta)_{s-s}]_{1R}\}$	0.45002	0.44805	0.44771	0.44873	0.38734
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{Overall}\}$	0.38364	0.38366	0.38476	0.38387	0.35517
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{Overall}\}$	-0.17647	-0.17508	-0.17505	-0.17360	-0.17443
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{Overall}\}$	0.12317	0.12464	0.12452	0.12616	0.12803
Pressure rise coefficient (rotor only static-static), $\{[(\psi)_{s-s}]_{1R}\}$	0.23729	0.23737	0.23658	0.23659	0.21050

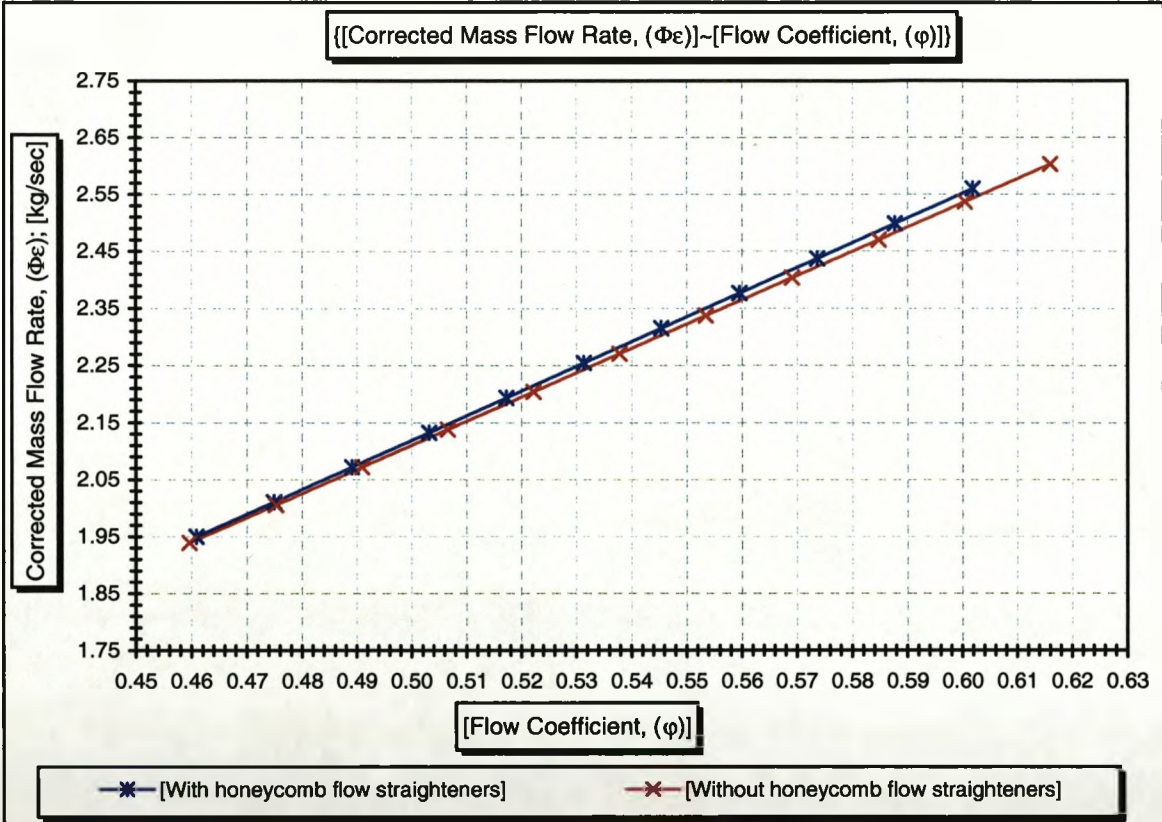
[Table (8-1)]- The obtained results from the rotor only compressor builds with and without the honeycomb flow straighteners

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.603	2.536	2.470	2.404	2.337	2.271	2.205	2.138	2.072	2.005	1.939
Flow coefficient, (ϕ)	0.616	0.600	0.585	0.569	0.553	0.538	0.522	0.507	0.491	0.475	0.460
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.543	2.478	2.412	2.347	2.282	2.217	2.151	2.086	2.021	1.955	1.890
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.659	2.689	2.740	2.791	2.829	2.844	2.835	2.805	2.764	2.726	2.714
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.365	0.385	0.404	0.421	0.437	0.452	0.466	0.478	0.490	0.500	0.509
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.744	0.752	0.757	0.760	0.761	0.759	0.756	0.749	0.741	0.730	0.717
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.272	0.290	0.307	0.321	0.333	0.344	0.352	0.358	0.363	0.365	0.365
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.570	0.595	0.617	0.634	0.647	0.656	0.661	0.662	0.658	0.651	0.639
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.207	0.230	0.250	0.268	0.284	0.297	0.308	0.316	0.322	0.325	0.326
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.040	0.111	0.175	0.231	0.281	0.323	0.358	0.385	0.406	0.419	0.425
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.012	0.044	0.073	0.100	0.124	0.146	0.166	0.183	0.197	0.210	0.219
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.661	0.681	0.697	0.709	0.716	0.719	0.717	0.711	0.701	0.686	0.667
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.241	0.263	0.283	0.300	0.314	0.325	0.334	0.340	0.343	0.343	0.340

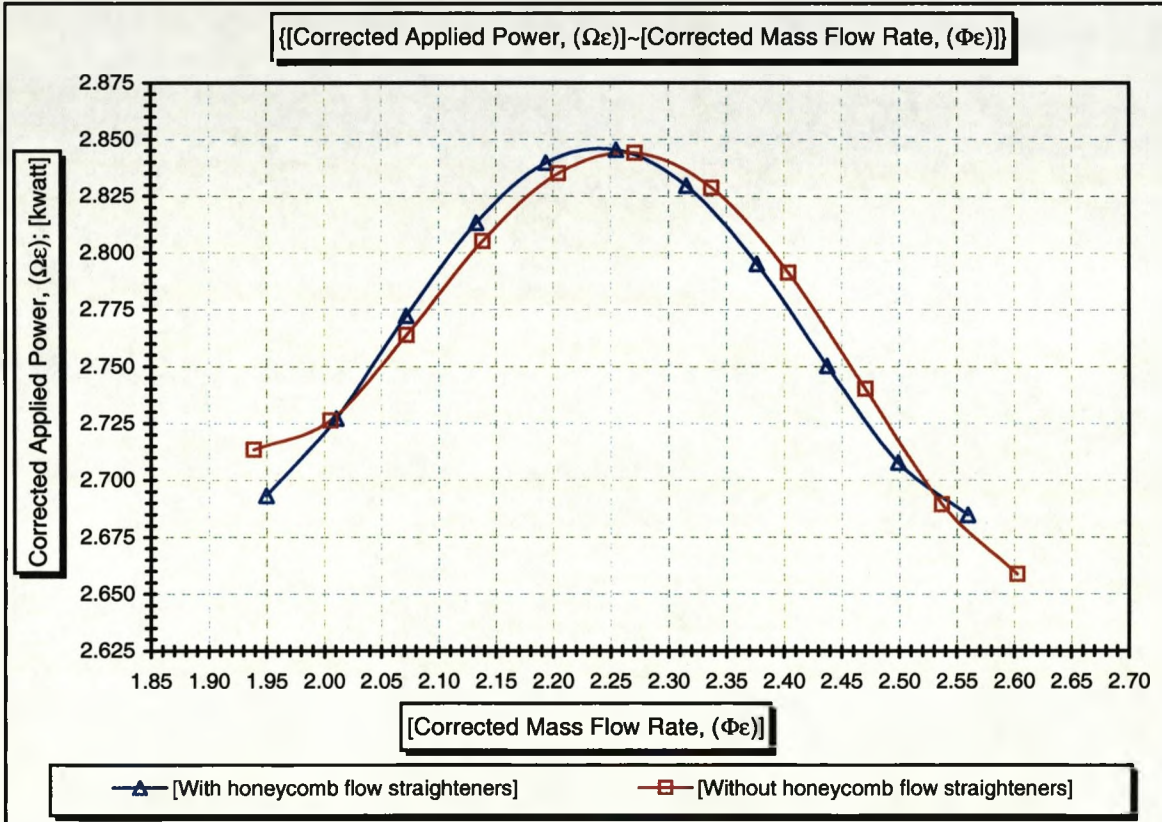
[Table (8-2)]- The obtained results from the one-stage compressor build without the honeycombs flow straighteners, $[(\text{IGV})+(1\text{R})+(1\text{S})-(\text{H})]$

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.559	2.498	2.437	2.376	2.315	2.255	2.194	2.133	2.072	2.011	1.950
Flow coefficient, (ϕ)	0.602	0.588	0.574	0.560	0.545	0.531	0.517	0.503	0.489	0.475	0.461
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.532	2.471	2.410	2.349	2.287	2.226	2.165	2.104	2.043	1.981	1.920
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.685	2.708	2.750	2.795	2.830	2.845	2.840	2.813	2.772	2.727	2.693
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.378	0.395	0.412	0.428	0.444	0.458	0.470	0.481	0.491	0.498	0.502
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.710	0.722	0.731	0.738	0.743	0.746	0.747	0.746	0.742	0.737	0.729
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.267	0.286	0.303	0.318	0.331	0.342	0.351	0.358	0.363	0.366	0.368
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.573	0.593	0.609	0.623	0.635	0.644	0.650	0.654	0.655	0.653	0.649
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.216	0.236	0.253	0.269	0.283	0.295	0.305	0.314	0.320	0.325	0.328
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.074	0.133	0.186	0.235	0.279	0.318	0.351	0.379	0.403	0.421	0.434
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.024	0.052	0.077	0.101	0.123	0.144	0.162	0.179	0.194	0.207	0.219
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.673	0.684	0.692	0.698	0.702	0.704	0.704	0.701	0.697	0.690	0.681
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.254	0.271	0.287	0.301	0.313	0.323	0.331	0.337	0.341	0.343	0.344

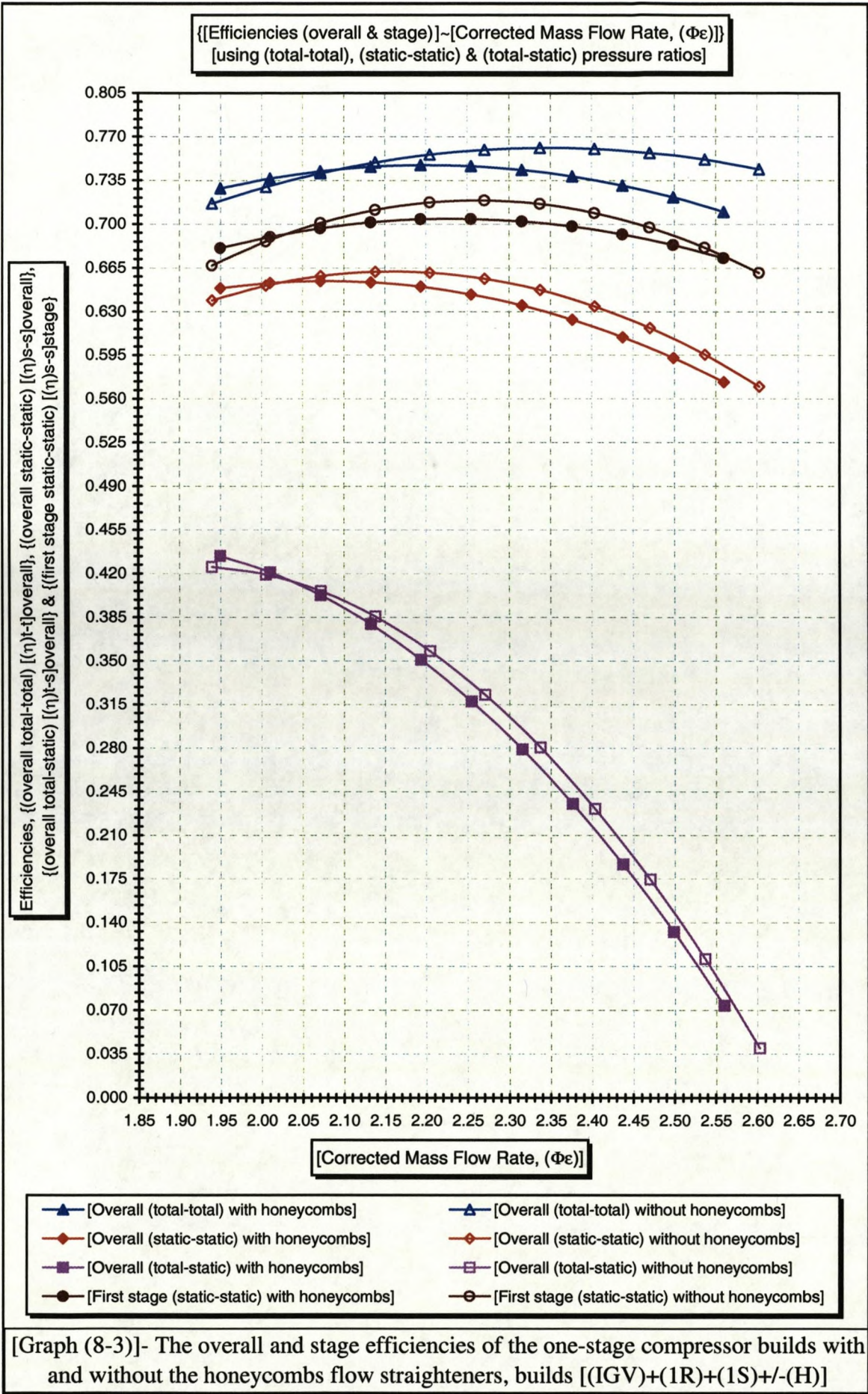
[Table (8-3)]- The obtained results from the one-stage compressor build with using the honeycombs flow straighteners, [(IGV)+(1R)+(1S)+(H)]

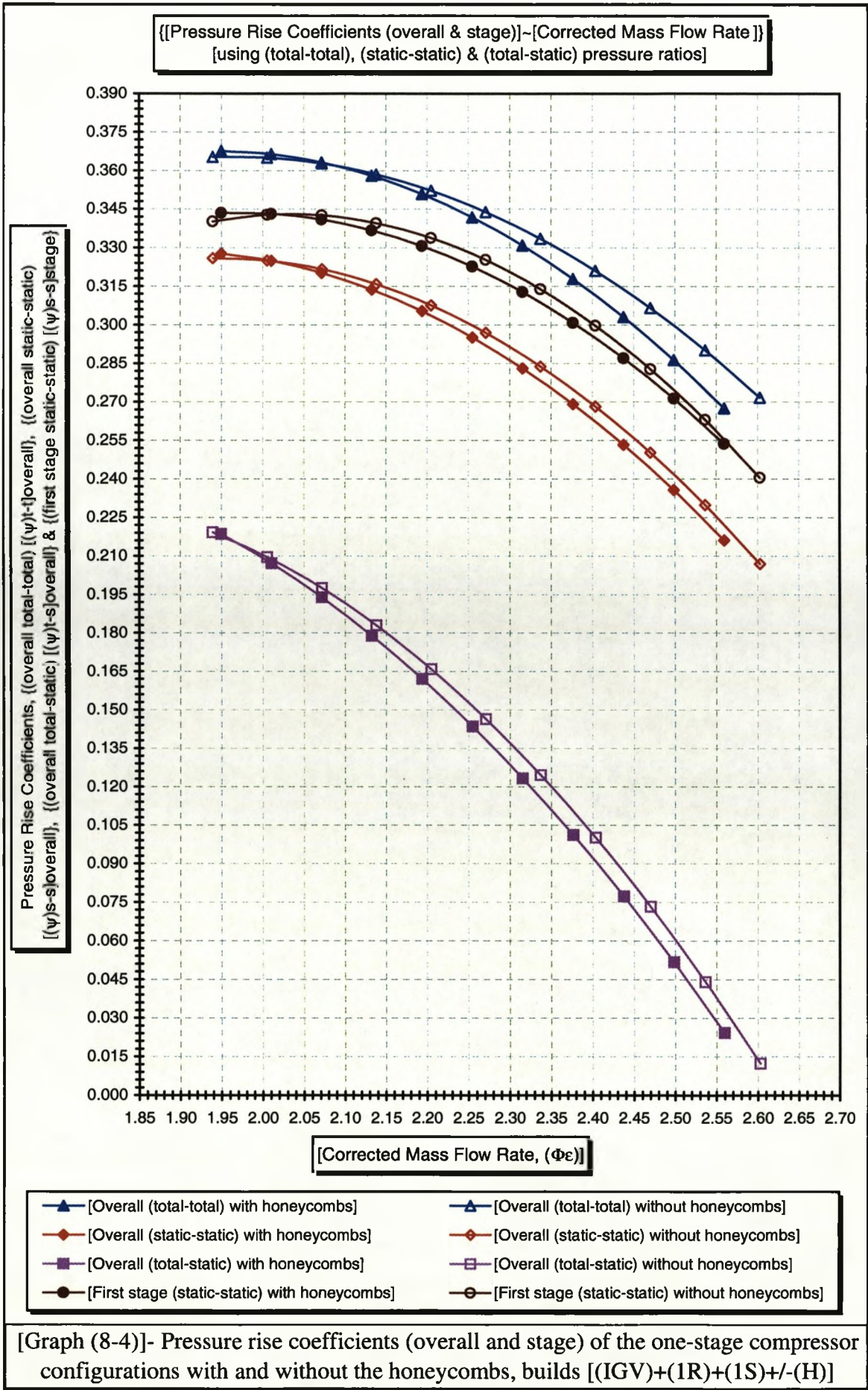


[Graph (8-1)]-Comparison of corrected mass flow rates in one-stage compressor builds with and without the honeycomb air straighteners, builds [(IGV)+(1R)+(1S)+/-(H)]



[Graph (8-2)]- The corrected applied power characteristics of one-stage compressor with and without use of the honeycombs flow straighteners, builds [(IGV)+(1R)+(1S)+/-(H)]





Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.569	2.507	2.446	2.384	2.322	2.260	2.199	2.137	2.075	2.014	1.952
Flow coefficient, (ϕ)	0.602	0.588	0.573	0.559	0.544	0.530	0.516	0.501	0.487	0.472	0.458
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.568	2.506	2.444	2.382	2.320	2.258	2.197	2.135	2.073	2.011	1.949
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.605	2.658	2.713	2.762	2.800	2.824	2.830	2.819	2.791	2.750	2.698
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.369	0.387	0.405	0.422	0.439	0.455	0.469	0.481	0.491	0.498	0.502
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.721	0.733	0.742	0.749	0.753	0.755	0.755	0.752	0.747	0.740	0.730
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.264	0.284	0.302	0.318	0.331	0.343	0.352	0.360	0.365	0.368	0.369
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.592	0.609	0.624	0.636	0.646	0.652	0.656	0.657	0.655	0.650	0.643
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.216	0.236	0.254	0.270	0.284	0.296	0.306	0.314	0.319	0.323	0.324
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.074	0.136	0.193	0.243	0.286	0.324	0.355	0.380	0.398	0.410	0.416
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.024	0.054	0.080	0.105	0.127	0.147	0.164	0.179	0.192	0.203	0.211
Efficiency (first stage static-static), $\{[(\eta)_{s-s}]_{1\text{Stage}}\}$	0.688	0.699	0.706	0.711	0.714	0.714	0.712	0.707	0.700	0.690	0.678
Pressure rise coefficient (first stage static-static), $\{[(\psi)_{s-s}]_{1\text{Stage}}\}$	0.252	0.271	0.288	0.302	0.314	0.325	0.332	0.338	0.342	0.343	0.342

[Table (8-4)]- The flow characteristics of the one-stage compressor configuration (before relocation of the test rig), build [(IGV+1R+1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	3.057	2.948	2.839	2.730	2.621	2.512	2.403	2.294	2.185	2.076	1.967
Flow coefficient, (φ)	0.706	0.681	0.656	0.630	0.605	0.580	0.554	0.529	0.504	0.478	0.453
Mass flow rate, $(\Phi); [\text{kg/sec}]$	3.052	2.943	2.833	2.723	2.614	2.504	2.395	2.285	2.176	2.066	1.956
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	4.598	5.127	5.595	5.999	6.333	6.596	6.781	6.886	6.907	6.839	6.679
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.526	0.618	0.705	0.787	0.864	0.936	1.002	1.064	1.121	1.172	1.219
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.781	0.805	0.824	0.839	0.849	0.854	0.855	0.851	0.843	0.830	0.813
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.406	0.499	0.585	0.663	0.733	0.796	0.851	0.899	0.939	0.972	0.997
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.554	0.619	0.674	0.719	0.754	0.778	0.793	0.798	0.792	0.777	0.751
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.277	0.385	0.484	0.574	0.653	0.724	0.784	0.835	0.877	0.909	0.931
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.064	0.204	0.325	0.429	0.514	0.582	0.631	0.661	0.674	0.668	0.644
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.007	0.132	0.247	0.353	0.449	0.534	0.610	0.677	0.733	0.779	0.816
Efficiency (first stage static-static), $\{[(\eta)_{s-s}]_{1\text{Stage}}\}$	0.174	0.201	0.224	0.243	0.258	0.270	0.277	0.281	0.281	0.276	0.268
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{1\text{Stage}}\}$	0.085	0.125	0.162	0.195	0.225	0.251	0.274	0.294	0.310	0.323	0.332
Efficiency (second stage static-static), $\{[(\eta)_{s-s}]_{2\text{Stage}}\}$	0.235	0.246	0.255	0.262	0.267	0.270	0.271	0.270	0.267	0.262	0.255
Pressure rise coefficient (second stage static-static), $\{[(\psi)_{s-s}]_{2\text{Stage}}\}$	0.122	0.153	0.181	0.207	0.230	0.251	0.269	0.284	0.297	0.307	0.315
Efficiency (third stage static-static), $\{[(\eta)_{s-s}]_{3\text{Stage}}\}$	0.243	0.254	0.262	0.268	0.271	0.272	0.271	0.267	0.261	0.253	0.242
Pressure rise coefficient (third stage static-static), $\{[(\psi)_{s-s}]_{3\text{Stage}}\}$	0.125	0.158	0.187	0.212	0.235	0.254	0.269	0.282	0.290	0.296	0.298

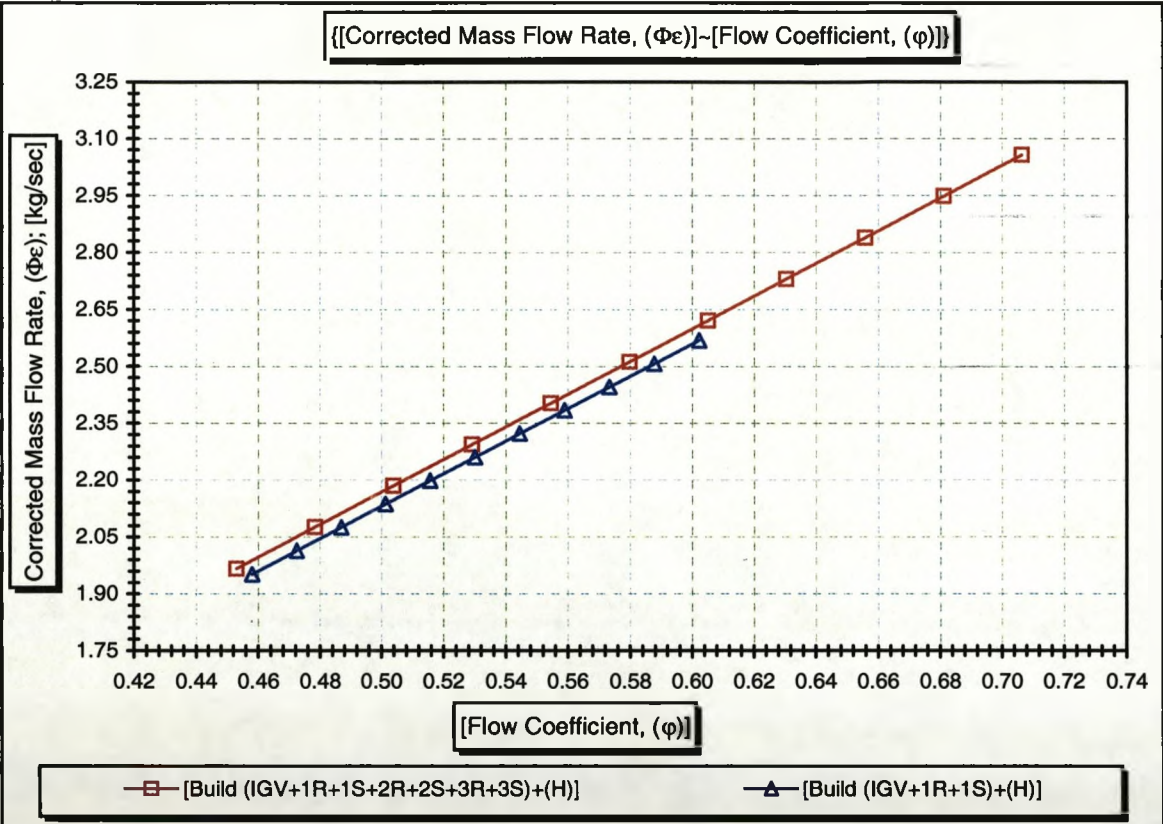
[Table (8-5)]- Flow characteristics of the three-stage compressor (before relocation of the test rig), build [(IGV+1R+1S+2R+2S+3R+3S)+(H)]

Flow coefficient at the stall point, $[(\Phi)_{SP}]$	0.458
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.949
Corrected mass flow rate at the stall point, $[(\Phi_E)_{SP}]$; [kg/sec]	1.952
Maximum corrected applied power, $[(\Omega E)_{Max}]$; [kwatt]	2.830
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi_E)/(\Omega E)_{Max}]$; [kg/sec]	2.207
Corrected applied power at the stall point, $[(\Omega E)_{SP}]$; [kwatt]	2.698
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi_E)/(\Omega E)_{SP}]$; [kg/sec]	1.952
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.502
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi_E)/(\Psi)_{Max-Overall}]$	1.952
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.755
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi_E)/[(\eta)_{Max}Overall-(t-t)]\}$	2.237
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.730
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.657
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi_E)/[(\eta)_{Max}Overall-(s-s)]\}$	2.148
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.643
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.416
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi_E)/[(\eta)_{Max}Overall-(t-s)]\}$	1.952
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.416
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.714
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi_E)/[(\eta)_{Max}Stage-(s-s)]\}$	2.287
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.678
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.369
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi_E)/[(\psi)_{Peak}Overall-(t-t)]\}$	1.957
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.369
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.324
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi_E)/[(\psi)_{Peak}Overall-(s-s)]\}$	1.952
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.324
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.211
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi_E)/[(\psi)_{Peak}Overall-(t-s)]\}$	1.952
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.211
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.343
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi_E)/[(\psi)_{Peak}Stage-(s-s)]\}$	2.009
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.342

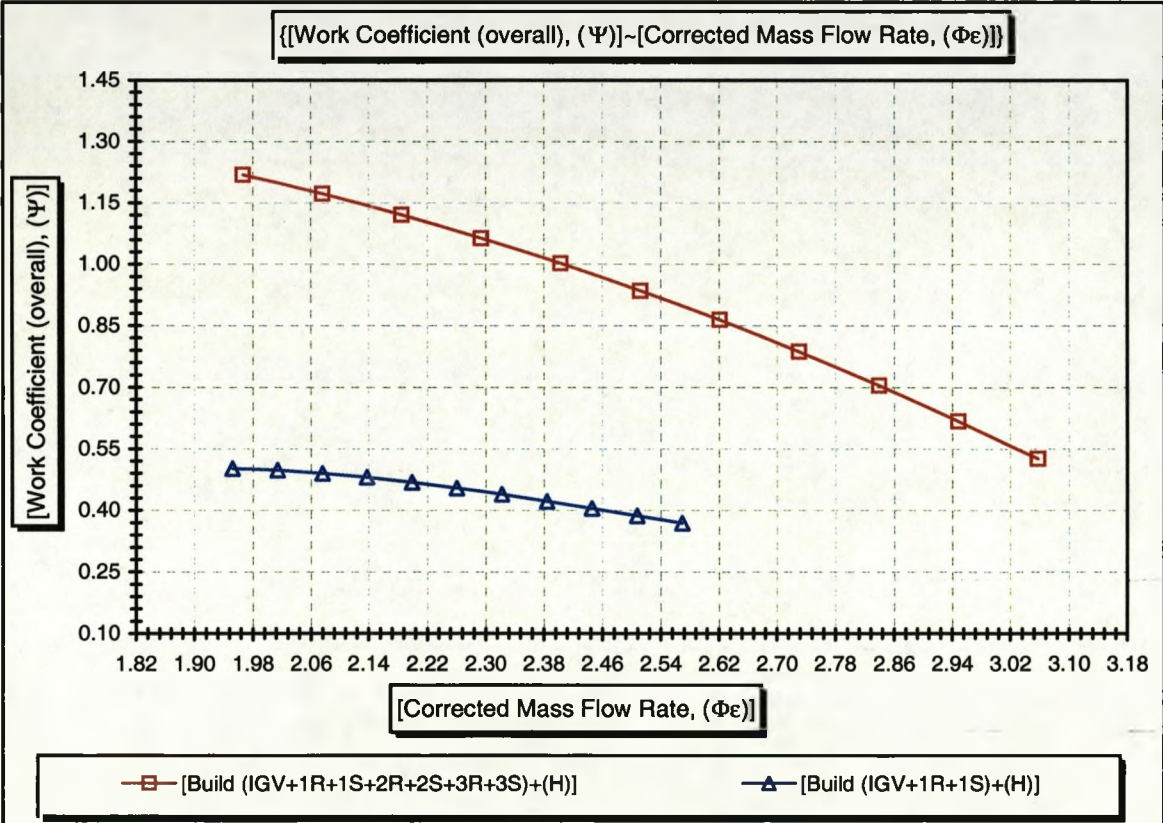
[Table (8-6)]- Some flow characteristics results related to the particular points of the one-stage compressor configuration (before relocation of test rig), build [(IGV+1R+1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.453
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.956
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.967
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	6.910
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.214
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	6.679
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.967
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	1.219
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.967
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.855
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.438
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.813
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.798
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.298
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.751
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.674
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	2.164
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.644
Maximum efficiency (first stage static-static), $\{[(\eta)_{Max}]_{1Stage-(s-s)}\}$	0.281
Corrected mass flow rate @ maximum efficiency (first stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{1Stage-(s-s)}\}$	2.248
Efficiency (first stage static-static) at the stall point, $\{[(\eta)_{SP}]_{1Stage-(s-s)}\}$	0.268
Maximum efficiency (second stage static-static), $\{[(\eta)_{Max}]_{2Stage-(s-s)}\}$	0.271
Corrected mass flow rate @ maximum efficiency (second stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{2Stage-(s-s)}\}$	2.402
Efficiency (second stage static-static) at the stall point, $\{[(\eta)_{SP}]_{2Stage-(s-s)}\}$	0.255
Maximum efficiency (third stage static-static), $\{[(\eta)_{Max}]_{3Stage-(s-s)}\}$	0.272
Corrected mass flow rate @ maximum efficiency (third stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{3Stage-(s-s)}\}$	2.520
Efficiency (third stage static-static) at the stall point, $\{[(\eta)_{SP}]_{3Stage-(s-s)}\}$	0.242
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.997
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.967
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.997
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.931
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.967
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.931
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.816
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.967
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.816
Peak pressure rise coefficient (first stage static-static), $\{[(\psi)_{Peak}]_{1Stage-(s-s)}\}$	0.332
Corrected mass flow rate @ peak pressure rise coefficient (first stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{1Stage-(s-s)}\}$	1.967
Pressure rise coefficient (first stage static-static) at the stall point, $\{[(\psi)_{SP}]_{1Stage-(s-s)}\}$	0.332
Peak pressure rise coefficient (second stage static-static), $\{[(\psi)_{Peak}]_{2Stage-(s-s)}\}$	0.315
Corrected mass flow rate @ peak pressure rise coefficient (second stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{2Stage-(s-s)}\}$	1.967
Pressure rise coefficient (second stage static-static) at the stall point, $\{[(\psi)_{SP}]_{2Stage-(s-s)}\}$	0.315
Peak pressure rise coefficient (third stage static-static), $\{[(\psi)_{Peak}]_{3Stage-(s-s)}\}$	0.298
Corrected mass flow rate @ peak pressure rise coefficient (third stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{3Stage-(s-s)}\}$	1.967
Pressure rise coefficient (third stage static-static) at the stall point, $\{[(\psi)_{SP}]_{3Stage-(s-s)}\}$	0.298

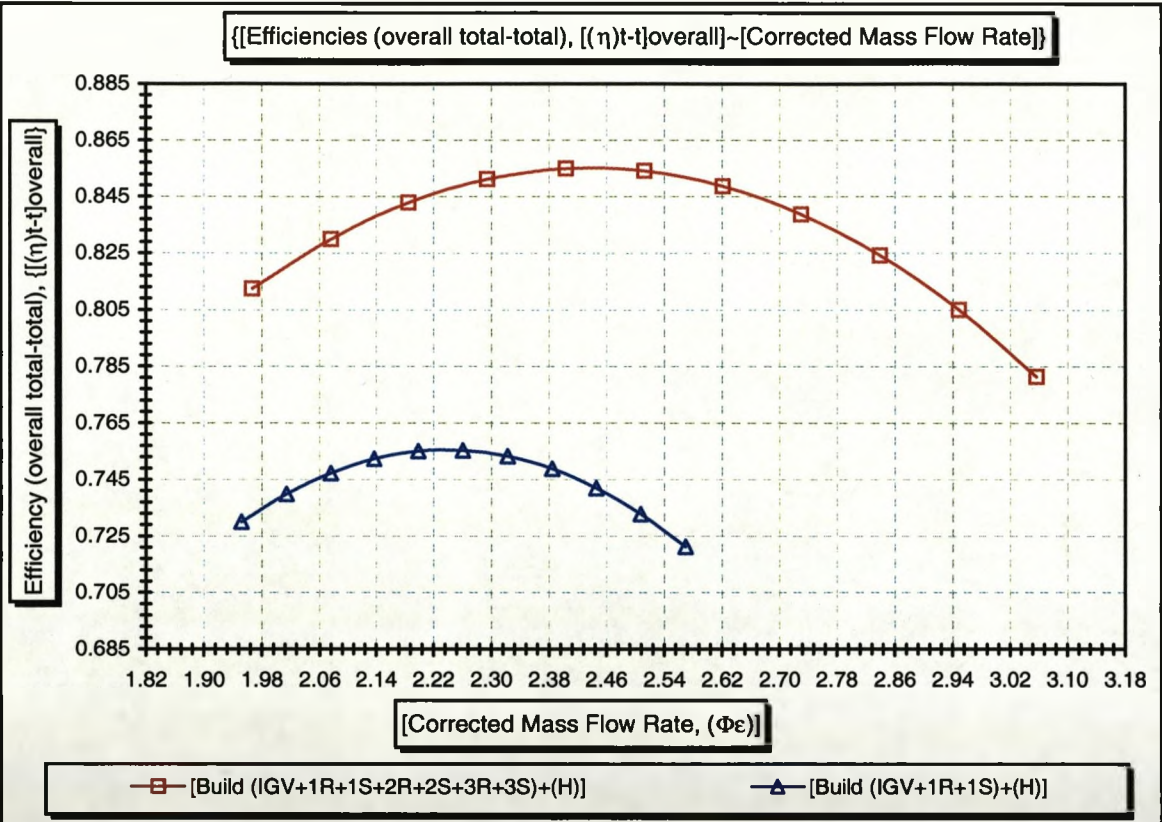
[Table (8-7)]- Some flow characteristics results related to the particular points of the three-stage compressor configuration, build [(IGV+1R+1S+2R+2S+3R+3S)+(H)]



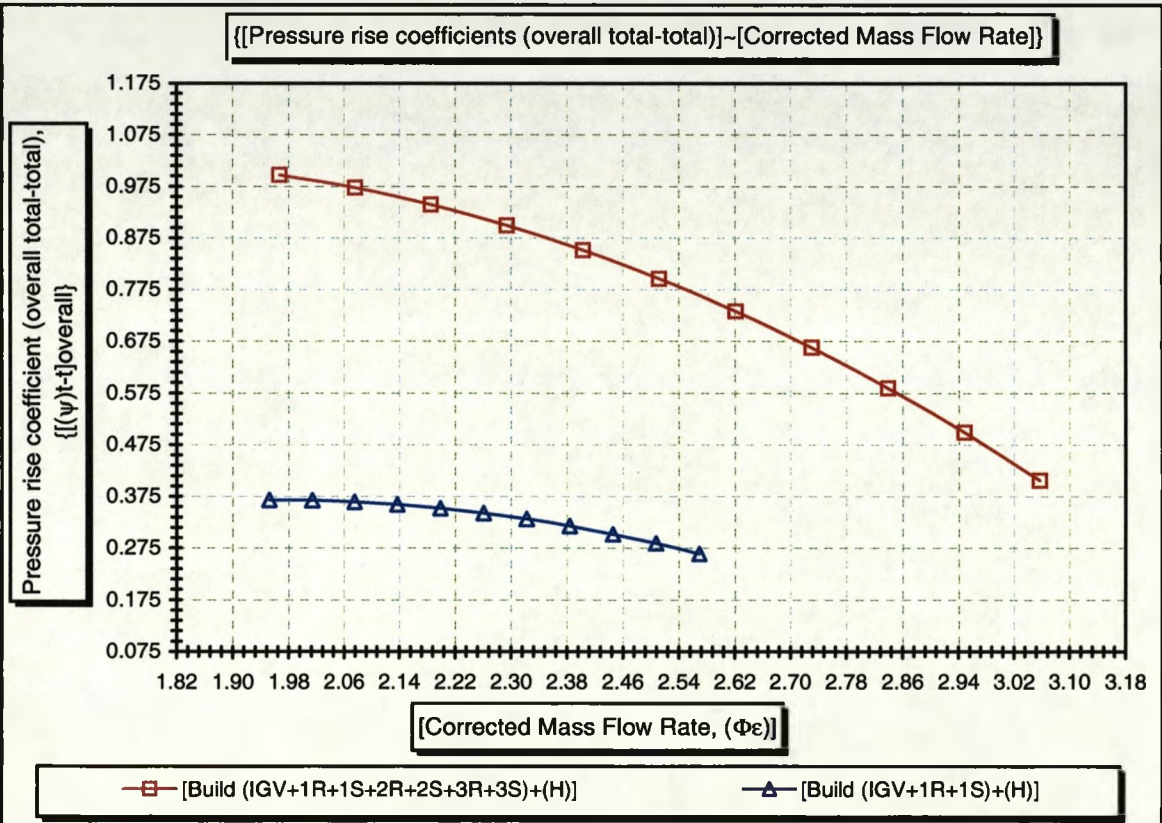
[Graph (8-5)]- Comparison between the corrected mass flow rates obtained from the flow characteristics of the three-stage and the one-stage compressor configurations



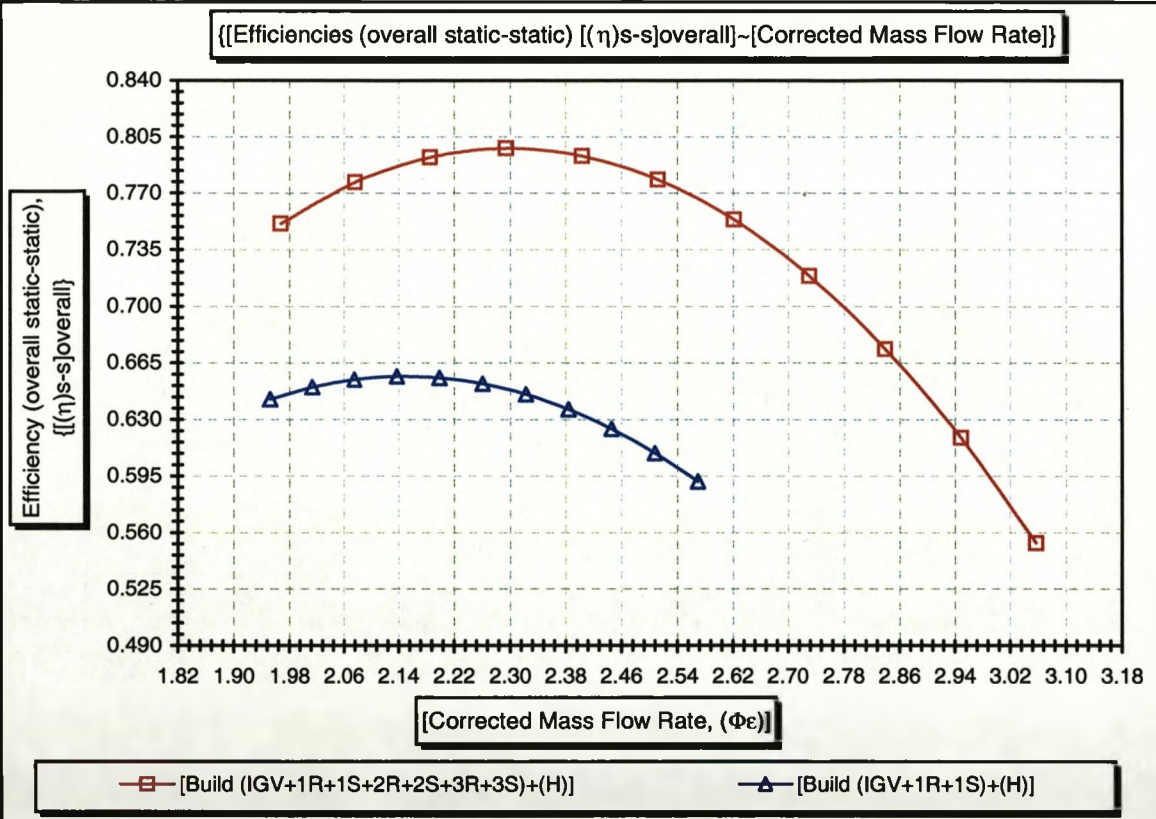
[Graph (8-6)]- Comparison in the overall work coefficients corresponding to the flow characteristics of the three-stage and the one-stage compressor configurations



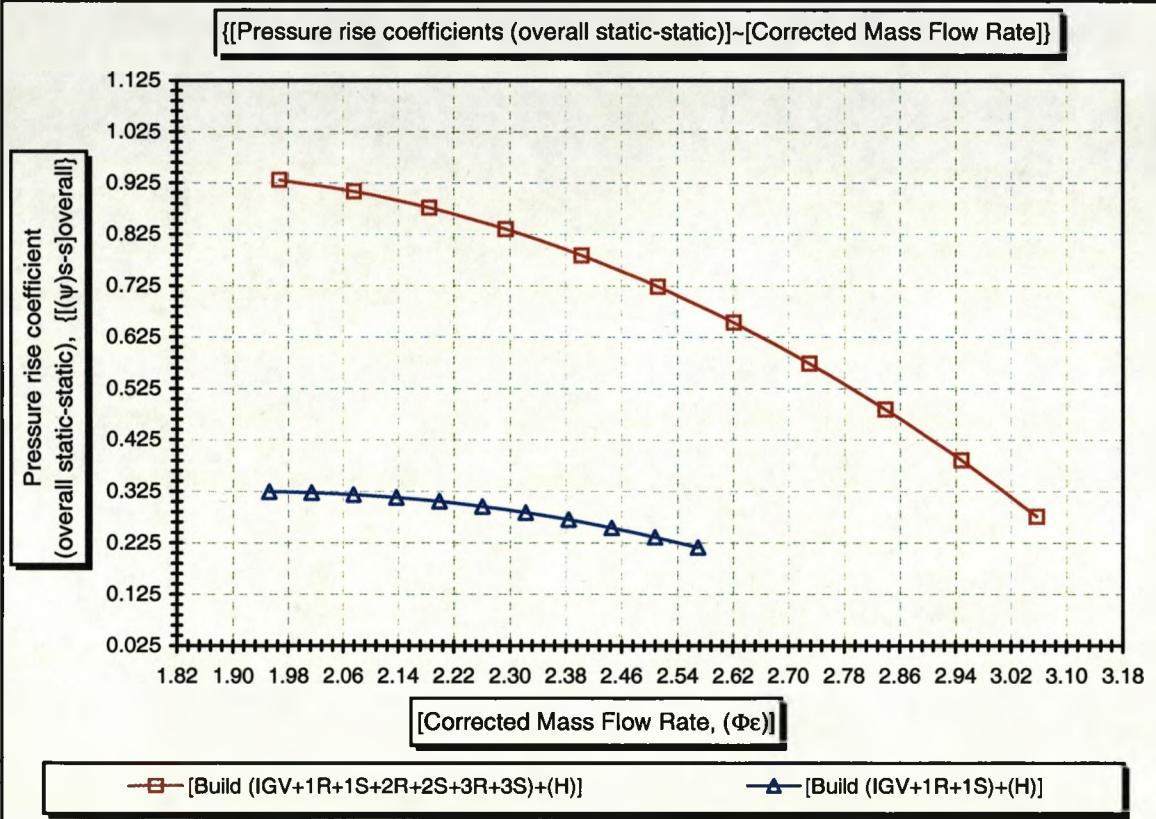
[Graph (8-7)]- Comparison of the efficiencies in terms of the overall total-total pressure ratio corresponding to flow characteristics of three-stage and one-stage compressor builds



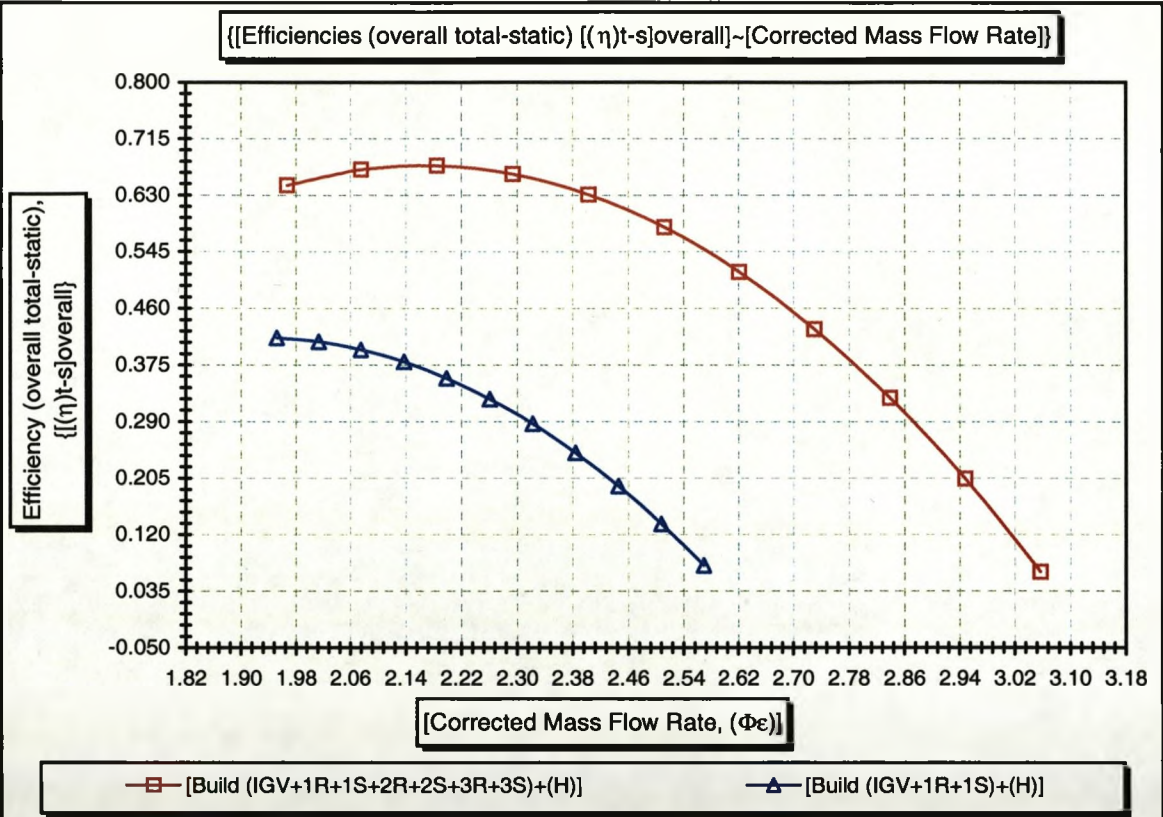
[Graph (8-8)]- Comparison of the overall total-total pressure rise coefficients related to flow characteristics obtained from three-stage and one-stage compressors configurations



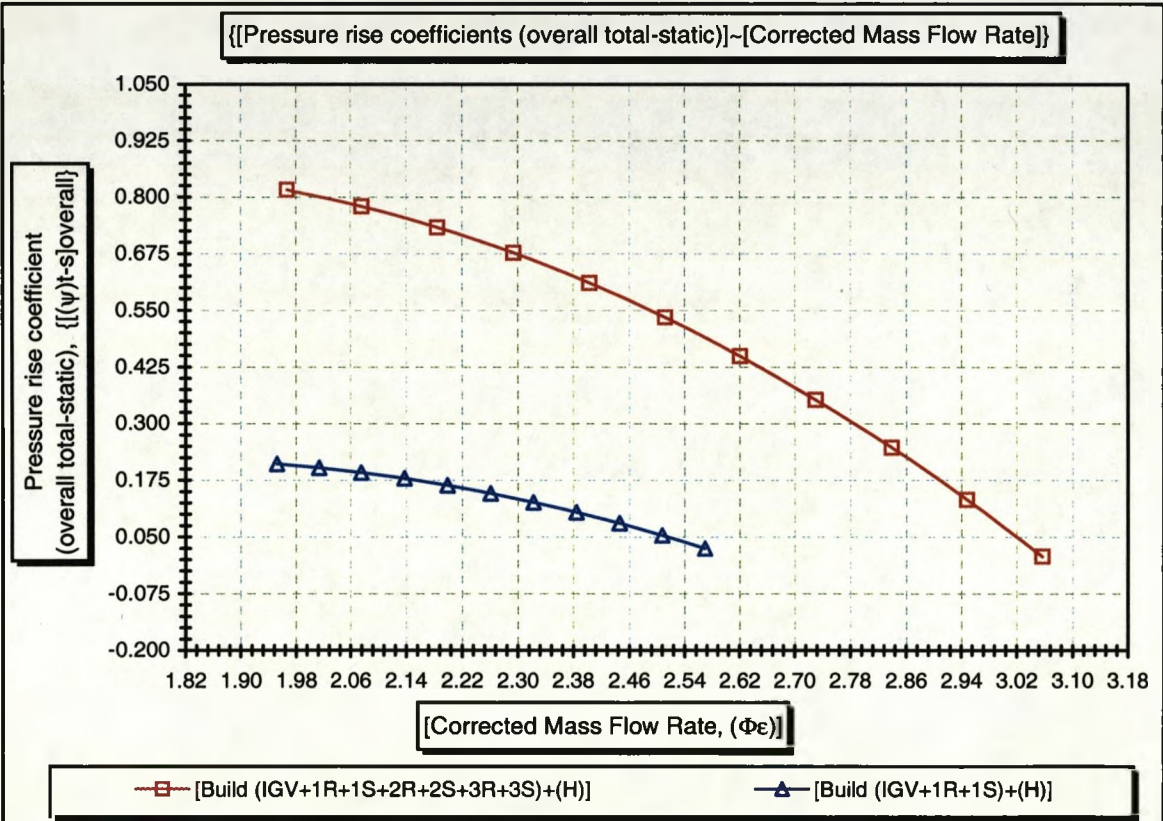
[Graph (8-9)]- Alterations of the overall static-static efficiencies obtained from the flow characteristics of the three-stage and the one-stage compressor configurations



