



**Cascade Testing and CFD Applied to Gas
Turbine Performance Improvement with
Compressor Cleaning**

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**Cascade Testing and CFD Applied to Gas Turbine Performance
Improvement with Compressor Cleaning**

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Abstract

With the growing interest in life cycle costs for heavy-duty gas turbines, equipment operators are investigating the trade off between performance improvements and associated maintenance costs. One of the key factors leading to performance degradation during plant operation is compressor fouling especially in harsh environments. These results from the adherence of dust or sand particles mixed with small oil droplets to compressor blade surfaces. The result is a reduction in compressor pressure ratio and an overall loss in mass flow, compressor efficiency and, therefore, overall power output.

To restore this power loss, an increased fuel flow rate with the consequent higher firing temperature (TET) is necessary. This will seriously reduce the creep life of the HP turbine blades.

The research described here is divided into three sections:

The first section focuses on the simulation and diagnostic analyses of a typical clean and fouled industrial gas turbine. The analysis tool is a performance code (Turbomatch) developed at Cranfield University.

The performance degradation was based on the following engine which is currently operating in the Libyan Desert, using field data available from the company. A single shaft industrial gas turbine (11 MW) engine

The study comprised a simulation of the “clean” engine performance of a typical small gas turbine used for power generation. The study also examined the economic impact of compressor fouling on operating costs due to increased fuel flow rate and the reduced creep life of the HP turbine. Additionally, the study included an economical analysis of various types of compressor online washing techniques.

The real engine data of a Sulzer Type 7 gas turbine (11MW) have been collected from a Libyan oil field over a period of four months without compressor washing. Corrected to standard ambient conditions the results are then compared with the clean engine

performance. A case study shows that as a result of compressor fouling, the compressor efficiency decreases about 2.5%, and Heat Rate increases by 7% and power output falls by 10%. Consequently, the cost of power losses in a one year period is around \$600,000.

The second section involves high fidelity CFD (two dimensional and three dimensional) simulations of the fouling mechanism of the ninth stage of an axial compressor.

This HP9 stage was tested in No.3 Compressor Test Facility at Rolls-Royce. The comparison between the experimental data and the ANSYS CFX is shown in chapter 4.

This research contains a description of the numerical analyses carried out on the HP9 test case to include the effects of variable amounts of roughness both in terms of spatial distribution and equivalent grain size. The examination of the effects of adding roughness to selected regions of the rotor blade suction side has shown the relative importance of the leading edge end on the blade suction side over other regions of the blade. The relative impact was quantified in terms of the wake velocity defect, magnitude and wake width. However when the radial distribution is considered the differences between clean and roughened blades is less apparent excepting for the case where the entire suction side of the blade is roughened.

The pressure distribution around the blades was also examined. This part of the work has shown the benefits of flow analysis using CFD in a very small region of the flow field. In this case close the analysis is confined to the leading edge and trailing edge of a compressor blade. It also demonstrates clearly the disadvantages of fouling in reducing pressure rise through increases of total pressure loss and therefore reduced efficiency.

It has been established that the leading edge and the concave surfaces of the rotor blade are the most sensitive regions to the fouling.

It is emphasised that experimental verification of this trend would be very difficult to achieve with a cascade rig because both rotor and stator would need static pressure

tapings very close to one another at the leading edge of the blade. This would be extremely difficult to achieve in practice.

In the third section experimental work has been undertaken on a cascade facility in the Gas Turbine Laboratory. This details measurements of the cascade the pressure losses as a result of increasing levels of fouling on the blades. In addition, further experiments are undertaken to recover performance loss due to fouling through a washing technique.

This has facilitated a rigorous estimation of the magnitude of compressor stage inefficiency caused by fouling. By observing the actual distribution of fouling in a typical compressor, it is now possible to estimate more accurately the overall impact of the fouled performance on overall compressor efficiency.

Furthermore an experimental study has been carried out based on a need in practice for online compressor washing; an investigation was undertaken by using a different spray nozzle pressure. The detergent used for the washing experiments was a water based cleaner diluted with demineralised water in a ratio of 1:4. The fouled compressor efficiency after cleaning with the current project washing scheme has been improved by 2.2 %.

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Alhamdulillah (Thanks God)

*GOD grant me the serenity
To accept the things
I cannot change
Courage to change
the thing I can
And the wisdom
to know the difference*

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To my beloved sons, Suhaib and Hammam

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Nomenclature

Latin symbols

b	Chord, m
b/t	solidity
C_{μ}	Constant (=0.09)
c	Blade chord
C_0	Stream velocity at the distance upstream, m/s
d_p	Particle diameter, m
E	Empirical constant (=9.793)
E	Entrainment coefficient
f_r	Roughness function
h	Distance between the stream centreline and limit trajectory at the upstream infinity, m
K	Roughness height
k	Turbulent kinetic energy
L	Characteristic size of the body, m
M	Cascade inlet Mach number
i	incidence angle, deg
P	Total pressure
p	Static pressure
Re	Reynolds number based on inlet velocity and chord length
St_k	Stokes number
t	Blade pitch
u_p	Mean fluid velocity at point P
u_{τ}	Friction velocity
y_p	Distance from point P to the wall
y^+	Y plus

Greek symbols

α	Air flow angle
ζ	Stagger angle
θ	Camber angle
κ	Von Karman constant (=0.4187)
μ	Micro
μ	Dynamic viscosity
ρ	Density

ρ_p	Particle density, kg/m ³
τ_w	Wall shear stress
ω	Total pressure loss coefficient
w_1	flow velocity in the inlet of the blade, m/s
W_2	flow velocity in the outlet of the blade, m/s
β_b	stagger angle with the axis u, deg
β_1	flow angle with the axis w, deg
β_1'	blade inlet angle, deg
β_2'	blade outlet angle, deg
β_2	flow outlet angle, deg
χ_1	angle between the chord and the tangent to the profile centreline in the inlet of the blade, deg
χ_2	angle between the chord and the tangent to the profile centreline in the outlet of the blade, deg
δ	deviation, deg

Subscripts

1	Upstream of cascade
2	Downstream of cascade
amb	Ambient
mic	Micron

Abbreviations

CFD	Computational Fluid Dynamics
SST	Shear stress transport
TET	Turbine Entry Temperature
PMS	Preventative maintenance schedules
NOC	National Oil Corporation
CBM	Condition Based Maintenance
Eq	Equation

Chapter 1 – Introduction

1. Introduction

This thesis discusses the effect of fouling and fouling mechanisms on gas turbine performance and, in particular, on the axial compressor. An analysis of the impact of environmental conditions, and consequent fouling of a stationary gas turbine have been investigated to determine the optimum online compressor wash frequency. Furthermore, analysis of the impact of blade roughness on the compressor efficiency and the pressure ratio as a function of the location of the fouling has been completed.

Typical large gas turbines for power generation can have an inlet mass flow of over 600kg/s of air and a power output of nearly 300MW. Under on-site conditions this could mean up to 800m³/s of air entering the gas turbine, carrying with it fine sand and dust particles, insects, oil, soot and unburned hydrocarbons, even sea water and salt. Some of the larger and stickier particles adhere to the compressor blade surfaces which changes the shape of the blade and its angle of attack. Such an adverse change in the aerodynamics of the system will result in decrease of the mass flow rate and the isentropic compressor efficiency. In turn this will lead to reduced power output and thermal efficiency and an increase in heat rate (turbine temperature) which will reduce the creep life of the turbine blades.

Usually in an attempt to recover these losses of power and reduce the threat to creep life, a compressor cleaning system is used. Accordingly a series of experiments on a compressor blade cascade using online washing technique was undertaken to investigate improvements in compressor efficiency, thermal efficiency and power output. The results are shown in Chapter 7.

1.1 Background

The gas turbine is based on the Brayton thermodynamic cycle and, like nearly all engines, converts the heat of combustion to mechanical work. This transformation of energy uses air as the working fluid and comprises three basic stages:

1. Compression; air is sucked into the compressor and as it is forced through the compression stage its pressure rises significantly, possibly by a factor of 30.
2. Combustion; the high pressure air passes into the combustion stage where injectors inject a steady flow of fuel. After mixing with fuel the combustible mixture moves to the combustion chamber where combustion takes place. The energy from combustion increases both the enthalpy and temperature of the air.
3. Expansion; the final turbine section transforms the enthalpy of hot products of combustion into mechanical work by moving the rotor blades and thus driving the shaft. The rotating shaft allows work to be extracted from the engine and is also used to drive the compressor and necessary external equipment.

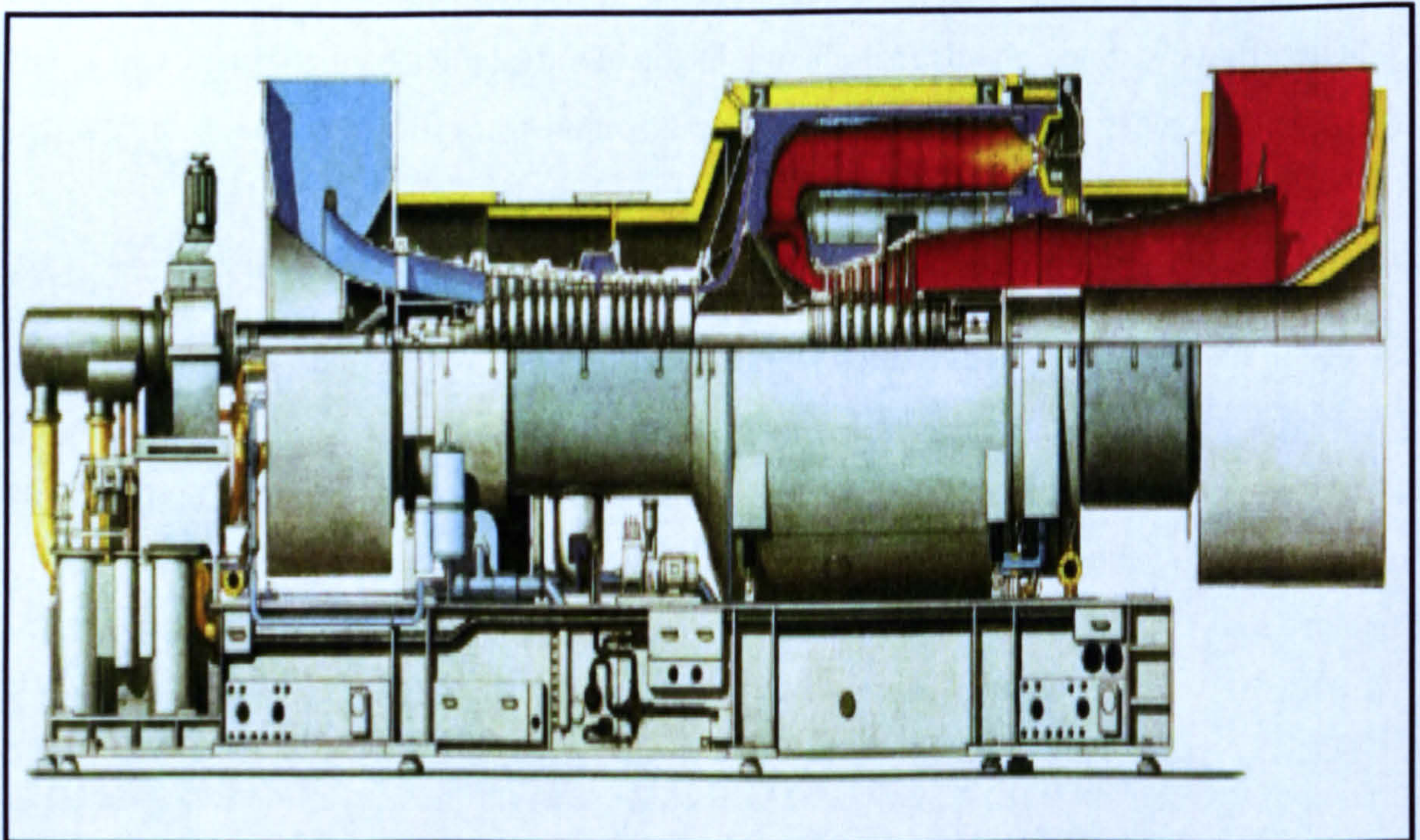


Figure 1-1: Cross-section of single shaft industrial gas turbine Sulzer Type 7 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The gas turbine has a host of applications: but here we are concerned only with the shaft being used to drive electrical generators, or mechanical equipment such as propellers, pumps, etc.

This thesis reports the investigation, using experimental and computer simulation techniques, of how compressor washing can help remove fouling mechanisms and recover lost power for a single shaft Sulzer Type 7 industrial gas turbine used for electricity generation sited in a power plant. A cross-section of Sulzer Type 7 engine is shown in Figure 1-1.

The massive ingestion of air by a gas turbine means that even if the filters reduced the contamination rate to 3ppm, the turbine could take in – on a daily basis - as much as 4.5kg of foreign material, and this shows why airborne particles such as sand are so important in performance deterioration.

The axial compressors have many advantages but tend to be more complex than other designs with the result that they are very sensitive to the small changes in airfoil shape that can result from fouling. Performance degradation due to fouling generally occurs because of build-up on blade surfaces changing their aerodynamic shape and roughening the surfaces, and erosion and abrasion of the blade giving an increased tip clearance. However, rain water will be ingested, and in coastal regions saltwater spray. Diakunchak (1991) has estimated compressor fouling is responsible for about 75% of all gas turbine performance loss.

1.2 Motivation

Presently the Libyan Oil and Gas and Power Generation industries are undergoing a re-evaluation of their approach to machinery maintenance according to the National Oil Corporation (NOC). Preventative maintenance schedules (PMS) are now replacing prescribed maintenance based exclusively on hours of operation or a scheduled time interval since the last service. PMS guidelines were developed on the basis of field experience to provide adequate maintenance schedules to maintain machinery in peak operational condition.

Condition Based Maintenance (CBM) is a procedure that claims to reduce overall maintenance costs by directing maintenance resources towards that plant and machinery that need it to ensure continued production and safety of operatives. CMB eliminates the necessity of much non-essential maintenance that previously would have been performed as part of a routine maintenance schedule. CBM is particularly suitable for use with gas turbines to monitor fouling build up on rotating and stationary aerofoils and determine when compressor washing should take place.

Fouling is one the most important of a number of contributors to the degradation of gas turbine performance degradation (Boyce M. P., 1994). This is particularly true of a desert environment where sand “gets everywhere” and mixes with oil vapour before entering the turbine. Figure 1-2 shows turbine compressor blade fouling that occurred in just such an environment. Fortunately, the deterioration of gas turbine performance deterioration due to fouling can be partially retrieved through washing and thus auxiliary systems have been developed specifically for this purpose.

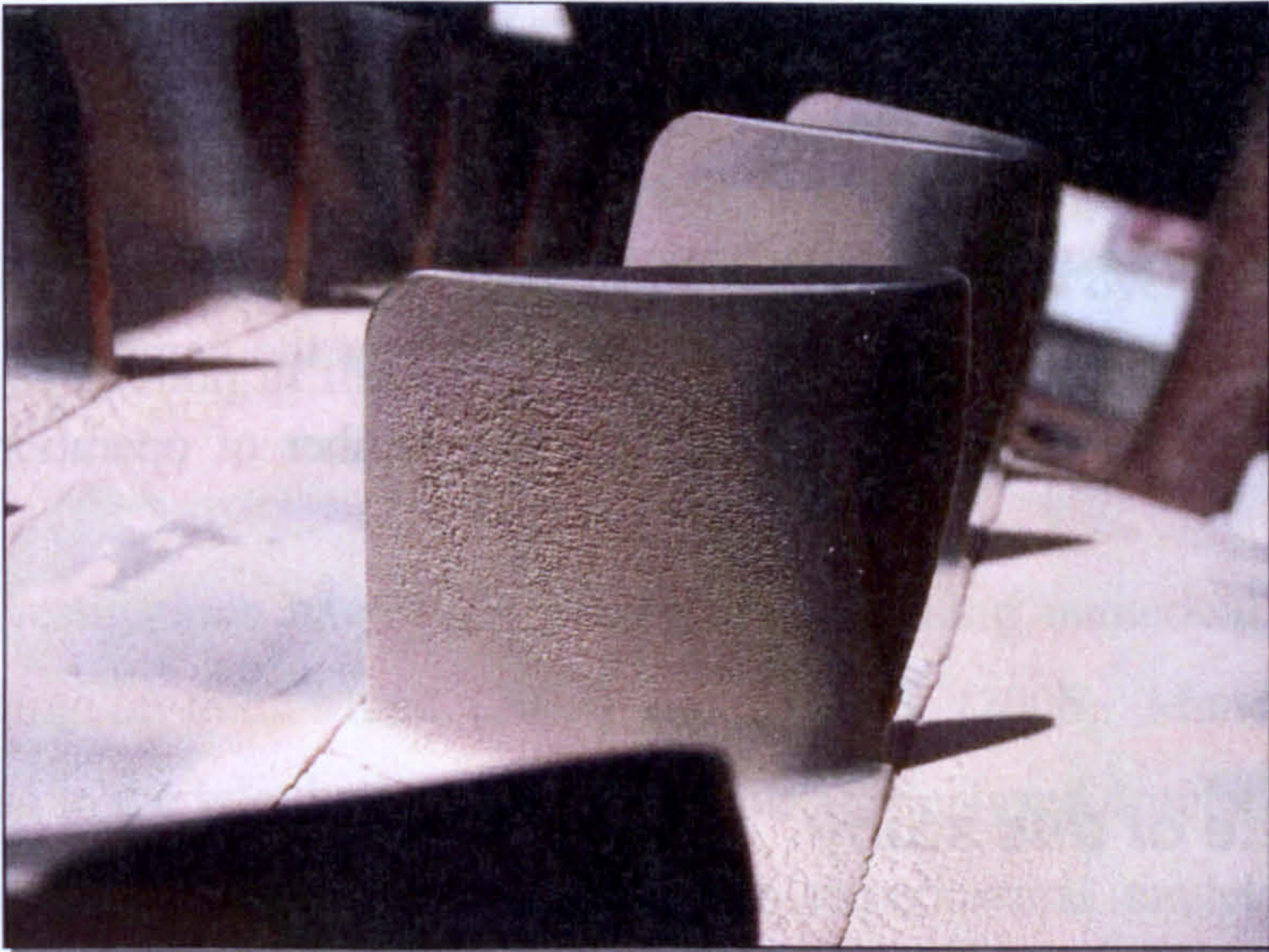


Figure 1-2: Heavy Fouled Compressor Blades in the Desert Environment (Courtesy of Arabian Gulf Oil Company (AGOCO), Nafora Field, 2008)

With a fouled compressor a gas turbine is more difficult to start, the peak power is reduced as is the thermal efficiency. Importantly the compressor stall margin is also reduced, which may lead to serious engine damage. Unfortunately, even a badly fouled compressor may not show signs of fouling, so any parameter intended for use as a fouling indicator must be very sensitive to fouling. It should also be a reliable method which detects fouling sufficiently early for turbine performance to be restored by the relatively simple and direct process of compressor cleaning (Boyce, 1994; Saravanamuttoo and Seddigh, 1990).

The full mechanisms of fouling in gas turbine compressors have not yet been adequately investigated.

The washing of the compressor is, of course, performed with the gas turbine off-line. The essence of the procedure is for the turbine to rotate slowly while a detergent solution is sprayed in. After that water is sprayed in to remove the detergent.

This crank washing requires operators to:

- 1) Secure (tag-out) several gas turbine subsystems,
- 2) Pressurise the wash tanks containing the appropriate solutions,
- 3) Perform the crank wash (both detergent and water),
- 4) Clean up and dispose of the residual wash chemicals (hazardous materials),
- 5) Line up and activate the previously secured subsystems (tag-in),

- 6) Start and run the turbine for several minutes to dry interior surfaces and components,
- 7) Check turbine parameters to determine to what degree performance was recovered.

The crank washing process is expensive, time consuming, requires generator down-time, and subjects the turbine to additional wear and tear. Washing the compressor only when it is required is therefore preferable to cleaning at a set number of operational hours or time interval.

1.3 Importance of this study

The literature review shows that in previous studies fouling has been associated with arbitrary factors related to the power output of the engine. This author believes it is necessary to relate the mechanisms of fouling with real physical parameters such as changes to blade surfaces. The importance of this is demonstrated by following example: If the annual electricity produced by a particular gas turbine of 300MW/h (Power Plant) is 2.65×10^6 MWh per year¹. That is US\$132 million of electricity sales per year. Suppose fouling reduces the compressor pressure ratio by 1% for the entire 4 years that is typical of the period between consecutive engine overall maintenance, engine production will have fallen short by 650,000 MWh. This represents US\$32 million loss due to unsold electricity.

The sums of money involved have gained the interest of companies and governments to study the deterioration mechanism in industrial gas turbines. An outcome of this combined interest is this research sponsored by the Libyan Oil and Gas Industry. Cranfield University, a leader in gas turbine development are collaborating with Recovery-Motor Power Ltd. a leading commercial company in on-line compressor washing.

¹Price of 1MWh =50 USD

1.4 Previous Collaboration

This thesis has grown out of previous research work carried out by the Gas Turbine Performance Engineering Group (GTPE) in Cranfield University over the last ten years. In particular, the collaboration of the Group with R-MC (Recovery-Motor Cleaning Ltd) into on-line washing.

Previous PhD investigations have been presented concerning numerical CFD analysis of compressor washing on-line: Mund (2006) and Mustafa (2006). Mund examined water droplet distribution in the inlet bell mouth whilst Mustafa analyzed droplet trajectory. In both projects the necessity to validate the results with experimental models was mentioned. Further experimental studies of compressor fouling mechanism have been carried out by Osvaldo (2007) and by Dimitrios (2008).

The research described here will investigate for the first time an experimental study of compressor cascade online washing. This new area explored in this investigation will be under the supervision of Dr. Kenneth Ramsden and Professor Pericles Pilidis. Technical support was given by Mr Paul Lambart from R-MC.

1.5 Software description

The speed of computational technology has increased to such an extent that today, it is possible to solve “n equations with n unknowns” in a reasonable computational time. This has meant rapid developments in Computational Fluid Dynamics (CFD) algorithms, particularly in the area of numerical solutions to the Navier-Stokes equations. In this research programme which required the computational analysis of flow patterns ANSYS CFX was used. This is a powerful state of the art code written in C to solve CFD models and has the capacity to process complex geometries with structured and unstructured meshes in two and three dimensions. Furthermore, the maintenance procedure is simulated through Computational Fluid Dynamics (CFD) investigations. As simulating tool, CFX version 10.0 has been used (particularly, CFX-TurboGrid, CFX-Pre-Processing, CFX-Solver and CFX-Post-Processing), a commercial code suitable for CFD analyses on turbomachinery components. A ‘k- ϵ ’ turbulent model, with the RNG option (Reynolds Normalized G) has been chosen.

Engine performance was obtained from the simulation program TURBOMATCH, a program written in FORTRAN by staff and researchers of the Gas Turbine Performance Engineering (GTPE) to describe thermodynamic gas turbine performance. The code is flexible and it is possible to add-on subroutines which simulate new engine configurations. The program library includes nine compressor and turbines maps to determine design or off-design point for engine operation. Using pre-programmed sub-routines called “codewords” and “bricks” it is possible to evaluate likely degradation of engine performance from predicted parameters of fuel consumption, output power, overall efficiency, etc.

1.6 Thesis Aims and Objectives

The Aims of this thesis are

- To investigate fouling mechanisms within the compressor section of a gas turbine engine,
- To validate the CFD results obtained from the experimental work, and
- To experimentally examine the on-line washing of the cascade tunnel to determine the optimum on-line compressor wash frequency.

The objectives of this research were:

1. To set up an experimental cascade facility to study the effect of the changes to blade aerodynamics and the mechanism of fouling.
2. To obtain experimental information of compressor washing on-line to validate previous theoretical studies.

To achieve these objectives, this research established the following programme:

Part I: Simulation and diagnostics of an industrial gas turbine engine using the Turbomatch performance code.

1. Specification of operating conditions of an industrial gas turbine.
2. Analysis and simulation of the engine in both clean and degraded cases.
3. Analysis of the effect of ambient temperature on engine performance.

4. Investigation of the effect of compressor fouling on engine performance.
5. Estimation of turbine blade creep life.
6. An economic analysis based on increased TET, fuel flow rate and power loss.

Part II: Simulation study undertaken on a stage (Rotor-Stator) axial flow compressor.

To investigate the physical mechanism of compressor fouling using CFD tools to better understand the physical mechanisms of compressor fouling for different blade roughness and height.

1. Analysis of blade aerodynamics with modifications of blade surface roughness (fouling).
2. Analysis of gas turbine degradation with compressor fouling.
3. Comparison of experimental results with CFD model predictions.
4. Investigation of the blade regions most sensitive to the fouling, using a CFD study.
5. Techno-economic study of compressor washing on-line.

Part III: Experimental investigation of a cascade compressor with fouled blades before and after online cleaning.

1. Design of the test rig based on the blade aerodynamics.
2. Study three dimensional CFD predictions for smooth clean cascade blades.
3. An experimental study of compressor on-line washing.
4. Comparison of experimental results with CFD model predictions for cleaned and fouled compressor cascade.

1.7 Contribution to knowledge

The result of this research describes the effect of fouling applied at different regions on the suction surface of both the rotor and stator of a high speed axial compressor stage using CFD tools. In particular, the research presents three dimensional predictions for an actual compressor stage applying various levels of fouling.

For the fouling cases considered, pressure distributions in front of and at the rear of the blades were estimated and analysed in order to examine the aerodynamic performance of the compressor stage. Furthermore an analysis of the impact of blade roughness on the stage efficiency as a function of the location of fouling has been carried out. Different fouling levels (roughness heights) have been applied on different regions on the rotor-stator stage compressor to investigate which region on the blade is the most sensitive to the fouling.

It has been demonstrated that the leading edge and the concave surfaces are the most sensitive regions to the fouling.

A CFD analysis one chord downstream of the blades indicates that the pressure loss coefficient increases as roughness level increases of course. Furthermore, the occurrence of trailing edge flow separation increases as roughness increases.

In support of this CFD research programme an experimental examination of the effect of roughness on compressor blades has been undertaken. As a result, a systematic series of cascade experiments using varying levels of surface roughness have facilitated a rigorous estimation of the magnitude of compressor stage inefficiency caused by fouling. By observing the actual distribution of fouling in a typical compressor, it is now possible to estimate more accurately the overall impact of its fouled performance on overall compressor efficiency.

The result of this research has been to show for the first time experimental investigation on compressor cascade online washing using a novel cleaning detergent from R-MC Company.

1.8 Thesis Structure

The thesis is separated into two main sections. The first section (Chapters 1, 2 and 3) presents a study of the fouling mechanism and blade aerodynamics. The second section (Chapter 4, 5, 6, 7 and 8) presents the results of the application of CFD studies and experiment to a real case to evaluate engine performance and the effectiveness of compressor washing on-line.

Chapter 2 presents an extensive review on the state-of-the-art on measurement of compressor performance degradation due to fouling, localization and the impact of roughness on loss of compressor efficiency and mass flow and deviation (change on the compressor map) in the compressor.

Chapter 3 show the performance result for a single shaft gas turbine engine working in a desert environment, and the impact of this harsh environment on engine performance is investigated. Furthermore real data and a case study of a clean and a fouled engine over a period of 3 to 4 months has been used to investigate the compressor fouling effect on the gas turbine.

Chapter 4 describes the development of a simple, meanline analysis of a single-stage axial compressor model (HP9), which has hydrodynamically smooth blades. A clean stage has been studied using CFD tools and validated to the Experimental results.

Chapter5 shows the scenario of the fouling mechanism of the compressor stage. A different level of roughness has been applied on the compressor stage (rotor-stator) using CFD. The suction surfaces of the rotor and stator has been divided into different regions to investigate the sensitive region to the fouling.

Chapter 6 presents CFD results obtained for fouling of the compressor cascade facility by applying different levels of surface roughness on the blades. The CFD model was used to investigate the particle deposition on the blade surface and the changes produced in the surface roughness. In addition, this model was used to study the consequences of changes to blade aerodynamics.

Chapter 7 presents the most important contribution of this research: the results of a series of experiments on the compressor cascade to investigate the effectiveness of compressor washing on-line.

Chapter 2 – Literature Review

2.1 Compressor Fouling

Fouling is a serious problem and is defined as degradation of flow capacity and efficiency caused by adherence of particular contaminants to the gas turbine engine airfoil and annulus surfaces. The result of fouling is build-up of unwanted material which disadvantageously changes the shape of the airfoil and airfoil inlet angle, increases surface roughness, and reduces the open area of the airfoil throat. The consequence is reduced performance. Fouling can occur in both the compressor and the turbine, with compressor fouling more common and with more serious effects on engine performance. Typically, between 70% and 85% of all gas turbine engine performance losses accumulated during operation are attributable to compressor fouling (Diakunchak I. S., 1992).

The contaminating particles and droplets that foul the compressor enter with the inlet air, cooling air, fuel, fuel additives and the water/ steam used for NO_x control. This is a particular problem for the oil and gas industry where sticky hydrocarbon aerosols are invariably present, and sand and salt water often present. These adhere to the surface of the compressor blades and rapidly degrade flow capacity and reduce efficiency. Operation of a gas turbine engine on heavy or crude oil will have a corrosive effect on the hot end gas path components. The corrosion plus deposits on the flow path surfaces will result in engine performance deterioration.

Fouling can be partially reversed by cleaning, but this will require turbine downtime. Unfortunately even when using high effectiveness air filters, fine particles less than 5µm will combine with oil vapour, sea salt, water, etc and adhere as deposits on the compressor blade surface. This changes their aerofoil shape, reduces airflow and increases temperature which could result in over-heating of the blades. Not only is power output reduced but turbine blade life shortened.

Fortunately, compressor washing may restore all or part of the loss in performance. Figure 2-1 shows the performance degradation as a result of compressor fouling - which begins as soon as the engine is in operation – and the effect of cleaning by washing.

All compressors are susceptible to fouling. The degree and rate of fouling depend on many factors including: type and condition of the air-borne contaminants, the local environmental conditions (e.g. presence of sand, humidity, etc.), compressor design, compressor airfoil loading, and airfoil surface smoothness/coating (Seddigh, F., and Saravanamuttoo, H. I. H., 1991). Thus fouling rates vary significantly with to site location due to the surrounding environment and climate. Indeed, weather has been shown to have a substantial effect on fouling rate. Fouling not only reduces compressor efficiency, but also reduces the compressor surge margin and may result in compressor surge (Mezheritsky, A. D., and Sudarev, A. V., 1990).

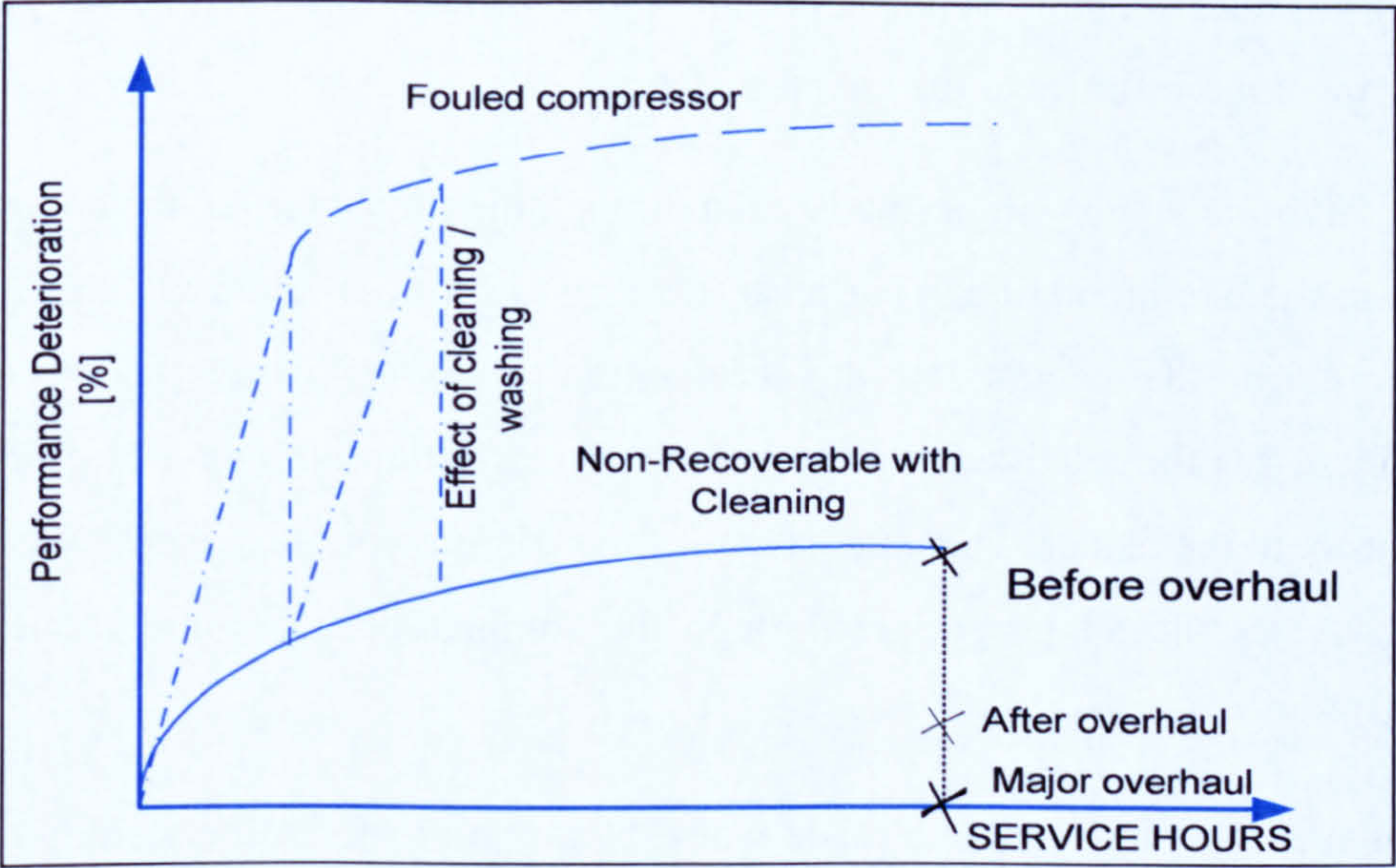


Figure 2-1: Typical compressor performance deterioration and recovery with cleaning (Diakunchak I. S., 1992)

2.1.1 The Nature of Compressor Fouling

The adherence of hard particles (e.g. dust, ash, dirt, sand, rust less than 10µm in diameter) to airfoil and annulus surfaces, increase surface roughness and change the geometric shape of the components in the air stream; contaminants of diameter larger than 10µm can also cause erosion (Kurz, R. and Brun, K., 2000) , (Meher-Homji C. B., 1990) and (Stalder J. P., 1998). The effects of soft particles such as unburned hydrocarbons are limited to fouling only.

The deposition rate on the leading edge is about ten times greater than for the rest of the blade surface, for both rotor and stator (Bouris, 2002). Fouling is deposited on the blade pressure side, leading edge and rotor pressure surface by impact (Boyce et al., 1994), (Meher-Homji C. B., 1990). Foulants also stick to the suction side, diffusion dominates particle deposition on the rotor suction side and both stator surfaces, thus deposits are typically lighter and smaller (Bouris, 2002).

Deposits are typically $\text{PH} \leq 4$, increasing the risk of pitting corrosion (Stalder J. P., 1998). Water-soluble compounds can contain chlorides which cause corrosion or are themselves hygroscopic (moisture absorbing). Water-soluble compounds can be rinsed away, but some will be imbedded in water-insoluble compounds. Water insoluble compounds are mostly hydrocarbon residues. Fouling deposits become more difficult to remove if left untreated, as aging binds them more firmly to the airfoil surfaces.

Liquid droplets and vapour will normally exist through the first stages of the compressor, and these increase the likelihood of salt particles adhering to surfaces (Stalder J.-P. a., 1994). The static pressure drop that occurs with the acceleration of the airflow as it passes through the bellmouth increases the relative humidity, which increases the likelihood of precipitation of contaminants onto the blades. The latter stages of the compressor are subject to relatively less fouling because as the air progresses through the compressor it becomes hotter and drier (Zaba, 1980).

Compressor fouling deterioration is made worse by any oil leaks in the vicinity of the blade surfaces. Oily substances (hydrocarbons) in the air flow act as a form of glue fixing dirt particles to compressor airfoils and shroud surfaces. Subject to high temperatures within the turbine these oils form a thick coating baked onto the surfaces. In addition to soot particles produced in the combustor, can also accumulate on turbine flow path surfaces (Stalder J. P., 1998) and (Zaba, 1980). Hot-end fouling reduces the firing temperature and the overall turbine efficiency.

2.1.2 Sensitivity and Susceptibility to Fouling

For axial compressors the inherent sensitivity to fouling is determined by the air inlet velocity at the inlet guide vanes and compressor pressure ratio; although aerodynamic and geometric

characteristics are also important. Researchers have discussed whether smaller gas turbines (<3MW) are more sensitive to fouling than large engines (Tarabrin, A.P. et al, 1998); (Saravanamuttoo, H.I.H.; Muir, D.E.; Marshall, D.J., 1988)). The consensus is that stage loading is the major factor whether the gas turbine is large or small (Saravanamuttoo, H.I.H.; Matthee, F.A.H., 1985), (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985) and (Saravanamuttoo & Seddigh, 1990). It is now generally agreed that fouling will be more detrimental to compressors with heavily loaded stages than those with lightly loaded stages. It follows that fouling will build up most rapidly and has most impact on the performance of gas turbines operating at peak load.

Smooth airfoil surfaces, or those with coatings, are less susceptible to fouling and respond better to cleaning by washing (Chow, 1995), (Martin, 1998). Up to a certain point an increase in humidity increases the fouling rate: above a certain humidity level, the water droplets in the air that condense out as the air accelerates through the bellmouth will tend to wash the blades, and power losses due to fouling may be reduced (Diakunchak I. S., 1991), (Stalder J. P., 1998). Fouling rates typically increase as the ambient temperature increases (Haq, Inamul and Saravanamuttoo, H.I.H., 1991); (Kurz, R. and Brun, K., 2000)

2.1.3 State of the Art in Compressor Fouling, Detection and Localization

Performance degradation of gas turbine engines increases with operating time because the condition of engine components that become fouled, eroded, corroded, etc., will get progressively worse with increasing operating time (Upton, 1974), (DeGreef, 1978), (Bammert K. a., 1980) and (Haub G. L., 1990). Usually fouling is gradual and the consequent reduction in gas turbine performance is also gradual, but there comes a level of fouling when, if the compressor isn't washed, deterioration of performance accelerates (Boyce et al., 1994), (Meher-Homji C. B., 1990) and (Stalder J. P., 1998).

Three types of engine performance deterioration may be listed:

1. Performance deterioration recoverable with cleaning/washing, such as deposits on the blades;

2. Performance deterioration which is not-recoverable with cleaning/washing, due to baked-on contaminants that must be physically scraped off the airfoils (during an overhaul); and
3. Permanent performance deterioration such as blade leading edge and tip erosion, which is not recoverable even after an overhaul and the refurbishment of all clearances, replacement of damaged parts, etc.

Recoverable Deterioration

Performance deterioration due to fouling is mostly recoverable (Boyce M. e., 1993), (Boyce et al., 1994), (Kurz, Rainer; Burn, Klaus, 2000). Recoverable performance loss is due mainly to compressor fouling which reduces flow capacity and compressor efficiency. Recoverable performance loss can be restored through online or offline compressor washing. (Peltier R. V., 1995) reported that after 700 operation hours for a 21.9MW gas turbine engine typically 1% of power loss was fully restored using a crank wash. The following methods are used in cleaning the flow path components of gas turbine engines, without the necessity of engine disassembly:

1. On-line dry cleaning of compressors with rice husks, pecan shells, or some other dry abrasive material.
2. On-line (at base load or part power) wash of compressor with water, water mixed with detergent, or some other suitable fluid, which is preferably nontoxic, non-flammable, and biodegradable.
3. Soak wash (with the engine not running) with a suitable fluid.

This last method is probably the most efficient for cleaning hot-end components coated with deposits as a result of running on crude oil fuel, without opening up the engine. Smooth airfoil surfaces or those with coatings which reduce surface roughness and are less susceptible to fouling will generally give better response to cleaning or washing.

One recoverable deterioration is when the inlet filter becomes iced-up. The consequent performance deterioration is likely to be even more sudden and severe than with normal compressor fouling. However such a phenomenon is temporary and, assuming it does not result in compressor damage, will not affect the rate of performance deterioration due to fouling.

Non-recoverable with Cleaning/Washing

Non-recoverable performance loss caused by the wearing out of engine components can only be restored by extensive disassembly and repairing of the engine. (Peltier R. V., 1995) has reported that engine performance of a 21.9MW gas turbine engine was fully restored after such an overhaul.

Even regular cleaning/washing, will leave some surface deposits that will continue to detract from the performance of the component. A number of faults developed within the engine will not be addressed by cleaning/washing, these include: flow path damage, surface erosion/corrosion, tip and seal clearance increase, etc., and the resulting performance deterioration will remain and probably worsen with time.

Permanent Performance Deterioration

Completion of a major overhaul would be expected to return the engine to the conditions of the initial performance acceptance test. However, invariably the engine performance is not quite restored to the "as new" condition because of such factors as cylinder distortion (eccentricity in clearances and increased leakage paths), surface roughness of flow path components due to erosion or rust scale deposits, distortions causing loss of aerodynamic performance and increased leakage, airfoil untwist, etc. However, under normal circumstances the performance deterioration is small.

2.1.4 Factors Affecting Performance Deterioration

Contaminants, air-borne contaminants are listed below:

1. Hard particles (causing erosion and fouling): sand, rust, dirt, dust, soot particles.
2. Soft particles (causing fouling): oil, unburned hydrocarbons, soot, air-borne industrial chemicals, fertilizers, herbicides, insecticides, flying insects, air-borne salts.

About 80 percent of particulate contaminants are below 2µm in diameter. If an evaporative cooler is used, then waterborne contaminants, such as dirt and salts, may be introduced and result in fouling and corrosion. Contaminants affecting the hot end include fuel-borne contaminants produced as a result of the combustion process, contaminants in water or steam

injected for NO_x control, and any particles, such as rust scale, etc., introduced through the cooling air system piping.

- **Fouling Detection**

Any condition based system for compressor cleaning must include close monitoring of engine parameters and a system that can accurately predict performance degradation as a result of fouling. The ideal parameter for monitoring fouling would:

- 1) provide an accurate, repeatable indication of compressor condition,
- 2) be unaffected by changes in ambient conditions or process load,
- 3) provide simple and quick data collection, and
- 4) require no specialist technical skills to interpret the measured data.

These characteristics were not found for any parameters or combination of parameters tested (Boyce et al., 1994), (Haq, Inamul and Saravanamuttoo, H.I.H., 1991). However, change in compressor pressure ratio or air intake depression did provide a good assessment of compressor fouling when corrected to standard conditions.

- **Comparison of Parameters to Detect Fouling**

Compressor Temperature Rise Coefficient ($\tau_c = T_{T,out} / T_{T,in}$): is not considered a parameter that can reliably identify the effect of fouling (Arnulfi, 1993), (Kurz, Rainer; Burn, Klaus, 2000). τ_c is affected by fouling, but not uniformly, its affected value can be higher or lower than for the “as new” condition. A small, moderately loaded Solar Centaur gas turbine compressor experienced no change in τ_c with up to 20% of the stages fouled. It then experienced a small increase in τ_c as additional stages were fouled (Saravanamuttoo, H.I.H. and Aker, G.F., 1989). Experiments on a large, heavily loaded GE LM-2500 compressor showed a drop in τ_c as the the front stages were fouled, there was then a small increase in τ_c as the rear stages were fouled (Saravanamuttoo, H.I.H. and Aker, G.F., 1989).

Exhaust Gas Temperature (EGT): Compressor fouling or erosion or fuel nozzle deterioration or oil leaks or turbine fouling may individually or in combinations increase EGT for a given power output. Thus, EGT is not a good parameter to monitor compressor fouling.

Compressor Efficiency (η_c): is considered neither sensitive nor accurate enough to determine the degree of fouling (Zaba, 1980); (Haq, Inamul and Saravanamuttoo, H.I.H., 1991), (Kurz, R. and Brun, K., 2000), (Saravanamuttoo, H.I.H. and Aker, G.F., 1989). Also, there are too many variables associated with accurately measuring the gas turbine power output for it to be a good indicator of fouling location.

Compressor Pressure Ratio ($PR = P_{T,out} / P_{T,in}$): is considered one of the more effective parameters for measuring compressor deterioration. PR always decreases with increased fouling, but the airflow at the compressor outlet is hot and non-uniform, making an accurate measurement of total outlet pressure difficult.

Air Intake Depression (ΔP_{intake}). ΔP_{intake} is the static pressure drop across the bellmouth, assuming total pressure is constant across the bellmouth. Originally proposed by Scott (1979), ΔP_{intake} is simple and inexpensive to measure (if there is room for an additional pressure tap at the compressor inlet). A number of authors (Boyce et al., 1994), (Diakunchak I. S., 1991); (Haq, Inamul and Saravanamuttoo, H.I.H., 1991), (Scott, 1979) consider this may be the most accurate and sensitive of all parameters for the assessment of fouling.

To measure the mass flow through the inlet three measurements are necessary: The temperature (T_T), total pressure (P_T) and static pressure (P) at the bellmouth throat. Thus, monitoring ΔP_{intake} is a method to accurately determine the corrected mass flow. ΔP_{intake} is simpler to measure than intake PR or η_c . The corrected mass flow, $\dot{m}_{a,corr}$, can be compared with the corrected speed, N_{corr} , as an indicator of fouling.

The decrease of ΔP_{intake} relative to a “clean engine” is an excellent indicator of fouling. When a compressor fouls, the decrease in mass flow rate is roughly proportional to the square root of the relative change of ΔP_{intake} (Scott, 1979). Hence, ΔP_{intake} is sensitive to changes in mass flow rate. Oil pumping stations in desert conditions, where mid-day temperature exceeded 44°C and dense clouds of airborne sand particles were almost always present, have successfully used ΔP_{intake} monitoring to determine when to wash (Haq, Inamul and Saravanamuttoo, H.I.H., 1991). In pipeline applications where fuel economy is critical, compressors were washed when a 3% reduction in ΔP_{intake} was detected – about 1.5% decrease in baseline mass flow (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985). Cleaning at this stage of degradation restored ΔP_{intake} (and mass flow) to the original level.

Once the level of deterioration has been found, it is possible to perform a cost-benefit to find the most economical time to wash. However, if maximum power is always required, even a small amount of fouling can pose significant problems, as the power reduction due to fouling is greatest for peak load. In such conditions there is little to be gained from on-condition compressor cleaning; frequent, regularly scheduled washing should be the standard (Boyce et al., 1994), (Saravanamuttoo, H.I.H. and Aker, G.F., 1989).

2.1.5 Fouling Simulation

To predict the overall effect on compressor performance of fouling of any single stage requires that each stage be modelled correctly as a stage in its own right and as a component of the compressor (Boyce et al, 1994). In this is achieved then simulation exercises can provide a quantitative estimate of how severely engine performance will suffer from fouling of various magnitudes and at different locations. The model can then be used to determine the optimum economical duration between compressor washes or, for on-condition washing, to set the fouling threshold limit (Boyce et al., 1994), (Saravanamuttoo, H.I.H. and Aker, G.F., 1989), (Stalder J.-P. a., 1994).

- **Linear Progression of Fouling:**

(Saravanamuttoo, H.I.H. and Aker, G.F., 1989) have investigated the so-called " $k_1:k_2$ " model of fouling for the compressor stages on a GE LM2500 and Solar Centaur oil pipeline gas turbines. Given that fouling impacts on the front stages of the compressor more than on the rear stages the authors reduced the stage flow coefficients by a fixed percentage, $k_1\%$, for each stage as the flow passed through the compressor: from $k_1\%$ for the last (n^{th}) stage to $nk_1\%$ for the first stage. Stage efficiency was decreased by a factor of $k_2\%$ in the same way. Initially, for the first stage a 1% reduction in flow coefficient was made. For the next step, a 2% reduction in the flow coefficient was assumed for the first stage and a 1% reduction for stage two was made. This process was repeated until all n stages were included in the process. It has been shown that if the fall in efficiency for front stage fouling is 1% then there is a 0.25% reduction in stage efficiency. Hence, the model was (1:0.25). This linear progressive fouling model was found to work well up to the middle stages (Saravanamuttoo H. a., 1997)

2.1.6 The Mechanism of Fouling

The mechanism of entrainment of particles by a surface of a body situated in the stream of an air-aerosol mixture was described by (Fuks, 1955). (Tarabrin, A.P. et al, 1998) also noted that the deposition of particles on the surface of a blade takes place under the action of inertial forces acting on the particles and forcing them to move across the curved stream lines (the particle trajectory deviates from the stream lines). On the other hand, when particles move in the flow path of compressor the centrifugal inertial forces make them move to the periphery of the compressor passage.

The coefficient of entrainment or the separation factor E according to (Fuks, 1955), is determined as a ratio:

$$E = \frac{\text{number of particles colliding with the surface of the body}}{\text{number of particles which could fall on the body surface if the stream lines were not deviated by the body}}$$

In this case it is assumed that all the particles stick to the body when they collide with it. In Figure 2-2 one can see the trajectories of the particles and the streamlines of the flow around a cylindrical body. If the particles are uniformly distributed in the air at a certain distance from the body, and the radius of the particle is negligible if compared with the radius of the cylinder, then

$$E = h/L \quad (1)$$

Where L is the characteristic size of the body and h is the distance between the stream centreline and limit trajectory at the upstream infinity

The limit trajectory is a trajectory of the particles that are still seized by the cylinder. In the general case, E can be presented as a function (Tarabrin, A.P. et al, 1998):

$$E = f(St_k, d_p/2L, Re_f) \quad (2)$$

Where

$$St_k = \frac{\rho_p d_p^2 C_0}{18\mu 2L}; \quad Re_f = \rho 2L C_0 / \mu$$

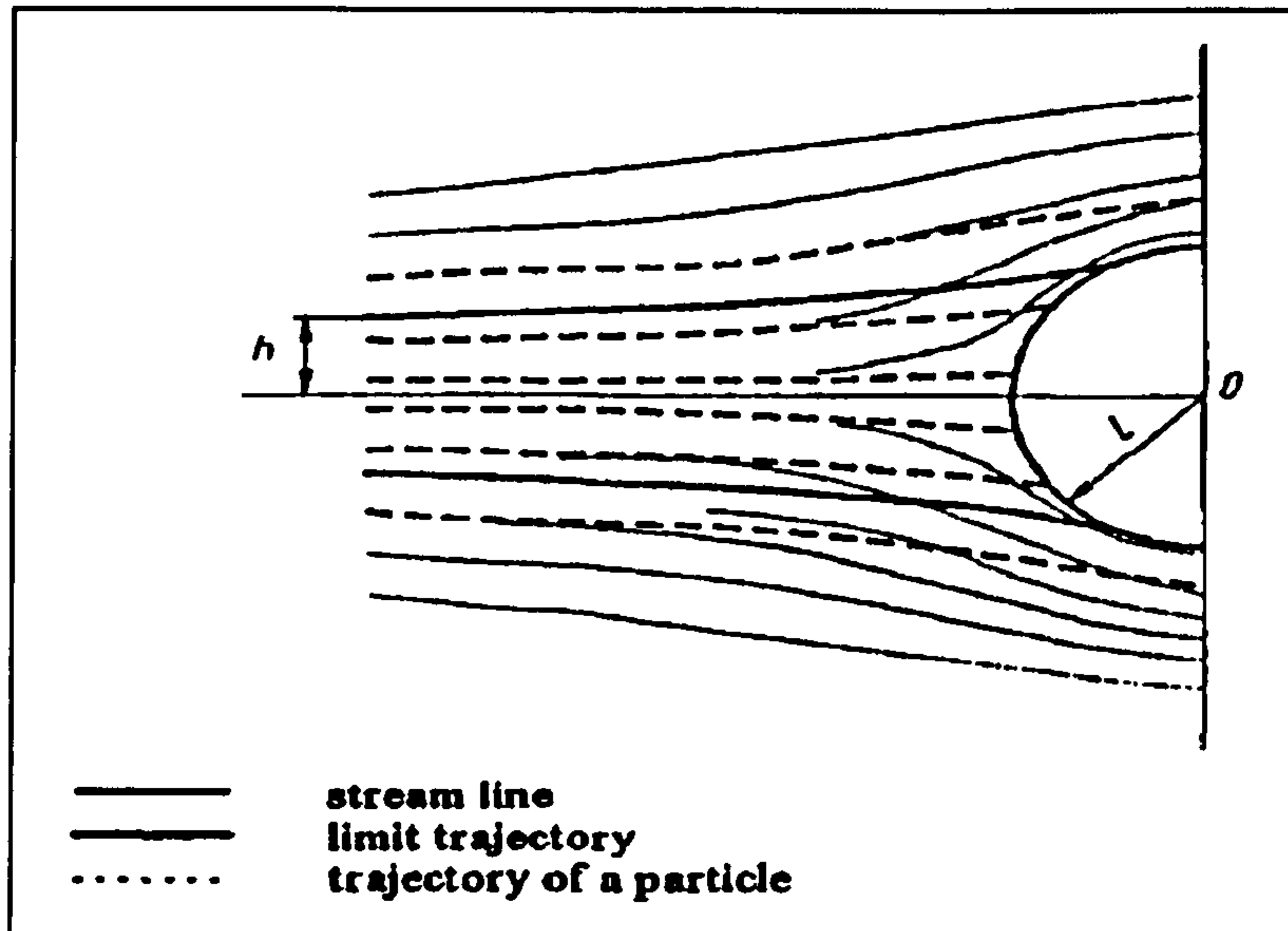


Figure 2-1: Inertial deposition of particles on cylinder surface (Tarabrin, A.P. et al, 1998)

$$L = b \sin (\beta_b - \beta_1) \quad (3)$$

In case of a cascade, Figure 2-3 can be taken for the characteristic size (Tarabrin, A.P. et al, 1998). The coefficient of entrainment for the cylinder with $L = b \sin (\beta_b - \beta_1)$ within the unbounded stream will be:

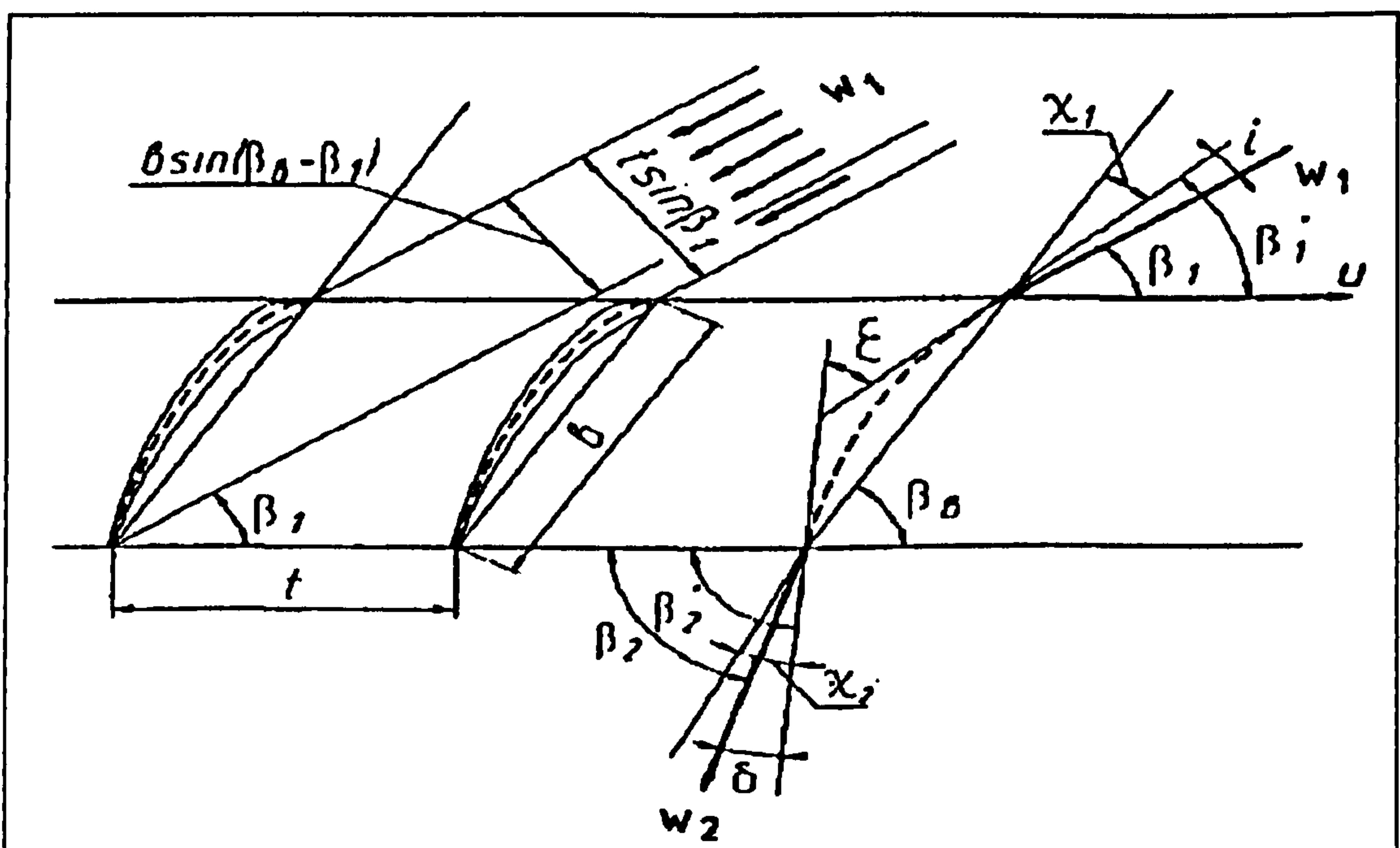


Figure 2-2: Cascade of profiles of axial compressor (Tarabrin, A.P. et al, 1998)

$$E = h / (b \sin (\beta_b - \beta_1)) \quad (4)$$

If in the cascade, the flow path is bounded by two adjacent profiles with pitch t , the cascade entrainment coefficient becomes:

$$E_c = \frac{\text{number of particles sticking to surface}}{\text{number of particles in the flow for one pitch of the cascade}} \quad (5)$$

Then

$$E_c = h / (t \sin \beta_1) = h / (b \sin(\beta_b - \beta_1)) (b/t) \times (\sin(\beta_b - \beta_1) / \sin \beta_1) \quad (6)$$

Or

$$E_c = E(b/t) (\sin(\beta_b - \beta_1) / \sin \beta_1) \quad (7)$$

It can be seen from Figure 2 that

$$\beta_1 = \beta'_1 - i = \beta_b - \chi_1 - i \quad (8)$$

$$\beta_2 = \beta'_2 - \delta = \beta_b - \chi_2 - \delta \quad (9)$$

From Eqs. (8) and (9) for a profile with a mean (skeleton) line of circular arc

$$\frac{1}{2}(\beta_2 = \beta'_2) = \beta_b - \frac{1}{2}(\delta + i) \quad (10)$$

Taking into account that for the cascade situated in the mean radius of a compressor stage and for the optimal regime of work $i \approx 0^\circ$ and $\delta \leq 8^\circ - 10^\circ$;

$$\beta_b \cong \frac{1}{2}(\beta_1 + \beta_2) \quad (11)$$

$$\beta_2 - \beta_1 = (\beta_2 - \beta_1)/2 = \Delta\beta/2 \quad (12)$$

The entrainment coefficient E for an endless cylinder at large Reynolds numbers can be defined theoretically by calculating its potential flow. Based on the data obtained from the calculation of the potential flow around the cylinder reported by (Fuks, 1955) within the full range of the Stokes numbers, characteristic for axial compressors, the dependence Eq. (2) at large Reynolds numbers can be approximated by the formula:

$$E = (1 + 0.77/St_k)^{-1} \quad (13)$$

For the cascade the Stokes number equals:

$$St_k = \frac{\rho_p d_p^2 w_1}{18 \mu b \sin(\beta_B - \beta_1)} \quad (14)$$

Considering Eqs. (12) and (13), Eq. (7) can be written in the form:

$$E_c = (1 + 0.77/St_k)^{-1} \frac{b \sin(\Delta\beta/2)}{t \sin\beta_1} \quad (15)$$

As the head (work in a stage) of the compressor stage increases, the flow turning angle ($\Delta\beta$) and the solidity (b/t) of the cascade also increase. Therefore, the high-head stages are more sensitive to fouling than the low-head ones.

If we compare geometrically similar compressors, (b/t) is constant, then it follows from Eqs. (14) and (15), when the size of the chord decreases, the Stokes number and the entrainment coefficient increase, that is, a physical model smaller than the real compressor is more sensitive to fouling than a full-scale one. Besides that, when the velocity w_1 increases, the value of E_c also increases. It is evident that Eq. (15) defines the inertial deposition of the particles on the windward side of the blade facing the flow.

There is also deposition of particles on the leeward side of the blade profile as a result of swirls and turbulence. (Tarabrin, A.P. et al, 1998) has reported deposit formation on gas turbine compressors blades and claims to have shown the fouling affects the first five to six compressor stages. The fouling of the blades on both the concave and the convex sides decreases as the air passes from stage to stage. As Figure 2-4 and Equation (15) show, the separation coefficient E_c increases with increase of angle of attack and this leads to an increase in the number of whirls on the convex side of the blade and, consequently, the deposition of more particles on both sides of the blade. In deriving Equations (1) - (15) it was assumed that all particles that collide with the surface of the blade stick to it. But not all the particles that reach the blade stick to it, a number rebound and are taken away by the flow. Factors influencing the adhesion of the particles are the local humidity, the presence of oily evaporation, etc. In case of the rotational flow, E_c will be influenced by the centrifugal force. However, these effects are not taken into consideration in Equation (15).

Thus, regarding the sensitivity of the cascades and stages of axial compressors to fouling it can be suggested:

- under conditions of aerodynamic and geometrical similarity, a smaller compressor is more sensitive to fouling than a large one;
- sensitivity to fouling of the axial compressor stage increases when the stage head grows;
- the degree of particles deposition on blades increases with increase in angle of attack.

Having determined the main factors influencing the sensitivity of the axial compressor to fouling, the next section of the present chapter attempts to find a generalized criterion for the sensitivity, and to develop a mathematical model for evaluating the change in the compressor performance as fouling progresses.

As the pressure ratio of a compressor stage increases the flow turning angle and chord to space ratio (c/s) of the cascade also increases. Therefore, the high-head stage is more sensitive to fouling than the low-head one.

Figure 2-4 and 2-5 shows that fouling is present on both the pressure and suction surfaces of the rotor and stator blades. It has been clearly shown that the suction surface is more affected by the deposit formations, than pressure surface.

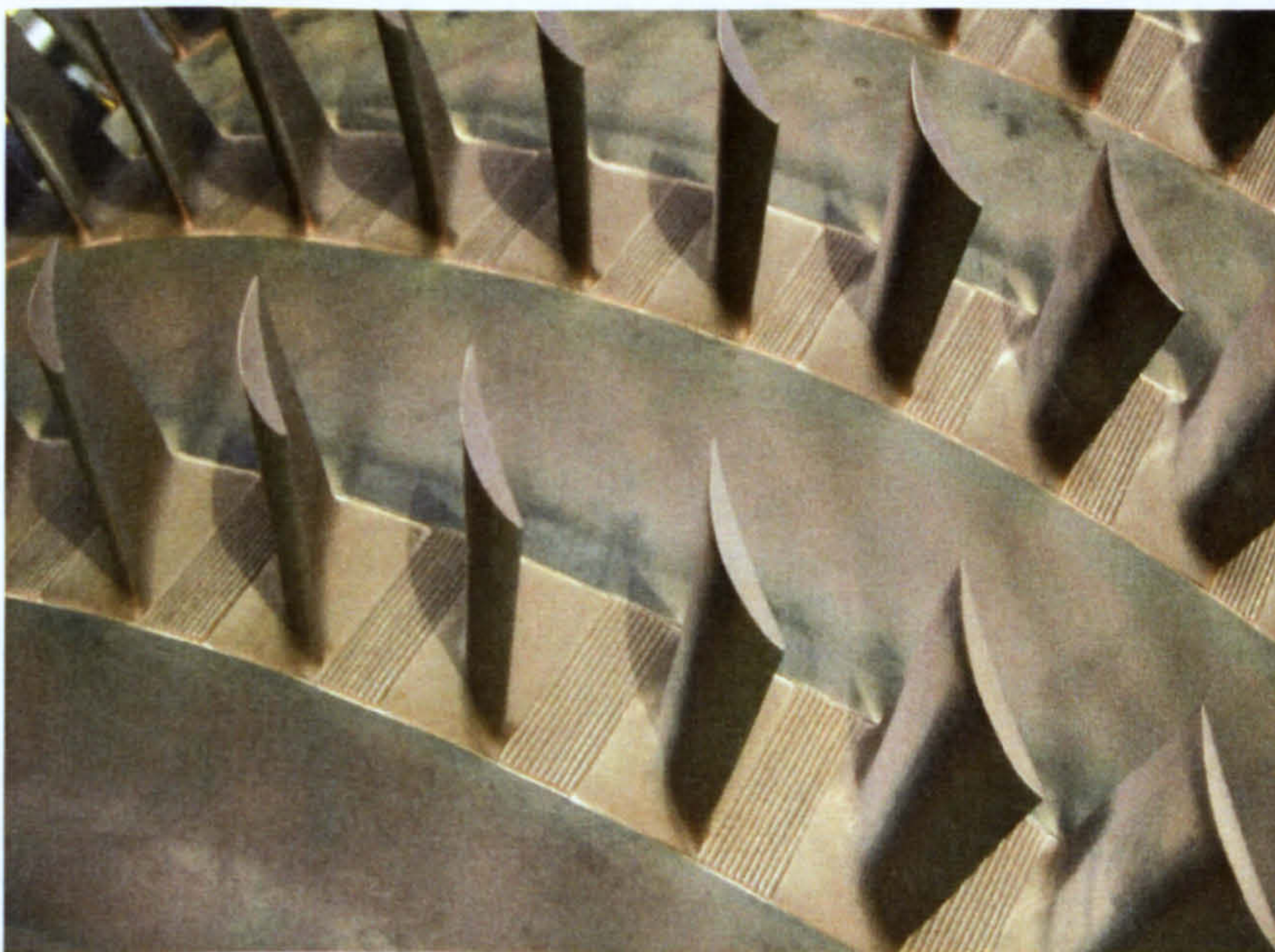


Figure 2-3: Compressor rotor blades fouling of Sulzer type 7 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)



Figure 2-4: Compressor stator blades fouling of Sulzer type 7 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

2.1.7 Sensitivity of Engine Size to Fouling

There are many publications, (for example Aker and Saravanamuttoo 1988; Sedigh and Saravanamuttoo, 1990; Tarabrin 1996) that describe the sensitivity of a turbine to fouling. Most of them confirm that a smaller engine shows more deterioration due to fouling than that a larger one.

Some mathematical simulations of fouling in multistage compressors have recently appeared. Field operating data has revealed that the first stage of a compressor fouls the most, constituting 40-50% of the total fouling effect, and the degree of fouling diminishes from the front to the rear stages.

Recent mathematical models have been developed for the estimation of the fouling effect on the compressor performance. These methods allows one to obtain a clearer idea on how fouling affects the dynamics of the change in the performance of a compressor and a gas turbine unit, and to correlate fouling with changes in such parameters as inlet turbine temperature, compressor mass flow, and the decrease in output and efficiency, etc. Some authors have attempted to use these models to try to determine a criterion for sensitivity of a compressor to fouling.

Aker and Saravanamuttoo (1989) and Seddigh and Saravanamuttoo (1991) developed mathematical models for calculating the performance of the axial compressor with fouled blades. Based on the operational data, the authors stated that the number of fouled stages constitutes $\bar{z}_f = z_f/z = 40\text{-}50$ percent of the total number of the compressor stages. In axial compressors of heavy-duty gas turbines, the number of compressor stages is generally from 8 to 20. In this case, if the percentage ratio is the same (e.g., 50 percent), the number of the fouled stages will range from 4 to 10, i.e., it will depend directly on the total number of stages. In this connection, if we use the mathematical model proposed by these authors, the results of the calculation of the compressor main parameters will also depend on the number of fouled stages, considering that z_f is the same. Besides that, it can be shown that the index of sensitivity of the compressor to fouling proposed by Seddigh and Saravanamuttoo (1991) is directly connected with the number of the compressor stages. In fact,

$$\frac{N_e}{\dot{m}c_p\Delta T_{stg}^*} = \left(\frac{T_3^*\eta_c\eta_t}{T_{in}^*\pi^{(\kappa-1)/\kappa}} - 1 \right) z \tag{16}$$

If we compare the engines with equal π, T_{in}^*, T_3^* we can see that the value $N_e/(\dot{m}c_p\Delta T_{stg}^*)$ mainly depends only on z . It is evident that in this case the changes occurred due to fouling in N_e and other parameters, will also be determined mainly by a number of compressor stages with $\bar{z}_f = \text{const.}$

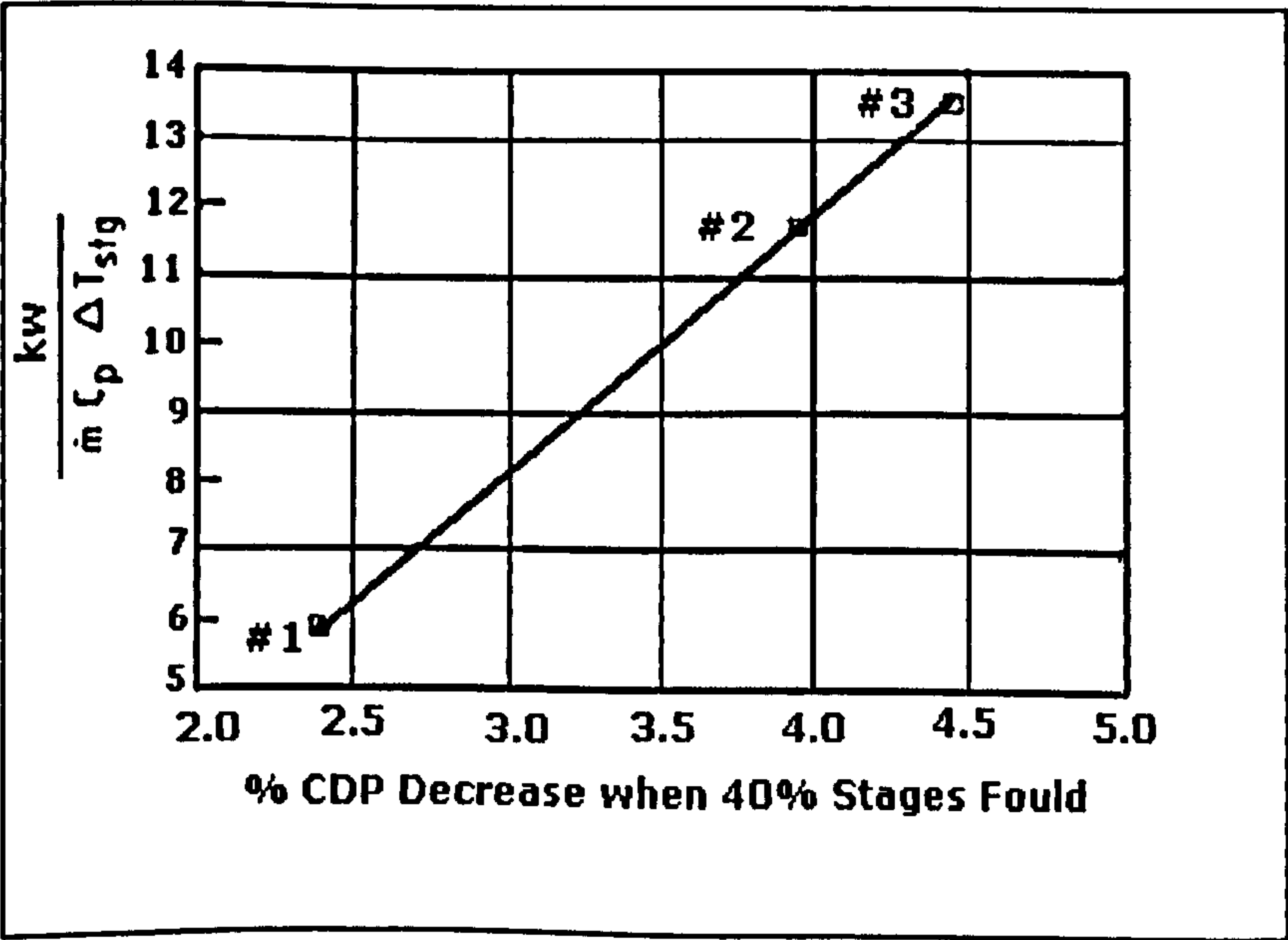


Figure 2-5: Variations of power with percentage CDP decrease of fouled stages (Kolkman, 1993)

Figure 2-6 shows the variation of this index against the percentage decrease in CDP for three different size engines, due to 40 percent fouling of the stages. The variation of this index with the sensitivity of an engine to fouling is linear and consistent. This index incorporates important specification data for an engine, i.e., the engine size manifested by its design power output, its design mass flow rate and its stage loading (Kolkman, 1993).

Tarabrin (1996) has established another index of compressor sensitivity to fouling (ISF), and suggested that, the following equation will reflect qualitatively the sensitivity of the axial compressor to fouling under similar environmental conditions.

Tarabin also noted that a compressor of a smaller size is more sensitive to fouling than that of a larger size. Finally, he noted that sensitivity to fouling of an axial compressor increases when the stage pressure ratio increases.

As follows from the foregoing statements, the sensitivity of the compressor stage to fouling depends mainly on the following parameters:

$$\text{Sensitivity to fouling} = f \left(\frac{b}{D_c}, C_z, u, H_{th}, D_c, \rho_{stg} \right) \quad (17)$$

The conclusions made at the end of the first paragraph that the index of compressor sensitivity to fouling (ISF) has been presented as:

$$ISF = \frac{\dot{m} c_p \Delta T_{stg}^*}{(1-r_h^2) D_c^3} 10^{-6} \quad (18)$$

2.1.8 Cost of Fouling

Although the axial compressor deterioration is shown to be less effective to the engine performance than other components (Pilides, 2001), there is significant costs to associated with compressor fouling due to resulted performance deterioration. Furthermore, high specific fuel consumption as well as higher firing temperature (TET) is needed in order to maintain demanded power output. There is no doubt that compressor fouling leads to significantly increased fuel cost and reduced hot gas path component life due to engine operating at higher firing temperature. Consequently, reduced intervals between major overhauls may be necessary, which results in additional maintenance costs. As suggested by (Diakunchak, 1992), a compressor fouling corresponding to the decrease in power output of

3% and an increase in heat rate of 1 %, results in increased operating costs of a small engine (46.500kw) of typically about 1.5 million dollars over a three year period.

2.2 The Effect of Fouling on the Compressor Performance

Many studies have shown that the main effect of compressor fouling is a reduction of mass flow, and that the percent reduction increases with operational speed (Boyce et al., 1994), (Saravanamuttoo H. a., 1997), (Saravanamuttoo, H.I.H.; Muir, D.E.; Marshall, D.J., 1988), (Saravanamuttoo, H.I.H. and Aker, G.F., 1989), (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985), (Saravanamuttoo & Seddigh, 1990), (Zaba, 1980). Fouling also reduces the compressor pressure ratio PR, and adiabatic efficiency, η_c , causing a decrease in gas turbine power output and an increase in heat rate (Hill, 1992), (Saravanamuttoo, H.I.H.; Muir, D.E.; Marshall, D.J., 1988), (Zaba, 1980). Fouling reduces the operating range (on the compressor map) and shifts the operating point to lower mass flow rates at a given corrected speed. Fouling also decreases the specific work, w, which results in a further drop in power output. Roughness makes the constant speed lines steeper on the compressor map, thus the range of mass flow rates covered by the characteristic curves decreases. The attainable pressure ratios are reduced (stall line drops) (Bammert K. a., 1980), (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985).

For a single-shaft gas turbine experiencing typical compressor fouling in the front stages, the percent reduction in air inlet mass flow is greater than the percent change in compressor pressure ratio, which is in turn greater than the percent decrease in compressor adiabatic efficiency. For any gas turbine undergoing front stage fouling, the percent decrease in airflow is typically 2-3 times greater than the percent decrease in efficiency (Boyce M. e., 1993), (Boyce M. , 1995), (Hill, 1992), (Kurz, Rainer; Burn, Klaus, 2000) and (Saravanamuttoo, H.I.H.; Muir, D.E.; Marshall, D.J., 1988).

2.2.1 Compressor Performance Degradation

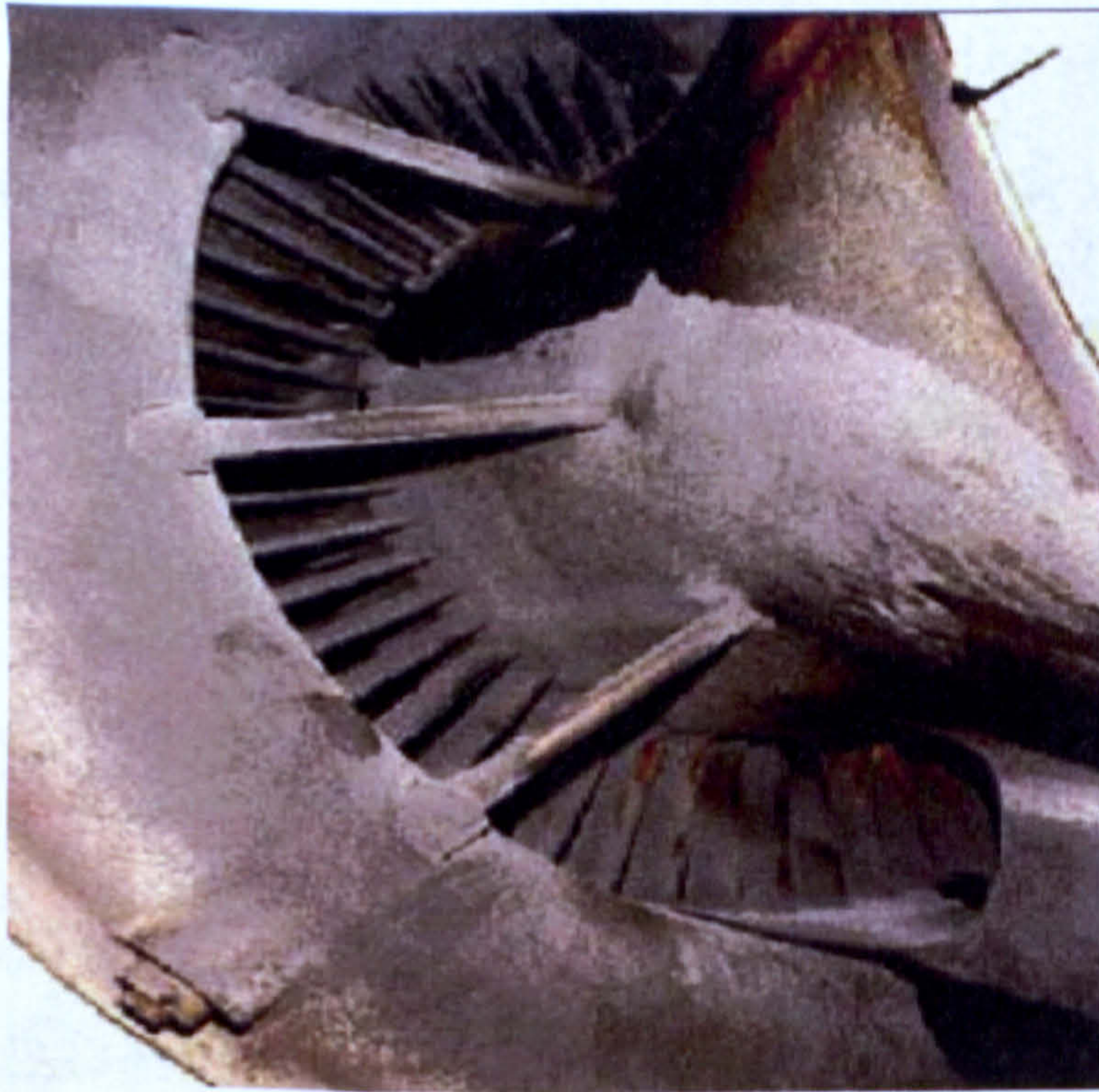


Figure 2-6: Example of heavy dust fouling in axial compressor (Syverud E. , 2007)

The increasing of fuel cost and the more request of electric power on the energy market pushed the owners of power plants to look for more power and reliability from their plant. The gas turbine plants nowadays have the need to control the performance of the engine, to define the performance trend during the normal operation and to try to define the health condition of each component. In order to avoid failures of power plant component, the gas turbine monitoring became an important topic of research for many companies and universities. The monitoring of compressor condition is important research to increase the reliability of this component and to improve the knowledge of compressor maintenance systems.

The state of a gas turbine compressor is affected by the environmental conditions of the site. With increasing operating time, degradation of the compressor can be registered in terms of reduced performance. Apart from abrasion and erosion of blade surfaces, compressor fouling is the main cause of the reduction of compressor efficiency and compressor inlet air mass flow (Schneider, Demircioglu, & Franco, 2009). During its operation, the modern gas turbine

ingests homogenous amounts of small particles along with the air it sucks in. For instance, with airborne foulants running at 10 ppm in the inlet air, a GE Frame 9 FA gas turbine would ingest 225 tons of dirt in a given year (Kalyanaraman, 2009).

The degradation of the gas turbine compressor has a direct influence on the gas turbine power plant efficiency and power. The power consumption of the compressor of a heavy duty gas turbine amounts to 50% of the turbine power (Kalyanaraman, 2009), so a little reduction of compressor efficiency is equivalent a big reduction of electric power generated.

Fouling is defined as degradation of flow capacity and efficiency caused by adherence of particular contaminants to the gas turbine engine airfoil and annulus surfaces (Syverud E. , 2007). The kinds of pollutant are: hydrocarbons, oil, lack, coal, salt, ash, fertilizing, insect, pollens, sand, and spores as shown in (Figure 2-7). Other compressor fouling is the internal gas turbine oil leaks; leakage from the front bearing of the axial compressor is a common cause. Oil leaks combined with dirt ingestion causes serious fouling problems (Figure 2-9).

The result of fouling is the building up of material on the compressor components, whose effects are:

- abrasion of rotating surface and material corrosion (if particles are above 10 to 20 μm) (Meher-Homji C. C., 2009),
- wet corrosion, caused by atmospheric interactions between humidity and hydrous acid, salt and other corrosive substances (Syverud E. B., 2007),
- high temperature corrosion, caused by fuel contamination, sulfates, metals and metal oxides,
- change in the shape and of the inlet angle of the airfoil,
- increases surface roughness,
- reduction of mass flow rate, caused by reduction of effective flow area, and then a reduction of mass flow coefficient Φ (Syverud E. a., 2006),
- variation of air-stream deviation angle,
- variation of speed profile and thickness of boundary layer (Syverud E. a., 2006),
- air stream 3D separation.
- in some cases, blade natural frequencies can be affected by the increase in mass due to dirt build up. Excessive dirt on the blades can also cause imbalance, and a consequent increase in running speed vibration (Diakunchak I. S., 1991).

- partial blockage of cooling passages of hot section stators and blades can be caused by fine particle size foulants (typically less than $5\ \mu\text{m}$). Cement dust, coal dust and fly ash can be responsible for this problem (Diakunchak I. S., 1991).

In other words, the effect of compressor fouling is a drop in airflow, compressor ratio and compressor isentropic efficiency, which results in a “re-matching” of the gas turbine and compressor causing a drop in power output and thermal efficiency, as shown in Figure 2-8. In extreme cases, fouling can also result in surge problems, as it tends to move the compressor surge line to the right; i.e. toward the operating line (Meher-Homji C. C., 2009).

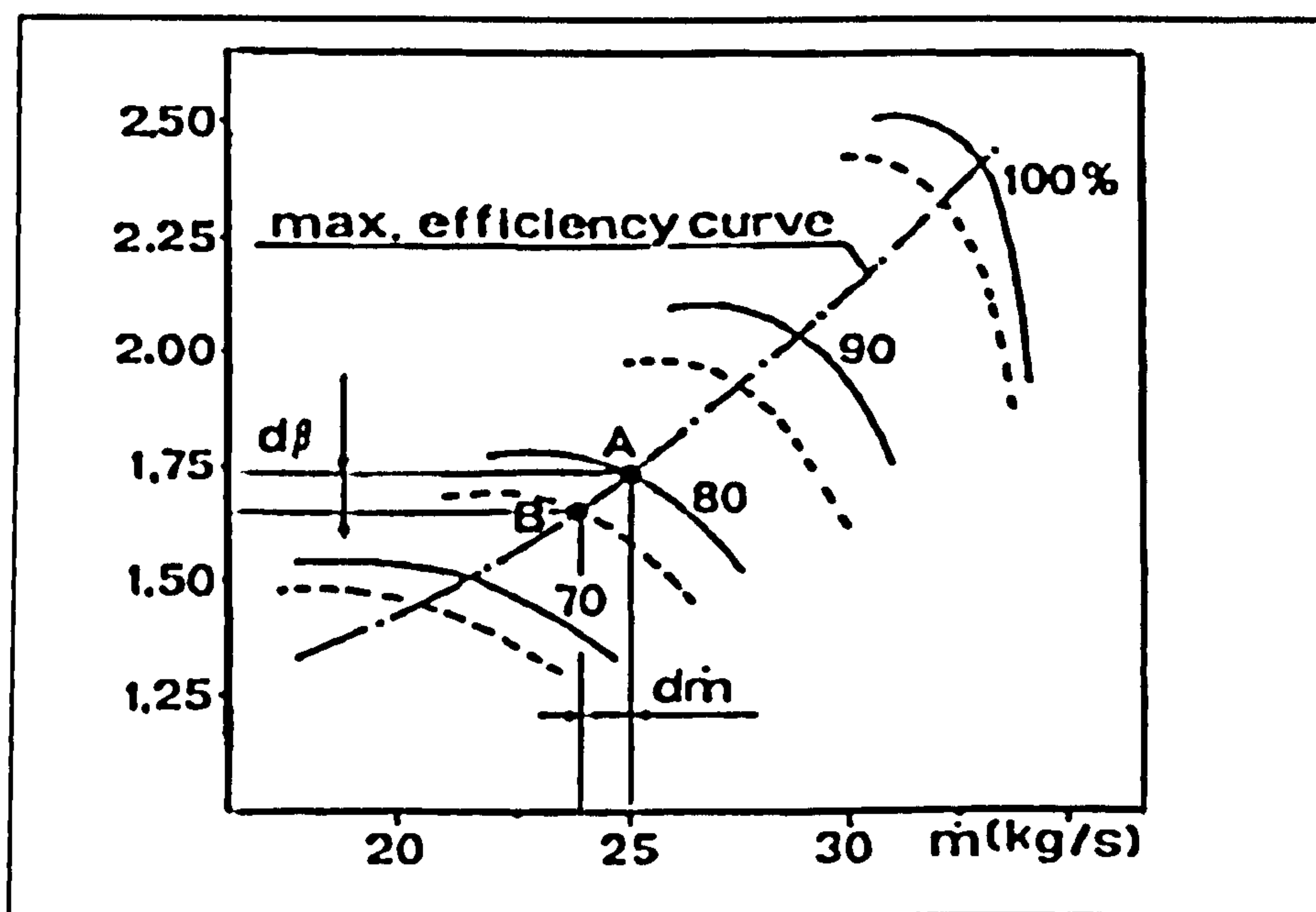


Figure 2-7: Compressor curve. Ideal performance in continuous line, fouled condition in dashed curve (Massardo, 1990)

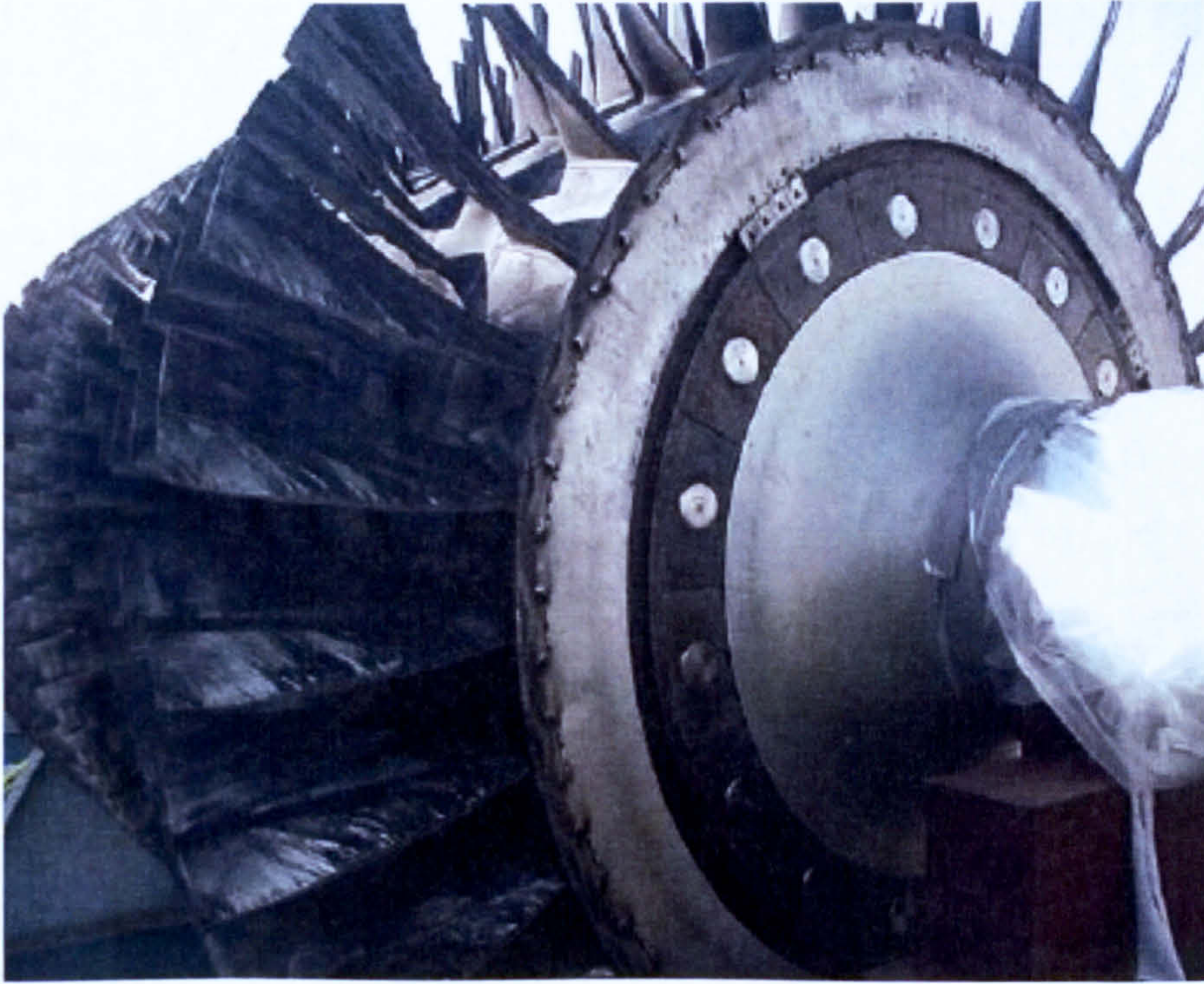


Figure 2-8: Example of oily deposits on axial compressor blades from No. 1 bearing oil leakage on a large heavy duty gas turbine (Meher-Homji C. C., 2009).

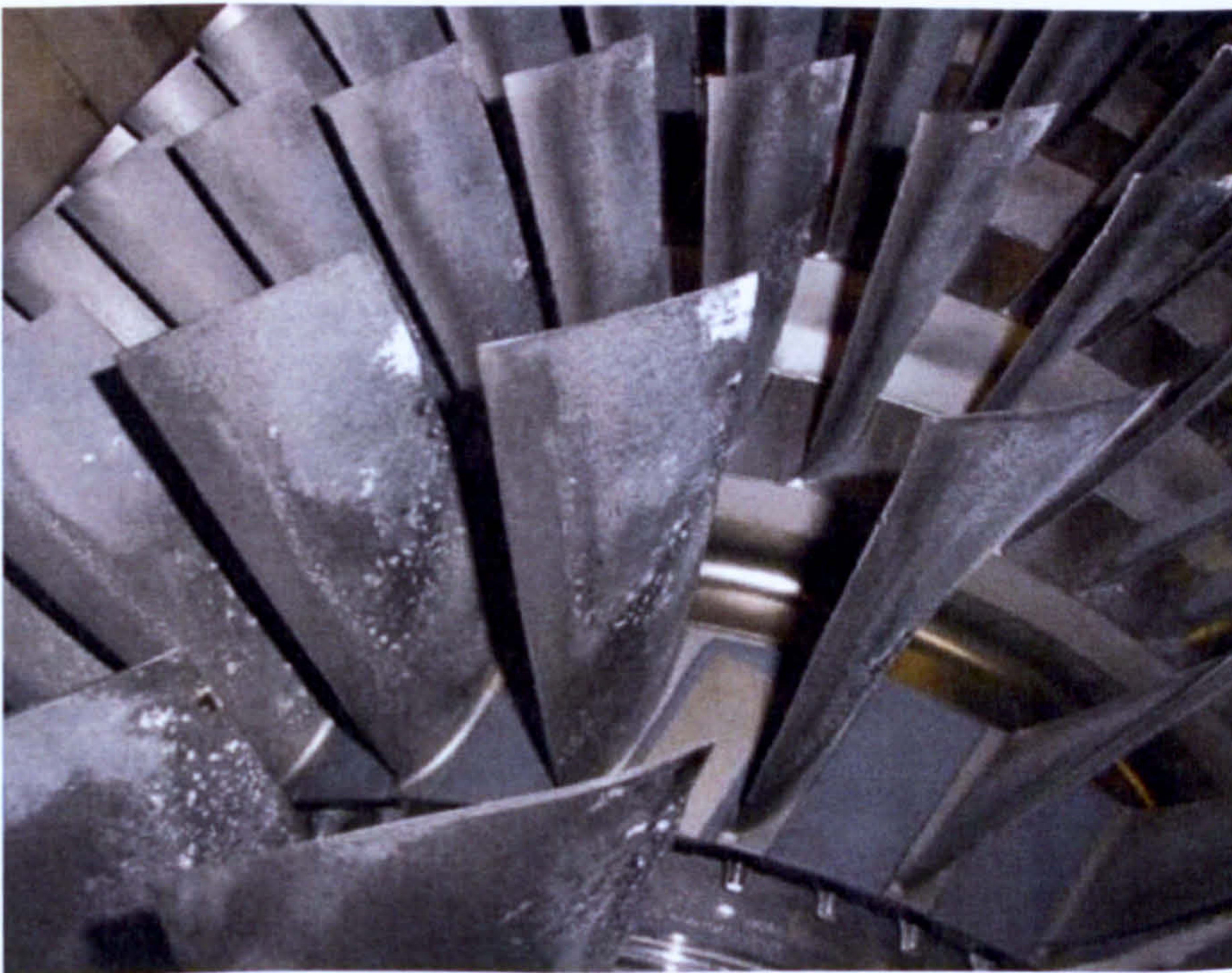


Figure 2-9: Salt deposits on compressor blades (Meher-Homji C. C., 2009).

2.2.2 Salt or Soluble Dust Deterioration

The marine environment may be cause of specific type of fouling, mainly due to salt deposits. A comprehensive study on this topic is presented in (Syverud E. B., 2007). Such a work is based on hypothesis of dust crystalline distribution that doesn't cause any permanent damage to the geometry, for example leading or trailing edge bluntness, chord changes, or increased tip clearances (Figure 2-11). A quasi-one-dimensional multi stage model was used. No adjustments were made to take off-design effects into account, therefore results and correlations are valid close to the design point. The main conclusions relevant to this review are:

- Bigger salt deposits were found in the stator rather than in the rotor.
- The stages with much fouling were the initial ones, in particular the 2nd, 3rd and 4th.
- An off-shore plant of 22 MW electric power can ingest 20 g of salt water per day, in particles above 2 μm size. With 6 g per day the performance might reduce, and with 30 g per day the loss becomes 10% of the total power output per day.
- A good washing depends on the fouling solubility, how much water evaporates, the wet surfaces and the wetting time (Syverud, E.; Brekke, O.; Bakken, L.E., 2007).

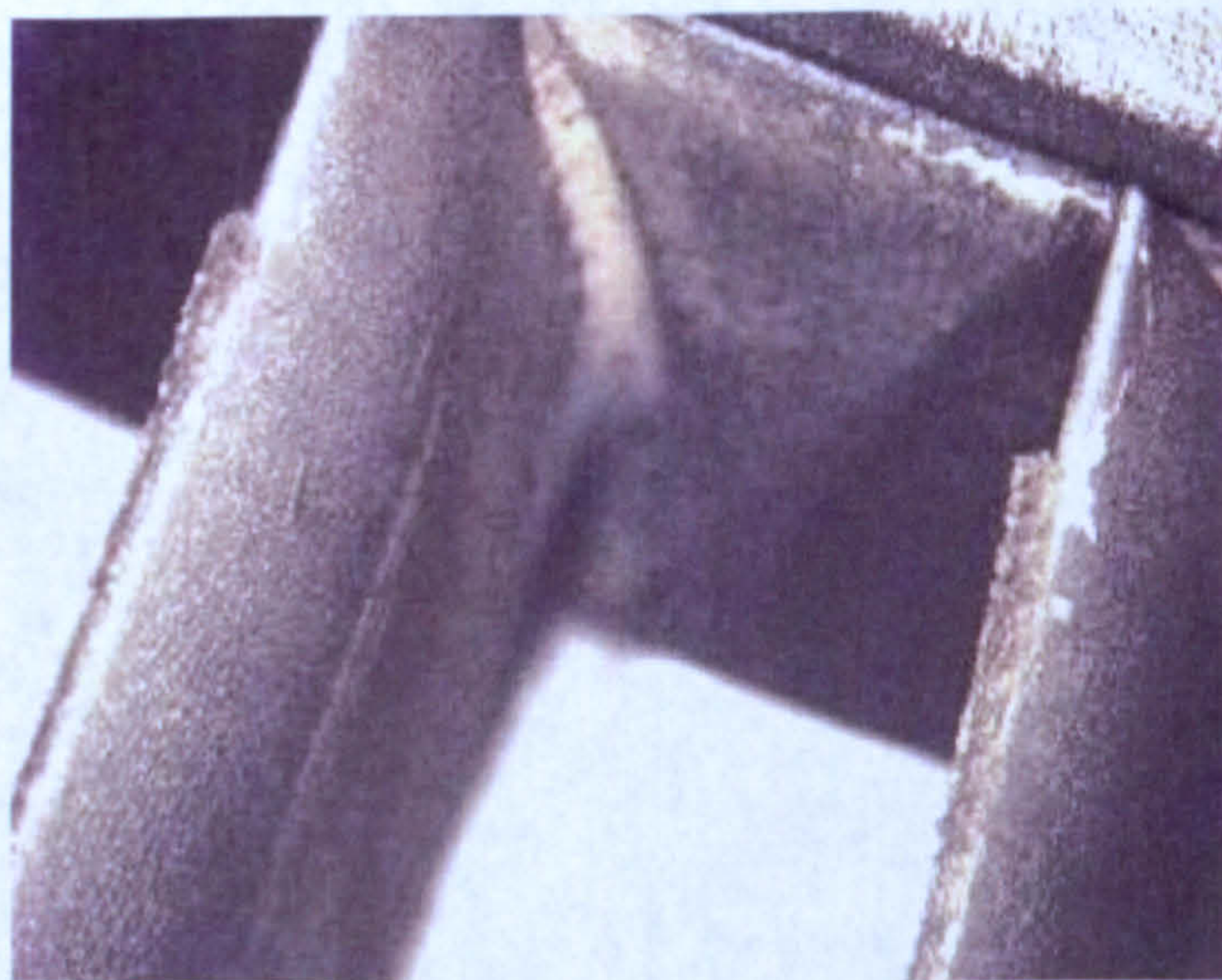


Figure 2-10: Salt deposits at the leading edge of the second-stage stator vanes. Hub is up in this image (Syverud E. a., 2006).

2.2.3 Degradation Effects on Industrial Gas Turbines

Several mechanisms cause the degradation of gas turbines. Fouling is caused by the adherence of particles to airfoils and annulus surfaces. Hot corrosion is the loss or deterioration of material from components exposed to hot combustion gases, caused by chemical reactions between the component and certain contaminants. Corrosion is caused both by inlet air contaminants and by fuel and combustion derived contaminants (Kurz, Rainer, 2009). Erosion is the abrasive removal of material from the flow path by hard or incompressible particles impinging on flow surfaces. Abrasion is caused when a rotating surface rubs on a stationary surface. Damage may also be caused by foreign objects striking the flow path components. While some of these effects can be reversed by cleaning or washing the engine, others require the adjustment, repair, or replacement of components. Industrial engines, assuming an appropriate air filtration system is installed, are probably more subject to fouling caused by smaller particles, corrosion, and possibly lube oil. The exception may be engines subject to water injection in the compressor inlet (Brun, 2005), where an incorrectly sized system can generate water droplets of a size sufficient to cause blade erosion.

2.2.4 Degradation of Components

Three major effects determine the performance deterioration of the gas turbine compressor: increased tip clearances, changes in airfoil geometry, and changes in airfoil surface quality. While the first two effects typically lead to non-recoverable degradation, the latter effect can be at least partially reversed by washing the compressor (Stalder J. P., 1998).

Typically, a degraded compressor also will have a reduced surge or stall margin (Spakovszky, 1999), (Brun, 2005) and (Graf, 1998) showed data for an axial compressor, where increased clearances caused reduced surge margin and reduced efficiency. In this case, the clearance was increased from 2.9% (design value) to 4.3%. This lead to an increase in surge flow coefficient of about 20%, a reduction in design pressure coefficient of 12%, and a loss of design point efficiency of 2.5 points. Notably, the loss in pressure coefficient became negligibly closer to choke flow. Similar results are reported by (Smith, G. D. J., and Cumpsty, N. A., 1984) where an increase in the clearance from 1% to 3.5% reduced the pressure coefficient by 9% (Frith, 1992) tested a helicopter gas turbine with compressor

blades cropped to simulate increased clearances. A 3% crop on the axial compressor stages reduced airflow by 4.6% and pressure ratio by 3%. The compressor discharge temperature is reported to remain unchanged, which indicates a reduction in compressor efficiency by about 2.5%.



Figure 2-11: Compressor Fouling Due to Erosion in a desert environment (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The compressor pressure ratio and the compressor flow are not independent, and the compressor efficiency is determined by the resulting compressor operating point. Increases in tip clearance as well as deteriorated airfoils will shift the pressure ratio-flow relationship for a given operating speed to lower flow rates, as well as to lower efficiencies (Figure 2-12).

The combustion system is not likely to be the direct cause for performance deterioration. The combustion efficiency will usually not decrease, except for severe cases of combustor distress. However, deterioration could potentially lead to a variation in the combustor exit temperature profile (Figure 2-13).

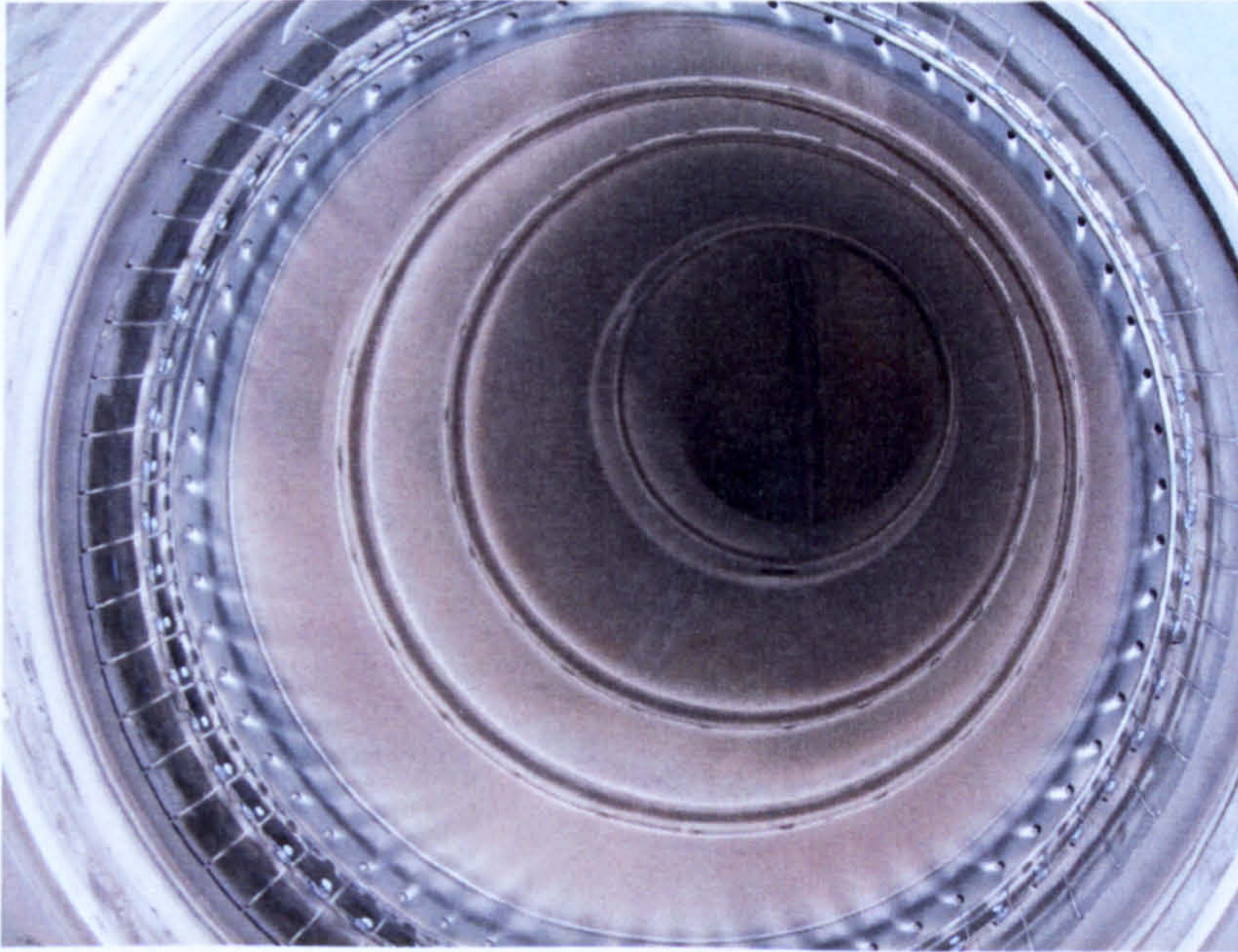


Figure 2-12: Combustion Chamber (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

Just as the compressor section, the turbine section experiences the following issues that result in degradation: increased tip clearances, changes in airfoil geometry, and changes in airfoil surface quality. Maintenance of tip clearances is, in particular, a problem in the turbine section, due to the extreme changes in temperature between a cold engine and an engine accelerating to full load. In many designs, the stationary components expand at a different rate than rotating components. Many new turbine designs use abradable seals to minimize these clearances (Figure 2-14).

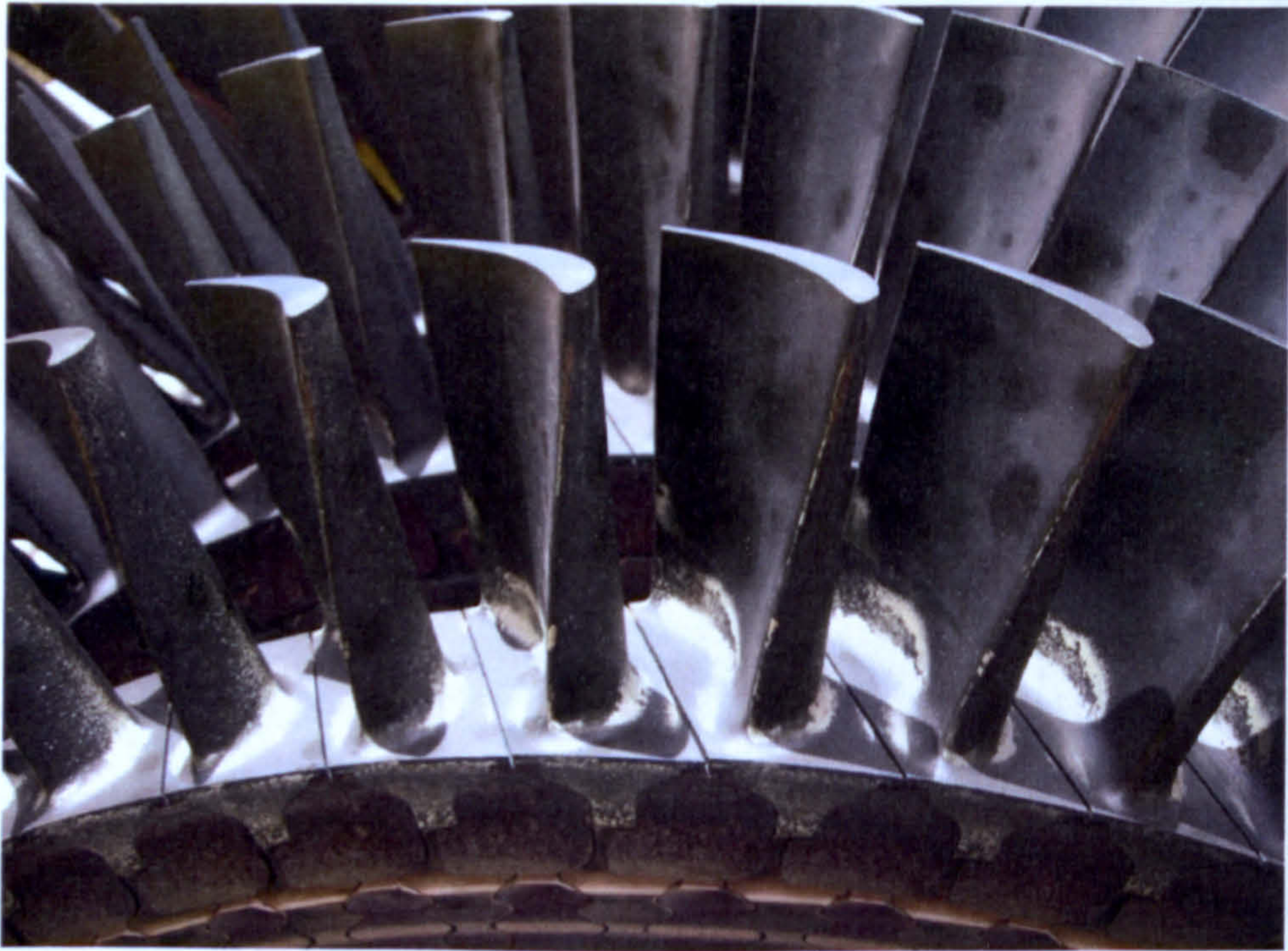


Figure 2-13: Turbine Blades Erosion for Sulzer Type 7 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

However, the most severe case, which is usually after a hot restart, will determine the minimum clearance for the engine. In addition to a reduced efficiency, added clearances will also increase the axial flow blockage, and thus will cause reduced flow and increased velocities in the main flow. (Radtke, F., and Dibelius, G., 1980) reported a reduction in efficiency of a multistage turbine by 0.6% when they increased the radial clearances from 0.5% of the blade height at the rotors and 0.4% of the blade height at the stators to 0.8% of the blade height at rotors and stators.

Corrosion tends to alter the flow path in two ways. It increases the surface roughness, but it may also remove material, in particular, at the leading edges and the trailing edge of the airfoils. Especially the turbine nozzles, operating at or near choked conditions, are very sensitive to changes in the flow area. Furthermore, changes in the flow capacity of the turbine section will subsequently alter the operating points for the engine compressor. Increased surface roughness causes thicker boundary layers on the blades and sidewalls, and thus may reduce the flow capacity, especially near choking conditions. (Boyle, 1994) found for a two stage turbine efficiency losses of 2.5% for a $10.2\ \mu\text{m}$ surface roughness when compared with smooth blades. He also found that the most pronounced differences appear at the optimum

operating point at the turbine, whereas the far off-optimum efficiency was almost the same for rough and smooth blades. It should be noted, that the losses due to clearances were in the same order of magnitude as the profile losses. However, if the degradation of the turbine section leads to material removal, especially in the nozzle area, it can be seen the opposite effect: The flow capacity increases for any given pressure ratio. Because the flow capacity of any nozzle is limited by the effective throat area, erosion of the trailing edge causes the throat area to increase and the exit flow angle to become more axial. This means a reduction of turning in the stator and the rotor, which will lead to reduced work extraction for this stage and to an increased flow capacity. Since the turbine nozzles constitute a flow restriction, any change in the flow capacity of the turbine section will also impact the operating points of the engine compressor.

