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CRANFIELD INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL ENGINEERING

Ph.D. Thesis

I.W.EAMES

A Theoretical and Experimental Study
of a Novel Refrigerant Compressor

Supervisor:

Prof. R.L.Elder

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degree of Doctor of Philosophy.

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SUMMARY

The aim of this thesis is to evaluate the findings of research undertaken into the design manufacture and testing of a novel refrigeration compressor for use in packaged air conditioner units of the type used in railway passenger transport systems.

A novel refrigerant compressor concept was envisaged consisting of a 'small scale' centrifugal compressor driven by a high frequency induction motor mounted on a common shaft with an impeller supported in aerodynamic bearings. Compared with conventional units this unique combination of 'state-of-the-art' compressor, bearing and motor technologies potentially provides refrigerator designers with improved system performance and better compressor reliability with significant reductions in weight and physical size at reduced capital and running costs.

A prototype unit incorporating the principle features of the novel compressor concept was designed, manufactured, tested and evaluated. A detailed description of the prototype compressor unit is included in Chapter 1. The concept is compared with conventional compression systems and key areas of research requiring detailed investigation are identified. These include; compressor stage performance, electric motor cooling, stress analysis of the motor, drive shaft and bearing system design studies and a computer simulation of a refrigerator system incorporating the proposed compressor unit. Each of these critical aspects of the study are discussed and evaluated within the remainder of the thesis. Conclusions are drawn and recommendations made for future research in Chapter 8. Because of the unique design of the compressor unit a significant proportion of these detailed investigations lay outside documented practices.

Chapter 2 evaluates the results of a literature survey into performance of centrifugal compressors. Recommendations are made for the modification of the present compressor unit directed towards improving aerodynamic performance.

The design of the electric motor is described in Chapter 3 where investigations into the mechanical integrity of the

motor and the performance of the cooling system are evaluated.

An investigation into the design of the drive shaft and bearing assemblies is evaluated in Chapter 4. Undamped lateral modes of vibration of the drive shaft assembly were obtained using a finite element program based on the M-P technique. The natural torsional modes of vibration were estimated using the 'equivalent shaft' method. External and internal sources of machine vibration are evaluated. These include; aerodynamic buffeting, imbalance of magnetic forces and out-of-balance mechanical forces. The design of a hybrid bearing system used for the prototype compressor unit is described and evaluated.

The manufacture of the prototype compressor unit is described and evaluated in Chapter 5. Aspects of the manufacture considered are; surface finishes, tolerancing, heat treatments and balancing processes. Conclusions are drawn and recommendations are made.

Testing of the compressor unit is described and evaluated in Chapter 6. The compressor was tested in both air and refrigerant environments. The construction of a 'Vapour Loop' test facility is described. Development problems encountered during testing are discussed and recommendations for future research. Results of testing were encouraging and demonstrated the feasibility of the novel compressor concept, the subject of this study.

A computer simulation study of the behaviour of a refrigeration system incorporating the prototype compressor is described and evaluated in Chapter 7. Calculations were based on steady state performance models for the compressor, evaporator, condenser, suction line throttle, hot gas bypass valve, expansion valve and motor. It was concluded that the compressor unit would operate satisfactorily when fitted into a refrigeration system with the same design point characteristics as conventional packaged air conditioner units. This theoretical study also predicted the effect on the refrigerator system and compressor part-load performance resulting from changes in compressor speed, evaporator and condenser capacity. A 'single-line' operating characteristic was shown to provide higher compressor efficiencies compared with a 'floating-point' type of control system. The results of the simulation study indicated that values of C.O.P. for a

refrigeration system incorporating the novel compressor compare favourably with those of commercial units and with values which must be achieved by a commercially competitive air conditioner unit. It was concluded that future research should include an investigation into the design and application of a microprocessor based proportional type control system for the compressor unit.

The theoretical and experimental research evaluated in this thesis confirms the feasibility of the novel compressor concept. The results have laid the foundation for the development of a production machine which could have applications in a wide variety of process industries including refrigeration, air conditioning and heatpump uses.

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I would like to acknowledge the craftsmanship of the technical staff in the manufacture and testing of the prototype compressor unit and for their advice in its design.

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NOMENCLATURE

The following table lists nomenclature not otherwise defined within the text. The nomenclature used in the Appendices is included at the end of the relevant Appendix.

CHAPTER TWO

H_{ad}	Adiabatic enthalpy rise
m	Mass Flow Rate, (kg/s)
P	Pressure, (Pa)
R	Radius, (m)
V	Velocity Relative to Impeller, (m/s)
V_w	Whirl Velocity Relative to Casing, (m/s)
W	Velocity Relative to Casing, (m/s)
β, β'	Vane Angle; see Figure 2.1(a)
α	Air Angle

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1	Impeller Inlet
2	Impeller Outlet
w	Whirl Component
m	Mean Height at Impeller Inlet
R	Radial Component

CHAPTER THREE

C	Air Velocity over Motor Casing, (m/s)
C_p	Specific Heat at Constant Pressure, (kJ/kg/K)
E	Youngs Modulus, (MPa)
e	Strain
$F(r)$	Force at Radius r, (Pa)
k	Coefficient of Linear Expansion, (m/°C)
L	Rotor Length, (m)
m	Mass Flow Rate, (kg/s)
N	Number of Cooling Passages
P	Pressure, (Pa)
P_r	Compressor Pressure Ratio
Q	Heat Transfer Rate, (kW)
R	Gas Constant, (kJ/kg/K)

Nomenclature Continued:

R_o, R_i	Outer and Inner Radius, (m)
r	Radius
T	Temperature, (K)
ΔT	Change in Temperature, (K)
V	Volume Flow Rate, (m ³ /s)
V_R	Rotor Volume, (m ³)
W_P	Pumping Power, (kW)
	Stress, (Pa)
	Poissons Ratio
	Rotational, (rads/s)
	Density, (kg/m ³)
	Angle, (rads)
	Cooling Passage Aspect Ratio

Sufficies

o	Outer Radius
i	Inner Radius
θ	Tangential Component
z	Co-ordinate
w	Motor Winding

CHAPTER FOUR

C	Bearing Clearance, (m)
CFM	Cubic Feet per Minute
D	Bearing Diameter, (m)
d	Orifice Diameter, (m)
F_c	Centrifugal Load, (Pa)
e	Bearing Eccentricity
g	Gravitational Constant
K	Bearing Stiffness, (N/m)
K'	Dimensionless Bearing Stiffness
L	Bearing Length, (m)
M_B	Out-of-Balance Moment, (N-m)
m	Mass of Rotor, (kg)
n	Number of Orifices
P_s, P_{amb}	Supply and Ambient Pressures, (Pa)
R_B, R_J	Radius of Bush and Journal, (m)
R_o, R_i	Outer and Inner Radius, (m),

Nomenclature Continued:

r	Radius, (m)
t	Time, (s)
U	Velocity, (m/s)
V	Volume Flow Rate, (m ³ /s)
W	Bearing Load, (Pa)
w	Rotational Speed, (rads/s)
μ	Viscosity, (N.s/m ²)
θ	Angular Component, (rads)

Sufficies

Opt	Optimum Value
-----	---------------

CHAPTER SEVEN

C_p	Specific Heat at Constant Pressure, (kJ/Kg/K)
F	Mathematical Function
h_i	Heat Transfer Coef. inside Heat Exchanger Tubes
m_r	Refrigerant Mass Flow
N	Rotational Speed, (rpm)
N	Dimentionless Speed
PSF	Pressure Ratio Scale Factor
Q	Heat Transfer Rate, (kW)
T	Temperature, (K)
U	Overall Heat Transfer Coefficient, (W/m ² /°C)
V	Volumetric Flow Rate, (m ³ /s)
VSF	Volume Flow Rate Scale Factor
e	Heat Exchanger Effectiveness
$\eta_c \eta_m$	Compressor and Motor Efficiencies
τ	Torque, (N-m)
ρ	Density, (kg/m ³)

Nomenclature Continued:Sufficies

a	Ambient Temperature
c	Condenser
e	Evaporator
od,dp	Off-Design and Design Point
m	electric motor

INTRODUCTION

A NOVEL FLUOROCARBON COMPRESSOR UNIT

The aims of this project were to design, manufacture and test a novel fluorocarbon compressor and to assess the performance of the unit in an air conditioner refrigeration system by using computer based mathematical models.

The novel refrigerant compressor concept was envisaged as consisting of a 'small scale' centrifugal compressor driven by a high frequency induction motor. The motor and compressor impeller were to be mounted on a common drive shaft supported by aerodynamic bearings. This unique combination of 'state-of-the-art' compressor, bearing and motor technologies would offer refrigeration system designers an advanced compressor which would increase system performance and reliability at lower capital and running costs than conventional compressors.

Wood¹ in a discussion document, commented that the compressor, (the subject of this study), would be required to produce a refrigeration effect of 7 Tons of Refrigeration, (24.6 kW), with condenser and evaporator temperatures of 135°F, (57.8°C), and 40°F, (4.5°C), respectively. These data approximate to the design point data of the UP31 packaged air conditioning unit manufactured by Stone Platt for railway carriage roof mounted application reported by Graham². The design point data for the UP31 unit are listed in Table I.1 with data for the larger RP40 unit manufactured by the same company for comparison.

Some preliminary research had been undertaken by Elder and Walters³ who had tested a highly modified turbocharger impeller and volute on Freon 11, a CFC refrigerant. Figure I.1 and I.2 show the compressor characteristics produced by Elder and Walters³. Freon 11 was used because it has a large specific volume at normal ambient temperatures compared with other commonly used air conditioning refrigerants such as Freon 12 or Freon 22. This makes Freon 11 particularly suitable for use with centrifugal machines. A disadvantage in using Freon 11 is that evaporator and suction line pressures tend to be sub-atmospheric and therefore air and moisture leakage into the system may take place. This can affect the performance of a refrigerator, promote corrosion and encourage the chemical breakdown of the refrigerant.

Techniques have been developed to remove air and moisture from refrigeration systems but these add to the capital and maintenance costs. There are a number of considerations when selecting refrigerants for particular applications. Duffield and Hodgeth⁴ presented an interesting and detailed review for heat pump applications.

From the Freon 11 refrigerant data given in the p/h diagram in Figure I.3, it was estimated that compressor pressure ratio 6 and mass flow rate of 633 kg/s would be required to meet the design point condition suggested by Wood¹. These results equate to an isentropic power consumption of 6 kW.

It was necessary to find a centrifugal compressor which would have a duty similar to that suggested by Wood¹ and Graham². Elder and Walters^{3,5} had investigated the performance of a highly modified Holset H2A turbocharger compressor compressing Freon 11. Figure I.1 and I.2, derived by Elder and Walters, show the pressure ratio/flow and efficiency performance/flow relationships resulting from their experiments. For comparison Figures I.4 and I.5 are included which show the performance characteristics for the same compressor operating on air. Elder and Walters used an air turbine to drive the compressor impeller. Using Woods¹ design point requirements, (inlet temperature of 4.5°C, inlet pressure of 0.49 bar and mass flow rate of 633 kg/s, the corrected volume flow rate through the H2A compressor would be 130 ccfm equating to a maximum pressure ratio of 3.9 at 60,000 rpm, (from Figure I.1). This result showed a mismatch between the design point duty suggested by Wood¹ and the modified Holset H2A compressor tested by Elder and Walters³. To achieve the 'design point' inlet conditions and pressure ratio of 6, (suggested by Wood), the compressor speed would have risen to 70,000 rpm whilst the mass flow through the compressor increased to 0.247 kg/s and the refrigeration effect would have increased from the required 24.6 kW to 34.6 kW. It was concluded from this study that the duty provided by the modified H2A compressor matched more closely the performance of Stone Platts RP40 packaged air conditioner rather than the smaller UP31 unit as shown from the design data listed in Table I.1. The duty flow rate could be achieved by throttling the compressor inlet flow and the design point discharge pressure maintained by increasing the compressor speed which would also have increased the overall pressure ratio. The problem of overcapacity was not considered a significant factor for the present study. The resulting prototype compressor would allow further development. Because of the availability of

the Holset modified H2A compressor tested by Elder and Walters it was decided to incorporate this into the design of the prototype compressor.

Wood¹ had suggested the isentropic efficiency of the compressor would need to be greater than 70% if it was to compete with existing reciprocating machines which were reported by Wood to have values of the order of 75%. Assuming an electrical conversion efficiency of 85%, (a typical value), an annual utilization of 30% and an electricity cost of 5.2p/kW.hr, (1988 UK commercial rate), the additional annual running cost resulting from a reduction in isentropic efficiency from 75% to 70% would be £92. If the efficiency was to fall to 60% then the annual running costs would increase by £321. The differential in running costs are significant when compared with the purchase price of a conventional compressor with a similar duty. An ASPERA type C , 8 kW, reciprocating compressor designed for air conditioning would cost approximately £1077, (ex VAT), at 1986 prices, HRP⁴.

Elder and Walters³ demonstrated an isentropic efficiency of 70% at a pressure ratio of 6 for a modified Holset H2A compressor operating on R11 at a rotational speed of 71000 rpm. This result was encouraging and supported the case for further research which led to the present study. As part of this study an examination of the results produced by Elder and Walters showed that the reported efficiency values were over estimated by approximately 10%. This was due to the assumption that enthalpy change across the compressor could be accurately calculated by assuming the gas to be ideal. The efficiency characteristics were redrawn using real gas data in Figure I.6. The reasons for this short fall in efficiency and the modifications to the compressor required to improve the performance were investigated. The results of this study are evaluated in Chapter 2.

A significant factor for undertaking the research into this unique compressor was the potential for a low cost and reliable machine. Manufacturing costs are difficult to estimate, Dieter⁷, (see also Matousek⁸ , French⁹ and Asimow¹⁰). A common approach is to use the basic material cost and to multiply this by a constant factor to cover production and then to multiply the result by a further factor which depends on the batch quantity. A production cost of £73.2 was estimated for a batch quantity of 200

units. Table I.2 shows the breakdown of the costings. The purchase price of an equivalent conventional reciprocating machine, (and Aspera type C), was £1077, H.R.P.⁶; experience suggests that the production costs are probably less than 50% of this. One reason for this cost difference is the quantity of material used in the proposed scheme compared with that used in conventional machines. Not included were the additional costs which would be incurred by a requirement to modify the electrical supply frequency for the novel compressor.

An additional factor for initiating this study was that the compressor design would have the potential for reduced weight. Weight is important because it affects not only the manufacturing costs of the unit but also installation costs. A light weight compressor would require a lighter supporting subframe structure; which is particularly important for application in the railway carriage roof structure.

Refrigerant compressors require a high degree of operational reliability. Often the owner is looking for machines which have minimal maintenance implications as with the compressors used in domestic refrigerators. This would require the novel centrifugal compressor to be at least as reliable as the machine it replaces. The reliability of modern hermetic reciprocating compressors has been improved in recent years and are generally reliable. The main failings in the reciprocating compressor are discussed in a later section. It could be argued that if the cost of the compressor unit was low in comparison with the present equivalent reciprocating machines then a greater degree of unreliability may be tolerated. This argument can not be supported because it was felt that users would not wish to be constantly repairing their system; time spent in the repair shop means loss of revenue for the carriage owner. Labour costs are also a significant factor in repair and maintenance budgets.

Reliability is largely a function of the number of dynamic components in a machine. If components are positioned in series, such that the failure of any individual component results in the failure of the machine, then reliability is equal to the product of individual component reliability measured as the probability that a component will not fail. However, if components are positioned in parallel a failure of one component does not result in a failure of the whole machine. Reliability is thus quantified by the product of

the unreliabilities of individual, redundant and parallel components measured as the probability of component failure. Machine reliability depends very much on the evolutionary stage of individual components. High reliability is often given because individual components are developed over many years. Once development is complete the reliability of the novel compressor would be greater than that of present reciprocating machines because it would have fewer and less heavily loaded dynamic components.

During the present study areas of technology were identified which required further research for the innovative concepts embodied in the novel compressor unit to be realised.

The highly modified centrifugal compressor wheel and volute, chosen to be included in the prototype compressor unit and previously tested by Elder² was shown to have a relative low isentropic efficiency. In order that the compressor unit could compete favourably against existing reciprocating machines it would be necessary for it to have an isentropic efficiency of approximately 70%, Wood¹. So far efficiency values of approximately 60% have been demonstrated, Walters and Elder³. It was necessary, therefore, to identify probable reasons for this shortfall and possible ways to improve the efficiency of the compressor. A study into the performance of centrifugal compressors was undertaken. The results of this study are discussed in Chapter 2 in which some recommendations are given for the future development of the compressor impeller and volute.

The reduced physical size of the electric motor imposed by the high rotational speed and the resulting high energy density meant that cooling would be a critical problem. This aspect of the study is discussed in Chapter 3 where it is concluded that the majority of the cooling could be undertaken by the compressor suction gas drawn through the motor casing. This was augmented by the addition of special cooling passages through the stator and by external cooling via the motor casing. The design of the special cooling passages was optimised in terms of their number, aspect ratio and positioning by considering heat exchange effectiveness and pumping loss.

The high rotational speed of the compressor drive motor imposed severe mechanical stress constraints on the design of the laminated rotor. Once the overall dimensions of the rotor had been decided upon, from both mechanical and electrical stand points, it was found that due to its

laminated construction the rotor would tend to loosen on the drive shaft at high rotational speeds which could cause fretting and the loss of dynamic balance. This was primarily because aluminium conductor rods, (squirrel-cage), used to pre-load the laminations, would have had insufficient strength to provide the necessary pre-load. Calculations showed that steel rods would have been too massive; causing the fitment holes to fail due to excessive stressing at the design speed of the compressor. It was concluded from the study that the necessary pre-load could be provided by a torque loaded slotted nut.

Knowledge of the dynamic behaviour of the drive shaft was identified as an area of particular interest in this study. Because of the high rotational speed of the drive shaft and the use of gas bearings it was important to acquire accurate estimates of the critical speeds to ensure that they did not fall within the normal operating speed range of the compressor. In view of importance of this knowledge to the mechanical integrity of the design a finite element program based on the Myklesad-Prohl method was written to determine the lateral critical speeds. The results of this analysis are included in Chapter 4.

The bearing system was identified as an important element in the design of the compressor unit. Hybrid thrust and journal bearing systems were devised which gave a degree of operational flexibility during the testing. This study is included in Chapter 4.

In order to prove feasibility of the novel compressor concept a prototype unit was manufactured and tested. The manufacture of the unit is described in Chapter 5 and the experimental results are described and evaluated in Chapter 6. Many lessons were learned from this aspect of the study which will be of value to future related studies.

A compressor unit forms a single component of a refrigeration system. A study into the behaviour of the compressor when installed into a refrigerator was undertaken in order to gain an understanding of the operating requirements placed on the compressor unit. A computer based simulation model was developed to enable this part of the study to be carried out. This research enabled a more precise definition of the compressor design requirements to be made than had been previously available, Wood¹. The results of this study are described and evaluated in Chapter 7.

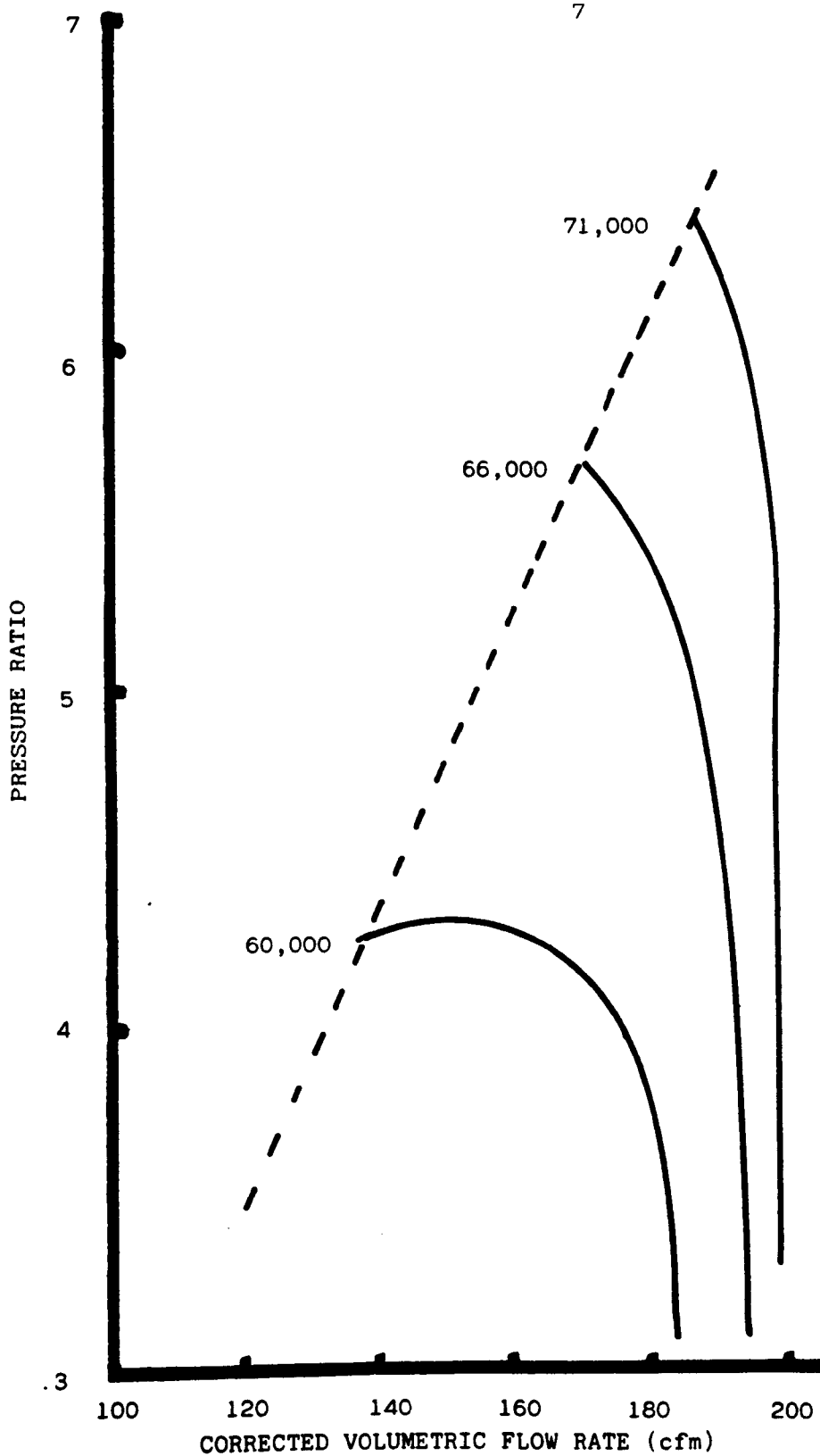


**FIG. I.1 COMPRESSOR PRESSURE RATIO CHARACTERISTIC
(FREON TEST RESULTS)**

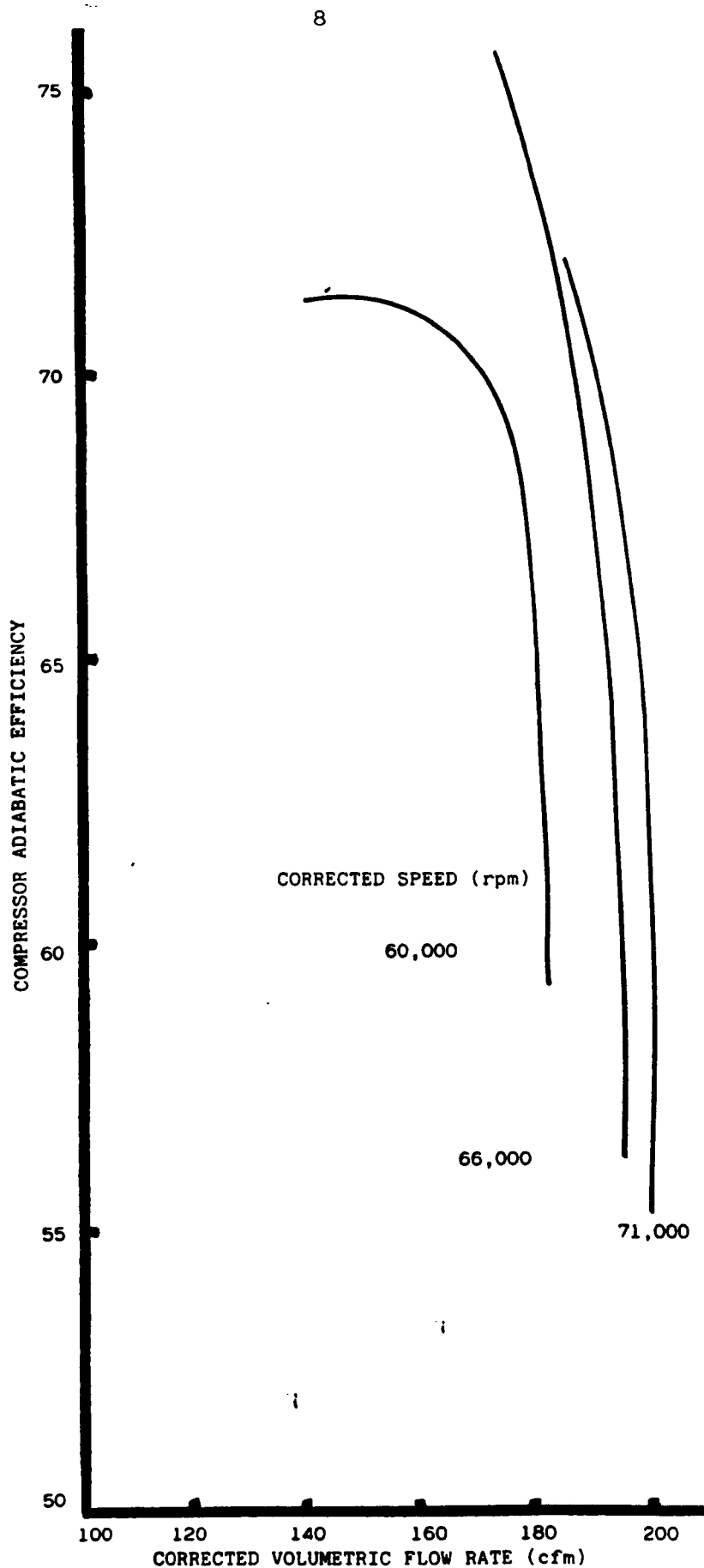


FIG. 1.2 COMPRESSOR EFFICIENCY CHARACTERISTIC (FREON TEST)

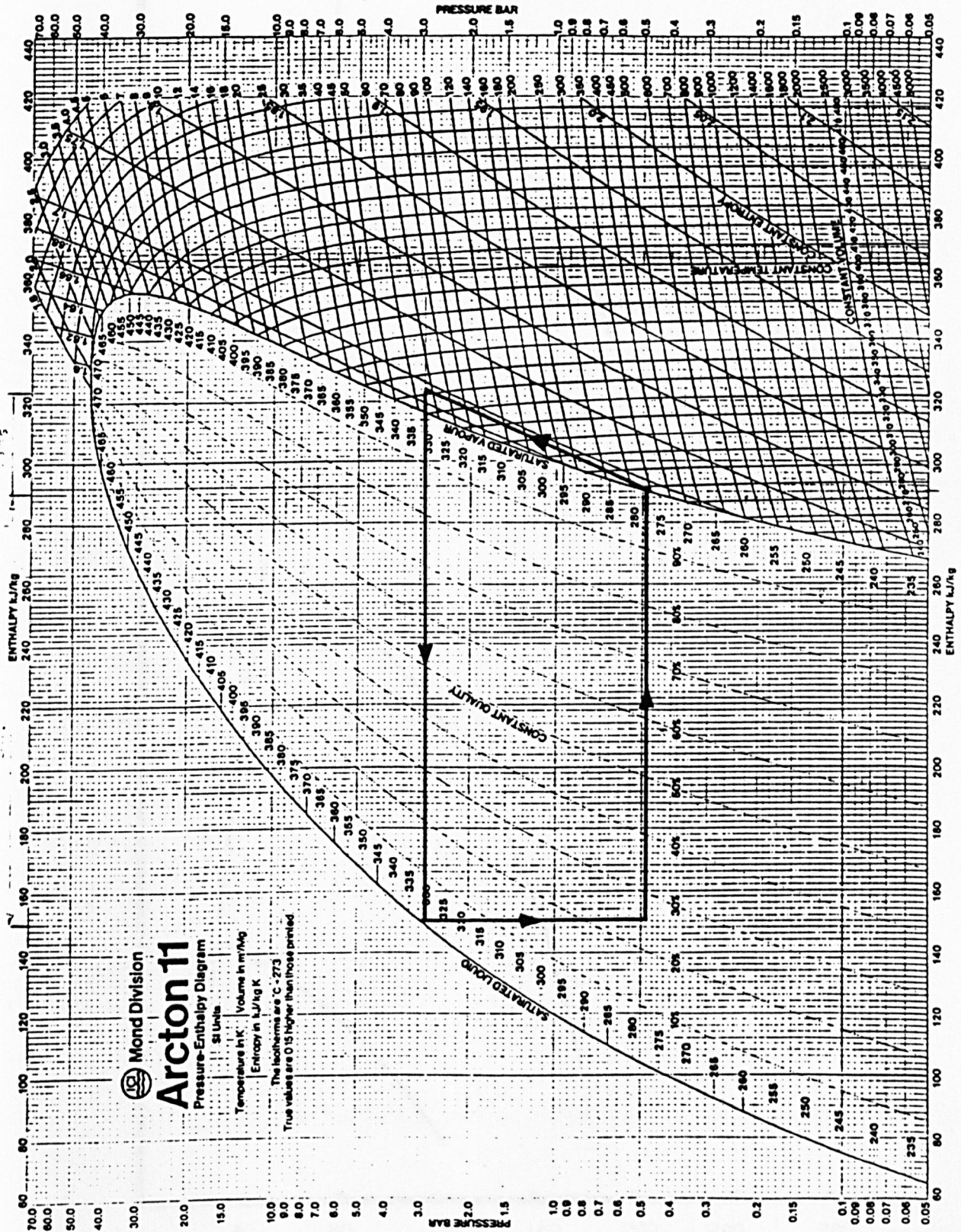


FIG. I.3 P/h DIAGRAM OF AN IDEAL VAPOUR COMPRESSION REFRIGERATION CYCLE

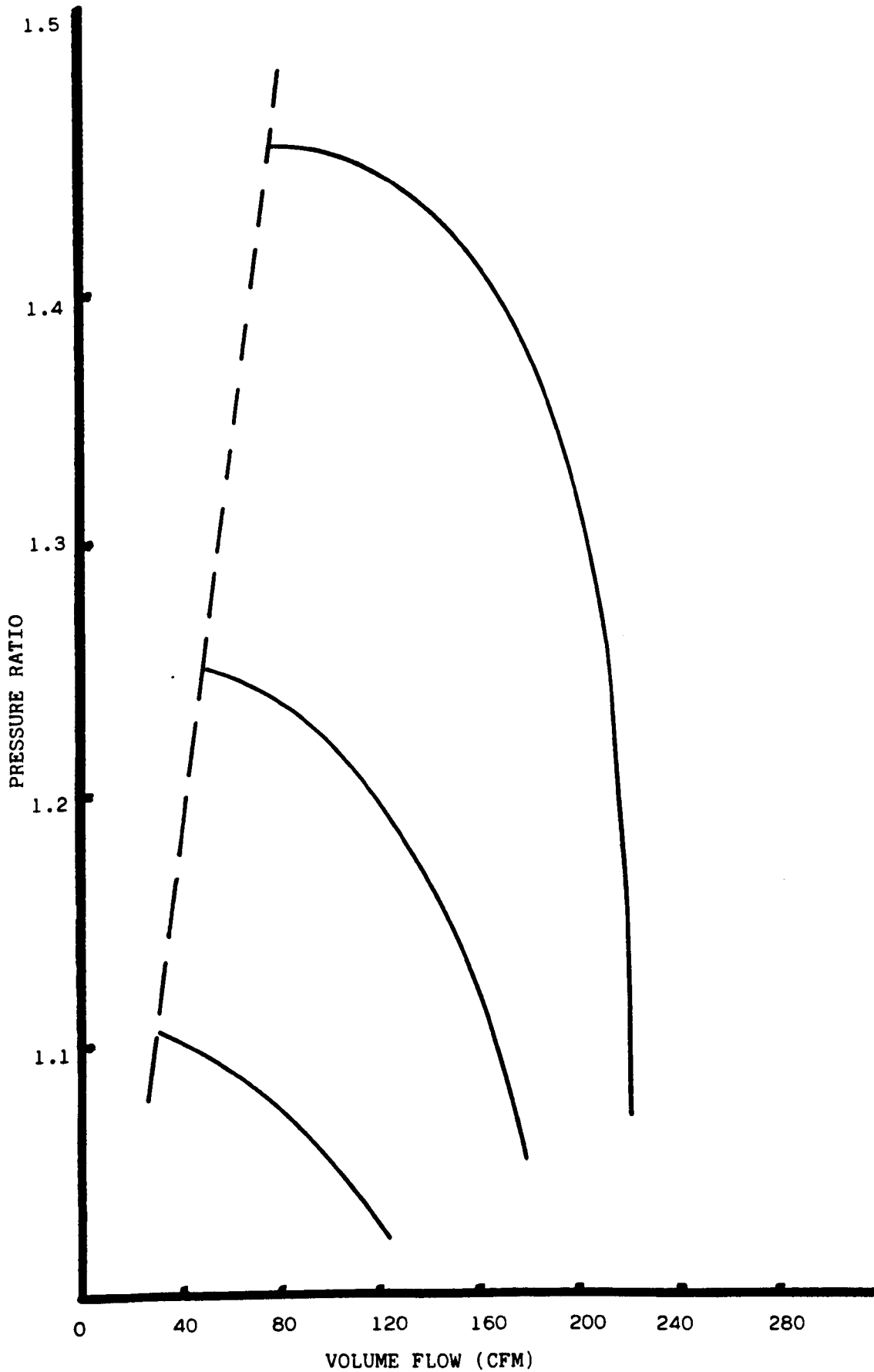


**FIG. 1.4 COMPRESSOR PRESSURE RATIO CHARACTERISTIC
(AIR TEST RESULTS)**

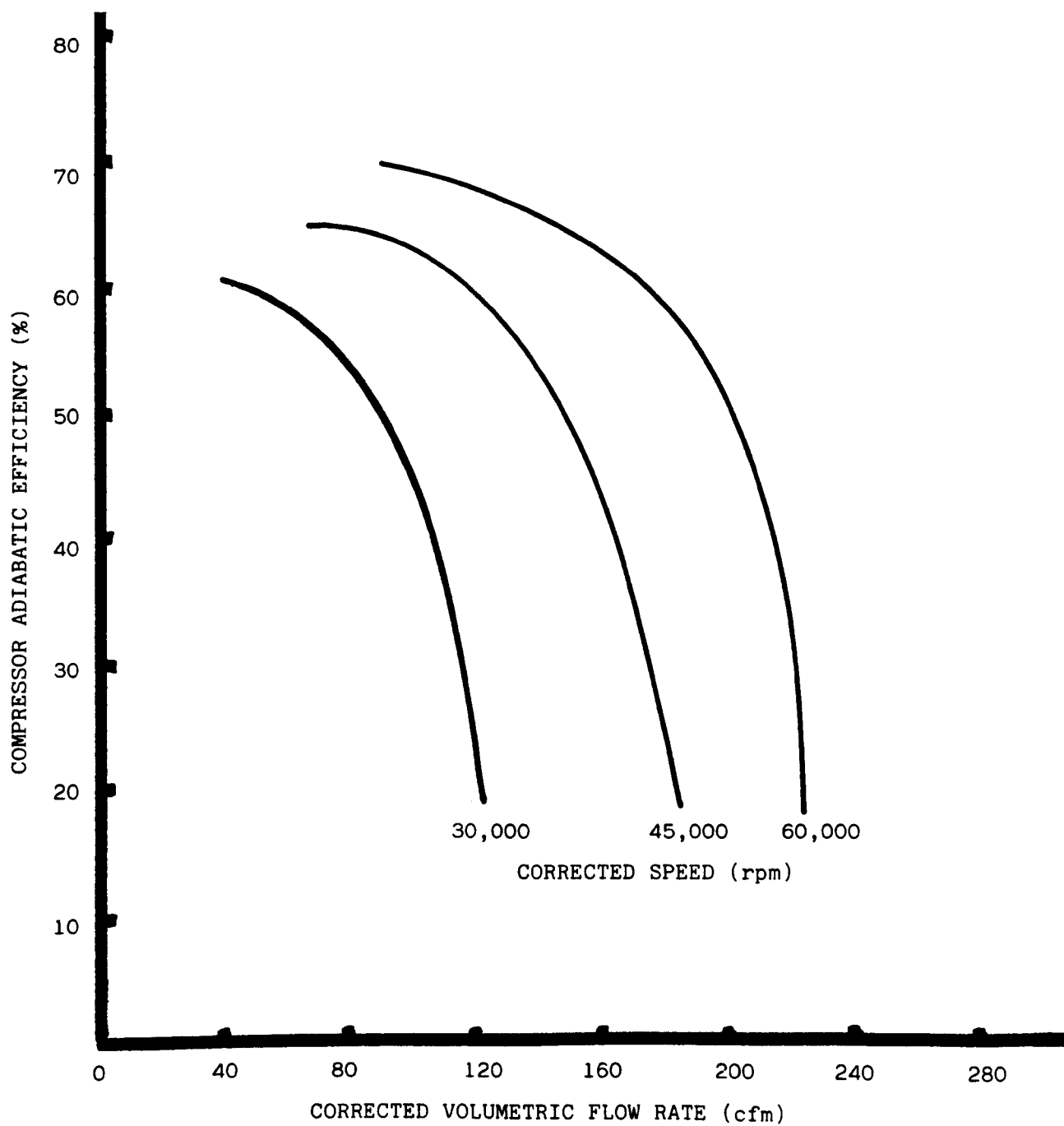


FIG. I.5 COMPRESSOR EFFICIENCY CHARACTERISTIC
(AIR TEST DATA)

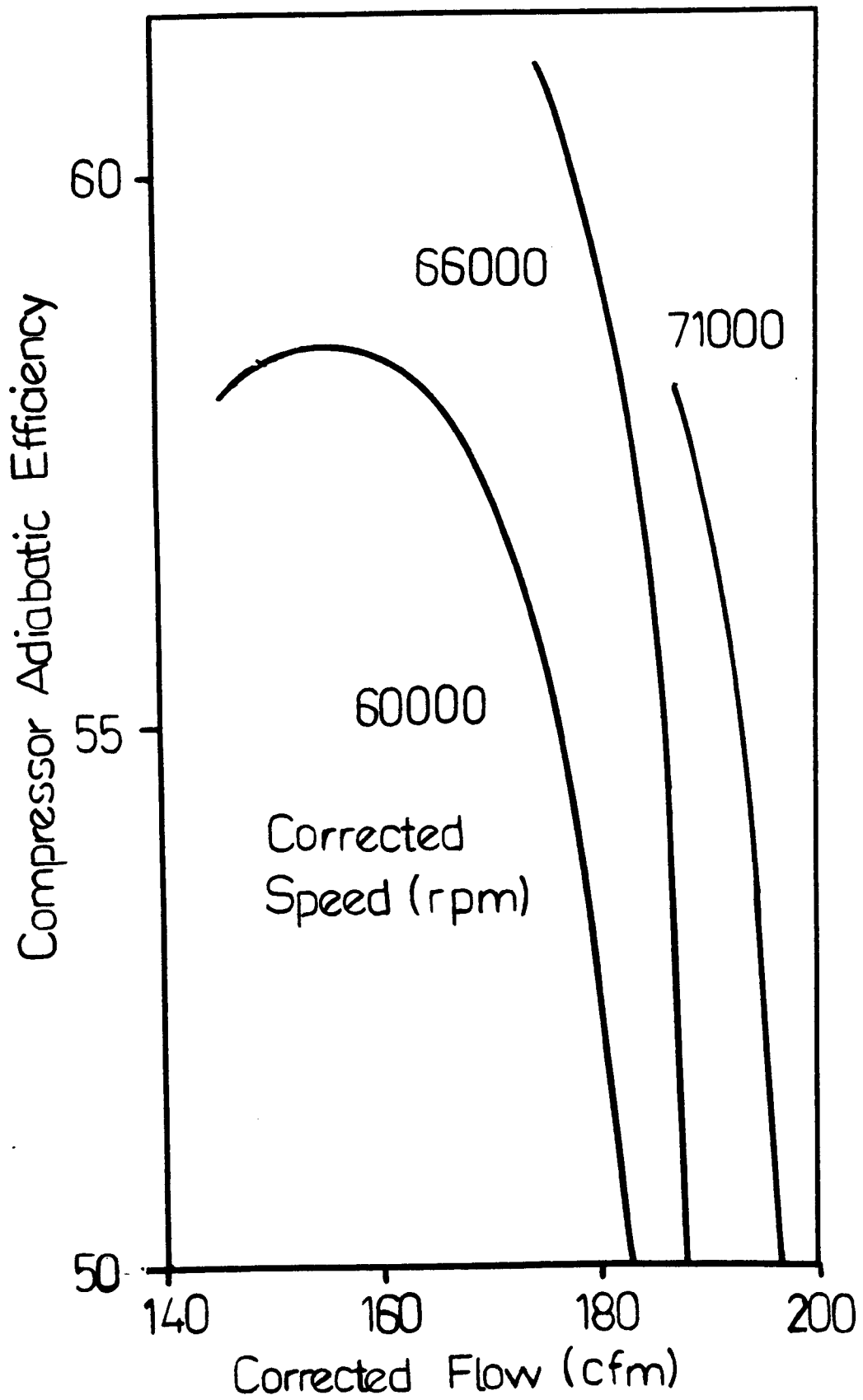


FIG. I.6 (FREON 11) COMPRESSOR EFFICIENCY CHARACTERISTIC

	RP40	UP31
Ambient temperature	33.3°C DB 72% r.h.	30°C DB 50% r.h.
Interior	26.5°C DB 66% r.h.	25°C DB 48% r.h.
Units per coach	2	1
Passengers per coach	300	72
Unit cooling capacity	34,400 kcal/h	21,000 kcal/h
Evaporator air flow	6300 m ³ /h	3800 m ³ /h
Evaporator entering air	28°C DB 75% r.h.	27.2°C 49% r.h.
Condenser air flow	15,300 m ³ /h	11,000 m ³ /h
Condenser entering air	33.3°C DB 72% r.h.	30°C DB 50% r.h..
Evaporator sat. outlet	7.5°C	3.6°C
Compressor sat. suction	6.94°C	2.5°C
Compressor sat. delivery	50°C	50°C
Power input compressor	13.7 kW	8.5 kW
Condenser fan motor	2.2 kW	2.3 kW
Evaporator fan motor	1.4 kW	1.2 kW
Supply	440 V 3ph 60Hz	415 V 3 ph 50 Hz

TABLE I.1 DESIGN POINT DATA FOR EXISTING AIR CONDITIONING SYSTEMS

Item	Material	Mass (kg)	Material Cost (£)	Batch Production Cost (£)
Motor	Steel	3.4	2.55	5.00
Casing	Al	1.4	5.6	11.20
Brg				
Support Assembly (2 off)				
Thrust Brg	Al	1.3	5.2	10.40
Support Assembly				
Rotor	EN 24	0.7	0.54	1.6
Drive				
Assembly				
Motor Ass.	Cu and Trans-former steel Transil 315	10.30		45.00
TOTALS		17.7	13.9	73.2

TABLE I.2 BREAKDOWN OF PREDICTED COSTS FOR DESIGN

CHAPTER ONE

BACKGROUND AND DISCUSSION

OF COMPRESSION SYSTEMS

1.1 Introduction

The Introduction to this thesis included a description of a novel compressor unit and identified the potential benefits of this unique concept and the inherent technological problems to be overcome. Following an iterative design process, which included wide ranging studies of relevant technologies, much of which lay outside documented experience, a suitable design scheme was settled upon. The aim of this chapter is to provide an overall description of the novel refrigerant compressor and to compare the new concept with conventional compressor designs.