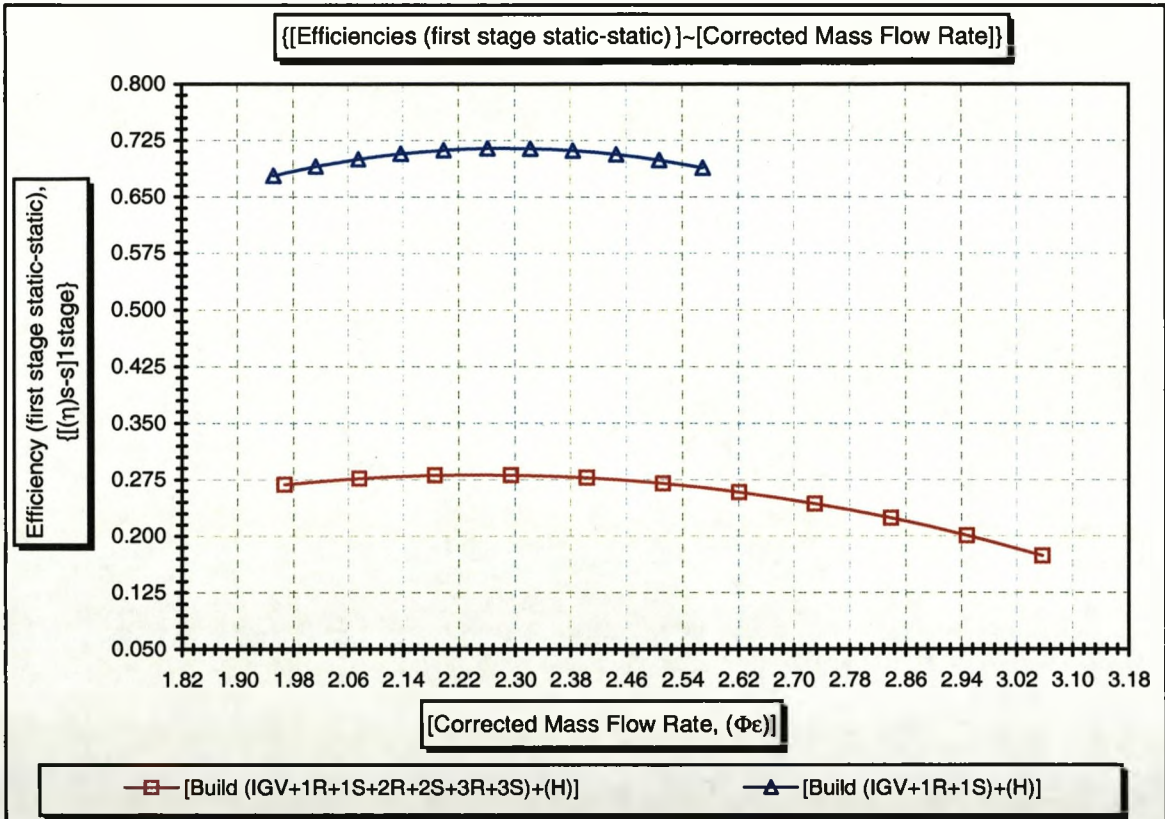
[Graph (8-10)]- Difference between the overall static-static pressure rise coefficients achieved from the flow characteristics of the three-stage and one-stage compressors



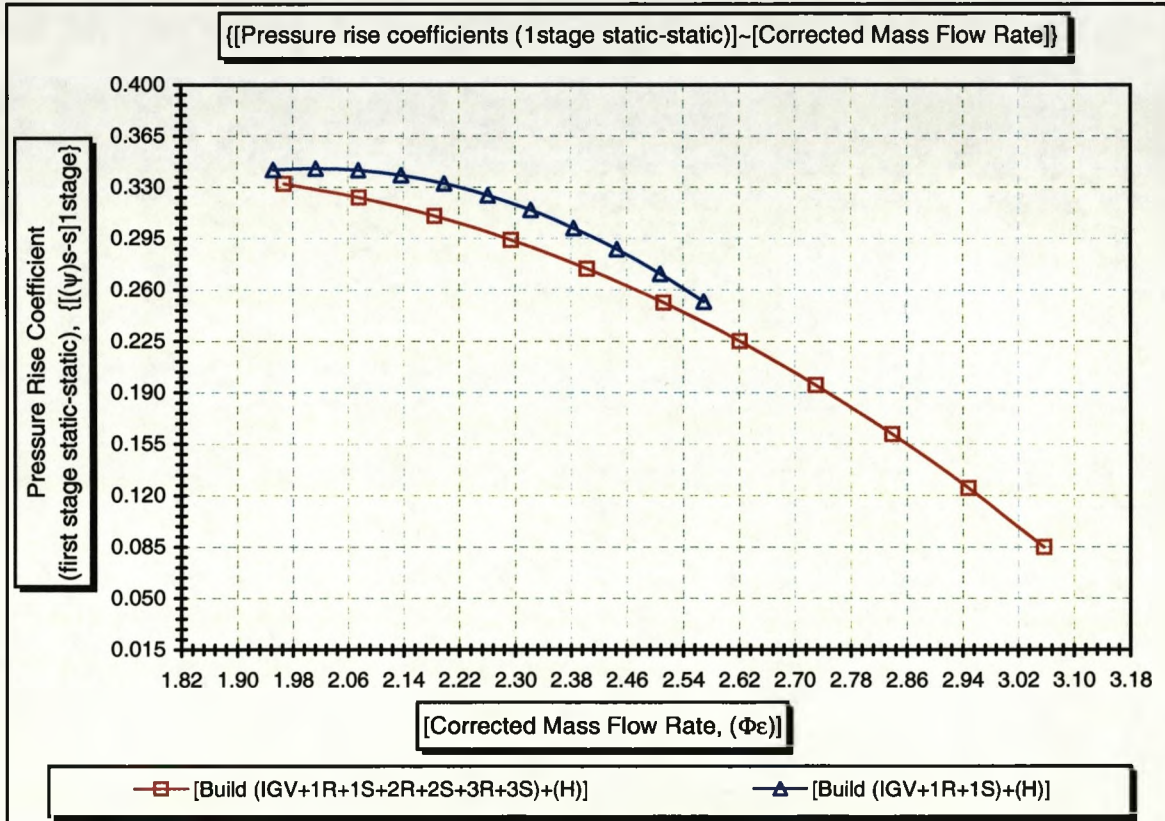
[Graph (8-11)]- The efficiencies alterations which were calculated in terms of the overall total-static pressure ratios, of the three-stage and the one-stage compressor builds



[Graph (8-12)]- The overall total-static pressure rise coefficients which were achieved from the flow characteristics of the three-stage and one-stage compressors configurations



[Graph (8-13)]- changes of the first stage static-static efficiencies obtained from the flow characteristics of the three-stage and the one-stage compressor configurations



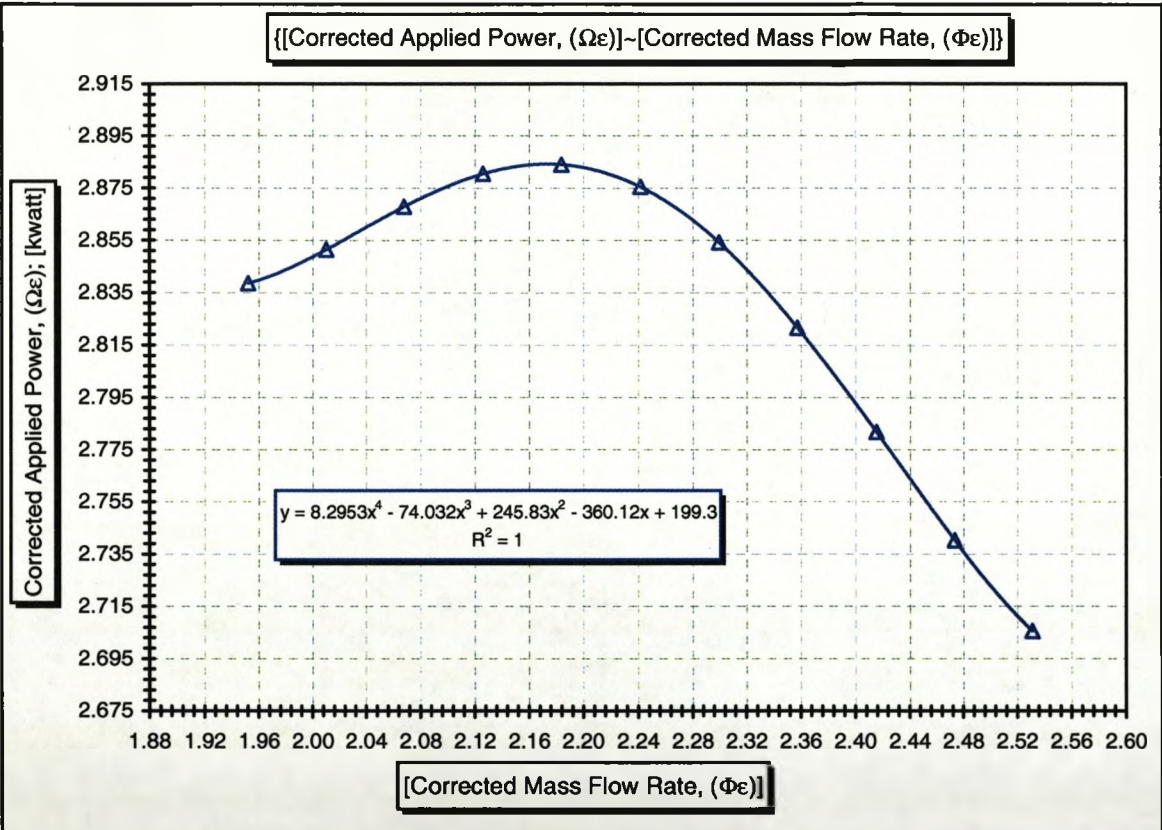
[Graph (8-14)]- Alterations of the first stage static-static pressure rise coefficients obtained from the flow characteristics of the three-stage and the one-stage compressors

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.531	2.473	2.415	2.357	2.299	2.241	2.184	2.126	2.068	2.010	1.952
Flow coefficient, (ϕ)	0.592	0.579	0.565	0.552	0.538	0.525	0.511	0.498	0.484	0.471	0.457
Mass flow rate, $(\Phi); ([\text{kg/sec}]$	2.534	2.475	2.416	2.358	2.299	2.240	2.181	2.123	2.064	2.005	1.946
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.706	2.740	2.782	2.822	2.854	2.875	2.884	2.880	2.868	2.851	2.839
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.383	0.400	0.416	0.432	0.447	0.462	0.477	0.491	0.503	0.515	0.526
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.715	0.728	0.738	0.744	0.746	0.744	0.739	0.729	0.716	0.699	0.679
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.273	0.292	0.308	0.322	0.334	0.344	0.352	0.357	0.360	0.360	0.358
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.555	0.579	0.599	0.615	0.626	0.634	0.639	0.639	0.635	0.627	0.615
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.211	0.231	0.250	0.266	0.281	0.293	0.303	0.312	0.318	0.322	0.325
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.079	0.139	0.192	0.239	0.279	0.313	0.340	0.361	0.376	0.384	0.386
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.028	0.056	0.082	0.105	0.126	0.145	0.161	0.175	0.188	0.197	0.205
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.672	0.685	0.695	0.702	0.705	0.705	0.702	0.695	0.685	0.672	0.655
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.256	0.274	0.290	0.304	0.316	0.326	0.334	0.340	0.344	0.346	0.345

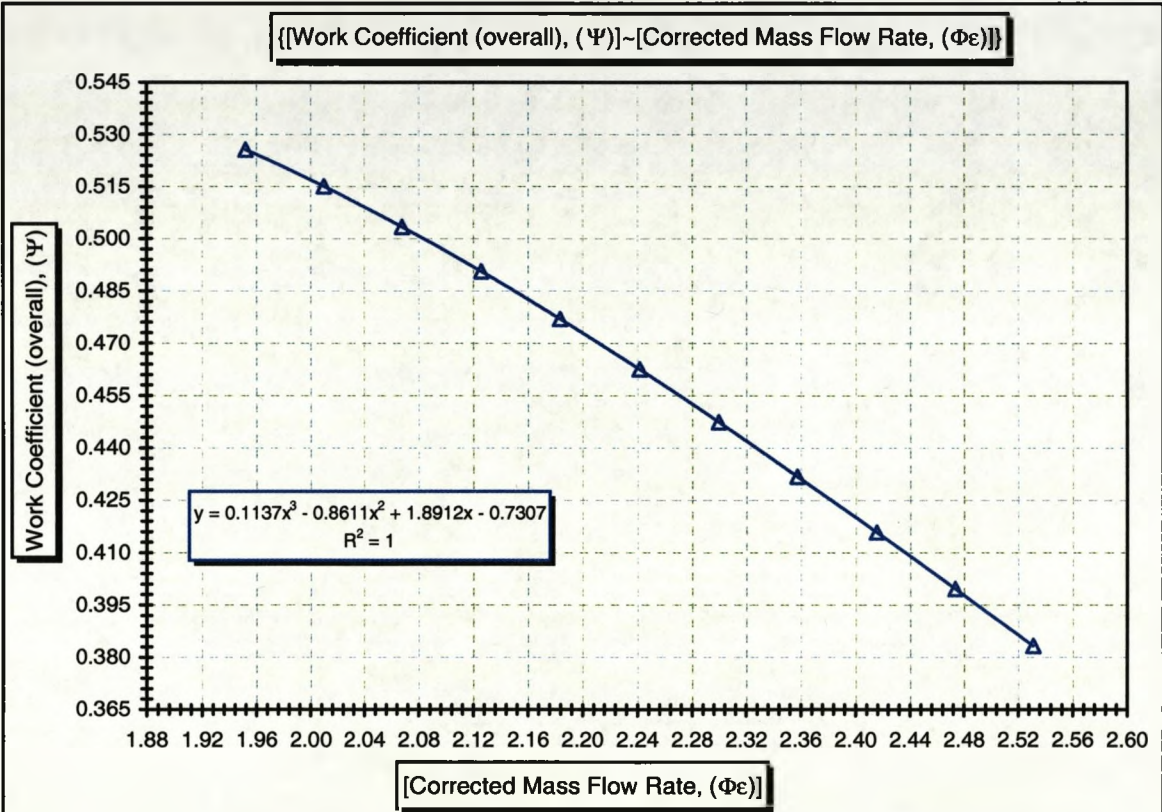
[Table (8-8)]- The flow characteristics of the one-stage solid casing (datum) compressor configuration, build [(IGV)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.457
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.946
Corrected mass flow rate at the stall point, $[(\Phi_E)_{SP}]$; [kg/sec]	1.952
Maximum corrected applied power, $[(\Omega E)_{Max}]$; [kwatt]	2.884
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi_E)/(\Omega E)_{Max}]$; [kg/sec]	2.173
Corrected applied power at the stall point, $[(\Omega E)_{SP}]$; [kwatt]	2.839
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi_E)/(\Omega E)_{SP}]$; [kg/sec]	1.952
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.526
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi_E)/(\Psi)_{Max-Overall}]$	1.952
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.746
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.297
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.679
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.639
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.154
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.615
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.386
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.964
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.386
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.706
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.271
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.655
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.360
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	2.026
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.358
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.325
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.952
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.325
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.205
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.952
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.205
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.346
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.987
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.345

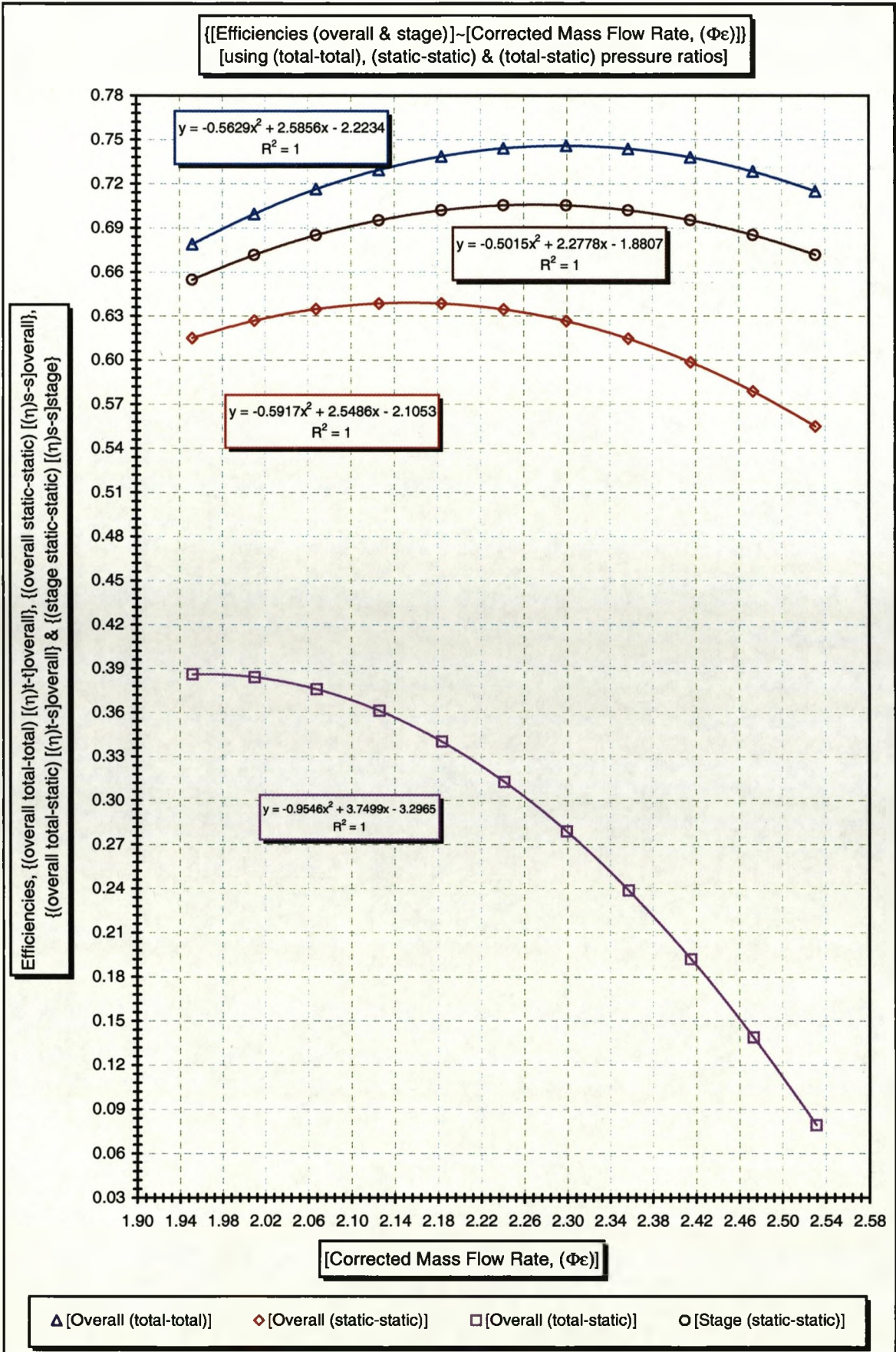
[Table (8-9)]- The flow characteristics results corresponding to the particular points of the one-stage solid casing compressor (datum) configuration, build [(IGV)+(1R)+(1S)+(H)]



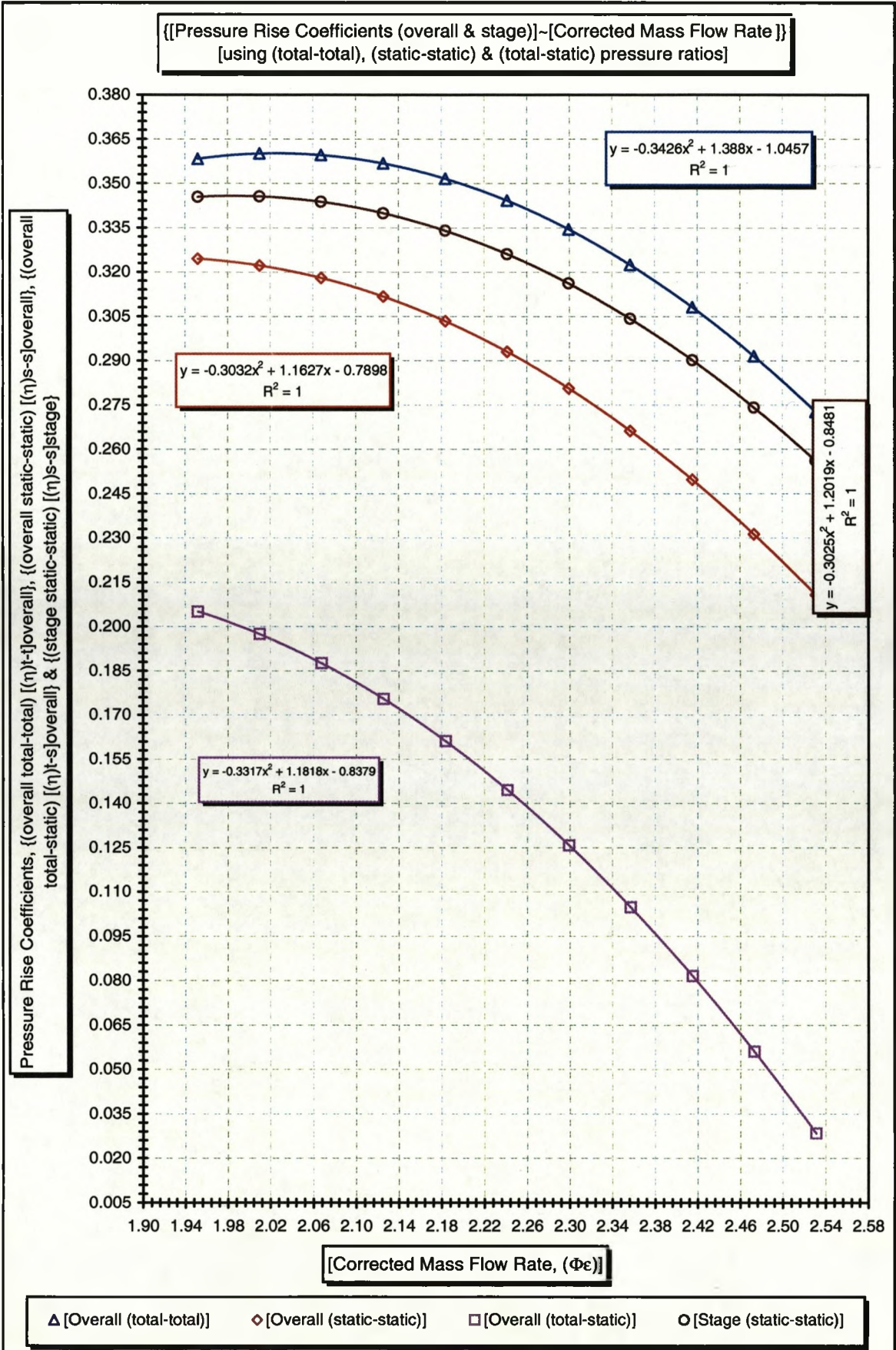
[Graph (8-15)]- The corrected applied power that obtained from the flow characteristics of the one-stage solid casing compressor configuration, build [(IGV)+(1R)+(1S)+(H)]



[Graph (8-16)]- The overall work coefficient, obtained from the flow characteristics of the one-stage solid casing compressor (datum configuration), build



[Graph (8-17)]- The overall and stage efficiencies obtained from flow characteristics of one-stage solid casing compressor (datum configuration), build [(IGV)+(1R)+(1S)+(H)]



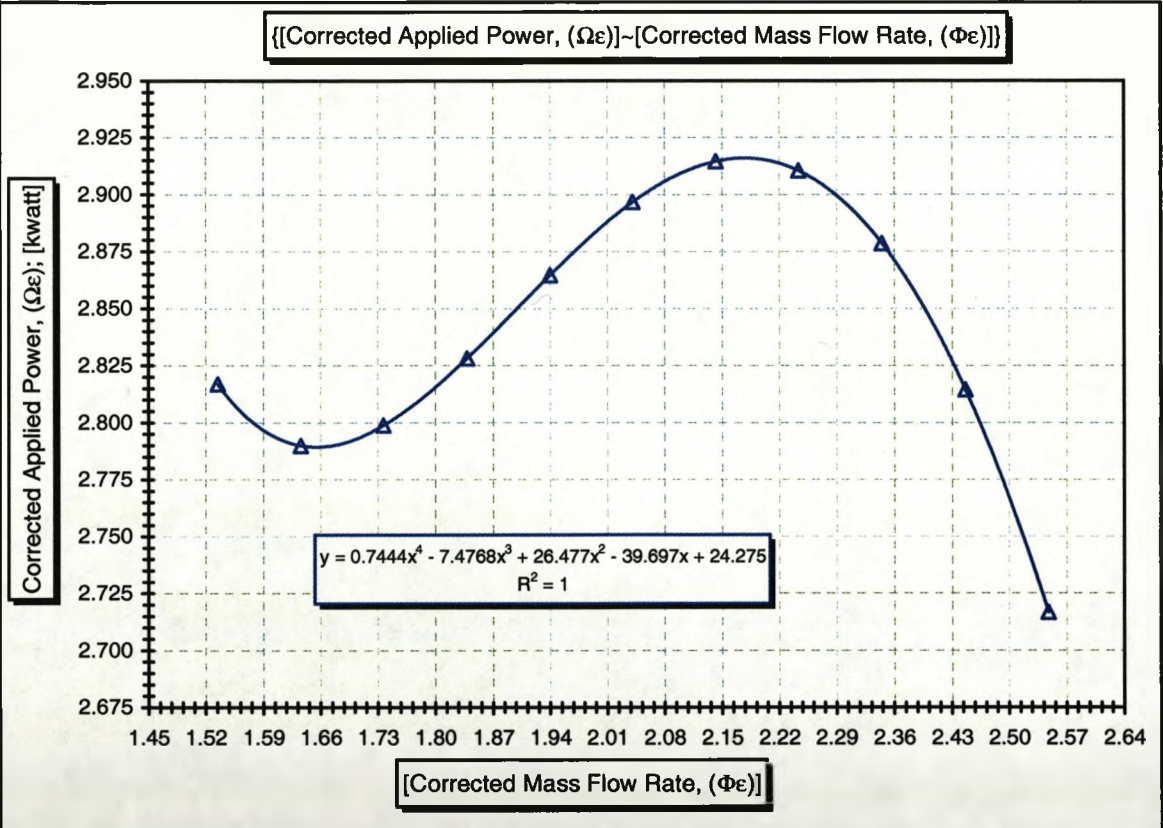
[Graph (8-18)]- The overall and stage pressure rise coefficients obtained from the flow characteristics of the one-stage solid casing compressor, build [(IGV)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.548	2.446	2.345	2.244	2.142	2.041	1.940	1.838	1.737	1.636	1.534
Flow coefficient, (ϕ)	0.593	0.569	0.546	0.522	0.499	0.475	0.452	0.428	0.405	0.381	0.358
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.529	2.428	2.327	2.226	2.125	2.024	1.923	1.823	1.722	1.621	1.520
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.717	2.815	2.878	2.910	2.915	2.897	2.865	2.828	2.799	2.790	2.817
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.380	0.414	0.442	0.466	0.487	0.507	0.529	0.553	0.581	0.616	0.658
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.726	0.735	0.738	0.733	0.721	0.702	0.675	0.642	0.602	0.555	0.500
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.280	0.303	0.323	0.338	0.349	0.356	0.359	0.357	0.351	0.341	0.327
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.559	0.589	0.611	0.624	0.628	0.623	0.610	0.588	0.557	0.517	0.469
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.214	0.243	0.268	0.289	0.305	0.317	0.324	0.326	0.324	0.318	0.307
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.084	0.169	0.241	0.300	0.346	0.379	0.399	0.406	0.400	0.381	0.348
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.029	0.071	0.109	0.141	0.169	0.192	0.209	0.222	0.231	0.234	0.233
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.674	0.687	0.694	0.693	0.686	0.671	0.650	0.622	0.586	0.544	0.495
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.259	0.284	0.304	0.320	0.333	0.341	0.345	0.346	0.342	0.335	0.323

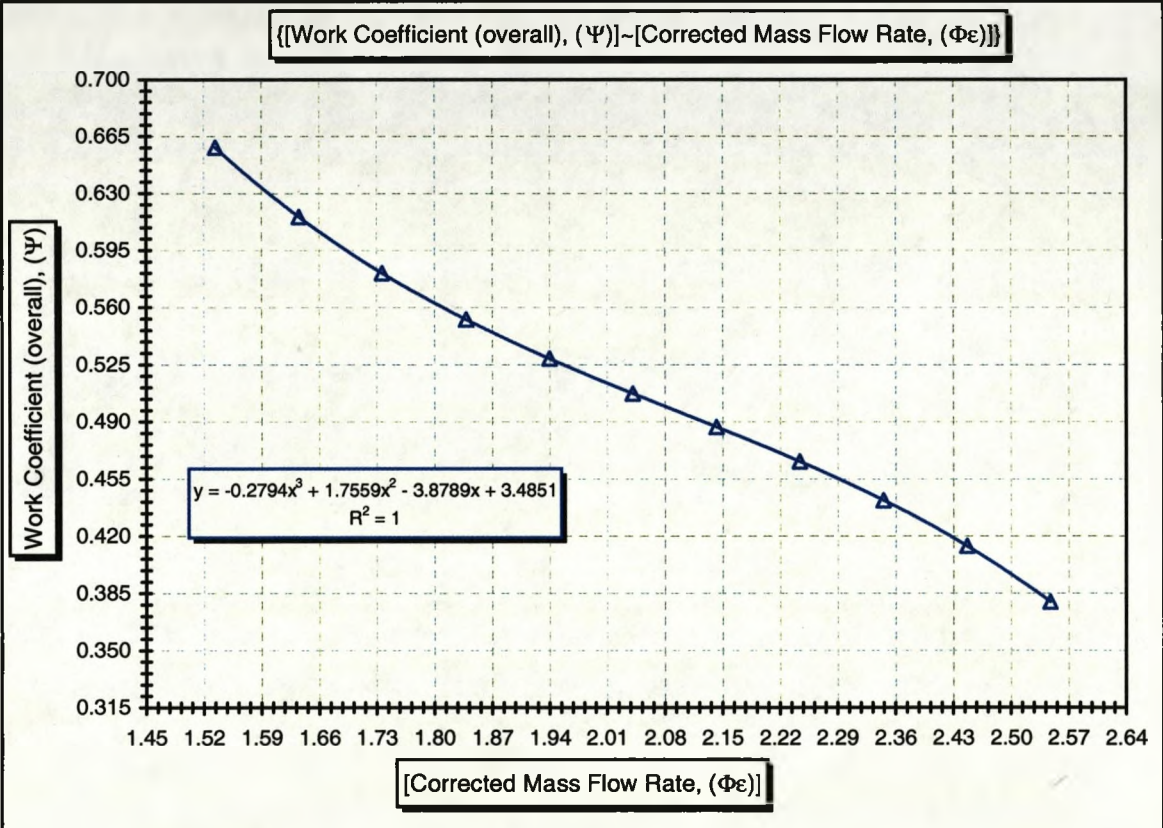
[Table (8-10)]- The flow characteristics of the one-stage compressor with casing treatment configuration of (0.232) rotor blade tip axial chord exposure ratio, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.358
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.520
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.534
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.916
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.177
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.817
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.534
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.658
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.534
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.738
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.365
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.500
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.628
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.145
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.469
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.406
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.835
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.348
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.694
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.302
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.495
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.359
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.927
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.327
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.326
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.829
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.307
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.234
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.616
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.233
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.346
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.879
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.323

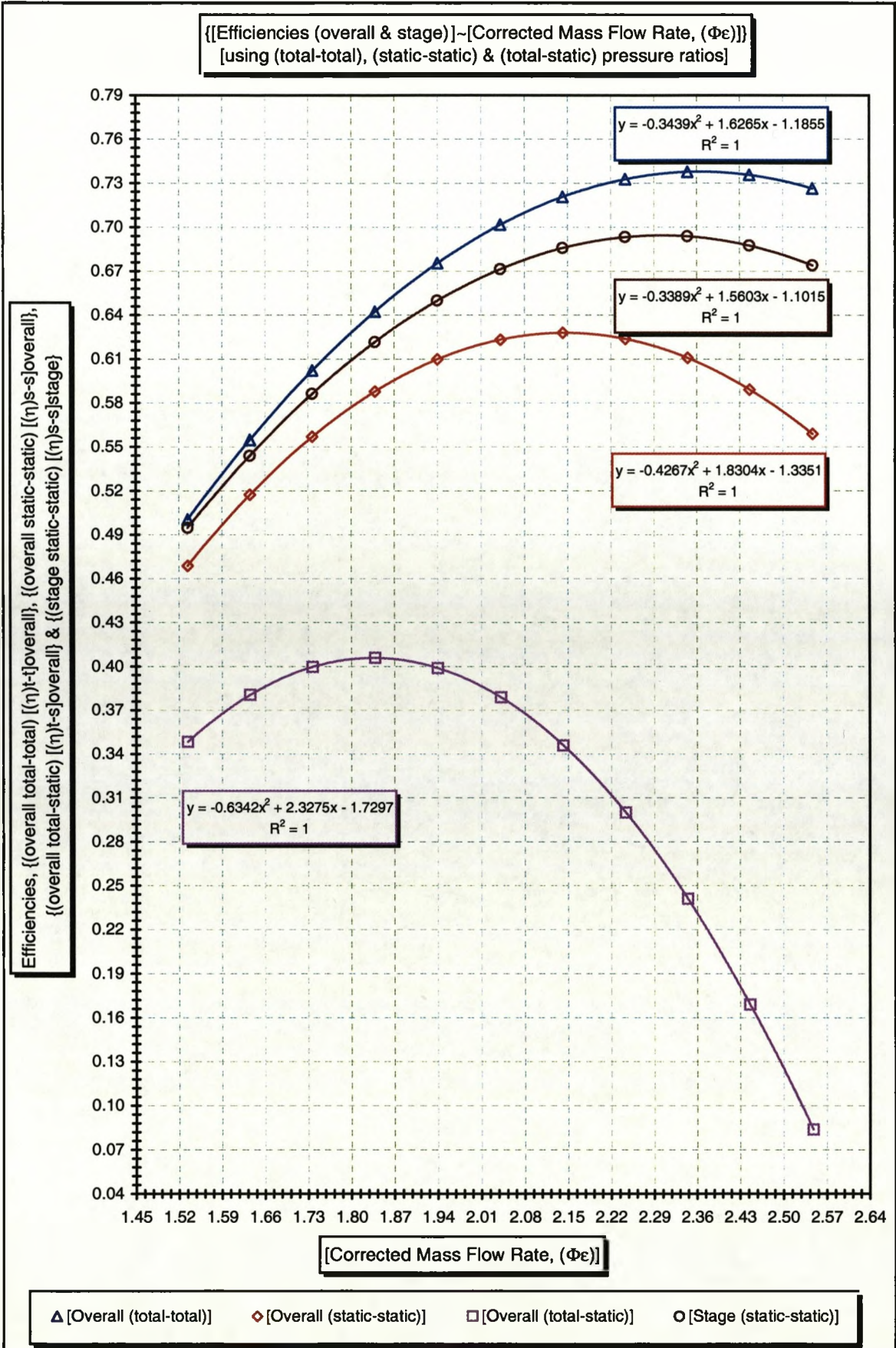
[Table (8-11)]- Characteristics of the one-stage compressor including the casing treatment with (0.232) rotor tip exposure ratio, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]



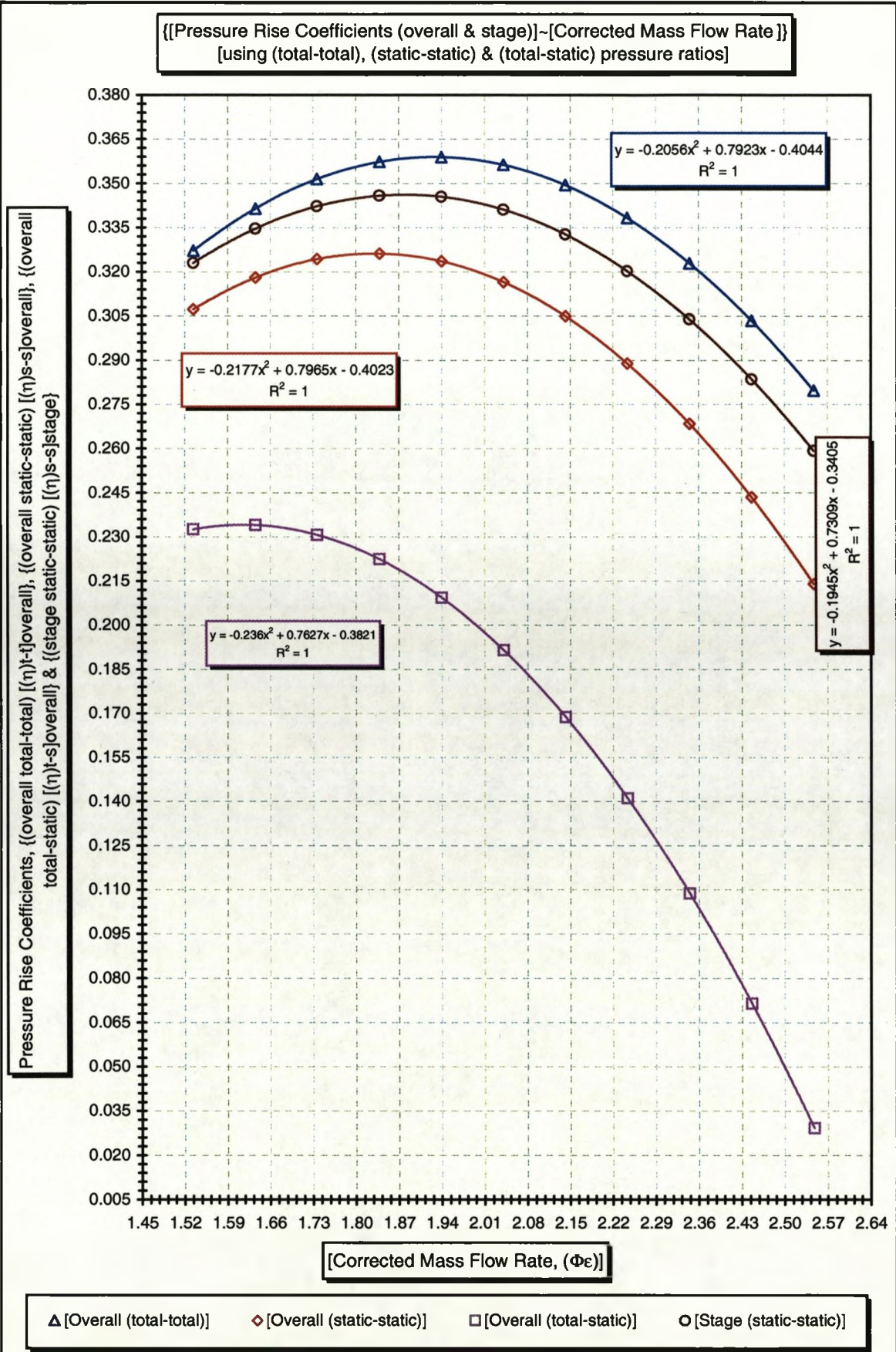
[Graph (8-19)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]



[Graph (8-20)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]



[Graph (8-21)]- The overall and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]



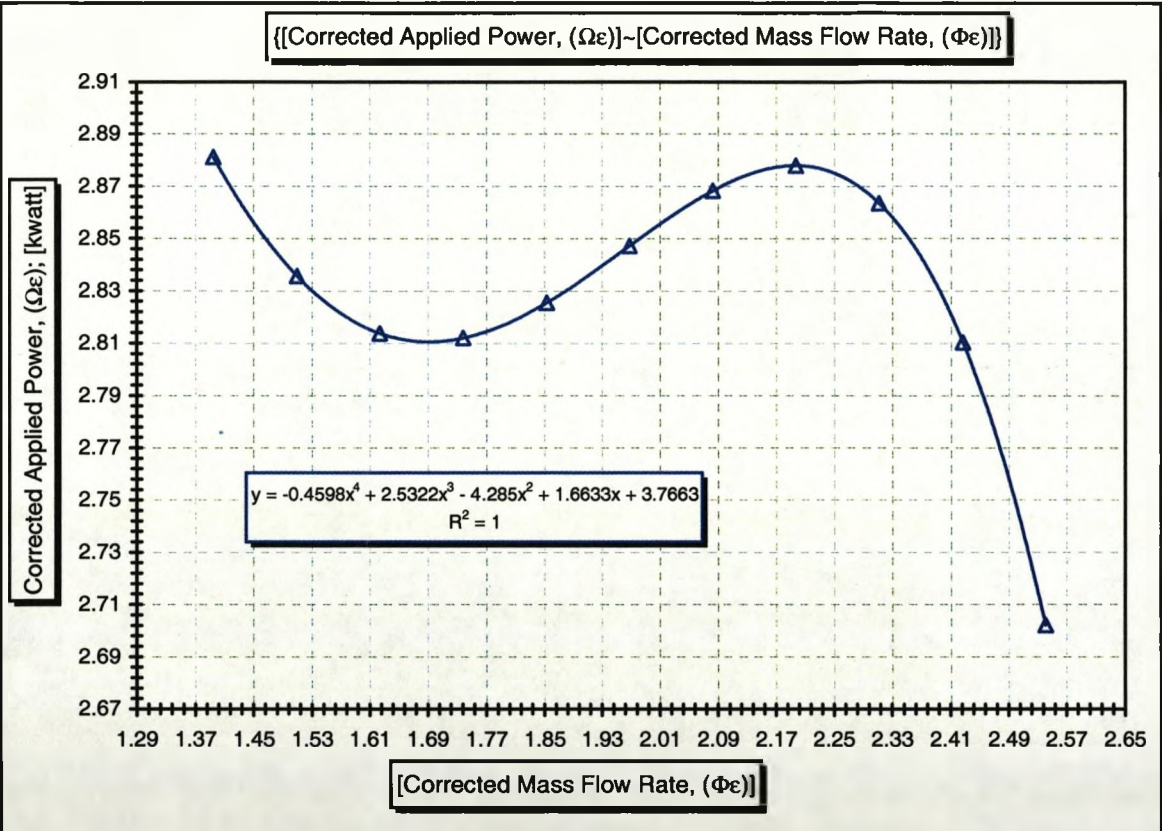
[Graph (8-22)]- The overall and stage pressure rise coefficients against flow rate of one-stage compressor with treatment, build [(IGV)+(TC-23.2%)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.541	2.426	2.311	2.197	2.082	1.968	1.853	1.738	1.624	1.509	1.394
Flow coefficient, (ϕ)	0.593	0.567	0.540	0.513	0.486	0.459	0.432	0.406	0.379	0.352	0.325
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.498	2.386	2.273	2.161	2.048	1.936	1.823	1.711	1.598	1.485	1.373
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.702	2.810	2.864	2.878	2.868	2.847	2.826	2.812	2.814	2.836	2.881
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.384	0.419	0.448	0.474	0.497	0.522	0.550	0.584	0.626	0.678	0.744
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.722	0.730	0.731	0.723	0.707	0.683	0.650	0.610	0.561	0.504	0.439
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.282	0.305	0.325	0.340	0.351	0.357	0.360	0.358	0.352	0.341	0.327
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.555	0.589	0.612	0.624	0.626	0.617	0.597	0.566	0.525	0.473	0.410
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.215	0.246	0.273	0.294	0.311	0.323	0.329	0.331	0.328	0.320	0.306
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.088	0.179	0.256	0.317	0.363	0.394	0.410	0.410	0.395	0.365	0.320
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.032	0.077	0.117	0.152	0.181	0.205	0.224	0.237	0.244	0.247	0.243
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.662	0.682	0.691	0.691	0.681	0.662	0.632	0.593	0.544	0.486	0.417
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.257	0.285	0.308	0.326	0.339	0.347	0.350	0.348	0.340	0.328	0.311

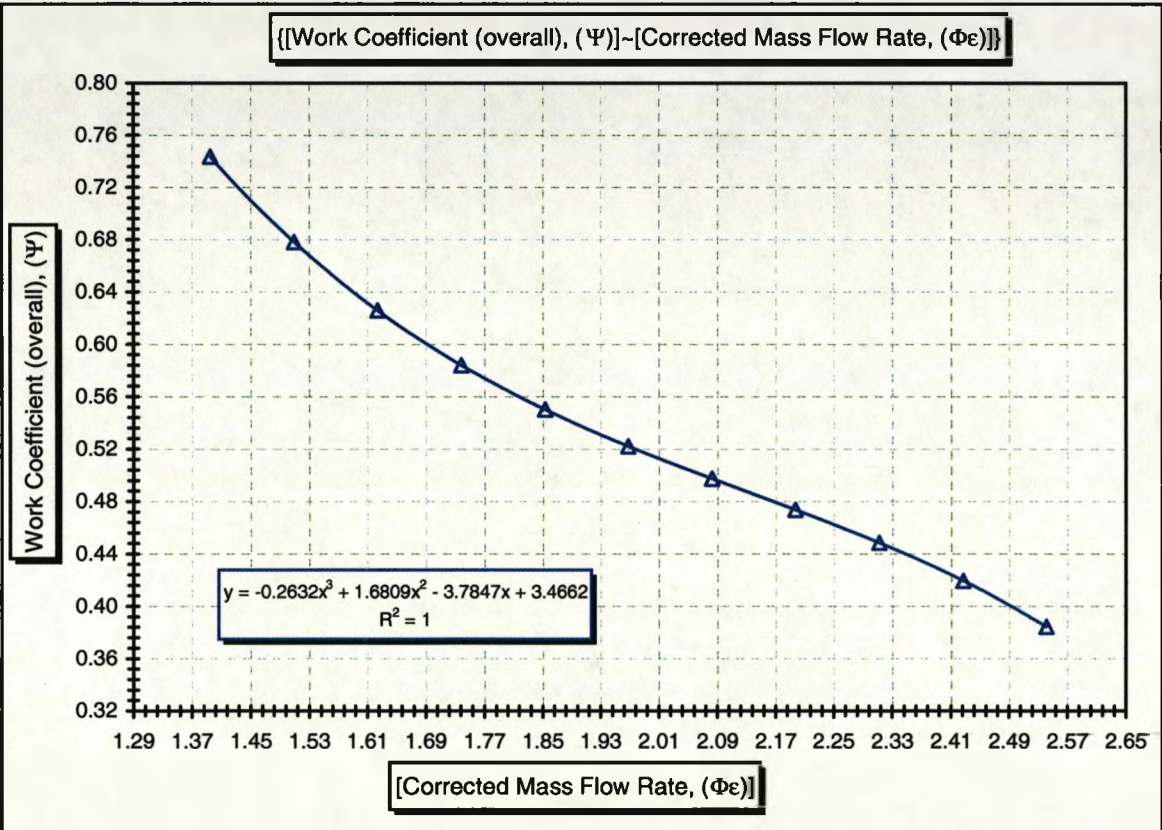
[Table (8-12)]- The flow characteristics of the one-stage compressor with casing treatment configuration of (0.333) rotor blade tip axial chord exposure ratio, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.325
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.373
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.394
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.878
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.196
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.881
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.394
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.744
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.394
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.731
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.363
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.439
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.626
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.122
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.410
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.412
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.793
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.320
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.693
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.255
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.417
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.360
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.846
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.327
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.331
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.756
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.306
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.247
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.520
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.243
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.350
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.846
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.311