- **Air Filtration Systems**

Fouling of inlet filters occurs progressively over time. This leads to increased pressure drop in the inlet system, and as a result, reduced engine power and efficiency. Figure 2-15 shows the relative impact of the pressure loss in the inlet system on power and efficiency. Self cleaning filters, where appropriate, or changing of filter pads or cartridges can reverse this pressure loss. It must be noted that air filtration systems are always a compromise between filtration effectiveness, pressure loss, and size or cost of the system. Also, the filtration system has to be appropriate for the type of contamination that is expected. Some types of filters are very effective for small particle sizes, some are specifically designed for high dust loads, and others are effective in keeping droplets (with potentially dissolved contaminants) out of the engine.

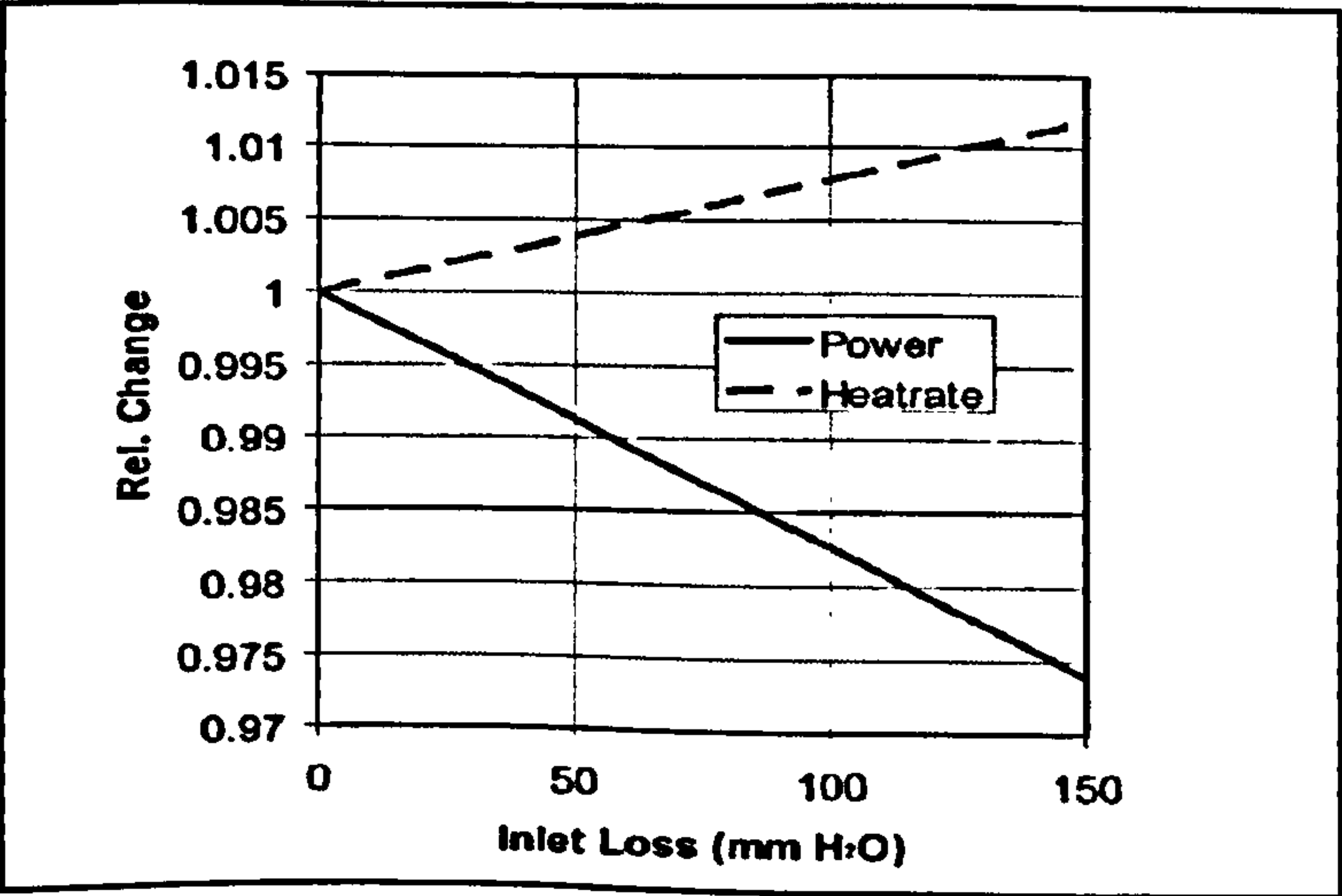


Figure 2-14: Impact of inlet pressure loss on engine power and heat rate (Kurz & Brun, 2009)

- **Reduced Compressor Efficiency**

Compressor degradation in the overall engine environment will yield different results for single and two shaft engines. Due to the fixed speed of the single shaft machine in combination with a usually choked turbine nozzle, loss of compressor efficiency will mostly affect compressor pressure ratio, but only to a very limited degree the flow through the machine. A two shaft engine with a compressor with reduced efficiency will exhibit significant changes in flow and pressure ratio. The reduction in flow was observed in several studies (Syverrud E. B., 2005), (Millsaps, 2004). According to (Stalder J. P., 1998); the pressure ratio-flow relationship of the gas turbine compressor operating points remains unchanged because it is determined by the turbine section, but the engine will have to run faster, and the compressor will consume more power for any point on the pressure-flow line once it deteriorates.

The impact of reduced compressor efficiency on full load power and heat rate (HR) of a two shaft engine depends on the ambient temperature and seems to be more severe at warm ambient temperatures (Figures. 2-16 and 2-17). For a temperature topped two shaft engine, the gas producer speed is reduced for the degraded engine (because for a given pressure and flow, the compressor now consumes more power). For single shaft engines, where the compressor speed is kept constant, the impact on power and heat rate is almost the same for all ambient conditions. Also, the reduction in performance is less pronounced than for a two shaft engine. Some of the impact of degradation can be reduced for two shaft engines by re-adjusting the controls to allow the engine to reach full firing temperature.

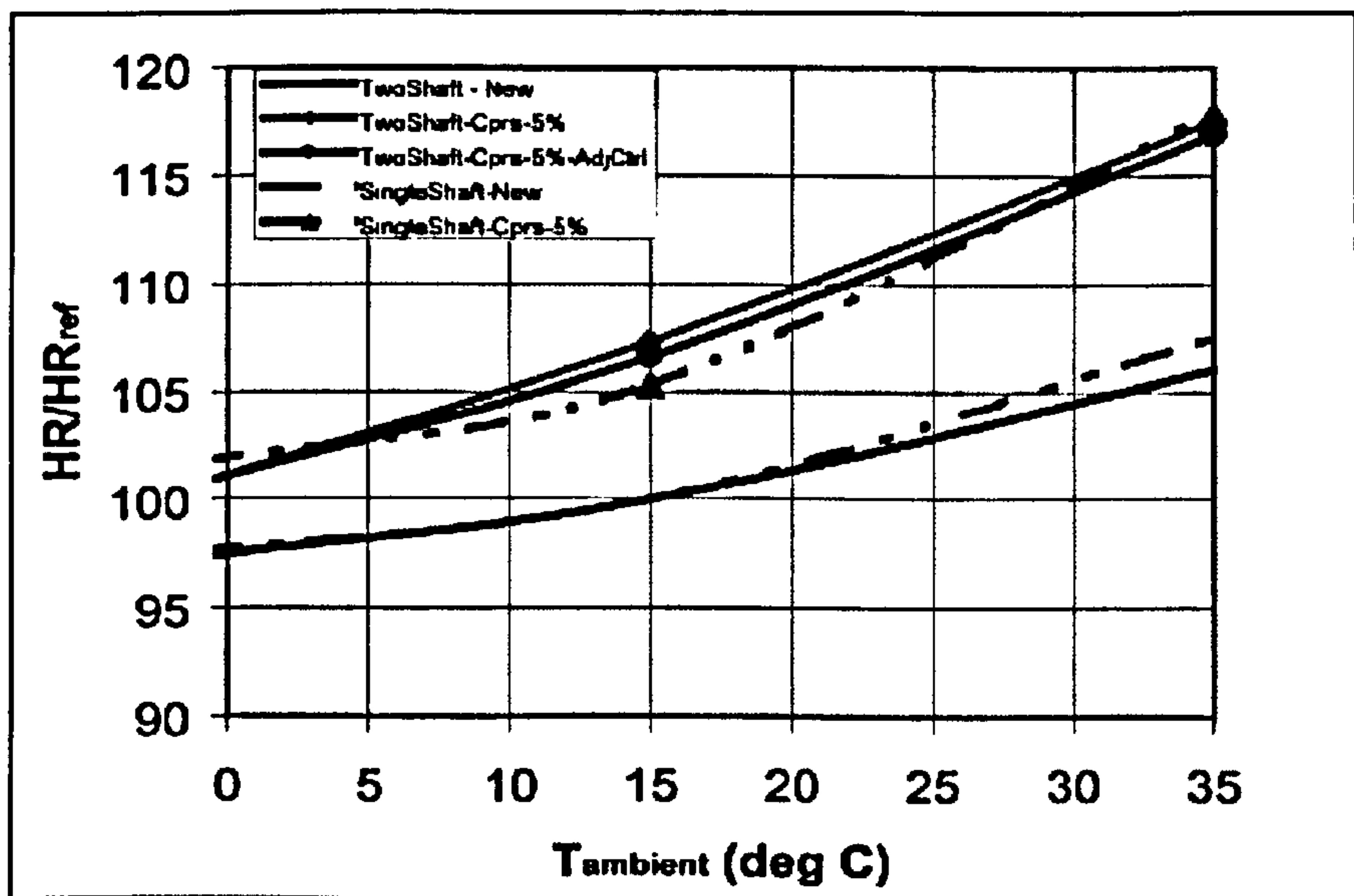


Figure 2-15: Impact of reduced compressor efficiency (reduction of 5%) on the full load heat rate of a single shaft and a two shaft gas turbine for a range of ambient temperatures (Kurz & Brun, 2009)

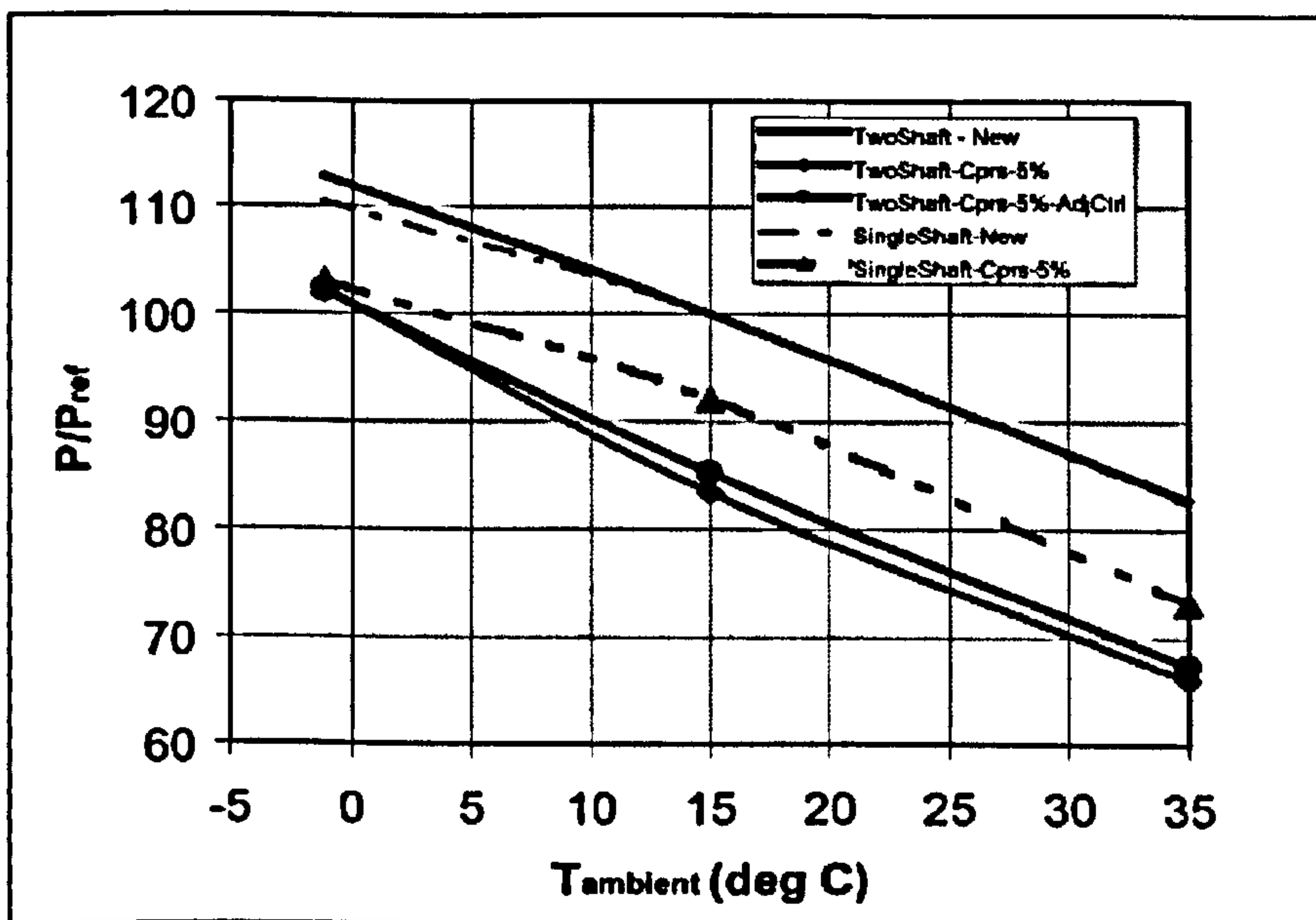


Figure 2-16: Impact of reduced compressor efficiency (reduction of 5%) on the full load power of a single shaft and a two shaft gas turbine for a range of ambient temperatures (Kurz & Brun, 2009)

Both (Tarabrin, 1998) and (Meher-Homji C. B., 2004) found that twin spool and three spool engines seem to be particularly susceptible to performance deterioration. For all types, the relative degradation in power is far more pronounced than the degradation in heat rate for a two shaft engine.

For an engine with standard combustion, at constant load, a reduction in compressor efficiency will cause the gas generator speed, and with it the compressor discharges pressure

and the airflow, to drop. At the same time, the firing temperature increases (to maintain the required power output) as well as the heat rate (Figure 2-18).

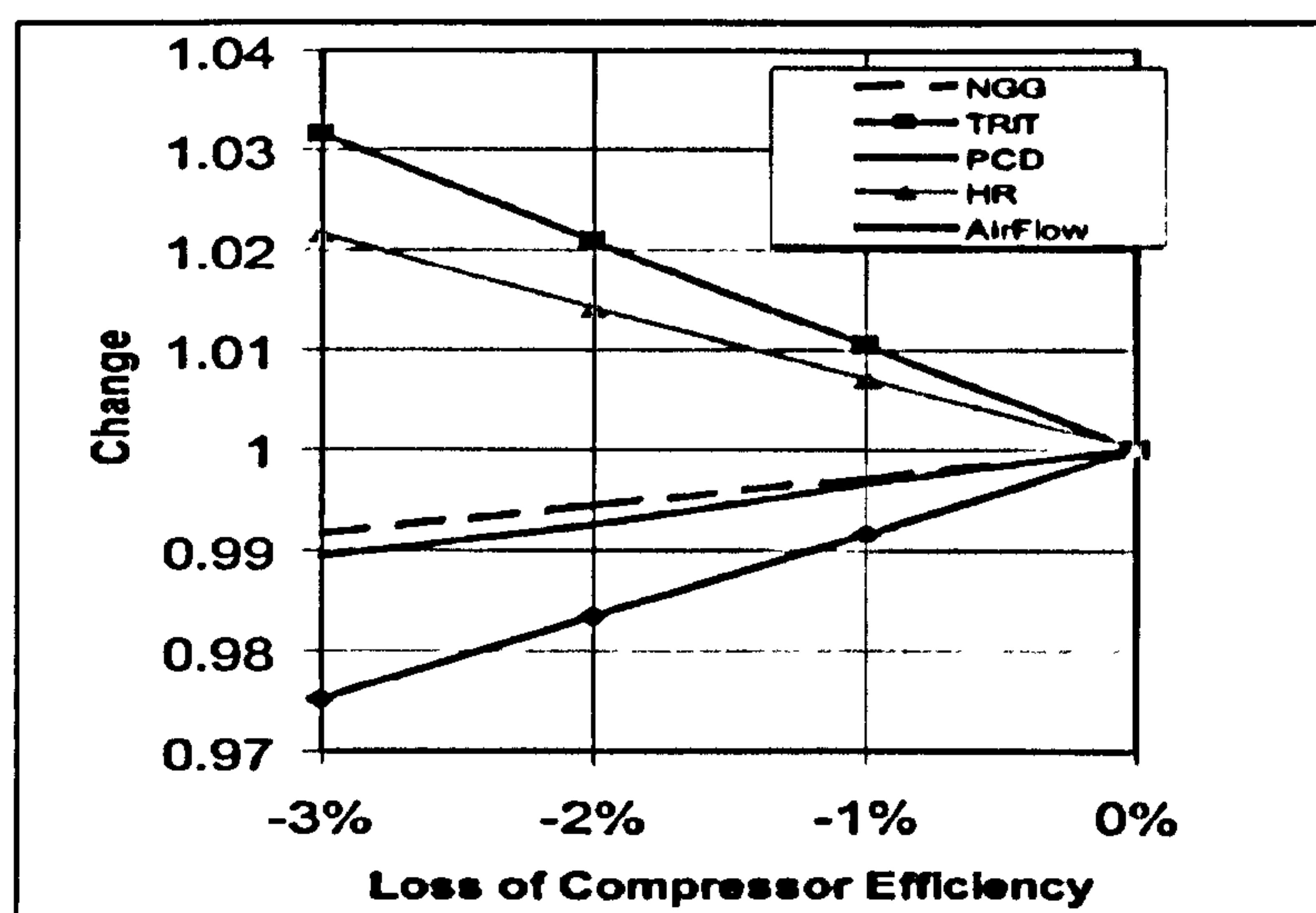


Figure 2-17: Impact of reduced compressor efficiency (at constant gas turbine power) on gas generator speed (NGG), firing temperature (turbine rotor inlet temperature (TRIT)), compressor discharge pressure (CDP), heat rate, and airflow for a two shaft engine (Kurz & Brun, 2009)

(Syverrud E. B., 2005) performed tests on a J-85 gas turbine, where they deteriorated the compressor performance by spraying salt water in the engine inlet. The deposits caused increased surface roughness on the compressor airfoils. They found that the majority of deposits occur on the first stage and become insignificant after about the fourth compressor stage. The deterioration shifted the compressor operating line to both a lower flow rate and a lower pressure ratio, which confirms the simulation results. The data show further that the degradation not only leads to reduced stage performance, but also to additional losses because individual stages no longer operate at their design flow coefficients. The operating points of the deteriorated engine were consistently at lower flow coefficients than for the clean engine. This also leads to additional efficiency reductions due to the movement of the stage operating points away from the stage design point.

(Syverrud E. B., 2005) were able to show the direct impact of the blade surface roughness on added profile losses and the increase in sidewall boundary layers due to the deposits on the deteriorated compressor performance. Their data also show that the compressor condition cannot be separated from the turbine section of the engine, since the turbine flow capacity determines the operating point of the compressor.

- **Reduced Compressor Flow Capacity**

Next, we consider the impact of reduced compressor flow capacity, which can be the result of fouling or increased clearances. (Spakovszky, 1999) and (MacLeod, 1991) related increased rotor clearances to an increase in flow blockage, thus reduced compressor flow capacity. (Khalid, 1998) investigated the effect of component deterioration on overall engine performance on a single shaft turboprop engine (thus operating at a constant speed). Besides the expected loss in engine efficiency, they found a significant reduction in compressor flow capacity with increased compressor clearances, but no significant change in compressor efficiency.

It has been found that the same level of compressor flow blockage leads to more power degradation in a two shaft engine than in a single shaft engine. It is interesting that both for single shaft and for two shaft engines there is a very small increase in heat rate due to the compressor blockage at higher ambient temperatures. Only for temperatures below 10 °C, the heat rate starts to increase slightly at about the same rate for single shaft and two shaft machines. Also, the loss in available Power is more pronounced at low ambient temperatures. This is probably due to higher Mach numbers at low ambient temperatures, which make the compressor more sensitive to changes in flow capacity. Compressor deterioration by itself will usually cause higher power losses than losses in heat rate, because a higher compressor exit temperature (due to lower efficiency) at a fixed firing temperature will reduce the possible fuel flow. An operating point at part load (i.e., below maximum firing temperature and below maximum gas generator speed) can still be maintained with a degraded engine, albeit at a lower firing temperature and a higher gas generator speed than for the new condition. The relative loss in efficiency is significantly lower than for an engine at full load for the same amount of degradation. Airflow and compressor discharge pressure change very little.

- **Changes in Gas Generator Turbine Flow Capacity**

Degradation effects on the flow capacity of the gas generator turbine can either increase or reduce the flow capacity: The reduction in the available flow path is either due to added surface roughness or eroded leading edges, both leading to thicker boundary layers and thus a reduced effective throat area (Kurz R. , 1995). The throat area increases when the nozzle trailing edges erode, thus increasing the flow capacity of the turbine. Both for single and two shaft engines, changes in gas generator flow capacity lead to a pronounced change in the

operating points of the compressor. Increased gas generator flow capacity moves the engine compressor operating points further into the choke region and thus leads to lower compressor efficiency, whereas a reduced flow area has the opposite effect. The throat area in a gas turbine has a larger detrimental effect than a reduction in flow area.

- **Reduced Gas Generator Turbine Efficiency**

Now we will review the case of an engine with reduced gas generator turbine efficiency due to fouling, or simply due to the fact that a change in nozzle geometry has altered the operating point of the turbine section. Rather severe degradation of the gas producer turbine nozzle occurs when material is removed due to erosion or corrosion. In this case, the turbine efficiency will drop (Kurz R. , 1995), while its flow capacity increases. Both full load power and heat rate deteriorate, but again, the loss in power is more pronounced than the increase in heat rate. We also find a stronger deterioration at high ambient temperatures.

2.2.5 Influence of Components on Engine Performance

The performance of gas turbine engine components (particularly compressor and turbine) may affect overall engine performance significantly. According to Pilidis et al. (2001), compressor degradation has a smaller effect on engine overall efficiency than degradations in other components.

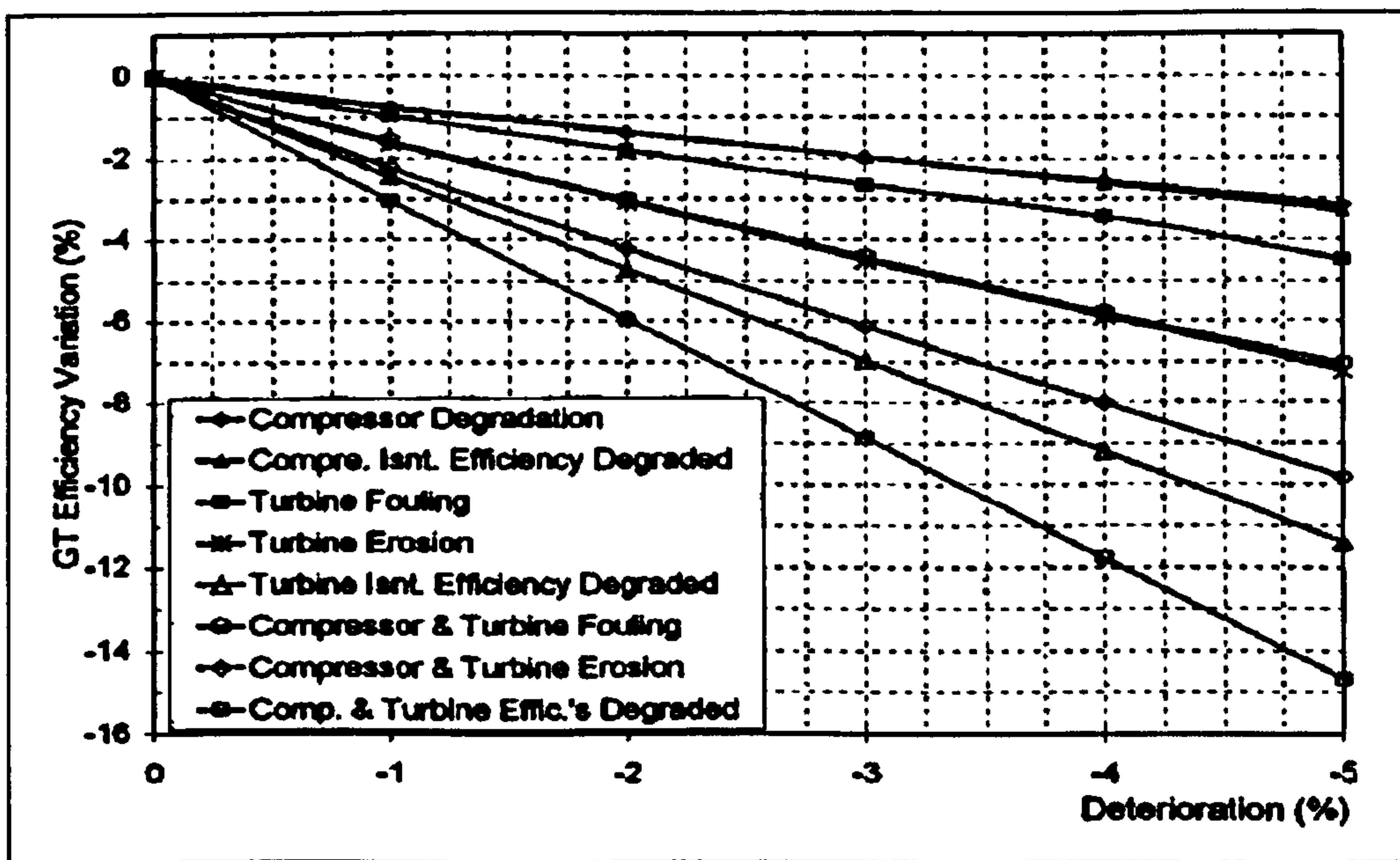


Figure 2-18: Effect of different component degradations on the engine thermal efficiency (Zwebek & Pilidis, 2001)

Figure 2-19: shows that the loss of isentropic efficiency due to turbine fouling has a stronger effect on engine performance than that of the compressor. Simultaneous compressor and turbine efficiency degradations shown in Figure 2-19 (Zwebek & Pilidis, 2001) have the most severe effect on engine thermal efficiency.

2.2.6 Indication of Compressor Fouling

As discussed earlier, compressor fouling normally occurs due to foreign deposits on the airfoils. In petro-chemical operations, this is the result of the oil and other hydrocarbons in the atmosphere caused by neighbouring process plant. An example of compressor fouling is shown in Figure 2-20.

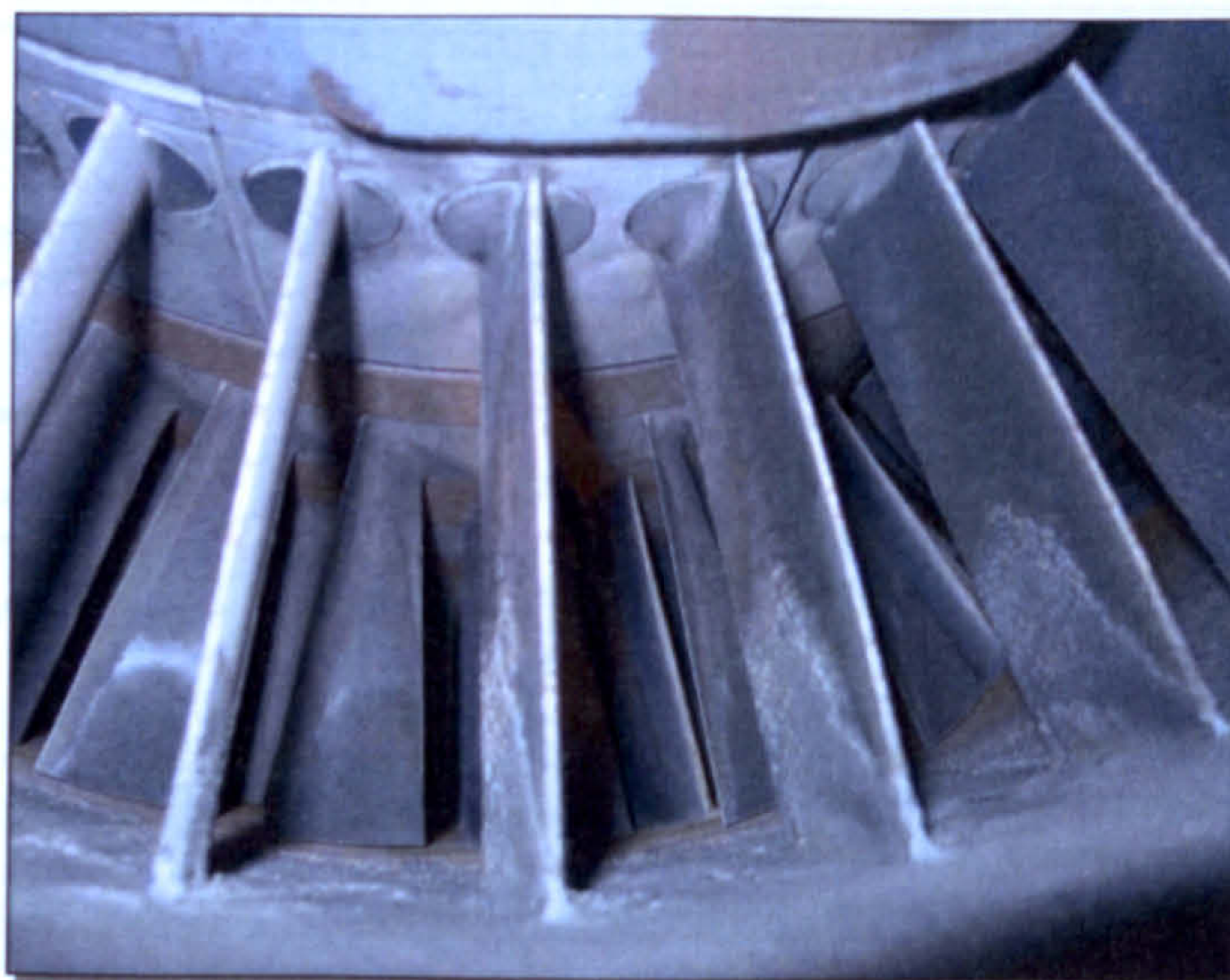


Figure 2-19: the hydrocarbon deposit in compressor IGV and blades (Courtesy of Arabian Gulf Oil Company (AGOCO), Nafora Field, 2008)

In any event, compressor fouling is indicated by a drop in compressor discharge pressure (CDP). This will also manifest itself as a reduction in load capacity at a constant ambient condition and compressor operating condition. A compressor efficiency drop of 2 percent is common as a result of fouling.

For a field engineer or operator, compressor fouling may be indicated by a drop in compressor discharge pressure at constant speed and load. Another indication of fouling may be the reduction in load capacity of the compressor. However, small drop of compressor discharge pressure, such as 1 psig decrease in compressor discharge pressure, is not easy to detect with field instrumentation.

There are many methods that have been established to determine performance deterioration due to compressor fouling, such as those described by I.S. Diakunchak, (1991). One of these methods is the monitoring of combustion shell pressure because the combustion shell pressure is one of the parameters used by the control system to keep the engine running at full load. However, the accuracy of the method is affected by several factors such as ambient temperature, ambient pressure, intake pressure loss, and output power.

2.3 The Effect of Surface Roughness

2.3.1 Definition of Roughness

Fluid moving across a surface is subjected to resistance if the surface is rough. The resistance due to roughness depends on the density distribution of the roughness elements (number per unit area), the shape and height of the roughness elements, and their geometrical arrangement over the surface (Cumpsty, 1989).

The shape and density distribution of the protrusions will vary significantly from one practical application to another. Therefore, it is convenient to correlate a given mean roughness height, k , with a standard roughness. It is common to adopt (Cumpsty, 1989) equivalent sand roughness, k_s . Roughness element height, k , is converted to equivalent sand roughness, k_s , by the equation:

$$5.7 \log \left(k_s / k \right) = 8.5 - B \quad (1)$$

Where B is a dimensionless roughness function based on the Reynolds number of the roughness elements. For a hydrodynamically rough surface, $B = 8.5$, so $k = k_s$. For a hydrodynamically smooth surface, B is typically less than 8.5, so $k_s > k$.

2.3.2 Hydro-dynamically Smooth Versus Rough

Roughness will degrade performance if a blade is hydro dynamically rough. The Reynolds number of the roughness elements is calculated to estimate if a compressor blade is rough, using

$$Re_k = (Uk/\nu) \quad (2)$$

Where U is the free stream velocity, k is the roughness element height, and ν is the kinematic viscosity of air. If this Reynolds number is less than 100, the roughness peaks are contained within the viscous sublayer and the blade is hydro dynamically smooth. (Schlichting, 1987) gave the criterion for hydraulic smoothness on flat plates a value of Roughness Reynolds Number < 100 , based on equivalent sand grain roughness and free stream velocity.

$$Re_k < 100 \quad [C_f = f(Re)] \quad \rightarrow \quad \text{Smooth} \quad (3)$$

$$100 < Re_k < 1000 \quad [C_f = f(Re, k)] \rightarrow \text{Transitional} \quad (4)$$

$$Re_k > 1000 \quad [C_f = f(k)] \rightarrow \text{Fully Rough} \quad (5)$$

While this is strictly true for flat plates which have no pressure gradient, it also applies, at least approximately, for airfoils where $dP/dx \neq 0$. For typical parameters, such as $U=200$ m/s and $\nu = 1.5 \cdot 10^{-5} \text{ m}^2/\text{s}$, this means that if the blade has a surface finish of $1 \text{ } \mu\text{m}$ ($39 \text{ } \mu\text{in}$), it is very smooth. It is not beneficial to make airfoils any smoother than this.

2.3.3 Effect of Roughness on the Boundary Layer

The surface quality of the blading greatly affects the efficiency of energy conversion in a turbomachinery (Bammert K. a., 1980). Fouling causes an increase in surface roughness and changes the blade shape if a sufficiently thick layer is added. Added roughness, when the blade is hydro dynamically smooth, increases the momentum boundary layer thickness, δ_2 , and increases the risk of flow separation. When the tips of roughness elements protrude through the viscous sublayer in the turbulent regime, the skin friction, τ_w , is increased (Kurz, R. and Brun, K., 2000), and separation conditions are reached before the trailing edge of the chord.

Since the suction surface of an airfoil sees greater local velocities than the pressure surface, the suction side is more sensitive to roughness. The boundary layer and viscous sublayer are thinner on the suction surface, so roughness elements are more prone to create disturbances and “trip”, or transition the flow to turbulence early, particularly on the adverse pressure gradient part of the blade. Rotor blades are similarly most sensitive when roughness accumulates toward the leading edge.

2.3.4 Profile Loss

The total pressure loss produced by the blades in the cascades, away from the end wall, is often called profile loss (Cumpsty, 1989). For compressible flow, profile loss is defined as

$$\omega = \frac{P_{T2} - P_{T1}}{P_{T1} - P_1} \quad (6)$$

in total pressure across an airfoil divided by the difference between the upstream total and static pressure. For smooth blades in a cascade, profile loss, ϖ , is a function of blade profile, solidity, S , Mach number, M , and incidence, i . A small displacement of the flow separation point in the upstream direction significantly increases the losses (Bammert K. a., 1980). The wake region becomes more extensive and pronounced, leading to greater total pressure losses. The main influence of blade roughness appears around optimum incidence angles, while the “far-off-optimum performance” is hardly affected, as discussed by (Bammert K. a., 1980). It is clear that added roughness on the pressure side has a very small effect compared to the suction side, as discussed by (Kurz, R. and Brun, K., 2000). (Boyce M. e., 1993) found an increase in profile losses for NACA 65(12)06 compressor cascades from 2% ($\varpi = 0.02$) with $k_s/C = 0.3 \cdot 10^{-3}$, to 10% ($\varpi = 0.10$) with $k_s/C = 5 \cdot 10^{-3}$, Where k_s is the equivalent sand roughness and C is the chord.

2.4 Analysis of Gas Turbine Performance Deterioration

The ingestion of particles suspended in the air such as sand, dust, water droplets, ice, fly-ash, salt, etc., is the single most important factor causing performance deterioration in all types of gas turbines. Since gas turbines use roughly half a ton of air for each horsepower output for every 24 hours of operation, even if one part per million (ppm) enters the compressor, a 10,000 HP unit will ingest ten pounds of foreign material in a single day! With such a high rate of foreign particle ingestion, even a highly efficient filtering system can only mitigate but not eliminate the problem of gas turbine performance deterioration. Industrial and marine gas turbines foul due to continued operation in an environment that almost always contains dust particles, pollen, salt, and other impurities. The fouling problem is amplified if there are internal oil leaks near the blade surfaces. (Lakshminarasimha, 1994)

From operational, economic, and safety considerations, gas turbine performance deterioration has emerged as a very important topic of research. Fault classification, quantification, and identification are three essential steps in any turbine engine deterioration model development. Since gas turbines have been in use for over four decades, fault classification is based on a

significant base of reported operational problems. Generally, performance deterioration in gas turbines is due to:

1. Fouling: due to minute dust particles, pollen, salt spray and insects that are deposited on blade surfaces.
2. Erosion: of blade surfaces caused by particulate ingestion.
3. Tip clearance increase: of blade tips caused by particulate ingestion.
4. Water ingestion—during rain.
5. Foreign Object Damage (FOD): caused by hailstones, run way gravel, and bird ingestion (Lakshminarasimha, 1994).

2.4.1 Analysis of Axial Compressor Fouling

Fouling of compressors in gas turbine power plant is perhaps the most significant operational problem faced by gas turbine operators. Aerodynamically, an axial flow compressor is a very sensitive component, requiring close manufacturing tolerances on the blading for the baseline performance to be achieved. Fouling of the blade surfaces reduces flow through the compressor, compressor pressure ratio, and its efficiency. This causes a reduction in gas turbine performance, decreasing power available and increasing the heat rate. Figure 2-21 shows the effect of fouling in a large gas turbine unit ((Meher-Homji C. B., 1990). Reduction in performance in the beginning is gradual but an exponential reduction in performance of gas turbine follows if the compressor is not washed in time. Performance deterioration due to fouling is, for the most part, recoverable.

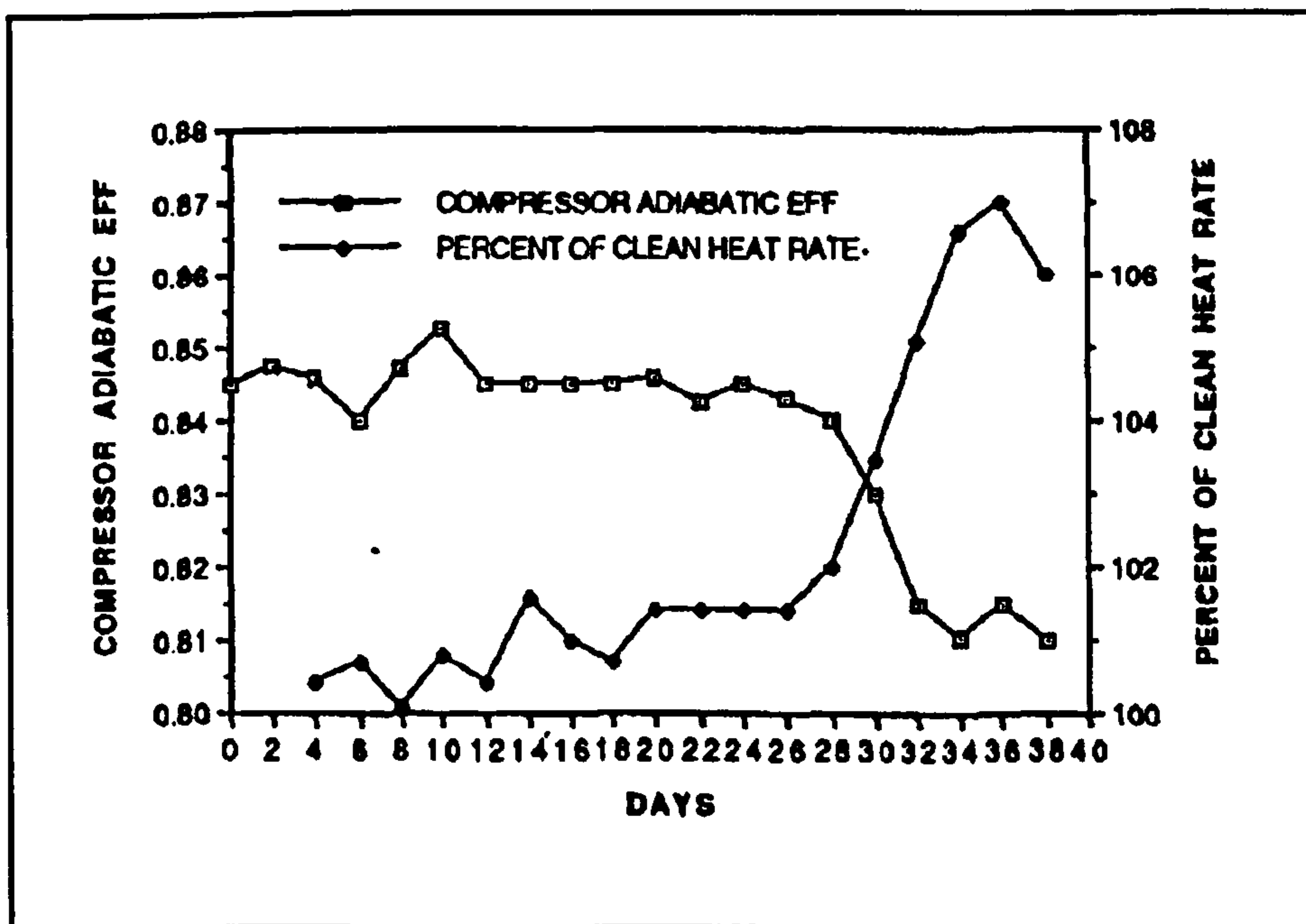


Figure 2-20: Effects of fouling on an industrial gas turbine (Lakshminarasimha, 1994)

The type of wash depends on the fouling characteristics. Opinions have varied in the past as to whether compressor fouling affects the early stages, which are directly exposed to incoming air, or the later stages, where deposits may bake on because of the higher temperatures, making them difficult to remove by washing. Quantification of the effects of fouling is important from both detection and washing schedule considerations. Very few publications quantify the effects of the basic factors affecting performance deterioration. (Lakshminarasimha, A. N., 1984), (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985) and (Muir, 1989) are among the few publications to quantify the effects of fouling. Compressor performance is simulated by the method of stage stacking.

Fouling in individual stages is simulated by modifying the stage characteristics. Since the performance of the whole stage is represented by meanline conditions at rotor inlet and outlet, only the overall effect of fouling on a stage can be simulated. Prime factors causing the blade profile change, the amount and distribution of deposition along the blade span, which describe the process of fouling, are not considered. Two factors that describe the effects of fouling, reduction in the flow coefficient ($\delta\Phi/\Phi$, or the flow) and the efficiency coefficient ($K\eta$, which describe the loss of efficiency due to blade profile change) are used to simulate fouling. For a given set of values of these two factors, stage characteristics could be modified to simulate a desired level of fouling. Various degrees and types of fouling in a stage can be obtained by giving different combinations of values to these factors.

2.4.2 Results of Fouling Simulation

A series of different fouling cases, which included fouling severity and fouling severity, were carried out on LM2500 compressor section. Figure 2-22 represents the effect of fouling stage location on compressor mass flow rate with $\delta\Phi/\Phi = 3$ percent and $K\eta = 1$ percent. Reduction of compressor mass flow rate when either of the two stages gets fouled increases with compressor speed, with the first stage having the strongest influence. The reason for this is that the front stages affect the operating point of all succeeding downstream stages. The last stage had no effect at all and is therefore not shown in the figure. The reduction of mass flow rate due to two levels of fouling severity $\delta\Phi/\Phi = 3$ and 2 percent with $K\eta = 1$ percent is shown in Figure 2-22. A reduction of the compressor mass flow rate is greater at higher speed, thus indicating fouling severity can be more easily noticed at higher gas generator speeds than at low off-design speeds. Reduction of the compressor delivery pressure was also noticed in all the cases studied.

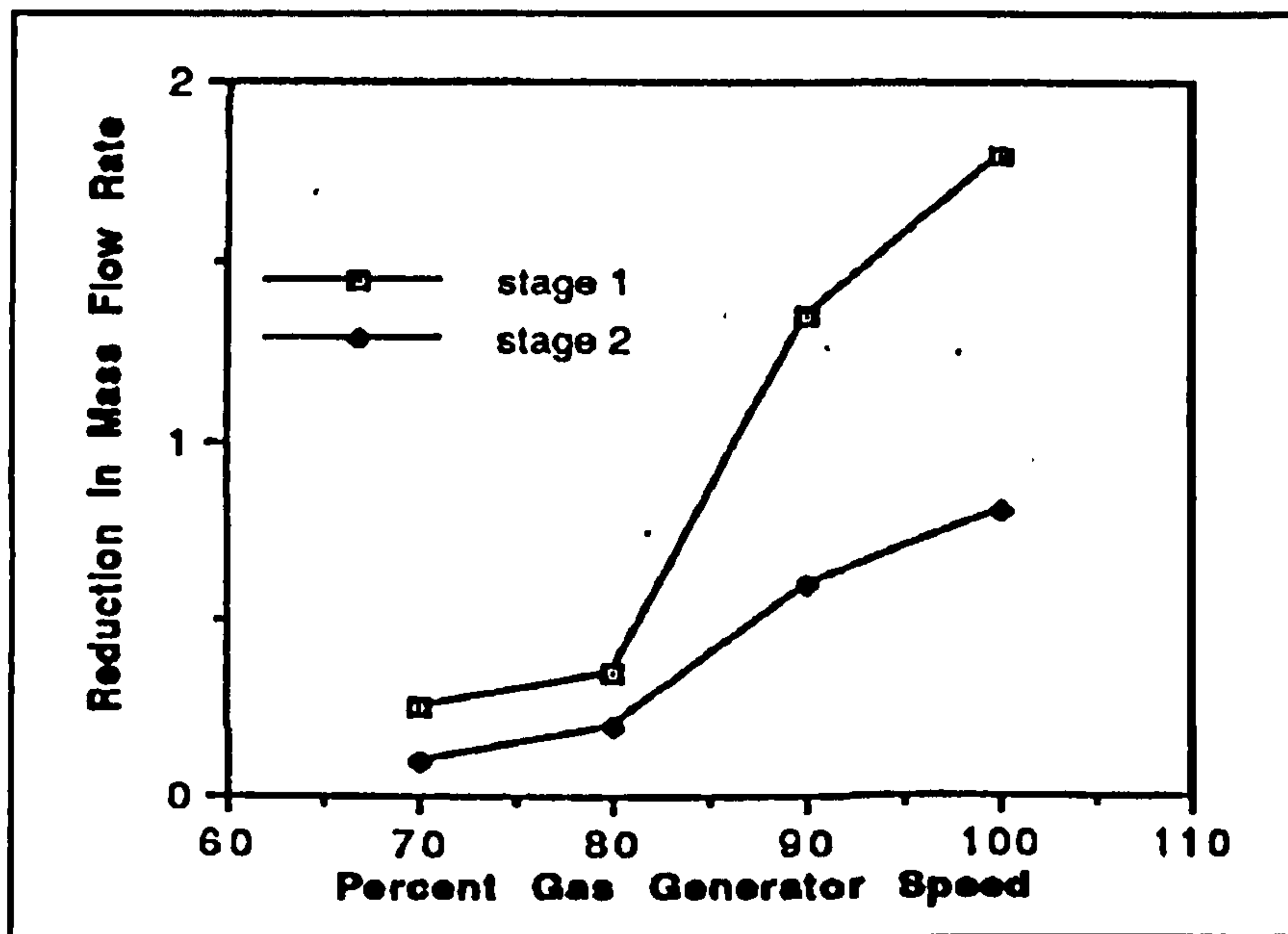


Figure 2-21: Effect of fouling location on gas generator masss flow rate (Lakshminarasimha, 1994)

2.4.3 Deterioration Effects on Engine Parameters

The power developed by a turbine is given by:
$$\text{Power} = \dot{m}C_pT_3 \left(1 - \frac{1}{\left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}}} \right)$$

The pressure P3 entering the turbine is essentially the compressor discharge pressure minus the pressure drop in the combustion chamber. As the mass flow rate and the compressor delivery pressure drop due to deterioration, the work output of the turbine drops. In order to maintain the power output, the fuel control system would increase the fuel flow rate and hence the turbine inlet temperature. Since deterioration also causes the compressor efficiency to drop, the turbine work to maintain the same flow rate and compressor delivery pressure increases. These factors reduce the life of hot section components. To illustrate the variation of the overall engine parameters due to fouling, the simulated compressor characteristics were matched with a gas generator turbine and a power turbine. The variations of power and specific fuel consumption as functions of compressor speed are shown in Figures 2-23 and 2-24. The significant drop in power observed is primarily due to a reduction in air flow capacity of the compressor. It should be noted that a nominal 1 percent drop in compressor efficiency (if first stage) resulted in about 3 percent increase in SFC, due to changes in both pressure ratio and efficiency. Using the fouling simulation, a fault matrix was created for fouling. The results or parameter variations that indicate that fouling reduces the compressor mass flow and compressor delivery pressure, and increases the heat rate and SFC, are in good agreement with the fault matrix obtained from a database of field data for the fouling cases indicated in Table 2 (Boyce M. P., 1984).

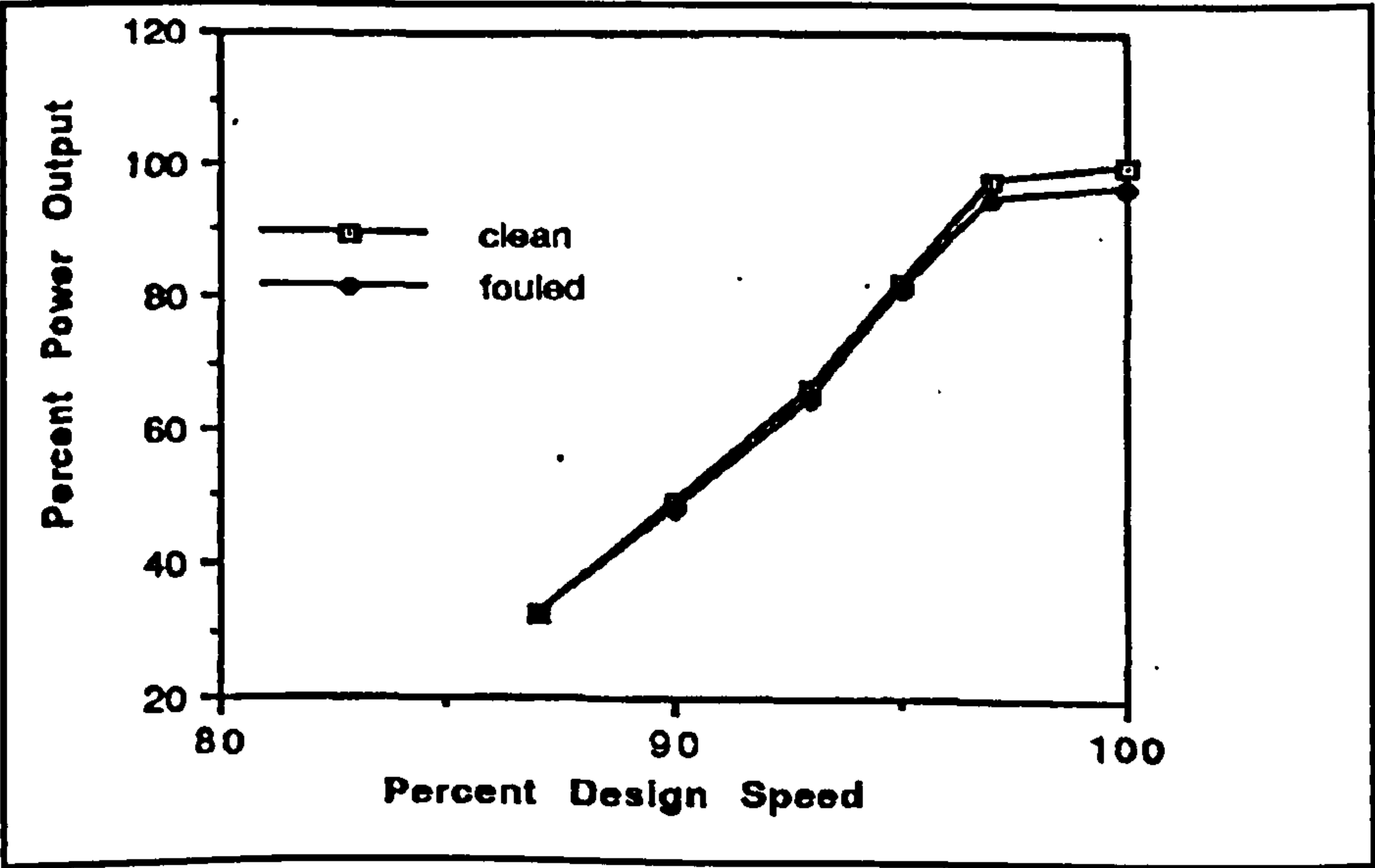


Figure 2-22: Effect of fouling on power (Lakshminarasimha, 1994)

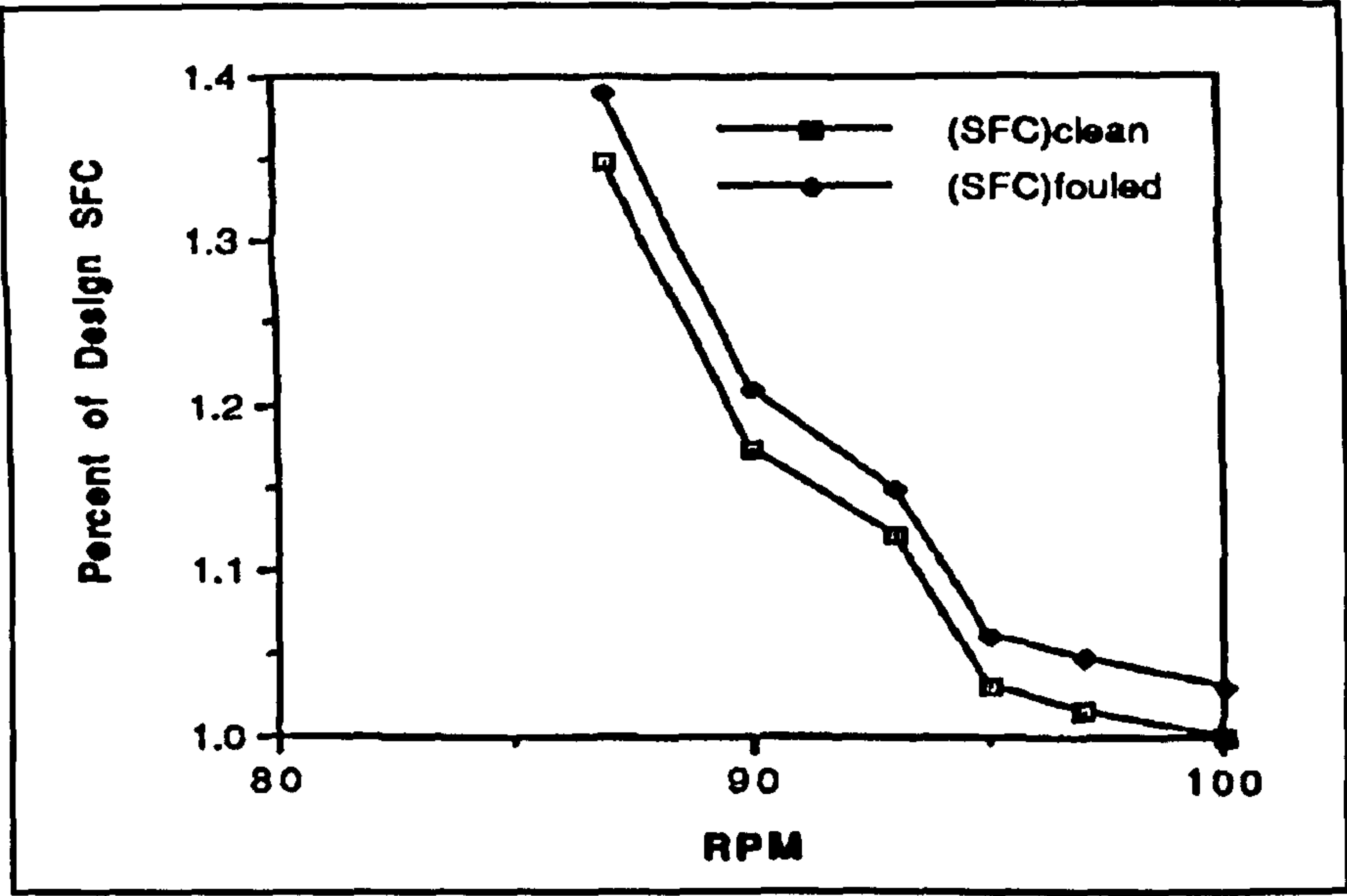


Figure 2-23: Effect of fouling on SFC (Lakshminarasimha, 1994)

Table 1: Compressor diagnostic matrix (Boyce, 1984)

	η_c	P_2/P_1	T_2/T_1	Mass Flow	Vibration	ΔT Bearing	Bearing Pressure	Bleed Chamber Pressure
Clogged filter		↓		↓				
Surge	↑	Variable		↓	Highly fluctuating	↑	↑	Highly fluctuating
Fouling	↓	↓	↓	↓	↑			
Damaged blade	↓	↓	↓	↓	↑			Highly fluctuating
Bearing failure					↑	↑	↓	

2.4.4 Prediction of Compressor Degradation Due to Sand Ingestion

Erosion of compressor blades due to operation in particulate environments is a serious problem for the manufacturers and users of industrial and aeronautical gas turbines, because of drastic degradations in performance, mostly through blunting of blade leading edges, reduction of chord and increase of tip clearance and surface roughness.

The reality of the danger posed by sand ingestion to the front compressor stages of gas turbines has been dramatically illustrated during Desert Shield/Desert Storm operations in extremely erosive desert environments polluted with particulates varying in size from sub-microns to several millimetres, and with quartz as the most abundant erosive constituent. The ingestion of sand particles and their abrasive impacts are known to cause considerable erosion near the leading edge and tip of axial compressor blades. (Balan and Tabakoff, 1984) studied experimentally the effects of sand ingestion on an axial compressor and found severe erosion of the leading edge and pressure side of rotor blades, with increased roughness. Also, Dunn et al. carried out tests on a full-scale engine and found similar characteristics during blades removal; increase of tip clearance and erosion of blade leading edges, which resulted in a substantial performance drop in both HP and LP compressors. Traditional procedures (based on experiments) for assessing turbomachinery erosion and performance degradation

remain very expensive and revealed limited information; consequently, research efforts are more concentrated on numerical simulations, because of the lower cost and time, especially at early design stages.

2.4.5 Field Measurements of Desert Dust Deposition (Libya)

North Africa is the largest global source of atmospherically transported desert dust. Advances in satellite technology allow dust plumes to be spatially and temporally monitored, but little is known of near-surface dust deposition, as field based studies are rare. Figure 2-25 presents the results from a year-long field-based dust monitoring study of three zones across Libya, ranging from the Mediterranean coast to the Sahara desert. Monthly deposition rates are compared with meteorological data, particle size distribution and mineralogy to establish potential source areas. Comparison of annual dust deposition rates with previous studies in North Africa shows that areas of Libya have the highest dust deposition rates on record.

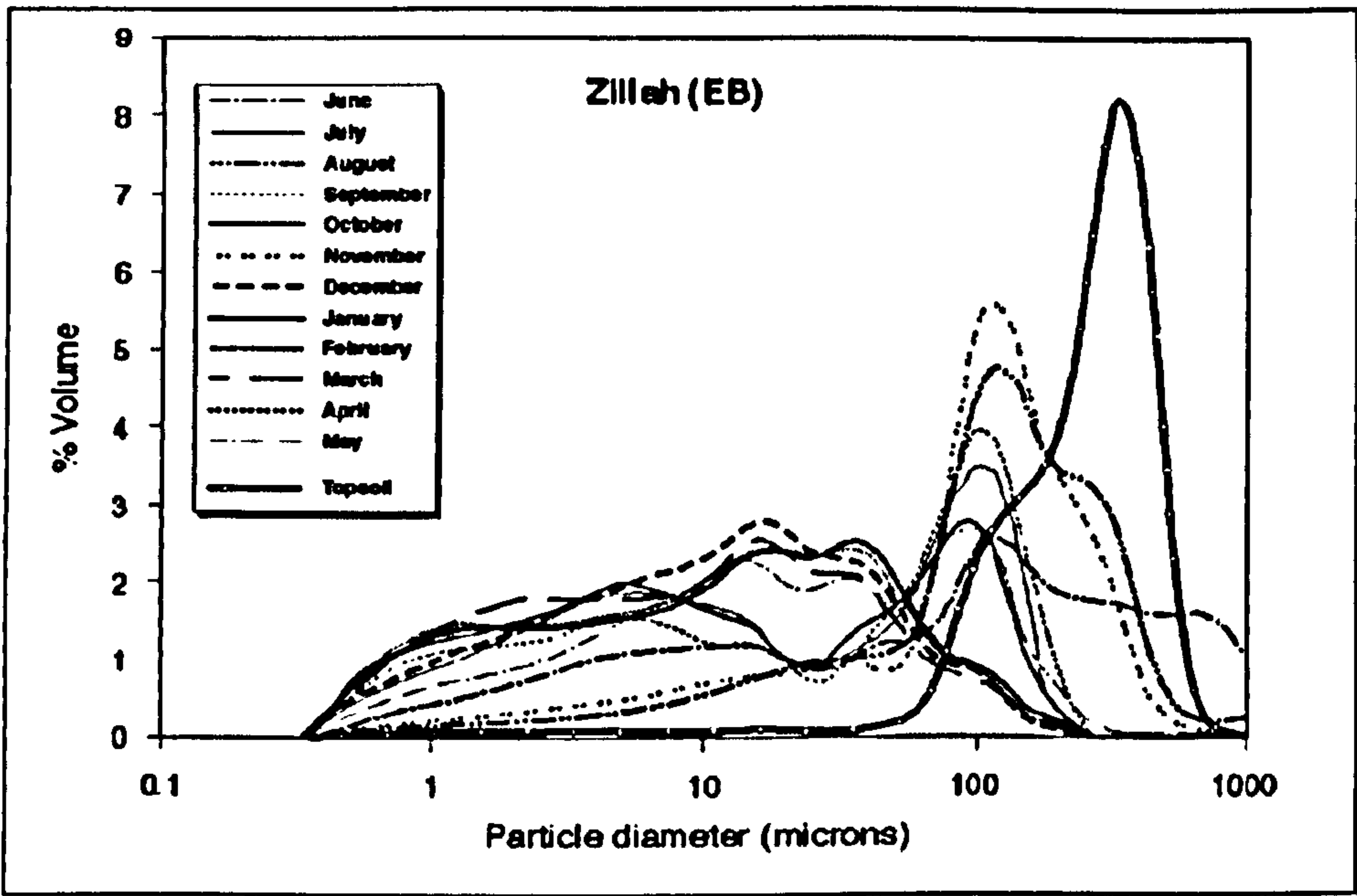


Figure 2-24: Measurement of sand particles in Libya desert (Sarah L. O'Hara, 2006)

2.5 Causes of Compressor Fouling

The air used for the operation of gas turbines always contains a certain amount of aerosols; airborne solid and liquid particles. Their quantity greatly increases under unfavorable conditions: dust and sand storms, dirty industrial environments, etc. To reduce the adverse

effect of fouling on the compressor blading and its performance, an effective filtration system can be installed in the compressor inlet. Theoretically, complete inlet air purification can be obtained, but there is no purpose to this in view of great expense and additional losses in output and efficiency. Therefore the main function of inlet filters is to protect the blading from erosion, and compressor fouling cannot be completely eliminated. It is noted (Diakunchak I. S., 1992) that compressor blading fouling is caused mainly by dust particles of about 2 μm and less in diameter and their mass constitutes up to 80-90 percent of the total dust mass behind the filter. The sources of fouling of the compressor inlet air are described in (Diakunchak I. S., 1992). Fouling of the compressor blades is caused by the "soft" aerosols contained in the air: dirt, dust, pollen, insects, oil and water vapour, sea water salt, sticky industrial chemicals, unburned hydrocarbons, soot particles, etc. Some of these contaminants are products of the operating process of the engine main components and auxiliaries: oil leaks from compressor seals and bearings, salt and mechanical impurities from the water-evaporative cooling system, etc. Some of these particles falling on the blade surface adhere to it, stick to each other, and produce a deposit. This leads to changes in aerodynamic form, profile roughness, and angle of attack. The deposits do not considerably change the throat of a compressor profile cascade. The compressor fouling causes a decrease in its mass flow, efficiency, pressure ratio, and surge margin. For example, as was noted (Diakunchak I. S., 1992) compressor fouling resulted in a drop of about 5 percent in the mass flow, 2.5 percent in the efficiency, and 10 percent in the output. Further, and to enhance this, General Electric found (Hoeft, 1993) that an airflow reduced by 5 percent will reduce output by 13 percent and increase heat rate by 5.5 percent.

2.5.1 Corrosion, Erosion, Damage

Corrosion is the loss of material from flow path components caused by the chemical reaction between these components and contaminants that enter the gas turbine with the inlet air, fuel, or injected water/steam. Salts, mineral acids, and reactive gases such as chlorine and sulfur oxides, in combination with water, can cause wet corrosion, especially of the compressor airfoils. Coating of compressor airfoils will protect them against wet corrosion and prevent the resulting performance deterioration. Elements such as sodium, vanadium, and lead, in metallic or compound form, can lead to high-temperature corrosion of turbine airfoils. The result is a loss in both the service life of turbine components and in engine performance. Even

in gas turbines running on clean gas fuel there will be hot-end component surface oxidation, which will result in the formation of a rough scale and hence performance deterioration.

Erosion is the abrasive removal of material from the flow path components by hard particles suspended in the air or gas stream. Particles causing erosion are usually 20 μm or more in diameter. The erosion of airfoil surfaces results in increased surface roughness, changes in the inlet metal angle (hence change in airfoil incidence), change in airfoil profile, change in airfoil throat opening, and increase in blade tip and seal clearances. The results of the above changes are increased losses and therefore a decrease in performance. Occasionally erosion results in reduced airfoil trailing edge thicknesses. This may be beneficial to performance, but is unacceptable from mechanical integrity considerations.

Damage is often caused by large foreign objects striking the flow path components of the gas turbine engine. These objects enter the engine with the inlet air or are the result of pieces of the engine itself breaking off and being carried downstream. Pieces of ice breaking off from the inlet of the compressor, or large pieces of carbon breaking off from fuel nozzles, can also result in damage to internal components. The resulting damage can lead at worst to a catastrophic engine failure or at best to nonrecoverable (with cleaning) engine performance deterioration.

2.5.2 Engine Component Performance Deterioration

Inlet

Fouling of high-efficiency filters occurs progressively with time. The blockage of the filter pads results in increased inlet loss and hence in engine performance deterioration. This performance loss is recovered completely when new filter pads are installed. The performance deterioration for an industrial gas turbine engine with increasing inlet loss is For a 6 in. (150 mm) H_2O inlet loss increase, the engine output power decreases by about 2.6 percent and the heat rate increases by about 1 percent.

Compressor

Some information available from the open literature indicates that fouling causing a 5 percent reduction in inlet flow will also reduce the compressor efficiency by about 2.5 percent (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985). The resulting decrease in engine power will be about 10 percent. Usually the fouling trend is assumed to have a linear characteristic with time. Experimental tests on the compressor of a small turboprop gas turbine engine indicated that the prime effect of fouling is on the inlet air flow rather than on compressor efficiency, and that the reduction in flow varies with operating speed (being higher at design speed). Site test data obtained on a large industrial gas turbine indicated that compressor fouling resulted in a 5 percent reduction in inlet mass flow and 1.8 percent reduction in compressor efficiency. This amount of fouling would reduce the engine output power by about 7 percent and increase the heat rate by about 2.5 percent.

Bleed valve leakage, flange and horizontal joint leakage, and improper positioning of the inlet guide vanes will have a deleterious effect on the compressor performance. The compressor control bleed valves have a tendency to bind in the open position or to leak during full-load operation. The latter is the more likely occurrence, which will get progressively worse with time and result in a performance penalty. The improper stagger of the inlet guide vanes will also result in a decrease in compressor performance. The performance deterioration due to "nicks," tip and seal rubs, erosion, damage, etc., may be significant in some cases, but usually it is considerably less than that due to compressor fouling. It should be pointed out that compressor fouling and other compressor performance deterioration causes have no effect on engine firing temperature.

Hot End

Under "hot end" are included: combustion system, turbine, and exhaust diffuser. The combustion system is not likely to be the direct cause of performance deterioration with time. Irrespective of the fuel used (whether natural gas, distillate oil, or even crude oil) and even if the fuel nozzles are coated with carbon deposits, the combustion efficiency will not decrease. However, carbon deposits breaking off from the nozzles and soot produced as a result of incomplete fuel burning will result in performance deterioration. Also, changes in the combustor outlet pattern factor may result in a temporary or permanent deformation of

downstream components, which in turn will result in performance deterioration. Hot end cylinder warping, which will increase with time, may result in flange and horizontal joint leakage, and hence in performance deterioration. Fouling of the turbine airfoils and annulus, surface erosion, "nicks" and deformation, blade tip and seal land rubs, and increased leakage and cooling flows will result in performance deterioration. Experiments carried out on a multistage axial turbine showed that both reduction in airfoil profile thickness (as caused by erosion) and increase in airfoil profile thickness (resulting from surface deposits) result in a significant performance reduction (Bammert, K., and Stobe, H., 1970). Any changes in the exhaust diffuser performance can be included in the overall turbine efficiency changes. A decrease of 1 percent in the overall turbine efficiency in an engine such as the CW251B12 will result in about 2.5 percent decrease in output power and a similar increase in heat rate. In addition, because of a decrease in the turbine temperature drop, the engine will be underfired by about 12°F (6.7°C) (note that the engine is controlled to a specified exhaust temperature). This will result in a further 1.2 percent decrease in output power and about 0.2 percent increase in heat rate. Thus the cumulative result of a 1 percent decrease in turbine efficiency with time is about 3.7 percent decrease in output power and about 2.7 percent increase in heat rate.

Gearbox, Generator, Auxiliaries, and Exhaust System

The components included under this heading will also suffer performance deterioration with time and as a result will affect the overall engine performance. It is more difficult to detect the performance losses in these components or to predict their rate of increase with time. The exception is the exhaust system loss. The increase in exhaust loss can be measured and steps can be taken to remedy the situation if the loss becomes excessive. Figure 2 shows the effect of exhaust loss increase on the CW251B12 engine performance. It should be noted that an increase in exhaust loss will result in a reduction in the firing temperature and hence a further reduction in engine performance. Thus a 6 in. (150 mm) H₂O increase in exhaust loss will result in about a 4.2°F (2.3°C) decrease in the firing temperature. Hence the total engine performance deterioration will be a 1.4 percent decrease in power and a 1.1 percent increase in heat rate.

Overall Plant

The overall plant performance deterioration is the result of the combined effect of all of the performance shortfalls of the individual components described previously. Past experience

with gas turbines has shown that after 3000 operating hours the output power loss can be as high as 20 percent under unfavourable conditions and as low as 2 percent under the most favourable conditions ((Scheper, G. W., et al., 1978). For instance, a 2 in. (50 mm) H₂O increase in inlet loss, a 5 percent decrease in inlet mass flow, a 1.8 percent decrease in compressor efficiency, and a 0.5 percent decrease in turbine efficiency, will result in a 10 percent decrease in output power and a 4.2 percent increase in heat rate. The control system can also affect the perceived performance deterioration. If the control system is not properly adjusted, the engine may be operated below the desired base load setting. This will lead to decreased output power and higher heat rate.

2.5.3 Effect of Erosion on Performance

While low-pressure compressor erosion losses are due to surface roughness and tip clearance, erosion losses in high-pressure compressor are created by blade length reduction and loss of air seal materials (Tabakoff & Lakshminarasimha, 1990) in addition to increase in roughness. The physical blade damages due to the erosion for the compressor reported by (Tabakoff & Lakshminarasimha, 1990) are summarized below.

Table 2: blade damage due to erosion (Tabakoff & Lakshminarasimha, 1990)

Stage	Region	Comment
1	tip	leading- and trailing-edge blunting
1	midspan	leading- and trailing-edge blunting
1	hub	leading- and trailing-edge blunting
3	tip	chord reduction
6	tip	chord reduction, airfoil thinning
8	tip	chord reduction
8	midspan	leading-edge blunting
11	tip	great chord reduction, leading- and trailing-edge thinning
11	midspan	chord reduction, trailing edge thinning and loss of camber
13	tip	great chord reduction, trailing-edge loss of camber
13	midspan	great chord reduction, trailing-edge loss of camber

From the above description and from the many detailed cascade and single-stage compressor tests, typical regions of the blade erosion can be depicted as shown in Figure 2-26. In general, the blade erosion can be described as blunting of leading edge, thinning of the trailing edge, and as a general increase in the surface roughness.

These profile and surface quality changes alter the blade boundary layer development, causing an increase in the total pressure loss across the blade row and a decrease in the overall efficiency of the power plant.

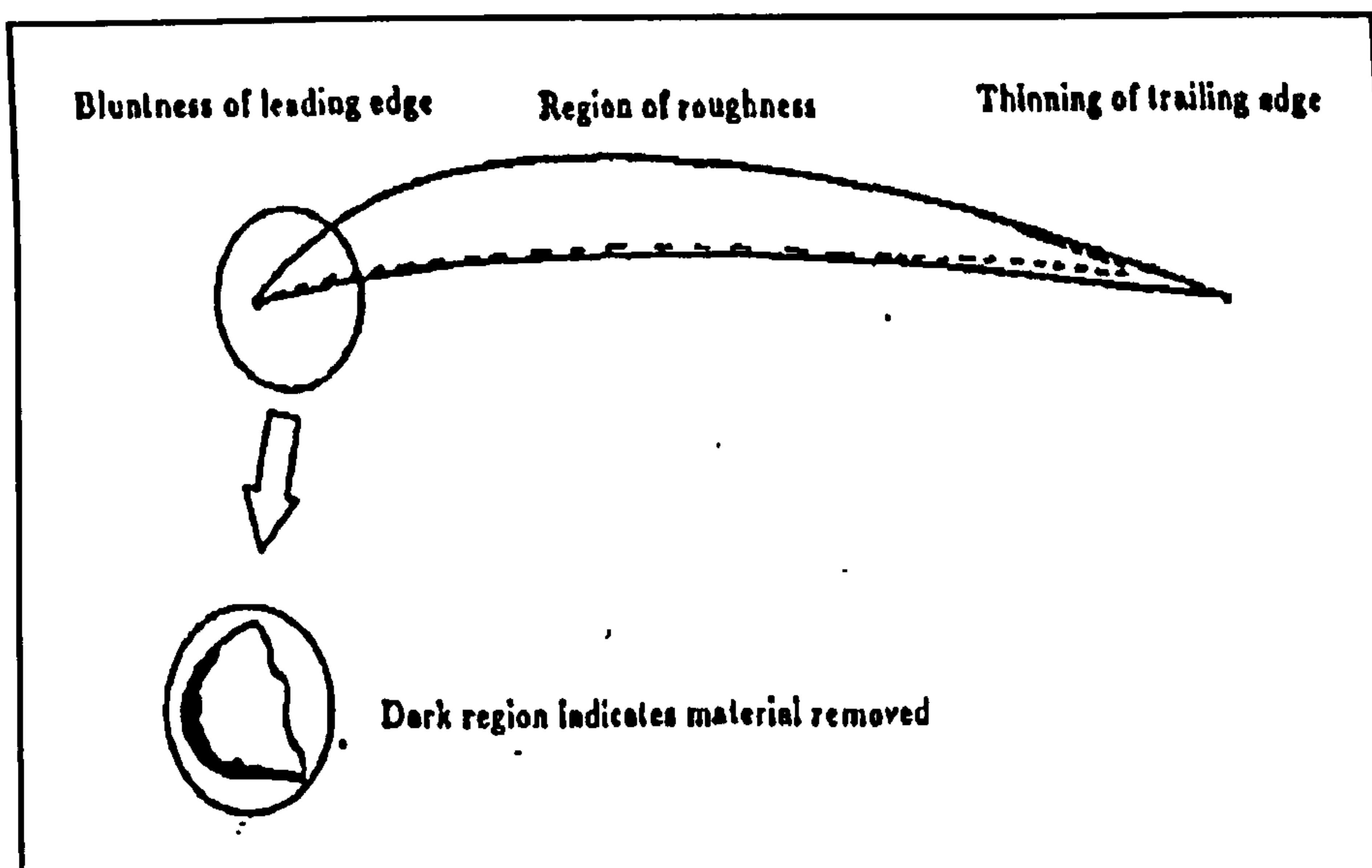


Figure 2-25: Typical erosion pattern of a compressor blade (Tabakoff & Lakshminarasimha, 1990)

2.5.4 Methods of Detection of Performance Deterioration

Detection of the extent of compressor fouling or engine performance deterioration is necessary before the appropriate actions can be taken to restore the performance shortfall. A side benefit of monitoring the compressor fouling or engine performance is the early detection of potential engine mechanical problems, and their resolution before the occurrence of a catastrophic engine failure. Economic considerations may enter into the determination of compressor cleaning frequency. On-line washing usually requires running the engine at part load, while soak washing necessitates shutting down the engine for a considerable period of time. In both cases there is a loss in power, and hence in revenue. Therefore, washing more frequently than required is a wasteful procedure. However, if compressor washing is delayed too long, the resulting performance deterioration will also cause a loss in revenue. Thus it is important to be able to detect compressor fouling and to institute an optimum compressor washing frequency. The optimum will depend on the particular situation at each installation. As a general recommendation, the compressor should be cleaned/washed when the estimated mass flow decrease reaches the 2 to 3 percent level (Diakunchak I. S., 1992).

Compressor Fouling

Compressor fouling, and hence engine performance deterioration, can be detected using the following methods:

Change in Combustion Shell Pressure

The combustion shell pressure is one of the parameters used by the control system to maintain the engine at base load. Therefore, it is a readily available and accurate measurement. When the compressor is fouled the inlet mass flow is reduced, and as a result the engine matches at a lower pressure ratio. Therefore, the amount of compressor fouling can be deduced at any time by comparing the actual site combustor shell pressure to the expected pressure if the compressor were clean.

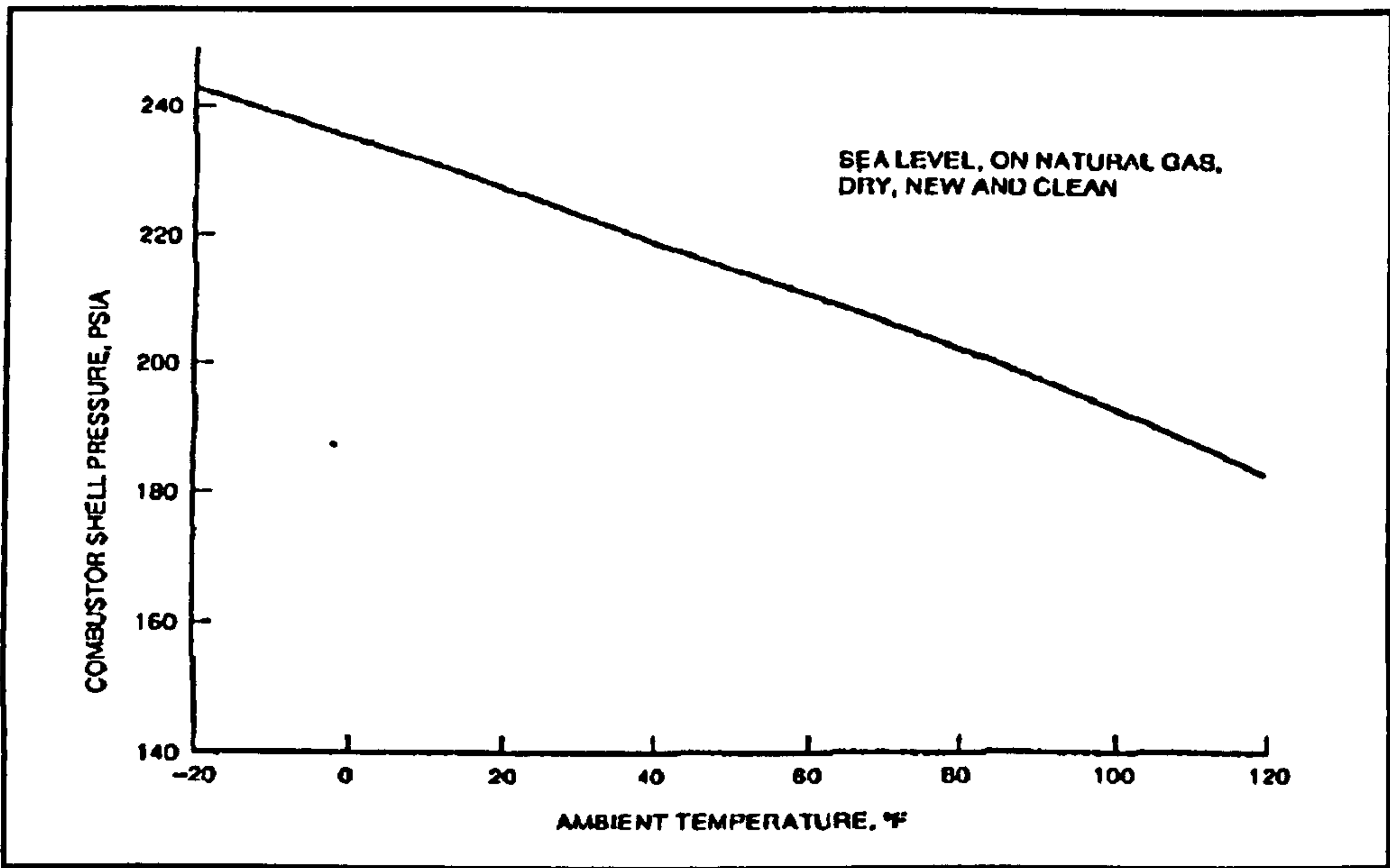


Figure 2-26: Combustor shell pressure versus ambient temperature (Diakunchak I. S., 1992)

One complication is that the combustor shell pressure varies with barometric pressure, inlet loss, compressor inlet temperature, and output power (assuming operation not at base load). If the assumption is made that the engine is operated at base load and that the inlet loss and barometric pressure variations are small, then the expected or "clean" combustor shell pressure variation with inlet temperature is a unique line (a typical curve is shown on Figure 2-27 for an engine at sea level on natural gas fuel). To check for fouling at any time, the site measured combustor shell pressure can then be plotted on this curve. If the point is below the

line, then the compressor is assumed to be fouled. Each percentage point that the actual pressure is below the line corresponds to a one percent reduction in inlet flow.

Cost of Fouling/Performance Deterioration

There is a significant cost to the gas turbine engine operator associated with fouling and the resulting performance deterioration. To make an approximate estimate of the cost, the following assumptions are made (Diakunchak I. S., 1992):

1. Simple Cycle CW251B12 Econopac Engine, on natural gas fuel, no injection.
2. Net output power = 46,500 kW
Net heat rate = 10,550 Btu/kW-h.
The above are average yearly values.
3. Engine on base load operation for 8000 hours per year.
4. Cost of electricity is \$0.04/kW • h.
5. Cost of natural gas fuel is \$2/MBtu
6. Improper cleaning, maintenance, etc., will result in an average yearly decrease in power of 3 percent and increase in heat rate of 1 percent.

The resulting yearly power shortfall = $0.03 * 46,500 * 8000 = 11.2 * 10^6$ kW.h. The cost of the power shortfall = $11.2 * 10^6 * 0.04 = \$446,400$. The resulting yearly excess fuel flow = $0.01 * 10550 * (1 - 0.03) * 46,500 * 8000 = 38.07 * 10^9$ Btu. The cost of the excess fuel = $2 * 38.07 * 10^9 / 10^6 = \$76,140$. The total yearly cost of the assumed average performance deterioration is \$522,500. Over the three year operating period between major overhauls the total cost will be about 1.5 million dollars. In the case of a combined cycle application there will be an additional cost due to reduced steam production in the waste heat boiler. Although compressor fouling results in an increase in exhaust temperature (at the same ambient temperature), the reduced exhaust flow has a greater negative effect. Thus the net effect is reduced steam production as a result of compressor fouling.

Prediction of Performance Deterioration with Time

At one installation site test data indicated that up to 8 percent power loss occurred over a two-week period. Most of this power shortfall could be recovered by cleaning the compressor with new catalyst. At another installation, after incorporation of coated compressor airfoils, the fouling rate was reduced considerably and on-line washing of the compressor recovered all of the power shortfall. At a third installation, after six months of continuous operation

without compressor cleaning, the engine experienced a 4.3 percent loss in power. A series of on-line, at reduced load, water washes recovered all but 0.5 percent of the power shortfall. It should be pointed out that this engine had a good inlet filtration system and coated compressor stator airfoils.

In order even to make an attempt at the prediction of performance deterioration, some ground rules or assumptions must be made. In the prediction of the likely engine performance deterioration with time the following assumptions were made:

1. Types of fuel used: (a) natural gas, (b) distillate oil, and (c) heavy or crude oil.
2. Clean environments
3. The engine will start its service life at time zero in the "new and clean" condition with zero performance shortfalls
4. Continuous base load operation for three years (24,000 hours or equivalent service hours) before a major overhaul. During the overhaul the engine is refurbished to almost "as new" condition, i. e., tip and seal clearances restored, damaged parts removed, "nicks" blended out, flow path surfaces cleaned, etc.
5. Good filtration system used, clean operating environment (no oil leaks, no corrosive chemicals, no sand), no major foreign object damage, coated compressor airfoils.
6. Proper operating and maintenance procedures
7. Effective and regular cleaning/washing of the compressor over the operating period

Based on the above assumptions, the previously described engine component performance deteriorations, and some actual field test data, the overall engine performance deterioration with time can be estimated very approximately. The assumed shape of the typical performance deterioration versus service hours curve is shown on Figure 2-28. This curve shows results of frequent compressor cleaning and also the non-recoverable, with cleaning, performance deterioration line. It should be pointed out that the non-recoverable with cleaning performance deterioration will apply equally to power and heat rate. This can be explained by the fact that once the effect of compressor fouling (which has a more significant influence on air flow and hence power, than on compressor efficiency and heat rate) has been removed then the degradation of power and heat rate will be approximately similar (Diakunchak I. S., 1992).

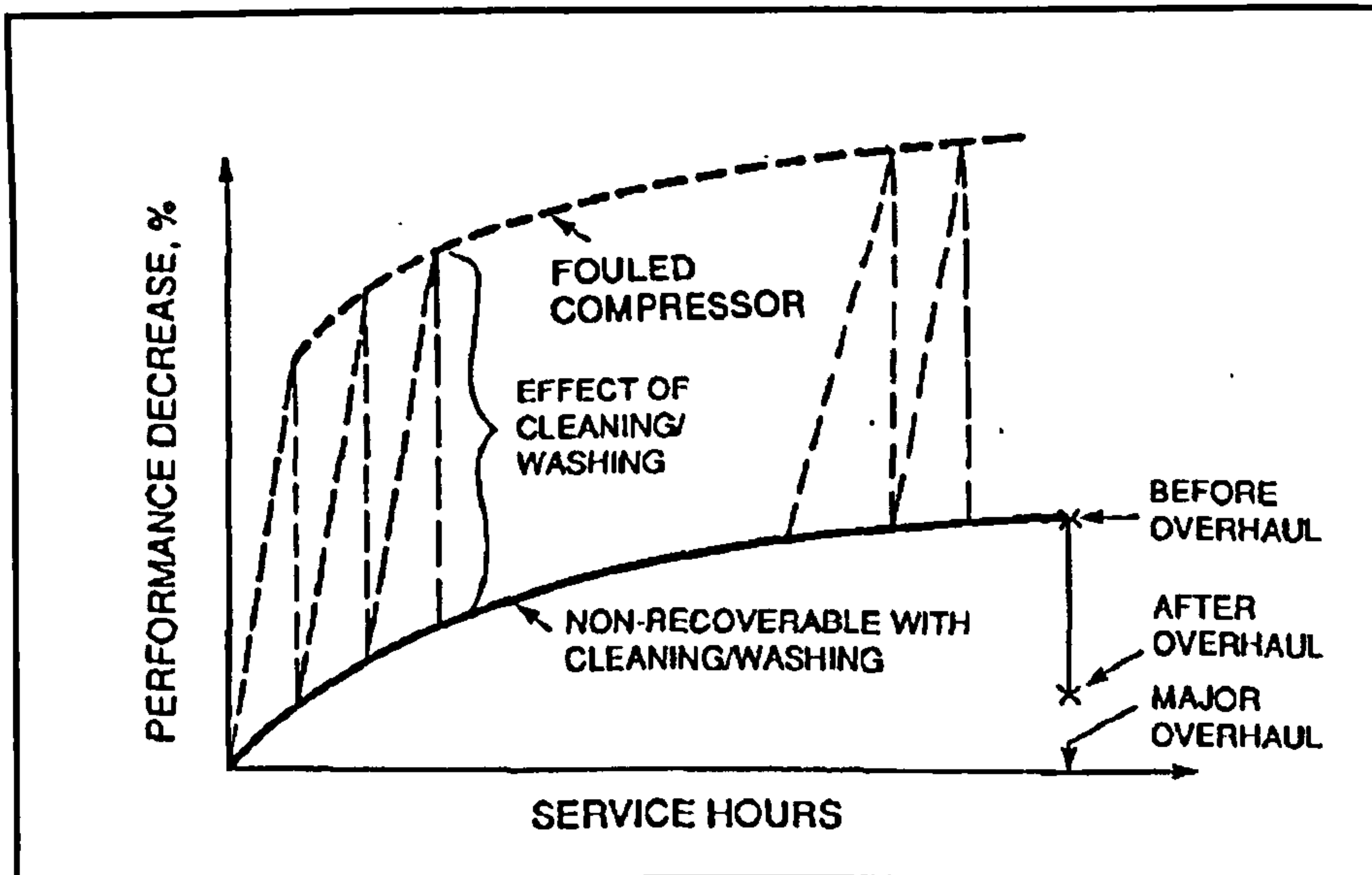


Figure 2-27: Typical performance deterioration (Diakunchak I. S., 1992)

2.6 Protection Against Degradation

While engine degradation cannot entirely be avoided, certain precautions can clearly slow the effects down. These precautions include the careful selection and maintenance of the air filtration equipment and the careful treatment of fuel, steam, or water that are injected into the combustion process. It also includes the observance of manufacturer's recommendations regarding shutdown and restarting procedures. The site location and environment conditions, which dictate airborne contaminants, their size, concentration, and composition, need to be considered in the selection of air filtration. Atmospheric conditions, such as humidity, smog, precipitation, mist, fog, dust, oil fumes, and industrial exhausts, will affect the entire engine. Fuel quality will impact the hot section. Given all these variables, the rate of degradation is impossible to predict with reasonable accuracy. While the rate of deterioration is slowed by frequent online washing, thorough on-crank washing can yield a more significant recovery (Byington, 2003). Online washing will usually only clean the first few stages of the compressor, because the increase in air temperature during compression will evaporate the washing fluid. The online washing process therefore can transport contaminants from the front stages of the compressor to the rear stages or the turbine section. No matter how good the washing, the rear stages of the compressor will not get cleaned. If the compressor blades can be accessed with moderate effort, for example, when the compressor casing is horizontally split, hand cleaning of the blades can be very effective.

2.7 Intake Filtration

2.7.1 Need for Filtration

By their very nature all gas turbines (GTs) consume vast quantities of air. For example, the GE MS9001F GT, when running at full power, consumes nearly 600kg/s of air. Naturally filtration is required to protect the turbine against contaminants in the air that degrade performance and shorten life, in particular, blockages in the cooling passages, corrosion, erosion and fouling. As the technology of the GT has advanced, and become ever more complex, the necessity for adequate filtration has increased.

Gas turbine inlet filters remove particles, such as dirt, dust, soot, etc., from the compressor inlet flow. Selection of the particular filtration methods depends on the location and environment. In a desert environment where there is ongoing potential contamination, and sand-storms are possible, self cleaning filters are recommended to prevent sand particles entering with the air into the turbine. In such a situation filter porosity will decrease with time, causing an increase in pressure loss. Such an occurrence results in reduced power output, and increased heat rate and fuel consumption.

2.7.2 Intake filters performance

Air intake systems have different designs for different applications; industrial, marine, aeronautical, etc. But in all of them atmospheric air passes through the filter to the compressor. The filter efficiency will affect the performance degradation of the GT because it will play a significant role in determining fouling, corrosion, etc. The benefits of improved filtration include lower fuel consumption, high engine performance and lower emissions, although at high intake velocities decisions on filtration porosity become complicated because of the very large pressure drop across the filter.

Typically the size of contaminant particles varies with location of the turbine. For example, in a desert environment 50% of contaminants consist of particles smaller than $1.5\mu\text{m}$. Figure 2-29 shows the efficiency of filter performance with particle size for two types of filter. Typical component contaminant sizes have been given by Giampaolo as: (MSME, PE Tony Giampaolo, 2003)

- ($> 5.0\mu\text{m}$) dust, rain, fog, chemicals and metals.
- ($1 - 2\mu\text{m}$) dust, salts, fog, soot, chemicals, minerals and metals.
- ($0.3 - 0.5\mu\text{m}$) hydrocarbon emissions, smog.

The density of dust particles depends on their chemical composition and can vary from 1.8g/cm^3 to for carbon black to about 3g/cm^3 for dust containing iron oxide (Arosczyk, Wake, & Connor, 1993).

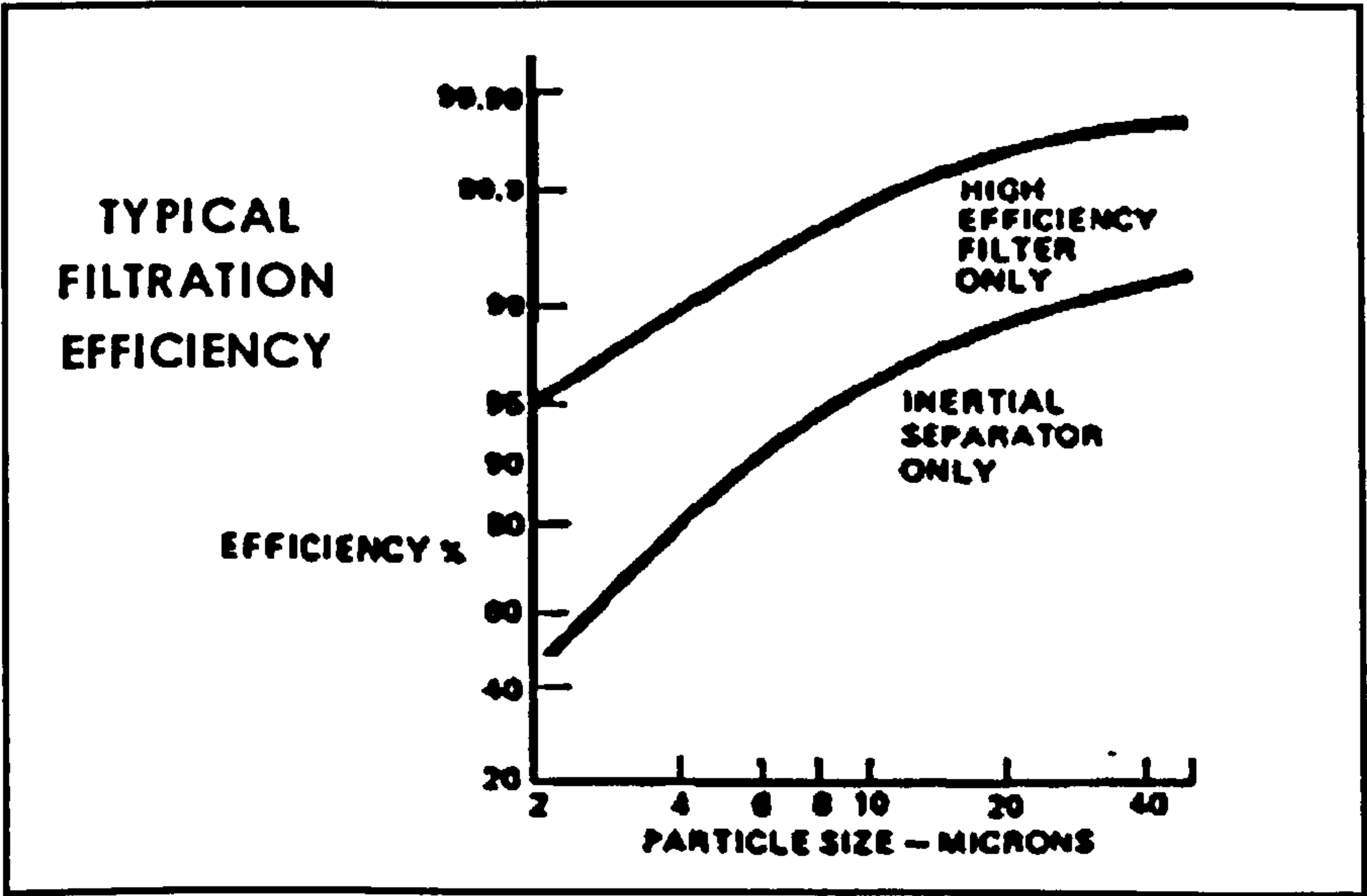


Figure 2-28: Typical Filtration Efficiency (Loud, 1991)

Filter efficiency is a function of air speed through the filter, surface properties of the filter, the physical properties of the air contaminants, whether the airflow is smooth or pulsating and the design of the filter system itself. The ingestion of airborne sea salt or sand through filter when the turbine is running can result in corrosion damage because contaminants such as salt when combined with high temperature, attacks turbine blades and nozzle materials (Loud, 1991). Deposition of contaminants is a cause of fouling, etc and in order to maintain high performance and efficiency, Giampaolo has stated that the following must be checked: that the chosen filter is compatible with local conditions, the correct filter has been installed, there has been proper maintenance of the filter elements and that local conditions have not changed since the filter was last replaced (MSME, PE Tony Giampaolo, 2003)

2.7.3 Intake losses

The intake filters of an industrial GT become blocked due to the accumulation of airborne particles. This leads to decreased filter porosity which eventually results in an increase in pressure drop across the filter and thus reduction in the inlet mass flow that will affect engine performance.

The pressure drop is the most important factor in the set of filter design. A low pressure drop is essential for optimal engine performance. Typically, maximum allowed engine power loss due to restriction of the engine air induction system is 2-3%. The most important factors affecting the pressure drop across filter media are air velocity, filter porosity and filter fibre diameter (Aroszyk, Wake, & Connor, 1993).

2.7.4 Types of Filters

Selection of filtration should match the on-site environment of the GT, and economic and maintenance considerations.

The selection of air filter should meet basic requirement such as the level of filtration needed, cost effectiveness and stable collection efficiency. Examples of filtration systems used on GTs are inertial separator, pre-filter, high efficiency filter and self cleaning filter. In a desert environment small particles of same will accumulate on the filter medium and reduce filter porosity increasing pressure drop and decreasing over-all efficiency (MSME, PE Tony Giampaolo, 2003). The three main types of filters are:

1. Inertial separators

Inertial separators, Figure (2-30), are the simplest and cheapest type of air filter. While recommended for situations where the dust concentrations are high, it is mainly effective with larger particles, greater than 20 μ m in diameter. As the dusty air enters the filter it has a rotational motion imposed on it, this centrifugal action throws the dust particles outwards where they can be separated and removed from the inlet flow. In many filters a bleed of 5 to 10% of the inlet flow is used for this purpose. The higher the air velocity the more effective the filter, and it is this condition that makes this filter not very effective at removing smaller particles. Hence inertial separators are not very good at preventing compressor fouling which is generally caused by small particles with diameter of 5 μ m or less, but is good at preventing

the erosion caused by larger particles. (Daikunchak, 1992) has assessed typical pressure loss of inertial filters as about 250Pa. However the inertial separator does have one important advantage, the pressure loss does not increase with time in service.

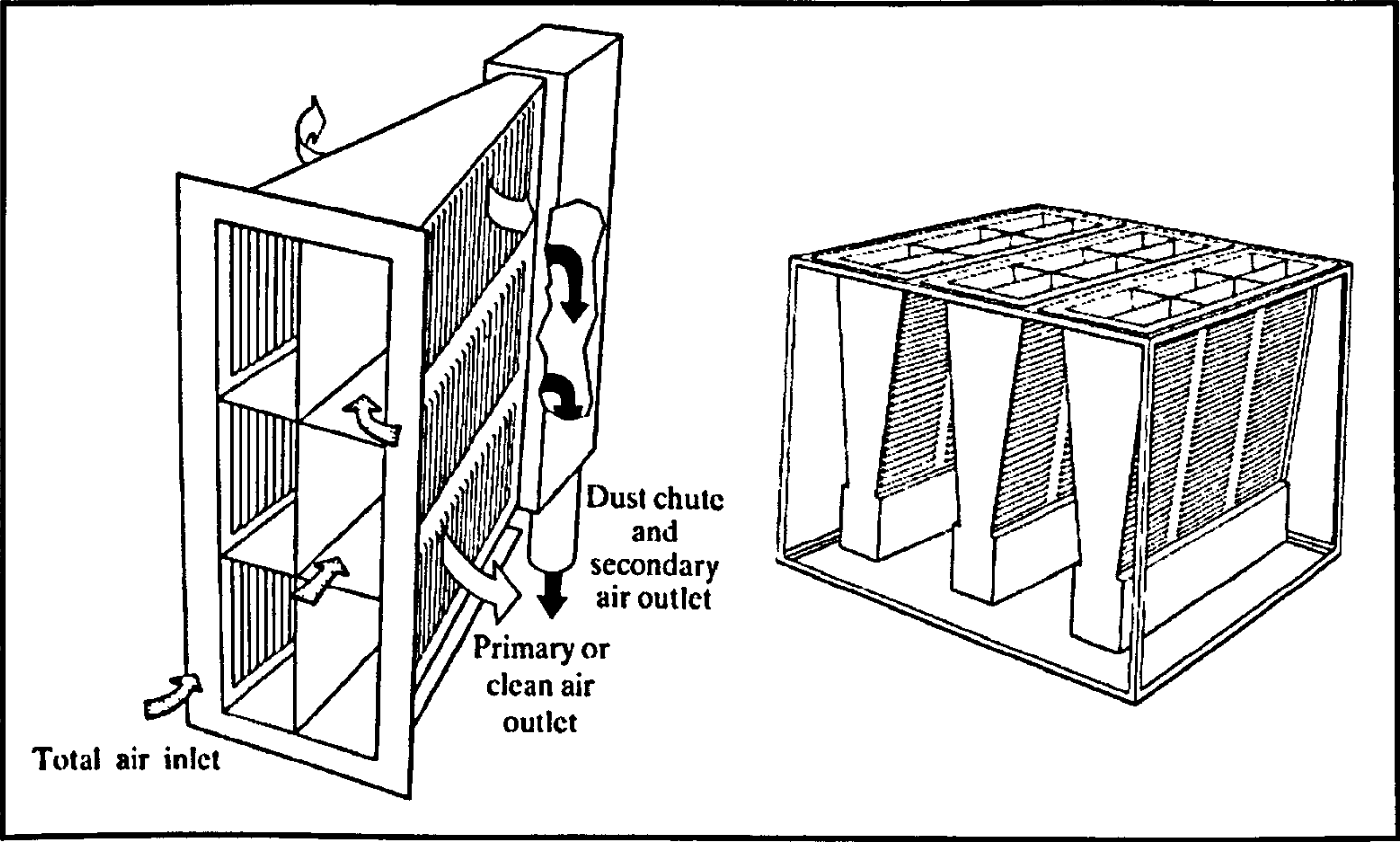


Figure 2-29: inertial separator (Dickenson T. , 1998)

Pre-filters

The pre filter, see Figure 2-31 is usually shaped like a panel that is installed directly in front of another filter so it is usually of very low initial cost. Unfortunately it is usually relatively expensive to maintenance cost and of low efficiency. Pre-filters tend to be limited to systems handling air volumes of less than 20m³/s. Another type of pre-filter is the automatic roll which is often slightly better than a panel filter.

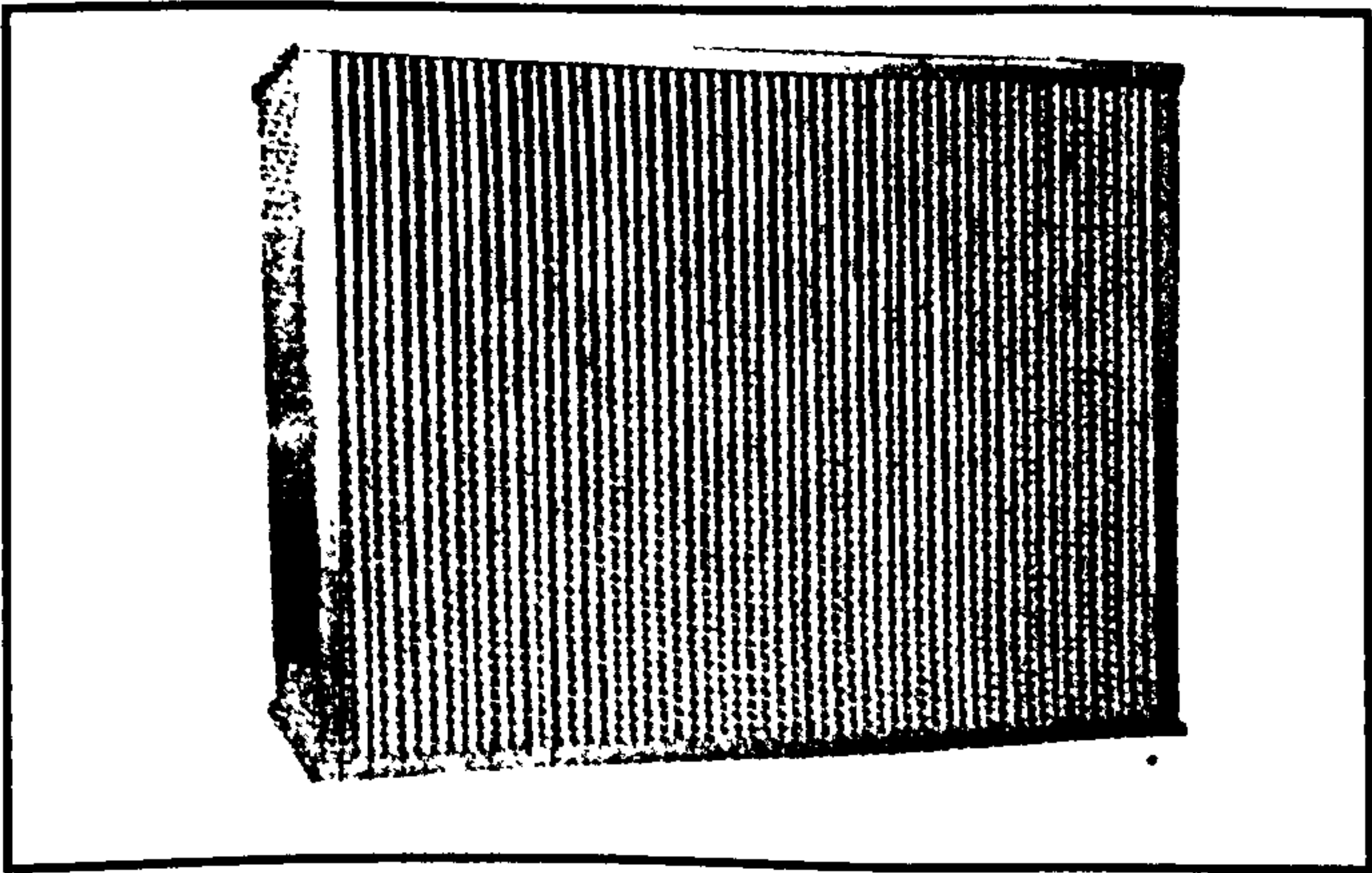


Figure 2-30: Pre-filter panel type (Dickenson T. , 1998)

2. High efficiency filters

High efficiency filters for particles smaller than $1\mu\text{m}$ tend to be made from fibre glass, PTFE, or treated paper. There are two types of high efficiency filter: the rectangular panel and cylindrical cartridge. Figure 2-32 shows a high efficiency rectangular panel filter which is of typical construction, using pleated media to increase available surface area. The rectangular panel filter comes as a replaceable unit element usually in a metal frame which is clipped into place. Unfortunately the air seals of such units have not been shown to be reliable.

Panel filters use filter media with fibres small enough in diameter to capture sub-micron particles at high efficiencies without the necessity of electrostatic attraction the particles are actually trapped within the body of media itself. The medium must be sufficiently porous for the he particles to penetrate it. Once inside the body of the filter those dust particles which come into contact with the fibres and attach themselves and are filtered out. The four mechanisms of by which the dust particles move into contact with the fibres are: diffusion (the smallest particles), interception (particle follows the air stream until the bump into a fibre), impaction (larger particles have sufficient momentum to proceed in a more straight line path until they impact on a fibre), and sieving (the largest particles are too large to pass between adjoining filter fibres).

The pressure drop across high efficiency filters will depend not only on their construction and installation but also the quality and quantity of air passed through. As dust accumulates the pressure drop also increases, relatively slowly at first but more rapidly as the filter become increasingly blocked (Loud, 1991).

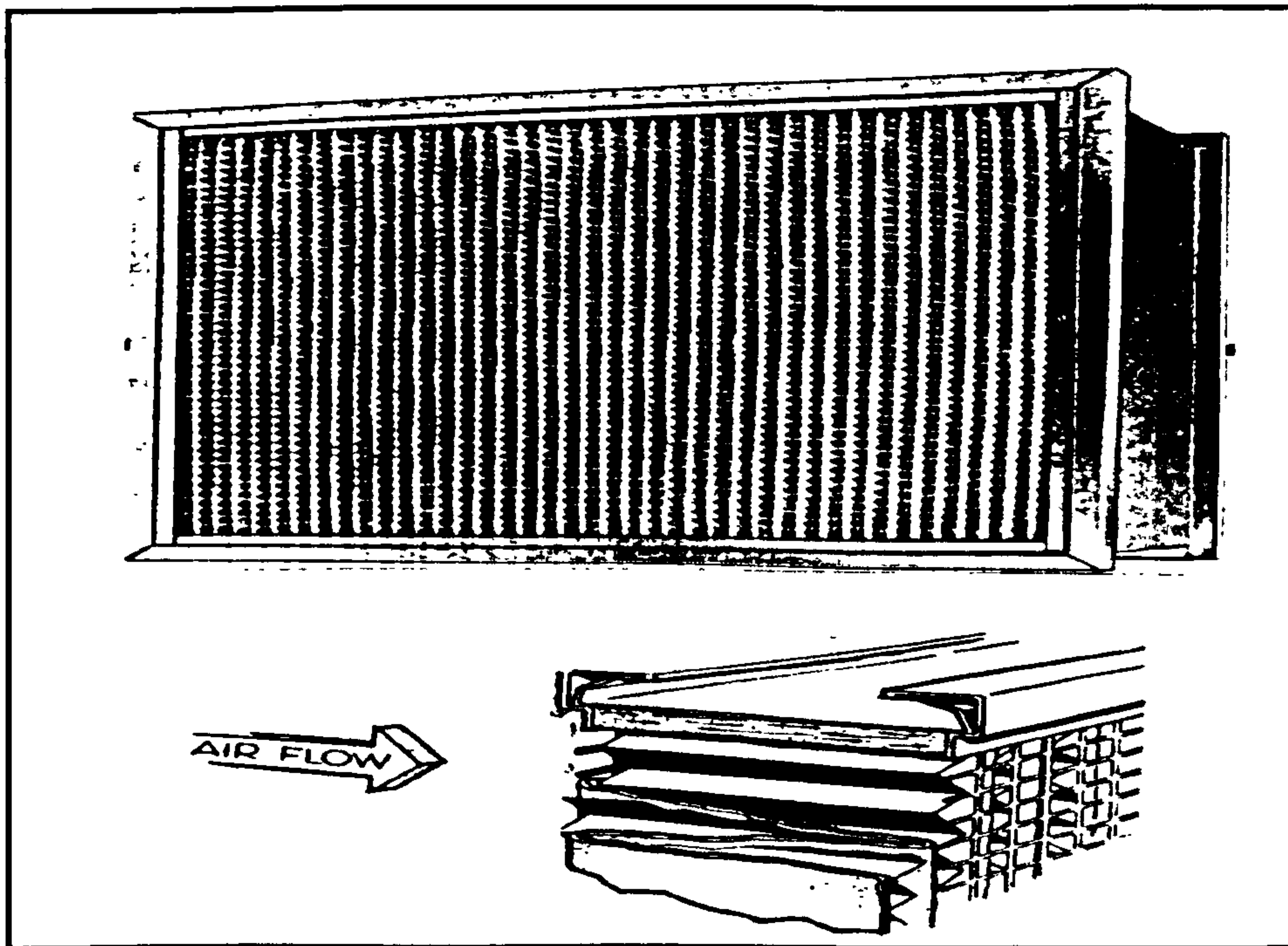


Figure 2-31: Rectangular panel high efficiency filter (MSME, PE Tony Giampaolo, 2003)

3. Self cleaning filter

In the 1970s a self cleaning inlet filter, see (Figure 2-33) was developed as a high efficiency filtration system with low cost maintenance. Air velocities are set below about 3m/s through filter element (Loud, 1991). Self cleaning air filtration systems have a differential pressure controller to measure the pressure drop across the filter. When the pressure drop reaches a pre-set level the filter was designed to automatically clean itself. Of course, self-cleaning can be initiated manually. After a full clean the filter unit pressure drop should reach its stabilized value.