1.2 A Description of the Compressor Unit

The prototype compressor unit was a semi-hermetic centrifugal type. Dynamic elements were supported in hybrid gas journal bearings, (mounted on compliant elastomer 'O' seals), and a plane hybrid thrust bearing. The compressor impeller, (a modified vehicle turbocharger type), was driven by a three-phase squirrel-cage type induction motor. A general assembly drawing of the prototype compressor is shown in Figure 1.1. A photograph showing the components of the compressor is included in Figure 1.2 and a drawing of the proposed compressor is shown in Figure 1.3. Manufacturing drawings for individual components are included in Appendix A.

1.2.1 Motor Casing

The motor casing, shown as item 10 in Figures 1.1 and 1.2 was manufactured from a length of mild steel tube, (to B.S.3601/B.S.3602), 6.625 inches O.D. and 0.625 inch wall thickness. The end flanges welded to the casing body were

made from standard blanks: 2 x 9 inches O.D. x 6 inches I.D. x 0.5 inches thick mild steel blanks, (to B.S.4360 grade 43A). Mounting points for the electrical junction box, eight 3/4 BSP connections, (for the motor coolant gas), and two 5mm connections for thermocouple leads and journal bearing touch-down detector leads. The casing was supported on a mild steel foot assembly, (item 21), held in place by a saddle clamp. Flanges welded to each end of the casing provided attachment for the forward and rear bearing support assemblies, (items 8 and 5). Elastomer 'O' rings provided sealing between the flanges and the support assemblies. Pins fitted to each flange ensured accurate drive shaft and motor alignment. A bearing housing alignment tool, (Pt.No PC15044), was used to ensure correct assembly. To ensure that the casing was gas tight, Viton 'O' seals were fitted to each end of the casing, (Size Number 257). Seal seatings were designed to B.S.1806:1962. The induction motor, (item 33 in Figure 1.2), was located by two 1.25 inch B.S.W. brass screws fitted through the casing. Provision was made for the fitment of electrical cables and instrumentation wiring.

1.2.2 Bearing Support Assemblies

The forward and rear bearing support assemblies are shown in Figures 1.1 and 1.2 as items 8 and 5. They were each located by means of dowel pins, (item 19 in Figure 1.1) and secured to the casing by six 5/16 BSF mild steel bolts, nuts and spring and plane washers, (items 23, 24 and 25). The location pins, (item 19 x 2 off), were made to different diameters to ensure that the casing was fitted correctly. Both support assemblies were machined from mild steel.

The forward bearing support assembly provides attachment points for the aluminium alloy compressor volute and back plate shown as items 31 and 16 in Figures 1.1 and 1.2. Provision was provided for the fitting of a mild steel labyrinth gas seal, (item 18); mild steel thrust bearing plates, (items 12, 13, 17 and 22), the forward journal bush, (item 28), and for two drive shaft displacement transducers.

The rear bearing support assembly, (item 5), provided location for the rear bearing bush, (item 26), two displacement transducers, an eddy current pulse generator used to measure the rotational speed of the shaft and a mild steel end cap, (item 7).

1.2.3 Drive Shaft Assembly

The drive shaft is shown in Figure 1.1 and 1.2 as item 2. The shaft assembly was supported on journal bearings, (items 26 and 28 in Figures 1.1 and 1.2), between the forward and rear support assemblies, (items 8 and 5), The journal bearing bushes were machined from a alloy steel, (items 28 and 26), each supported on a pair of size 127, Vitol, 'O' seals, (item 27).

The drive shaft, (item 2 in Figures 1.1 and 1.2.), was manufactured from EN24 steel. Mounted on the shaft were; a Holset compressor impeller which was secured by a 5/16 inch BSF stiffnut and plane washer, (items 30 and 32 in Figures 1.1, 1.2 and 1.4), mild steel spacer, (item 20), mild steel 'special' nut, (item 15), a further mild steel spacer, (item 14), carbon steel thrust bearing disc, (item 11), steel rotor, (item 9), slotted nut and lock washer, (items 3 and 4). Two flats were machined at the rear end of the shaft to enable a spanner to be used to hold the shaft during assembly and provide a reference for the magnetic speed transducer.

The special nut, (item 15), retained the thrust bearing washer, (item 11), and spacer, (item 14). The nut had a left hand thread to prevent unlocking during operation. The nut is tightened using a peg spanner, (part number PC15040/3).

The labyrinth seal, (item 18), consisted of six glands machined into the gas seal body on an inner diameter of 50.1mm. Each gland, in section, resembled a modified B.S.W thread form with a zero helix angle and a 0.25mm flat machined on the thread form apex. The radial clearance between the gas seal and the special nut, (item 15), was 0.05mm.

A light press fit was used to attach the laminated rotor, (item 9 in Figures 1.1, 1.2 and 1.4), to the drive shaft. The rotor was held in place by a slotted nut, (item 3), and lock-washer, (item 4).

1.2.4 Thrust Bearing Assembly

The thrust bearing assembly consisted of items 11, 12 and 13 in Figures 1.1 and 1.2. The bearing assembly consisted of a 75mm diameter thrust washer with a 16 mm diameter central hole. The washer was held in place by a spacer, (item 14), and a special nut, (item 15). Axial thrust was transferred from the drive shaft to the casing via the thrust washer, (item 11), and a pair of thrust plates, (item 13), which maintained the axial location of the shaft. A running clearance of

0.003 inches between the thrust washer, (item 11), and the thrust plates, (item 13), was maintained by a spacer, (item 12). Compressed gas was fed to the bearing via radial drillings and pressure ports, (8 per thrust plate), equally spaced on a 1.9 inch P.C.D.. There were eight 0.03 inches diameter pressure ports in each thrust plate. Seitz 'watchmakers' jewels were used to ensure the geometrical precision of the gas ports.

The thrust bearing pack plus spacers were secured to the internal forward face of the bearing support assembly, (item 8), by eight 1/4 inch BSF cap-head screws.

1.2.6 Impeller Shroud

The impeller volute, (Figures 1.1, 1.2 item 31), was attached to a prepared flange on the forward end of the bearing support assembly, (item 8). The volute was held in place by steel washers and 1/4 inch BSW bolts. The back plate, (item 16 in Figure 1.1 and 1.2), was clamped between the impeller shroud and the forward bearing support assembly gas seal, (item 18).

1.3 Comparison of the Novel Compressor Unit with Conventional Machines

Two groups of compressor used in air conditioning refrigeration systems were identified; aerodynamic, (turbo), and positive displacement. Figure 1.5 shows the types of compressor making each group. The potential application of a particular compressor depends upon the required flow rate and pressure rise. Table 1.1 lists typical air flow and discharge pressure rise for the compressors, Laing¹².

A way of relating the performance of various types of compressor is to compare their specific flow. Figure 1.12 shows the distribution of specific speed, (defined in Figure 1.6), for the types of compressor considered. It can be seen from Figure 1.6 that positive displacement compressors tend to have low specific speed values, reciprocating compressors having values less than 150. Centrifugal compressors are shown in Figure 1.6 to have specific speeds ranging for 40 to 350, Laing¹². For the present study the specific speed of a compressor operating with a rotational speed of 70,000 rpm and a flow rate of 5.7 cubic metres per minute, (200 cfm), with a discharge pressure of 2.5 bar, was estimated to be 280 placing the operation well within the specific speed range of centrifugal compressors. Operating a compressor at 3600 rpm, (a representative speed for conventional

machines), with the same flow rate and discharge pressure reduces the specific speed to 11 which is well within the range of all positive displacement machines.

Reciprocating machines were shown to be the most common type of compressor used in air conditioning refrigeration systems. They are used individually or in banks which allows for a degree of capacity modulation. A particular disadvantage of reciprocating compressors is their susceptibility to liquid ingestion, (slugging), which can cause damage to the bearings, valves and connecting rods. The application and theory of reciprocating compressors is discussed in text books, notably Chlumsky¹¹ who provides an excellent introduction to all types of reciprocating and rotary compressors. Dossat¹³ and Thevenot¹⁴ considered refrigeration systems applications. Others worthy of note are; Ophulus¹⁵ Horn and Scarf¹⁶ and Square et al¹⁷ who discuss the design considerations for heat pump applications. Scott and Davis¹⁸ investigated the effects of compressor performance on air conditioning systems. Because of the market position held by reciprocating machines they offer the greatest challenge to the general application to the present novel compressor. A literature study was undertaken to identify weaknesses in reciprocating compressor design in order that the advantages of the present concept may be more fully identified.

An evaluation of the literature indicated that an advantage of the sliding vane compressor is the high degree of tolerance to liquid slugging, relatively low starting torque and low noise output compared with reciprocating machines. The design of the sliding vane compressor limits the system pressure ratio and part-load performance tends to be low because of this, Buick¹⁹.

For larger flow rates the Roots Blower offers the same advantages as the vane compressor over the reciprocating machine. Pressure ratios are typically 1.14 for operation on air. Given the greater molecular weight of CFC refrigerants the pressure ratio would still be unsuited to most refrigeration system applications particularly those found in air conditioning and heat pumps. Some machines are designed to operate with the benefit of an oil free compression process.

The 'screw' compressor offers similar advantages as the Roots Blower but with better potential pressure ratio. Laing¹² and O'Neill^{20,21} provide a detailed evaluation of the history, operation and application of the screw compressor. In air conditioning applications compressor pressure ratios of 5 to 6 are thought to be typical. With these high pressure ratios oil injection during the compression process is necessary to prevent excessive gas leakage between the helical compartments. For the application of these machines it is necessary to fit an oil separator in the discharge line in order to keep the quantity of oil in the remainder of the system to acceptable levels. In addition oil coolers and scavenge pumps may be required. The cost of this equipment would be prohibitive at the duty considered in this study.

Centrifugal compressors have traditionally been selected for large duty industrial and commercial applications. Dossat¹³ suggested that the usual duty is between 123 kW and 3.5 MW. Fluids most often used are Freon 11 and Freon 113 because of their large specific volumes at ambient temperatures. The theory and operation of centrifugal compressors is well documented most notably by Ferguson²² and Stepanoff²³. A literature survey and study was undertaken in order to provide an understanding of the causes of low isentropic efficiency and potential methods of improving the performance of centrifugal compressors.

The manufacturing technology of small scale centrifugal compressors has been well developed by Holset, of Huddersfield, amongst others. This makes the units relatively inexpensive compared with the other compressor types, (eg reciprocating, screw and vane), of similar capacity and which generally have more complex component parts from a manufacturing standpoint.

The type of compressor unit described in this thesis has three primary advantages; high power/weight ratio, low noise and vibration, low cost. Centrifugal compressors are intrinsically lighter in weight than other types, with a similar capacity, due to the high rotational speed. The weight of the drive motor is proportional to the volume which is inversely proportional to rotational speed. Because a high speed induction motor was selected for the present study the use of a speed reduction gearbox between the compressor and motor was avoided. For modern railway coach applications low machinery noise output is important for

passenger comfort. Because of the high rotational speed of a centrifugal compressor the noise generated tends to be at higher frequencies and consequently is easier to attenuate with conventional silencing methods. Compared to conventional reciprocating machines centrifugal compressors have far lower vibration levels which makes them particularly attractive to system owners and users.

Maintenance costs are also an important consideration in the selection of refrigeration compressors. The absence of valves in the centrifugal compressors reduces the peak pressures at compressor discharge and consequently reduces the rate at which the refrigerant decomposes. In the proposed design the use of gas bearings throughout and the resulting absence of lubricating oil further reduce periodic maintenance costs. However, the use of Freon 11 may produce an increase in inspection frequencies due to the possible induction of moist air because of the subatmospheric pressures found in parts of the system; notably in the evaporator and compressor suction line. With few moving parts and the absence of valves the novel compressor would reduce the number of failures and increase reliability compared with reciprocating machines.

A semi-hermetic design was proposed for the prototype machine. This would remove drive shaft sealing problems between the motor and compressor and provide relatively easy access to the internal components compared with an hermetic unit. Hermetic compressors are preferred by manufacturers because the design avoids tampering by inexperienced service engineers.

1.4 Sources of Inefficiency in Reciprocating Compressors

Because of the strong market position held by reciprocating machines in the refrigeration industry they offer the greatest challenge to the general application of the novel compressor unit considered. A literature review was undertaken to identify weaknesses in reciprocating compressor design in order that the advantages of the present concept may be identified.

A literature survey has shown that until the mid-nineteen seventies the majority of air conditioning refrigeration

compressors were designed primarily for summertime air-conditioning with some heating requirements during the winter, Squarer et al¹⁷. The fuel crisis in the 1970's and early 1980's appears to have focused the attention of researchers on improving the design of reciprocating refrigerant compressors particularly for heat pump application. Hamilton et al²⁴ identified three major areas of research; reduction in sales price, increase in capacity and thermodynamic performance, improvement in component life. A reason for the sudden rise in refrigerant compressor design effort was the increased application of small heatpumps in which compressors are required to withstand greater mechanical stresses from increased pressure ratios, Horn and Scharf¹⁶. A spin-off from this research activity has been an increase in the reliability in all reciprocating compressors applications.

Bearing loads increase with compressor pressure ratio. In addition, excessive coking of valve gear may result from oil breakdown at higher discharge temperatures, Dossat¹³. Coking of plate valves can result in gas leakage during the compression process resulting in a gradual fall off in volumetric efficiency. Centrifugal compressors do not suffer from this problem and are therefore more suited to high temperature application. Furthermore centrifugal compressors carry the load symmetrically about the rotor therefore journal bearing loads depend on the weight of the drive shaft and rotational speed rather than duty. Thrust bearing will, however, increase with load unless impeller forces can be balanced, as in the present case with the use of a special nut and labyrinth seal.

Using a 2 kW reciprocating compressor Jacobs²⁵ showed that a 1.5 psi pressure drop in the suction manifold caused a 1.5% reduction in compressor efficiency. Whereas a 4 psi pressure loss in the discharge manifold resulted in a 1% loss. This result is thought to describe the importance of suction line sizing in the design of a refrigeration system. The use of oil to lubricate reciprocating compressors often requires that gas velocities in the suction line are greater than a critical value determined at the lowest anticipated evaporator operating pressure, (temperature). Failure to observe this requirement can result in lubricant not being returned to the compressor. The use of gas bearings proposed for the novel compressor removes this requirement with the result that larger diameter suction and discharge pipe lines

may be selected with resulting improvements in system performance.

In reciprocating compressors, pressure pulsations are caused within suction and discharge manifolds. These pulsations are due partly to the intermittent nature of flow through the manifold and partly due to the natural, (organ pipe), frequency of the manifold chamber. It is possible to design manifolds so that the pressure pulsations produce a supercharging effect on the compressor. Jacobs²⁵ reported that suction-line pressure pulsations reduced the volumetric efficiency of his test machine by 12%; due in part to their effect on the action of the valve plates. Centrifugal compressors benefit because they do not normally suffer from pressure pulsations in the suction and discharge lines, (however see 'surge' instability discussed in Chapter 2).

In refrigeration systems some heat transfer to the vapour in the suction-line is desirable to ensure as little refrigerant as possible enters the cylinder which can cause damage to the valves, bearings and connecting rods. According to Hughes et al²⁶ suction-line heating accounts for 20-30% reduction in mass flow in reciprocating compressors and therefore excessive heat transfer should be avoided. Centrifugal compressors are reported to be more tolerant to liquid ingestion and therefore suction-line superheat temperatures may be reduced from 8-10°C, (commonly used for reciprocating machines), to 3-4°C with a resulting improvement in capacity. The use of suction gas for motor cooling further reduces the capacity of compressors. A detailed discussion of this aspect is given in Chapter 3.

Valve leakage in reciprocating compressors reduces their volumetric efficiency due to discharging gas back into the suction manifold during the compression stroke as inducing gas from the discharge manifold during the induction stroke. Hughes et al²⁶ suggested a typical reduction in volumetric efficiency of 2%. Lubricating oil acts to reduce gas leakage past valves and pistons and entrained oil acts as a heat sink during compression thereby reducing compressor work, (a technique used in the design of some screw compressors). In relation to reciprocating compressors Hughes et al²⁶ suggested that the proportion of oil in the gas flow discharged from the cylinder is typically 1-2%. The potential improvement in efficiency resulting from cooling the gas during compression is attractive. Oil cooling in the present study was discounted because the use of gas bearings

removed the need to have oil in the system and because the potential improvements in performance were thought to be of second order compared with the disadvantages of having oil in the system. It was also felt that liquid injection into an high speed impeller may result in excessive erosion damage if maintained.

During a defrost cycle, when the evaporator is artificially heated, reciprocating compressors can be subjected to excessive quantities of refrigerant ingestion. Hermetic compressors are tolerant to liquid ingestion into the compressor casing. Problems do occur if excessive quantities of liquid enter into solution with the lubricating oil. Repeated flooding can cause a severe fall in oil temperature which can result in progressive damage caused by excessive foaming and loss of oil from the compressor casing. Oil dilution can also result in high wear rates and piston seizure due to a rapid reduction in cylinder-piston clearances, according to Horn and Scharf¹⁴. Liquid ingestion can be controlled by fitting suction line accumulators.

Scott and Davis²⁷ from their research into air conditioning system simulation reported that system COP improvements in the order of 0.5 to 0.6 in the longer term from improvements in the performance of reciprocating compressors. Table 1.4 summarizes the losses found in typical reciprocating compressors measured by Jacobs²⁸.

1.5 Electric Drive Motor

The majority of refrigerant compressors today are driven by electric motors because they offer the user relatively low cost, low noise, minimal maintenance and ease of starting and stopping, Stoecker and Jones²⁹. One disadvantage is that they tend to be constant speed machines although this objection is being removed particularly for large commercial and industrial systems where the economies of scale make the introduction of switched winding techniques and variable frequency supply viable.

A further consideration to electric motor drives is the overall energy cost particularly in the larger systems. In the U.K. the C.E.G.B. supply electricity to the user with an approximate efficiency of 30%²⁹. A refrigerant compressor driven by an electric motor may produce a seasonal heating C.O.P. of 2.5. If primary energy is taken into account this value would fall to approximately 0.8. In order to improve

the overall C.O.P., (sometimes referred to as the Primary Energy Ratio), direct drive from some form of heat engine has been used in some cases. An additional benefit of a heat engine drive would be the relative ease with which compressor speed is modulated to give capacity control.

Until recently a drawback of electric motors was that they tended to be constant speed machines. More recently solid state electronic speed controllers have been developed for a wide range of powers. These controllers in general are limited to controlling supply frequencies to 50 Hz, Reynolds Ltd³⁰ and Allspeed Ltd³¹. The 'switched winding' technique is an alternative method used to vary the motor speed. In the present case this method was discounted because switching from 2-pole to 4-pole operation would halve the compressor speed which would reduce the maximum pressure ratio of the compressor from 6 to 1.2 and halve the volumetric flow rate. This degree of capacity modulation was outside the operating requirements of the compressor.

There are a number of different types of electric motor available. The polyphase induction motor, invented by Nikola Tesla (Trans AIEE vol 5 1888 p305), is at present the only practical option for driving hermetic and semi-hermetic refrigeration compressors. Various types are available though in general squirrel cage induction motors are most often used. Being brushless motors they require less maintenance than other types and size for size are less expensive than wound rotor or synchronous motors. Squirrel cage induction motors are manufactured in a range of frame sizes. Single phase units range from 50W to 5kW, whilst three phase motors are available from 250W upwards, though they are not often used below 750W, Alger³². A further reason for not using slip-ring motors is the possibility of electrical arcing from brushes which could cause a refrigerant to breakdown forming acids resulting in corrosion, Dussat¹³.

1.6 Alternative Compressor Drives

Research into gas engine powered refrigeration compressors for heatpumps was stimulated by the energy crisis of 1974, Pegley and Riek³³. It is anticipated that there will be a continually increasing application of relatively inexpensive modified automotive engines to drive refrigeration compressors particularly for heat pumps.

A disadvantage in adopting conventional vehicle engines to drive refrigeration compressors is thought to be the relatively short interval between servicing. However, Pegley and Rieke³³ observed that in most cases these engines operate in a relatively clean atmosphere and use low coking natural gas which would ensure that servicing frequencies are far greater than those we experience with motor vehicle engines.

Engine noise was not considered an important factor against the use of reciprocating i/c engine drives in commercial and industrial environments. However, noise is an important feature when considering domestic applications. Strong³⁴ identified noise, vibration, part-load operation, size, pollution, reliability and maintenance periods as the main disincentives against the adoption of this type of engine in domestic settings. Of these noise, vibration and physical size are the only basic disadvantages though the reliability of the turbo-compressor/expander proposed by Strong³⁴ and developed by him had far greater potential than conventional reciprocating engines used to drive refrigeration compressors.

Rankine/Rankine cycle heatpumps like that developed by Strong³⁴ have been the subject of study, Energy Technical Services Unit^{35,36}. A Rankine/Rankine heatpump cycle was also suggested by Angelino, Ferrari et al³⁷ to provide 1Mw of heating with a theoretical power cycle efficiency of 31%.

King³⁸ in a review of Rankine/Rankine heatpump research in the USA reported the work of Mechanical Technology Incorporated which has included both solar and gas powered boilers. This research utilises a turbine to drive a centrifugal compressor. In a case study MTI's turbomachinery package consisted of an axial flow steam turbine, an alternator and a refrigeration compressor coaxially mounted in gas bearings with a mechanical shaft seal to separate the steam and refrigerant compartments. In this particular case the refrigerant used was R11. This research has similarities with the current project and MTI's failure to develop a reliable turbo-machine was of some concern.

It was interesting to note other recent developments of miniature turbine driven centrifugal refrigerant compressors, King³⁸ particularly the Honeywell/ Barber/ Nichols/ Lennox, (steam driven), system developed for 10kW and 90kW capacity and Airesearch/Dunham Bush R11 system being developed for the 10kW, 90kW and 270kW capacity, and finally the Carrier Corporation 90kW, R-113 system operating on gas bearings.

This project was at the conceptual phase in 1979 but nothing appears to have been published since.

1.7 Conclusions

A novel compressor concept was described consisting of a 'small scale' centrifugal compressor driven by a high frequency, (1200hz), induction motor mounted on a common drive shaft supported in aerodynamic bearings. The reason for undertaking the research reported in this thesis was ultimately to provide refrigeration engineers with a low cost, low weight, reliable compressor, with the potential for high performance, suitable for use in refrigerators, heat pumps and air conditioning refrigeration systems, particularly railway carriage applications where low noise and vibration and robustness are important factors, Graham². A complete description of the compressor unit was given in Section 1.2 and the novel concept was compared with conventional compressors such as vane, screw and Roots types in Section 1.3, in terms of cost, reliability and application.

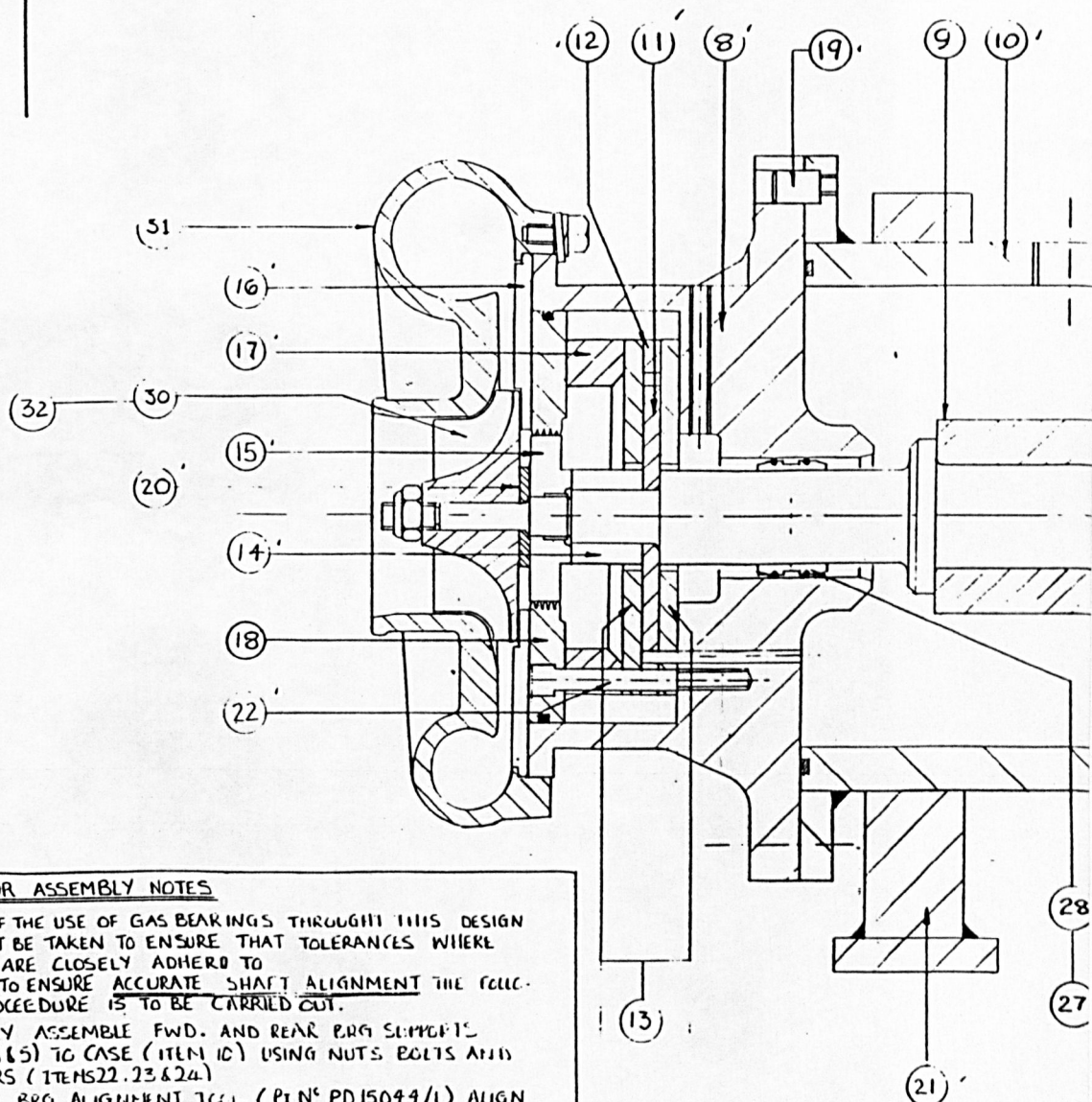
Because of the dominant market position held by reciprocating compressors a special study was made to compare them with the novel compressor unit. This study, described in Section 1.4, identified weight, cost, maintainability, vibration, performance and reliability as areas where the novel compressor gained over conventional reciprocating machines.

Objections to the use of electric motor drives on energy conservation grounds were discussed, however, application to the compressor unit considered in this study is unchallenged.

This study identified the development of solid state electronic speed controller to provide approximately 11kW, 3-phase, 415 volts AC, over a frequency range of approximately 900 to 1200 hz.

For the successful commercial application of the novel compressor unit critical areas of technology were identified. These were; compressor efficiency, motor cooling, laminated rotor design, drive shaft and bearing system design. These aspects of this study are discussed in Chapters 3 and 4. The manufacture of the prototype compressor unit is described in Chapter 5 and testing is described and evaluated in Chapter 6. In order to provide

future researchers with a more precise definition of the operational requirements of the novel compressor unit proposed in this report a computer based simulation program was written which modelled the behaviour of the compressor when installed in an air conditioning system refrigerator. This aspect of the study is evaluated in Chapter 7.

**COMPRESSOR ASSEMBLY NOTES**

1. BECAUSE OF THE USE OF GAS BEARINGS THROUGHOUT THIS DESIGN CARE MUST BE TAKEN TO ENSURE THAT TOLERANCES WHERE SPECIFIED ARE CLOSELY ADHERED TO
2. IN ORDER TO ENSURE ACCURATE SHAFT ALIGNMENT THE FOLLOWING PROCEDURE IS TO BE CARRIED OUT.
 - (i) LOOSELY ASSEMBLE FWD. AND REAR BRG SUPPORTS (ITEM 8 & 5) TO CASE (ITEM 10) USING NUTS BOLTS AND WASHERS (ITEMS 22, 23 & 24)
 - (ii) USING BRG ALIGNMENT TOOL (PTN^S PD15044/1) ALIGN BRG SUPPORT ASSEMBLIES ENSURING DISPLACEMENT PROBE FIXINGS AND LOCATION PILOT HOLES ARE ALIGNED
 - (iii) TIGHTEN NUTS AND BOLTS (ITEMS 22 & 24) EVENLY WITHOUT DISTURBING ALIGNMENT.
 - (iv) USING 5MM DIA TWIST DRILL AND PILOT HOLE (2 EACH END) BORE HOLES FOR LOCATION PINS IN FLANGE
 - (v) USING A 10MM DIA END MILL COUNTER BORE LOCATION HOLES TO A DEPTH OF 20MM AS SHOWN ON THIS Dwg
 - (vi) USING A "1/4" BSF TAP PREPARE LOCATION HOLES IN FLANGE FOR FIXING LOCATION PINS (ITEM 19)
 - (vii) FIT LOCATION PINS. REASSEMBLE AND CHECK ALIGNMENT

3. FITTING OF GAS SEAL (ITEM 18)

- (i) ASSEMBLE THRUST PLATES, SPACERS & G. USING C.H.S. (BOFF, ITEM 22). MEASURE
- (ii) REMOVE SPACER (ITEM 17) AND GRIND FACES. REASSEMBLE AND CHECK THA