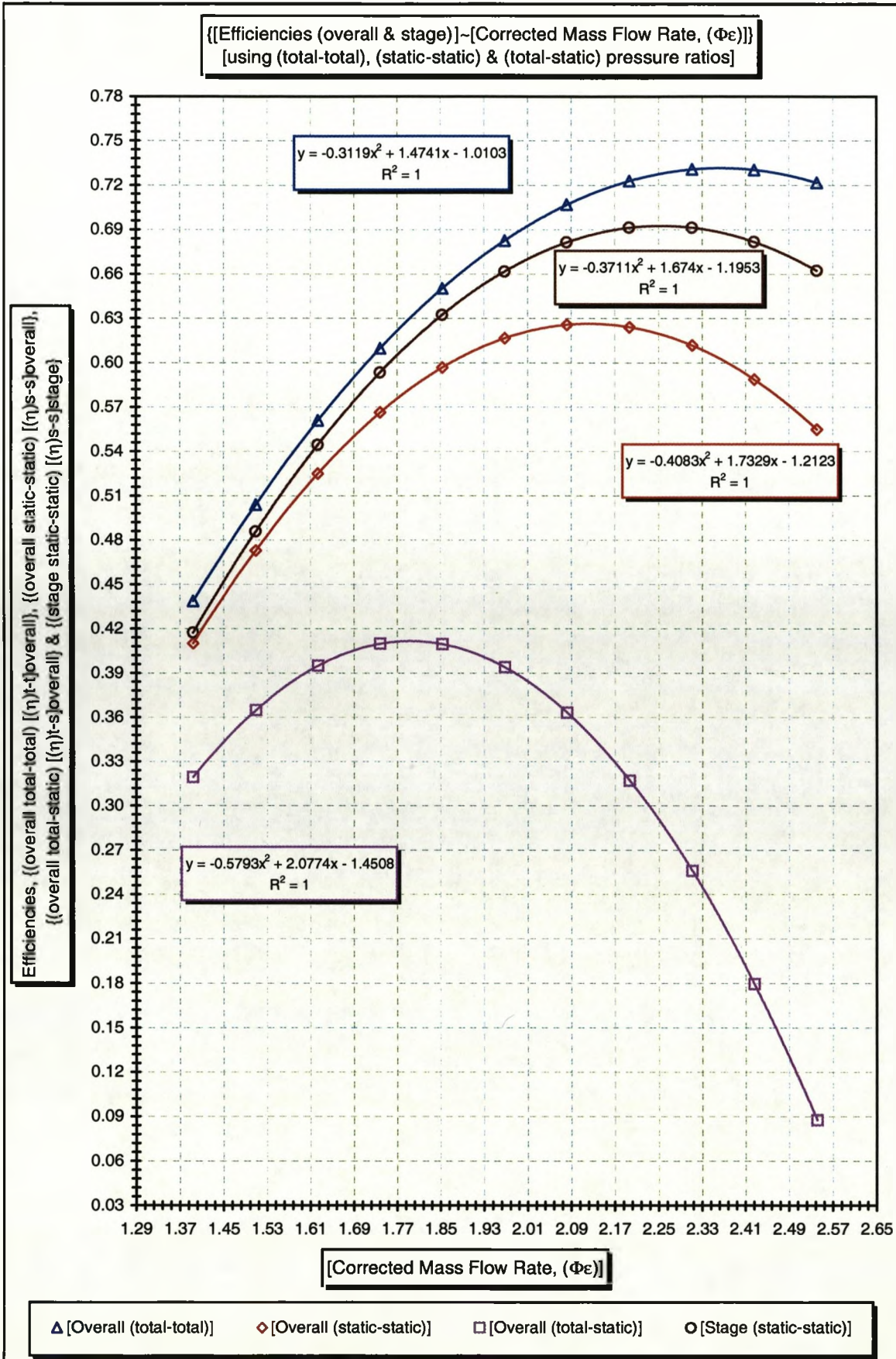
[Table (8-13)]- The characteristics of the one-stage compressor with (0.333) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)]



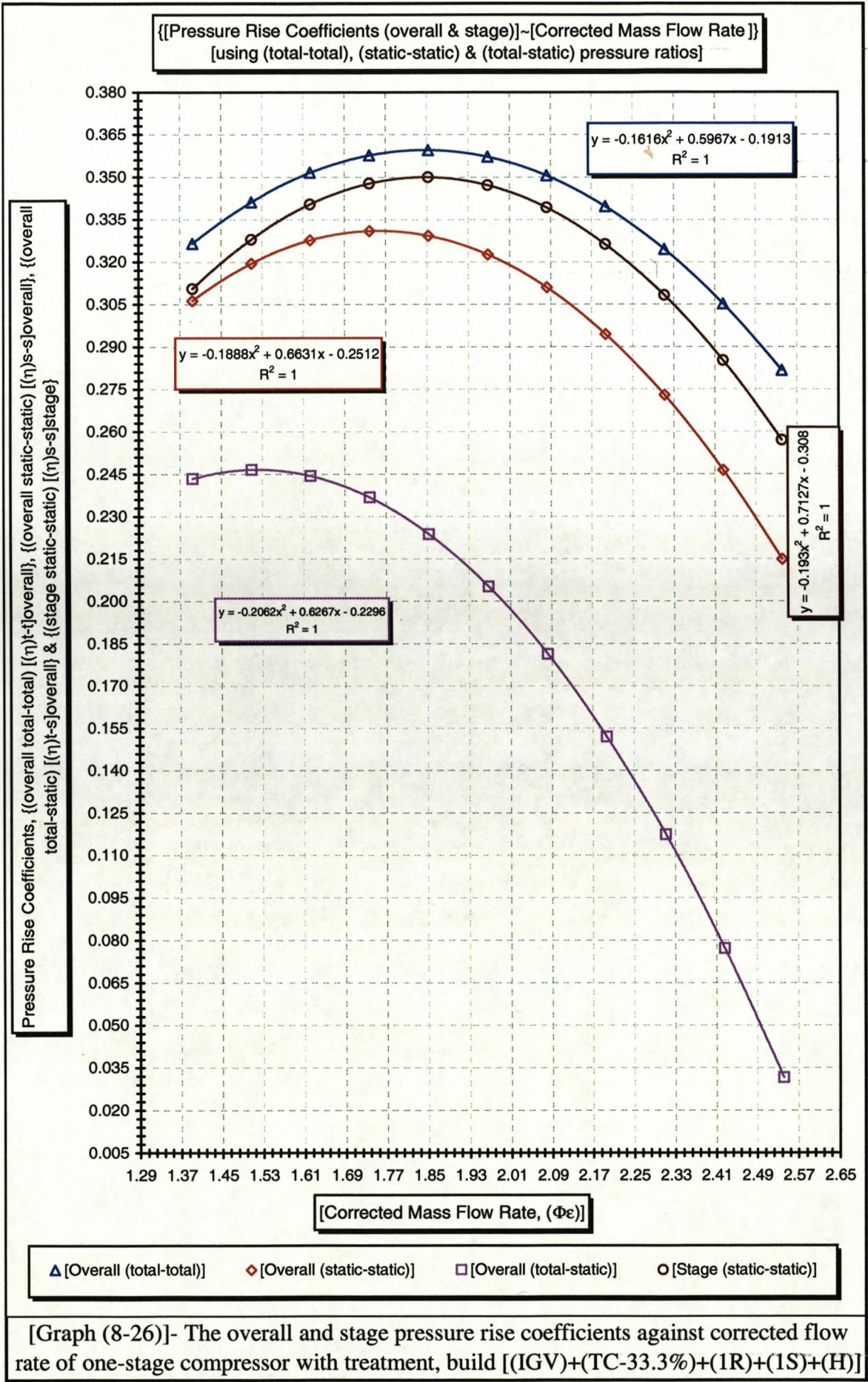
[Graph (8-23)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)]



[Graph (8-24)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)]



[Graph (8-25)]- The overall and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-33.3%)+(1R)+(1S)+(H)]

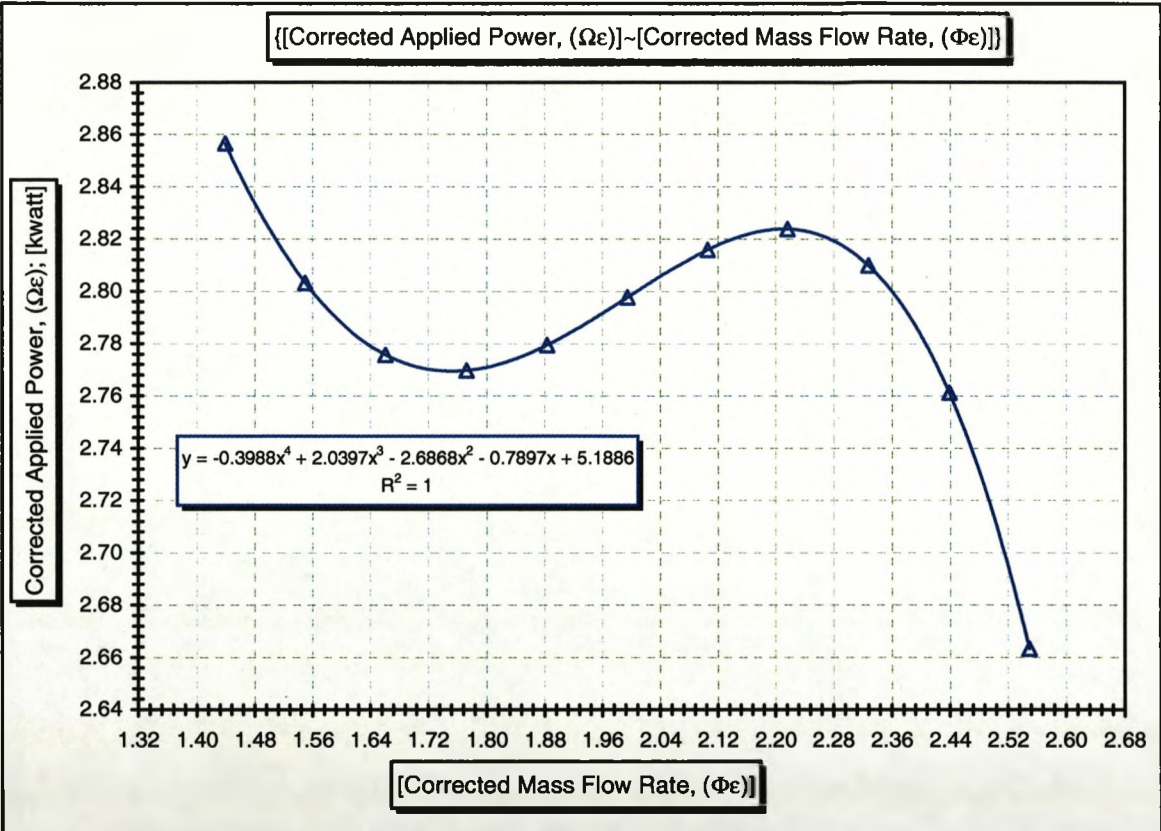


Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.549	2.438	2.327	2.216	2.105	1.994	1.883	1.772	1.661	1.550	1.439
Flow coefficient, (ϕ)	0.592	0.566	0.540	0.515	0.489	0.463	0.437	0.411	0.386	0.360	0.334
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.546	2.434	2.323	2.212	2.100	1.989	1.877	1.766	1.654	1.543	1.432
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.663	2.761	2.810	2.824	2.816	2.798	2.780	2.770	2.776	2.803	2.856
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.371	0.405	0.432	0.456	0.478	0.501	0.527	0.558	0.597	0.646	0.707
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.752	0.760	0.760	0.753	0.738	0.715	0.685	0.647	0.602	0.549	0.488
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.284	0.306	0.325	0.340	0.351	0.359	0.363	0.364	0.361	0.354	0.344
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.568	0.603	0.628	0.643	0.647	0.641	0.625	0.598	0.560	0.513	0.454
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.284	0.306	0.325	0.340	0.351	0.359	0.363	0.364	0.361	0.354	0.344
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.086	0.182	0.262	0.327	0.377	0.412	0.431	0.435	0.424	0.397	0.355
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.030	0.075	0.115	0.151	0.181	0.206	0.226	0.241	0.251	0.256	0.255
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.679	0.702	0.716	0.719	0.713	0.696	0.669	0.633	0.586	0.529	0.462
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.254	0.283	0.307	0.326	0.341	0.350	0.355	0.355	0.350	0.340	0.325

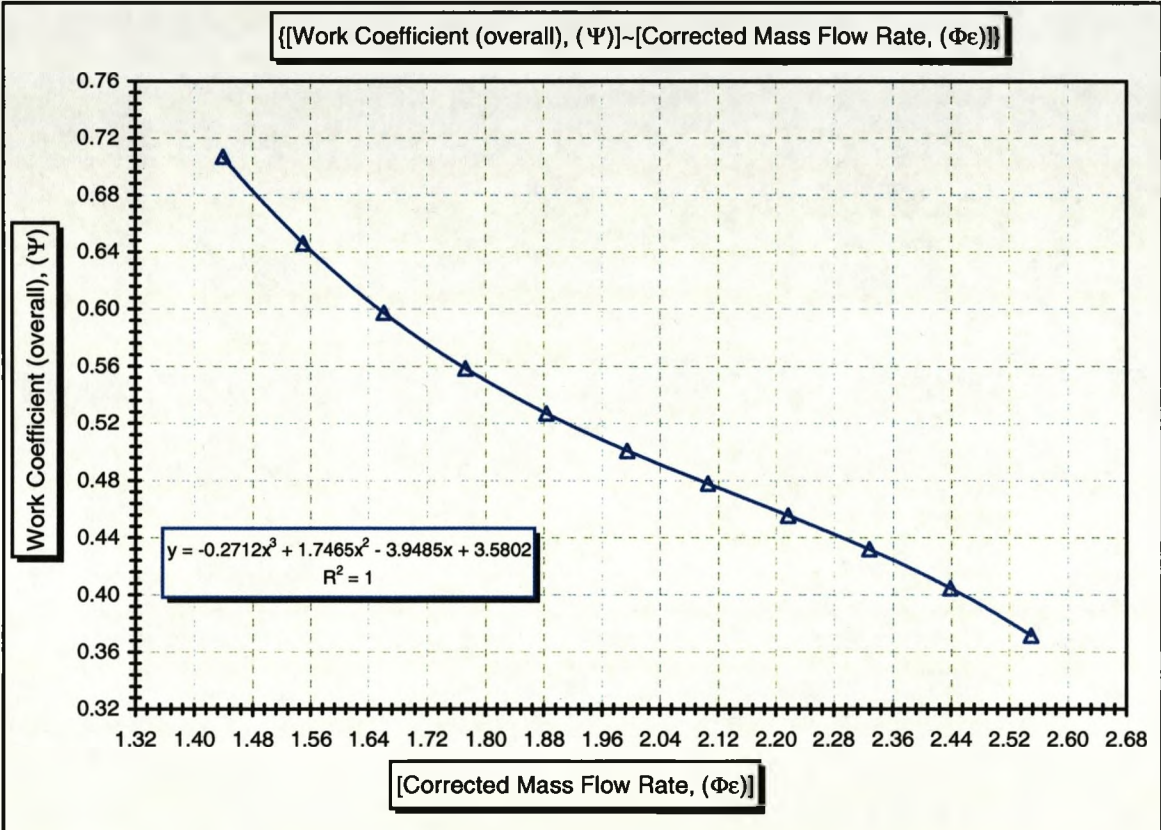
[Table (8-14)]- The flow characteristics of the one-stage compressor with casing treatment configuration of (0.434) rotor blade tip axial chord exposure ratio, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\varphi)_{SP}]$	0.334
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.432
Corrected mass flow rate at the stall point, $[(\Phi_E)_{SP}]$; [kg/sec]	1.439
Maximum corrected applied power, $[(\Omega E)_{Max}]$; [kwatt]	2.856
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi_E)/(\Omega E)_{Max}]$; [kg/sec]	1.439
Corrected applied power at the stall point, $[(\Omega E)_{SP}]$; [kwatt]	2.856
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi_E)/(\Omega E)_{SP}]$; [kg/sec]	1.439
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.707
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi_E)/(\Psi)_{Max-Overall}]$	1.439
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.761
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.378
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.488
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.647
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.116
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.454
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.435
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.799
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.355
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.720
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi_E)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.234
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.462
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.364
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.810
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.344
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.335
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.720
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.321
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.256
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.500
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.255
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.356
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi_E)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.831
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.325

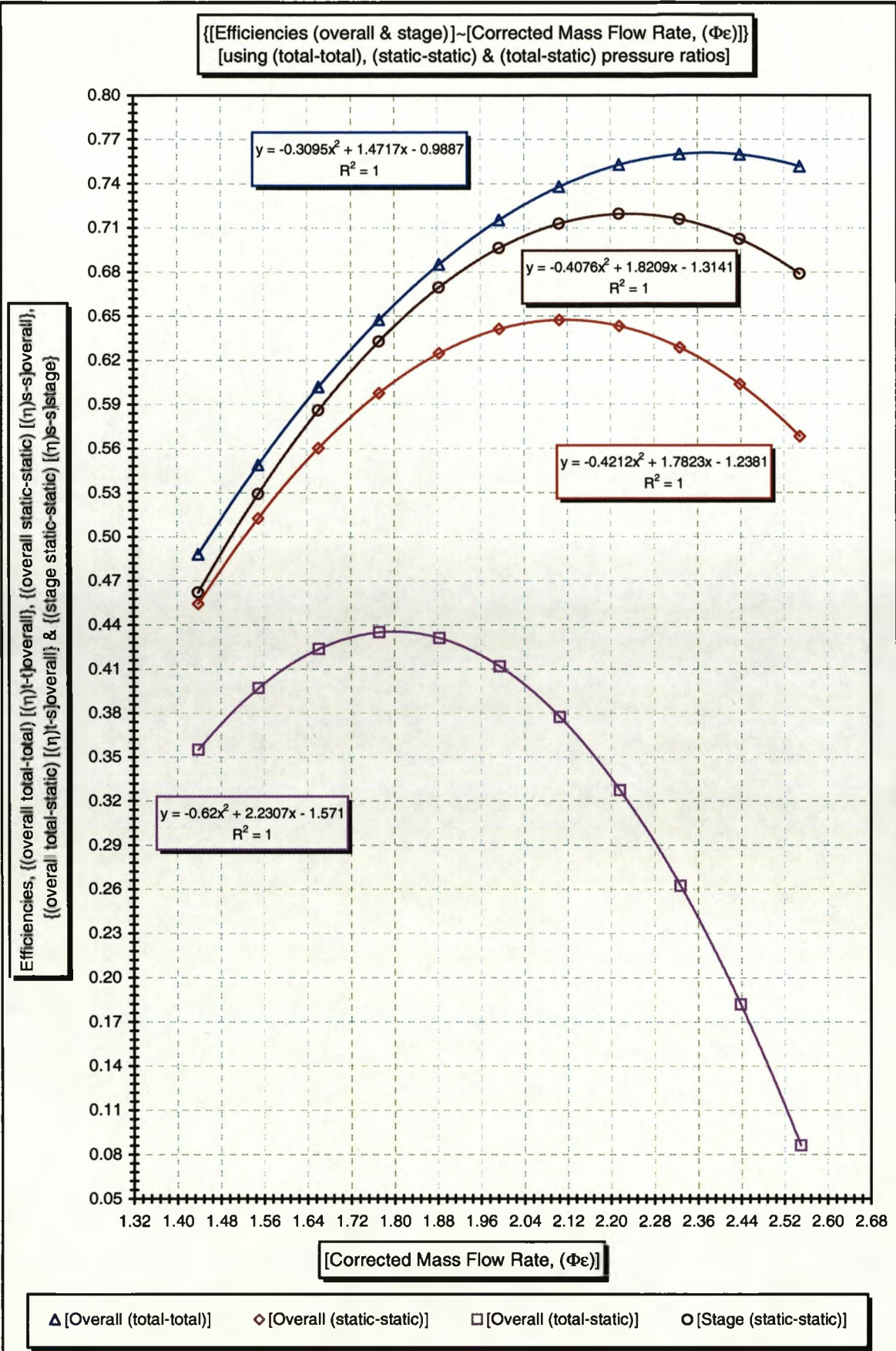
[Table (8-15)]- The characteristics of the one-stage compressor with (0.434) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]



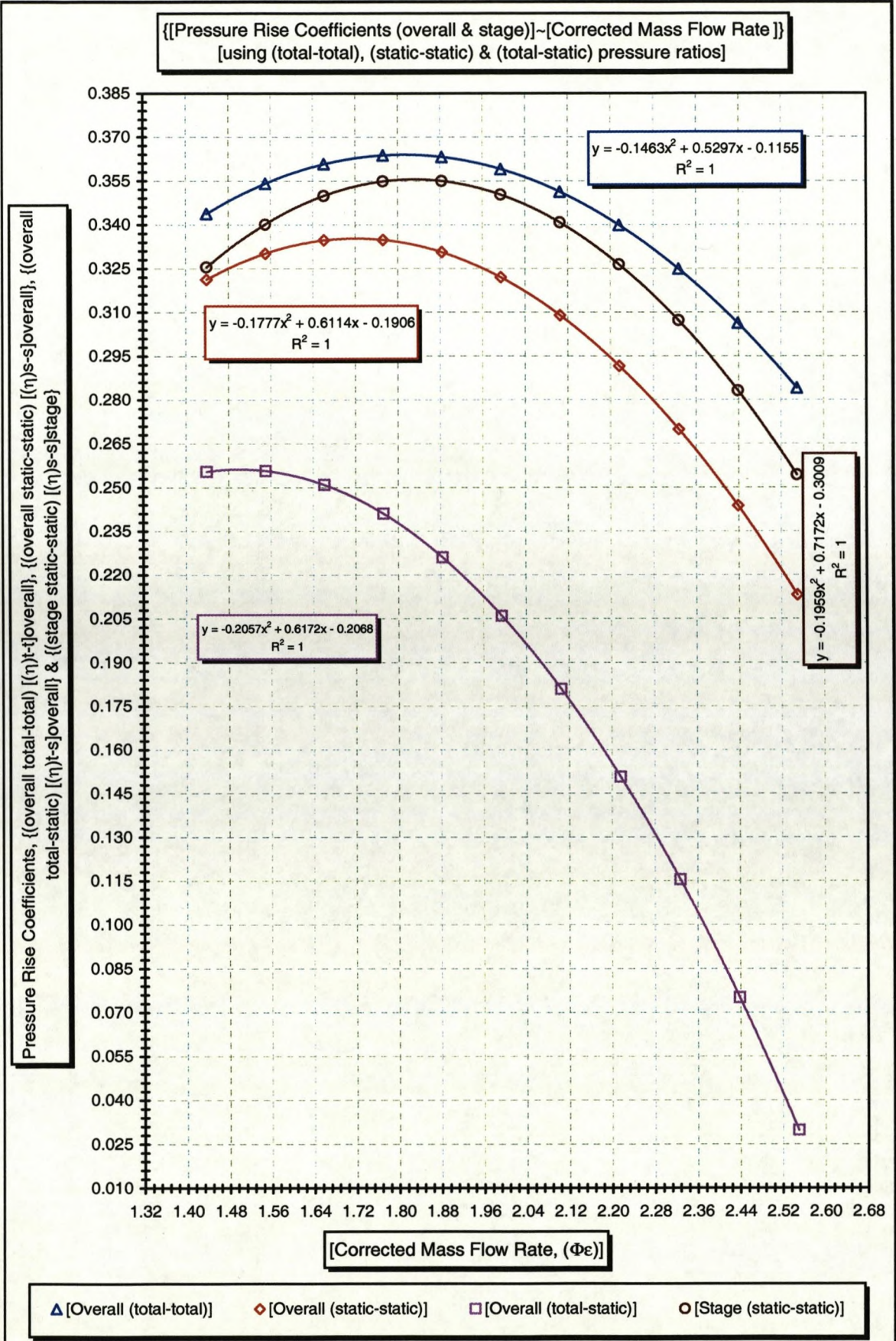
[Graph (8-27)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]



[Graph (8-28)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]



[Graph (8-29)]- The overalls and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]



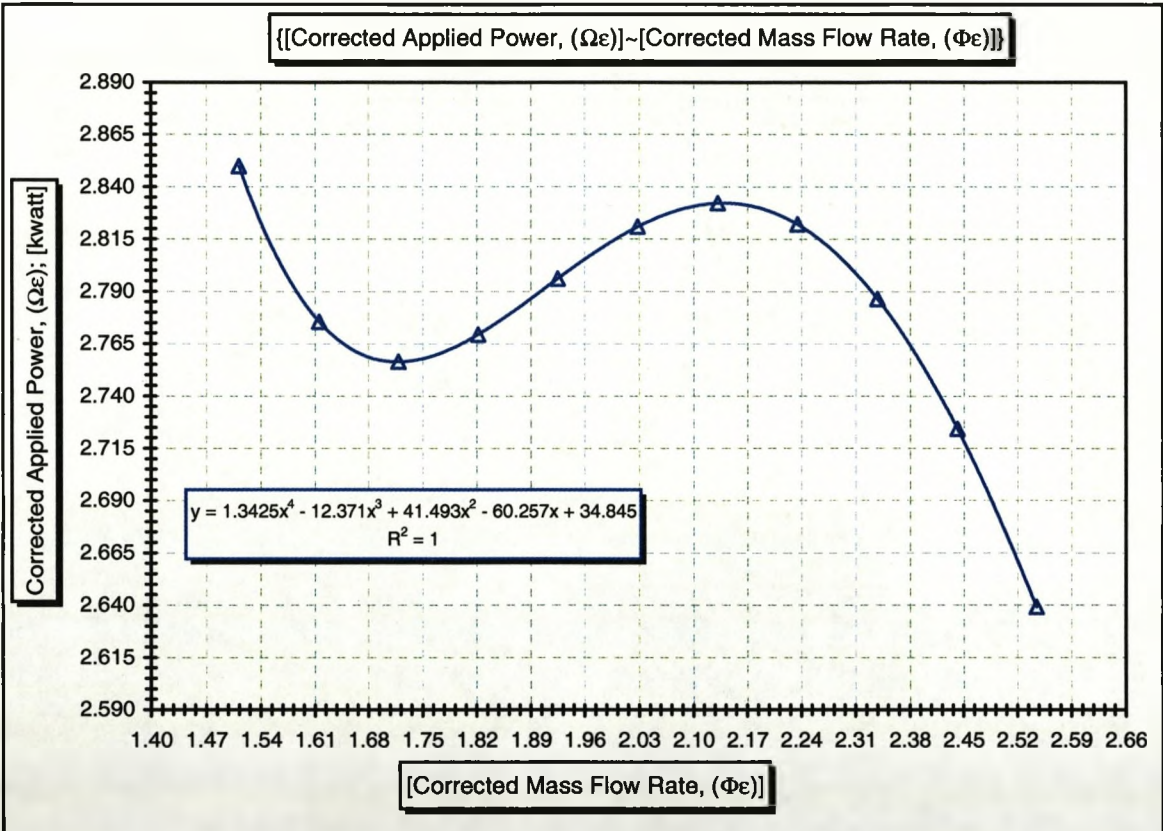
[Graph (8-30)]- The overalls and stage pressure rise coefficients against corrected flow rate of one-stage compressor with treatment, build [(IGV)+(TC-43.4%)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.544	2.441	2.338	2.235	2.131	2.028	1.925	1.822	1.718	1.615	1.512
Flow coefficient, (ϕ)	0.591	0.567	0.543	0.519	0.495	0.471	0.447	0.423	0.399	0.375	0.351
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.566	2.462	2.358	2.254	2.150	2.045	1.941	1.837	1.733	1.629	1.525
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.639	2.724	2.786	2.822	2.832	2.821	2.796	2.770	2.756	2.776	2.850
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.366	0.401	0.428	0.451	0.472	0.493	0.516	0.543	0.576	0.618	0.670
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.751	0.761	0.764	0.760	0.748	0.730	0.704	0.672	0.632	0.586	0.532
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.281	0.304	0.323	0.339	0.352	0.361	0.367	0.369	0.367	0.362	0.353
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.570	0.603	0.627	0.643	0.649	0.646	0.635	0.614	0.584	0.546	0.498
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.212	0.241	0.266	0.288	0.305	0.319	0.329	0.336	0.338	0.337	0.331
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.083	0.176	0.255	0.320	0.372	0.409	0.433	0.442	0.438	0.420	0.388
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.029	0.072	0.111	0.146	0.176	0.202	0.223	0.239	0.251	0.259	0.262
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.680	0.704	0.719	0.725	0.721	0.709	0.687	0.656	0.616	0.566	0.508
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.252	0.281	0.305	0.325	0.340	0.351	0.357	0.359	0.356	0.349	0.337

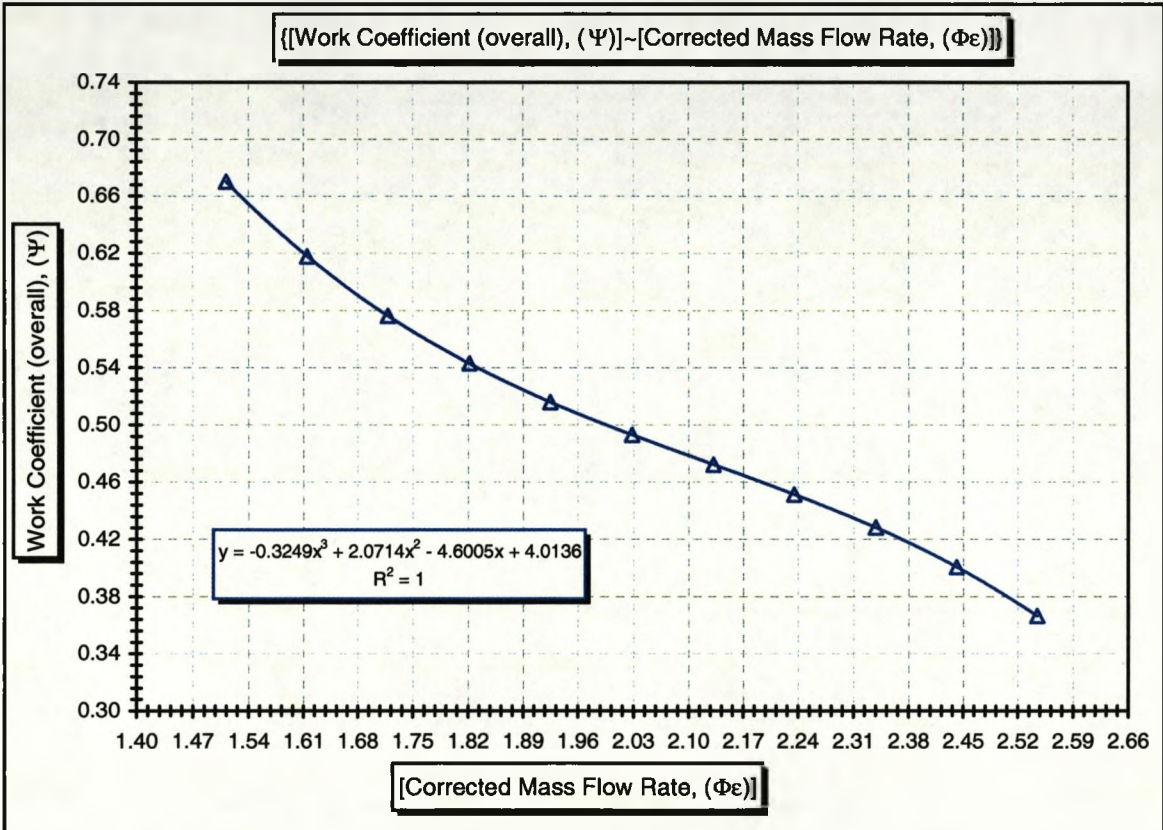
[Table (8-16)]- The flow characteristics of the one-stage compressor with casing treatment configuration of (0.535) rotor blade tip axial chord exposure ratio, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\Phi)_{SP}]$	0.351
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.525
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.512
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.850
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	1.512
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.850
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.512
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.670
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.512
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.764
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.348
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.532
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.649
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.111
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.498
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.442
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.802
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.388
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.725
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.222
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.508
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.369
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.816
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.353
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.338
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.705
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.331
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.262
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.512
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.262
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.359
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.835
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.337

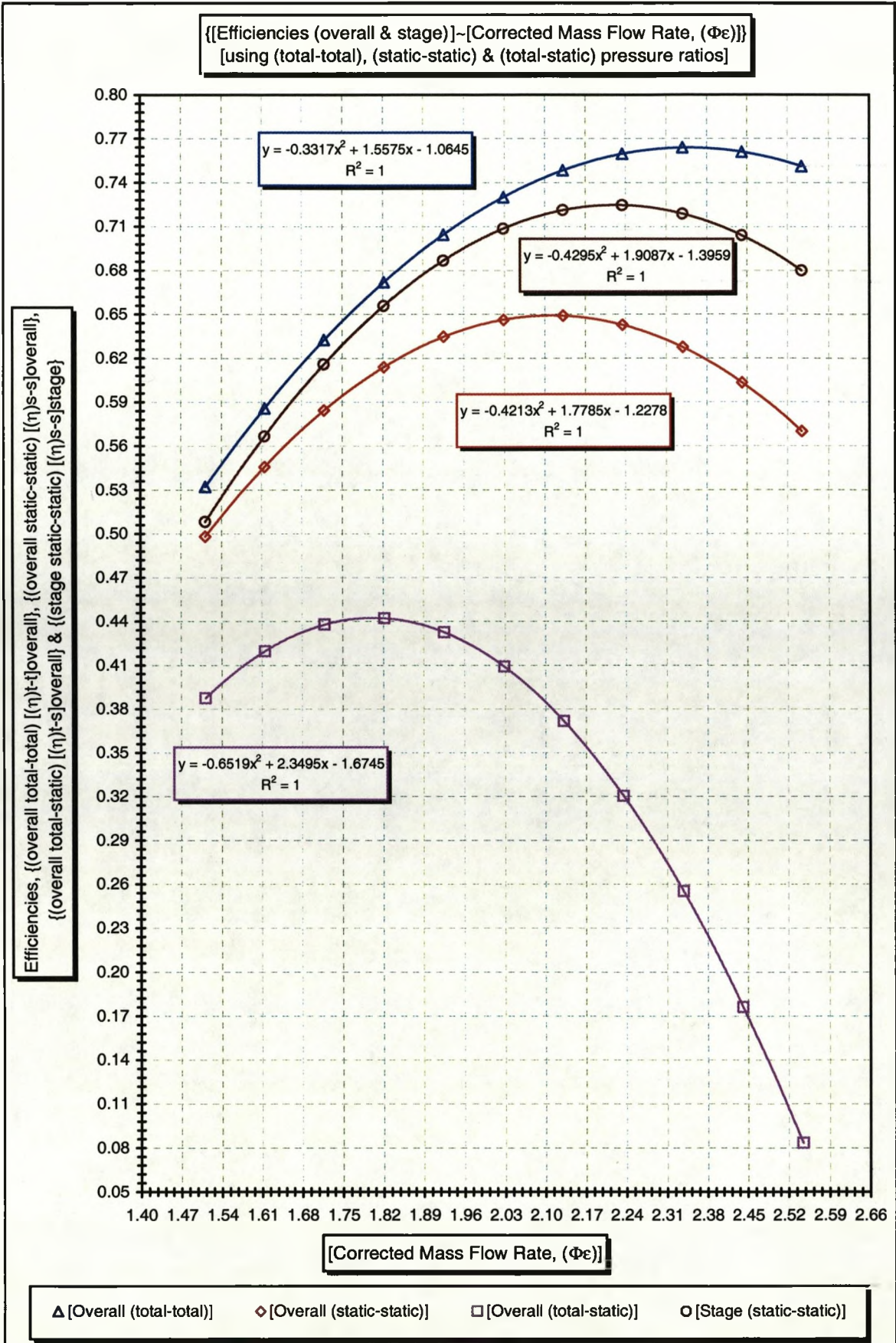
[Table (8-17)]- The characteristics of the one-stage compressor with (0.535) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]



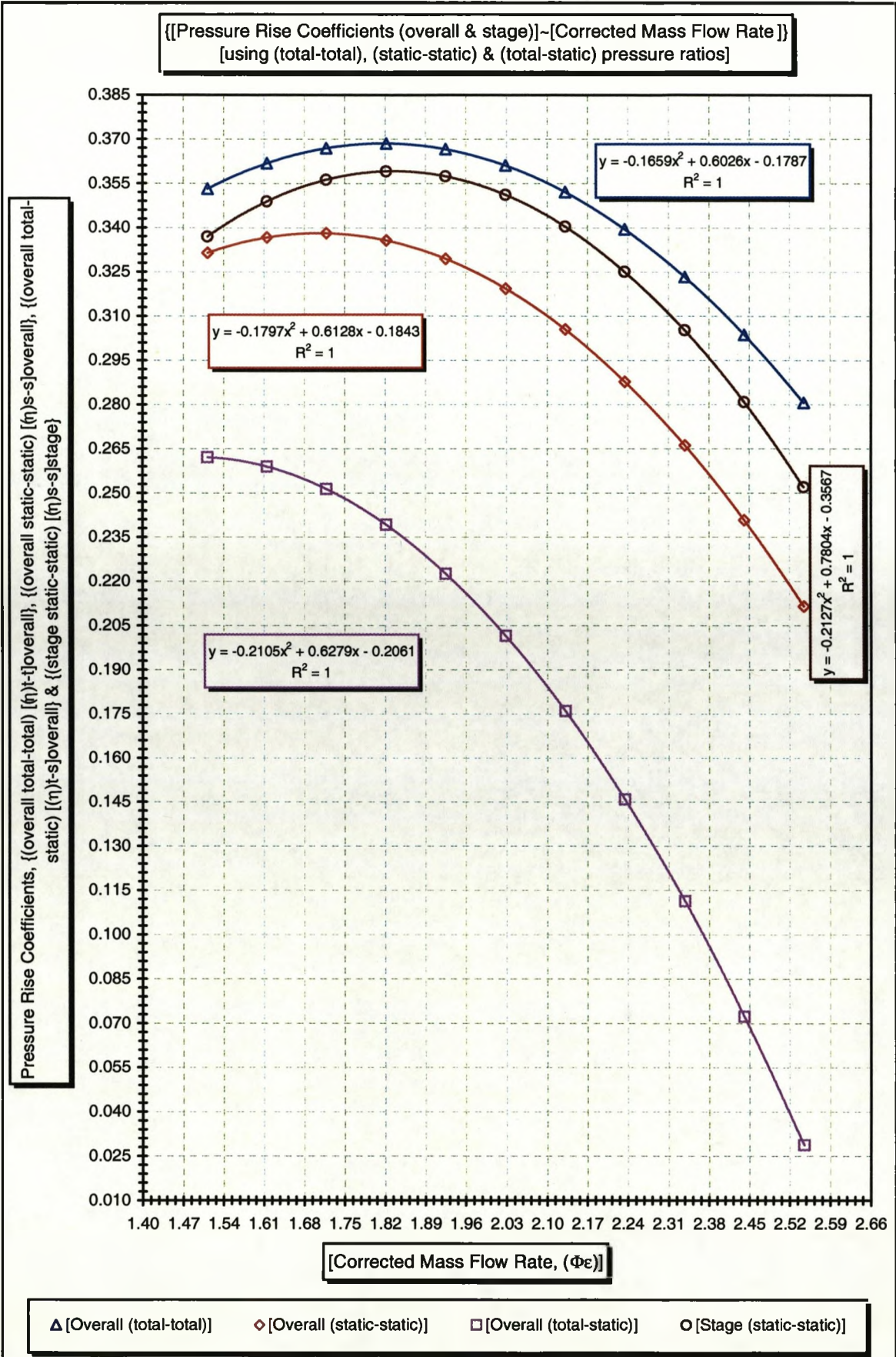
[Graph (8-31)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]



[Graph (8-32)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]



[Graph (8-33)]- The overalls and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]



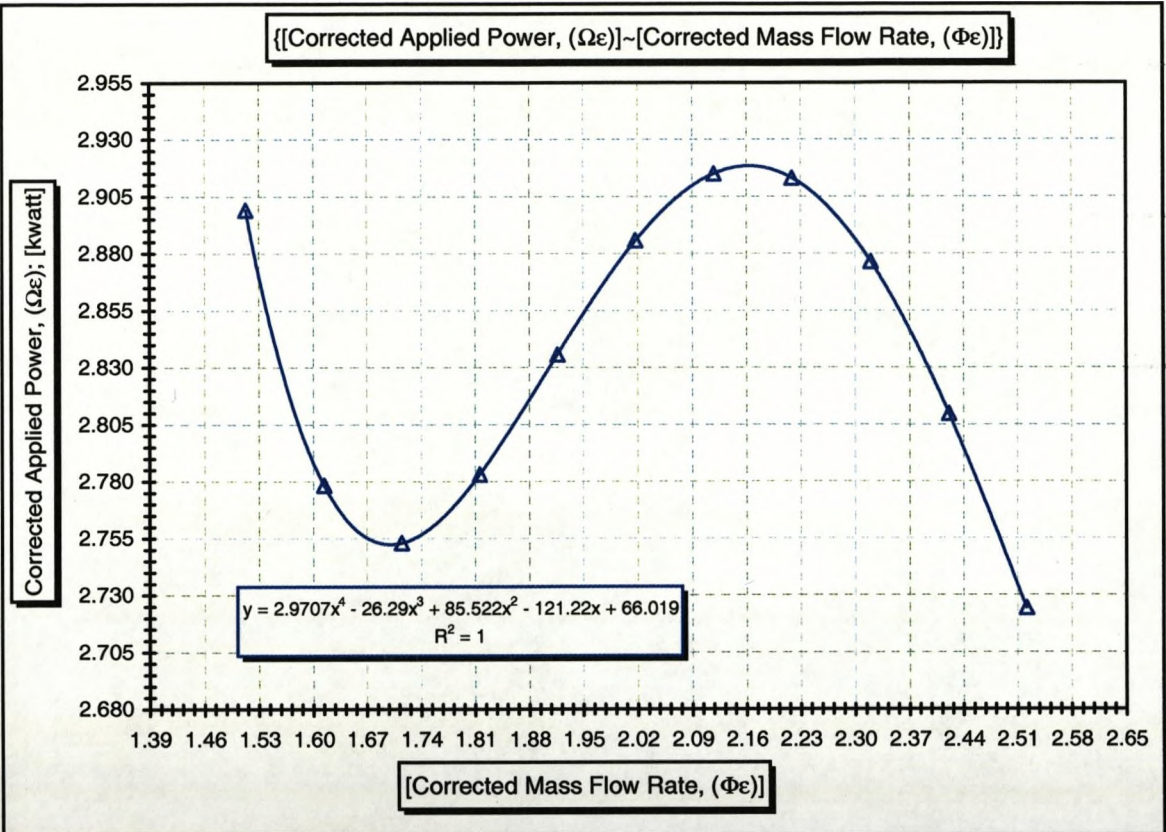
[Graph (8-34)]- The overalls and stage pressure rise coefficients against corrected flow rate of one-stage compressor with treatment, build [(IGV)+(TC-53.5%)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.521	2.421	2.320	2.219	2.118	2.017	1.916	1.816	1.715	1.614	1.513
Flow coefficient, (ϕ)	0.587	0.564	0.541	0.517	0.494	0.471	0.448	0.424	0.401	0.378	0.355
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.498	2.397	2.296	2.195	2.094	1.993	1.892	1.790	1.689	1.588	1.487
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.724	2.810	2.877	2.913	2.915	2.886	2.836	2.783	2.753	2.778	2.899
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.381	0.420	0.449	0.472	0.491	0.510	0.530	0.555	0.587	0.631	0.687
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.722	0.732	0.735	0.732	0.723	0.707	0.684	0.655	0.620	0.578	0.529
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.283	0.306	0.324	0.340	0.352	0.361	0.367	0.370	0.369	0.365	0.357
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.547	0.579	0.604	0.620	0.628	0.627	0.618	0.600	0.574	0.539	0.496
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.214	0.243	0.268	0.289	0.307	0.321	0.331	0.338	0.341	0.340	0.336
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.085	0.175	0.251	0.314	0.364	0.400	0.423	0.433	0.430	0.413	0.383
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.031	0.075	0.114	0.148	0.178	0.203	0.224	0.241	0.253	0.260	0.263
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.646	0.674	0.692	0.701	0.700	0.690	0.671	0.642	0.603	0.555	0.498
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.251	0.281	0.307	0.327	0.343	0.354	0.360	0.361	0.358	0.349	0.337

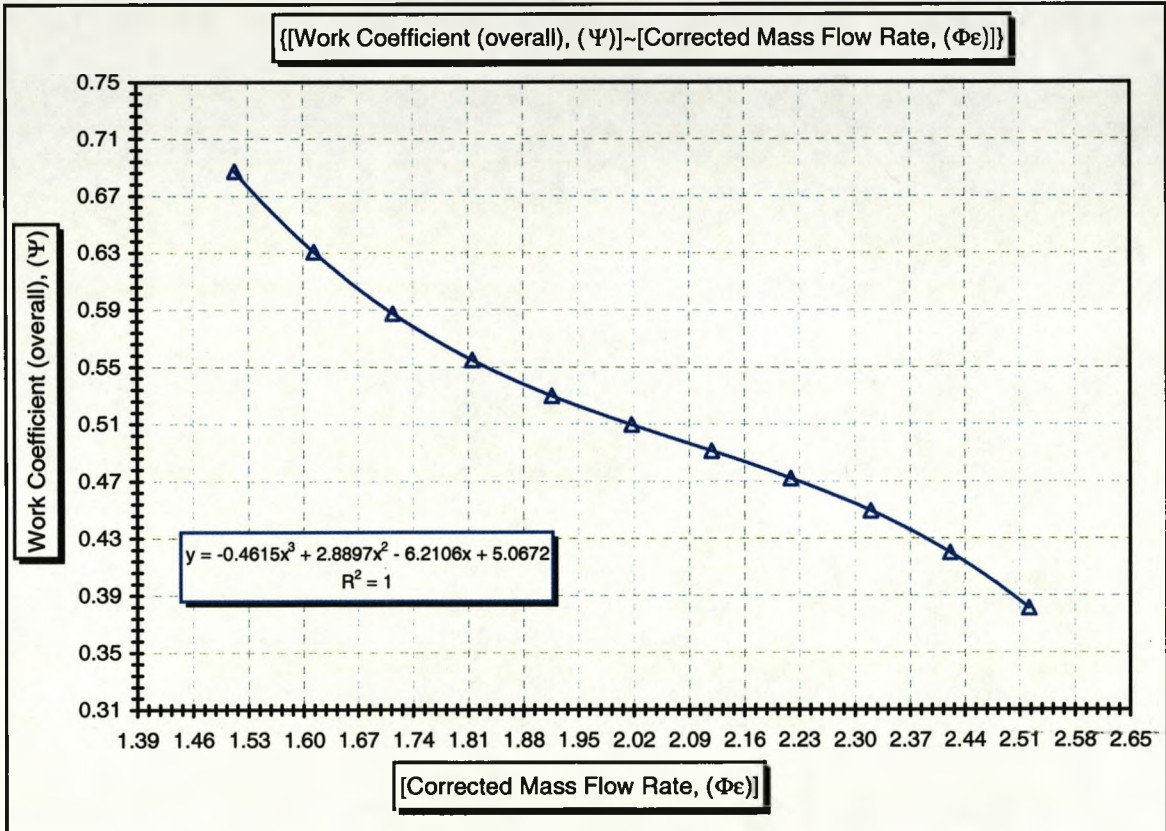
[Table (8-18)]- The flow characteristics of the one-stage compressor with casing treatment build of (0.636) rotor blade tip axial chord exposure ratio, configuration [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\Phi)_{SP}]$	0.355
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.487
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.513
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.918
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.163
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.899
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.513
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.687
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.513
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.735
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.317
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.529
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.628
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.077
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.496
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.433
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.791
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.383
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.702
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.175
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.498
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.370
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.791
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.357
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.341
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.679
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.336
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.263
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.513
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.263
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.361
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.837
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.337

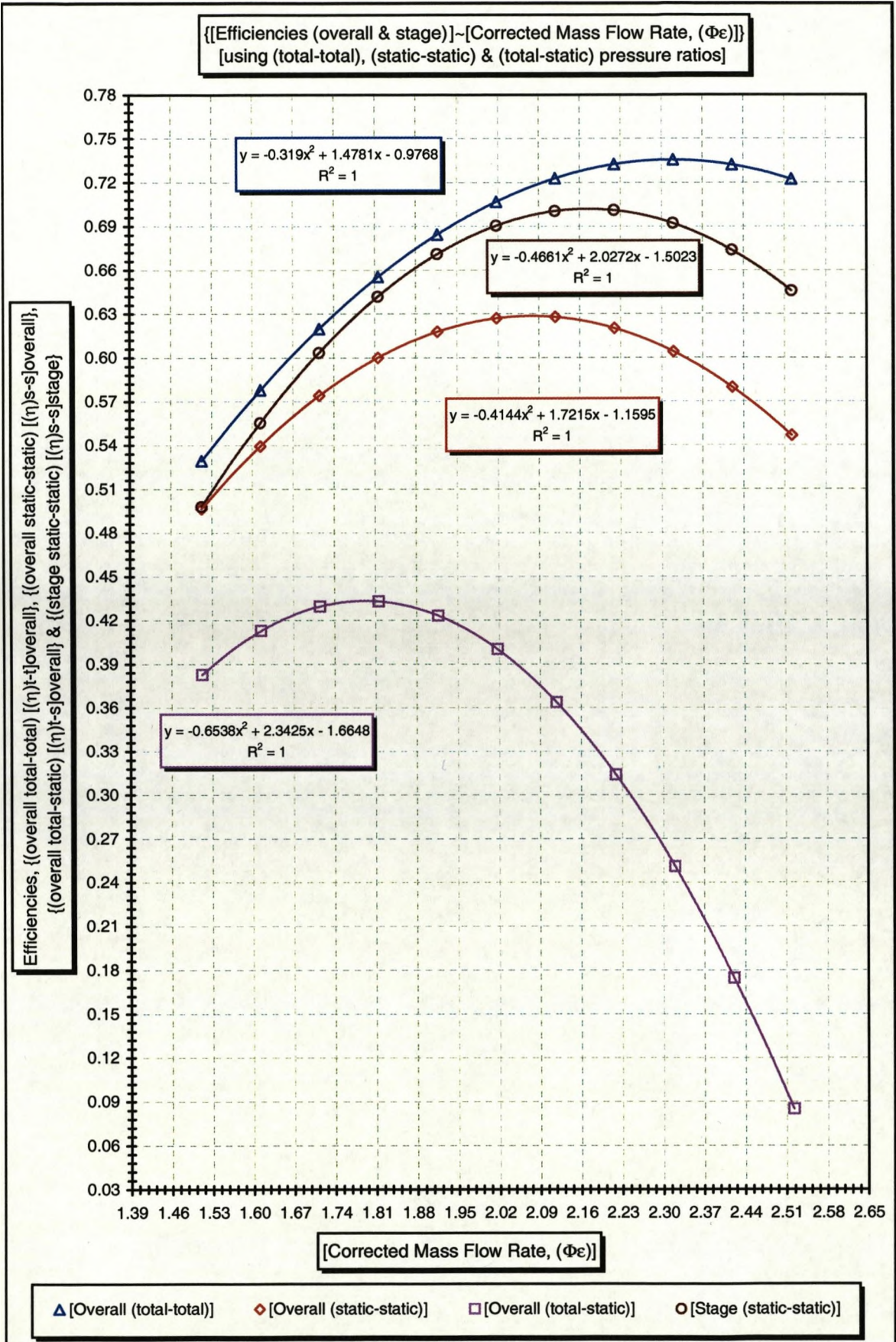
[Table (8-19)]- The characteristics of the one-stage compressor with (0.636) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]