The dust filtered from the air flowing through it will collect as a cake on the outside face. The dust is removed by the cleaning process. Unfortunately, a small fraction of the dust always becomes imbedded within fibre matrix of the filter (Loud, 1991).

Figure 2-33 shows principle of operation of the self cleaning filter. As soon as the pressure in the clean air plenum falls to a set-point a compressed air valve opens and strong reverse pulse of air is automatically initiated for less than 1 second. This travels through the filter to blow the dust off its face. In fact the counter-flowing air is controlled so that it starts at the top of the filter and works its way downwards. The dust usually settles in a hopper from where it can be removed.

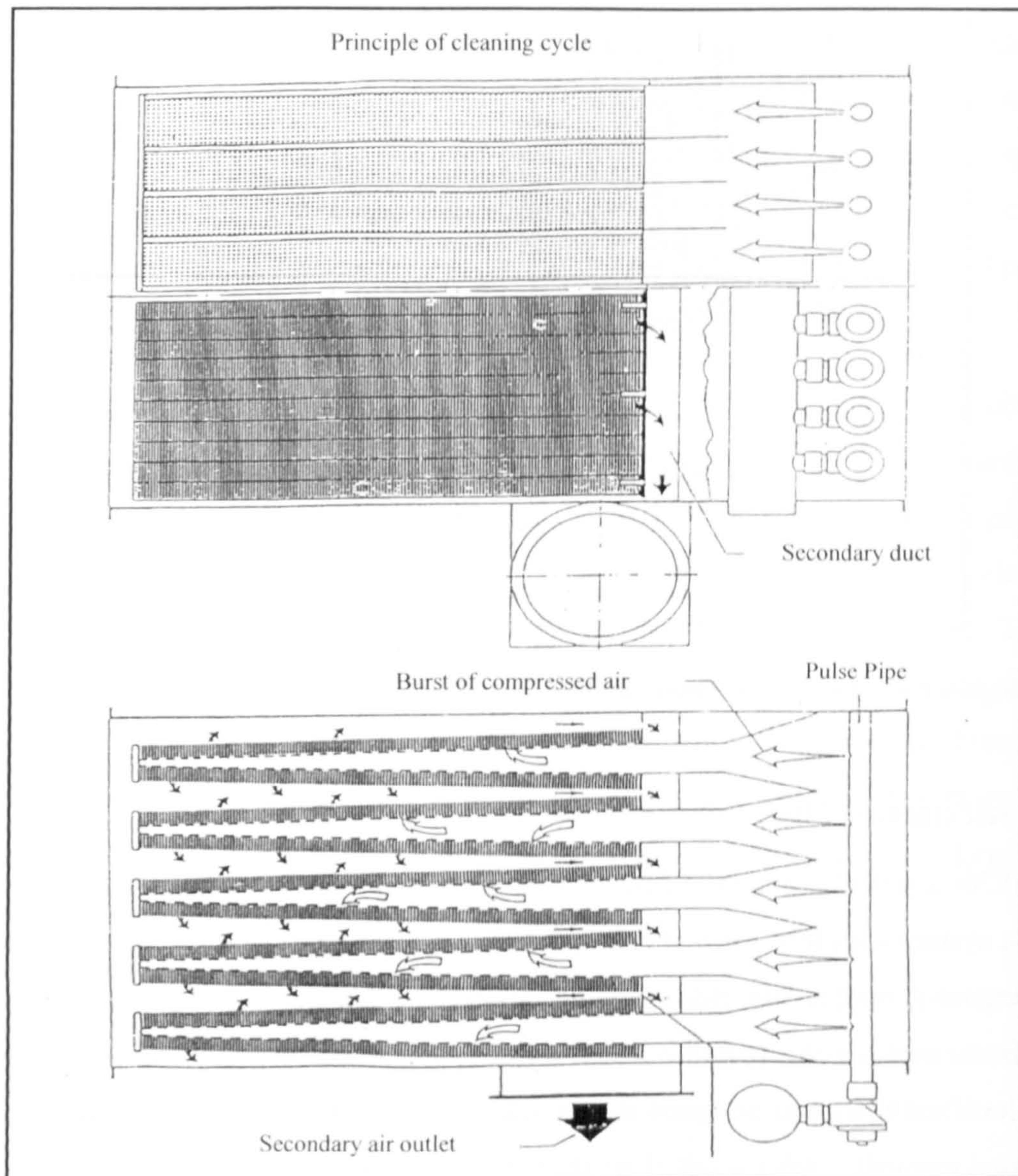


Figure 2-32: Operation of Self Cleaning Filter (Loud, 1991)

This type of filter is very effective in desert environments where dust and sand can rapidly accumulate see (Figure 2-34). In arctic regions there is snow, and in marine applications there is salt, both of which will have an effect on intake filtration.



Figure 2-33: Self Cleaning Filters in a Desert Environment (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

2.8 Compressor Cleaning

Previously GT cleaning was achieved by crank soak washing and/or by the injection of solid substances such as rice husks or nutshells with the unit on-line and running at full speed (McDermott P. E., 1991).

This is known as cleaning by soft erosion, but now with the inclusion of coated compressor blades for further protection against pitting corrosion, wet cleaning is predominantly used. In addition, ashes and solid cleaning substances may block the complex turbine cooling system if they enter the GT air cooling stream. In the 1980's a strategy had to be developed for the time periods between on-line washing with the combination of off-line washing (Stalder J.-P., 2001). It is also believed by many users that on-line washing could completely take over from off-line washing. Hoeft has stated that reduction of inlet airflow by 5% because of fouled compressor blades, reduced the output by 13% coupled with an increase in the heat rate by 5.5% (Hoeft, 1993).

2.8.1 Compressor Washing Techniques

The benefits of compressor cleaning have long been recognized and there have been many research studies published in this field. Crank soak cleaning and the use of solid compounds such as nutshells or rice husks have been replaced by wet on-line cleaning. An important factor in this process is the influence of the time interval between on-line washing events (Stalder J.-P., 2001).

Compressor fouling has a significant impact on GT operation because the costs of power loss and increased fuel have increased dramatically. To avoid this penalty completely, compressor washing using both on-line and off-line techniques must be employed. This section deals with the different kinds of compressor cleaning and the effect of the various detergents used. In particular, the advantages and disadvantages of these techniques and the best solution of the problem are discussed.

The section is divided into sections as follows:

- Types of compressor cleaning,
- Washing materials,

- Liquid cleaning, and
- Compressor cleaning frequency,

2.8.2 Types of compressor cleaning

Several types of compressor cleaning have been developed over recent years. The initial method required that the engine is disassembled and cleaned manually. This is very time consuming and expensive (Mund, F. and Pilidis, P., 2004). However, there are times when this method must be used, in particular in overhauls. However complementary methods are available:

- Abrasive compressor cleaning, and
- Wet compressor washing.

• Abrasive compressor cleaning

The abrasive cleaning method is relatively old and not used so much now, it works by injecting solid compounds such as rice husks and nutshells or spent catalyst into the GT. This is achieved by simply throwing the cleaning material into the GT intake or via an air compressor. This ensures that the method is inexpensive and simple.

However, such a method can cause a number of problems in the GT (McDermott P. E., 1991):

1. Blade erosion.
2. Bearing damage.
3. Blockage of blade cooling systems.
4. Blockage of bleed and instrument ports.
5. Re-depositing the dirt to the later compressor stages.

Due to these detrimental effects, abrasive cleaning methods are being discontinued and even prohibited by some regulations.

• Wet compressor cleaning

There are two kinds of compressor washing systems, the on-line and the off-line, and both inject a quantity of distilled water or a mixture of distilled water and a special detergent into

the compressor while it is turning to remove deposits. The two systems have of virtually the same components. That is, they both require the injection nozzle, the manifold, the control unit, and the cleaning material. Figure 2-35 shows a typical diagram of a liquid washing system.

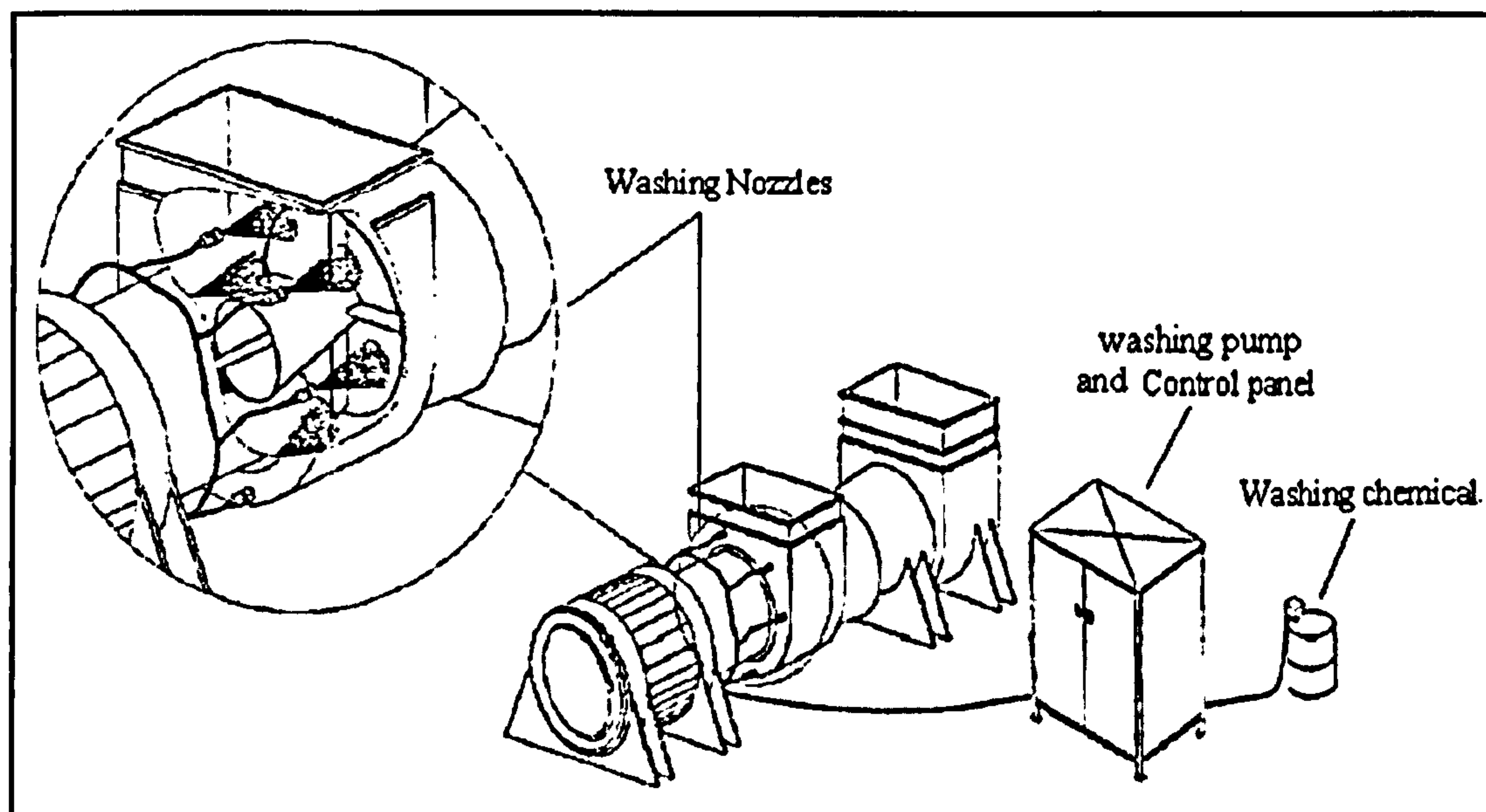


Figure 2-34: Diagram of a typical liquid compressor wash system (Mund, F. and Pilidis, P., 2004)

The use of demineralised (DI) water in both methods is of prime importance, so that no alkali or hard water scale get deposited in the GT. A typical specification for dematerialized water usually limits the conductivity and pH of the water, and limits the proportions of silica, calcium, magnesium, sodium, potassium, fluorine and chlorine, dissolved in the water (Stalder J.-P. , 2001).

A. Off-line Crank Washing

Off-line washing can be more closely controlled than on-line washing and so is considered more effective, but is more time intensive and disruptive to operations. With this washing process the GT is turned over at very low speed and detergent injected into the inlet. There then follows a prescribed period called the “soaking period” after which there is a DI water rinse (Figure 2-36). The cleaner is mainly surfactant (a wetting agent) which reduces the surface tension of the cleaning solution enabling it to wet, penetrate and disperse unwanted deposits. Mezheritsky (1990), Abdelrazk (1991) and Stalder (1998) have found that, to be most effective at removing oily deposits, crank washing with cleaning solvents should be carried out at relatively cold temperatures ($\sim 15^{\circ}\text{C}$) or with water heated to nearly boiling.

The water injected into the GT drains off the blades rapidly, reducing the contact time during the off-line soaking period.

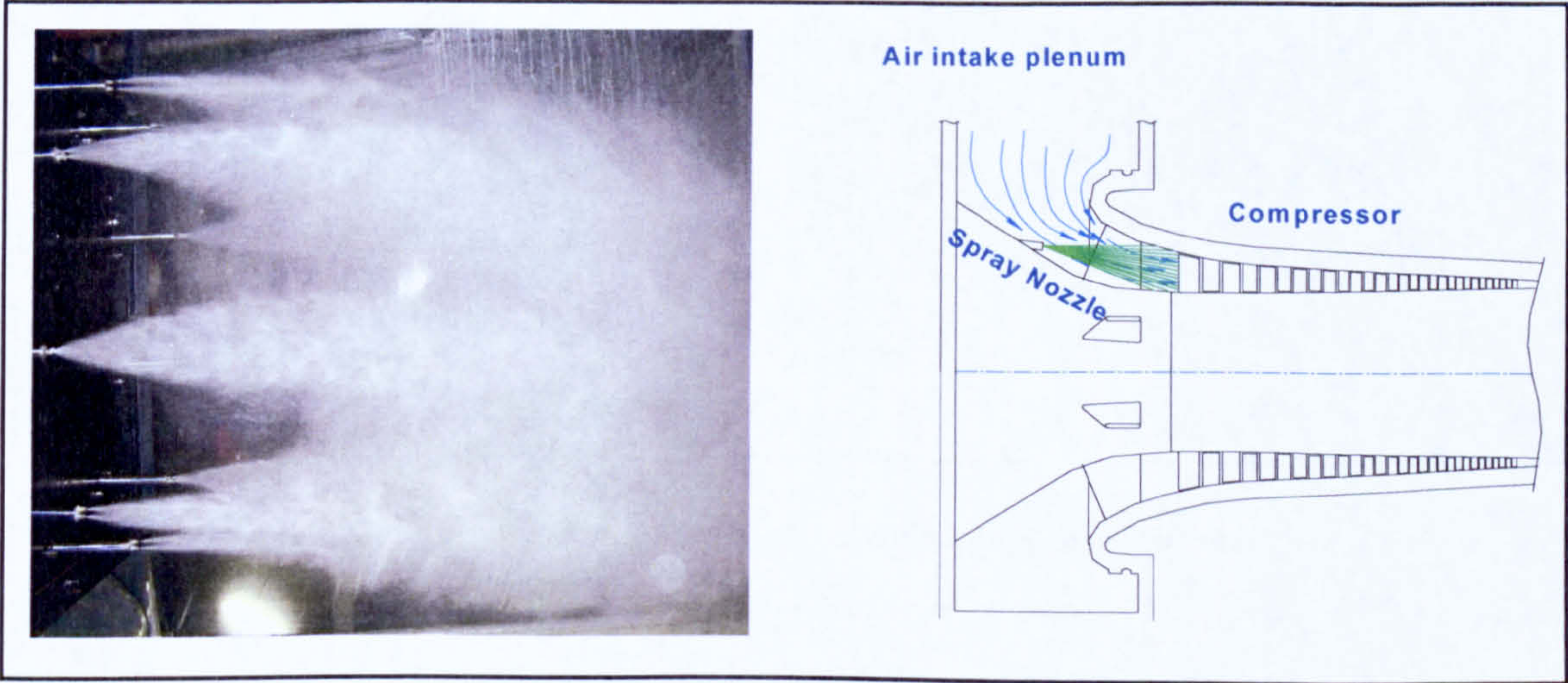


Figure 2-35: Cleaning Fluid Spraying through the Air Intake (Courtesy of R-MC Company, UK)

B. On-line Washing

DI water and/or cleaning solution is injected into the GT at the inlet under part-load operation (Figure 2-37). This is less disruptive and time intensive than off-line crank washing because little or no cleanup is required. However, on-line washing is not as effective as crank washing due to increased rotational speed – centrifugal forces spread the solution to the casing surface after only a couple of stages. The overall effect is more like a “steam cleaning” (Haub B. L., 1990), (Kolkman, 1993), (Stalder J. P., 1998). The rate of fouling can be significantly slowed by frequent on-line water washing. More significant performance recovery is obtained through off-line crank washing. No matter how good the wash, the rear stages of the compressor will not get cleaned; Hand cleaning or overhaul is necessary in such cases (Haub B. L., 1990).



Figure 2-36: Different Spray Nozzles Online-Offline Washing (www.siemens.com)

C. Optimum Regimen

Frequent on-line cleaning increases the allowable time interval between off-line cleaning operations. On-line washing with water should be conducted at short intervals (every 3 days to weekly). On-line cleaning with detergent cleaners should be conducted once a week at most. A general recommendation for crank washing is to clean/wash when the estimated inlet air flow decreases 2-3% (Scott, 1979). Furthermore, at least four off-line cleanings per year are needed to remove salt-laden deposits on the downstream stages. This rule also holds for a peak load unit running only a few times a week (Abdelrazik A. a., 1991), (Diakunchak I. S., 1991), (Kolkman, 1993) and (McDermott P. E., 1991).

Some authors recommend an on-line wash every 200 hours and soak/crank-wash every 1000 hours (Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N., 1985). Others suggest a maximum interval between on-line washes of five days, thereby assuring little ageing of the deposition layer (Mezheritsky, 1990). In general, operators should perform an on-line wash frequently to remove salt deposits, regardless of performance. If the on-line wash is not effective at restoring performance, a crank wash should be performed see (Figure 2-38). The cleaning liquid should be viscous enough to cling to the blades during the spraying and soaking period, then be easily dissolved and removed thereafter during the rinse. Engine cranking at very slow speed is preferable while injecting detergent (as slow as possible), (Haub B. L., 1990), (Peltier R. V., 1995) and (Stalder J. P., 1998).



Figure 2-37: Compressor Blades before and after Cleaning (Haub B. L., 1990),

2.8.3 Washing Materials

Nowadays, there are two ways to classify compressor cleaning methods. The first is the previously discussed: on-line and off-line cleaning. The second is via the injected material. They can be classified as either chemical or DI water compressor cleaning. Each method has its advantages and disadvantages as shown in Table 3 (Mund, F. C. and Pilidis, P., 2006).

Table 3: Relative advantages of different compressor cleaning methods

Method	Advantages	Disadvantages
Manual cleaning brushes, washing agent	Very effective	Shut down of engine, laborious
Grit blasting: charcoal, rice, nut shells, synthetic resin particles	Simple and fast. No engine downtime Effective in cold environments	Less effective at rear stages of GT and oily deposits. Possible clogging of internal cooling passages. Erosion. Increased surfaces roughness Damage to blade coatings
Soak, crank, off-line washing DI water, washing agents	Very effective	Shutdown of engine
On-line washing, DI, washing agents	No interference with load profile Extends intervals of crank washes	Less effective Cannot replace off-line washing (complementary)

Chemical Cleaning

For use against oily deposits chemical cleaning is particularly effective (Stalder J.-P. , 2001). This is because the chemical used can contains surfactant which reduces its surface tension allowing better penetration and removal of the deposits. The detergent used (and DI water) must be used with the approval of the GT's manufacturer. Because foam is such a good carrier of dirt, the amount of foam generated by a compressor cleaner is often taken as an indication of the degree of effectiveness of the surfactants. This is because the foam will ensures a better distribution of the cleaning solution and its more effective penetration of the unwanted deposits. The foam will also be easily rinsed off during the off-line rinse cycle.

Some detergents contain corrosion inhibitors which neutralize the influence of salts. The benefit of this is that it will eliminate the need for the rinse cycle in the off-line cleaning which might be important for installations located in remote areas.

Chemical detergents come in either of two forms: water based or solvent based (Stalder J.-P. , 2001). Water based detergents have a broader range of application as they are very effective against oily soils, water soluble salts and particulates. Solvent based are more effective than water based against oily soils deposits but it is virtually ineffective against other types and is toxic.

The disadvantage of the chemical cleaning is that the detergent is expensive and might not give the required results if it is not properly formulated or injected.

2.8.4 Liquid Cleaning Influence Factors

According to (Mund, F. and Pilidis, P., 2004), there are several factors that influence liquid cleaning. These parameters must be taken into consideration while designing or even selecting a proper cleaning system. For example, droplet size and velocity are critical so that good and erosion-free cleaning is achieved. The most important factor is the injection pressure since it will have an impact on most of the other factors (Paul Lambert et al., 2004).

2.8.5 Compressor Cleaning Frequency

Accordingly to (Mund, F. and Pilidis, P., 2004), more and more GT operators have become convinced of the financial benefits of compressor cleaning and have started to fight compressor fouling. Many operators choose their washing frequency based on GT performance. Basically, the operator monitors engine performance relative to its baseline performance. Once the allowed performance drop is reached, a compressor wash is performed.

In the pursuit of maximum financial benefits from compressor cleaning, many operators applied optimization methods to obtain the optimum washing frequency and a combination of on-line and off-line compressor washing methods.

As indicated earlier, off-line compressor washing is more effective than on-line washing. However, it requires a complete shutdown and a cooling period. However, on-line washing does not restore the full GT capability and so there is always a downward trend in performance. Therefore, the use of both methods is necessary in the most economical combination to minimise the rate of performance degradation.

In the past few years, there has been a move to use optimization techniques in conjunction with GT diagnostics and prognostic systems. This has allowed GT operators to understand the GT performance more accurately and optimize washing frequency. Thus, without an accurate performance calculation, the optimal washing frequency will not be reached (McDermott P. E., 1991).

Another field of optimization is the compressor wash operating cost, and some work has been undertaken in this field (Paul Lambert et al., 2004) because lower consumption of washing liquid injection will significantly reduce the required equipment size, volume, and cost of washing skids and will consequently reduce overall costs. It will also decrease the effluent water to be disposed. Such an optimization can be reached by the proper combination of a good system design, a good cleaner and good maintenance.

2.8.6 Combination and Frequency of On-line and Off-line Washing

In the middle of the 1980's a long term test was conducted on the effect of on and off-line washing at various intervals. It was carried out on 100 MW combined cycle plant (two GTs of 30.7 MW site output and one 38.6MW steam turbine). The plant was in an industrial environment and the interested parties included the chemical and food processing industries. When combined these industries emit dust, salts and fine sprays into the surrounding air. Natural gas was used for the running the GTs with no fouling of hot section. The power measurements were taken with the GT being run in a controlled mode at base load hot gas inlet temperature. The results in (Figure 2-39 and Figure 2-40) have been corrected to new and clean guaranteed conditions. The following results show the outcome during the test.

- The power output deterioration without cleaning inclines to stabilise with increased operating hours. The unit test validates the figures that the output degradation was stabling at 90% base load (new and clean).
- After off-line washing (soak and rinse procedure) the power recovery was noticeably higher than following an on-line wash.
- On-line washing was carried out at time intervals between 700, 350 and 120 operating hours. The plant's performance is significantly higher at shortened washing intervals, therefore preventing the increase of power deterioration.
- Combining the two washing methods is the most effective and economical.
- By assessing the performed measurements and extending the operating time to 8000 hours it was calculated that performance could be enhanced equivalent to \$450,000 per year.

This would be managed with the combination of on and off-line compressor washing, on one 30MW GT. However operational and maintenance costs of the system were not considered in Figure 2-39 (Stalder J.-P. , 2001).

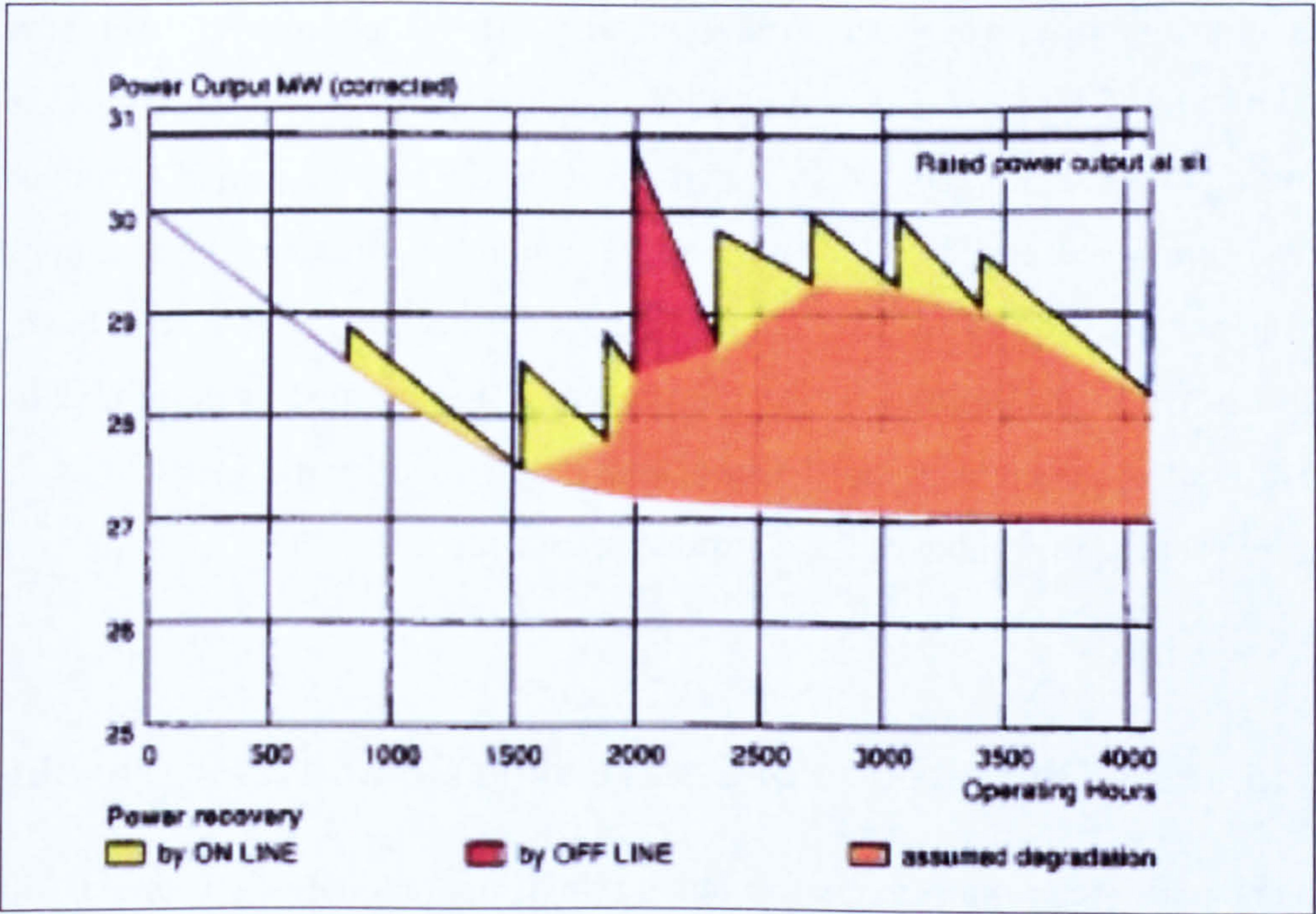


Figure 2-38: Likely effects of on-line and off-line compressor wet cleaning on power output (Stalder J.-P. , 2001)

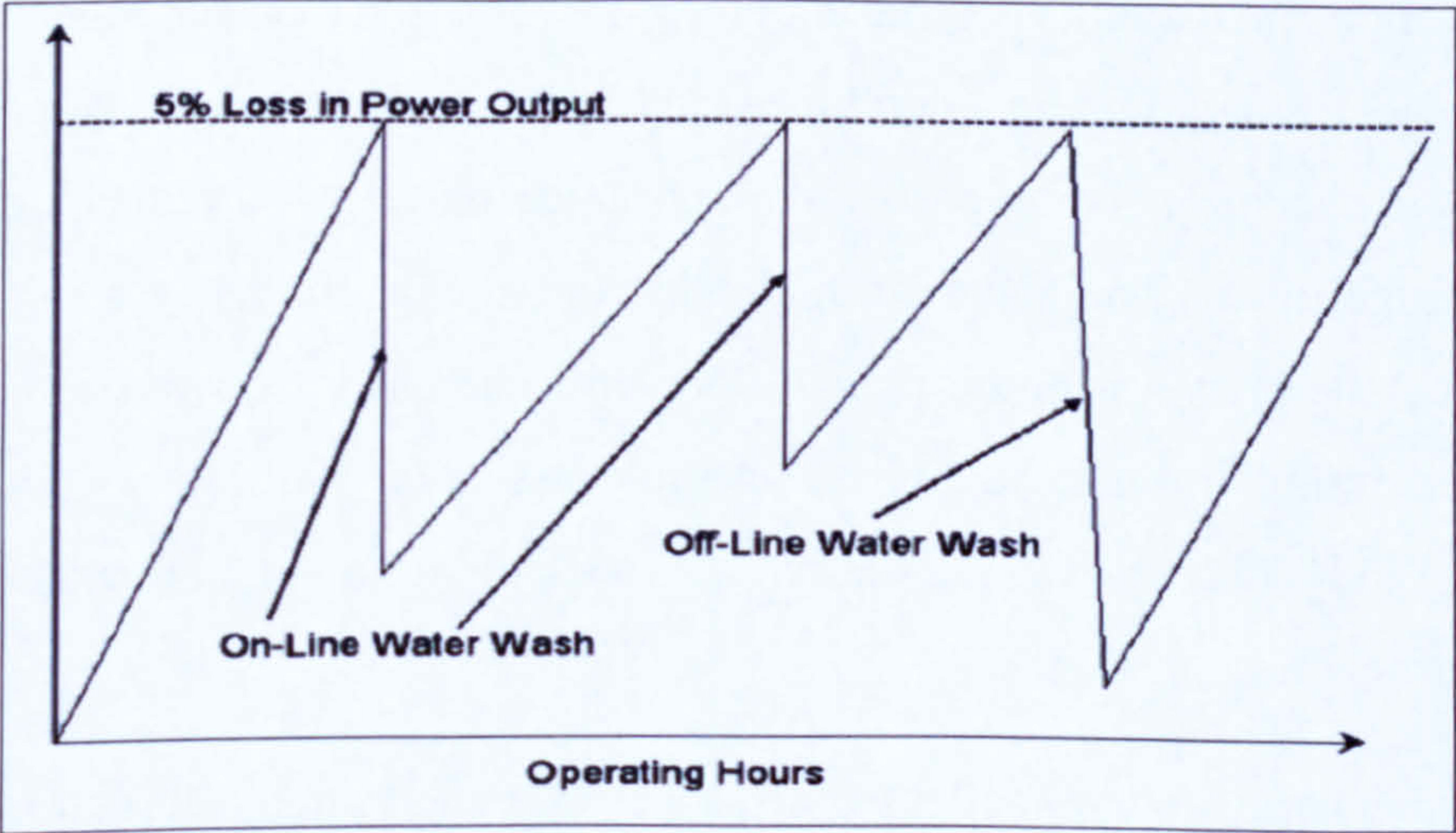


Figure 2-39: Effect of compressor water wash on power output (Boyce M. P., 2007)

2.8.7 Considerations on the Economical Profitability

For over 4000 operating hours when compressor cleaning is not performed the average power degradation is up to 10%. Diakunchak demonstrated that in these or similar conditions the

production profitability when performing on and off-line compressor wet cleaning was improved by \$1,175,000 over a period of 8000 operating hours which represents sizable extra profit. During the program \$20,000 was spent on chemical cleaner a small cost in comparison to the much improved profitability. Furthermore without an on and off-line compressor wet cleaning there would be extra loss of profitability because of the reduced steam produced which occurs with the fouling of the compressor in a combined cycle application. Steam production is affected more by the reduced mass flow than by the increase of the exhaust temperature because of fouling of the compressor (Stalder J.-P. , 2001).

2.8.8 Economic Optimisation of Gas Turbine Compressor Washing

Power plant parts deteriorate in performance whilst operational. As a direct result power output is lost and/or fuel consumption is increases, and consequently plant owners suffer reduced income. Operating costs and emissions rise with the increases fuel consumed, and equipment life cycle costs are increased. As a result of performance deterioration, higher firing temperatures increase the component creep life for a given power demand. A small 1% reduction in fuel consumption in a combined cycle power generation can result in more than a US\$1m reduction in operating costs, and CO₂ emissions may be reduced by 50 tons per year for a typical plant. The aim is to maintain compressor efficiency at high levels, thus saving fuel for the given operating points. Pollutants such as dust, hydrocarbon sprays and salts produced during normal operations results in degradation of efficiency (Hovland, G. and Antoine, M., 2004). Three common options are available to the plant operator in relation to compressor efficiency:

1. Continue normal operation letting efficiency degrade from the current level.
2. Perform on-line washing of the compressor.
3. Perform an off-line wash.

On-line washing where chemicals and water are injected into the air intake of the compressor does not require the plant to close down; this will clean the first stages of the compressor before the water evaporates. Off-line washing occurs after plant shut down. This method requires the blades of all of the compressor stages to be immersed in water and chemicals and because evaporation does not occur the water and chemicals stay on the compressor blades much longer. Therefore off-line washing normally restores the efficiency to a higher level.

However performing off-line washing requires plant shut down and, therefore, with planned regular on-line washing depending on the price of power sold and cost of fuel bought may prove more economical (Hovland, G. and Antoine, M., 2004).

2.9 On-line Water Wash Test

With the advantages of longer operating intervals and reduced degradation, on-line water washing has become increasingly interesting to operators of industrial GTs. Successful on-line washing requires close attention to the GT flow path geometry, the operating profile, and the nature of the airborne fouling at the compressor inlet (after filtration).

On-line washing equipment is readily available and many patents exist relating to GT water wash. There is currently no consensus on a recommended method for effective on-line water washing because important system parameters such as droplet velocity, droplet size, and fluid injection rate vary from one system to another. This makes it difficult for operators to select the best on-line water wash system for their application. Two recent publications give a historical review of on-line washing systems and a classification of available systems (Stalder J.-P. , 2001), (Mund, F. and Pilidis, P., 2004).

Today's on-line washing equipment for GT engines can be categorized into two main pressure ranges: Low pressure - fluid pressures up to 10 bar and high pressure - fluid pressures above 50 bar. The atomized droplets produced by high pressure systems have a droplet diameter typically less than 150 μ m and resemble the water occurring naturally in clouds and fog. Low pressure systems will have larger, drizzle-like droplets with diameter ranging from 100 to 500 μ m and even larger which may cause blade erosion in the compressor (Mund, F. and Pilidis, P., 2004), (Asplund, 1997). Thus low pressure systems can be augmented with air nozzles to generate smaller droplet sizes, resembling the droplets of the high pressure systems.

The fluid injection rate (water-to-air ratio) has an impact on the internal surface wetting of the compressor. Due to possible control instabilities, flame-out, or blade erosion the water-to-air ratio has generally been kept low. A typical on-line washing system for aero derivative engines has fluid injection in the range from 0.2% to 0.8% (mass based) (Mund, F. and Pilidis, P., 2004), (Asplund, 1997).

2.9.1 Engine On-line Water Wash Equipment

Accelerated engine deterioration was done through the ingestion of atomized saltwater (Caguiat, 2002). Further details of the deterioration method, equipment and results are given by (Syverud E. B., 2007).

The on-line water wash system consisted of a up to 18 nozzles positioned 770mm in front of the compressor inlet guide vane. Two manifolds were used, the first consisted of 12 air assisted, flat spray nozzles producing a mean droplet size of 25 μ m. The second had 18 interchangeable full cone spray nozzles which could produce a mean droplet size of either 75 or 200 μ m. To maintain comparable flow rates the numbers of nozzles in each test was varied. Figures 2-41 and 2-42 show the two manifolds, with water flow rates of about 4.4 l/min. The first manifold had 12 nozzles producing water droplets of mean diameter 25 μ m, and the second had 8 nozzles producing droplets of mean diameter 75 μ m. The volume flow rates were 4.4 l/min and 4.3 l/min respectively. Both arrangements gave good droplet coverage across the required area, and the direction of the spray from the injectors and their closeness to the GT bellmouth gave good control of the droplet size at the compressor face. The larger drops had the greater injection velocity.

Following (Syverud, E. and Bakken, L. E., 2007) and applying ISO Standard 9272–2 (Calculation of average particle sizes/ diameters and moments from particle size distribution – the droplet sizes were measured in still air using a Malvern laser diffraction spray analyzer. Because the nozzles were close and tilted towards the centerline, there was a slight variation in the size of the drops with the number of nozzles in use. Due to multiple reflections and scattering of the laser beam droplet diameters could not be measured for the largest flow rates. The droplet diameters shown in Table 4 were measured at the centre of the nozzle arrangement for fully developed spray, 440mm in front of the nozzles.

Figure 2-43 shows a schematic diagram of the water wash system using the air assisted nozzles shown in Figure 2-41. The manifold shown in Figure 2-42 was operated at full water pressure (24 bar), with no air assistance. Here tap water with no additives or detergents was used in all tests but for field on-line washing, the water quality must conform to the requirements of the GT manufacturer.

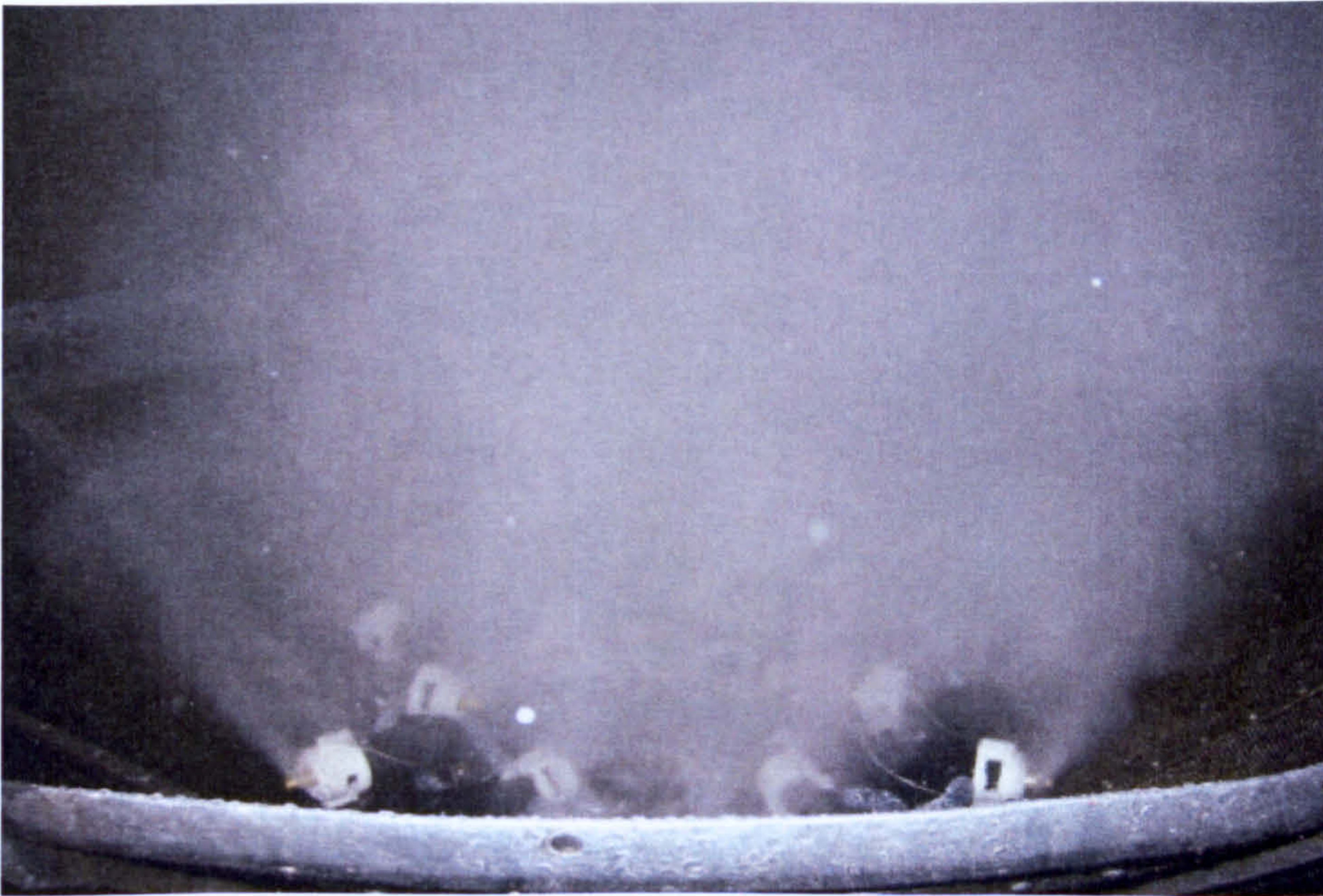


Figure 2-40: Water wash manifold with 12 nozzles giving 25 µm droplets at 4.4 l/min (Syverud, E. and Bakken, L. E., 2007)

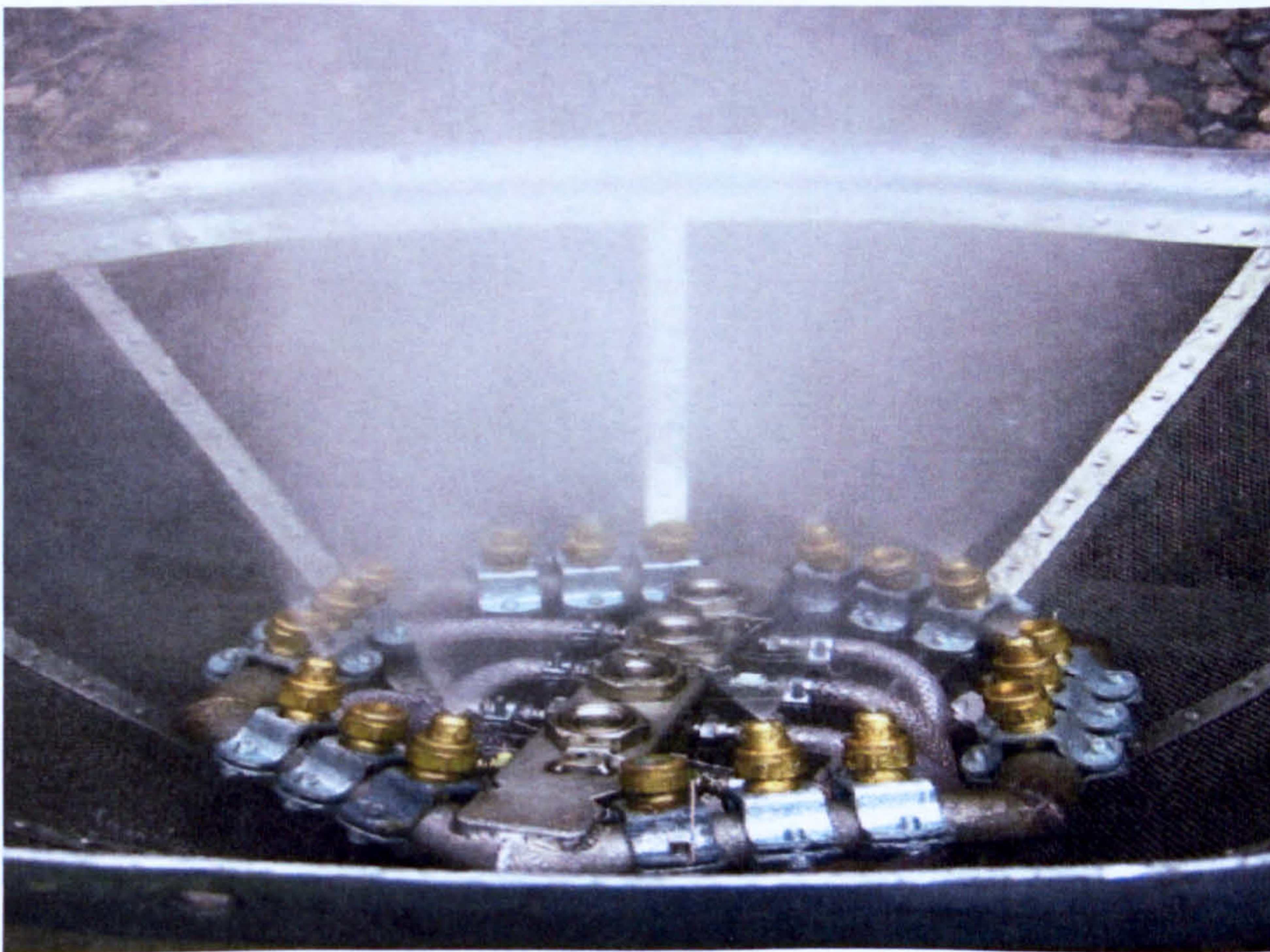


Figure 2-41: Water wash manifold with eight nozzles giving 75 µm droplets at 4.3 l/min (Syverud, E. and Bakken, L. E., 2007)

Table 4 Measurement of water droplet size for nozzle arrangements is reported in table 1 together with the droplet size data provided by the manufacturer (Figures 2-41 and 2-42) after (Syverud, E. and Bakken, L. E., 2007)

Table 4: Comparison between measured and the manufacture data of droplet size

Test number	Droplet size (μm_{VMD})		Water flow rate (l/min)	Water-to-air ratio (%)
	Manufacturer's data	Measured diameter		
1	200	131	8.8	0.87
2	200	139	4.4	0.43
3	75	83	9.1	0.89
4	75	93	4.3	0.42
5	25	39	4.4	0.43

Results

The engine performance deterioration and restoration was analyzed through changes in the intake depression and in the compressor stage work coefficient. Intake depression is defined as the deviation between the total and static pressure at engine inlet.

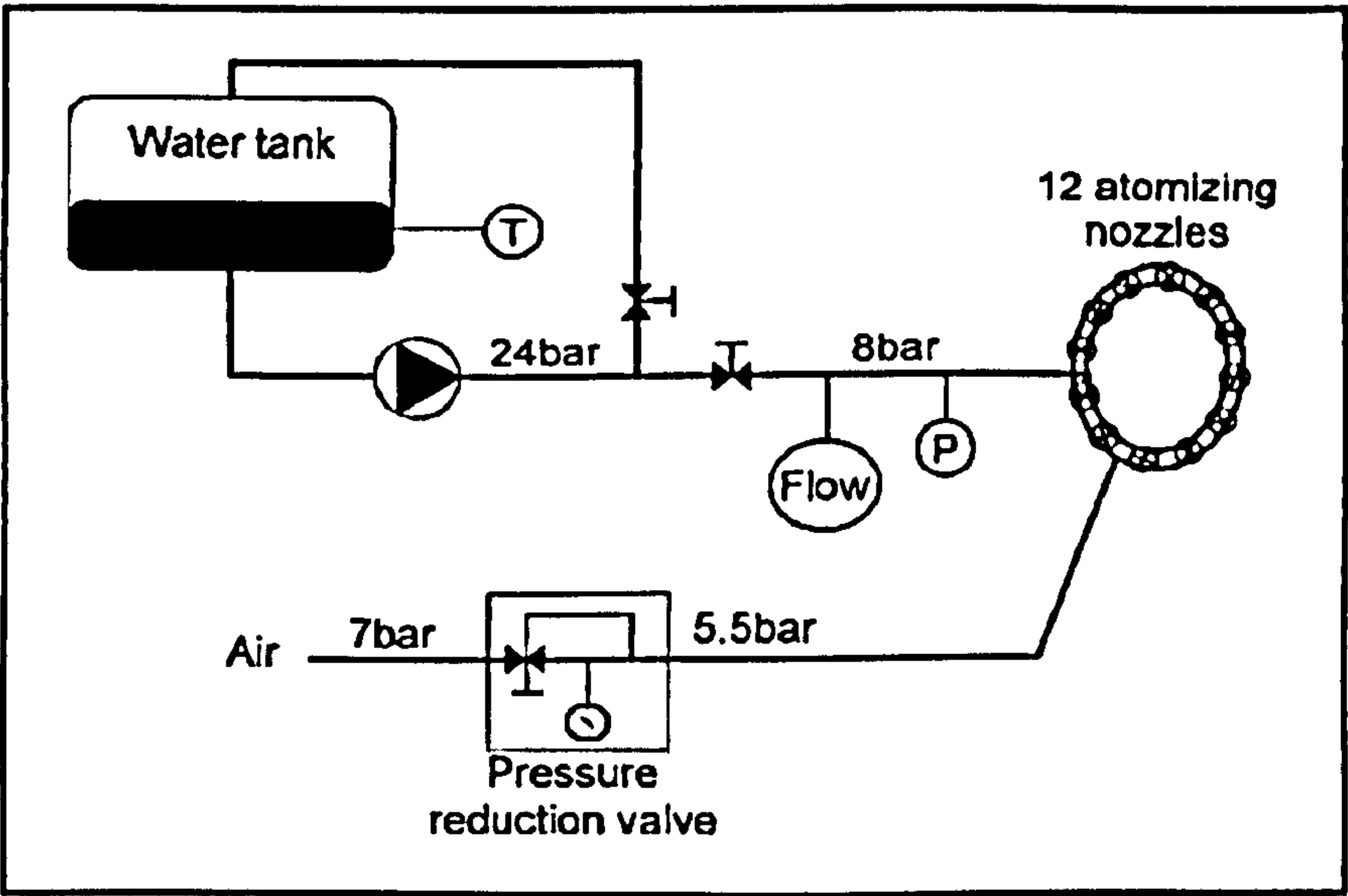


Figure 2-42: Water wash system schematic (Syverud, E. and Bakken, L. E., 2007)

- **Water-to-Air Ratio.**

A GE J85-13 GT was operated without stability or structural problems at water injection rates of 40 l/min at all speeds from 70% to full speed. This represents water-to-air ratios (by mass) of over 3% at full speed and 7% at 70% speed. Such high water-to-air ratios are not recommended for on-line water washing because they may cause excessive blade loads and erosion damage. Figure 2-44 shows the change in intake depression as a function of corrected engine shaft speed for three different water-to-air ratios with 200µm droplet size and 1 min water injection time. Figures 2-45 and 2-46 show the change in stage work coefficient for the same cases as given in Figure 2-44. Stage four performance is not restored with the smallest water-to-air ratio of 0.43% (Syverud, E. and Bakken, L. E., 2007).

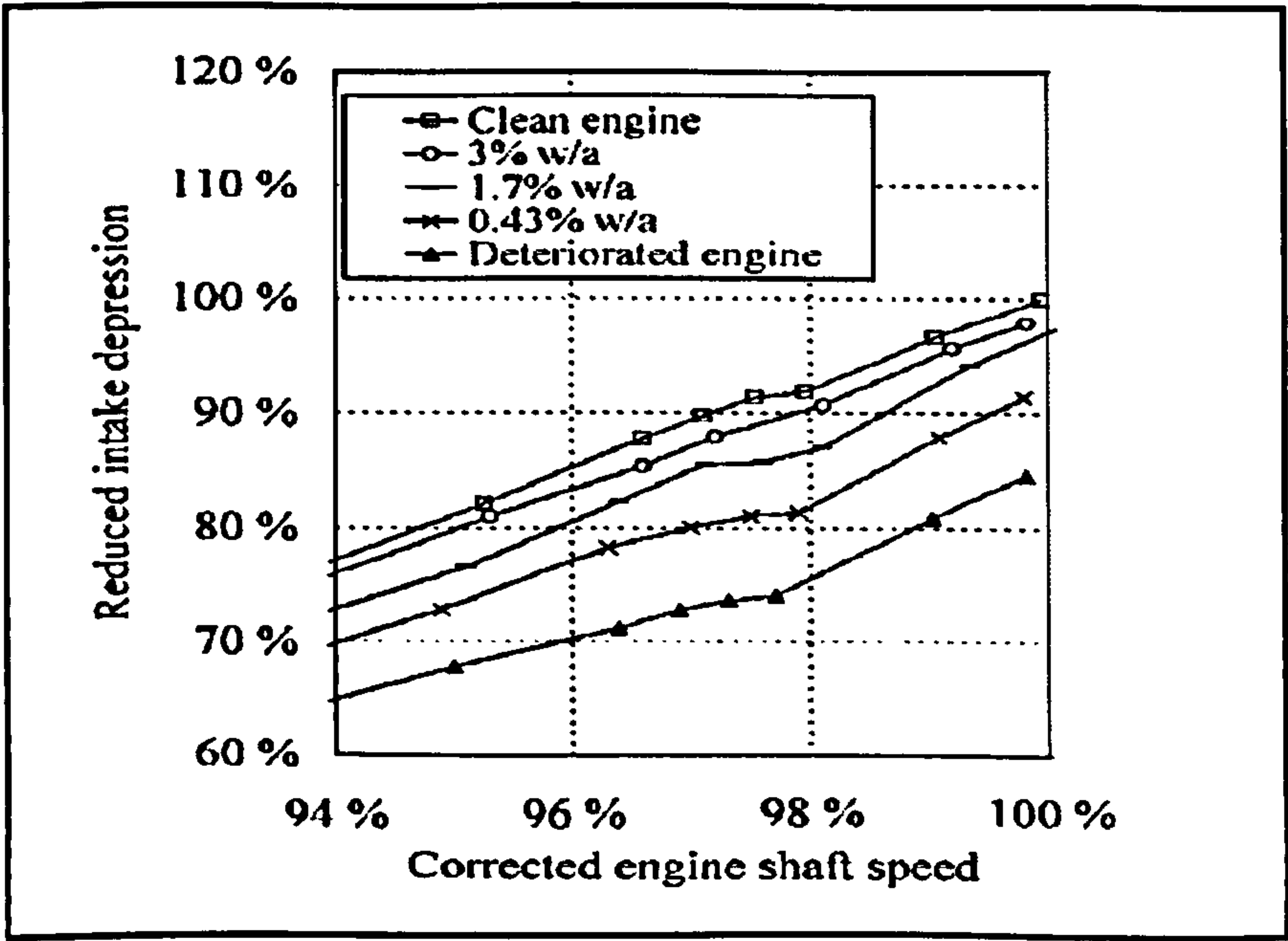


Figure 2-1: Intake depression recovery after 60 s water washes with 200 µm droplets (Syverud, E. and Bakken, L. E., 2007)

The remaining salt deposits reduce the effective flow area and caused a reduction in the stage six flow coefficient. With a 3% water-to-air ratio the performance of the sixth stage is restored. The tests clearly document the impact of increased water-to-air ratio on the compressor performance recovery. At low water-to-air ratios, performance recovery is limited.

- **Water Injection Time**

Increasing the water injection time of on-line water wash is expected to improve performance recovery when the engine is fouled with water soluble components such as salt (Stalder J.-P. , 2001).

Although an equivalent mass of water is used in the 1.7% 1/min wash and the 0.43% 4 min wash, the increased water injection time at the lower flow rate fails to recover engine performance. With the lower water-to-air ratio of 0.43%, the water seems to evaporate in stage four and the expected benefit from increased soaking time is therefore not apparent. The tests show that water injection time is only an issue in the front stages where the wash fluid is not evaporated. For rear stage performance recovery, it is better to inject the water in a shorter time, hence, at a higher water-to-air ratio (Syverud, E. and Bakken, L. E., 2007).

Table 5: Analysis of water used for on-line water wash

Parameter	Value
Total matter	< 1 mg/l
pH	7.52
Sodium	2.1 ppm by weight
Potassium	0.63 ppm by weight
Chloride	1.7 ppm by weight

Figure 2-47 compares the change in intake depression for three different cases with 200µm droplet size. The total water injected in the 4 min water wash at 0.43% water-to-air ratio is equivalent to the total water injected during the 1 min water wash at 1.7% water-to-air ratio. The case of the 0.43% 1/min water wash case is included for reference. The impact on stage four and six work coefficients are given in Figs. 2-45 and 2-46, respectively. Although an equivalent mass of water is used in the 1.7% 1/min wash and the 0.43% 4 min wash, the increased water injection time at lower flow rate fails to recover engine performance (Syverud, E. and Bakken, L. E., 2007).

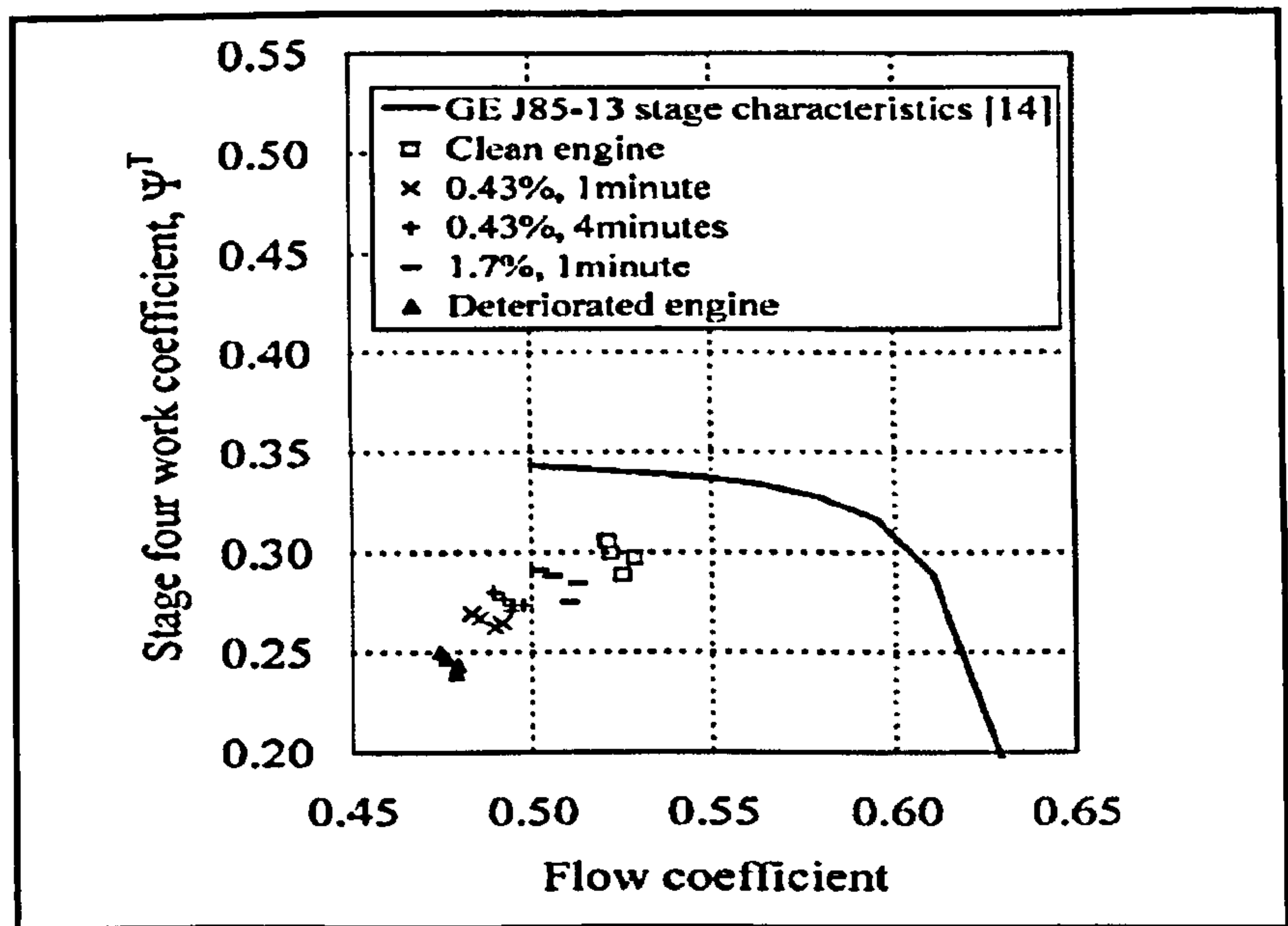


Figure 2-2: Stage 4 work coefficient after water wash with 200µm droplets at different water injection times (Syverud, E. and Bakken, L. E., 2007)

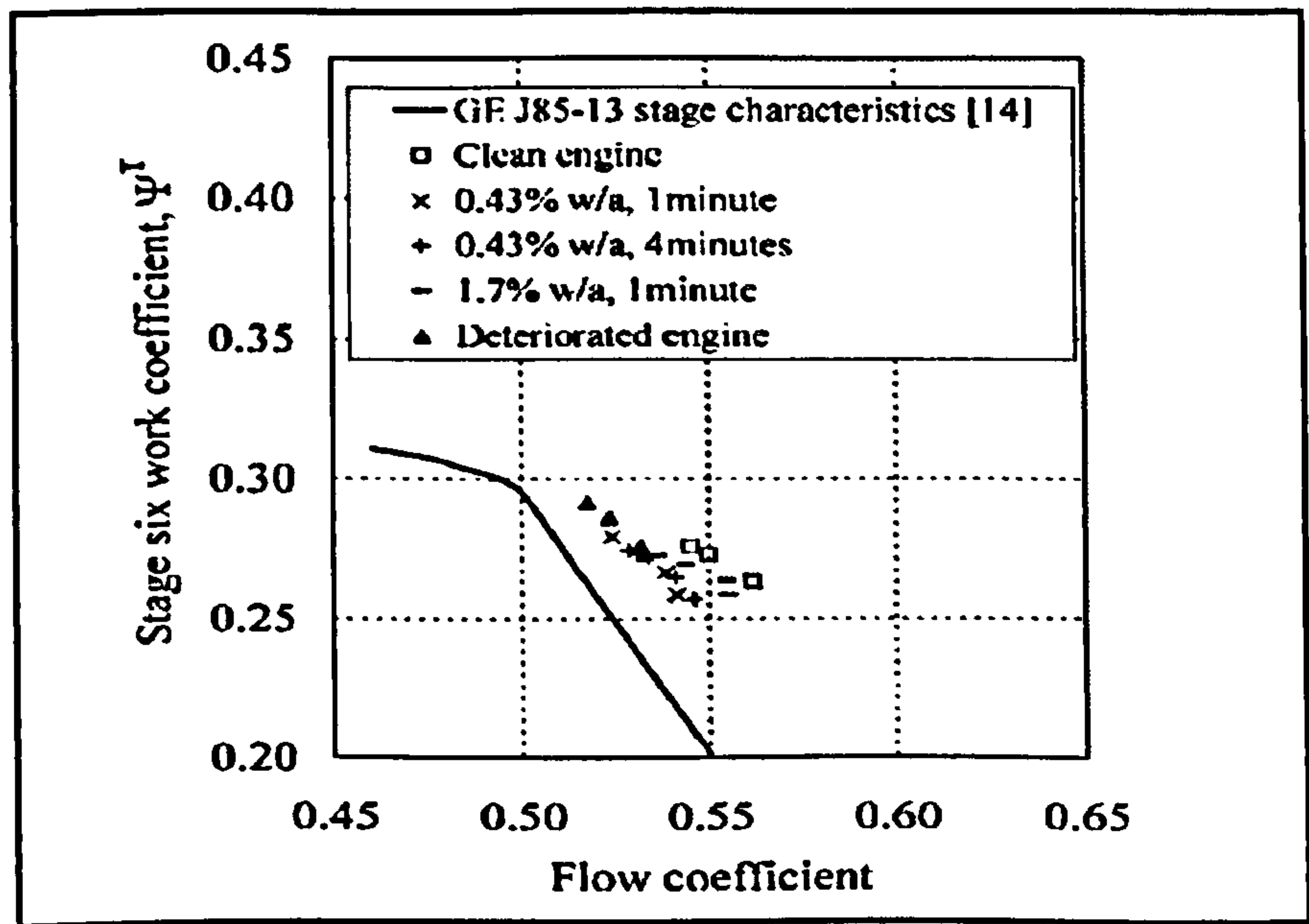


Figure 2-3: Stage 6 work coefficient after water wash with 200µm droplets at different water injection times (Syverud, E. and Bakken, L. E., 2007)

With the low water-to-air ratio of 0.43%, the water seems to evaporate in stage four and the expected benefit from increased soaking time is therefore not apparent. The tests show that water injection time is only an issue in the front stages where the wash fluid is not evaporated.

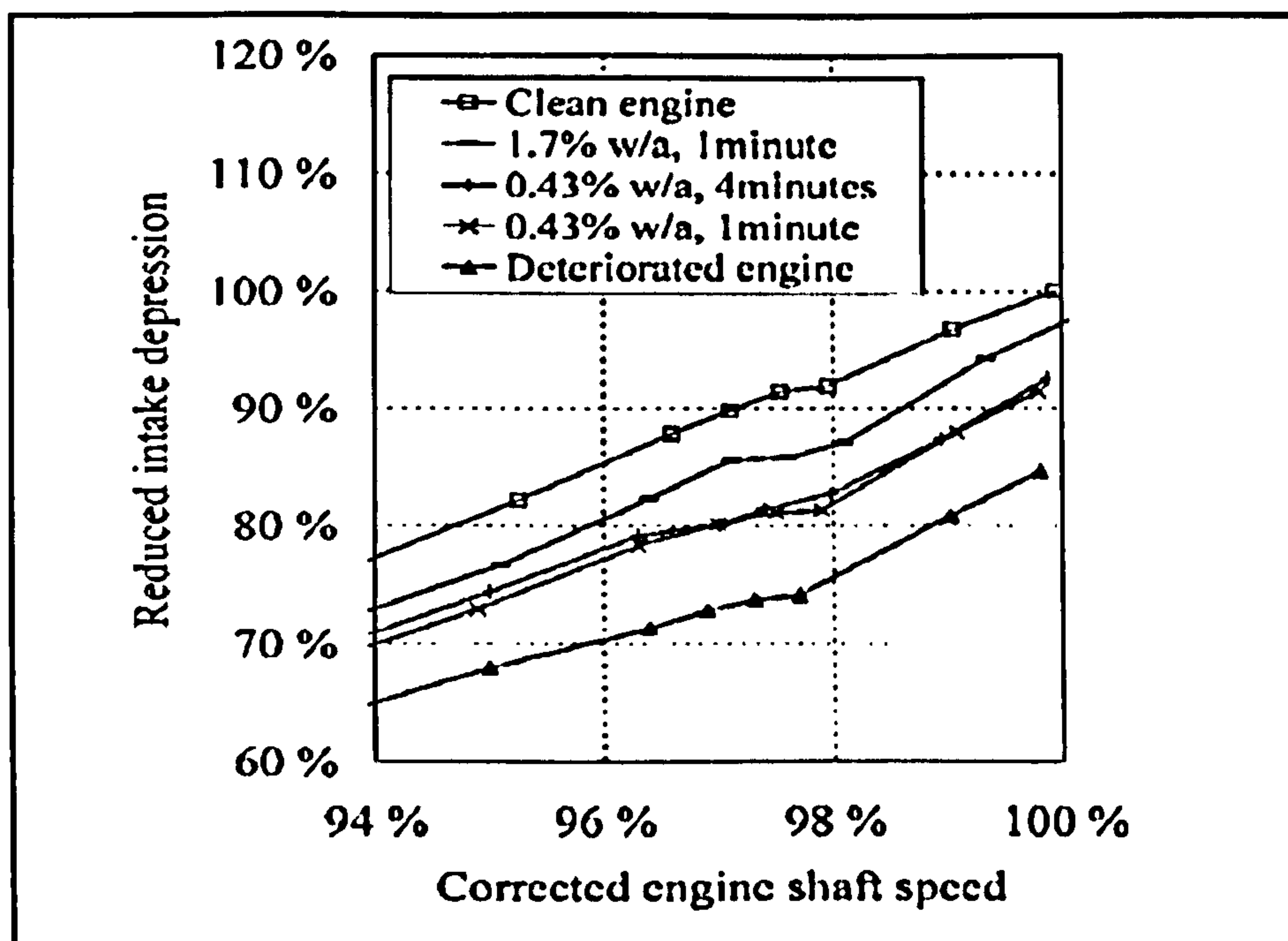


Figure 2-4: Intake depression recovery at different water injection times with 200 μ m droplets (Syverud, E. and Bakken, L. E., 2007)

2.9.2 Droplet Size

Droplet size is considered important for performance recovery in the rear compressor stages. Water flow path, surface wetting, evaporation rate, and erosion are all affected by the droplet size. Figures 2-48 compares the intake depression restoration and the stage four and stage six work coefficient recovery after 1 min on-line water wash at a water-to-air ratio of 0.4% and with various droplet sizes (Syverud, E. and Bakken, L. E., 2007).

The 75 μ m droplets are the most effective in recovering the overall engine performance, and in cleaning the fourth stage; however, the smaller (25 μ m) droplets seem to redeposit the fouling in the sixth stage and cause heavy deposits in the sixth stage, increasing sixth stage deterioration. The tests show that small droplets increase the fouling in the rear stages.

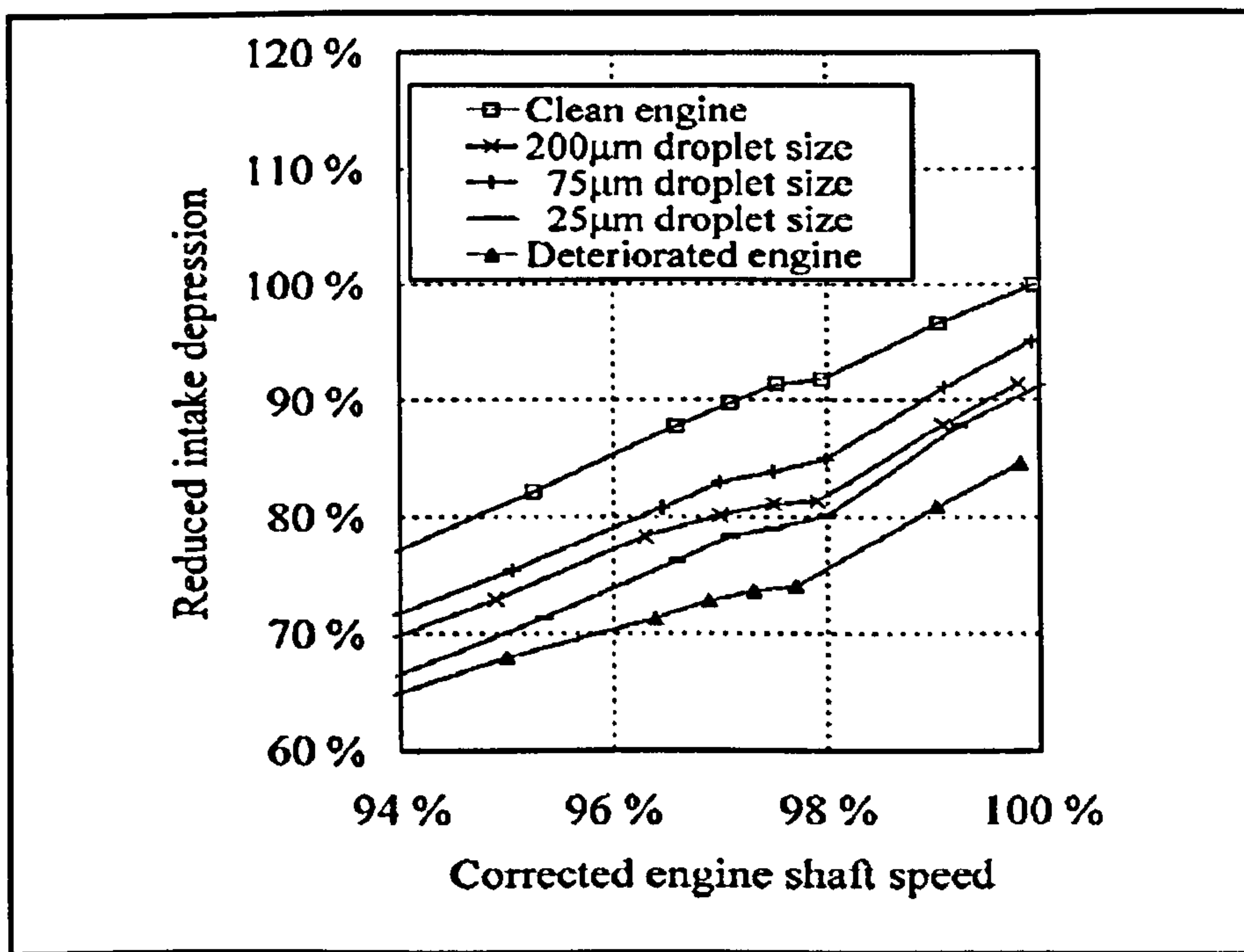


Figure 2-5: Intake depression recovery after 1 min water wash with various droplet sizes at 0.4 % w/a (Syverud, E. and Bakken, L. E., 2007)

- **Salt Distribution After Water Wash**

Further evidence of the movement of salt from the front stages to the aft stator vanes is given in Figure 2-49. The figure compares the salt deposits on the stator vanes after degradation to the salt distribution after a 1 min water wash with 75µm droplets and 0.42% water-to-air ratio. The salt deposits were measured as chlorine content and converted to weight distribution of salt (sodium chloride).

After degradation, immeasurably small traces of salt were found on the aft stages. However, after the water wash, the salt had moved from the front stages, aft, re-deposited on all stages from stage four. Rotor deposits were not possible to measure due to practical limitations; however, from visual observation, the salt deposits after water wash were found to be heaviest on the suction side of stages 3, 4, and 6 and on the pressure side of stage 5.

- **Velocity Triangles**

As a visualization of the change in stage loading with deterioration and on-line water wash, the velocity triangles for stage 4 in the clean, deteriorated and washed conditions are shown in Figure 2-50. The data are for the 98% operating point, where the engine shaft speed varied from 16,204 to 16,162 and 16,208 rpm for clean, deteriorated, and washed conditions,

respectively. This change in blade speed was negligible when comparing the velocity triangles. The water wash data are for 75 μ m droplets, 0.42% water-to-air ratio, and 1 min water injection time. The data are calculated based on the assumptions above for the calculation of stage work (Syverud E. B., 2007). As discussed above, the deviation angles are assumed constant for the clean and deteriorated cases. The velocity triangles show that the deterioration causes a reduction in the flow coefficient and a change in the stage work coefficient.

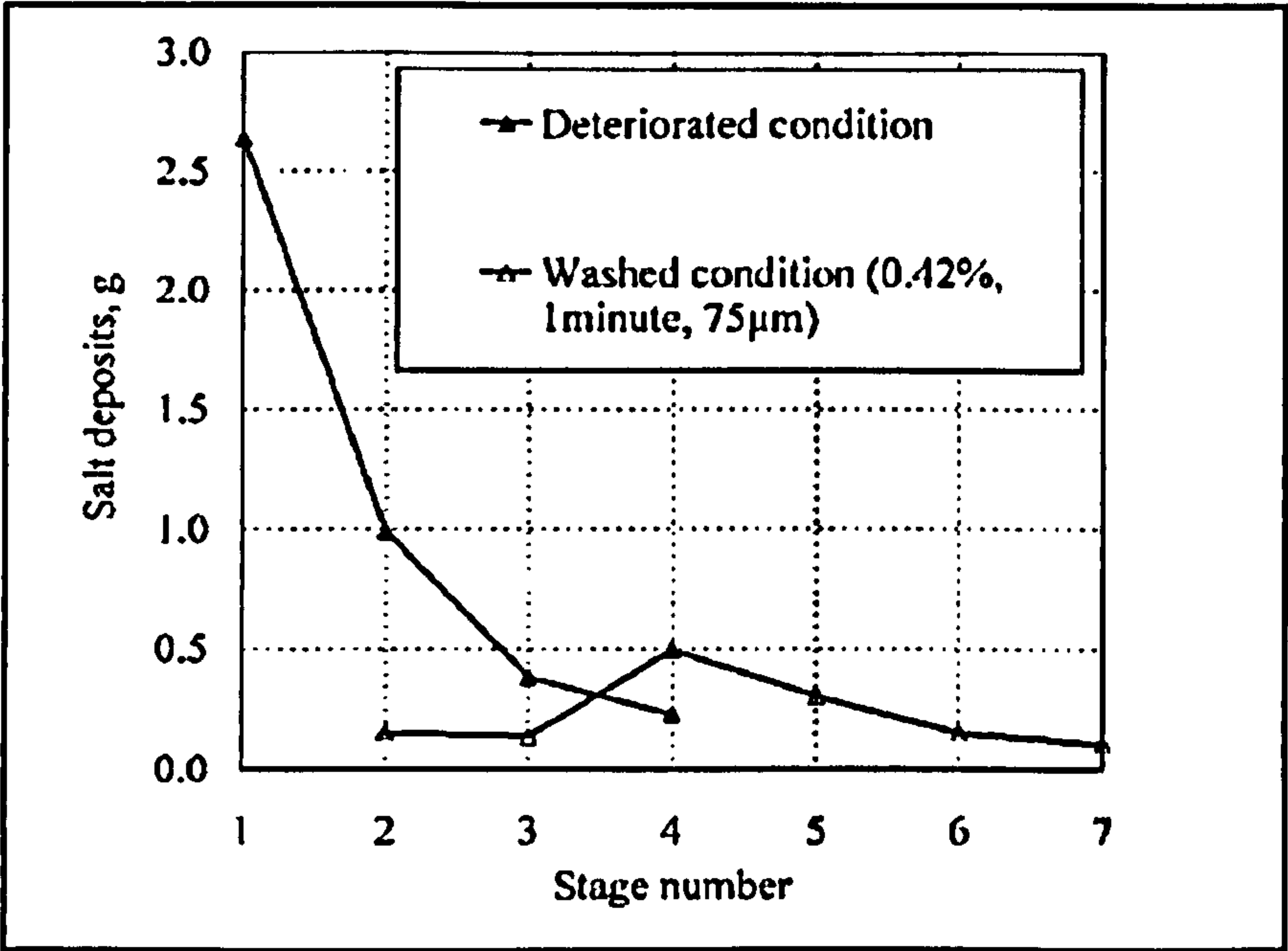


Figure 2-6: Weight distribution of salt deposits on stator rows (Syverud, E. and Bakken, L. E., 2007)

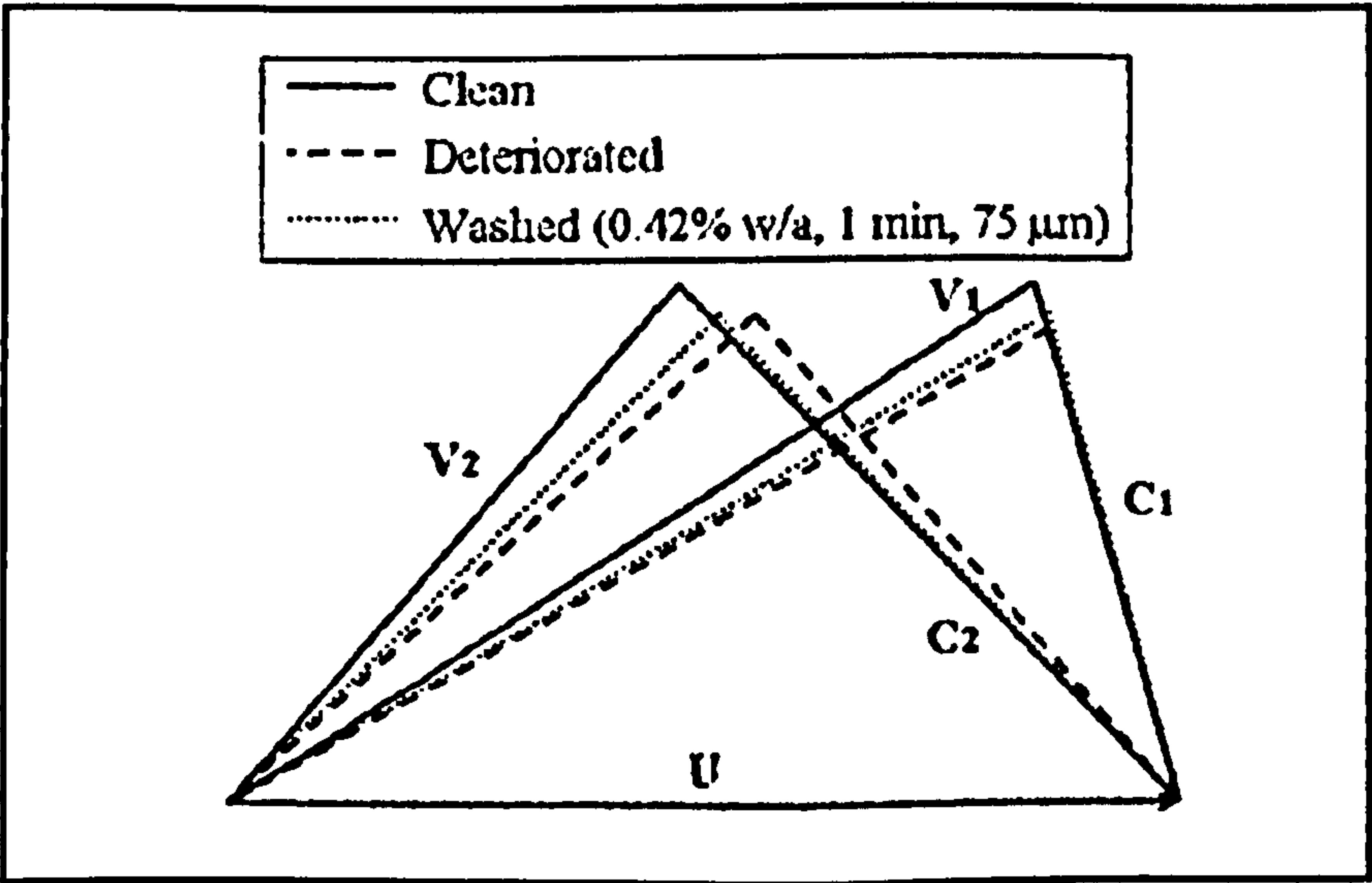


Figure 2-7: Stage 4 velocity triangles at 98% shaft speed (Syverud, E. and Bakken, L. E., 2007)

2.10 Field Experience of On-line Washing

GT operators now accept on-line compressor washing as an established practice. For example the Netherlands Division of Electrabel have operating five GE Frame 9-FA GT units since 1995, these are subject to a daily on-line wash and one off-line wash every year when the units are shut down for the annual inspection and maintenance. The on-line water wash (OLWW) systems used is the Turbotect Mk1 nozzle design and was supplied by GE themselves (Oosting, J. and Stalder, J. P., 2007).

The OLWW regime was suspended in mid-2001 after receiving a recommendation from original equipment manufacturer (OEM) that the first stages of the compressor be inspected for erosion marks. This recommendation was issued because erosion had been seen on the compressor blades in some engines where the OLWW system had been used frequently. Simultaneously, a number of other GTs at the Netherlands Division of Electrabel had undergone their first major overhaul which allowed examination of the first row blades of the compressor for signs of erosion. In both cases only minor erosion was found after more than seven years of operation, during which period there had been daily on-line washes. However, because of erosion concerns the OEM-recommended more frequent inspections and that measures be taken to reduce the rate of erosion due to droplet impingement. Electrabel instigated an independent investigation to look for ways of reducing erosion rates while the on-line washing continued.

This project presents the results of the investigation which covered 24 months of routine on-line compressor washing. One outcome was that the Turbotect Mk3 on-line water wash system confirmed that it mitigated erosion risk on the leading edge of the R0 compressor blade, and so decreased the number of blending operations over the life time of the R0 compressor blades. This system for on-line compressor cleaning of large GTs achieved a lower rate of power degradation compared to the previous Mk system (somewhere between 30% and 40%) and substantially improved cleaning effectiveness (Oosting, J. and Stalder, J. P., 2007).

- **Plant Description**

The 2,470 MW Eemshaven power plant complex is located in the North East Netherlands. The site comprises two different plants: the first one has been operational since 1977 and is a steam turbine power plant which was upgraded with a Siemens V 94.2 GT to give a

combined total output of 697 MW. An extension with a new combined cycle plant of 1,775 MW started commercial operation in 1995. It comprises 5 x Frame 9-FA engines each of 225 MW, and each with a heat recovery boiler and one steam turbine on the same shaft. The total combined output per unit is 355 MW. The plant is fired with natural gas from the Groningen field (Pfleger, 1998). The Eems GTs are among the fleet leaders in the Frame 9-FA engine population. Until 2000 the plant used to be operated at base load. However, because of recent natural gas price increases the plant is now operated in cycling mode with many more starts and shut downs. As of September 2006 the units have logged the following fired hours and starts:

Table 6: Fired hours and starts as of September 25th, 2006.

Unit	EC-3	EC-4	EC-5	EC-6	EC-7
Fired hours	78,191	77,086	80,222	76,737	71,274
Fired starts	744	834	636	599	702

- Plant Environment**

The Eemshaven power plant is located directly on the shore of the North Sea. The main concerns for air inlet filtration are the coastal marine atmosphere with salt-laden winds and high humidity, as well as seasonal dust concentrations due to agricultural activities.

2.10.1 Compressor Cleaning Regime

Since the beginning of commercial operation in 1995 and until June 2001 all GT compressors at Eems were washed on-line, daily in summer and 3 times per week in winter for approx. 30 to 40 minutes. The anti-icing heat exchanger is started for on-line compressor wash when the ambient temperature is below +15°C. Additionally an antifreeze agent is added to the cleaning solution when the ambient temperature is below +10°C. Based on the local ambient temperature profile, the winter OLWW period covers on average about 7 months and the summer period approx. 5 months of each calendar year.

As a result of the first request received from the OEM in June 2001 to inspect at the first opportunity and then annually for erosion on the R0 compressor blades, the on-line washing regime was first reduced to 3 times per week for 10 minutes during summer and winter. Later

in 2001 the OEM recommended to wash on-line only up to 5 minutes daily (total 35 min. weekly) until 100 OLWW hours are reached. However, in September 2002 after inspection results from two rotors which showed no unexpected roughness on the R0 blades, measured by molds as instructed by the OEM, the washing regime was extended again to 3 times per week for 17 minutes in summer and twice per week for 17 minutes in winter; see Table 7 below.

Until 2001 a single compressor off-line wash used to be made on average every year on each engine, when it is shut down for the annual inspection or for maintenance outage. For instance, a performance drop of about 4 to 5 MW was measured prior to June 2001 outage on unit EC-3 after some 10,000 operating hours. Interestingly, a higher power drop due to compressor fouling resulted after reducing the on-line cleaning regime, as recommended in the TIL letters in 2001. The consequence of this reduced OLWW regime was that the number of off-line compressor washes was doubled to two per year. As mentioned above, the new strategy for compressor off-line cleaning recently introduced is to wash off-line when the CC (GT + ST) has reached a combined 4 MW power degradation (approx. 1.1%), or at first opportunity when a unit shuts down.

From visual inspection it can generally be stated that after some 8,000 operating hours the compressor blading is always in a relative good condition. Throughout the compressor rotor stages, only a thin layer of deposits covers the blade surfaces. The deposits are smooth, slightly oily/sooty, not sticky, and can be cleaned off easily (Oosting, J. and Stalder, J. P., 2007).

Table 7: On-line water wash, frequency and duration

Time period	Summer	Winter
1995 up to June 2001	Daily 35 min.	3 x 35 min/week
June 2001 to Aug 2002	3 x 10 min/week	3 x 10 min /week
Sept. 2002 onwards	3 x 17 min/week	2 x 17 min /week

2.10.2 Original On-line Water Wash (OLWW) Design Data

As mentioned above the Eems engines were originally supplied with the Mk1 OLWW systems. Both the OEM and the Mk1 OLWW nozzle systems are of the low pressure type.

Table 8: OLWW nozzle systems comparison (Oosting, J. and Stalder, J. P., 2007)

GE Frame 9-FA	OEM OLWW	Mk1
No. of nozzles	18	38
Water mass flow	38gpm / 144 l/min	7.7 gpm / 29.3 l/min
Water pressure	100 psi / 6.9 bar	58 psi / 4 bar

The design philosophy of Turbotect is to position a relatively large number of nozzles in the air inlet casing, in both the up-stream side around the cone and on the down-stream side around or above the bell mouth. This ensures a better spatial distribution of the injected droplets and consequently provides improved wetting and cleaning characteristics as shown in Figure 2-51.

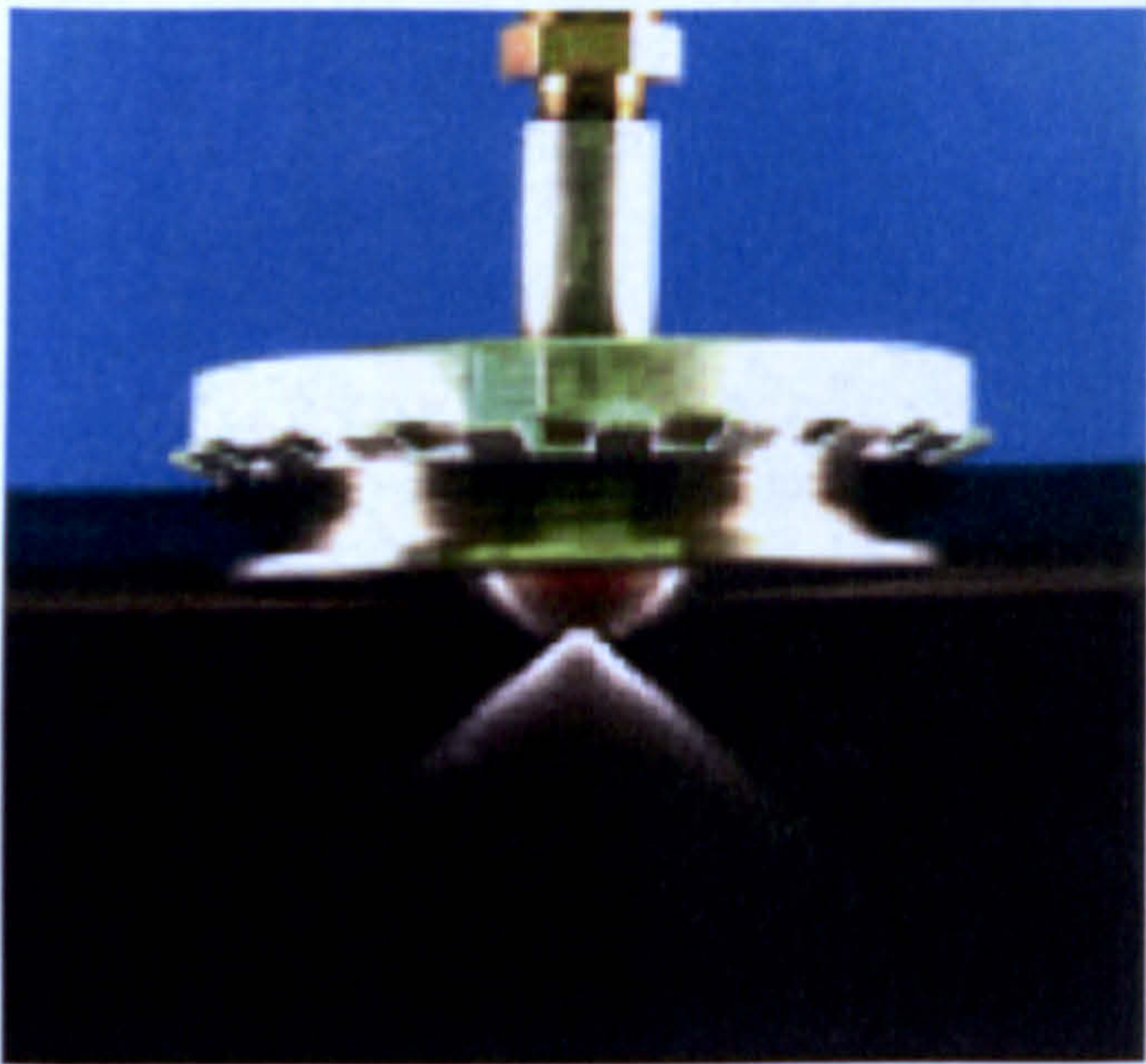


Figure 2-8: The Mk1 on-line injection nozzle (patented) allowing orientation of the spray (Oosting, J. and Stalder, J. P., 2007)



Figure 2-9: Molds applied on leading edge of R0, 1st stage, compressor blades on Unit EC-6 (Oosting, J. and Stalder, J. P., 2007)

- **Roughness after 7 years OLWW on unit EC-7**

Boonstra presented in 2003 plant experience gained between 1995 and 2002 with OLWW and the measured R0 compressor blade roughness on Unit EC-7. Mold measurements of the R0 leading edge (see Figure 2-52) were made at the major inspection after 51,936 operating hours. The measured roughness mean value was 8.9 mils (226 micron). Based on the washing regime practiced over this time period, the number of compressor on-line washes was 1,500 for a combined total of 790 wash hours. The average wash time was approx. 31 minutes per wash. The total cleaning solution injected with the Mk1 OLWW system over this time period was 1,389m³. The roughness measurements made on the other GT engines were in the same range (Oosting, J. and Stalder, J. P., 2007).

- **Initial R0 leading edge roughness after blending operation**

Blending means polishing of the roughness followed by shot peening. A mold measurement was made to establish initial surface roughness on the R0 compressor blades after blending. A freshly blended R0 compressor blade can have an average initial roughness of up to 2.6 mils (65 micron) before entering service.

Table 9: Initial roughness on blended R0 compressor blades

	Average measured roughness on R0 blade leading edge
After polishing	25 micron / 1 mils
After shot peening	65 micron / 2.6 mils

The logistics required for regular mold measurements be taken and roughness evaluations made with special optical apparatus, including scheduling, blade disassembling, refurbishing operation and re-blading are considerable. Factors such as overall costs for blending, non-availability, personnel involvement and time spent are significant.

2.10.3 Leading edge roughness measurement of R0 compressor blades

As instructed by the OEM, the R0 compressor blade roughness was measured by taking molds at 100 OLWW hours which corresponds, to the practiced OLWW regime (frequency and duration), to the interval between two hot gas path inspections (HGPI). The R0 compressor blades are uncoated. The results are tabulated for Unit EC-3 to EC-6 in table 10 below (Oosting, J. and Stalder, J. P., 2007).

Table 10: Measured roughness on R0 blades after blending

Unit	EC-3	EC-4	EC-5	EC-6
OLWW	Mk1	Mk1	Mk1	Mk1
Date	17 Oct 05	29 Nov 04	20 Jan 05	11 May 04
Op. hours	21,900	17,869	20,821	23,352
Meas. avg. roughness	182 μ m 7.2 mils	170 μ m 6.7 mils	147 μ m 5.8 mils	199 μ m 7.8 mils
OLWW Hrs.	92 hrs.	73 hrs.	82 hrs.	107 hrs.

The roughness measurements are the average of 3 measurements made per mold and from 12 blades selected evenly over the circumference. Roughness measurements were also made at

the fillet radius of the blades and their results were always lower than the ones measured at the leading edge near the root. Averages of all performed leading edge roughness measurements were below the OEM criterion of 10mils (254micron).

- **Unit EC-6: OLWW upgrade from the MK1 to the MK3 nozzles**

In a move to further reduce the erosion rate by OLWW and to improve the cleaning efficiency, the on-line water wash system for unit EC-6 was upgraded to the Mk3 on-line wash nozzle system (US Patented), specifically designed for use in large GTs as shown in Figure 2-53 and 2-54.

Table 11: Design data comparison of the OLWW nozzle systems for the Frame 9-FA

OLWW System	Mk1	Mk3
No. of nozzles	38	30
Water mass flow	7.7 gpm / 29.3 l/min	4.6 gpm / 17.6 l/min
Water pressure	58 psi / 4 bar	58 psi / 4 bar
Air pressure	—	58 psi / 4 bar
Air mass flow	—	6.12 Nm ³ / min 216 ft ³ / min

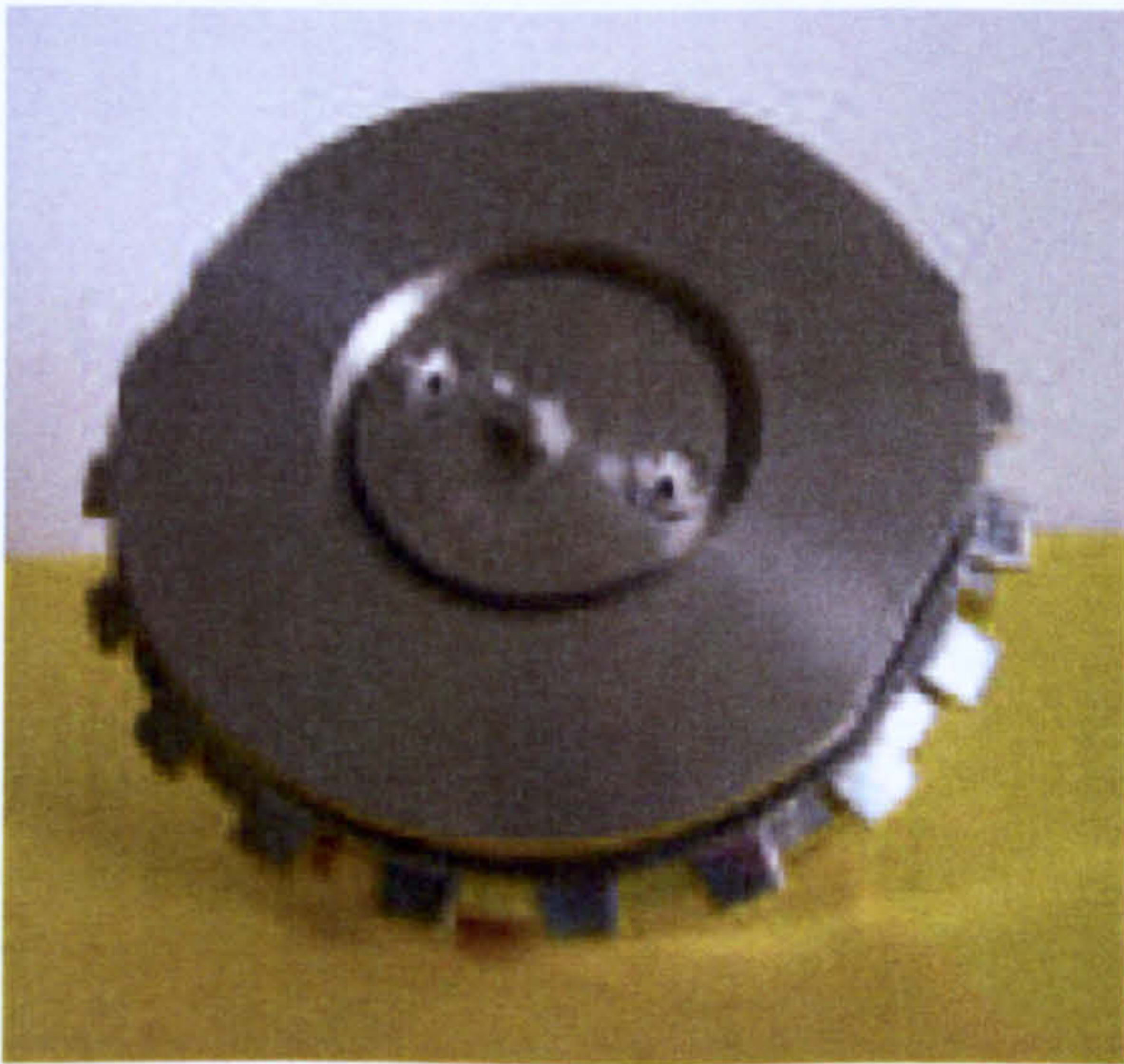


Figure 2-10: The Mk3 on-line nozzle assembly (patented). The air nozzles are located on each side of the water nozzle (Oosting, J. and Stalder, J. P., 2007)



Figure 2-11: The water spray of the Mk3 on-line nozzle pictured at base load (Oosting, J. and Stalder, J. P., 2007)

Table 12: Erosion increment on EC-6 with the Mk3 nozzle system and regular OLWW

OLWW on EC-6	Up to May 2004	June 2004 onwards	
OLWW system:	Mk1	Mk3	
R0 blade; operational hours accumulated before blending in 2001	17,336		
Start date of new operating period	In 2001	10 June 04	Dec. 05
Date erosion meas.	11 May 04	14 Dec. 05	30 Jun 06
Add. operating hrs.	+ 23,352	+ 11,986	+ 3,710
Total since blending	23,352	35,338	39,048
Grand total R0 blade	40,688	52,674	56,384
Average measured roughness	199 μ m 7.8 mils	208 μ m 8.2 mils	190 μ m 7.5 mils
No. of washes	384	+ 181	+ 51
No. of wash hours	107 hrs.	+ 52 hrs.	+ 15 hrs.
Erosion increment over the period:		+ 9 μ m + 0.35 mils	- 18 μ m - 0.7 mils
Per 100 wash hours:		17 μ m/0.67mils	
Per 1,000 op. hours:		0.75 μ m/0.03mils	

- Unit EC-6: Discussion of the achieved results in erosion increment

The roughness measurements as tabulated in tables 8 and 10 above are plotted in Figures 2-55 and 2-56. The measured roughness at approximately 20,000 operating hours form a cluster, indicating that they have a similar erosion rate pattern for all engines and well below the 10 mils (254 micron) limit. The roughness pattern shows a steep rate of erosion increase at the early beginning of cumulative OLWW wash hours and thereafter the rate of erosion flattens out (Oosting, J. and Stalder, J. P., 2007).

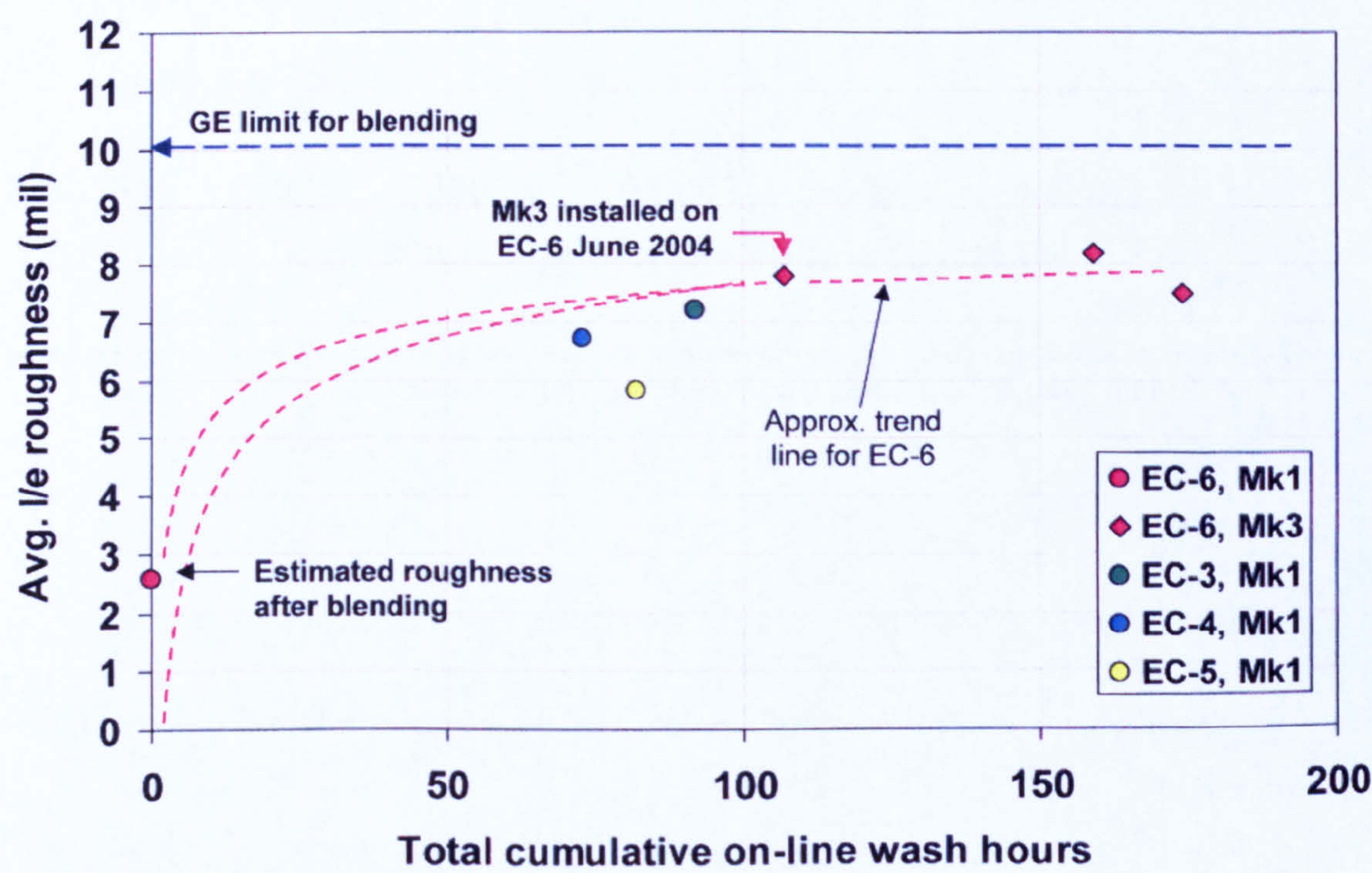


Figure 2-12: Plots for the measured R0 compressor blade roughness on leading edge versus OLWW hours and GT operating hours (Oosting, J. and Stalder, J. P., 2007)

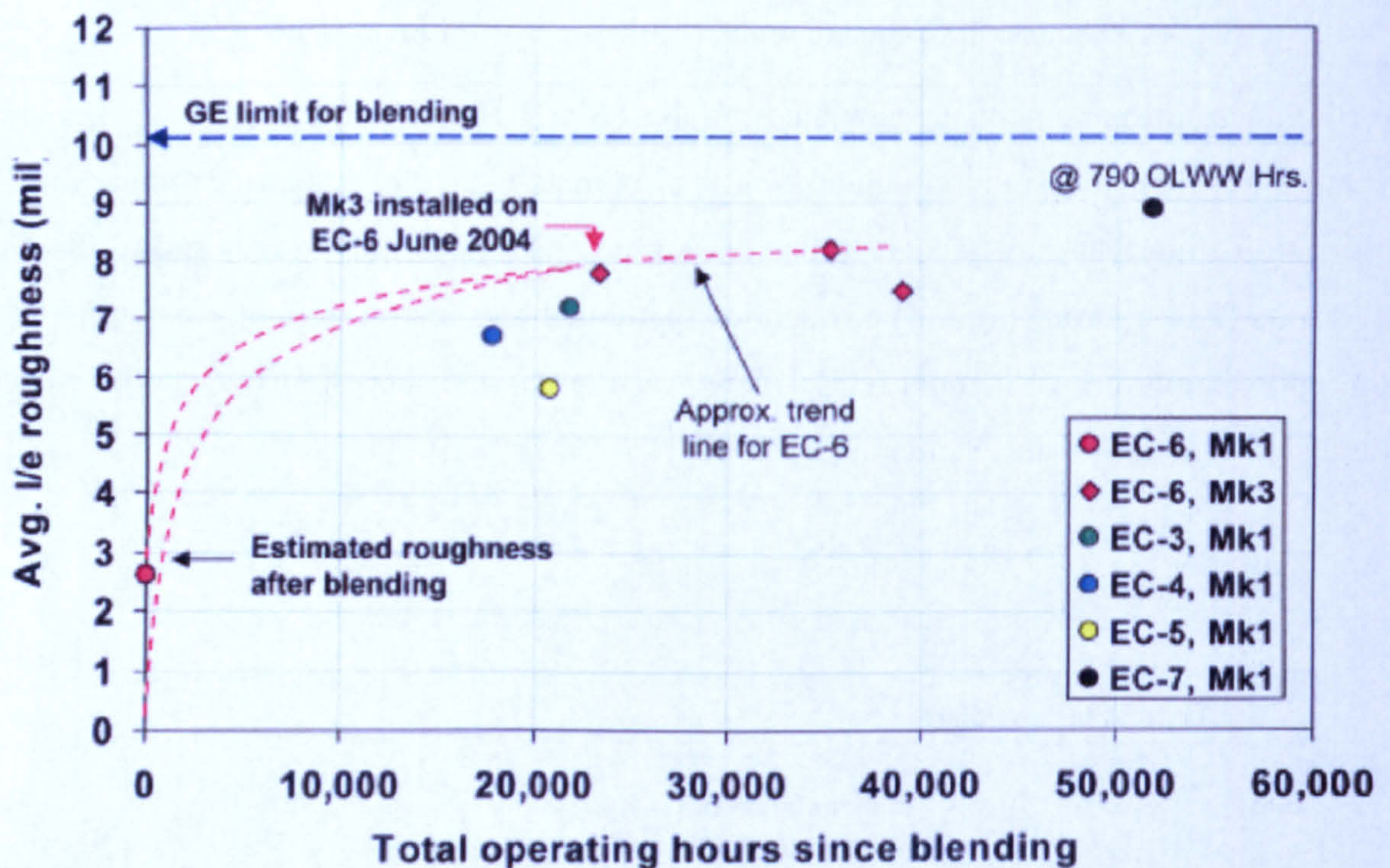


Figure 2-13: Plots for the measured R0 compressor blade roughness on leading edge versus OLWW hours and GT operating hours (Oosting, J. and Stalder, J. P., 2007)

- Off-line compressor washing on unit EC-6, power degradation and recovery

Table 13 below shows the current power degradation and recoveries after off-line compressor washing on Unit EC-6.

Table 13: Power degradation and recovery after off-line compressor washing

Period	Operating hrs.	P[MW] cor. clean	P[MW] cor. fouled	Power degradation
Jun 04-Jun 05	7,902	359 MW	343 MW	16 MW 4.4%
Jun 05-Dec 05	4,083	357 MW	348 MW	9 MW 2.5%
Dec 05-Mar 06	1,922	353 MW	343 MW	10 MW 2.8%

2.10.4 On-line Compressor Washing and Improved Nozzle System Performance

Table 14: OLWW system performance comparison between the Mk1 and the novel Mk3 nozzle systems

Unit No. OLWW System	EC-5 Mk1	EC-6 Mk3	EC-7 Mk1
No. of nozzles	38	30	38
Volume flow	29.3 l/min	17.6 l/min	29.3 l/min
Weekly wash regime	3 x 17 min	3 x 17 min	daily 20 min
Op. hrs. in the period	1,283 hrs.	1,683 hrs.	1,686 hrs.
Power drop in period	14.3 MW	10.9 MW	11 MW
Rate of degradation MW per 1,000 Hrs.	11.1 MW	6.5 MW	6.5 MW

Table 14 is very interesting and three important observations can be made:

1. Comparing units EC-5 and EC-6, both with the same wash regime - the rate of power degradation with the novel Mk3 nozzle system is about 30 to 40% less than with the Mk1 nozzle system.
2. Comparing units EC-6 and EC-7 - the rate of power degradation is virtually the same for both machines, even though EC-7 with the Mk1 nozzle system has been washed more frequently and at a higher mass flow rate.
3. Comparing units EC-5 and EC-7, both with Mk1 nozzles - the rate of power degradation is almost halved when daily on-line washing is performed. This result confirms the comment made by the OEM, that daily water washes are more effective in mitigating performance loss than extended interval washes (Oosting, J. and Stalder, J. P., 2007).

2.11 Historical Perspective of Compressor On-line Washing Developments

This section presents the developments in on-line washing over the last four decades. Because on and off-line washing methods are complimentary the development of the two systems remains closely related. There has been a slow shift from off-line washing to on-line washing beginning in the early 1970's. During spray injection the rotational speed of the GT has risen from crank speed to increasingly higher levels, and the wash has been performed at higher and higher loads (Mund. F. C. and Pilidis, 2006).