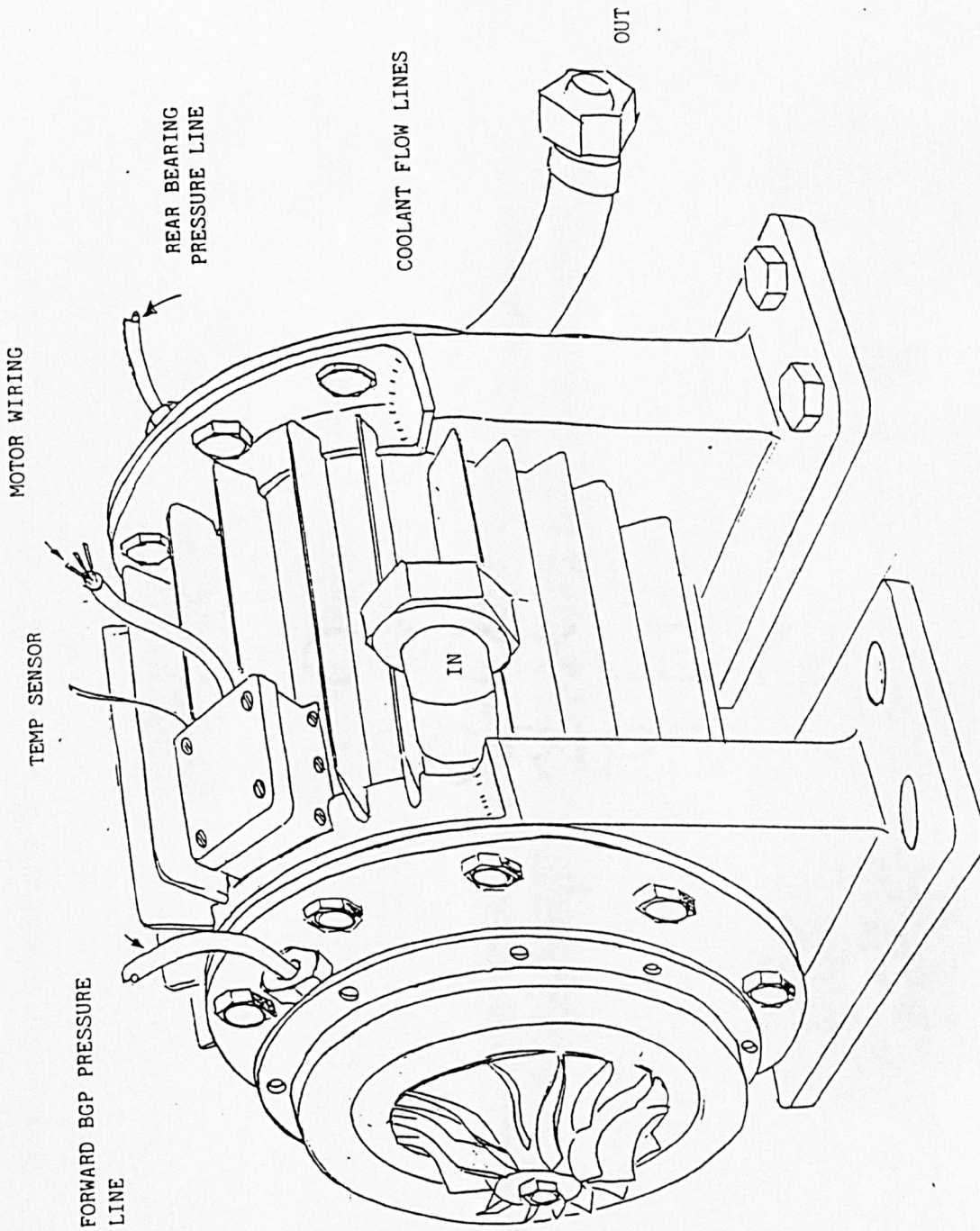


FIG. 1. 3 AN EARLY SCHEME OF PROTOTYPE COMPRESSOR

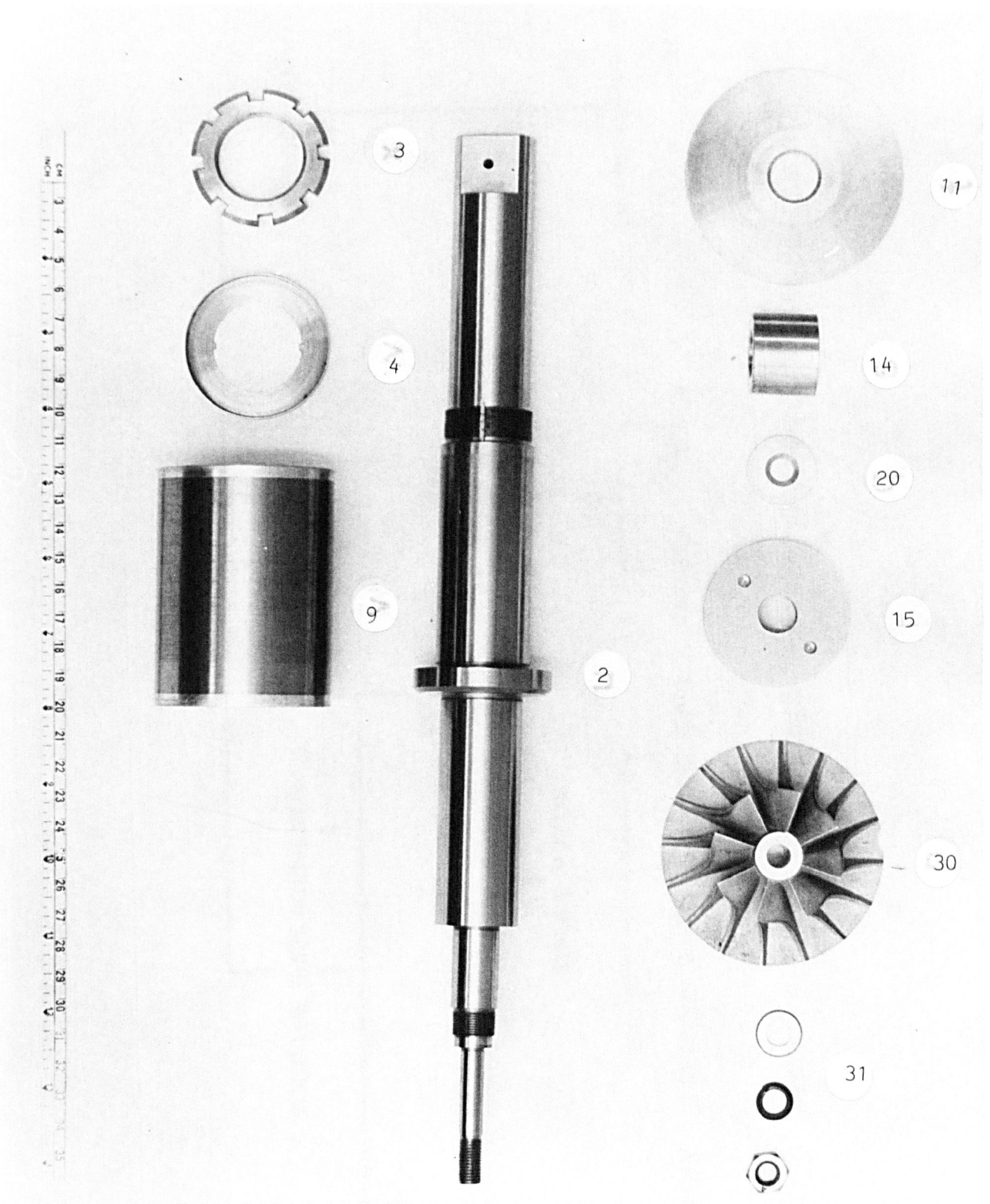


FIG. 1.4 DISMANTLED ROTOR SHAFT ASSEMBLY

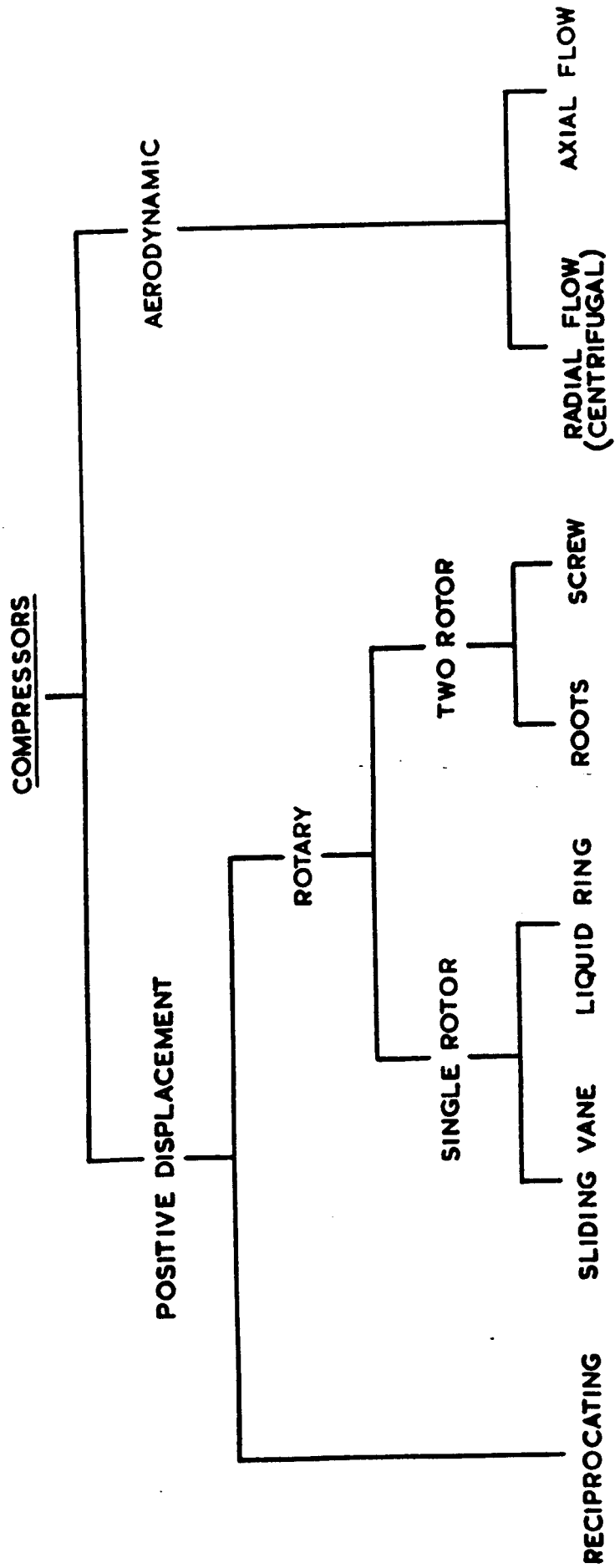


FIG. 1.5 CHART SHOWING RELATIONSHIPS BETWEEN COMPRESSOR TYPES

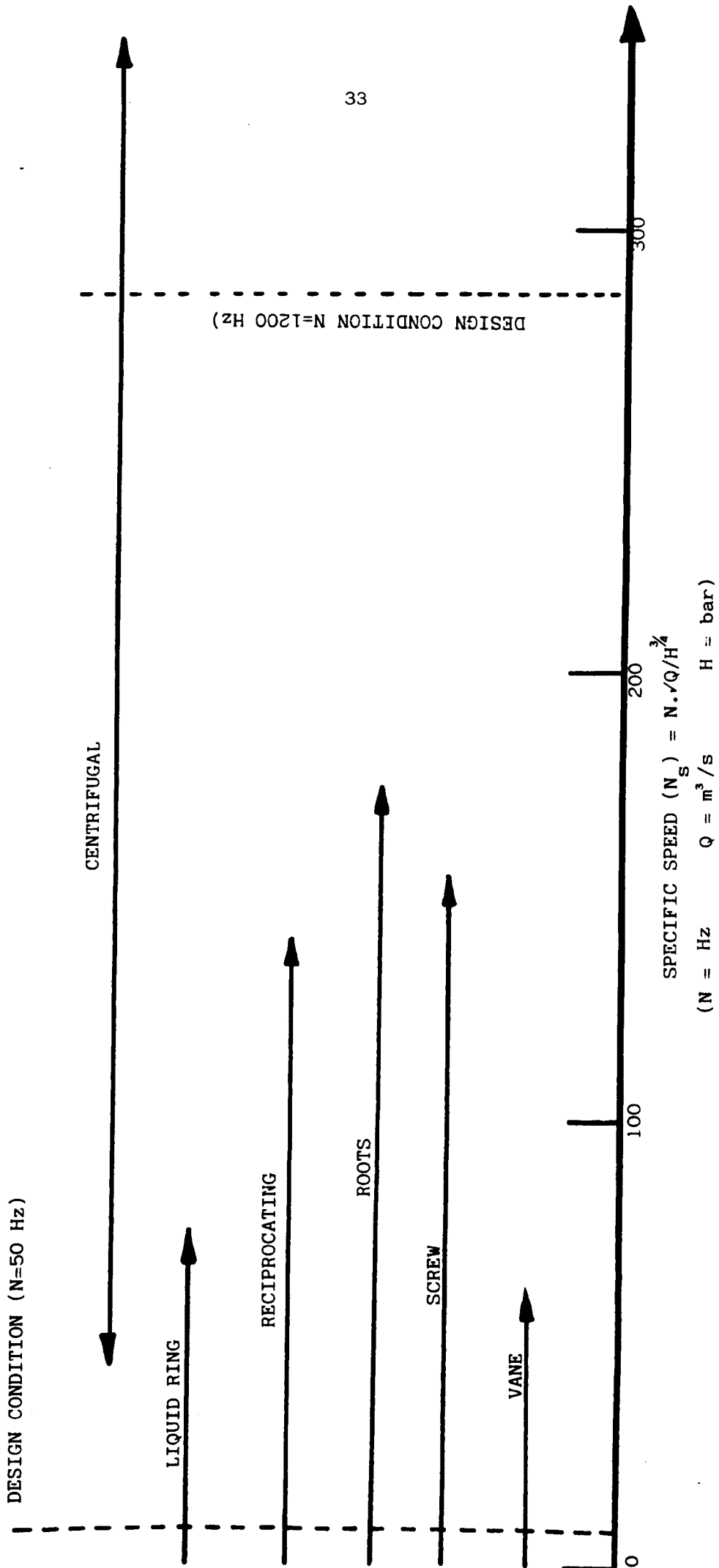


**FIG. 1.6 COMPARISON OF SPECIFIC SPEED RANGES FOR VARIOUS TYPES OF COMPRESSOR
 (BASED ON DATA GIVEN IN TABLE 1)**

COMPRESSOR TYPE	VOLUME RANGE		NORMAL PRESSURE RANGE			
	m ³ /hr	cfm	SINGLE STAGE		MULTI STAGE	
			kg/cm ² g	psig	kg/cm ² g	psig
AXIAL	34,000 UPWARDS	20,000 UPWARDS	0-0.35	0-5	7	100
CENTRIFUGAL	3,400-170,000	2,000-100,000	0-1	0-15	10.5	150
LIQUID RING	85 - 8,500	50 - 5,000	0-1.4	0-20	7.0	100
RECIPROCATING	UP TO 17,000	UP TO 10,000	0.7-7	10-100	1,000 +	15,000 +
ROOTS	85-25,500	50-15,000	0-0.7	0-10	1.7	25
SCREW OIL FREE	250-34,000	150-20,000	0.7-3.5	10-50	17.5	250
SCREW OIL INJECTED	170 - 2,500	100 - 1,500	0.7-21	10-300	21.0	300
SLIDING VANE	85 - 3,400	50 - 2,000	0.7-7	10-100	8.8	125

TABLE 1.1 COMPRESSOR VOLUME AND PRESSURE RANGES

Refrigerant: R22
Conditions: 45°/130°/65°
RPM: 3450

Category	Perf. Btu/W-Hr	% Loss	Measurement Technique
Theoretical Maximum	15.2		
Motor Efficiency	12.5	18	
Superheat @ 140°F	10.7	12	Thermocouples locate at the suction port
Suction Manifold @ 1.5 psi Drop	10.5	1.3	Pressure transducers
Underpressure	9.8	4	Pressure transducers
Overpressure	8.9	6	Pressure transducers
Clearance Volume @ (8% CV)	8.8	.6	Calorimeter
Leakage .(2%)	8.7	.6	Flowmeter
Discharge Manifold @ 4 psi Drop	8.6	.6	Pressure transducers
Friction (Estimated)	8.4	1.3	
TOTAL	8.4	44.4	

TABLE 1.2 TYPICAL COMPRESSOR LOSSES

CHAPTER TWO

CENTRIFUGAL COMPRESSOR PERFORMANCE

2.1 Introduction

The aims of this chapter are to report the results of a study into the performance of centrifugal compressors and to present recommendations for the future development of a high efficiency compressor impeller and volute suitable for use in the present compressor unit.

There has been considerable research into the performance of centrifugal compressors. In this case the problem centred on improving adiabatic efficiency behind the 60% value demonstrated by Elder and Walters³ for the impeller and volute utilized in this study. The study, therefore, focussed on the causes of inefficiency and ways in which the efficiency of centrifugal compressors might be improved. An important reason for wanting to improve the efficiency of the compressor is that refrigerator system performance, measured in terms of Coefficient of Performance, (C.O.P.), is directly proportional to the efficiency of the compressor.

Inefficiencies in centrifugal compressors arise from thermodynamic irreversibilities introduced into the compression process by friction, acoustic shocks and turbulent mixing.

Table 2.1 lists a typical distribution of loss within an 8:1 pressure ratio, (80% isentropic efficiency), compressor stage, Came³⁹.

Table 2.1

source	% loss
Prewirl vanes	0.3
Leakage	1.3
Vaned diffuser	5.4
Impeller	9.4
Vaneless space	1.1
Windage	2.4
Exit duct	0.4

Table 2.1 DISTRIBUTION OF LOSSES IN CENTRIFUGAL COMPRESSOR.

Fixed or rotatable inlet guide vanes , (IGV), provide some control over the degree of swirl imposed on the flow as it enters the impeller eye. By swirling the flow in the compressor intake the gas velocity relative to the impeller is reduced which reduces the pressure losses within inducer section resulting from friction and acoustic shocks. It is common practice to fit IGV's to CFC refrigerant compressors because of the relatively low sonic velocity of these fluids compared with lower molecular weight gases such as air and steam. The fitting of IGV's also increases the range of operation of a compressor by increasing its choke margin. Pressure losses within IGV's are attributable mainly to frictional flow resistance imposed by vane surface area, but some loss results from turbulent mixing in wake region down stream of the vanes.

Pressure losses caused by acoustic shocks may be as little as 1% of stagnation pressure when the flow Mach number within the inducer section is less than 1.2, Came³⁹. However, shock waves standing in the inducer throat may cause the flow to separate from the suction surfaces of the vanes down stream of the inducer which can lead to a supersonic expansion within the throat caused by the reduction in effective flow area. When the flow diffuses down stream of the inducer the resultant acoustic shock can be much stronger than it otherwise would have been resulting in significant pressure losses⁴⁰. Fitting I.G.V.'s can remove this potential problem.

Boundary layers along vane surfaces and annulus walls thicken rapidly in a diffusing flow. Shroud boundary layers are scrubbed constantly by the spinning impeller churning and mixing the flow. This results in internal entropy gains , a rise in stagnation temperature and pressure losses. According to Dean⁴¹ the inducer section where the majority diffusion should take place and is the region from which the flow usually separates to form the characteristic jet-wake flow within the remainder of the impeller.

Pressure losses also occur within the radial diffuser section. These result from a combination of turbulent mixing and acoustic shock. In the case of a vaned diffuser additional losses occur under part load conditions when flow incidence angle varies from the design value.

When a disc rotates in a body of fluid a resistive torque is

set up. This increases the power absorbed by the compressor and reduces the overall efficiency. The problem of disc friction is well documented and design charts are available, Ferguson⁴².

Gas leakage to the low pressure side of the compressor results in a loss of efficiency. Leakage usually occurs at sections where the rotor shaft passes through the casing to the low pressure side of the compressor. The fitting of seals between the two compartments will often prevent leakage. Labyrinth seals are frequently used for this purpose, as in the present case.

2.2 Slip Factor

Centrifugal compressors increase the pressure head of the process fluid through a combination of centrifugal and angular accelerations. Energy is transferred to the flow partly by the change in velocity combined with a centrifuge action. The energy transfer is predicted by the Euler equation given below using the nomenclature described in Figure 2.1(a).

$$\Delta H_{ad} = U_2 V_{w2} - U_1 V_{w1} \quad 2.1$$

where ΔH_{ad} is equal to the theoretical work done by the impeller on the fluid.

In terms of air angles:

$$\Delta H_{ad} = U_2^2 - U_2 W_2 \sin \beta_2 - U_1 V_1 \sin \alpha_1 \quad 2.2$$

When no inlet whirl is used, as in the present case;

$$\Delta H_{ad} = U_2^2 - U_2 W_2 \sin \beta_2 \quad 2.3$$

If in addition the impeller vanes are purely radially disposed then equation 2.3 reduces thus:

$$\Delta H_{ad} = U_2^2 \quad 2.4$$

In practice the flow angle at the impeller will not equal the vane angle, (β_2), but will vary tending to reduce the whirl velocity component. This phenomenon, known as 'slip', is

the inability of the vanes to give perfect directional guidance to the flow. Came³⁹ suggested that it is partly due to the circulation of the flow within the impeller channels. This is shown in Figure 2.1(b).

The reduction in the whirl velocity, caused by flow slippage, reduces the work done by the impeller. To account for this a 'slip factor', (σ), has been defined. Assuming no pre-whirl at the impeller inlet and radially disposed vanes, then from Equation 2.4;

$$H_{ad} = \sigma U_2^2 \quad 2.5$$

The gas flow at the impeller tip is highly three-dimensional consisting of distinct jet and wake regions. Dean⁴² has taken the concept of slip factor a stage further by allocating separate values to each region:

$$\sigma_{JET} = V_{w,JET} / U_2 \quad 2.6$$

$$\text{and } \sigma_{WAKE} = V_{w,WAKE} / U_2 \quad 2.7$$

Because the velocity of the wake region is approximately zero, (relative to the vanes), according to Dean⁴³;

$$\sigma_{WAKE} \approx 1 \quad 2.8$$

Variations in σ_{JET} result from slippage between the impeller vane and the flow.

Due to the complexity of the flow through centrifugal impellers slip factors are determined empirically. A number of quasi-empirical relationships have been devised. Stanitz⁴⁴ suggested for the radial vaned impeller;

$$\sigma = 1 - 2/Z \quad 2.9$$

For values of vane angles, (β'), other than 90° Stanitz suggested;

$$\sigma = 1 - (0.63 / Z) / (1 - (V_{R2} / U_2) \cot \beta'_2) \quad 2.10$$

Note β'_2 is measured relative to the impeller tip diameter tangent and is equal to 90°

Other widely used slip factor correlations are those due to Stodola and Busemann, both reported by Ferguson²². Stodola theory assumes that the flow is frictionless and has constant vorticity; (Helmholz law). It is assumed that the flow through the impeller channels has zero vorticity relative to the casing, shown schematically in Figure 2.2. The largest theoretical vortex has a length dimension equal to the vane passage width at the impeller tip. Slip velocity is therefore assumed equal to this theoretical vortex velocity at the impeller tip which is given by:

$$\text{Eddy Velocity} = w \cdot x / 2$$

Where w = rotational frequency of the impeller, (rads per sec).

According to Stodola the eddy velocity is equal to the slip velocity shown in Figure 2.2 as $x = 2 r_2 \sin \alpha / Z$, then from Equation 2.3;

$$\sigma = 1 - \frac{(\pi / Z) \sin \beta'_2}{(1 - (V_{R2}/U^2) \cot \beta'_2)} \quad 2.11$$

Busemann assumed a two-dimensional flow through a logarithmic spiral vane passage. Busemann's correlation is given as:

$$\sigma = \frac{B - C(V_{R2}/U_2) \cot \beta'_2}{1 - (V_{R2}/U_2) \cot \beta'_2} \quad 2.12$$

where B and C are functions of r_2 / r_1 , β'_2 and Z .

Ferguson²² reporting the work of Csandt and also Wislicenus suggest that when;

$$r_2 / r_1 \geq \exp((2 \pi / Z) \sin \beta'_2)$$

(in the present case $r_2 / r_1 = 1.69$), then $C = 1$ and $B = f(Z, \beta'_2)$. Using this approach values of B may be determined from Figure 2.3 and slip factor determined from Equation 2.12.

Balje suggests the following alternative correlation;

$$\sigma = \frac{Z}{3.183 + Z} \quad 2.13$$

A comparison is made in Table 2.2 between experimental results and the slip factor values determined from the correlations so far described.

A conclusion drawn from Figure 2.4 is the accuracy with which the relatively simple theories predict the results of such a complex flow regime as exists within the impeller channels of a centrifugal impeller.

For the present case, of a 16 vaned, 76mm diameter radial vaned impeller, if a speed of 60,000 rpm is chosen to give a tip speed, (U_2), of 239 m/s then the maximum theoretical work done by the impeller, based on Equation 2.4, would be 57.2 kJ/kg. Applying the Stodola model previously described then the eddy velocity at the impeller tip would be 47 m/s giving a predicted whirl velocity, (V_{w2}), of 192 m/s and a slip factor of 0.8. The theoretical work done by the impeller is therefore 45.8 kJ/kg, (at 60,000 rpm).

The effects of fluid leakage and friction between the impeller and casing increases the work input above the value indicated above. A work input factor, (ψ), or the energy transfer coefficient, Casey⁴⁵ is introduced to take these factors into account. This is defined in the following;

$$\psi = \frac{\text{Theoretical Work Done}}{\text{Actual Work Done}}$$

$$\text{so that Actual Work Done, } \Delta H = \frac{\sigma U_2^2}{\psi} \quad 2.14$$

Cohen et al⁴⁶ suggested that the values of work input factor lie between 1.035 and 1.04.

The work done on a gas is equal to the change in stagnation enthalpy across the stage;

$$\Delta H = C_p (T_{2\cdot} - T_1) \quad 2.15$$

where $T_{2\cdot}$ = actual compressor discharge temperature

The isentropic efficiency of a compressor is defined as;

$$\eta_c = \frac{(T_2 - T_1)}{(T_{2\cdot} - T_1)} \quad 2.16$$

Where T_2 = isentropic compressor discharge temperature

$$\text{Noting that } \frac{P_2}{P_1} = \left(\frac{T_2}{T_1} \right)^{\gamma/(\gamma-1)} \quad 2.17$$

Combining Equations 2.14, 2.15, 2.16 and 2.17 rearranging the resulting equation then the pressure ratio across an impeller is given;

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta_c \sigma U_2^2}{C_p T_1 \psi} \right)^{\gamma/(\gamma-1)} \quad 2.18$$

Equation 2.18 shows the relationship between isentropic efficiency, work input factor and slip factor. For the compressor impeller used in this project, operating on the refrigerant Freon 11 and taking the maximum values of isentropic efficiency and volumetric flow at 60,000 rpm of 58.5% and 155 cfm, (Figure 1.6), and assuming; a work input factor of 1.0375, a slip factor of 0.8, (from the previous calculation), an impeller inlet temperature of 288K and representative thermodynamic values for Freon 11 then;

$$\frac{P_2}{P_1} = \left(1 + \frac{0.585 \times 0.8 \times 239^2}{587 \times 288 \times 1.0375} \right)^{1.116/0.116} = 3.91$$

From Figure 1.2 the pressure ratio of the present compressor at 60,000 rpm and 155 cfm is given as 4.3 which is 9% greater than the result calculated previously. Elder and Walters³ had found experimental values of 'slip factor' for a range of speed and flow conditions. These are shown in Figures 2.4 and 2.5 for air and refrigerant Freon 11 respectively. At a flow rate of 155 cfm and at 60,000 rpm the experimental value of slip factor was found to be 0.8, from Figure 2.5. Although this value is similar to that calculated using the Stodola model previously described the definition of 'slip factor' used by Elder and Walters was the ratio between actual compressor work and the square of the impeller tip speed. Using the definitions described in Equations 2.5 and 2.14 the vertical axis in Figures 2.4 and 2.5 is the product of work input factor and slip factor. Because the values of work input factor are close to unity then the actual slip factor values are probably not more than 5% less than those shown in Figures 2.4 and 2.5.

Increasing slip factor will not improve the thermodynamic performance of an impeller but only increase the work that

an impeller can do by increasing the potential enthalpy rise across the impeller. Increasing the number of vanes will increase slip factor. However, the additional surface area would increase pressure losses due to friction and present a greater blockage to the flow. In the present case increasing the number of vanes from 16 to 32 would increase the slip factor from 0.8 to 0.9, by Equation 2.11. Assuming that the isentropic efficiency was to remain as before at 58.5% and the work input factor at 1.0375 then the pressure ratio would increase from 3.9 to 4.6, by Equation 2.18. In fact it would be unlikely for the either work input factor or isentropic efficiency to remain constant as both are dependent upon frictional effects within the impeller which must increase with surface area.

Efficiency, work input factor and slip factor are not independent. The number of vanes is an important parameter in each case and an optimum number of vanes has been shown to exist. Ferguson²² reporting the work of Pfeiderer, gave the following empirical formula for the optimum vane number;

$$Z = 6.5 \left(\frac{R_2 + R_1}{R_2 - R_1} \right) \sin \beta_m \quad 2.19$$

$$\text{where } \beta_m = \beta_{2/2} + \beta_{1/2}$$

For the present case $R_2 = 38 \text{ mm}$, $R_1 = 22.5 \text{ mm}$ and $\beta_m = 78^\circ$. Substituting these values into Equation 2.19 gives an optimum vane number of 25. From the results of an experimental investigation Kearton and Varley, reported by Ferguson²², have shown that the optimum number of vanes would be less than that given by Equation 2.19 and their results suggest a factor of 0.66 which, for the present case reduces the optimum number to 16 which is equal to that used for the current impeller.

The impeller outlet vane angle effects slip factor as indicated in Figure 2.6. An increase in vane angle from 90° , (inclining the vane forward), shows a corresponding increase in slip factor. By reducing the vane angle the loading on each vane is reduced and the isentropic efficiency of the compressor stage can be improved. This effect can be quantified in terms of stage reaction, (Λ), which measures the degree of diffusion across the impeller by comparing the static enthalpy rise to the total enthalpy rise. The variation in stage reaction, defined by Equation 2.20, with vane angle, (β'_2), and flow coefficient is shown in

Figure 2.7.

It is concluded that if the impeller vane angle is reduced from the present value of 90° the slip factor will be reduced and with it the work done by the impeller. However, a reduction in vane angle will increase the stage reaction, (Equation 2.20 and Figure 2.7), and with this the isentropic efficiency, (Equation 2.21). A change in maximum pressure ratio may occur which entails increasing the impeller tip speed by either increasing the diameter of the impeller or the rotational speed.

Slip factor described so far does not take into account part load performance of the impeller. This was discussed by Watson and Janota⁴⁷. Variation in Slip Factor with rotational speed and mass flow was investigated by Whitfield whose results are reported by Watson and Janota⁴⁷. These are repeated in Figure 2.8. It was concluded that slip factor variation with rotational speed is not marked. This is confirmed by Elder and Walters³ results in Figure 2.4 and 2.5 where it is shown that slip, (or more correctly the product of slip factor and work input factor), is more dependent on mass flow than on rotational speed.

Figure 2.9⁴⁷ describes the relationship between Energy Transfer Coefficient, (the product of slip and work input factors), vane back-sweep angle (β_2) and swirl parameter, (V_{w2}/V_{R2} defined in Figure 2.1b), for values of slip factor of 0.9 and 0.85. Figure 2.9 shows that energy transfer coefficient is proportional to the swirl parameter and increasingly so as the degree of backsweep increases. It may be concluded from this that, in the present case, there is a need to reduce the value of V_{R2} , by increasing diffusion within the impeller to increase the ratio V_{w2}/V_{R2} , (swirl parameter). Because of the fall in work input factor with increasing vane backsweep it is thought necessary to increase impeller rotational speed to compensate for the reduction in work input. This results in higher mechanical stresses resulting from increased centrifugal load which is proportional to the square of rotational speed. The correct choice of blade sweep angle must be a compromise between the cost of design and manufacture and improved performance.

2.3 Flow Within the Impeller

Pressure losses occur as the flow expands from the impeller tip across the vaneless space. To minimise this loss it is usual to diffuse the flow as much as possible within the impeller passages. This is achieved by carefully specifying the height of the vanes so that the relative velocity, (V_{R2}) , and thus pressure losses are kept to an acceptable level whilst simultaneously maintaining a wide operating range. It was shown earlier that a fall in Swirl Parameter, (V_{R2} / V_{W2}) , will produce a corresponding reduction in Work Input Factor, however, in the present case this may be offset by an improvement in isentropic efficiency. Watson and Janota⁴⁷ commenting on the findings of Dean and Young, suggested an optimum value of Swirl Parameter of 0.5. Research reported by Watson and Janota⁴⁷ (Jansen, and Senoo and Kinoshita), found that a vaneless diffuser was stable down to a Swirl Parameter value of 0.25. This is equivalent to an air exit angle, (α_2) , of 76° . A similar value is suggested in an unpublished centrifugal compressor design method⁴⁸.

There are several methods for gauging flow diffusion reported in the literature. A frequently used approach is to compare the static enthalpy to the total enthalpy rise across the impeller. This ratio is known as the Stage Reaction, (Λ) , which was discussed in Section 2.2. Referring to Figure 2.10 stage reaction is given by;

$$\Lambda = \frac{h_2 - h_1}{H_7 - H_1} = \frac{1 - (V_{R2} / U_2)^2 \operatorname{Cosec}^2 \beta'_2}{2(1 - (V_{R2} / U_2) \cot \beta'_2)} \quad 2.20$$

$$\text{Also } \eta_c = \frac{H_{4.} - H_1}{H_3 - H_1} = \frac{H_{4.} - H_1}{H_7 - H_1}$$

$$\eta_R = \frac{H_{10} - H_1}{H_7 - H_1}$$

$$\text{and } \eta_D = \frac{h_{3.} - h_1}{V_2^2 / 2}$$

From which;

$$H_{4.} - H_1 = (h_2 - h_1) + \eta_D V_2^2 / 2 - (1 - \eta_R)(H_7 - H_1)$$

Dividing throughout by $(H_2 - H_1)$ and simplifying yields;

$$\eta_c = \lambda + \eta_D (1 - \lambda) - (1 - \eta_R) \quad 2.21$$

The variation of Stage Reaction with Flow Coefficient, (V_{R2}/U_2) , for values of impeller exit blade angle are shown in Figure 2.7, Ferguson²². It was concluded from Equation 2.21 that the reaction of the impeller should be increased by increasing both the vane height at the impeller tip and the vane sweep. The effect will be to reduce V_{R2}/U_2 , increase stage reaction, (λ) , and therefore improve the efficiency of the impeller and stage.

If Equation 2.21 is differentiated with respect to Flow coefficient, (V_{R2}/U_2) , then for a given vane angle the maximum stage efficiency is given when;

$$(V_{R2}/U_2) = (1 - \sin \beta'_2) \tan \beta'_2 \quad 2.22$$

This result reported by Ferguson²² is due to Stanitz. The validity of this analysis was thought questionable because of the implicit relationship between Stage Reaction and impeller isentropic efficiency. An alternative approach is to compare the relative flow velocities at inlet and outlet to the impeller, (W_1/W_2) , (termed impeller diffusion factor). This ratio should lie in the range 1.4 to 2.5 with flow velocities based on one-dimensional flow; Dean⁴³, Casey⁴⁵, Ferguson²² and Came³⁹. The value of this coefficient for the impeller used in the present research at a rotational speed of 66,000 rpm, pressure ratio of 5.6 and corrected flow rate of 0.264 kg/s, was estimated to be 1.16 which suggested that compressor efficiency could be improved if more diffusion was carried out within the impeller section. This could be achieved by increasing the vane tip height towards the impeller tip. This would reduce the Flow Coefficient, (V_{R2}/U_2) , and increase both Stage Reaction and Isentropic efficiency from to Equation 2.21. This conclusion is supported by the research of Dean⁴³ who defined the following diffusion parameter.

$$DR = W_{1,t} / W_{tip}$$

where DR = diffusion ratio

$W_{1,t}$ = Relative velocity at inducer tip

W_{sep} = Relative velocity at separation
point on the suction surface/shroud
corner with the impeller

Dean⁴³ suggested that the majority of diffusion within an impeller takes place in the axial section down stream of the inducer inlet. Because of this flow separation is more likely to occur at the bend in the impeller channel than elsewhere through the combined effects of changing flow direction and an adverse pressure gradient. Dean⁴³, therefore, used relative velocity at separation as the denominator in his diffusion parameter. Dean and Young⁴⁰, developed this approach by using a relative Mach numbers ratio instead of velocity. This parameter is termed by Dean as 'Impeller Mach-number Ratio', MR. Variation in efficiency and MR is shown in Figure 2.11. As suggested earlier, stage efficiency is shown to increase with diffusion although this effect tends to flatten off at higher values of MR. The experimental measurements required to quantify the diffusion factors described so far are made only with difficulty because the flow is highly three-dimensional and time dependent. Dean and Young identified this problem with their Diffusion Ratio (DR).

Amongst the thermodynamic properties which might be used to quantify impeller diffusion performance, static pressure rise between the inlet and exit planes appears to be the most promising. Time-averaged pressure readings and standardized probe locations would assist in comparing designs. This alternative diffusion parameter would simply be the static-pressure ratio across the impeller.

2.4 Reynolds Number Effects on Compressor Performance

It is thought that the magnitude of slip velocity at the impeller tip is proportional to the ratio between inertial and viscous forces in a flow. If inertial forces are large then the resistance within the flow to adopt the same angular speed as the impeller will be greater. Viscous effects tend to dampen eddies which suggests slip factor is a function of impeller wheel Reynolds number.

Wheel Reynolds number is given by:

$$Re = 2\rho W_{zh}/\mu$$

In terms of mass flow rate:

$$Re = \dot{m} / (\mu_1 R_2 \sin \beta'_2) \quad 2.23$$

An examination of the Reynolds number expression shows that vane angle and R_2 are functions of compressor geometry. Because of this slip becomes a function of the ratio between mass flow rate and viscosity. It is not usual to plot Slip Factor against vane tip Reynolds number. However, if it is assumed that viscosity is constant over the range of results shown in Figures 2.4 and 2.5, then these approximate to graphs of Slip Factor against Reynolds number. If this is the case Figures 2.4 and 2.5 supports the view that Slip Factor falls with increasing Reynolds number.

The influence of Reynolds number on the efficiency of centrifugal compressors was studied by Pampreen and reported by Dean⁴³. Figure 2.12 shows Pampreen's results from which the empirical relationship in Equation 1.16 is derived.

$$\eta_c = 1 - 2.16 Re^{-0.164} \quad 2.24$$

Pampreen defines Reynolds number as follows;

$$Re = 2 \rho_1 r_2 U_2 / \mu_1 \quad 2.25$$

Where ρ_1 and μ_1 refer to the inlet conditions.

The results shown in Figure 2.12 indicate that the correlation expressed in Equation 2.24 will have a reasonable accuracy up to $Re = 4.0 \times 10^6$. For Reynolds numbers greater than this the correlation appears to break down. For the present compressor operating on air at an inlet condition of 15°C and 1 bar, the Reynolds number based on Equation 2.25 is 2.3×10^6 . Substituting this value into Equation 2.24 suggests an isentropic efficiency of 80.5% which is approximately 10% greater than found by experiment. This result is thought not to indicate that Pampreen's correlation is invalid but rather what could be reasonably anticipated from a well designed compressor.

Based on Pampreen's definition the Reynolds number for the impeller used in the present research when operating on Freon 11 at 66,000 rpm with an inlet temperature of 280K and pressure of 0.55 bar, (representative design-point conditions), would be 12.4×10^6 which, on substitution into

Equation 2.24, yields a potential efficiency of 85%. The present optimum value measured by Elder and Walters³, at the same rotational speed was 60%, however, as can be seen from Figure 2.12, the Pampreen correlation breaks down when Reynolds numbers are greater than 4.0×10^4 . Because of this no reliable conclusion may be drawn from the result.

Pampreen's results in Figure 2.12 indicate that an increase in Reynolds number will tend to improve the efficiency of the compressor. Assuming that the inlet density and viscosity remain constant then the only methods by which the Reynolds number can be increased are by increasing tip radius or rotational speed. Assuming the rotational speed is fixed then increasing radius will increase the tip speed simultaneously. By doubling the impeller radius the efficiency predicted by Equation 2.24 would increase from 85% to 86.7%. Neglecting the four fold increase in mechanical stress incurred, the potential efficiency improvement could not be justified on economic grounds.

The breakdown in the Pampreen correlation, in Figure 2.12, at high Reynolds number values is similar to the variation of pipe-wall friction factor with Reynolds numbers in standard pipeline pressure loss calculations. At high Reynolds numbers friction factor becomes independent of Reynolds number for 'rough walled' pipe flow. For most applications Wiesner⁴⁷ suggests that Reynolds number effects may not be applicable to centrifugal compressors. Wiesner's research data, (Figure 2.13), is based on the diffuser entry Reynolds number. As indicated by Pampreen's work above a certain Reynolds number the performance parameters shown in Figure 2.13 tend to become independent of Reynold number which supports Wiesners view.

A survey of Efficiency vs Re correlations is given in Table 2.3, Wiesner⁴⁷. In the equation shown at the head of Table 2.3 the quantity 'a' is defined as the fraction of total loss which is assumed to be independent of Reynolds number. Casey⁴⁸, suggested that there is no general agreement as to the correct value for either constant in the Wiesner correlation, (a or γ as may be seen from Table 2.3).

In a search for an alternative correlation Casey⁴⁸ applied a dissipation loss approach in which adiabatic efficiency is related to changes in an equivalent pipe friction factor as defined by the Darcy's friction formula, Massey⁵⁰. It is well known that friction factors are independent of Reynolds

number when the value of the latter increases beyond a limiting value dependent on the roughness of the pipe wall. This would suggest that the part-load efficiency would tend to a constant value behind a certain critical value of Reynolds number when in fact the efficiency tends to fall sharply as the operating point of a compressor moves away from the 'design-point'. This is exemplified in Figure I.6 for the present compressor. The friction factors given in the Moody diagram are also based on a fully developed flow whereas the flow field within the impeller and diffuser channels is three-dimensional. A friction factor correlation is therefore inappropriate in the present case and that the correlation described by Casey⁴³ is only suitable for compressors with a relatively low pressure rise and fans.

For the present compressor the Reynolds number at a representative operating condition was calculated to be 70×10^4 which suggests a constant friction factor value. At low flow rates it is probable that pressure losses are proportional to the kinetic energy of the flow whilst at higher volumetric flows Mach number effects become increasingly significant and produce the characteristic limit in volume flow.

It is concluded that changes in impeller Reynolds number at a representative design-point condition will not affect compressor efficiency directly through changes in friction factor which would remain independent of flow rate. Pressure losses may be reduced by keeping down velocities relative to the impeller. This would have the effect of increasing Stage Reaction and Diffusion Coefficient.

2.5 Effect of the Diffuser on Compressor Performance

The effect of the diffuser efficiency on the overall isentropic efficiency of a compressor is described by Equation 2.21. The factors which affect the efficiency of the diffuser section are now considered.

The purpose of the diffuser section is to recover the static pressure from the flow leaving the impeller as efficiently as possible. Vaned and vaneless diffusers are used. In the former case the diffusion process is controlled by vanes which guide the flow radially whilst in the latter the flow

expands freely between parallel walls before entering the volute region.

One-dimensional isentropic flow analysis through a vaneless diffuser shows that the tangent of the angle between any radius and the flow direction is proportional to the ratio of the density at the point of interest and the fluid density at entry, (ρ_2). This result produces logarithmic spiral streamlines. Furthermore from conservation of momentum;

$$\frac{V_{wr}}{V_{w2}} = \frac{R_2}{R} \quad 2.26$$

Where;

V_{wr} = Tangential velocity at radius R

R_2 = Impeller tip radius

V_{w2} = Tangential velocity at impeller tip

Equation 2.26 shows that the diffuser outlet radius must be twice the inlet radius if the velocity of the flow is to halve. Dean and Young⁴⁰, suggest a practical design value for the diffuser radius ratio of between 1.4 and 2. In the present case this ratio is approximately 2.08, (158mm/76mm), which indicates a halving in the tangential velocity between the impeller tip and the entry to the volute. From velocity triangles constructed at a representative design point, (described in the Section 2.6), and by using a one dimensional compressible isentropic flow analysis, it was estimated that the static pressure ratio across the diffuser would be approximately 2 when processing the Freon 11. This value assumes that no pressure losses are incurred. Frictional effects tend to reduce this value due to heating. This is shown in Figure 2.10.

Vaneless diffusers tend to be larger than vaned types for the same pressure recovery but have the attraction of low capital cost coming from their relative simplicity. Ferguson²² suggested that there is no significant evidence that vaneless diffusers are potentially less efficient than vaned types with similar pressure recovery. Some researchers disagree and suggest potential efficiency improvements between 2% and 10% depending on the compressor pressure ratio. Higher values offering larger potential improvements. Elder²¹, testing various compressor builds, has shown it possible to improve the maximum isentropic efficiency from

70% to 75% by using a vaned diffuser shown in Figure 2.14. It is interesting to note from Elders data that the volumetric flow range was not significantly altered by using a vaned diffuser, which can introduce additional flow blockage and unfavourable incidence angles at part-load conditions.

Vaneless diffusers offer greater tolerance to fouling and erosion than vaned types. Backflow into the impeller is also less likely with a vaneless diffuser because of the smaller adverse pressure gradients encountered. This leads to a compressor which is more aerodynamically stable in operation and less likely to enter a surge condition.

There is indication from the literature that vaneless diffusers offer wider flow ranges by improving surge and choke tolerance compared with vaned types. It has been noted from available data that the efficiency of a vaned compressor falls off steeply as the operating point moves away from the design conditions, Elder²¹, and Elder and Walters²³. This is thought to be due to the sensitivity of the diffuser vanes to changes in incidence.

In the vaned diffuser the flow is directed radially outwards by vanes. This increases the rate at which diffusion takes place and thus reduces the required radial distance needed to produce the same pressure recovery as a vaneless type. This feature is important where compressor size plays a key factor in design such as in aero engine compressors. In the present study this factor is thought to be unimportant.

Brown and Bradshaw²², correlated vaned diffuser performance with a conventional conical diffuser analogy on the basis of equivalent cone angle. They found for their research machine that an equivalent angle of 6 degrees gave best performance. The underlying simplicity of this approach offers a realistic and simple design criteria in what is a difficult theoretical area. It is interesting to compare Brown and Bradshaws result with the 6-8 degree included angle recommended for optimum performance of conical diffusers.

Between the impeller tip and leading edges of the vanes there is a 'vaneless' space. The purpose of this is to reduce flow interaction between stationary vanes and rotating impeller. Dean⁴³ has suggested that this space needs to be no more than one-tenth of the cord length of the diffuser vane. Measurements taken within the vaneless space by Elder and Gill²³, have shown that the flow in this region

is highly three-dimensional with a strong time dependent component. This feature produces rapid oscillations in diffuser vane leading edge incidence. Relatively high velocities within the vaneless space region often results in the formation of an acoustic shock wave systems. Figure 2.15⁴³, shows a typical instantaneous pressure contour map within this region. Shock-waves at the leading edges of diffuser vanes extend radially inwards disturbing the flow pattern within the impeller, Elder and Gill⁴³. To even out the flow oscillations it is usual to design the diffuser with fewer vanes than the impeller, Ferguson²².

Watson and Janota⁴⁷, identified certain design parameters; throat blockage, diffuser stagnation pressure loss, pressure recovery coefficient, diffuser area ratio, inlet Mach number and throat area ratio. A full discussion of the relationship between these parameters and their effect on compressor performance is given by Watson and Janota in their excellent book. Also Dean and Young⁴⁰, have supplied much information in this area of research.

Various types of vaned diffuser have been proposed. Figure 2.16 identifies four types of aerofoil section. Other types of diffuser not discussed here are the rotating wall type, the pipe diffuser and variable geometry diffusers⁴⁰.

In the present research the compressor unit was operated with a vaneless diffuser. This choice was thought to offer a wider operating regime than would have been found with a vaned compressor but at the cost of lower isentropic efficiency. In future studies the possibility of variable geometry vane arrangement should be considered. Variable incidence vanes is one option whilst movable diffuser walls which provide an increase in flow area offer a wider flow range and therefore a larger number of potential applications for the compressor unit.

2.6 Volute Design

The volute is used to collect the fluid as it leaves the diffuser and guide it, with minimum pressure loss, to the outlet pipe. A typical volute design is shown in Figure 2.17.

A volute may be designed using a free vortex method due to

Pfleiderer, described by Ferguson²². This method assumes that the angular momentum within the volute remains constant, (ie neglecting friction). An alternative approach described by Ferguson is to assume that the average pressure and velocity are independent of angular position. Ferguson suggests this method produces compressors with higher isentropic efficiencies than those designed using the free vortex approach.

2.7 Flow instabilities

Aerodynamic stall is a common feature of the flow through all types of turbo compressors and results wherever blade incidence to the flow varies greatly from design values. Incidence losses increase when the flow rate varies from design value due to turbulent mixing along suction and pressure surfaces of impeller and diffuser vanes. A reduction in mass flow rate causes the flow to leave the impeller at an increasing angle (α_2). In a parallel walled vaneless diffuser the flow will try to maintain this angle throughout the expansion process. However, compressibility and viscous effects tend to increase this angle and eventually the flow is swept back inwards towards the impeller forming a vortex within the diffuser. This type of stall may be stationary or rotational.