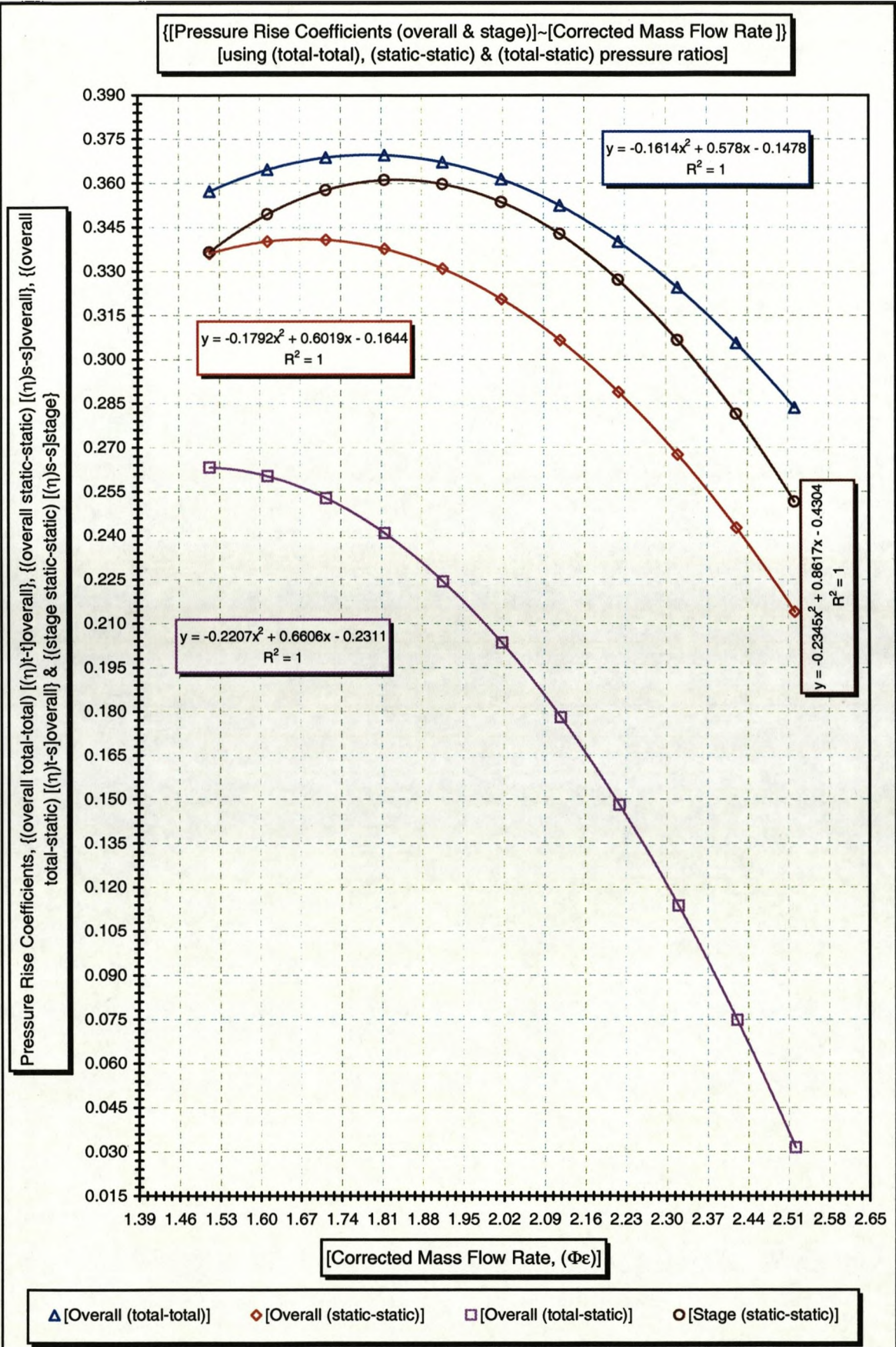
[Graph (8-35)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]



[Graph (8-36)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]



[Graph (8-37)]- The overalls and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]



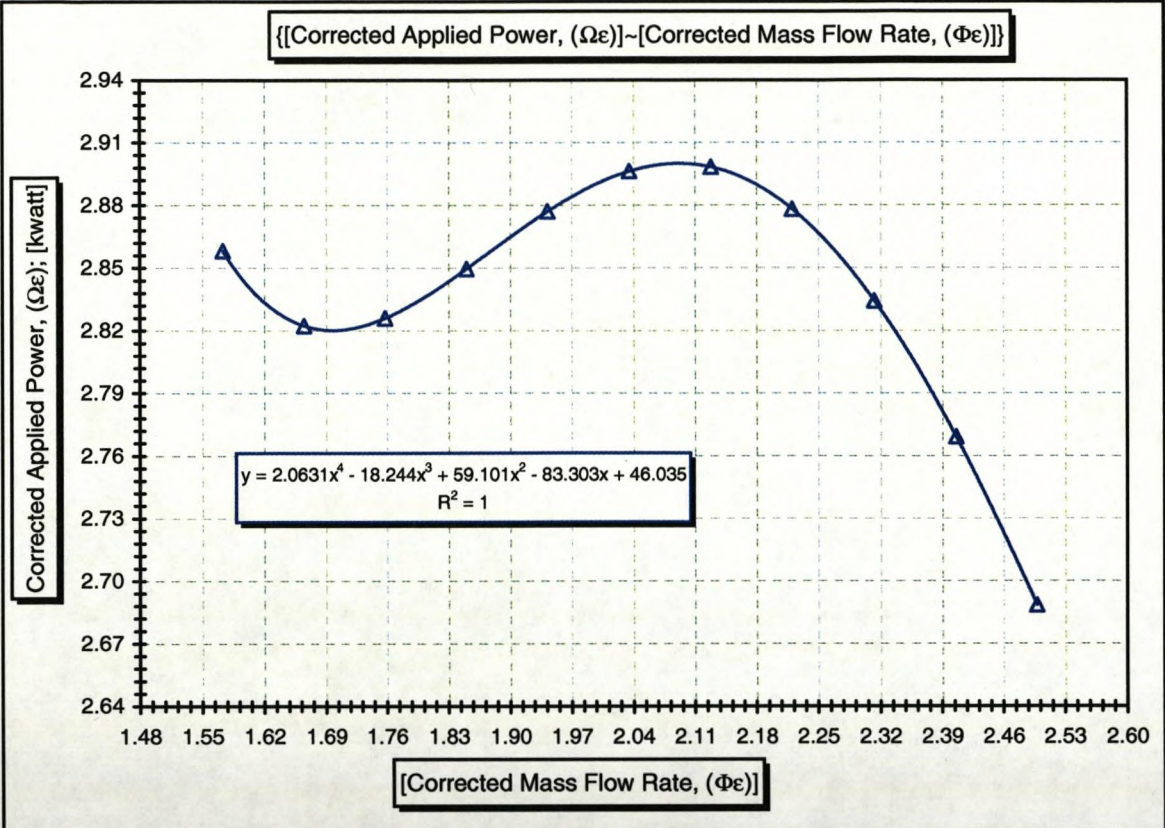
[Graph (8-38)]- The overalls and stage pressure rise coefficients against corrected flow rate of one-stage compressor with treatment, build [(IGV)+(TC-63.6%)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]$; [kg/sec]	2.498	2.405	2.313	2.220	2.128	2.035	1.943	1.850	1.758	1.665	1.573
Flow coefficient, (ϕ)	0.582	0.561	0.539	0.518	0.496	0.474	0.453	0.431	0.410	0.388	0.366
Mass flow rate, (Φ) ; [kg/sec]	2.467	2.375	2.284	2.192	2.100	2.009	1.917	1.826	1.734	1.642	1.551
Corrected applied power, $[(\Omega\epsilon)]$; [kwatt]	2.688	2.769	2.834	2.878	2.898	2.896	2.877	2.850	2.826	2.822	2.858
Work coefficient (overall), $[(\Psi)_{Overall}]$	0.385	0.418	0.445	0.468	0.489	0.510	0.531	0.554	0.581	0.614	0.654
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Overall}\}$	0.721	0.727	0.727	0.723	0.714	0.699	0.680	0.656	0.626	0.592	0.552
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{Overall}\}$	0.282	0.302	0.320	0.335	0.347	0.356	0.362	0.366	0.366	0.364	0.359
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{Overall}\}$	0.546	0.576	0.598	0.614	0.622	0.623	0.617	0.603	0.582	0.554	0.519
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{Overall}\}$	0.212	0.239	0.264	0.285	0.302	0.317	0.328	0.335	0.339	0.340	0.338
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{Overall}\}$	0.092	0.173	0.243	0.302	0.349	0.384	0.409	0.422	0.424	0.414	0.393
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{Overall}\}$	0.034	0.074	0.110	0.142	0.170	0.195	0.215	0.232	0.245	0.255	0.260
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{Stage}\}$	0.649	0.673	0.689	0.696	0.696	0.687	0.670	0.645	0.612	0.570	0.520
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{Stage}\}$	0.252	0.280	0.304	0.324	0.339	0.350	0.357	0.359	0.356	0.350	0.339

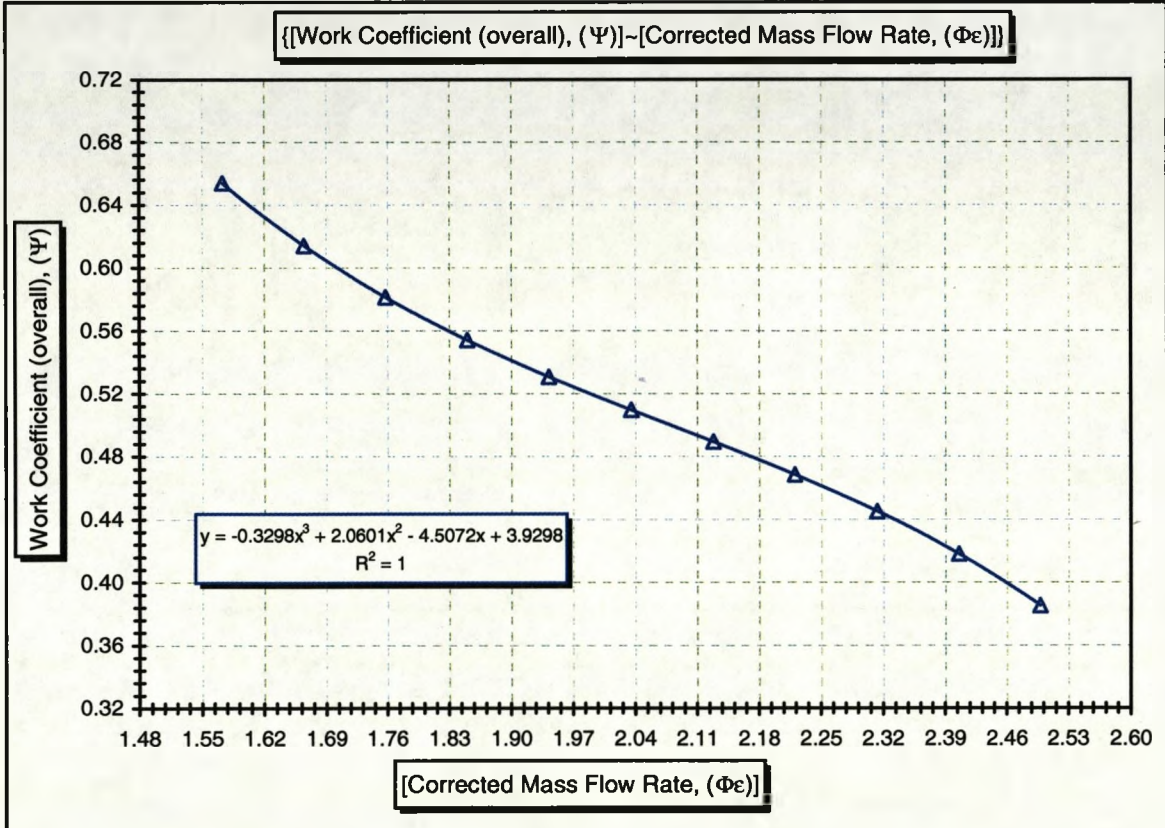
[Table (8-20)]- The flow characteristics of the one-stage compressor with casing treatment build of (0.737) rotor blade tip axial chord exposure ratio, configuration [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.366
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.551
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.573
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.900
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.091
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.858
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.573
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.654
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.573
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.728
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.346
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.552
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.624
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.069
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.519
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.425
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.789
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.393
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.697
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.180
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.520
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.366
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.787
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.359
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.340
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.688
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.338
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.260
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.573
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.260
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.359
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.851
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.339

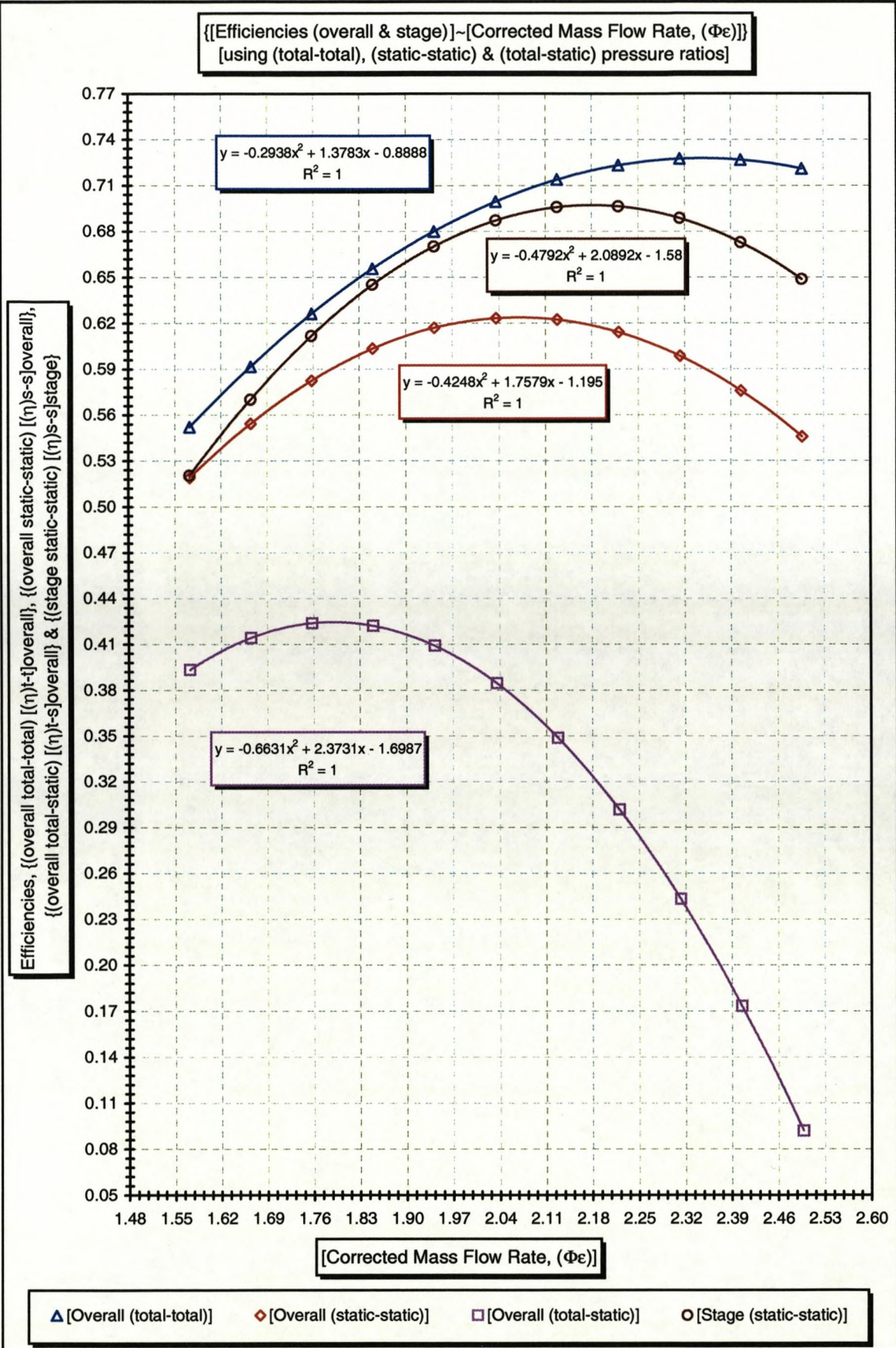
[Table (8-21)]- The characteristics of the one-stage compressor with (0.737) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]



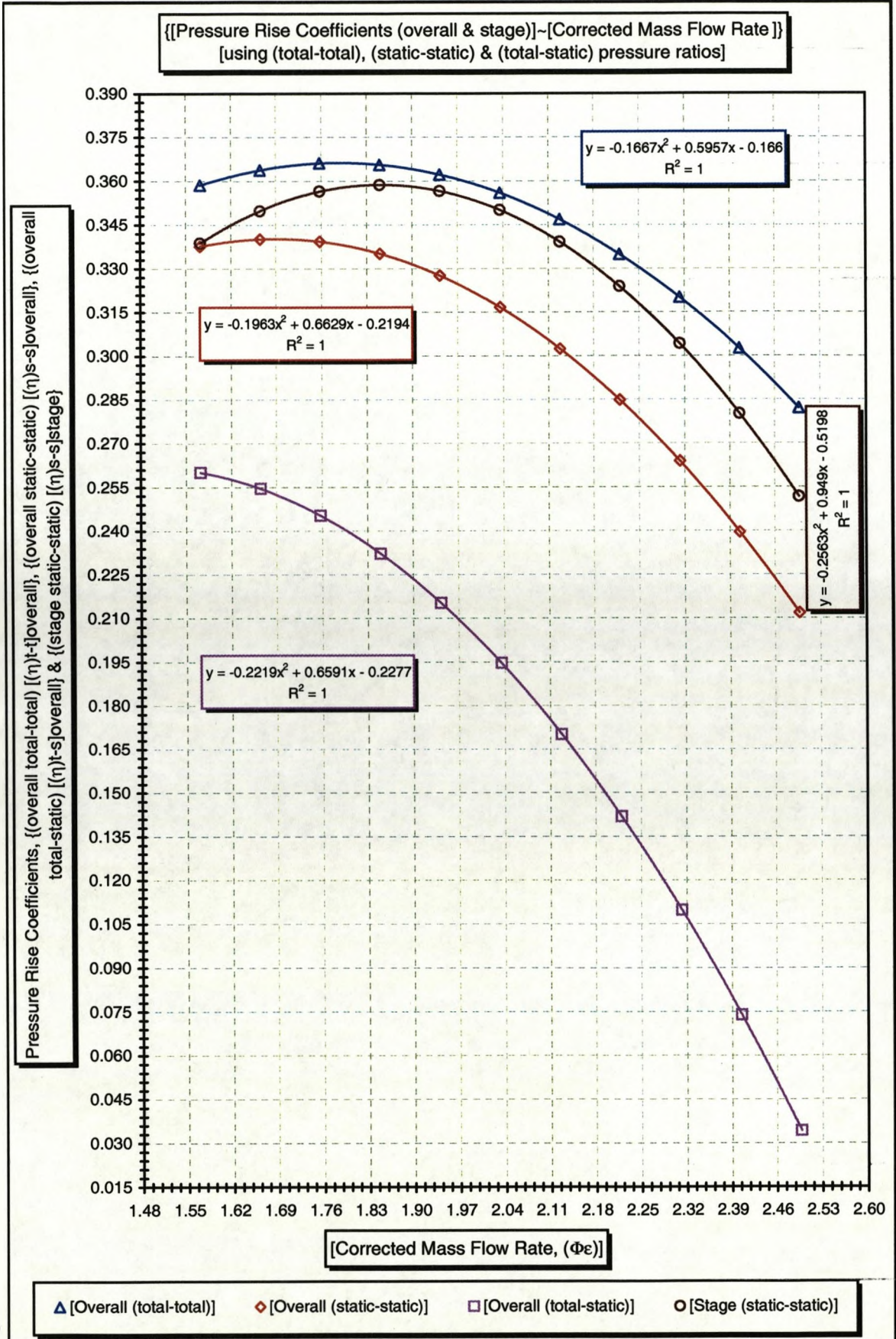
[Graph (8-39)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]



[Graph (8-40)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]



[Graph (8-41)]- The overalls and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]



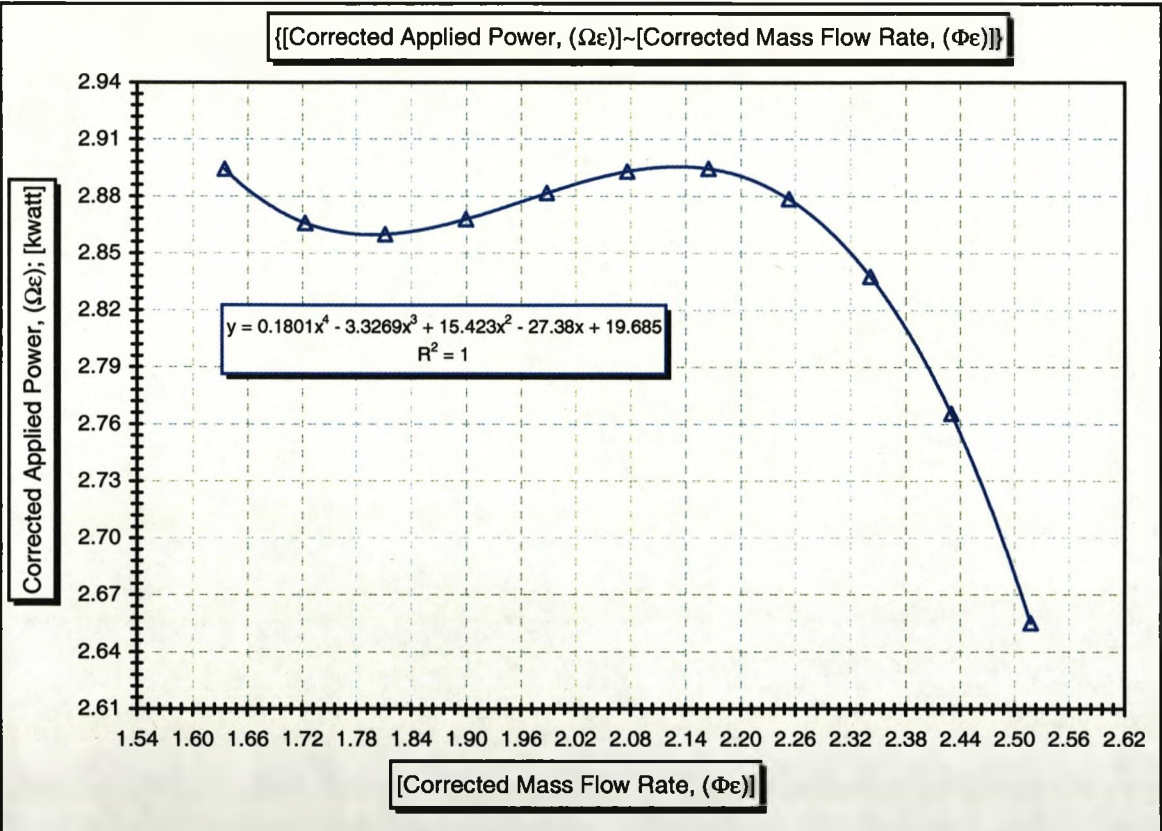
[Graph (8-42)]- The overalls and stage pressure rise coefficients against corrected flow rate of one-stage compressor with treatment, build [(IGV)+(TC-73.7%)+(1R)+(1S)+(H)]

Corrected mass flow rate, $[(\Phi\epsilon)]; [\text{kg/sec}]$	2.518	2.430	2.341	2.253	2.165	2.076	1.988	1.900	1.811	1.723	1.635
Flow coefficient, (ϕ)	0.585	0.564	0.543	0.523	0.502	0.482	0.461	0.441	0.420	0.399	0.379
Mass flow rate, $(\Phi); [\text{kg/sec}]$	2.478	2.391	2.304	2.216	2.129	2.042	1.954	1.867	1.779	1.692	1.605
Corrected applied power, $[(\Omega\epsilon)]; [\text{kWatt}]$	2.655	2.766	2.838	2.878	2.894	2.893	2.882	2.868	2.860	2.866	2.894
Work coefficient (overall), $[(\Psi)_{\text{Overall}}]$	0.374	0.405	0.432	0.455	0.475	0.495	0.516	0.538	0.564	0.594	0.630
Efficiency (overall total-total), $\{[(\eta)_{t-t}]_{\text{Overall}}\}$	0.710	0.716	0.717	0.713	0.705	0.693	0.676	0.654	0.628	0.597	0.562
Pressure rise coefficient (overall total-total), $\{[(\psi)_{t-t}]_{\text{Overall}}\}$	0.270	0.289	0.307	0.321	0.334	0.343	0.350	0.354	0.356	0.355	0.352
Efficiency (overall static-static), $\{[(\eta)_{s-s}]_{\text{Overall}}\}$	0.560	0.587	0.607	0.620	0.627	0.627	0.620	0.607	0.587	0.561	0.528
Pressure rise coefficient (overall static-static), $\{[(\psi)_{s-s}]_{\text{Overall}}\}$	0.212	0.238	0.261	0.281	0.297	0.311	0.321	0.328	0.333	0.334	0.331
Efficiency (overall total-static), $\{[(\eta)_{t-s}]_{\text{Overall}}\}$	0.073	0.151	0.219	0.276	0.323	0.360	0.386	0.403	0.409	0.404	0.390
Pressure rise coefficient (overall total-static), $\{[(\psi)_{t-s}]_{\text{Overall}}\}$	0.026	0.063	0.096	0.127	0.154	0.177	0.198	0.215	0.229	0.240	0.248
Efficiency (stage static-static), $\{[(\eta)_{s-s}]_{\text{Stage}}\}$	0.667	0.686	0.698	0.702	0.699	0.689	0.672	0.647	0.615	0.576	0.530
Pressure rise coefficient (stage static-static), $\{[(\psi)_{s-s}]_{\text{Stage}}\}$	0.252	0.278	0.300	0.318	0.332	0.342	0.348	0.350	0.348	0.342	0.332

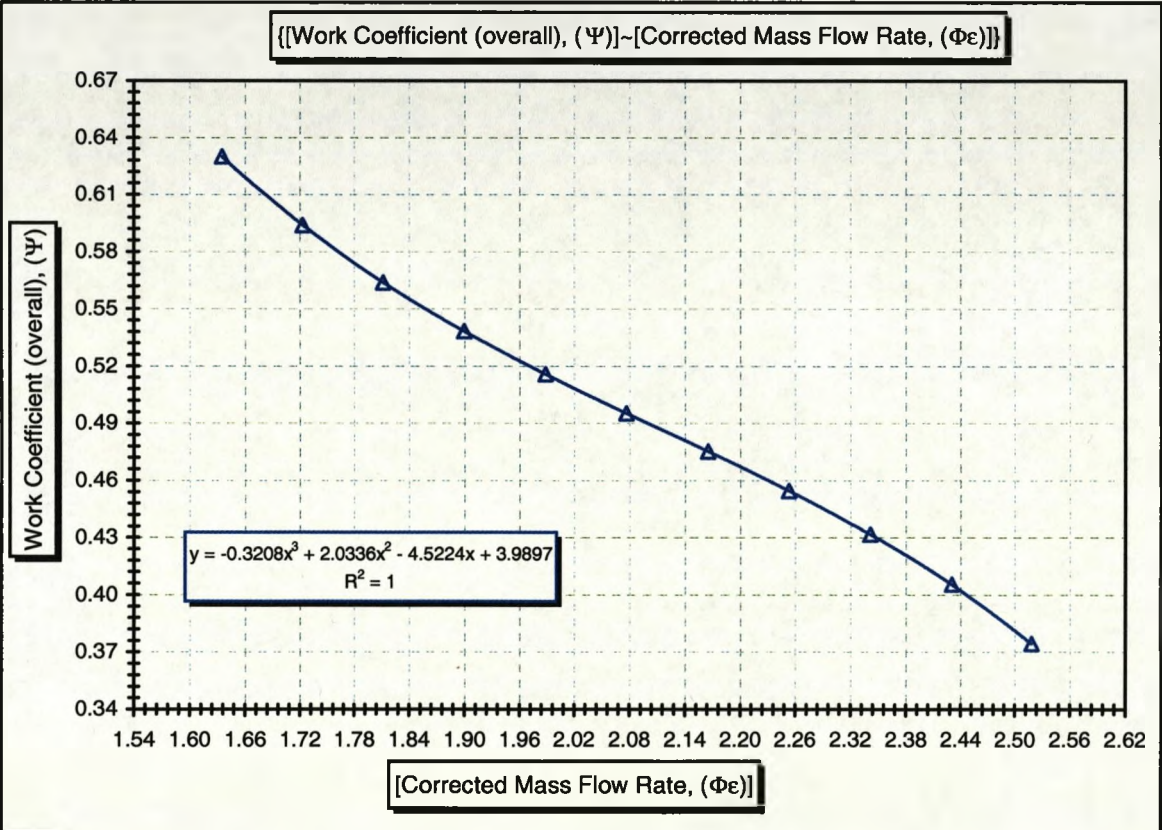
[Table (8-22)]- The flow characteristics of the one-stage compressor with casing treatment build of (0.838) rotor blade tip axial chord exposure ratio, configuration [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]

Flow coefficient at the stall point, $[(\phi)_{SP}]$	0.379
Mass flow rate at the stall point, $[(\Phi)_{SP}]$; [kg/sec]	1.605
Corrected mass flow rate at the stall point, $[(\Phi\epsilon)_{SP}]$; [kg/sec]	1.635
Maximum corrected applied power, $[(\Omega\epsilon)_{Max}]$; [kwatt]	2.895
Corrected mass flow rate @ maximum corrected applied power, $[(\Phi\epsilon)/(\Omega\epsilon)_{Max}]$; [kg/sec]	2.131
Corrected applied power at the stall point, $[(\Omega\epsilon)_{SP}]$; [kwatt]	2.894
Corrected mass flow rate @ corrected applied power at stall point, $[(\Phi\epsilon)/(\Omega\epsilon)_{SP}]$; [kg/sec]	1.635
Maximum work coefficient (overall), $[(\Psi)_{Max-Overall}]$	0.630
Corrected mass flow rate @ maximum overall work coefficient, $[(\Phi\epsilon)/(\Psi)_{Max-Overall}]$	1.635
Maximum efficiency (overall total-total), $\{[(\eta)_{t-t}]_{Max-Overall}\}$	0.717
Corrected mass flow rate @ maximum efficiency (overall total-total), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-t)}\}$	2.365
Efficiency (overall total-total) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-t)}\}$	0.562
Maximum efficiency (overall static-static), $\{[(\eta)_{Max}]_{Overall-(s-s)}\}$	0.627
Corrected mass flow rate @ maximum efficiency (overall static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(s-s)}\}$	2.119
Efficiency (overall static-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(s-s)}\}$	0.528
Maximum efficiency (overall total-static), $\{[(\eta)_{Max}]_{Overall-(t-s)}\}$	0.409
Corrected mass flow rate @ maximum efficiency (overall total-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Overall-(t-s)}\}$	1.804
Efficiency (overall total-static) at the stall point, $\{[(\eta)_{SP}]_{Overall-(t-s)}\}$	0.390
Maximum efficiency (stage static-static), $\{[(\eta)_{Max}]_{Stage-(s-s)}\}$	0.702
Corrected mass flow rate @ maximum efficiency (stage static-static), $\{(\Phi\epsilon)/[(\eta)_{Max}]_{Stage-(s-s)}\}$	2.244
Efficiency (stage static-static) at the stall point, $\{[(\eta)_{SP}]_{Stage-(s-s)}\}$	0.530
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{Peak}]_{Overall-(t-t)}\}$	0.356
Corrected mass flow rate @ peak pressure rise coefficient (overall total-total), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-t)}\}$	1.797
Pressure rise coefficient (overall total-total) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-t)}\}$	0.352
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{Peak}]_{Overall-(s-s)}\}$	0.334
Corrected mass flow rate @ peak pressure rise coefficient (overall static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(s-s)}\}$	1.738
Pressure rise coefficient (overall static-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(s-s)}\}$	0.331
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{Peak}]_{Overall-(t-s)}\}$	0.248
Corrected mass flow rate @ peak pressure rise coefficient (overall total-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Overall-(t-s)}\}$	1.635
Pressure rise coefficient (overall total-static) at the stall point, $\{[(\psi)_{SP}]_{Overall-(t-s)}\}$	0.248
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{Peak}]_{Stage-(s-s)}\}$	0.350
Corrected mass flow rate @ peak pressure rise coefficient (stage static-static), $\{(\Phi\epsilon)/[(\psi)_{Peak}]_{Stage-(s-s)}\}$	1.898
Pressure rise coefficient (stage static-static) at the stall point, $\{[(\psi)_{SP}]_{Stage-(s-s)}\}$	0.332

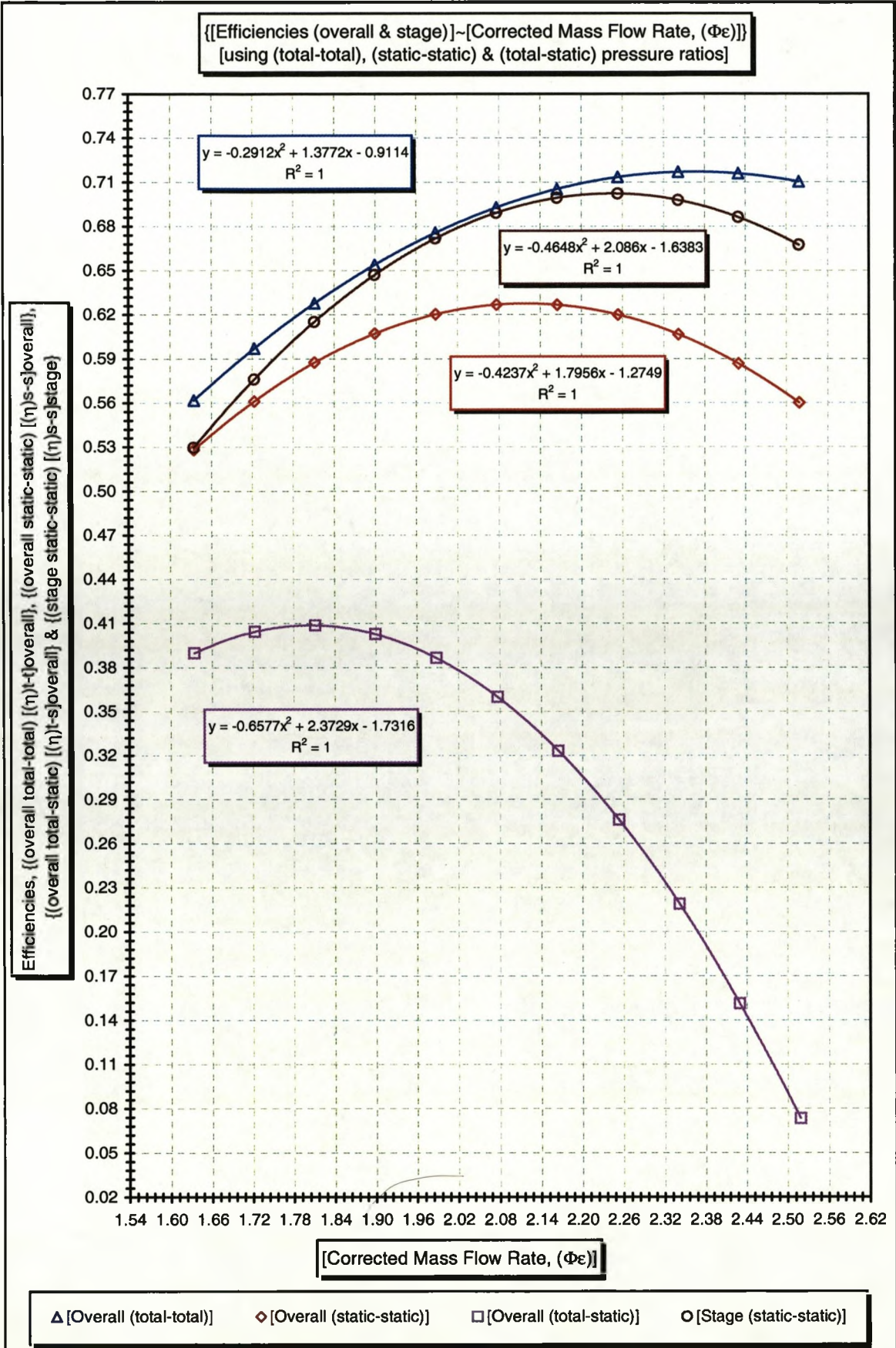
[Table (8-23)]- The characteristics of the one-stage compressor with (0.838) rotor tip axial chord exposure ratio casing treatment, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]



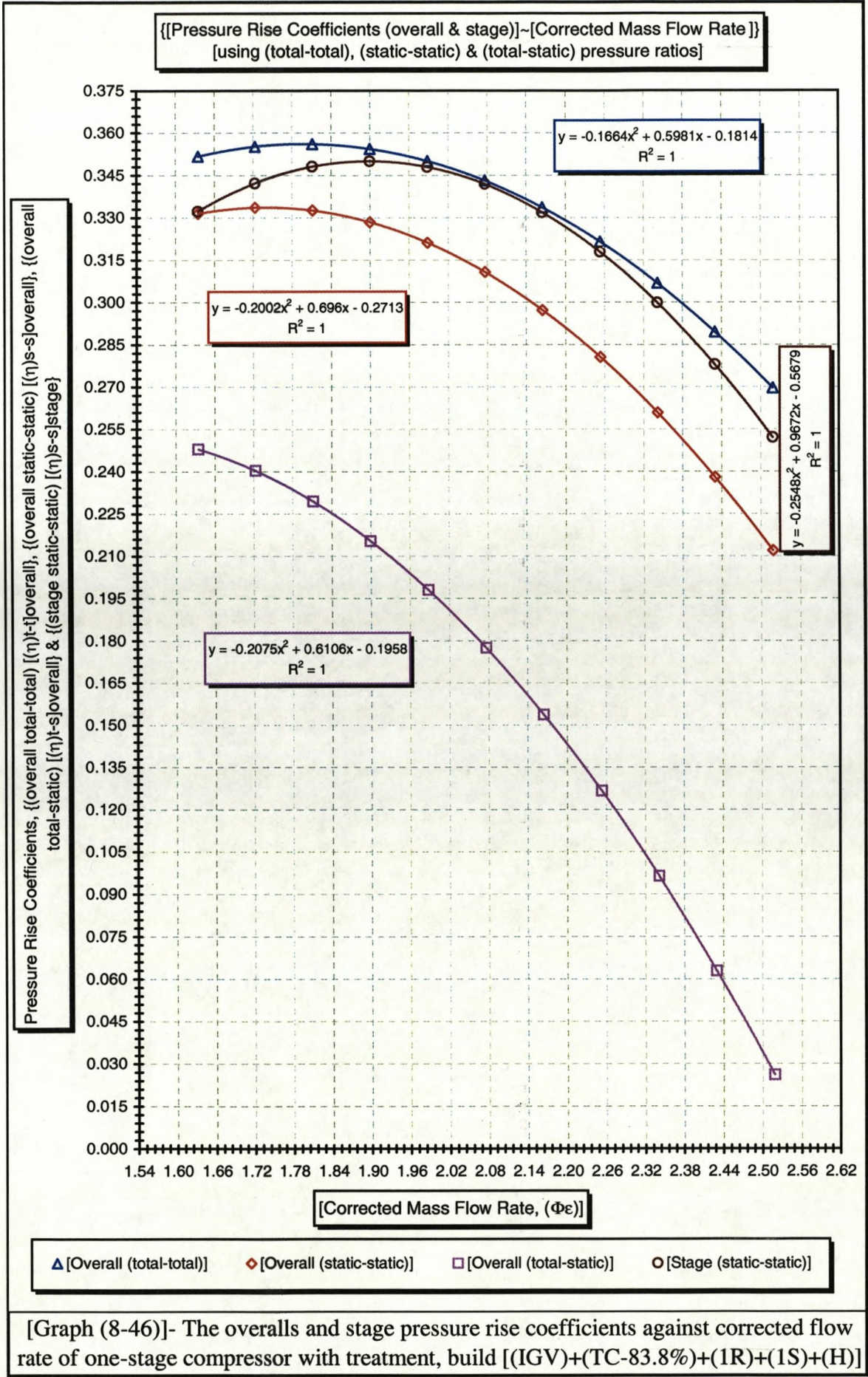
[Graph (8-43)]- The corrected applied power versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]



[Graph (8-44)]- The overall work coefficient versus the corrected mass flow rate of the one-stage compressor with casing treatment, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]



[Graph (8-45)]- The overalls and stage efficiencies versus the corrected mass flow rate of one-stage compressor with casing treatment, build [(IGV)+(TC-83.8%)+(1R)+(1S)+(H)]



CHAPTER NINE

9)- Final Analysis (Improvement or Deterioration), Discussions and Conclusions:

9.1)- Introduction:

This chapter presents an analysis summarising the results of the study and highlights, improvements that have been achieved, as well as instances where no benefits have been obtained. The results are evaluated in terms of flow characteristics, efficiencies, and pressure rise coefficients.

Although some previous investigators used several definitions in order to assess the performance of the casing treatments, it appears that the stall margin improvement in terms of different flow characteristics is common to almost all of the prior studies. Gains or losses in both efficiency and pressure rise coefficient are also among the assessment parameters that have received considerable attentions.

9.2)- General Discussions:

The results obtained from the applied anti-stall casing treatments with different rotor blade tip axial chord exposure ratios (λ_{RTAC}) indicated that in the all cases, the one-stage compressor rig, was able to continue normal operation at much lower mass flow rates than that of the datum build.

The results obtained from the casing treatment configurations incorporated seven different values of rotor tip exposure ratios showed that the nature of the stall changed from the abrupt to progressive, except in the case of the build with the (23.2%) rotor tip exposure. Similarly all of the casing treatment configurations except that of the (0.232) rotor exposure ratio enabled operation at much lower mass flow rates. All of the configurations led to reductions in noise levels during their normal operation, but again only the six which featured progressive stall actually reduced noise levels when operating near the stall point.

However, four configurations with (0.333, 0.434, 0.535 and 0.636) rotor exposure ratios provided for further closure of the compressor test rig throttle in comparison to the other two with (73.7% and 83.8%) rotor blade exposures.

As described in the previous chapter, the casing treatments along with all of the rotor exposure ratios except the (0.232) value was tested by reducing the mass flow rate until the point at which the static pressures upstream and downstream of the inlet guide vanes were equalised. This point was actually before the onset of stall for these builds.

This decision to stop data acquisition when the static pressures were equalised was based on the fact that at the lower mass flow rates, further closing of the compressor throttle prevented the beneficial functioning of the inlet guide vanes (IGVs). This was due directly to the occurrence of higher static pressure downstream of the IGVs. Therefore with a positive value of $[(\Delta p)_{IGVs}]$ achieved from the inlet guide vanes control volume, the performance of the compressor started to deteriorate quite sharply.

Under near stall conditions it appeared that some stall cells developed at the inlet guide

vanes. Therefore even with a small decrease in the mass flow rate, the compressor was faced with an unrecoverable rotating stall, which in a very short time led to the development of surge along with associated loud noise levels.

Some of the most significant flow characteristics results obtained from the one-stage compressor with the solid and treated casings builds used in the comparison procedures are presented in [Appendix (C)]. A trendline based on the polynomial equations of power six or seven was fitted to each individual flow characteristic obtained from the examination of the compressor incorporating the casing treatment configurations.

The corresponding coefficients of the aforementioned fitted equations including the values of each flow characteristic, which were determined in relationship to rotor exposures of the applied casing treatments from these equations are also shown in [Appendix (D)]. The maximum or minimum values of each flow characteristic in addition to the corresponding rotor tip exposure were also presented in this Appendix. The flow characteristic values, which were calculated corresponding to a range of (0.232 to 0.838) rotor tip exposure ratio with an increment of (0.101) are also illustrated in [Appendix (E)].

Moreover, some of the most important improvement or loss characteristic parameters that were calculated using the flow characteristics of the one-stage compressor with casing treatment configurations as well as those belonging to the datum build, and were applied in the comparison procedures are presented in [Appendix (F)].

The magnitudes of the gains or losses in the characteristics in addition to the maximum or minimum values, which were calculated from the fitted polynomial equations of power six, corresponding to all the rotor blade tip exposures are shown [Appendix (G)]. The improvements or losses in the characteristics which were determined at increments of (1.01%) corresponding to rotor tip exposures range of (23.2% to 83.8%) are illustrated in [Appendix (H)].

9.3)- Improvements or Deterioration in terms of the Flow Rates:

The characteristics obtained in terms of the corrected mass flow rate in comparison with those of the normal mass flow rates were shown to provide slightly better results. This was confirmed by both the normal and the corrected mass flow rate characteristics obtained from all casing treatment builds when compared to those achieved from the solid casing configuration.

The characteristics of the corrected mass flow rate (Φ_e) versus the flow coefficient (ϕ) of the solid casing in addition to those belonging to the treated configurations with different values of rotor exposure ratios are shown in [Graph (9-1)].

9.3.1)- Improvements of the Corrected Mass Flow Rates at the Stall Points:

The characteristics of the mass flow rates (Φ), flow coefficients (ϕ) and the corrected mass flow rates (Φ_e) were plotted according to their values at a point close to stall conditions against the corresponding rotor blade exposure ratios (λ_{RTAC}). The results all showed desirable benefits in terms of flow rates.

All of the above characteristics confirmed that the casing treatment build featuring the (0.333) rotor exposure ratio provided the best results. The flow coefficient characteristics of the different casing treatment configurations against the corresponding rotor blade exposures are shown in [Graph (9-2)].

A trendline fitted to these characteristics based on a polynomial equation of power six

showed that a casing treatment configuration with an optimum value of (35.14%) rotor exposure can operate at the lowest flow coefficient of (0.325) at close to stall conditions.

9.3.2)- Improvements in terms of (CMFR) corresponding to Maximum (CAP):

The characteristics of the corrected mass flow rates (CMFRs) corresponding to the stall point of all the compressor builds with casing configuration confirmed that the optimum results of 1.394 [kg/sec] occurred at (0.347) rotor exposure ratio.

In order to find out whether the corrected mass flow rate corresponding to the maximum value of an individual characteristic was able to achieve similar performance, it was decided to consider the characteristic of corrected mass flow rates at the maximum values of the corrected applied powers (CAPs) of the treatment builds.

This characteristic, which showed slightly different trendline with respect to the corrected mass flow rate at near the stall conditions, indicated that the optimum results of 1.396 [kg/sec] corrected mass flow rate can be achieved with a rotor blade exposure of (37.22%). The characteristics of the corrected mass flow rate corresponding to peak values of the corrected applied power $[(\Phi\epsilon)/(\Omega\epsilon)_{\max}]$ are presented in [Graph (9-3)].

9.3.3)- Improvements in Stall Margin in terms of Corrected Mass Flow Rates:

In order to compare the flow rates obtained from the casing treatment configurations and those belonging to the solid casing build it is necessary to define a parameter called the stall margin improvement. This can be expressed in terms of the normal and corrected mass flow rates as well as the flow coefficient values, all at the stall conditions. In the current investigation the stall margin in terms of the corrected mass flow rates at near to the stall point was calculated using the following formula¹.

$$[(\Delta\Phi\epsilon)_{SP/(CMFR)}] = \frac{[(\Phi\epsilon)_{SP-SC}] - [(\Phi\epsilon)_{SP-TC}]}{[(\Phi\epsilon)_{SP-SC}]} = 1 - \frac{[(\Phi\epsilon)_{SP-TC}]}{[(\Phi\epsilon)_{SP-SC}]} \quad [9-1]$$

Where $[(\Phi\epsilon)_{SP-SC}]$ and $[(\Phi\epsilon)_{SP-TC}]$ are the corrected mass flow rates at stall conditions for the solid casing build and for the casing treatment configurations, respectively. The characteristics of the stall margin improvement in terms of the corrected mass flow rates versus the relevant rotor blade tip axial chord exposure ratios are shown in [Graph (9-4)].

It can be concluded from the above graph that the best value of the stall margin improvement achieved was (28.56%) corresponding to (34.74%) rotor exposure. This means that, in order to achieve the greatest improvement in the stall margin in terms of the flow rates, the optimum value of rotor blade exposure ratio is about (0.347).

It is worth noting that all compressor builds with casing treatments were able to continue normal operation at lower mass flow rates compared with the solid casing at near the stall conditions. But, with further reduction of mass flow rate, the build with (0.232) rotor exposure ratio immediately experienced a loud noise and an abrupt stall occurred. In contrast, other casing treatment configuration just showed progressive stall.

Although there was no opportunity to identify the exact locations of the stall cells occurrence in the experimental set-up, the loud noise in the region of the bellmouth and inlet guide vanes suggested that the stalls started in this zone.

¹ Some of the previous investigators at Cranfield University used the similar formula, but with an insignificant difference of which the stall margin improvements were calculated in terms of the flow coefficients.

With the exception of the build with (23.2%) rotor exposure, the other configurations were somehow able to operate under progressive stall conditions. This capability was however greater for the builds with (33.3%, 43.4%, 53.5% and 63.6%) rotor exposures. These results however, confirmed that the initial idea of using the casing treatment with a chamfered section at the injection part of the treatment, was a good idea. These were used in the design of the spacer rings located after the main part of the casing treatment.

However, it can be concluded that in order to achieve the best results in term of the stall margin improvement, it is necessary to have an effective vane-recessed tubular-passage casing treatment with about (35%) rotor tip exposure. This is in close agreement with the results achieved by Azimian (1987), who reported that by applying a similar type treatment to a single-stage fan, optimum results had been achieved at (35.5%) rotor blade exposure.

9.4)- Improvements or Deterioration in terms of Work Coefficients:

The overall work coefficients $[(\Psi)_{\text{overall}}]$, obtained from the compressor builds with and without the casing configurations indicated that the build with (0.333) rotor blade tip exposure ratio had shown the highest value of the work coefficient at a point close to the stall condition.

The characteristics of the maximum value of overall work coefficient against the rotor tip exposures obtained from the different configurations of the one-stage compressor builds with casing treatments are presented in [Graph (9-5)].

In order to determine the rotor tip exposure ratio corresponding to a possible peak value of the maximum overall work coefficient a trendline based on a polynomial equation with a power of six was fitted to the above characteristic. The results indicated that a casing treatment configuration with a rotor tip exposure of (33.79%) was able to provide a peak value of the maximum overall work coefficient of (0.741).

The characteristics of the overall work coefficients versus the corresponding flow coefficients of the solid casing configuration and the casing treatment builds with different values of the rotor exposure ratios are shown in [Graph (9-6)].

9.5)- Increases or Decreases in Corrected Applied Power:

It is worth noting that the corrected applied power characteristics obtained from the calculation results of the compressor with stepped and solid casing configurations indicated that the datum build behaviour was noticeably different in comparison with the casing treatment builds.