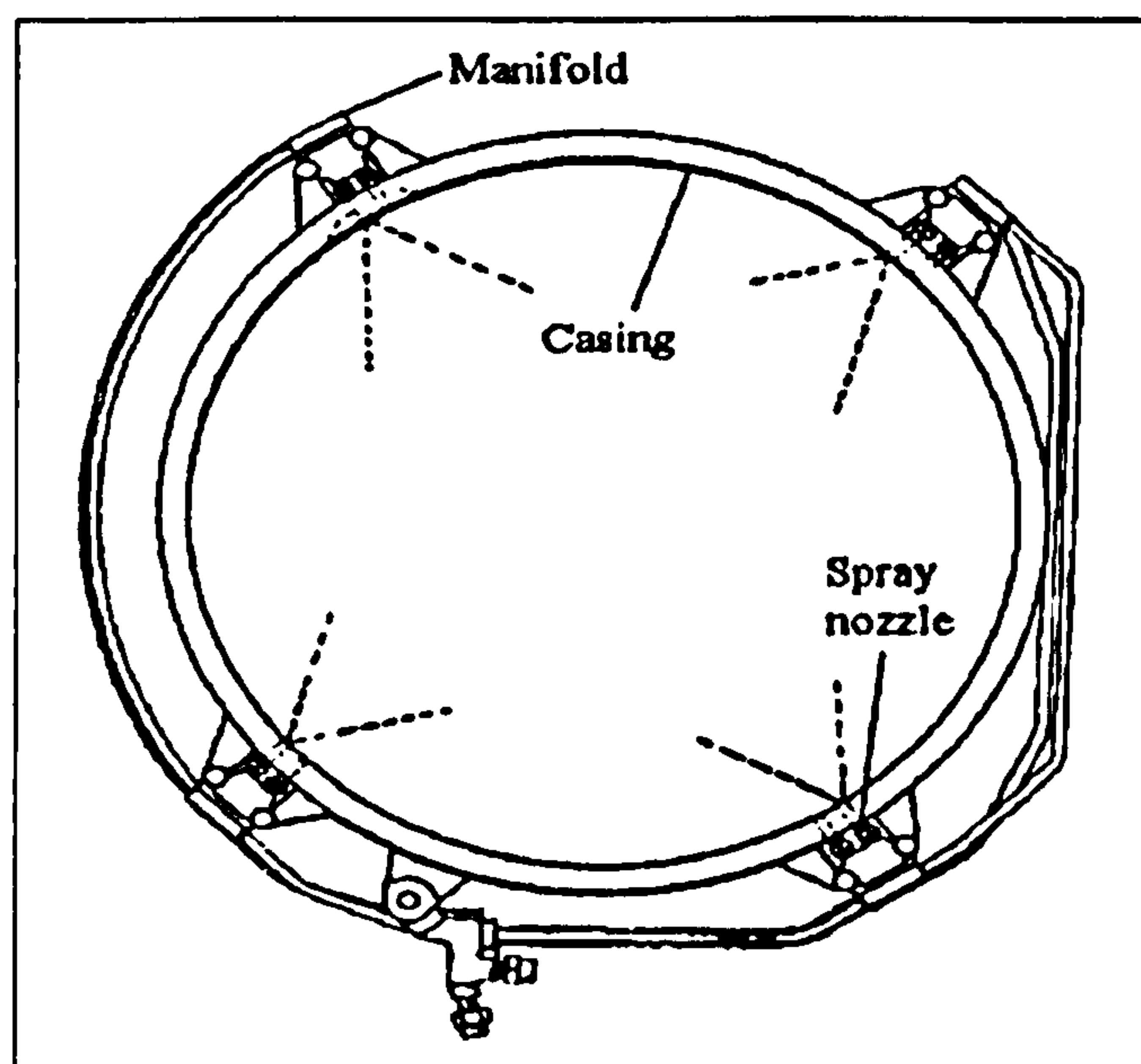


Figure 2-14: Removable spray manifolds (Mund. F. C. and Pilidis, 2006)

In the 1970's the first publications referring to compressor washing were made available. The foremost concern was that the spray systems would alter the airflow pattern and cause loss of compressor due inlet distortion. In 1970 a patent was filed mentioning cleaning by spraying fluid into the intake air stream of a compressor. This used a removable manifold assembly with four nozzles where the placement of the nozzles was specified on order that there would be no interference to the airflow (Figure 2-57). It was deemed that the location of the nozzles was ideal when they did not extend into the airflow

During the 1980 washing methods were established which did not detract from the GT load profile. The emphasis was on the properties of the injected sprays: chemical properties of the washing agents were optimised and attempts were made to describe effective spray patterns qualitatively. In 1986 McDermott patented a method and equipment for cleaning GT engines, describing the full or near full load operating conditions (McDermott P. , 1991). He claimed that test results were taken when the engine was running at full speed and the compressor at a maximum speed. In 1989 the phrase “on-line washing”, was attributed by (Thames et al., 1989) to not interfere with continuous GT operation. It was also stated that on-line washing relied on keeping the engine clean as opposed to making good once it had deteriorated. The economic value of extended time intervals between off-line washes, via on-line washing was acknowledged. It was generally observed that spray characteristics were important for achieving a good clean and to avoid component damage, but spray patterns were only described qualitatively.

A first hint toward blade stresses and subsequent failure due to the impingement of large droplets from bulky wash sprays was found in a patented off-line washing apparatus in 1980 (Hornak, 1980). Mc-Demott (McDermott P. , 1991) claimed for his proposed washing system that the formation of spray sheets that were centrifuged toward the casing of the compressor was avoided by fluid injection across the air stream as opposed to parallel injection, as illustrated in Figure 2-58. The spray manifold was suggested to be opposite the compressor on the bellmouth side. (Becker, B. and Bohn, D., 1984) reported that droplets must be ejected over the entire blading cross section to avoid blade erosion.

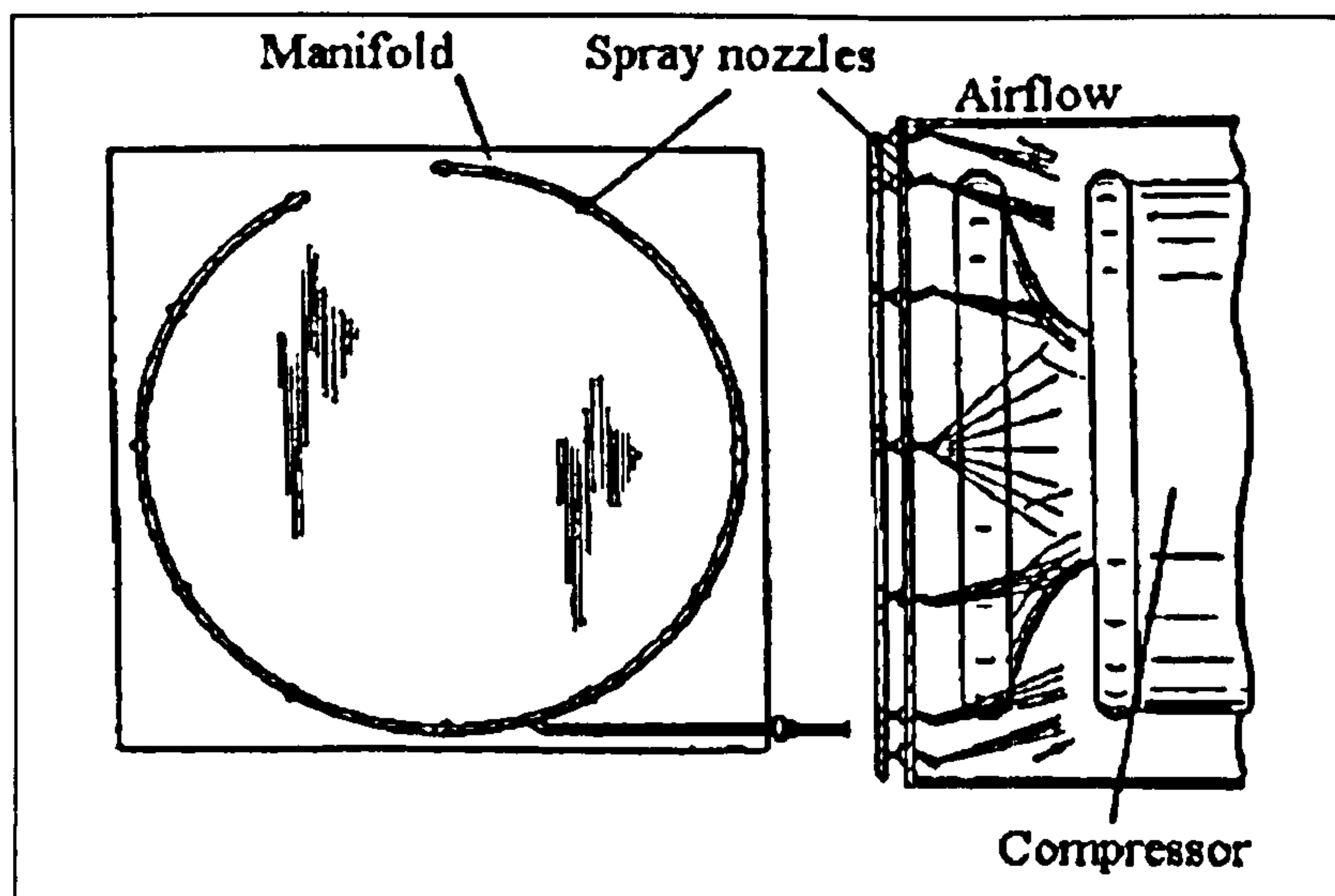


**Figure 2-15: Spray nozzle manifold (left) and installation opposite the compressor (right)
(McDermott P. , 1991)**

In the 1990's there was a rising concern about the loosening of spray nozzles because of vibration and corrosion. Changes of injector locations and safer nozzles fittings were used to ensure parts did not enter the compressor during the operation. In general it was understood that there was less likelihood of a loose nozzle invading the compressor if the spray nozzles were placed further away from the compressor i.e. installation opposite the compressor were favoured. A nozzle using a ball joint was patented (Kolev, S. and Robben, R., 1993) and it is shown in Figure 2-59 (right), they claimed it to be more secure. And in Figure 2-59 (left), the spray nozzles were set at a significant distance from the compressor inlet.

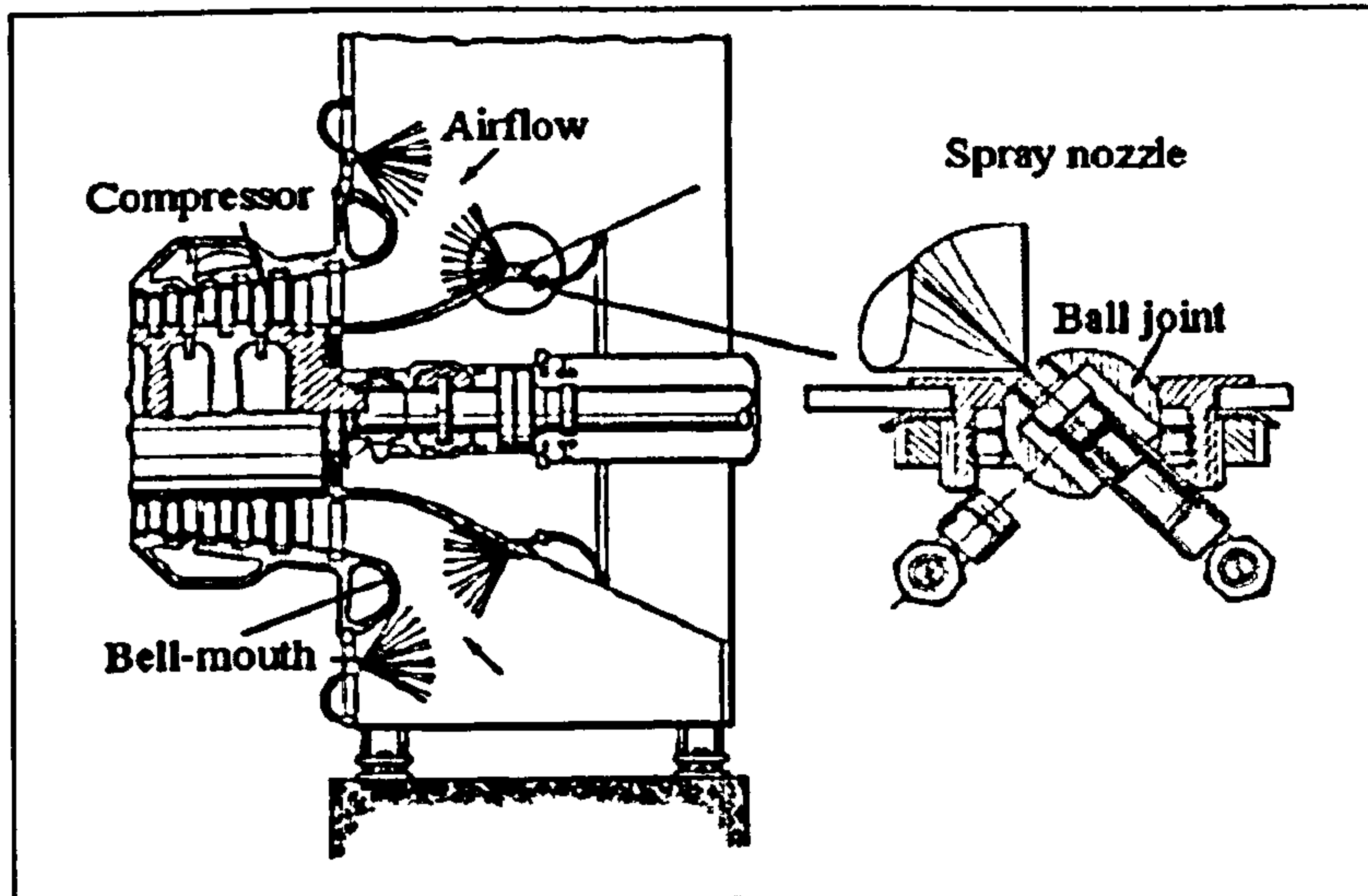


Figure 2-16: Washing device and nozzle (Mund. F. C. and Pilidis, 2006), side view

In the early 1990's attention was directed to qualitatively describing droplet size to ensure the droplets followed the airflow to avoid damage to the blades. This initiative was led by the military research programme. Although a general numerical description of the droplet size distributions could be given by certain size parameters such as the Sauter mean diameter, the information available fell short of a numerical definition of the size distribution referred to diameter ranges. Droplet sizes in the range of 125-200 μm and 80-200 μm were logged by Patterson and Spring (Patterson, J. S. and Spring, S. K., 1992) but these were dependant on the GT and also the washing system. A marked higher mean droplet diameter of 800 μm was recorded and a washing method was patented for aero engines two years later which claimed to produce droplet sizes that followed the airflow and, likewise, the path of the contaminants (see Figure 2-60 left). To achieve droplet size of between 120 and 250 μm with a high injection velocity of 100-126m/s to combat the centrifugal effect of the rotating blades it was necessary to operate at a pressure between 5MPa and 8MPa. Alternative pressure levels, flow rates, and droplet sizes were recommended depending on the existence of inlet guide vanes. An adaption of the system for industrial engines is shown in Figure 2-60 (right), this looked at the common washing practice for aviation and marine applications.

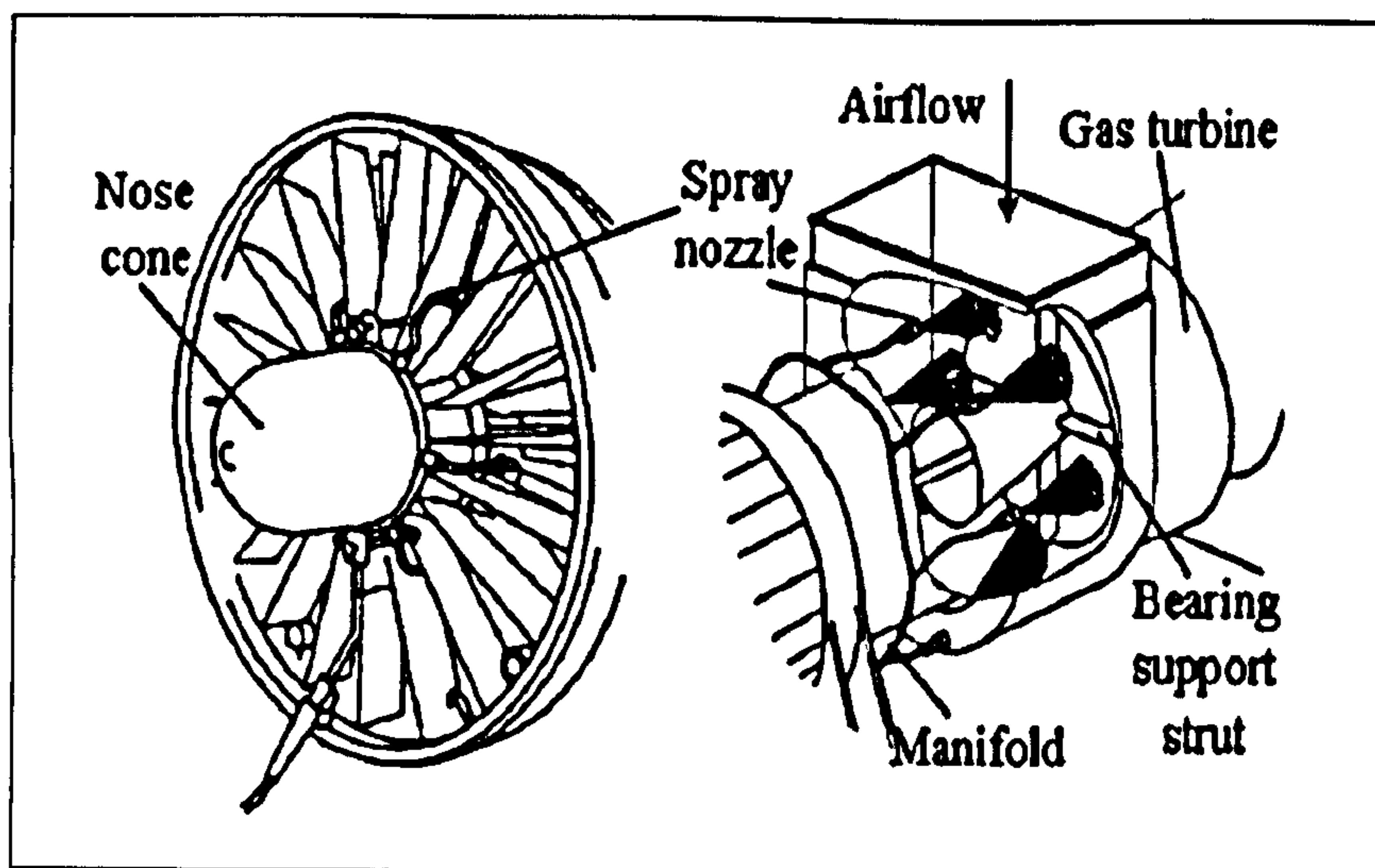


Figure 2-17: Washing device for aero-engines and adaptation for a stationary application (Asplund, 1997)

Chapter 3 – GT9 Performance Simulation

3. Field date studying of GT9 (Sulzer Type 7)

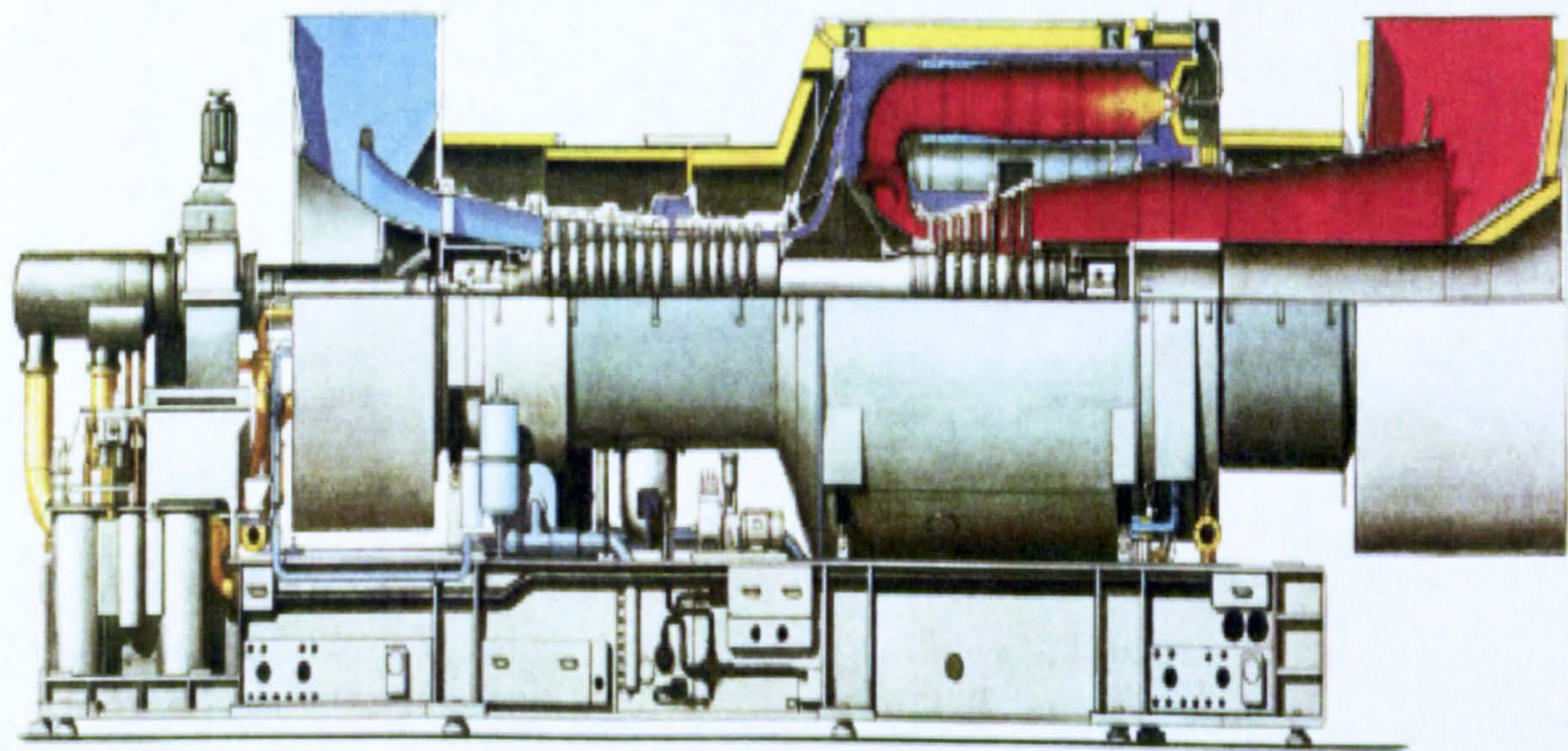


Figure 3-1: Single Shaft Gas Turbine operating in oil Field (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

3.1 Design Data of GT Sulzer Type 7

The following table contains the main data for the Type 7 GT based on the ISO standard reference conditions. The data are only valid for natural gas firing and $\text{CH}_4 > 85\% \text{ Vol.}$

ISO Conditions:

Ambient temperature	15°C
Barometric pressure	1013 mbar
Intake system pressure losses	0 mmWG
Exhaust system pressure losses	0 mmWG

Main data		Unit
Nominal turbine speed	6400	1/min
Power at gas turbine coupling	11000	kW
Efficiency at gas turbine coupling	24.8	%
Compressor pressure ratio	7.55	-
Air intake volume	51.9	m ³ /s
Turbine inlet temperature (hot gas)	approx. 925	°C
Exhaust gas temperature	493	°C
Exhaust gas massflow	64.3	kg/s

3.2 Cycle Description

A 13 stage axial compressor delivers air through a radial diffuser into the 9 combustion chambers located equally around the periphery of the turbine. The fuel injected into the centre of each combustion chamber mixes with the compressed air and is ignited by an electrical discharge flame torch.

The combustion gases flow through the 6 stage turbine expanding the energy and therefore driving the rotor and generator. The expanded gases are finally exhausted through the casing.

- **Design Details**

3.2.1 Base Frame

The gas turbine is supported on a welded steel base frame. A combination of three rigid and three heavy duty spring loaded dampers support the turbine on this structure. A fixed point under the conical air inlet casing, and a keyway under the main housing of the combustion chamber maintain the longitudinal position of the machine relative to the base frame. The ten feet of the base frame stand

on a concrete foundation slab, the base being held in the axial direction by a fixed point at the distributing gear end of the unit.

3.2.2 Compressor and Turbine

A row of guide vanes at the compressor inlet imparts a twist to the air flow, lowering the Mach number at the blade tip of the first moving row. The blade profiles and pitch are selected so as to avoid resonance over the whole operating speed range - due allowance having also been made for the possibility of blade excitation caused by row interference of the flow pattern - again over the whole operating range of the machine.

The compressor rotor blades are fixed by a system which requires no specially designed cut-outs in the trapezium shaped rotor slots in order to insert the blades. The results in significantly lower operational stresses in the components see (Figure 3-2).

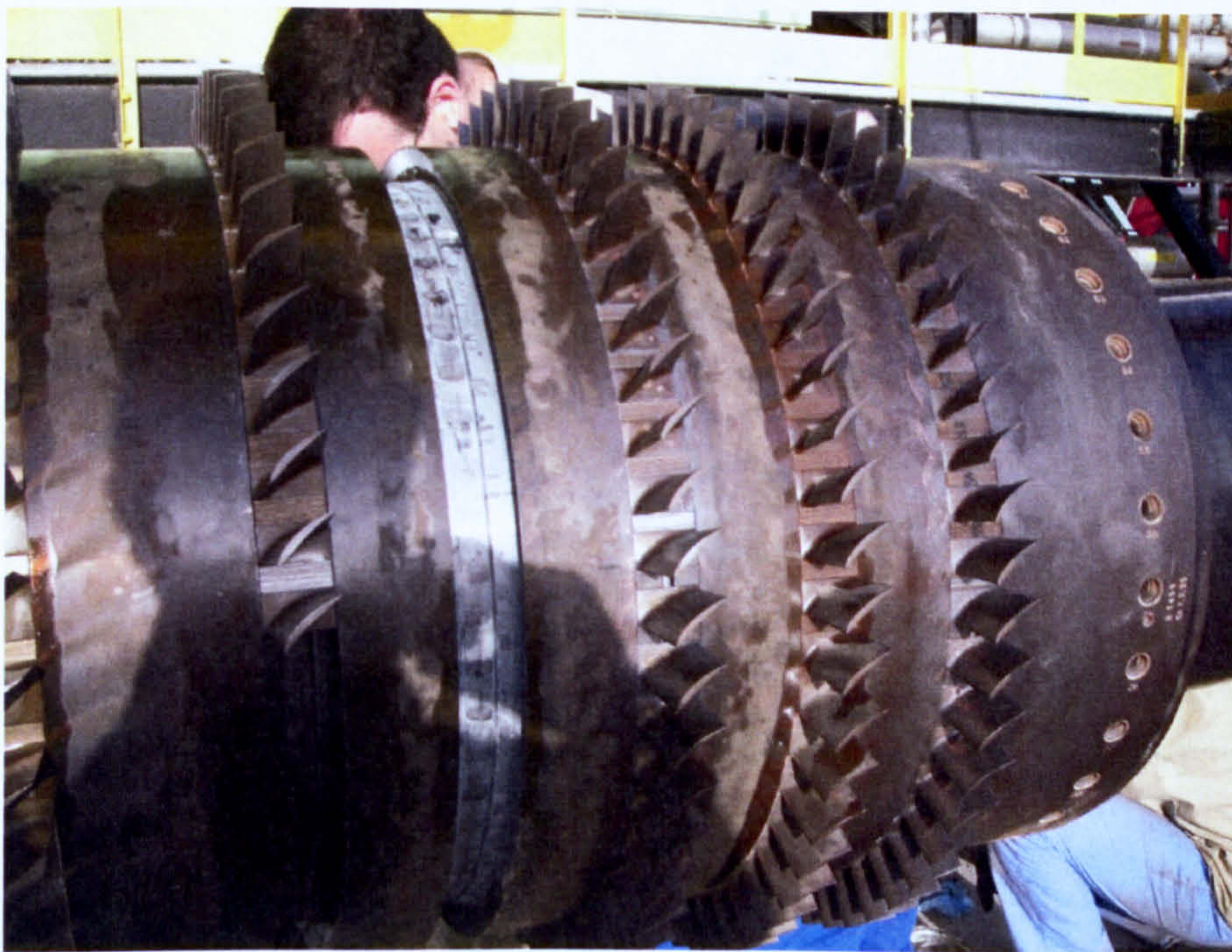


Figure 3-2: Compressor rotor blades installation (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

Prior to final assembly, the stator blades of the compressor are fitted into special mountings according to the required blade angle. These mountings are then fixed in the grooves of the horizontally split cast iron blade support. Owing to its relatively small dimensions, the compressor rotor is manufactured from the solid. The ferritic discs of the six-stage turbine are all identical and are held in alignment by Gleason-Curvic

toothed couplings. They are bolted to the compressor by a central tie rod which is tensioned hydraulically during assembly.

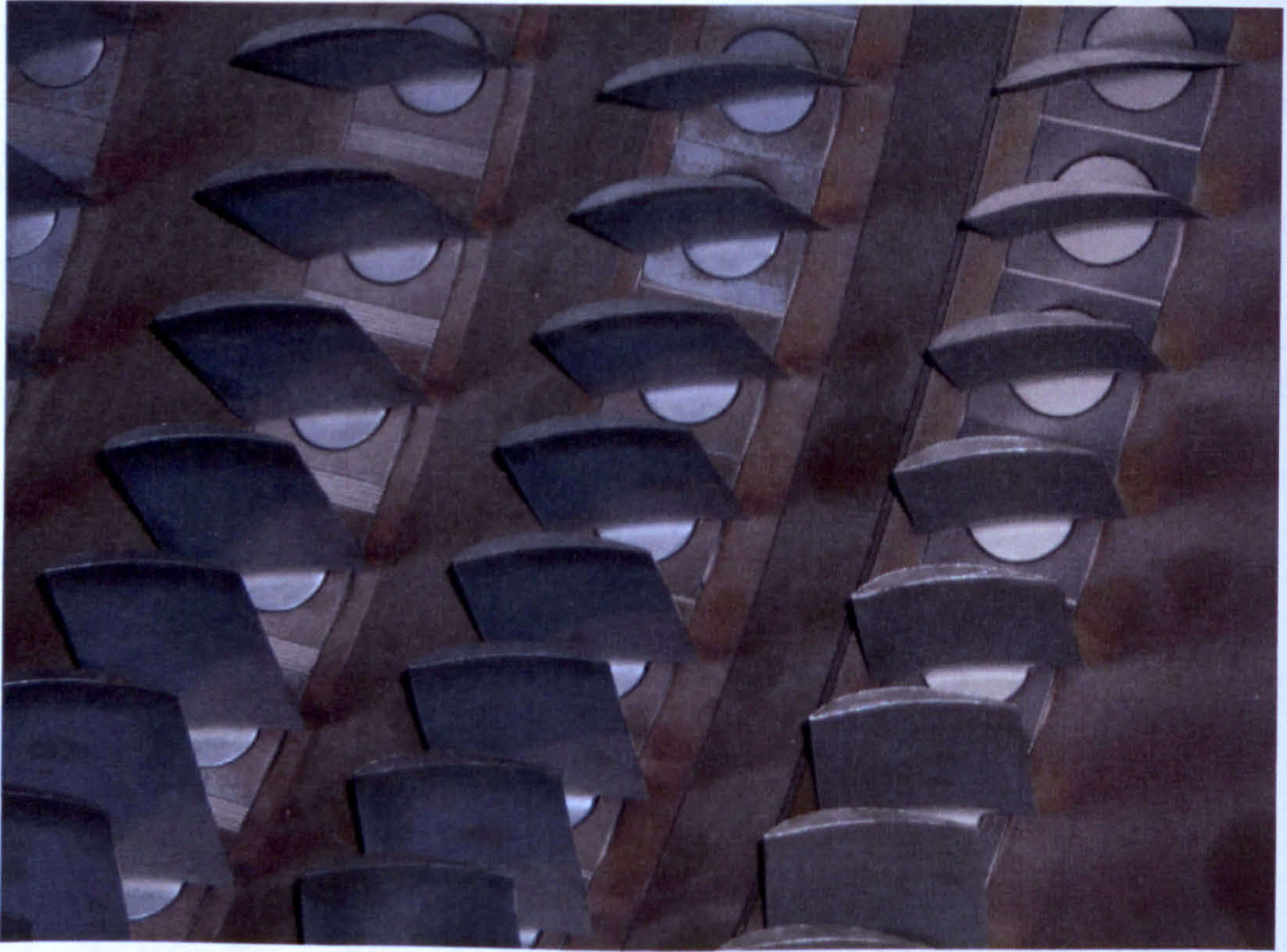


Figure 3-3: Fixed stator blades of the compressor (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The turbine rotor blades are held in fir tree shaped fixings. The design also includes a certain radial distance between the root of the blade and the beginning of the aerodynamic profiled section over which the combustion gases flow. This radial distance is not in the hot combustion gas flow region and is cooled by the air bleeding out of the stationary stator blades immediately upstream and downstream of the rotor blade. The heat conduction from the rotor blades to the main body of the rotor is thus kept to a minimum.

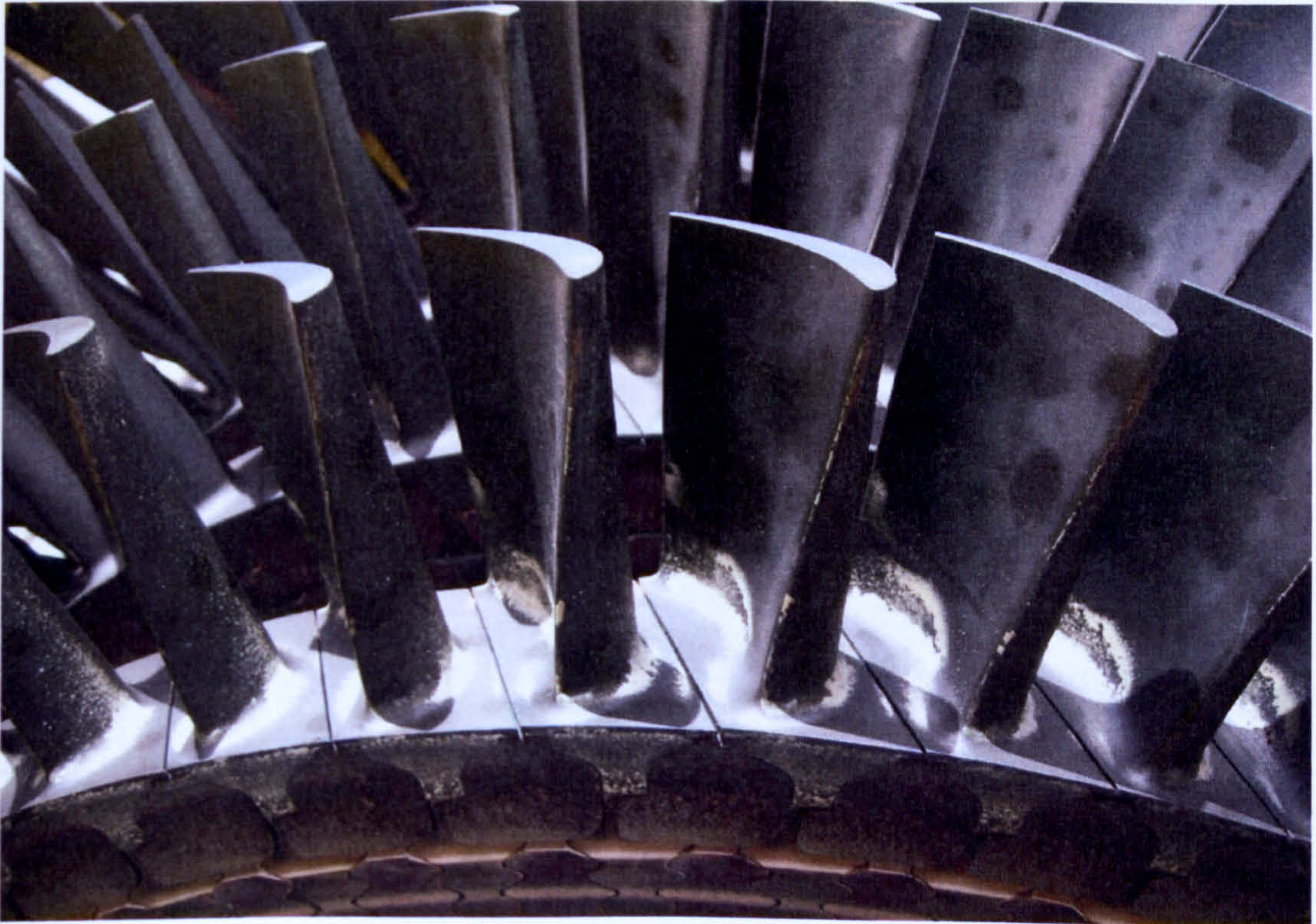


Figure 3-4: Turbine rotor blade (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The turbine stator blades are grouped in several segments. The first stage segments comprise three blades in each batch and are of precision cast design. The second to sixth stage segments, however, comprise four blades in each batch and are of welded design. The blades of the first and second stages are air-cooled *by* a through flow system. This is assisted by having additional film cooling holes on the blade inside trailing edge and also on the same edge extremity. The blades of the third to fifth stages are cooled *by* through flow only.

The turbine stator blade support is made of alloyed cast steel and, in order to assure material stability and good dimensional concentricity under working conditions, it is machined to a constant relatively small thickness. The stator blade support is fixed to the hot gas casing and into the intermediate casing. It is free to expand radially and its concentricity with respect to the rotor can be adjusted during erection by means of eccentric discs.

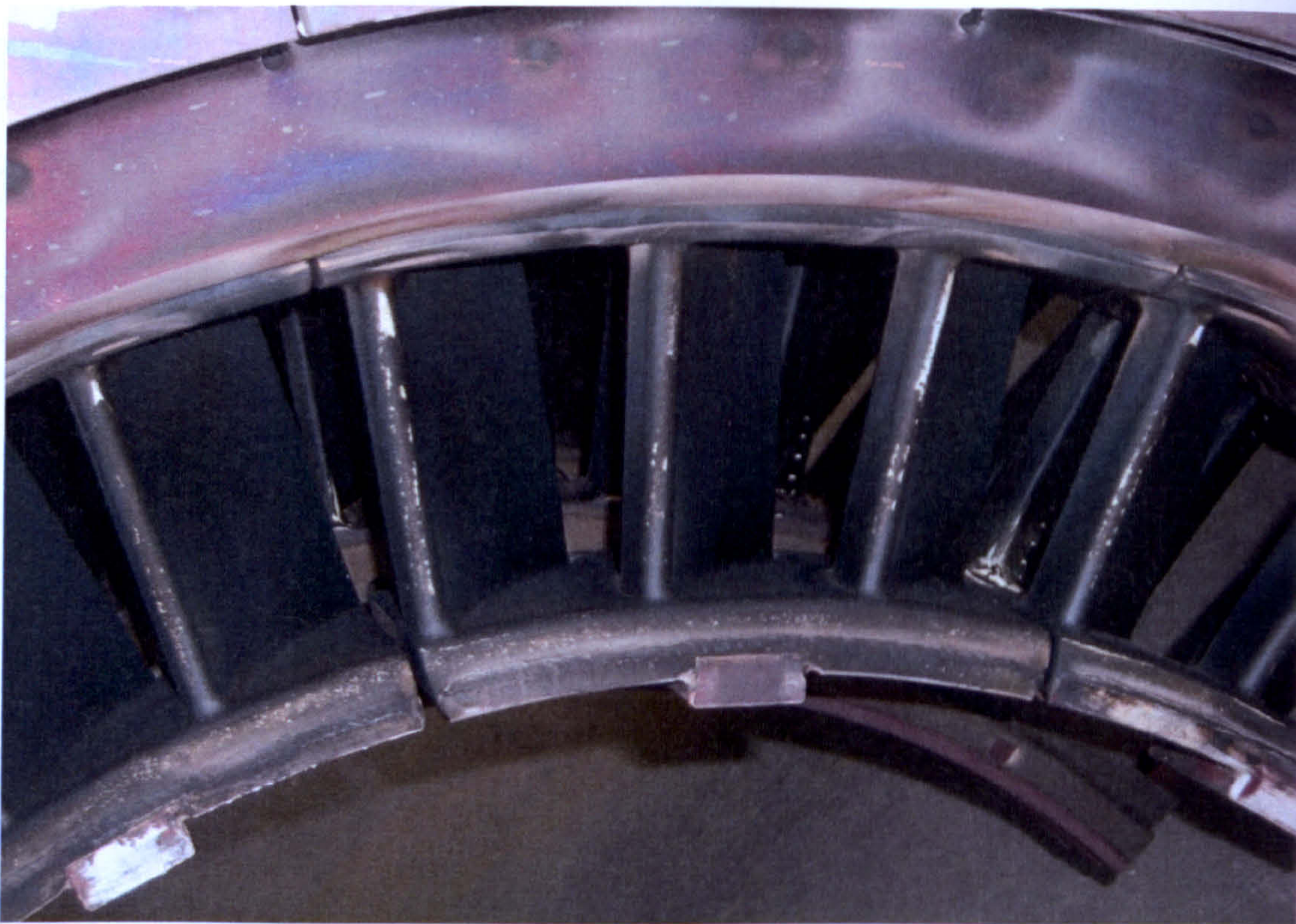


Figure 3-5: Turbine stator blades (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

Concentricity of the compressor casing with reference to the rotor can be achieved by the relative movement of two oblique intermediate rings. These are located between the flange on the pressure side of the compressor and the diffuser - thus modifying the casing alignment if necessary.

3.2.3 Combustion Chambers

Nine combustion chambers are located around the gas generator turbine stator. The air is delivered via the diffuser into the horizontally split combustion chamber housing, thus resulting in a good distribution of air to each combustor (see Figure 3-6). The internal flame tubes are surrounded by a concentric liner thus cutting down the radiant heat transmission and bringing about an effective and uniform cooling characteristic. The same principle is applied to the entire hot gas casing, including the stator blade support features.



Figure 3-6: Combustion Chambers (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The combustion chamber heads are provided with fuel gas nozzles (burners for liquid fuels or dual-fuel firing are also available). Ignition is effected by means of two high energy spark ignitors mounted in diametrically opposed combustion chambers. The fuel gas/air mixture in the other chambers is ignited through spontaneous cross-firing. Fire-eye detection systems are used to supervise ignition start-up and continuous running conditions.

3.2.4 Bearings

The rotor is supported at each end by white metal radial bearings, the axial thrust bearing being located adjacent to the radial bearing at the compressor and of the unit. The rotor is connected through a toothed coupling to the distributing gearbox, which contains all the mechanically driven auxiliary systems.

The rotor is connected through a toothed coupling to the distributing gearbox, which contains all the mechanically driven auxiliary systems.

3.2.5 Auxiliary Systems

- **Lube oil system**

Brief description

The oil system of the gas turbine generator package includes the lube oil system, the control oil system and the jacking oil system. Each system serves a distinct and separate function, the systems are fundamentally interrelated in as much they are all feed from the lube oil tank. The piping system is partly integrated in the gas turbine base frame; this frame also contains the lube oil tank. The distinct functions of the systems are automatically controlled and supervised by the GT- control system.

The lube oil system, in addition to its main function of providing lubricating oil for the GT- bearings, the accessory drive gear, the reduction gear, the couplings and the generator bearings, it also supplies oil to the control oil- and the jacking oil system.

The control oil system supplies servo oil to operate the starting clutch, the turbo-coupling filling valve and the blow-off valve. The jacking oil system supplies high pressure oil to the GT and Generator bearings in order to reduce the initial starting torque, when the shaft train begins to rotate.

The various systems are briefly described below:

A) Integrated into the base frame are the following:

- Oil reservoir with a capacity of approximately 9000 l which corresponds to a settling time of about 5 minutes. (Oil specification see: Lube Oil System).
- Electric heating system with a rating of 36 kW.
- Auxiliary lubricating oil pump driven by an a.c. motor, double filter, pressure control valves, temperature control bypass valve.
- High pressure (jacking) lube oil pump for lifting the rotor to reduce the breakaway torque (driven by an a.c. motor).

- Emergency lube oil pump driven by a d.c. motor. This pump is only started when there is no a.c. supply available during the cooling period after a shut down.
- Auxiliary control oil pump with a.c. motor and pressure reducing valve.
- Oil vapour extraction system, comprising an ejector (for normal operation) and a fan with an a.c. motor drive for the cooling period, both facilities being served by a common oil separator.
- Three terminal boxes, one for the power cables to the auxiliaries, one for the control system cables, and one for the oil temperature measurements on the bearings, combustion air, and combustion gases.

B) The supervisory equipment and indicating instruments are either mounted on the base frame locally or on the instrument rack.

C) The distributing gearbox is connected to the compressor end of the shaft via a gear-type coupling. The box contains all the necessary facilities for coupling and driving as indicated below:

- Starting motor (possible methods of starting include electric motor, hydraulic motor, or expansion turbine (gas or steam)).
- Main control oil pump
- Power transfer shaft to the main reduction gearbox
- Main lube oil pump
- Barring motor for slow turning of the rotor during the cooling period

Reduction Gear

The main reduction gearbox is designed to handle all the power capable of being produced by the turbine when connected to the driven machine.

Couplings

The two gear-type couplings, one between the rotor and distributing gearbox and one between the distributing gearbox and reduction gearbox are both suitable for the continuous transmission of the maximum power capable of being developed by the machine.

Starting System

There are several possible methods of starting the turbine set, i.e. by electric motor, hydraulic expander using oil from a central station pump, expansion turbine (gas or steam), diesel set. The approximate maximum starting load is 400 kW.

Air Inlet System

The suction air intake ducting is shop assembled and contains an inertia-type filter, with an electric motor driven dust extractor fan – for particularly dusty atmospheric condition where compressor performance has to be safe-guarded and cleaning kept to minimum, moving screen filters can be installed.

Exhaust System

The exhaust gas system consists of a stack, transition piece, and supporting structural steelwork. The expansion joint between the exhaust casing and the stack allows for any thermal expansion place.



Figure 3-7: GT9 Power Plant (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

3.3 Preventive Maintenance of GT9

3.3.1 Actual Data

The performance analysis, with reference to fouling, efficiency and power output, is based on the actual log sheet and the operating data readings. If it becomes suspicious that the gas turbine suffers from any loss of power output or efficiency, which cannot be reduced by dry-compressor cleaning, a performance check is indicated. At least once per month a complete set of readings is recommended.

- **Exhaust Gas Temperature**

A continuously logged exhaust temperature distribution is an essential way to detect and observe developing problems, cohering with the combustion system (e.g. fuel injectors, combustion chambers, etc.).

The change of the exhaust gas temperature (EGT) distribution over a certain period of time is more significant than the deviation of the measured EG temperatures compared to the average EGT. The manufacturer recommends to print out the actual reading data log sheet

once a week, to have a complete recording of the EGT distribution. The control system performs an automatic calculation of the average temperature and supervision of the temperature deviation, and protects the unit from damage caused by abnormal deviation.

- **Mechanical Conditions**

The mechanical condition of the package, specially the bearing vibrations, are part of the periodical recordings, based on the actual log sheet. The recording of the mechanical condition is recommended to be done once a week.

- **Daily Reading**

The daily log sheet is a summary of the most important gas turbine data over a time period of 24 hours. Every four hours a new data set is added and saved on the Logsheet archive. By using the daily log sheet you get a quick overview of the condition and operating status of the gas turbine in a very short time. The log sheet allows you to see the history of the machine over months and years.

- **Data Reading at shutdown**

This log sheet is intended to ease troubleshooting. It contains the same values as the actual log sheet. The data however will be stored as soon as the shutdown occurs. This gives you the possibility to see all actual values at that certain moment of the shutdown. These data are stored until a new shutdown occurs.

- **Control Parameters**

It is the purpose of the logged control parameters, to watch the trouble free performance of the analogue/digital functions. The readings listed on the actual log sheet and the PC 670 - log sheets allow the following important checks:

- **GT speed measurement**
- **Fuel control valve position**
- **Calculated hot gas temperature K4**

The readings and corresponding function checks are recommended to be done once per month.

Calculation formula for speed value TP1:

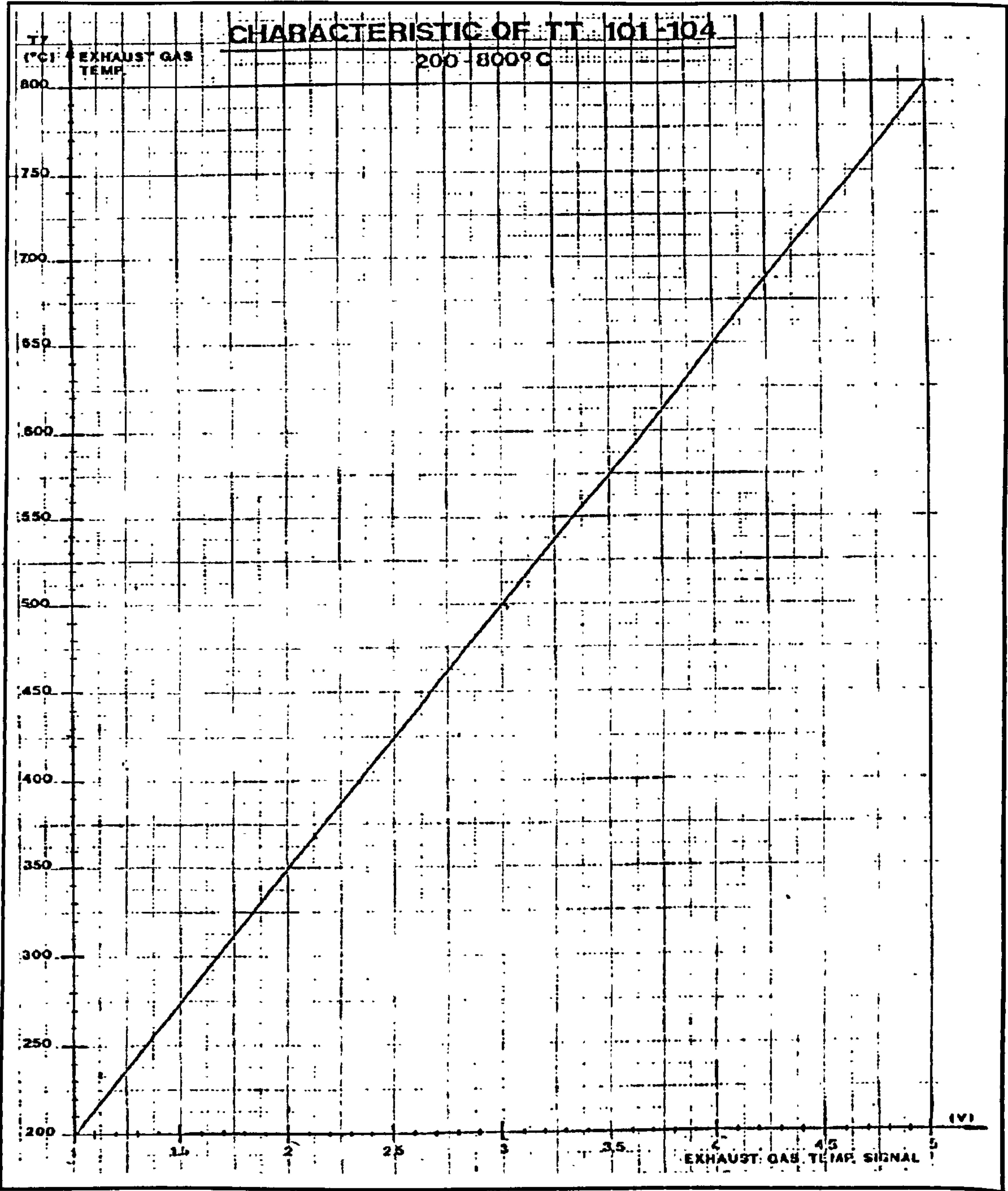
$$\text{GT Speed [rpm]} = 2000[\text{rpm/V}] \times (\text{TP1 [V]} - 1)$$

Where TP1 is a temperature transducer for measuring T7 and can be read it from the diagram 1 in the next page.

Calculation formula for K4 (TET) value TP2:

$$\text{K4[}^{\circ}\text{C]} = \text{Exhaust gas temperature T7 [}^{\circ}\text{C]} + K \times \text{Combustion chamber pressure [bara]},$$

Where $K = 20,0 \text{ }^{\circ}\text{C / bara}$



Daigram 3-1: Exhaust Gas Temperature Measurement (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

3.4 Preventive routine maintenance

3.4.1 Compressor dry cleaning

The power output of a gas turbine plant installation can be considerably reduced by dirty compressor blades. The degree of blade fouling is primarily dependent on the site location (i.e. a dirty atmosphere) and the amount of maintenance that is given to the inlet air filtering plant.

The indication that a compressor needs cleaning can be summarized as follow:

- *Increase of K4 (TET), which is the inlet gas temperature to the first row of blades at the gas generating turbine.*
- *Increase of T7, which is the average exhaust gas temperature.*
- *Drop in combustion chamber absolute pressure.*

All the above items imply a gradual fall-off of overall plant thermal efficiency and hence in order to accurately predict when a compressor needs cleaning, it's necessary to keep a long time record of all the above readings and trends.

A visual check of the condition of the first compressor stator blades can be made by removing the manhole cover on the air intake casing when the machine is shut down and stands still. A hand held battery powered inspection torch is normally sufficient for this purpose.

If the compressor blades is found to be in a clean condition (operational experience is required here to judge the degree of fouling), and the machine still exhibits one or all of the above mentioned symptoms, then the source of the power loss lies somewhere else in the overall system. In this case assistance should be requested from MAN Turbo.

3.4.2 Procedure for dry cleaning of compressor

Any changes in the load that occur during the cleaning process should be taken up by the remaining machines or the grid automatically (except in the condition when there is an emergency shut down or when the change in the load is more than the available spare capacity left on the remaining machines).

- Execute the compressor cleaning only when the turbine is operating on part load, 60% of full load or lower if possible.
- Run the machine under steady conditions for at least 15 minutes to stabilize all the readings and then take a set of "Before Cleaning" data as given example in the following table:

	Before cleaning	After 1 st injection	After 2 nd injection	After 3 rd injection
GGT speed rpm				
C.C pressure bara				
Gas inlet temperature %				

Before the cleaning process is started, both lube oil filters should be cleaned. After each injection check the differential pressure across the filter and observe the lube oil pressure. Change and clean the used filter immediately to avoid any problems, which may be caused to low lube oil pressure.

The dry cleaning is done to deliver the blades from the normal operation deposition. Some rice particles get with the airflow to the oil system. That is the reason while the filters should be cleaned.

3.4.3 Cleaning process

1. Insert around 1 kg of ordinary clean rice granules into the rice box as shown (Figure 3-8 and Figure 3-9).
2. Open the valve to inject the rice into the compressor and close the valve again, after all rice is injected.
3. Print out the second set of readings (1st. after injection).
4. Repeat this process until no significant betterment is evidently recognized with respect to the previous corresponding readings.
5. Plot the relation between the above mentioned parameters and the number of injections.

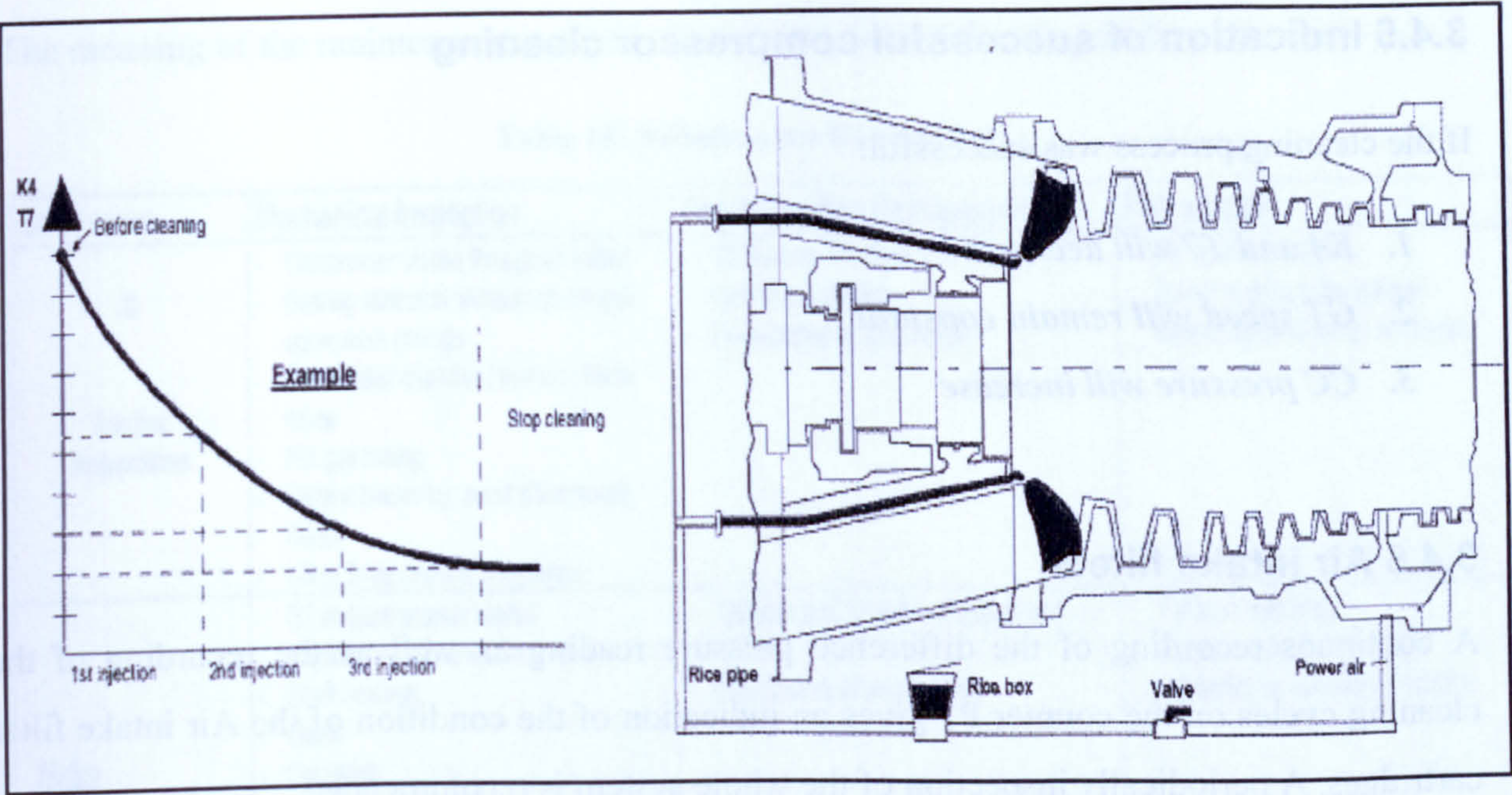


Figure 3-8: Dry compressor cleaning system

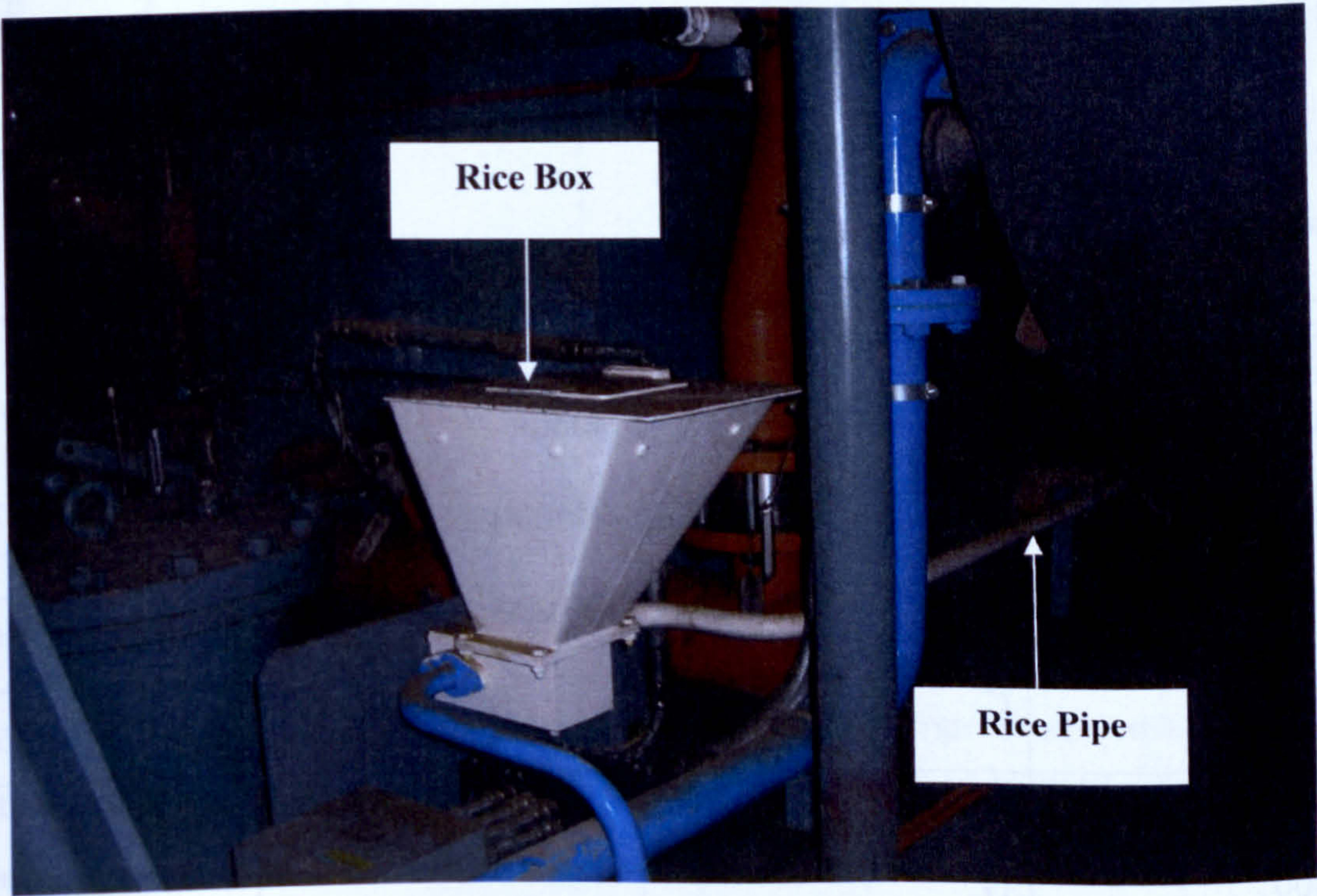


Figure 3-9: Installation of dry cleaning system in front of sulzer type 7 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

3.4.5 Indication of successful compressor cleaning

If the cleaning process was successful:

- 1. *K4 and T7 will decrease*
- 2. *GT speed will remain constant*
- 3. *CC pressure will increase*

3.4.6 Air intake filters

A continuos recording of the difference pressure reading as well as the recording of the cleaning cycles on the counter P1 gives an indication of the condition of the Air intake filter cartridges. A periodically inspection of the whole system is recommended.

3.5 Heavy Maintenance

3.5.1 Standard overhaul interval

Diesel oil and Dual fuel units

Accumulated Operating Hours	Maintenance Code	Accumulated Operating Hours	Maintenance Code
8'000	B1	88'000	B
18'000	B	100'000	D
28'000	C		
40'000	B		
50'000	D		
64'000	B		
75'000	C		

B1 = First Inspection B = Inspection C = Minor overhaul D = Major overhaul
E = General overhaul

The meaning of the maintenance codes is described on the following table.

Table 15: Maintenance Description

Code name	Mechanical inspection	Control system / instrumentation	Replacement
<div>B</div> <div>Routine Inspections</div>	<ul style="list-style-type: none">Compressor blades through air intakeBearing attainable without removing of upper main casingsCombustion chamber / burners / flame tubesHot gas casingTurbine blades by use of a boroscopeGearsFilters to be cleaned or replaced	<ul style="list-style-type: none">Verification / check of control and supervision systemRecalibration of instruments	<ul style="list-style-type: none">For liquid fuel firing: Burner nozzles to be replaced depending on condition of nozzles
<div>C</div> <div>Major Inspections</div>	<ul style="list-style-type: none">GT and compressor bladesRepackaging of compressor rotorShaft bearingsGearsCouplingsAlignmentAuxiliariesRebalancing of rotor if necessary	<ul style="list-style-type: none">Verification of control and supervision systemRecalibration of instruments	<ul style="list-style-type: none">For liquid fuel firing: Burner nozzles to be replaced depending on condition of nozzles
<div>D</div> <div>Overhauls</div>	<ul style="list-style-type: none">According to "C" - InspectionNon destructive test of compressor blading	<ul style="list-style-type: none">According to "C" - Inspection	<ul style="list-style-type: none">GT rotor disks respective rotorsGT rotor bladesGT stator and stator bladesFlame tubesGas burner front partBurner nozzlesHot gas casingRoller bearings of all full-time operated auxiliaries etc.

3.5.2 Turbine main-components life time

Replacement of hot parts

The following operation hours are only a rough clue. Thus the operating hours depend on the operation conditions and therefore on the maintenance of the gas turbine and its auxiliary systems. To control the normal running conditions you need the complete long time readings of all mentioned systems.

To reach the given duration of life the gas turbine needs regular service and maintenance of all systems. Thus make sure that this work will always be done carefully by entitled personnel only!

every 50'000 hours: - 1st turbine rotor row

every 30'000 hours: - flame tubes (gas firing)

3.5.3 Gas Turbine Operating Instructions

Process: Overview turbine

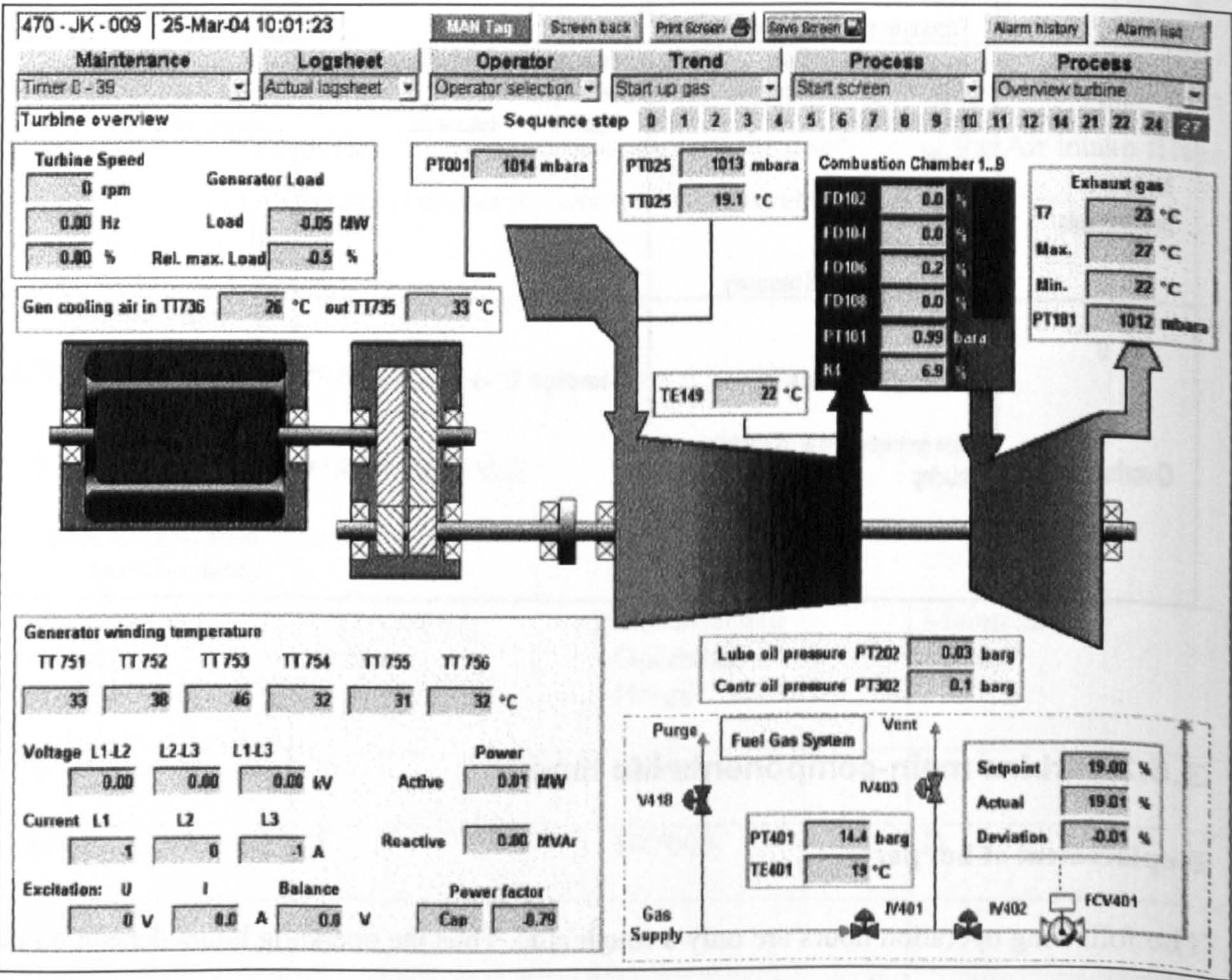


Figure 3-10: Process overview of GT9 (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

This page gives you an overview from the most important values from the turbine, such as speed, K4 value, exhaust gas temperature T7, load, etc. The indications in the combustion chambers change their colour when the flame eyes detect a flame. For the fuel control valve (FCV) the set point, actual value and the deviation are indicated. Some general values concerning the generator are placed on this screen as well.

3.5.4 Process air system and ventilation of GT9

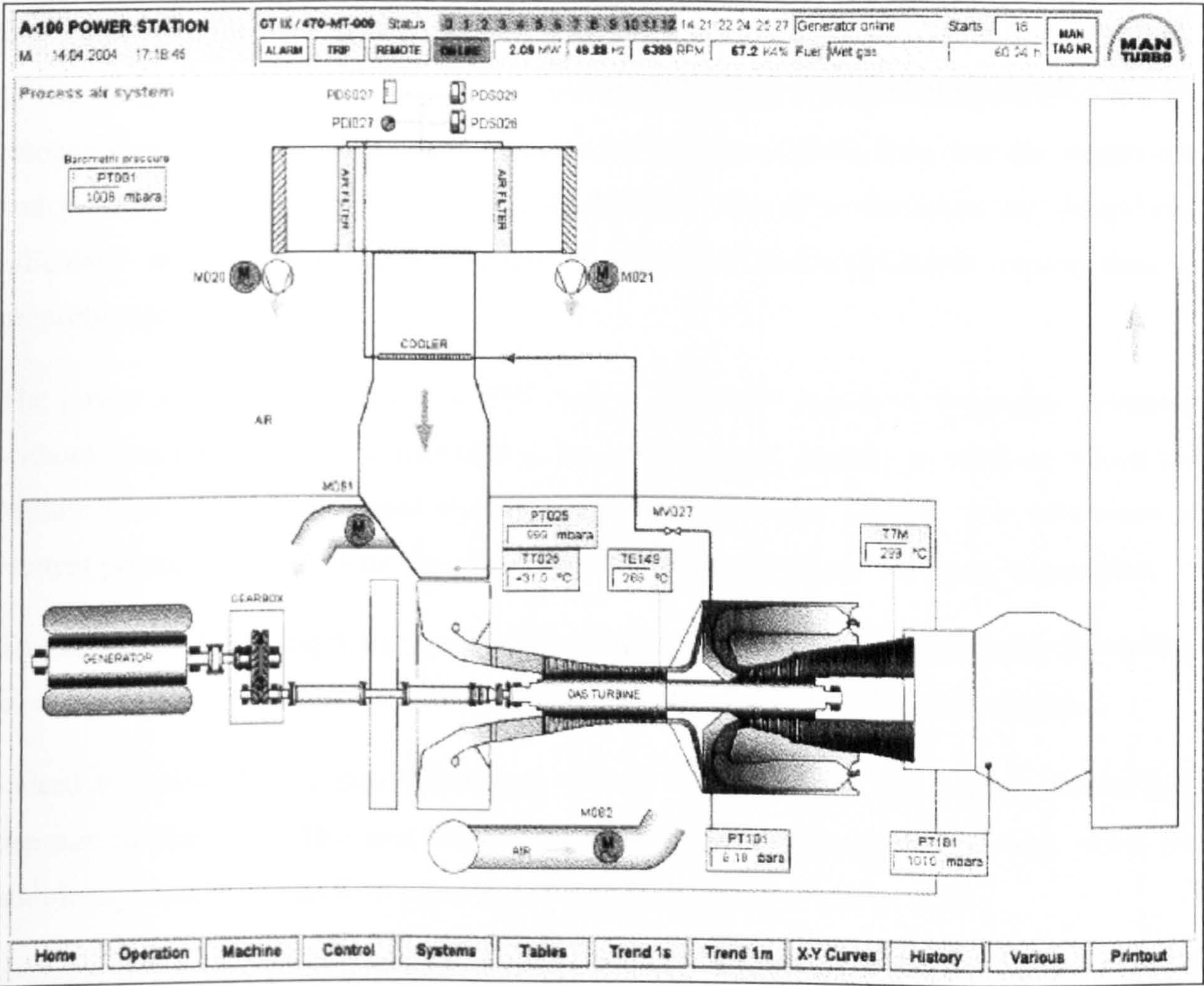


Figure 3-11: Process air system and ventilation (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

Process air system

The gas turbine process air system includes the air intake- and the exhaust gas system, the cooling- and sealing air system. The intake system consists of a two stage air cleaner system and filters the intake air of the compressor during operation (Figure 3-11).

The exhaust gas system directs the gas leaving from the turbine exhaust through a casing into the stack. To prevent the transfer of heat from the gas stream to vital components within the turbine and to minimize thermal stresses, cooling air is tapped from the compressor and piped to the turbine components.

Throughout the gas turbine there are several clearances between stationary and rotating parts. For sealing matters pressurized sealing air from the compressor is used to avoid leakage.

Some amount of air, tapped from the compressor, is used for the lube oil system (oil mist ejector), fuel system and flame detector cooling.

3.6 Gas Turbine Performance Simulation Using Turbomatch

3.6.1 Introduction of the engine working environment

In hostile environments the compressor component of a gas turbine is susceptible to fouling through deposits of oil and fine sand particles that pass through the filter into the compressor and possibly combined with salt. The combination over time decreases the compressor efficiency increasing compressor power demand and reducing output power demand proportionately.

The power loss can be as much as 10% over a 12 month period of the engine operating without cleaning. This fact is increased in hot environment typically in north of Africa and Middle East where the ambient temperature reaches 50 degree Celsius. The only ways to recover power loss are to clean the compressor regularly or increase the firing temperature.

The impact of increasing firing temperature is to increase fuel flow and reducing the turbine blade creep life. The reduction in creep life is more significant than fuel flow increase.

Indeed typically of 25 degree which may recover the full power will half the life of high pressure turbine rotor. This cost the primary reason for the compressor cleaning, whilst the additional benefits of fuel flow cost offset this to some extent (Figure 3-12).

Through this work a more accurate prediction of axial compressor fouling has been produced and aligned to observations seen in the field (GT Sulzer type 7).

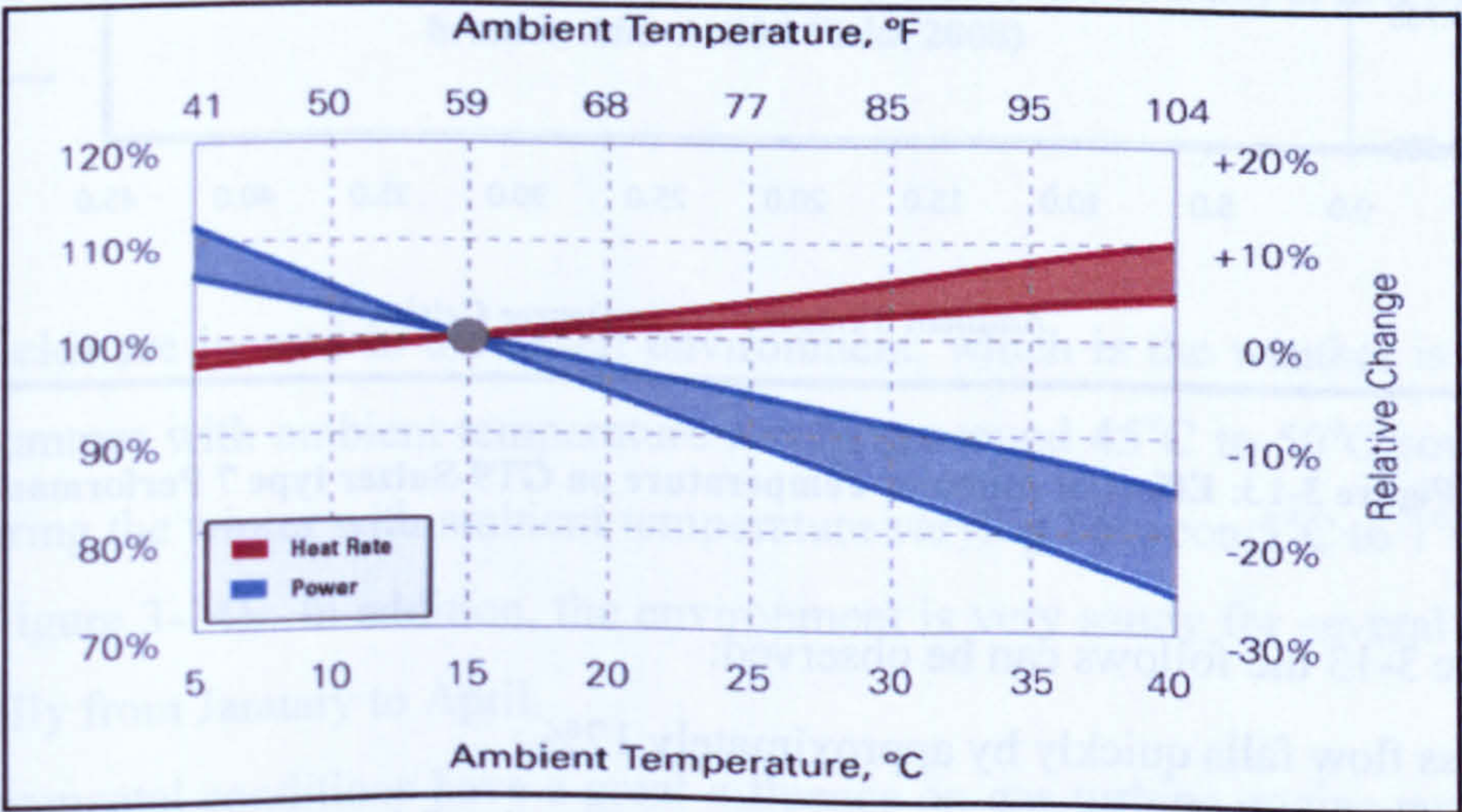


Figure 3-12: Ambient temperature effect on gas turbine performance

3.6.2 Influence of ambient temperature on the engine performance

The variation of ambient temperature has a significant effect on the change of gas turbine performance as shown in Figure 3-13. A different simulation of 11MW engine (Sulzer Type 7) has been studied with the Turbomatch Code developed from gas turbine performance group at Cranfield University.

This study shows as the ambient temperature increases to 45°C the power output decreases dramatically by approximately 25% and the heat rate increases by 10%, where the turbine inlet temperature remaining constant.

The variation of the ambient temperature in the engine operating location (desert environment) is shown in figure 2 over the whole year.

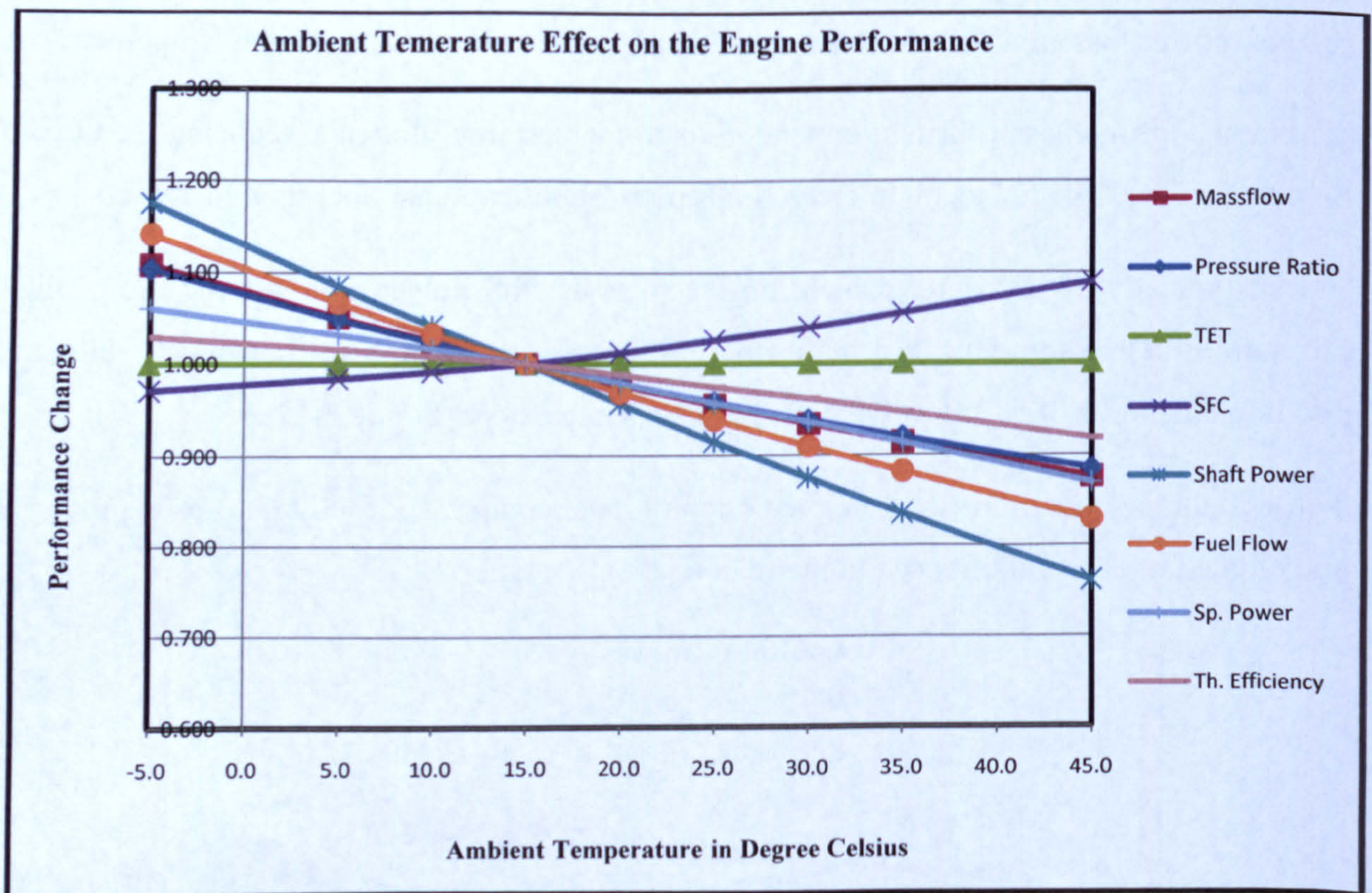


Figure 3-13: Effect of Ambient Temperature on GT9-Sulzer type 7 Performance

From Figure 3-13 the follows can be observed:

- The mass flow falls quickly by approximately 12%.
- The pressure drops by approximately 11 % as a result of the reduction in mass flow.
- The SFC increases slightly from to 10%

- The shaft power drops faster around 25%

There are two likely ways to react the changes in ambient temperature, by continuing the same power output, or by increasing the TET by increase the fuel flow in order to recompense the mass flow losses, that's occurs due to the reduction in the air density which decreases as the ambient temperature increase .The compressor needs more power from the turbine to compress the hot air, the fuel consumption increase and the thermal efficiency drops as a result of the reduction in power output.

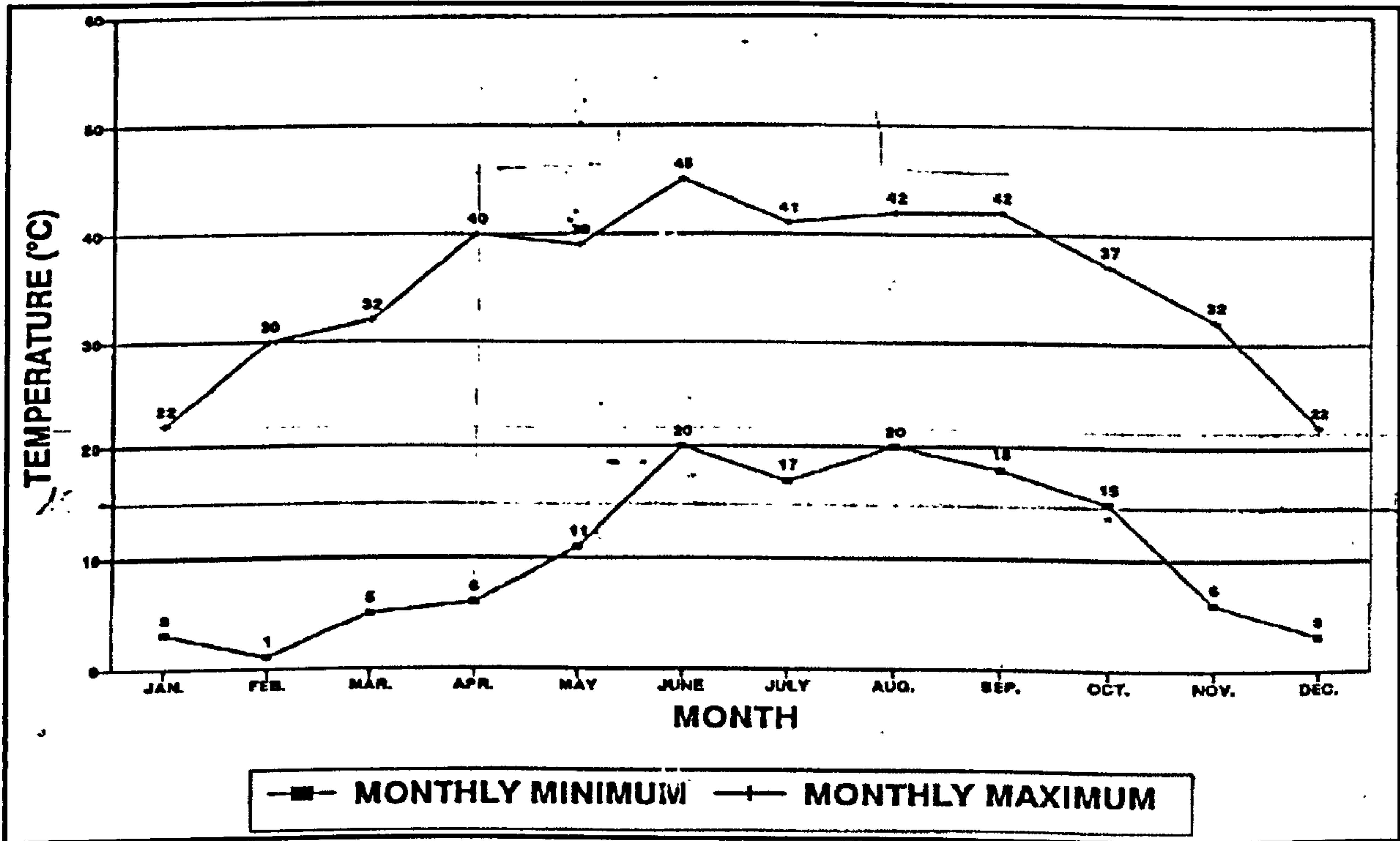


Figure 3-14: Ambient temperature variation in a desert environment (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

All the oil fields are located in the desert environment, which is the weather is hot and dry during the summer with ambient temperature reaching around 45°C to 50°C sometimes and very cold during the winter with ambient temperature varying between 5°C to 1°C during the nights see (Figure 3-14). In addition, the environment is very sandy for several months in a year, especially from January to April.

These environmental conditions have a great influence on gas turbine engine performance in two ways. Firstly, the change in ambient temperature and secondly the dust and sand that

leads to severe compressor fouling, corrosion and erosion. In addition, the atmosphere is laden with hydrocarbon particles from neighbouring plants.

3.7 Field Data Analysis

The data of the gas turbine engine Sulzer type 7 (11MW) has been collected from the power station in Libyan oil field. An actual data reading of power outlet, temperatures and pressures were taken and compared with the simulation data of turbomatch code. It has been demonstrated that the results of the code are closer to the actual field data (see table 16).

Table 16: Power output compression

Ambient Temperature [°C]	Field Data Power output	Turbomatch Code Power output	Deviation in %
15	10.24	10.40	1.56
30	8.89	8.61	-3.31
45	7.75	7.46	-3.67
50	7.37	6.95	-5.68

3.7.1 Observation of Exhaust Gas Temperature

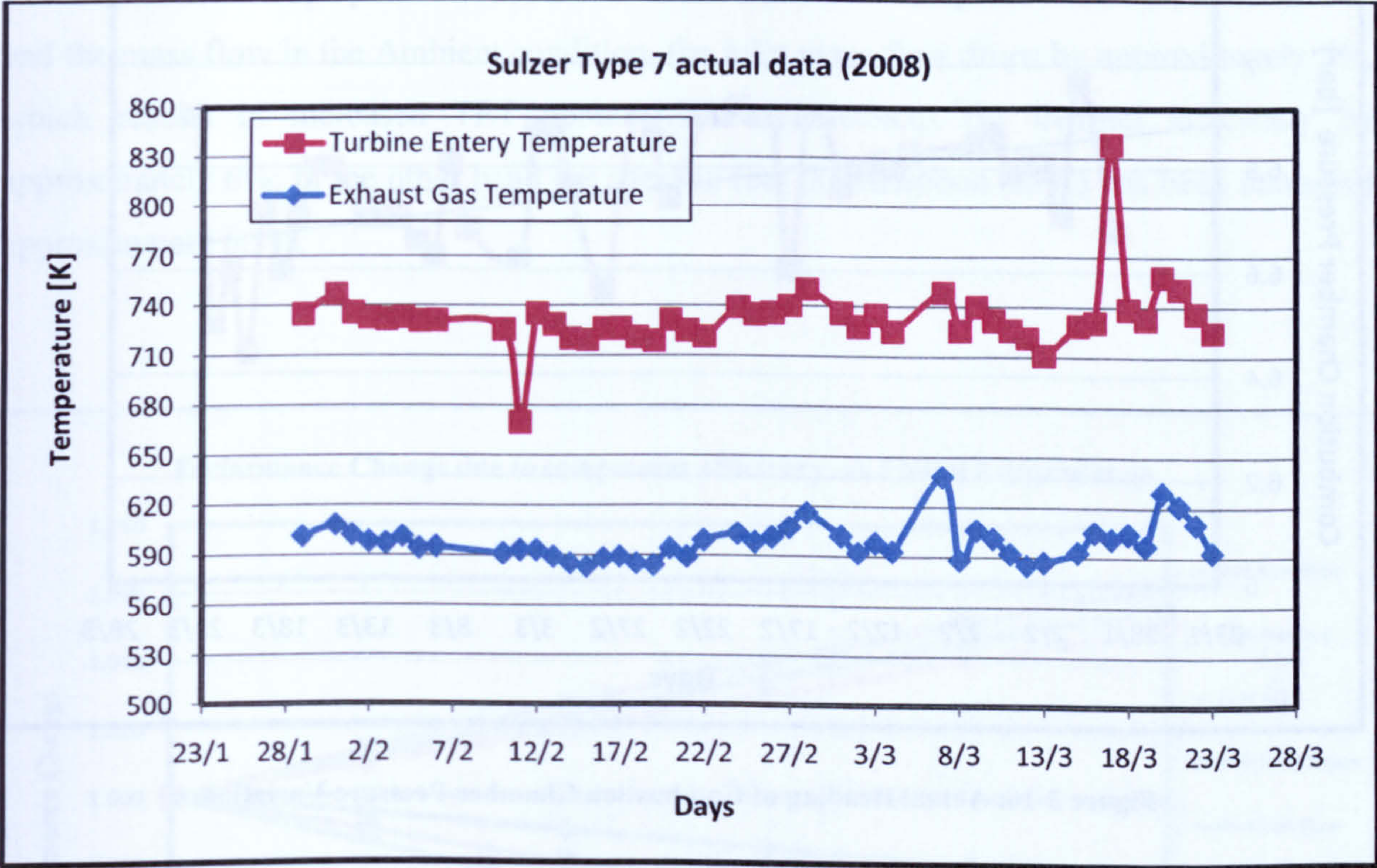


Figure 3-15: Actual reading of the temperature variation

As it discussed in the previous section 3.4 the indication of compressor fouling observed with the increase of exhaust gas temperature (EGT), increase of turbine entry temperature (TET) and decrease in combustion chamber pressure (CCP) see (Figures 3-15 and 3-16).

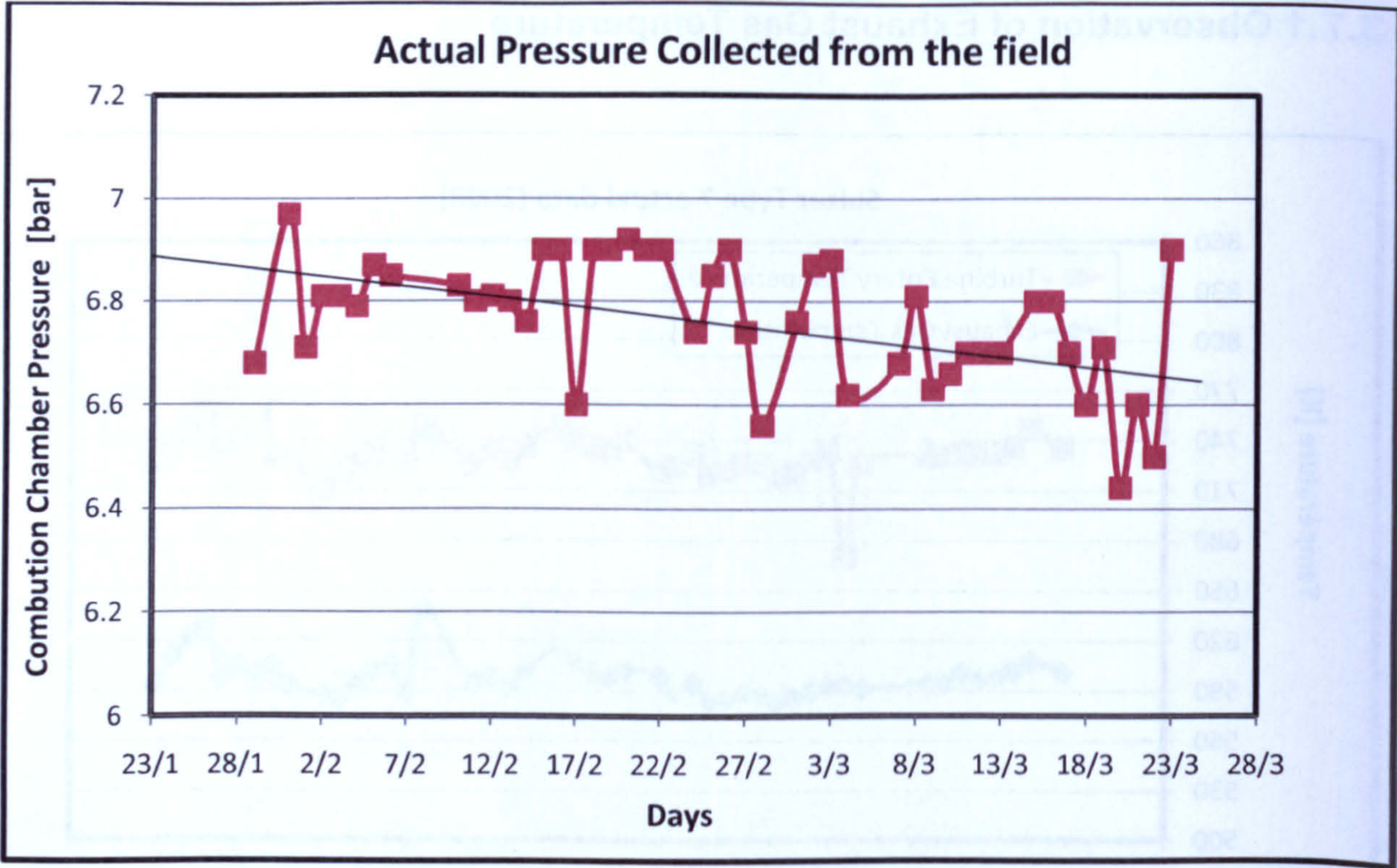


Figure 3-16: Actual Reading of Combustion Chamber Pressure Variation

3.8 Simulation of Clean and Fouled Gas Turbine

Gas turbine engine performance is extremely sensitive to the degradation of the components such as the compressor, combustor and turbine. The performance starts to deteriorate as soon as the engine is started for the first time, the inlet mass flow and the isentropic efficiency will degraded and is simulated separately. The representation of component degradation cases in table 1 implanted in the turbomatch code to predict the effect of the compressor fouling and erosion on engine performance.

Table 1: compressor implanted degradation

Physical Fault	NDMF	EFF	Ratio	Range
Compressor fouling	Drop	Drop	(1.0 : 0.5)%	0.0-(-5.0) %
Compressor EFF	-	Drop	-	0.0-(-2.5) %
Compressor erosion	Drop	Drop	(1.0 : 0.5)%	0.0-(-5.0) %

3.8.1 Compressor isentropic efficiency degradation

The Figure 3-17 displays the effect of the deterioration in compressor isentropic efficiency and the mass flow in the Ambient condition, the inlet mass flow drops by approximately 5%, which results in increased TET about 5.9% and reduces the thermal efficiency by approximately 6%. In the other hand the specific fuel consumption (SFC) has been increased approximately 6.5%.

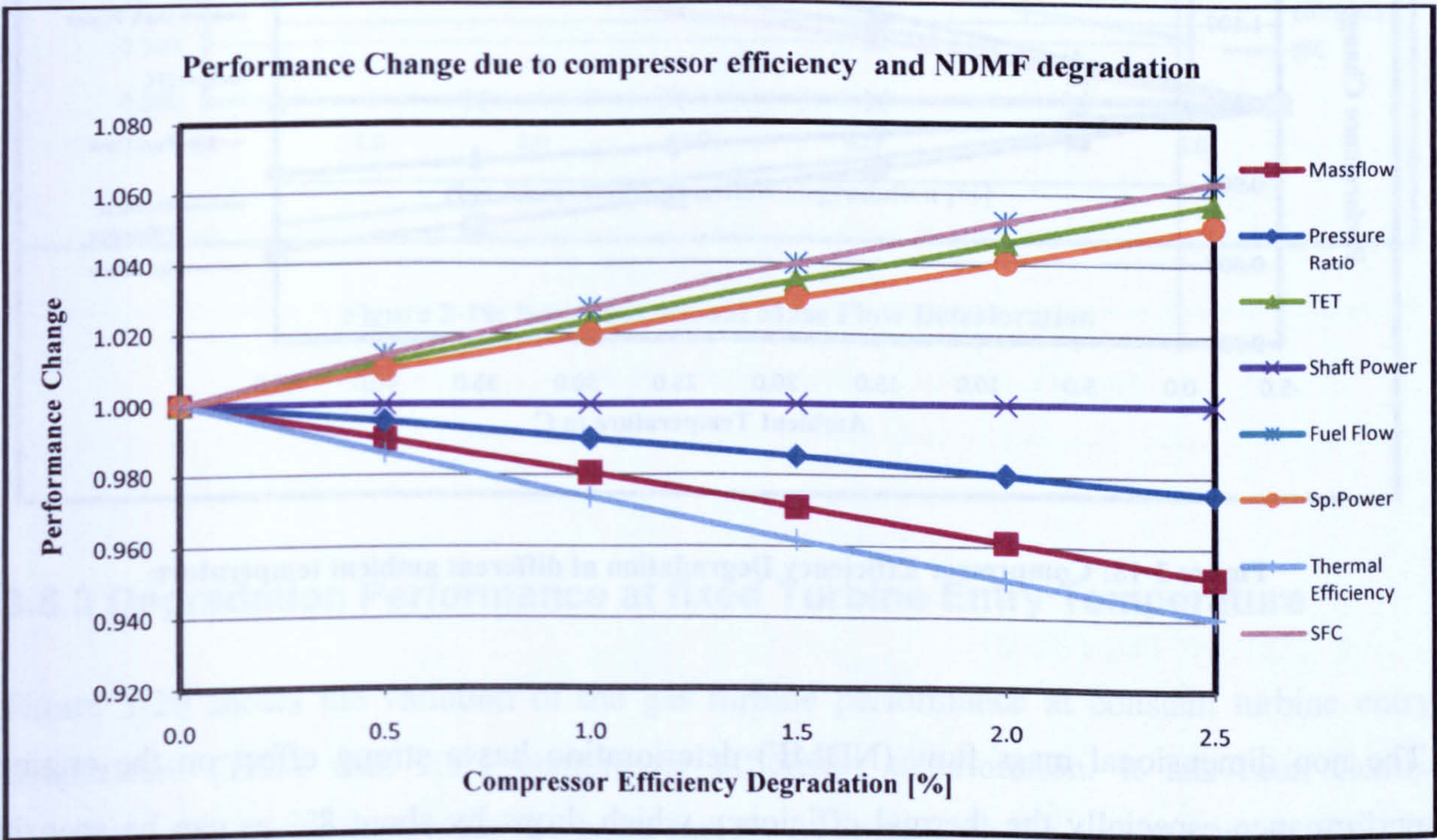


Figure 3-17: Compressor Efficiency and Mass flow Deterioration

3.8.2 Degradation Performance at fixed Power Output

The assumed operation mode in this work that the power output of the Gas Turbine cycle is maintained constant. This means that as the compressor inlet mass flow is reduced, the firing temperature must be increased to the given power outlet. As shown in figure 3-18 the turbine entry temperature (TET) increases around 10% at ambient temperature 15°C than increases rapidly with increase of ambient temperature.

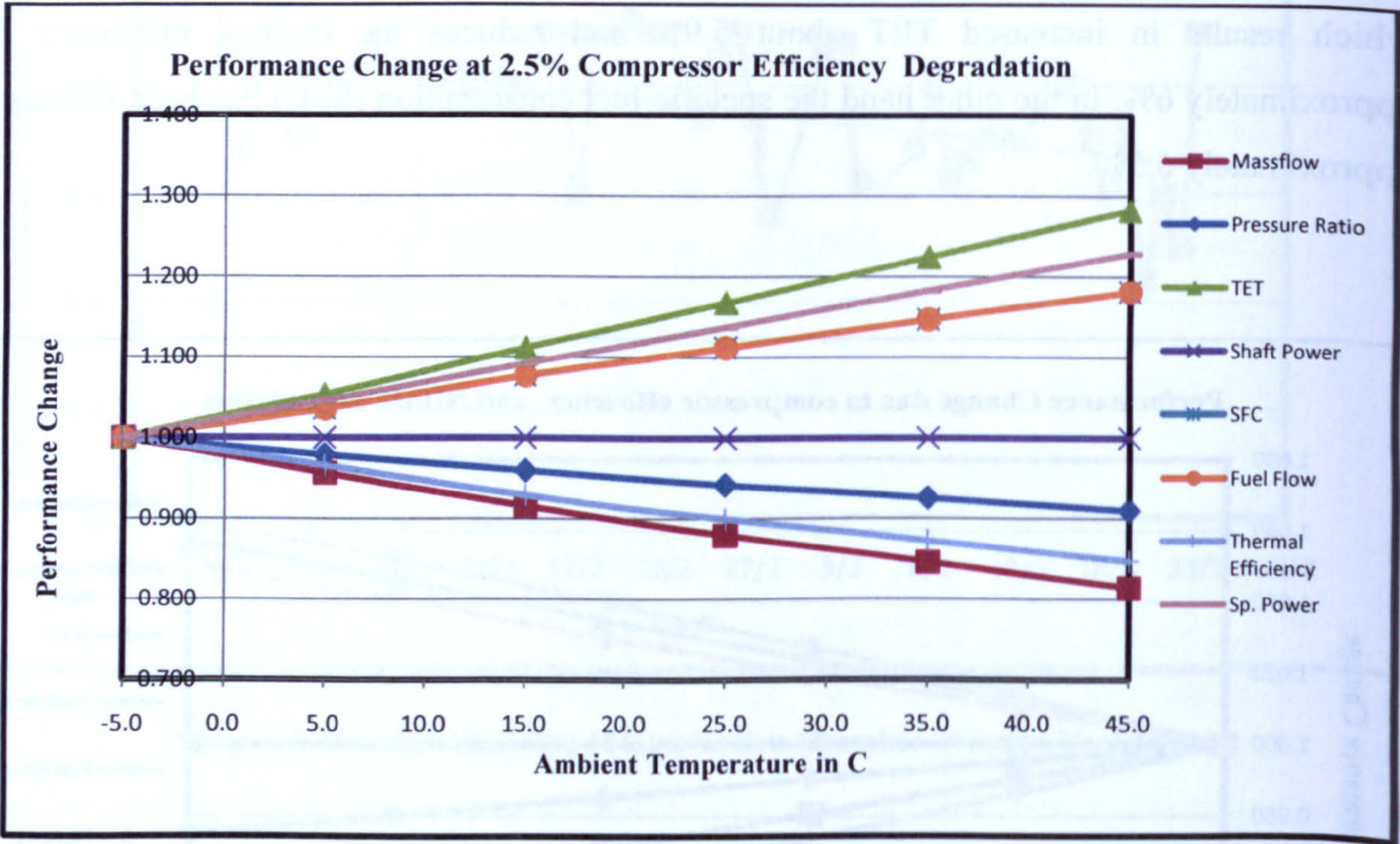


Figure 3-18: Compressor Efficiency Degradation at different ambient temperature

The non dimensional mass flow (NDMF) deterioration has a strong effect on the engine performance especially the thermal efficiency which drops by about 8% as can be seen in Figure 3-19.

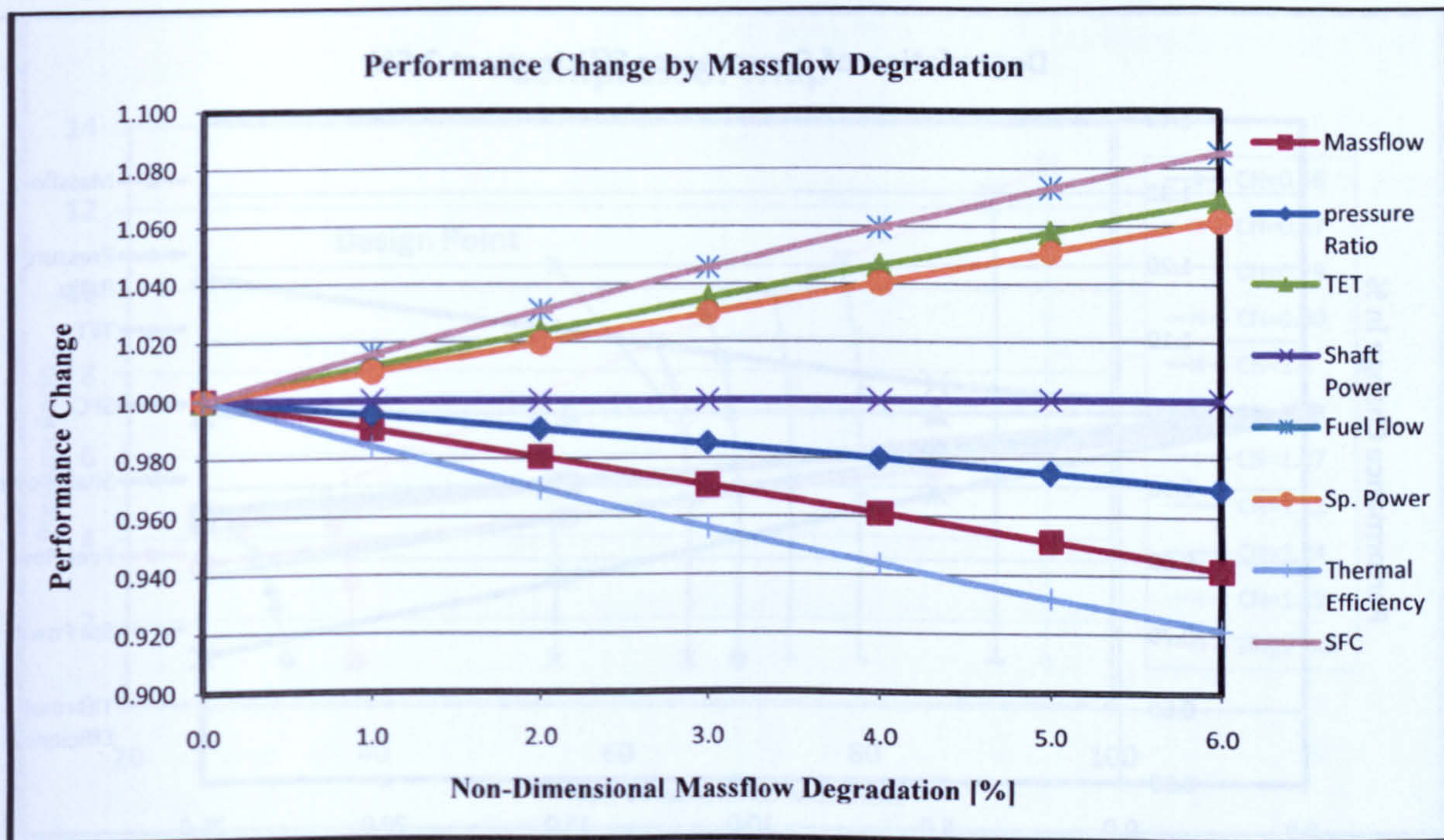


Figure 3-19: Non-Dimensional Mass Flow Deterioration

3.8.3 Degradation Performance at fixed Turbine Entry Temperature

Figure 3-20 shows the variation of the gas turbine performance at constant turbine entry temperature (TET) with 2.5% compressor efficiency deterioration. It has been clearly illustrated that the power output decreases rapidly by increasing the ambient temperature. at 15°C power output drops 25% and the specific fuel consumption rises by 10%.

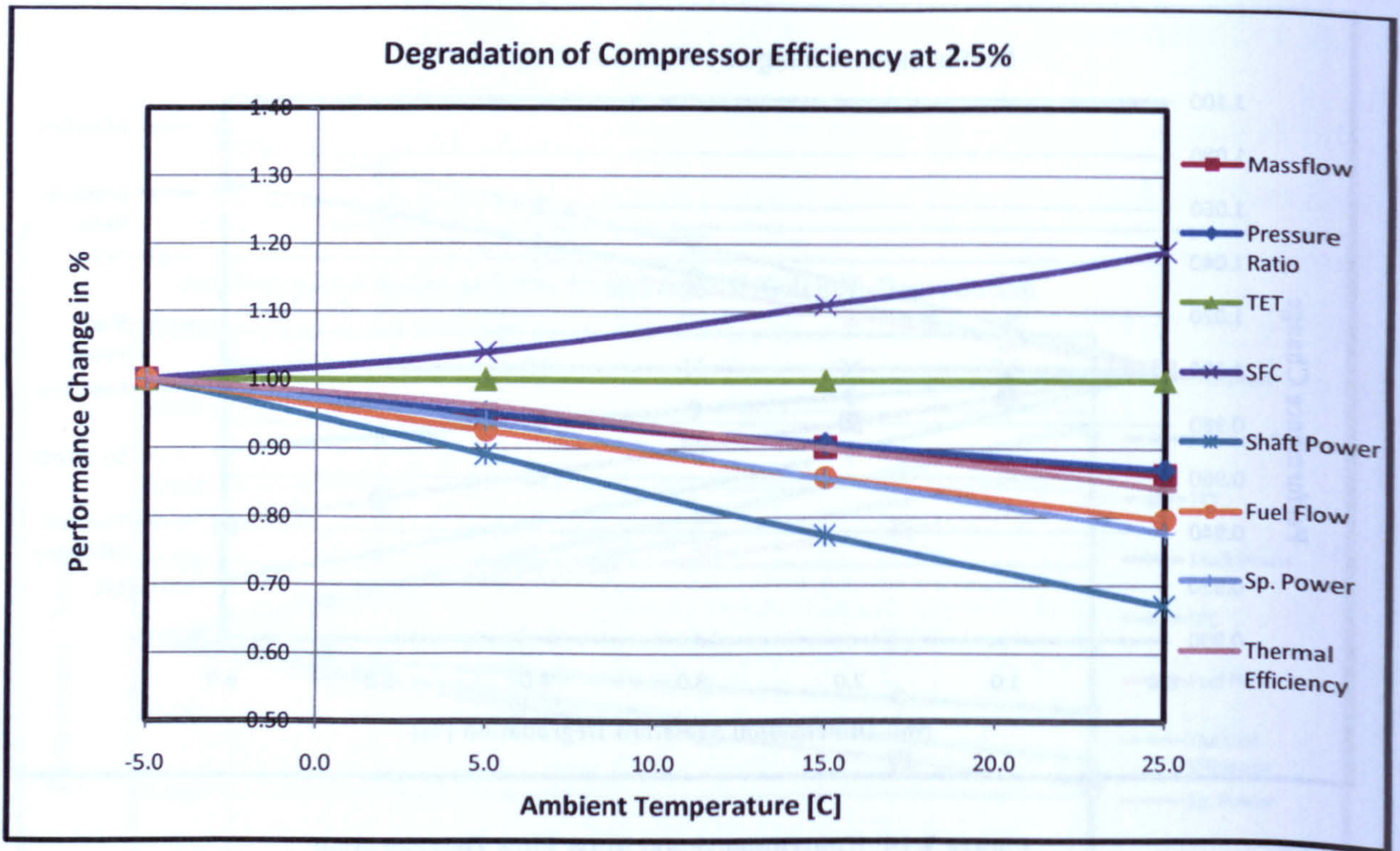


Figure 3-20: Performance Deterioration by Compressor Fouling

3.8.4 Compressor Map

The Figures 5 and 6 described the compressor characteristics, which show the design point of the gas turbine compressor Sulzer type 7. This is characteristic map of the compressor calculated at different non-dimensional speed (CN).