A second form of flow instability is rotating stall. Rotating stall, described by Ferguson²², has been observed in impeller vanes and diffuser vanes by Dean and Young⁴⁰. Although it need not be an unstable feature, Ferguson²² reporting the work of Emmons, suggests that rotating stall may be an indicator of the onset of the surge condition.

Compressor surge is characterised by an audible mechanical noise emanating from within the compressor. Surge noise is widely held to originate from aerodynamic buffeting of impeller and diffuser vanes which produce an oscillating load on the vanes which in time may result in fatigue failure. The instability is thought to be not very well understood though it is generally accepted to originate from the interaction of the compressor and pipe system. Qualitatively, surge appears when the compressor is unable to support down stream pressure rise. When this occurs flow rate falls to zero and can reverse. A fall in flow rate is accompanied by an out of phase reduction in upstream pressure. As pressure falls flow is re-established and the process is repeated. The frequency with which this process proceeds is known to rise with compressor speed. Elder and

Gill²³, identify the following factors which influence the aerodynamic stability of centrifugal compressors.

- (i) Gradient of the pressure rise characteristic
- (ii) Impeller blade sweep-back and inlet swirl.
- (iii) Number of impeller and diffuser vanes.
- (iv) Pressure recovery within the vaneless space.
- (v) Diffuser channel pressure recovery and collector type
- (vi) Inducer incidence changes with flow rate.

Elder and Gill²³ view the stability performance of a compressor not as a whole but as a series of discrete parts each with its own pressure rise characteristic. Each regime within the compressor has its own flow regions of instability. A region of a compressor entering an unstable condition may induce instability in other regions which results in the whole machine entering a surge condition. According to Ferguson²², ideal conditions for surge exist when a compressor outlet main line is long, gas velocities are low and there is a constant pressure restriction, (such as a gas flow governor), at the far end away from the compressor.

Dean and Young⁴⁴, suggested that changes in system design do not significantly alter surge margins established in tests which are usually an accurate indicator of the performance of a compressor in practice. It is thought that this only applies if the system on which the tests were carried out is not significantly different in terms of volume and gas velocity according to Ferguson²².

2.8 Recommendations for Future Compressor Developments

The results of a refrigerator system simulation study, described in Chapter 7, indicated the compressor design point data listed in Table 2.4. The operating parameters of the compressor used in the present study required to satisfy the design point data are listed in Table 2.5 and the relevant geometrical data in Table 2.6. From the data in Tables 2.4 to 2.6 the thermodynamic conditions at the impeller inlet and discharge stations were calculated. These are listed in Table 2.7 and the resulting velocity triangles

are shown in Figure 2.18 and 2.19.

The aim of the analysis presented in this section was to suggest possible ways of improving the isentropic efficiency of the compressor unit. The study indicated that this aim might be partially achieved by reducing the vane angle, (α_2), at the impeller discharge and by increasing the diffusion within the impeller.

Stage reaction, (Λ), defined in Equation 2.20, is a measure of diffusion and related to the stage isentropic efficiency by Equation 2.21. The optimum value of Flow Coefficient, (V_{m2}/U_2), was also shown to be a function of vane outlet angle, (α_2), by Equation 2.22 which on substitution into Equation 2.20 makes stage reaction, (Λ), a function of vane angle only, (β'_2). Thus the stage isentropic efficiency becomes a function of vane outlet angle alone if values of diffuser and impeller isentropic efficiencies are assumed to be constant, Equation 2.21. The variation in optimum values of flow coefficient and stage reaction, with vane outlet angle are shown in Figure 2.20 and 2.21 respectively and the resulting variation in stage isentropic efficiency with vane outlet angle is shown in Figure 2.22 for two separate rotor and impeller isentropic efficiencies of 90% and 85% , (and what are believed to have been the values of the present impeller), 77% and 69.4%. The former values are considered to be the best that is achievable for a well designed compressor, (Ferguson²²), and later values were estimated by substituting the present efficiency of 60% into Equation 2.21 along with the corresponding value of stage reaction, (Λ), and by assuming that the diffuser would be ten percentage points less efficient than the impeller.

A further effect of changing vane angle, (β'_2), is to alter the slip factor, (σ). Equation 2.11, due to Stodola, was used to determine the variation in slip factor with vane angle, (β'_2). The optimum number of impeller vanes, (Z), was estimated from Equation 2.19 by assuming constant values for the impeller inlet and tip diameters, (D_1 and D_2), of 45mm and 76.13mm, which are the same as the compressor utilized in this study. Also an inlet flow relative Mach number of 0.8 was assumed throughout in order to reduce inducer losses. The value of β_1 and the optimum number of impeller vanes were calculated from Equation 2.19. The variation in slip factor, (σ), and optimum vane number, (Z), with vane outlet angle, (β'_2) are shown in Figures 2.23 and 2.24 respectively.

In terms of the impeller vane and air angles, (β'_2 , α_1), the actual work done by an impeller is given by Equation 2.27.

Actual Work Done =

$$(U_2^2 - U_2 V_{R2} \tan(\beta'_2) - U_1 V_{A1} \tan(\alpha_1)) \quad 2.27$$

The isentropic efficiency of the stage is defined by Equation 2.28.

$$\text{Isentropic Work Done} = \eta_c \times \text{Actual Work Done} \quad 2.28$$

Combining Equations 2.27 and 2.28 gives the isentropic work done by an impeller in terms of isentropic efficiency, slip factor and work done factor. The variation in isentropic work done with vane angle, (β'_2), was then estimated. The results of this analysis are shown in Figure 2.25 for a range of rotational speeds and assumed values of impeller and diffuser efficiencies, (η_R, η_D).

The design point value of isentropic work done was estimated from Equation 2.29 to be 32 kJ/kg. This result has been included in Figure 2.26 as a horizontal line 'A'-'A'. Suitable combinations of rotational speed and vane angle are given at the intersection of the constant speed line and line 'A'-'A'.

Isentropic Work Done =

$$C_p T_1 ((P_3 / P_1)^{\gamma/C_p} - 1) \quad 2.29$$

For the study it was decided that the design point speed of the motor would be 70,000 rpm and therefore any future compressor impeller developments would ideally be optimised at this rotational speed. The operating speed had been selected primarily on the compressor data provided by Elder and Walters³, from their earlier study into the performance of the compressor impeller and volute used in this study which included an optimization of the motor from both mechanical and electrical standpoints. The results of this study are described in Chapter 3.

Selecting a design point speed of 70,000 rpm and impeller and diffuser efficiencies of 83.5% and 77.2%, (the mean of the extreme efficiency cases used so far), places the design condition at point 'O' in Figure 2.25 with a vane outlet angle, (β'_2) of 54°.

The improvements in diffuser and impeller efficiency are thought to be realistic in view of the increase in Stage Reaction from 0.44 to 0.55 and the reduction in Flow Coefficient from 0.35 to 0.26. The optimum number of impeller vanes reduces from 16 to 12 whilst the Slip Factor is reduced from 0.8 to 0.75. These values were taken from Figures 2.20 to 2.24 inclusive. The resulting isentropic efficiency was estimated to be 75% from Equation 2.21. Figures 2.26 and 2.27 show the recommended velocity triangles at the impeller discharge and inlet. The inlet velocity triangles were calculated by using the radial equilibrium condition, ($V_{w1}r = \text{constant}$).

2.9 Conclusions

The aim of this Chapter was to identify those factors which affect the efficiency of a centrifugal compressor. Amongst those factors which play a key role in this are; stage reaction, (λ), work input factor, (ψ), vane outlet angle, (α_2), inducer inlet relative Mach number and diffuser radius ratio, (R_3/R_2). The major sources of irreversibility were shown to be the impeller and diffuser as seen in Table 2.1, Came³⁹.

The results of this study indicate that the isentropic efficiency of a compressor would be increased from 60% to 75% by using a backward inclined impeller with a vane outlet angle of 54° and by reducing relative inlet Mach number from 0.95 to 0.8 at the impeller inlet vane mean height, assuming that optimum values of flow coefficient and stage reaction and vane number are used, Figure 2.20, 2.21 and 2.24. The resulting thermodynamic and geometrical data for the recommended compressor are listed in Tables 2.8 and 2.9.

The results of this study are indicative of the direction of future research towards a more efficient compressor impeller and volute presently available. The performance of a refrigerator is proportional to the efficiency of the compressor and it is therefore important for the future application of the compressor unit, described in this thesis, that such a study be undertaken.

This Chapter evaluates a study into the performance of centrifugal compressors with the aim of directing future research towards improving the isentropic efficiency of the present compressor unit.

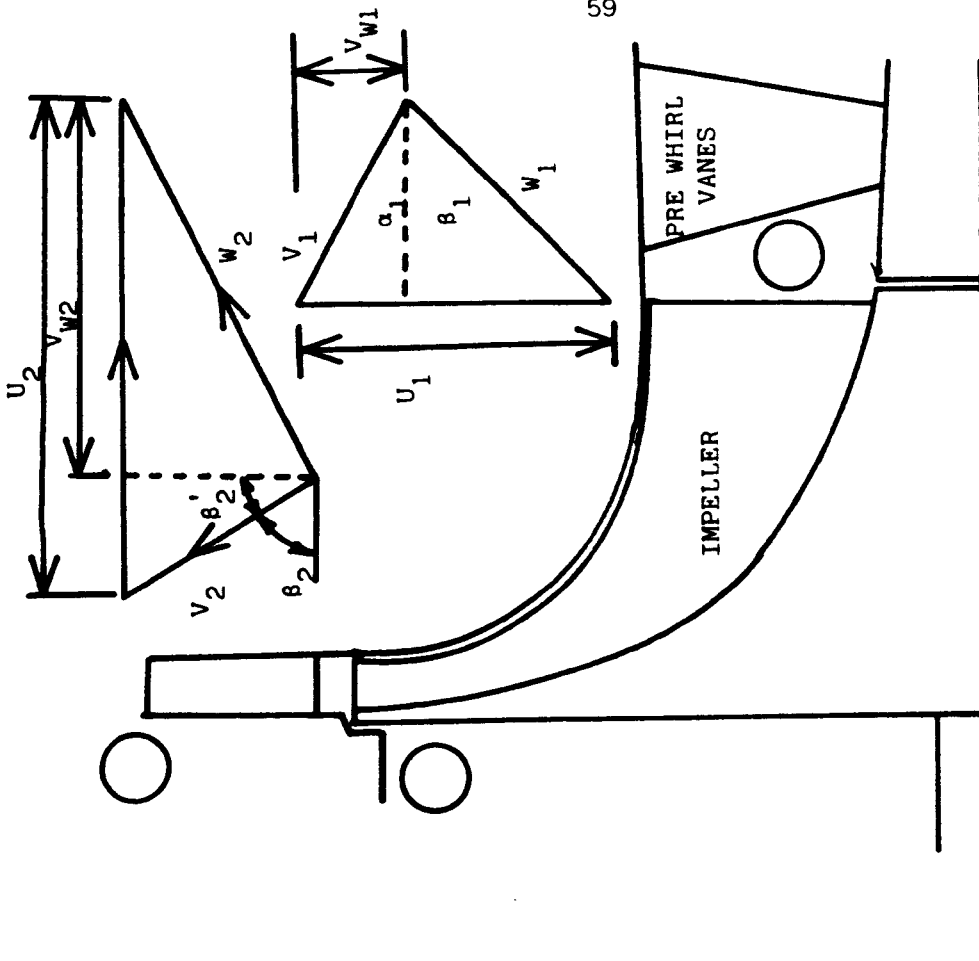


FIG. 2.1 (a) NOMENCLATURE FOR VELOCITY TRIANGLES

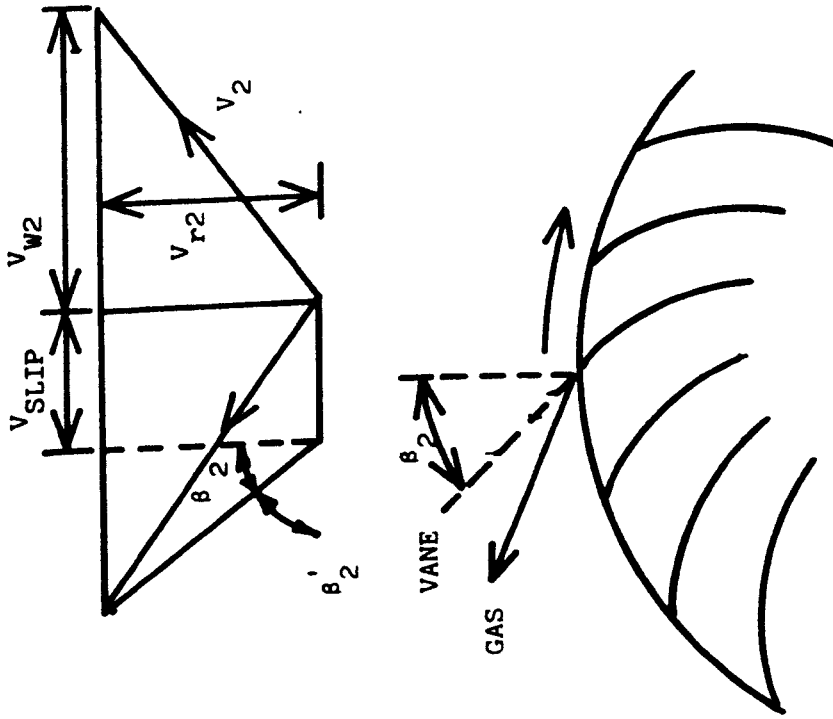


FIG. 2.1(b) EFFECT OF SLIP ON RELATIVE FLOW AT IMPELLER EXIT

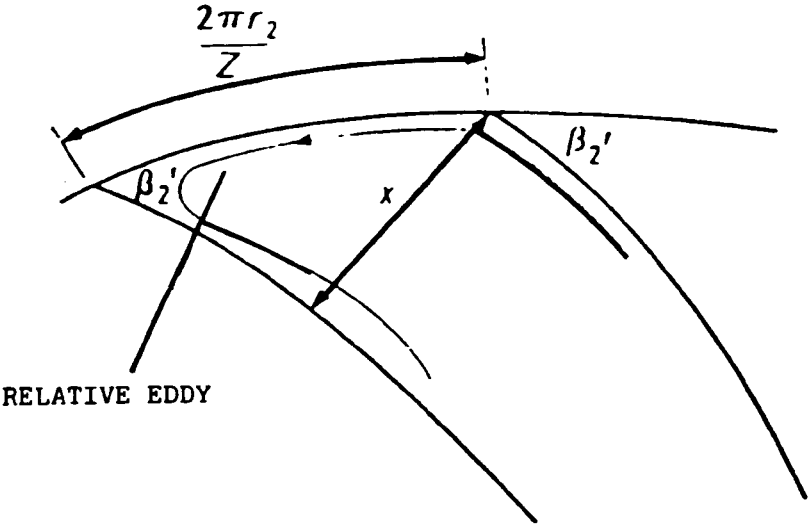


FIG. 2.2 STODOLA SLIP VELOCITY

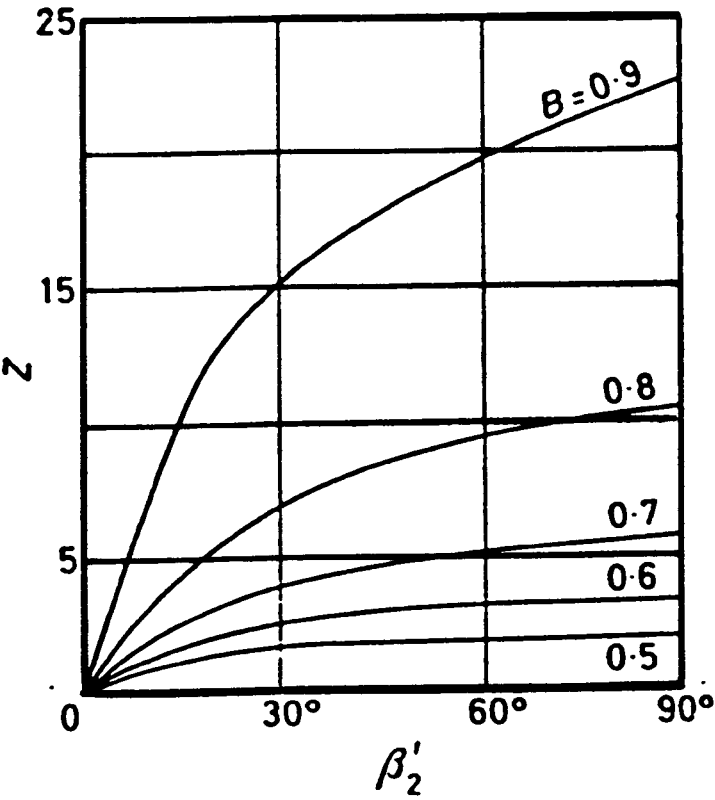


FIG. 2.3 HEAD CORRECTION FACTORS FOR RADIAL IMPELLERS

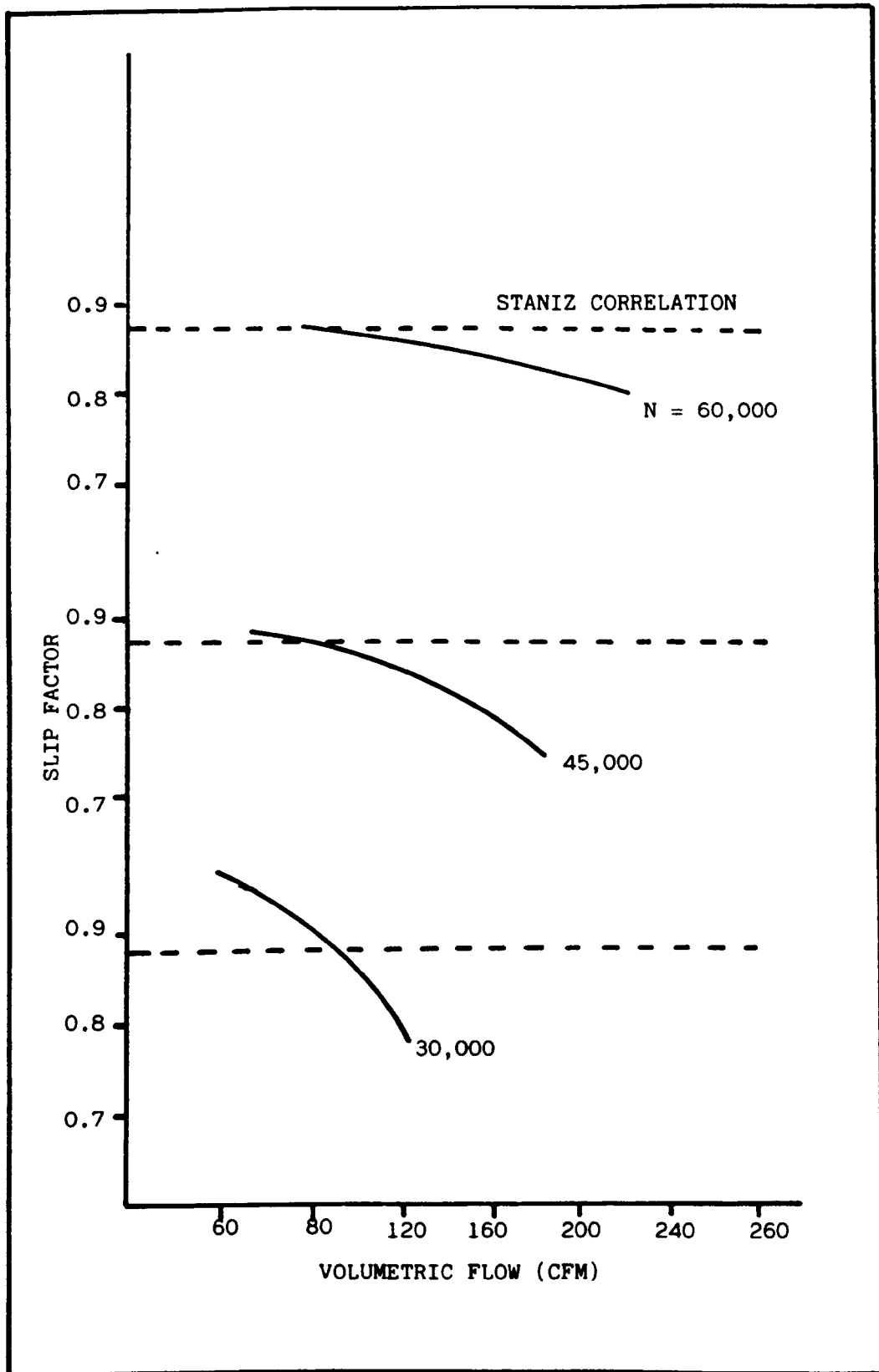


FIG. 2.4 CENTRIFUGAL COMPRESSOR AIR PERFORMANCE TEST RESULTS

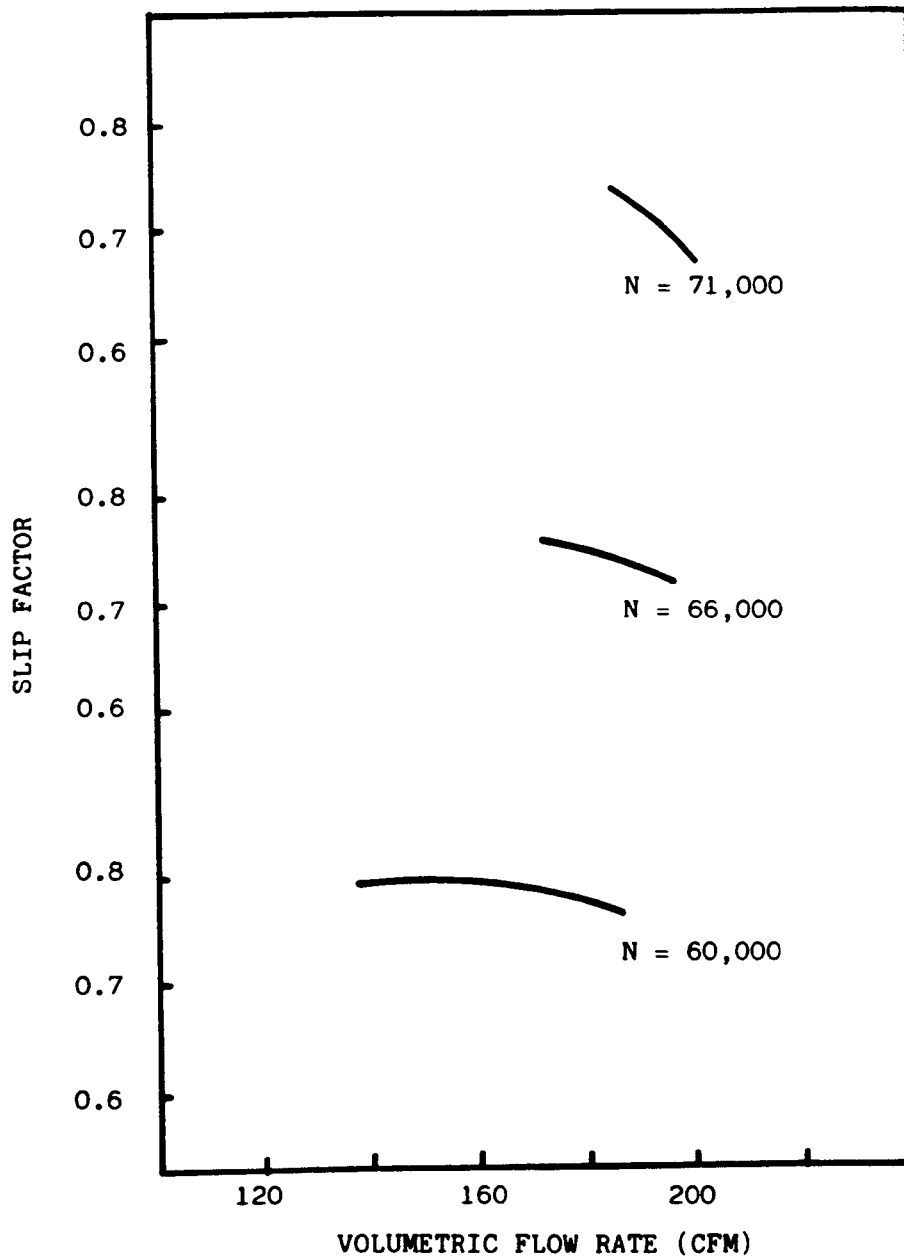


FIG. 2.5 RESULTS OF CENTRIFUGAL COMPRESSOR FREON PERFORMANCE TEST

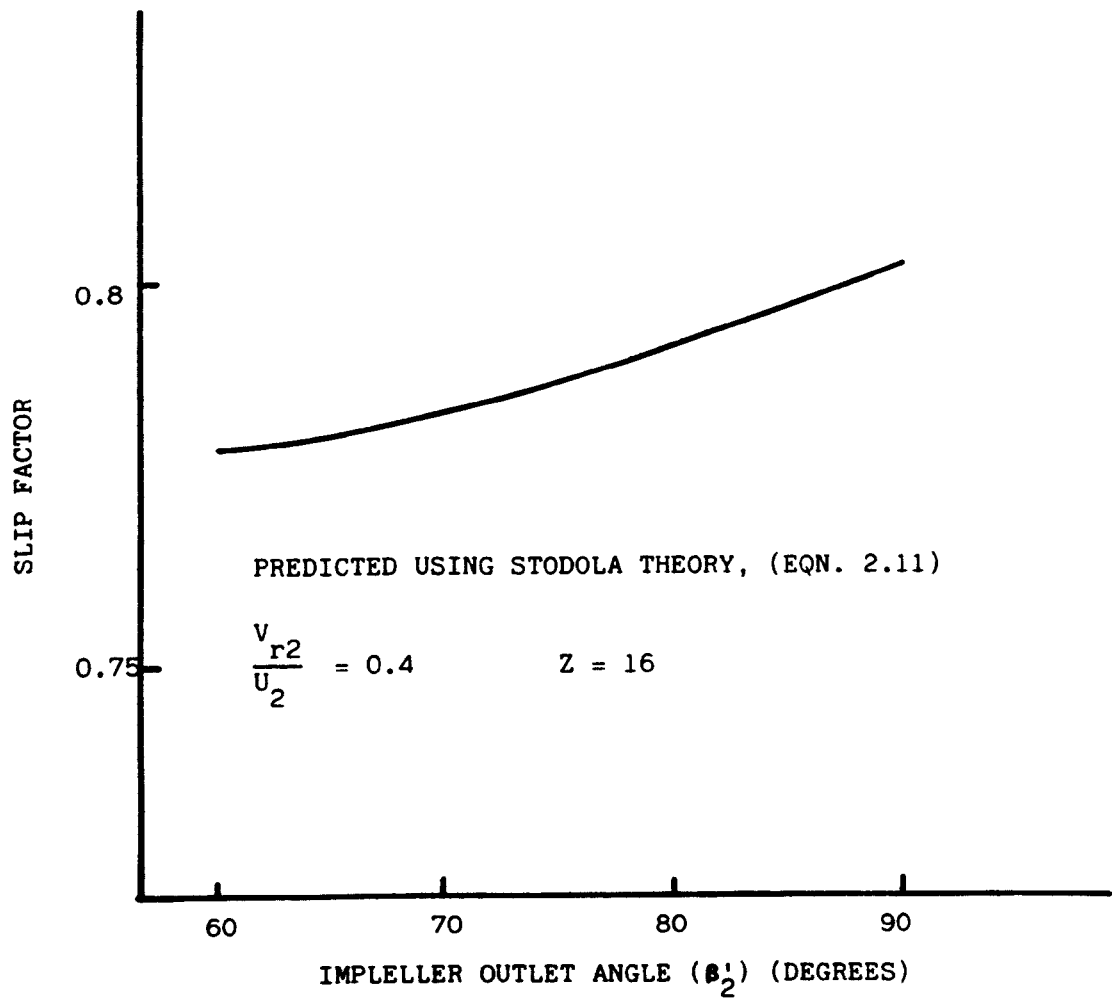


FIG. 2.6 VARIATION IN SLIP FACTOR WITH IMPELLER OUTLET ANGLE

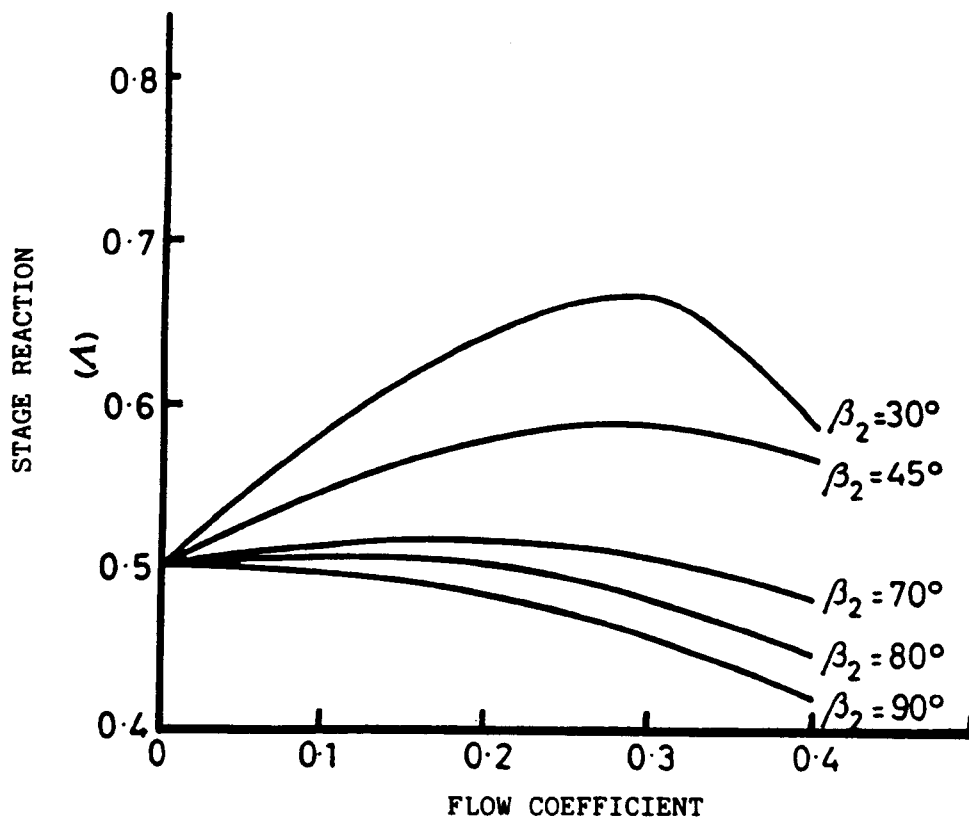


FIG. 2.7 VARIATION OF STAGE REACTION (A) WITH FLOW COEFFICIENT

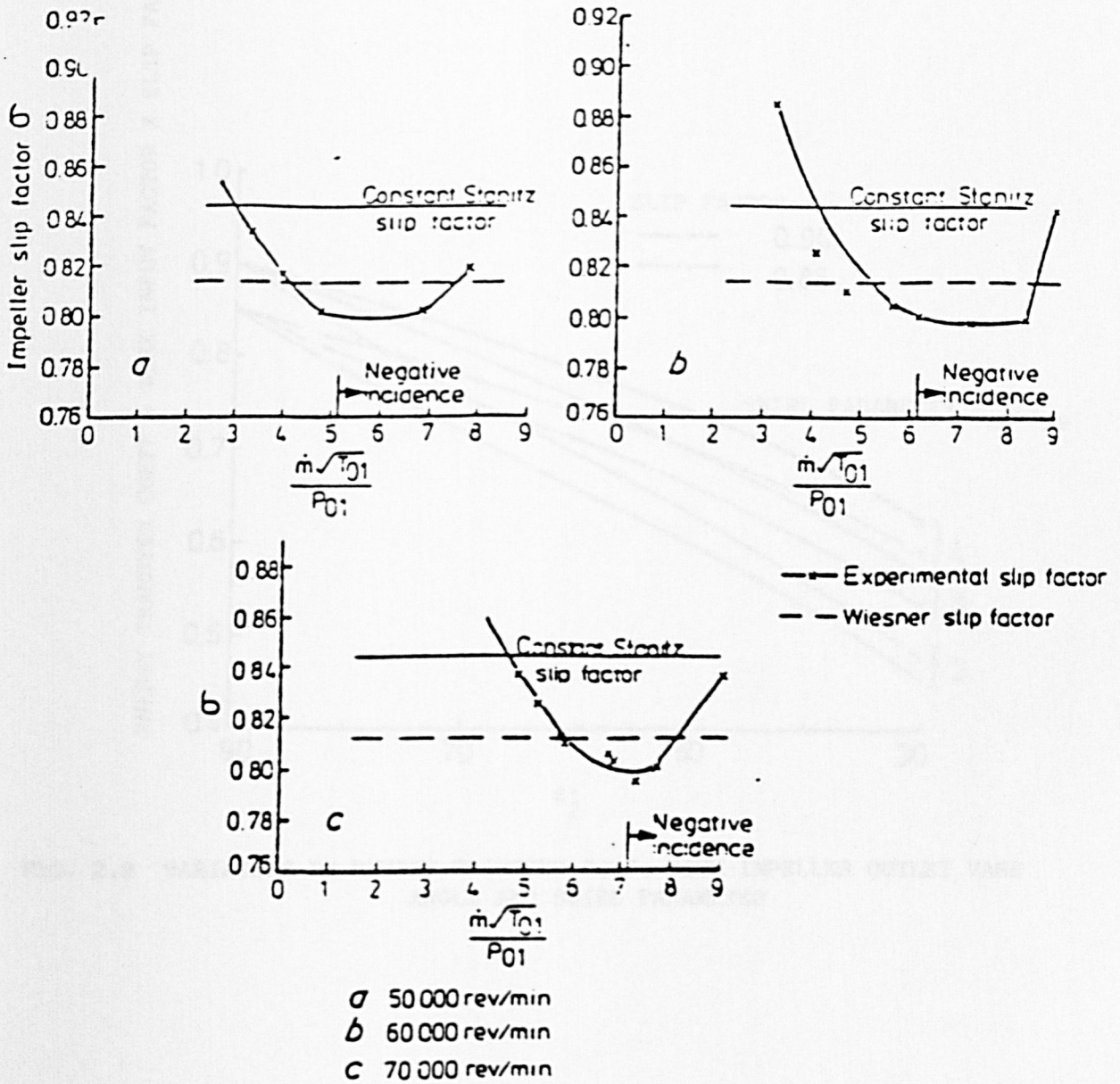


FIG. 2.8 EXPERIMENTAL SLIP FACTORS DUE TO WHITFIELD

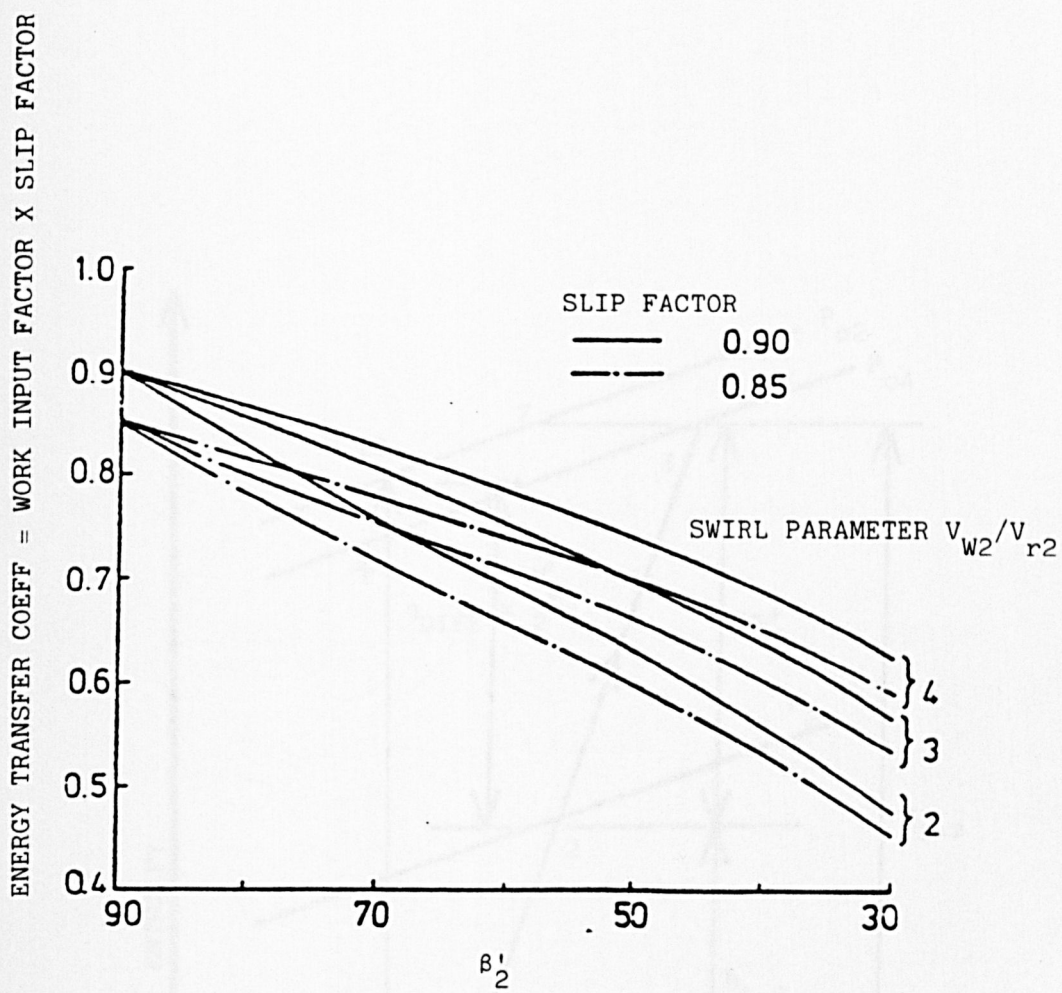


FIG. 2.9 VARIATION IN ENERGY TRANSFER COEF. WITH IMPELLER OUTLET VANE ANGLE AND SWIRL PARAMETER

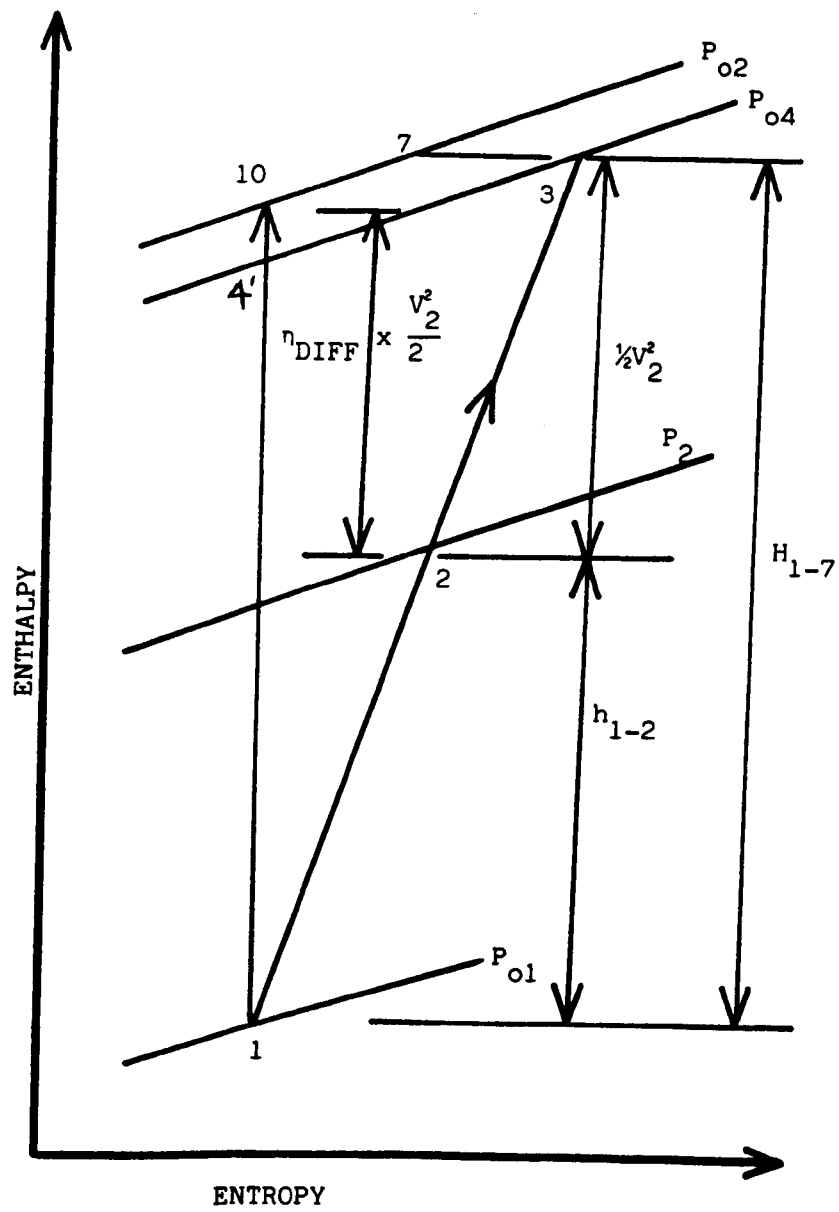


FIG. 2.10 IRREVERSIBLE COMPRESSOR STAGE

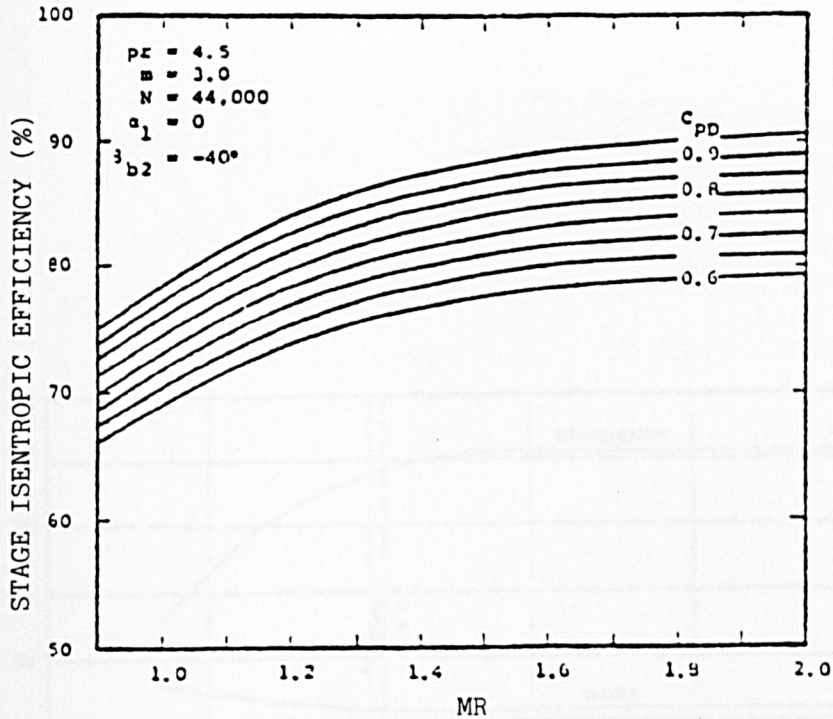


FIG. 2.11 INFLUENCE OF IMPELLER AND DIFFUSER PRESSURE RECOVERY PERFORMANCE UPON STAGE AND IMPELLER EFFICIENCY

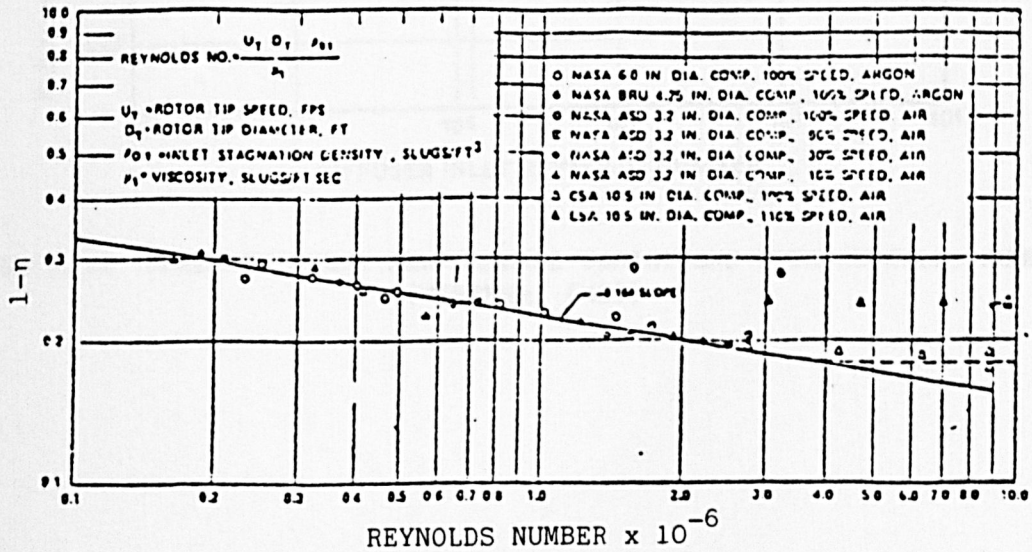


FIG. 2.12 THE INFLUENCE OF REYNOLDS NUMBER ON THE EFFICIENCY OF CENTRIFUGAL COMPRESSORS (PAMPREEN, (45))