These characteristics indicated that, except for the builds with (0.232 and 0.636) rotor exposure ratios, the compressor configurations with casing treatments showed lower values of corrected applied power in comparison with the solid casing build at fully choked conditions. This situation changed at near the stall circumstance, since only the build with (23.2%) rotor exposure showed a lower corrected applied power value than that achieved with the datum build.

Moreover, each build with casing treatment had shown two corresponding maximum values of the corrected applied power. The first was located between the choke and the stall condition and the second just was at near the stall point. It is worth noting that the first maximum values of the above parameter relating to the compressor configurations with (0.232, 0.636 and 0.737) rotor tip exposure ratios were higher in comparison with the corresponding second peak values.

On the other hand, the highest values of the corrected applied power of the compressor builds with casing treatment of (0.333, 0.434, 0.535 and 0.838) rotor tip exposure ratios took place at near the stall condition. This again was in contrast to observations from the solid casing configuration.

The characteristics of the corrected applied power obtained from the solid casing build and the casing treatment configurations with different values of the rotor tip exposures against the corresponding flow coefficients are illustrated in [Graph (9-7)].

9.6)- Improvements or Deterioration in terms of the Efficiencies:

The performance of a compressor can be evaluated using a number of different characteristic parameters. The efficiency in terms of the overall and individual stage pressure ratios, can be considered as one of the most significant parameters when comparing several compressor builds, as well as evaluating a compressor rig performance under some different configurations.

The overall efficiencies $[(\eta)_{\text{Overall}}]$, and stage efficiency $[(\eta)_{\text{Stage}}]$, defined in terms of the corresponding total-total and static-static pressure ratios, are widely used in experimental investigations. As stated in chapter four, the current compressor test rig was equipped with conventional slow-response wall static tapings at the compressor inlet and outlet. It also had provision for using inter-row wall static measurements at the gaps between two adjacent blade cascades. As a result of this it was possible to consider two additional efficiency characteristics, calculated in terms of the overall total-static and stage static-static pressure ratios, respectively.

The efficiency characteristics of the datum build, in addition to those of the casing treatment configurations with different values of rotor exposures, were calculated in terms of the overall total-total, overall static-static, overall total-static and stage static-static pressure ratios. These characteristics versus their corresponding flow coefficients are shown in [Graph (9-8)], [Graph (9-9)], [Graph (9-10)], and [Graph (9-11)], respectively.

9.6.1)- Characteristics in terms of the Maximum and Near Stall Efficiencies:

In order to evaluate the characteristics of each of the above-noted efficiencies in more detail, it is more convenient to consider the individual characteristics in terms of their corresponding maximum $[(\eta)_{\text{Max}}]$, values as well as the values calculated at a point close to the stall conditions $[(\eta)_{\text{SP}}]$.

These considerations were then followed with the assessment of the corrected mass flow rates corresponding to the points at which the highest values of each efficiency characteristic was achieved. The values of the maximum efficiency characteristics of the compressor rig with treatment configurations, calculated in terms of the overall total-total, stage and overall static-static, and the overall total-static pressure ratio were plotted against treatment rotor tip exposure, as illustrated in [Graph (9-12)].

The equivalent characteristics for efficiency values at the points close to the stall conditions are shown in [Graph (9-13)]. The characteristics of the corrected mass flow rates corresponding to the maximum values of overall and individual stage efficiencies $[(\Phi\epsilon)/(\eta)_{\text{Max}}]$ are also presented in [Graph (9-14)].

By analysing the characteristics obtained from the aforementioned overall and stage maximum efficiencies relating to different rotor tip exposure ratios, it was found that, in the majority of cases, there were significant similarities. This was an encouraging result.

The characteristics of the maximum efficiency in terms of the overall total-total pressure ratio indicated that the highest value of (0.765) occurred at a rotor tip exposure of (49.16%). The characteristic in terms of the stage static-static pressure ratio was able to provide a maximum efficiency of (0.725) corresponding to (49.89%) rotor tip exposure.

The overall static-static and overall total-static maximum efficiencies were also able to achieve peak values of about (0.651) and (0.442), corresponding to (49.02%) and (52.52%) of the rotor blade axial chord exposures, respectively. However, it seems that the best results in terms of the maximum efficiencies can be achieved with a casing treatment configuration possessing a rotor tip exposure in the range of (49% to 53%).

These results indicated that in order to maximise both the overall and the stage efficiencies, it is recommended that a casing treatment configuration with a rotor exposure ratio of (0.501) should be applied.

9.6.2)- Improvements or Deterioration in terms of the Maximum Efficiencies:

As noted in chapter six, all previous researchers who carried out their studies in this area at Cranfield University stated that the improvements and deterioration in the values of the peak efficiency depend significantly upon the values of rotor tip exposure ratios. It is worth noting that although the optimum values for rotor tip exposure suggested by these investigators differed significantly, they were all located in the range of (40%) to (70%).

In order to specify the optimum values of the rotor exposure ratio of the casing treatment designed for the current investigation, it is necessary to define a formula², based on the values of the maximum efficiency of the compressor build configurations with and without casing treatments. The associated improvements or losses of the maximum efficiency $[(\Delta\eta)_{ME}]$ can be determined, using the following equation:

$$[(\Delta\eta)_{ME}] = [(\eta_{Max})_{SC}] - [(\eta_{Max})_{TC}] \quad [9-2]$$

Where; $[(\eta_{Max})_{SC}]$ and $[(\eta_{Max})_{TC}]$ are the maximum efficiencies of the solid casing build and the casing treatment configurations, respectively.

The above equation can be used to determine changes in maximum efficiencies calculated in terms of the overall total-total, overall static-static, overall total-static and the stage static-static pressure ratios. The characteristics of the overall and stage maximum efficiency improvements or losses calculated based on the maximum efficiency of the datum build versus the corresponding values of the rotor blade tip axial chord exposure is presented in [Graph (9-15)].

The gain and loss characteristics, determined in terms of the overall total-total as well as the stage and overall static-static pressure ratios, indicated that the compressor builds with (0.434) and (0.535) rotor exposure ratios were the best configurations. These builds were able to provide higher values of the maximum efficiencies in comparison with the performance achieved from the solid casing examination. The results obtained for the values of the maximum efficiencies in terms of the overall total-static pressure ratios showed that all compressor builds with casing configurations performed better than the datum build.

The variations in improvements and losses to the peak efficiencies were in fair agreement with each other, with the exception of the maximum efficiency curve obtained from the

² The previous investigators at Cranfield University also used this equation.

overall total-static pressure ratio. It will be discussed subsequently that the optimum value of the rotor exposure ratio obtained from this characteristic does not differ much from the other curves.

Although the reason behind above variations was explained in the previous chapter, it is worth mentioning it briefly here again, since it is quite noticeable. The major reason for the above noted dissimilarity is due to the fact that all overall total-static characteristics were calculated in terms of the corresponding pressure ratios based on the inlet total (ambient) and outlet static pressures.

The latter were measured by four wall static tapings at the downstream of the third stage stator blades on the same cross section at which the Cobra type total pressure probe was also positioned.

The improvement or loss characteristics in terms of the overall total-total efficiencies showed some dissimilarity for the compressor build with (0.838) rotor exposure ratio casing treatment in comparison with the overall and stage static-static characteristics. It seems that this difference was due to the fact that the value of the (83.8%) rotor exposure was probably too high, since the compressor operating losses at lower mass flow rates was significantly increased.

However, the above noted characteristics in terms of the overall total-total and overall static-static pressure ratios indicated that with the applications of (49.16%) and (49.02%) rotor tip exposures, improvements of about (1.985%) and (1.165%) in the values of the corresponding maximum efficiencies could be achieved, respectively.

Meanwhile, the similar characteristics, calculated in terms of the overall total-static and stage static-static pressure ratios confirmed that the greatest improvements in the values of the maximum efficiencies were (5.625%) and (1.958%), corresponding to rotor tip exposures of (52.52%) and (49.89%), respectively.

Thus, it can be concluded that in order to gain some improvements in the maximum efficiencies, the value of the rotor tip exposure ratio had to be selected in a range of (0.49 to 0.52). However, an optimum value of (51.15%) rotor tip exposure can be suggested by averaging the aforementioned results, with the aim of obtaining a compressor build with casing treatment providing the maximum efficiencies.

9.7)- Improvements or Deterioration in terms of Peak Pressure Rises:

Another possibility for consideration of the compressor performance can be achieved by examining the characteristics of the compressor pressure rise coefficients, which are widely used in both theoretical and experimental research.

In order to assess the performance of the one-stage compressor with the casing treatment configurations in comparison to those results obtained from the datum build, the pressure rise coefficients in terms of the overall and stage pressure ratios were calculated.

As with the procedure conducted for the efficiencies, the characteristics of the pressure rise coefficients were also calculated in terms of the overall total-total $[(\psi_{t-t})_{\text{Overall}}]$, overall static-static $[(\psi_{s-s})_{\text{Overall}}]$, overall total-static $[(\psi_{t-s})_{\text{Overall}}]$ and stage static-static $[(\psi_{s-s})_{\text{Stage}}]$, using the corresponding pressure ratios.

The characteristics of the overall and stage pressure rise coefficients of the datum and the treated casing configurations, versus their corresponding flow coefficients are shown in [Graph (9-16)], [Graph (9-17)], [Graph (9-18)], and [Graph (9-19)], respectively.

9.7.1)- Characteristics of the Peak and Near Stall Pressure Rise Coefficients:

To provide an appropriate evaluation of the pressure rise coefficient characteristics obtained from all the compressor builds incorporating casing treatment configurations, it was decided to calculate every individual pressure rise coefficient in terms of its corresponding peak $[(\psi)_{\text{Peak}}]$ value as well as the value calculated at the points near to the stall conditions $[(\psi)_{\text{SP}}]$.

As with the procedure carried out for the efficiencies, the peak and near stall values of the pressure rise coefficients were calculated in terms of the overall total-total, overall static-static, overall total-static and the stage static-static pressure ratios. The characteristics of the pressure rise coefficients according to the peak and near stall values, obtained from the compressor rig with casing treatments and plotted against the corresponding treatment rotor exposures, are presented in [Graph (9-20)] and [Graph (9-21)], respectively.

This part of the assessment will now be followed by a presentation of the beneficial characteristics of the corrected mass flow rates obtained from the points at which the peak values of the pressure rise coefficients occurred. The characteristics of the corrected mass flow rates corresponding to the peak values of the overall and stage pressure rise coefficients $[(\Phi\epsilon)/(\psi)_{\text{Peak}}]$ are illustrated in [Graph (9-22)].

The characteristics of the peak pressure rise coefficients indicated that the compressor build featuring the casing treatment with (0.636) rotor exposure ratio was the best configuration in terms of the pressure rise coefficients. This compressor treatment build was able to provide the highest value for pressure rise in comparison the performance achieved from the datum configuration.

In order to specify which casing treatment configurations were able to produce the best pressure rise performance for the one-stage compressor rig, it was necessary to analyse the overall and stage characteristics obtained from the peak pressure rise coefficients and their corresponding rotor tip exposures.

It is worth noting that as with the maximum efficiencies, the peak pressure rise coefficients also showed some similarities. However, these results also indicated that, in order to benefit from the highest value of the peak pressure rise coefficients, the optimum rotor tip axial chord exposure has to be chosen in the range of (61% to 68%). Moreover, these results are in good agreement with those reported by the most of previous researchers, working in this area in this area at Cranfield University.

The results obtained from the characteristics of the peak pressure rise coefficients in terms of the overall total-total pressure ratio show that the highest value of (0.370) was achieved with a rotor tip exposure of (61.54%). Additionally, the peak pressure rise coefficient, obtained in terms of the overall static-static pressure ratio was calculated as (0.341) with a (67.67%) rotor tip exposure.

The characteristics of the peak pressure rise coefficient, determined in terms of the overall total-static and stage static-static pressure ratios had maximum values of about (0.263) and (0.361), corresponding to (61.87%) and (63.95%) of the rotor blade axial chord exposures, respectively.

From these results it can be concluded that in order to benefit from a treatment configuration that can provide the greatest performance in terms of the pressure rise coefficient, it is recommended to use a casing treatment with a rotor tip axial chord exposure ratio of (0.637).

9.7.2)- Improvements or Deterioration in Peak Pressure Rise Coefficients:

The majority of previous investigators conducting their research in the field of the vane-recessed casing treatment at Cranfield University reported that the rotor exposure ratio has a significant influence on the compressor flow characteristics and performance such as pressure rise coefficients.

Most of these researchers claimed that a minimum value of rotor exposure ratio is essential in order to achieve an improvement in pressure rise coefficients. They also state that the treated casings with more than (50%) rotor exposure were able to provide considerable performance improvements.

However, as noted in chapter six, due to the results obtained from testing different compressor builds incorporating casing treatments with various rotor tip exposures, it was suggested that the treatment configurations with rotor tip exposure ratios in the range of (0.5-0.7) would provide optimum results in terms of pressure rises.

By applying the values of the peak pressure rise coefficients, obtained from the examination of the one-stage compressor builds with and without casing treatments, in the following equation³, the associated improvements or losses in terms of the peak pressure rise coefficients $[(\Delta\psi)_{PPRC}]$ can be calculated as follows:

$$[(\Delta\psi)_{PPRC}] = \frac{[(\psi_{Peak})_{TC}] - [(\psi_{Peak})_{SC}]}{[(\psi_{Peak})_{SC}]} = \frac{[(\psi_{Peak})_{TC}]}{[(\psi_{Peak})_{SC}]} - 1 \quad [9-3]$$

Where; $[(\psi_{Peak})_{SC}]$ and $[(\psi_{Peak})_{TC}]$ are the peak pressure rise coefficients for solid casing (datum build) and for the treated casing configurations, respectively.

The above formula can be used with all values of the peak pressure rise coefficients, calculated based on the overall total-total, overall static-static, overall total-static and the stage static-static pressure ratios.

The improvement or loss values associated with the overall and stage peak pressure rise coefficients of the compressor builds with casing treatment configurations were then calculated based on the similar parameter of the solid casing build. The above-noted improvement or loss characteristics were plotted against the corresponding values of the rotor tip axial chord exposures and are illustrated in [Graph (9-23)].

The above improvement and loss characteristics, calculated in relation to the overall total-static, overall static-static and stage static-static pressure ratios, indicated that all compressor builds with treatment configurations were able to increase the machine performance significantly in terms of the peak pressure rise coefficient.

The aforementioned results, mean that the all one-stage compressor builds incorporating the casing treatment configurations improved the datum build compressor performance, in terms of the above pressure rise characteristics. The similar characteristics in terms of the overall total-total pressure ratio showed that except for the casing treatment builds with (0.232, 0.333 and 0.838) rotor blade tip exposure ratios, the treatment configurations were able to improve the compressor performance significantly in terms of the peak pressure rise coefficients.

However, the characteristics of the improvements and losses, in terms of the all overall as well as stage peak pressure-rise coefficients indicated that the compressor builds with

³ Previous researchers at Cranfield University have used similar formulae.

(0.535, 0.636 and 0.737) rotor exposure ratios were the best configurations. Moreover, this meant that these builds were able to provide higher values of the peak pressure rise coefficients in comparison to those achieved with the solid casing.

It is worth noting that the changes in the peak pressure rise coefficients obtained in terms of the overall total-static pressure ratio showed some dissimilarities in characteristic values as well as the trendline characteristics in comparison to the other overall and stage characteristics.

However, these differences are due to the pressure values, measured through the wall static tappings downstream of the third stage stator blades. As was discussed in previous chapters, the ambient pressure in the test area was considered as the inlet total pressure during the current experimental investigation.

The dissimilarity in the trendlines is not overly significant. The optimum value of the rotor tip exposure ratio achieved from the peak pressure rise coefficients improvement and loss characteristics in terms of the overall total-static pressure ratio was shown to be very close to results achieved for the overall and stage characteristics.

The greatest improvements in the peak pressure rise coefficients were (2.698%, 4.442%, 5.164%, & 28.451%) in terms of the overall total-total, stage and overall static-static and overall total-static, respectively. These improvements were determined with (61.54%, 63.94%, 67.66% & 61.87%) of the rotor tip axial chord exposures, respectively.

It can be concluded that for the objective of improving the performance of the compressor by increasing the peak pressure rise coefficients, it is necessary to apply a vane-recessed treatment with a rotor tip exposure ratio in range of (0.61) to (0.68). By averaging the four values of the above-noted rotor exposures, a value of (63.75%) can be suggested to obtain the highest values of peak pressure rise coefficient in comparison to the datum build.

9.8)- Improvements or Deterioration in terms of the Flow Rates and Pressure Rises:

Different values of improvements and losses are achieved when different equations are utilised for the specification of compressor performance with and without casing treatment. It has already been stated that the largest improvement in stall margin in terms of the corrected mass flow rates was calculated of (28.56%) with an associated rotor tip exposure of (34.74%). Meanwhile the improving of the stall margin in terms of the flow coefficients was found to be of (28.92%) with a corresponding rotor exposure of (35.14%).

On the other hand, the characteristic of the maximum efficiencies calculated in terms of the overall and stage pressure ratios have shown that the peak values of the maximum efficiencies obtained corresponded to a rotor tip exposure range of (49.02% to 52.52%). Moreover, the overall and stage flow characteristics results obtained in terms of the peak pressure rise coefficients showed that the best results can be achieved with rotor tip exposures in the range of (61.54% to 67.66%).

The above results meant that it would be difficult to specify an optimum value of the rotor tip exposure ratio in the range of (0.347 to 0.676). To the author's knowledge, this area of the investigation received insufficient consideration from other researchers. This lack of coverage was not only apparent from researchers working in the field of the vane-recessed treated casings, but also from others investigating different types of the casing treatments.

However, in order to provide appropriate evidence for the previously described relationship, the improvement in stall margin has been considered using the pressure rise

coefficients and corrected mass flow rates (both at near the stall conditions) obtained from the solid casing build and treated casing configurations.

9.8.1)- Improvements in Stall Margin in terms of (CMFRs) and (PRCs), Both at Near Stall Points:

The stall margin improvement in terms of the pressure rise coefficients (PRCs) at near the stall points and the corresponding corrected mass flow rates is a significant parameter that can be used to assess the performance of the compressor with and without casing treatments.

The improvement of the stall margin using the pressure rise coefficients and the corrected mass flow rates both at near the stall conditions $\{\Delta[(\Phi\epsilon)_{SP} * (\psi)_{SP}]\}$ was defined by the following equation:

$$\Delta[(\Phi\epsilon)_{SP} * (\psi)_{SP}] = \frac{\{[(\Phi\epsilon)_{SP-SC}] * [(\psi)_{SP-SC}]\} - \{[(\Phi\epsilon)_{SP-TC}] * [(\psi)_{SP-TC}]\}}{\{[(\Phi\epsilon)_{SP-SC}] * [(\psi)_{SP-SC}]\}} \quad [9-4]$$

Where; $[(\Phi\epsilon)_{SP-SC}]$ and $[(\Phi\epsilon)_{SP-TC}]$ are the corrected mass flow rates at near the stall points for the solid casing build and for the casing treatment configurations, respectively. Additionally; $[(\psi)_{SP-SC}]$ and $[(\psi)_{SP-TC}]$ are also the pressure rise coefficients at the stall conditions for the datum configuration and for the casing treatment builds. This expression, albeit with slight differences, has been used by the previous researchers⁴.

Equation [9-4], can be applied, using the overall and stage peak pressure rise coefficients, calculated in terms of the overall total-total, overall static-static, overall total-static and the stage static-static pressure ratios.

The improvements or losses associated with the overall and stage pressure rise coefficients at the stall conditions and their corresponding corrected mass flow rates of the compressor with casing treatments relative to the similar parameters belonging to the solid casing build were then calculated. These characteristics were then plotted against the corresponding values of rotor tip axial chord exposures, as are illustrated in [Graph (9-24)].

As can be seen from these characteristics, all compressor builds incorporating casing treatments were able to improve the stall margins in terms of the overall total-total as well as the stage and overall static-static pressure rise coefficients and corresponding corrected mass flow rates at the stall points.

Meanwhile, the characteristic determined in terms of the overall total-static pressure rise coefficients, indicated that except for the compressor builds with (0.737 and 0.838) rotor tip exposure ratio treated casings, the other treatment configurations were able to offer better performance than that of the solid casing.

Moreover, all above-noted stall margin improvement characteristics indicated that the compressor builds with (0.232, 0.333 and 0.434) rotor exposure ratio casing treatments, were the best configurations.

However, the highest values of the stall margin improvements achieved in terms of the pressure rise coefficients and the corresponding corrected mass flow rates at stall points, were (34.68% and 35.53%) in relationship to the overall total-total and stage static-static pressure ratios, respectively.

⁴ (Azimian A. R., Elder R. L. & McKenzie A. B., 1990) and (Pundhir D. S., 1994).

The rotor blade tip exposure values associated with these stall margin improvements were calculated as (32.30% and 33.36%), respectively. The maximum values of the above stall margin improvements in terms of the overall static-static and overall total-static pressure ratios were determined to be (32.36% and 15.39%), with the corresponding rotor exposures of (32.57% and 31.14%), respectively.

It is possible to conclude that in order to achieve the highest values of the stall margin improvements in terms of the pressure rise coefficients and corresponding corrected mass flow rates at near the stall points. It is necessary to use a casing treatment configuration with a rotor blade tip axial chord exposure ratio in range of (0.311) to (0.326).

By averaging the four above-noted rotor blade exposures, an optimum value of (32.35%) can be derived. A casing treatment with aforementioned exposure can be used in order to maximise values of the stall margin improvements in terms of the pressure rise coefficients at stall points and corresponding corrected mass flow rates.

9.9)- General Conclusions:

This section presents a summary of the conclusions that can be drawn from the results of the current investigation. The study examined the first stage, including the inlet guide vanes, of a three-stage axial-flow compressor, and involved the application of casing treatments with seven different values of rotor tip exposure ratios. The most important points arising from the investigation are as follows:

- 1)- A vane-recessed-recessed tubular-passage casing treatment has the potential to significantly delay the initiation of stall in both single-stage and multistage axial-flow compressors.
- 2)- In the all cases, the compressor rig incorporating the casing treatments was able to continue operating normally at much lower mass flow rates than that of the solid casing configuration.
- 3)- The compressor with casing treatments along with all of the rotor exposure ratios except the (0.232) value was tested by reducing the mass flow rate until the point at which the static pressures upstream and downstream of the inlet guide vanes were equalised. This point was actually before the onset of stall for these builds.
- 4)- The nature of the stall changed from abrupt to progressive in six compressor configurations with casing treatments, except in the case of the build with (0.232) rotor tip exposure ratio.
- 5)- Four configurations with (0.333, 0.434, 0.535 and 0.636) rotor exposure ratios provided further closure of the compressor test rig throttle in comparison to the other two with (73.7% and 83.8%) rotor blade exposures.
- 6)- All of the compressor incorporated casing treatment configurations led to reductions in noise levels during their normal operation, but again only the six which featured progressive stall actually reduced noise levels when operating near the stall point.
- 7)- With the exception of the compressor configuration with (0.232) rotor exposure ratio, the other builds were able to operate, albeit with a reduction in performance, under progressive stall conditions. This capability was however greater for the builds with (33.3%, 43.4%, 53.5% and 63.6%) rotor exposures.
- 8)- Near stall conditions, some stall cells developed at the inlet guide vanes. Therefore even with a small decrease in the mass flow rate, the compressor was faced with an

unrecoverable rotating stall, which in a very short time led to the development of surge along with associated loud noise levels.

- 9)- All of the casing treatment configurations except that of the (23.2%) rotor exposure enabled operation at much lower mass flow rates, when the stall cells started to develop.
- 10)- The characteristics obtained in terms of the corrected mass flow rate in comparison with those of the normal mass flow rates were able to provide slightly better results.
- 11)- All flow characteristic results confirmed that the casing treatment build featuring the (0.333) rotor exposure ratio provided the best results. About (28.89%) stall margin improvements in terms of flow coefficients were achieved from using this compressor configuration.
- 12)- The best value for stall margin improvement in terms of the corrected mass flow rate was (28.57%) and this was achieved with the (0.347) rotor exposure ratio.
- 13)- In order to achieve the maximum improvements in stall margin in terms of flow rates, the optimum value of the rotor tip axial chord exposure has to be about (35%).
- 14)- The occurrence of loud noise in the region of the bellmouth and inlet guide vanes suggested that the stall cells were developed in this zone, particularly near the leading edges of the IGVs.
- 15)- The datum build behaviour was noticeably different to those achieved from the casing treatment configurations when compared in terms of the corrected applied power characteristics.
- 16)- Each build with the casing treatment showed two corresponding maximum values of the corrected applied power. The first was located between choke and stall conditions and the second was achieved close to the stall point.
- 17)- The first maximum values of the corrected applied powers relating to the compressor configurations with (0.232, 0.636 and 0.737) rotor tip exposure ratios were higher in comparison to the corresponding second peak values.
- 18)- The highest values of the corrected applied power of the compressor build with (0.333, 0.434, 0.535 and 0.838) rotor tip exposure ratios casing treatments were achieved at points close to the stall condition. This was in contrast to the observations from the datum build.
- 19)- The compressor build incorporating the casing treatment with rotor tip exposure ratio of (0.333) showed the highest value of the work coefficient at a point close to the stall condition.
- 20)- The compressor build with (0.535) rotor exposure ratios was the best configuration in terms of efficiency gain and loss characteristics. This build was able to provide highest values of the maximum efficiencies in comparison with the performance achieved from the solid casing.
- 21)- In order to achieve the best results in terms of the maximum efficiency, a casing treatment configuration possessing a rotor tip exposure in the range of (49% to 53%) has to be applied into the compressor with the solid casing.
- 22)- An optimum value of the rotor tip axial chord exposure ratio of about (0.51) is suggested in order to obtain a compressor build with a casing treatment that offers maximum efficiencies.

- 23)- The compressor build with casing treatment of (0.636) rotor exposure ratio was the best configurations in terms of the pressure rise coefficients. This treatment build was able to provide the highest value of pressure rise in comparison to the characteristics achieved from the datum build.
- 24)- To benefit from the peak value of the pressure rise coefficients, the optimum rotor tip exposure has to be chosen in the range of (61% to 68%).
- 25)- An optimum value of the rotor tip axial chord exposure ratio of about (0.64) can be recommended for a casing treatment which is able to present the highest values for peak pressure rise coefficient in comparison to that of the datum build.

Finally it must be stressed that in order to achieve maximum benefit from a vane-recessed tubular-passage casing treatment in a multistage axial-flow compressor it is necessary to carefully consider all of the design recommendations contained in chapter seven. These include the formulas and related casing treatment design procedures. It is worth noting that some of the novel design recommendations including a number of the presented equations for the casing treatment geometry and shapes are unique to the current investigation.

9.10)- Suggestions and Recommendations for Future Investigations:

Much encouraging progress has been made to date in the study of the use of vane-recessed casing treatments as a passive anti-stall device. However it must be emphasised that this research area is still in its early stages and more work needs to be carried out in order to be confident that the optimum performance from this type of casing treatment can be fully realised.

The author hopes that the following suggestions and recommendations can provoke some new ideas for future investigations. These recommendations could be applied firstly to a low-speed axial flow compressor and with some modification to a high-speed machine as well.

It is recommended to start with a four or five-stage constant-annulus low-speed axial-flow compressor, as the test rig. This must have the facility to provide for different values of hub to tip diameter ratio. The compressor rig would have to incorporate inlet guide vanes, a plenum chamber to measure the static pressure of the exit flow, and several total pressure probes.

A vane-recessed casing treatment with complete tubular-passage and a set of fully curved vanes has to be designed according to the compressor geometry by using the design recommendations presented by previous researchers in addition to the guidelines compiled in chapter seven of the current study.

It is proposed that the next phase of investigation should feature a completely novel tubular-passage treatment. It is recommended that both the inner diameter of the outer ring and the outer diameter of the inner ring have to be designed using optimum curved shapes across the axial lengths of both.

The first part of the investigation should involve testing the compressor rig with all stages present in order to establish the base performance of the machine. This would be used to inform the subsequent design process.

Once a casing treatment design has been established this should be tested using a compressor rig configuration which includes only the inlet guide vanes and the first stage rotor blades.

The tests should involve rig builds utilising both the original solid, and the treated casings. Although one option would be to follow this with testing the second, third, fourth, and fifth stage rotors, with and without the casing treatment, this is not the preferred approach. Rather it is recommended that in order to reduce the necessary technician support, that tests are conducted on each stage in turn using rotors and stators in place, with and without the casing treatment applied.

It is worth noting that the tests have to be carried out, using the inlet guide vanes in all cases. The stagger angles of the moving and stationary blades of the compressor have to be fixed as designed values.

It is strongly recommended that the compressor test rig include fast-response total pressure measuring devices, located at both the inlet and outlet of the compressor as well as between each row of rotating and stationary blade cascades.

It is also recommended that the characteristics of the compressor builds with and without casing treatment configurations are calculated in terms of the overall total-total and overall total-static pressure ratios, in addition to the individual stage and rotor only total-total pressure ratios. This would provide a significant opportunity to have a more comprehensive set of results for purposes of comparison between the builds with and without casing treatments.

The test have to be carried out using the builds with and without the casing treatments in order to find the optimum values of the rotor blade tip axial chord exposure ratios, corresponding to each individual rotor and stage configuration.

It is also recommended that the same examination procedures should be followed using different values of hub to tip ratios as well as with different numbers of recessed guide vanes inside the treated casings.

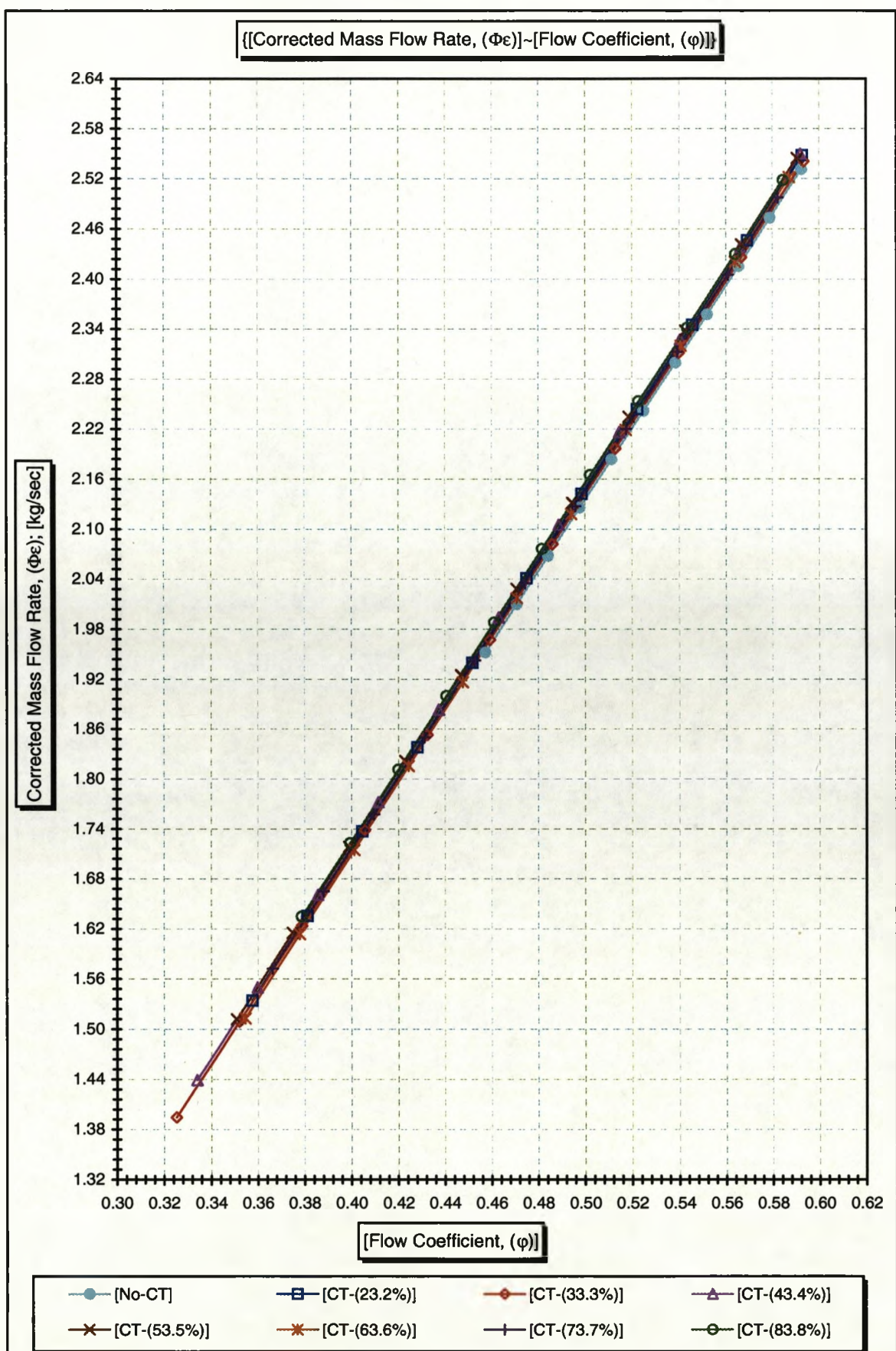
With an appropriate set of results, a matrix of data, obtained from the different flow characteristics will be available in order to analyse the performance of the designed casing treatments.

A similar approach based on the above recommendations can also be applied to test a high-speed multistage axial flow compressor.

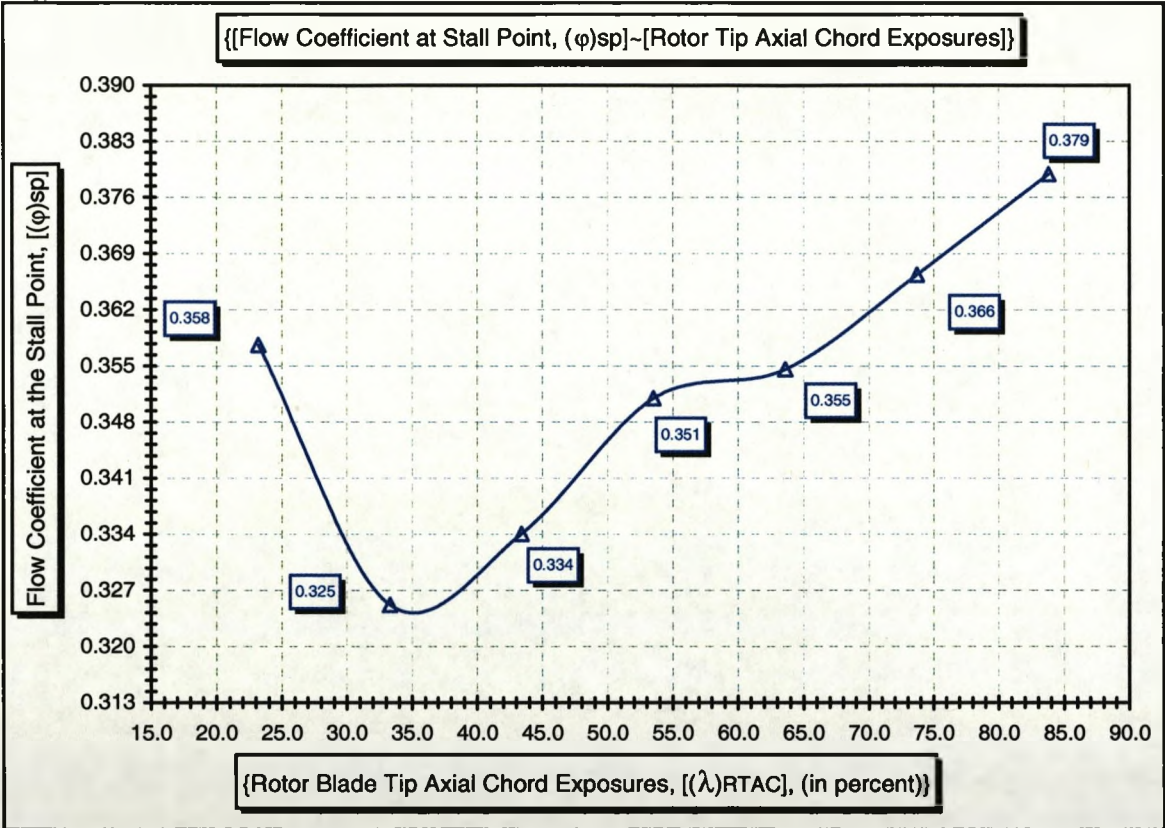
The author would like to end the current dissertation by presenting two wishes. Firstly to wish all the best to the previous researchers who conducted their studies in this area and have provided some valuable guidelines for the current experiments. Secondly to wish all success to future investigators who are going to pursue their studies in the same field as the present research.

Finally it is my sincere hope that the results of the current investigation will be of benefit to all humankind.

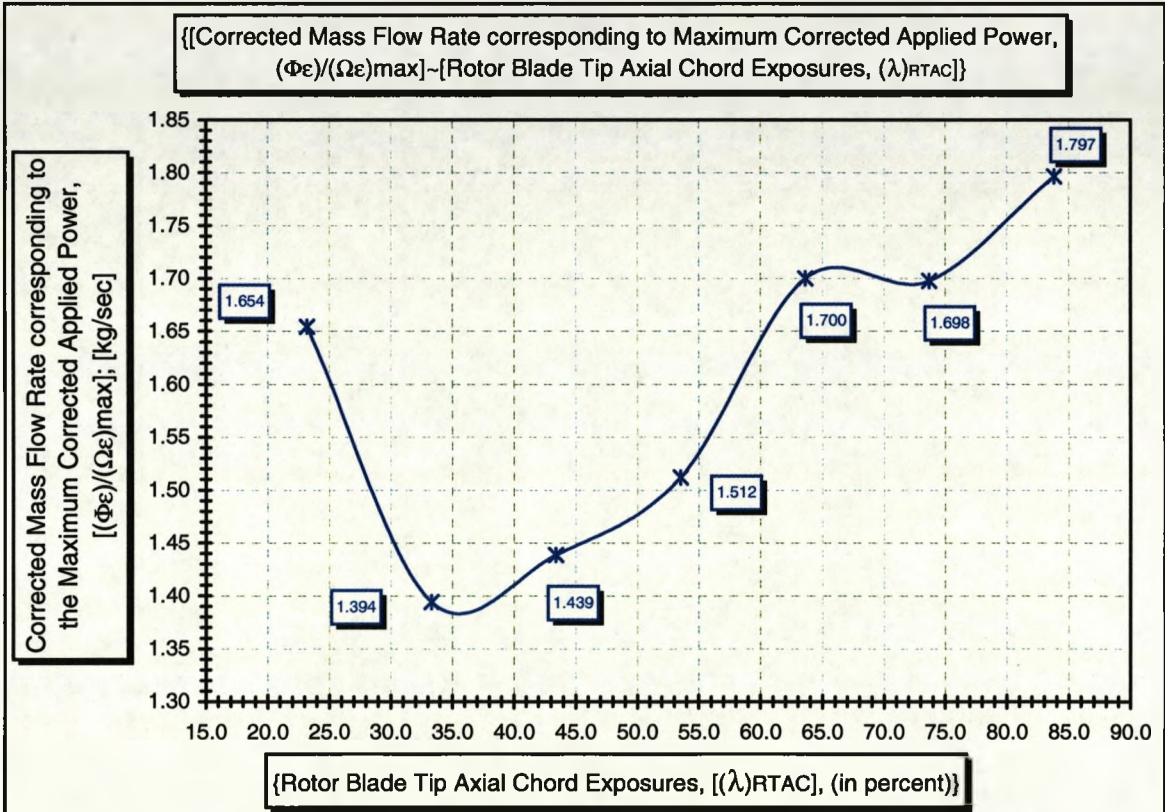
Mohammad Akhlaghi
Department of Power, Propulsion and Aerospace Engineering
School of Engineering
Cranfield University, England, UK
November, 2001



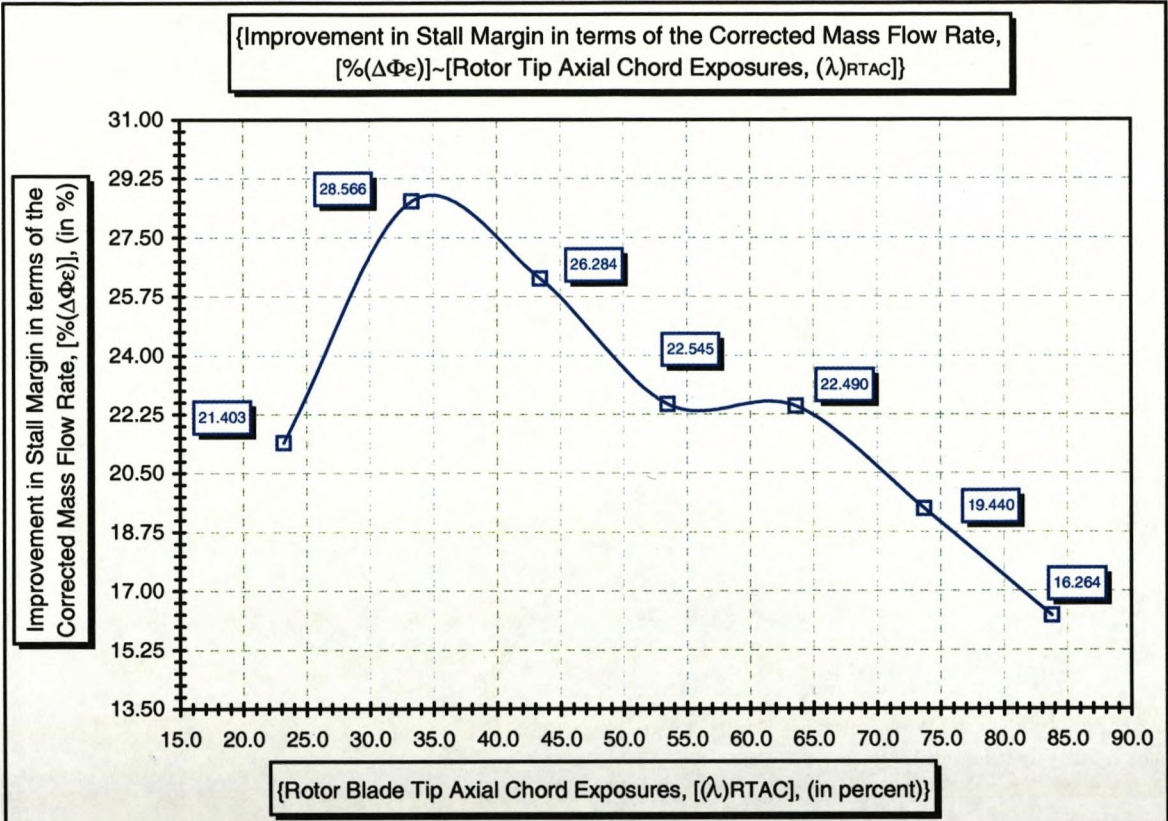
[Graph (9-1)]- The characteristics of the corrected mass flow rate versus flow coefficient of datum build and casing treatment configurations with different rotor exposure ratios



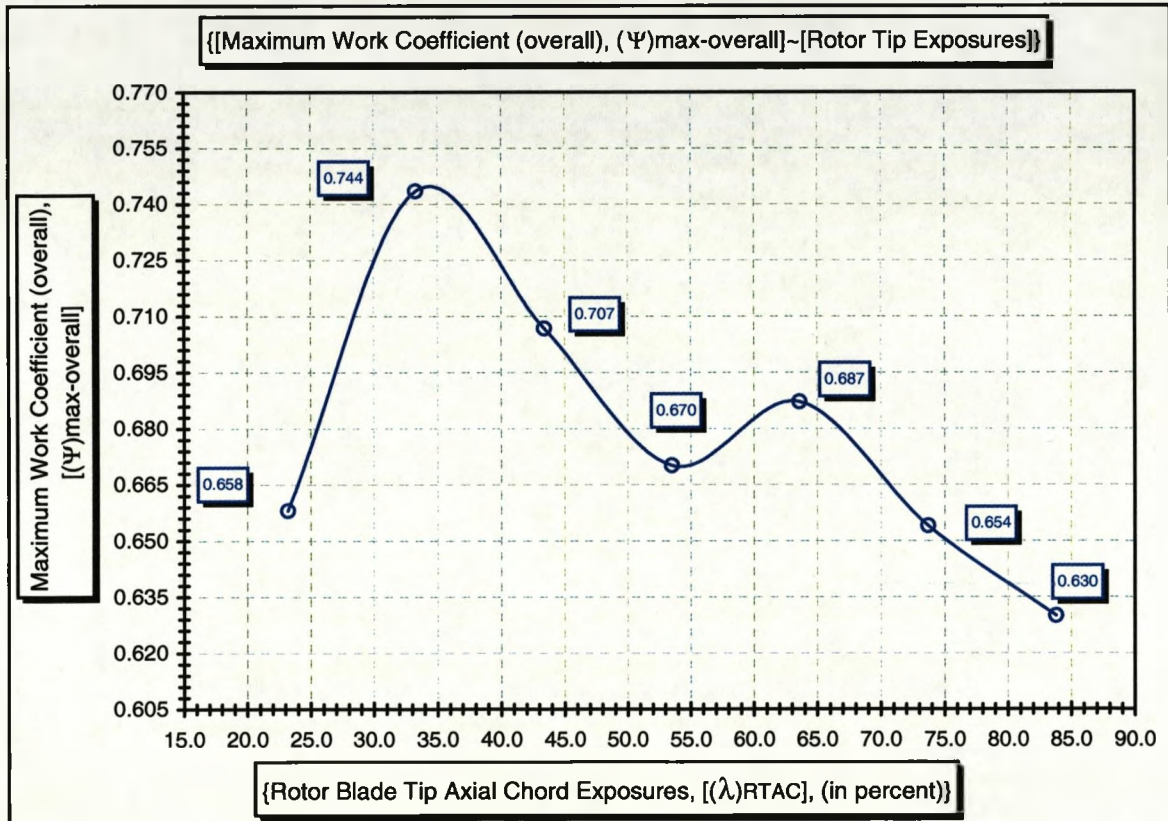
[Graph (9-2)]- The characteristic of flow coefficient near the stall points obtained from the results of different casing treatment builds against the corresponding rotor exposures



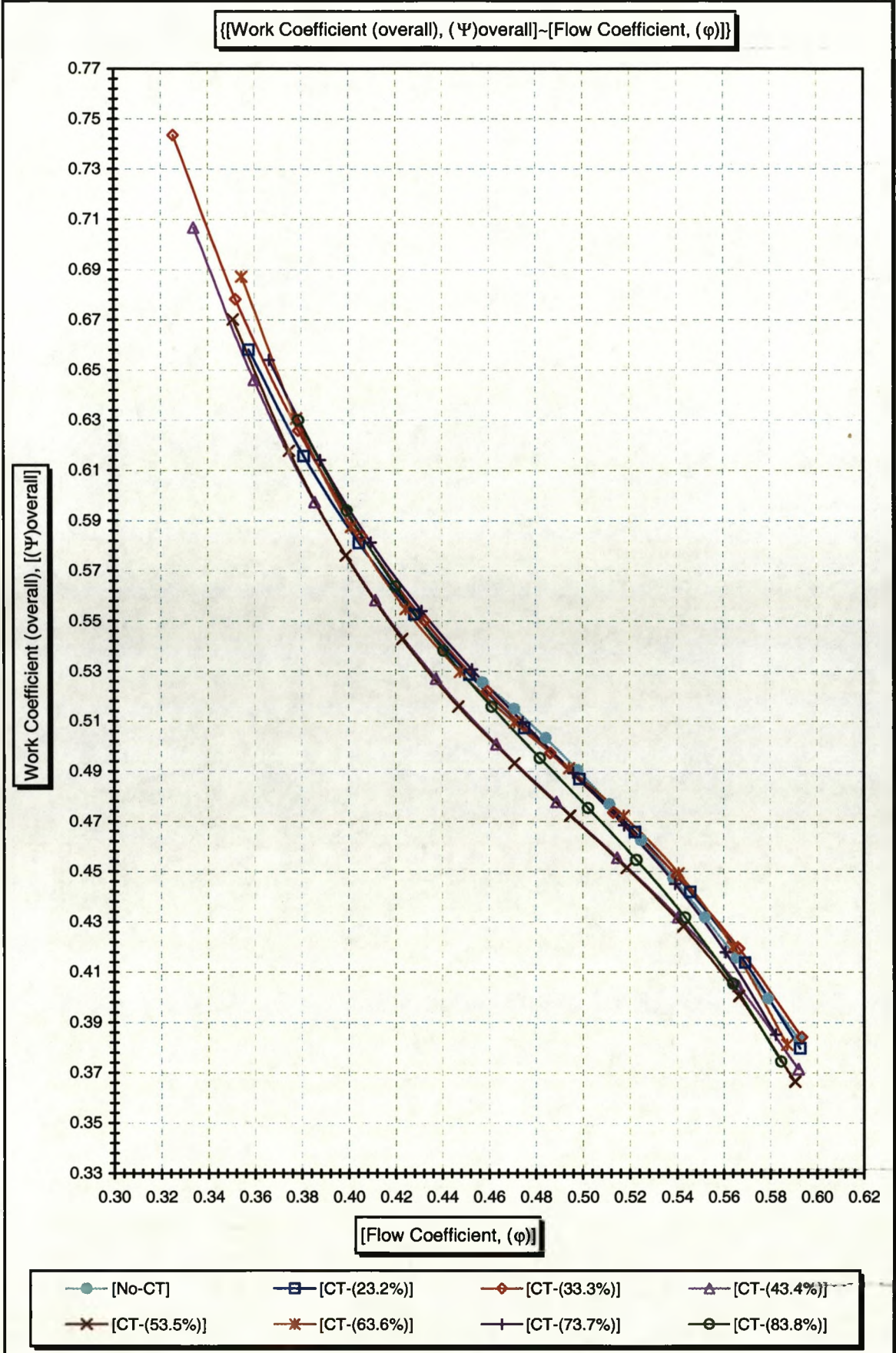
[Graph (9-3)]- The characteristics of the corrected mass flow rate corresponding to peak values of the corrected applied power of treatment builds versus relevant rotor exposures