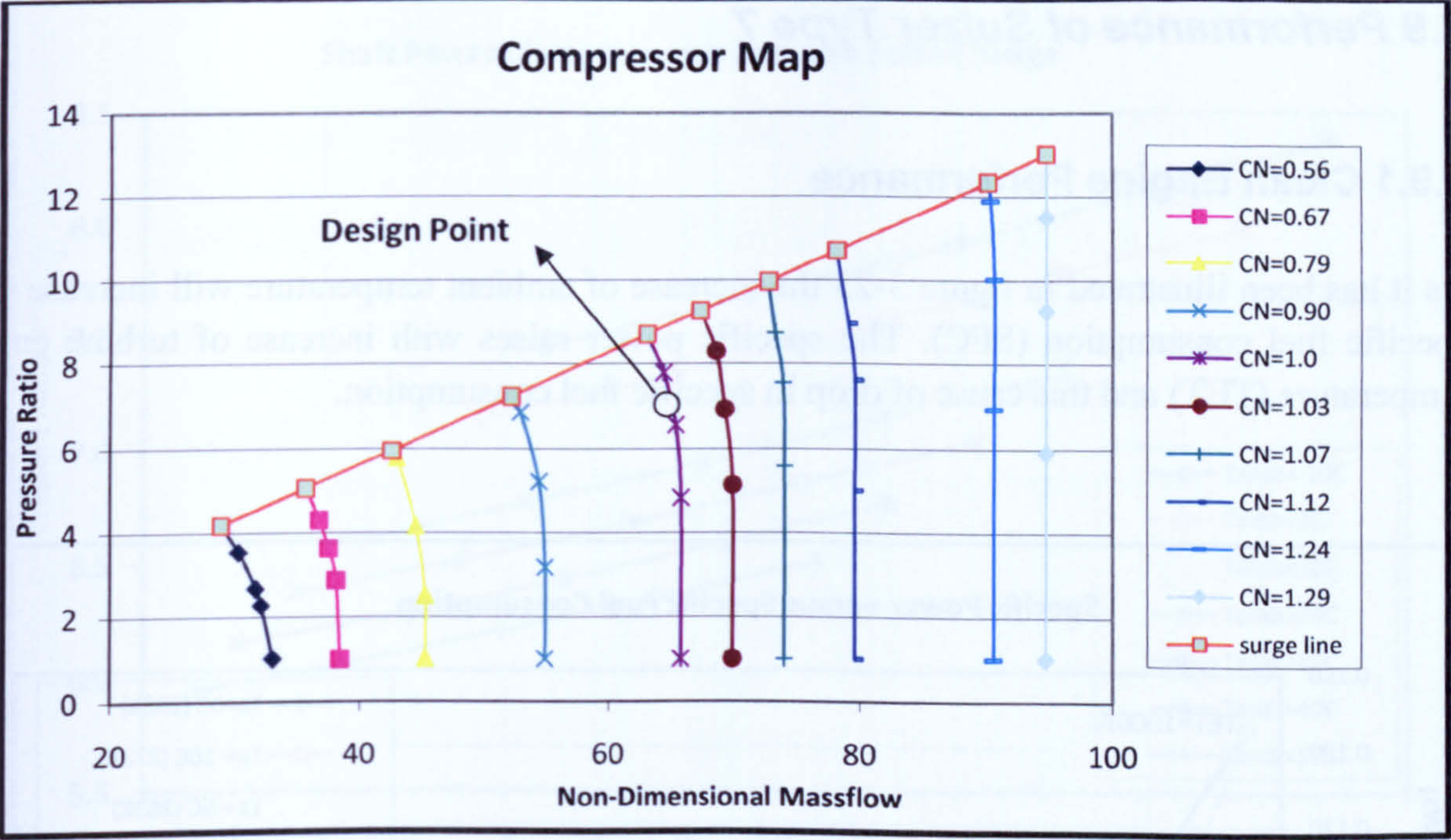


Figure 3-21: Compressor Map of Sulzer Type 7

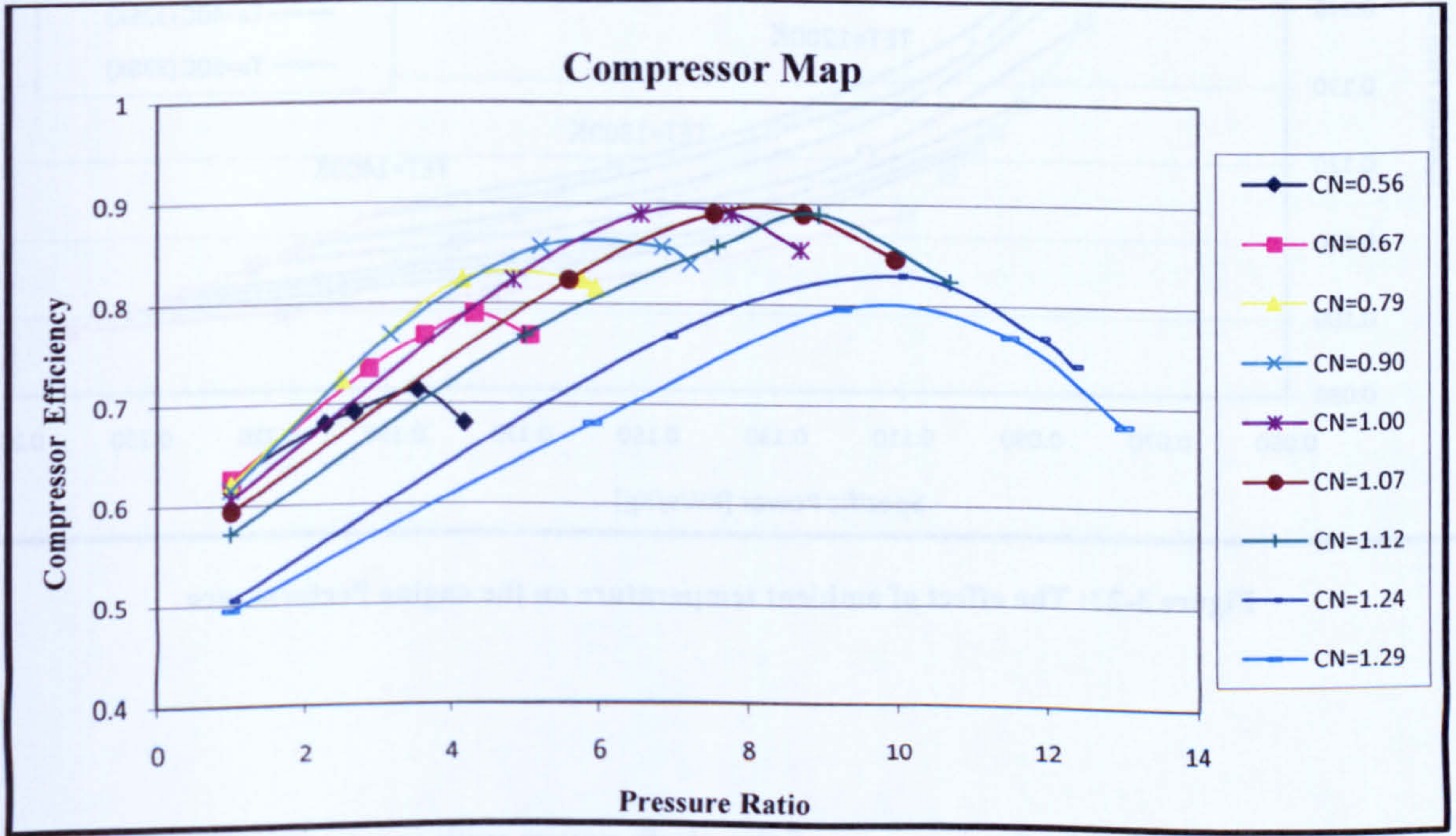


Figure 3-22: Compressor Efficiency

3.9 Performance of Sulzer Type 7

3.9.1 Clean Engine Performance

As it has been illustrated in figure 3-23 the increase of ambient temperature will increase the specific fuel consumption (SFC). The specific power raises with increase of turbine entry temperature (TET) and this cause of drop in specific fuel consumption.

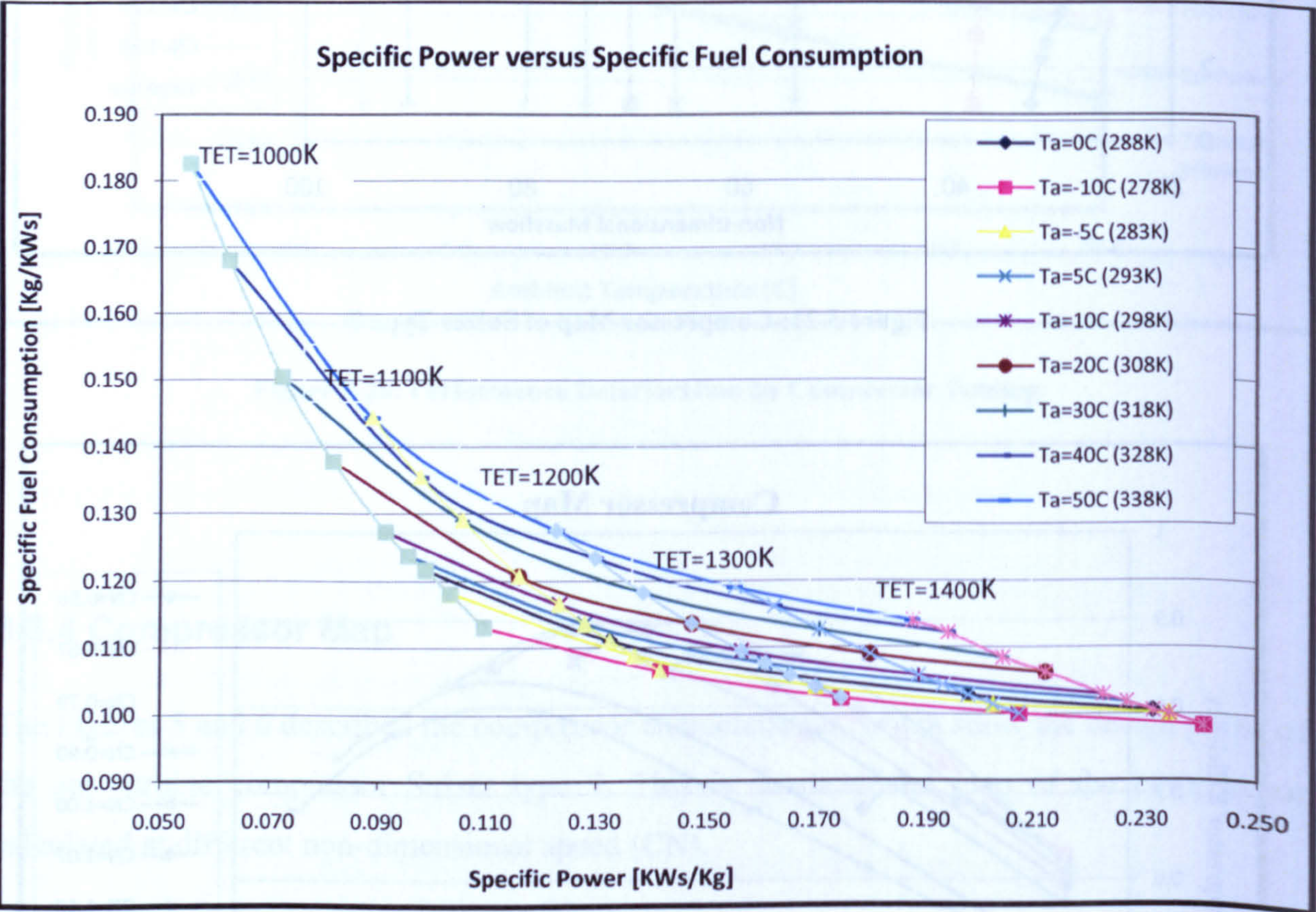


Figure 3-23: The effect of ambient temperature on the engine Performance

Figure 3-24 shows clearly the increase of the shaft power with increasing the pressure ratio and as the ambient temperature drops the power output will increase to reach the maximum power at the lowest ambient temperature.

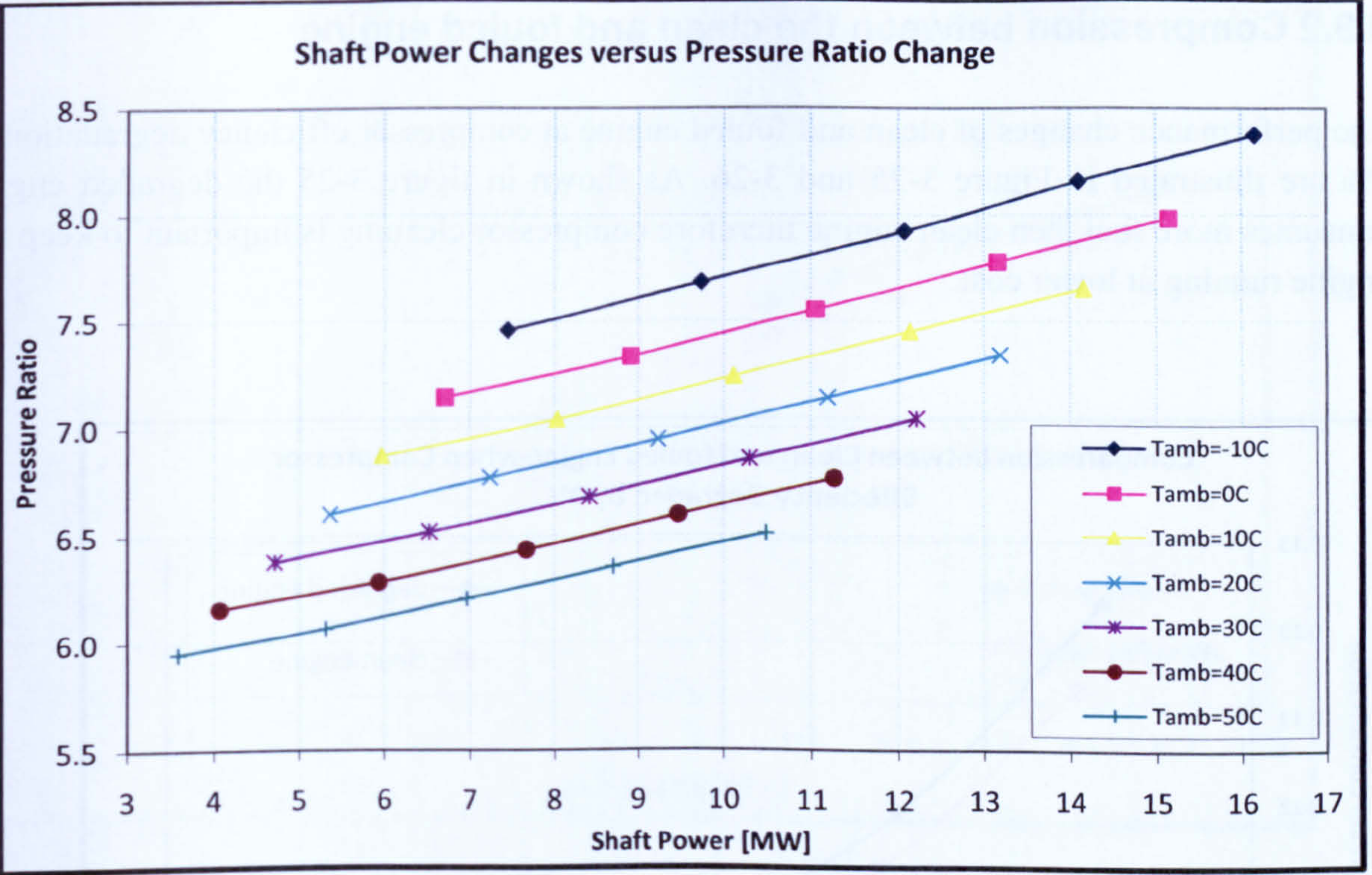


Figure 3-24: Increasing of Power with the Pressure Ratio

3.9.2 Compression between the clean and fouled engine

The performance changes of clean and fouled engine at compressor efficiency degradation of 3% are illustrated in Figure 3-25 and 3-26. As shown in figure 3-25 the degraded engine consumes more fuel then clean engine therefore compressor cleaning is important to keep the engine running at lower cost.

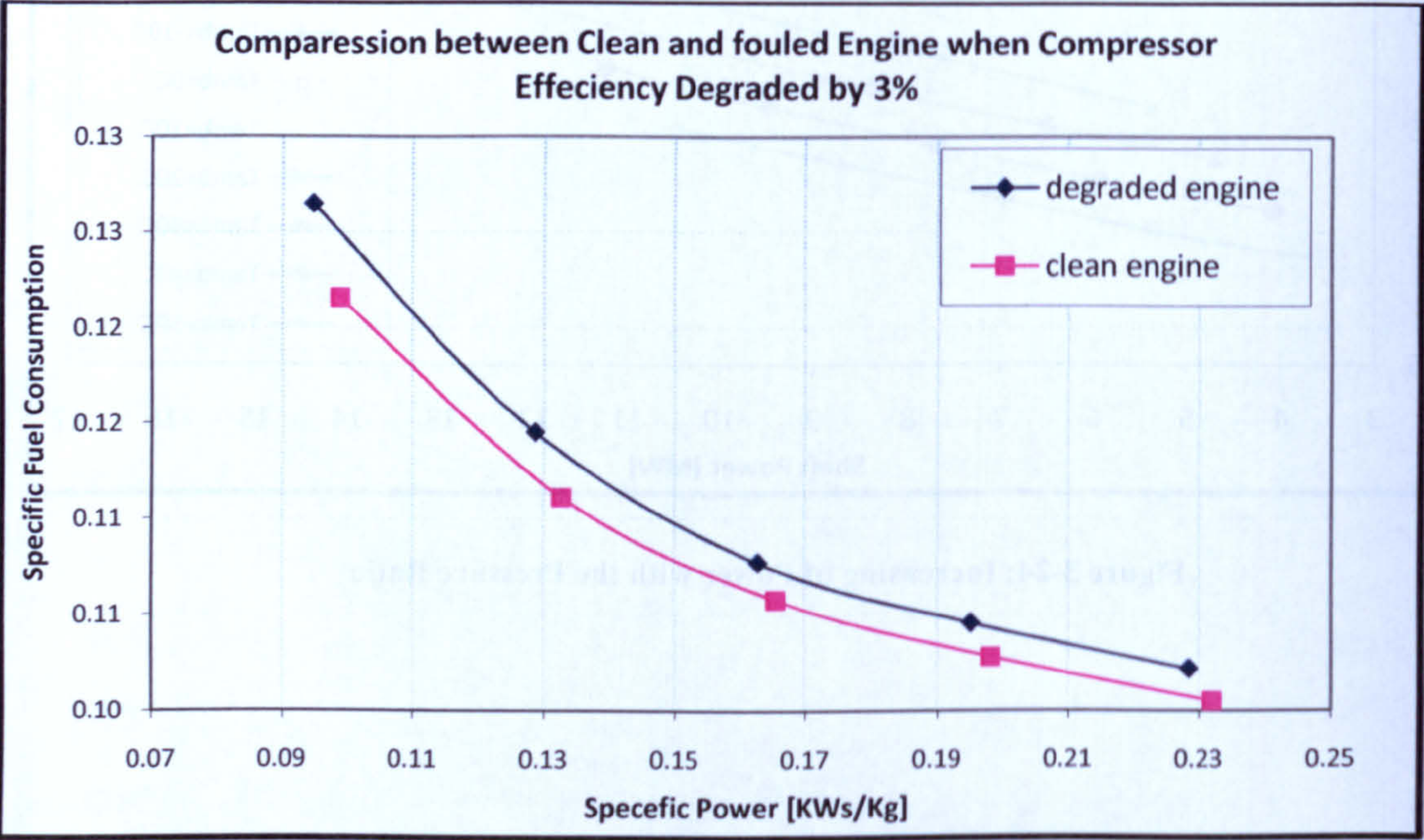


Figure 3-25: Compression between Clean and Fouled Engine Sulzer Type 7

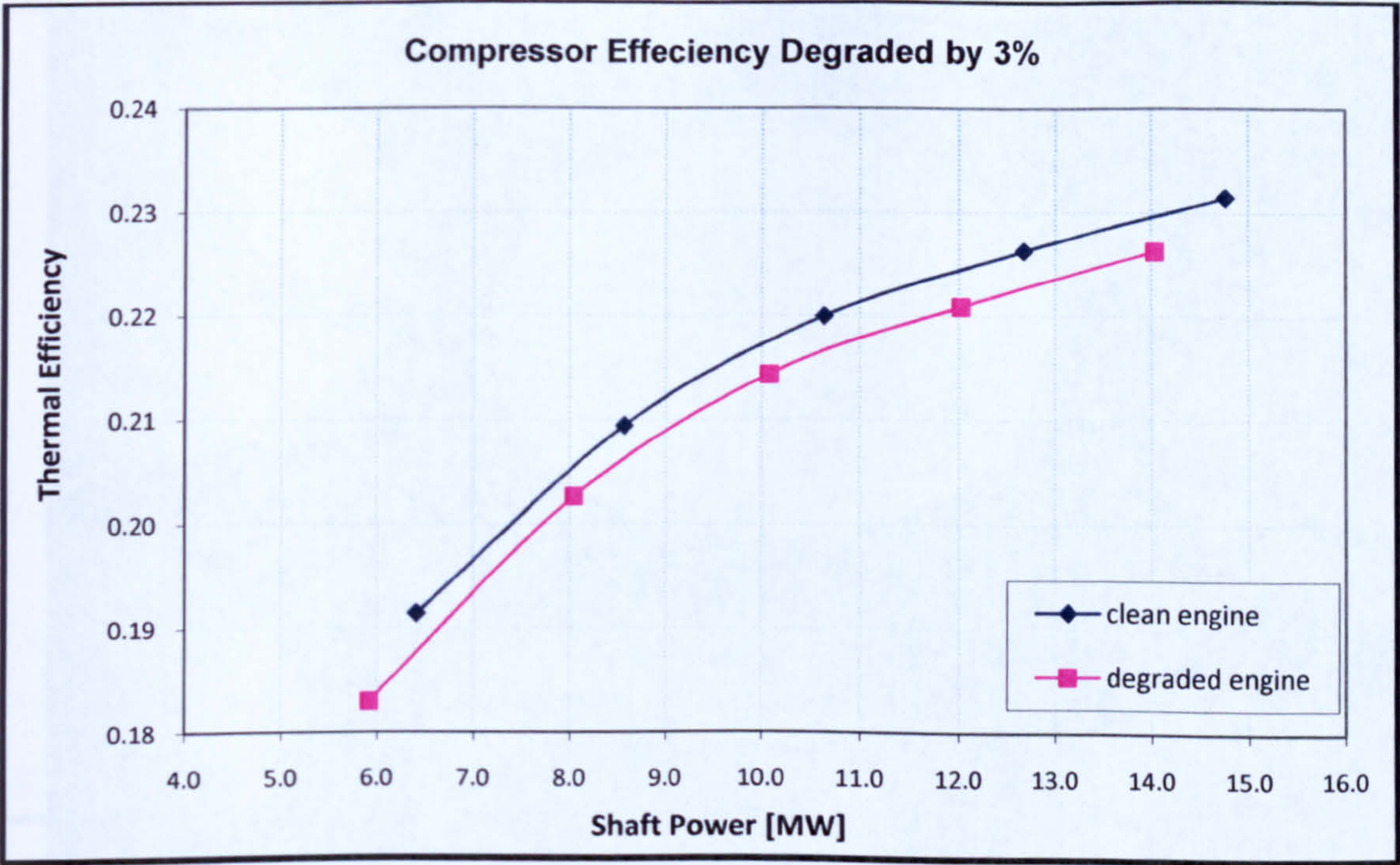


Figure 3-26: Simulation of clean and degraded engine

3.10 Inlet Filtration System

The purpose for installing air inlet filtration is to reduce atmosphere pollution, as mentioned above several kilos of pollution in the ppm range going through the filters will enter the engine and stick on the blades. To choose the proper filtration system, the drop in pressure cross the filter elements must be at minimum, in general the selection of a filtration system is largely dependent of the site location and operating condition as shown in (Figure 3-27).

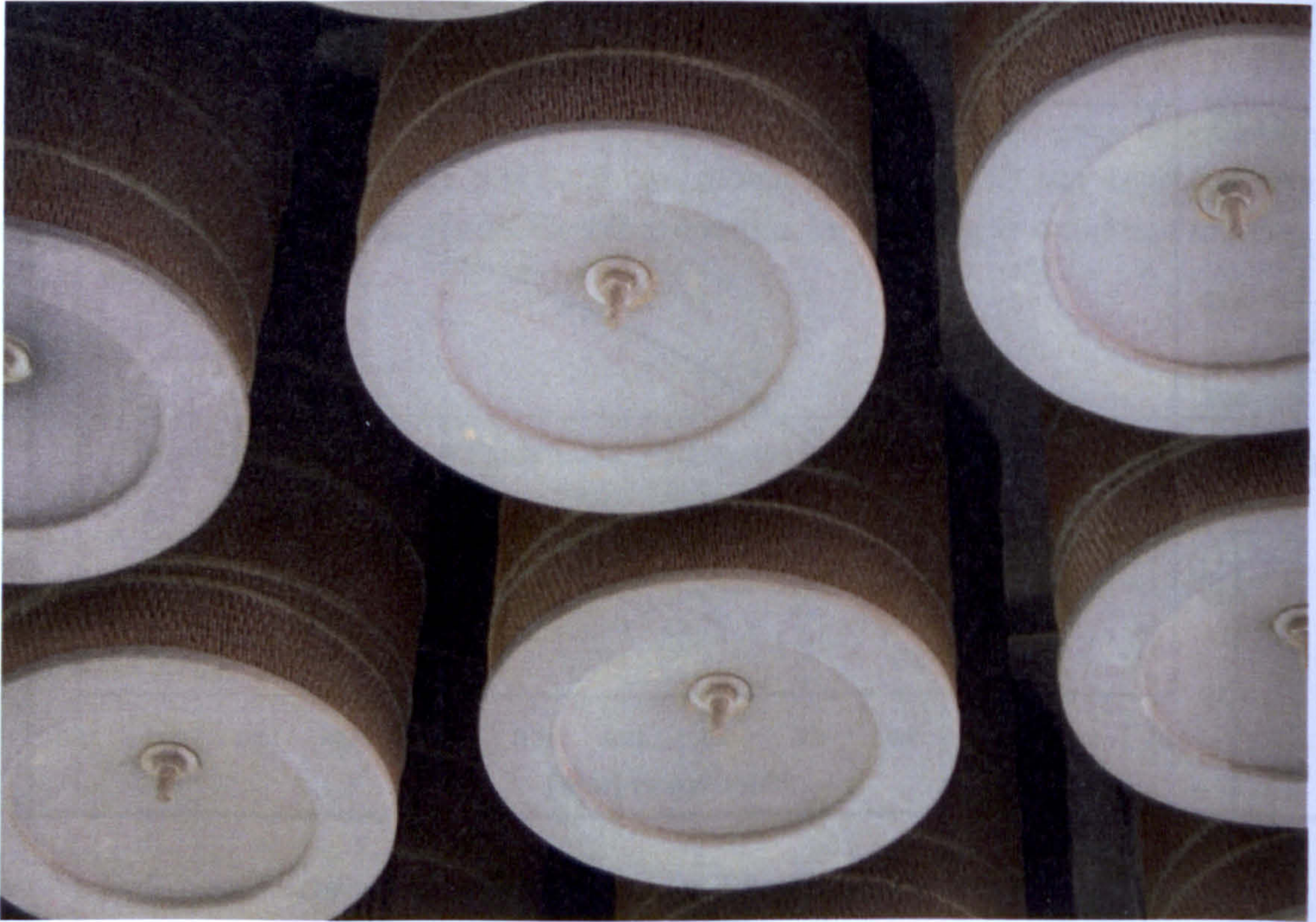


Figure 3-27: Self Cleaning Filtration System (Courtesy of ENI Oil – Libyan branch, Abu-Attifel Field, 2008)

The main types of filters are as follows:

Pre-filters - these are medium filters usually made of cotton fabric and are used to extend the life of a high efficiency filter further downstream. Inertial - this type removes the larger particulates from the inlet air. High efficiency media - these filters remove smaller dirt particles from the inlet air.

Marine or Demisters - these filters are used in marine environment to remove both moisture and salt.

Self-cleaning - these are composed of a number of high efficiency media filters. Air is drawn through the media at low velocity and at predetermined pressure drop across the system. A reverse blast of air removes built-up dirt on the filters and lowers the pressure drop. This filter is used in the Sulzer type 7 and is the most common standard for gas turbines used in desert environment (Figure 3-27). In addition performance calculation has been obtained to investigate the effect of intake pressure drop of the engine performance (Figure 3-28).

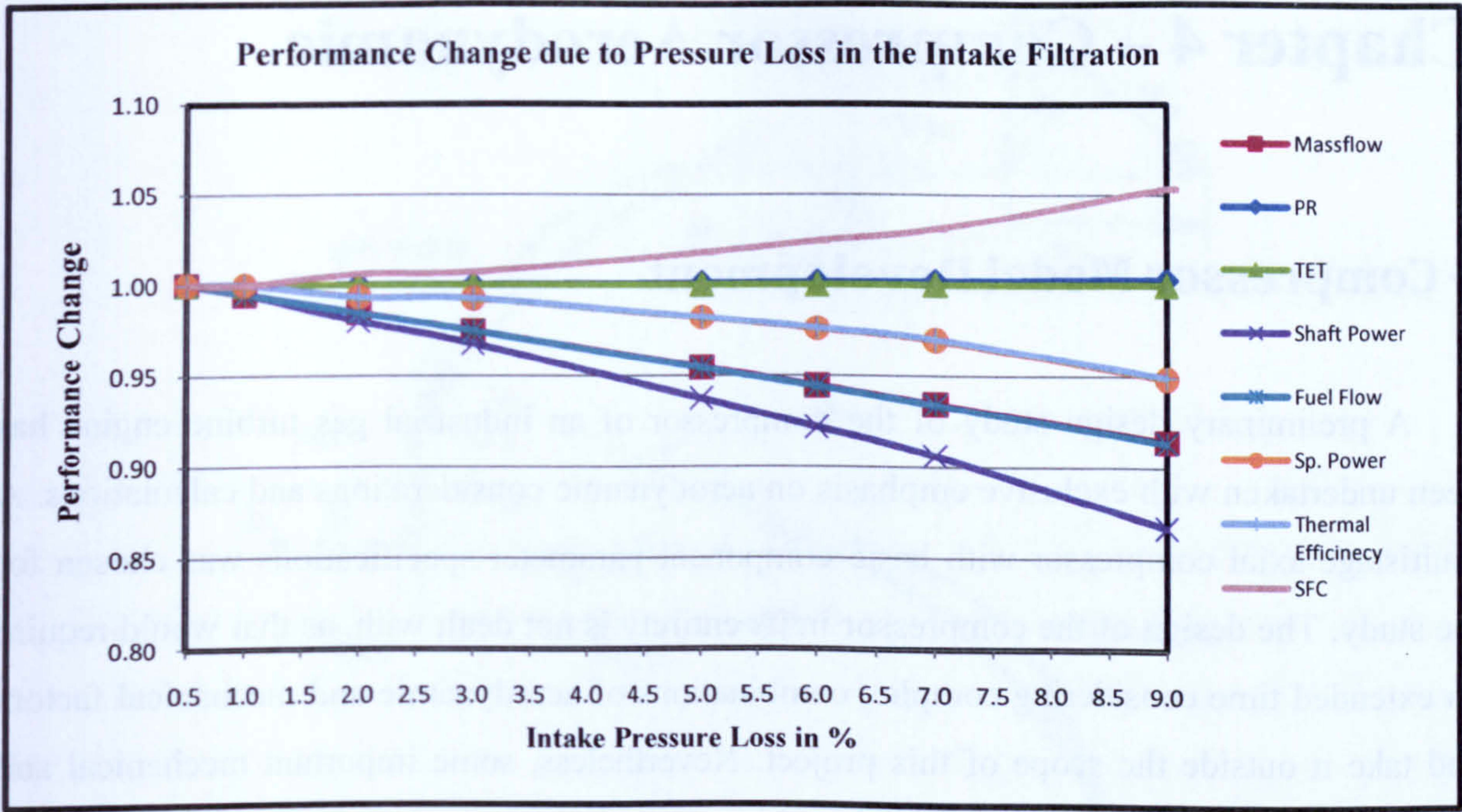


Figure 3-28: Intake Filtration System

Chapter 4 – Compressor Aerodynamic

4 Compressor Model Development

A preliminary design study of the compressor of an industrial gas turbine engine has been undertaken with exclusive emphasis on aerodynamic considerations and calculations. A multistage axial compressor with basic component parameter specifications was chosen for the study. The design of the compressor in its entirety is not dealt with, as that would require an extended time considering complex combinations of aerodynamic and mechanical factors and take it outside the scope of this project. Nevertheless, some important mechanical and aerodynamic constraints have been included to make the preliminary design more realistic.

The compressor stage of turbo-machinery works by transferred energy from moving blades to a moving fluid by means of changing swirl, or tangential velocity. In multi stage compressors, the first stage is usually preceded by static inlet guide vanes which introduce an appropriate level of swirl into the first stage of the machine. Both the moving rotor blades and the stator blades in any given compressor stage have been designed to diffuse and slow down the moving fluid which transfers its kinetic energy (KE) into internal energy. As a result, both static enthalpy and static pressure of the fluid rise.

Figure 4-1 shows the velocity triangles for an axial compressor stage. At the rotor inlet, the fluid flow is directed so that the fluid approaches the rotor with a high relative velocity and relative KE. In the rotor passage, this flow is diffused so that the relative KE is reduced. At the same time, the rotating blades do work on the fluid which increases its total enthalpy. Thus the static pressure and static enthalpy of the fluid are increased by a combination of two effects: the external work input from the blades which increases the total enthalpy of the fluid, and the internal transfer of KE as the velocity is reduced. This is shown schematically in Figure 4-2.

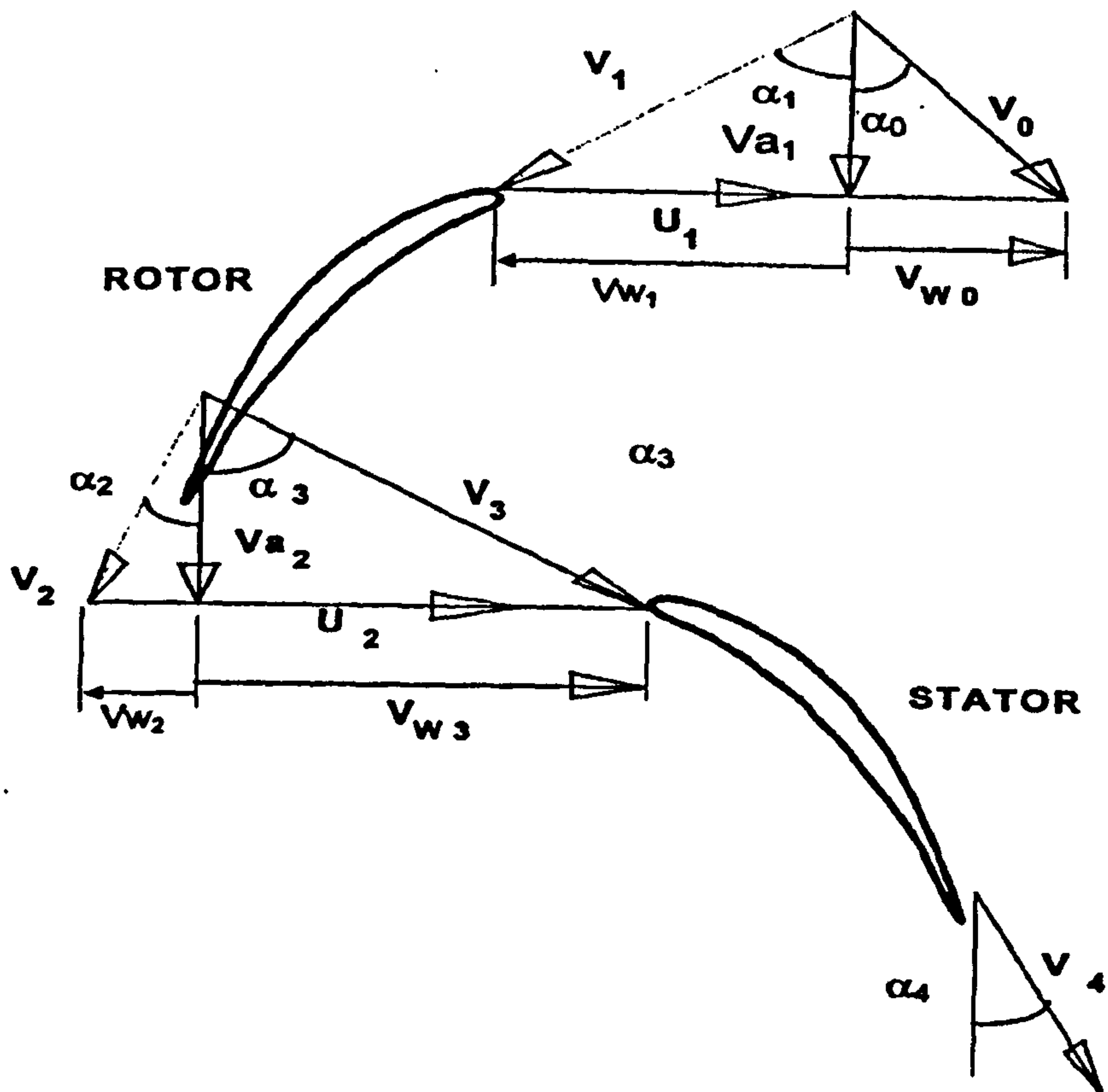


Figure: 4-0-1: Single stage velocity triangle (Ramsden, 2002)

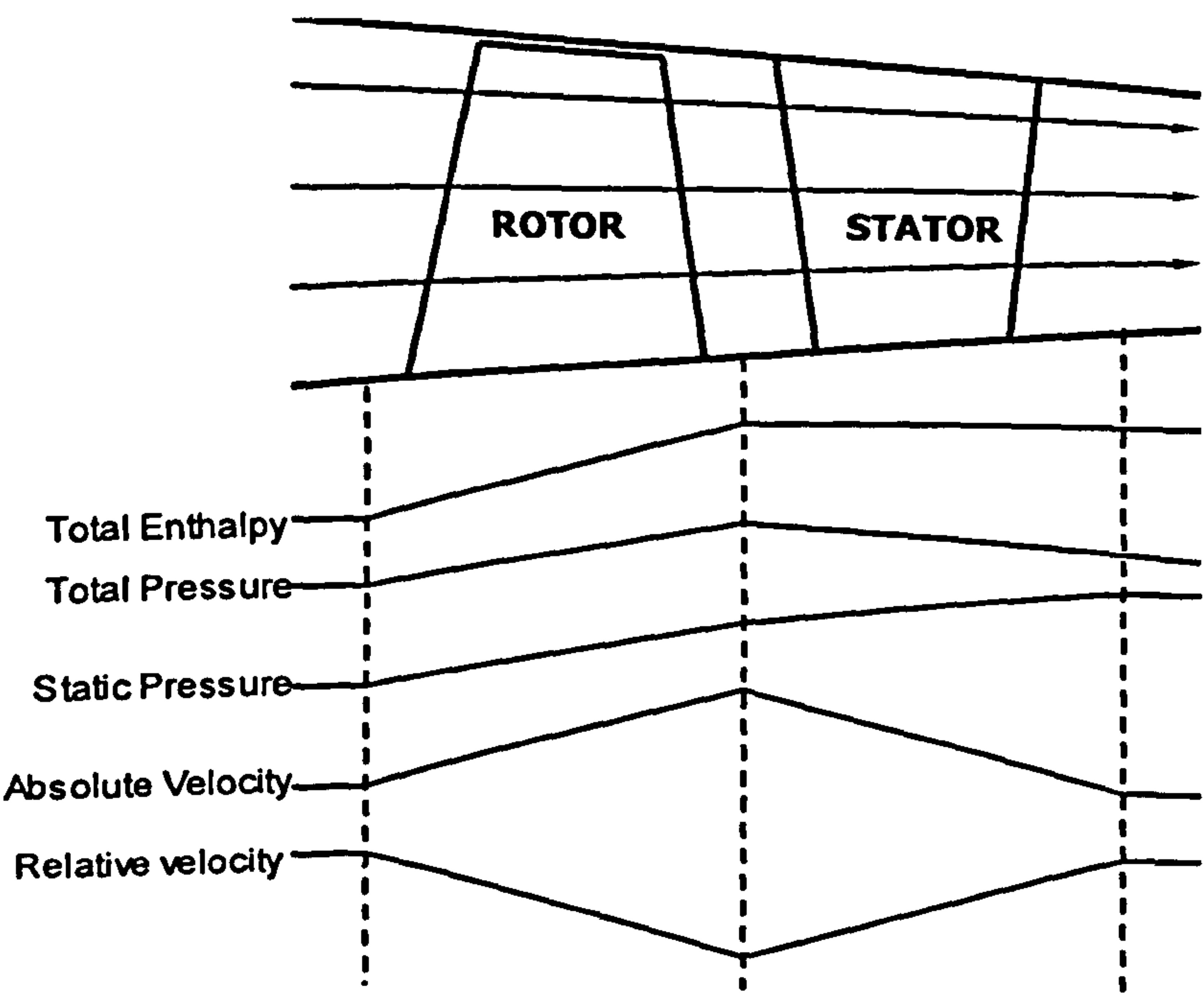


Figure 4-2: Schematic of changes in fluid properties and velocity through an axial compressor stage

In the stator, similar processes take place. The velocity of the fluid as it approaches the stator blades is at its maximum. In flowing through the stator passage it is slowed down and its KE is again transformed into internal energy with a resulting rise in static pressure. The fluid leaves the stator with a lower velocity than it entered the inlet. Unlike the rotor the stator is fixed and no external work transfer occurs, so there is no change in the total enthalpy across the stator.

4.1 Blade Aerodynamic Design

Real flow within axial flow compressors is extremely complex. Figure 4-3 is the well-known representation of (Farmakalides, 1992) which has been widely used to demonstrate the complexity of the flow existing within the passage of the rotor blades. The 3D nature of the flow within axial flow compressors requires the employment of 3D Navier-Stokes numerical solutions. However before the required computational power was available a limited measure of the three-dimensional flow field could be obtained in terms of inter-related 2D surfaces.

Figure 4-4 describes the system proposed by (Wu, 1952) in which S1 and S2 represent stream-surfaces which twist through the blades. This analysis approach has rarely been attempted however, because the stream-surfaces change shape with each iteration.

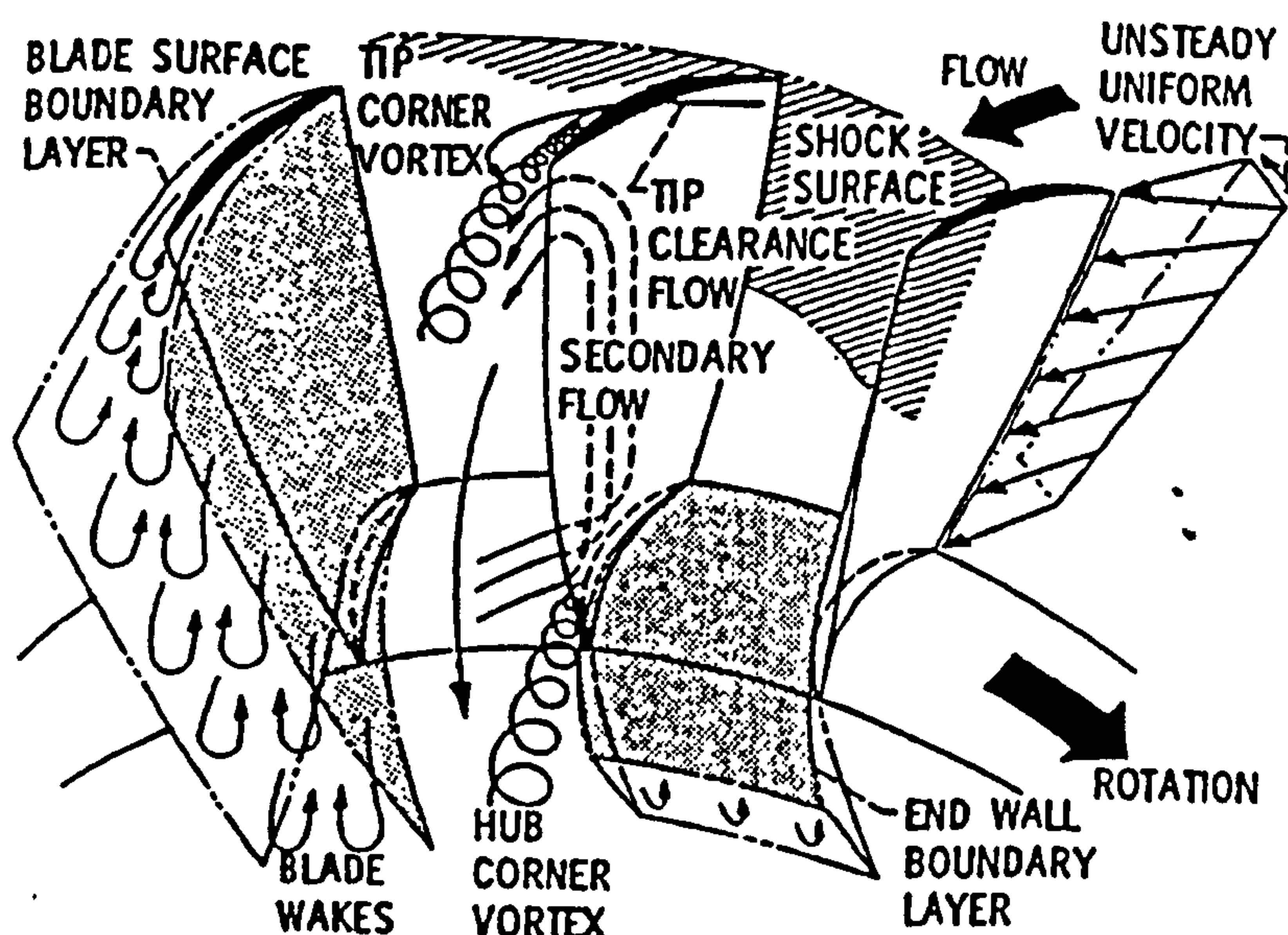


Figure 4-3: Compressor Blade Row Flow Field Characteristics (Farmakalides, 1992)

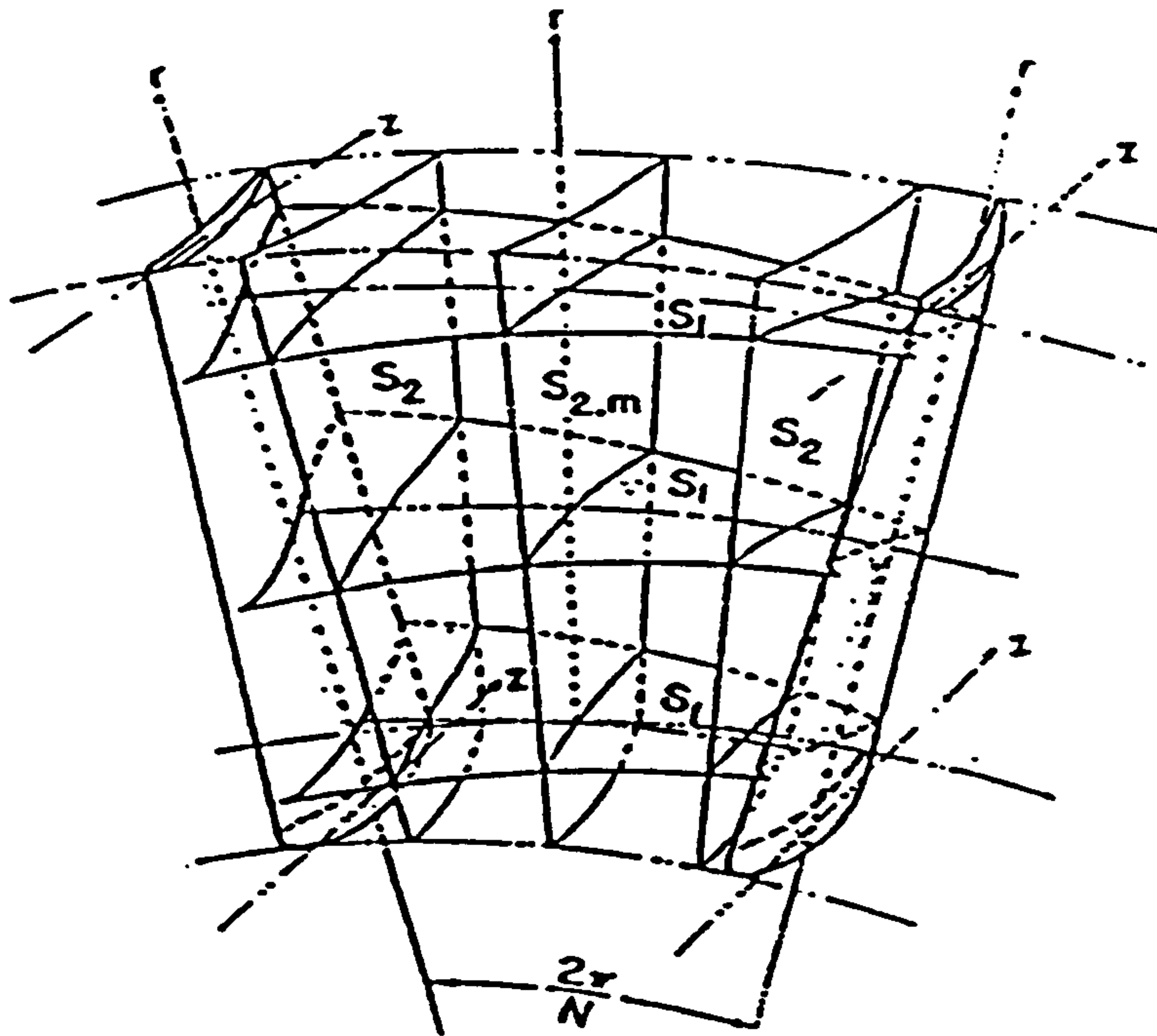


Figure 4-4: Relative Stream Surfaces S1 and S2 (Wu, 1952)

A more generally adopted approach can be seen in Figure 4-5 which shows the flow on cylindrical blade to blade surfaces along the span of the blade and on a mid-pitch meridional plane (a plane which contains the axis of rotation and is swept around that axis). The analysis techniques used for these planes are progressed through successive developments.

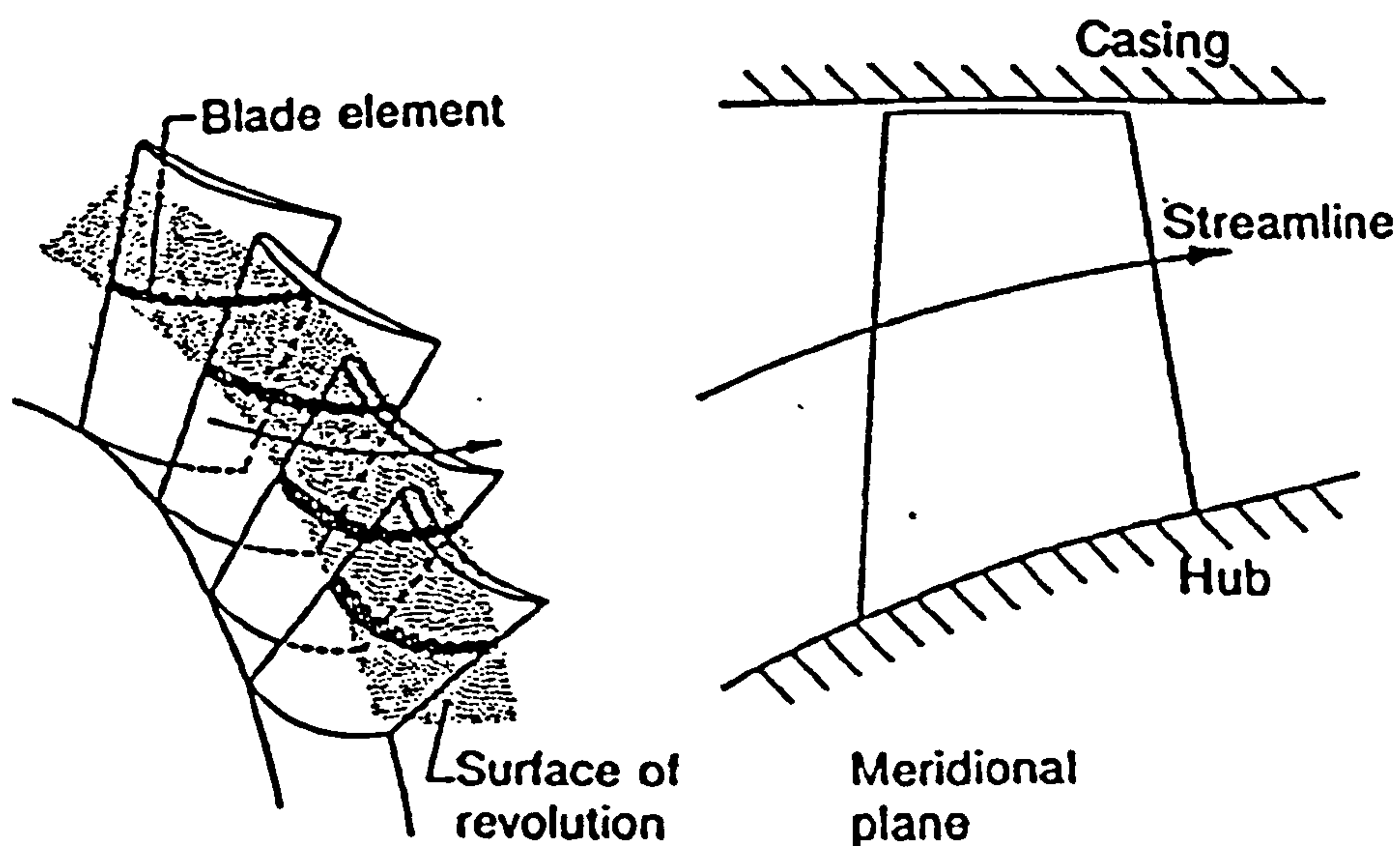


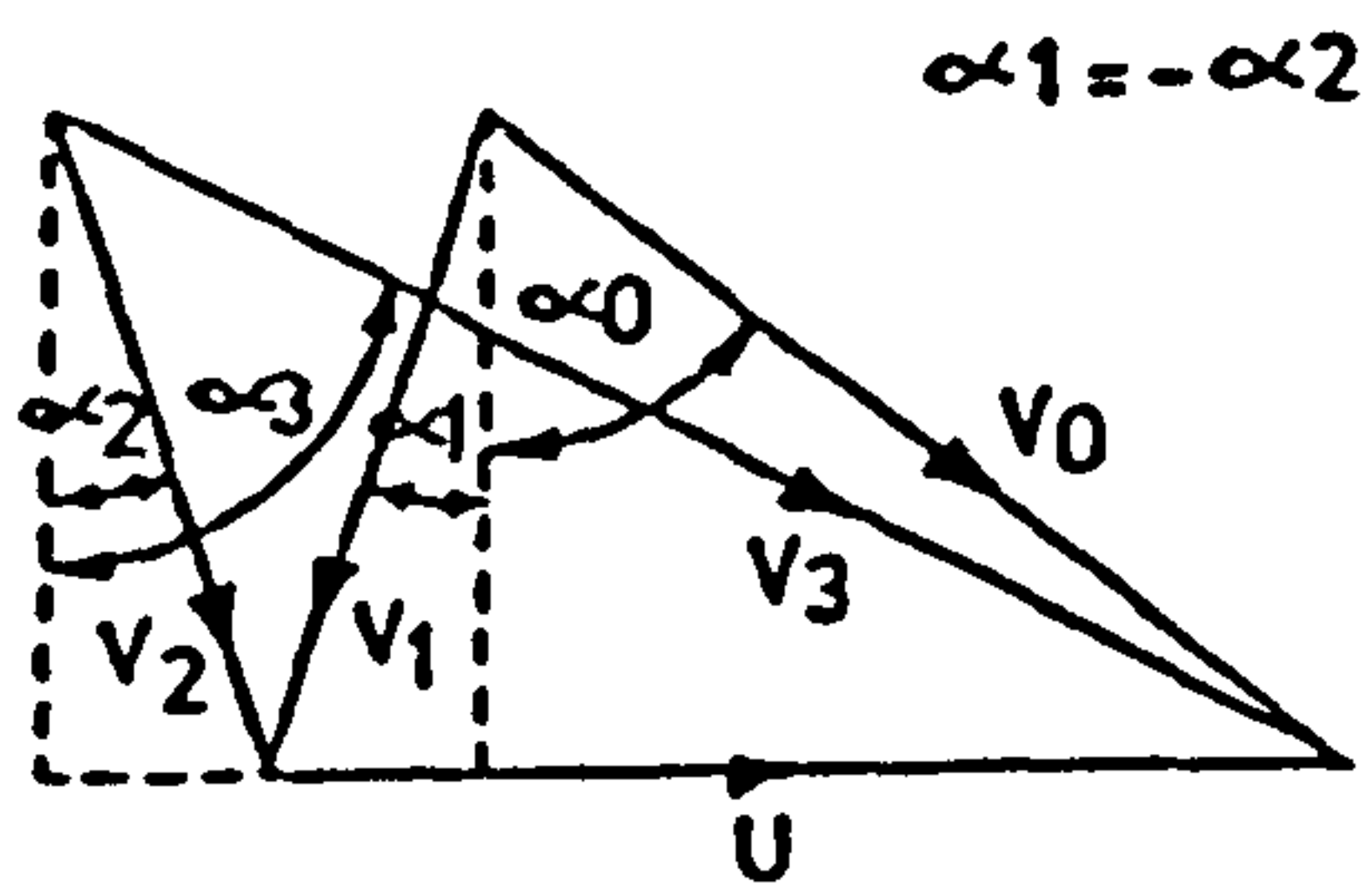
Figure 4-5: Conventional description of flow in compressors on surface of revolution (blade to blade surface) and on meridional plane (Howell, 1948)

In recent years computer software packages have been developed which calculate 3D flows numerically – either solving the compressible Euler equations by treating the flow as inviscid, or fully solving the compressible Navier Stokes equations for defined boundary conditions. These numerical techniques depend heavily on empirical data and levels of accuracy vary with what aspects of the flow the package predicts.

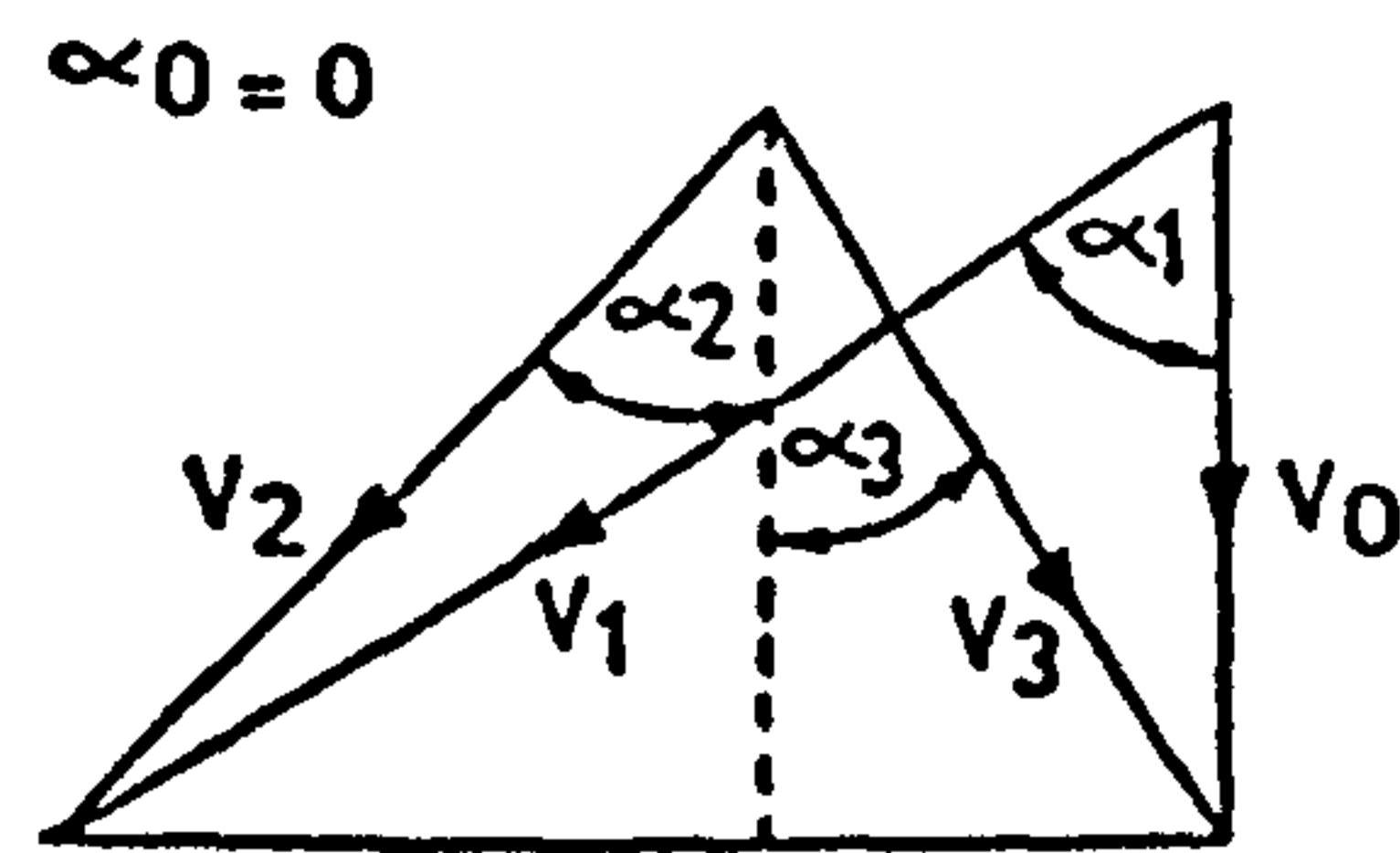
Axial compressor design begins with a parametric study at a reference radius – usually the mid-radius, as in this investigation. To meet a specific duty (defined by mass flow rate and pressure ratio) requires selection of suitable work and flow coefficients for best stage conditions as governed by the physical dimensions, number of stages and rotational speed of the compressor. These determine the choice of what is termed the “stage degree of reaction”. That is the ratio of the static temperature rise of the rotor to the total temperature rise of the stage, which indicates what proportion of the static pressure rise of the stage is consumed by the rotor. Figure 6.6 shows the velocity triangles for four different degrees of reaction for different work and flow coefficients. 50% reaction is identified by symmetrical velocity triangles and this type of design minimises the adverse static pressure rise gradient on either of the rotor or stator blade surfaces.

The outcomes of a reaction cannot be fully assessed using simple 2-D analysis with cascade data. (Wall, 1976) considered secondary flow effects and concluded that stator cascade performance could not accurately predict rotor cascade performance. Stators are fixed and directly affected by the end wall boundary layers. There is an inherent static pressure gradient in stators which acts to push secondary flows on the outer radially inwards. This will act to feed flow into the inner sections of the blades which are already more loaded than the outer ones and so have the effect of thickening the boundary layers on the free-stream blade surfaces. The opposite will be true for the rotors since a low axial velocity acquires a higher whirl velocity at the rotor exit than the main stream. With rotors the secondary flow will tend to move outwards, reducing the load on the inner diameter and free-stream blade surface boundary layers and thickening the boundary layer on the outer diameter.

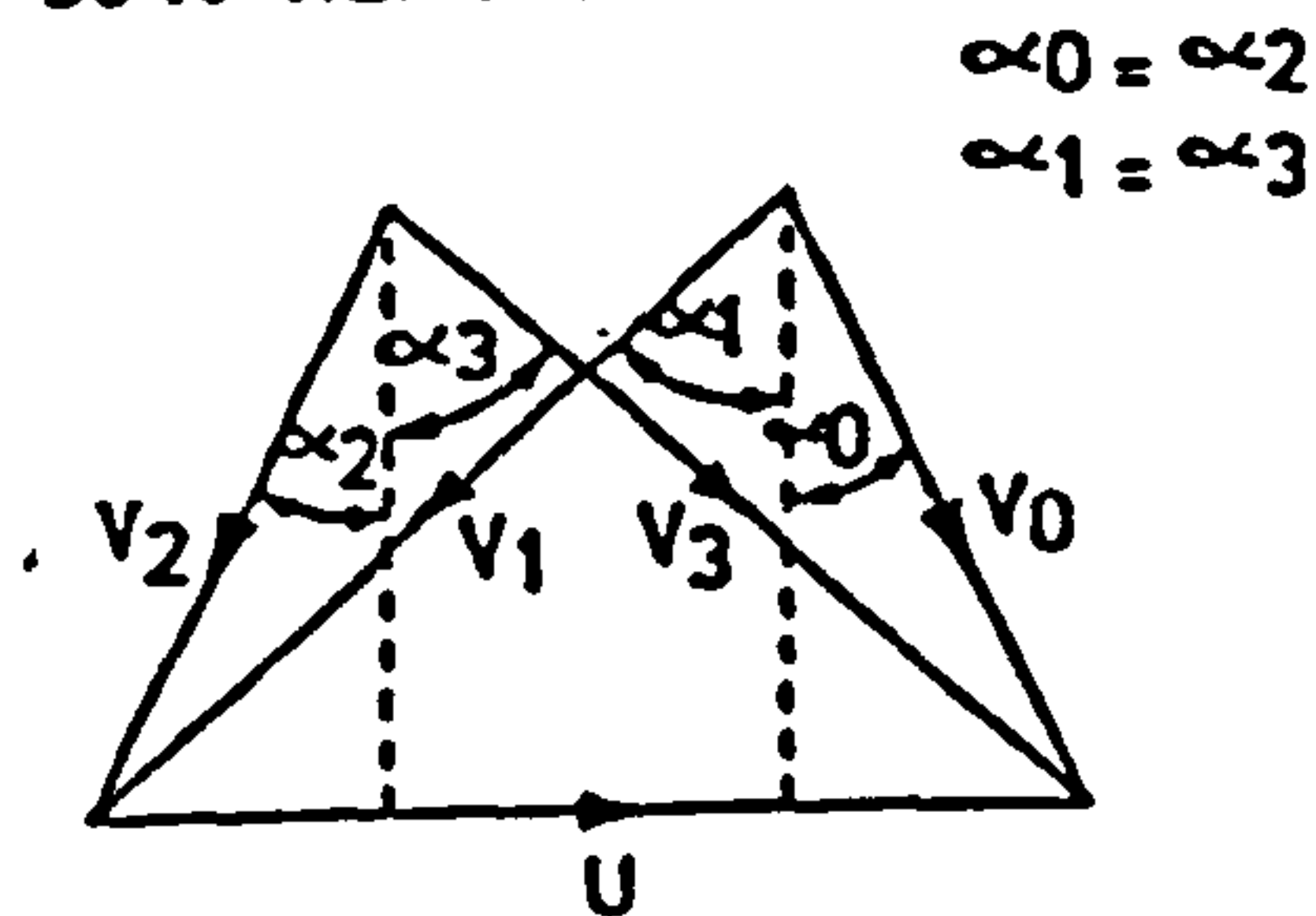
a) 0% REACTION



c) FREE VORTEX ($\sim 80\%$ REACTION)



b) 50% REACTION



d) 100% REACTION

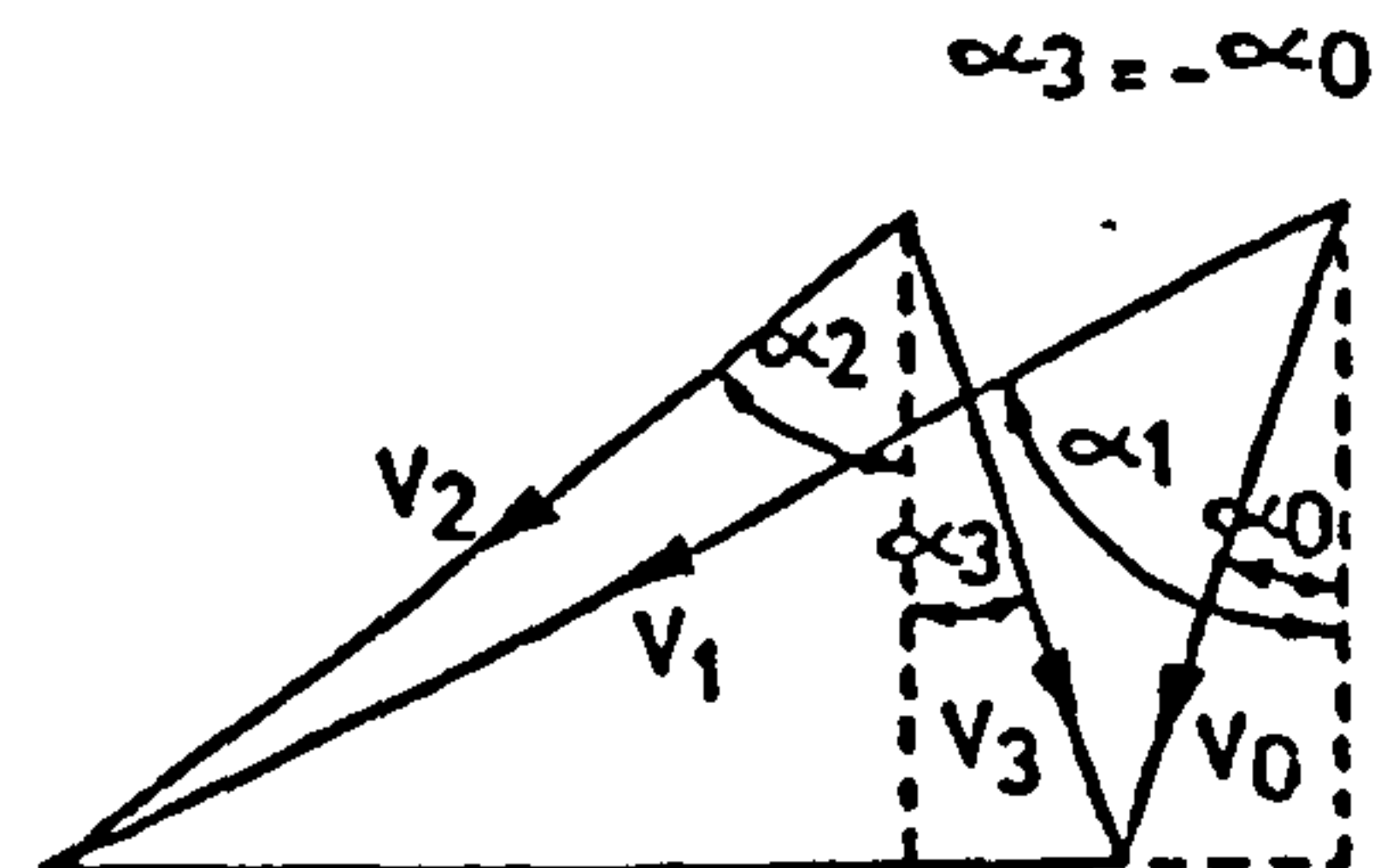


Figure 4-6: Velocity triangles for different reactions (Leiblein, 1960)

To maximise efficiency industrial compressors which run at a constant speed should have variable stator blades to match the different mass flow rates and give the design pressure rise. Such considerations are outside the scope of this study.

4.2 Computational Fluid Dynamics

Computational fluid dynamics (CFD) is applicable to a wide range of problems in the field of fluid flow. Progress in CFD simulations has been possible because of the rapid increase in computer power and speed and the advances made in techniques for the solution of ordinary and partial differential equations. However, despite these advances the success of CFD is very much dependent on the physical models applied which can only be found from a comprehensive understanding of the relevant physical phenomena (cfd-online).

Whenever turbulence is present in a fluid flow it appears to dominate all other flow phenomena. Successful modelling of turbulence is a pre-requisite for high quality numerical solutions. Unfortunately no generally applicable turbulence model exists and there still exist

many open questions on how turbulence should be modelled. Nevertheless, the cost of our ignorance is immense, which had made CFD modelling of e.g. turbine compressors, economically extremely attractive and in the last decade CFD has become a major tool in engineering. Today CFD seems capable of solving industrial problems at moderate cost and turnaround times. Its increased use will depend on how accurate complex turbulent flows can be modelled. However, turbulent flows contain small fluctuations. The resolution of which requires fine grids and small time steps, such that a direct simulation becomes unfeasible for high Reynolds numbers.

4.3 Numerical Modelling Approach

This chapter describes the various steps taken to model an axial-flow compressor for which a number of flow measurements have been performed and are available in the open literature. This numerical study seeks to offer a validation of the numerical code, and of the user of the software, by comparing the numerical predictions with the experimental data. The results of the numerical simulation were examined and compared with the available experimental data which include both integrated quantities, pressure and temperature ratios and adiabatic efficiency, and radial distributions, Mach number, total temperature, total pressure and yaw angle. The various procedures and details of the calculations used to investigate this test case are outlined in this chapter.

4.2.1 ANSYS CFX Description

ANSYS CFX is the commercial CFD code used in the present work. Although the code is capable of solving diverse and complex multi-dimensional fluid flow problems, its historical employment has mostly been in the solution of turbomachinery problems. The fluid solver provides solutions for compressible or incompressible, steady-state or transient, laminar or turbulent single-phase fluid flow in complex geometries. The software uses block-structured non-orthogonal grids with grid embedding and grid attaching to discretize the domain. The software system has additional capabilities which can predict subsonic, transonic and supersonic compressible flows including temperature solutions in solid regions of the domain, for laminar or turbulent flow.

ANSYS Turbo Grid

ANSYS Turbogrid is a powerful tool that lets the designers and analysts of rotating machinery create high quality hexahedral meshes, while preserving the underlying geometry. These meshes are used in the ANSYS work flow to solve complex blade passage problems. It is designed for the creation of periodic blade geometry meshes with a minimum of user intervention, where:

- A number of pre-defined grid topology templates are available, each well suited for a certain class of turbo-machines
- Periodic boundaries are managed automatically by ensuring both physical and topological boundaries.
- Grid attachment between sub-grids or multi-block domain and between corresponding periodic boundaries is automatically performed during the mesh creation.

4.2.2 HP9 Compressor Stage modelling

HP9 is a single or two-stage rig representing of the later stages of a civil HP compressor. A suitable inlet velocity profile is generated by spoiler rings on the hub and casing upstream of the rotor, but no attempt is made to reproduce representative inlet turbulence. The rig has been used for evaluating a wide variety of blading. The RD case is a single, close-coupled stage, with moderate loading and high reaction. It has conventional blade profile with moderate values of space/chord ratio. The blading was designed assuming a flat velocity profile at rotor inlet and does not have any end-bends.

The rig has provision for radial and area traverses between rows in addition to conventional inlet fixed instrumentation. However, in common with many other compressor tests, the rotor exit data are subject to considerable uncertainty and should be interpreted accordingly. No laser anemometry or turbulence data are available.

The main parameters are:

Pressure ratio = 1.24

Tip speed = 251.3 m/s (288K inlet)

Design speed (100%) = 9262.5 rev/min

Outer diameter (constant) = 518.2 mm at 288K inlet Temperature

Hub/tip ratio (inlet) = 0.84

Space/chord ratio

(mid height) = 0.055 rotor, 0.7 stator

Stage inlet/ exit swirl = 0

Rotor inlet EWBL thickness = 10-20% annulus height

Reynolds No (mid-height) = $\sim 0.7 \times 10^6$ rotor
= $\sim 0.3 \times 10^6$ stator

Rotor blade number = 41

Stator blade number = 73

Where; EWBL = end wall boundary layer

A general aspect of the facility containing the HP9 rig is shown in Figure 5-7

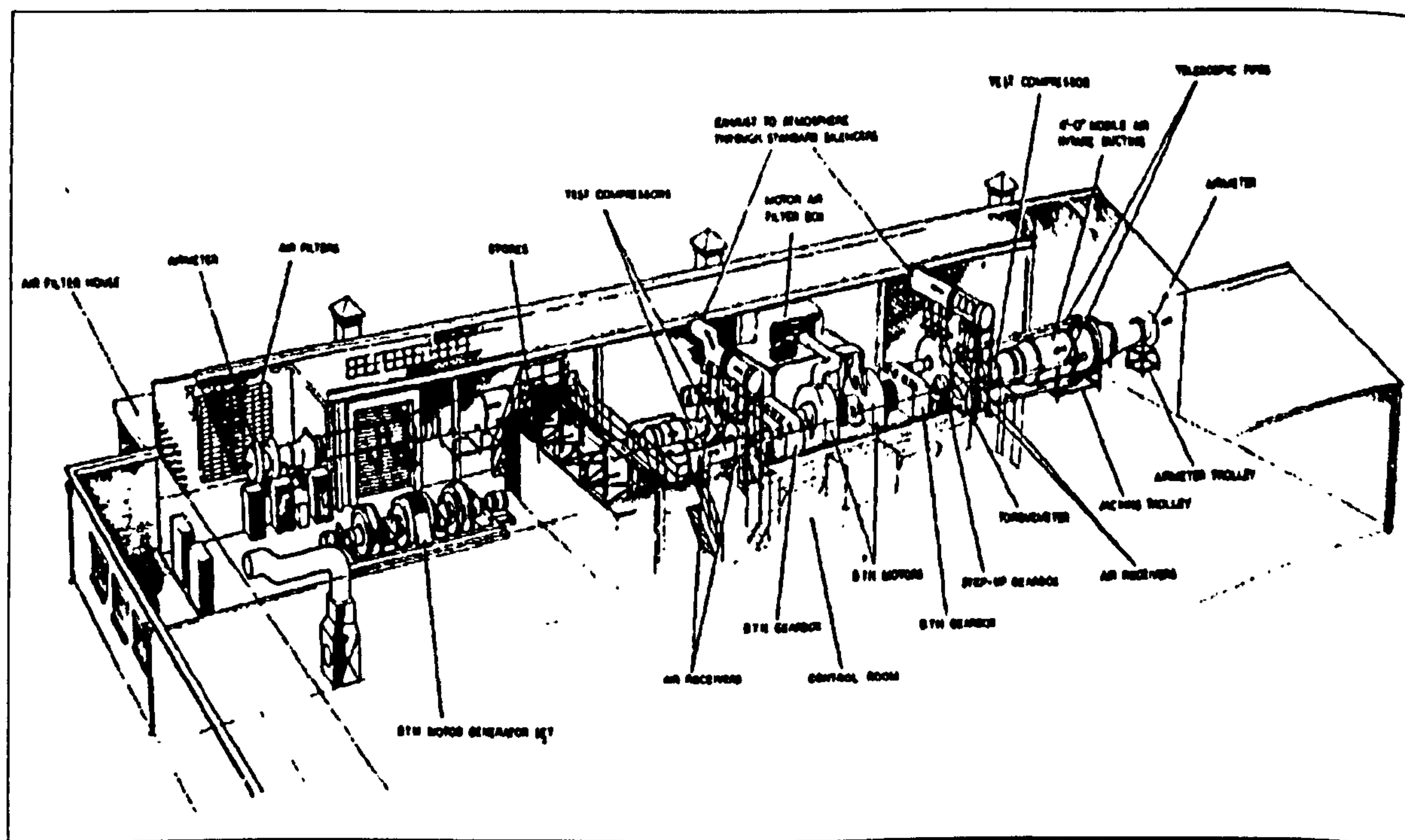


Figure 5-7: Rolls-Royce Derby no.3 compressor test facility

The reported experimental data cover three flow conditions denoted as max flow, mid-chic and near surge. The mass flows associated with these points are, 9.70, 8.82 and 7.80 kg/s respectively.

4.3 Mesh generation

The ANSYS TurboGrid package was used to generate the grid for the blade domain employing a double-block grid template shown in figure 57 which is designed to produce a high quality mesh for low-blade stager using H/J/C/L-type grid blocks to mesh the blade and the passage with grid elements inside the blade being blocked-off in order to form the blade surface. The files required by the CFX-TurboGrid contain data points in Cartesian coordinates for the hub, shroud, and the 9 blade profile curves in between. Both hub and shroud curves are extended upstream of the blade leading edge and downstream of the blade trailing edge. The calculations of inlet and outlet conditions are taken at stations similar to those where the measurements were taken in the experimental rig.

The computational domain is shown in figure 5-8. The rotor blade has a fixed tip clearance of 0.5mm and the mesh consists of about 500,000 nodes per passage. The grid was checked for minimum and maximum skew angle, aspect ratio and negative volume, and necessary adjustments were made to satisfy the conditions required by the CFX-Solver Manager. The stage computational domain is presented in figure 5-8. This computational domain was used to perform initial steady state simulation of the HP9 stage. A mixing plane is used for the interface between the rotor and stator domains.

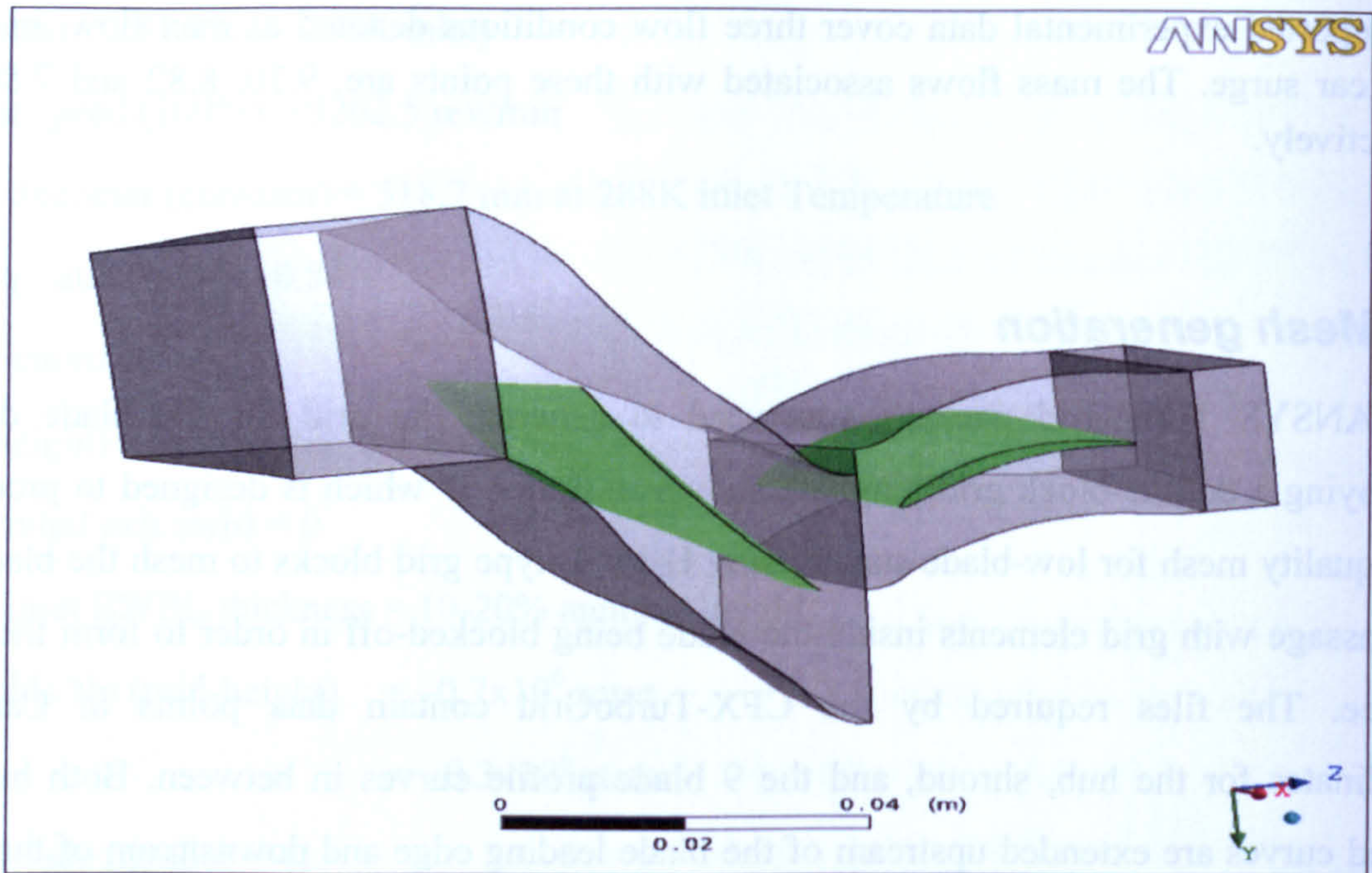


Figure 4-8: R-R HP9 computational domain

4.4 Results and discussion

The initial set of computations refers to the max flow regime where the machine passes 9.70 Kg/s. These simulations correspond to an approximate solution since they do not include the inlet profile which represents the thick endwall boundary layers which are generated in the rig by the spoiler rings on the hub and casing upstream of the rotor. Figure 4-9 describes a blade-to-blade contour plot of velocity, relative in the rotor and absolute in the stator, taken at mid-span. The air flow enters the blade passage with a velocity of about 230 m/s but on the rotor blade suction surface the velocity reaches a value of some 300 m/s. The wakes in both rotor and stator are fairly thin and the peak of the velocity distribution in the rotor suction side is located relatively downstream chordwise. These are indicative of flow at or near the design point. Nevertheless the stagnation point in the rotor is seen to be tilted towards the suction surface of the blade and this, and the local acceleration seen in the pressure surface, corresponds to a negative incidence condition. This situation usually corresponds to a high flow regime where the axial velocity, and hence the mass passed, is higher than design.

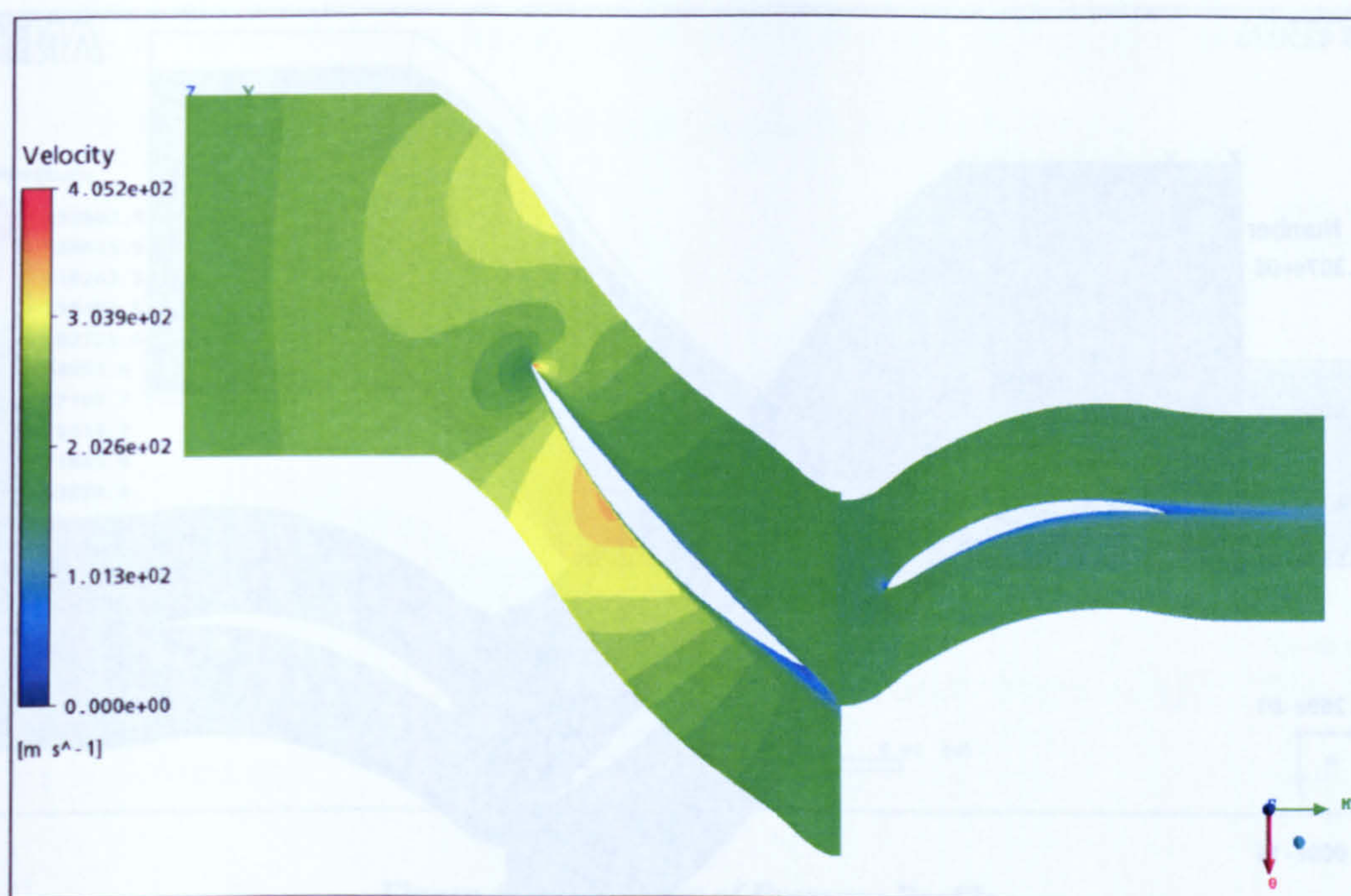


Figure 4-9: Contours of Velocity Magnitude

The high velocities of the test case, seen in the previous figure are associated with sizeable Mach number values. In Figure 4-10 there is evidence of a small patch of transonic flow near the leading edge and a more extensive region in the suction surface.

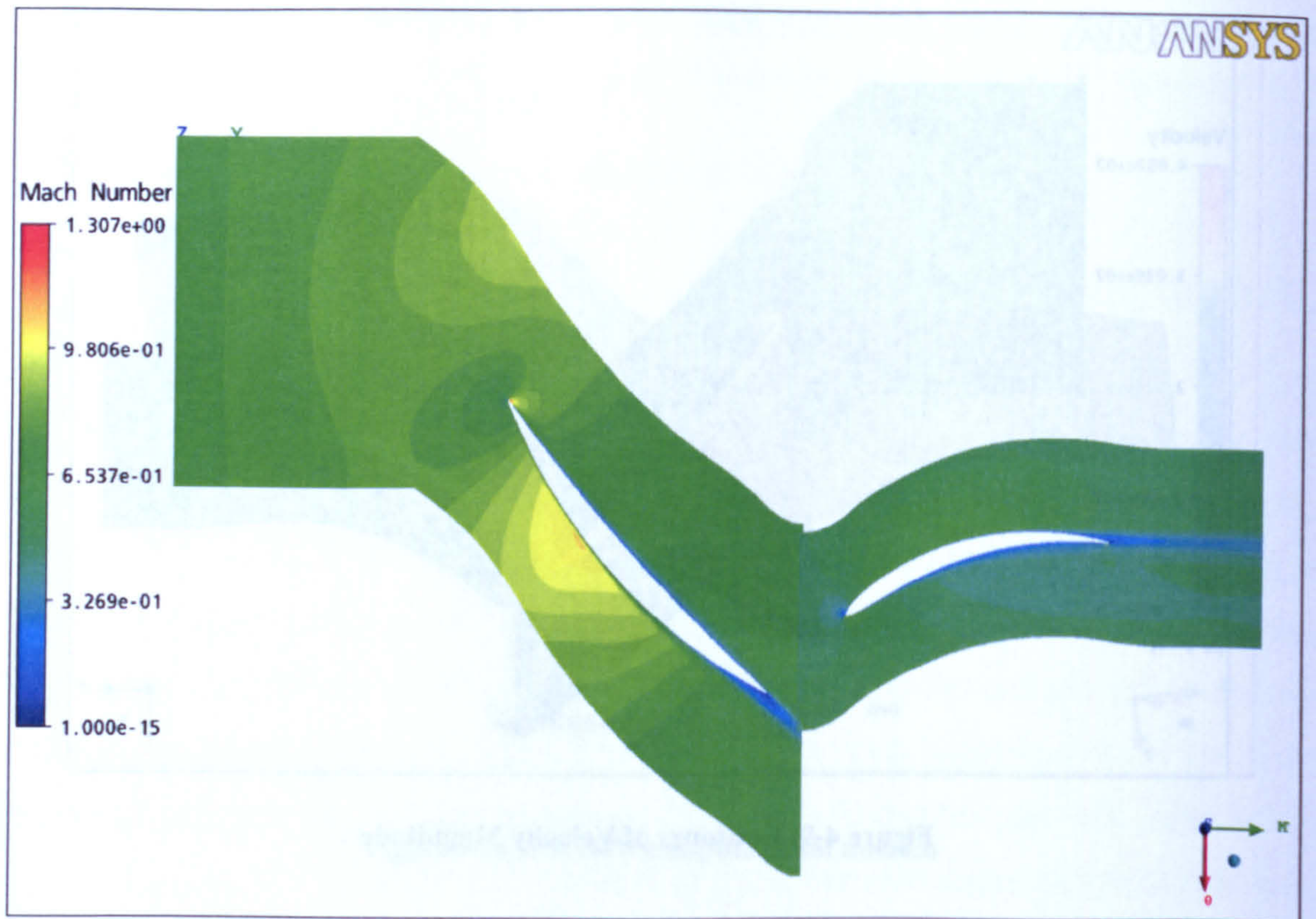


Figure 4-10: Mach number Profile

The static pressure at the entrance of the rotor passage is approximately 81000 Pa and at the exit it is close to 97000 Pa, corresponding to a pressure rise of 16000 Pa. The mid-span pressure field is shown in figure 4-11. An additional pressure increase of lower value (13000 Pa) takes place along the stator. Therefore 55% of the stage pressure increase is produced by the rotor and 45% by the stator.

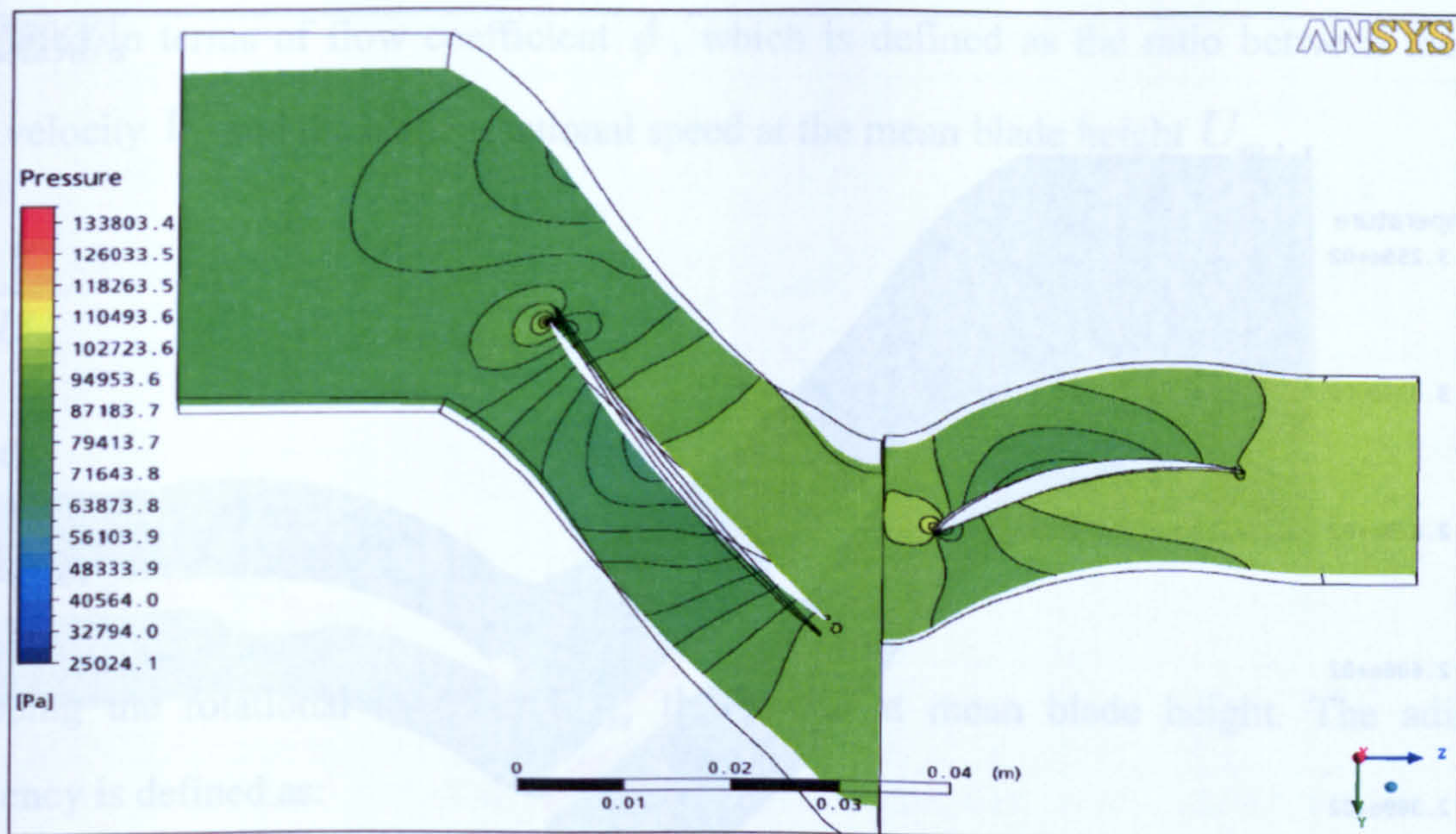


Figure 4-11: Contour of Pressure Profile

Examining the static temperature distribution along the stage passages, Figure 4-12, it is possible to observe temperature variations that are due to the general compression effects and those that occur as function of local flow features. An illustration of the first point is the gradual downstream increase in temperature that can be seen in the stage. However there are local variations, such as the temperature peak associated with the leading edge and with the turbulent wake.

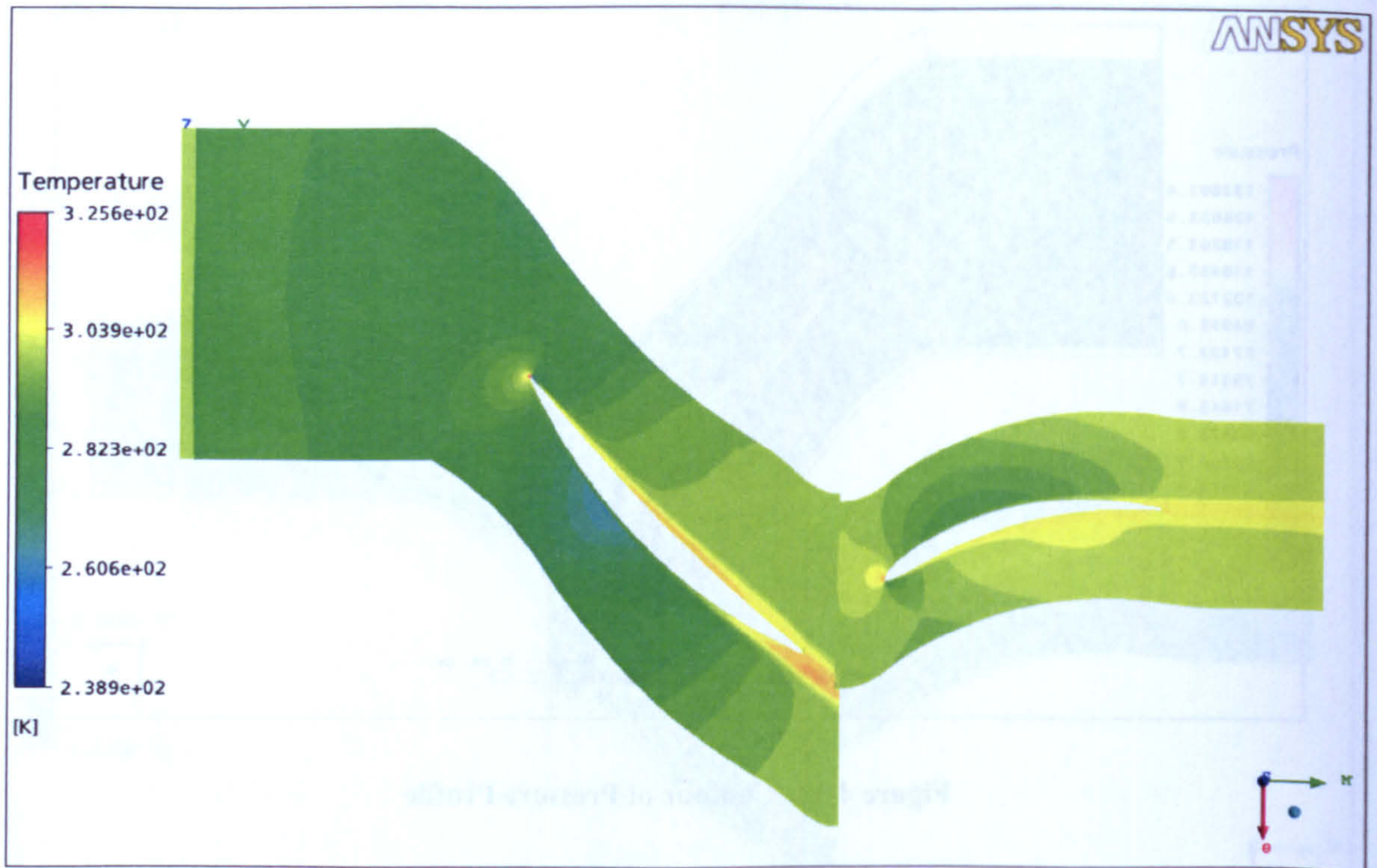


Figure 4-12: Contour of Temperature Profile

4.5 Comparison of CFD results with experiments

The performance of axial flow compressor is usually represented through the pressure flow characteristics and efficiency of the machine. Total and static pressure rise across the stage, mass flow, rotational speed and work input must be computed to carry out a performance investigation. This section will discuss the general techniques used in the present work to calculate some of the parameters mentioned above and compare with the HP9 experimental data. The source is AGARD AR 275, of 1990, by Ginder and Harris.

4.5.1 HP9 compressor overall performance

The flow results employed in the analysis of the HP9 compressor were computed by area averaging the typical flow quantities on planes located at rotor inlet and stator exit. The location of these planes coincides with the data collection planes described in source mentioned above. The pressure ratio, adiabatic efficiency η and temperature rise ratio are

calculated in terms of flow coefficient ϕ , which is defined as the ratio between the axial flow velocity V_a and the blade rotational speed at the mean blade height U_m ,

$$\phi = \frac{V_a}{U_m}$$

where

$$U_m = \Omega r_m$$

Ω being the rotational speed and r_m the radius at mean blade height. The adiabatic efficiency is defined as:

$$\eta = \frac{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{T_2}{T_1} - 1\right)}$$

where P_1 is the total pressure upstream the rotor, P_2 is the total pressure downstream the stator, T_1 is the total temperature upstream the rotor and T_2 is the total temperature downstream of the stator.

The overall predicted performance of the HP9 compressor using ANSYS CFX solver, in the form of efficiency, pressure ratio and total temperature rise ratio is presented and compared to the experimental data at max and mid-chic flow conditions as shown in Table 4-1. It can be seen that the numerical simulation results are in good agreement with the experimental data.

Table 17: Overall performance comparison

Flow Condition		PR	η	TTR
Max flow	CFD results	1.175	0.871	0.0543
Max flow	Experimental data	1.196	0.857	0.0612
Mid-chic	CFD results	1.228	0.876	0.0682
Mid-chic	Experimental data	1.236	0.883	0.0707
Near Surge	CFD results	1.210	83.2	0.0711
Near Surge	Experimental data	1.267	85.1	0.0822

4.5.2 Travers Results Validation

The radial profiles of the total pressure P_t , total temperature T_t , Mach number and yaw angle at the rotor outlet station are calculated and compared to the experimental data. These quantities were area-averaged along the pitch for each radial position from the hub to shroud and are presented with the equivalent experimental data in Figures 4-13 to 4-16. Figure 4-13 describes the variation of total pressure downstream of the rotor. The degree of agreement is not very good and this is thought to be related to the use of uniform values at inlet rather than the experimental profile. Nevertheless in absolute terms the discrepancy is not as significant as the scale of the graph may suggest.

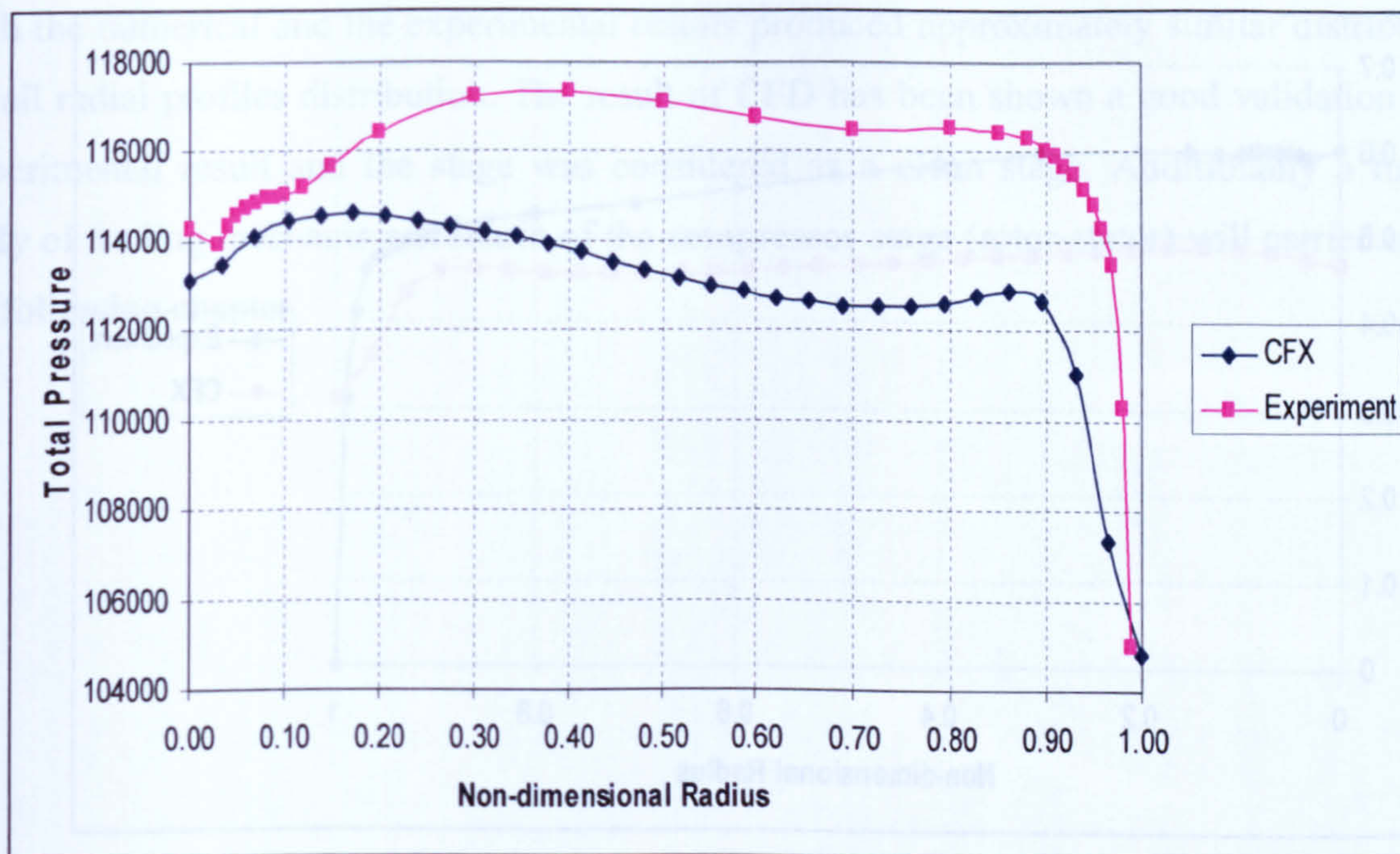


Figure 4-13: Rotor exit radial pressure traverse

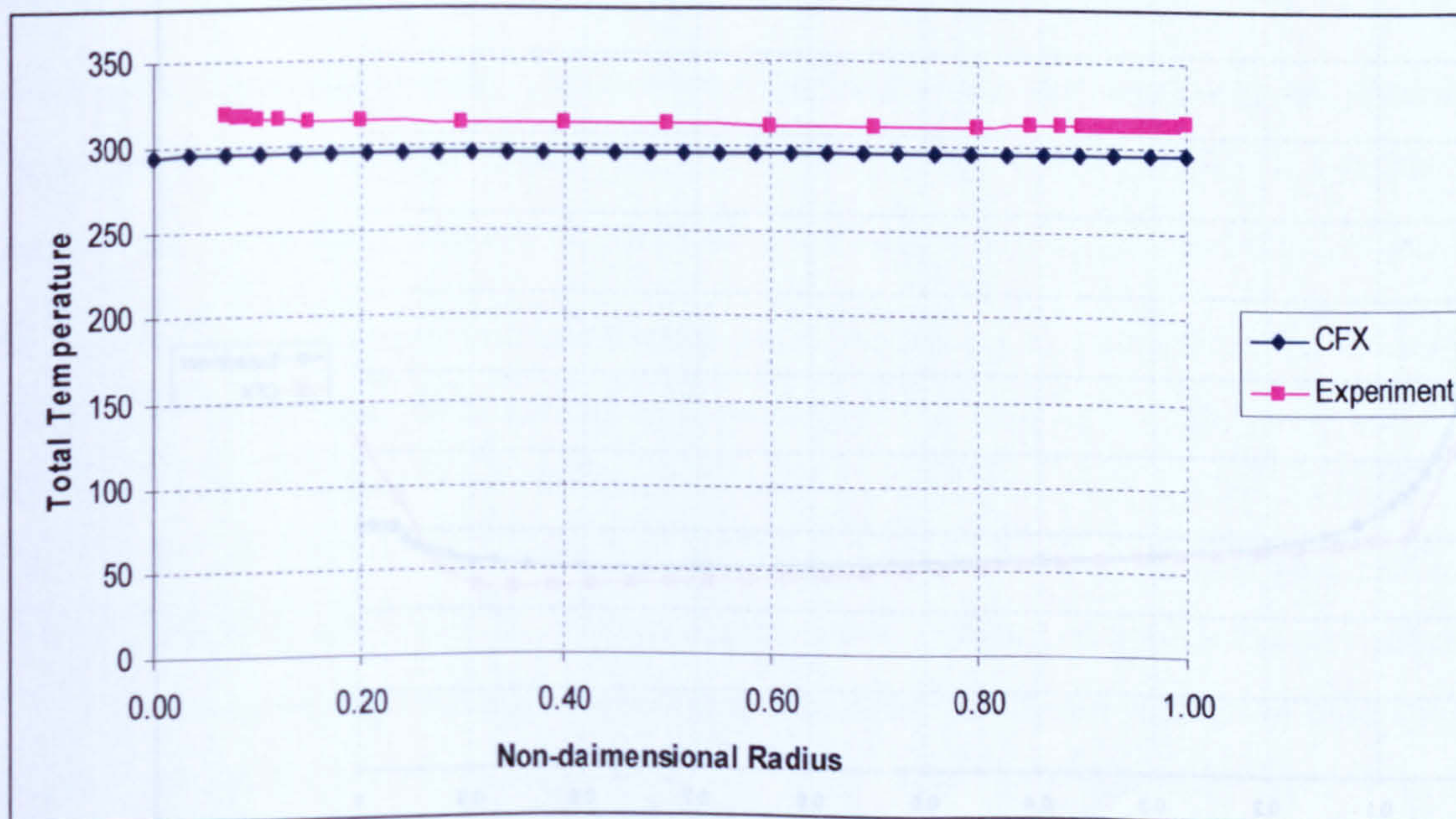


Figure 4-1: Rotor exit radial temperature traverse

Both the total temperature, Figure 4-14, and yaw angle, Figure 4-16, show good agreement between the experimental and computational values. The Mach number, Figure 4-15, demonstrates an agreement between the experimental and computational values.