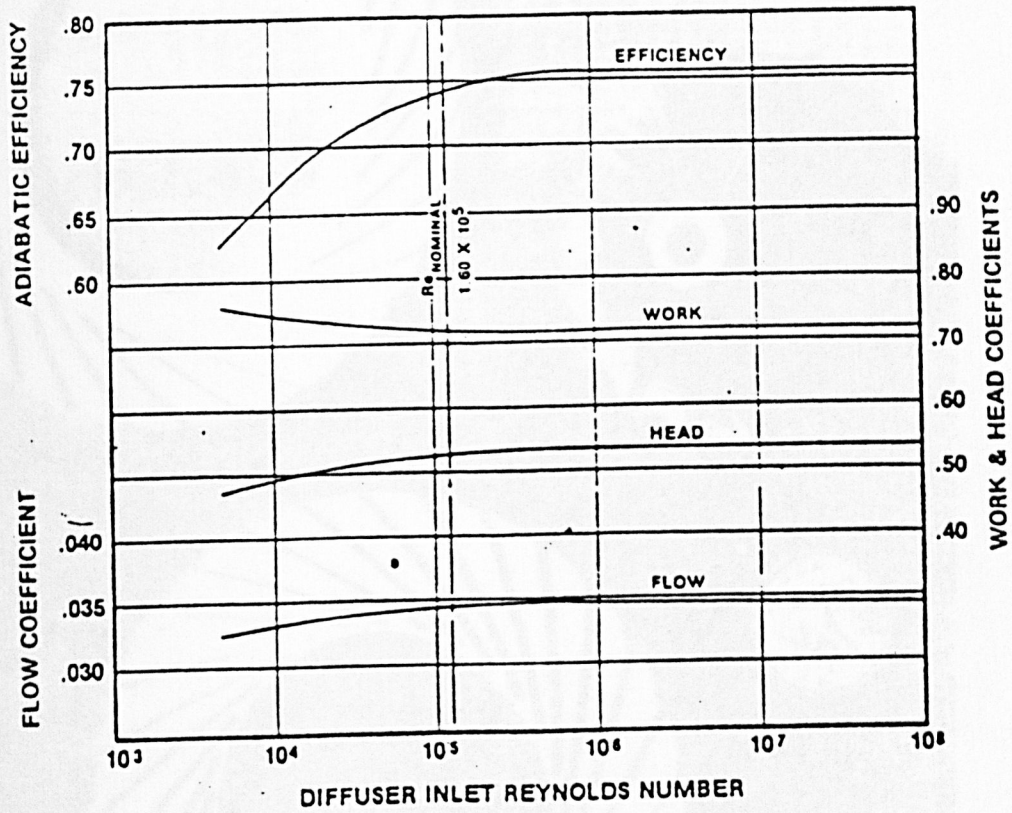


FIG. 2.13 PREDICTED PEAK PERFORMANCE VARIATIONS WITH REYNOLDS NUMBER
(WIESNER, (48))

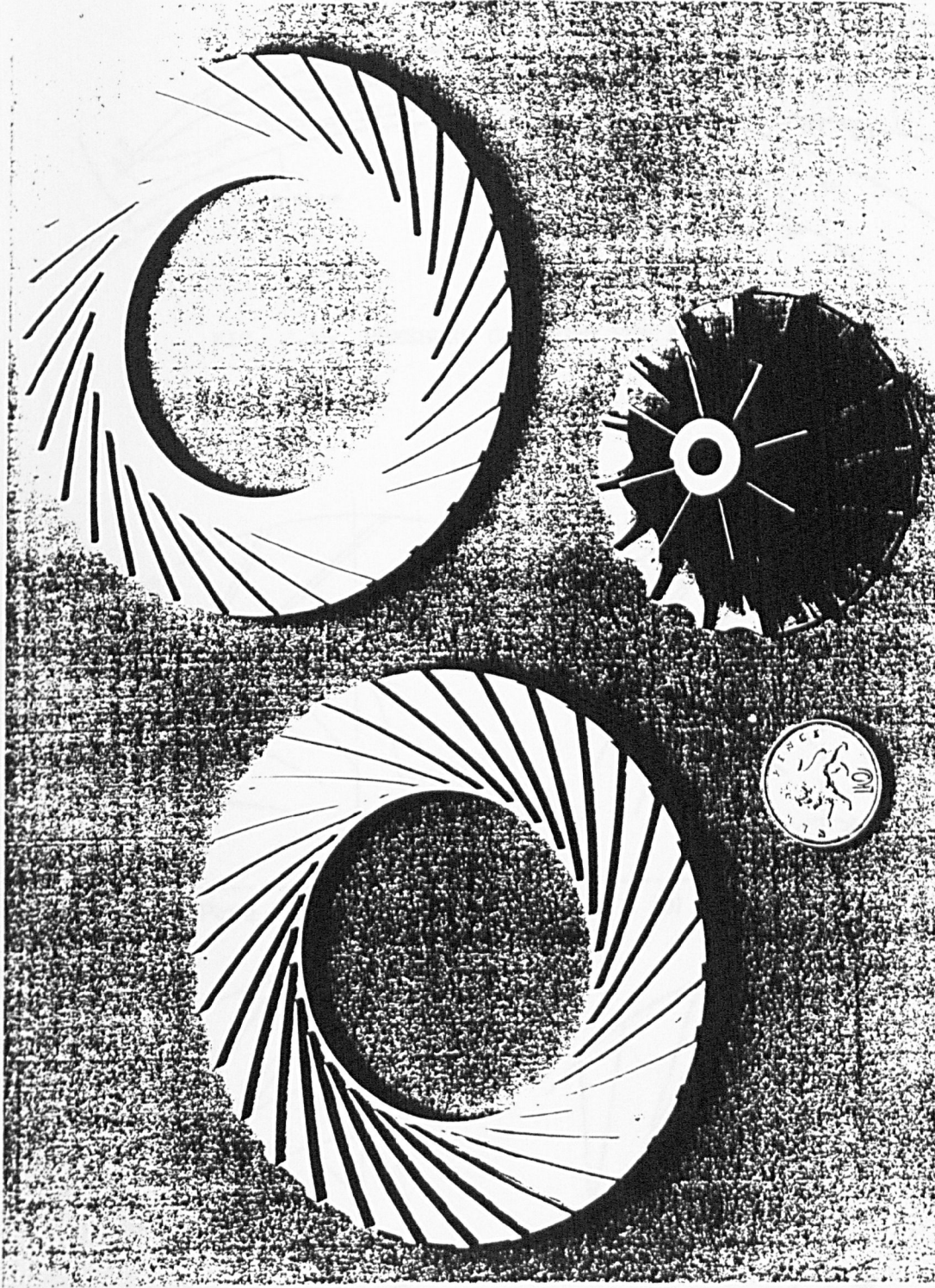


FIG. 2.14 VANED DIFFUSERS TESTED BY ELDER

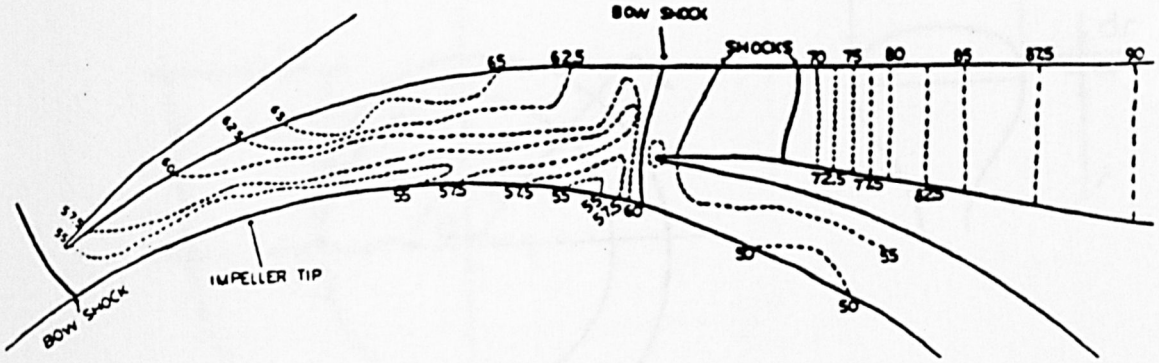


FIG. 2.15 PRESSURE CONTOURS FOR A VANED DIFFUSER

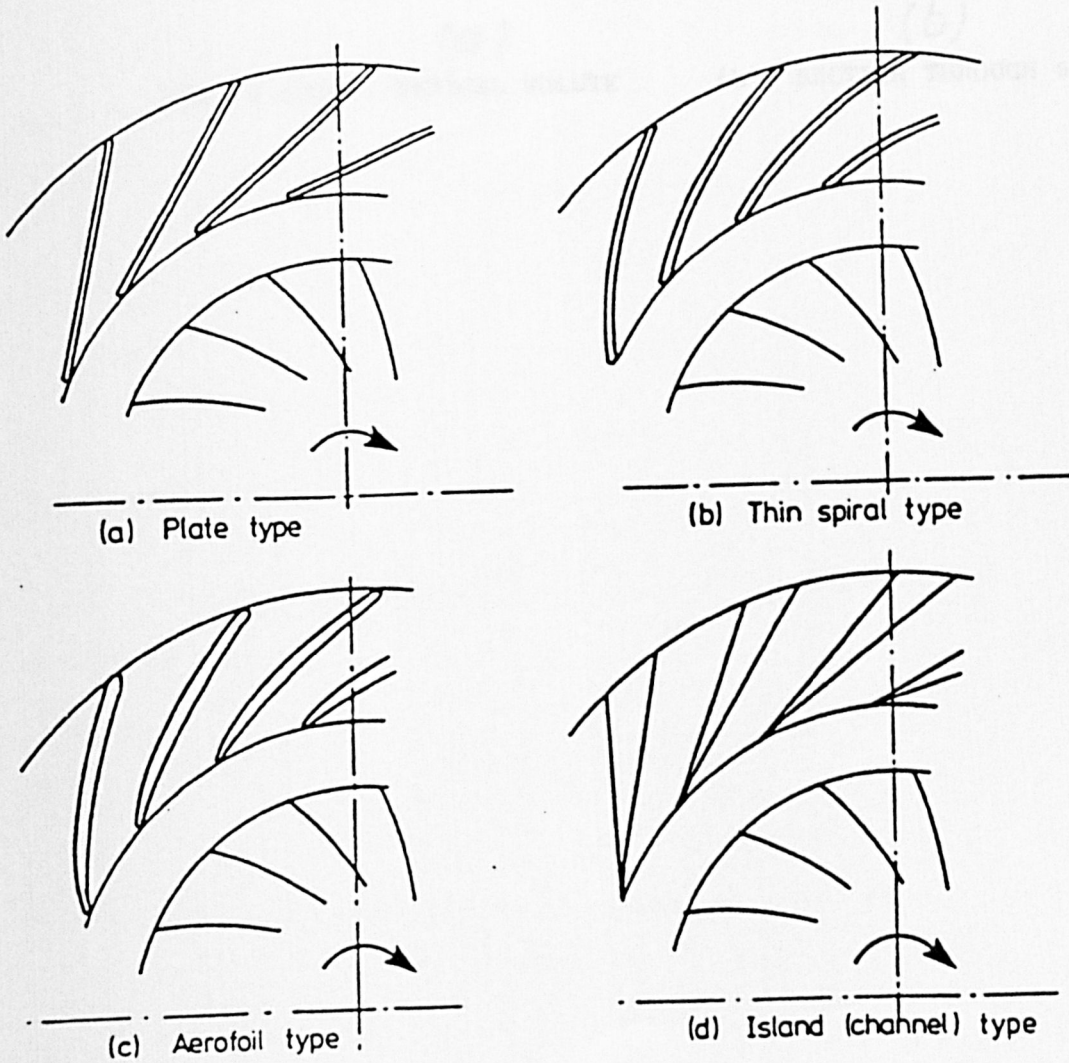


FIG. 2.16 VANED DIFFUSER TYPES

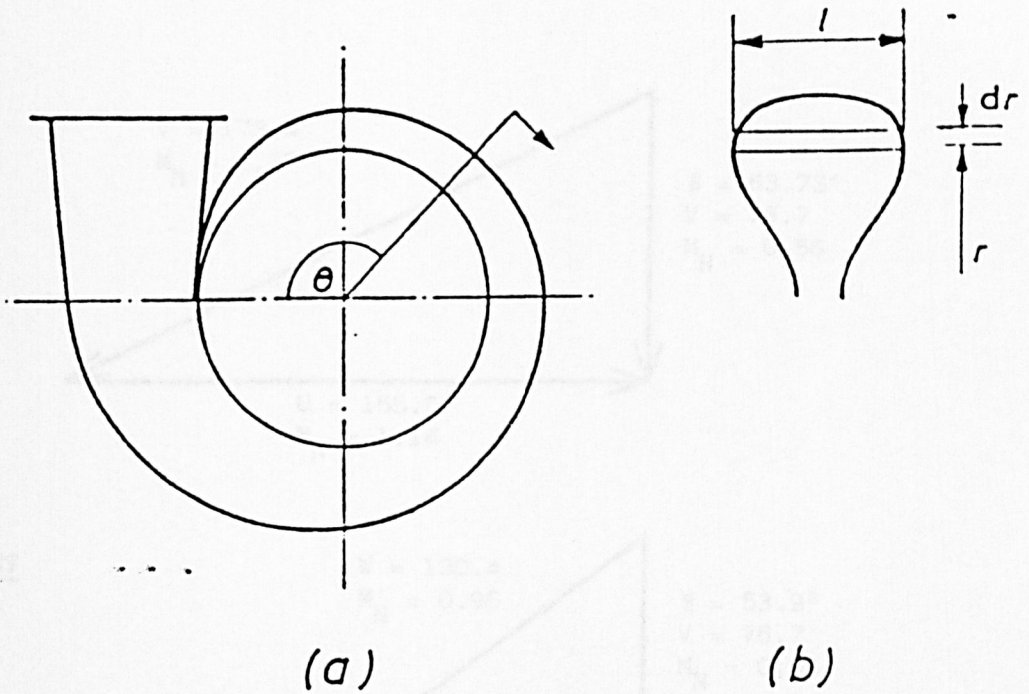


FIG. 2.17(a) TYPICAL VOLUTE

(b) SECTION THROUGH θ

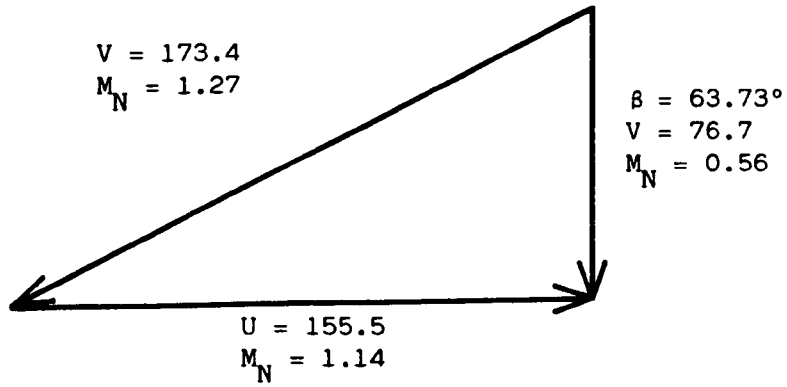
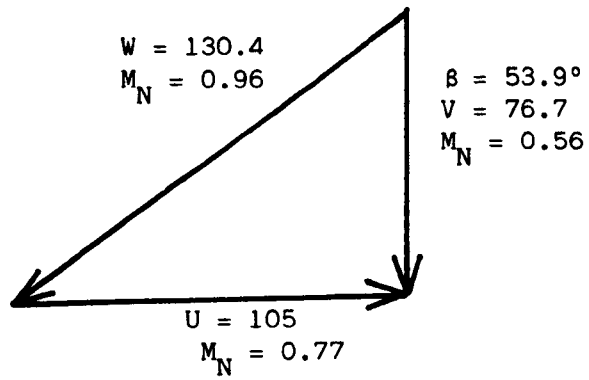
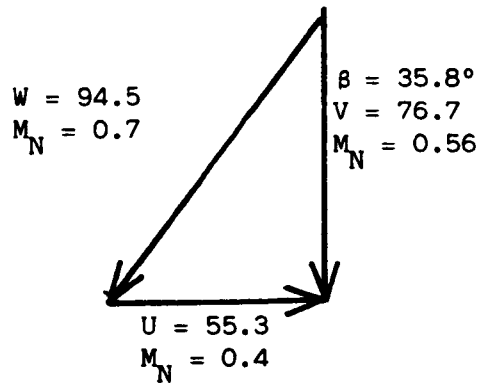
TIPMID HEIGHTROOT

FIG. 2.18 INLET VELOCITY TRIANGLES OF PRESENT IMPELLER (velocities in m/s)

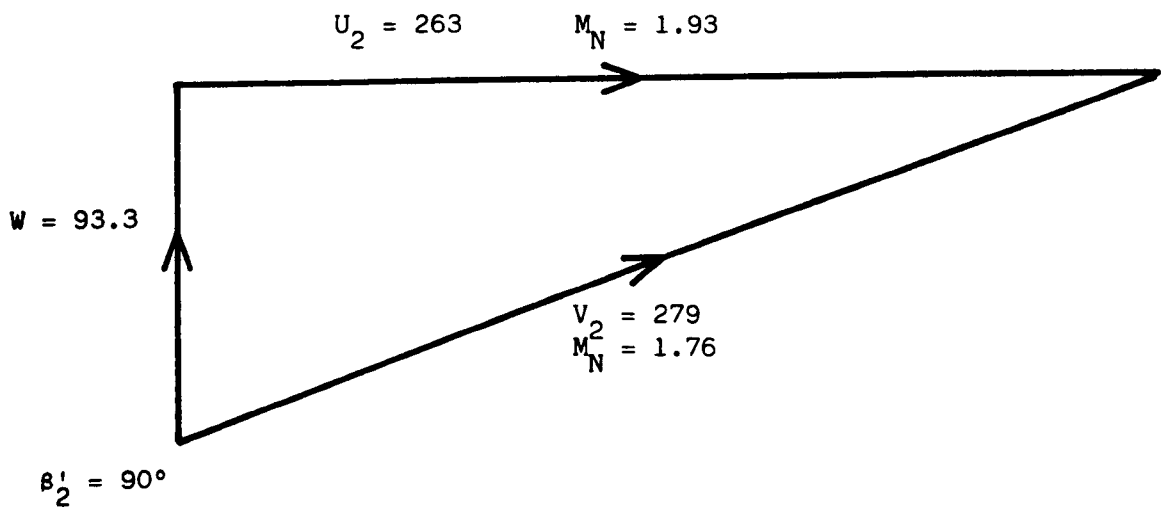


FIG. 2.19 PRESENT IMELLER TIP VELOCITY TRIANGLES

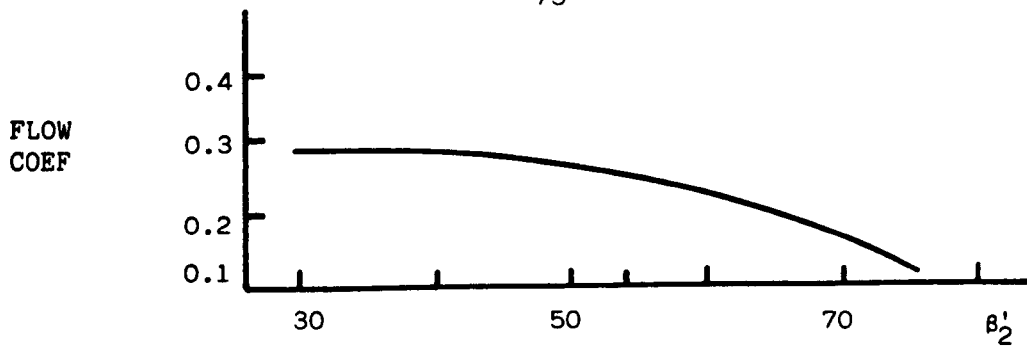


FIG. 2.20 VARIATION IN FLOW COEF WITH IMPELLER OUTLET ANGLE (ALL SPEEDS)

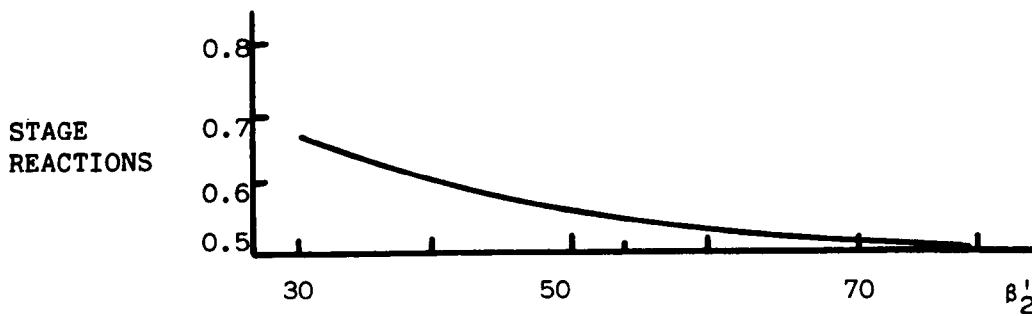


FIG. 2.21 VARIATION IN STAGE REACTION WITH VANE OUTLET ANGLE (ALL SPEEDS)

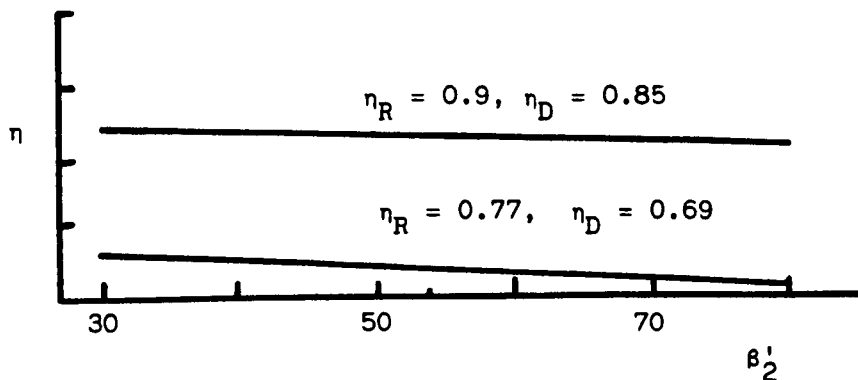


FIG. 2.22 VARIATIONS IN STAGE EFFICIENCY WITH VANE OUTLET ANGLE (ALL SPEEDS)

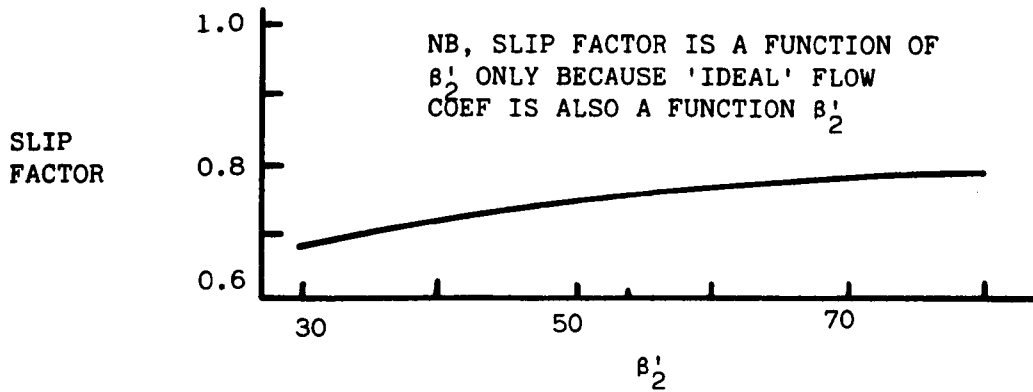


FIG. 2.23 VARIATION IN SLIP FACTOR WITH VANE OUTLET ANGLE (ALL SPEEDS)

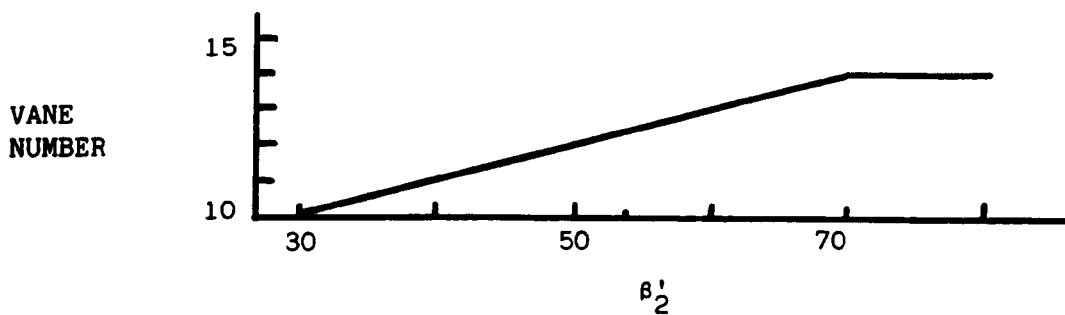


FIG. 2.24 VARIATION IN IDEAL VANE NUMBER (Z) WITH VANE OUTLET ANGLE

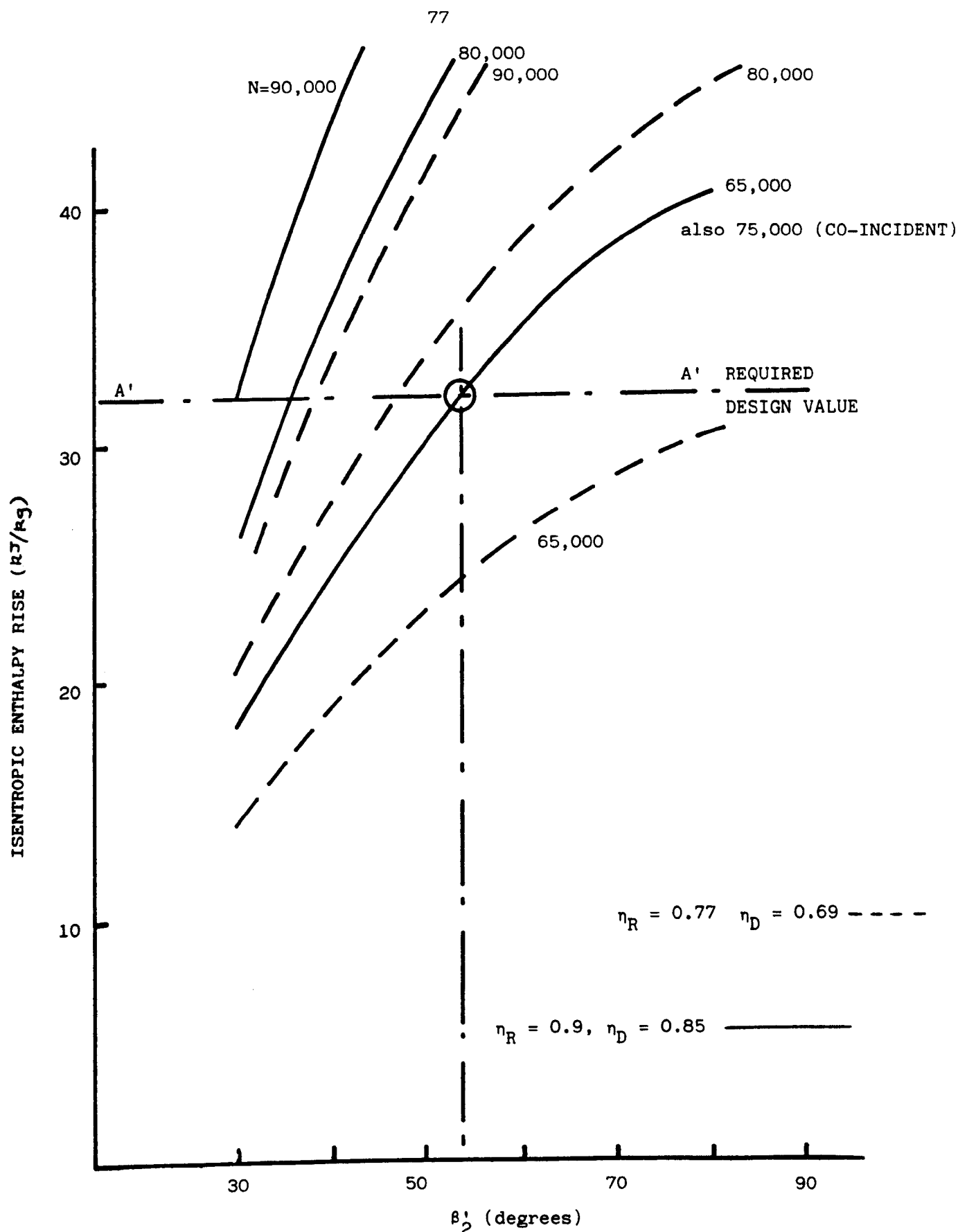


FIG. 2.25 VARIATION IN ISENTROPIC ENTHALPY RISE WITH VANE OUTLET ANGLE FOR VARIOUS ROTATIONAL SPEEDS, IMPELLER AND DIFFUSER EFFICIENCIES

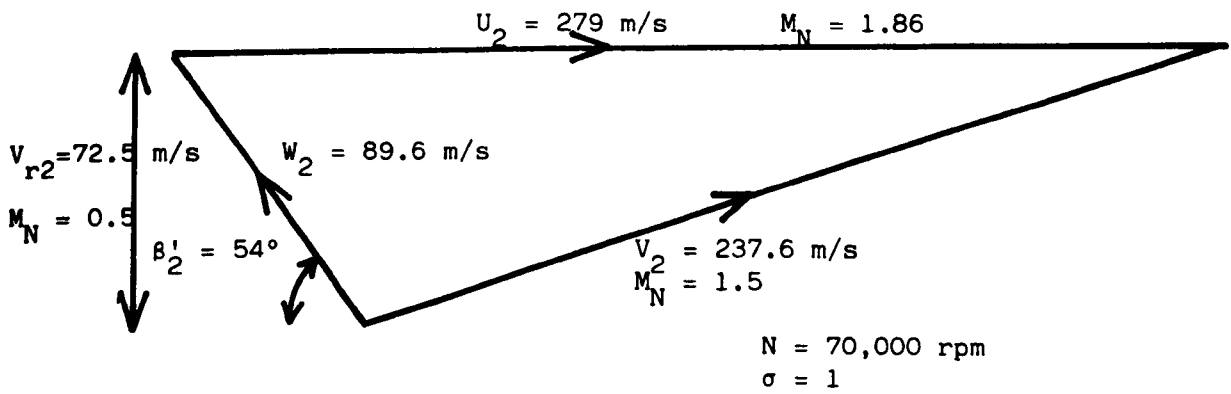
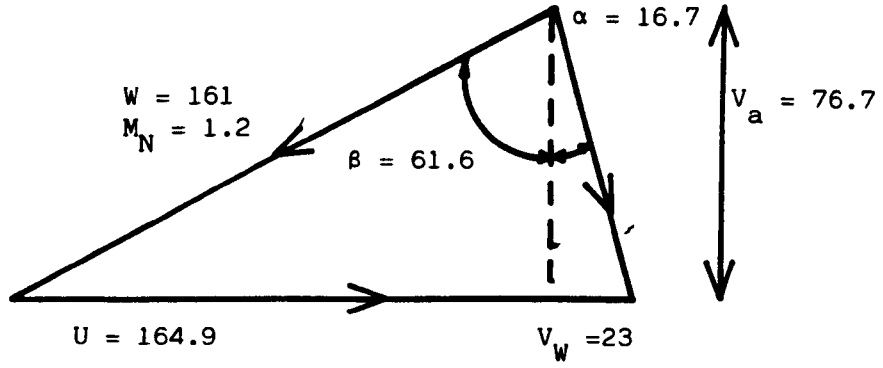
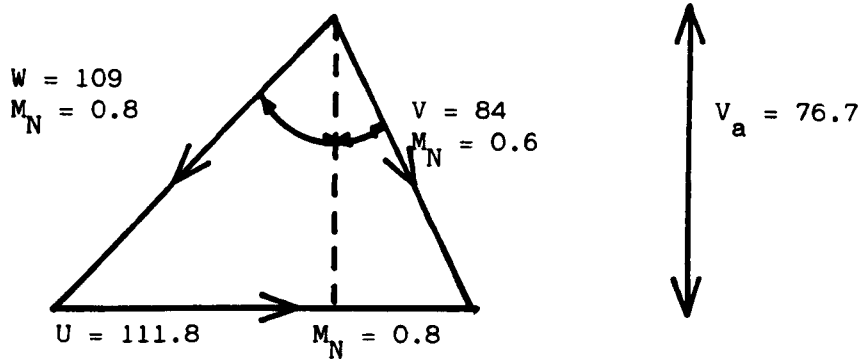


FIG. 2.26 RECOMMENDED IMPELLER TIP VELOCITY TRIANGLE

TIP



MID HEIGHT



ROOT

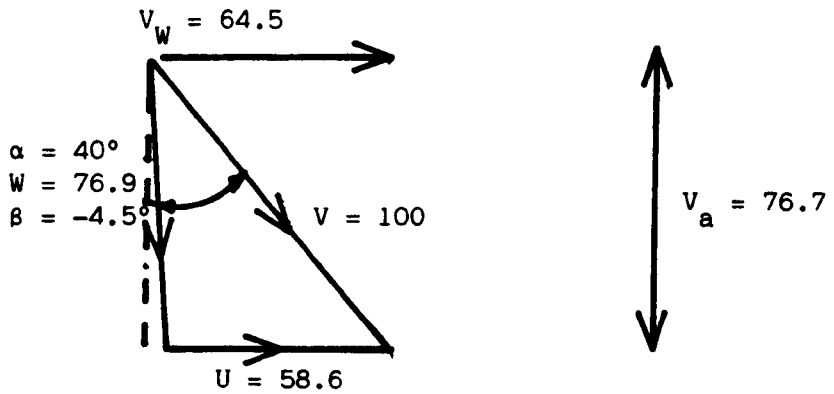


FIG. 2.27 RECOMMENDED IMPELLER INLET VELOCITY TRIANGLES

Ref.	β_2	Z	r_2/r_1	Exptl σ	Busemann ¹ σ	Stodola ² σ	Balje ⁵ σ	Stanitz ³ σ
6	20	6	2.24	0.77	0.76	0.79	0.64	0.61
6	20	6	2.24	0.76	0.76	0.77	0.64	0.58
6	20	7	1.85	0.76	0.71	0.75	0.63	0.54
6	24	6	2.77	0.78	0.76	0.73	0.68	0.59
15	23.5	6		0.75	0.73	0.73	—	0.57
7	27	5	3.1	0.68	0.69	0.68	0.63	0.55
8	30	6	1.66	0.53	0.62	0.60	0.58	0.49
8	30	6	1.66	0.64	0.67	0.65	0.58	0.55
8	35	6	1.66	0.56	0.63	0.60	0.58	0.55
9	35	18		0.88	0.88	0.86	—	0.85
10	40	8	1.76	0.75	0.74	0.67	0.65	0.68
10	40	16	1.76	0.84	0.86	0.84	0.79	0.84
10	40	32	1.76	0.95	0.92	0.92	0.88	0.92
9	50	16		0.909	0.81	0.80	—	0.83
9	57.5	13		0.85	0.81	0.71	—	0.78
9	64.2	16		0.87	0.83	0.79	—	0.85
	90	16	1.42	0.87	0.84	0.80	0.79	0.88
9	90	18		0.9	0.86	0.83	—	0.89
9	90	18		0.9	0.86	0.82	—	0.89
9	90	19		0.9	0.87	0.83	—	0.9

TABLE 2.2 COMPARISON OF EXPERIMENTAL AND THEORETICAL SLIP FACTORS (NB FOR REFS REFER TO FERGUSON ())

$$\frac{1 - \eta_T}{1 - \eta_{TOM}} = a + (1 - a) \left(\frac{Re_{TOM}}{Re_T} \right)^{\gamma}$$

YEAR	SOURCE	REF.	INDEPENDENT LOSS FRACTION $\bar{a}$	DEPENDENT LOSS FRACTION (1-a)	EXPONENT γ	MACHINE TYPE
1925	MOODY	22	0.25	0.75	0.33	PROPELLER TURBINES
1930	ACKERET/MÜHLEMANN	2,22	0.50	0.50	0.20	HYDRAULIC TURBINES
1942	MOODY	22	0.00	1.00	0.20	PUMPS
1947	PFLIEDERER	3	0.00	1.00	0.10	PUMPS
1951	DAVIS, KOTTAS & MOODY	4	0.00	1.00	VARIOUS	ALL TURBOMACHINES
1954	HUTTON	15,22	0.30	0.70	0.20	KAPLAN TURBINES
1958	ROTZOLL	6	0.00	1.00	VARIABLE	PUMPS
1960	WIESNER	12	0.50	0.50	0.10	CENTRIFUGAL COMPRESSORS
	FAUCONNET	15	0.24	0.76	0.20	
1961	O'NEILL & WICKLI	15	0.00	1.00	VARIABLE	CENTRIFUGAL COMPRESSORS
			0.00	1.00	0.20	AXIAL COMPRESSORS
1966	ASME CODE PTC-10	21	0.00	1.00	0.10	CENTRIFUGAL COMPRESSORS
1971	MASHIMO, ET AL	25	0.25 MIN.	0.75 MAX.	0.20	CENTRIFUGAL COMPRESSORS
1973	SAPIRO	26	$0.54 - (1 - \frac{NS}{120})^2$	$0.46 - (1 - \frac{NS}{120})^2$	VARIABLE 0.20 MAX.	CENTRIFUGAL COMPRESSORS
1974	MASHIMO, ET AL	28	0.15-0.57	0.43-0.85	0.20-0.50	CENTRIFUGAL COMPRESSORS

TABLE 2.3 HISTORICAL SUMMARY FOR WIENSERS EFFICIENCY CORRELATION ()

Design Point Data

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Compressor Pressure Ratio	5.6
Inlet Stagnation Pressure	0.55 bar
Inlet Stagnation Temperature	280.5 K
Mass Flow Rate	0.264 kg/s
$R = 60.5 \bar{C}_p =$	

TABLE 2.4Present Compressor Parameters Satisfying the Design Point Requirements

Rotational Speed	66,000 rpm
Corrected Flow Rate	173 c.f.m.
Isentropic Efficiency	60%

TABLE 2.5Geometric Data of Present Compressor

<u>Impeller Inlet</u>		<u>Impeller Discharge</u>	
Vane Tip Dia	45 mm	Vane Tip Dia	76.13 mm
Vane Hub Dia	16 mm	Vane Heights	2.56 mm
Vane Thickness	0.635 mm	Vane Thickness	1.00
Number of Vanes	8	Number of Vanes	8 (full), 8 ₂ (half)
Effective Flow Area	$1.242 \times 10^{-3} \text{ m}^2$	Effective Flow Area	$5.71 \times 10^{-4} \text{ m}^2$
Vane Angle (β_1)	68°	Vane Angle (β_2)	90°

TABLE 2.6Present Compressor Thermodynamic Data

	<u>Inlet (mean height)</u>	<u>Impeller Discharge</u>
$\dot{m}$ (kg/s)	0.264	0.264
A (m ²)	1.242×10^{-3}	5.718×10^{-4}
T (K)	280.5	374
P (bar)	0.55	3.08
V (m/s)	76.72	279
W (m/s)	130.4	93.27
M_{REL}	0.956	0.219

$$R = 60.5 \text{ J/kg/K} \quad C_p = 587 \text{ J/kg/K} \quad \gamma = 1.116$$

TABLE 2.7

Thermodynamic Data for Recommended Compressor

	<u>Inlet</u> (mean height)	<u>Impleller Discharge</u>
$\dot{m}$ (kg/s)	0.264	0.264
$A(m^2)$	1.242×10^{-3}	8.86×10^{-4}
T (K)	280.5	357
P (bar)	0.55	3.08
V (m/s)	83.9	237.6
W (m/s)	109.12	89.6
M_{REL}	0.8	0.528

R , C_p and γ as for Table 2.7

TABLE 2.8

CHAPTER THREE

MOTOR DESIGN

3.1 Introduction

The design of the electric drive motor was a key technological area for study. The motor was a three phase, two pole, squirrel cage, induction type. Because of the required power output and high rotational speed the design from both electrical and mechanical standpoints was non-standard and the design lay outside documented experience. The electrical design of the motor was carried out by Ron Hill²⁴. Significant mechanical design features were the integrity of the laminated rotor and the effectiveness of the motor cooling system.