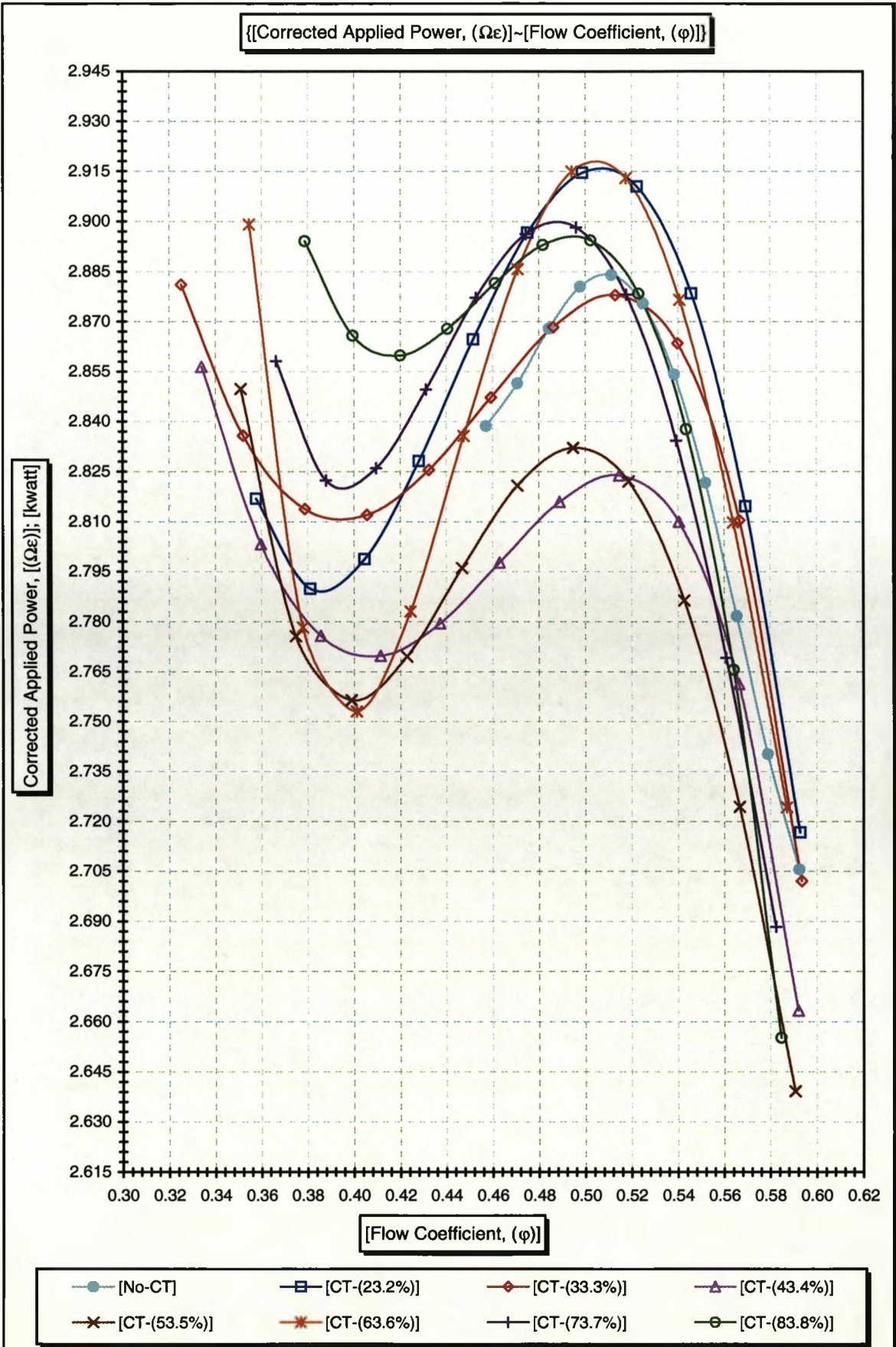
[Graph (9-4)]- The characteristic of the stall margin improvement in terms of the corrected mass flow rate near the stall points of treatment builds against relevant rotor



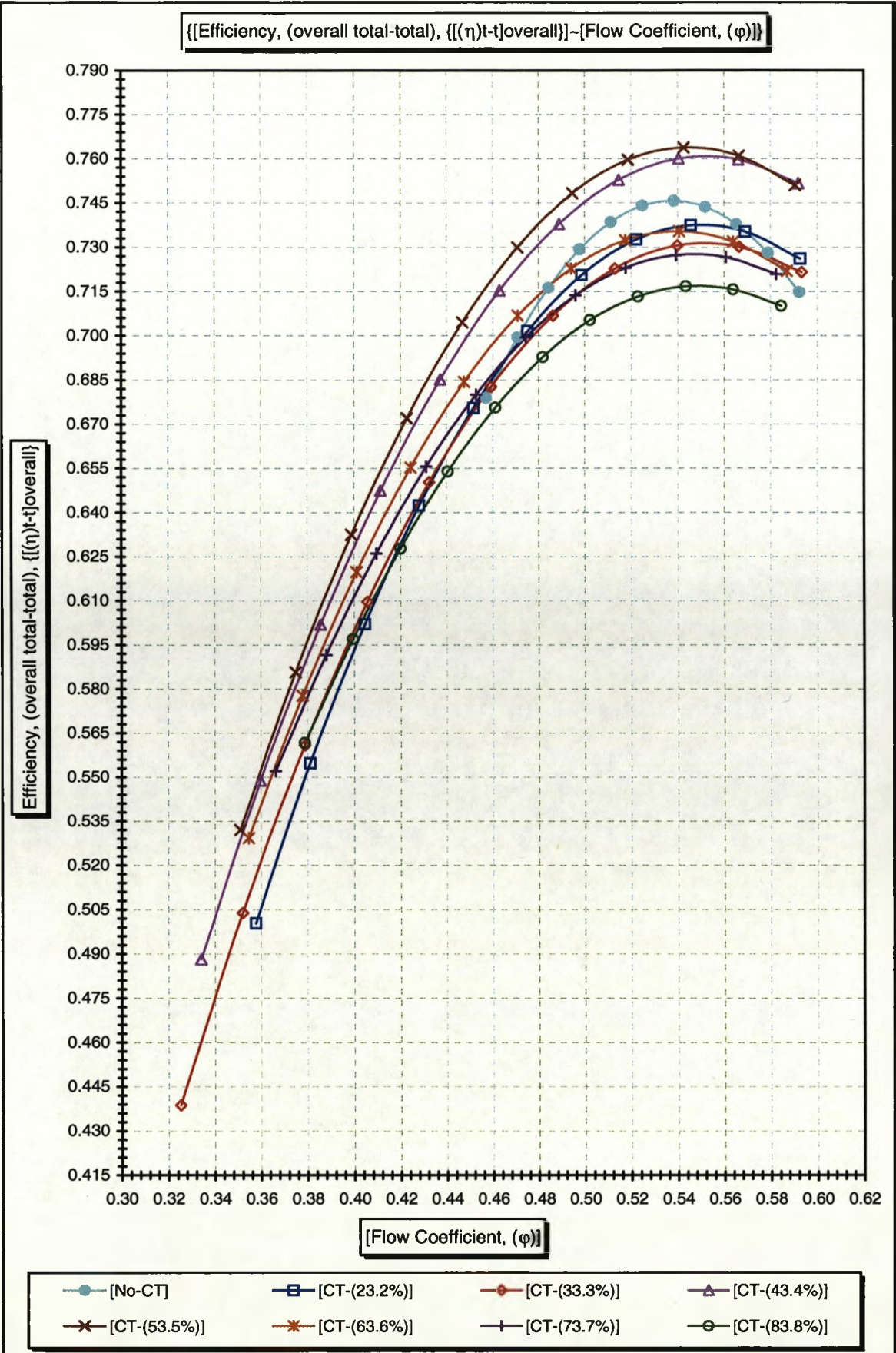
[Graph (9-5)]- The characteristics of the maximum value of the overall work coefficients versus the rotor exposures obtained from different casing treatment configurations results



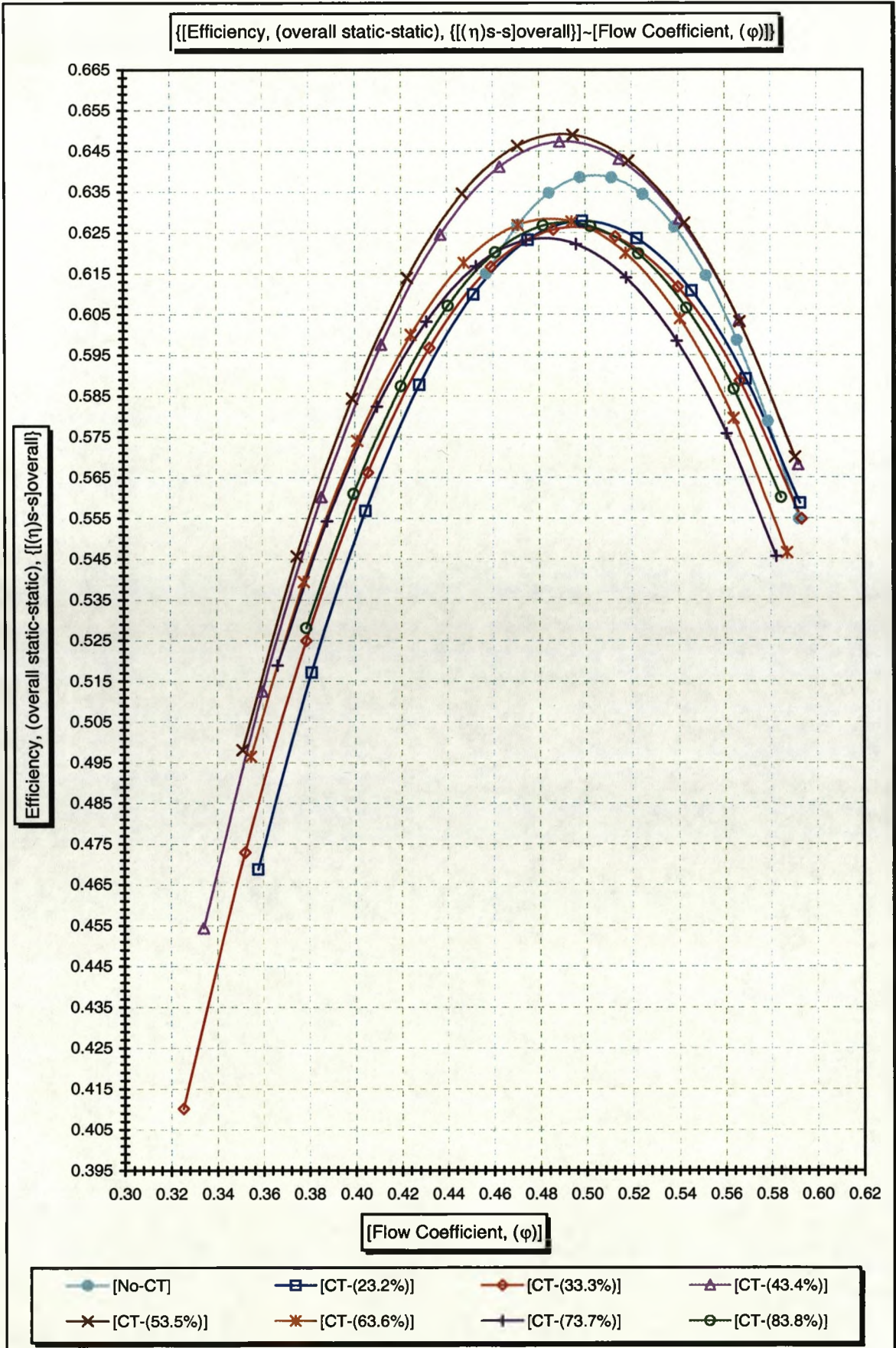
[Graph (9-6)]- Characteristics of the overall work coefficient against the flow coefficient of the datum build and the casing treatment configurations with different rotor exposures



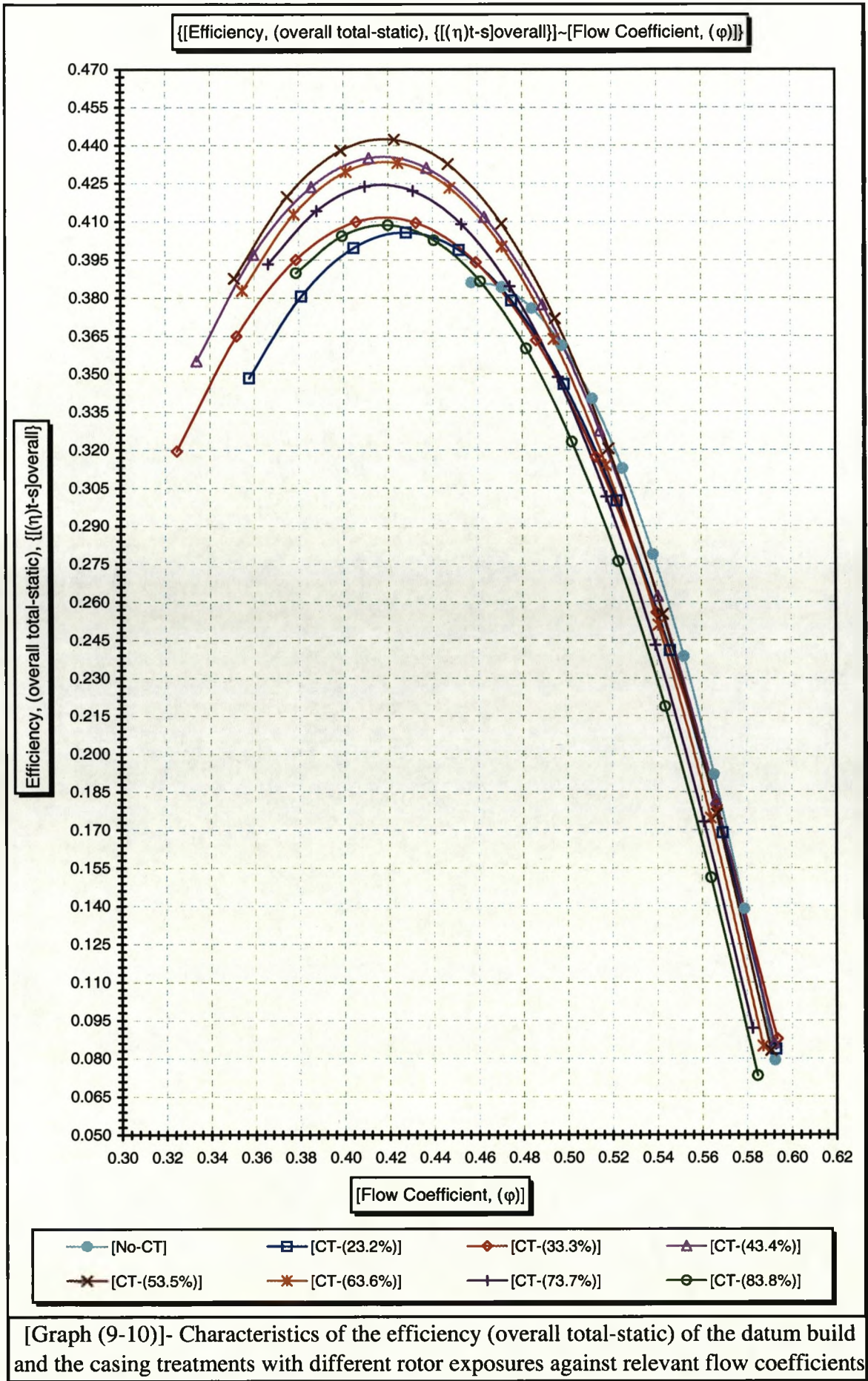
[Graph (9-7)]- Characteristics of the corrected applied power versus the flow coefficient of the datum build and the casing treatment configurations with different rotor exposures

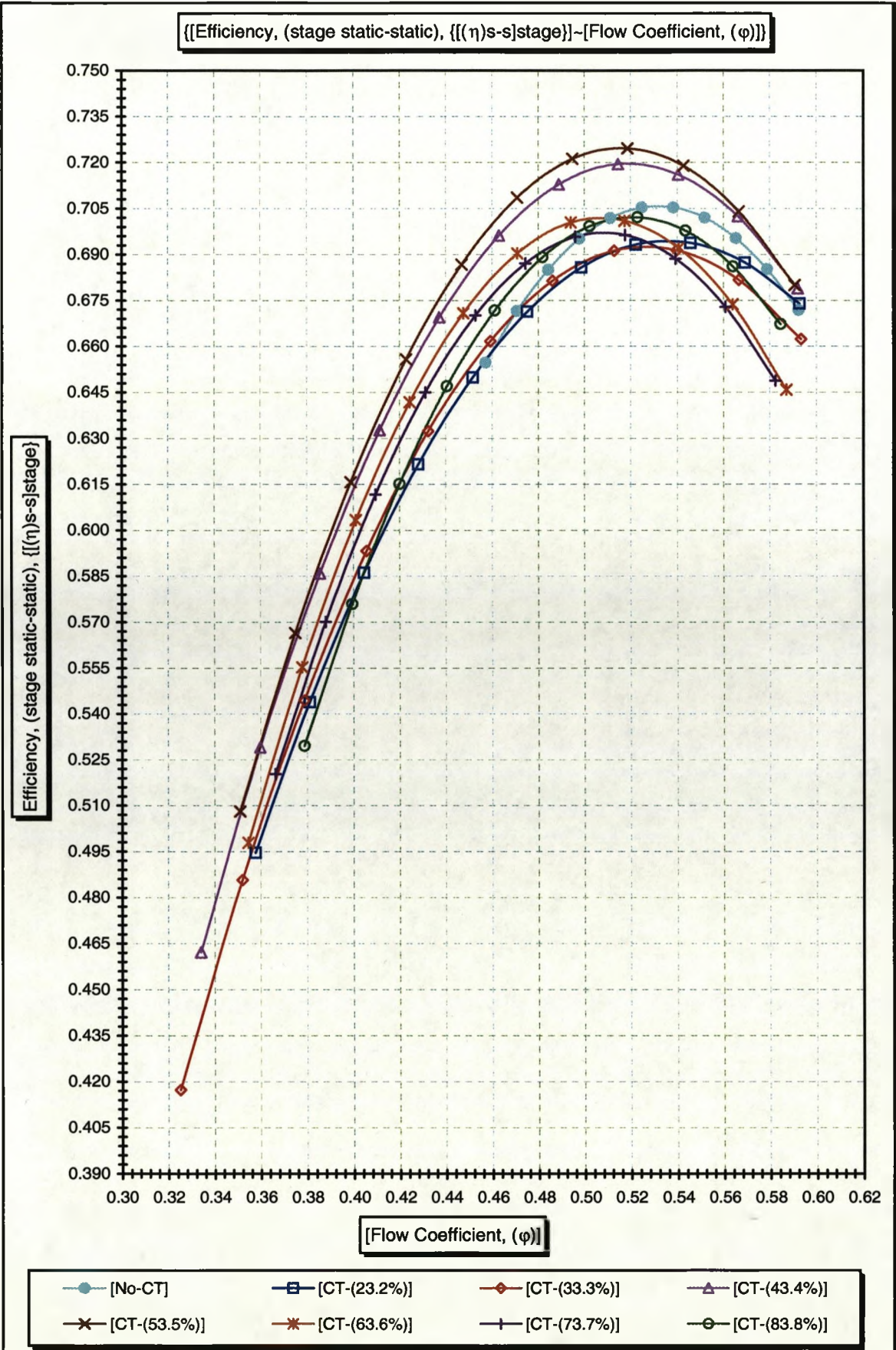


[Graph (9-8)]- Characteristics of the efficiency (overall total-total) of casing treatments with different rotor exposures and the solid casing build versus relevant flow coefficients

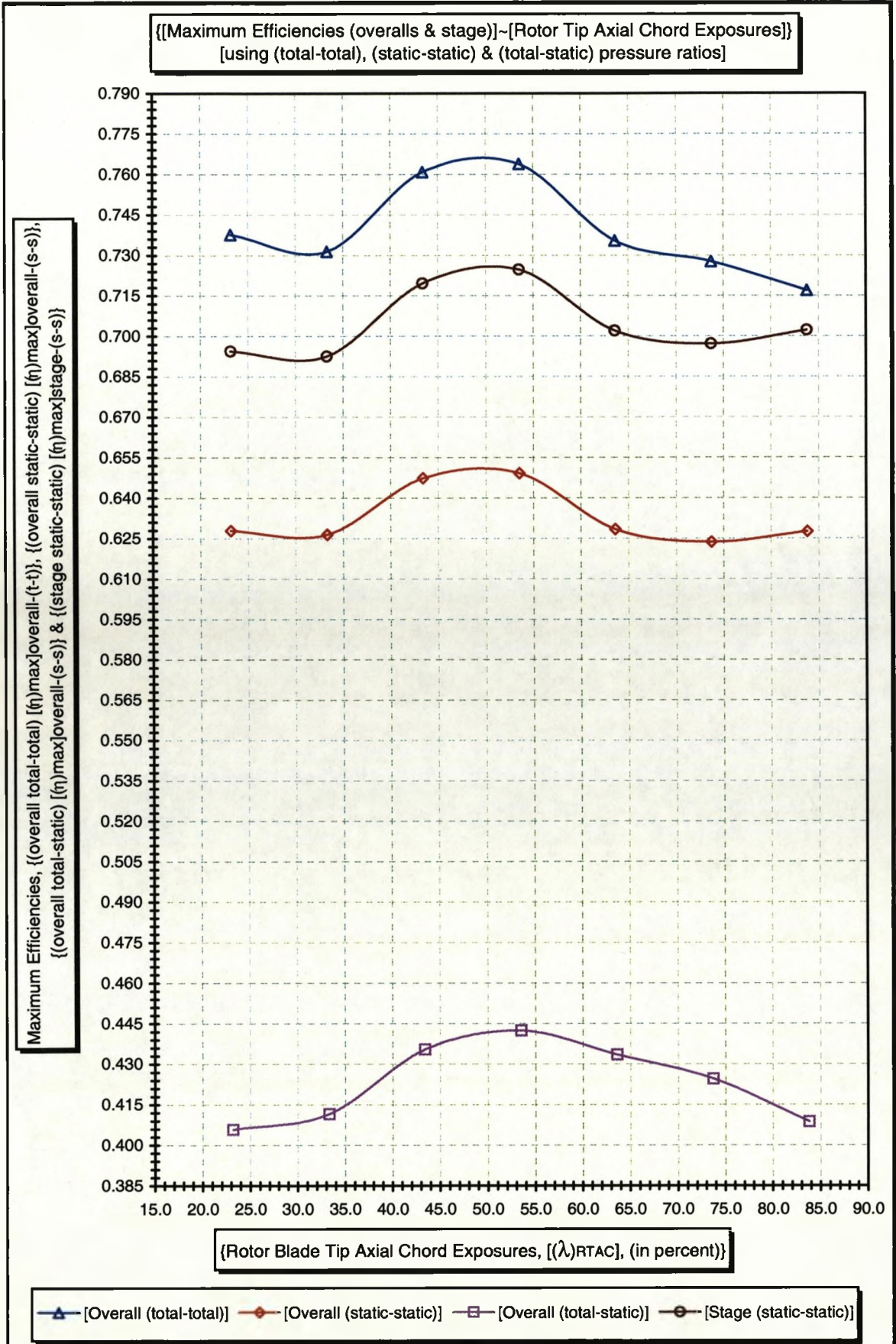


[Graph (9-9)]- Characteristics of the efficiency (overall static-static) of the datum build and the casing treatments with different rotor exposures against relevant flow coefficients

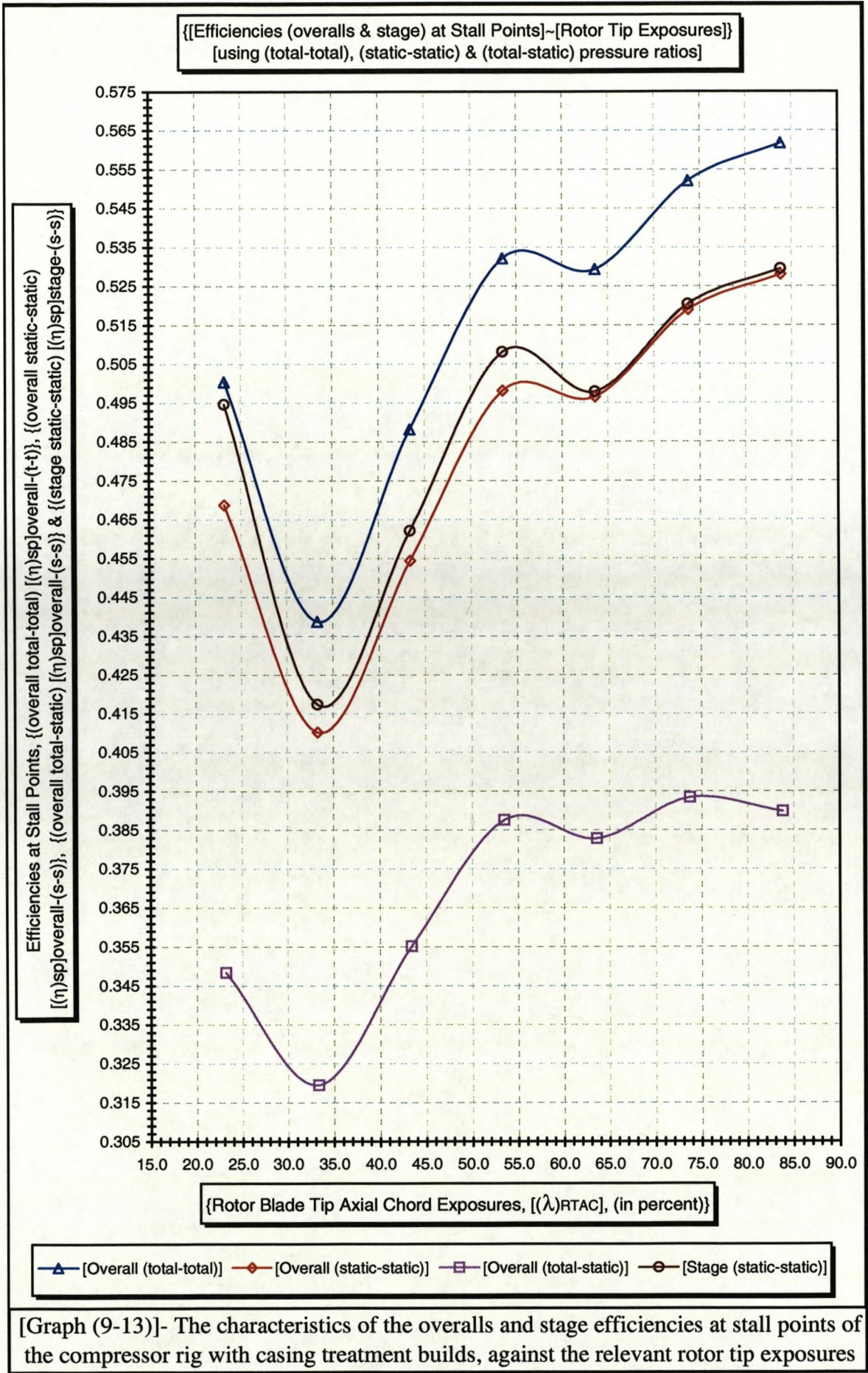


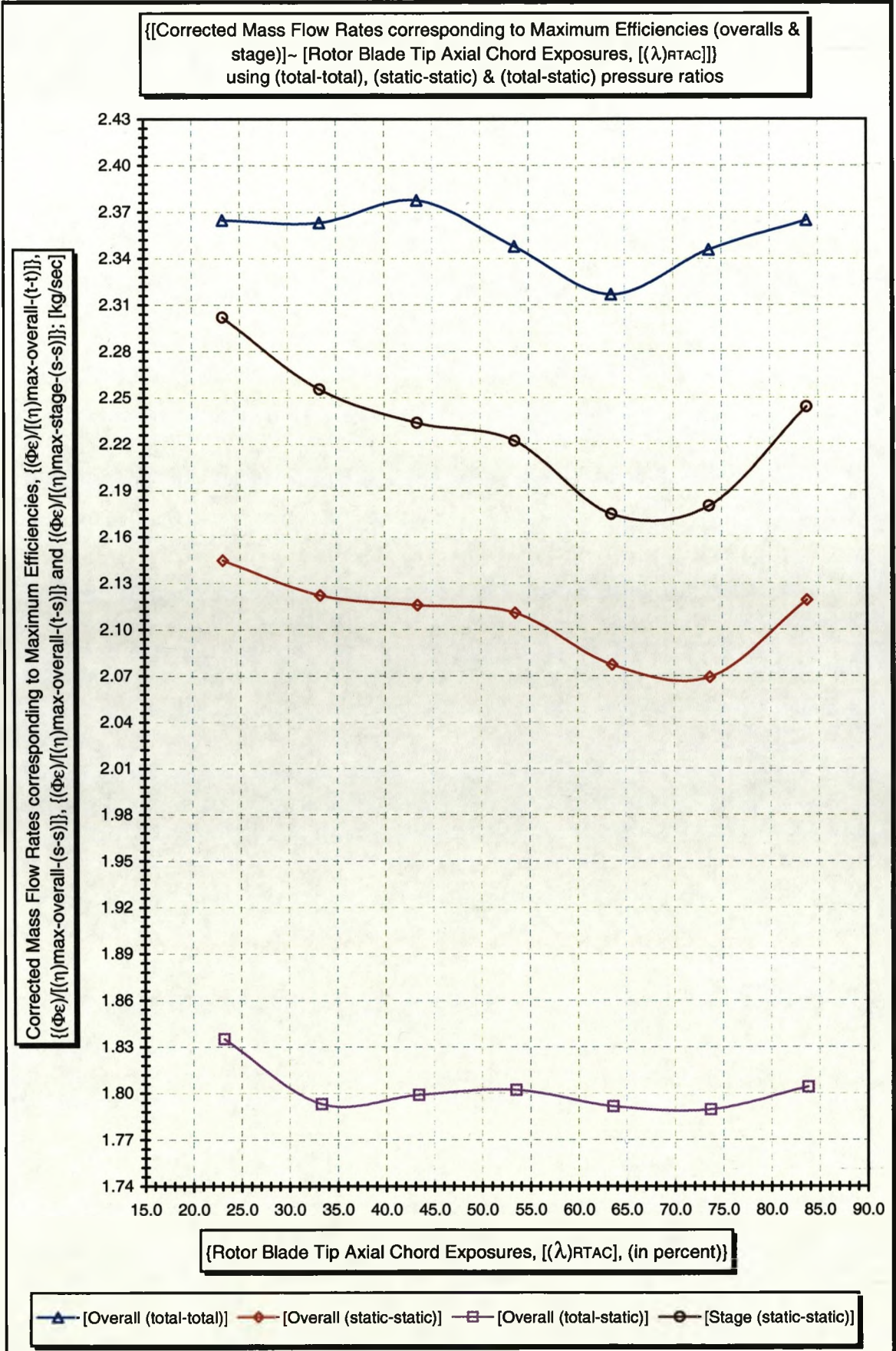


[Graph (9-11)]- Characteristics of the efficiency (stage static-static) of the solid casing build and the casing treatments with different rotor exposures versus flow coefficients

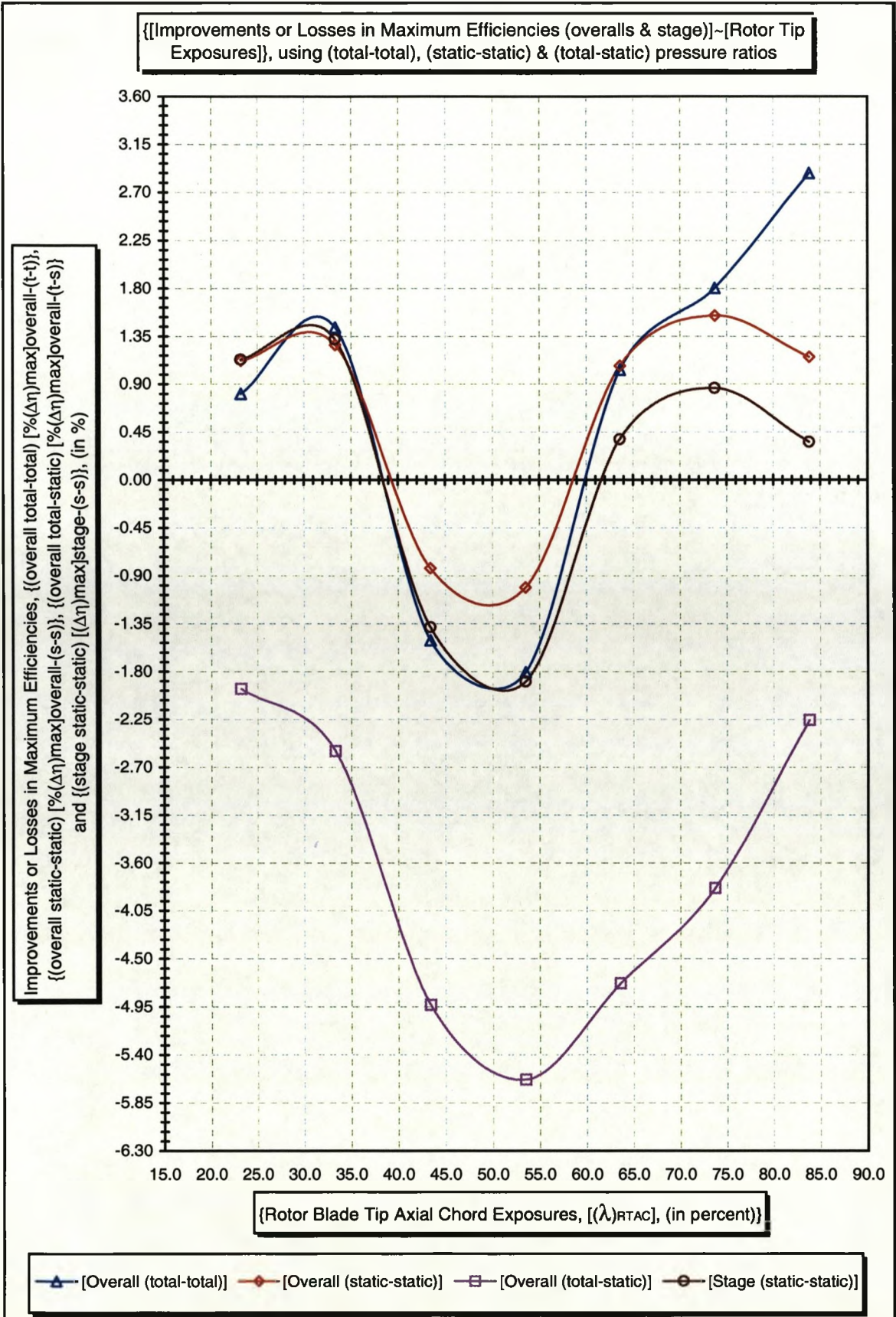


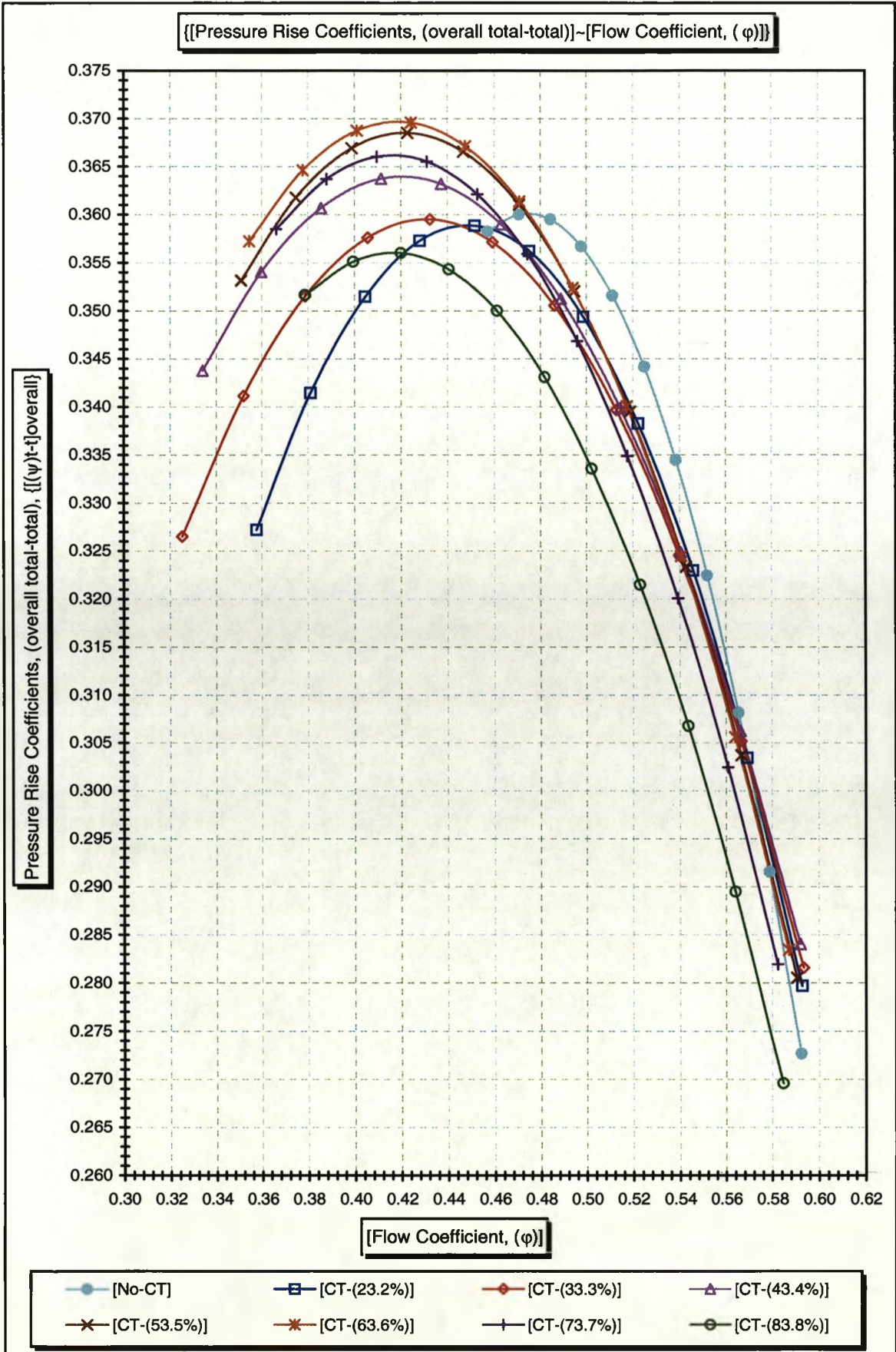
[Graph (9-12)]- The characteristics of the overalls and stage maximum efficiencies of the compressor rig with casing treatment builds, versus the corresponding rotor tip exposures

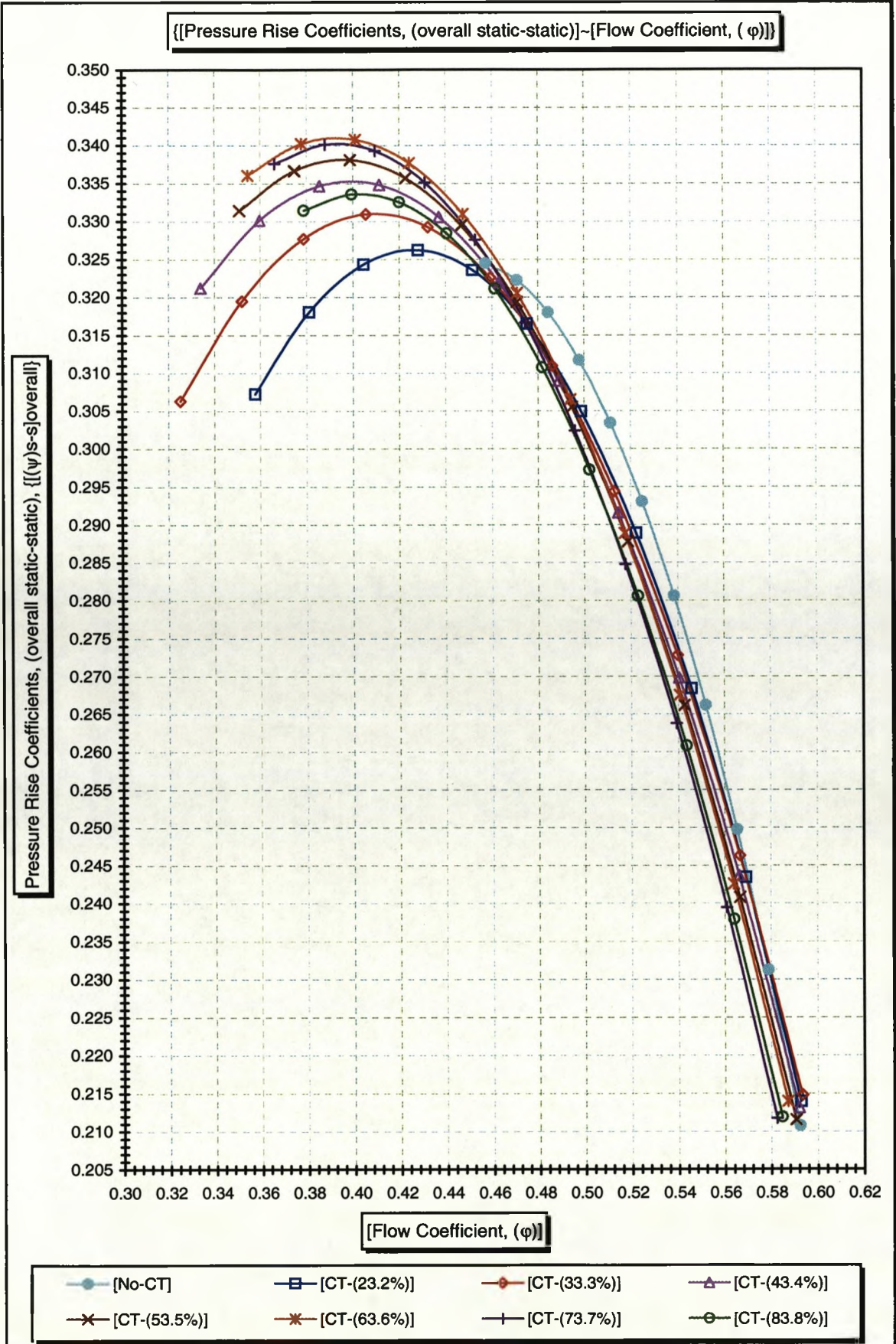




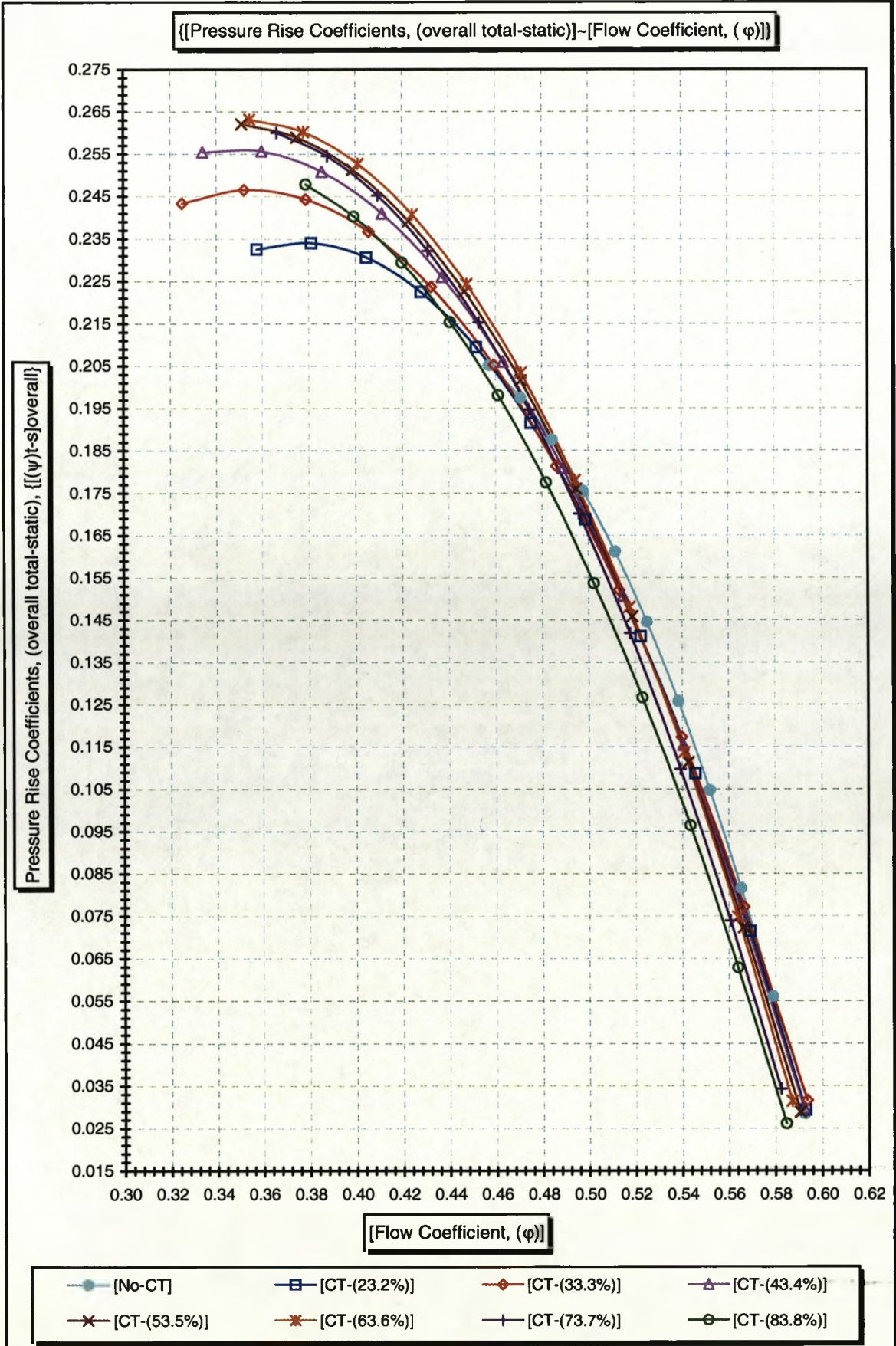
Graph (9-14)]- Characteristics of the corrected mass flow rates at the maximum values of the overalls and stage efficiencies of the treatment builds versus relevant rotor exposures