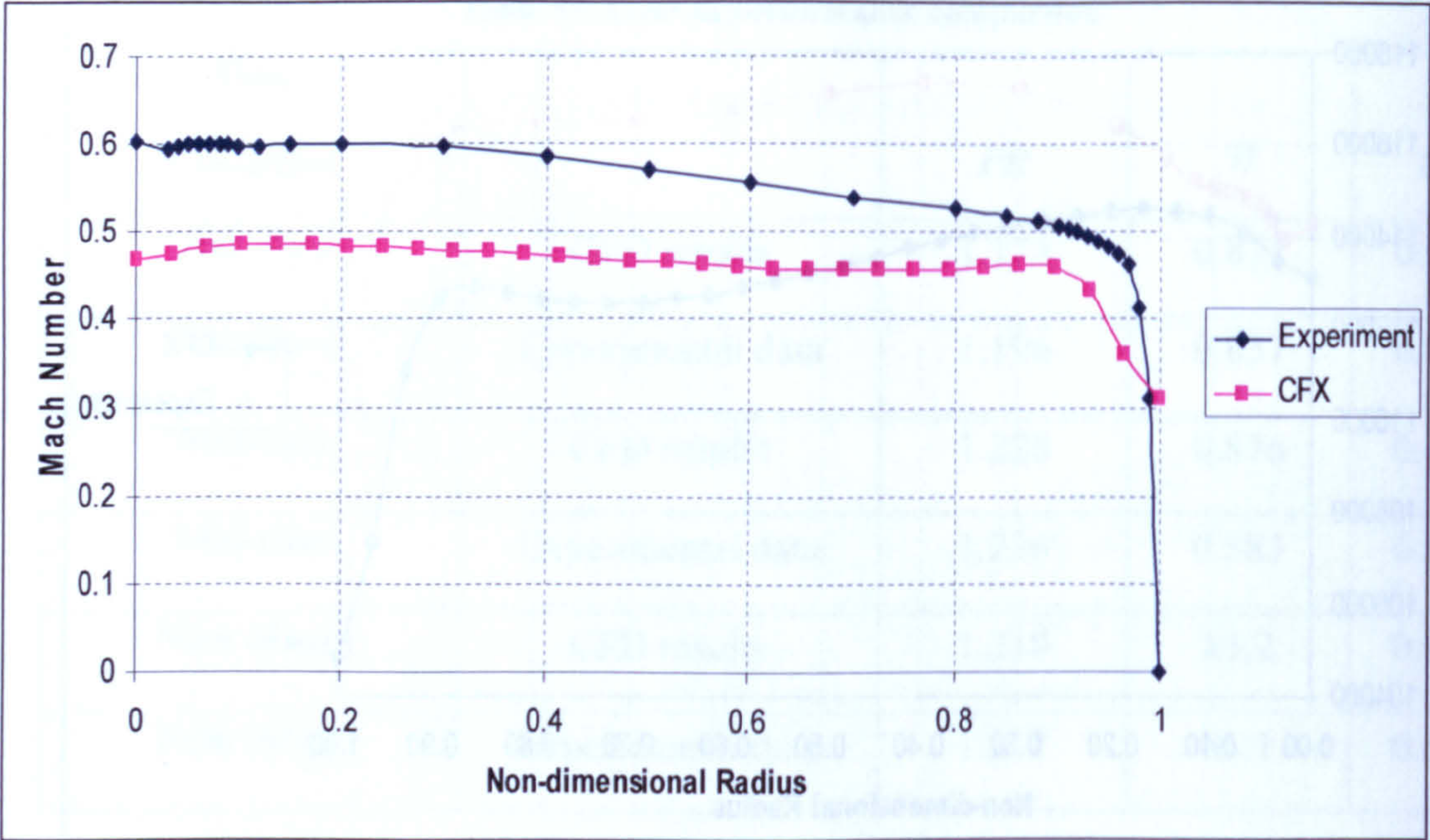


Figure 4-2: Rotor exit radial Mach number traverse

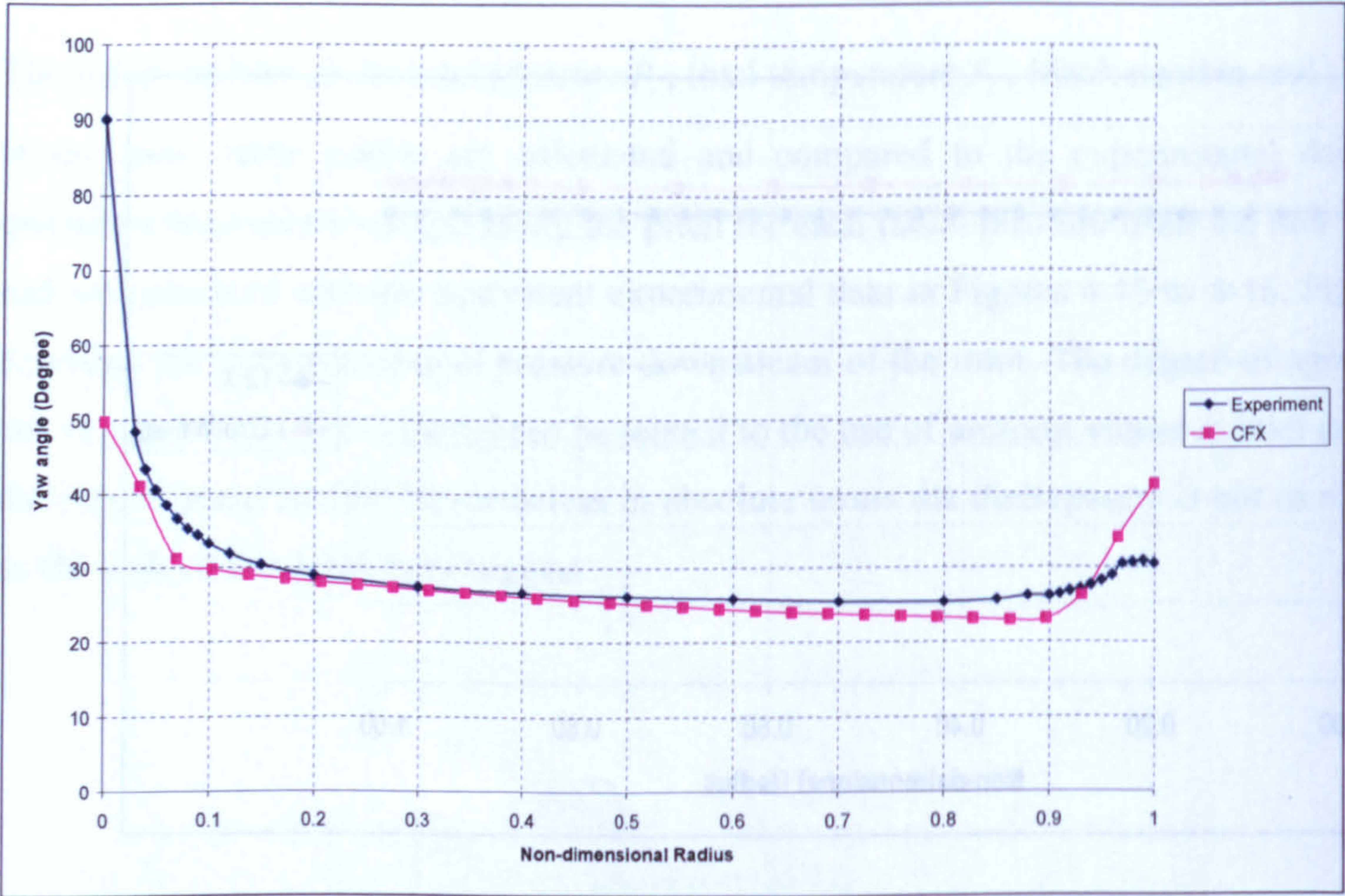


Figure 4-3: Rotor exit radial Yaw angle traverse

Both the numerical and the experimental results produced approximately similar distribution for all radial profiles distribution. The result of CFD has been shown a good validation with Experimental result and the stage was considered as a clean stage. Additionally a further study of fouling mechanism scenario of the compressor stage (rotor-stator) will carried out in the following chapter.

Chapter5 – CFD Modelling of Clean and Fouled Axial Compressor Stage

5.1 Introduction

In the present work, the software package ANSYS-TurboGrid was used for grid generation while the CFD simulations mainly employed the code ANSYS-CFX. However some limited use was also made of CFX-TASCflow. Within the ANSYS-CFX package the pre-processor CFX-Pre was used for preparing the computational model for solution, including the setting of appropriate boundary conditions. The solution of the Navier-Stokes equations was performed with CFX-Solver. The post-processing of the results was carried out with CFX-Post including the use of the dedicated turbomachinery module, TURBO-POST.

Also used in this project was the commercial code CFX-TASCflow (originally developed by AEA Engineering Ltd). TASCflow was employed to divide the surface of the blades, of both the rotor and stator, into different regions so that different roughness levels could be applied to different zones of the blades to investigate fouling of the compressor stage.

A number of parameters (e.g., aspect ratio, maximum and minimum angles) can be adjusted when generating the grid to determine the quality and density of the mesh. Figure 5-1 shows a typical grid used in the present simulations for a single blade. Hexagonal meshes of three densities, denoted as fine, medium and coarse, with 185,133, 98,592 and 47,125 nodes respectively per passage, were employed. In all cases the grid density was carefully adjusted to optimize node distribution in regions of high flow gradients.

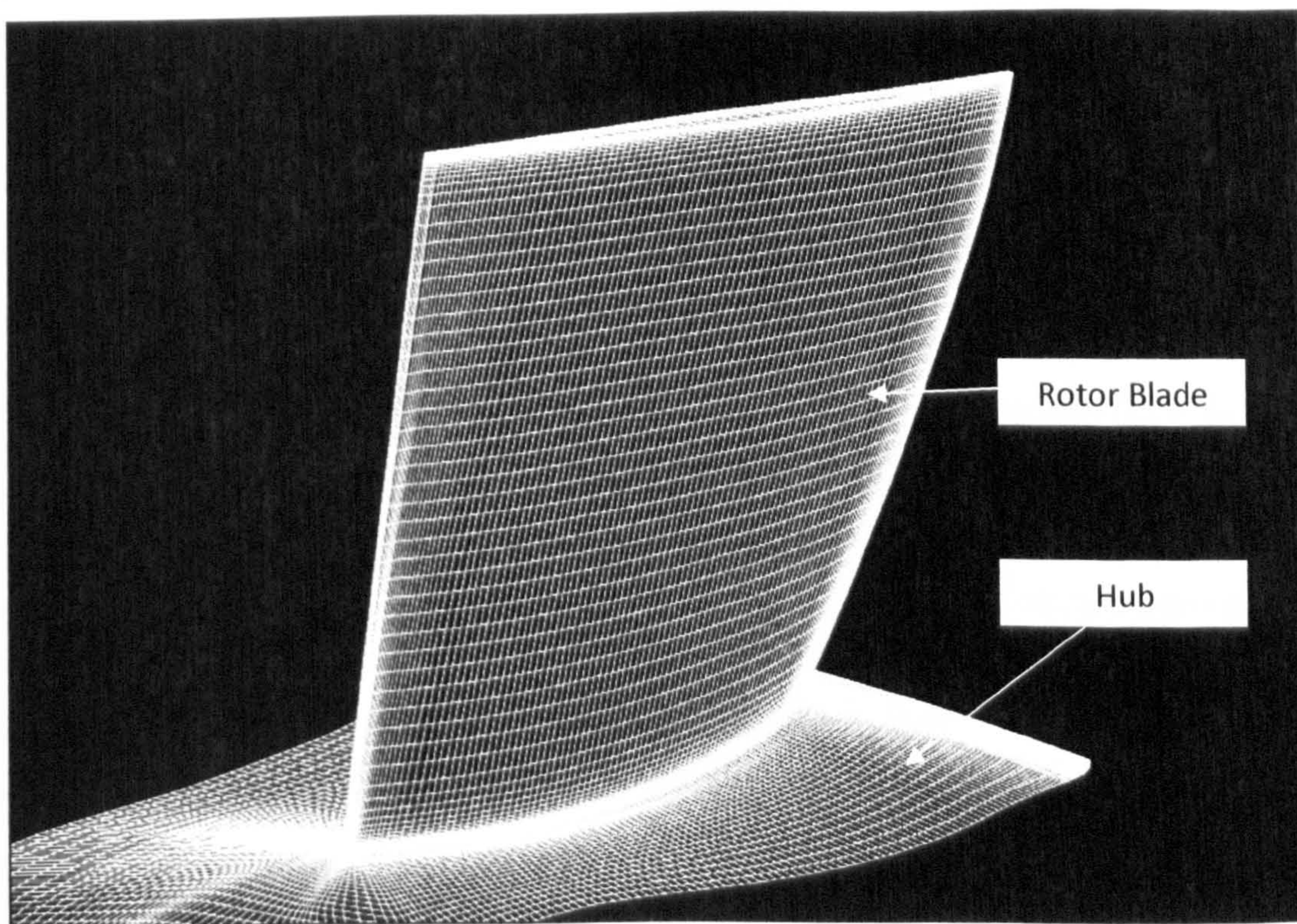


Figure 5-1: Typical Grid on Rotor Blade.

Flow in an axial compressor with rows of stationary and moving blades experiences rotor-stator interactions which cause periodic fluctuations of properties. This rotor-stator interaction necessitates the requirement of a dedicated numerical modelling approach. CFX offers a particularly efficient simulation of systems with both stationary and rotating components. It has the capability of performing steady-state analyses using local reference frames which may be stationary or rotating, as appropriate. A control surface approach and a general intersection algorithm are designed and used in such a way as to provide for maximum robustness and accuracy.

The program was used to analyze several blade rows in combination using a mixing plane formulation, known in the code as the 'Stage Interface' approach. The 'Stage Interface' allows adjacent blade rows in different rotational frames of reference (i.e., a rotor-stator) to be coupled together by circumferentially averaging pressure and velocity but maintaining spanwise gradients. This interface is especially useful because it enables multiple blade rows to be modelled using just a single blade in each row. Energy, mass and momentum are fully conserved across the interface, even for blade rows with significantly different pitch ratios.

ANSYS-CFX is a general purpose commercial CFD package for rotating machinery applications that solves the Navier-Stokes equation using a conservative finite volume and a pressure-based coupled solver. This code and its predecessor, TASCflow from which it acquired many features, have been successfully used to predict turbo-machinery performance for nearly two decades.

In this project, ANSYS-CFX was used to solve the continuity, momentum and turbulence equations for fluid flow in three and a half stages of an axial compressor. The blade surfaces, hub and shroud were modelled as walls with a no-slip boundary condition being imposed. Air at 288K entry temperature was used as the operating fluid for steady-state dry case conditions.

5.2 Turbulence modelling

Since the current problem is associated with turbulent conditions, the exact Navier-Stokes equations are recast in a Reynolds-averaged format. These equations contain the Reynolds stresses, which generate the need for a turbulence closure model in order to obtain a solution. Initially for the dry case, the standard k- ϵ turbulence model was employed. This model solves two additional transport equations, one for the turbulent kinetic energy and the other accounting for the dissipation of this energy.

The standard k- ϵ model is used in most turbulent flow prediction calculations because of its economy, robustness and reasonable accuracy for a wide range of flows. However, the model does not perform well with non-equilibrium boundary layers, it has a tendency to under-predict the amount of separation and to predict the onset of separation too late. Because separation has such a strong influence on the performance of turbine blades and aerodynamic bodies it also strongly influences wall heat transfer and multi-phase phenomena. Under prediction of separation will usually result in optimistic prediction of machine performance which could be dangerous, as with, for example, the prediction of wing stall on airplanes. A number of different strategies were attempted to improve convergence with this model, including the lowering of under-relaxation factors. However, none improved the level of convergence and therefore further simulations using this turbulence model were abandoned. For these reasons both the k- ω and the standard k- ϵ turbulence model were used for the

simulations of the clean compressor stage. For the roughness cases examined the $k-\epsilon$ model and the BSL model were selected.

5.3 CFD modelling

The CFD code (CFX) is employed to obtain numerical predictions of the axial compressor single stage. The simulations are steady state and three-dimensional. The fluid volume of the stage has been meshed with structured grids created by the mesh generator Turbo grid.

The $k-\omega$ and the standard $k-\epsilon$ turbulence model were used for the simulations of the clean compressor stage. For the roughness cases examined the $k-\epsilon$ model and the BSL model were selected. The passage grid is shown in Figures 5-2, A and 5-3. It consists of an H-grid with an O-grid fitted to the regions of the blade that contain the blades' boundary layer. The O-grid contains 100 nodes around the blade and 50 nodes spanwise.

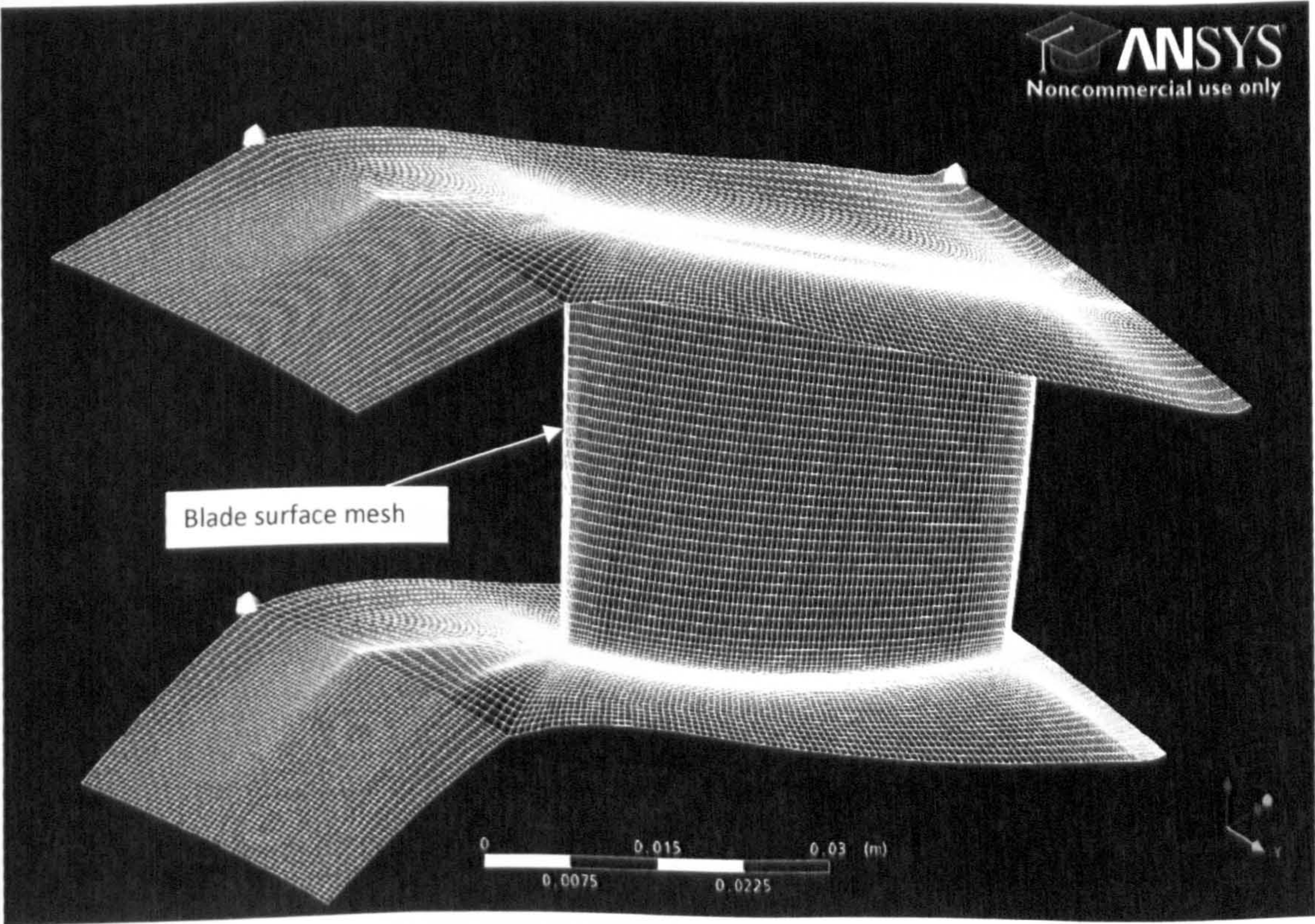


Figure 5-2: Compressor Rotor blade mesh and its hub and shroud

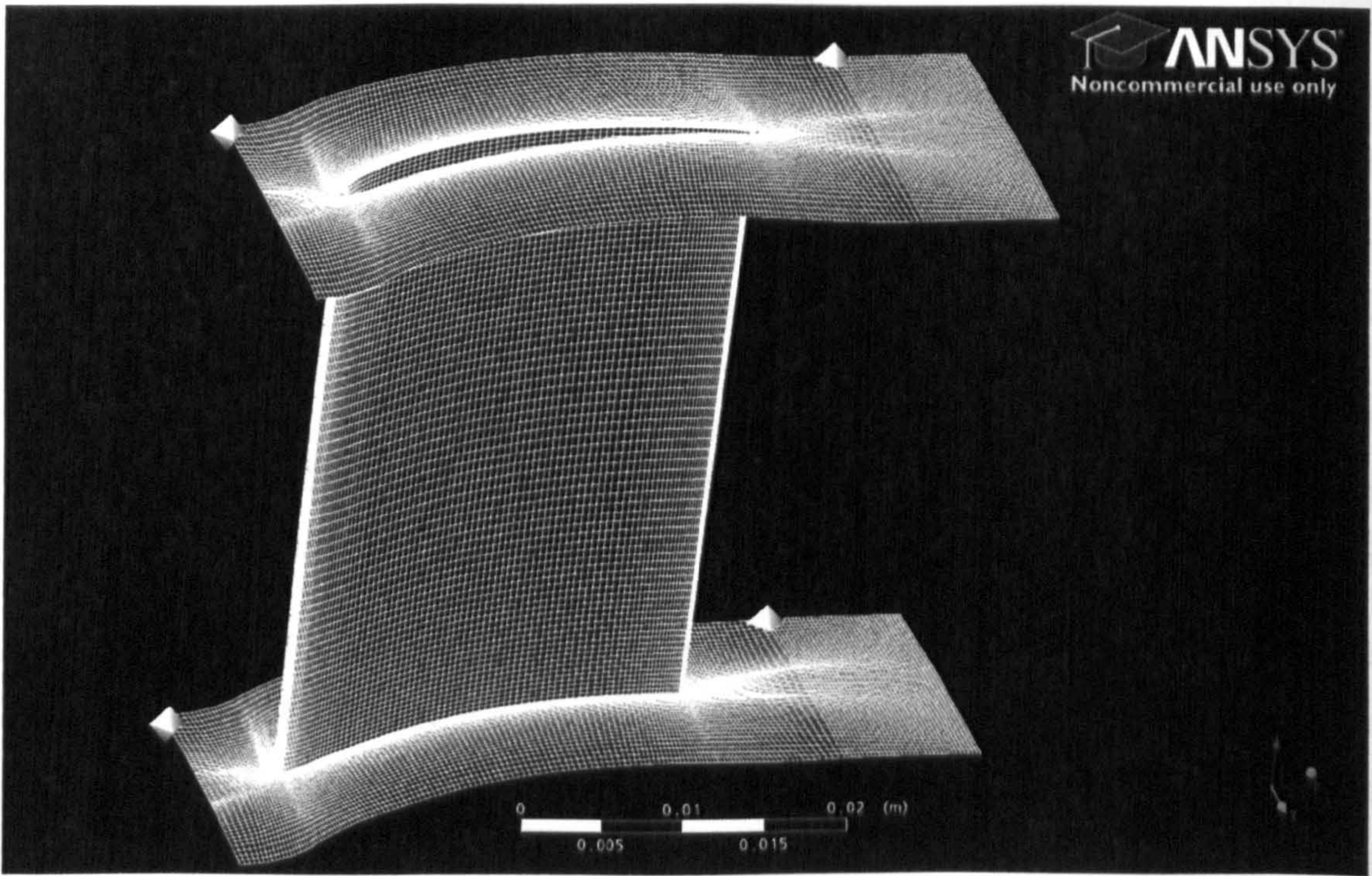


Figure 5-3: Compressor stator blade mesh

For the smooth blades the distribution of points in the near wall region is denser than for the roughened cases in order to resolve the boundary layer in the viscous sub-layer. An important parameter when simulating wall bounded viscous flows is y^+ the dimensionless distance from the wall of the first mesh node in the flow away from the wall. The y^+ provides information on the near wall resolution of the mesh. The y^+ values, equation (5-1), along the blade surfaces are below 5. For this case the enhanced wall treatment is employed.

$$y^+ = \frac{\rho u_\tau y_p}{\mu} \dots\dots\dots (5-1)$$

For all the roughened blade cases standard wall functions are used. According to the code manual, the wall roughness effects can be included through the law-of-the-wall modified for roughness as follows:

$$\frac{u_p u^*}{\tau_w / \rho} = \frac{1}{\kappa} \ln \left(E \frac{\rho u^* y_p}{\mu} \right) - \Delta B \dots\dots\dots (5-2)$$

$$\text{where } u^* = C_\mu^{1/4} k^{1/2} \dots\dots\dots (5-3)$$

$$\text{and } \Delta B = \frac{1}{\kappa} \ln f_r, \dots\dots\dots (5-4)$$

For a sand-grain roughness and similar types of uniform roughness elements ΔB has been found to be well-correlated with the non-dimensional roughness height K_s^+ .

$$K_s^+ = \rho K u^* / \mu \dots\dots\dots (5-5)$$

According to the CFX manual, for a uniform sand-grain roughness, the parameter K which is the physical roughness height can reach the value of the height of the sand-grain.

For a non-uniform sand-grain the roughness height can be equal to the grain mean diameter. For the present CFD analysis the fouling grains were assumed to be distributed uniformly on the different regions of the blade surfaces and their mean diameter (size) was used as the roughness height input for the simulation.

The simulation was run with the following boundary conditions: total pressure inlet (3 different cases), 97200 Pa for near surge flow condition, 96500 Pa mid-chic, 96200 Pa for the max flow condition and total temperature 288 K. The turbulence kinetic energy and the dissipation rate were set at default values.

The stage outlet was set as a pressure outlet boundary. Considering the three different inlet conditions, three different values of outlet static pressures were used which are 114500 Pa for near surge flow, 105500 Pa for mid-chic flow and 99000 Pa.

The pre-processing tasks, such as the setting of the inlet and outlet boundary conditions, pressure, temperature and velocity profiles, were performed with the dedicated module of the code, CFX-Pre. Figure 5-4 shows the rotor-stator domain modelled in CFX-Pre in order to apply different boundary conditions. The rotor blade suction surface has been divided into 7 patches span-wise and 3 regions stream-wise and the stator divided into 5 patches span-wise and 3 patches stream-wise. In these patches different levels of roughness on different locations of both rotor and stator can be specified. These patches were initially specified in CFX TASCflow as shown in figure 5-4.

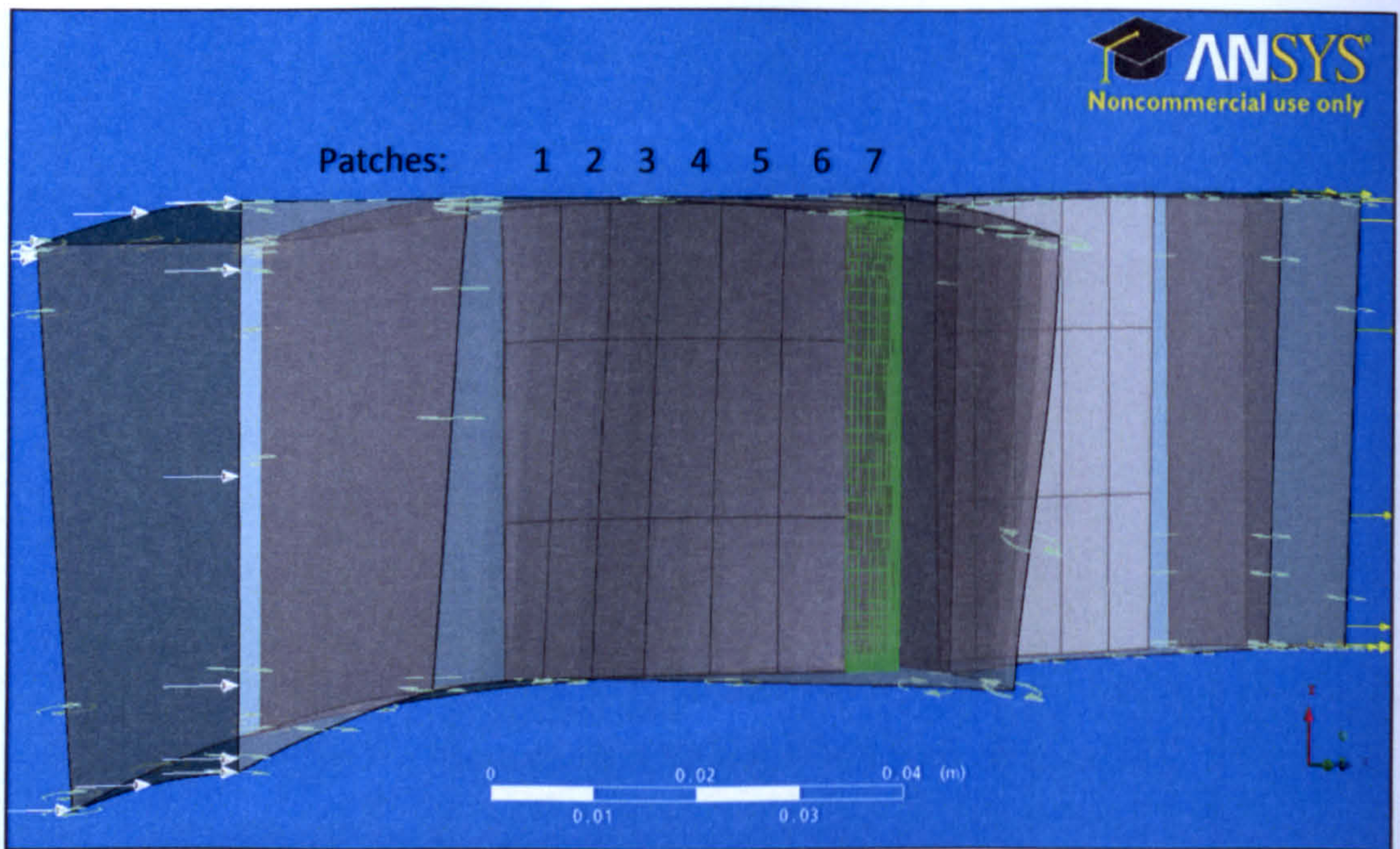


Figure 5-4: Compressor stage (Rotor-Stator) with rough surfaces

Once the solutions are obtained the results are extracted in CFX-Post the dedicated post-processor module of the CFD code. In this part of the code the results can be analysed, ie graphs can be assembled to illustrate the variation of the parameters (pressure, temperature, velocity and e.g.) as shown in the next figures.

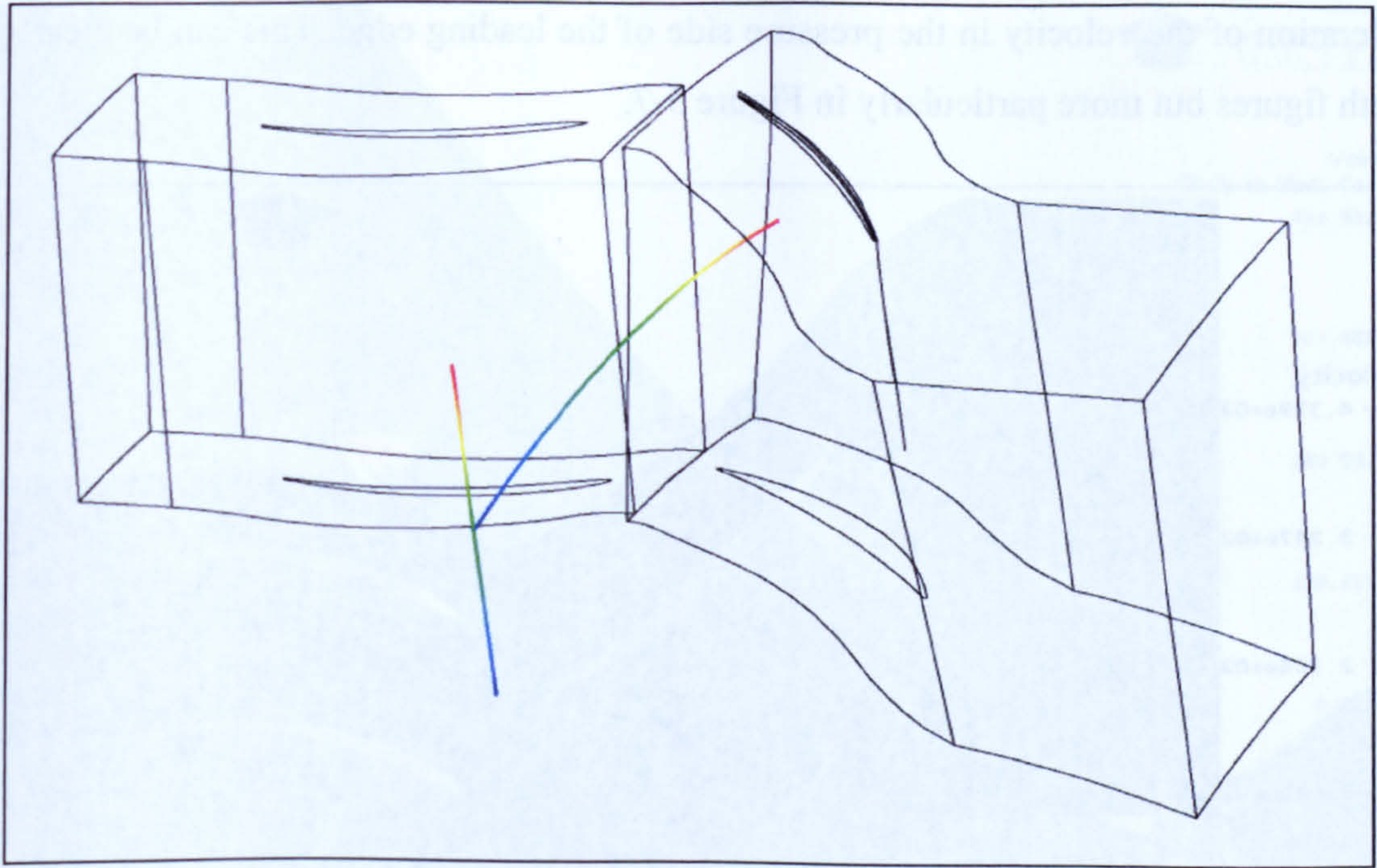


Figure 5-5: Traverse measurement of the rotor blade

Figure 5-5 above shows a wireframe representation of a stage domain together with two lines representative of the spatial representations, in a figurative format, of radial and pitchwise data collection lines. The blue and red ends of the lines are indication of the start and end of the plots. Therefore for the radial distribution plots the data originates at the hub and terminates at the shroud. Likewise the pitchwise data is plotted from the pressure side of the blade towards the suction side.

5.4 Clean Stage Qualitative Analysis

5.4.1 Velocity Distribution, Max flow Condition, Clean Stage

Figure 5-6 and 5-7 show the velocity distribution contours, relative velocity on rotors and absolute on stators, on a mid-span plane. As can be observed in Section 5.0 the rotor blade is exposed to negative incidence. When considering the triangle of velocities at the rotor inlet and the flow described in this plot, the axial flow vector, and hence the mass flow, is larger than would be the case at design point. For this reason the stagnation point is displaced closer to the suction side and, given the direction from which the flow approaches, there is a strong

acceleration of the velocity in the pressure side of the leading edge. This can be clearly seen in both figures but more particularly in Figure 5-7.

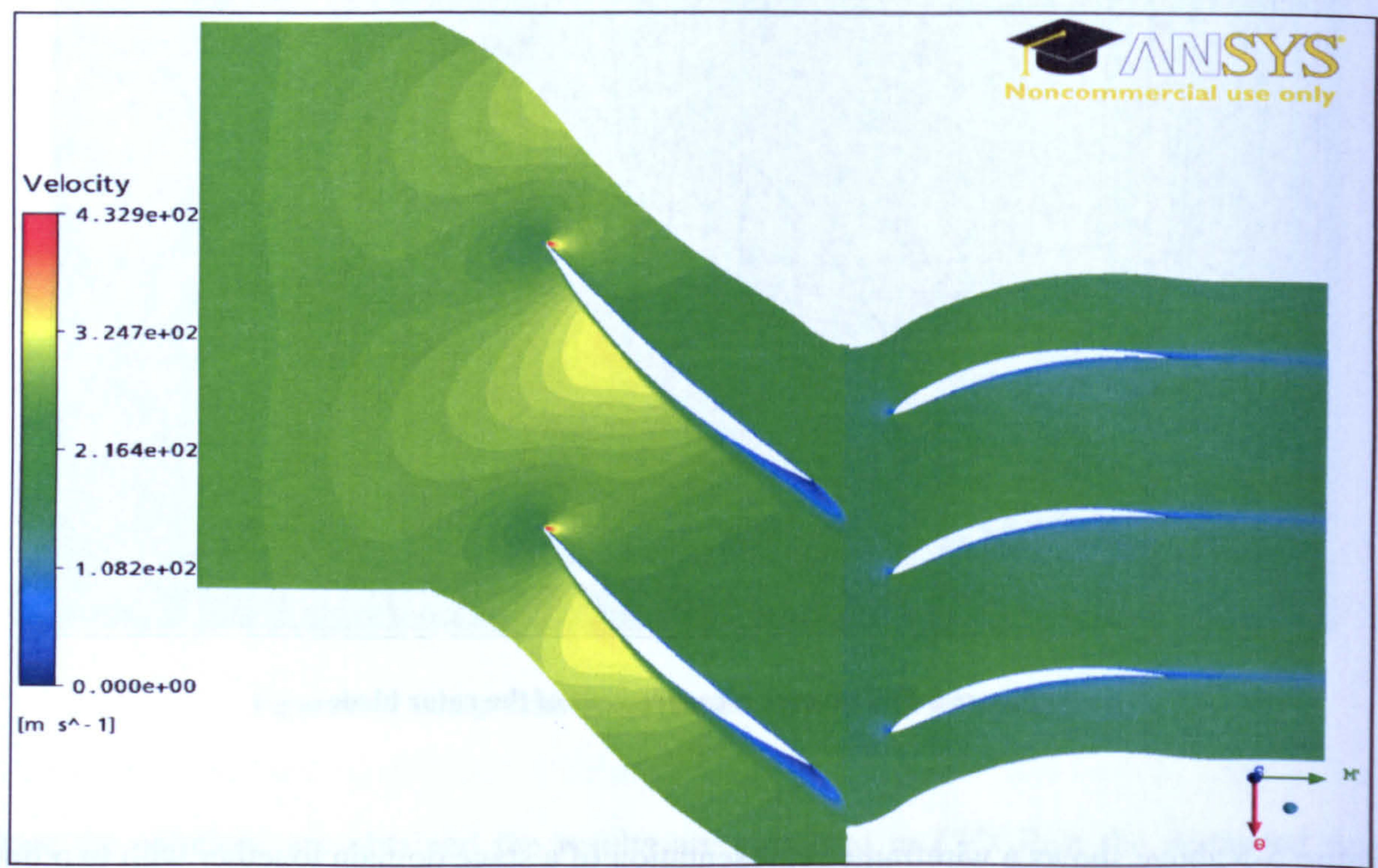


Figure 5-6: Compressor stage contours of velocity (m/s) at max flow condition

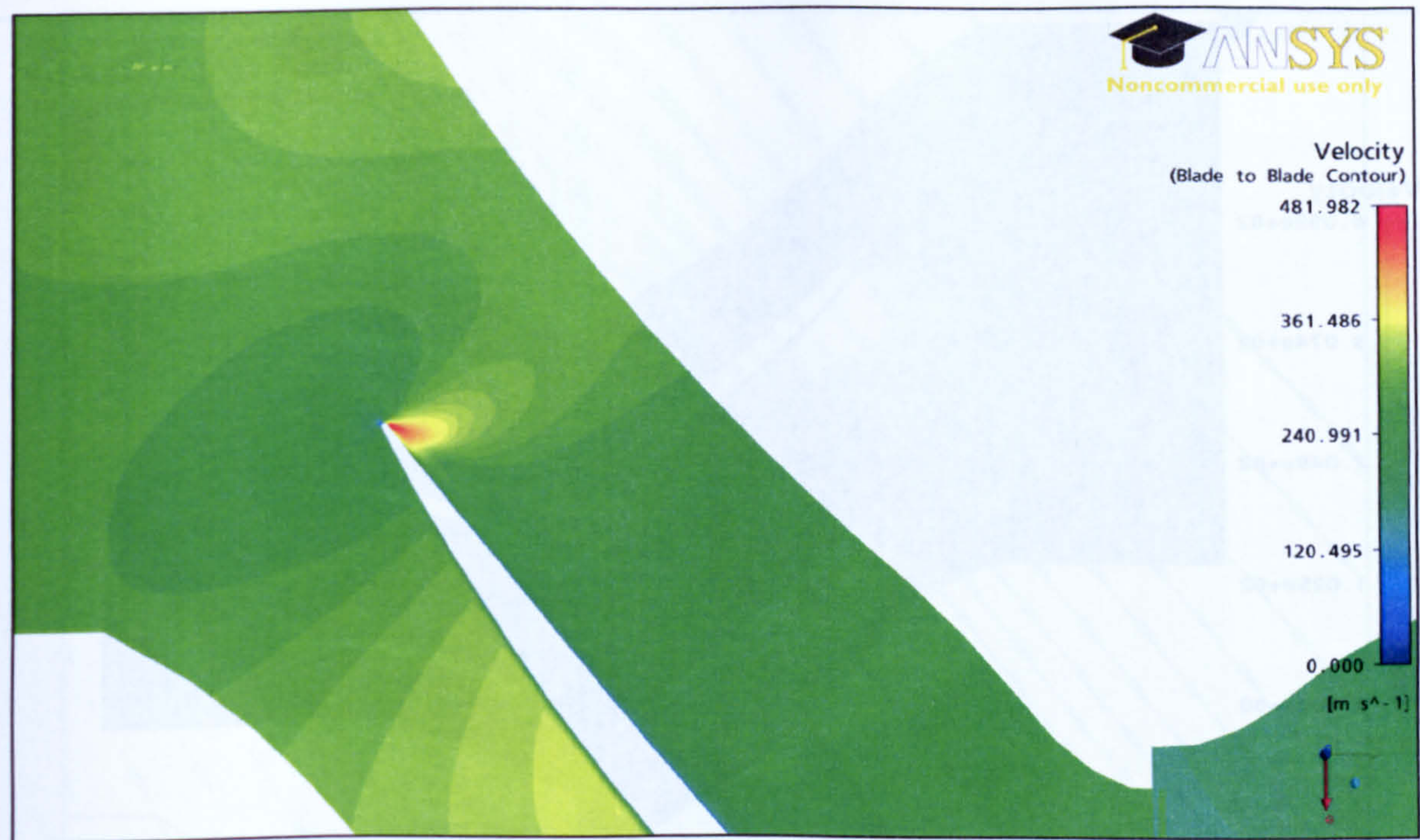


Figure 5-7: Velocity contour distribution on the rotor inlet at max flow

5.4.2 Velocity Distribution, Near-Surge Flow Condition, Clean Stage

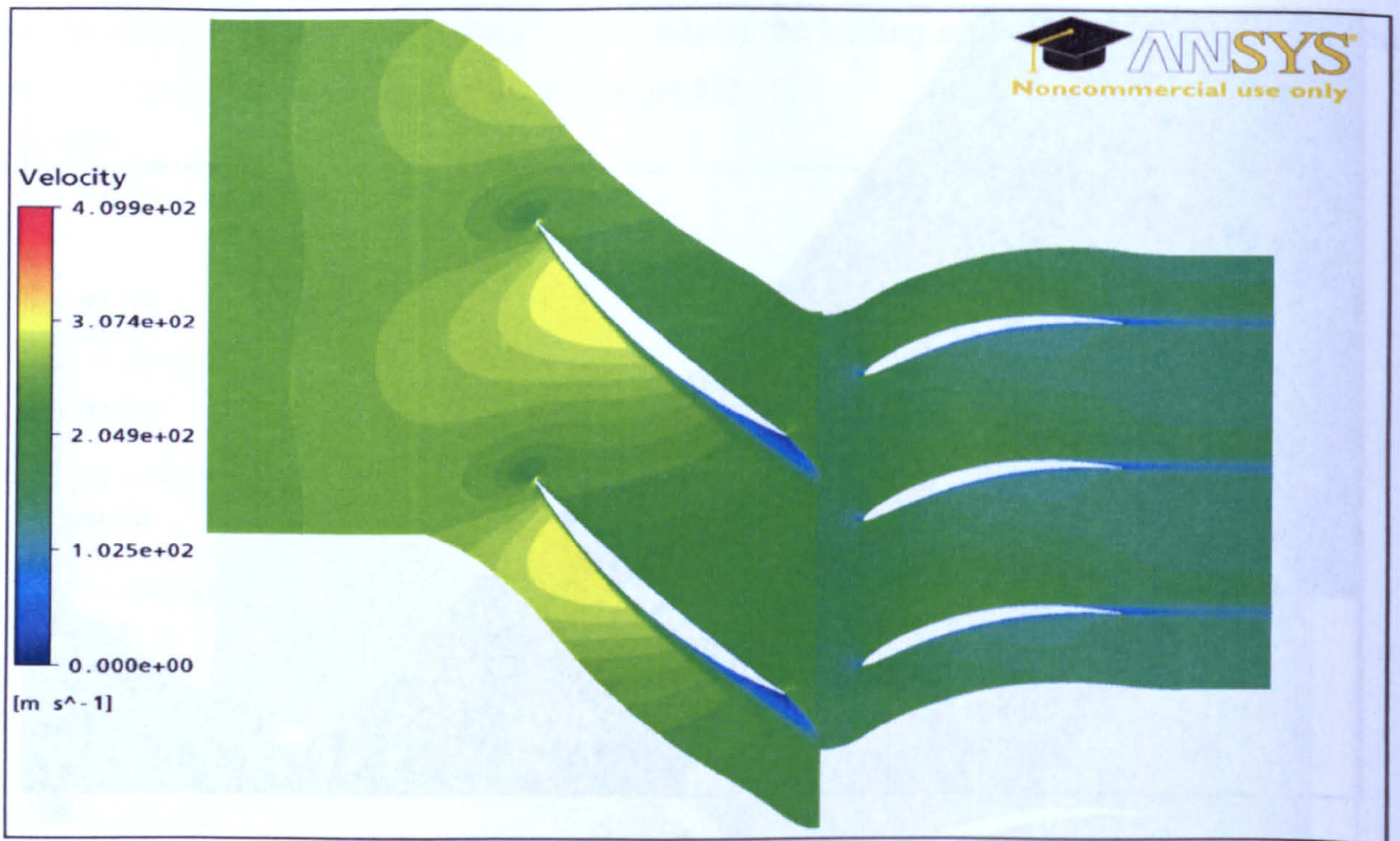


Figure 5-8: Velocity contour distribution on the stage at near surge flow

As can be seen in Figure 5-8 the stagewise velocity distribution at near surge is fairly similar to what was observed at max-flow. That includes a region of separation on the rotor blades that is common to both regimes. However the negative incidence observed in the rotor flow at max-flow is unsurprisingly not present in this point of the characteristic. The rotor velocity field associated with the higher work of the near surge point is characterised by the displacement of the locus of the peak velocity in the suction side forward towards the leading edge as can be see by comparing Figures 5-8 and 5-6. As can be seen from the Figures 5-9 and 5-10 the velocity vectors direction attaches the leading edge with different angle of incidence.

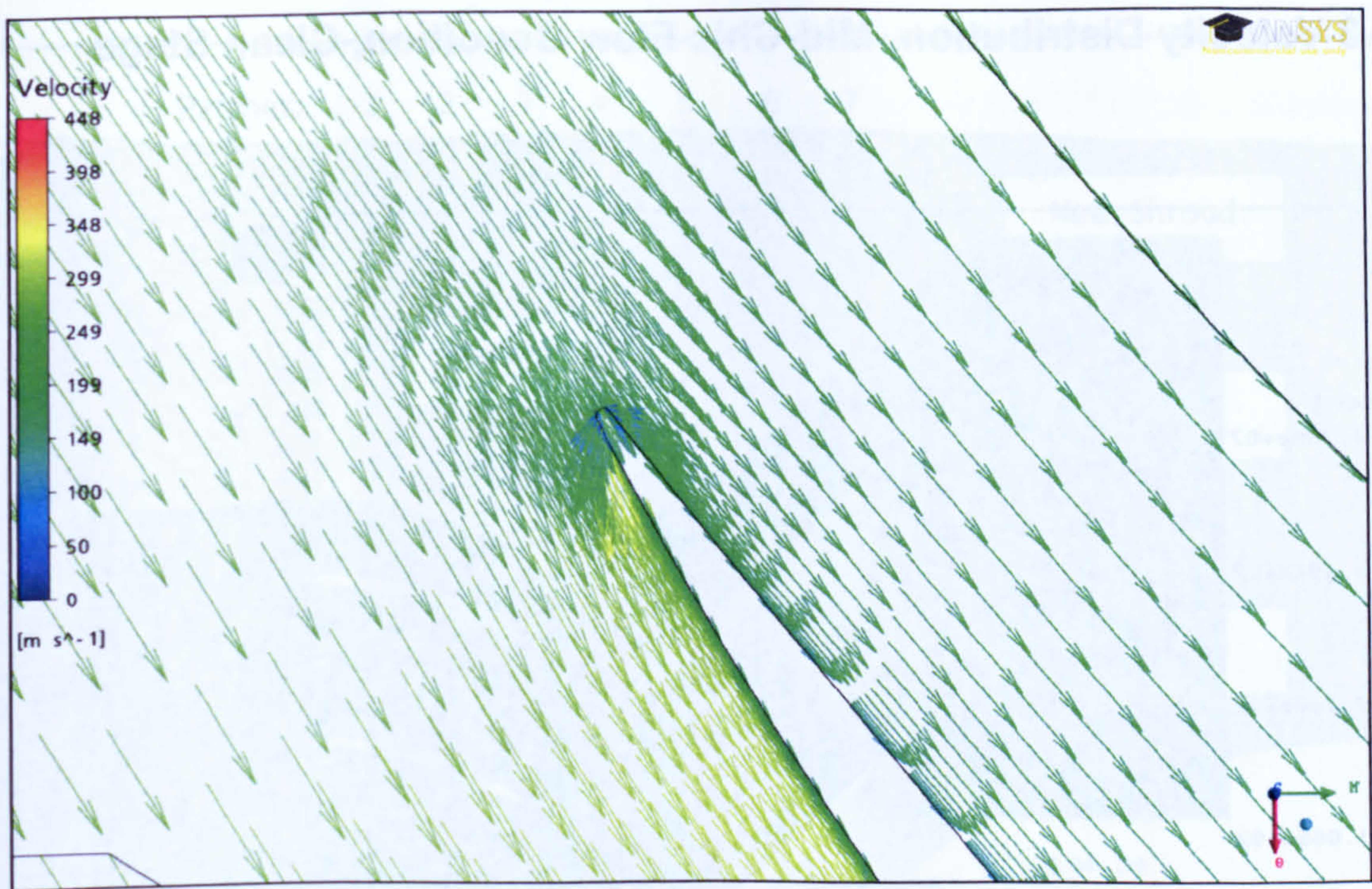


Figure 5-9: Velocity vectors on the rotor leading edge at near surge flow condition

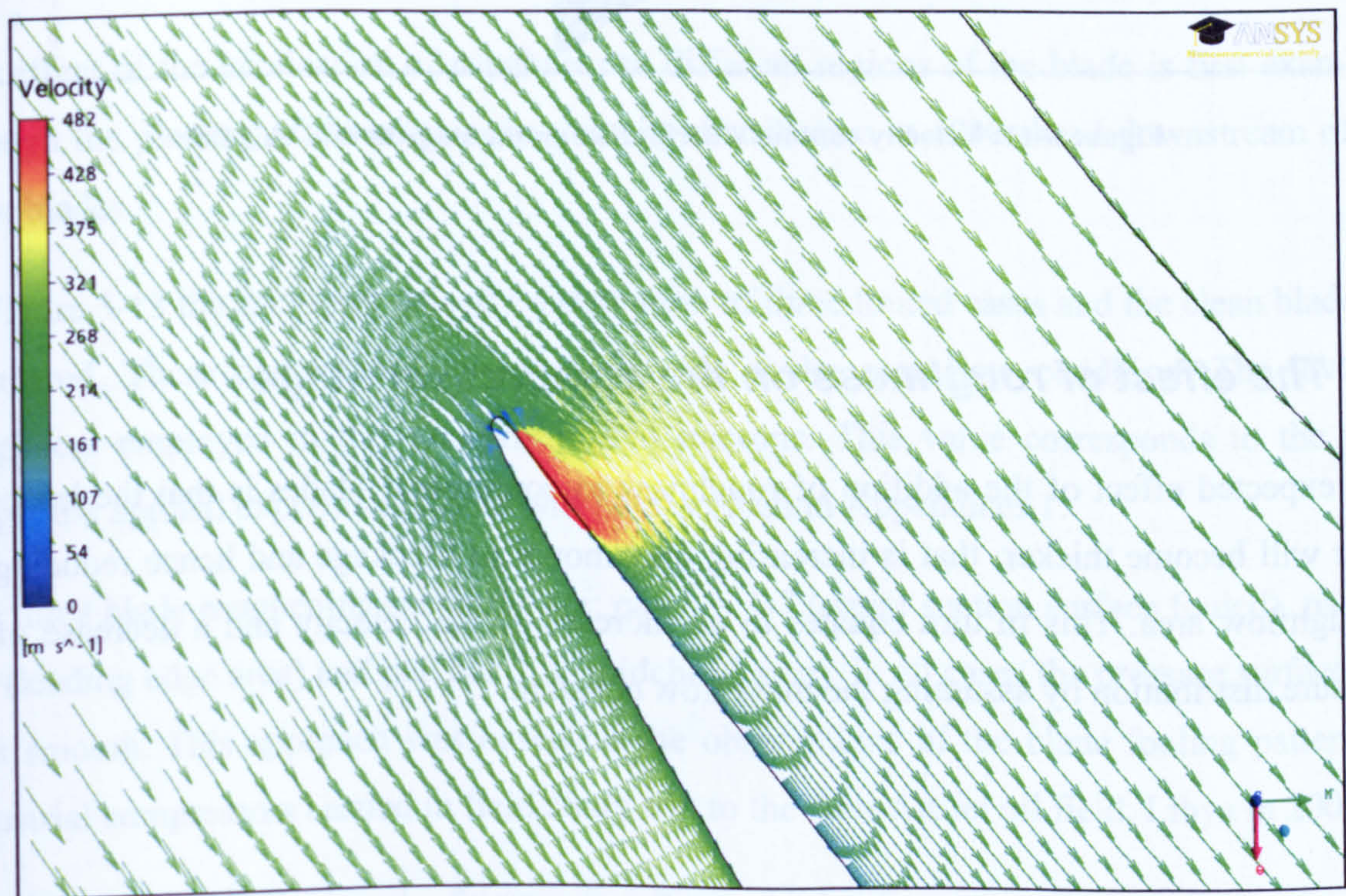


Figure 5-10: Velocity vectors on the rotor leading edge at max flow condition

5.4.3 Velocity Distribution, Mid-Chic Flow Condition, Clean Stage

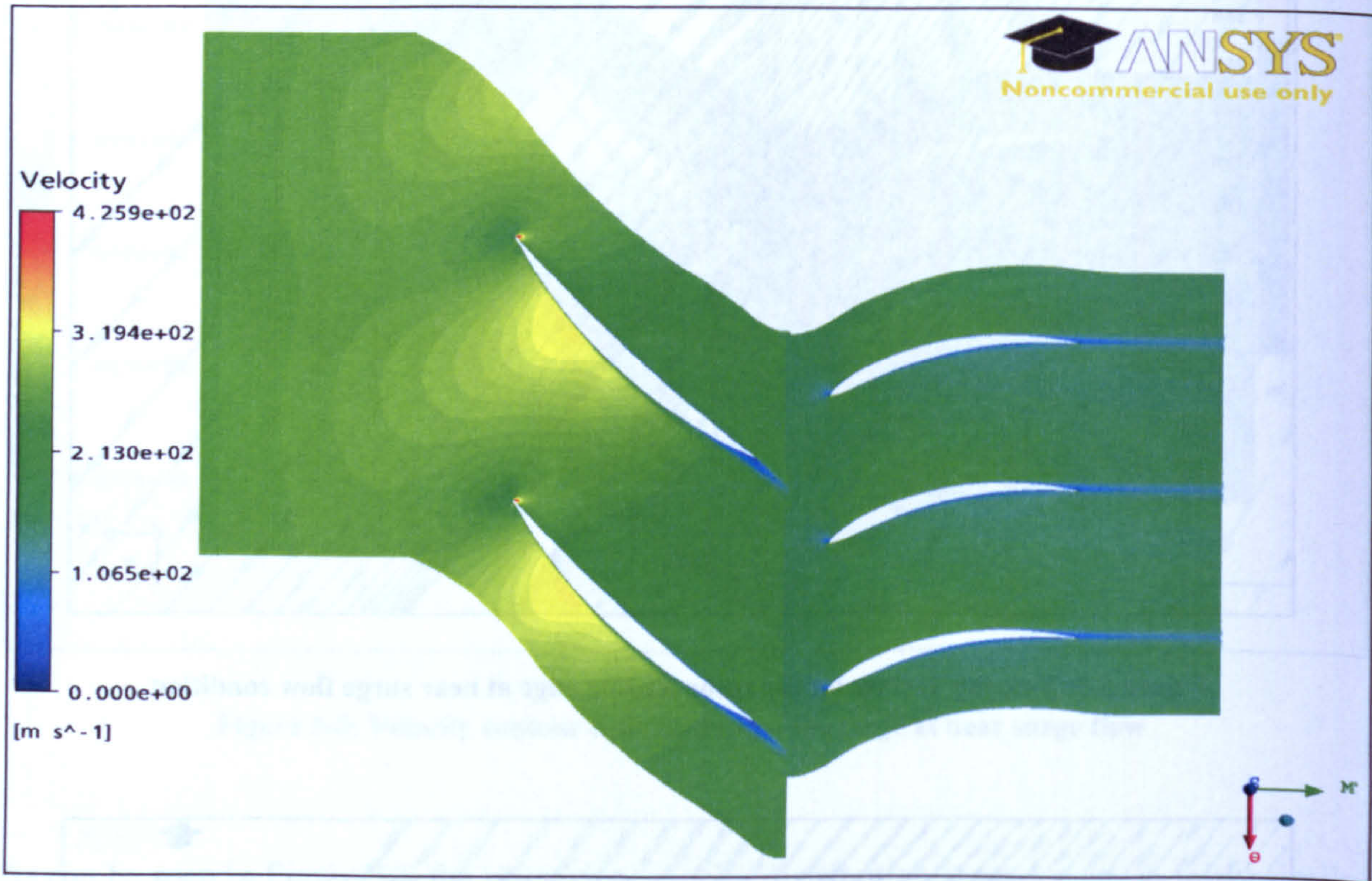


Figure 5-11: Velocity contour distribution on the stage at mid-chic flow

5.5 The effect of roughness on the compressor stage

The expected effect of the addition of roughness to compressor blades is that the boundary layer will become thicker, that is increasing the amount of blockage and hence reducing the throughflow area. This in turn equates to an increase of the velocity and a decrease of the pressure distribution by assuming the mass flow constant.

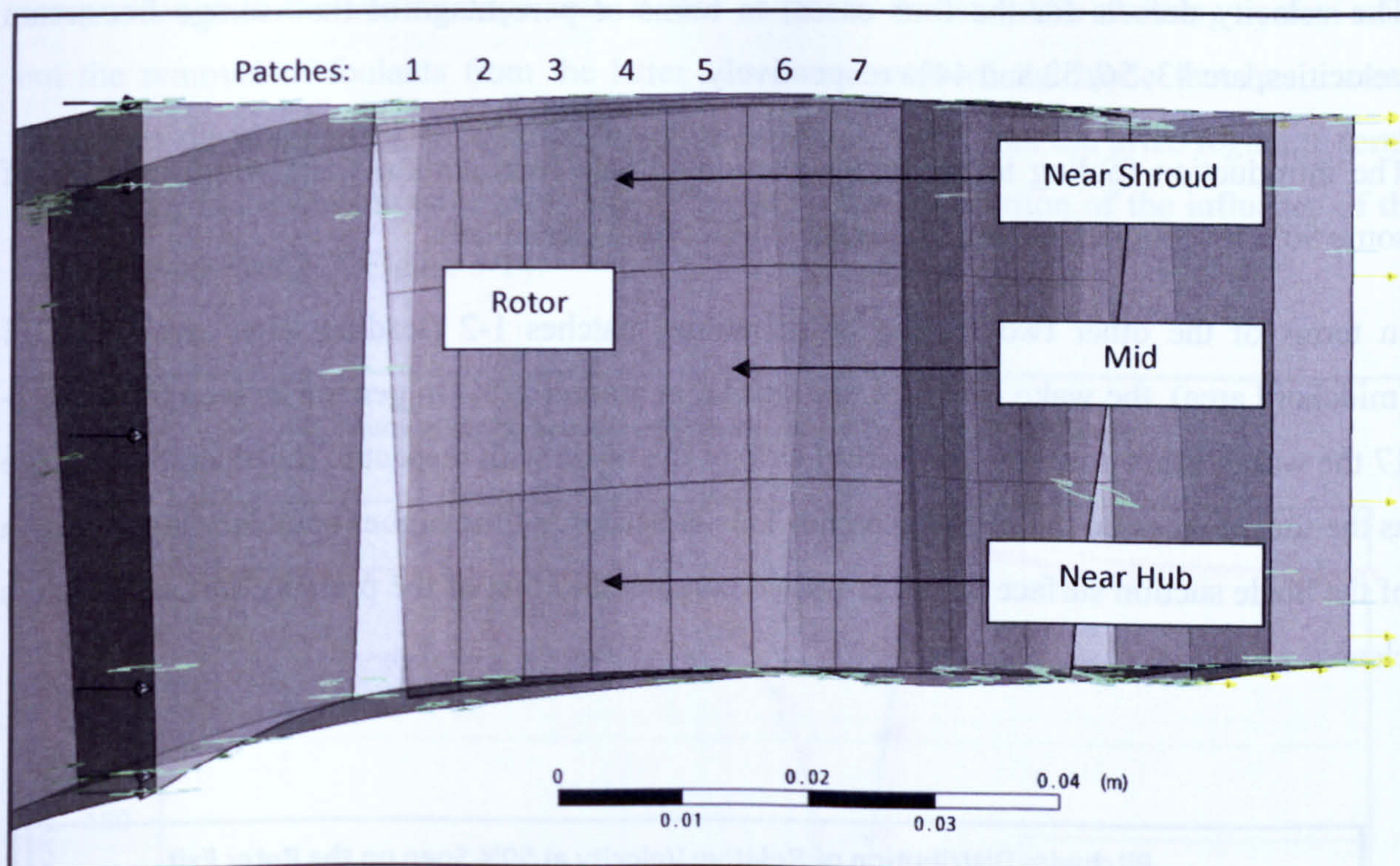


Figure 5-12: Rotor blade roughened in different regions

The effect of the application of roughness to different regions of the blade is first examined through the plotting of the pitch-wise velocity distribution at 10% chord downstream of the rotor blades.

In Figure 5-13 the mid-span velocity distribution of three fouled cases and the clean blade are compared. These comparisons were carried out at the max flow condition. The level of roughness employed in this analysis is 254 microns. This value corresponds to the peak roughness applied experimentally as will be described in Appendices 1.

The three blade roughening scenarios are: patches 1-7 (entire suction surface fouled), patches 1-2 (leading edge area) and patches 3-4 (midchord area). In all cases the pressure surface was kept smooth. This approach was based on the observations of the blade fouling patterns in industrial compressors carried in field study trip to the Abu-Attifel oil field, Libya in 2008.

In all the four studied cases the freestream velocity in the passage remains almost constant at around 230 m/s, Figure 5-17. The velocity defect in the wake drops from 130 m/s for the clean blade to 100 m/s when the entire suction surface is fouled, rough patches 1-7. In the case of the roughened midchord area, patches 3-4, the peak velocity defect is 125 m/s. When only the leading edge area is roughened, patch 1-2, the minimum wake velocity is 110 m/s.

The velocity defects for the four cases, in terms of percentage of the average freestream velocities, are 43, 56, 52 and 44% respectively.

The introduction fouling to the entire suction surface represents a wake width increase of some 56% by comparison with the wake width of the clean blade.

In terms of the other two fouling distributions, patches 1-2 (leading edge area) and 3-4 (midchord area), the wake increases are similar at around 22%. In general as seen in figure 5-17 the wakes move towards the suction side of the blade with respect to the clean blade wake as the fouled areas on the blade increase. This is because of the higher boundary layer growth of the blade suction surface which is fouled compared to that of the pressure surface which is clean.

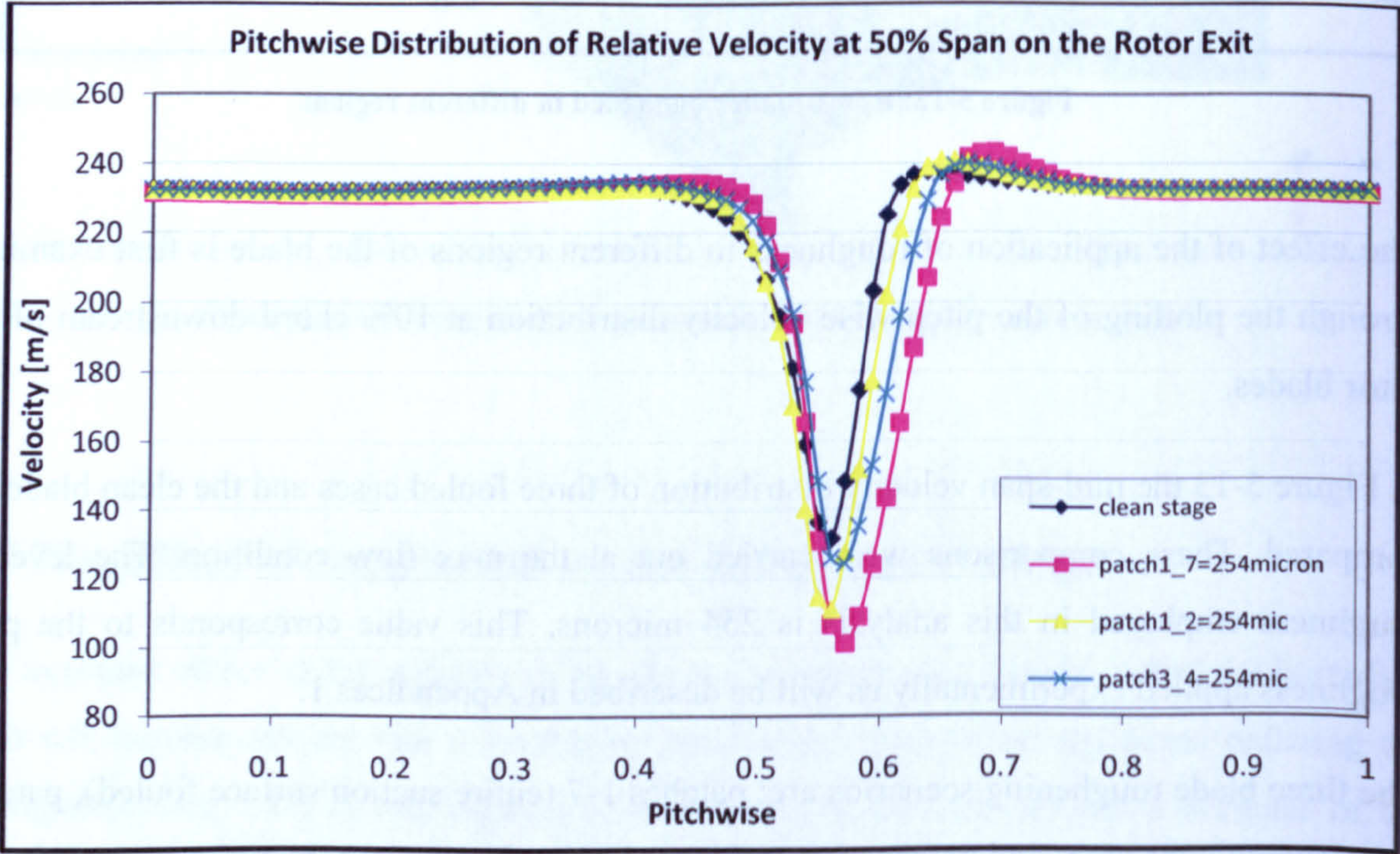


Figure 5-13: Velocity wakes of the rotor exit at different roughness

In Figure 5-14 the velocity distribution is the wake at 75% of the span is compared for the clean bade case and two roughness cases, patches 1-2 (leading edge area) and patches 6-7 (trailing edge area), see Figure 5-12. The structure of the wake is broadly unchanged between the clean blade case and the situation where the trailing edge area is roughed. This confirms the relative importance of the impact of roughness in the first part of the blade chord by

comparison with the trailing edge area. It also confirms that should a cleaning system miss out the removal of foulants from the latter chordwise areas of the blade that has a small impact on the blade blockage distribution. Conversely it shows that the prize region in terms of cleaning is the leading edge area. This is borne by the observation of the influence of the fouling in patches 1-2, Figure 5-14.

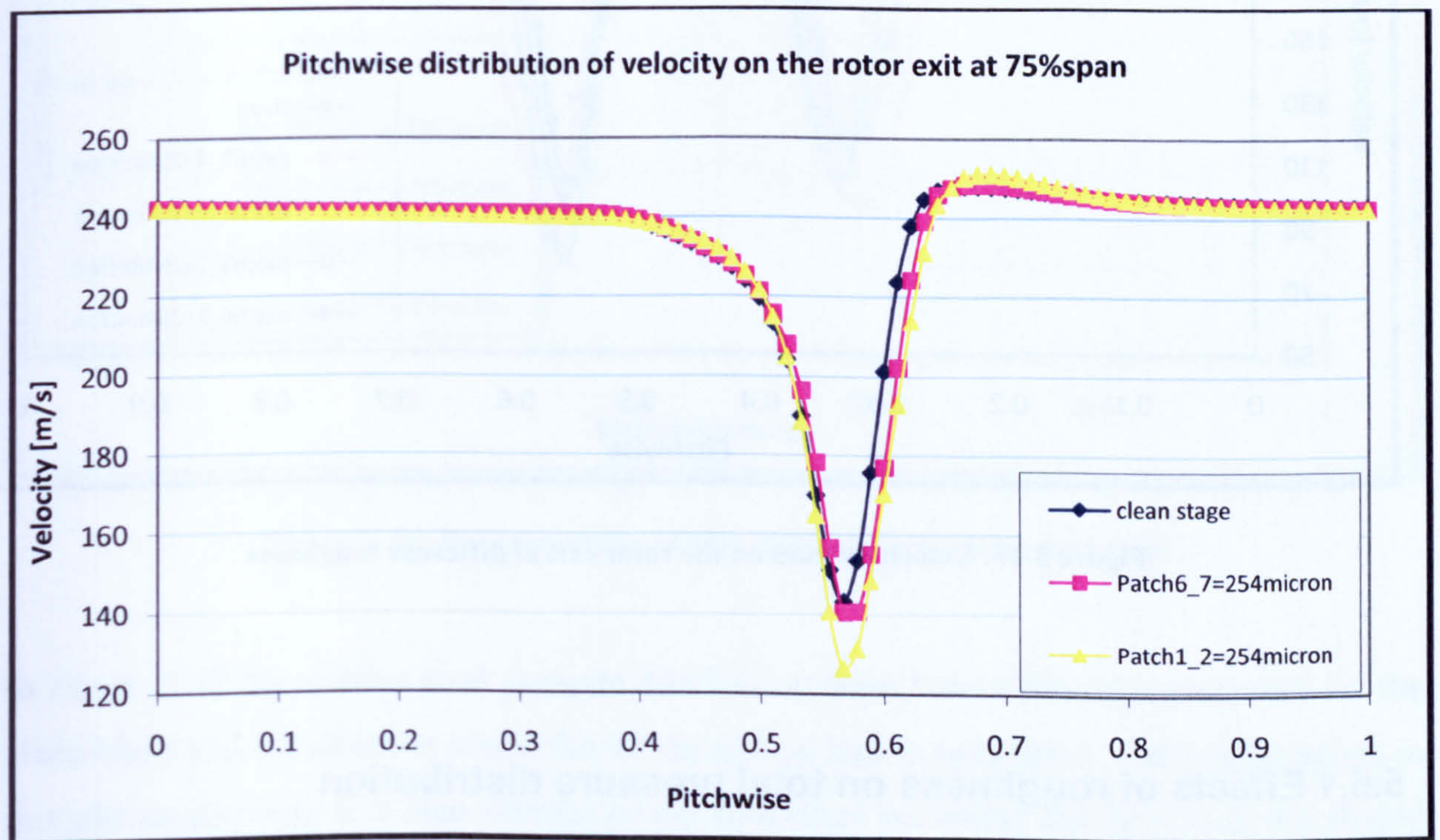


Figure 5-14: Velocity wakes on the rotor exit at leading and trailing edge by 75% span-wise

In Figure 5-15 the velocity wakes on the rotor exit at 25% span and different roughness distributions is plotted. The observations made for the mid-span case are broadly applicable to this plot.

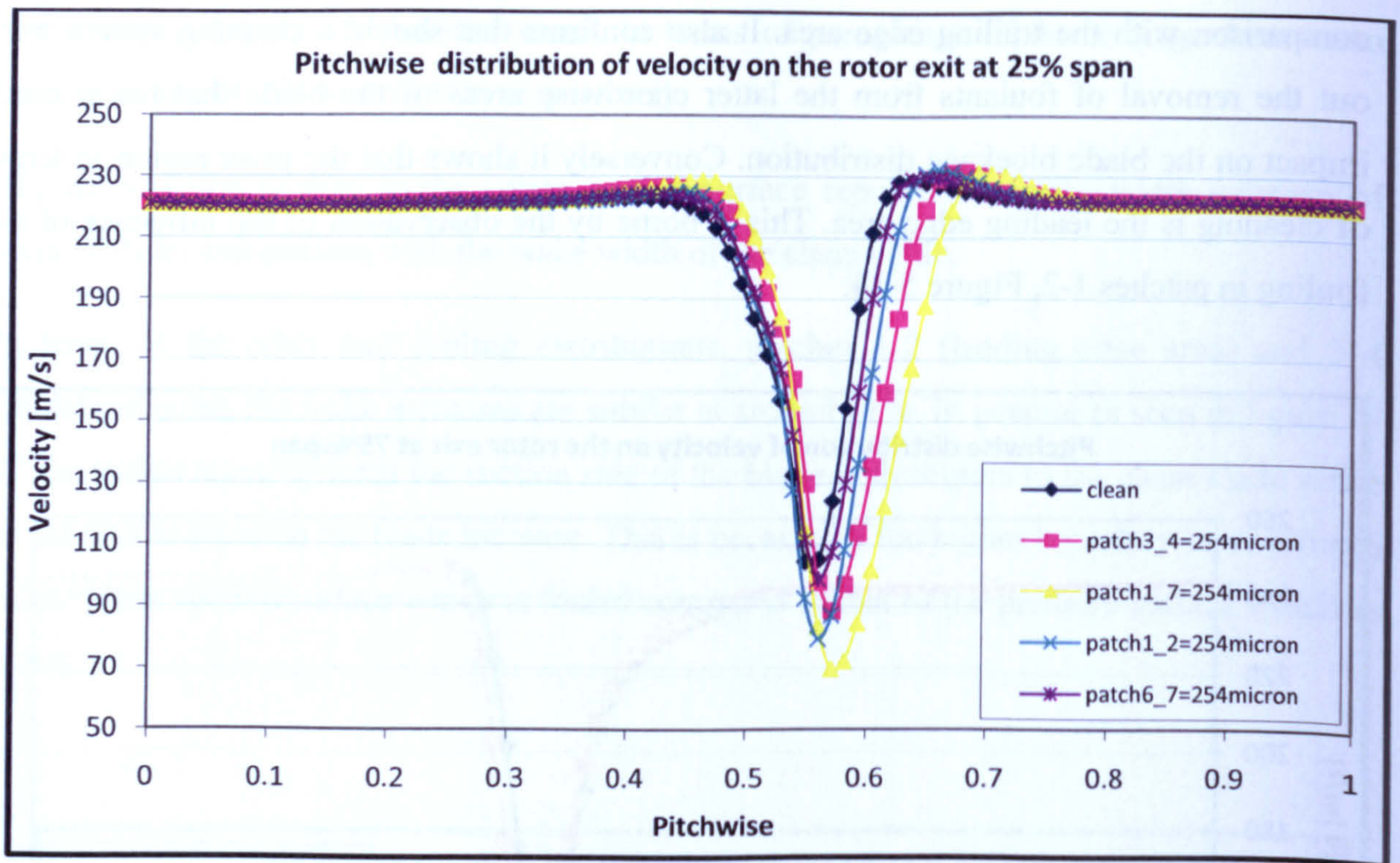


Figure 5-15: Velocity wakes on the rotor exit at different roughness

5.5.1 Effects of roughness on total pressure distribution

The compressor stage total pressure downstream of the blades is examined next. The pressure distribution at 10% of the chord downstream of the rotor blade is shown in Figure 5-16.

The total pressure defect in the wake can be analysed in terms of the wake width and minimum value. The total pressure defect in increasing order of wake width is as follows: clean blade, patch 6-7 (trailing edge), patch 3-4 (mid-chord), patch 1-2 (leading edge) and patch 1-7 (entire suction surface). The minimum total pressure values in the wakes are not too dissimilar, ranging from about 77000 Pa for the clean case to around 73500 Pa for the patch 1-7 (entire suction surface).

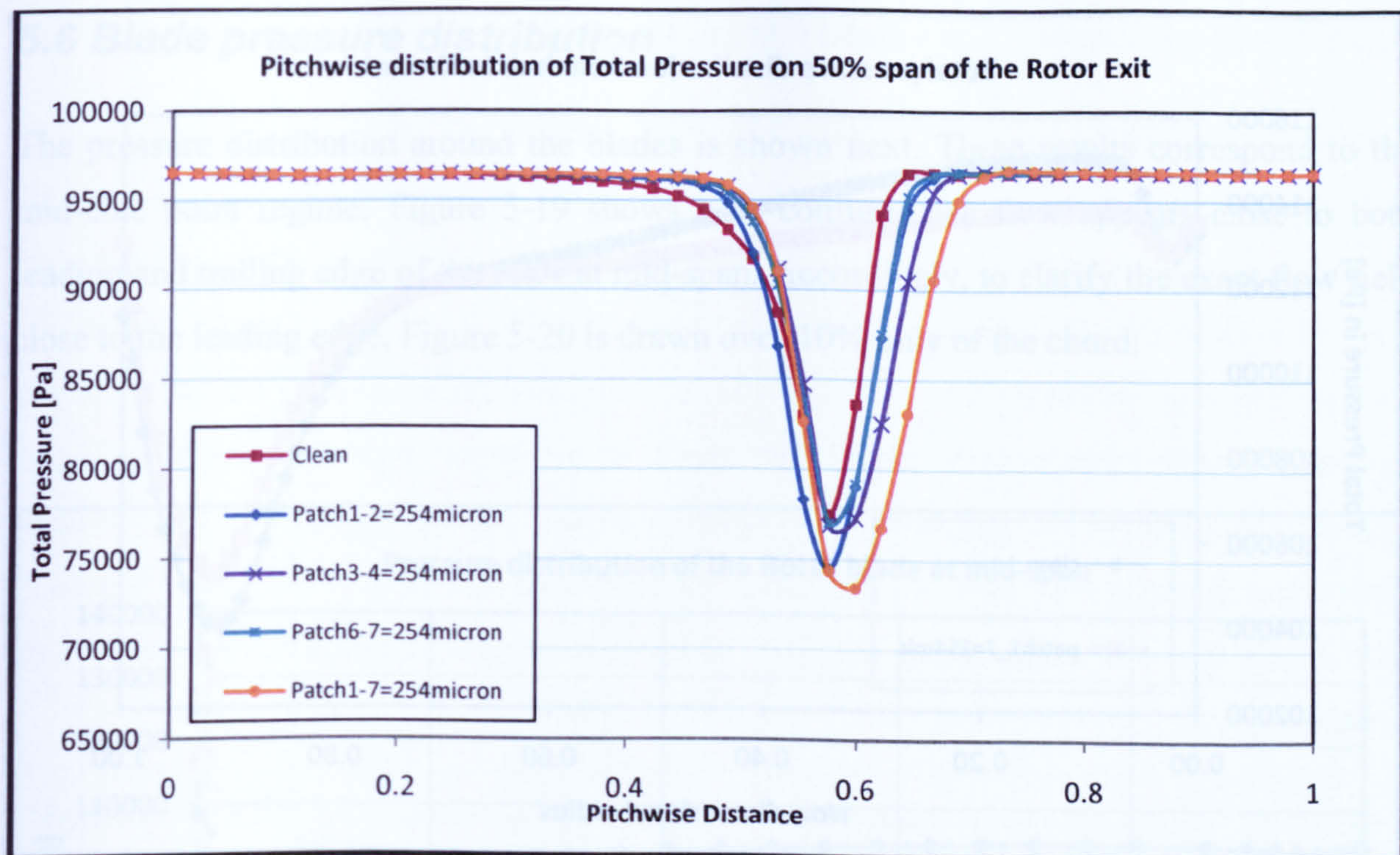


Figure 5-16: Pressure wakes of different roughness at the rotor exit

In Figure 5-17 the relative total pressure distribution from hub to shroud is presented for the clean blade and the situation where the whole suction side is roughened. Each of the points in this plot corresponds to a area average of the equivalent pitchwise distribution of the plotted quantity. The first three quarters of the blade span presents a shallow total pressure decrease with the two configurations showing fairly similar trends and absolute values. The last fifth of the blade span is occupied with a sharp decrease of total pressure, of the order of some 8000 Pa or 7% of the average value. This dip in total pressure is associated with the tip clearance vortex.

The equivalent distribution for the other roughness distributions, with the patch naming convention as described above, is shown in Figure 5-18. As can be seen in this plot the radial structure of the total pressure distribution is fairly similar for all cases and essentially independent of the blade roughness arrangement.

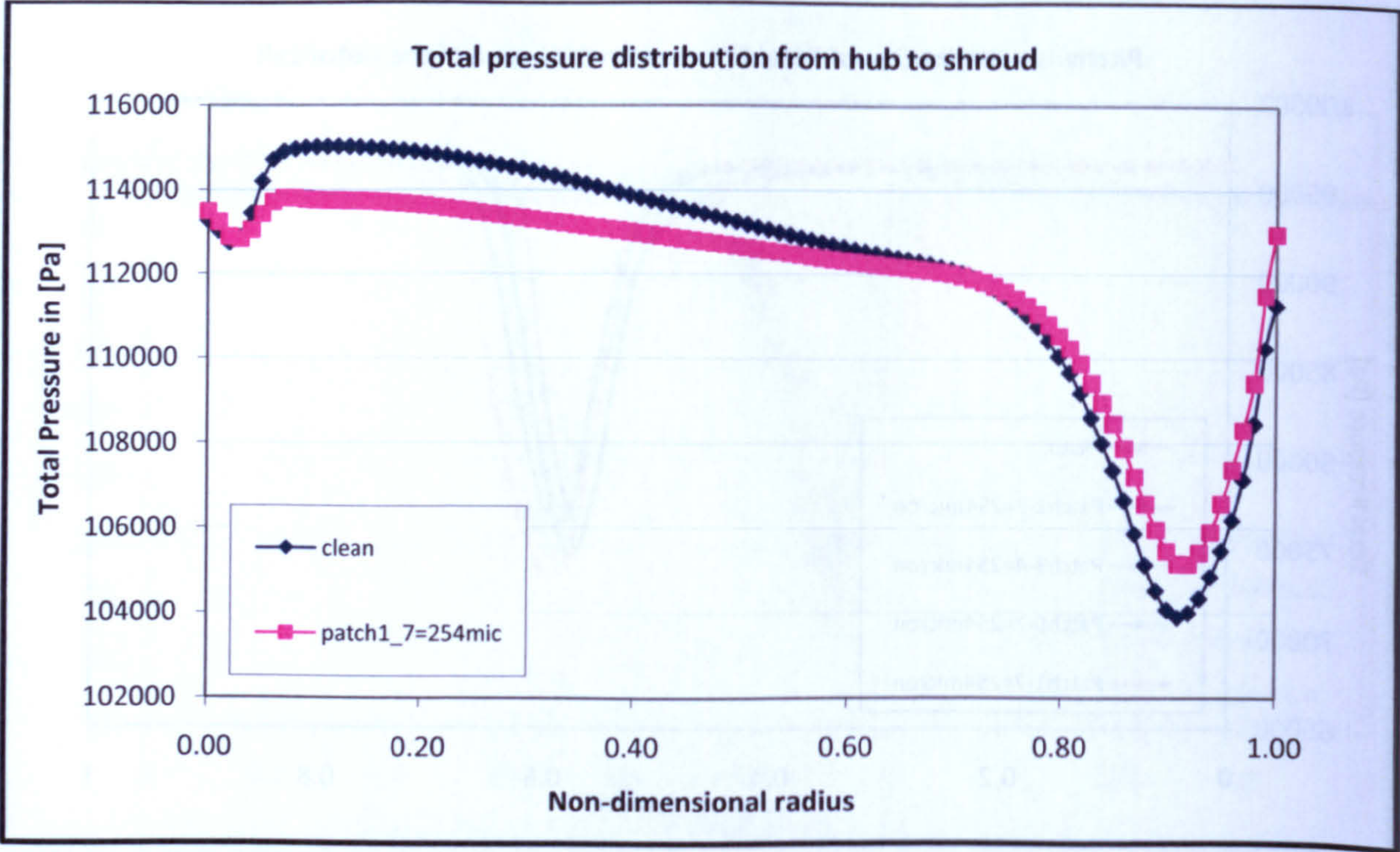


Figure 5-17: Total pressure distribution of the rotor at stream-wise distance

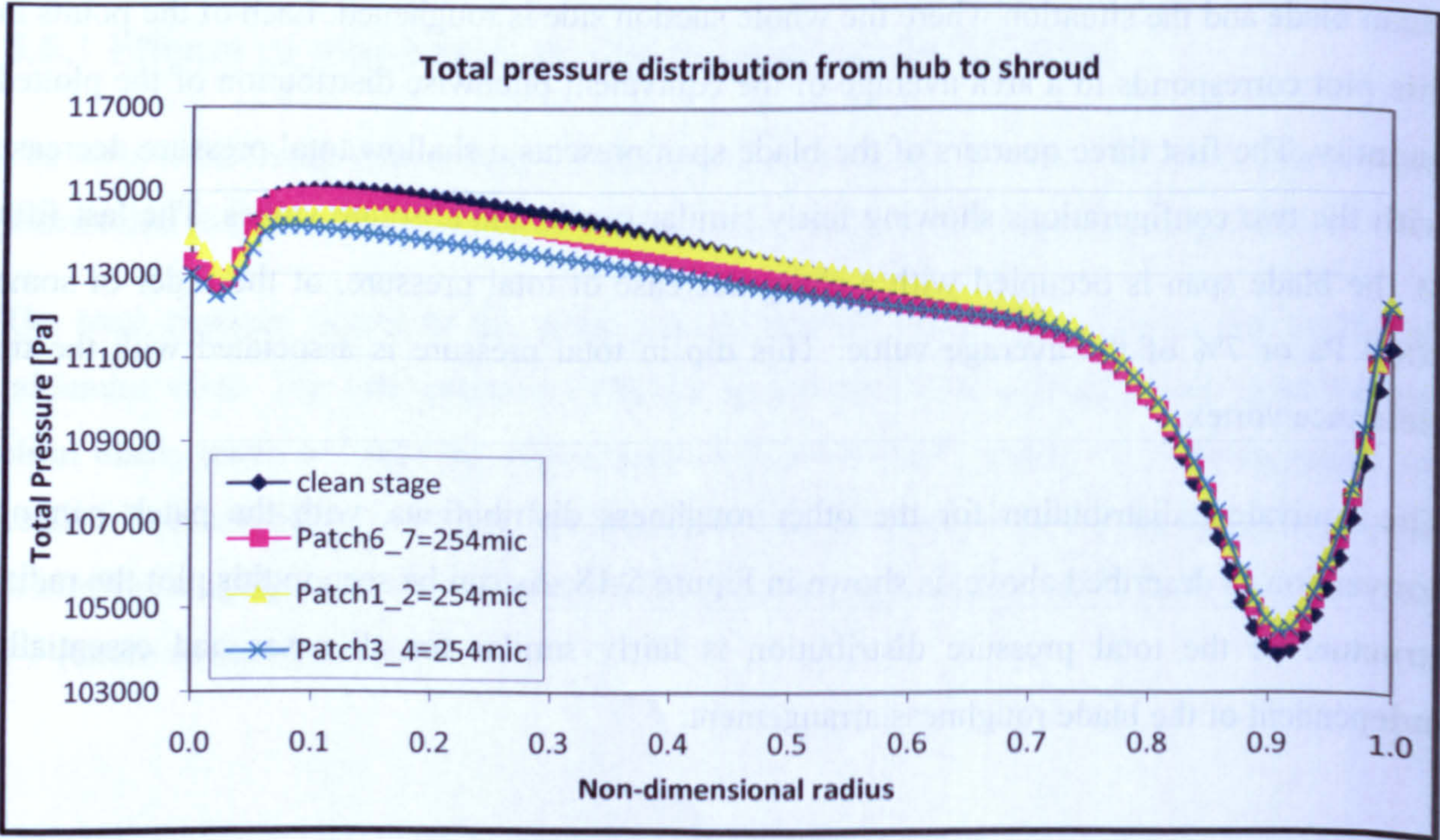


Figure 5-18: Total pressure distribution at different roughness on stream-wise location

5.6 Blade pressure distribution

The pressure distribution around the blades is shown next. These results correspond to the mid-chic point regime. Figure 5-19 shows how confused the flow appears close to both leading and trailing edge of the rotor at mid-span. Accordingly, to clarify the exact flow field close to the leading edge, Figure 5-20 is drawn over 10% only of the chord.

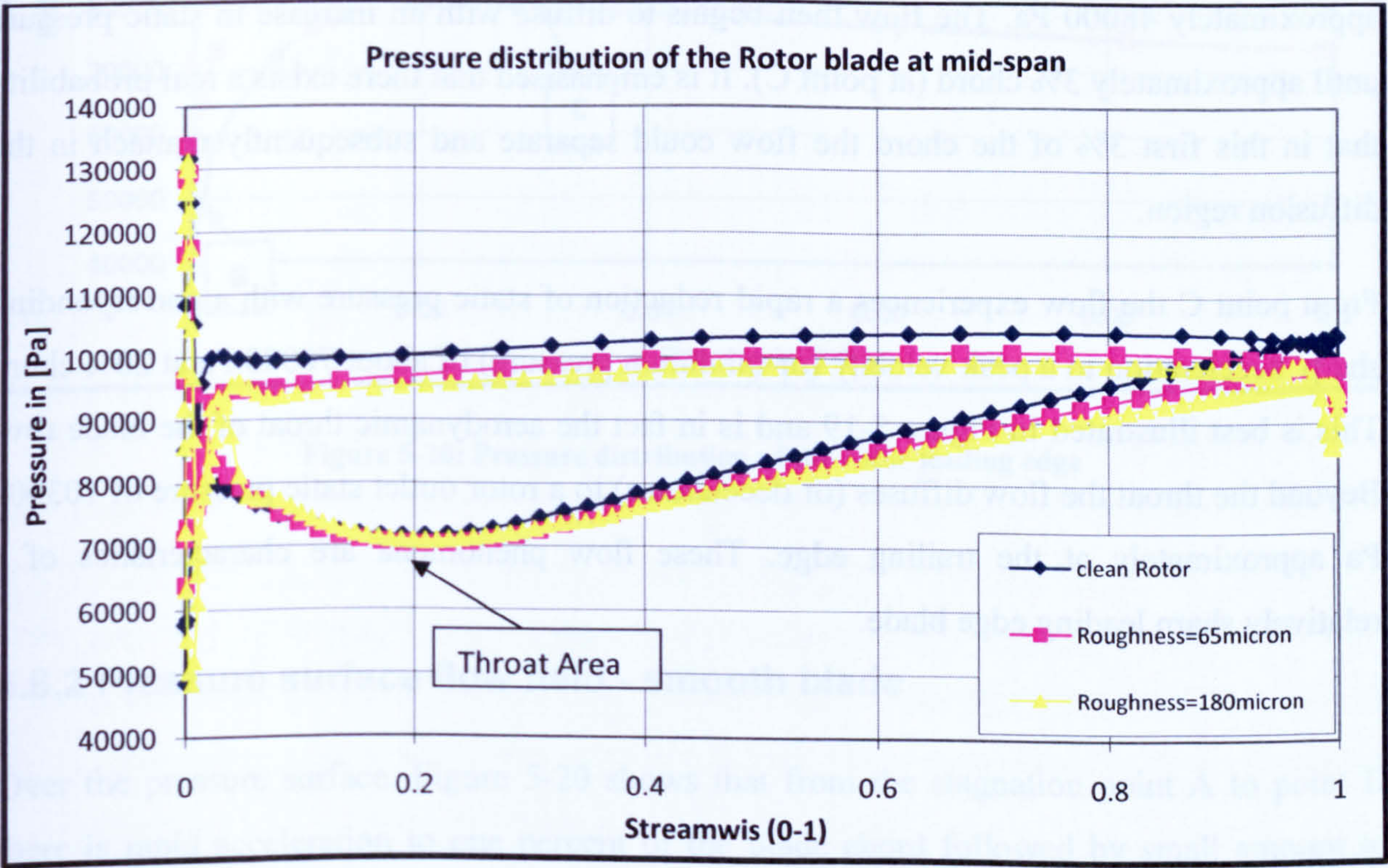


Figure 5-19: Pressure distribution on the Rotor blade in mid chic flow condition

The stagnation point for the clean blade is at approximately half of 1% on the pressure side of the blade (point B). This is also a point at which the flow field divides between pressure and suction surfaces. The position of the stagnation point is displaced to the pressure surface rather than at the geometric leading edge because of the stream-wise curvature in the upstream flow field generated by the pressure waves caused by the blade presence diverting the flow ahead of the leading edge.

5.6.1 Suction surface flow field - smooth blade

Considering is first the suction surface flow beginning at the stagnation point (A on the enlarged picture) of the flow in Figure 5-20. This point, in fact, appears very close to the leading edge at approximately one half of one percent of the chord where the corresponding stagnation (also the static) pressure is approximately 135000 Pa. Beyond this point the flow then proceeds to accelerate very rapidly with an associated pressure drop around the actual geometric leading edge (0% chord) to reach a minimum pressure (at point B) of approximately 48000 Pa. The flow then begins to diffuse with an increase in static pressure until approximately 3% chord (at point C). It is emphasised that there exists a real probability that in this first 3% of the chord the flow could separate and subsequently reattach in the diffusion region.

From point C the flow experiences a rapid reduction of static pressure with a corresponding sharp acceleration to a peak velocity (or minimum pressure) of about 70000 Pa at 20% chord. This is best illustrated in Figure 5-19 and is in fact the aerodynamic throat of the blade row. Beyond the throat the flow diffuses (or decelerates) to a rotor outlet static pressure of 103000 Pa approximately at the trailing edge. These flow phenomena are characteristics of a relatively sharp leading edge blade.

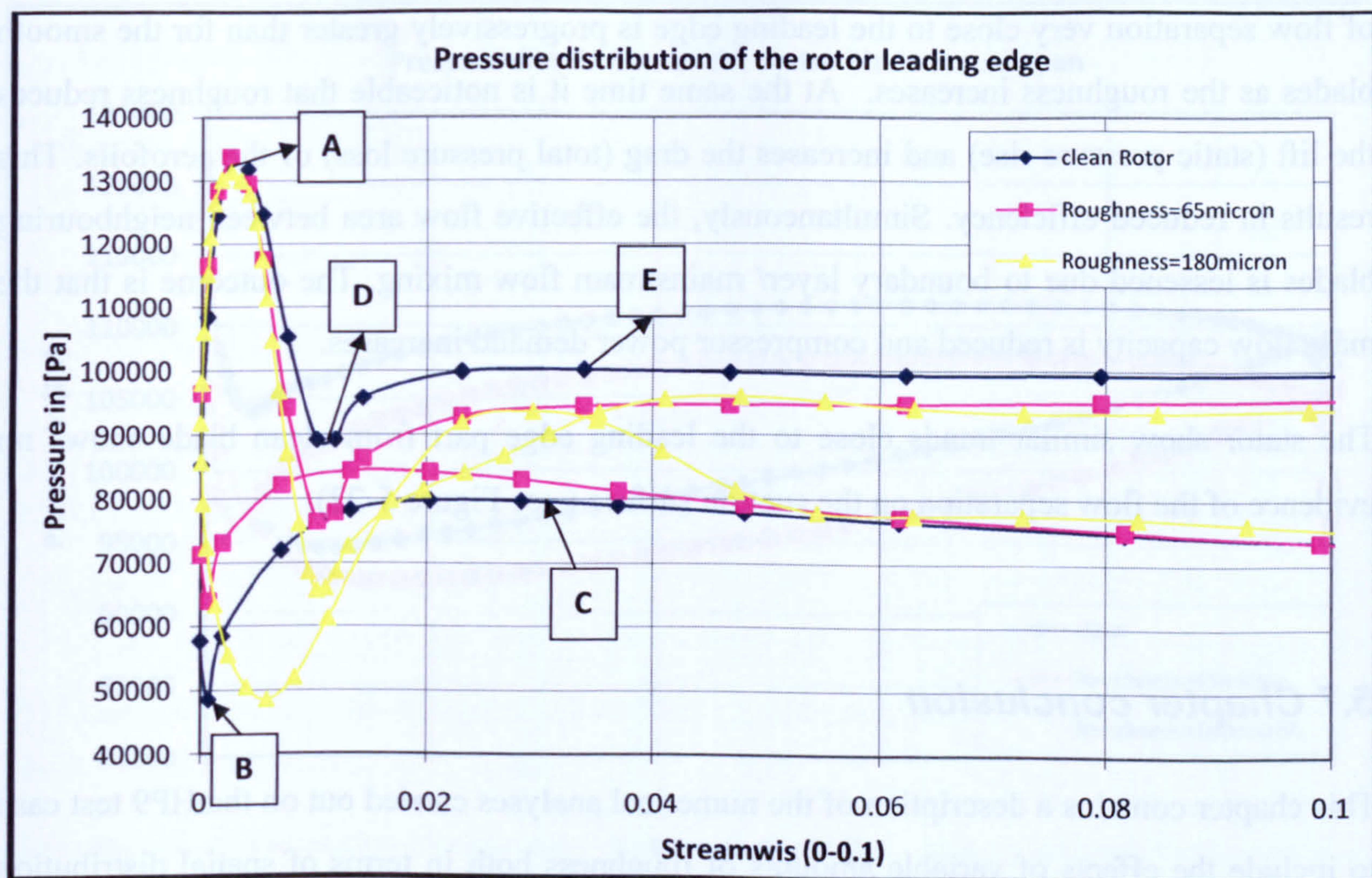


Figure 5-20: Pressure distribution of the rotor leading edge

5.6.2 Pressure surface flow field - smooth blade

Over the pressure surface, Figure 5-20 shows that from the stagnation point A to point D there is rapid acceleration to one percent of the blade chord followed by small amount of diffusion from (D to E). Again this can lead to a separation followed by a reattachment. However, in the author's opinion, this is much less likely since the amount of the pressure rise is quite small. Downstream of approximately the 2.5% blade chord position (point E), the flow remains at almost constant static pressure to the trailing edge (Figure 5-19).

5.6.3 Flow field analysis – roughened blade

The effect of the roughness on pressure distribution is shown on Figure 5-19 and 5-21. The red curves correspond to a roughness of 65 microns and the yellow curve to 180 microns. Reference to Figure 5-20 in particular shows that the effects at the leading edge referred to in the clean case above are exaggerated in both roughness cases. This means that the likelihood

of flow separation very close to the leading edge is progressively greater than for the smooth blades as the roughness increases. At the same time it is noticeable that roughness reduces the lift (static pressure rise) and increases the drag (total pressure loss) of the aerofoils. This results in reduced efficiency. Simultaneously, the effective flow area between neighbouring blades is lessened due to boundary layer/ mainstream flow mixing. The outcome is that the mass flow capacity is reduced and compressor power demand increases.

The stator show similar trends close to the leading edge part from clean blade shows no evidence of the flow separation on the suction surface (see Figure 5-22).

5.7 Chapter conclusion

This chapter contains a description of the numerical analyses carried out on the HP9 test case to include the effects of variable amounts of roughness both in terms of spatial distribution and equivalent grain size. The examination of the effects of adding roughness to selected regions of the rotor blade suction side has shown the relative importance of the leading edge end on the blade suction side over other regions of the blade. The relative impact was quantified in terms of the wake velocity defect, magnitude and wake width. However when the radial distribution is considered the differences between clean and roughened blades is less apparent excepting for the case where the entire suction side of the blade is roughened.

The pressure distribution around the blades was also examined. This part of the work has shown the benefits of flow analysis using CFD in a very small region of the flow field. In this case close the analysis is confined to the leading edge and trailing edge of a compressor blade. It also demonstrates clearly the disadvantages of fouling in reducing pressure rise through increases of total pressure loss and therefore reduced efficiency.

It is emphasised that experimental verification of this trend would be very difficult to achieve with a cascade rig because both rotor and stator would need static pressure tapings very close to one another at the leading edge of the blade. This would be extremely difficult to achieve in practice.

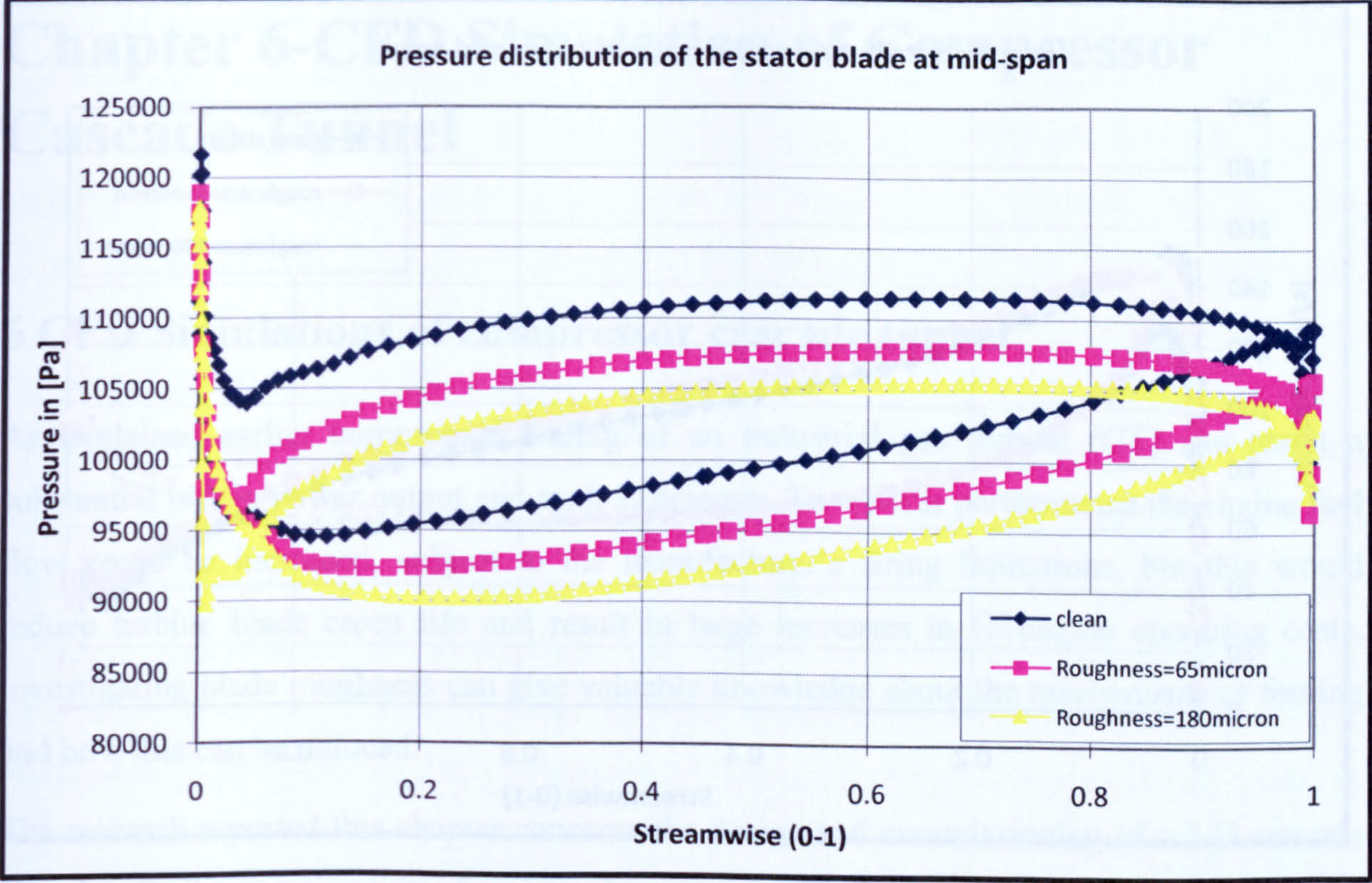


Figure 5-21: Pressure distribution of the Stator blade

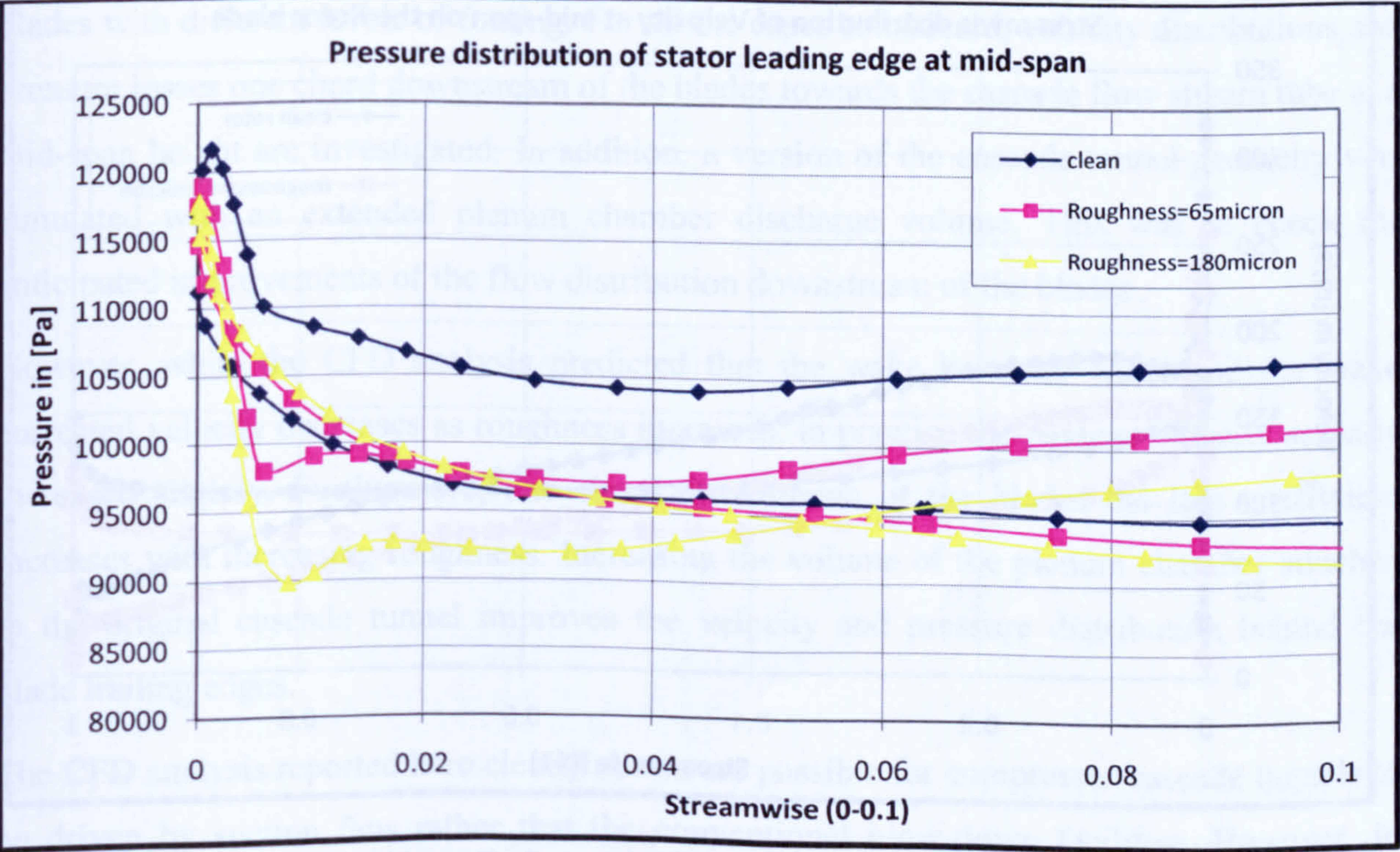


Figure 5-22: Pressure distribution of the stator leading edge (0-0.1) % stream-wise

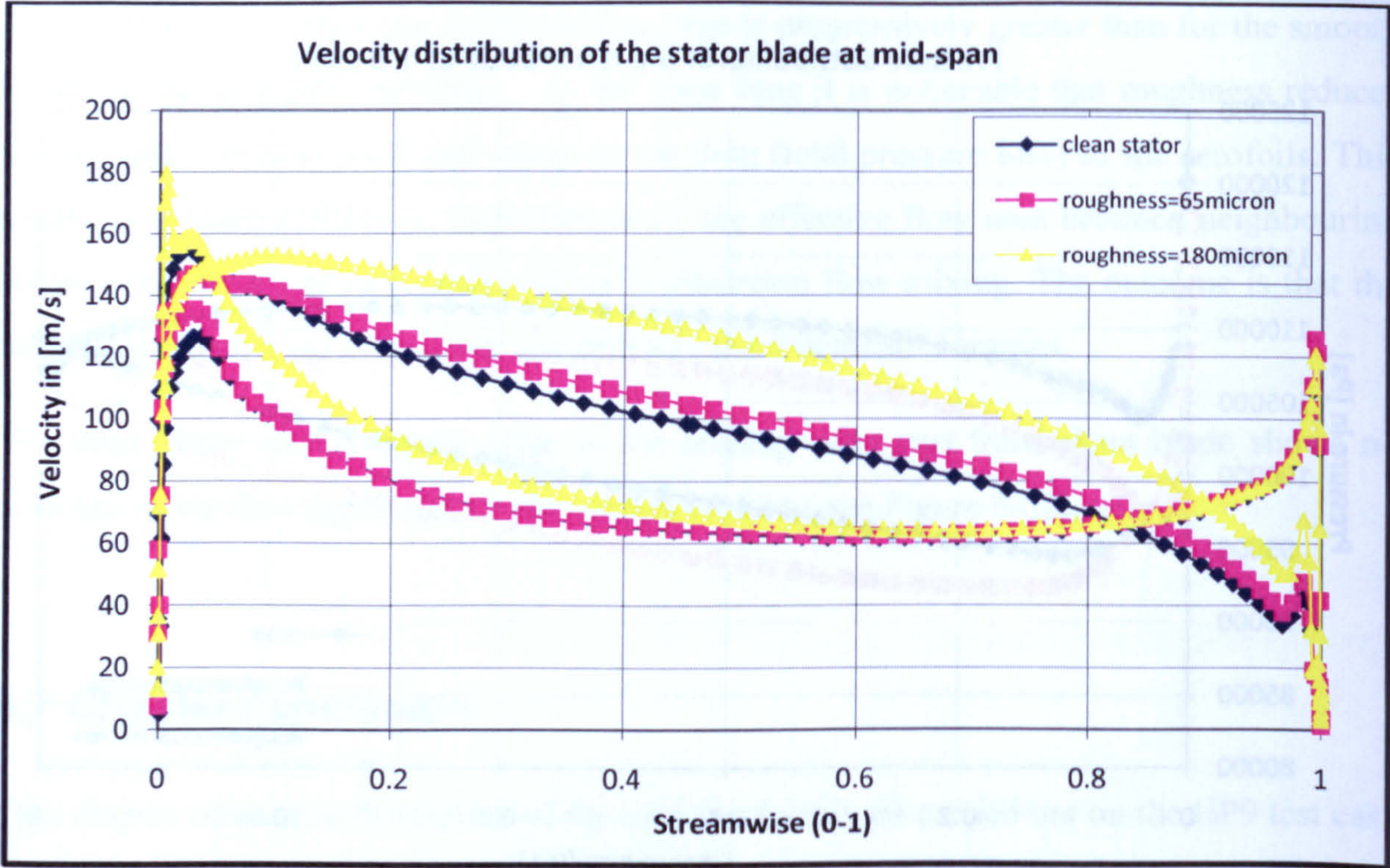


Figure 5-23: Velocity distribution on the Stator blade

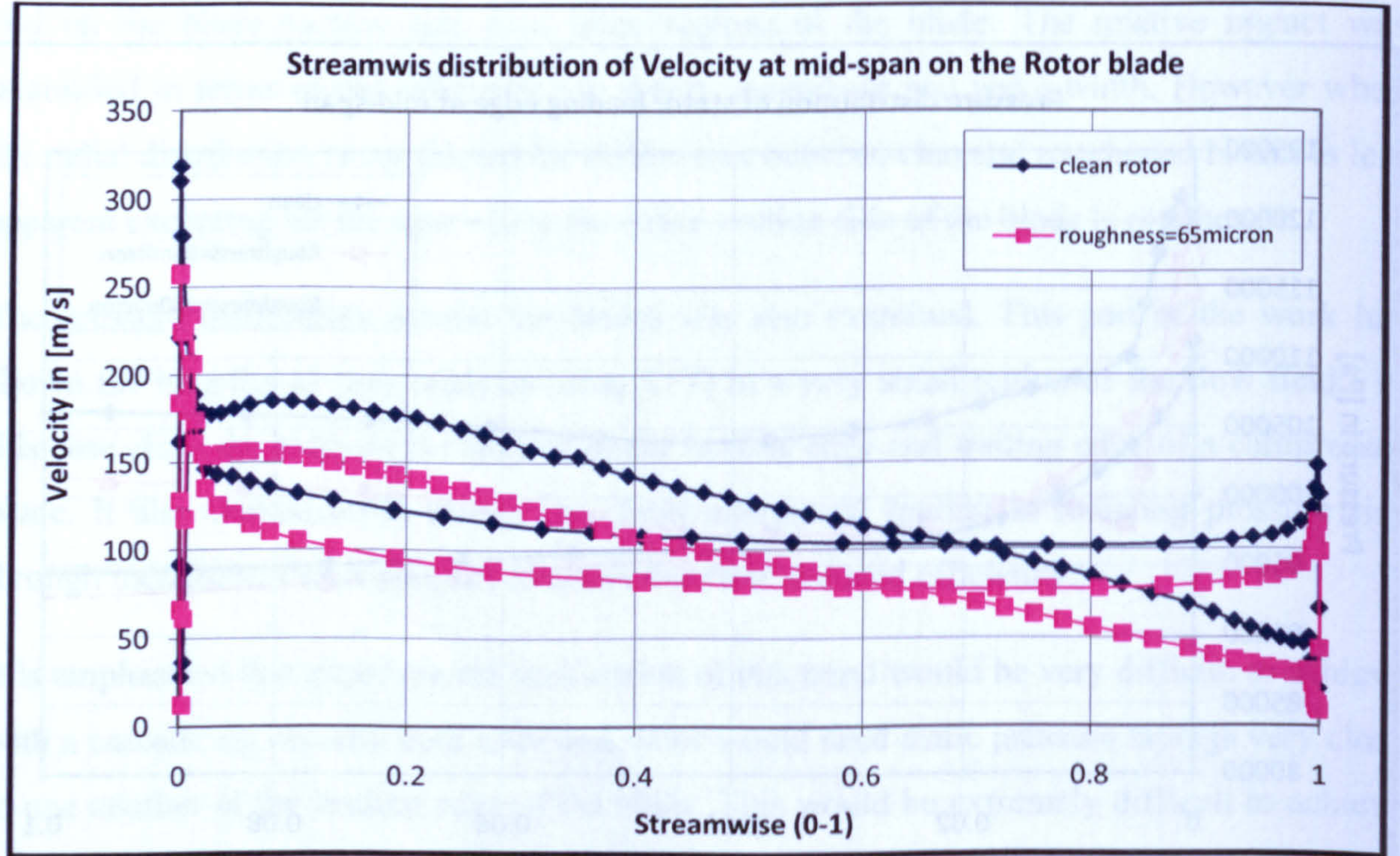


Figure 5-24: Velocity distribution on the Rotor blade

Chapter 6-CFD Simulation of Compressor Cascade Tunnel

6 CFD Simulations of compressor cascade tunnel

As explained earlier compressor fouling of an industrial gas turbine (GT) can result a substantial loss in power output and cycle efficiency. To recover performance the engine fuel flow could be increased, subject to the manufacturer's firing limitations, but this would reduce turbine blade creep life and result in large increases in GT engine operating costs. Investigating blade roughness can give valuable knowledge about the mechanisms of fouling and how this can be reduced.