The motor laminations were manufactured from Transil 315 which is a silicon steel suitable for electrical transformers. The material has similar mechanical properties to those of mild steel and has a 0.1% proof stress of 370 MPa. Conductor rods were manufactured from an aluminium alloy, 2014A. This is a solution and precipitation heat treated alloy with a 0.2% PS of 370 MPa and a UTS of 435 MPa.

The novelty of the motor design, particularly the high power density, (resulting from the high rotor speed), provided an invaluable study into the cooling of the motor. An indication of the magnitude of the problem compared with conventional 50 Hz motor designs is given by comparing surface area. Cooling effectiveness is proportional to the surface area of the motor. For a constant power output the power density, (power/volume), of a motor is inversely proportional to the rotational speed. The power density of the experimental motor would be 24 times that of a standard 50 Hz machine and the surface area would be one-eighth if geometric proportions were maintained. Thus the difficulty of cooling the present electric motor would be eight times greater than for a conventional 50 Hz type. A literature study revealed no material concerned with the problems of cooling refrigerant compressor motors. The lack of available data was thought to make this part of the study unique.

The quantity of heat generated by an electric motor is related to the efficiency by which electrical energy is converted to mechanical work. The efficiency characteristic shown in Figure 3.1 is thought typical of a polyphase induction motor, Alger³².

Class A insulation, proposed for the present motor, allows a maximum winding temperature of 160°C. Because of the rate of refrigerant decomposition at such temperatures and the improvement in efficiency resulting from a lower winding resistance, (with the lower temperature), it was decided, after consultation with Hill³⁴, that a maximum temperature of 90°C would be used for design purposes.

A further critical area for research was the mechanical integrity of the laminated rotor. The primary cause of mechanical stress was the dynamic effects produced by the high rotational speed, (70,000 rpm). Secondary effects included:

- i) Axial preload required to prevent movement of the laminations.
- ii) The effect of the conductor rods on laminated rotor strength.
- iii) The effect of heating on preload.
- vi) The transfer of drive from rotor to drive shaft.

These aspects of the study are considered in Sections 3.2 to 3.8.

3.2 Dynamic Stress in Laminated Rotor

For a solid hollow cylinder rotating about its central axis it would be conventional to adopt a plane strain model because the radial strains are resisted by the axial tensile stresses. In this design the cylinder is laminated and so axial tensile stresses could not be supported. Although individual laminations may support such a stress the rotor-pack cannot if compression stresses incurred during assembly are neglected. For this reason a plane stress model is more appropriate. This requires that all stresses are concentrated in the radial plane. Plane stress and plane strain methods are described in Appendix D.

When considering dynamic stresses in a rotating circular disc with a central hole it is assumed that the maximum stress for the plane stress condition are, Lancaster et al²⁴;

$$\sigma_{\max} = \frac{\rho \omega^2}{4} ((3+\nu) R_o^2 + (1-\nu) R_i^2) \quad 3.1$$

Applying the Tresca yield criteria the principle stresses at the inner radius are;

$$\sigma_{\min} = \sigma_r \quad \text{and} \quad \sigma_{\max} = \sigma_\theta$$

$$\text{at } R = R_i \quad \sigma_r = 0$$

$$\text{Therefore} \quad \sigma_{\min} = \sigma_\theta \quad \text{at } R = R_i \quad 3.2$$

During the preliminary electrical design study Hill²⁴, calculated that the rotor would require an outside diameter of 66mm and an internal diameter of 37.75mm. Preliminary stress calculations were carried out using the plane stress model which showed that the hoop stress at the inner radius would exceed the yield stress by 34 MPa at the design rotational speed. Although a small amount of material yield may not be detrimental to the safe operation of the rotor it was felt that any permanent deformation would disturb the dynamic balance of the rotor. In order to maintain the dimensional stability of the rotor it was decided that the tangential stress at the inner radius of the rotor at 1.05 times the design maximum operating speed, (73,500 rpm), should be less than the yield stress, (370 MPa), which is in agreement with the requirements of the Tresca failure criteria. From discussions with Hill²⁴ it was decided that the diameter to length ratio of the motor, (L/D_o), should be approximately 1.5 and the volume, (V), of the rotor should be similar to the volume of the initially designed rotor, (ie., 2.264 x 10⁻⁴ m³).

Applying these conditions the rotor volume is given by:

$$V_r = 1.5 \pi (R_o^2 - R_i^2) \quad 3.3$$

Rearrangement gives;

$$R_i = \left(R_o^2 - \frac{V_R}{1.5 \sqrt{R_o}} \right)^{1/2} \quad 3.4$$

Substituting Equations 3.2 and 3.4 into 3.1 and rearranging gives;

$$R_o^2 = \frac{\sigma_{YIELD}}{e\omega^2} + \frac{V_R (1 - \nu)}{6 \sqrt{R_o}} \quad 3.5$$

Substituting known values gives $R_o = 32.75$ mm

from which;

$$D_o = 65.5 \text{ mm}$$

$$D_i = 36.8 \text{ mm}$$

$$L = 98.3 \text{ mm}$$

These dimensions were almost the same as those calculated by Hill²⁴, given earlier.

3.3 Two Dimensional Stress Model

The calculations described in the previous section do not allow for the presence of conductor rods which were thought to increase local stress levels. In addition to a central hole each lamination had twentytwo 5mm diameter equally spaced holes set about its periphery. This is shown in Figure 3.2. The inclusion of these holes causes the stress distribution within the laminations to be altered from that suggested by the plane stress model.

A mathematical two-dimensional model was used to determine the stresses in the rotor laminations. Figure 3.2 describes the model and the nomenclature used. The lamination is divided into three regions, A, B and C, shown in Figure 3.2. Equations were developed to estimate the stress and strain in each region. The stresses at the interface of the regions are adjusted to give strain compatibility. In this way the stress distribution was determined. The derivation of the equations used in this model is described in Appendix E.

The equations for calculating the stresses in regions A and C, derived in Appendix E, are;

$$\sigma_R = \rho \omega^2 \frac{(3+\nu)}{8} \left[R_o^2 + R_i^2 - \frac{R_o^2 R_i^2}{r} - r^2 \right] - \frac{R_i^2 R_o^2}{r^2} \frac{(Z_o - Z_i)}{R_o^2 - R_i^2} - \frac{Z_i R_i^2 - Z_o R_o^2}{(R_o^2 - R_i^2)} \quad 3.6$$

$$\sigma_\theta = \rho \omega^2 \frac{(3+\nu)}{8} \left[R_o^2 + R_i^2 + \frac{R_o^2 R_i^2}{r^2} - \frac{(1-3\nu)}{(3+\nu)} r^2 \right] + \frac{R_i^2 R_o^2}{r^2} \frac{(Z_o - Z_i)}{(R_o^2 - R_i^2)} - \frac{(Z_i R_i^2 - Z_o R_o^2)}{(R_o^2 - R_i^2)} \quad 3.7$$

where:

$$\sigma_r = Z_o \text{ at } r = R_o$$

$$\sigma_r = Z_i \text{ at } r = R_i$$

Radial growth compatibility between each region is a necessary condition. During rotation laminations will be strained by an amount given by:

$$\Delta r = \frac{r}{E(\sigma_{\theta} - \nu \sigma_r)} \quad 3.9$$

The derivation of Equation 3.9 is detailed in Appendix E.

Because of its slender circumferential dimension it was assumed that Region B would be subjected to radial stresses only and as such should behave as a simple rod in tension with the addition of body forces due to rotation. The centrifugal force, (Fr) , at any radius within Region B, (Figure 3.2), is given by:

$$F(r) = \int_r^d [\omega^2 \rho r^2 \theta] dr - \int_r^d (\omega^2 2\rho r \sqrt{C^2 - (B-r)^2}) dr \quad 3.10$$

From the geometry of Region B the stress is given by:

$$\sigma_r \approx \frac{F(r)}{r\theta - 2\sqrt{C^2 - (B-r)^2}} \quad 3.11$$

and the radial growth of region B is given by;

$$\Delta r_B = \frac{1}{E} \int_A^B \sigma_r dr \quad 3.12$$

The effect of conductor rods on stress within the laminations was thought to be an important factor as it was felt possible for a conductor rod to tear the wall between the hole and the outer skin of the rotor. A reasonable approximation of the effect on stress within region B, due to the centrifugal load of the conductor rods, was given by assuming the centrifugal force produced by each conductor rod is resisted by region B. The centrifugal force acting on a conductor rod is given by:

$$F_2 = \pi C^2 B \rho \omega^2 t \quad 3.13$$

(per unit length of rod)

where t = lamination thickness

Under the action of centrifugal force, the hoop like, Region C would expand to a diameter greater than if it were restrained by Region B. This imposes an additional load on Region B causing it to extend radially. The force, (F_3), would initially have an unknown value necessary to maintain strain compatibility between regions B and C. Applying the principle of superposition the stress at any radius within region B is given by adding Equation 3.10 and 3.14 to the compatibility force, F_3 .

$$\sigma_r = \frac{F_1 + F_2 + F_3}{r\theta - 2\sqrt{C^2 - (B-r)^2}} \quad 3.14$$

The solution of Equation 3.14, described in Appendix E., was made using the geometrical data of the rotor based on Equation 3.1, ie. $D_o = 64.2$ mm and $D_i = 36.8$ mm, and the conductor rod data supplied by Hill²⁴. The results, listed in Table 3.1, showed that the tangential stress exceeds the yield point stress to a radius of 22 mm.

Reducing the external diameter of the lamination to 62 mm and the internal diameter to 34 mm, the tangential stress throughout the lamination is reduced. Calculations were repeated using these dimensions. The results listed in Table 3.2 showed that the tangential stress still exceeded the yield point stress over part of the diameter.

Following a number of redesigns the following geometry was found to satisfy the imposed stress conditions, (ref Figure 3.2):

A = 23.35 mm
 B = 25.85 mm
 C = 2.5 mm
 D = 28.35 mm

From these results the required dimensions of the laminated rotor were estimated:

$D_o = 58.7$ mm
 $D_i = 31.75$ mm
 $L = 88.05$ mm (for $L/D = 1.5$)

With the exception of the length of the laminated rotor, which was reduced to 77 mm, these dimensions were subsequently used in the manufacture of the rotor.

A 5% overspeed condition was assumed and calculations were repeated based on the above geometrical data. Table 3.3 lists these results and Table 3.4 lists the results of

calculations at design point speed. Table 3.5 and 3.6 list results of calculations assuming 10% and 15% overspeed conditions respectively.

The predicted stress distribution, in a lamination, is shown in Figure 3.3 constructed from data listed in Table 3.4.

3.4 Pre-Load in Laminated Rotor

The laminations were held in place by a 'squirrel cage' arrangement of conductor rods which pass through each lamination and through shorting rings at each end of the laminated pack. This is shown schematically in Figure 3.4. The laminated pack was compressed and the conductor rods were riveted at each end placing the assembly under compression. Motor acceleration causes the centrifugal forces to strain the laminations in the axial direction. Without sufficient axial pre-load the laminations can become free to move causing the dynamic balance of the assembly to alter. To determine the magnitude of pre-load it was necessary to determine the strain in the rotor due to centrifugal load. Hill⁵⁷ proposed that the conductor rods should be used to pre-load the rotor during assembly. It was estimated that the stress in the conductor rods to produce the required pre-load would cause the rods to plastically deform.

3.5 Thermal Effects on Pack Pre-Load

During the assembly of laminated pack, shown schematically in Figure 3.4, a pre-load would be introduced which would produce a compressive stress into each lamination of $-E_s e_z$ and a tensile stress in the conduction rods of $E_s e_z$. If the laminated rotor were subjected to uniform temperature rise, ΔT , then the compressive stress in the pack would reduce:

$$\sigma_z = E_s [K_a + K_s] \Delta T = E_s e_z \quad 3.15$$

where k_a and k_s are the coefficients of linear expansion for steel and aluminium respectively.

The laminated structure of the rotor would not resist tensile loads and therefore $\sigma_z < 0$. Rearrangement of Equation 3.15 shows that for σ_z to remain negative then;

$$\Delta T < \frac{e_z}{(K_a - K_s)} \quad 3.16$$

Using the conductor dimensions selected by Hill²⁷, it was calculated that the maximum strain imposed by the conductor tie rods on the laminations would be 4.8×10^{-4} .

Given: $k_a = 1.34 \times 10^{-5} / ^\circ\text{C}$ and $k_s = 2.34 \times 10^{-5} / ^\circ\text{C}$. Therefore the rise in temperature required to reduce to pre-load to zero would be 48°C , from Equation 3.16. If the laminated pack were to be assembled at an ambient temperature of 20°C and if the temperature of the rotor was to be increased to 68°C , during operation, then the pre-load would be completely removed and the laminations would be free to move on the conductors.

The combined effect of increasing temperature and dynamic loading would act together to reduce pre-load. It was concluded that the conductor tie-rods themselves could not impose sufficient pre-load in the laminated pack during assembly to prevent the laminations moving on the drive shaft during normal operation and it would be necessary to provide additional preload by means of a special securing nut as shown in Figure 3.4.

3.6 Assembly Preload

A proposed method of assembling the motor rotor is shown schematically in Figure 3.4. To ensure a compressive pre-load is maintained throughout the effects of dynamic and thermal strains were taken into account. It was determined that the thermal strain, due to differential expansion of the steel shaft and the aluminium conductors, was given by:

$$e_a = \frac{(K_a - K_s) \Delta T}{(EA)_a + \frac{(EA)_s}{1}} \quad 3.17$$

In the present case;

$$\begin{array}{ll}
 E_s = 200 \text{ GPa} & E_a = 69 \text{ GPa} \\
 A_s = 1.134 \times 10^{-3} \text{ m}^2 & A_a = 4.32 \times 10^{-3} \text{ m}^2 \\
 K_s = 1.34 \times 10^{-5} / ^\circ\text{C} & K_a = 2.34 \times 10^{-5} / ^\circ\text{C}
 \end{array} \quad 3.18$$

Assuming $\Delta T = 100^\circ\text{C}$

$$e_a = -8.838 \times 10^{-4} \text{ and } e_s = +1.161 \times 10^{-4}$$

From which;

$$\sigma_a = -61 \text{ MPa and } \sigma_s = 23 \text{ MPa} \quad 3.19$$

This result showed that an axial clearance of $9.18 \times 10^{-4} \text{ m}$ would have existed between the rotor laminations and aluminium shorting ring if no compressive pre-load were introduced during assembly.

To ensure that the rotor laminations would remain under a compressive load throughout it was necessary to include axial strain equal to the sum of the thermal strain and the dynamic strain. As dynamic strain varies with radius the maximum value was be used. This was estimated to be -6.03×10^{-4} occurring at the inner radius of each lamination.

$$\begin{aligned}
 \text{Therefore assembly strain} &= 6.03 \times 10^{-4} + 9.18 \times 10^{-4} \\
 &= 6.1218 \times 10^{-4}
 \end{aligned}$$

This strain would have introduced a compressive stress 122 MPa during assembly. As shown in Section 3.5 the aluminium conductors would have supported only 94.4 MPa or 77% of the required load. The remaining pre-load was provided by a steel securing nut shown in Figure 3.4. Under these circumstances a securing nut would need to provide an axial load of 46.7 kN.

To enable the tie rods to be fitted it was necessary to compress the rotor pack by an amount equal to the sum of the strains undergone by both the rotor and the tie rods when the assembly is complete, ie., $e_r + e_c$

$$\text{Conductor Strain} = 370/69 \times 10^{-3} = 5.362 \times 10^{-3}$$

$$\text{Rotor Strain} = 94.4/200 \times 10^{-3} = 4.72 \times 10^{-4}$$

$$\text{Total strain} = 5.834 \times 10^{-3} \text{ m/m}$$

The amount of stress induced in the rotor to produce the required strain would have caused the rotor to plastically deform. This would have been unacceptable because the effect would have been to remove some of the necessary pre-load. To enable the rotor to be assembled with the correct pre-load the laminated rotor required to be compressed to its elastic limit, (giving 1.85×10^{-3} of strain). If the conductor rods were heated to 190°C before assembly and fixing then the correct amount of tension would have been achieved, (thermal expansion compensating for elastic strain). There was a practical disadvantage to this approach. Unless the conductor rods could have been fixed to the shorting rings without reduction in temperature then the required pre-load would not be achieved. It is also felt that there would be practical problems in trying to simultaneously secure twenty-two conductor rods. For these reasons the rotor was assembled with the assembly at room temperature with the rotor compressed to the elastic limit.

The resulting strain induced by following the above procedure was:

$$e_s = \frac{\frac{\sigma_{\text{yield}}}{E_s}}{1 + \frac{(EA)_s}{(EA)_a}} \quad 3.20$$

Substituting known values gives;

$e_s = 1.729 \times 10^{-3}$. After assembly the resulting strain is 1.135×10^{-4} ,

but the required strain was 6.1218×10^{-4} . The strain deficit was made up by the securing nut shown in Figure 3.4. To enable this strain to be produced it was necessary for the securing nut to provide an axial load of 211 kN, (21 tons). It was estimated that the torque loading required during the assembly of the compressor was 386 N-m, (285 lbf-ft).

3.7 Rotor Bursting Speed

For the purpose of this analysis it was assumed that the rotor would mechanically fail when the tangential stress at the outer surface layer exceeded the yield stress of the material; in this case 370 MPa. Applying the plane stress model and allowing σ_θ to equal σ_{yield} , (370 MPa), gave a bursting speed of 89,870 rpm.

3.8 Discussion

The two-dimensional stress model described in Section 3.3 assumed that the magnitude of the tangential stress would be zero within Region B which suggested that the calculations over estimate the radial stress and over estimate the mean radial stress at the interfaces between Region B and Regions A and C. Thus the stress model over estimates the tangential stress at the circumference of the rotor lamination central hole. It was concluded, therefore, that a rotor designed using the two dimensional model described in this section would be satisfactory from a mechanical strength standpoint.

The stress distribution in the laminations, shown in Figure 3.3, indicates a sharp fall and rise in tangential stress which is unrealistic and comes from the assumption that the circumferential stress in region B is zero. The stresses would tend to distribute themselves so that there would be a

less abrupt transition in stress values between the three regions.

It is believed that the two-dimensional stress model described provides a useful simple means for the design of the present high speed rotor. The use of finite element programmes in later studies may provide a more exact solution.

The rotational speed at which plastic deformation would first occur was thought to be under estimated due to an over estimation of tangential , (hoop), stress in the lamination central hole. It was not known by what amount the 'first deformation' speed was underestimated.

It had been proposed to use a key to transfer the drive from the rotor to the drive shaft. This is standard practice for low speed machines, (ie 50Hz). Because of the relatively high stress concentration at the periphery of the laminated rotor the addition of the keyway in the design would have required the overall dimensions of the rotor to be far less than it otherwise would have been. It was decided to leave the keyway out of the design and to transmit the necessary torque via friction between the rotor and the shaft. The torque required to transmit 12.5 kW, (maximum input power less electrical losses), at 70,000 rpm is 1.7 N-m. The torque is transmitted from the rotor to the drive shaft via the frictional resistances between the rotor shorting plates, the slotted nut, (see Figure 3.4) and the location shoulder. The axial load provided by the securing nut needed to transmit this power was determined from simple clutch theory to be 22kN. This was taken into account during the estimation of the securing nut torque load.

It was understood that there would be some delay in the delivery of the electric motor. So that testing could proceed it was decided to design a dummy rotor to fit in place of the laminated rotor. This was manufactured in mild steel because of the similarity between the UTS values of mild steel and Transil 315.

3.9 Motor cooling

It was estimated that 2kW of heat would be rejected by the motor. This was based on experimental data for the centrifugal compressor determined by Elder and Walters³ and assuming an electrical conversion efficiency of 80%. In practice iron and copper losses and magnetic hysteresis are the main sources of heating. Primary copper losses, ($I^2 \times R$), can be determined from the length of conductor used in the windings. Material resistivity and cross sectional area and current effect this. Slip losses result from heating produced as current flows increase in squirrel cage bars as slip increases.

Core losses, (or iron loss), originate from eddy current heating of the stator and rotor cores, Alger³². In the present case core losses were reduced by using well annealed transformer steel, (3% silicon Transil 315), and by ensuring a high standard of fabrication.

Eddy current losses are caused by the running clearance between the rotor and the stator particularly with high frequency supplies, as in the present case, Alger³². The use of aerodynamic bearings kept this clearance to 0.8 mm thus keeping flux leakage to an acceptable level, Hill³⁴.

The following sections evaluate the potential application of natural and forced convective cooling of the motor casing, (indirect cooling), and internal cooling techniques, (direct cooling).

3.10 Natural Convection and Radiant Cooling

Natural convective and radiant cooling offered the simplest solution to maintaining the motor winding temperature within acceptable limits particularly if this could be achieved without the need for external finning. To assess the feasibility of this approach a one-dimensional radial conduction with internal heat generation model was developed to determine the steady state temperature of the motor windings assuming natural convection and radiant heat transfer from the outer casing. This is described in Appendix H. The results of this analysis, plotted in Figure 3.5, show that with an expected heat loss of 2 kW the core temperature of the motor winding would be in the order of 800°C which would be unacceptable for class 'A' insulation

which is limited to 160°C.

3.11 Forced Convective Cooling

Appendix H describes a mathematical model to examine the feasibility of using forced convection from the motor casing to cool the motor. The results of this study, shown in Figure 3.6, indicate that with a mean air velocity of 41 m/s and a cooling rate of 2 kW the winding temperature would be 500°C which was unsatisfactory.

A problem with forced convention would have been a possibility of significant pressure loss in the cooling flow. It was estimated that pressure losses would have been approximately 2 kPa for a cooling air flow temperature rise of 5°C. The fan power to maintain this flow would be in the order of 700 watts which is an additional energy cost to the compressor system.

The addition of fins to the motor casing surface was investigated. The analysis is described in Appendix H. Several fin heights, thicknesses and spacings were assumed. Figure 3.7 shows the results of this analysis for a flow velocity of 40 m/s and ambient temperature of 20°C.

Alpha in Figure 3.7, is the ratio of effective heat transfer area to the area of the original casing. In order to dissipate 2 kW of heat it was estimated that the core temperature of the motor would be 215°C with an effective heat transfer area of 4 times the original area of the casing. This temperature was still 55°C above the maximum acceptable value of 160°C. Further calculations showed that doubling both flow velocity and effective area would not reduced the temperature to an acceptable level. In addition the study showed that this would have produced a significant increase in pressure loss through the fins resulting in greater fan power consumption. Also an increase in the number of fins was thought to be impractical from a manufacturing standpoint because of the increasing flimsyness. It was concluded from the study that some form of internal cooling would be necessary.

3.12 Comparison of Internal Motor Cooling Techniques

The use of compressor discharge gas to cool the motor was investigated. It was estimated that a typical discharge

temperature would be in the order of 70°C. providing a 20 degree thermal potential, (ie 90-70). Compared with an assumed suction gas temperature of 13°C which would provide a 77 degree potential, (ie 90-13). The heat transfer coefficient between the windings and the gas would need to have been 385% greater, (ie 77/20), for the discharge gas. Even if higher winding temperatures were acceptable the flow velocity and pressure losses across the windings necessary to produce the required heat transfer would make discharge gas cooling prohibitive and therefore unsuitable for the present case.

A disadvantage of the suction cooling technique is that the heat added to the gas causes the density to decrease prior to entering the compressor impeller. Under constant speed and pressure ratio conditions centrifugal compressors are constant volume machines and the mass flow through the compressor can be estimated from;

$$\dot{m} = \rho \dot{V} \quad 3.21$$

By assuming the gas to be ideal then;

$$\dot{m} = \frac{P \dot{V}}{RT} \quad 3.22$$

For a constant inlet pressure the ratio of mass flow between the suction gas cooling and discharge gas cooling may be approximated by;

$$\frac{\dot{m}_1}{\dot{m}_2} = \frac{T_2}{T_1} \quad 3.23$$

where: T_1 = compressor inlet temperature without suction gas motor cooling
 T_2 = compressor inlet temperature with suction gas motor cooling
 $\dot{m}_1$ = refrigerant mass flow without suction gas motor cooling
 and $\dot{m}_2$ = refrigerant mass flow with suction gas motor cooling

The heat added to the suction gas during motor cooling is given by;

$$\dot{Q} = \dot{m}_2 C_p T_1 (T_2 / T_1 - 1) \quad 3.24$$

Therefore
$$\frac{T_2}{T_1} = \frac{\dot{Q}}{\dot{m}_2 C_p T_1} + 1$$

From 3.23
$$\frac{\dot{m}_1}{\dot{m}_2} = \frac{T_2}{T_1}$$

Therefore
$$\frac{\dot{m}_1}{\dot{m}_2} = \frac{\dot{Q}}{\dot{m}_2 C_p T_1} + 1 \quad 3.25$$

Rearranging gives;

$$\frac{\dot{m}_2}{\dot{m}_1} = 1 - \frac{\dot{Q}}{\dot{m}_1 C_p T_1} \quad 3.26$$

Using the data produced by the simulation program described in Chapter 7 to give some indication of the percentage reduction in mass flow and by assuming a representative motor efficiency of 80% and taking $C_p = 502 \text{ J/kg/K}$ and $T_1 = 283\text{K}$ gives;

$$\frac{\dot{m}_2}{\dot{m}_1} = 1 - \frac{2000}{0.143 \times 502 \times 283} = 0.9$$

Therefore the use of suction line cooling in this case indicates 10% reduction in mass flow rate. Because the mass flow rate appears in both the numerator and denominator of the coefficient of performance parameter its value would be unaffected though the refrigeration effect would reduce by a

similar percentage as the mass flow rate. This would not occur with discharge gas cooling.

A further method of internal motor cooling of hermetic compressor motors is to spray quantities of liquid refrigerant into the low pressure motor casing where the latent heating of the refrigerant is used to cool the motor. The coolant may be then drawn into the compressor along with the normal suction gas. The study showed that a disadvantage of this method was that it reduces the refrigeration effect of an evaporator and in similar proportions increases the compressor work. Although these effects tend to reduce the C.O.P. of the system, the reduction is thought to be less than for conventional suction or discharge gas cooling because less refrigerant would be required to provide the same cooling due to a significant increase in convective heat transfer coefficient anticipated from a boiling/evaporation process and the latent heat capacity being far greater than the specific heat. An additional benefit of this approach is that cooling can be supplied only when required which may be an advantage under part-load and compressor cycling conditions

3.13 Suction Line Cooling

The study indicated that suction gas cooling would be technically less complicated to install than liquid injection. Also the experimental facility used in testing the compressor consisted of a 'vapour loop' system and with the absence of condensate, liquid injection was not considered a feasible option for the prototype unit.

To encourage the coolant to flow across the windings and remove the heat at source it was decided that cooling passages should be fitted through the stator. Figure 3.8 shows a schematic view of the gas flow through the motor casing. Refrigerant gas would enter the motor casing via pipes through four equally spaced drillings at the rear of the motor casing, 'a', where it would be distributed at the rear of the casing. The gas would then flow across the motor windings, 'b', picking up heat along the way before entering cooling passages at 'c'. Some gas would flow through the clearance between the rotor and stator, 'd'. The ratio of flow between these two pathways was estimated to be 3:1 in favour of the cooling passages. No account was taken of the rotation of the gas between the stator and laminated rotor.

Having flowed through the stator passages the gas would enter the annular space at the forward end of the motor casing, 'e', where it would flow over the motor windings before leaving the casing through four equally spaced ports, 'f', leading to the impeller inlet.

3.14 Cooling Passage Specification

Cooling passages were fitted through the stator as shown in Figure 3.8. In order to provide efficient cooling these passages were positioned as close as possible to the stator windings. A constraint in positioning was the avoidance of excessive eddy current heating produced by the passages themselves particularly at corners and free edges. This may have offset any benefit produced by reducing the thermal conduction path between the passages and the windings. From discussions with Hill⁵⁴, it was decided that the cooling passages should be positioned on a radius no smaller than 48 mm, leaving sufficient space between the passages and winding channels as shown in Figure 3.9.

A study described in Appendix H showed that for high rates of heat transfer the aspect ratio of the passages needed to be large. This is because, for a constant flow rate, Nusselt number and heat transfer area both increase with aspect ratio. However, pressure losses also increase with aspect ratio which placed a constraint on the design. Equation 3.28, 3.29, and 3.30 derived in Appendix H and listed below, allow the heat transfer, pressure loss and pumping power to be calculated for changes in aspect ratio, (λ), passage width, (l), and length, (L), and the number of passages, (N), for a given mass flow, ($\dot{m}$).

$$\Delta P = \frac{1.215 \times 10^{-3} [1+\lambda]^{1.2} \cdot L}{l^{4.8} \lambda^3} \left[\frac{\dot{m}}{N} \right]^{1.8} \quad 3.28$$

$$\dot{Q} = \dot{m} C_p [T_w - T_1] \left\{ 1 - \exp \left[\frac{-(1+\lambda)^{1.2}}{199 l^{0.8} \lambda} \cdot L \cdot \left[\frac{N}{\dot{m}} \right]^{0.2} \right] \right\} \quad 3.29$$

$$W_p = \frac{\dot{m}^{2.8} (1+\lambda)^{1.2} L}{2963 \lambda^{4.8} N^{1.8}} \quad 3.30$$

Figures 3.10, 3.11 and 3.12 show the variation in heat transfer, (Q), and pumping power, (W_p), with the number of cooling passages, (N), passage aspect ratio, (λ), and width, (l). The stator length, (L), used in the calculations was equal to the finished dimension and the stator winding temperature, (T_w), was set at 90°C. The refrigerant gas inlet temperature was assumed to be 10°C, which was 5 to 10 degrees higher than would be found in practice. Refrigerant flow rate, ($\dot{m}$), and compressor power were determined using the compressor performance characteristic at the design speed of 70,000 rpm, taken from the results of a simulation model described in Chapter 7. The anticipated heat rejection rate at the design condition was estimated by assuming an 80% electrical conversion efficiency at the design point, which was thought to be a low side value but would allow for a margin of error in the calculations. In order to ensure the thermal performance of the passages convective cooling from the exterior surfaces of the stator and windings was neglected.

DESIGN DATA

$\dot{m} = 0.1433 \text{ kg/s}$	$L = 0.076 \text{ m}$
$T_w = 90^\circ\text{C}$	$T_1 = 10^\circ\text{C}$
$C_p = 502 \text{ J/kg/K}$	$Q = 1.77 \text{ kW}$

The required design point cooling rate of 1.77 kW was plotted in Figure 3.10, 3.11 and 3.12. Figure 3.10 shows that for a passage width of 6mm, the aspect ratio of the passages

would need to be 5 for 20 cooling passages decreasing to 3.7 for 50 passages. This trend suggested that fewer holes with high aspect ratios would be better than many holes with small aspect ratios. In addition the fewer holes would have reduced manufacturing problem. An examination of the pumping power characteristic in Figure 3.10 indicated that the pumping power for 20 holes would have been 2.5 kW whilst for 50 holes the power consumption would have been 350W, for the same heat transfer. Clearly a compromise between pumping power and the number and dimensions of the cooling passages was required.

The manufacture of the cooling passages provided a further constraint. It was decided that the cooling passages should be punched through the laminations. A 1mm x 10mm passage was initially envisaged. Calculations showed that the stress in a punch to produce these holes would be in the order of 570 MPa, which is less than half the yield stress of most common tool steels. The critical buckling length of the punch was also calculated using the previous dimensions and found to be 34 mm at the required load. The manufacturers were concerned about the restriction in tool design resulting from this length of punch and suggested a minimum punch thickness of 1.5 mm. As the cooling passage height in itself was not critical it was decided to fix this dimension at 1.5 mm, which reduced the design problem to two variables; number and width. Referring to Figures 3.10, 3.11 and 3.12 it is seen that the solution lay on a vertical line constructed at the aspect ratios equating to a passage height of 1.5 mm, (ie 4, 5.3 and 6.7 respectively).

From Figure 3.10 it was seen that by increasing the number of passages from 20 to 50 would increase the cooling rate from 1.5 to 1.8 kW. The latter solution was preferable because the pumping power was much lower and the cooling rate greater than the total required. In Figure 3.11 the cooling rate was shown to increase to 1.6 Kw for 20 passages whilst for 50 the cooling rate would have been 1.9 kW. The pumping power in for 20 passages is more acceptable at 600W.

After further consideration it was decided to specify forty cooling passages with a width of 8 mm and a height of 1.5 mm. Referring to Figure 3.11 it is seen that this cooling passage geometry would provide 1.85 kW of cooling with a pumping power of 250W. To fit the cooling passages into the space available it was decided to position 20 passages equally spaced on a 96 mm PCD and a further 20 passages on a

105 mm PCD. The layout of the laminations is shown in Figure 3.9.

3.15 Off Design Performance of Motor Cooling System

It was considered that cooling efficiency would fall under part-load conditions, particularly at higher power settings. In other words it was possible that an increase in heat rejection caused by increasing the power demand on the motor may not have been matched by an increase in the performance of the cooling system. Assuming the refrigerant to be a real gas, compressor power is given by the following equation:

$$\text{Compressor Power} = \frac{\dot{m} Z C_p T_1}{\eta_c} \left(P_r^{\frac{\gamma-1}{\gamma}} - 1 \right) \text{ watts}$$

Assuming a constant pressure ratio of 4, an inlet temperature of 283 K, compressibility factor of 1.12, specific heat value of 502 J/kg/K, specific heat ratio of 1.137 and worst case isentropic efficiency of 52% then compressor power is a linear function of mass flow.

$$\text{Compressor Power} = 55630 \dot{m} \text{ Watts}$$

Assuming a worst case motor efficiency of 80% then the heat rejection rate from the motor is given by:

$$\text{Heat Rejection Rate} = 13900 \dot{m} \text{ Watts}$$

This value was thought to over estimate the required cooling performance over the operating range of the motor because of the low values of motor efficiency and isentropic efficiency.

The above relationship is plotted on to Figure 3.13 in which the 'worst case' heat rejection by the motor may be compared with the potential performance of the cooling passages. This simplified analysis suggested that the heat rejected by the motor would have exceed the cooling system performance. The

difference is not thought to be significant and because a large amount of cooling would have occurred at the windings at each end of the stator pack, (probably at least half), the effect would have been to reduce the available heat to be collected by the refrigerant flowing through the stator cooling passages. It was concluded therefore that the design of the cooling passages would prove satisfactory. A photograph showing the motor cooling passages is included in Figure 3.14.

3.16 Conclusions Resulting from the Motor Design Study

Two critical areas in the design of the motor were identified; motor cooling and laminated rotor stressing. This chapter has been concerned with both these aspects of the project.

The mechanical integrity of the laminated rotor was investigated. A plane strain model, ($\epsilon_z = 0$), was thought to be inappropriate for determining the stresses in a laminated rotor because the laminations can not support axial tensile loads. A plane stress model, ($\sigma_z = 0$), was considered to be more appropriate. The optimum rotor dimensions were calculated and found to be; $D_o = 64.2$ mm, $D_i = 36.8$ mm and $L = 98.3$ mm. These were later altered because of the stress concentration resulting from the circular holes cut out of the laminations to enable the fitment of the conductor rods.

For design purposes a 5% rotor over-speed condition was assumed at the design point which would have allowed the laminated rotor to operate safely without plastic deformation to 73,500 rpm. The rotor bursting speed was estimated in to be 89,870 rpm.

The presence of conductor rods in the laminated assembly were thought to increase the maximum dynamic hoop-stress at the inner diameter of the rotor. A two-dimensional stress model was devised which provided a better estimate of the operating stresses than that provided by the plane stress model. The rotor dimensions were re-calculated using this two-dimensional model. The results were $D_o = 58.7$ mm, $D_i = 31.75$ mm and $L = 88.05$ mm. It was later shown from electro-magnetic considerations that the length of the laminated rotor could be reduced to 77 mm. The outer diameter was increased to 59 mm and the inner diameter altered to 32 mm for manufacture.

The axial pre-load during the assembly of the rotor was necessary to prevent the rotor laminations from becoming displaced during normal operation. Analysis showed that the necessary pre-load could not be provided by the conductor rods alone. To ensure adequate axial pre-load it was necessary for the laminated rotor to be compressed by a load of 211 kN, (21tons), whilst the conductor rods were fitted through the shorting plates and riveted. When the laminated rotor was fitted to the drive shaft additional pre-load was provided by a special nut, shown schematically in Figure 3.4, which was be torque loaded to 386 N-m.

It was estimated that the performance of the cooling system would need to be in the order of eight time more effective than that of conventional 50 Hz machines. Although the maximum winding temperature for class A insulation was 160°C, because of the possibility of chemical breakdown of the refrigerant, (Freon 11), it was decided to limit the winding temperature to 90°C, David²².

The study showed that combined natural convective and radiant cooling would not be sufficiently effective. Analysis showed that the maximum core temperature would have been 800°C had this method been adopted. Forced convection was shown to reduce the stator core temperature of the motor to 500°C. With the addition of fins to the motor casing it was shown that this would have reduced to 215°C. In this case the pressure losses in the cooling air flow, (ideal fan power), were found to be in the order of 700 watts which was thought to be excessive. It was concluded that some form of direct cooling would be required.