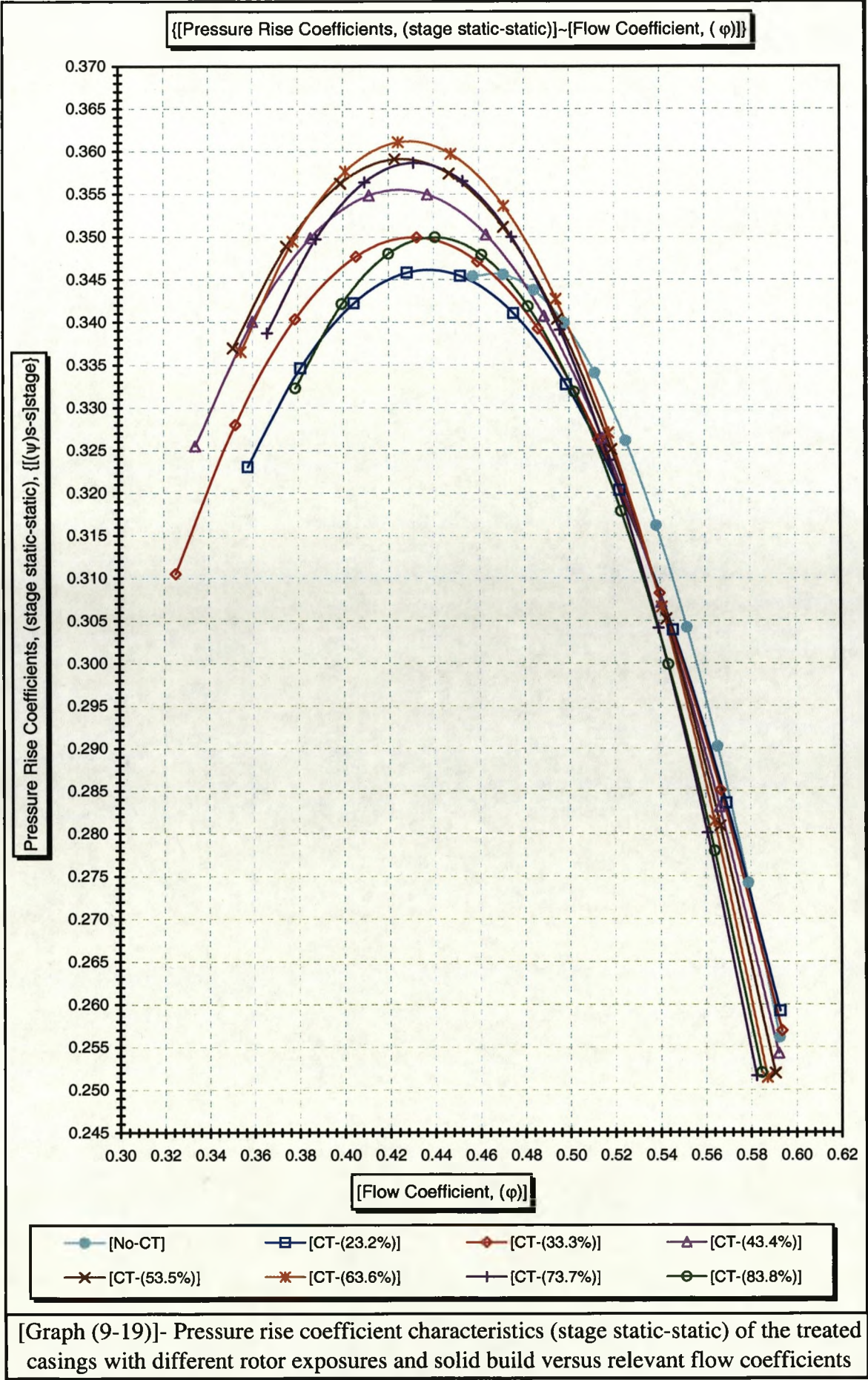


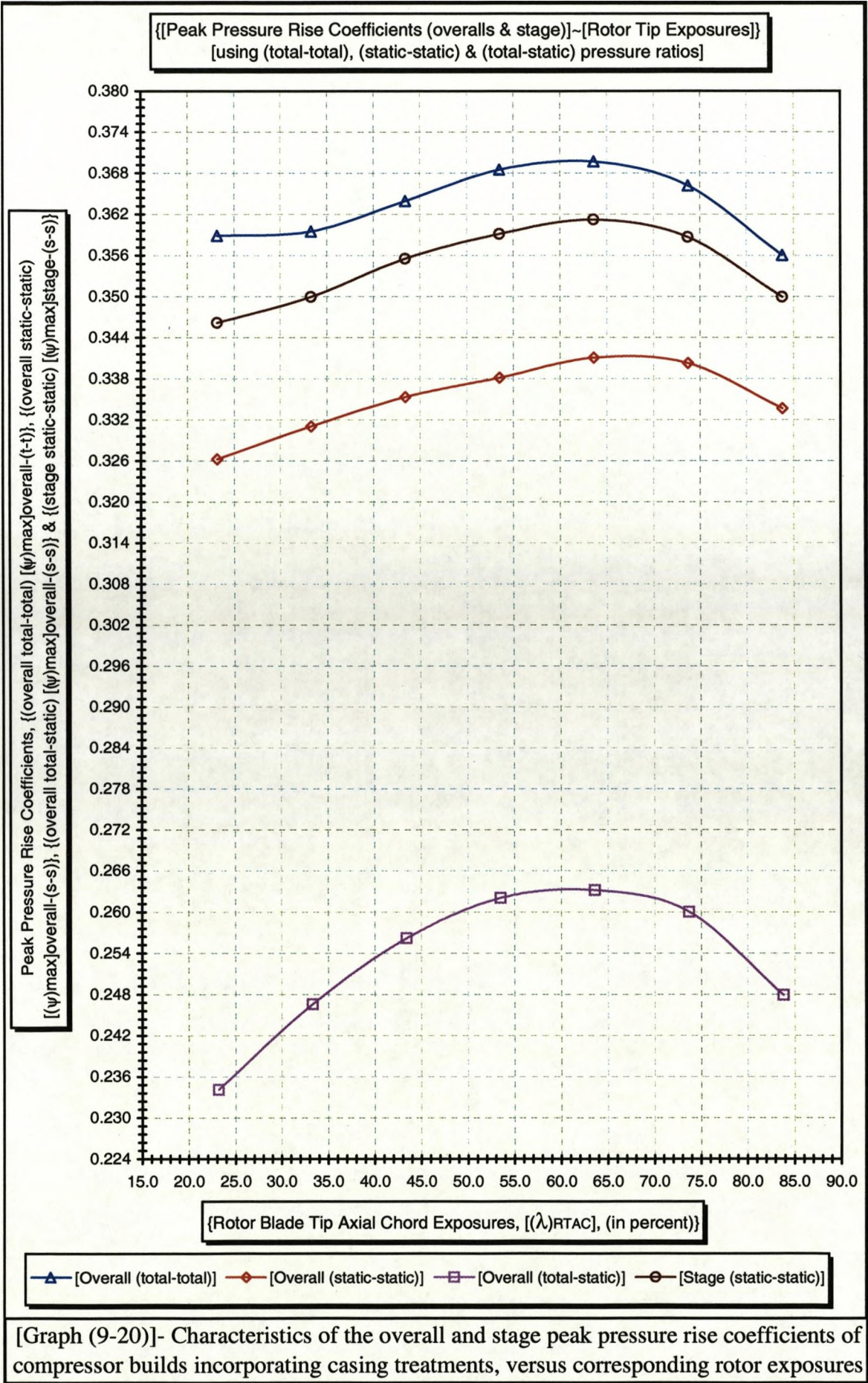


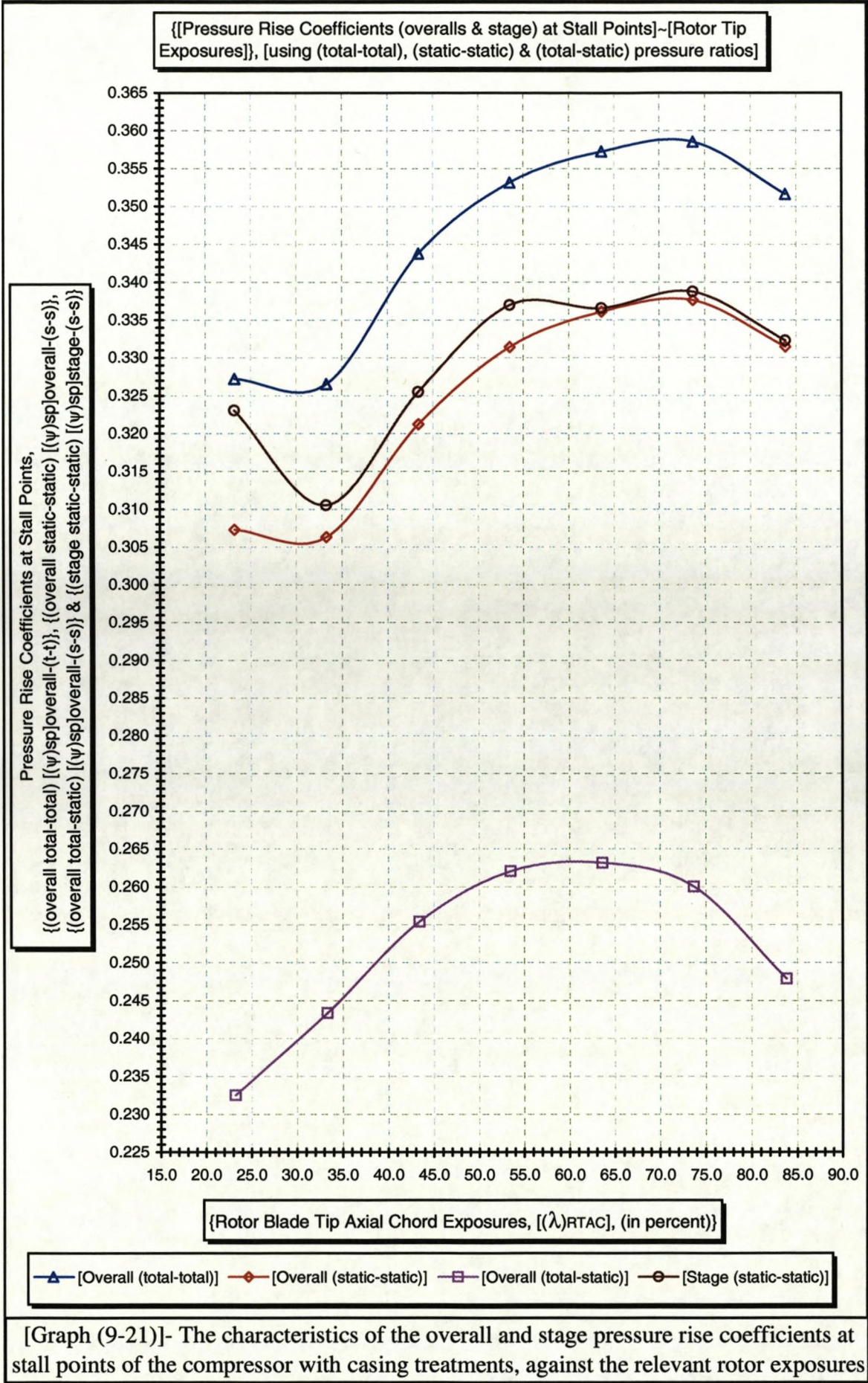
[Graph (9-17)]- Pressure rise coefficient characteristics (overall static-static) of treated casings with different rotor exposures and solid build versus relevant flow coefficients

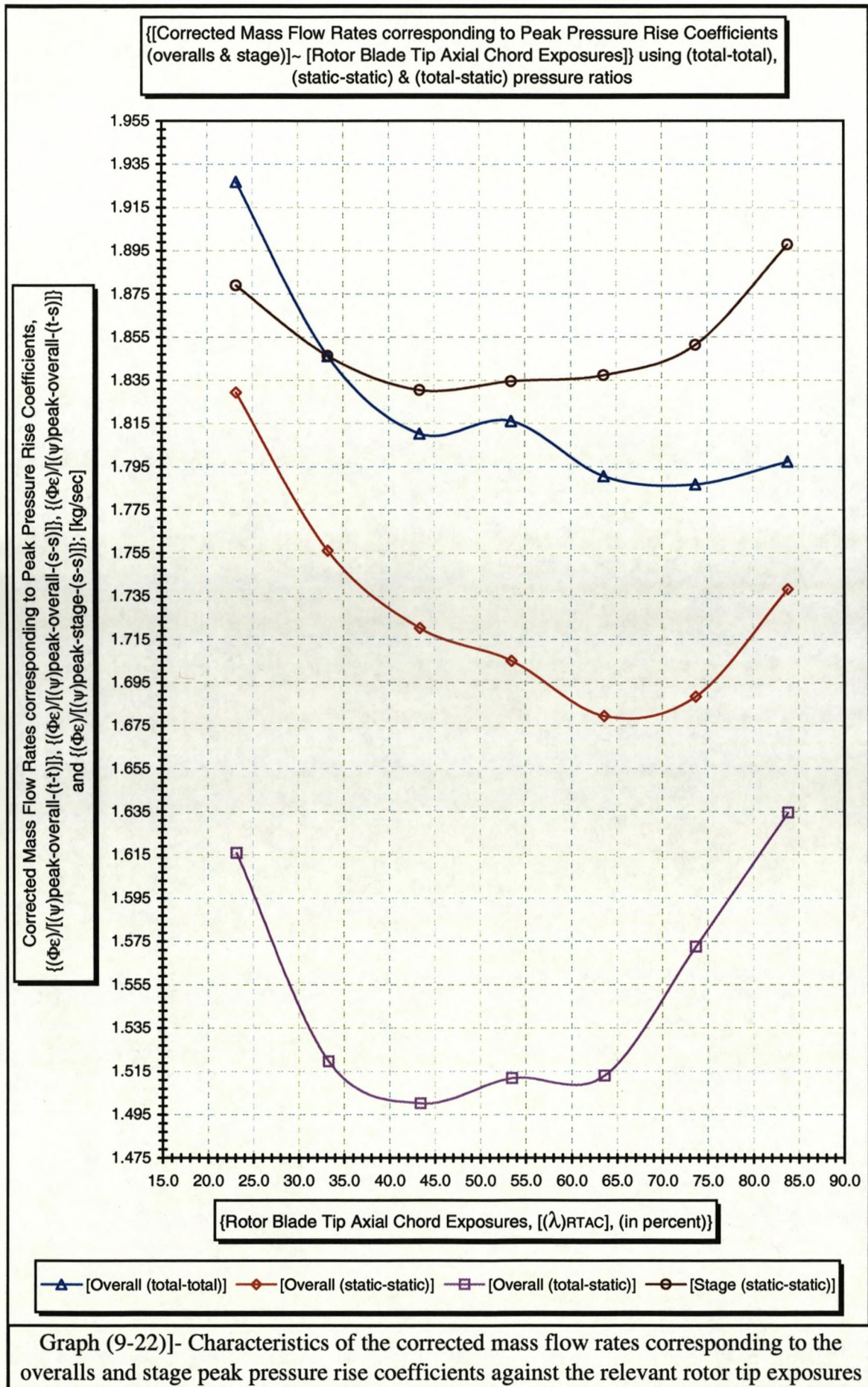


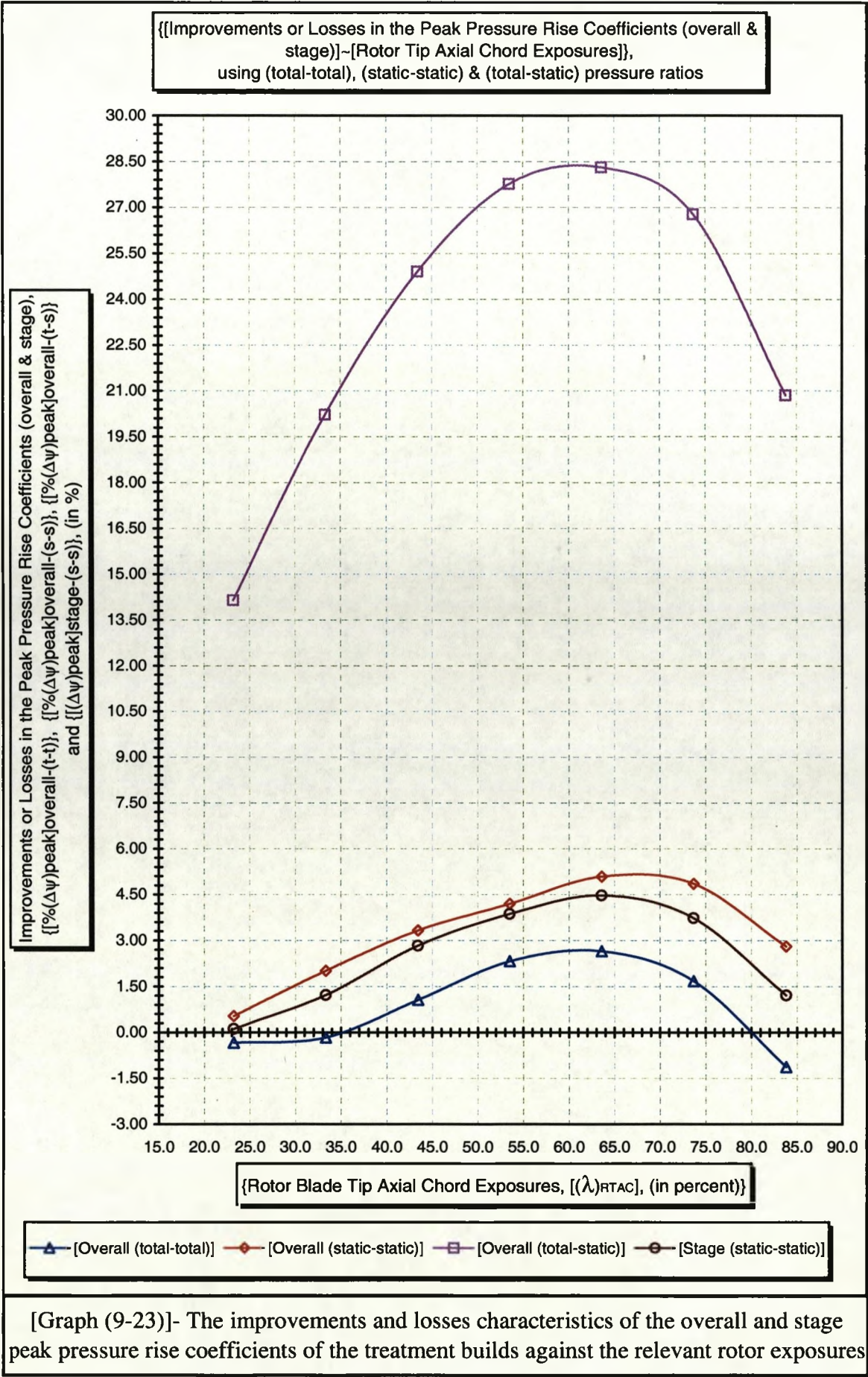
[Graph (9-18)]- Pressure rise coefficient characteristics (overall total-static) of the treated casings with different rotor exposures and solid build versus relevant flow coefficients

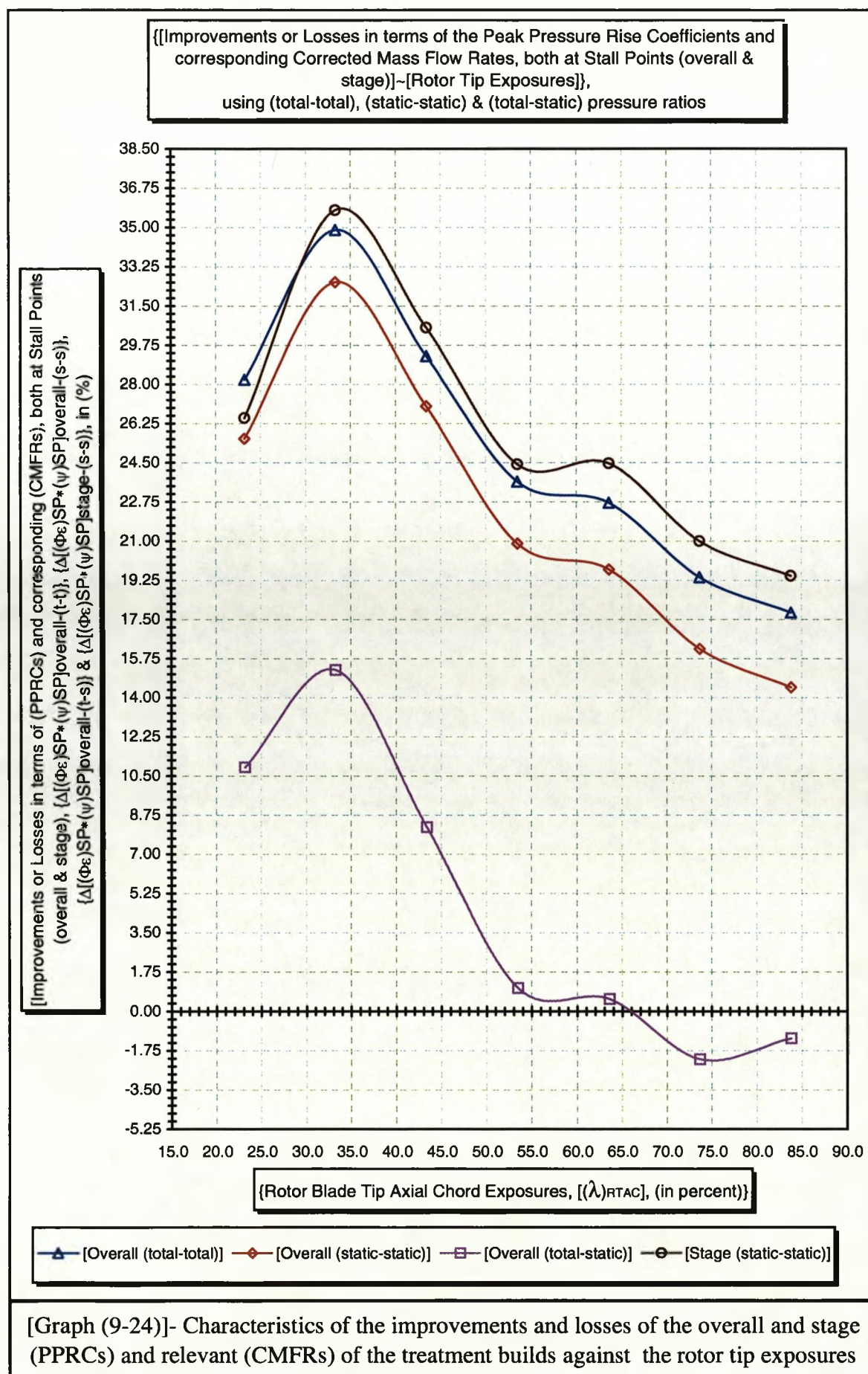












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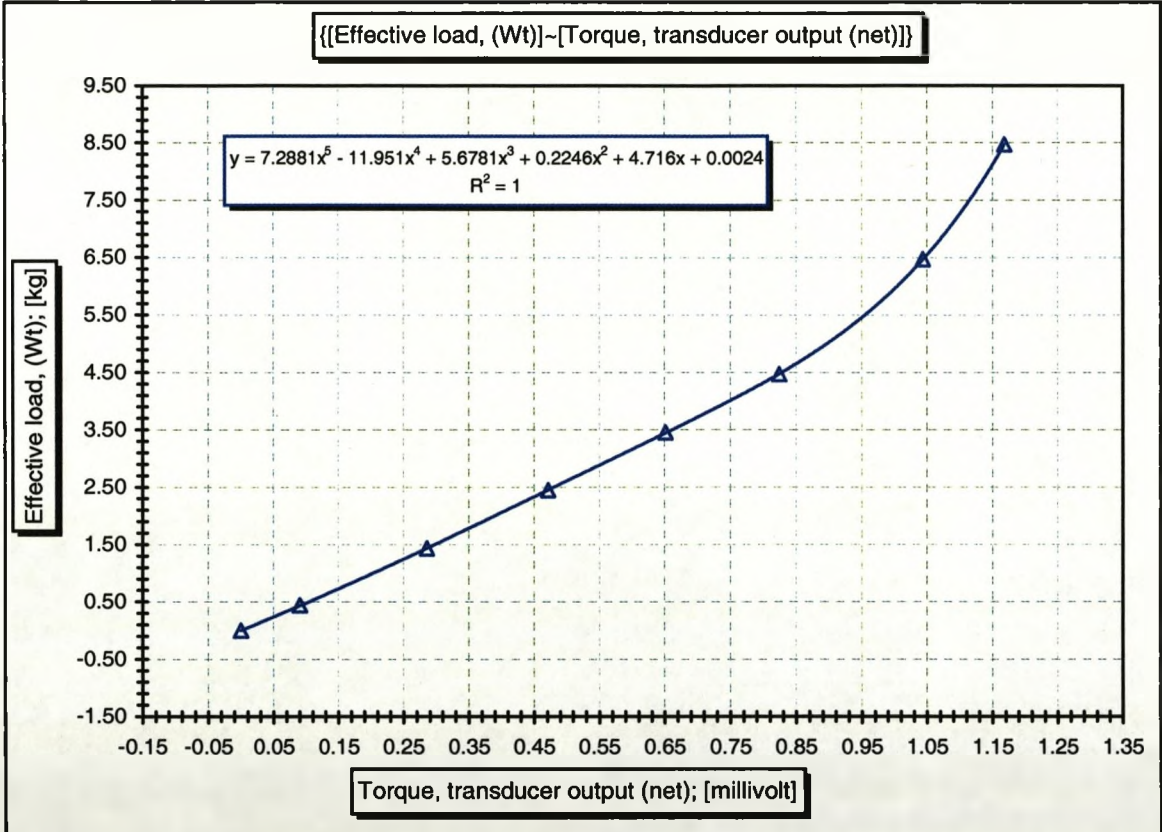
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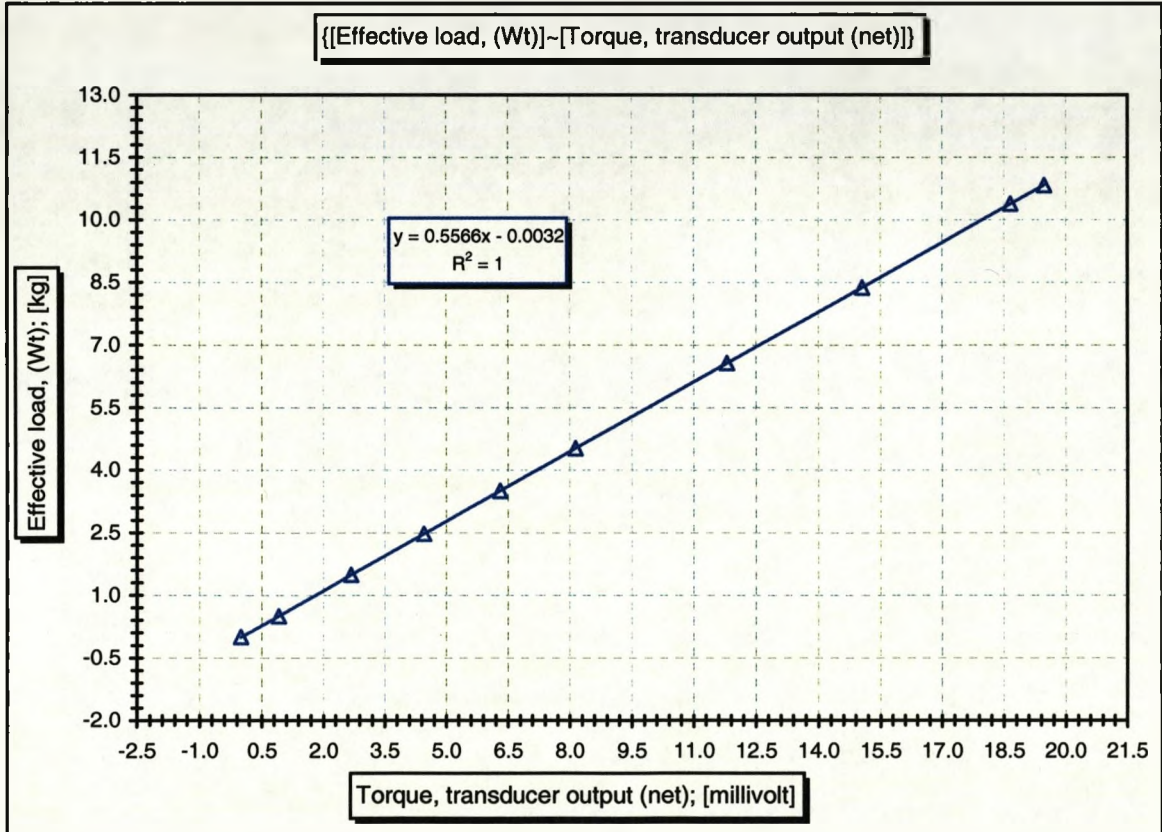
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Situation of the torque calibration factor with average value of 0.18149 [millivolt/kg]							
1)- The transducer was not connected to the motor's stator. 2)- The transducer was located out of the test rig. 3)- The air was not fitted to the rig's air bearing. 4)- The operating domain of the transducer output had not been fixed in appropriate magnitudes.							
Torque, transducer output (raw); [millivolt]							
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	[Average]
0.00	-0.001	0.015	0.015	0.016	0.016	0.015	0.013
0.440	0.101	0.100	0.104	0.103	0.109	0.102	0.103
1.430	0.295	0.301	0.298	0.300	0.299	0.298	0.299
2.450	0.478	0.481	0.483	0.486	0.486	0.491	0.484
3.450	0.654	0.661	0.659	0.671	0.665	0.673	0.664
4.470	0.831	0.832	0.834	0.846	0.831	0.849	0.837
6.470	1.048	1.053	1.054	1.063	1.056	1.062	1.056
8.470	1.174	1.174	1.183	1.183	1.186	1.186	1.181
Torque, transducer output (net); [millivolt]							
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	Net [mean]
0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
0.440	0.102	0.085	0.089	0.087	0.093	0.087	0.091
1.430	0.296	0.286	0.283	0.284	0.283	0.283	0.286
2.450	0.479	0.466	0.468	0.470	0.470	0.476	0.472
3.450	0.655	0.646	0.644	0.655	0.649	0.658	0.651
4.470	0.832	0.817	0.819	0.830	0.815	0.834	0.825
6.470	1.049	1.038	1.039	1.047	1.040	1.047	1.043
8.470	1.175	1.159	1.168	1.167	1.170	1.171	1.168
Torque Calibration Factor (Individual); [millivolt/kg]							
Load [kg]	C.F. [1]	C.F. [2]	C.F. [3]	C.F. [4]	C.F. [5]	C.F. [6]	C.F. [mean]
0.440	0.232	0.193	0.202	0.198	0.211	0.198	0.206
1.430	0.207	0.200	0.198	0.199	0.198	0.198	0.200
2.450	0.196	0.190	0.191	0.192	0.192	0.194	0.192
3.450	0.190	0.187	0.187	0.190	0.188	0.191	0.189
4.470	0.186	0.183	0.183	0.186	0.182	0.187	0.184
6.470	0.162	0.160	0.161	0.162	0.161	0.162	0.161
8.470	0.139	0.137	0.138	0.138	0.138	0.138	0.138
Torque Calibration Factor (Average), [(TCF)-ave]; [millivolt/kg]							0.18149

Situation of the torque calibration factor with average value of 1.79969 [millivolt/kg]							
1)- The transducer was connected to the motor's stator. 2)- The transducer was located on the test rig. 3)- The air was not fitted to the rig's air bearing. 4)- The operating domain of the transducer output had not been fixed in appropriate magnitudes.							
Torque, transducer output (raw); [millivolt]							
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	[Average]
0.00	0.000	0.001	0.001	0.000	0.000	0.001	0.001
0.505	0.915	0.913	0.913	0.912	0.913	0.914	0.913
1.490	2.691	2.690	2.690	2.689	2.690	2.691	2.690
2.475	4.456	4.455	4.454	4.454	4.455	4.454	4.455
3.505	6.304	6.303	6.305	6.303	6.305	6.303	6.304
4.525	8.136	8.135	8.137	8.134	8.136	8.137	8.136
6.565	11.798	11.799	11.800	11.799	11.798	11.799	11.799
8.380	15.057	15.059	15.060	15.059	15.060	15.057	15.059
10.380	18.652	18.654	18.653	18.654	18.653	18.653	18.653
10.830	19.469	19.469	19.468	19.468	19.467	19.467	19.468
Torque, transducer output (net); [millivolt]							
Load [kg]	[1]	[2]	[3]	[4]	[5]	[6]	Net [mean]
0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
0.505	0.915	0.912	0.912	0.912	0.913	0.913	0.913
1.490	2.691	2.689	2.689	2.689	2.690	2.690	2.690
2.475	4.456	4.454	4.453	4.454	4.455	4.453	4.454
3.505	6.304	6.302	6.304	6.303	6.305	6.302	6.303
4.525	8.136	8.134	8.136	8.134	8.136	8.136	8.135
6.565	11.798	11.798	11.799	11.799	11.798	11.798	11.798
8.380	15.057	15.058	15.059	15.059	15.060	15.056	15.058
10.380	18.652	18.653	18.652	18.654	18.653	18.652	18.653
10.830	19.469	19.468	19.467	19.468	19.467	19.466	19.468
Torque Calibration Factor (Individual); [millivolt/kg]							
Load [kg]	C.F. [1]	C.F. [2]	C.F. [3]	C.F. [4]	C.F. [5]	C.F. [6]	C.F. [mean]
0.500	1.812	1.806	1.806	1.806	1.808	1.808	1.808
2.505	1.806	1.805	1.805	1.805	1.805	1.805	1.805
4.505	1.800	1.800	1.799	1.800	1.800	1.799	1.800
6.035	1.799	1.798	1.799	1.798	1.799	1.798	1.798
7.570	1.798	1.798	1.798	1.798	1.798	1.798	1.798
9.065	1.797	1.797	1.797	1.797	1.797	1.797	1.797
10.100	1.797	1.797	1.797	1.797	1.797	1.797	1.797
11.090	1.797	1.797	1.797	1.797	1.797	1.797	1.797
12.030	1.798	1.798	1.798	1.798	1.798	1.797	1.798
Torque Calibration Factor (Average), [(TCF)-ave]; [millivolt/kg]							1.79969



[Graph (A-1)]- The effective load requirement corresponding to the torque calibration factor status with average of (0.18149)



[Graph (A-2)]- The effective load requirement corresponding to the torque calibration factor status with average of (1.79969)

[Comments]	[Data]
Static pressure/(measuring)-(inlet guide vanes-upstream)-Tappings (1&2), [Ps-(IGV-US)/(1-2)]; [cm-H2O]	80.00
Static pressure/(measuring)-(inlet guide vanes-upstream)-Tappings (3&4), [Ps-(IGV-US)/(3-4)]; [cm-H2O]	80.10
Static pressure/(measuring)-[(inlet guide vanes-downstream)=(first stage rotor-upstream)]-Tappings (1&2), [Ps-(IGV-DS)/(1-2)]=[Ps-(1R-US)/(1-2)]; [cm-H2O]	79.80
Static pressure/(measuring)-[(inlet guide vanes-downstream)=(first stage rotor-upstream)]-Tappings (3&4), [Ps-(IGV-DS)/(3-4)]=[Ps-(1R-US)/(3-4)]; [cm-H2O]	80.20
Static pressure/(measuring)-[(first stage rotor-downstream)=(first stage stator-upstream)]-Tappings (1&2), [Ps-(1R-DS)/(1-2)]=[Ps-(1S-US)/(1-2)]; [cm-H2O]	62.40
Static pressure/(measuring)-[(first stage rotor-downstream)=(first stage stator-upstream)]-Tappings (3&4), [Ps-(1R-DS)/(3-4)]=[Ps-(1S-US)/(3-4)]; [cm-H2O]	61.80
Static pressure/(measuring)-[(first stage stator-downstream)=(second stage rotor-upstream)]-Tappings (1&2), [Ps-(1S-DS)/(1-2)]=[Ps-(2R-US)/(1-2)]; [cm-H2O]	58.80
Static pressure/(measuring)-[(first stage stator-downstream)=(second stage rotor-upstream)]-Tappings (3&4), [Ps-(1S-DS)/(3-4)]=[Ps-(2R-US)/(3-4)]; [cm-H2O]	57.40
Static pressure/(measuring)-[(second stage rotor-downstream)=(second stage stator-upstream)]-Tappings (1&2), [Ps-(2R-DS)/(1-2)]=[Ps-(2S-US)/(1-2)]; [cm-H2O]	56.70
Static pressure/(measuring)-[(second stage rotor-downstream)=(second stage stator-upstream)]-Tappings (3&4), [Ps-(2R-DS)/(3-4)]=[Ps-(2S-US)/(3-4)]; [cm-H2O]	56.80
Static pressure/(measuring)-[(second stage stator-downstream)=(third stage rotor-upstream)]-Tappings (1&2), [Ps-(2S-DS)/(1-2)]=[Ps-(3R-US)/(1-2)]; [cm-H2O]	56.70
Static pressure/(measuring)-[(second stage stator-downstream)=(third stage rotor-upstream)]-Tappings (3&4), [Ps-(2S-DS)/(3-4)]=[Ps-(3R-US)/(3-4)]; [cm-H2O]	56.80
Static pressure/(measuring)-[(third stage rotor-downstream)=(third stage stator-upstream)]-Tappings (1&2), [Ps-(3R-DS)/(1-2)]=[Ps-(3S-US)/(1-2)]; [cm-H2O]	56.80
Static pressure/(measuring)-[(third stage rotor-downstream)=(third stage stator-upstream)]-Tappings (3&4), [Ps-(3R-DS)/(3-4)]=[Ps-(3S-US)/(3-4)]; [cm-H2O]	56.70
Static pressure/(measuring)-(third stage stator blades-downstream)-Tappings (1&2), [Ps-(3S-DS)/(1-2)]; [cm-H2O]	56.70
Static pressure/(measuring)-(third stage stator blades-downstream)-Tappings (3&4), [Ps-(3S-DS)/(3-4)]; [cm-H2O]	56.80
Static pressure/(measuring)-(venturi flow meter-upstream), [Ps-(VFM-US)]; [cm-H2O]	57.20
Static pressure/(measuring)-(venturi flow meter-downstream), [Ps-(VFM-DS)]; [cm-H2O]	63.80
Static pressure/(measuring)-(Cobra pressure probe at third stage stator blades-downstream-right hand side hole), [Ps-(3S-DS)/(Ps-R)]; [cm-H2O]	54.70
Static pressure/(measuring)-(Cobra pressure probe at third stage stator blades-downstream-left hand side hole), [Ps-(3S-DS)/(Ps-L)]; [cm-H2O]	54.70
Total pressure (measuring)-(third stage stator blades-downstream)-(Cobra pressure probe-centre hole) at mid blade height, [Pt-(3S-DS)/(Pt-C)]; [cm-H2O]	50.20
Static pressure/(measuring)-manometer datum at mid blade height, [Ps-(datum)]; [cm-H2O]	75.20
Temperature at test area (ambient temperature), (T-amb); [c]	12.20
Temperature at intake of compressor bellmouth (inlet temperature), (T-inlet); [c]	12.50
Temperature at upstream of inlet guide vanes, [T-(IGV-US)]; [c]	12.80
Temperature at outlet of first stage stator blades row, (first stage stator-downstream), [T-(1S-DS)]=(T-outlet); [c]	14.10

Temperature at inlet of venturi flow meter, (venturi flow meter-upstream), [T-inlet-(VFM-US)]; [c]	14.30
Torque (Transducer-running), (Torque); [milivolt]	4.510
Pressure at test area (ambient pressure)=Total pressure at the intake of compressor bellmouth (inlet pressure), [(P-inlet)=(P-amb)]; [millibar]	1017.070
Static pressure (gauge)-(inlet guide vanes-upstream), [Ps-(IGV-US)]; [Pa]	-237.96574
Static pressure (gauge)-[(inlet guide vanes-downstream)=(first stage rotor blades-upstream)], [Ps-(IGV-DS)=Ps-(1R-US)]; [Pa]	-235.51248
Static pressure (gauge)-[(first stage rotor blades-downstream)=(first stage stator blades-upstream)], [Ps-(1R-DS)=Ps-(1S-US)]; [Pa]	642.75281
Static pressure (gauge)-[(first stage stator blades-downstream)=(second stage rotor blades-upstream)], [Ps-(1S-DS)=Ps-(2R-US)]; [Pa]	839.01321
Static pressure (gauge)-[(second stage rotor blades-downstream)=(second stage stator blades-upstream)], [Ps-(2R-DS)=Ps-(2S-US)]; [Pa]	905.251095
Static pressure (gauge)-[(second stage stator blades-downstream)=(third stage rotor blades-upstream)], [Ps-(2S-DS)=Ps-(3R-US)]; [Pa]	905.251095
Static pressure (gauge)-[(third stage rotor blades-downstream)=(third stage stator blades-upstream)], [Ps-(3R-DS)=Ps-(3S-US)]; [Pa]	905.25110
Static pressure (gauge)-(third stage stator blades-downstream), [Ps-(3S-DS)]; [Pa]	905.25110
Static pressure (gauge)-(venturi flow meter-upstream), [Ps-(VFM-US)]; [Pa]	883.17180
Static pressure (gauge)-(venturi flow meter-downstream), [Ps-(VFM-DS)]; [Pa]	559.34214
Static pressure (gauge)-(Cobra pressure probe at third stage stator blades-downstream-right hand side hole), [Ps-(3S-DS)/(Ps-R)]; [Pa]	1005.83455
Static pressure (gauge)-(Cobra pressure probe at third stage stator blades-downstream-left hand side hole), [Ps-(3S-DS)/(Ps-L)]; [Pa]	1005.83455
Total pressure (gauge)-(pressure probe at third stage stator blades-downstream-centre hole) at mid blade height, [Pt-(3S-DS)/(Pt-C)]; [Pa]	1226.6275
Pressure at test area (ambient pressure)=Total pressure at the intake of compressor bellmouth (inlet pressure), [(P-total-inlet)=(P-amb)]; [Pa]	101732.12
Venturi flow meter-downstream pressure (absolute), [P-abs-(VFM-DS)]=[Ps-(VFM-DS)]+(P-amb)]; [Pa]	102291.46
Venturi flow meter-upstream pressure (absolute), [P-abs-(VFM-US)]=[Ps-(VFM-US)]+(P-amb)]; [Pa]	102615.29
Venturi flow meter throat (absolute pressure) to inlet (absolute pressure) pressures ratio, [(PR)-abs-(VFM)]=[P-abs-(VFM-DS)]/[P-abs-(VFM-US)]	0.99684
Specific heats ratio (Kapa=Cp/Cv)	1.40015
Venturi flow meter outlet (throat) cross section diameter, [d-(vfm-throat)]; [m]	0.25400
Venturi flow meter inlet cross section diameter, [d-(vfm-inlet)]; [m]	0.40640
Venturi flow meter outlet (throat) to inlet cross sections diameters ratio, (Rd)=[d-(vfm-throat)]/[d-(vfm-inlet)]	0.62500
Expansion coefficient (Epsilon)	0.99790
Venturi flow meter discharge coefficient, (CD)	0.97800
Venturi flow meter outlet (throat) cross section area, [A-(vfm-throat)]=[PI()*[d-(vfm-throat)^2]/4]; [m^2]	0.05067

Universal gas constant (R-universal); [Joul/kg*K]	287.05200
Temperature (total) at the inlet of venturi flow meter, (venturi flow meter-upstream), [T-inlet-(VFM-US)]; [K]	287.450
Pressure (absolute) drop in venturi flow meter, [Delta.Pabs-(VFM)]=[Pabs-(VFM-US)-[Pabs-(VFM-DS)]]; [Pa]	323.82966
Mass flow rate at venturi flow meter outlet (throat) cross section area, [(M. Akhlaghi) calculation], [M-(VFM-throat)]; [kg/sec]	1.52460
The standard sea level atmospheric pressure [(P-standard)=(P-atm)]; [Pa]	101325.00
Temperature (total) at the intake of compressor bellmouthinlet (inlet temperature), (T-inlet); [K]	285.650
The standard sea level atmospheric temperature [(T-standard)=(T-atm)]; [K]	288.150
Corrected mass flow rate (M-corrected)=[(M)*(P-standard)/[P-total-(inlet)]]*Sqrt[(T-inlet)/(T-standard)]; [kg/sec]	1.51189
Total pressure (absolute)-(pressure probe at third stage stator blades-downstream-centre hole), [P-total-(3S-DS)]=[Pt-(3S-DS)/(Pt-C)]+[P-amb]; [Pa]	102958.75
Mean total pressure (absolute) in compressor annulus, [P-total-(mean)]=[(1/2)*{[P-total-(3S-DS)]+[P-amb]}]; [Pa]	102345.44
Temperature (total) at the upstream of inlet guide vanes, [T-(IGV-US)]; [K]	285.950
Temperature (total) at the outlet of first stage stator blades row, (first stage stator-downstream), [T-(1S-DS)]=T-outlet; [K]	287.250
Mean temperature (total) in compressor annulus, [T-(mean)]=[T-(IGV-US)]+[T-(1S-DS)]/2; [K]	286.600
Mean density at the compressor constant annulus, [(Roh)-mean]=[P-total-(mean)]/[(R-universal)*T-(mean)]; [kg/m^3]	1.24403
Compressor annulus outside diameter, [d-(out-annulus)]; [m]	0.40640
Compressor annulus inside diameter, [d-(in-annulus)]; [m]	0.28440
Area cross section of compressor annulus, [A-(annulus)]=[PI()/4)]*{[d-(out-annulus)^2]-[d-(in-annulus)^2]}; [m^2]	0.06619
Axial flow velocity of air at compressor annulus cross section, [Va-(annulus)]=[M]/{[(Roh)-mean]*[A-(annulus)]}; [m/sec]	18.51491
Mean diameter of the compressor bladings, [D-(mean)]; [m]	0.33530
Running Speed (N); [rpm]	3000.00
Rotating blades circumferential velocity at meam diameter, [U-(mean)]=[PI()*D-mean)*N/60]; [m/sec]	52.66880
Flow coefficient at compressor annulus (mean diameter reference), [(Va/U)mean]=[Va-(annulus)]/[U-(mean)]	0.35153
Corrected rotational velocity (N-corrected)=[(N)*Sqrt[(T-standard)/(T-inlet)]]; [rpm]	3013.09935
Overall (IGV+/-VRCT+1R+1S) temperature rise at first stage including inlet guide vanes row, [(Delta.T)overall]=[T-(1S-DS)]-[T-(IGV-US)]; [K]	1.300
Overall (IGV+/-VRCT+1R+1S) isentropic temperature rise, [(Delta.Ts)overall]=[T-inlet]*{[([P-total-(3S-DS)]/[P-amb])^(kapa-1)/(kapa))]-1}; [K]	0.98011
Sonic velocity of air under Peregrine compressor operating condition, (V-sonic)=[(kapa)*(R-universal)*(T-mean)]^0.5	339.39518
Mach number of air under compressor (IGV+/-VRCT+1R+1S) operating condition, (Mach)=[Va-(annulus)]/(V-sonic)	0.05455

Chord of the Peregrine compressor rotating blade, (C-rotor); [m]	0.03050
Dynamic viscosity of air under Peregrine compressor operating condition, (μ); [(N*sec)/(m ²)]	1.7955E-05
Reynolds number (based on rotor blade cord), (Re)=[((Roh)-mean)*(C-rotor)*(Va-(annulus))/(μ)]	3.913E+04
Effective load corresponding with the motor's stator torque (Wt)=[0.8076*(Torque)-(0.0091)]; [kg]	3.63318
Applied torque calibration factor, [(TCF)applied]=[(Torque)/(Wt)]; [milivolt/kg]	1.24134
Torque arm length, [L-(arm)]; [m]	0.25400
Applied Torque (net), [Taw-(net)]=[(Wt)*(g)*(L-arm)]; [N.m]	9.05572
Anagular velocity of the rotor blades , (Ω)=[(2*PI()*(N)/60)]; [rad/sec]	314.15927
Applied power, [(Power)-applied]=(Ω)*[Taw-(net)]/1000; [kwatt]	2.84494
Corrected applied power, [(Power)-corrected]=((P-standard)/[P-total-(inlet)])*{Sqrt[(T-standard)/(T-inlet)]}*{[(Power)]}; [kwatt]	2.84592
Overall (IGV+/-VRCT+1R+1S) work coefficient [(WC)overall]=(((Ω)*(Taw-net))/((M)*[(U-mean) ²]))	0.67268
Overall (IGV+/-VRCT+1R+1S) total-total pressure rise, (((Δ .P)t-t)overall)=[(P-total-(3S-DS))-(Pt-inlet)]; [Pa]	1226.6275
Overall (IGV+/-VRCT+1R+1S) total-total pressure rise coefficient (using pressure rise), (((PRC)t-t)overall/DP)=(((Δ .P)t-t)overall)/(((Roh)-mean)*[U-(mean) ²])	0.35545
Overall (IGV+/-VRCT+1R+1S) total-total pressure ratio, (((PR)t-t)overall)=[(Ptotal-(3S-DS))/(Pt-inlet)]	1.01206
Heat coefficient at the constant pressure attitude (Cp), [Joul/Kg*K]	1004.4100
Overall (IGV+/-VRCT+1R+1S) total-total efficiency, (((η)t-t)overall)=[(((M)*(Cp)*(T-inlet))/((Ω)*(Taw-net)))*(((PR)t-t)overall) ^{(((kapa-1)/kapa)-1)]}]	0.52756
Overall (IGV+/-VRCT+1R+1S) total-total pressure rise coefficient, (((PRC)t-t)overall)=[(((Cp)*(T-inlet))/[(U-mean) ²]))*(((PR)t-t)overall) ^{(((kapa-1)/kapa)-1)]}]	0.35488
Static pressure (absolute) at upstream of inlet guide vanes, [Ps-absolute-(IGV-US)]=[Ps-(IGV-US)]+[P-amb]; [Pa]	101494.16
Static pressure (absolute)-(third stage stator blades-downstream), [Ps-absolute-(3S-DS)]=[Ps-(3S-DS)]+[P-amb]; [Pa]	102637.37
Overall (IGV+/-VRCT+1R+1S) static-static pressure rise, (((Δ .P)s-s)overall)=[(Ps-absolute-(3S-DS))-[(Ps-absolute-(IGV-US))]; [Pa]	1143.2168
Overall (IGV+/-VRCT+1R+1S) static-static pressure rise coefficient (using pressure rise), (((PRC)s-s)overall/DP)=(((Δ .P)s-s)overall)/(((Roh)-mean)*[U-(mean) ²])	0.33128
Overall (IGV+/-VRCT+1R+1S) static-static pressure ratio, (((PR)s-s)overall)=[(Ps-absolute-(3S-DS))/(Ps-absolute-(IGV-US))]	1.01126
Overall (IGV+/-VRCT+1R+1S) static-static efficiency, (((η)s-s)overall)=[(((M)*(Cp)*(T-(IGV-US)))/((Ω)*(Taw-net)))*(((PR)s-s)overall) ^{(((kapa-1)/kapa)-1)]}]	0.49349
Overall (IGV+/-VRCT+1R+1S) static-static pressure rise coefficient, (((PRC)s-s)overall)=[(((Cp)*(T-(IGV-US)))/[(U-mean) ²]))*(((PR)s-s)overall) ^{(((kapa-1)/kapa)-1)]}]	0.33196
Overall (IGV+/-VRCT+1R+1S) total-static pressure rise, (((Δ .P)t-s)overall)=[(Ps-absolute-(3S-DS))-[(Pt-inlet)]]; [Pa]	905.25110
Overall (IGV+/-VRCT+1R+1S) total-static pressure rise coefficient (using pressure rise), (((PRC)t-s)overall/DP)=(((Δ .P)t-s)overall)/(((Roh)-mean)*[U-(mean) ²])	0.26232
Overall (IGV+/-VRCT+1R+1S) total-static pressure ratio, (((PR)t-s)overall)=[(Ps-absolute-(3S-DS))/(Pt-inlet)]	1.00890