The research reported this chapter concerns the design and commissioning of a 2-D cascade tunnel to facilitate estimation of pressure losses and associated compressor inefficiencies as a function of degree of fouling, as defined by the surface roughness. This chapter also presents preliminary 3-D CFD predictions for smooth clean cascade blades and for uniformly fouled blades with different levels of fouling. For all the cases considered, velocity distributions and pressure losses one chord downstream of the blades towards the cascade flow stream tube at a mid-span height are investigated. In addition, a version of the cascade tunnel geometry was simulated with an extended plenum chamber discharge volume. This was to check for anticipated improvements of the flow distribution downstream of the blades.

However, while the CFD analysis predicted that the wake becomes wider and the wake corrected velocity decreases as roughness increases, in practice the passage velocity actually increased slightly. Furthermore, one chord downstream of the blades the loss coefficient increases with increasing roughness. Increasing the volume of the plenum chamber attached to the original cascade tunnel improves the velocity and pressure distribution behind the blade trailing edges.

The CFD analysis reported here clearly shows are possible for compressor cascade tunnels to be driven by suction fans rather than the conventional blow-down facilities. However, in adopting suction fans, sufficient test bed space for exit plenum chambers must be made available. Early cascade experimental data strongly supports the validity of the CFD analysis.

6.1 Introduction

Fouling deposits on compressor blades cause an increase of blade surface roughness and boundary layer thickening which leads to a reduction of the blade passage flow area and air mass flow through the GT engine. Due to the drop in the air mass flow of the engine the compressor pressure ratio reduces and its isentropic efficiency falls. In the current study, the effect of surface roughness on the aerodynamic losses produced by compressor blades in a mildly compressible flow environment is considered for specified turbulence level and Reynolds number.

An understanding of the effect of levels of surface roughness in turbomachinery has been investigated by many researchers to improve GT performance when operating in fouled or eroded conditions. (Leipold, 2000) investigated the effect of surface roughness on the flow around a highly loaded compressor cascade by measuring blade profile pressure distribution and the local total pressures at the exit measuring plane of a smooth and a roughened blade for different inlet flow angles and inlet Reynolds numbers. They found that the surface roughness reduces slightly the separation bubble created at low Reynolds numbers and small inlet angles. Also, at higher Reynolds numbers roughness causes turbulent separation and high total pressure losses.

(Zhang, 2004) investigated the surface roughness effects on the aerodynamic performance of turbine blades for different inlet turbulence intensity levels of 0.9%, 5.5% and 16.2%. They found that changing the surface roughness condition, the effect on the integrated aerodynamic losses is significant, whereas changing the inlet turbulence intensity level the effect is small. The integrated aerodynamic loss magnitudes were determined by integrating profiles of exit free-stream stagnation pressure minus exit local stagnation pressure with respect to a normal coordinate measured from airfoil centerline in the traverse flow direction across the blade wake.

(Bammert K. a., 1980) carried out boundary layer development measurements in a cascade wind tunnel representing a section through the stator blades of a GT of 50% reaction. Testing smooth and roughened blades, the authors stated that surface roughness increases the friction factor and causes a shift of the transition point forward.

(Suder, 1995,) examined the performance deterioration of a high-speed axial compressor rotor by varying the airfoil surface roughness and thickness. They applied a smooth and a rough coating to different portions of the blade surface to determine the blade portions that

are most sensitive to thickness/roughness variations. At design speed, the authors found that the areas most sensitive to roughness are the leading edge and the front half of the suction surface. Based on flow field predictions generated using quasi and full 3-D Navier-Stokes codes they indicated that adding thickness to the rotor has a small impact in terms of performance. Blade roughness causes boundary layer thickening and passage blockage which reduces the diffusion process and the rotor pressure and temperature rise. Examining relative Mach number variations pitchwise at 15% chord distance behind the trailing edge at a blade height of 70% span, it was illustrated that an increase in the Mach number in the rear of the blade passage and in wake momentum deficit takes place as the roughness increases.

6.2 Cascade Tunnel

The cascade wind tunnel simulated through CFD is a suck down tunnel. Its cross-sectional area is 0.043m^2 and is designed for an air mass flow rate of approximately 5kg/s . This corresponds to an inlet Mach number of approximately 0.3 and a Reynolds number of approximately 3.8×10^5 . The test section comprises nine two-dimensional blades set at zero incidence with a span of 180mm and a chord length of 60mm. The pitch to chord ratio, s/c , is 0.8 and the pitch “s” is 4 mm. The blade stagger angle ζ is 36° deg and the blade camber angle θ is 30° .

The blades have a span to chord ratio (aspect ratio) of 3. The selection of the particular number of blades and aspect ratio has been made under consideration of the wall boundary layer effects. In a compressor cascade tunnel the increase in static pressure across the blades causes wall boundary layer thickening and contraction of the flow. Due to the contraction only of the flow the fluid accelerates. This contrasts with the diffusing action of the compressor cascade itself. In consequence, the sidewall boundary layer effectively reduces the static pressure increase that theoretically would be achieved. To reduce this effect (Dixon, 1998) and many others suggest that at least seven blades are needed for compressor cascade testing and that each blade has a minimum aspect ratio of three. (Pollard, 1967) have run some experiments with two low speed compressor cascade tunnels, one having an aspect ratio less than 3 and applying sidewall boundary layer suction and one with solid walls having a high blade aspect ratio greater than 3. The authors state that if porous sidewalls are used to control the sidewall boundary layer, a low aspect ratio will give good 2-D results. However

for the case of solid side wall, two-dimensionality of the flow can be obtained using aspect ratios equal or greater than 3. In parallel, their research shows that higher levels of free stream turbulence enhances earlier boundary layer transition and delays the increase in loss associated with a significant reduction in Reynolds number.

The cascade test section of the current project has nine two-dimensional blades of NACA 65 thickened profile with circular arc camber line. Figure 6-1 illustrates the corresponding cascade tunnel arrangement. A plenum chamber was installed behind the cascade test section. The ideal cascade experiment would discharge the airflow from the blading into an infinite space (e.g. the atmosphere). In the current project however, the facility employs a suction pump (fan). Accordingly the blading has been designed to discharge into the largest practical volume feasible to ensure a uniform exit static pressure. In addition, it is equally important, that the effect of the plenum chamber shape itself and its exit ducting to the fan does not distort the cascade discharge flow.

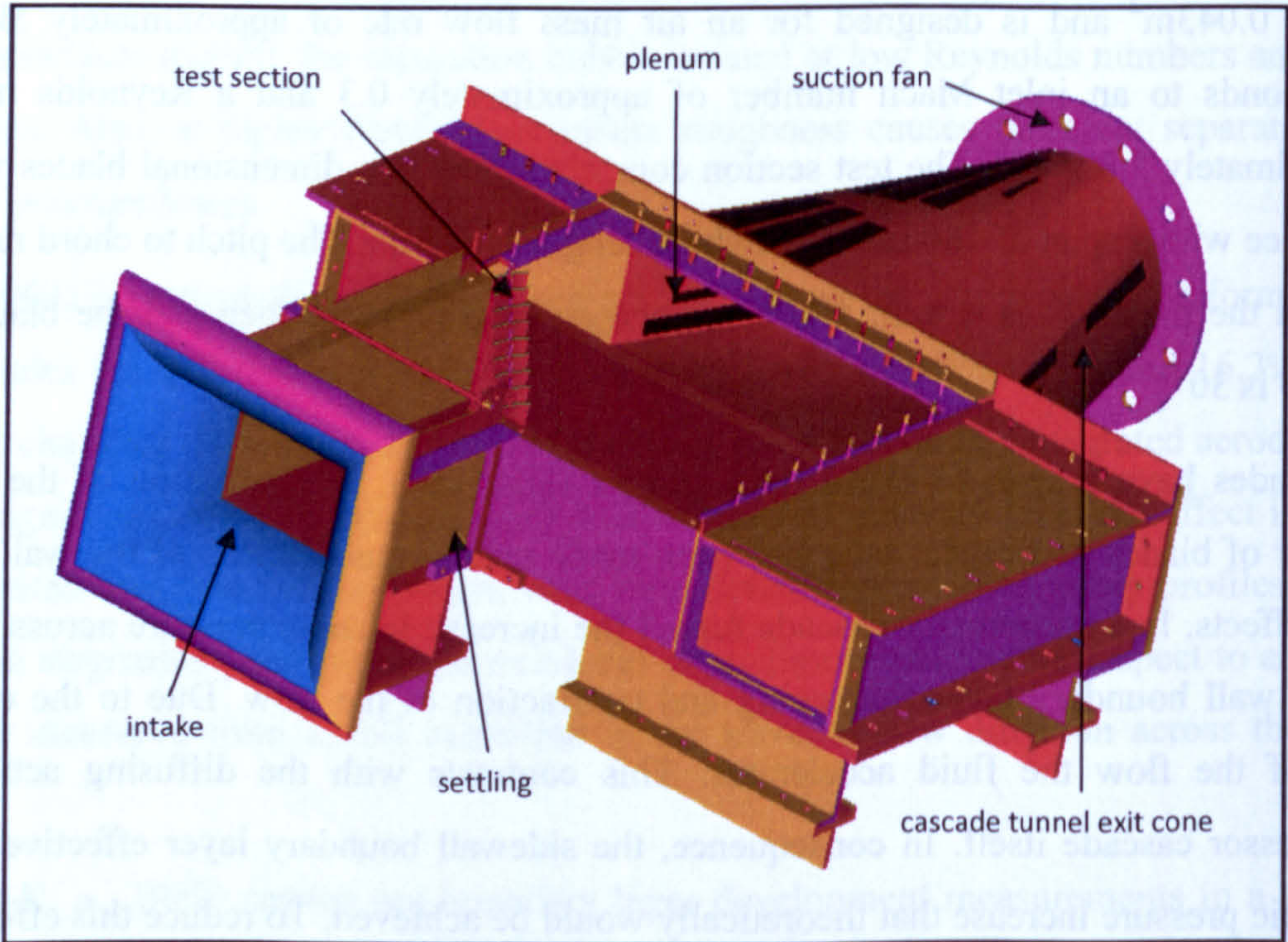


Figure 6-1 : Cascade tunnel design iso-view

6.3 Roughness and Measurement Parameters

The central three cascade blades are to be roughened by covering their surfaces with very thin double-sided sticky tape. In fact this does not significantly increase the thickness of the profile. Subsequently, carborundum grit is applied uniformly to roughen the blade surfaces of the central 3 blades. The carborundum grain sizes are defined with emery grade numbers which represent different sieve sizes through which the grains will pass. The grit numbers involved are 60, 120, 180 and 220 related to grain sizes of 254, 102, 76 and 63 microns. These correspond to K/c values of 0.0042, 0.0017, 0.0013 and 0.0010, respectively. The roughness height K is assumed to be equal to the average size of the carborundum grains.

CFD prediction measurements of blade total pressure loss coefficient ω in the area of the wake were undertaken for the case of smooth and roughened blades for different carborundum grit sizes. Traverse planes (see Figure 6-2) and lines at a mid-span height were created in Fluent Inc. software one chord upstream and one chord downstream of the blades. Stream tube vectors were evaluated to monitor the predicted pressures.

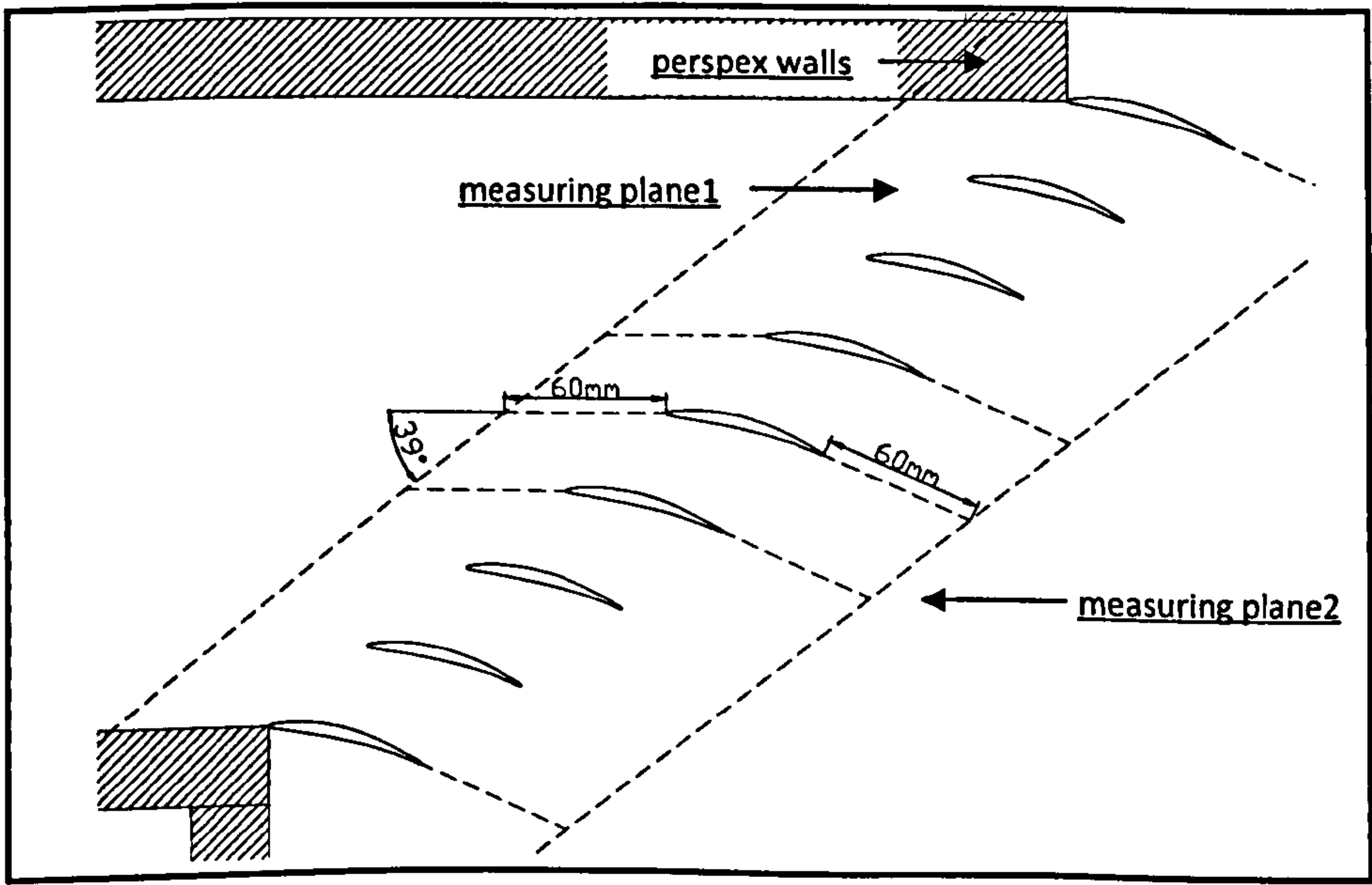


Figure 6-2: Cascade test section

The blade profile total pressure loss coefficient is obtained through analysis of the pressure data from the front and downstream blade traverses.

$$\omega = \frac{P_1 - P_2}{P_1 - p_1} \quad (1)$$

According to (Saravanamuttoo H. H., 1996), test results for subsonic compressor blade sections over a range of Mach numbers shows that at low Mach numbers, the losses at zero incidence are very low compared to those at high incidence and high Mach number. The CFD results reported here are at a Mach number of 0.3 and 0.4 and at a nominal incidence of zero. Accordingly, the corresponding losses are expected to be low and depend only secondarily on blade thickness. Clearly however, at high Mach number and incidence far away from zero, as expected the losses increase dramatically. In this case, blade thickening due to roughness significantly increases the pressure losses.

6.4 CFD Modelling

A computational solution using a commercial CFD package (Fluent 6.3) is used for the compressor cascade geometry of the current study. The simulation is 3-D and steady-state. The fluid volume of the whole cascade tunnel has been meshed with a combination of both structured and unstructured grids which are created using the pre-processor Gambit 2.4.

The SST $k-\omega$ model and the standard version of the $k-\epsilon$ turbulence model is used for the simulation but due to divergence detected in terms of the $k-\omega$ model when applying roughness, the $k-\epsilon$ model is selected. The internal passage grid of the cascade geometry is illustrated in Figures 6-3 and Figure 6-4. It consists of an H-grid with an extra O-grid around the cascade blades containing the boundary layer region. The O-grid around the cascade blades contains 176 x 50 grid points. The grid in the tunnel exit cone and in the area (settling chamber) between the inlet and the test section is unstructured.

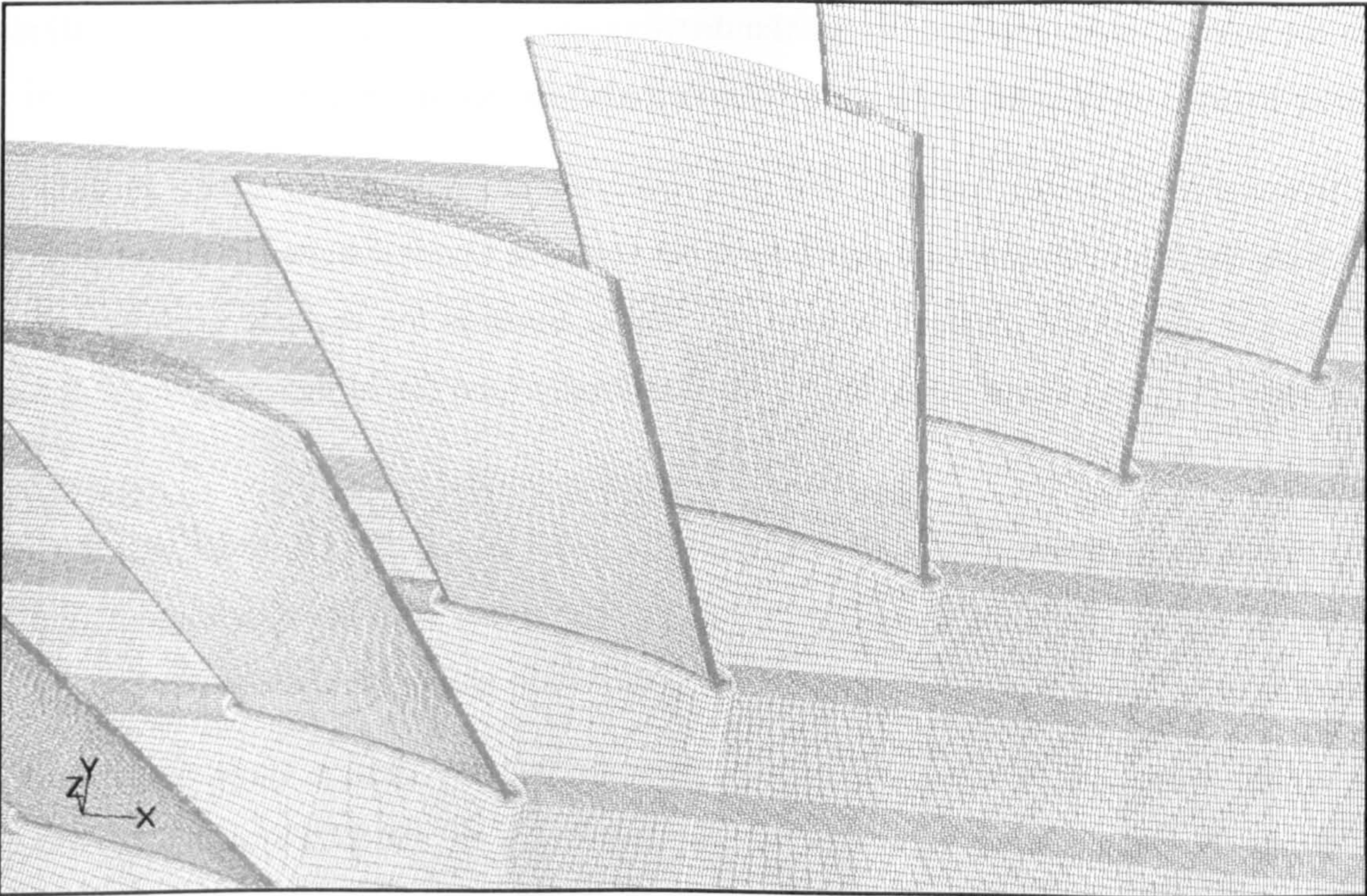


Figure 6-3: Computational mesh

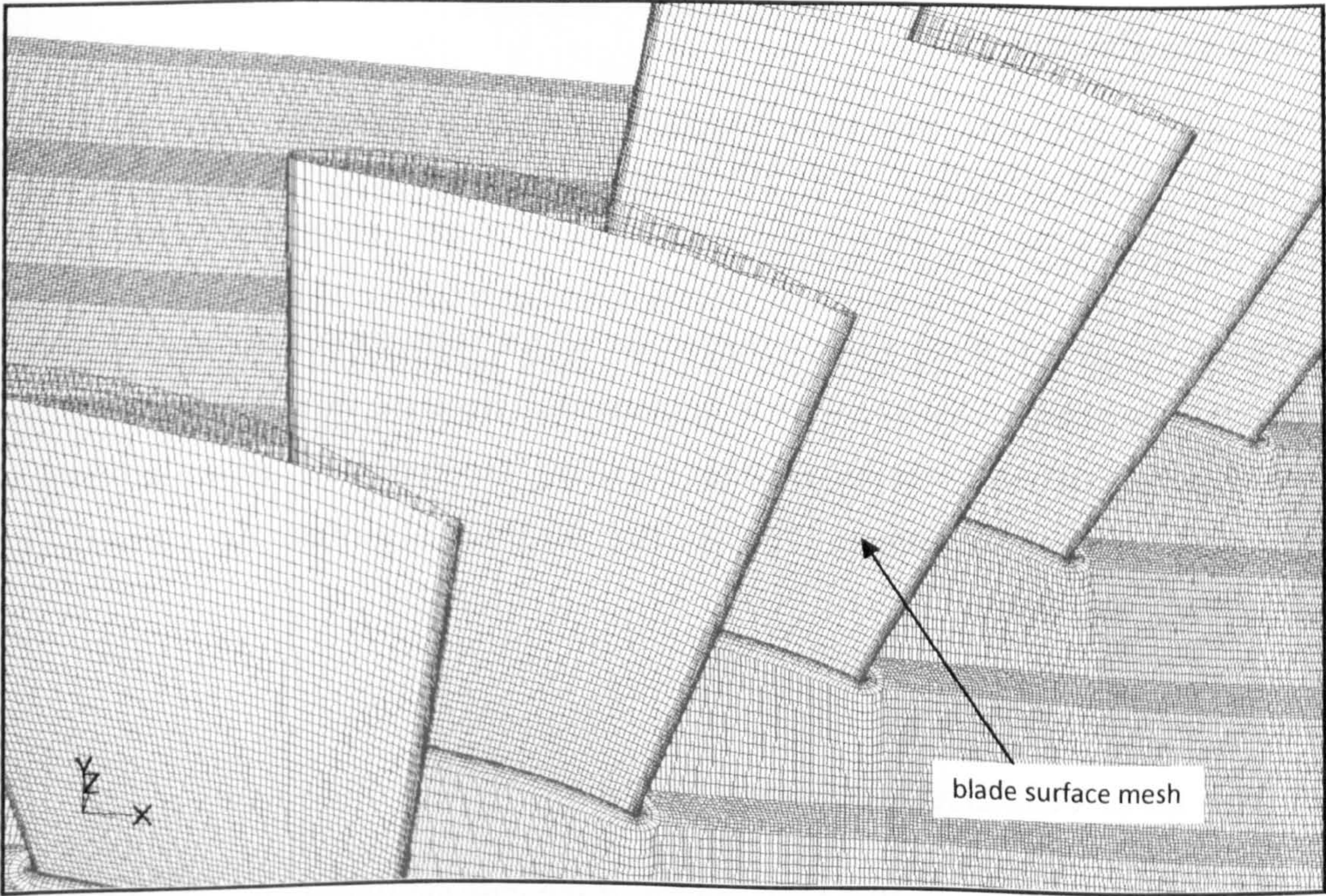


Figure 6-4: Blade surface meshing

For the smooth un-roughened blades the distribution of points in the near blade wall region is denser in order to resolve the boundary layer in the viscous sub-layer region.

Different meshes were examined and grid independency is achieved after 9 million cells by checking the total pressure loss coefficient between one chord distance in front of the cascade blades and one chord distance behind the blades towards the cascade outflow vector. The final grid involving about 10 million cells was finally used for further analysis of the cascade flow.

In order to simulate roughness effects the finest grid was modified in order that the centroids of the cell nodes close to the walls are at a distance higher than the roughness height. This is assumed to be the size of the particles uniformly spread along the blade surfaces.

An initial computation corresponding to the preliminary design stage was run with the following boundary conditions: a total pressure inlet of 101592Pa and total temperature 293.15K. The turbulence intensity was assumed to have a low value, 1%, with the length scale set at 1.8mm. The outlet of the cascade was treated as the pressure outlet boundary with a static pressure of 95583Pa. This assumes that the flow between the inlet and the outlet of the cascade is adiabatic.

All the other CFD runs employed boundary conditions extracted from the individual experiments that were carried out as part of this project. However the boundary conditions combinations remained as described above.

A second order discretization scheme is used for all the simulation including the turbulence parameters. The pressure based solver is implemented since the flow is initially assumed to be mildly compressible. Extra information regarding the numerical solver is found in the Fluent user's guide (Anonymous, 2007,). The SIMPLE algorithm for pressure-velocity coupling is also employed.

6.4.1 The CFD Velocity Analysis

Figure 6-5 illustrates the potential flow region of the cascade tunnel on a plane crossing the tunnel area at mid-span. A quite uniform wake distribution behind the smooth blades is obviously indicated. On checking the velocity distribution contours, a decrease from approximately 145m/s ($M=0.4$) to 105m/s is predicted as the flow propagates from the cascade test section inlet (one chord upstream of the blades) towards the exit of the cascade

(one chord downstream towards the exit flow stream tube). This implies a De Haller number which is the ratio of cascade blade exit flow velocity over inlet flow velocity, greater than 0.7, which is acceptable for designing a compressor cascade (Ramsden K. , 2006).

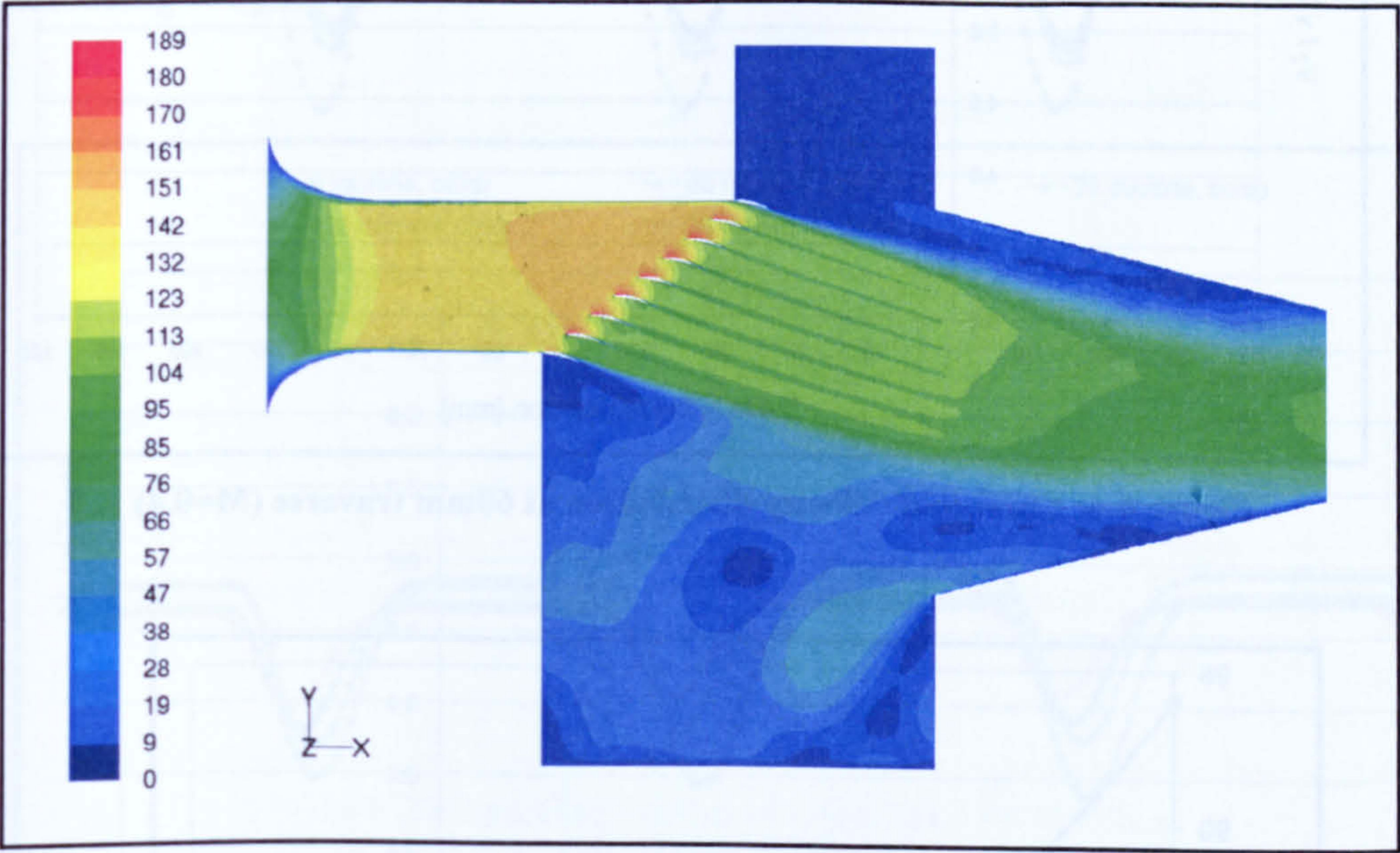


Figure 6-5: Velocity contours

Applying roughness to the three middle blades of the cascade one can see from Figures 6-6 and 6-7 the results emerging from a 60mm traverse line towards the exit cascade stream tube taken at a mid-span height. As the roughness particle size increases from 0 to 254 microns the corrected velocity (V_2 / T_{amb}) drops from 5.3 to 4.4 respectively and the wake velocity drops continuously but non-linearly.

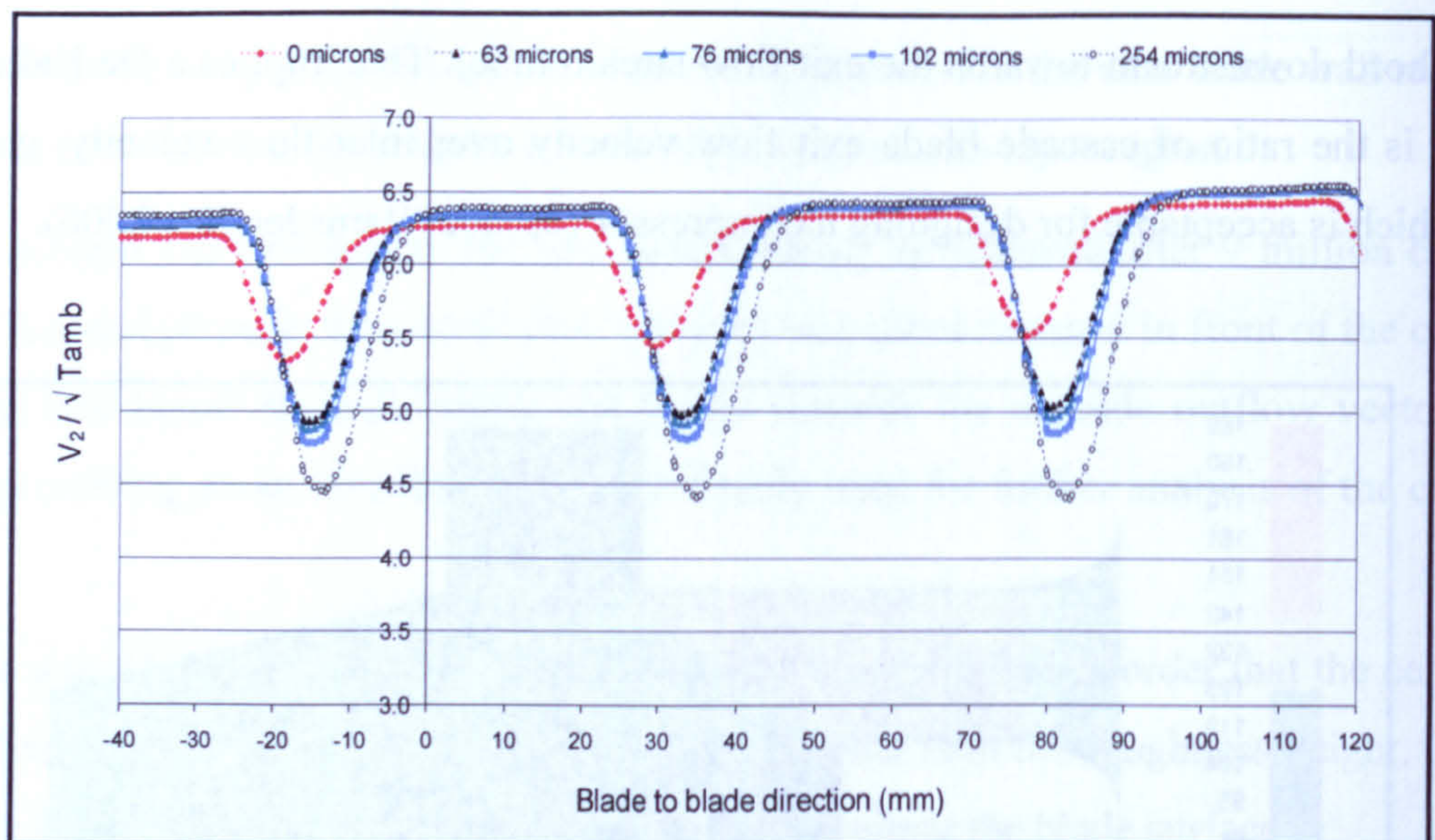


Figure 6-6: Corrected velocity distribution at 60mm traverse (M=0.4)

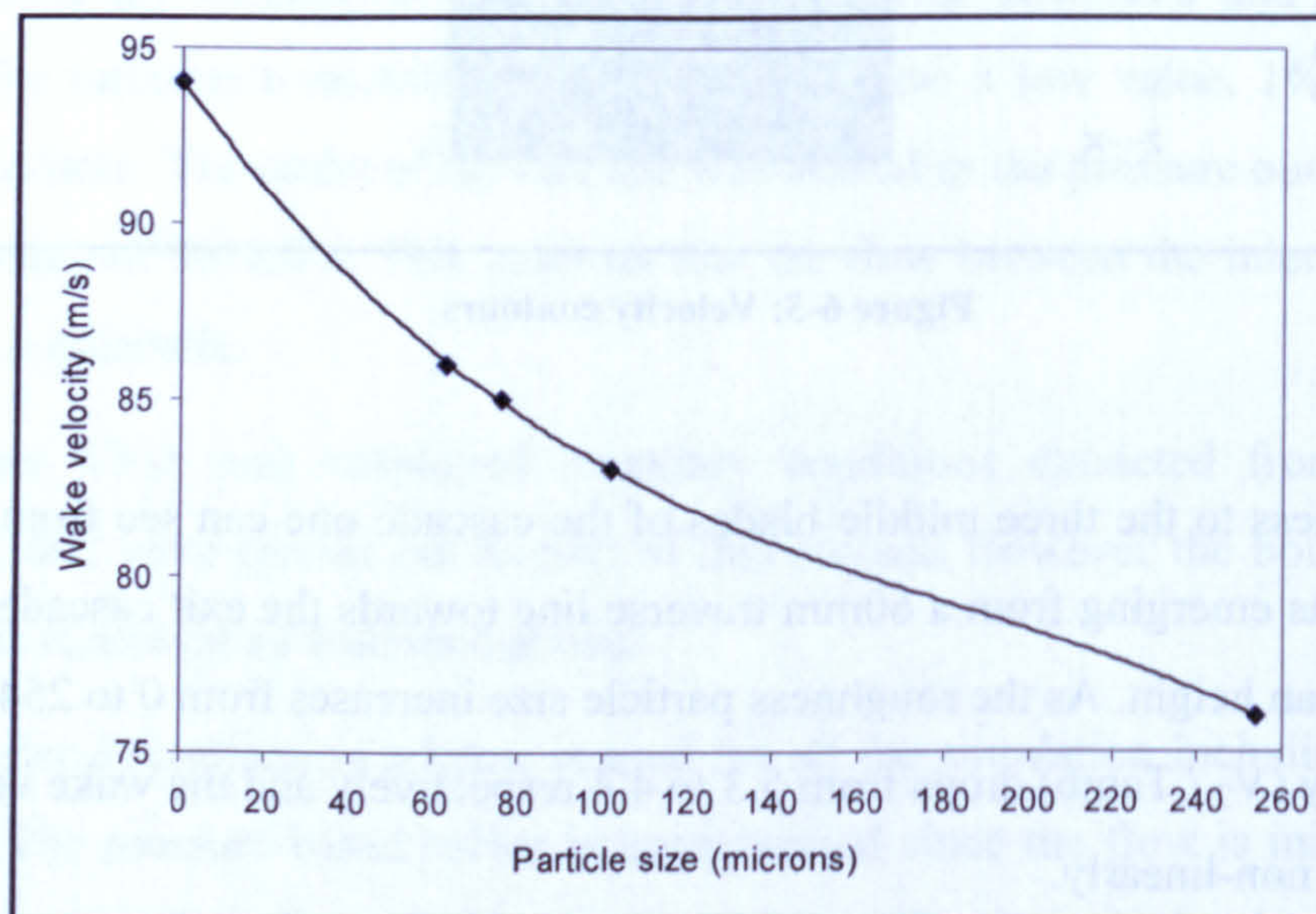


Figure 6-7: Wake velocity deterioration at 60mm traverse (M=0.3)

Results corresponding to cascade inlet Mach number 0.3 are shown in Figure 6-8. As the diameter of the particles increase gradually from 63 to 254 microns, the wakes behind the blades widen and the wake velocities decrease significantly compared to the existing passage velocities. For smooth blades the wake corrected velocity drops to about 4.0 relative to the passage corrected velocity of 4.7.

As particles size and therefore, roughness increases, the magnitude of the velocity in these passages increases. This is due to the reduction of the passage flow area maintaining continuity. In the worst case of 254 microns the corrected velocity in the wake falls to a value

of 3.43 from a value of 4.8 in the passage and this passage value was found to be in agreement with the one corresponding to the experimental work related to this project. In fact, as a result of increasing the particle size, the velocity increase in the potential core flow is significantly less than the velocity decrease of the wake. This difference increases with further increasing roughness.

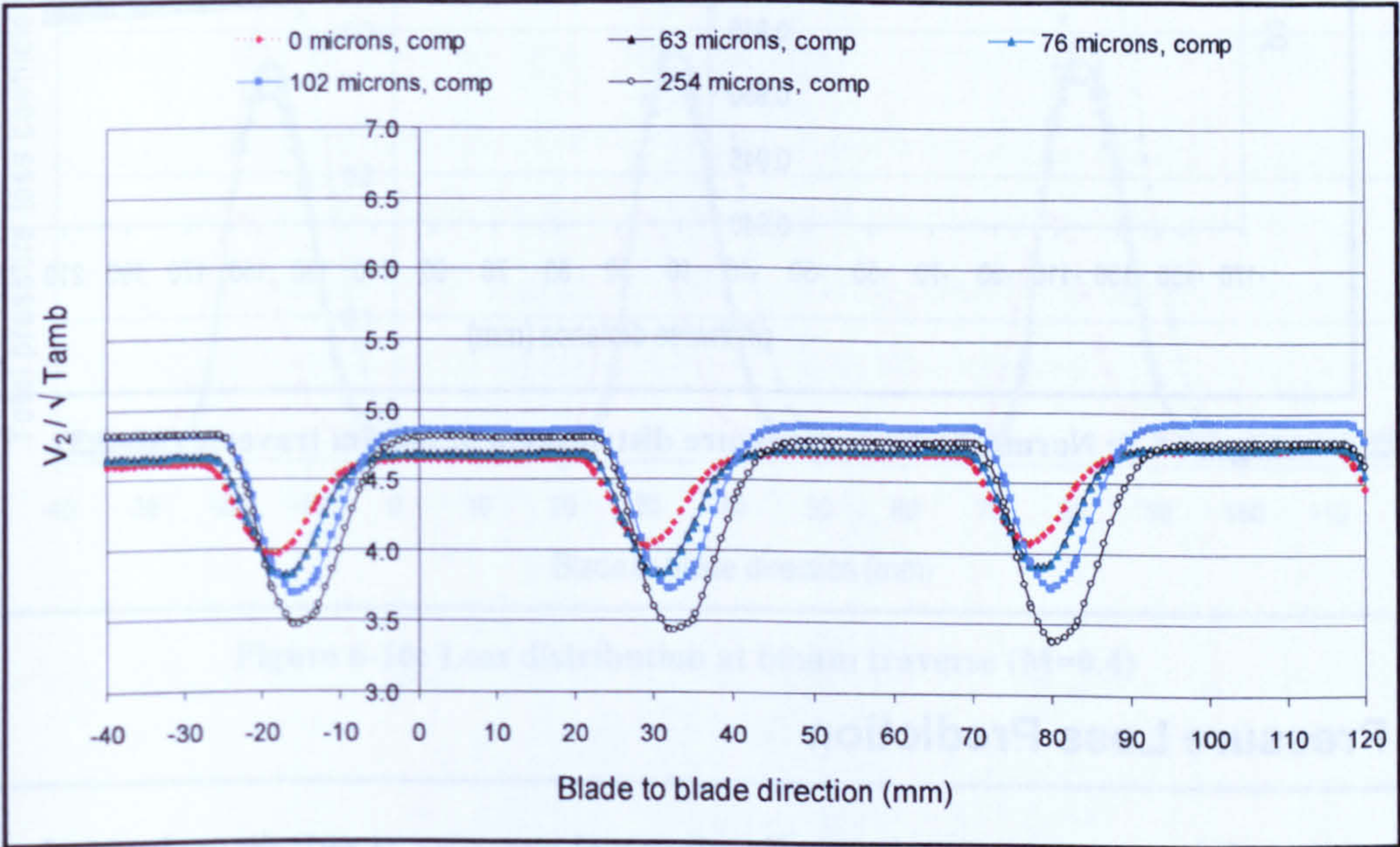


Figure 6-8: Corrected velocity distribution at 60 mm traverse (M=0.3)

At the trailing edge of the smooth blades, the boundary layer thickness is larger on the suction surface than on the pressure surface. Accordingly, the total wake thickness at the trailing edge is always biased to the suction surface side and there is movement of the roughened blade wakes to the right behind the blades (towards the trailing edge) as the roughness increases.

The fact is that at cascade exit, there is a small pitch-wise reduction in static pressure predicted. This is shown in Figure 6-9. This will, of course, cause the upstream flow to deviate towards the lower static pressure region and effectively progressively increase the values of exit flow angle α_2 at exit from the blades in the pitch wise direction. The pitch-wise pressure gradient existing behind the blades can be attributed to vortices existing at the right and left part of the cascade plenum chamber enhancing and retarding the flow.

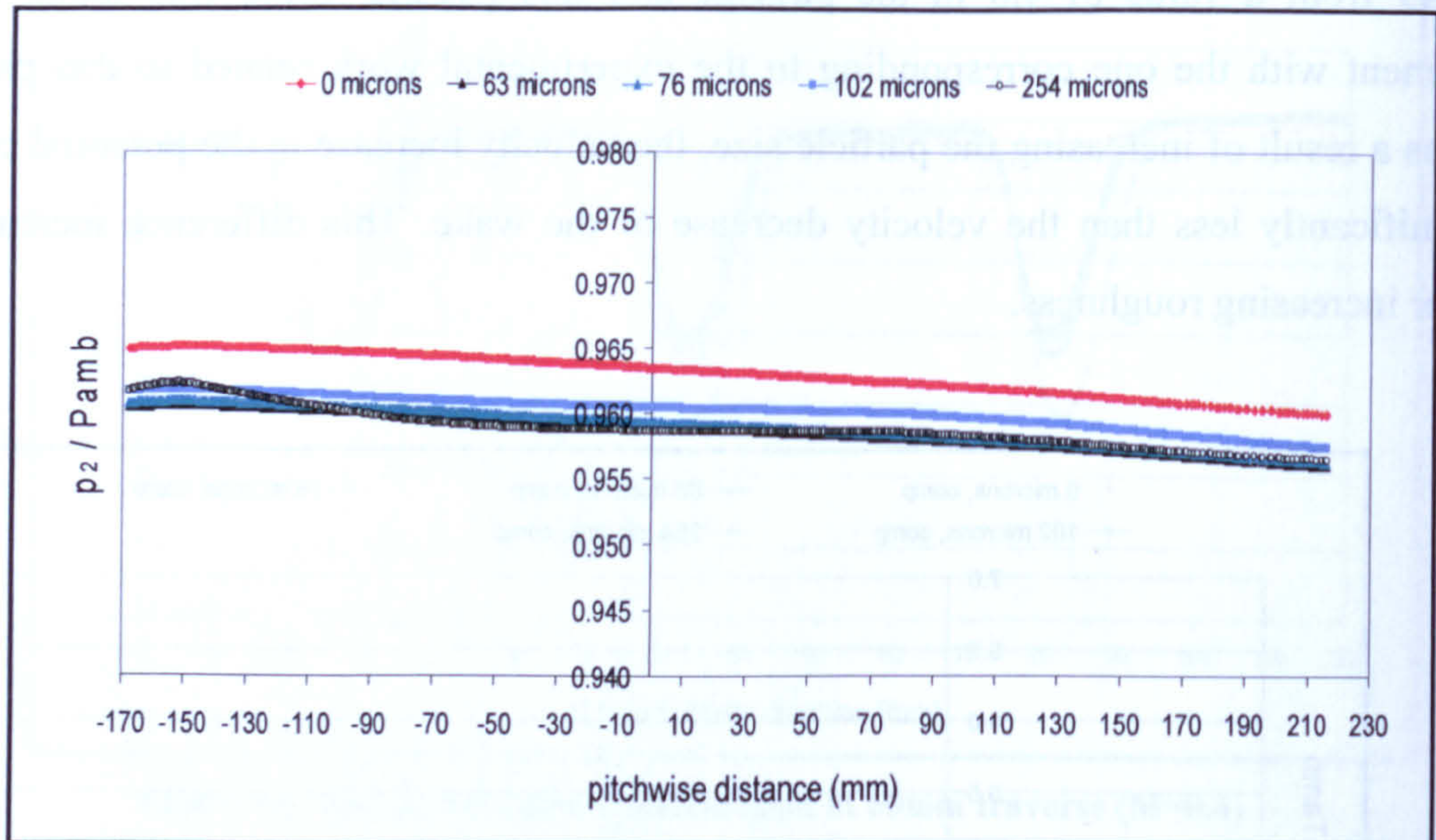


Figure 6-9: Normalized rear pressure distribution at 60 mm traverse ($M=0.3$)

6.4.2 Pressure Loss Prediction

The Fluent model cannot accurately predict the total pressure at exit from the blade row in the potential flow. The model does predict, however, a small total pressure rise in the potential flow between blade rows. This cannot, of course, be the real case. The reason for this has not been established. Accordingly the zero line in Figure 6-10 has been adjusted to correspond to no change in total pressure in the potential region across the cascade blade row (the flow in this region is assumed to be isentropic). From this figure it can be seen that the total pressure loss coefficient ω is predicted to increase as particle size increases. Loss increases were also reported by (Kind, 1998) as a function of increasing roughness height to chord ratio (K/c) of 0.0021, 0.0045 and 0.0063 at design incidence.

In Figure 10, following the increase in particle size from 0 to 254 microns, the total pressure loss coefficient increases from 0.16 to about 0.36, respectively. The loss increase in the blade passage is not significant between smooth clean blades and fully roughened blades as well. The total pressure loss distribution for inlet cascade Mach number of 0.3 (see Figure 6-11) got almost similar values with that corresponding to Mach number of 0.4 as far as the fouling level of 0 and 254 microns is concerned. However, the cases of 63, 76 and 102 microns were found to be higher in terms of total pressure loss coefficient for inlet Mach number of 0.4.

Experimental work done validated the results in terms of loss corresponding to inlet Mach number of 0.3.

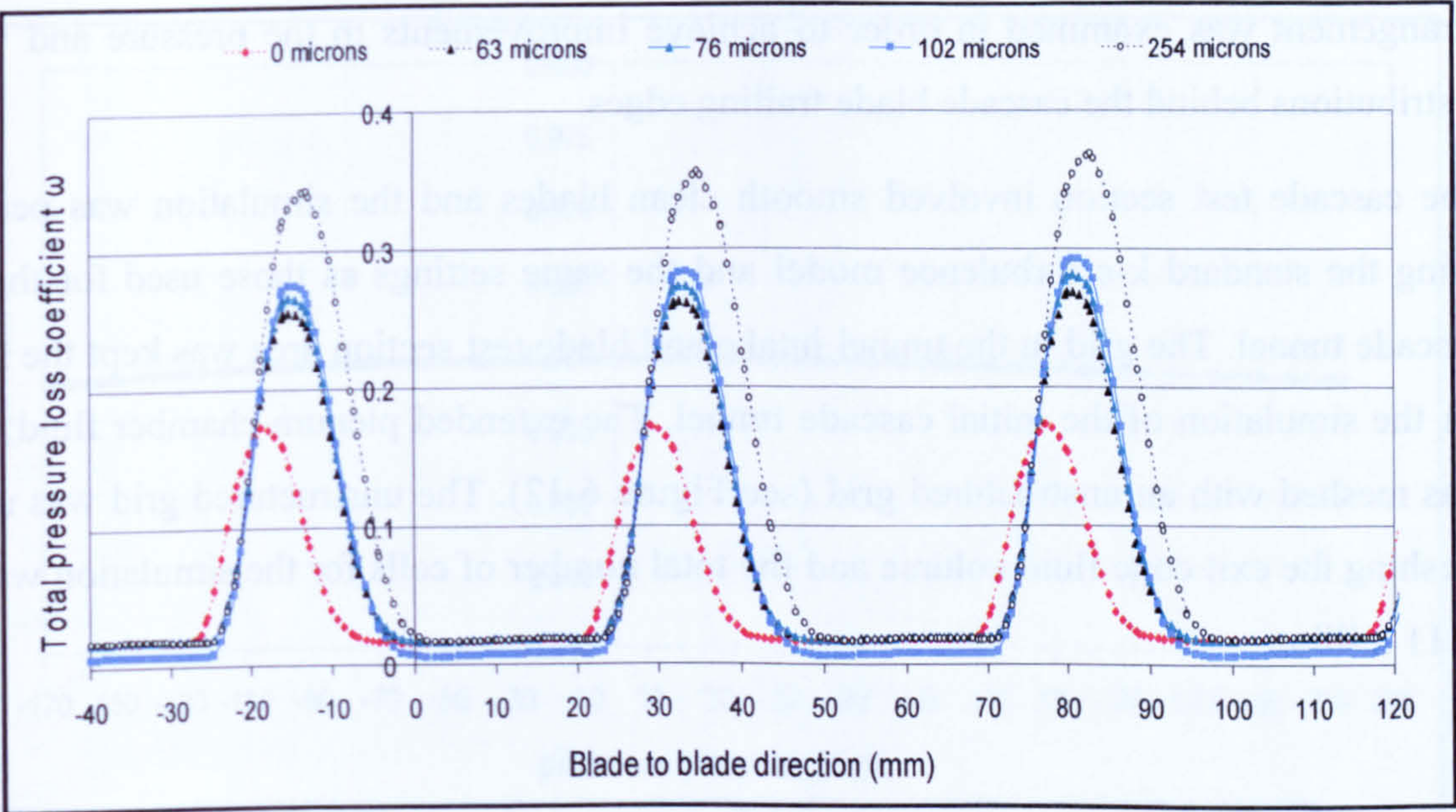


Figure 6-10: Loss distribution at 60mm traverse (M=0.4)

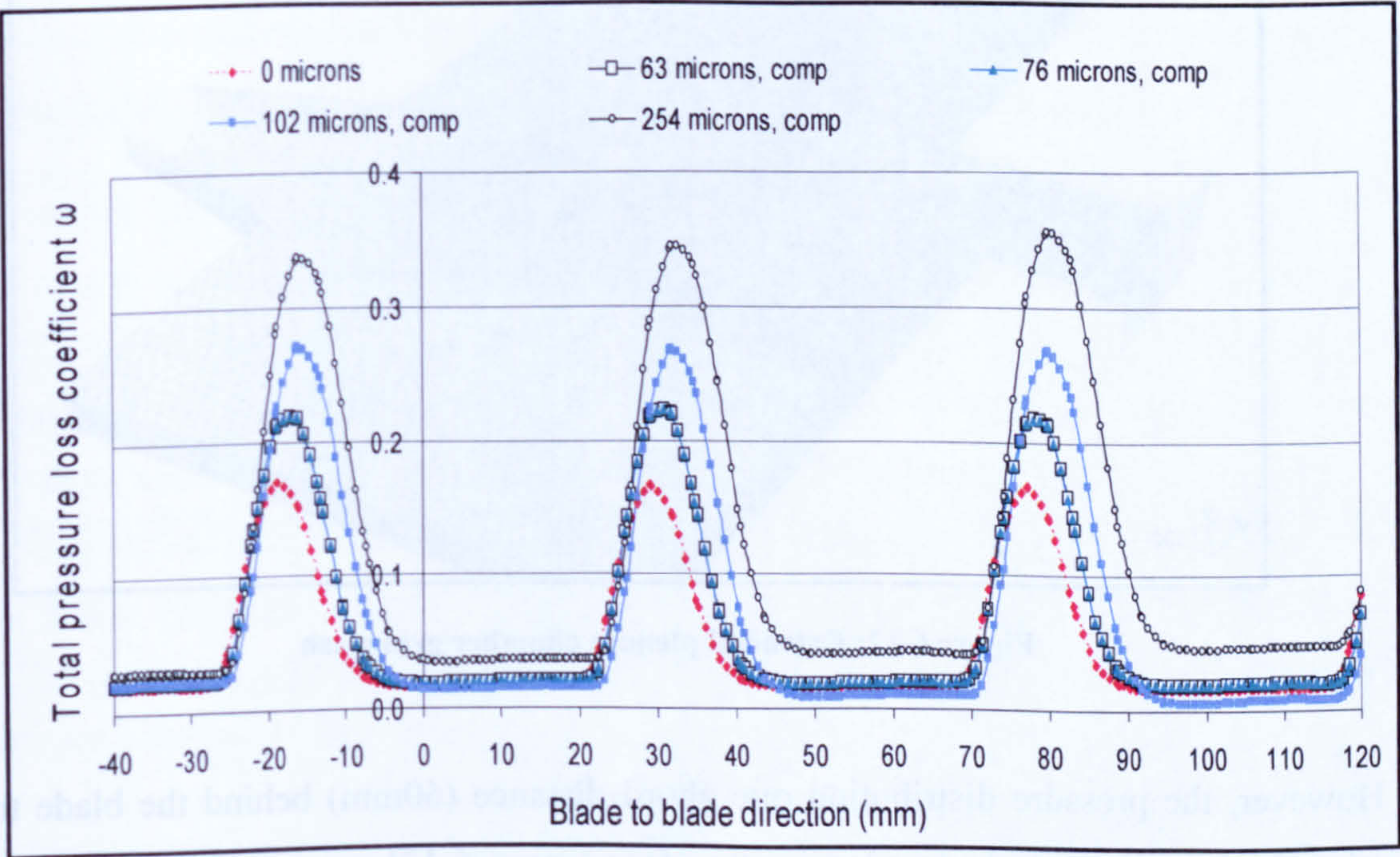


Figure 6-11: Loss distribution at 60mm traverse (M=0.3)

6.4.3 CFD Predictions of Plenum Chamber Size Increase

Based on the initial cascade tunnel geometry CFD analysis, a modified cascade tunnel with a larger plenum chamber was simulated having side walls extended by 1 meter. This arrangement was examined in order to achieve improvements in the pressure and velocity distributions behind the cascade blade trailing edges.

The cascade test section involved smooth clean blades and the simulation was performed using the standard $k-\epsilon$ turbulence model and the same settings as those used for the initial cascade tunnel. The grid in the tunnel intake and blade test section area was kept the same as for the simulation of the initial cascade tunnel. The extended plenum chamber fluid volume was meshed with an unstructured grid (see Figure 6-12). The unstructured grid was used for meshing the exit cone fluid volume and the total number of cells for the simulation was close to 11 million.

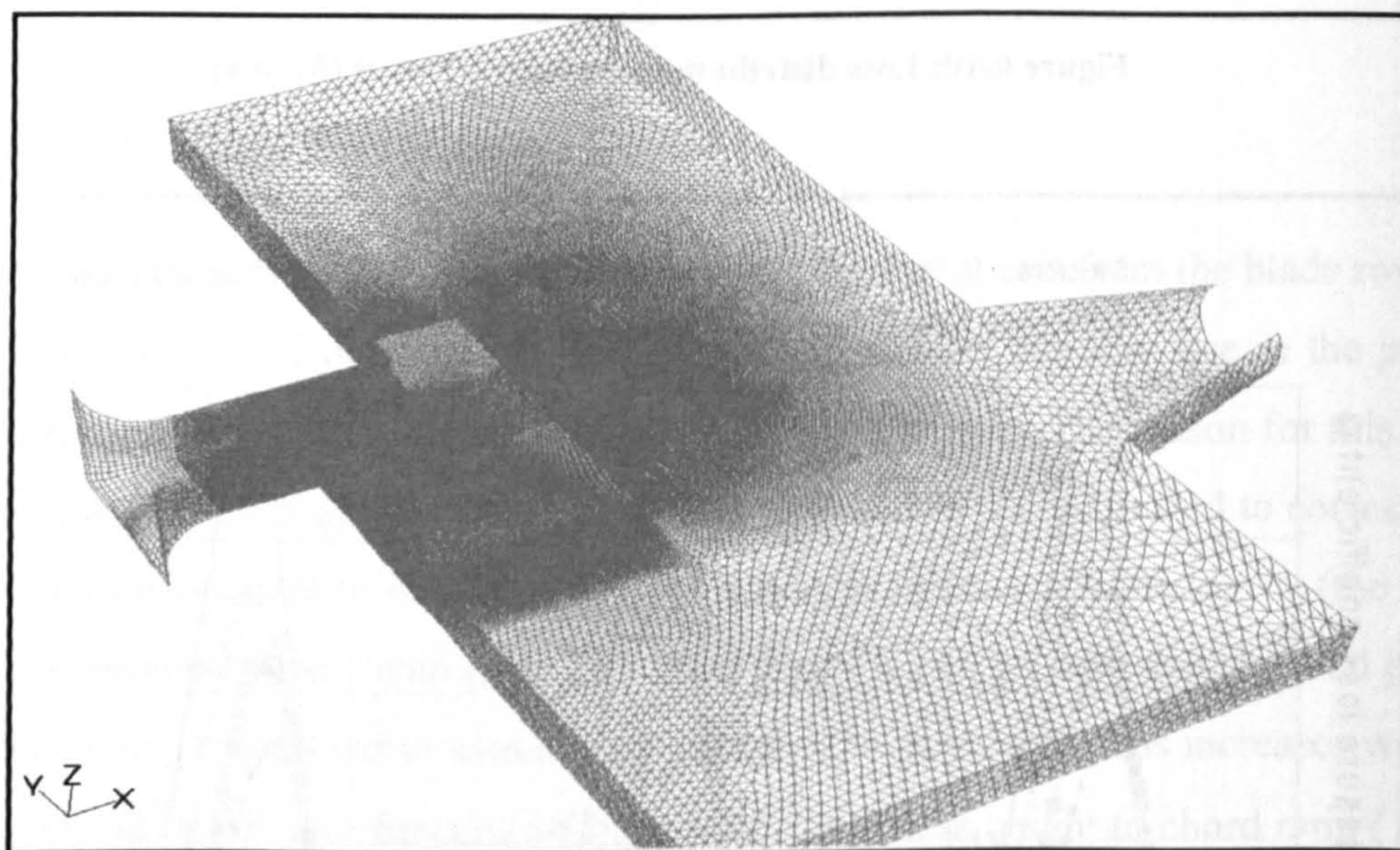


Figure 6-12: Extended plenum chamber grid mesh

However, the pressure distribution one chord distance (60mm) behind the blade trailing edges was predicted to be much smoother (see Figure 6-13) compared to those related to the initial cascade tunnel. At the rear traverse the velocity distribution was predicted to be especially uniform with an almost constant corrected velocity value of 4.8 undisturbed by the plenum chamber vortices referred to above. This extended plenum chamber

arrangement seems to provide better cascade flow results but the associated space limitations prevented its adoption.

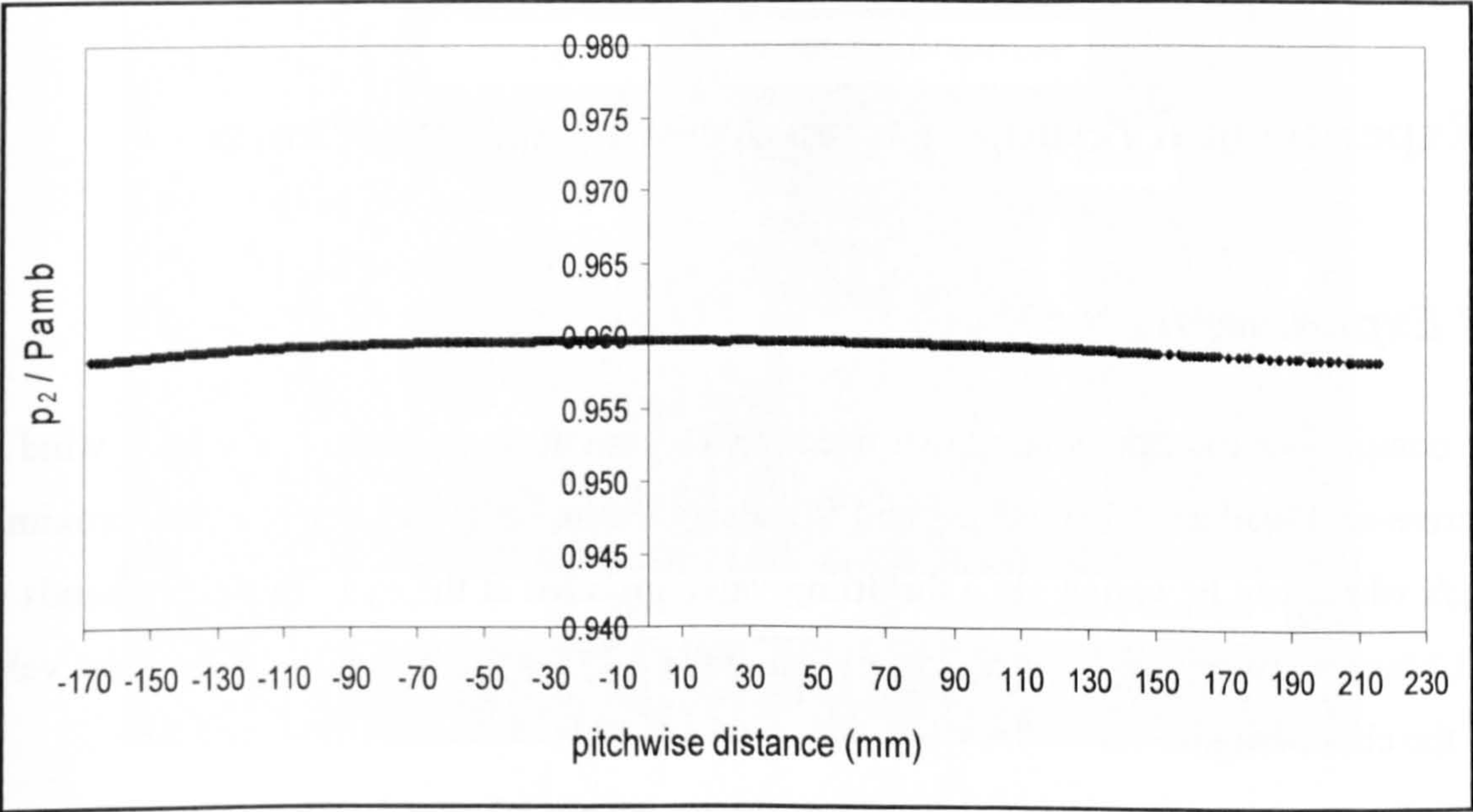


Figure 6-13: Extended cascade tunnel rear normalized pressure distribution (M=0.3)

Chapter 7 – Experimental Work of Online Compressor Cleaning

7 Experimental results of Compressor Cleaning Online

7.1 Experimental facility

The compressor cascade wind tunnel used for the current experiments is a suction wind type, its cross-sectional area is 0.043 m^2 and it is designed for air mass flow rate of approximately 5 kg/s which can be varied via a throttling valve installed at the exit. This corresponds to an inlet Mach number of 0.3 and a Reynolds number of 3.8×10^5 formed with the inlet velocity and the chord length.

The degree of turbulence and the mean velocity of the inlet air flow were measured with a hot wire anemometer as 2.245% and 52.6 m/s respectively. The test section of the cascade comprises nine two-dimensional blades of NACA 65 thickened profile with circular arc camber line. The blades are set at zero incidence, the blade span is 180 mm, the chord length is 60 mm and the pitch to chord ratio s/c is 0.8. The cascade blades represent a stator blade mean section of a gas turbine compressor with 50 percent reaction.

The blades have a span to chord ratio (aspect ratio) of 3. The selection of the particular number of blades and aspect ratio has been made under consideration of the wall boundary layer effects. In a compressor cascade tunnel the increase in static pressure across the blades causes wall boundary layer thickening and contraction of the flow. Due to the contraction only of the flow the fluid accelerates. This contrasts with the diffusing action of the compressor cascade itself. In consequence, the sidewall boundary layer effectively reduces the static pressure increase that theoretically would be achieved.

Figure 7-1 illustrates the corresponding cascade tunnel arrangement. A plenum chamber was installed behind the cascade test section. The ideal cascade experiment would discharge the airflow from the blading into an infinite space (e.g. the atmosphere). In the current project however, the facility employs a suction pump (fan). Accordingly the blading has been designed to discharge into the largest practical volume feasible to ensure a uniform exit static

pressure. In addition, it is equally important, that the effect of the plenum chamber shape itself and its exit ducting to the fan does not distort the cascade discharge flow.

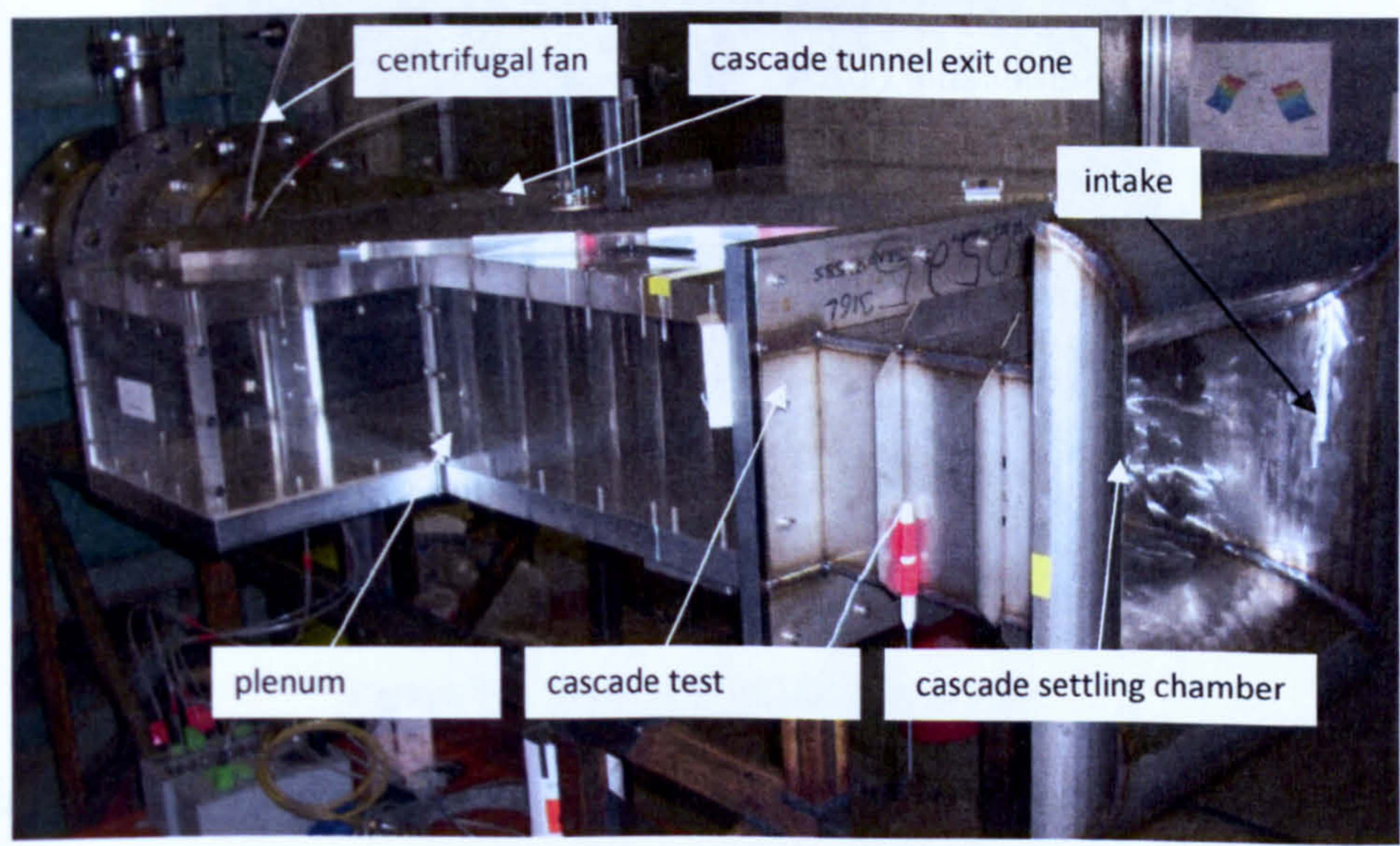


Figure 7-1: Actual cascade tunnel left side view

7.1 Cleaning fluid and injection nozzle characteristics

The cleaning fluid that is going to be used for the experimental work is the R-MC Power Guard that is water based cleaner diluted in a ratio of 1:4 with demineralised water. The density of the mixture of the cleaning fluid and water was calculated and it was found to be 992 kg/m³.

This particular fluid was used for compressor on-line washing of an Alstom Typhoon industrial gas turbine having ten compressor stages, compression ratio of 14.2, air inlet mass flow 18.7 kg/s and producing 4.7 MW of power. The washing system involved for the compressor cleaning of this gas turbine run at an operating pressure of 90 bar for 4 minutes comprising 6 flat fan spray nozzles delivering 1.25 lt/minute each (see figure 7-2). The cleaning mixture of fluid and water was heated at a temperature of 50°C as well.

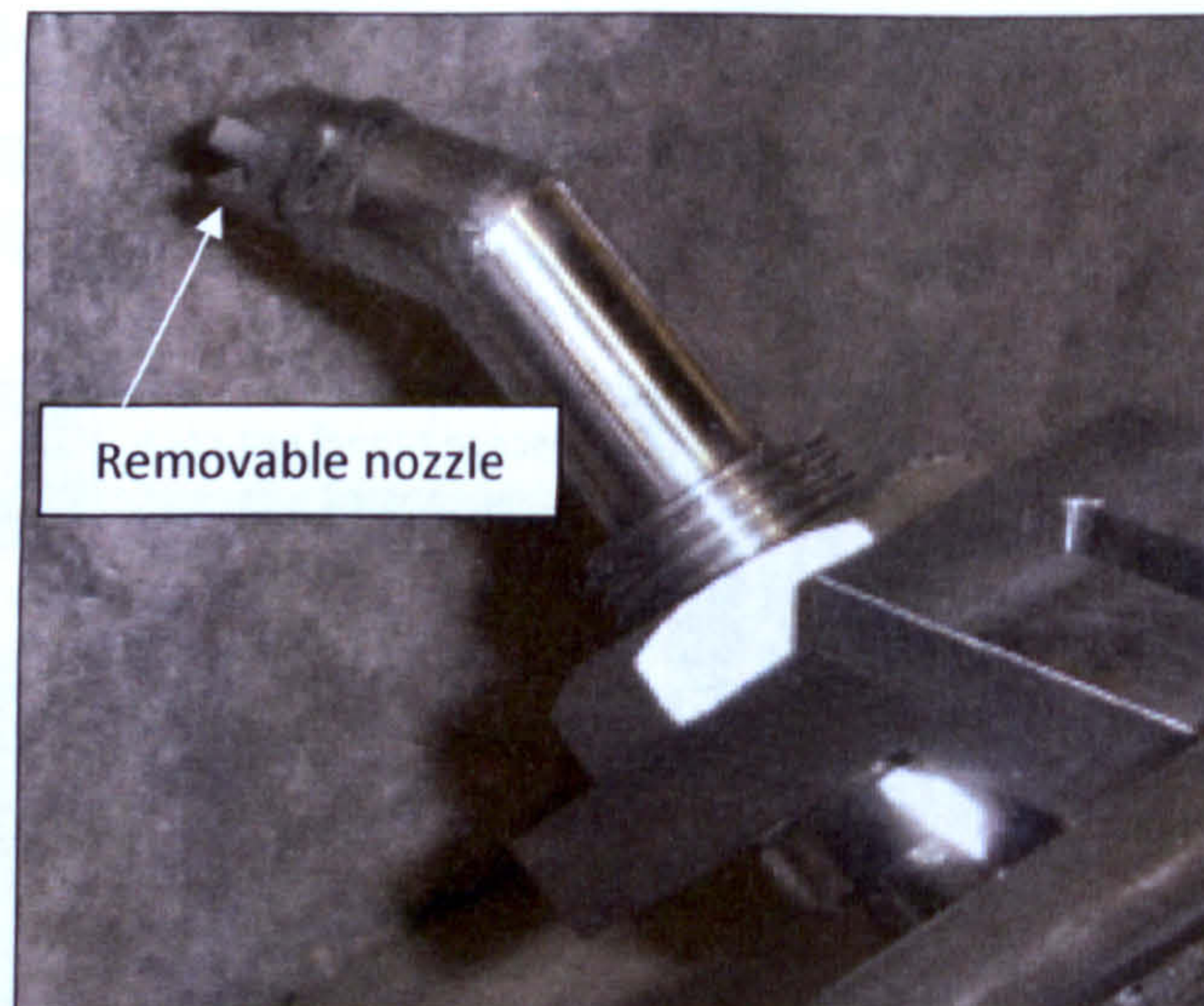


Figure 7-2: Flat fan spray nozzle.

Each of these injection nozzles used for the Alstom Typhoon compressor cleaning has an orifice width of 0.78mm and orifice height of 0.33mm capable of producing under operating pressure of 90 bar spray width angle of 85 degrees, spray height angle of 42 degrees and droplet diameters of between 50 and 150 μm . The washing mixture was heated to a temperature of 50°C in order to obtain more effective cleaning.

For the current experimental cascade washing a smaller nozzle tip having orifice height 0.2 mm and orifice width 0.51 mm was used. The nozzle tip was attached on the water metallic tube of the above flat spray nozzle providing a fluid mass flow rate level of approximately 0.0117 kg/s. This corresponds to a ratio of fluid mass flow rate over cascade tunnel air of 0.2 %. The same ratio was used for cleaning the compressor of an industrial gas turbine engine producing 165 MW of power and having an intake air mass flow rate of 370 kg/s.

7.2 Washing system part units

The washing system that is going to be used for the test rig experimental work is the ATOMAX MSC system of the R-MC company with just a few modifications. The rig washing system comprises a tank of 40 litres capacity of cleaning fluid (see Figure 7-3).

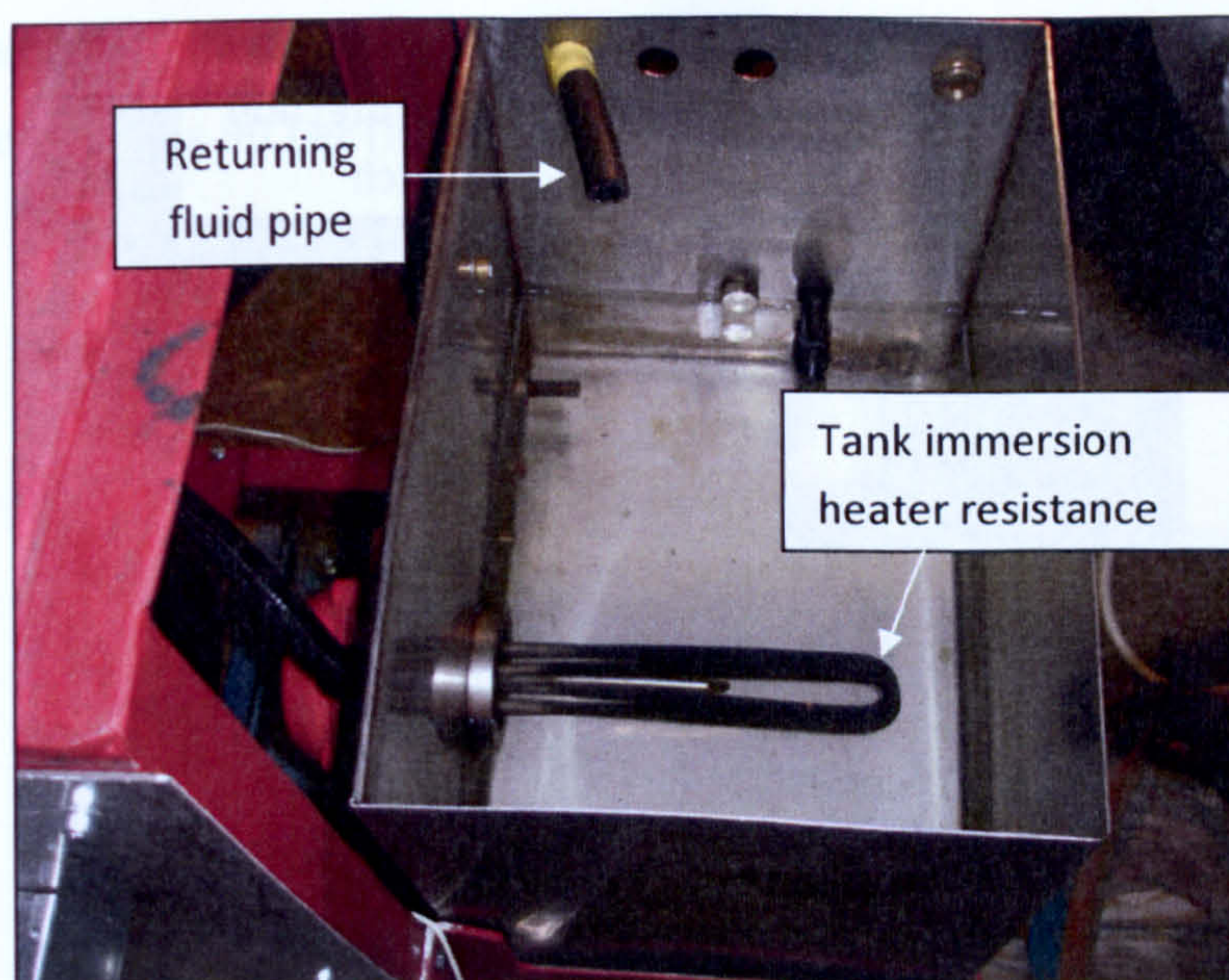


Figure 7-3: Test rig washing system tank.

The tank involves a three phase (440 Volt, 1x3 KW) immersion heater in order to heat up the cleaning fluid in it up to a temperature of 50 °C. In case the fluid level drops below the level of the heater a float switch for heater protection is involved and if below the intake of the pump another float switch is activated to prevent the pump from running dry (sees Figure 7-4).

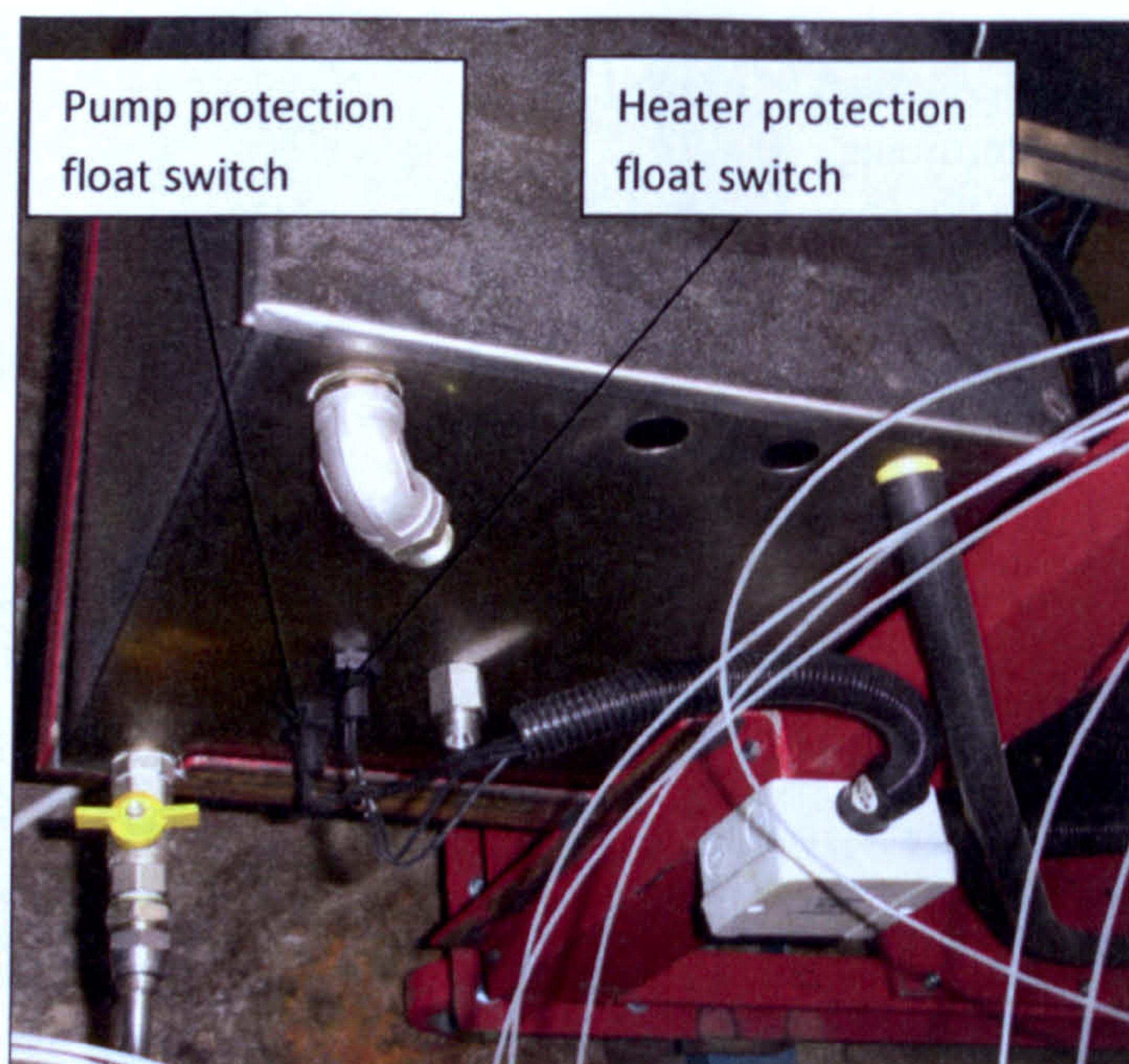


Figure 7-4: Tank float switches.

A temperature gauge is also fitted in the tank in order to display the temperature level of the fluid (see Figure 7-5).



Figure 7-5: Temperature gauge.

The washing fluid is directed to the injection nozzle via a three cylinder plunger pump (5.5HP) capable to deliver 150 bar maximum pressure at 1450 rpm. The pump (Figure 7-6) was driven by a three phase electric motor of 2.2 KW running at 1128 rpm. At the outlet of

the pump a pressure regulator existed in order to provide the injection nozzle with the desirable pressure which is 90 bar (Figure 7-7).

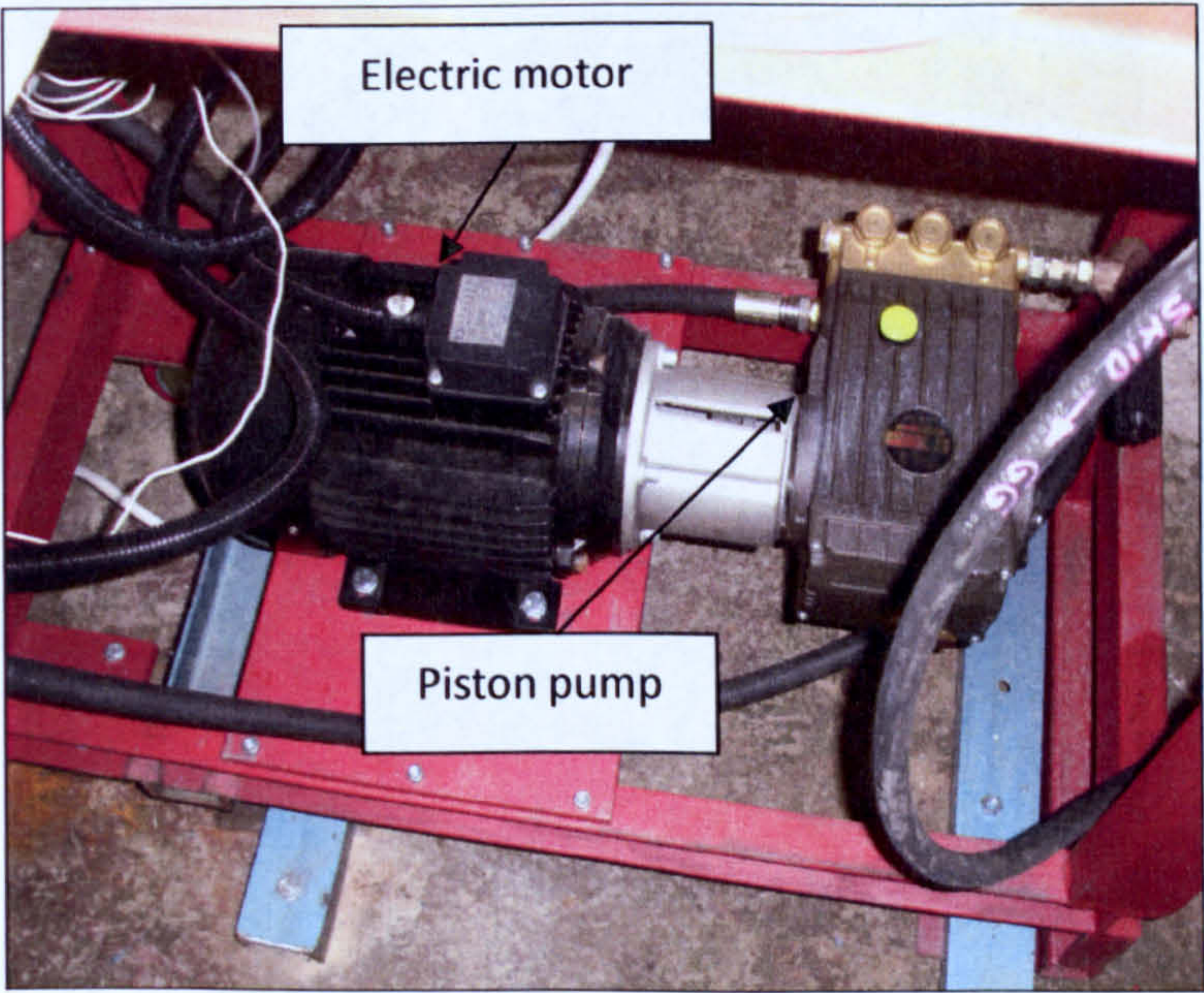


Figure 7-6: Pump and electric motor arrangement.

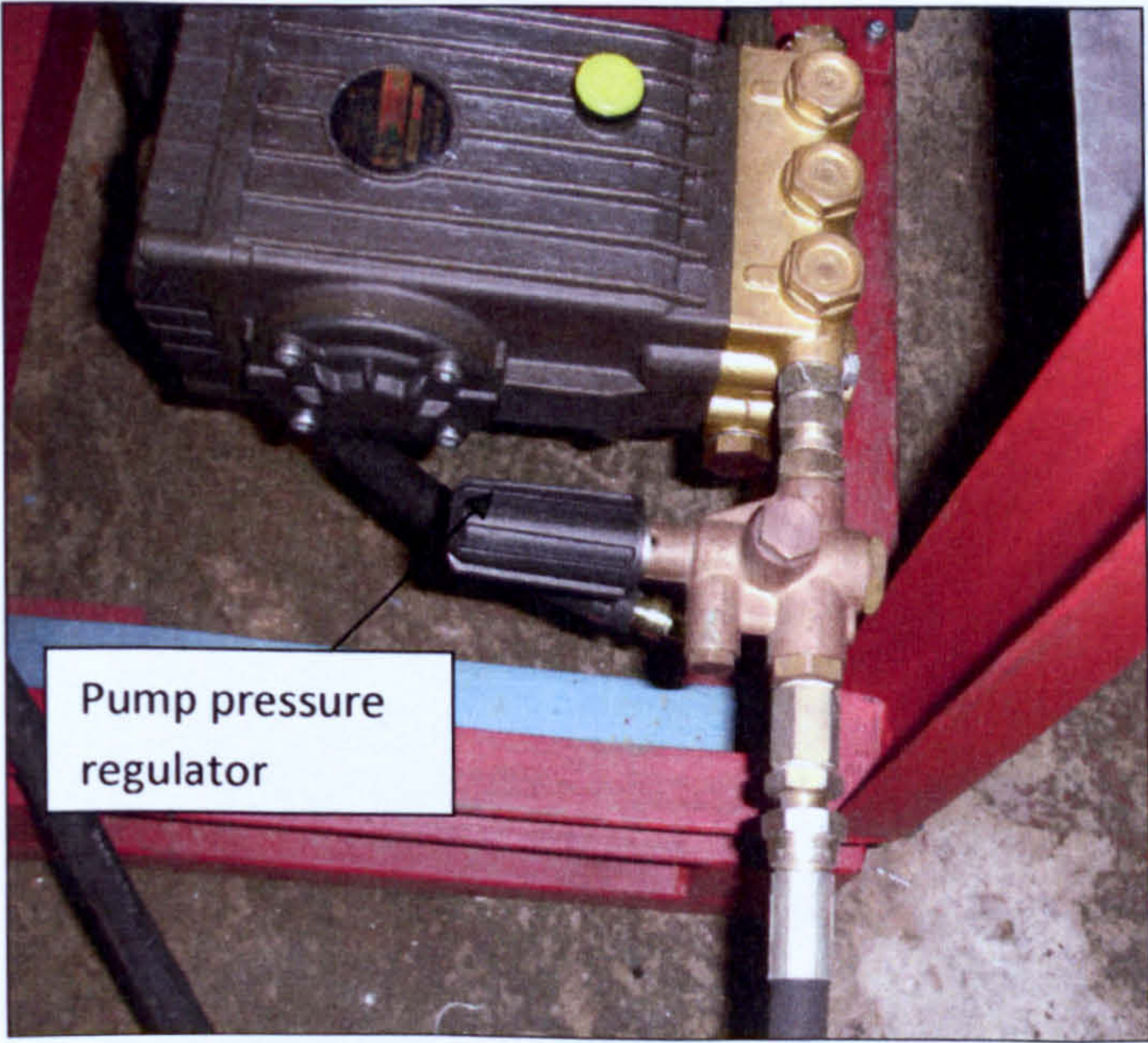


Figure 7-7: Pump pressure regulator.

The reading of the pressure is taken from a pressure gauge fitted next to the injection nozzle (see Figure 7-8).

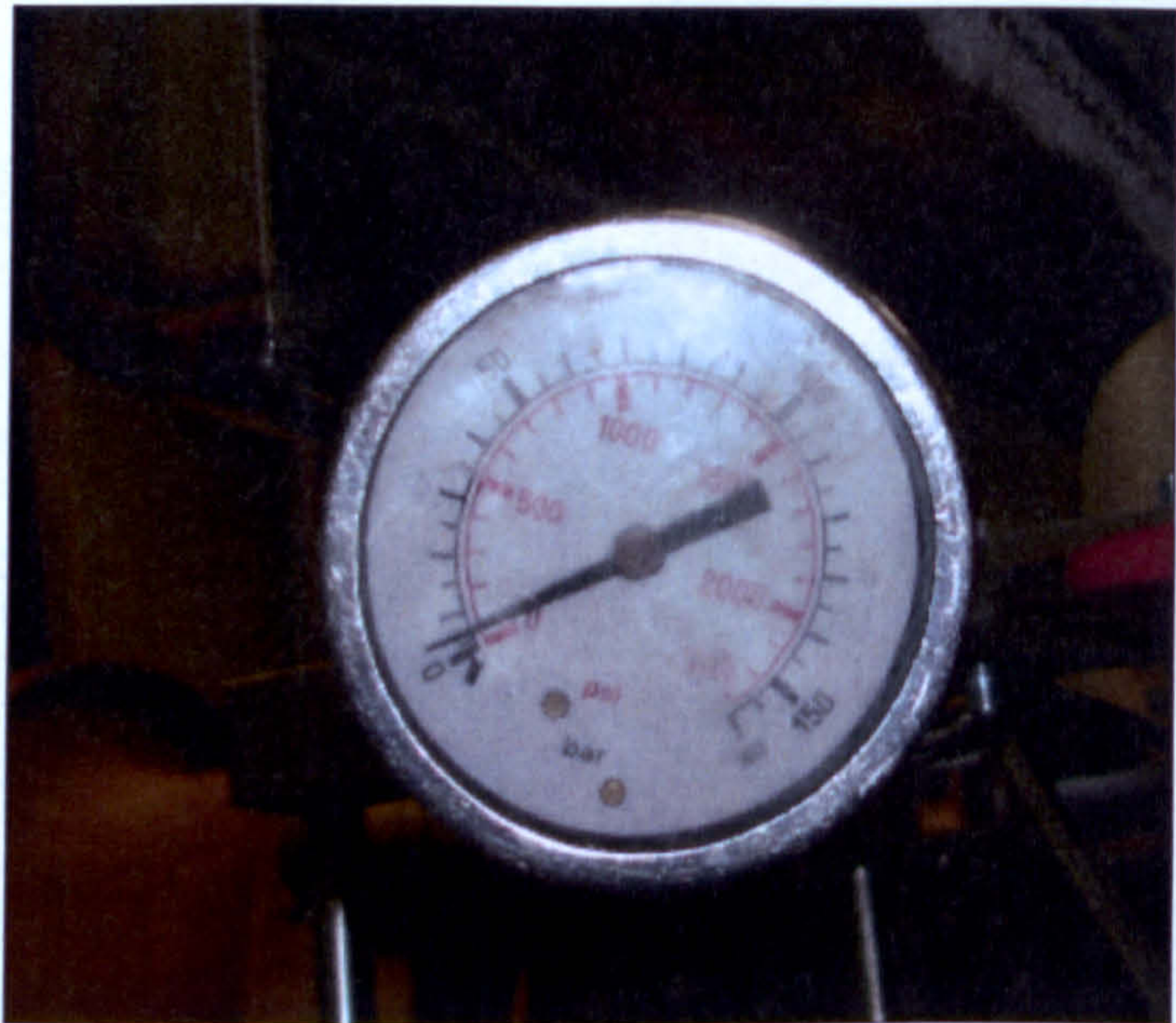


Figure 7-8: Pressure gauge.

In terms of the electrics three important switches are involved in the system. On the electric line there is initially the main power supply electric switch which supports the main pump switch and the heater normal and emergency switch (see Figure 7-9).



Figure 7-9: Main power switch.

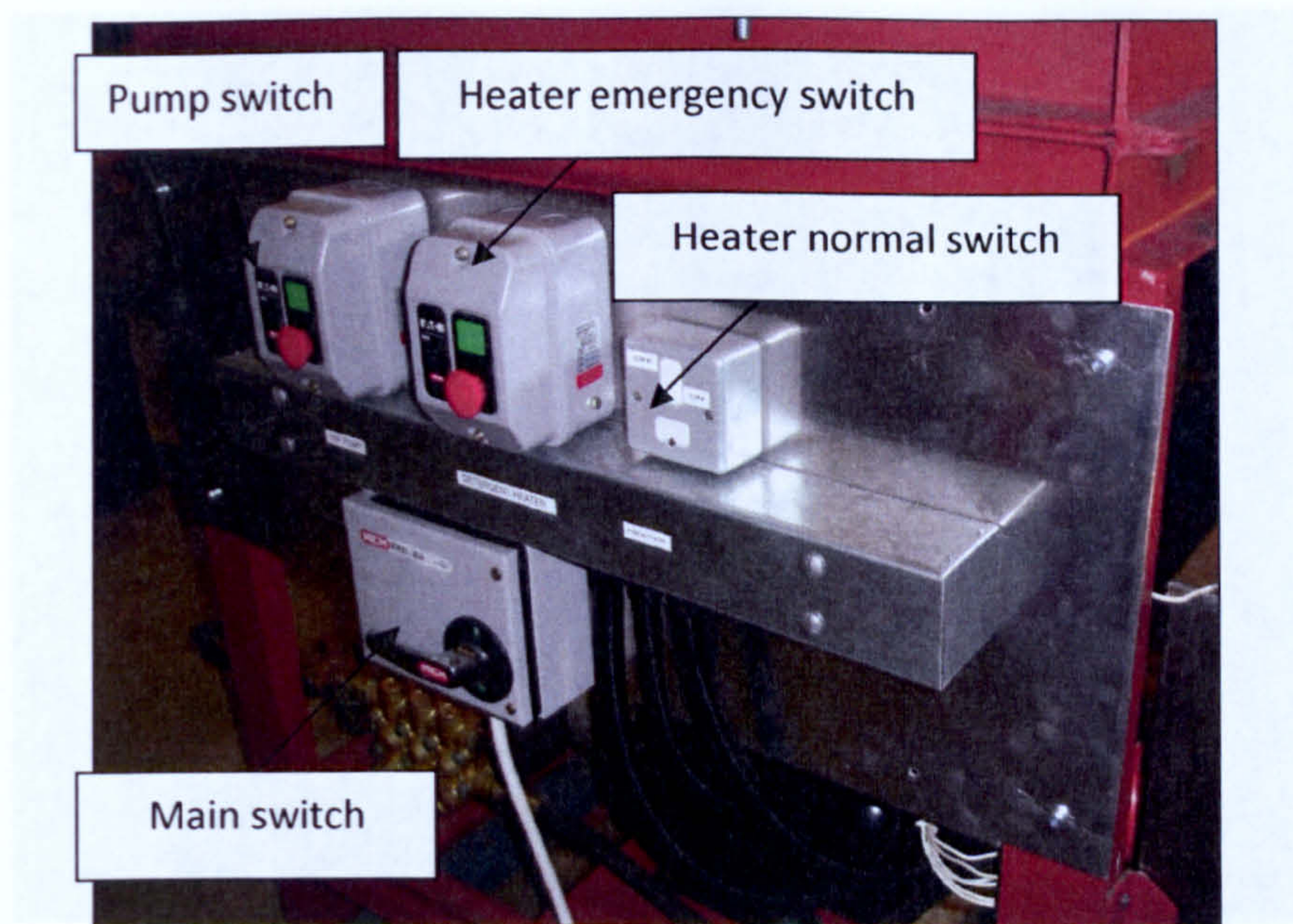


Figure 7-9: Main electric switches.

The nozzle of the injection system involves three degrees of freedom in terms of its movement with respect to the intake of a compressor cascade test rig. This flexibility in terms of movement comes out from the design of a nozzle adjustable support rack that can be used for various intake compressor cascade suction geometries (see Figure 7-10).

With this nozzle support arrangement different nozzle positions can be investigated with respect to the entry of the cascade intake in order to check their effect on blade washing.



Figure 7-10: Nozzle support rack

7.3 Cascade wash experimental work and performance

After fouling the three middle cascade blades with particles of average size diameter 102 microns, cascade washing took place using a flat fan nozzle shown in figure 7-10. The characteristics of the current washing system are the following:

- Droplet size approximately 150 microns
- System operating pressure 90 bar
- Washing fluid mass flow rate level with respect to air 0.2 %
- Injection velocity 122 m/s
- Spray angle approximately 65 degrees
- Water-detergent mixture temperature 30 degrees Celsius

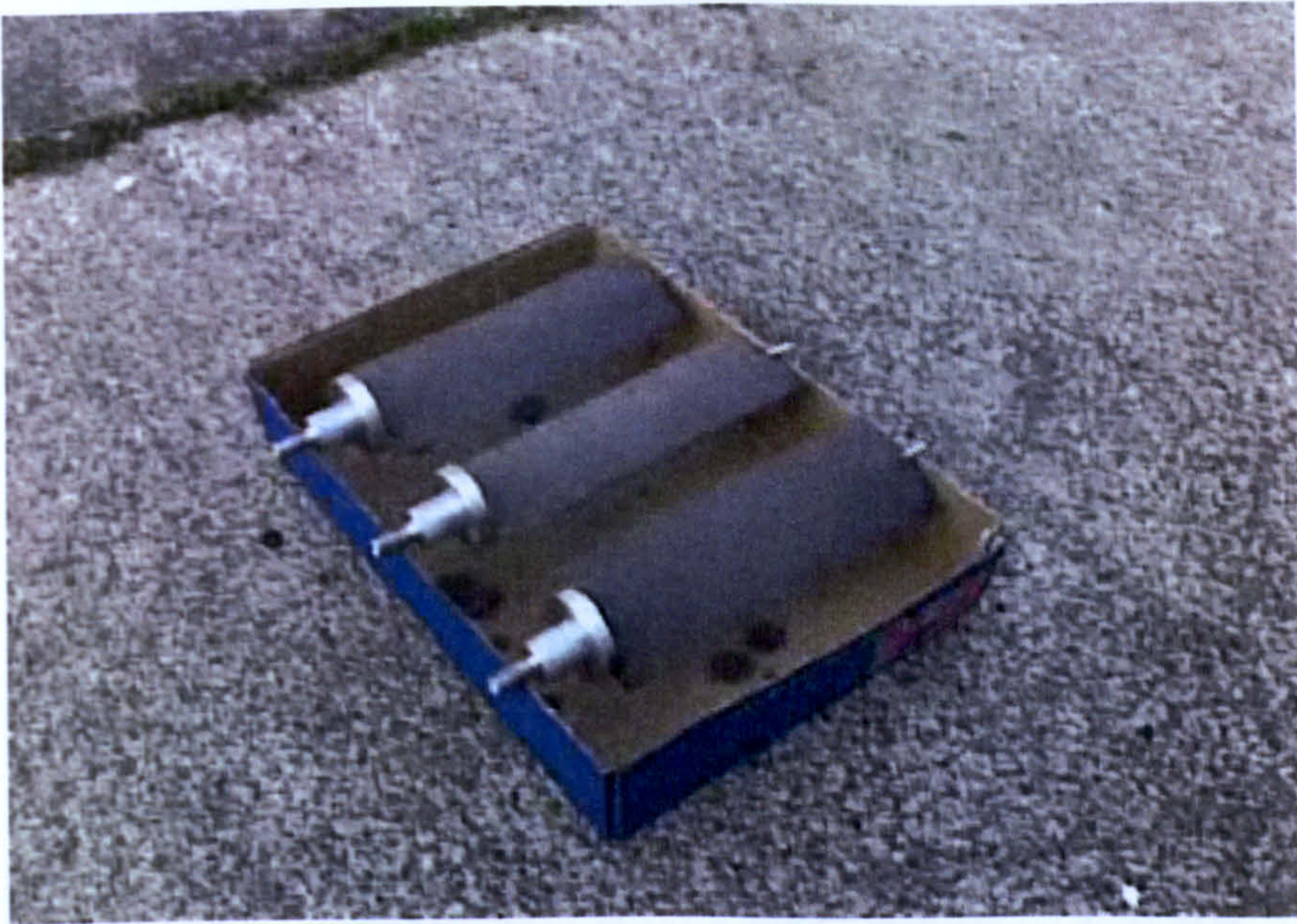


Figure 7-11: Blade before cleaning fouled with heavy viscose crud oil

As it shown in Figure 7-11 the three compressor blades are fouled with heavy crude oil and dried in the atmosphere. In Figure 7-12 the results of washing after 5 minutes are shown. The leading edges of the blades seemed to be partially cleaned and vortices appeared on the suction surface in the area between the blade root and the midspan. These vortices appeared behind the middle blade chord towards the trailing edge part and their clockwise motion helped cleaning the rear suction surface part of the blades. The vortices existed on blade 4 were found to be smaller.

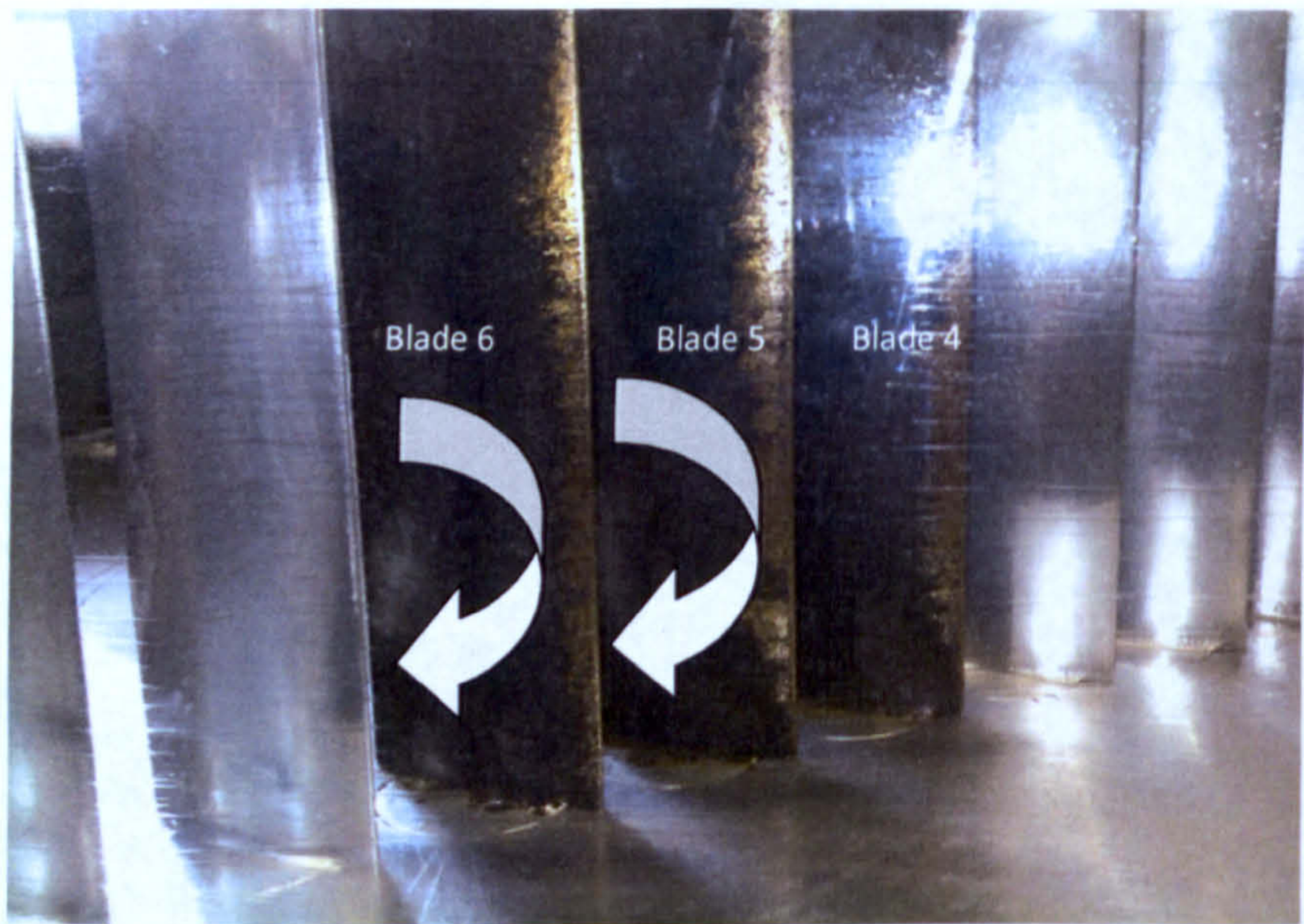


Figure 7-12: Washing results after 5 minutes running.

After ten minutes of washing the leading edge cleaning took place further downstream (see Figure 7-13). Continuing cleaning further more had no additional result on the blade surface deposit removal.

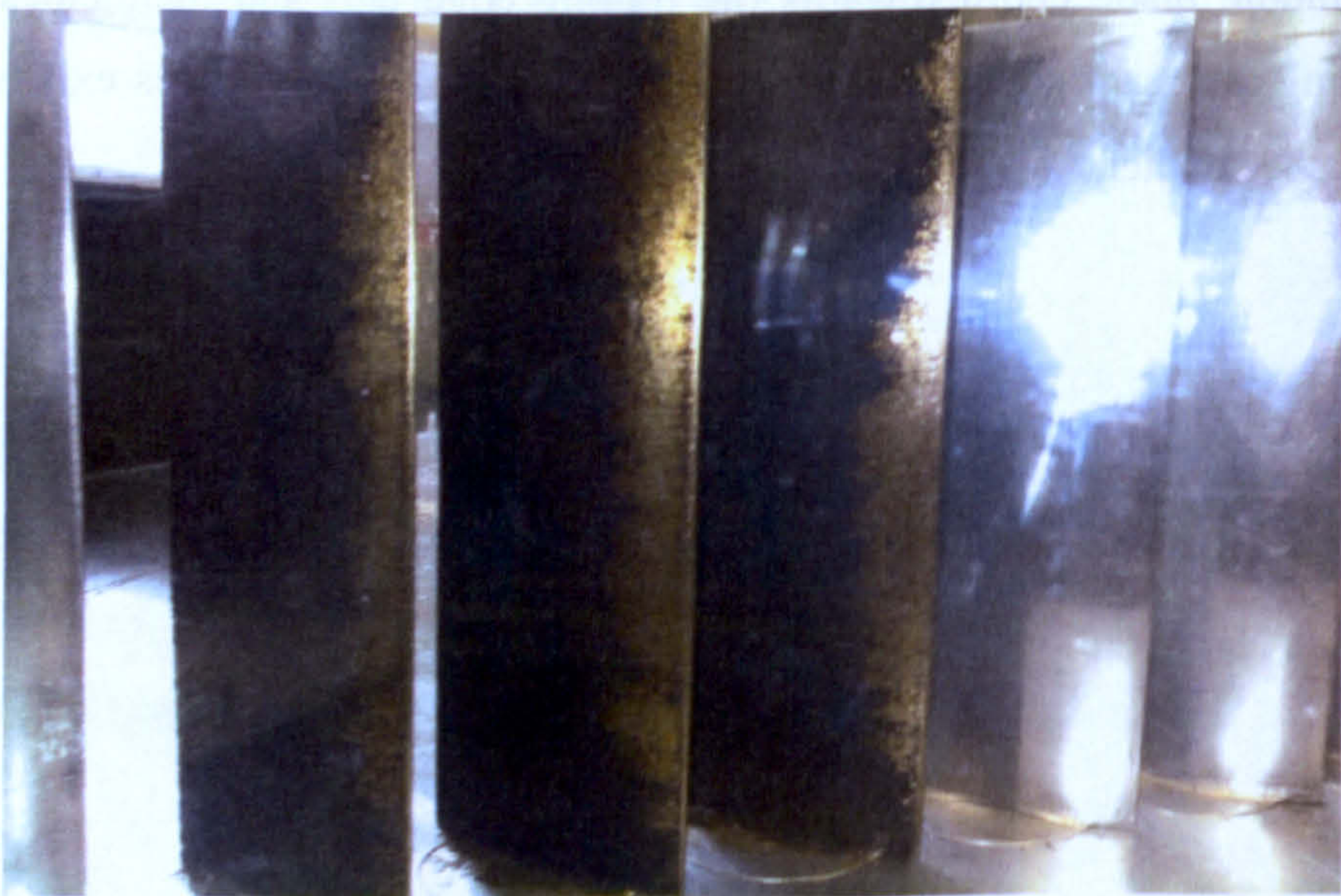


Figure 7-13: Washing results after 10 minutes running.