Three internal, (direct), cooling methods were compared. The use of compressor discharge gas was thought to be unsuitable because the temperature of this gas would have provided a low temperature potential, (20°C), compared with suction gas cooling, (77°C). The main disadvantage to suction gas cooling is that the mass flow through the compressor falls as the quantity of heat rejected by the motor increases due to a fall in vapour density. It was estimated that the mass flow would fall by approximately 10% because of suction gas heating. Direct injection of liquid condensate was investigated. This method compares favourably with suction gas cooling and it was concluded that because of the 'vapour loop type' experimental apparatus used during test runs this method would be unsuitable for the prototype compressor.

To improve the performance of the motor cooling system forty

cooling passages were specified. The cooling passages are shown in Figure 3.14. The part-load performance of the cooling passages was investigated. It was concluded that the motor cooling system would operate satisfactorily over the normal operating range. This was confirmed by experiments described in Chapter 6.

The study described in this chapter was necessary in order that some degree of confidence could be placed on the mechanical and thermal integrity of the novel electric motor. The study set the physical dimensions of the laminated rotor from electro-magnetic and mechanical stress considerations and provided an insight into the problems of motor cooling.

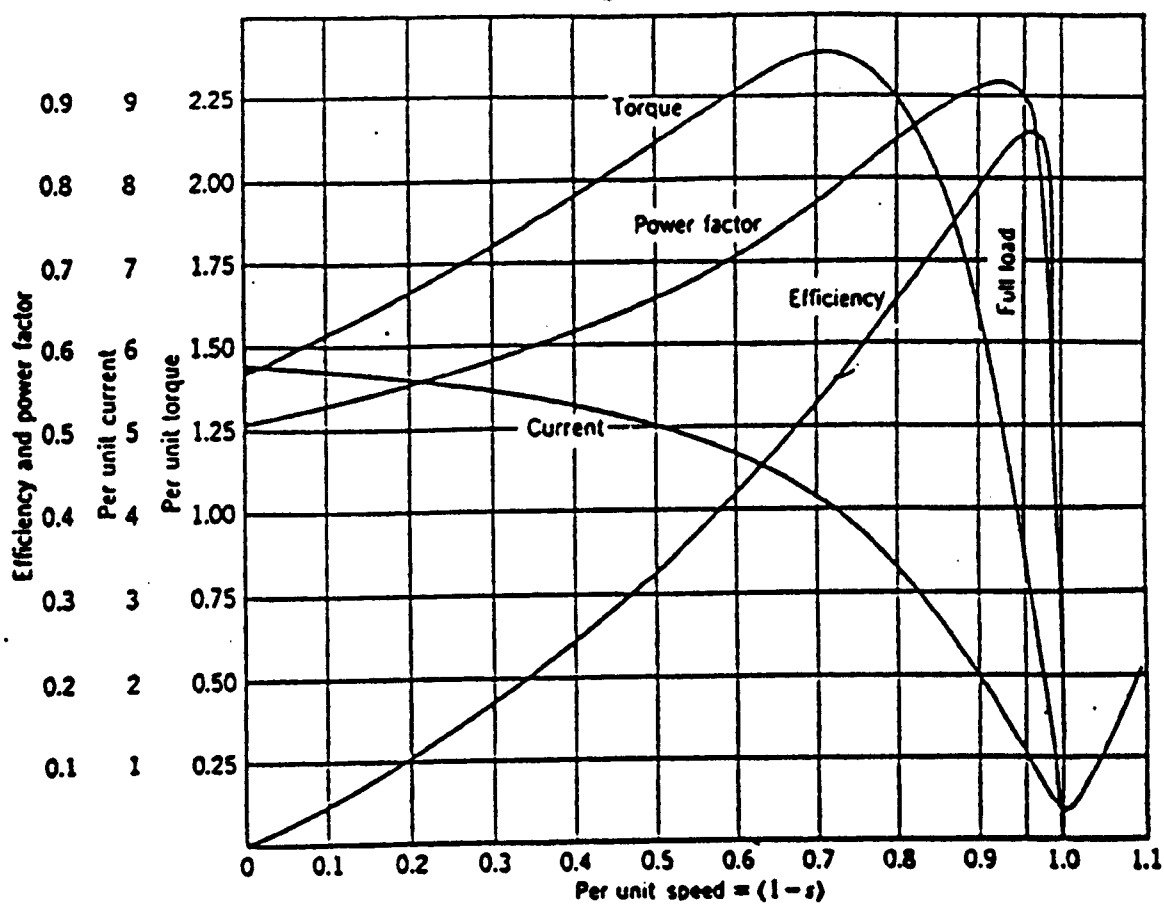


FIG. 3.1 PERFORMANCE CURVES OF A TYPICAL POLYPHASE INDUCTION MOTOR

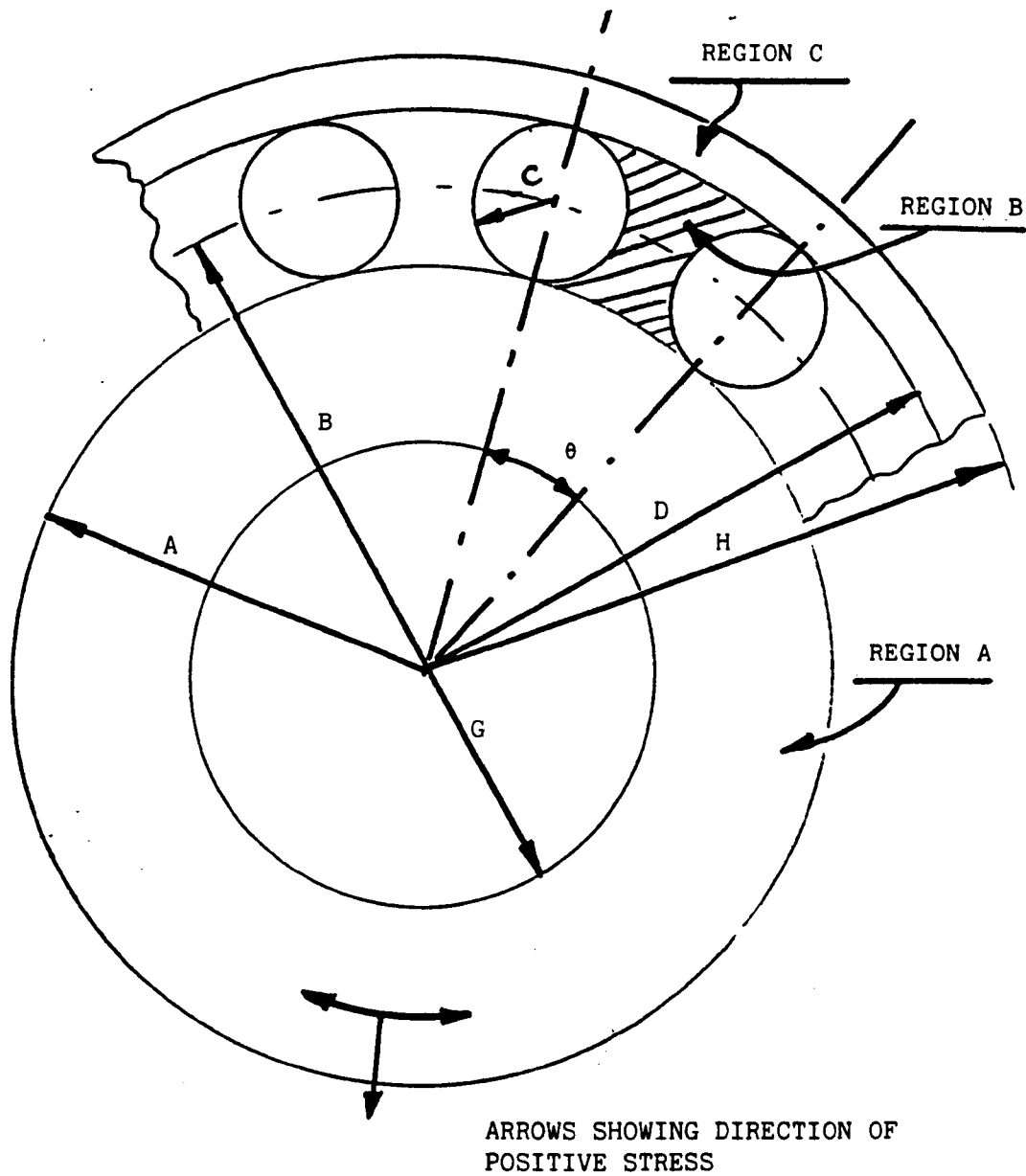


FIG. 3.2 REGIONS OF MODEL AND NOMENCLATURE USED

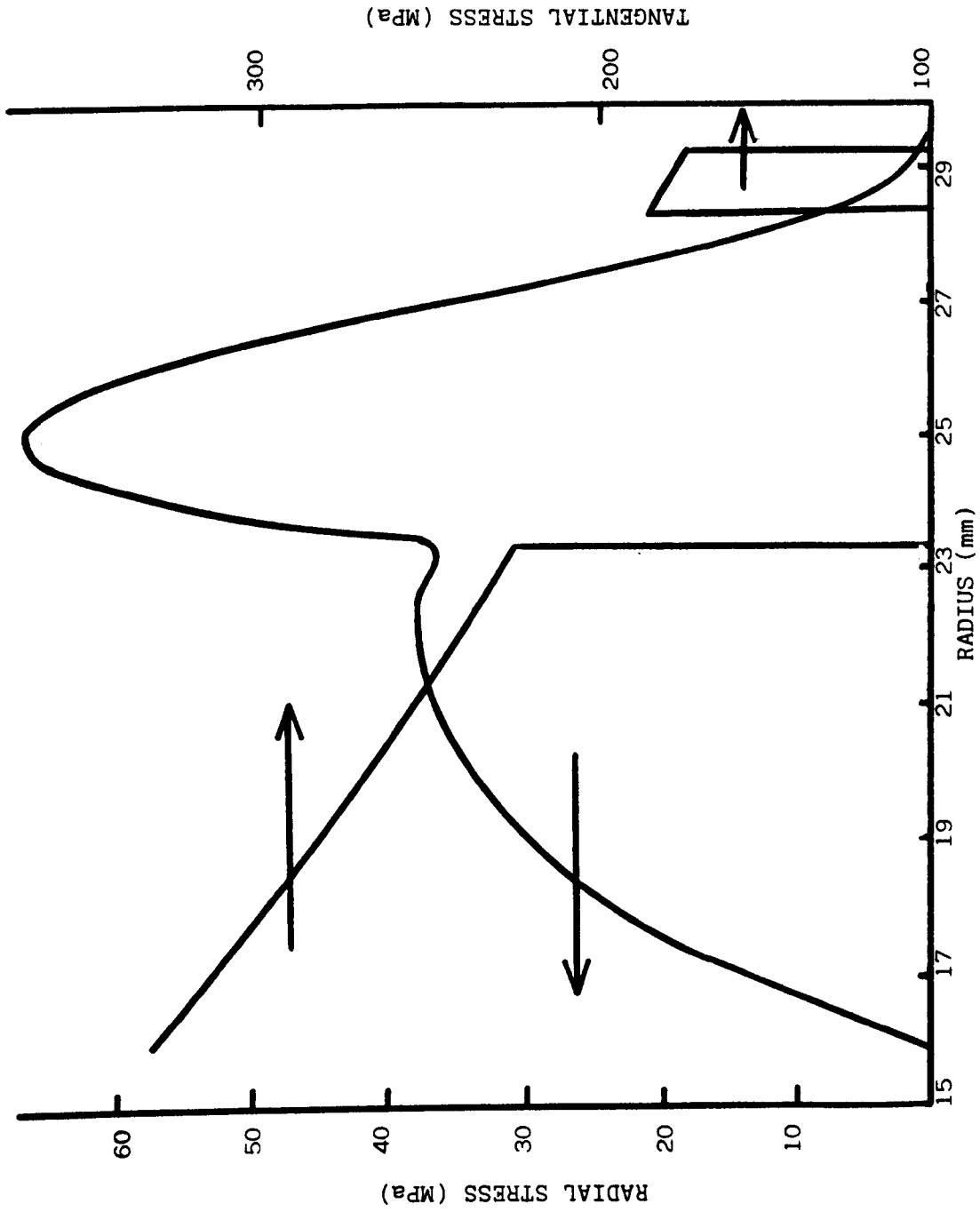


FIG. 3.3 STRESS DISTRIBUTION IN MOTOR LAMINATION ROTATING AT 70×10^3 RPM

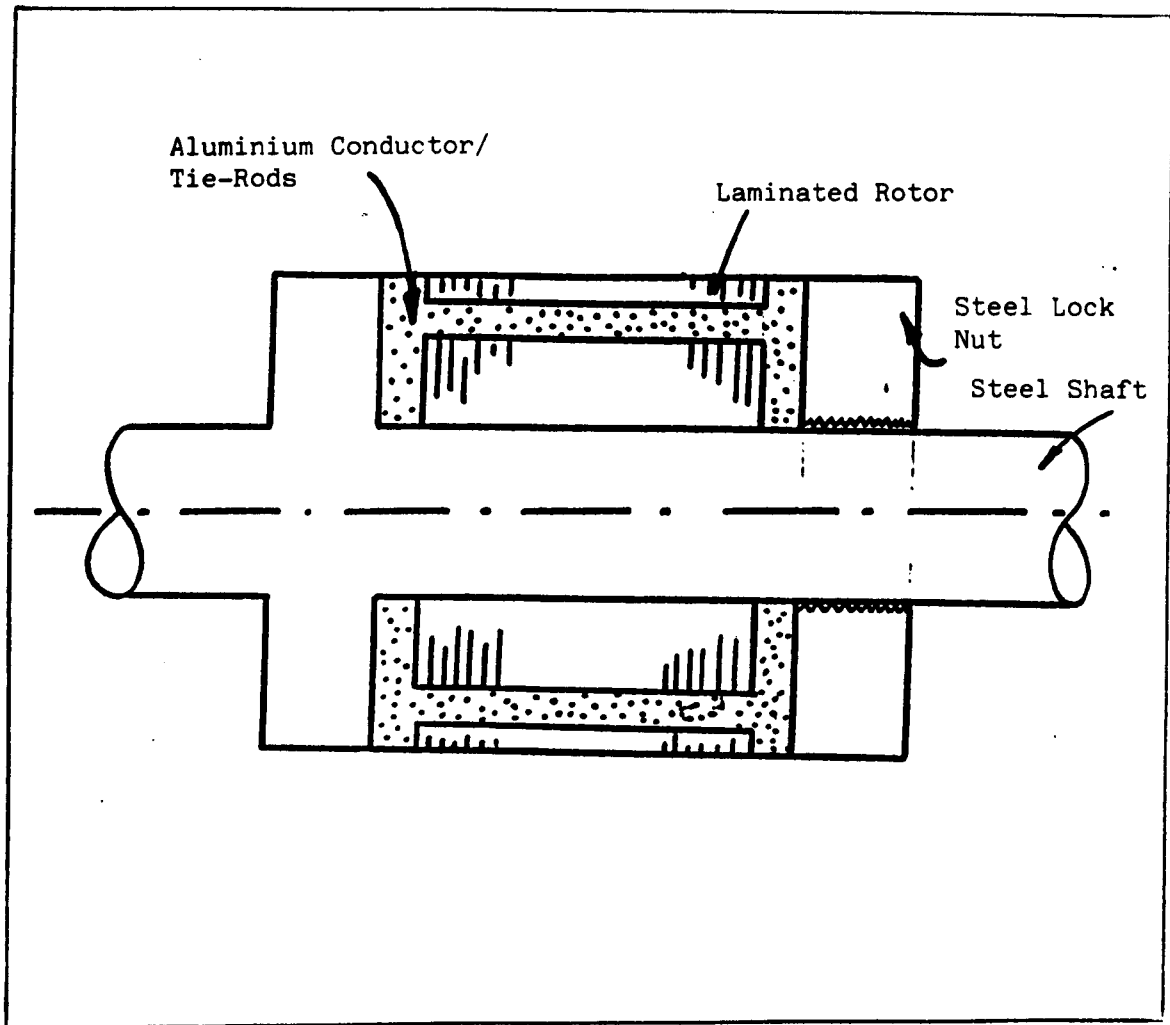
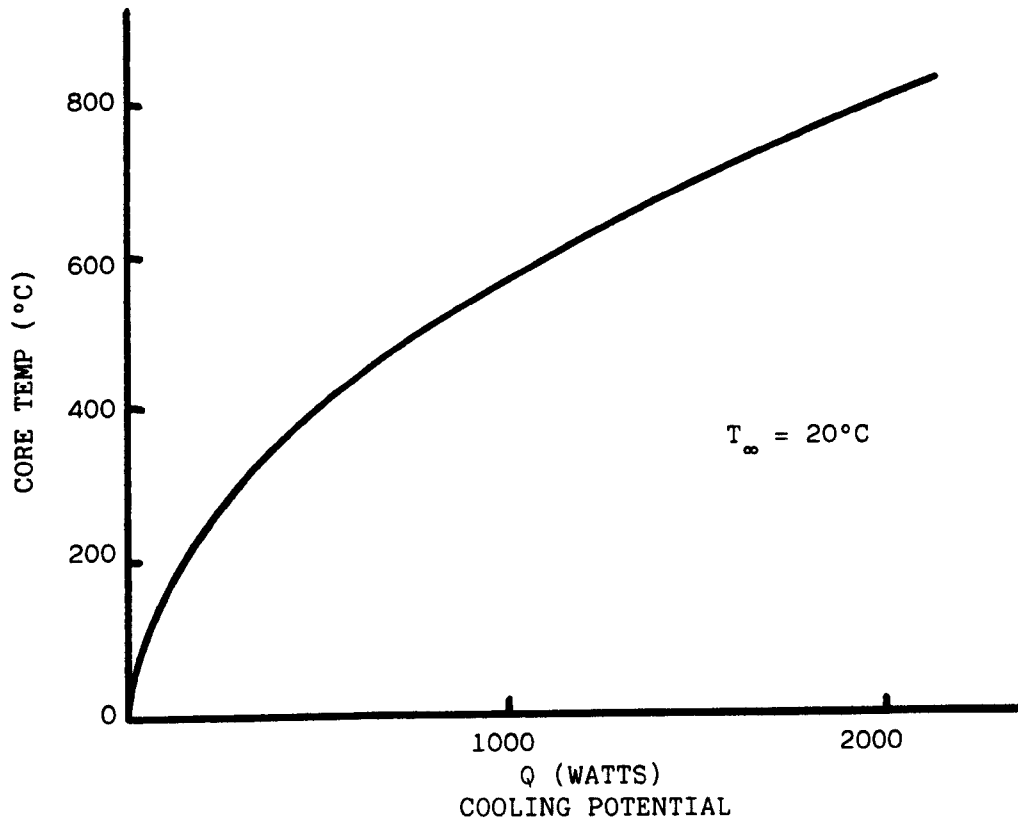
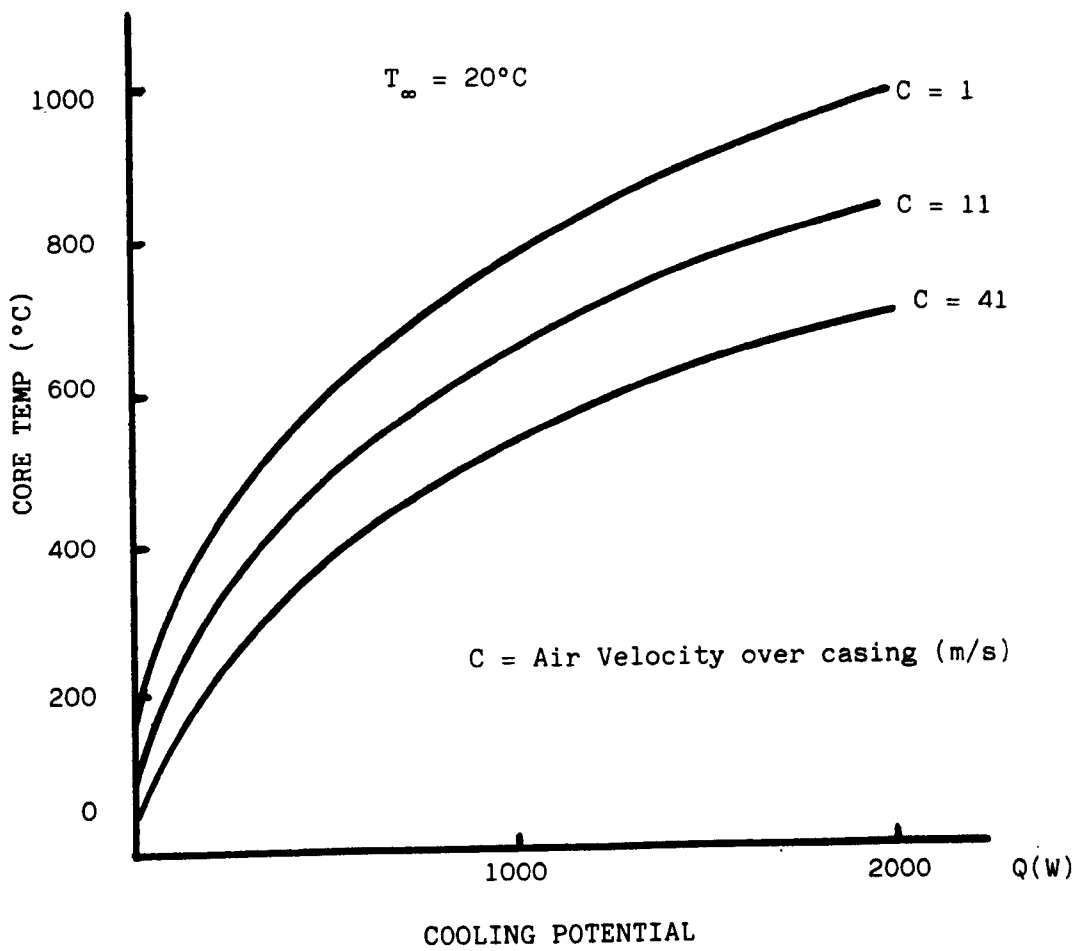
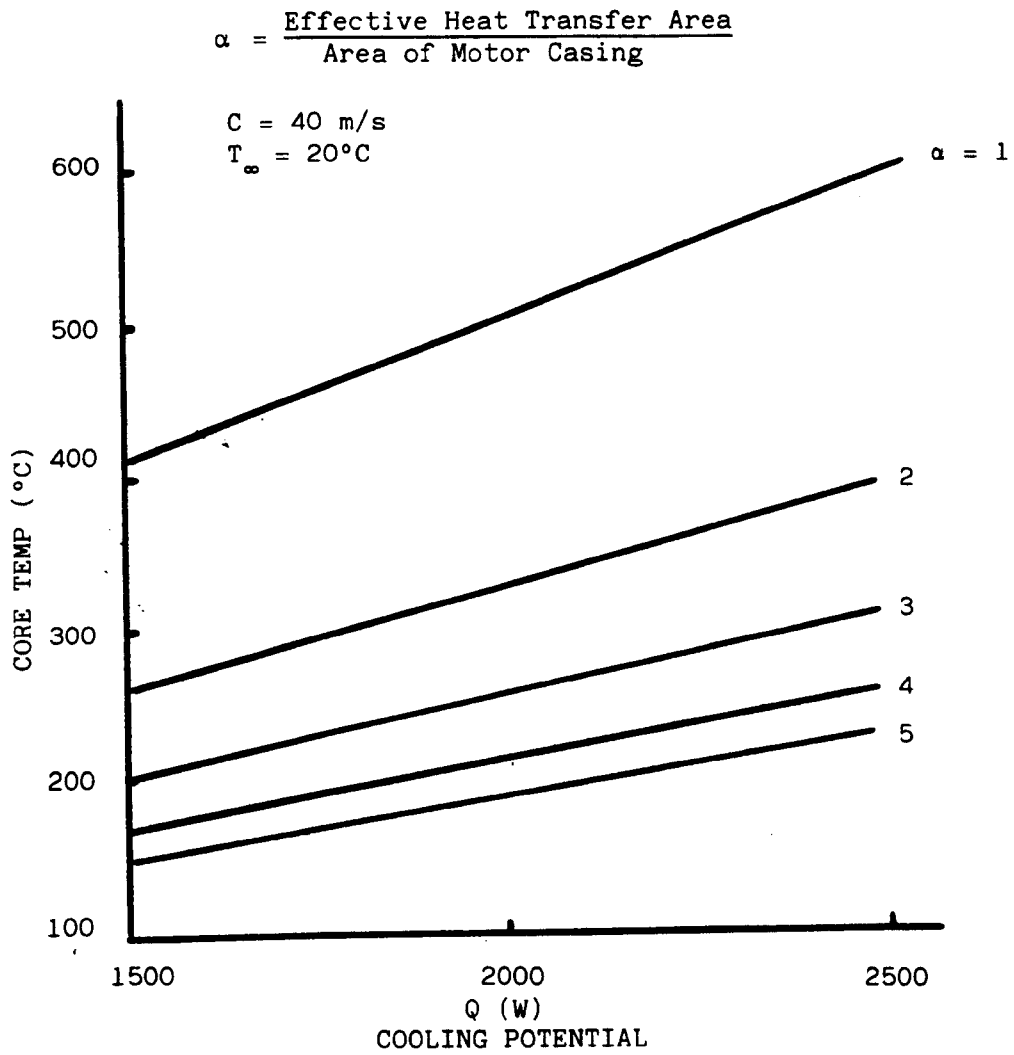