Overall (IGV+/-VRCT+1R+1S) total-static efficiency, $\{[(\text{Eta})_{t-s}]_{\text{overall}}\} = \{[(M) \cdot (C_p) \cdot (T_{\text{-inlet}})] / [(\Omega) \cdot (T_{\text{aw-net}})]\} \cdot \{[(PR)_{t-s}]_{\text{overall}}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.38977
Overall (IGV+/-VRCT+1R+1S) total-static pressure rise coefficient, $\{[(PRC)_{t-s}]_{\text{overall}}\} = \{[(C_p) \cdot (T_{\text{-inlet}})] / [U_{\text{-mean}}^2]\} \cdot \{[(PR)_{t-s}]_{\text{overall}}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.26219
Static pressure (absolute)-(first stage rotor blades-upstream), $[P_{\text{s-absolute}}(1R\text{-US})] = [P_{\text{s}}(1R\text{-US})] + [P_{\text{amb}}]$; [Pa]	101496.61
First stage (1R+1S) static-static pressure rise, $\{[(\Delta P)_{s-s}](1R+1S)\} = [P_{\text{s-absolute}}(3S\text{-DS})] - [P_{\text{s-absolute}}(1R\text{-US})]$; [Pa]	1140.76358
First stage (1R+1S) static-static pressure rise coefficient (using pressure rise), $\{[(PRC)_{s-s}](1R+1S)/DP\} = \{[(\Delta P)_{s-s}](1R+1S)\} / \{[(R_{\text{oh}})_{\text{-mean}}] \cdot [U_{\text{-mean}}^2]\}$	0.33057
First stage (1R+1S) static-static pressure ratio, $\{[(PR)_{s-s}](1R+1S)\} = [P_{\text{s-absolute}}(3S\text{-DS})] / [P_{\text{s-absolute}}(1R\text{-US})]$	1.01124
First stage (1R+1S) static-static efficiency, $\{[(\text{Eta})_{s-s}](1R+1S)\} = \{[(M) \cdot (C_p) \cdot (T_{\text{-IGV-US}})] / [(\Omega) \cdot (T_{\text{aw-net}})]\} \cdot \{[(PR)_{s-s}](1R+1S)\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.49243
First stage (1R+1S) static-static pressure rise coefficient, $\{[(PRC)_{s-s}](1R+1S)\} = \{[(C_p) \cdot (T_{\text{-IGV-US}})] / [U_{\text{-mean}}^2]\} \cdot \{[(PR)_{s-s}](1R+1S)\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.33125
Static pressure (absolute)-(inlet guide vanes-upstream), $[P_{\text{s-absolute}}(\text{IGV-US})] = [P_{\text{s}}(\text{IGV-US})] + [P_{\text{amb}}]$; [Pa]	101494.16
Static pressure (absolute)-(inlet guide vanes-downstream), $[P_{\text{s-absolute}}(\text{IGV-DS})] = [P_{\text{s}}(\text{IGV-DS})] + [P_{\text{amb}}]$; [Pa]	101496.61
Inlet guide vanes row (IGV) static-static pressure rise, $[(\Delta P)_{s-s}]_{\text{IGV}} = [P_{\text{s-absolute}}(\text{IGV-DS})] - [P_{\text{s-absolute}}(\text{IGV-US})]$; [Pa]	2.45326
Inlet guide vanes row (IGV) static-static pressure rise coefficient (using pressure rise), $\{[(PRC)_{s-s}]_{\text{IGV}}/DP\} = \{[(\Delta P)_{s-s}]_{\text{IGV}}\} / \{[(R_{\text{oh}})_{\text{-mean}}] \cdot [U_{\text{-mean}}^2]\}$	0.00071
Inlet guide vanes row (IGV) static-static pressure ratio, $\{[(PR)_{s-s}]_{\text{IGV}}\} = [P_{\text{s-absolute}}(\text{IGV-DS})] / [P_{\text{s-absolute}}(\text{IGV-US})]$	1.00002
Inlet guide vanes row (IGV) static-static efficiency, $\{[(\text{Eta})_{s-s}]_{\text{IGV}}\} = \{[(M) \cdot (C_p) \cdot (T_{\text{-IGV-US}})] / [(\Omega) \cdot (T_{\text{aw-net}})]\} \cdot \{[(PR)_{s-s}]_{\text{IGV}}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.00106
Inlet guide vanes row (IGV) static-static pressure rise coefficient, $\{[(PRC)_{s-s}]_{\text{IGV}}\} = \{[(C_p) \cdot (T_{\text{-IGV-US}})] / [U_{\text{-mean}}^2]\} \cdot \{[(PR)_{s-s}]_{\text{IGV}}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.00072
Static pressure (absolute)-(first stage rotor blades-upstream), $[P_{\text{s-absolute}}(1R\text{-US})] = [P_{\text{s}}(1R\text{-US})] + [P_{\text{amb}}]$; [Pa]	101496.61
Static pressure (absolute)-(first stage rotor blades-downstream), $[P_{\text{s-absolute}}(1R\text{-DS})] = [P_{\text{s}}(1R\text{-DS})] + [P_{\text{amb}}]$; [Pa]	102374.87
First stage rotor blades row (1R) static-static pressure rise, $[(\Delta P)_{s-s}]_{1R} = [P_{\text{s-absolute}}(1R\text{-DS})] - [P_{\text{s-absolute}}(1R\text{-US})]$; [Pa]	878.26529
First stage rotor blades row (1R) static-static pressure rise coefficient (using pressure rise), $\{[(PRC)_{s-s}]_{1R}/DP\} = \{[(\Delta P)_{s-s}]_{1R}\} / \{[(R_{\text{oh}})_{\text{-mean}}] \cdot [U_{\text{-mean}}^2]\}$	0.25450
First stage rotor blades row (1R) static-static pressure ratio, $\{[(PR)_{s-s}]_{1R}\} = [P_{\text{s-absolute}}(1R\text{-DS})] / [P_{\text{s-absolute}}(1R\text{-US})]$	1.00865
First stage rotor blades row (1R) static-static efficiency, $\{[(\text{Eta})_{s-s}]_{1R}\} = \{[(M) \cdot (C_p) \cdot (T_{\text{-IGV-US}})] / [(\Omega) \cdot (T_{\text{aw-net}})]\} \cdot \{[(PR)_{s-s}]_{1R}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.37946
First stage rotor blades row (1R) static-static pressure rise coefficient, $\{[(PRC)_{s-s}]_{1R}\} = \{[(C_p) \cdot (T_{\text{-IGV-US}})] / [U_{\text{-mean}}^2]\} \cdot \{[(PR)_{s-s}]_{1R}\}^{\{[(k_{\text{apa}}-1)/k_{\text{apa}}]-1\}}$	0.25526
Static pressure (absolute)-(first stage stator blades-upstream), $[P_{\text{s-absolute}}(1S\text{-US})] = [P_{\text{s}}(1S\text{-US})] + [P_{\text{amb}}]$; [Pa]	102374.87
Static pressure (absolute)-(first stage stator blades-downstream), $[P_{\text{s-absolute}}(1S\text{-DS})] = [P_{\text{s}}(1S\text{-DS})] + [P_{\text{amb}}]$; [Pa]	102571.13
First stage stator blades row (1S) static-static pressure rise, $[(\Delta P)_{s-s}]_{1S} = [P_{\text{s-absolute}}(1S\text{-DS})] - [P_{\text{s-absolute}}(1S\text{-US})]$; [Pa]	196.26040
First stage stator blades row (1S) static-static pressure rise coefficient (using pressure rise), $\{[(PRC)_{s-s}]_{1S}/DP\} = \{[(\Delta P)_{s-s}]_{1S}\} / \{[(R_{\text{oh}})_{\text{-mean}}] \cdot [U_{\text{-mean}}^2]\}$	0.05687
First stage stator blades row (1S) static-static pressure ratio, $\{[(PR)_{s-s}]_{1S}\} = [P_{\text{s-absolute}}(1S\text{-DS})] / [P_{\text{s-absolute}}(1S\text{-US})]$	1.00192

First stage stator blades row (1S) static-static efficiency, $\{[(\text{Eta})_{s-s}1S]=\{[(M)*(C_p)*[T-(IGV-US)]]/[(\Omega)*(T_{aw-(net)})]\}*\{[(PR)_{s-s}1S^{(kapa-1)/(kapa)}]-1\}$	0.08427
First stage stator blades row (1S) static-static pressure rise coefficient, $\{[(PRC)_{s-s}1S]=\{[(C_p)*[T-(IGV-US)]]/[(U-mean)^2]\}*\{[(PR)_{s-s}1S^{(kapa-1)/(kapa)}]-1\}$	0.05669
Temperature (total) at the test area (ambient temperature), (T-amb); [K]	285.35000
Density of ambient air at test area, $[(Roh)_{-amb}]=\{(P_{-amb})/[(R-universal)*(T_{-amb})]\}$; [kg/m ³]	1.24199
Density of air at Inlet of venturi flow meter, $[(Roh)_{-inlet-(VFM-US)}]=\{[P_{-abs-(VFM-US)}]/[(R-universal)*[T_{-inlet-(VFM-US)}]]\}$; [kg/m ³]	1.24362
Density of air at throat of venturi flow meter, $[(Roh)_{-thr-(VFM-DS)}]=\{[P_{-abs-(VFM-DS)}]/[(R-universal)*[T_{-inlet-(VFM-US)}]]\}$; [kg/m ³]	1.23970
Mass flow rate from [Peregrine compressor characteristics] procedure: Coefficient, (K)	8.85000
Static pressure of air entering the venturi (venturi flow meter-upstream), [Ps-(VFM-US)]; [Inch-H2O]	3.54331
Drop in static pressure between inlet and throat of venturi, $[\Delta.Ps-(VFM)]=\{[Ps-(VFM-US)]-[Ps-(VFM-DS)]\}$; [Inch-H2O]	1.29921
Pressure at the test area (ambient pressure), (P-amb); [Inch-Hg]	30.13106
Temperature (total) at the inlet of venturi flow meter, (venturi flow meter-upstream), [T-inlet-(VFM-US)]; [K]	287.45000
Mass flow rate at venturi flow meter throat cross section [Peregrine compressor manual procedure], (M1); [Lb/sec]	3.28004
Mass flow rate at venturi flow meter throat cross section [Peregrine compressor manual procedure], (M1); [kg/sec]	1.48783
Difference percentage of mass flow rate from (Peregrine compressor manual) procedure based on (M. Akhlaghi) calculation, (Del.M1%)	2.41193
Mass flow rate from the (Ballal-1968) procedure: Coefficient, (K)	8.95000
Venturi flow meter discharge coefficient (CD)	0.97800
Mass flow rate at venturi flow meter throat cross section [Ballal procedure], (M2); [kg/sec]	1.50464
Difference percentage of mass flow rate from (Ballal) procedure based on (M. Akhlaghi) calculation, (Del.M2%)	1.30924
Mass flow rate from the [(Giannassis)-(based on Pergrine notebook)] procedure; (K)	0.04330
Drop in mean static pressure of air through the venturi, $[\Delta.P_v]$; [Cm-H2O]	3.30000
Mean wall static pressure of air entering the venturi (Pv1); [Cm-H2O]	9.00000
Pressure at the test area (ambient pressure)=Total pressure at the intake of compressor bellmouth (inlet pressure), $[(P_{-inlet})=(P_{-amb})]$; [Pa]	101732.12
Temperature of air entering the venturi (Tv1); [K]	287.45000
Mass flow rate at venturi flow meter throat cross section [(Giannassis)-(based on Pergrine notebook) procedure], (M3); [kg/sec]	1.48617
Difference percentage of mass flow rate from (Giannassis) procedure based on (M. Akhlaghi) calculation, (Del.M3%)	2.52071
$[M-(VFM-throat)]=\{(\text{Epsilon})*(CD)*[A-(vfm-throat)]\}*\text{Sqrt}\{[2/(R-universal)]/[1-((Rd)^4)]\}*\text{Sqrt}\{([P_{-abs-(VFM-US)})/[T_{-inlet-(VFM-US)}]]*\Delta.P_{abs-(VFM)}\}$; [kg/sec]	

Rotor blade tip axial chord exposure ratio, $[\lambda_{RTAC}]$	No-CT	CT-0.232	CT-0.333	CT-0.434	CT-0.535	CT-0.636	CT-0.737	CT-0.838
Corrected mass flow rate at stall point, $[(\Phi E)_{sp}]$; [kg/sec]	1.95197	1.53418	1.39437	1.43892	1.51189	1.51298	1.57251	1.63451
Maximum efficiency (overall total-total), $\{[(\eta)_{max}]_{overall-(t-t)}\}$	0.74575	0.73766	0.73142	0.76082	0.76381	0.73541	0.72770	0.71693
Maximum efficiency (overall static-static), $\{[(\eta)_{max}]_{overall-(s-s)}\}$	0.63906	0.62785	0.62639	0.64734	0.64917	0.62836	0.62363	0.62750
Maximum efficiency (overall total-static), $\{[(\eta)_{max}]_{overall-(t-s)}\}$	0.38613	0.40577	0.41162	0.43546	0.44245	0.43344	0.42451	0.40868
Maximum efficiency (stage static-static), $\{[(\eta)_{max}]_{stage-(s-s)}\}$	0.70573	0.69441	0.69252	0.71956	0.72467	0.70192	0.69711	0.70217
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{peak}]_{overall-(t-t)}\}$	0.36013	0.35890	0.35952	0.36396	0.36851	0.36968	0.36618	0.35605
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{peak}]_{overall-(s-s)}\}$	0.32451	0.32624	0.33103	0.33530	0.33813	0.34102	0.34025	0.33362
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{peak}]_{overall-(t-s)}\}$	0.20510	0.23412	0.24658	0.25618	0.26205	0.26317	0.26003	0.24787
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{peak}]_{stage-(s-s)}\}$	0.34575	0.34615	0.34996	0.35553	0.35913	0.36121	0.35866	0.34995
Flow coefficient at stall point, $[(\phi)_{sp}]$	0.45729	0.35763	0.32518	0.33406	0.35096	0.35459	0.36638	0.37888
Pressure rise coefficient (overall total-total) at stall point, $\{[(\psi)_{sp}]_{overall-(t-t)}\}$	0.35826	0.32721	0.32653	0.34378	0.35315	0.35724	0.35853	0.35164
Pressure rise coefficient (overall static-static) at stall point, $\{[(\psi)_{sp}]_{overall-(s-s)}\}$	0.32451	0.30727	0.30633	0.32123	0.33143	0.33605	0.33761	0.33146
Pressure rise coefficient (overall total-static) at stall point, $\{[(\psi)_{sp}]_{overall-(t-s)}\}$	0.20510	0.23254	0.24334	0.25540	0.26205	0.26317	0.26003	0.24787
Pressure rise coefficient (stage static-static) at stall point, $\{[(\psi)_{sp}]_{stage-(s-s)}\}$	0.34539	0.32304	0.31052	0.32548	0.33699	0.33654	0.33874	0.33227

Coefficients of Polynomial Equations of Power (6) or (7)	A1*(X^7)	A*(X^6)	B*(X^5)	C*(X^4)	D*(X^3)	E*(X^2)	F*(X^1)	G*(X^0)
Corrected mass flow rate at stall point, $[(\Phi\epsilon)_{sp}]$; [kg/sec]	0.00E+00	-3.81E-10	1.15E-07	-1.36E-05	7.98E-04	-2.34E-02	3.02E-01	3.87E-01
Maximum efficiency (overall total-total), $\{[(\eta)_{max}]_{overall-(t-t)}\}$	0.00E+00	-9.56E-11	2.66E-08	-2.81E-06	1.36E-04	-2.86E-03	1.31E-02	9.21E-01
Maximum efficiency (overall static-static), $\{[(\eta)_{max}]_{overall-(s-s)}\}$	0.00E+00	-7.12E-11	2.02E-08	-2.20E-06	1.13E-04	-2.67E-03	2.30E-02	6.41E-01
Maximum efficiency (overall total-static), $\{[(\eta)_{max}]_{overall-(t-s)}\}$	0.00E+00	-2.72E-11	6.95E-09	-6.15E-07	1.77E-05	3.25E-04	-2.38E-02	7.00E-01
Maximum efficiency (stage static-static), $\{[(\eta)_{max}]_{stage-(s-s)}\}$	0.00E+00	-8.74E-11	2.50E-08	-2.74E-06	1.42E-04	-3.44E-03	3.14E-02	6.87E-01
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{peak}]_{overall-(t-t)}\}$	0.00E+00	-3.68E-12	1.17E-09	-1.48E-07	8.90E-06	-2.51E-04	2.93E-03	3.50E-01
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{peak}]_{overall-(s-s)}\}$	0.00E+00	8.49E-12	-2.74E-09	3.53E-07	-2.32E-05	8.16E-04	-1.41E-02	4.19E-01
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{peak}]_{overall-(t-s)}\}$	0.00E+00	-2.11E-12	5.30E-10	-5.17E-08	2.32E-06	-5.83E-05	2.18E-03	1.98E-01
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{peak}]_{stage-(s-s)}\}$	0.00E+00	5.51E-12	-1.83E-09	2.46E-07	-1.73E-05	6.71E-04	-1.30E-02	4.43E-01
Flow coefficient at stall point, $[(\phi)_{sp}]$	0.00E+00	-7.12E-11	2.17E-08	-2.59E-06	1.52E-04	-4.41E-03	5.47E-02	1.83E-01
Pressure rise coefficient (overall total-total) at stall point, $\{[(\psi)_{sp}]_{overall-(t-t)}\}$	0.00E+00	-1.79E-12	-5.02E-10	2.35E-07	-2.98E-05	1.70E-03	-4.40E-02	7.41E-01
Pressure rise coefficient (overall static-static) at stall point, $\{[(\psi)_{sp}]_{overall-(s-s)}\}$	0.00E+00	-6.97E-12	1.45E-09	-6.04E-08	-6.96E-06	7.56E-04	-2.44E-02	5.64E-01
Pressure rise coefficient (overall total-static) at stall point, $\{[(\psi)_{sp}]_{overall-(t-s)}\}$	0.00E+00	-2.45E-12	4.16E-10	2.22E-10	-4.27E-06	3.19E-04	-7.75E-03	2.92E-01
Pressure rise coefficient (stage static-static) at stall point, $\{[(\psi)_{sp}]_{stage-(s-s)}\}$	2.69E-12	-1.05E-09	1.69E-07	-1.45E-05	7.16E-04	-2.00E-02	2.92E-01	-1.38E+00

Rotor blade tip axial chord exposures, $[(\lambda)_{RTAC}]$, (in percent)	23.20	33.30	43.40	53.50	63.60	73.70	83.80	$(\lambda)_{RTAC}$	Max/Min
Corrected mass flow rate at stall point, $[(\Phi E)_{sp}]$; [kg/sec]	1.5376	1.3966	1.4394	1.5046	1.5216	1.5660	1.6235	34.7375	1.39470
Maximum efficiency (overall total-total), $\{[(\eta)_{max}]_{overall-(t-t)}\}$	0.7404	0.7331	0.7596	0.7622	0.7384	0.7254	0.7126	49.1598	0.76559
Maximum efficiency (overall static-static), $\{[(\eta)_{max}]_{overall-(s-s)}\}$	0.6295	0.6275	0.6465	0.6481	0.6302	0.6223	0.6249	49.0173	0.65072
Maximum efficiency (overall total-static), $\{[(\eta)_{max}]_{overall-(t-s)}\}$	0.4074	0.4128	0.4344	0.4423	0.4343	0.4238	0.4071	52.5174	0.44237
Maximum efficiency (stage static-static), $\{[(\eta)_{max}]_{stage-(s-s)}\}$	0.6964	0.6938	0.7186	0.7233	0.7041	0.6955	0.6990	49.9073	0.72537
Peak pressure rise coefficient (overall total-total), $\{[(\psi)_{peak}]_{overall-(t-t)}\}$	0.3590	0.3596	0.3640	0.3685	0.3697	0.3661	0.3559	61.5410	0.36984
Peak pressure rise coefficient (overall static-static), $\{[(\psi)_{peak}]_{overall-(s-s)}\}$	0.3264	0.3311	0.3352	0.3383	0.3409	0.3403	0.3337	67.6754	0.34128
Peak pressure rise coefficient (overall total-static), $\{[(\psi)_{peak}]_{overall-(t-s)}\}$	0.2342	0.2466	0.2562	0.2619	0.2634	0.2598	0.2476	61.8715	0.26345
Peak pressure rise coefficient (stage static-static), $\{[(\psi)_{peak}]_{stage-(s-s)}\}$	0.3464	0.3501	0.3553	0.3593	0.3611	0.3587	0.3499	63.9524	0.36112
Flow coefficient at stall point, $[(\phi)_{sp}]$	0.3583	0.3257	0.3340	0.3497	0.3560	0.3653	0.3770	35.1424	0.32503
Pressure rise coefficient (overall total-total) at stall point, $\{[(\psi)_{sp}]_{overall-(t-t)}\}$	0.3286	0.3275	0.3428	0.3535	0.3574	0.3582	0.3509	71.7200	0.35831
Pressure rise coefficient (overall static-static) at stall point, $\{[(\psi)_{sp}]_{overall-(s-s)}\}$	0.3082	0.3070	0.3205	0.3315	0.3363	0.3373	0.3308	72.0687	0.33737
Pressure rise coefficient (overall total-static) at stall point, $\{[(\psi)_{sp}]_{overall-(t-s)}\}$	0.2329	0.2436	0.2552	0.2620	0.2634	0.2598	0.2475	61.2932	0.26351
Pressure rise coefficient (stage static-static) at stall point, $\{[(\psi)_{sp}]_{stage-(s-s)}\}$	0.3226	0.3107	0.3254	0.3364	0.3372	0.3385	0.3322	73.5101	0.33849

$[\%(\lambda)_{\text{ETAC}}]$	$[(\phi)_{\text{sp}}]$	$\{[(\eta)_{\text{max}}]\}$ overall-(t-t)}	$\{[(\eta)_{\text{max}}]\}$ overall-(s-s)}	$\{[(\eta)_{\text{max}}]\}$ overall-(t-s)}	$\{[(\eta)_{\text{max}}]\}$ stage-(s-s)}	$\{[(\psi)_{\text{peak}}]\}$ overall-(t-t)}	$\{[(\psi)_{\text{peak}}]\}$ overall-(s-s)}	$\{[(\psi)_{\text{peak}}]\}$ overall-(t-s)}	$\{[(\psi)_{\text{peak}}]\}$ stage-(s-s)}	$[(\phi)_{\text{sp}}]$	$\{[(\psi)_{\text{sp}}]\}$ overall-(t-t)}	$\{[(\psi)_{\text{sp}}]\}$ overall-(s-s)}	$\{[(\psi)_{\text{sp}}]\}$ overall-(t-s)}	$\{[(\psi)_{\text{sp}}]\}$ stage-(s-s)}
23.20	1.534	0.738	0.628	0.406	0.694	0.359	0.326	0.234	0.346	0.358	0.327	0.307	0.233	0.323
24.21	1.512	0.735	0.627	0.406	0.693	0.359	0.327	0.235	0.346	0.353	0.326	0.306	0.234	0.321
25.22	1.491	0.734	0.626	0.405	0.692	0.359	0.327	0.237	0.347	0.348	0.326	0.306	0.235	0.318
26.23	1.473	0.732	0.625	0.406	0.691	0.359	0.328	0.238	0.347	0.344	0.325	0.305	0.236	0.316
27.24	1.456	0.731	0.625	0.406	0.690	0.359	0.328	0.239	0.347	0.340	0.325	0.305	0.237	0.315
28.25	1.441	0.730	0.624	0.406	0.690	0.359	0.329	0.241	0.348	0.336	0.325	0.305	0.238	0.313
29.26	1.428	0.730	0.624	0.407	0.690	0.359	0.329	0.242	0.348	0.333	0.325	0.305	0.239	0.312
30.27	1.417	0.730	0.624	0.408	0.690	0.359	0.330	0.243	0.349	0.331	0.325	0.305	0.240	0.311
31.28	1.408	0.730	0.625	0.409	0.691	0.359	0.330	0.244	0.349	0.328	0.325	0.305	0.241	0.311
32.29	1.400	0.730	0.626	0.410	0.691	0.359	0.331	0.245	0.349	0.327	0.326	0.306	0.242	0.311
33.30	1.394	0.731	0.626	0.412	0.693	0.360	0.331	0.247	0.350	0.325	0.327	0.306	0.243	0.311
34.31	1.391	0.733	0.628	0.413	0.694	0.360	0.331	0.248	0.350	0.324	0.328	0.307	0.245	0.311
35.32	1.391	0.735	0.629	0.416	0.696	0.360	0.332	0.249	0.351	0.324	0.329	0.308	0.246	0.312
36.33	1.393	0.738	0.632	0.418	0.699	0.361	0.332	0.250	0.352	0.325	0.331	0.310	0.247	0.313
37.34	1.398	0.742	0.634	0.421	0.702	0.361	0.333	0.251	0.352	0.325	0.333	0.311	0.248	0.315
38.35	1.403	0.746	0.637	0.423	0.706	0.361	0.333	0.252	0.353	0.327	0.335	0.313	0.250	0.317
39.36	1.410	0.749	0.639	0.426	0.709	0.362	0.334	0.253	0.353	0.328	0.337	0.315	0.251	0.318
40.37	1.418	0.753	0.642	0.429	0.712	0.363	0.334	0.254	0.354	0.330	0.339	0.317	0.252	0.320
41.38	1.425	0.756	0.644	0.431	0.715	0.363	0.335	0.255	0.355	0.331	0.341	0.318	0.253	0.322
42.39	1.432	0.759	0.646	0.434	0.718	0.363	0.335	0.255	0.355	0.333	0.342	0.320	0.254	0.324
43.40	1.439	0.761	0.647	0.435	0.720	0.364	0.335	0.256	0.356	0.334	0.344	0.321	0.255	0.325
44.41	1.445	0.762	0.648	0.437	0.721	0.364	0.336	0.257	0.356	0.335	0.345	0.322	0.256	0.327
45.42	1.453	0.764	0.649	0.438	0.722	0.365	0.336	0.258	0.356	0.337	0.346	0.324	0.257	0.328
46.43	1.461	0.765	0.650	0.439	0.724	0.365	0.336	0.258	0.357	0.339	0.347	0.325	0.258	0.330
47.44	1.470	0.765	0.651	0.440	0.725	0.366	0.337	0.259	0.357	0.341	0.348	0.326	0.259	0.331
48.45	1.478	0.766	0.651	0.441	0.725	0.366	0.337	0.260	0.358	0.343	0.349	0.327	0.259	0.332
49.46	1.487	0.766	0.651	0.442	0.726	0.367	0.337	0.260	0.358	0.345	0.350	0.328	0.260	0.333
50.47	1.494	0.766	0.651	0.442	0.726	0.367	0.337	0.261	0.358	0.347	0.351	0.329	0.261	0.335
51.48	1.501	0.766	0.651	0.442	0.726	0.368	0.338	0.261	0.359	0.348	0.352	0.330	0.261	0.336
52.49	1.507	0.765	0.650	0.442	0.725	0.368	0.338	0.262	0.359	0.350	0.352	0.331	0.262	0.336
53.50	1.512	0.764	0.649	0.442	0.725	0.369	0.338	0.262	0.359	0.351	0.353	0.331	0.262	0.337
54.51	1.515	0.762	0.648	0.442	0.723	0.369	0.338	0.262	0.359	0.352	0.354	0.332	0.262	0.337
55.52	1.516	0.760	0.646	0.442	0.722	0.369	0.339	0.263	0.360	0.352	0.354	0.333	0.263	0.338
56.53	1.516	0.757	0.644	0.441	0.719	0.369	0.339	0.263	0.360	0.353	0.355	0.333	0.263	0.338
57.54	1.515	0.754	0.642	0.440	0.717	0.369	0.339	0.263	0.360	0.353	0.355	0.334	0.263	0.338
58.55	1.513	0.750	0.639	0.439	0.714	0.370	0.340	0.263	0.361	0.353	0.356	0.334	0.263	0.337
59.56	1.512	0.747	0.637	0.438	0.711	0.370	0.340	0.263	0.361	0.353	0.356	0.335	0.263	0.337
60.57	1.511	0.743	0.634	0.437	0.708	0.370	0.340	0.263	0.361	0.353	0.356	0.335	0.263	0.337
61.58	1.510	0.740	0.632	0.435	0.706	0.370	0.341	0.263	0.361	0.354	0.357	0.335	0.263	0.337
62.59	1.511	0.738	0.630	0.434	0.704	0.370	0.341	0.263	0.361	0.354	0.357	0.336	0.263	0.337
63.60	1.513	0.735	0.628	0.433	0.702	0.370	0.341	0.263	0.361	0.355	0.357	0.336	0.263	0.337
64.61	1.517	0.734	0.627	0.433	0.701	0.370	0.341	0.263	0.361	0.355	0.358	0.336	0.263	0.337
65.62	1.521	0.733	0.626	0.432	0.700	0.369	0.341	0.263	0.361	0.356	0.358	0.337	0.263	0.337
66.63	1.526	0.732	0.625	0.431	0.699	0.369	0.341	0.263	0.361	0.358	0.358	0.337	0.263	0.337
67.64	1.532	0.731	0.625	0.430	0.698	0.369	0.341	0.263	0.361	0.359	0.358	0.337	0.263	0.338
68.65	1.539	0.730	0.624	0.429	0.698	0.369	0.341	0.262	0.361	0.360	0.359	0.338	0.262	0.338
69.66	1.546	0.730	0.624	0.429	0.697	0.368	0.341	0.262	0.360	0.361	0.359	0.338	0.262	0.338
70.67	1.553	0.730	0.624	0.428	0.697	0.368	0.341	0.262	0.360	0.363	0.359	0.338	0.262	0.339
71.68	1.560	0.729	0.624	0.427	0.697	0.367	0.341	0.261	0.360	0.364	0.359	0.338	0.261	0.339
72.69	1.566	0.729	0.624	0.426	0.697	0.367	0.341	0.261	0.359	0.365	0.359	0.338	0.261	0.339
73.70	1.573	0.728	0.624	0.425	0.697	0.366	0.340	0.260	0.359	0.366	0.359	0.338	0.260	0.339
74.71	1.579	0.727	0.624	0.423	0.697	0.365	0.340	0.259	0.358	0.368	0.358	0.337	0.259	0.338
75.72	1.585	0.726	0.624	0.422	0.697	0.365	0.339	0.258	0.357	0.369	0.358	0.337	0.258	0.338
76.73	1.591	0.725	0.624	0.420	0.698	0.364	0.339	0.257	0.357	0.370	0.357	0.337	0.257	0.338
77.74	1.597	0.724	0.624	0.419	0.698	0.363	0.338	0.256	0.356	0.371	0.357	0.336	0.256	0.337
78.75	1.603	0.723	0.624	0.417	0.698	0.362	0.338	0.255	0.355	0.373	0.356	0.335	0.255	0.337
79.76	1.609	0.722	0.625	0.416	0.699	0.361	0.337	0.254	0.354	0.374	0.355	0.335	0.254	0.336
80.77	1.616	0.720	0.625	0.414	0.700	0.360	0.336	0.252	0.353	0.375	0.355	0.334	0.252	0.335
81.78	1.622	0.719	0.626	0.412	0.700	0.359	0.335	0.251	0.352	0.376	0.354	0.333	0.251	0.334
82.79	1.628	0.718	0.627	0.411	0.701	0.357	0.335	0.249	0.351	0.378	0.353	0.332	0.249	0.333
83.80	1.635	0.717	0.627	0.409	0.702	0.356	0.334	0.248	0.350	0.379	0.352	0.331	0.248	0.332

Rotor blade tip axial chord exposure ratio, $[(\lambda)_{RTAC}]$	CT-0.232	CT-0.333	CT-0.434	CT-0.535	CT-0.636	CT-0.737	CT-0.838
Improvement in stall margin in term of corrected mass flow rate, $[\%(\Delta\Phi\epsilon)]$, (in percent)	21.4035	28.5661	26.2838	22.5454	22.4897	19.4399	16.2637
Losses or improvements in maximum efficiency, (overall total-total), $\{ \%[(\Delta\eta)t-t]-(\text{overall-max}) \}$, (in percent)	0.8083	1.4324	-1.5071	-1.8068	1.0335	1.8045	2.8814
Losses or improvements in maximum efficiency, (overall static-static), $\{ \%[(\Delta\eta)s-s]-(\text{overall-max}) \}$, (in percent)	1.1214	1.2678	-0.8278	-1.0101	1.0701	1.5437	1.1569
Losses or improvements in maximum efficiency, (overall total-static), (in percent), $\{ \%[(\Delta\eta)t-s]-(\text{overall-max}) \}$	-1.9640	-2.5487	-4.9332	-5.6318	-4.7307	-3.8382	-2.2553
Losses or improvements in maximum efficiency, (stage static-static), $\{ \%[(\Delta\eta)s-s]-(\text{stage-max}) \}$, (in percent)	1.1317	1.3210	-1.3831	-1.8941	0.3811	0.8621	0.3560
Improvements or losses in peak pressure rise coefficient, (overall total-total), $\{ \%[(\Delta\psi)t-t]-(\text{overall-peak}) \}$, (in percent)	-0.3400	-0.1679	1.0656	2.3273	2.6525	1.6814	-1.1331
Improvements or losses in peak pressure rise coefficient, (overall static-static), $\{ \%[(\Delta\psi)s-s]-(\text{overall-peak}) \}$, (in percent)	0.5340	2.0108	3.3261	4.1986	5.0881	4.8511	2.8068
Improvements or losses in peak pressure rise coefficient, (overall total-static), $\{ \%[(\Delta\psi)t-s]-(\text{overall-peak}) \}$, (in percent)	14.1497	20.2248	24.9033	27.7689	28.3131	26.7827	20.8540
Improvements or losses in peak pressure rise coefficient, (stage static-static), $\{ \%[(\Delta\psi)s-s]-(\text{stage-peak}) \}$, (in percent)	0.1149	1.2150	2.8264	3.8672	4.4691	3.7337	1.2144
Improvement in stall margin in term of flow coefficient, $[\%(\Delta\phi)]$, (in percent)	21.7928	28.8885	26.9469	23.2514	22.4570	19.8791	17.1464
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-total), $\{ \%[\Delta(\Phi\epsilon*\psi)t-t]-(\text{overall-SP}) \}$, (in percent)	28.2165	34.8942	29.2634	23.6512	22.7113	19.3800	17.8114
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall static-static), $\{ \%[\Delta(\Phi\epsilon*\psi)s-s]-(\text{overall-SP}) \}$, (in percent)	25.5776	32.5676	27.0282	20.8938	19.7315	16.1872	14.4694
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-static), $\{ \%[\Delta(\Phi\epsilon*\psi)t-s]-(\text{overall-SP}) \}$, (in percent)	10.8860	15.2457	8.2040	1.0370	0.5442	-2.1363	-1.1987
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (stage static-static), $\{ \%[\Delta(\Phi\epsilon*\psi)s-s]-(\text{stage-SP}) \}$, (in percent)	26.4905	35.7774	30.5325	24.4299	24.4763	20.9920	19.4451

Coefficients of Polynomial Equations of Power (6)	A*(X^6)	B*(X^5)	C*(X^4)	D*(X^3)	E*(X^2)	F*(X^1)	G*(X^0)
Improvement in stall margin in term of corrected mass flow rate, [%($\Delta\Phi\epsilon$)], (in percent)	1.95E-08	-5.89E-06	6.99E-04	-4.09E-02	1.20E+00	-1.55E+01	8.02E+01
Losses or improvements in maximum efficiency, (overall total-total), [%[($\Delta\eta$)t-t]-(overall-max)], (in percent)	9.56E-09	-2.66E-06	2.81E-04	-1.36E-02	2.86E-01	-1.31E+00	-1.75E+01
Losses or improvements in maximum efficiency, (overall static-static), [%[($\Delta\eta$)s-s]-(overall-max)], (in percent)	7.12E-09	-2.02E-06	2.20E-04	-1.13E-02	2.67E-01	-2.30E+00	-1.84E-01
Losses or improvements in maximum efficiency, (overall total-static), (in percent), [%[($\Delta\eta$)t-s]-(overall-max)]	2.72E-09	-6.95E-07	6.15E-05	-1.77E-03	-3.25E-02	2.38E+00	-3.13E+01
Losses or improvements in maximum efficiency, (stage static-static), [%[($\Delta\eta$)s-s]-(stage-max)], (in percent)	8.74E-09	-2.50E-06	2.74E-04	-1.42E-02	3.44E-01	-3.14E+00	1.85E+00
Improvements or losses in peak pressure rise coefficient, (overall total-total), [%[($\Delta\psi$)t-t]-(overall-peak)], (in percent)	-1.02E-09	3.26E-07	-4.11E-05	2.47E-03	-6.97E-02	8.14E-01	-2.69E+00
Improvements or losses in peak pressure rise coefficient, (overall static-static), [%[($\Delta\psi$)s-s]-(overall-peak)], (in percent)	2.62E-09	-8.45E-07	1.09E-04	-7.13E-03	2.51E-01	-4.35E+00	2.92E+01
Improvements or losses in peak pressure rise coefficient, (overall total-static), [%[($\Delta\psi$)t-s]-(overall-peak)], (in percent)	-1.03E-09	2.59E-07	-2.52E-05	1.13E-03	-2.84E-02	1.06E+00	-3.56E+00
Improvements or losses in peak pressure rise coefficient, (stage static-static), [%[($\Delta\psi$)s-s]-(stage-peak)], (in percent)	1.59E-09	-5.30E-07	7.12E-05	-5.01E-03	1.94E-01	-3.75E+00	2.81E+01
Improvement in stall margin in term of flow coefficient, [%($\Delta\phi$)], (in percent)	1.56E-08	-4.74E-06	5.66E-04	-3.31E-02	9.63E-01	-1.20E+01	6.00E+01
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-total), [%[$\Delta(\Phi\epsilon*\psi)$ t-t]-(overall-SP)], (in percent)	1.96E-08	-5.75E-06	6.50E-04	-3.52E-02	8.88E-01	-7.72E+00	1.57E+01
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall static-static), [%[$\Delta(\Phi\epsilon*\psi)$ s-s]-(overall-SP)], (in percent)	2.15E-08	-6.38E-06	7.37E-04	-4.12E-02	1.11E+00	-1.18E+01	4.23E+01
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-static), [%[$\Delta(\Phi\epsilon*\psi)$ t-s]-(overall-SP)], (in percent)	2.57E-08	-7.69E-06	9.02E-04	-5.19E-02	1.49E+00	-1.91E+01	8.60E+01
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (stage static-static), [%[$\Delta(\Phi\epsilon*\psi)$ s-s]-(stage-SP)], (in percent)	2.76E-08	-8.22E-06	9.56E-04	-5.42E-02	1.51E+00	-1.74E+01	6.98E+01

Rotor blade tip axial chord exposures, $[(\lambda)_{RTAC}]$, (in percent)	23.20	33.30	43.40	53.50	63.60	73.70	83.80	$(\lambda)_{RTAC}$	Max/Min
Improvement in stall margin in term of corrected mass flow rate, $[\%(\Delta\Phi\epsilon)]$, (in percent)	21.230	28.453	26.259	22.919	22.046	19.772	16.830	34.7375	28.54932
Losses or improvements in maximum efficiency, (overall total-total), $\{ \%[(\Delta\eta)t-t]-(\text{overall-max}) \}$, (in percent)	0.530	1.260	-1.382	-1.647	0.737	2.038	3.311	49.1598	-1.98459
Losses or improvements in maximum efficiency, (overall static-static), $\{ \%[(\Delta\eta)s-s]-(\text{overall-max}) \}$, (in percent)	0.952	1.159	-0.748	-0.900	0.887	1.679	1.421	49.0173	-1.16585
Losses or improvements in maximum efficiency, (overall total-static), (in percent), $\{ \%[(\Delta\eta)t-s]-(\text{overall-max}) \}$	-2.131	-2.664	-4.825	-5.615	-4.813	-3.765	-2.099	52.5174	-5.62402
Losses or improvements in maximum efficiency, (stage static-static), $\{ \%[(\Delta\eta)s-s]-(\text{stage-max}) \}$, (in percent)	0.932	1.190	-1.290	-1.761	0.165	1.022	0.669	49.9073	-1.96460
Improvements or losses in peak pressure rise coefficient, (overall total-total), $\{ \%[(\Delta\psi)t-t]-(\text{overall-peak}) \}$, (in percent)	-0.320	-0.149	1.063	2.312	2.670	1.659	-1.166	61.5410	2.69825
Improvements or losses in peak pressure rise coefficient, (overall static-static), $\{ \%[(\Delta\psi)s-s]-(\text{overall-peak}) \}$, (in percent)	0.583	2.044	3.280	4.254	5.058	4.857	2.819	67.6754	5.16747
Improvements or losses in peak pressure rise coefficient, (overall total-static), $\{ \%[(\Delta\psi)t-s]-(\text{overall-peak}) \}$, (in percent)	14.187	20.232	24.919	27.712	28.418	26.688	20.727	61.8715	28.45088
Improvements or losses in peak pressure rise coefficient, (stage static-static), $\{ \%[(\Delta\psi)s-s]-(\text{stage-peak}) \}$, (in percent)	0.182	1.266	2.767	3.916	4.443	3.737	1.213	63.9523	4.44423
Improvement in stall margin in term of flow coefficient, $[\%(\Delta\phi)]$, (in percent)	21.643	28.775	26.950	23.523	22.148	20.113	17.560	35.1424	28.92177
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-total), $\{ \%[\Delta(\Phi\epsilon*\psi)t-t]-(\text{overall-SP}) \}$, (in percent)	27.804	34.613	29.413	23.975	22.223	19.777	18.524	32.3092	34.67725
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall static-static), $\{ \%[\Delta(\Phi\epsilon*\psi)s-s]-(\text{overall-SP}) \}$, (in percent)	25.221	32.325	27.120	21.270	19.213	16.600	15.195	32.5730	32.36014
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (overall total-static), $\{ \%[\Delta(\Phi\epsilon*\psi)t-s]-(\text{overall-SP}) \}$, (in percent)	10.590	15.065	8.191	1.552	-0.118	-1.621	-0.343	31.1367	15.38479
Improvement in stall margin in term of corrected mass flow rate and pressure rise coefficient at stall point, (stage static-static), $\{ \%[\Delta(\Phi\epsilon*\psi)s-s]-(\text{stage-SP}) \}$, (in percent)	26.103	35.523	30.590	24.943	23.796	21.523	20.360	33.3612	35.52364

$[\%(\lambda)_{\text{ITAC}}]$	$[\%(\Delta\Phi_e)_{\text{max}}]$	$\{[\%(\Delta\eta)_{\text{t-t}}]$ (overall-max)}\}	$\{[\%(\Delta\eta)_{\text{s-s}}]$ (overall-max)}\}	$\{[\%(\Delta\eta)_{\text{t-s}}]$ (overall-max)}\}	$\{[\%(\Delta\eta)_{\text{s-s}}]$ (stage-max)}\}	$\{[\%(\Delta\psi)_{\text{t-t}}]$ (overall-peak)}\}	$\{[\%(\Delta\psi)_{\text{s-s}}]$ (overall-peak)}\}	$\{[\%(\Delta\psi)_{\text{t-s}}]$ (overall-peak)}\}	$\{[\%(\Delta\psi)_{\text{s-s}}]$ (stage-peak)}\}	$[\%(\Delta\phi)]$	$\{[\%(\Delta(\Phi_e+\psi)_{\text{t-t}})]$ (overall-SP)}\}	$\{[\%(\Delta(\Phi_e+\psi)_{\text{s-s}})]$ (overall-SP)}\}	$\{[\%(\Delta(\Phi_e+\psi)_{\text{t-s}})]$ (overall-SP)}\}	$\{[\%(\Delta(\Phi_e+\psi)_{\text{s-s}})]$ (stage-SP)}\}
23.20	21.40	0.808	1.121	-1.964	1.132	-0.340	0.534	14.15	0.115	21.79	28.22	25.58	10.89	26.49
24.21	22.54	1.031	1.237	-1.941	1.281	-0.371	0.689	14.82	0.202	22.91	29.44	26.84	11.84	28.07
25.22	23.59	1.218	1.330	-1.937	1.401	-0.390	0.842	15.48	0.294	23.93	30.54	27.98	12.67	29.51
26.23	24.54	1.370	1.401	-1.950	1.492	-0.400	0.994	16.12	0.391	24.87	31.51	28.99	13.39	30.80
27.24	25.40	1.486	1.449	-1.982	1.555	-0.399	1.144	16.75	0.494	25.72	32.36	29.88	14.00	31.95
28.25	26.17	1.566	1.475	-2.031	1.588	-0.387	1.293	17.36	0.601	26.47	33.09	30.64	14.49	32.95
29.26	26.83	1.610	1.478	-2.099	1.592	-0.364	1.439	17.96	0.714	27.13	33.70	31.28	14.87	33.81
30.27	27.41	1.619	1.459	-2.184	1.568	-0.331	1.585	18.55	0.831	27.71	34.18	31.79	15.13	34.52
31.28	27.89	1.593	1.418	-2.288	1.515	-0.287	1.728	19.12	0.954	28.19	34.54	32.17	15.29	35.08
32.29	28.27	1.530	1.354	-2.409	1.432	-0.233	1.870	19.68	1.082	28.59	34.78	32.43	15.32	35.50
33.30	28.57	1.432	1.268	-2.549	1.321	-0.168	2.011	20.22	1.215	28.89	34.89	32.57	15.25	35.78
34.31	28.73	1.271	1.140	-2.722	1.158	-0.088	2.151	20.76	1.358	29.07	34.83	32.52	15.00	35.85
35.32	28.74	1.030	0.962	-2.937	0.930	0.010	2.291	21.28	1.514	29.11	34.56	32.27	14.57	35.67
36.33	28.62	0.729	0.744	-3.185	0.653	0.123	2.431	21.79	1.679	29.03	34.11	31.85	13.98	35.30
37.34	28.40	0.387	0.499	-3.454	0.342	0.248	2.570	22.28	1.850	28.85	33.53	31.28	13.26	34.77
38.35	28.11	0.020	0.241	-3.734	0.013	0.381	2.706	22.76	2.024	28.59	32.85	30.62	12.45	34.12
39.36	27.75	-0.350	-0.020	-4.014	-0.321	0.519	2.840	23.23	2.198	28.28	32.11	29.89	11.58	33.39
40.37	27.37	-0.707	-0.269	-4.285	-0.642	0.660	2.969	23.68	2.369	27.94	31.34	29.13	10.68	32.63
41.38	26.98	-1.031	-0.495	-4.535	-0.936	0.800	3.094	24.11	2.532	27.59	30.59	28.37	9.803	31.87
42.39	26.61	-1.304	-0.686	-4.755	-1.189	0.936	3.213	24.52	2.686	27.25	29.88	27.66	8.965	31.16
43.40	26.28	-1.507	-0.828	-4.933	-1.383	1.066	3.326	24.90	2.826	26.95	29.26	27.03	8.204	30.53
44.41	25.95	-1.658	-0.934	-5.079	-1.536	1.195	3.431	25.27	2.956	26.64	28.68	26.42	7.462	29.93
45.42	25.57	-1.786	-1.023	-5.206	-1.670	1.331	3.529	25.63	3.078	26.27	28.07	25.76	6.672	29.27
46.43	25.15	-1.890	-1.094	-5.317	-1.785	1.472	3.620	25.97	3.194	25.88	27.43	25.08	5.853	28.57
47.44	24.72	-1.968	-1.147	-5.411	-1.879	1.613	3.706	26.29	3.305	25.46	26.79	24.38	5.026	27.86
48.45	24.28	-2.018	-1.180	-5.488	-1.950	1.753	3.789	26.59	3.410	25.03	26.16	23.69	4.211	27.15
49.46	23.85	-2.040	-1.192	-5.548	-1.996	1.889	3.870	26.88	3.510	24.60	25.56	23.02	3.429	26.47
50.47	23.44	-2.031	-1.182	-5.593	-2.015	2.017	3.950	27.14	3.605	24.20	24.99	22.39	2.701	25.84
51.48	23.08	-1.990	-1.149	-5.621	-2.005	2.134	4.030	27.37	3.696	23.83	24.47	21.81	2.045	25.27
52.49	22.78	-1.916	-1.092	-5.634	-1.966	2.239	4.113	27.58	3.783	23.51	24.02	21.31	1.484	24.79
53.50	22.55	-1.807	-1.010	-5.632	-1.894	2.327	4.199	27.77	3.867	23.25	23.65	20.89	1.037	24.43
54.51	22.40	-1.641	-0.886	-5.606	-1.771	2.404	4.292	27.93	3.951	23.06	23.38	20.59	0.727	24.20
55.52	22.35	-1.407	-0.713	-5.554	-1.589	2.473	4.393	28.06	4.037	22.94	23.20	20.38	0.546	24.10
56.53	22.35	-1.120	-0.502	-5.479	-1.360	2.535	4.500	28.17	4.122	22.86	23.10	20.25	0.468	24.10
57.54	22.40	-0.797	-0.264	-5.386	-1.098	2.587	4.607	28.25	4.204	22.81	23.05	20.19	0.464	24.18
58.55	22.47	-0.454	-0.011	-5.281	-0.818	2.629	4.713	28.32	4.279	22.78	23.04	20.15	0.510	24.29
59.56	22.55	-0.104	0.245	-5.167	-0.533	2.661	4.813	28.36	4.346	22.76	23.04	20.13	0.578	24.42
60.57	22.61	0.235	0.493	-5.051	-0.257	2.680	4.904	28.38	4.401	22.74	23.02	20.10	0.643	24.53
61.58	22.63	0.548	0.721	-4.936	-0.004	2.685	4.982	28.37	4.441	22.68	22.98	20.04	0.676	24.59
62.59	22.60	0.819	0.917	-4.828	0.214	2.677	5.045	28.35	4.465	22.60	22.88	19.92	0.652	24.59
63.60	22.49	1.034	1.070	-4.731	0.381	2.652	5.088	28.31	4.469	22.46	22.71	19.73	0.544	24.48
64.61	22.31	1.193	1.186	-4.644	0.506	2.616	5.118	28.26	4.458	22.27	22.47	19.47	0.348	24.26
65.62	22.07	1.315	1.281	-4.563	0.608	2.571	5.142	28.21	4.436	22.06	22.17	19.15	0.090	23.97
66.63	21.80	1.407	1.357	-4.484	0.688	2.515	5.157	28.14	4.403	21.82	21.83	18.79	-0.213	23.63
67.64	21.49	1.476	1.417	-4.406	0.750	2.446	5.161	28.06	4.357	21.56	21.47	18.40	-0.544	23.24
68.65	21.16	1.529	1.461	-4.327	0.795	2.363	5.153	27.95	4.297	21.29	21.08	17.99	-0.885	22.83
69.66	20.81	1.573	1.493	-4.244	0.827	2.265	5.130	27.81	4.221	21.01	20.70	17.59	-1.220	22.42
70.67	20.46	1.616	1.515	-4.156	0.847	2.149	5.091	27.63	4.128	20.72	20.33	17.19	-1.529	22.01
71.68	20.11	1.664	1.530	-4.061	0.858	2.014	5.032	27.40	4.016	20.43	19.97	16.82	-1.797	21.62
72.69	19.76	1.724	1.538	-3.955	0.862	1.859	4.953	27.12	3.886	20.15	19.65	16.48	-2.005	21.28
73.70	19.44	1.805	1.544	-3.838	0.862	1.681	4.851	26.78	3.734	19.88	19.38	16.19	-2.136	20.99
74.71	19.13	1.898	1.544	-3.711	0.856	1.483	4.728	26.39	3.562	19.61	19.14	15.93	-2.205	20.75
75.72	18.81	1.995	1.535	-3.577	0.840	1.266	4.587	25.95	3.373	19.34	18.93	15.70	-2.238	20.53
76.73	18.50	2.095	1.518	-3.436	0.814	1.031	4.428	25.47	3.165	19.08	18.72	15.48	-2.235	20.32
77.74	18.18	2.199	1.492	-3.288	0.778	0.777	4.250	24.94	2.940	18.80	18.54	15.28	-2.195	20.14
78.75	17.87	2.305	1.458	-3.133	0.732	0.505	4.055	24.37	2.697	18.53	18.38	15.10	-2.120	19.98
79.76	17.55	2.414	1.415	-2.971	0.677	0.214	3.841	23.75	2.436	18.26	18.23	14.94	-2.008	19.83
80.77	17.23	2.526	1.363	-2.803	0.611	-0.095	3.610	23.09	2.158	17.98	18.10	14.79	-1.860	19.71
81.78	16.91	2.642	1.303	-2.627	0.536	-0.423	3.360	22.39	1.861	17.71	17.98	14.67	-1.676	19.60
82.79	16.59	2.760	1.234	-2.445	0.451	-0.769	3.093	21.64	1.547	17.43	17.89	14.56	-1.455	19.51
83.80	16.26	2.881	1.157	-2.255	0.356	-1.133	2.807	20.85	1.214	17.15	17.81	14.47	-1.199	19.45