The lower surfaces of the blades after cleaning were almost unaffected as Figure 7-14 shows.



Figure 7-14: Lower surface washing results.

The velocity distribution behind the blades at 60 mm traverse along the cascade stream flow tube is shown in Figure 7-15. After washing the middle passage velocity reduced from 83.2 m/s to 82.6 m/s, hence only 0.65 %. This was happened because after blade washing the passage area increases and the flow velocity drops. However, after washing the wake of the middle passage blade 5 was not reduced in terms of depth. The depths of the wakes corresponding to blades 4 and 6 were reduced by 3.1 and 1.5 %, respectively as far as the parameter velocity is concerned. The widths of the wakes after wash would be expected to be reduced but at the present experimental results this was not clear.

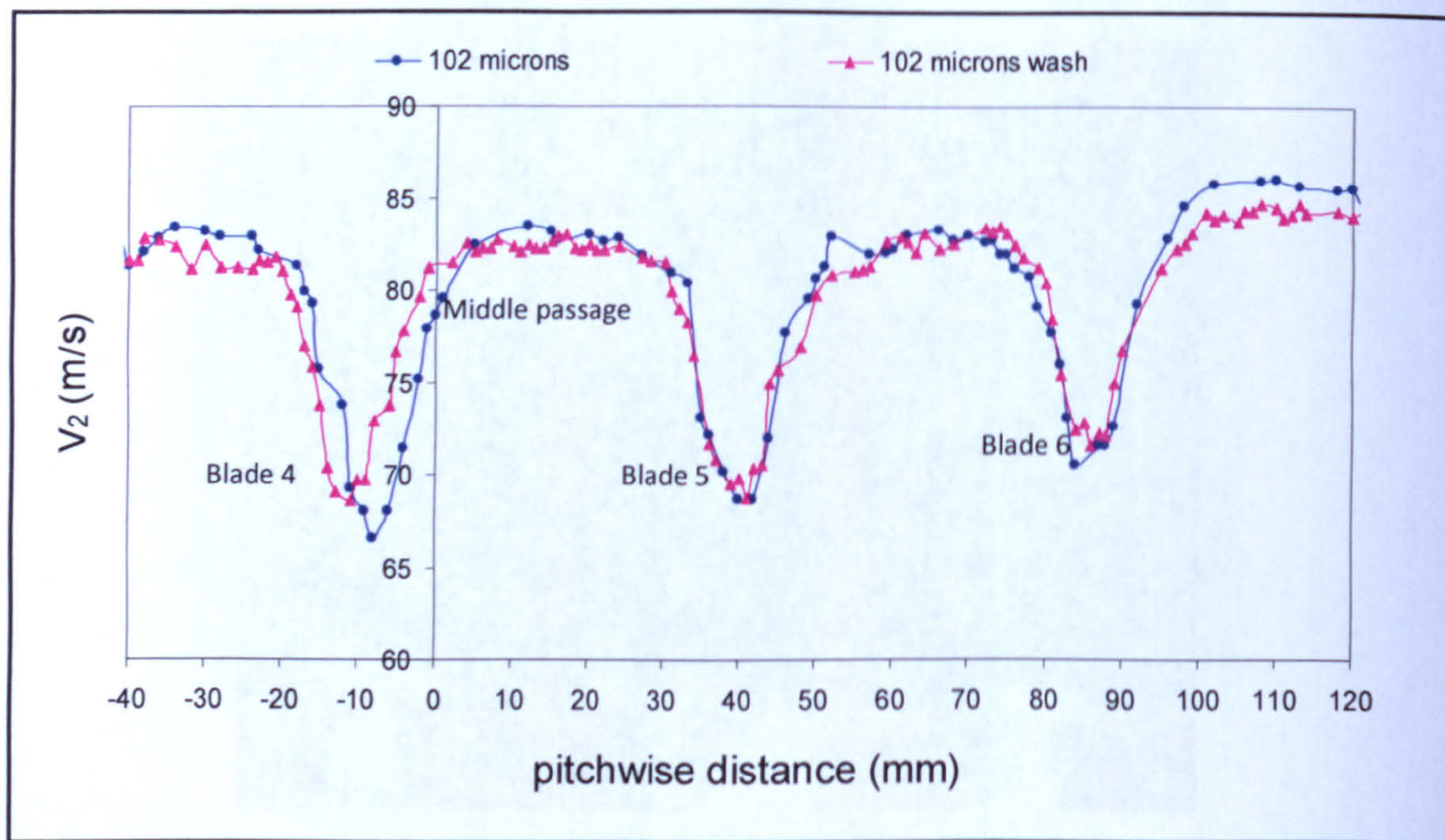


Figure 7-15: Velocity distribution before and after wash

Figure 7-16 shows the corrected velocity distribution results before and after wash. The middle passage corrected velocity falls from a value of 4.9 to 4.8 after wash, hence 2 %.

The blade No. 4 seems to be washed more effectively since it's wake becomes swallower compared to blades 5 and 6. Also, the wake of blade 4 moves to the left implying that the boundary layer thickness of the blade reduces.

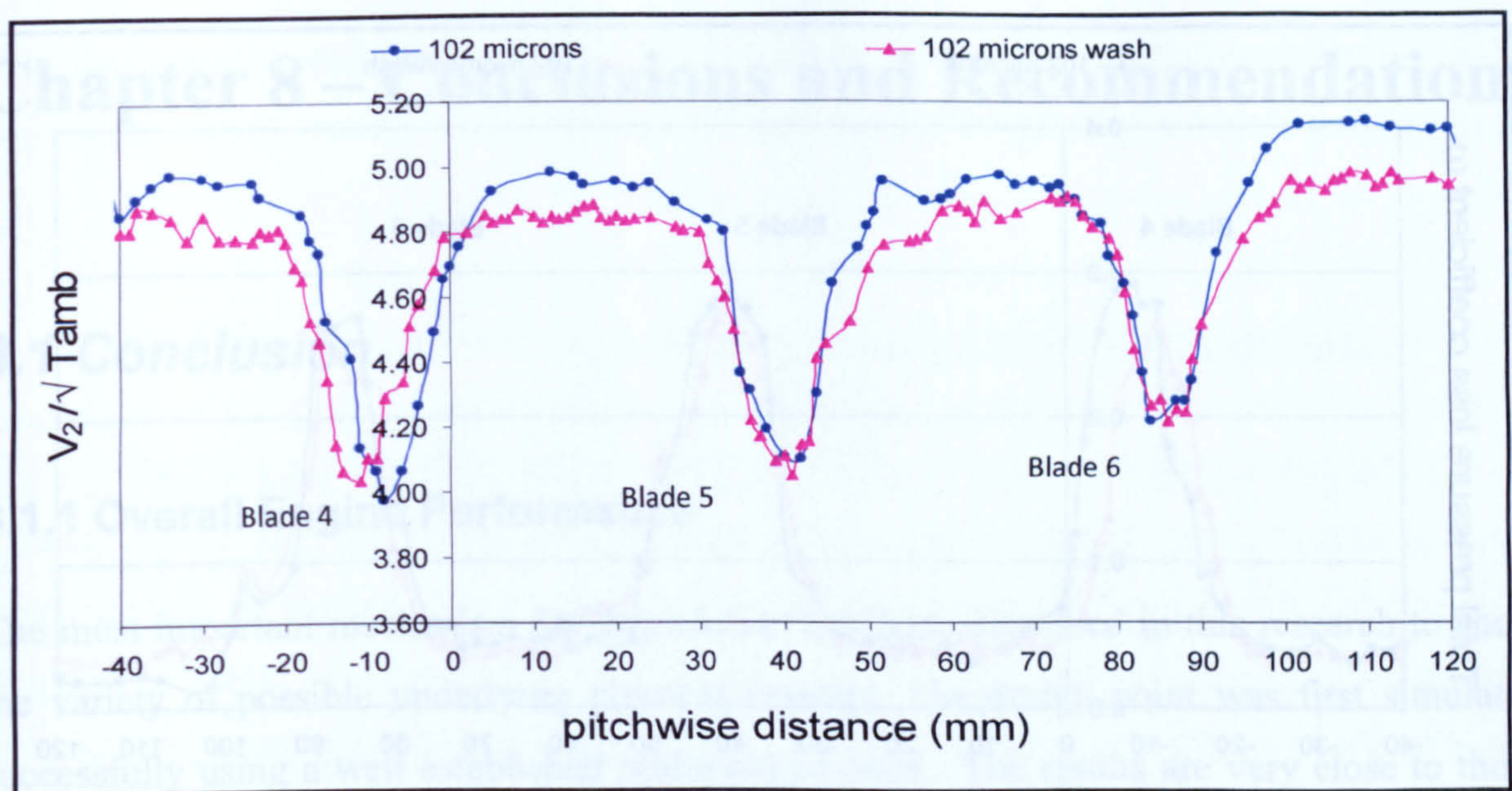


Figure 7-16: Corrected velocity distribution before and after wash

Figure 7-17 illustrates the pressure loss coefficient distribution behind the three middle cascade blades. The middle passage loss coefficient was found not to change significantly after washing. The loss coefficients related to the wakes of blades 5 and 6 were reduced very slightly after washing by almost 3.5 % both. However, the higher reduction in wake loss (9.3 %) after wash, was caused for blade 4. This implies that blade 4 was better washed and this probably can be attributed to the fact of the smaller trailing vortices created on blade 4 compared to blades 5 and 6.

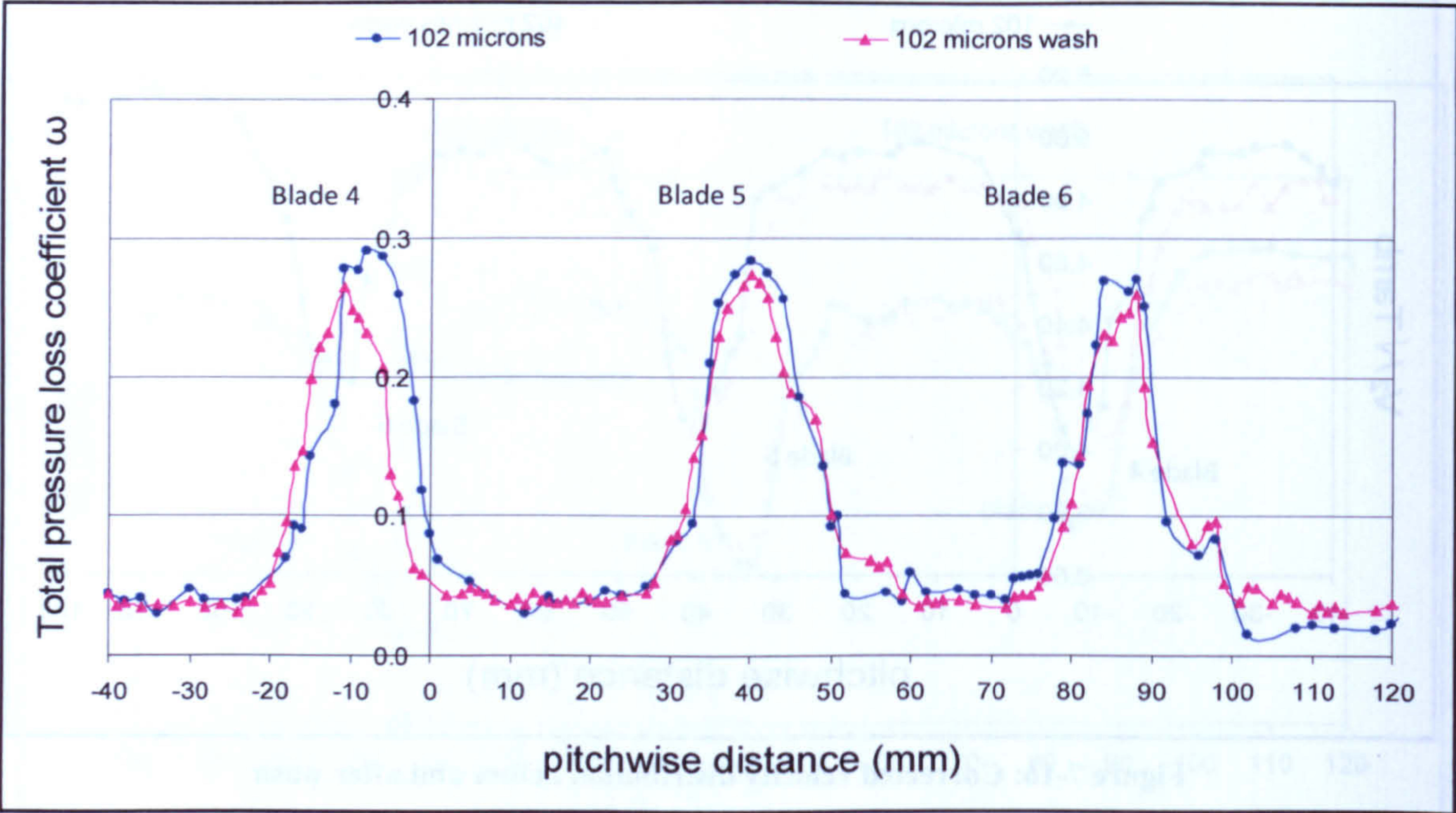


Figure 7-17: Loss distribution before and after wash.

Chapter 8 – Conclusions and Recommendation

8.1 Conclusion

8.1.1 Overall Engine Performance

The most important mechanism for degradation has been discussed in this research to show the variety of possible underlying physical reasons. The design point was first simulated successfully using a well established performance code. The results are very close to those available to the user in the field.

Off-design engine performance is carried out over a variety of ambient temperatures. It is demonstrated, as is well known, that at a given turbine entry temperature (TET) the power output decreases significantly and heat rate increases slightly as ambient temperature increases. In addition, simulation of compressor degradation is examined. It is concluded that compressor cleaning leads to dramatic improvements in gas turbine performance since power loss can be recovered and fuel consumption can be reduced more significantly than is commonly assumed. Furthermore, maintenance costs are substantially reduced with compressor cleaning, especially when the engine is shut down.

An important contribution in this study is that actual industrial engine data is used to enhance the understanding of fouling mechanisms and to indicate the true variation of compressor fouling with the time. So often, in reality, the gas turbine user when making a decision on whether to install cleaning equipment bases conclusions on unsubstantiated claims as to benefits by the vendor. This work assists the user in improving his confidence in the outcome of such purchases. This is now possible because the work reported here is based on the field observations. When coupled with experimental cascade results obtained in this research through actual roughness measurements, a more systematic and scientific analysis is enabled.

The field data used included daily readings of engine temperatures and pressures a period of time simultaneously with observations of exhaust gas temperature increases. Knowledge of the latter facilitates an estimation of firing temperature. This then has allowed the changing fouled state of the compressor to be quantitatively identified over a period of time. In

consequence, the resulting increase in fuel consumption at a fixed power demand has been identified.

8.1.2 Computational Fluid Dynamics (CFD)

A very important contribution in this work has been a computational fluid dynamic (CFD) study of a single stage (HP9) of an actual compressor is compared with experimental data. It is concluded that the CFD predictions using CFX closely corresponds to the measured cascade experimental work. Three different flow conditions (max flow, mid-chic and near surge flow) are studied using the CFD tools as a result the validation with experimental work has been achieved.

Against this background, the coupled CFD and cascade data obtained has demonstrated and described the detailed effect of fouling at on the suction surface of both rotor and stator of a typical high speed axial compressor stage. In particular, the research presents three dimensional predictions for an actual compressor stage applying various levels of fouling

For the fouling cases considered, pressure distributions at the leading and trailing edges of the high pressure stage compressor (HP9) have been estimated using CFD and analysed in order to examine the aerodynamic performance of the compressor stage with and without fouling.

This work has shown (Chapter 5 Section 5.6) the benefits of flow analysis with CFD in a very small region of the flow field. In this case close the analysis is confined to the leading edge and trailing edge of a compressor blade. It also demonstrates clearly the disadvantages of fouling in reducing pressure rise through increases of total pressure loss and therefore reduced efficiency.

It is emphasised that experimental verification of this trend would be very difficult to achieve with a cascade rig because both rotor and stator would need static pressure tapings very close to one another at the leading edge of the blade. This would be extremely difficult to achieve in practice.

According to the CFD analysis one chord downstream of the blades, the pressure loss coefficient increases by increasing the roughness level of course and increases the risk of a flow separation. In addition, engine performance deteriorates in terms of power output and

fuel consumption as the level of fouling increases. However, it is demonstrated that this performance deterioration can be substantially recovered through compressor cleaning.

In support of this CFD research programme an experimental examination of the effect of roughness on compressor blades has been undertaken. As a result, a systematic series of cascade experiments (See Appendix 1) using varying levels of surface roughness have facilitated a rigorous estimation of the magnitude of compressor stage inefficiency caused by fouling. By observing the actual distribution of fouling in a typical compressor, it is now possible to estimate more accurately the overall impact of its fouled performance on overall compressor efficiency.

Furthermore, online compressor cleaning of compressor cascade has been applied. It was found that the part of the upper blade surface close to the leading edge was mainly cleaned effectively (see Chapter 8). Also, the correlation of the washed compressor cascade with a real industrial engine after a fouling level of 102 microns showed that the polytropic efficiency increases by 2.2 % after washing.

8.1.3 Overall Efficiency Prediction

A further important contribution of this research has been an enhanced scientific analysis (than hitherto) of the effect of blade roughness on the stage by stage (polytropic) efficiency as a function of the stage by stage location of the blading. For example it was observed in the field data obtained that the level of roughness decreases as the flow proceeds through the compressor. Given this observation, it is probably typical of compressor fouling in general. With this in mind, a series of roughness levels are applied to the compressor stage by stage and diminishing in degree of roughness towards the rear of compressor. This assumption has allowed the polytropic efficiency of each stage to be evaluated and used to estimate the overall impact of compressor fouling on overall efficiency.

The rather useful result now is that the calculation of compressor overall (isentropic or adiabatic) efficiency degradation with actual observations of real compressor fouling can be more accurately evaluated. This in turn will lead to more reliable and more accurate calculations of increases in compressor work and loss of power output due to fouling can now

be made. Importantly also the outcome is to be able to better estimate levels of firing temperature increase necessary to restore lost performance due to fouling. Interestingly, a further benefit is that this knowledge can greatly enhance the users understanding of compressor cleaning benefits in extending turbine blade creep life (20 degrees centigrade of firing temperature increase can half the life of a typical high pressure turbine rotor)

8.1.4 Final Conclusion

Finally, the outcome of this research can allow the cleaning equipment supplier to improve his case in terms of confidence in the outcome when selling his equipment. In addition, the intending purchaser will better be able to make a decision for or against purchase on the basis of more informed levels of cost versus benefit.

8.2 Recommendation

Several improvements may be accomplished in this field. However, this would strongly increase the computational demand at very high levels; in this case, the simulations may be run only with very powerful facilities, constituted by several computational units.

Further, in this Thesis, water particles over the inlet surface of the axial compressor stage (HP9) have to be considered. Notwithstanding the fact that this approximation is, in most cases, by far acceptable, some experimental tests, analyzing the real water droplet positions when entering the compressor, may be very useful in improving the realism of the model.

The CFD was essential in this investigation to calculate the parameters that were not possible to measure in the experiment. However, it is important to have in mind the limitations of the CFD, because each result represents a possible solution that will be judged by the researcher and for that reason it is highly recommended to have a solid criterion of the physical phenomenon based on experimental or real data.

This information also will help to adjust the CFD model as it was in this case the macro scale information used to validate and adjust the CFD model. The computational time to process the information is another parameter to consider because this can represent a serious problem when a high precise solution is required. It is suggested to increase gradually the process of the CFD since very basic to complex models. Following this suggestion, it is possible to detect an error during the early stages and avoid long periods of time for incorrect or non-converged solutions.

The results of this research were studied in two dimensional conditions. However, to cover the whole area, it will be necessary to study rotational effect of the flow. The three dimensional deposition of the dust in the CFD represents a new challenger to investigate. One of the most important applications of the CFD is the possibility of simulating the flow in the whole compressor. So, further investigations could give an important result in studies that includes each particular stage of the compressor.

Also, the simulation of the mechanism of fouling by CFD represents a challenge for future investigations. In addition, the compressor washing on line should be analyzed with different fouling samples and this possibility could be studied in the test rig of this investigation.

The results of this investigation can be used as a guide of new projects. The real status of the compressor can only be understood with complete and solid background knowledge of the fouling phenomenon. The future information of compressor washing will result important in the total elimination of fouling.

Washing experiment of the compressor cascade can be done by employing washing nozzles of different orifice cross section which are already available, providing so different water to the air mass flow ratios. Different kind of detergents can also employed in order to check the most effective one of the compressor cleaning.

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10 Appendices

10.1 Appendix 1

“Experimental investigation of the influence of fouling on cascade characteristics and implications for gas turbine engine performance”

(Journal of Power and Energy, IMechE 2010)

Experimental investigation of the influence of fouling on cascade characteristics and implications for gas turbine engine performance

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Abstract: The performance of the compressor of an industrial gas turbine can suffer significantly from fouling due to the ingestion of particles like sand and dust. In very hostile environments when particles mix with oil vapour, the outcome is a substantial loss in power output and cycle efficiency due to compressor fouling. In order to recover this performance loss and subject to manufacturer's firing temperature limitations, the engine fuel flow could be increased. This however, would reduce the turbine blades creep life and result in large increases in gas turbine engine operating costs. Therefore, investigating the effects of blade roughness gives valuable knowledge about the mechanisms of fouling and how this can be reduced.

A suction type compressor cascade tunnel with a plenum chamber was employed for investigating fouling blade effects. The tests showed that such a testing arrangement allows the extraction of pressure and corrected velocity distribution data downstream of the blades that is comparable with what can be obtained from blow type cascade tunnels.

This study presents experimental results for smooth clean cascade blades and for uniformly fouled blades. For all the cases considered, mid-span corrected velocity distributions and pressure losses taken one chord downstream of the blades were investigated in order to identify the effects of fouling on the blades. The result of fouling on exit flow angle was investigated as well.

At the present study cascade clean and fouled cases were correlated with real engine performance. Results are obtained in terms of stage polytropic efficiency, thermal efficiency, useful power and compressor efficiency deterioration. Roughening the cascade blades uniformly with particles of 254 microns size, the compressor efficiency percentage deterioration reached a value of 7.7 %.

Keywords: compressor cascade, polytropic efficiency, thermal efficiency, useful power, compressor efficiency

1 INTRODUCTION

This paper describes the findings of a study which examined the influence of fouling on the behaviour of a cascade and by making use of these results the performance implications for gas turbine engines of exposure to airborne foulants.

Surface roughness effects on compressor blades strongly influence the performance of gas turbine engines. Gbadebo et al [1], applied surface roughness on stator blading of a single-stage low-speed axial compressor. With surface flow visualization and exit loss measurements it was shown that the three dimensional separation at the hub was increased by the presence of roughness. The separation identified by Gbadebo caused a significant reduction in the stage total pressure rise. In addition the experimental work illustrated that applying surface roughness between the leading edge and the location of the peak suction caused a significant reduction in stage performance, whereas applying surface roughness downstream of the suction peak had a virtually negligible effect. Applying roughness on

the blades the stage total pressure rise was reduced but at higher flow coefficients the effect of roughness was insignificant.

Nikuradse [2] investigated the flow in pipes which were roughened with sand grains and divided the flow into three regions. In the first region the roughness of the surface is completely covered by the laminar sublayer and it was noted that surface roughness has no influence on the flow. In the second region some roughness elements protruded through the sublayer causing an increase of the pressure drop coefficient. In the third region the roughness elements protrude completely through the sublayer and the pressure drop coefficient does not increase further.

Schaffler [3] investigated experimentally the effect of Reynolds number on four multistage axial compressors and reported three different regimes of operation characterized by the condition of the blades boundary layer, the laminar separation, the turbulent attached flow with hydraulically smooth blade surface and the turbulent attached flow with hydraulically rough blade surface. Schaffler defined a lower

critical Reynolds number below which laminar separation takes place. Above this critical Reynolds number the efficiency increases as the Reynolds number increases as well. This increase stops when the Reynolds number reaches the upper critical Reynolds number which is 5×10^5 and the blade surface behavior is hydraulically rough. Also, it is stated that the lower critical Reynolds number for standard production blades is equal or below the value of 10^5 . Below this value significant blade flow separation occurs and the blade can stall.

Bammert and Milsch [4] run experiments with compressor cascades consisting of roughened NACA 65 series blade sections in different geometrical variations and illustrated that loss coefficients rise by increasing the roughness grade defined as the ratio of sand grain size over the chord length. Also, they illustrated that increasing the surface roughness resulted in the flow turning angle decreasing.

Bammert and Woelk [5] investigated the influence of blade surface roughness on the aerodynamic behavior and characteristic of a three-stage axial compressor model, by using emery grain uniformly roughened blades. For a relative roughness $k_s/c = 4.51 \times 10^{-3}$, the authors reported a reduction in static pressure ratio of 30 % and a reduction in overall efficiency of 13 %. The equivalent sand grain roughness k_s represents the size of sand grains which give the same skin friction coefficients in internal passages as the roughness being evaluated [6].

Tarabrin et al. [7] analyzing computational results, reported a 4.5 % reduction in mass flow, 4 % reduction in pressure ratio and 2 % reduction in compressor efficiency, as successive stages of a gas turbine foul for the first six stages. Tarabrin et al. [8] reported that the gas turbine unit sensitivity to axial compressor fouling decreases by increasing the turbine entry temperature and keeping the compressor pressure ratio constant. Keeping the turbine entry temperature constant and increasing the compressor pressure ratio, the gas turbine unit sensitivity to fouling increases. They also, stated that a 1 % decrease in axial compressor efficiency of a single shaft gas turbine due to fouling can cause a reduction in useful power output of 2.82 %, keeping the turbine entry temperature and pressure ratio constant.

Kurz and Brun [9] developed a model for a two shaft gas turbine engine with power turbine in order to investigate engine performance degradation. By using 2.1 % loss in compressor efficiency, 5 % reduction in airflow, 5 % reduction in pressure ratio and a 0.5 % reduction in gas generator turbine efficiency, the authors reported a 8.6 % reduction in power and an efficiency drop of 3.5 %.

Zaba [10] investigated the effect of fouling on the performance of an industrial gas turbine. He stated that if all the compressor stages are equally and uniformly fouled, the percent reduction in the compressor volume flow rate is approximately equal to the percent reduction in the compressor efficiency. The author also, reports that fouling of the first stages has greater

impact on the compressor volume flow rate than fouling of the rear stages and percent change in the volume flow rate is greater than the percent change in the compressor efficiency. For the case he investigated experimentally, he found that for heavily fouled first stages the change in volumetric flow rate is approximately equal to 2.5 times the percent change in compressor efficiency.

Meher-Homji and Bromley [11] run performance simulations for a 39.6 MW industrial gas turbine with a firing temperature of 2020 °F. Imposing a 6 % deterioration in mass flow and 5 % deterioration in the compressor efficiency, the authors found that the power output drops by 5.5 MW, hence by 14.3 %.

Correlation between cascade test results and real compressor stage has been done by Howell [12]. However, correlation between uniformly fouled cascade and real uniformly fouled engine hasn't yet been investigated. For the present study cascade clean and fouled cases were compared and then they were correlated with real engine performance.

2 EXPERIMENTAL FACILITY

The compressor cascade wind tunnel employed for this investigation is a suction wind tunnel, with a cross-sectional area of 0.043 m² and is designed for air mass flow rate of approximately 5 kg/s which can be varied via a throttling valve installed at the exit. This corresponds to an inlet Mach number of 0.3 and a Reynolds number of 3.8×10^5 formed with the inlet velocity and the chord length. This value is above the critical Reynolds number of about 10^5 [3].

The degree of turbulence and the mean velocity of the inlet air flow were measured with a hot wire anemometer as 2.25 % and 52.6 m/s respectively. The test section of the cascade comprises nine two-dimensional blades of NACA 65 thickened profile with circular arc camber line. The blades are set at zero incidence, the blade span is 180 mm, the chord length 60 mm and the pitch to chord ratio s/c 0.8. The cascade blades represent a stator blade mean section of a gas turbine compressor with 50 percent reaction.

The blades have a span to chord ratio (aspect ratio) of 3. The selection of the particular number of blades and aspect ratio was made in order to minimize the influence of the cascade wall boundary layer effects on the test blade section.

In a compressor cascade tunnel the increase in static pressure across the blades causes wall boundary layer thickening and contraction of the flow. Due to the contraction of the flow the fluid accelerates. This contrasts with the diffusing action of the compressor cascade itself. As a consequence, the sidewall boundary layer effectively reduces the static pressure increase that theoretically would be achieved. To reduce this effect Dixon [13] and many others suggest that at least seven blades are needed for compressor cascade testing and that each blade has a minimum aspect ratio of three.

cascade to discharge to uniform static pressure. The dimensions of the settling chamber were dictated by size limitations of the test location.

When employing suction driven cascade experiments, exit conditions in terms of static pressure are always greatly influenced by the downstream discharge conditions.

The measured wakes were typically around 10 mm wide with the velocity defect falling from 83 to 73.5 m/s in the wake.

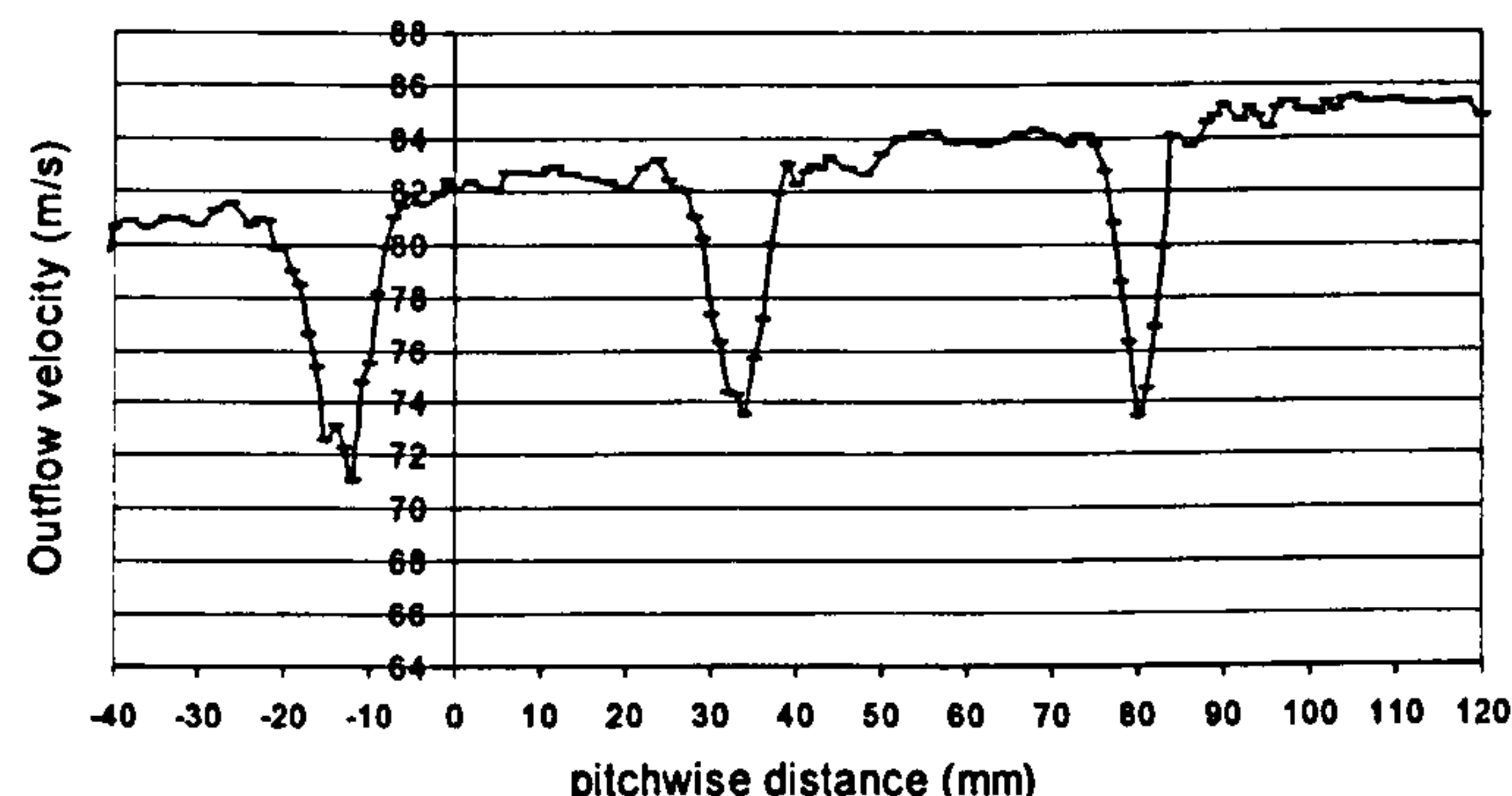


Fig. 3 Statistical average velocity distribution at 60 mm traverse

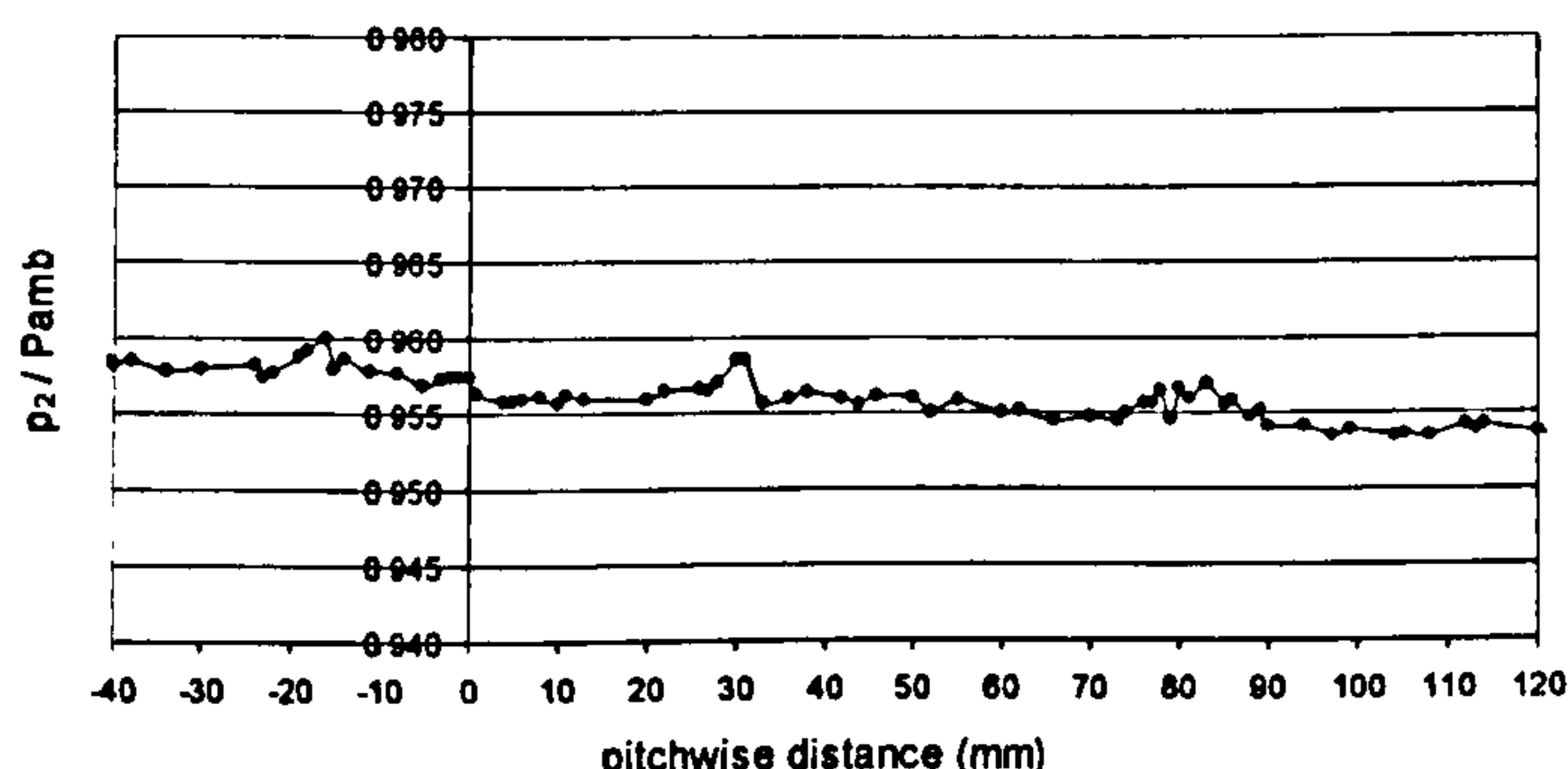


Fig. 4 Normalized rear static pressure distribution at 60 mm traverse

Figure 5 illustrates a statistical average of pressure loss coefficient values calculated at different ambient conditions. The wake loss coefficient of the middle blade reaches a maximum value of 0.16 and the left and right blade loss coefficients both reach a value of around 0.18. The loss coefficient corresponding to the three passages investigated is almost constant at a value of 0.04.

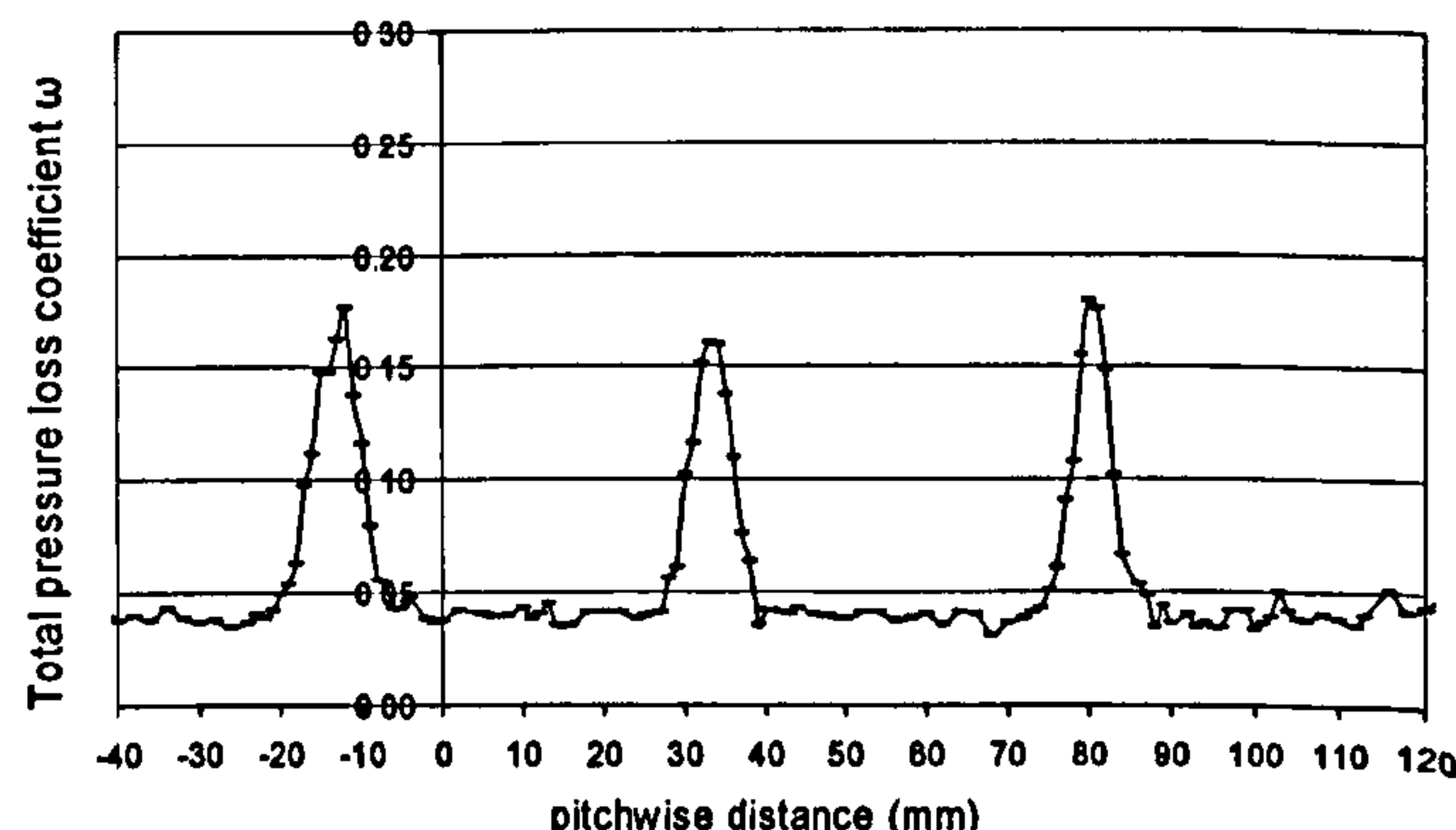


Fig. 5 Statistical average loss distribution at 60 mm traverse

It is important to note that all the cascade experiments have been undertaken at a nominal inlet Mach number of 0.3. In fact the range of Mach number varies from Mach 0.297 to Mach 0.31. Examples of typical variation of losses with Mach number over a range of incidences reported (Saravanamutto, [15] and NASA SP36, [16]) show that for Mach number less than 0.5 losses are independent of Mach number. For the test cases considered here, experimental examination of losses at Mach numbers less than 0.3 has not been undertaken. These values are somewhat lower than Mach 0.5 which is characteristically found in the first stage of industrial compressors. However the fan capacity did not allow the experiments to reach that number.

A statistical representation of the distribution of blade exit flow angle, α_2 , is illustrated in Fig. 6. It can be observed that the middle blade wake exit flow angle is approximately 36 degrees whilst in the passage the exit flow falls to 34 degrees. In addition, there is a small (one degree) decrease in the maximum value of the wake exit flow angle, from one side of the cascade to the other.

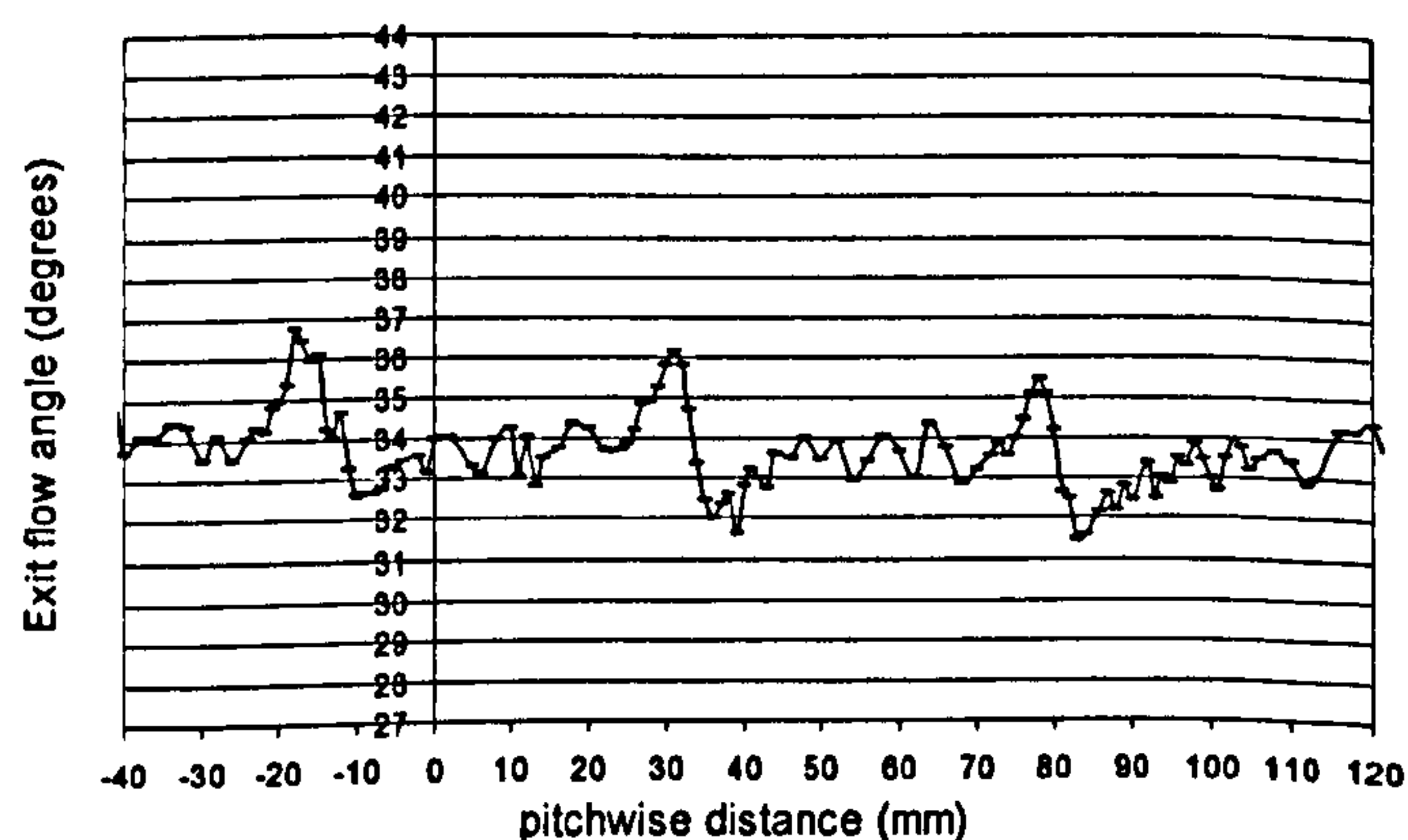


Fig. 6 Statistical average exit flow angle distribution at 60 mm traverse

5 EXPERIMENTAL RESULTS-ROUGHENED BLADES

Roughness was applied uniformly on the three middle cascade blades. The particles used were carborundum of 63, 76,

102 and 254 microns average size stuck with very thin double-sided sticky tape on the blades (see Fig. 7).

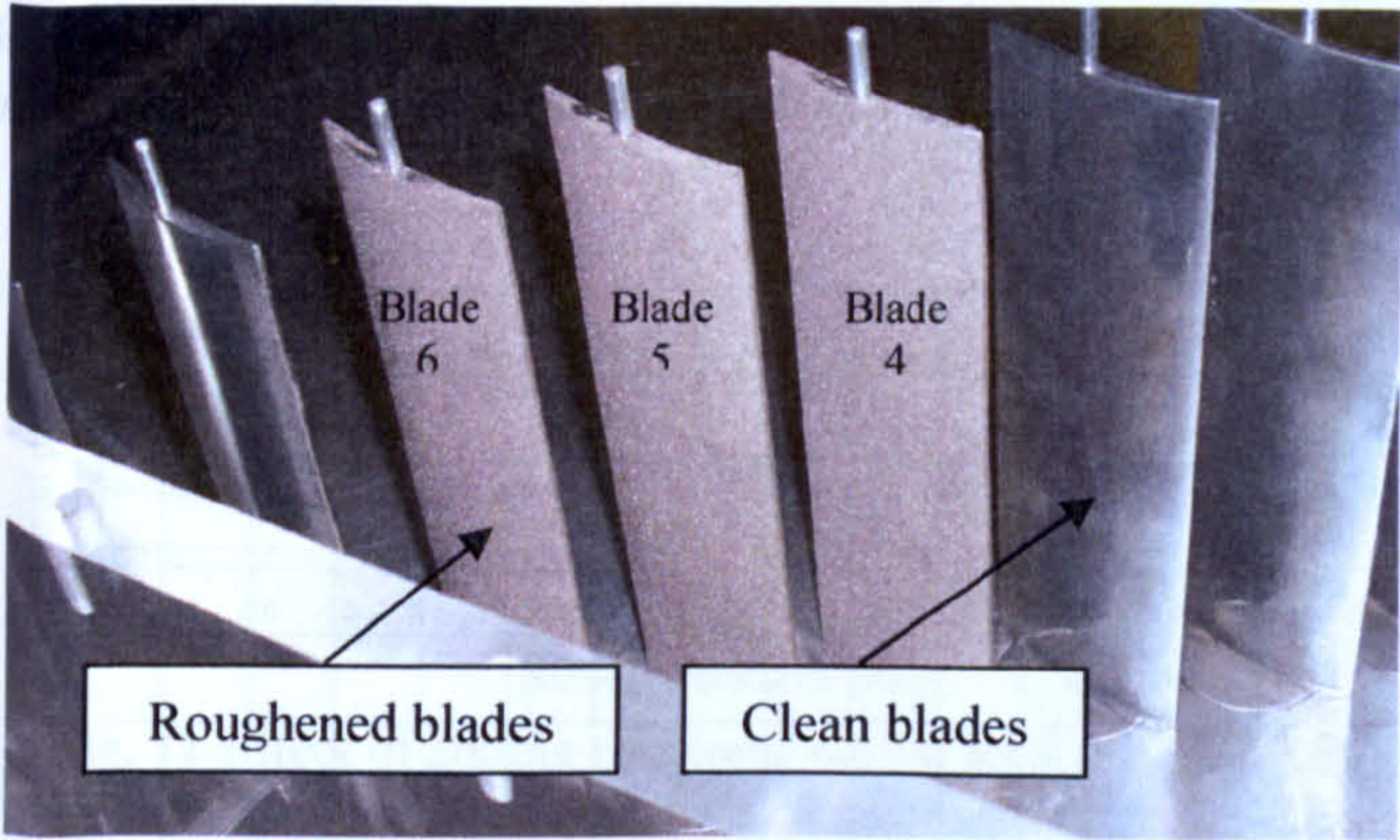


Fig. 7 Roughened blades illustration

Roughness is usually described by the arithmetic average value R_a , defined as the arithmetic mean of the absolute departures of the roughness profile from the mean line. This is the line fitted through the profile where the areas of the profile above and below this line are equal [3].

However this arithmetic mean value R_a is not sufficient in defining the hydrodynamic characteristic of the blade surface because a number of small roughness particles are fully submerged in the laminar sublayer and do not protrude or are felt by the turbulent flow. Nevertheless they must be taken into account in terms of the arithmetic mean value. Schaffler [3] used an effective roughness height k which describes the peaks rather than the average and is defined as the difference between the arithmetic averages of the ten highest peaks and the ten deepest grooves existing per millimeter length. The measuring length used by Schaffler was 5 mm.

Schaffler [3], after measuring the surface roughness of several blades, found a correlation between the effective roughness height, k and the arithmetic average height, R_a (see Eq. (2)):

$$k = 8.0 \cdot R_a \quad (2)$$

For this project the parameter R_z was used as well, which is the average height difference between the five highest peaks and the five lowest valleys within a sampling length of 0.8 mm [17].

A surface texture measuring device (Taylor Hobson Sutronic 25) was used involving a stylus following the surface of the blades in the streamwise and spanwise direction and taking roughness measurements at stations of 25, 50 and 75 % of the chord and 25, 50 and 75 % of the span. All the span and streamwise measurements were averaged and the results are presented in table 1 and 2 for upper and lower blade surfaces, respectively. The sampling lengths assessed from the measuring device were five and a mean value for every roughness parameter measured was calculated within an assessment length

l defined from the sum of the sampling lengths. This provides a better statistical estimate of the parameter's measured value.

Particle size	Upper surface spanwise roughness (μm)			Upper surface streamwise roughness (μm)		
	R_a	R_z	k	R_a	R_z	k
0 microns	0.42	2.56	3.76	0.6	4.67	5.34
63 microns	6.73	37.22	59.93	7.2	40.11	64.41
76 microns	10.02	51.78	89.20	10.11	51.33	90.00
102 microns	14.44	73.44	128.55	14.62	73.67	130.14
254 microns	22.71	114.00	202.13	21.47	107.44	191.05

Table 1 Upper surface roughness

Particle size	Lower surface spanwise roughness (μm)			Lower surface streamwise roughness (μm)		
	R_a	R_z	k	R_a	R_z	k
0 microns	0.37	2.11	3.36	0.47	3.22	4.15
63 microns	7.4	41.33	65.86	7.38	41.11	65.66
76 microns	10.4	53.22	92.56	10.11	51.11	89.99
102 microns	14.53	74.22	129.35	14.33	72.33	127.57
254 microns	24.04	122.78	214.00	22.44	109.22	199.76

Table 2 Lower surface roughness

The effect of fouling on the pitch-wise velocity distribution at one chord downstream of the cascade blades was investigated for clean and fouled blades. The corrected velocity, defined as the ratio of exit flow velocity to the square root of the ambient temperature ($V_2 / \sqrt{T_{amb}}$), in the middle of the passage remains almost constant at a value of 4.9, Fig.8. However, as the fouling level increases this ratio undergoes significant changes. Increasing the particle size to 63, 76, 102 and 254 microns the velocity ratio in the wake drops from 4.39 (clean blade) to 4.21, 4.14, 4.10 and 3.91 respectively. These values correspond to percentage falls of 10.4, 14, 15.5, 16.3 and 20.2 % from the passage value of 4.9. This indicates that the wake corrected velocity behind the roughened blades drops continuously though not linearly.

In addition the wake widens as the fouling level increases. However the increase is not gradual. For clean blades the wake is approximately 10 mm thick. However, for fouling levels of 63, 76 and 102 microns the wake thickness remains almost constant at 24 mm. Applying particles of 254 microns on the blades, the wake thickness increases considerably to about 39 mm.

As seen in Fig. 8 all the wakes move towards the right with respect to the 0 microns wake as the fouling level is increased. This happens due to the higher boundary layer growth of the suction surface of the blades compared to the pressure surface boundary layer. The effect of the pitchwise static pressure increment towards the right side of the cascade (see Fig. 4) behind the blades is assumed not to be responsible for this wake

shift towards the right side of the cascade. This is because the flow Mach number there is very low. For such cases probable incidence changes do not cause significant changes in the blade drag coefficient which is related to the thickness of the boundary layer affecting the blade wake.

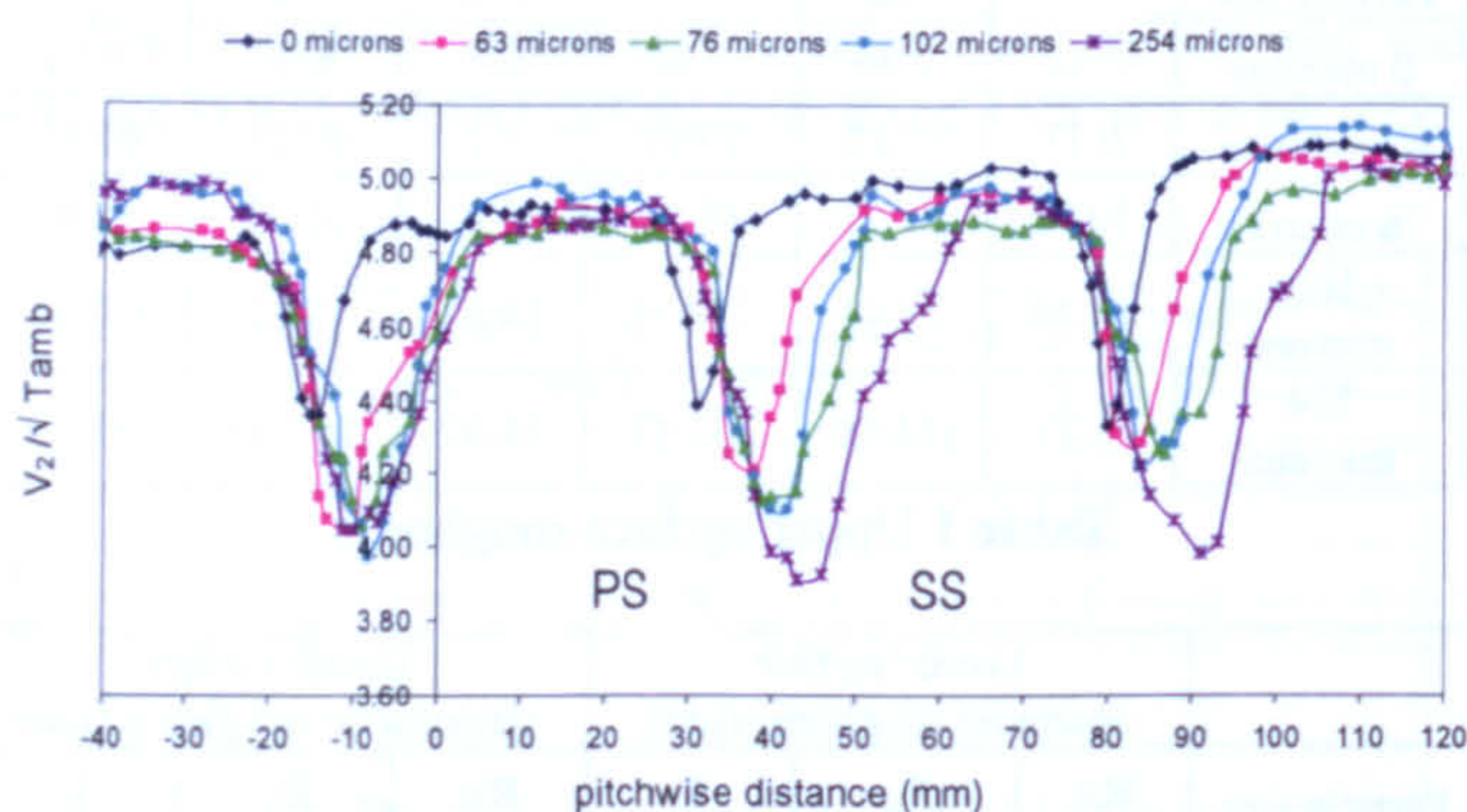


Fig. 8 Roughness wakes at 60 mm traverse

The pressure losses corresponding to the all three middle passages behind the blades were investigated. Figure 9 illustrates the pressure loss distribution one chord downstream of the blades towards the cascade exit streamtube. Taking into account the middle passage, as the fouling level applied on the blade surface increases from 0 microns (smooth blades) to 63, 76, 102 and 254 microns the total pressure loss coefficient increases from 0.16 to 0.25, 0.27, 0.285 and 0.35 respectively. This indicates that the wake total pressure loss coefficient behind the roughened blades rises continuously but not linearly. The passage loss varies smoothly around the value of 0.04 and it seems that the fouling does not affect this area significantly.

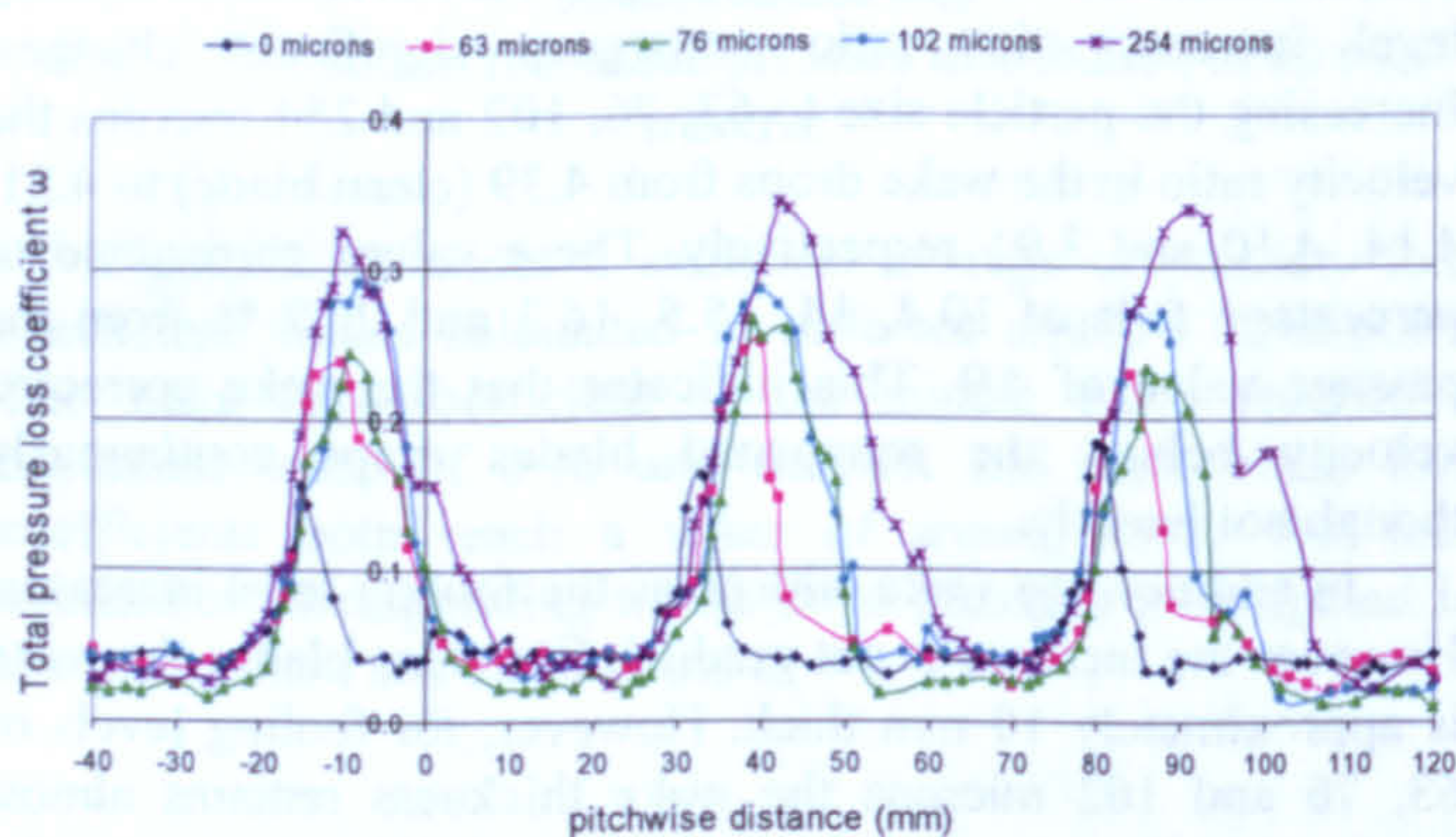


Fig. 9 Roughness cases loss distribution at 60 mm traverse

It was noted that as the fouling level increases from 0 to 76 microns the middle passage exit flow angle (one chord downstream) corresponding to the nulling point in the passage between blade 4 and 5 increases from 34 degrees to 37 degrees. Increasing the fouling level further to 102 and 254 microns the passage exit flow angle at the same point gets values of 38 and 39.5 degrees (see Fig. 10). Hence, the passage exit flow angle seems to increase in a lower proportion as the roughness

increases more than 102 microns. Up to the 102 microns the increase in exit flow angle is 4 degrees and from 102 microns to 254 microns, the increase is reduced to 1.5 degrees only. This can be attributed to the increase of the passage blockage which progressively reduces the deviation and therefore the exit flow angle.

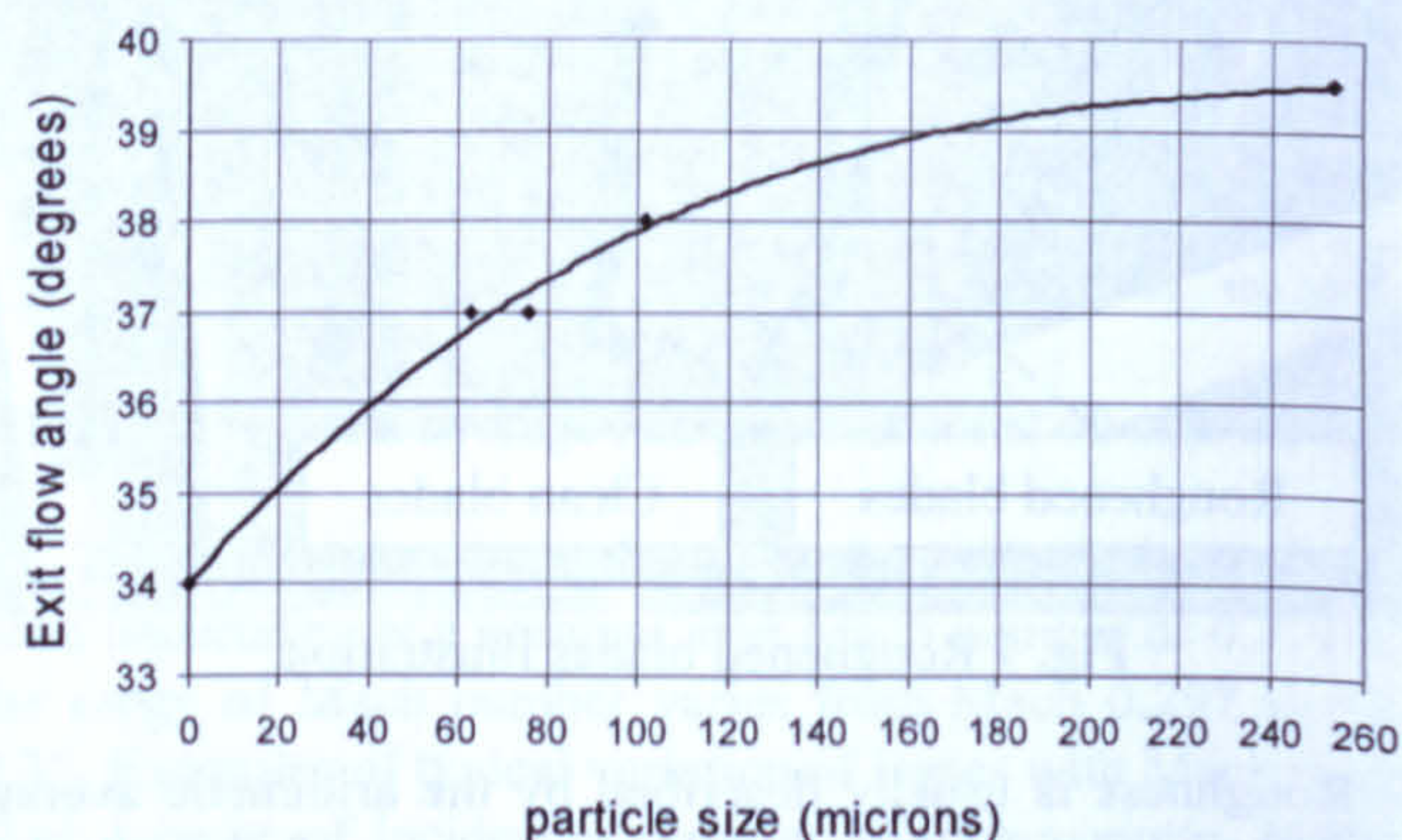


Fig. 10 Blade 4-5 passage exit flow angle versus particle size

6 BLADE THICKNESS EFFECT

The effect of blade thickness due to roughness has not been taken into account in the current study. Gbadebo et al [1] in order to separate the effect of thickness from the effect of roughness, performed tests by covering the leading edge / peak suction region with thin cardboard strips of similar thickness to that of the emery paper used for applying roughness. Comparing contours of stage pressure rise coefficient for thickened blades with these of smooth and roughened blades they found that the thickness has negligible contribution to wake thickening. According to Saravanamuttoo [15] test results for subsonic compressor blade sections shows that at low Mach numbers, the losses for zero incidence are very low compared to those at higher incidence and higher Mach number. The current cascade runs with an inlet Mach number of 0.3 and at a nominal incidence of zero degrees. The corresponding losses are expected to be low and depend only secondarily on blade thickness. However, at high Mach number and incidence far away from nominal, as expected, the losses increase dramatically. In this case, blade thickening due to roughness significantly increases the pressure losses.

The effect of adding thickness to the middle cascade blades by attaching the double sided sticky tape was investigated experimentally. Figure 11 shows that the addition of the double sided sticky tape has almost no effect in total pressure loss coefficient especially for the two side cascade blades.

assumed that the percentage deterioration in polytropic efficiency of this engine equals the percentage deterioration of the polytropic efficiency derived from the experimental cascade expressed as stage results. An assumption is made that the mid radius reaction is 50 %.

Another parameter used as input in the Turbomatch simulation was the percentage deterioration in the cascade passage non-dimensional mass flow $\Delta(W_1 T_1^{0.5}/P_1)_p$ obtained from the experimental results due to fouling. This was assumed to be equal to the percentage reduction in non-dimensional mass flow of the industrial gas turbine engine examined.

8 EXPERIMENTAL PERFORMANCE SIMULATION RESULTS

The polytropic efficiency was calculated for smooth and fouled blades of the current project compressor cascade tunnel, Fig. 12 showing that for fouling levels of 63, 76, 102 and 254 microns, the percentage deterioration in polytropic efficiency is 2.2, 3.9, 4.9 and 7.7 % respectively.

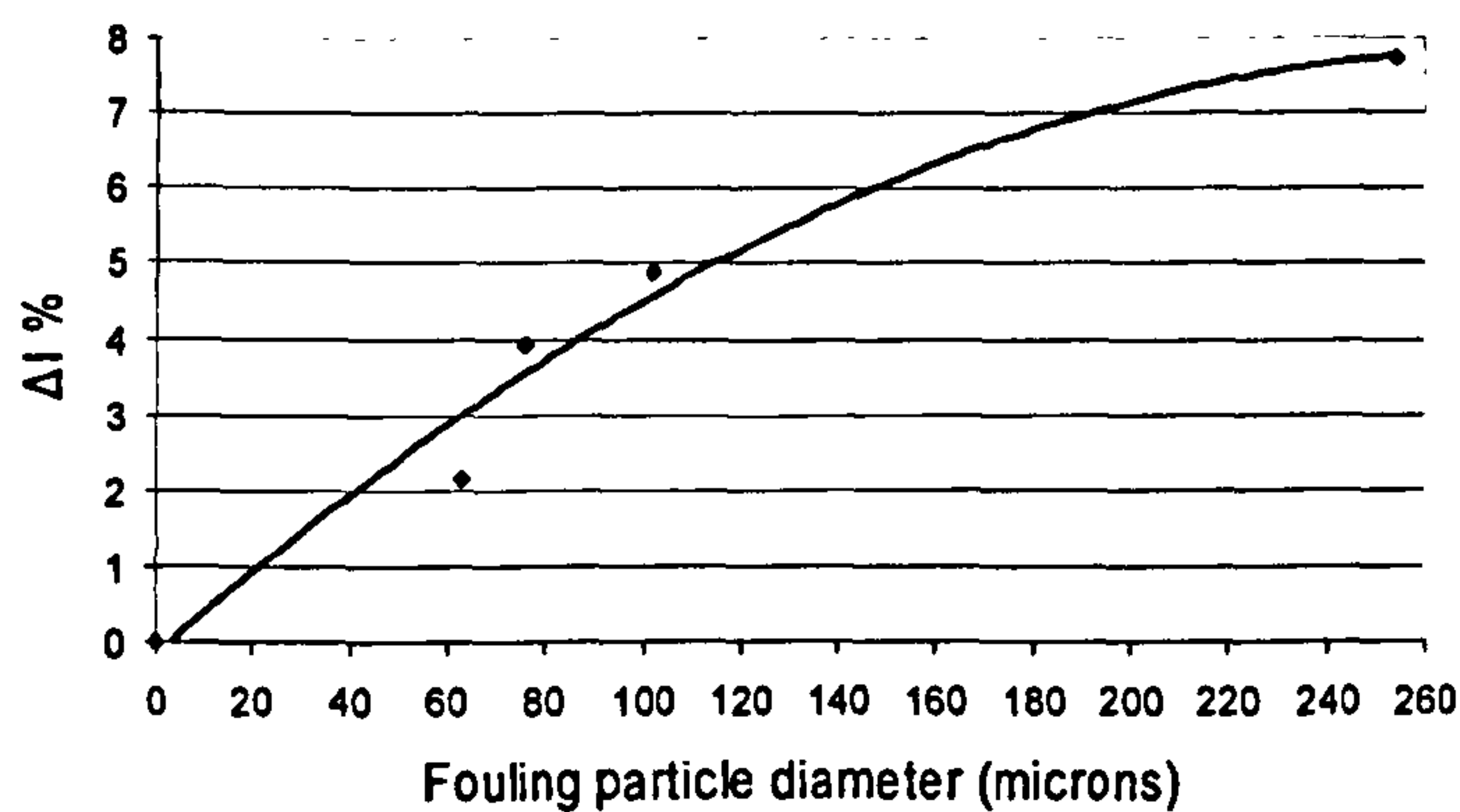


Fig. 12 Experimental percentage deterioration in polytropic efficiency due to fouling

Table 3 shows the percentage reduction of the cascade passage non-dimensional mass flow with respect to the fouling particle size applied on the blades.

For this study it was assumed that the compressor is fouled uniformly all the way through. Turbomatch performance simulation runs were carried out while varying the turbine entry temperature between 1100 and 1500 K combined with the effect of fouling. The thermal efficiency of the engine was examined first. The thermal efficiency η_{th} increases as the TET increases, Fig. 13, and this can be attributed to the parallel increase of the compressor pressure ratio, Fig. 14. Increasing the particle size from 0 to 254 microns for constant TET levels the pressure ratio was found to decrease by 1.3 %. However, keeping the fouling level constant and increasing the TET from 1100 K to 1500 K the pressure ratio increased by 8.3 %.

Particle size (microns)	$\Delta(W_1 T_1^{0.5}/P_1)_p$ %
0	0.000
63	0.4
76	0.5
102	0.7
254	1.7

Table 3 Cascade passage % reduction in non dimensional mass flow

As the fouling increases, the pressure ratio drops and the thermal efficiency decreases (i.e. referring to values at constant TET in Figs. 14 and 13 respectively). Considering the case of smooth blades (0 microns) the TET increases from 1100 K to 1500 K and the thermal efficiency increases by 15.5 %. Taking into account the case of the highest fouling of 254 microns for the range of the same TET increase, the thermal efficiency increases by 25.5 %. Therefore as the fouling particle size increases, the percentage thermal efficiency gain in the same level of TET increase gets higher.

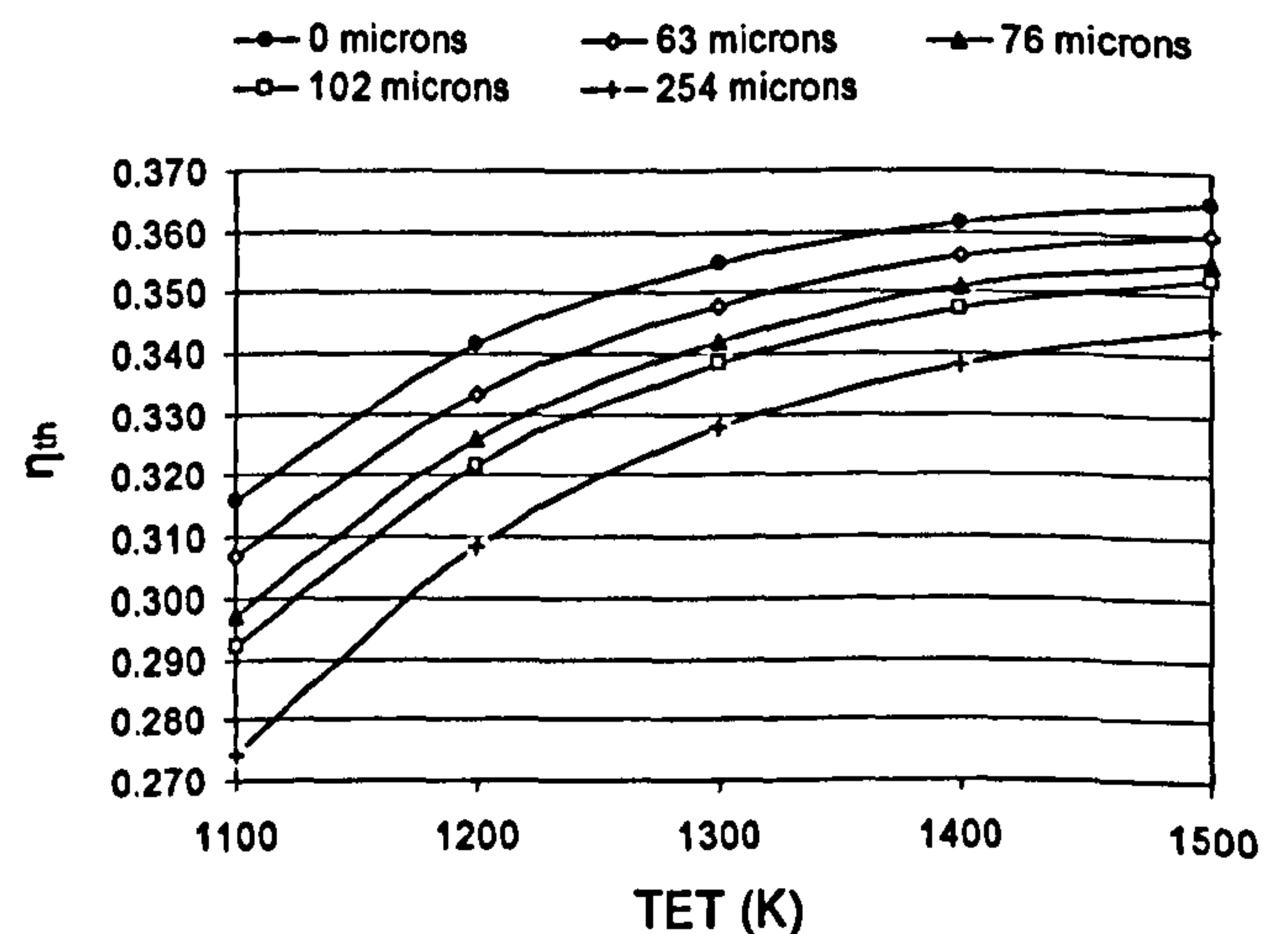


Fig. 13 Thermal efficiency versus TET

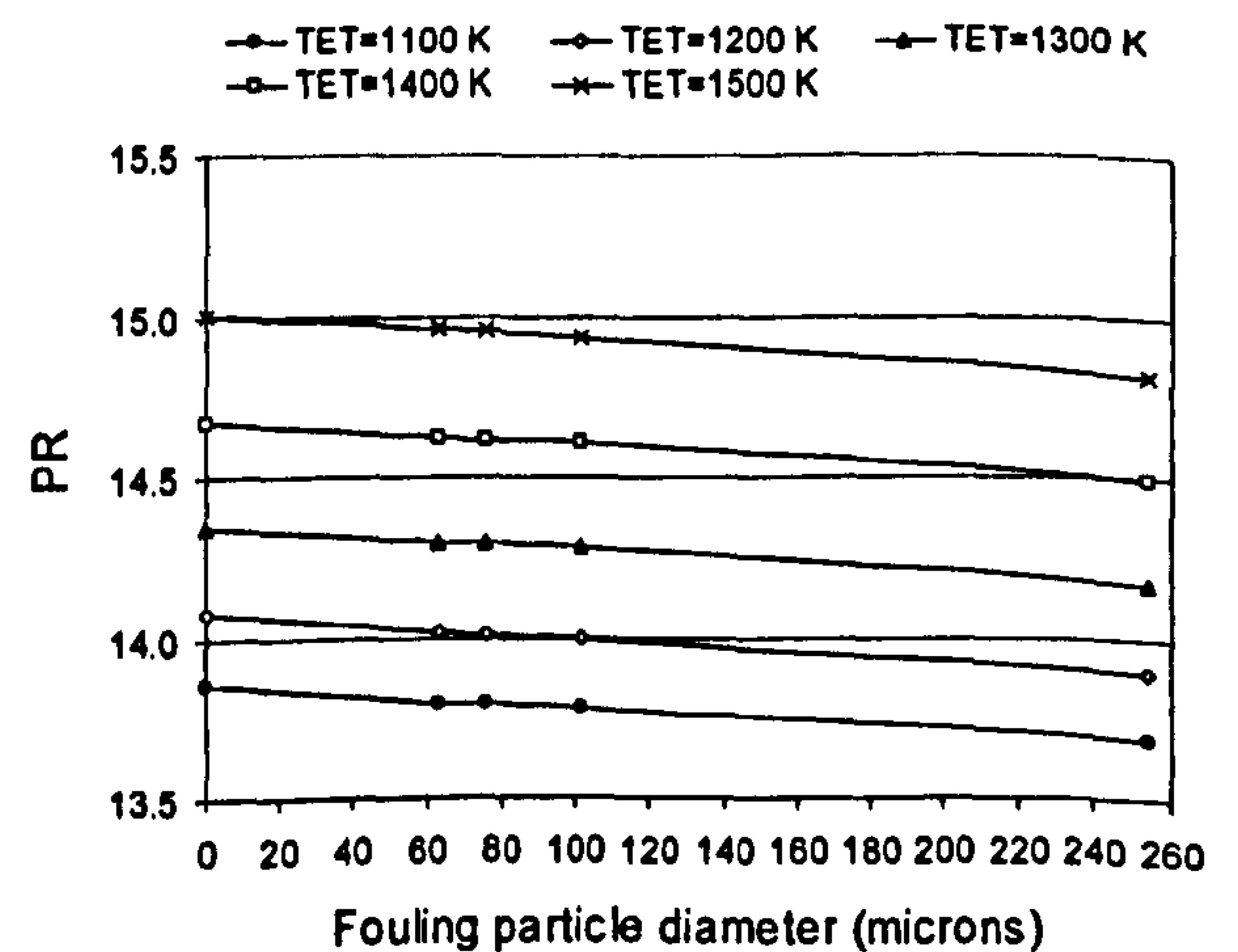


Fig. 14 Pressure ratio reduction due to fouling

As a result of the diverging constant pressure lines in the temperature-entropy diagram, Fig. 15, the useful power (UW) progressively increases with increasing turbine entry temperature. As a result of compressor fouling the compressor efficiency reduces and the compressor work increases. As TET increases the effect of increasing compressor work with increasing fouling decreases.

Finally, the outcome is that the engine performance measured by the thermal efficiency is less sensitive to component inefficiency as TET increases. This fact is illustrated by the reducing range of thermal efficiency change with increasing TET shown in Fig. 13. In conclusion the engine performance deterioration due to fouling is highest at low TET.

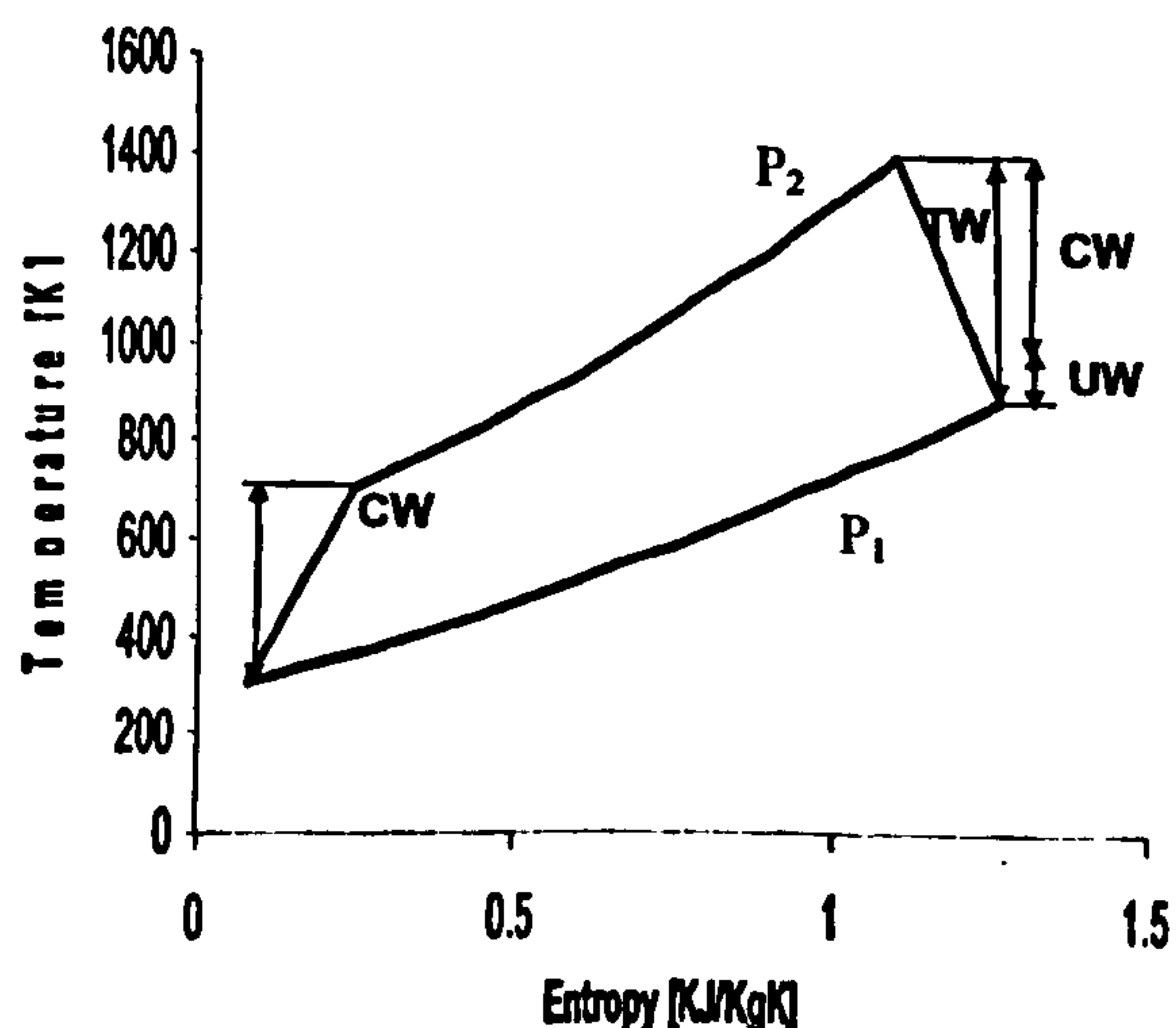


Fig. 15 Temperature versus entropy (Pilandis [23])

From the Turbomatch results obtained it was shown that as the fouling level increases the useful power of the gas turbine decreases but increasing the TET this drawback could be handled (see Fig. 16). Taking into account the case of 0 microns (smooth blades), incorporating a fouling level of 254 microns on the blades at the same TET of 1100 K the useful power drops by 19.2 % and only by 9.8 % for the level of 1500 K. In order for the engine to recover the original useful power at 0 microns after suffering from fouling level of 254 microns at the TET of 1100 K, it has to increase its TET by almost 50 K. For the case of 102 microns passing to 254 microns, the TET should increase less in order to recover the original useful clean engine output.

Also, for constant values of fouling level the useful power was increased linearly with respect to the TET increase. The decrease in useful power as the fouling level (particle size) on the blades increases is caused due to reduction of the engine mass flow capacity.

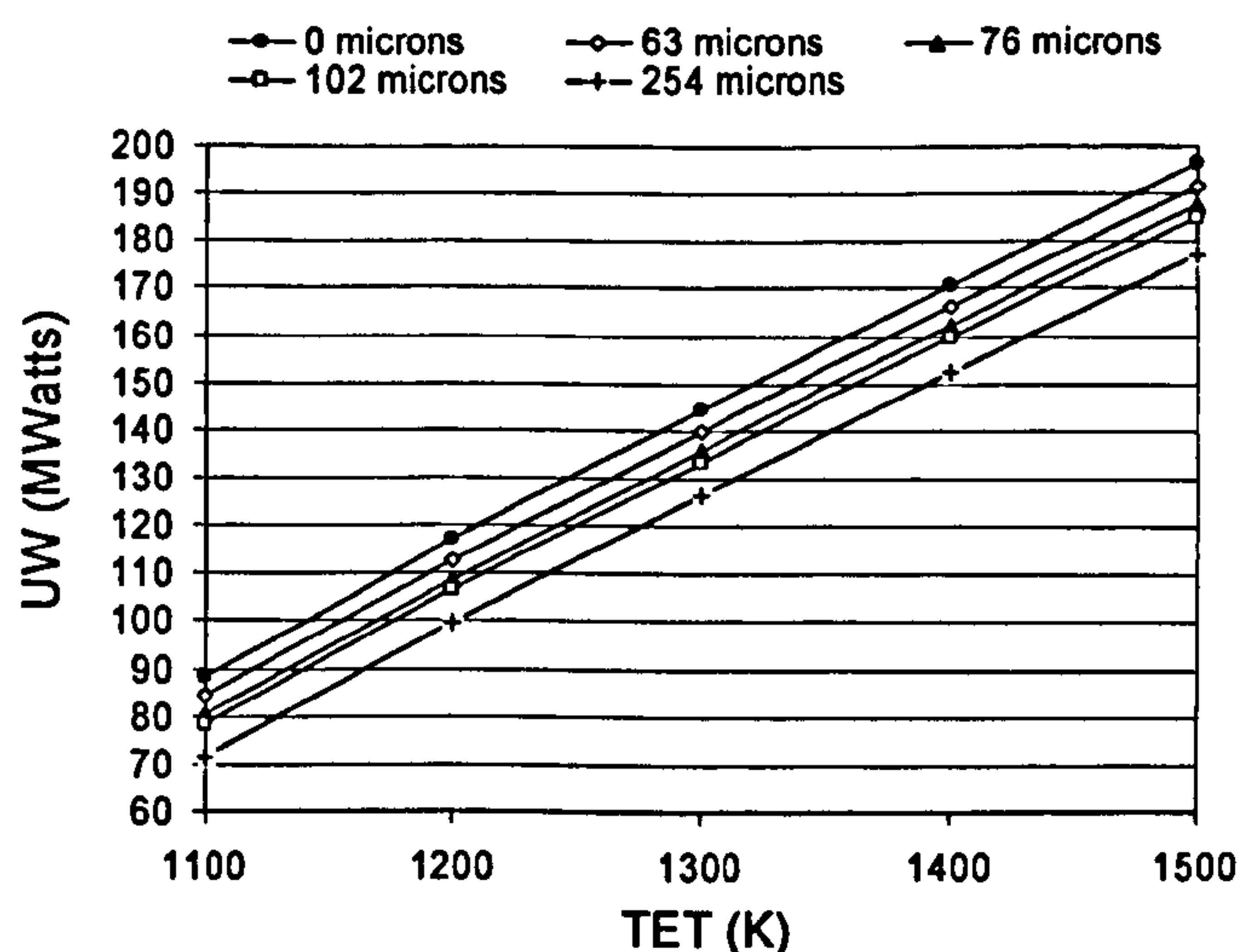


Fig. 16 Useful power versus TET

Using predicted data obtained with the Turbomatch code, Fig. 17 shows that for the industrial engine whose design point is at a TET of 1378 K, the rate of reduction of compressor efficiency with roughness particle size increase is nearly independent of TET. Referring to the TET value of 1100 K, as the particle size increases from 0 to 254 microns the compressor efficiency falls as stated earlier. Also, increasing the TET from 1100 K to 1500 K the compressor efficiency η_c keeps falling by almost 1.2 % for the smooth (0 microns) and all the fouling cases examined. This drop in efficiency is due to the fact that the engine non dimensional mass flow rate decreases as the TET increases. This happens because for a compressor constant speed running line, as the TET increases, the pressure ratio increases (moving from point A to point B) as Fig. 18 illustrates. However, the non dimensional mass flow rate of the engine falls following the trend of the efficiency line passing from the design point shown in Fig. 19.

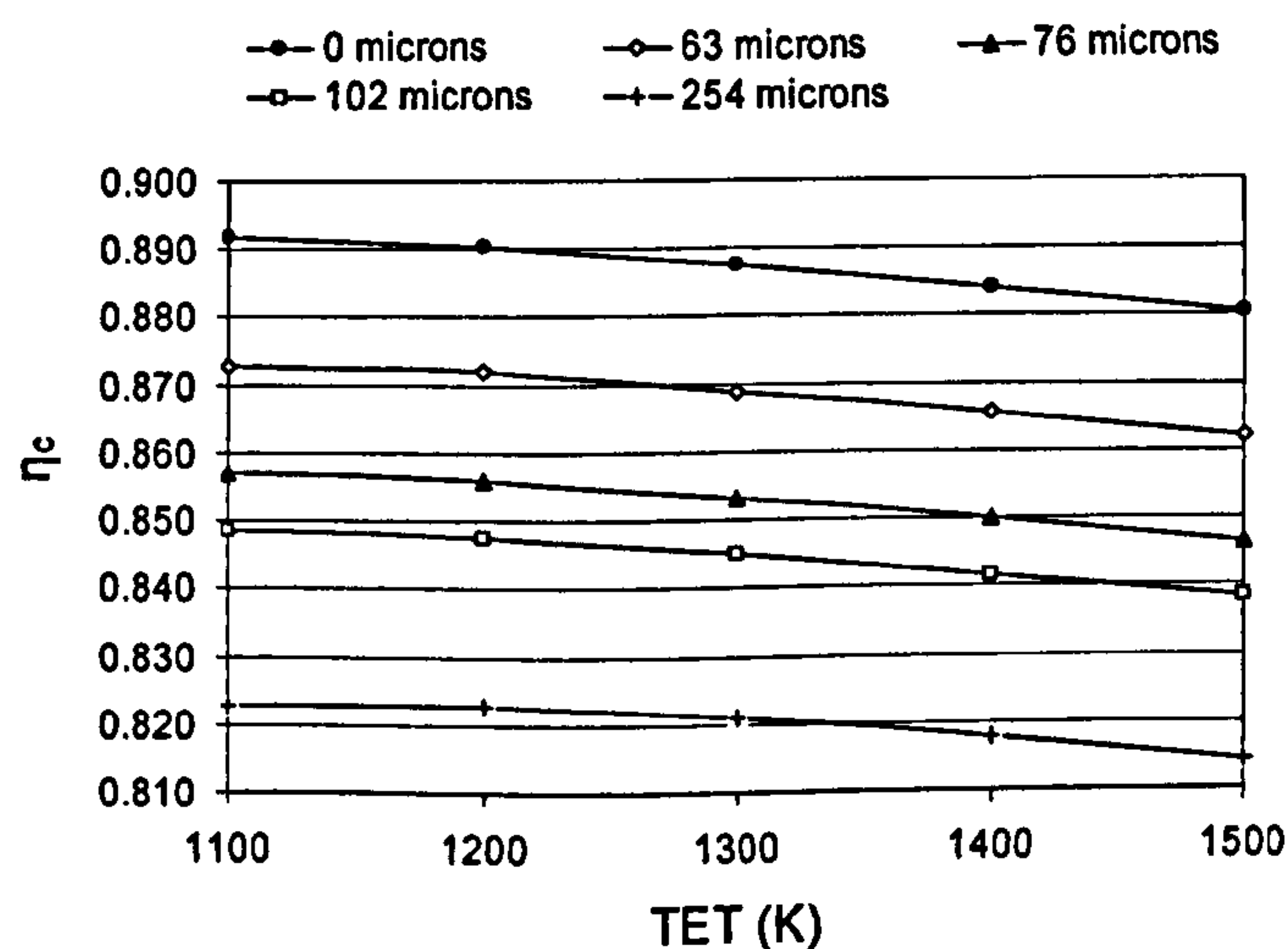


Fig. 17 Compressor efficiency versus TET

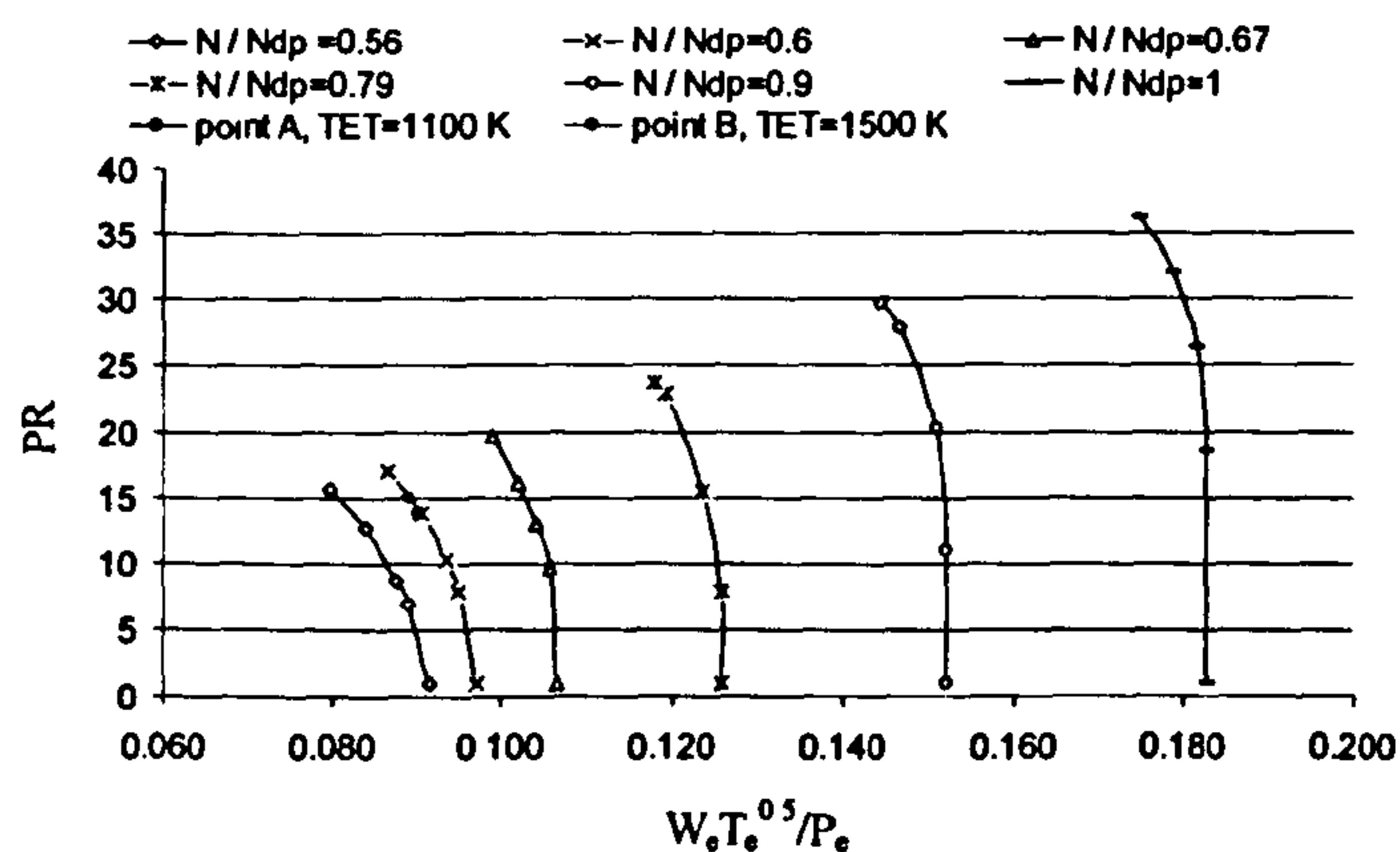


Fig. 18 Industrial engine pressure ratio versus non dimensional mass flow rate

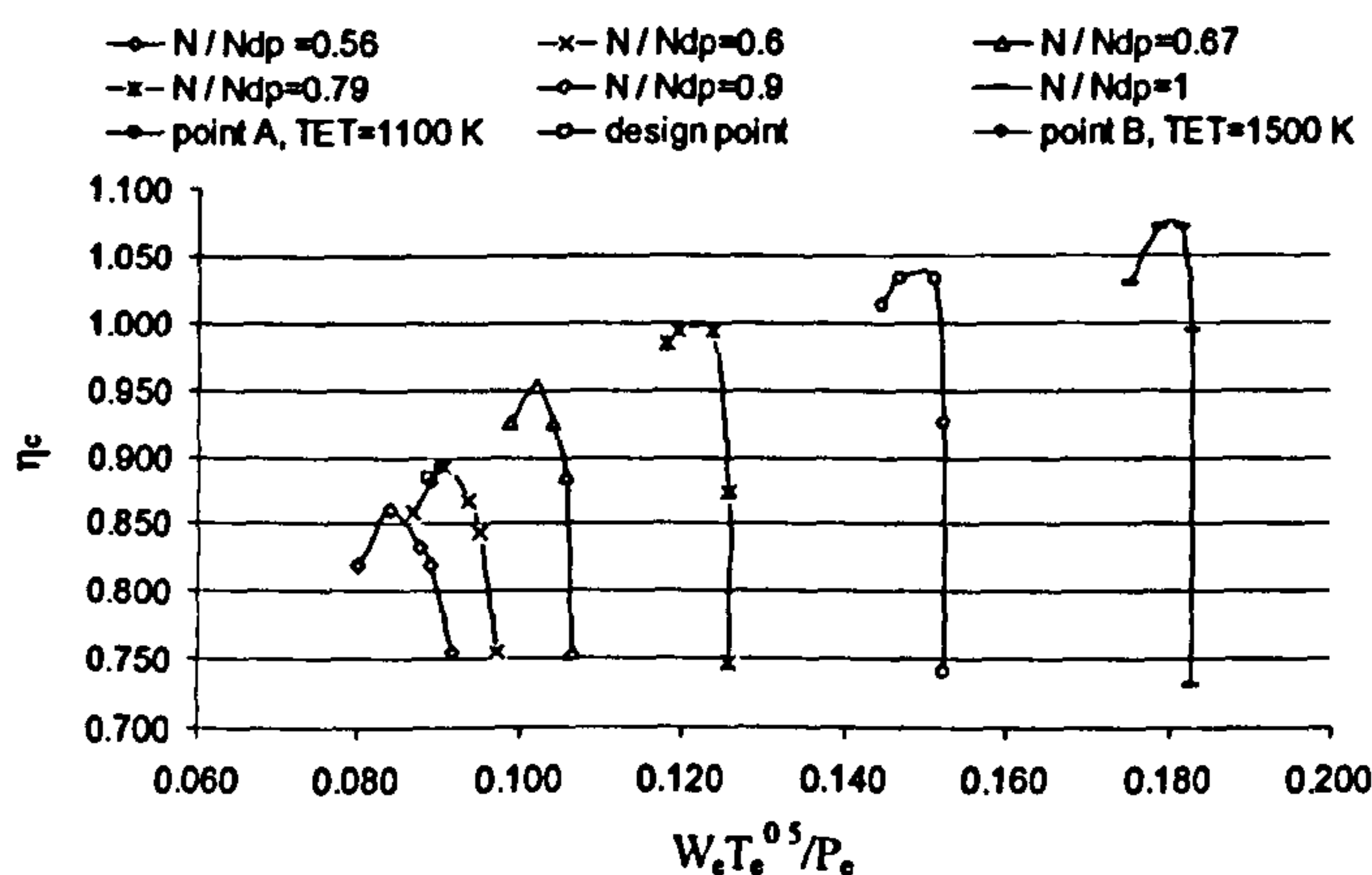


Fig. 19 Industrial engine compressor efficiency versus non dimensional mass flow rate

Keeping the TET constant and increasing the fouling level gradually, the performance drawbacks in terms of percentage compressor efficiency, thermal efficiency and useful power deterioration are illustrated in table 4. From the experimental cascade results correlated to the real uniformly roughened fouled engine via Howell's theory, it was found that by increasing the roughness up to a level of 254 microns the drawbacks in compressor efficiency can be as high as 7.7 %. Taking into account these deterioration percentages in terms of compressor efficiency, increasing the fouling towards the level of 254 microns at 1100 K turbine entry temperature, the percentage deterioration in the useful power ($\Delta UW(1100 \text{ K}) \%$) produced by the engine was found to be 19.2 %. This percentage deterioration was eliminated almost by half, as the TET was increased to 1500 K. For the same TET increase and fouling particle size of 254 microns, the percentage deterioration in thermal efficiency was eliminated by 57.6 %. For lower levels of fouling, and increasing the TET in the same range, similar trends in terms of useful power and thermal efficiency were found. Therefore it can be stated that one performance deterioration inhibitor when fouling presents, is the parameter TET which must be increased properly.

	0 microns	63 microns	76 microns	102 microns	254 microns
$\Delta \eta_c \%$	0	2.2	3.9	4.9	7.7
UW (1100 K) MW	88.2	84.2	80.4	78.3	71.3
$\Delta UW(1100 \text{ K}) \%$	0.0	4.6	8.9	11.3	19.2
UW (1500 K) MW	196.5	191.5	187.4	185.0	177.3
$\Delta UW(1500 \text{ K}) \%$	0.0	2.5	4.6	5.8	9.8
$\eta_{th} (1100 \text{ K})$	0.316	0.307	0.297	0.292	0.274
$\Delta \eta_{th} (1100 \text{ K}) \%$	0.0	2.9	5.9	7.6	13.2
$\eta_{th} (1500 \text{ K})$	0.365	0.360	0.355	0.352	0.344
$\Delta \eta_{th} (1500 \text{ K}) \%$	0.0	1.4	2.7	3.4	5.6

Table 4 Percentage efficiency and useful power reductions due to fouling of industrial gas turbine

9 CONCLUSIONS

The analysis of the cascade experiments showed that as a result of increasing the fouling level, the width and the depth of the wakes increased significantly. More significantly it was observed in all fouling cases that the wakes shifted towards the right side of the cascade due to the increased boundary layer growth on the suction surfaces of the blades. Increasing the roughness to 254 microns caused the total pressure loss coefficient (measured one chord downstream of the blades) to double ie, from 0.16 to 0.35. Also, by increasing the level of fouling, the wake corrected velocity defect and the total pressure loss coefficient rise measured downstream of the blades was observed to vary gradually though not linearly.

From the experimental results, it can be observed that in the middle of the blade passage, the increase in the exit flow angle when the fouling level was raised from 0 to 102 microns was reduced by almost half when fouling was further increased from 102 to 254 microns. This is due to the resulted added exit flow blockage related to the widening of the wakes as the roughness increases.

The paper presents a correlation between the compressor cascade tested and a real compressor stage. Howell's theory of correlating cascade data with compressor stages was employed. The performance simulation tool Turbomatch was used to examine a real engine subject to the fouling comparable to the four different fouling cases examined experimentally. The results showed that by increasing the fouling level, the deterioration in both polytropic and overall efficiencies of the compressor increases continuously. By increasing the fouling level to 254 microns, the percentage deterioration in compressor efficiency reached a value of 7.7 % when compared to the smooth blades.

As the turbine entry temperature increases, the thermal efficiency of the engine increases as well as a result of the increase in compressor pressure ratio. Also, for the same range of TET increase, the higher the level of fouling on the blades, the higher the gain in thermal efficiency is, compared to clean blades. Keeping the TET constant and increasing the fouling level, the pressure ratio drops and the thermal efficiency decreases.

The useful power increases linearly with the TET, and for constant TET, by increasing the fouling level, the power reduces due to the reduction in the mass flow capacity of the engine. At lower TETs, the decrease in useful power due to fouling is higher than that corresponding to higher TETs as a result of the extra margin in useful power incorporated as the diverging lines of the Temperature-Entropy diagram show.

The compressor efficiency falls when the fouling level on the blades increases due to the pressure ratio degradation caused. However, for the compressor examined, by increasing the TET from 1100 K to 1500 K, the efficiency falls by 1.2 % no matter how severely fouled the compressor may be. Again, in order for the engine to recover the loss in useful power due to the increasing compressor work caused by fouling, the TET must increase. Increasing the TET by 36.4 % (from 1100 to 1500 K) and keeping the fouling level constant, the engine percentage performance deterioration in terms of useful power and thermal efficiency was reduced by almost one half.

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NOTATION

Abbreviations

CW	Compressor power
PS	Pressure surface
PR	Pressure ratio
SS	Suction surface
TET	Turbine Entry Temperature
TW	Turbine power
UW	Useful power

Latin symbols

C_D	Drag coefficient
C_L	Lift coefficient
c	Blade chord
c_p	Specific heat at constant pressure
H	Blade height
I	Polytropic efficiency
K	Roughness height (particle size)
k	Roughness parameter
k_s	Equivalent sand grain roughness
l	Assessment length

N	Engine rotational speed
P	Total pressure
P_{amb}	Ambient pressure
p	Static pressure
R	Stage reaction
R_a	Arithmetic average roughness
R_z	ISO 10 point height roughness parameter
s	Blade pitch
T	Total temperature
T_{amb}	Ambient temperature
U	Blade speed
V	Velocity
W	Mass flow rate

Greek symbols

α	Air flow angle
β	Blade metal angle
Δ	Change in value
δ	Yaw probe wedge angle
η	Efficiency
λ	Work done factor
ρ	Density

Subscripts

0	Stator outlet
1	Rotor inlet, cascade inlet
2	Rotor outlet, cascade outlet
3	Stator inlet
4	Stator outlet
a	Axial, annulus
c	Compressor
dp	Design point
e	Engine compressor inlet
m	Mean
p	Profile, peak, passage, polytropic
s	Secondary, stage
th	Thermal

10.2 Appendix 2

Sulzer Type 7 gas turbine Off-Design simulation using turbomatch code

```
! INDUSTRIAL GAS TURBINE:
Sulzer Type 7 POWER GENERATION
50 Hz SIMPLE CYCLE
MODELLED BY A. GANNAN / AUGUST 2008
////
OD SI KE VA FP
-1
-1
INTAKE S1-2      D1-4      R100
COMPRES S2-3     D5-11     R102   V5  V6
PREMAS S3,4,22   D12-15
BURNER S4-5      D16-18     R104
MIXEES S5,22,6
TURBIN S6-7      D19-26,102,27      V19 V20
NOZCON S7-8,1    D28      R110
PERFOR S1,0,0    D19,29-31,110,100,104,0,0,0,0,0,0,0
CODEND

BRICK DATA ITEMS////
! INTAKE
1 0.0      ! ALTITUDE
2 0.0      ! ISA DEVIATION:Tamb=288.15 K, Pamb=1.01325 bar
3 0.0      ! MACH NUMBER
4 -1.0     ! PRESSURE RECOVERY
! COMPRESSOR
5 0.85     ! SURGE MARGIN
6 1.0      ! DESIGN SPEED
7 7.6      ! DESIGN PRESSURE RATIO
8 0.85     ! ISENTROPIC EFFICIENCY
9 1.0      ! ERROR SELECTION
10 5.0     ! COMPRESSOR MAP NUMBER
11 0.0     ! RELATIV TO DP VARIABLE STATOR ANGLE
! SPLITTER
12 0.96    ! LAMBDA (W)
13 0.0     ! DELTA (W)
14 1.0     ! LAMBDA (P)
15 0.0     ! DELTA (P)
! BURNER
16 0.05    ! PRESSURE LOSS 0%
17 0.99    ! EFFICIENCY
18 -1.0    ! FUEL FLOW (-1 = TET SPECIFIED. SEE SV DATA)
! POWER TURBINE
19 10600000.0 ! AUXILLARY POWER REQUIRED
20 0.8     ! NON DIMENTIONAL MASS FLOW
21 0.6     ! NON DIMENTIONAL SPEED
22 0.87    ! ISENTROPIC EFFICIENCY
23 -1.0    ! RELATIV ROTATIONAL SPEED
```



```

24 1.0      ! COMPRESSOR NUMBER
25 3.0      ! TURBINE MAP NUMBER
26 1000.0   ! POWER INDEX N
27 0.0      ! ANGLE
! NOZCON
28 -1.0     ! AREA FIXED
! PERFORMANCE
29 1.0      ! PROPELLER EFFICIENCY
30 0.0      ! SCALLING INDEX
31 0.0      ! REQUIRED DP NET THRUST OR POWER OUTPUT FOR PT

-1
1 2 64.3    ! MASS FLOW
5 6 1198.0  ! TURBINE ENTRY TEMPERATURE
-1
-1
5 6 1000.0 ! --New OD Calculation; DT=TAMB(STANDARD)-TAMB(ACTUAL)=0; TET =
1000.0K
-1
-1
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-1
-1
5 6 1200.0  ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
5 6 1300.0  ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
5 6 1400.0  ! OD Calculation; DT=0; TET = 1400.0K
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-1
-1
5 6 1300.0  ! OD Calculation; DT=-5; TET = 1300.0K
-1
-1
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-1
-1
5 6 1100.0  ! OD Calculation; DT=-5; TET = 1100.0K
-1
-1
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-1
5 6 1100.0  ! OD Calculation; DT=-10; TET = 1100.0K
-1
-1

```


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-1
5 6 1300.0 ! OD Calculation; DT=-10; TET = 1300.0K
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-1
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-1
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