FIG. 3.4 SCHEMATIC REPRESENTATION OF A PROPOSED ROTOR/SHAFT ASSEMBLY

**FIG. 3.5 NATURAL CONVECTIVE COOLING**

**FIG. 3.6 FORCED CONVECTION**

**FIG. 3.7 FORCED CONVECTION WITH FINS**

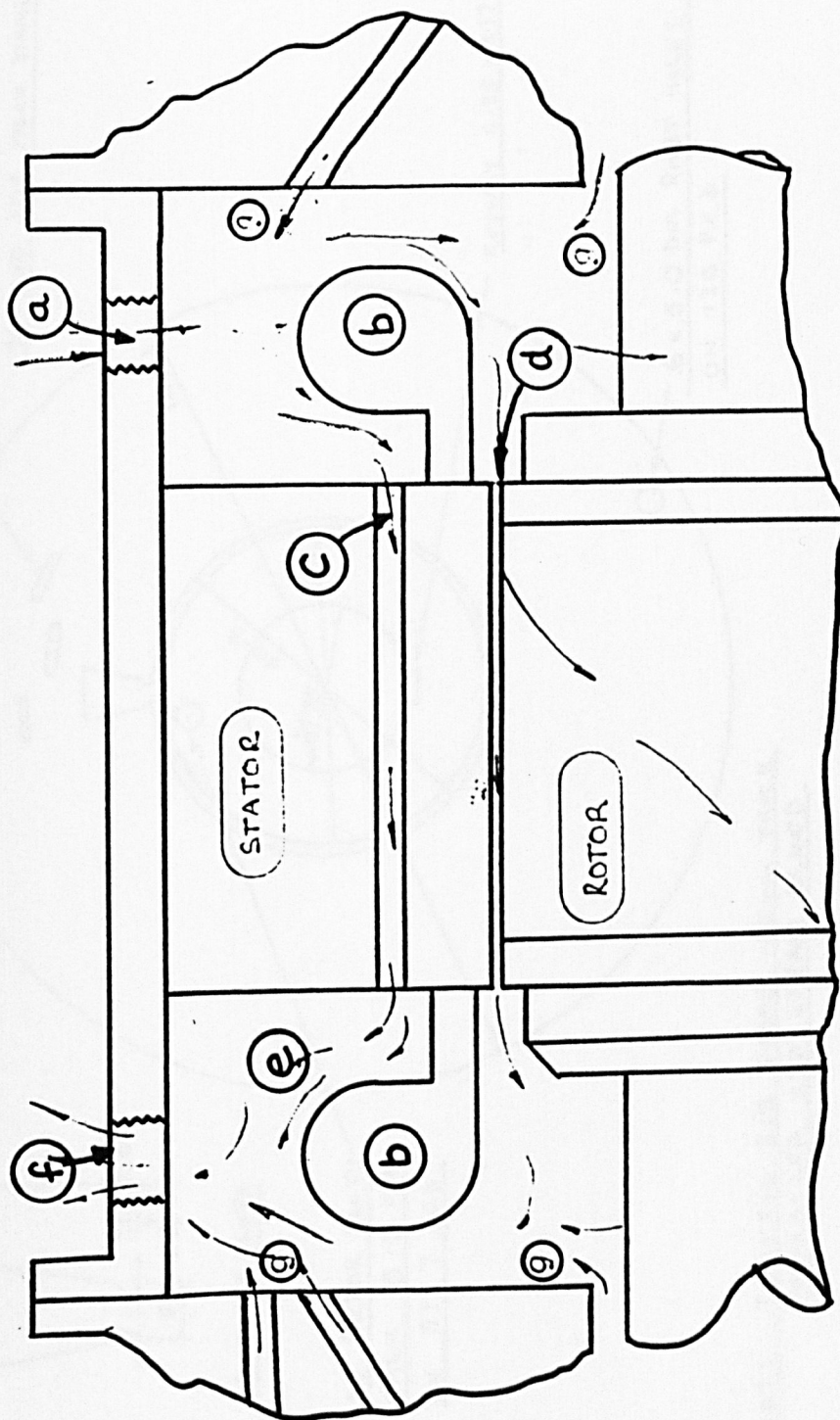


FIG. 3.8 GAS FLOW THROUGH MOTOR CASING

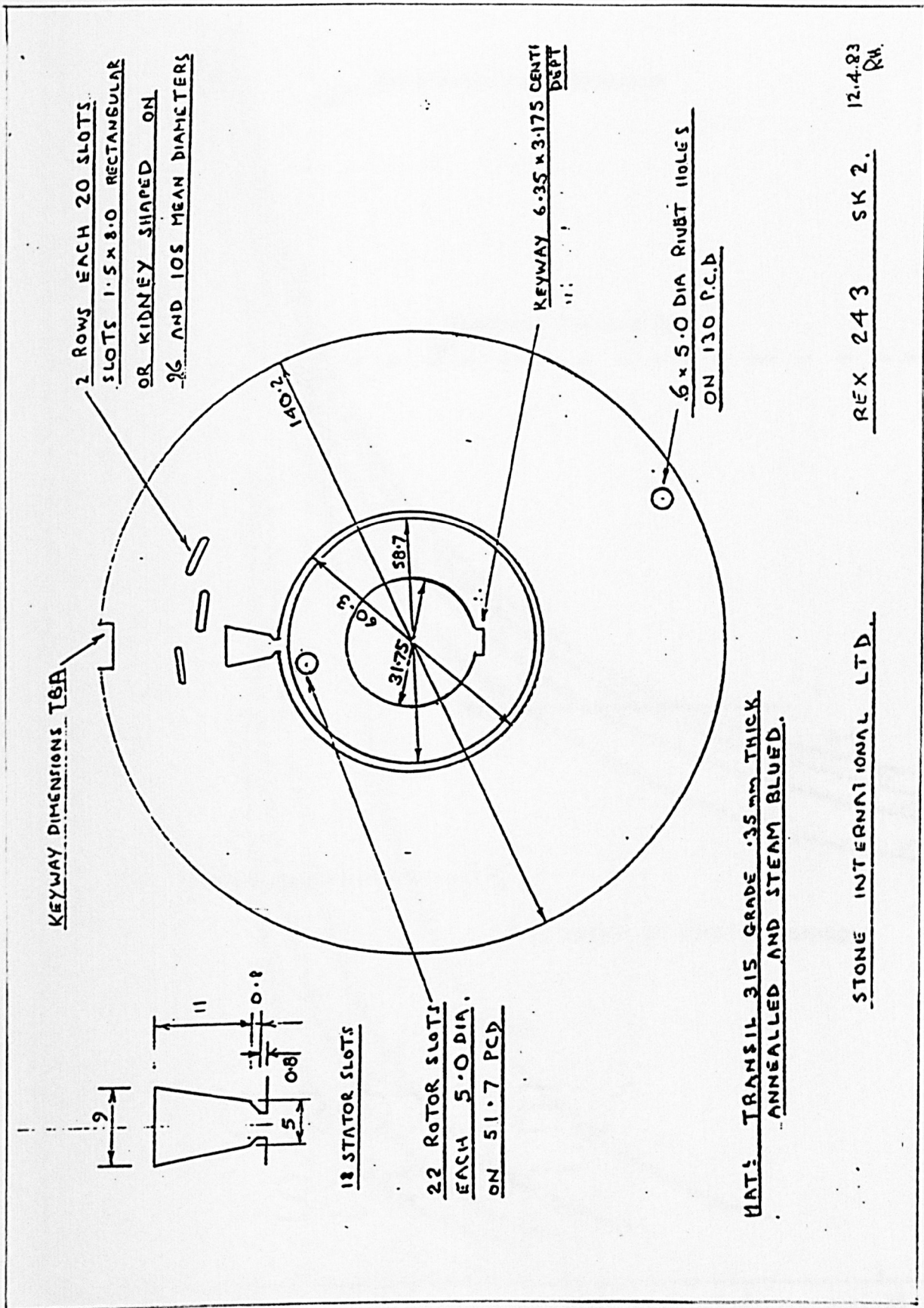


FIG. 3.9 LAYOUT OF MOTOR ROTOR AND STATOR

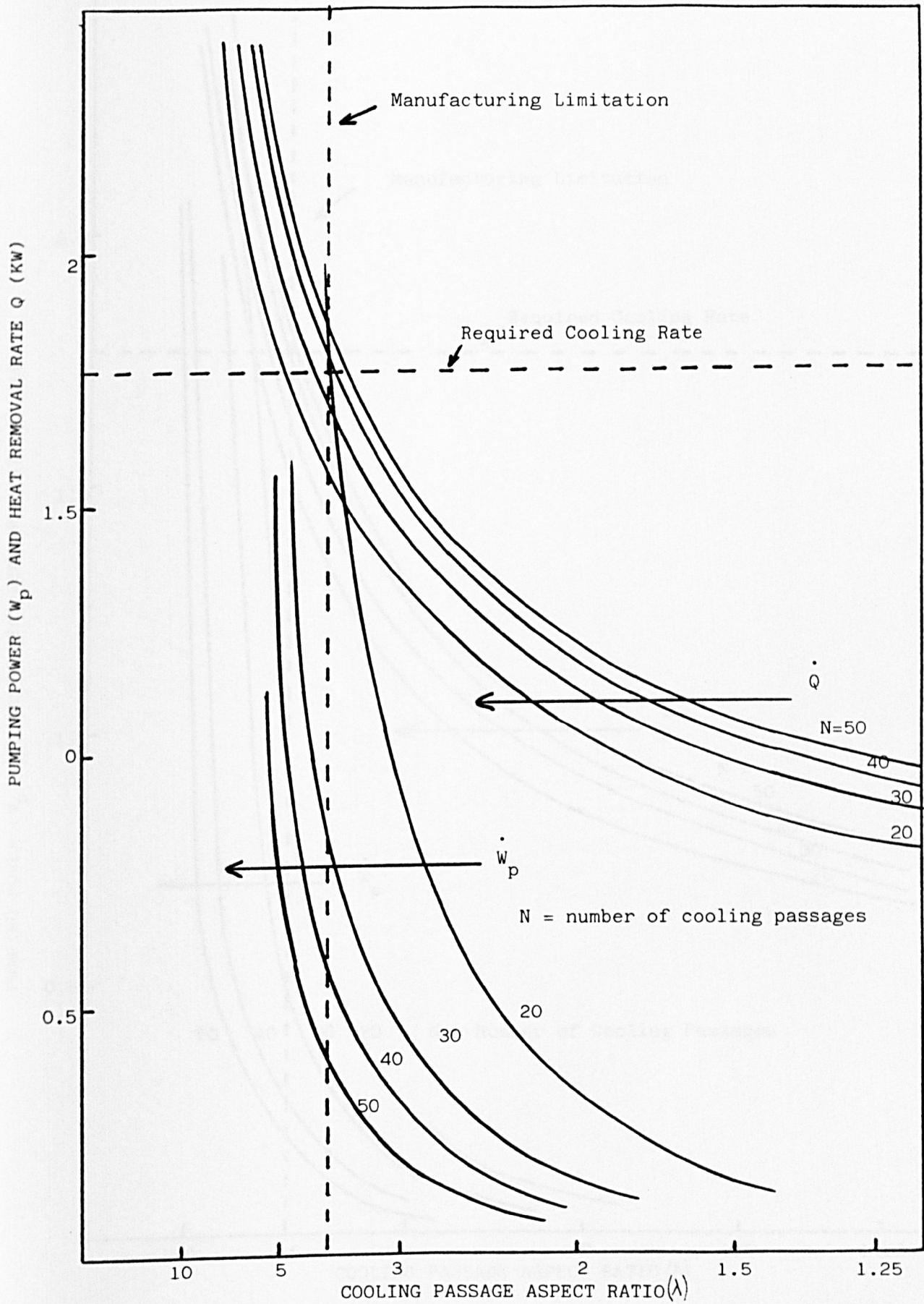


FIG. 3.10 MOTOR COOLING PASSAGE PERFORMANCE $l = 6$ mm

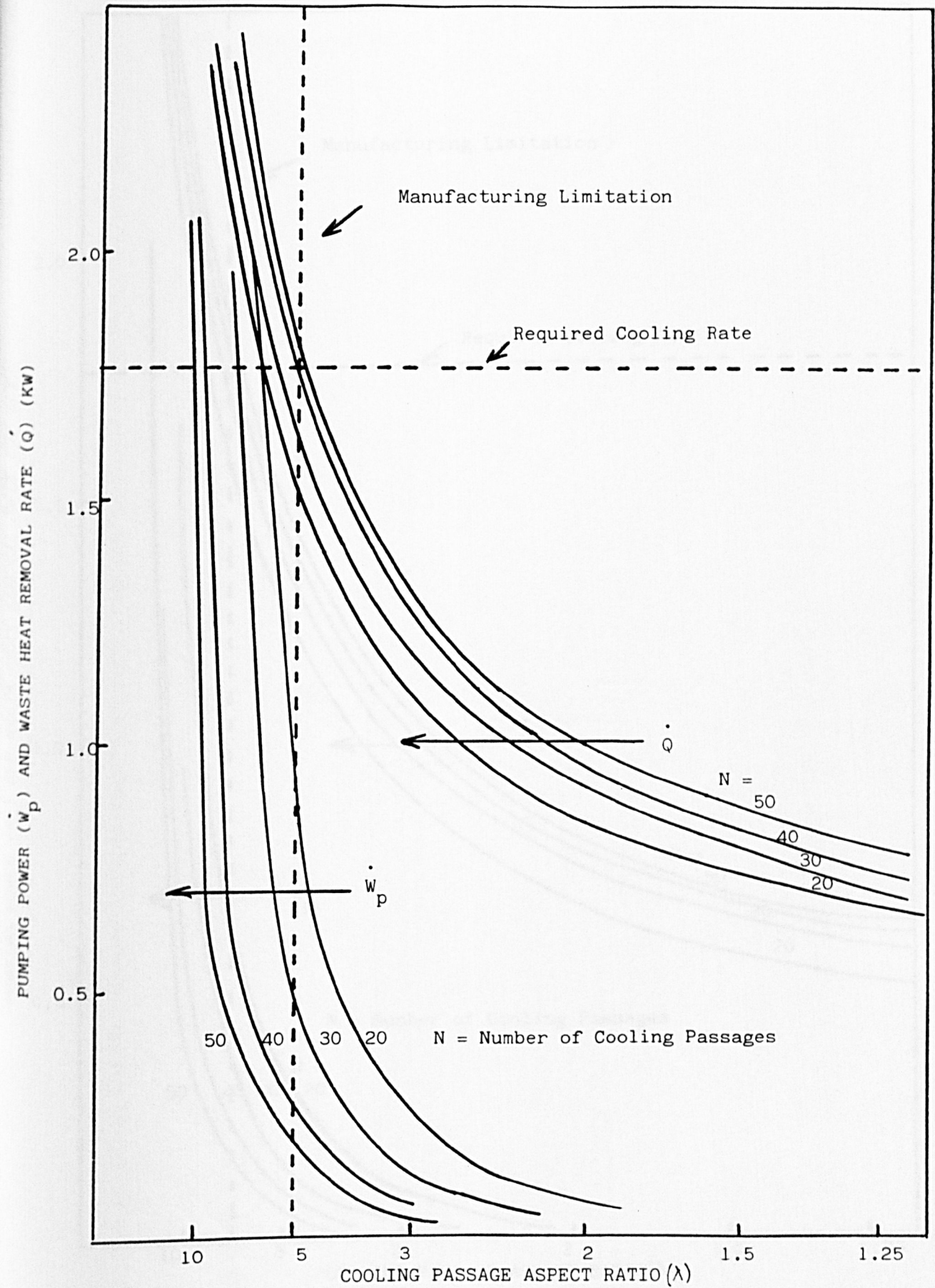


FIG. 3.11 MOTOR COOLING PASSAGE PERFORMANCE, $l = 8$ mm

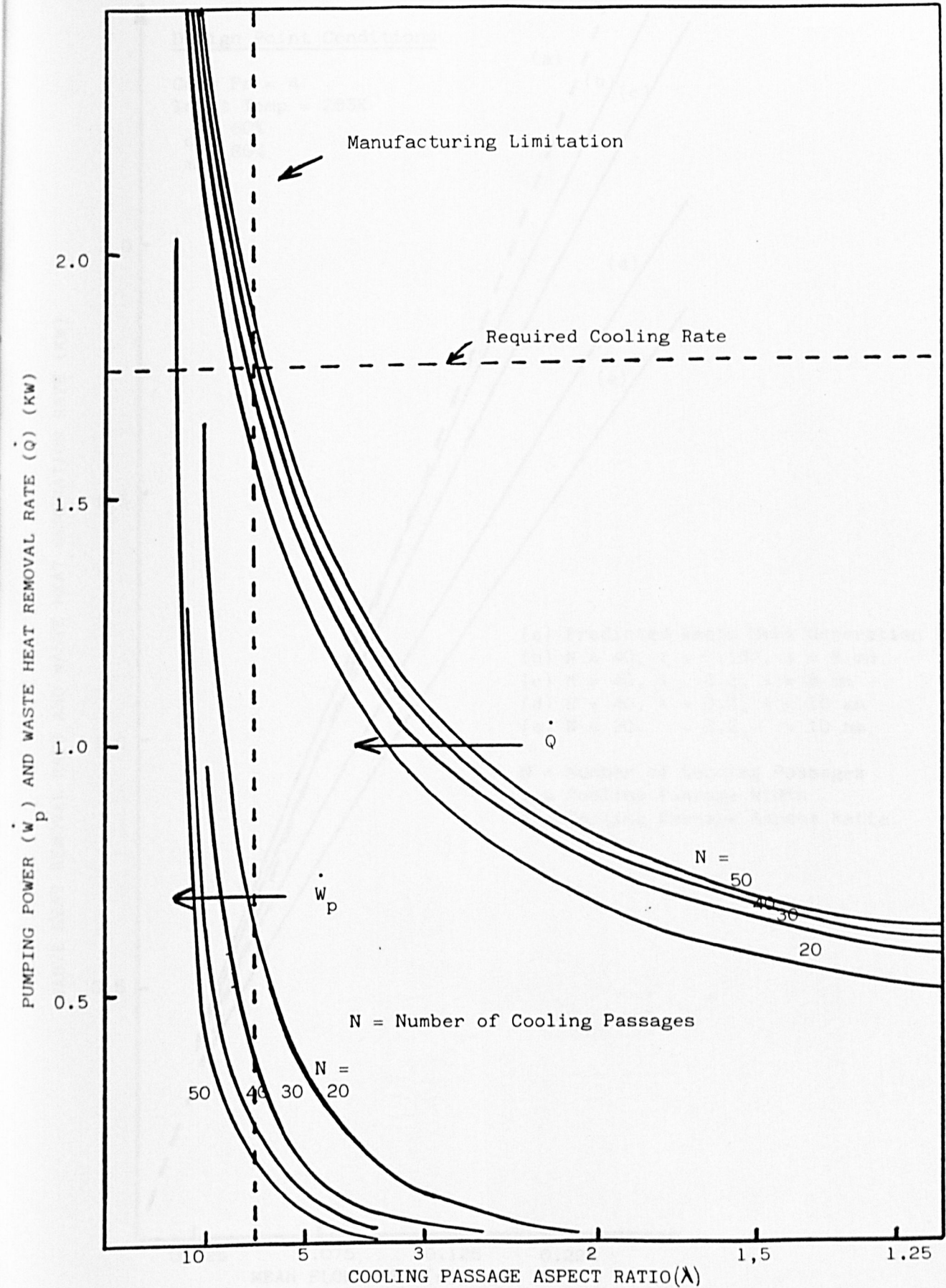


FIG. 3.12 MOTOR COOLING PASSAGE PERFORMANCE, $l = 10$ mm

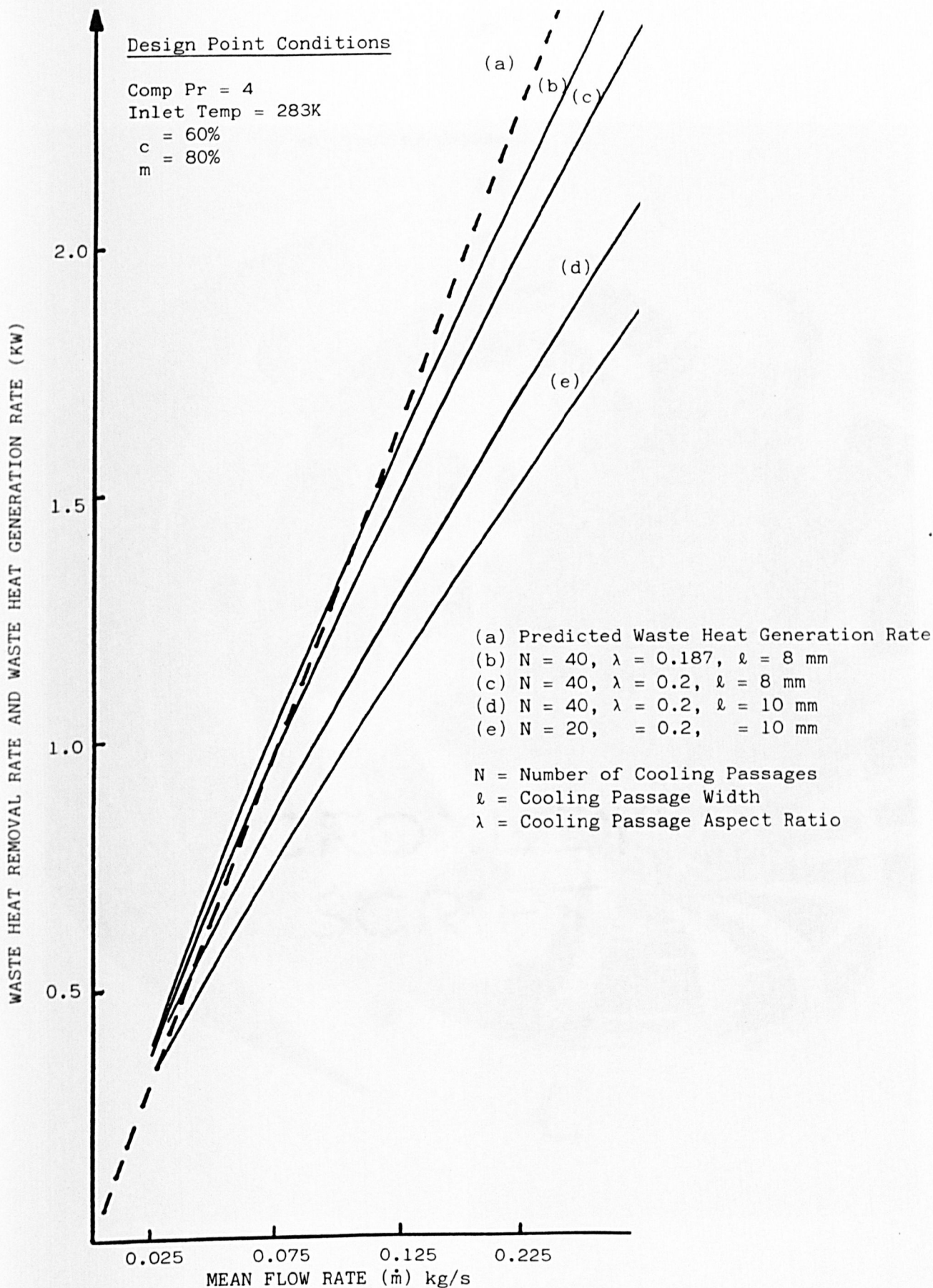


FIG. 3.13 PREDICTED PART LOAD PERFORMANCE OF MOTOR COOLING SYSTEM

40 Cooling passages

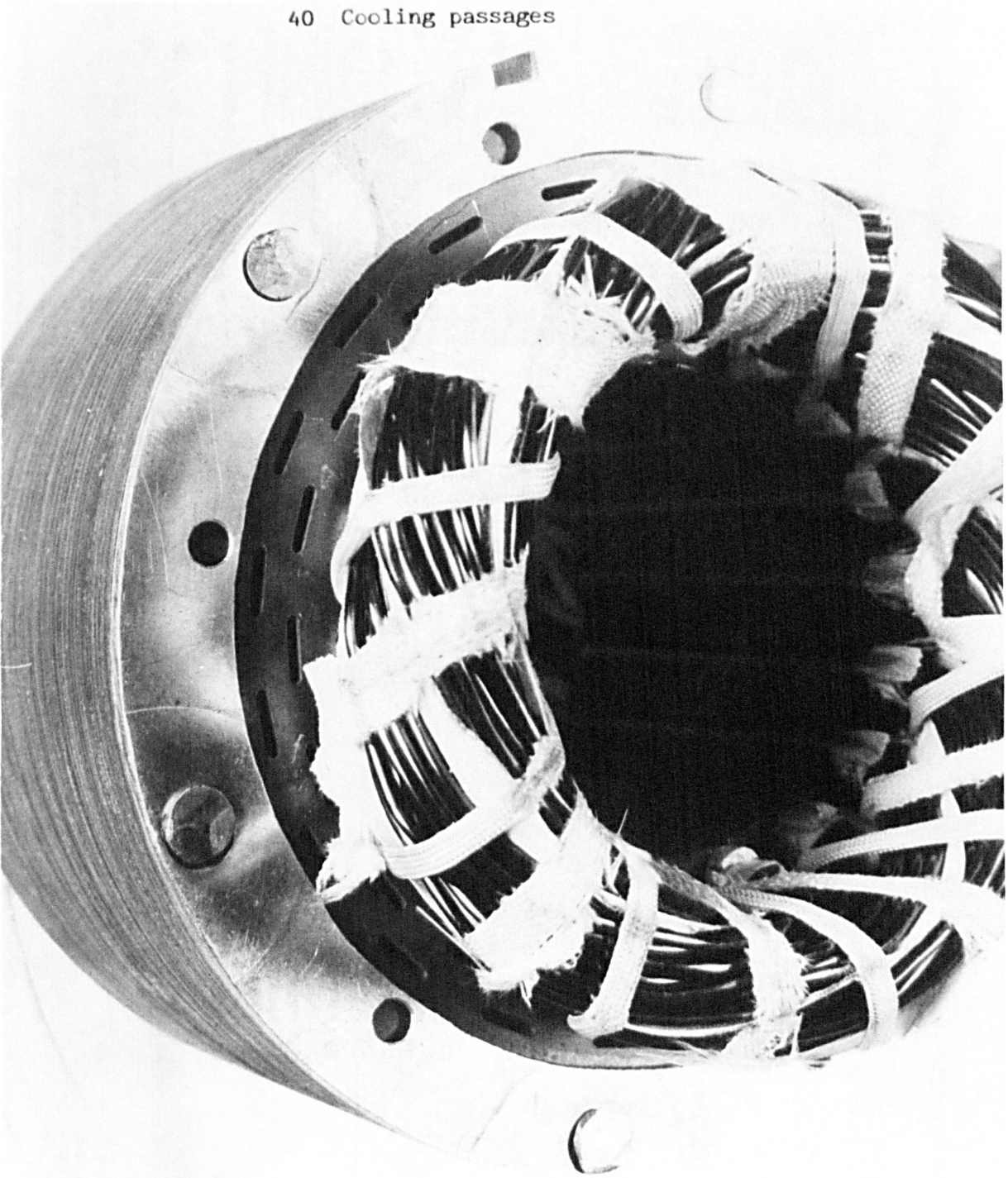


FIG. 3.14 PROTOTYPE INDUCTION MOTOR STATOR

RESULTS

ROTATIONAL SPEED = 73491.3865 RPM

G= 18.4 --- B= 28.5 --- H= 32.1 --- C= 2.5

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
18.4	0	370000000
19.17	12037984	370000000
19.94	21872681	370000000
20.71	29793250	370000000
21.48	36037973	370000000
22.24	40804625	370000000
23.01	44258469	369439041
23.78	46533464	356543698
24.55	47762131	344355447
25.32	48029407	332774355
26.09	47425706	321715005
26.86	47425706	
27.63	63398065	
28.4	70805187	
29.17	73360553	
29.94	70715723	
30.71	6227323	
31.48	52305626	
32.24	39815163	
33.01	27325357	
33.78	15757456	
34.55	6131390	
35.32	6131390	
36.09	5559756	
36.86	4978711	
37.63	4388328	
38.4	3788678	
39.17	3179320	
39.94	2561854	
40.71	1934816	
41.48	1298795	
42.24	652824	
43.01		
43.78		268064365
44.55		266801182
45.32		265541518
46.09		264285303
46.86		263032465
47.63		261782924
48.4		260536641
49.17		259293517
49.94		258053501
50.71		256816521
51.48		
52.24		

Table 3.1 Results of Stress Calculations

RESULTS

ROTATIONAL SPEED = 73491.3865 RPM

G= 17 --- B= 27.5 --- H= 31 --- C= 2.5

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
17	-1	370000000
17.8	12455347	370000000
18.6	22442857	370000000
19.4	30321633	358810761
20.2	36382139	353420074
21	40860343	339234711
21.8	43949729	326061190
22.6	45810151	313739655
23.4	46574615	302137102
24.2	46354512	291142139
25	45243700	2806660707
25.5	45243700	
26	62007013	
26.5	70602061	
27	74320346	
27.5	72396093	
28	65005418	
28.5	53733144	
29	40773421	
29.5	27927921	
30	16181301	
30.5	6474147	
31	6474147	233075194
31.5	5865747	231913569
32	5248503	230754899
32.5	4622431	229599115
33	3987749	228446151
33.5	3344371	227295944
34	2692411	226148427
34.5	2031932	225003539
35	1362997	223861218
35.5	685666	222721401
36	0	221584031

Table 3.2 Results of Stress Calculations

RESULTS

ROTATIONAL SPEED = 73471.3865 RPM

G= 15.875 --- R= 25.85 --- H= 27.35 --- C=

RESULTS

ROTATIONAL SPEED = 69976.741 RPM

G= 15.875 --- R= 25.85 --- H= 29.35 --- C=

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
15.87	0	370000000
16.62	11125961	358091843
17.37	20064375	341670025
18.11	27134953	326786919
18.86	32595350	313184873
19.61	36655075	300654378
20.36	39485875	289022664
21.1	4122715	278150827
21.85	42004592	267917813
22.6	41709394	258223751
23.35	41027416	248991346
23.85	41027416	
24.35	58589914	
24.85	68767991	
25.35	74635614	
25.85	73951806	
26.35	66654030	
26.85	54683521	
27.35	40930753	
27.85	27612957	
28.35	15782217	
28.85	6234795	
29.35	6234795	
29.85	5649823	
30.35	5055115	
30.85	4453738	
31.35	3842761	
31.85	3223252	
32.35	2595277	
32.85	1958901	
33.35	1314108	
33.85	661200	
34.35	0	

Table 3.3 Results of Stress Calculations

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
15.87	-1	341424936
16.62	10092885	324842063
17.37	18201344	309945054
18.11	24615401	296443884
18.86	29568785	284104824
19.61	33251553	272737819
20.36	35819524	262187048
21.1	37401426	252323784
21.85	38104354	243040931
22.6	38017993	234248003
23.35	37217911	225871836
23.85	33149685	
24.35	62564130	
24.85	67705499	
25.35	67085184	
25.85	60465025	
26.35	49608732	
26.85	37130224	
27.35	25049020	
27.85	14316796	
28.35	5655877	
28.85	5655877	
29.35	5125222	
29.85	4586641	
30.35	4040195	
30.85	3485950	
31.35	2923965	
31.85	2334299	
32.35	1777012	
32.85	1192162	
33.35	579806	
33.85	0	

Table 3.4 Results of Stress Calculations

RESULTS

ROTATIONAL SPEED = 77005.5277 RPM

G= 15.875 --- R= 25.85 --- H= 29.35 --- C= 2.5

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
15.87	0	370000000
16.62	12215421	370000000
17.37	22229089	370000000
18.11	27792023	358786077
18.86	35787105	343852111
19.61	40244360	330094623
20.36	4352376	317325023
21.1	45266931	305387513
21.85	46117705	294152474
22.6	46013184	283511361
23.35	45044844	273372717
23.85	4327072	
24.35	75721376	
24.85	81943976	
25.35	81193209	
25.85	73180830	
26.35	60041456	
26.85	44938716	
27.35	30316833	
27.85	17327621	
28.35	6845309	
28.85	6845309	
29.35	6203057	
29.85	5551212	
30.35	4899850	
30.85	4219046	
31.35	3538875	
31.85	2849408	
32.35	2150718	
32.85	1442874	
33.35	725945	
33.85	-1	
34.35		

Table 3.5 Results of Stress Calculations

RESULTS

ROTATIONAL SPEED = 80500.3702 RPM

G= 15.875 --- R= 25.93 --- H= 29.35 --- C= 2.5

RADIUS (MM)	RADIAL STRESS (Pa)	TANGENTIAL STRESS (Pa)
15.87	-1	370000000
16.62	13349425	370000000
17.37	24074133	370000000
18.11	32557727	370000000
18.86	39169350	370000000
19.61	43980396	360738355
20.36	47376940	346783504
21.1	49169253	333727790
21.85	50358906	321159761
22.6	50284762	309836792
23.35	49226527	298750940
23.85	49226527	
24.35	70298797	
24.85	82750878	
25.35	89551145	
25.85	88730681	
26.35	79974483	
26.85	65615321	
27.35	49110547	
27.85	33131260	
28.35	18936210	
28.85	7480785	
29.35	7480785	
29.85	6778911	
30.35	6066552	
30.85	5343794	
31.35	4610717	
31.85	3867402	
32.35	3113930	
32.85	2390377	
33.35	1576822	
33.85	793328	
34.35	0	

Table 3.6 Results of Stress Calculations

CHAPTER FOUR

DRIVE SHAFT AND

BEARING SYSTEM DESIGN

4.1 Introduction

This chapter evaluates the design of the drive shaft and gas bearing system. The first part is concerned with the mechanical integrity of the drive shaft and the latter part deals with the bearing system.

Briefly the drive shaft assembly, in Figures 1.1, 1.2 and 1.4, consisted of a shaft, (item 2), manufactured from EN 24 steel, supported on plain journal bearings, (items 26 and 28). A 'special' nut, (item 15), secured the thrust bearing washer, (item 11), via a spacer, (item 14), to the shaft. A second spacer, (item 20), between the nut and the impeller maintained the tip clearance. A laminated rotor, (item 9), was positioned by a slotted nut and lock washer, (items 3 and 4).

Because of the distance between journal bearing centres and the slenderness of the shaft, it was necessary to have accurate estimates of critical speeds to ensure that they did not coincide with the operating speed of the compressor. A finite element analysis was carried out to provide this based on the Myklestad-Prohl technique.

A reliable bearing design was required to enable testing to proceed without extensive bearing system development. For this reason it was proposed to support the shaft assembly in hybrid journal bearings. Bearing surfaces were treated with a 'Sulfinuz' anti-scuffing agent to reduce wear. When starting the compressor the bushes were pressurized using refrigerant vapour. When the compressor achieved 'take-off' speed of the bearings the gas supply was turned off. A photograph of the bushes is included in Figure 4.1. The design of the journal bearings is described in Sections 4.10 to 4.14.

A double acting hybrid thrust bearing was proposed. This consisted of a plane thrust washer between a pair of plane thrust plates. The bearing was designed to resist the axial loads imposed by the impeller. A photograph of the

dismantled bearing is included in Figure 4.2 and the location of the thrust bearing is shown in Figure 1.1, (item 11). The design of the thrust bearing is described in Sections 4.15 to 4.22.

4.2 Drive Shaft Material

The factors affecting material selection were; strength, cost, compatibility with the refrigerant, (Freon 11), and ease of manufacture. It was concluded that a plain carbon steel would be suitable from a strength standpoint, however, because of the potential problem of corrosion it was decided to use an alloy steel. Of the many types available 708 M40 and 817 M40 were considered to be best suited. Because 817 M40 contains nickel, (1.3-1.7%), it was felt this material would be more resistant corrosion than 708 M40 which contains no nickel.

4.3 Drive Shaft Vibration

The prototype compressor was bolted to a rigid test-bed and therefore external sources of vibration were not included in the analysis. Vibrations may be excited by; mechanical out-of-balance, aerodynamic buffeting and out-of-balance magnetic forces. Because of the symmetry and the aerodynamic smoothness of the shaft assembly aerodynamic buffeting forces were thought to be of secondary importance and were neglected. Rotor misalignment within the stator produces a periodic force emanating from the magnetic field which can excite the natural frequency of the shaft in a similar way to an out-of-balance mass. By maintaining a close tolerance on rotor alignment it was thought that this effect would be negligible. To maintain an acceptable degree of lateral vibration a residual out-of-balance moment of 3.9×10^{-6} N-m was selected, Federn²⁷.

4.4 Critical Speed Analysis

Because of the complexity of the shaft assembly and the importance of accurate estimations of critical speeds a finite element analysis was carried out based on the Myklestad-Prohl, (M-P), technique, Tse, Morse and Hinkle²⁸.

The application of the M-P technique is described in Appendix G. To apply this method the shaft assembly was divided into elements. Associated with each element was a

state matrix defining deflection, inclination, moment and shear force. Each element is related to its neighbours by a transfer matrix which allows mechanical forces to be transmitted between elements. The result is a series of simultaneous equations describing the dynamic behaviour of the shaft assembly. The M-P method is thought to be particularly useful because it allows for the shaft to be flexibly mounted and it takes account of gyroscopic stiffening.

A computer program, LAT-VIB, listed in Table G1, was written to solve the resulting simultaneous equations. The program was validated by comparing analytical solutions to standard problems against results produced by LAT-VIB. This is described in Appendix G along with instructions for the application of the program.

The shaft assembly was divided into 14 regions as shown in Figure 4.4. Each region was subdivided into elements. The total number of elements was increased after each calculation until convergent results were achieved. Material properties and boundary conditions were INPUTed to the program as well as values of journal bearing stiffness. Calculations were repeated for different number of elements and the results compared. These are listed in Table 4.1 which shows that the solutions converge as the number of elements is increased. From Table 4.1 the first five critical speeds are given as; 6,522, 7,860, 87,360, 146,940 and 207,840 rpm respectively.

The lower two frequencies are the rigid body critical speeds associated with the journal bearings. This was proved by separate calculations which assumed a simple spring-mass model of the shaft. The results of these calculations agreed with the M-P results for the first two critical speeds. The stiffness of the bearings was varied and calculations were repeated. It was found that the first two critical speeds decreased with bearing stiffness showing the dependence of these values on bearing characteristics, however, there was little change to the third, fourth and fifth critical speeds which indicated that the bearings system and shaft assembly would be dynamically decoupled. The values of the first two critical speeds, listed in Table 4.1, were estimated by assuming constant bearing stiffnesses. In practice stiffness increases with bearing eccentricity and therefore the results for the first two critical speeds are not reliable. The problem is further complicated by mounting the bushes in elastomer 'O' seals which provide additional damping which

would tend to reduce critical speeds.

For the purpose of modelling the drive assembly it was assumed that the flexural stiffness of the laminated rotor would be similar to that of a solid cylinder of the same length and overall diameter. This assumption was shown to be valid, (Appendix F), if the tensile stress in the rotor produced by whirling was less than the compressive, (preload), stress introduced during assembly. The preload calculation, described in Chapter 3, assumed that the laminations were to remain under compression to 105% of the design speed. Therefore the calculated values of the third to fifth critical speeds are only valid within the design operating speed range of the compressor. Above this speed the critical speeds would fall with the stiffness of the laminated rotor as the preload is removed with increasing speed. This factor was not taken into account because it not proposed to operate the compressor at speeds in excess of 70,000 rpm, the design speed.

Because the drive shaft was expected to accelerate quickly through the first two critical speeds and because vibration amplitude is a time dependent phenomena, they were not considered to be of particular significance. The remaining three critical speeds were outside the normal operating speed of the compressor by a reasonable margin.

4.5 Torsional Vibration

Under certain circumstances torsional vibration originating from the compressor impeller is possible. Air flow from the impeller tip is in the form of a jet which if periodically interrupted by diffuser vanes would impose a periodic twisting moment on the drive shaft against the direction of rotation. The forcing function coming from this would have a frequency equal to the product between rotational speed of the impeller and the number of radial vanes. In the design of the prototype compressor it was proposed not to include a radial vane diffuser and therefore this source of vibration was not considered further.

A further potential source comes from the magnetic force between the stator and rotor. This force produces rotation and is present in all electric motors. In this case the frequency of the magnetic force is equal to twice rotational speed of the motor because of the 2 pole design.

The critical torsional speed of the shaft assembly was calculated. This analysis, described in Appendix G, uses the standard equivalent shaft method, Case⁴¹. The analysis showed that the first critical speed was 1,274 hz. It was of some concern that the predicted frequency was close to the operating speed of the shaft assembly, (1,167 hz), which is approximately 90% of the critical speed. In subsequent designs the geometry of the shaft may be altered to increase the critical torsional speed. The critical torsional speed is inversely proportional to the length of a shaft. By reducing the length of the shaft from 307mm to 245mm it was estimated that the critical speed could be increased to 1600 hz which would be safely outside the operating range of the compressor.

4.6 Drive Shaft Stressing

The predominant stress in the drive shaft during operation was estimated to be the dynamic stress due to rotation. This was found to be three orders of magnitude greater than the stress due to bending, which had an estimated maximum value of 422 kPa, or torsion which had a maximum value of 2.8 MPa at the impeller drive end. Taking into account the complex loading due to bending, torsion and dynamic effects produced results which were an order of magnitude less than the maximum dynamic stress at the central point along the shaft where the electrical rotor is positioned.

The maximum stress in a solid shaft occurs along its central axis at its largest diameter. In the present case the maximum shear stress was found to be 44.5 MPa which is within limits imposed by the Tresca failure criteria for EN 24T steel of approximately 525 MPa. A stress concentration factor of 2 was anticipated at changes in section but even with this factor taken into account the shaft would operate with safety.

4.7 Reasons for Selecting Gas Bearings

The selection of gas lubricated bearings affects the operation and design of the refrigerator system. In the design of a refrigeration system using oil lubricated compressors care must be taken to ensure that lubricating oil, carried over with the exhaust gas is recovered from the evaporator. In the compressor suction line oil comes out of solution with the refrigerant to form a thin film along the

pipe walls. It is then drawn along the pipe wall and back to the compressor crankcase. Special care is usually taken to ensure that at the lowest evaporator temperatures, (when the mass flow rate is at its minimum), gas velocities are sufficient to recover the oil. This requirement can produce relatively large suction line pressure losses which under normal operating conditions reduce the overall performance of the system.

In larger refrigeration systems an oil sump in the evaporator is sometimes used to collect lubricant from where it is fed directly back to the compressor via an ancillary pump. Oil separators are sometimes fitted between the compressor and the condenser. In these ways large suction line pressures losses can be avoided but at the expense of a more complex system. The use of the refrigerant as the lubricant removes the need for these features. In the present RP40 and UP31 systems manufactured by Stone International the designers rely on the suction gas velocity to recover the lubricant.

Oils used in the lubrication of refrigeration compressors breakdown at temperatures found at compressor discharge valves, David²². Chemical reactions occur to produce long chain hydrocarbons such as varnishes and coke and result in a reduction in the lubricating quality of the oil. The effect is to limit compressor discharge temperature. Various maximum temperatures are suggested but 120°C appears to be a practical choice whereas for an oil free system a maximum temperature of 140°C is thought possible, David²². In the majority of refrigerator applications the limiting temperature is seldom exceeded. Where problems are thought to occur are in air-water heatpump applications. Maximum temperatures are unlikely to be exceeded in air-air, (air conditioning refrigerator), applications as in the present case.

A further disadvantage to the use of oil lubricants is that oil films on the inside surface of heat exchanger tubes increase thermal resistance. The effect of this is to reduce the performance of the refrigerator by increasing heat exchanger Approach Temperature Difference, (ATD).

The use of the refrigerant as a lubricant overcomes many of the problems associated with oil lubrication. No special consideration is required in the design of piping and therefore suction and pressure lines may be sized to keep pressure losses as low as is economically justifiable by the

cost of pipework. However, there are some disadvantages. In the case of the aerodynamic bearings the low viscosity of gases gives them a low load capacity compared with similar oil bearings. Therefore gas bearings tend to be larger and running clearances smaller which require smaller manufacturing tolerances.

4.8 Types of Aerodynamic Bearing

Various types of aerodynamic bearings have been developed to overcome the instability problems associated with plain bearings. One method described by Whipple⁴² is to cut shallow chevron or herring bone channels into the bearing surface of either the journal or bush. Knight⁴³ describes the use of this technique in the design of a miniature high-speed motor, Figure 4.5. These channels reduce the load capacity of bearings but increase their stability. An alternative to the plain journal bearing is the foil bearing described by Heshmat, Shapiro and Gray⁴⁴ and shown in Figure 4.6. In the foil bearing the journal rides on a thin gas film above a metal lamina itself supported by a compliant 'bump-foil'. The compliance of the bearing surface increases the fractional speed whirl frequency and provides an amount of damping not present in the plain journal bearing. Kerr⁴⁵ used a similar technique with use of elastomer 'O' seals to support a plain journal bearing. A further method is to use tilting-pad type bearings of the type shown in Fig 4.7.

The availability of standard design information was important when choosing between alternative bearing systems. A literature survey revealed far more information for the plain journal bearings than other types. Furthermore it was thought that the manufacture of plain bearings would be technically simpler than conical, tilting-pad, or foil types. The project was not concerned solely with the development of the bearing system. What was required was a well tried and documented bearing system. Because of the possibility of aerostatic instability of the plain journal bearing it was decided to add compliance to the bushes by mounting them in elastomer 'O' seals following the method investigated by Kerr⁴⁵. A further advantage was that 'O' seals would allow a small degree of misalignment between the forward and rear journals thereby providing a greater degree of tolerance in the manufacture and assembly than would otherwise have been possible.

The plain journal bearing system was further modified by adding pressure jets, shown in Figure 4.1, which allowed the drive shaft to be supported aerostatically during the 'start' and 'stop' procedures thus reducing scuffing damage.

4.9 Journal Bearing Loads

The static load on the journal bearings is equal to the weight of the shaft. The volume and mass of each region of the shaft shown in Figure 4.8 was estimated. Results are listed in Table 4.2. From this data the centre of gravity and shaft mass were determined to be 207.5 mm, (measured from the impeller end), and 1.5311 Kg, (3.376lbs), respectively. The drive shaft carries a number of masses, Figure 4.8. The effect on the weight and moment was determined. These results are listed in Table 4.3.

Mechanical constraints meant that the journal bearings would be located at stations 126mm and 290 mm, Figure 4.8. The forward and rear bearing reactions were estimated from the data in Tables 4.2 and 4.3, to be 20.0 N and 12.2 N respectively.

Dynamic loads originate from out-of-balance forces. Part 1 of B.S 5265 details nine balance quality grades. M.T.I⁴⁴ report some practical rules, based on US Navy specifications, for selecting acceptable levels of residual unbalance. These are;

Speeds below 150 rpm $M_B = 0.25W$
 Speeds between 150 and 1000 rpm $M_B = 5630W/N^2$
 Speeds above 1000 rpm $M_B = 4W/N$

In the above table rotor weight must be in pounds-force and the units of residual unbalance are in ounce-inches.

M.T.I. reporting the work of Federn⁵⁹ give acceptable limits of residual out-of-balance for various categories of machine. In the present case 'Quality Group A' was thought applicable. This group includes gyro-rotors, highspeed expanders and similar machines. The specification of group A machines requires a residual unbalance of 0.0015 to 0.0075

oz.in. per 100lb.wt of rotating assembly. In the present case the rotor assembly weighed 7.3 lbs, therefore, the residual imbalance should not have been greater than 0.00055 oz.in. (3.9×10^{-4} N-m).

At 70,000 rpm the dynamic load is:

$$\begin{aligned} F_c &= M_B \cdot \omega^2 / g \\ &= 3.9 \times 7330^2 \times 10^{-4} / 9.81 \\ &= 21.3 \text{ N} \end{aligned}$$

Variation in dynamic load with rotational speed is given in Figure 4.9.

Transitory loads occur during the service of the compressor.

Impulsive reactions occur in both radial and axial directions. British Railways Board⁴⁷ specifications for refrigeration systems give the maximum accelerations in each axis along with anticipated frequency spectra. Discussions with Ford⁴⁸ suggested that peak body accelerations for passenger carriages under normal operation seldom exceed:

Lateral	0.28 g
Vertical	0.2 g
Longitudinal	0.1g

Also shunting accelerations seldom exceed 7 g, British Railways⁴⁷. At this stage transitory loading was neglected because the prototype unit was to be bolted to a rigid test-bed thus eliminating external loads.

Total bearing load is given as a function of speed in Table 4.4.

4.10 Journal Bearing Design

To confirm the design of the drive shaft assembly and to allow the development of the bearing system to proceed whilst the electric motor was being manufactured the preliminary test-runs were carried with the compressor

operating on air.

The temperature and pressure of the lubricant gas are important parameters in the design of gas bearings. The variation in bearing capacity with ambient pressure and temperature is shown in Figure 4.10. Figure 4.10 was constructed from data provided by Raimondi⁴⁷. From Figure 4.10 it was concluded that the capacity of aerodynamic bearings is more sensitive to variations in pressure than temperature, however, the capacity of a gas-bearing increases with fluid viscosity which increases with temperature. For design purposes bearing ambient pressure was assumed to be 1 atm. During testing the minimum bearing temperature would be found when the compressor was started from cold. During the preliminary 'air' test runs dehumidification of the process air meant that the temperature of the air would be at approximately 0°C and for this reason the design point temperature was taken at this value.

Journal diameter was constrained by the internal diameter of the laminated rotor. The rear journal diameter needed to be less than the laminated rotor lock-nut thread diameter. This restricted the rear journal diameter to 28mm. To simplify manufacture it was decided to make the forward journal to a similar diameter.

A design point bearing eccentricity of 0.6 to 0.7 was reported to be most suitable, Gross⁷⁰, MTI⁴⁴ and Fuller⁷¹. For this reason a design point eccentricity of 0.7 was chosen. With larger eccentricities there is a danger of the journal rubbing against the bush. With smaller eccentricities the bearing becomes increasingly unstable.

Design charts for finite length aerodynamic bearings were used to select bearing lengths and clearances, Gross⁷⁴. The variables used in these charts are; Bearing Number, (Λ), Dimensionless load, (W'), and Eccentricity at constant values of L/D . Charts are provided for $L/D = 1$ and $L/D = 1/2$.

For the forward bearing:

$W = 33.2 \text{ N}$
 $e = 0.7$
 $D = 28 \text{ mm}, (R = 14 \text{ mm})$
 $P = 101325 \text{ Pa}$
 $\omega = 7330 \text{ radians per second}$
 (equating to a design speed
 of 70,000 rpm)

$$\text{For } L/D = 1 \quad W' = \frac{W}{D^2 \cdot P} = \frac{33.2}{0.028^2 \times 101325} = 0.415$$

From design chart⁹⁴ with $e = 0.7$

$$\frac{\Delta}{\phi} = 0.16 = \frac{\mu W \cdot R^2}{P \cdot C^2} = \frac{1.75E-5 \times 7330 \times 0.014^2}{101325 \times C^2}$$

From which; $C = 0.039 \text{ mm}, (1.53 \times 10^{-3} \text{ inch})$

$$\text{For } L/D = 1/2 \quad W' = \frac{2W}{D^2 \cdot P} = \frac{2 \times 33.2}{0.028^2 \times 101325} = 0.830$$

From design chart⁹⁴ with $e = 0.7$

$$\frac{\Delta}{\phi} = 0.85 = \frac{\mu W \cdot R^2}{P \cdot C^2} = \frac{1.75E-5 \times 7330 \times 0.014^2}{101325 \times C^2}$$

From which; $C = 0.017 \text{ mm}, (0.67 \times 10^{-3} \text{ inch})$

For the rear bearing;

$W = 20.4 \text{ N}$
 $e = 0.7$
 $D = 28 \text{ mm}, (R = 14 \text{ mm})$
 $P = 101325 \text{ Pa}$
 $w = 7330 \text{ radians per second}$
 (equating to a design speed
 of 70,000 rpm)

For $L/D = 1 \quad e = 0.7 \quad W' = 0.257 \quad \Delta/\phi = 0.097$
 From which $C = 0.0505 \text{ mm}, (1.99 \times 10^{-3} \text{ inch})$

For $L/D = 1/2 \quad e = 0.7 \quad W' = 0.514 \quad \Delta/\phi = 0.525$
 From which $C = 0.0217 \text{ mm}, (0.856 \times 10^{-3} \text{ inch})$

Because of the relatively large distance between bearing centres a small misalignment between the journal and bushes could have resulted in bearing rub and a reduction in

bearing load capacity. Larger bearing clearances are thought to alleviate this problem to a some extent, however, as shown by the previous calculations this is offset by an increase in bearing length. In addition the correcting moment which tends to align the bearing faces reduces with bearing length, Gross⁷⁴. On balance a shorter bearing length is preferred.

Manufacturing inaccuracies affect bearing load capacity. In the present case economic tolerances of $\pm 1 \times 10^{-4}$ inch on journal and bush diameters were selected. For the forward bearing with an $L/D = 1/2$ the worst effect of these tolerances would have been to reduce the diametrical clearance to 0.0146 mm, (0.47×10^{-3} inch). With such clearances the slightest misalignment between bushes could have resulted in the bearings binding. In view of this it was thought that larger clearance was required. For this reason it was decided to select an $L/D = 0.75$ for the rear bearing and to adjust the length of the forward bearing so that clearance of both bearings were the same. This approach was thought to compromise between the alignment problems of long bearings with large clearances and short bearings with small clearances. By making the clearance of both bearing the same it was thought that the journals and bushes would be more likely to align themselves under normal operating conditions than if different clearances were used.

For the rear bearing;

$$e = 0.7 \quad L/D = 1 \quad W' = 0.257 \quad \text{and} \quad L/D = 1/2 \quad W' = 0.514$$

$$\text{Therefore with } L/D = 0.75 \quad W' = \frac{0.514 + 0.257}{2} = 0.3855$$

From design charts⁷⁴

$$L/D = 1 \quad \mathcal{A}/6 = 0.097, \quad L/D = 1/2 \quad \mathcal{A}/6 = 0.525$$

Taking into account the exponential relationship between L/D and bearing number interpolation gave;

$$\text{For } L/D = 0.75 \quad \mathcal{A}/6 = 0.196$$

$$\text{From which } C = 0.035 \text{ mm, } (1.4 \times 10^{-3} \text{ inch})$$

For the forward bearing $W = 33.2$ N. The bearing number of the forward bearing would equal that of the rear because the clearances were to be equal, therefore;

$$L/D = 1/2 \quad \text{and} \quad \mathcal{A}/6 = 0.196$$

From the previous calculation $W' = 0.386$ for $L/D = 0.75$.

From design charts⁹⁴;

$$e = 0.7 \quad L/D = 1 \quad /6 = 0.196 \quad \text{then } W' = 0.5$$

$$\text{For the forward bearing; } W' = \frac{W}{(L/D) \cdot D^2 \cdot P} = \frac{0.418}{(L/D)}$$

Assuming a linear relationship between L/D and W' then;

$$\frac{0.386 - 0.5}{0.75 - 1.0} = \frac{W' - 0.386}{(L/D) - 0.75}$$

But $W' = 0.418/(L/D)$. Substituting this into the above equation and solving for L/D gave;

$$(L/D) = 0.9$$

SUMMARY

Forward Bearing: $L/D = 0.9$
 $e = 0.7$
 $C = 0.035 \text{ mm}$
 $D = 28 \text{ mm}$
 $L = 25.2 \text{ mm}$

Rear Bearing $L/D = 0.75$
 $e = 0.7$
 $C = 0.035 \text{ mm}$
 $D = 28 \text{ mm}$
 $L = 21 \text{ mm}$

The possibility that journal bearings designed for operation in air would function satisfactorily in Freon 11 was investigated. To test this possibility the eccentricities of the forward and rear bearings were calculated based on Freon 11 property data and the bearing geometric listed in Table 4.5.

Freon Data at a Saturated
Vapour Temperature of 0°C:

Ambient Pressure = 0.402 bar
Freon Viscosity = 13.1×10^{-4} Pa.s

From this data the following were determined;

	FWD BRG	REAR BRG
Dimensionless load	1.18	0.91
Bearing Number	2.29	2.29
L/D=(1/2)	0.9	0.83
L/D=(1)	0.75	0.7

Interpolation of the above data gave;

FWD BRG at L/D = 0.9 e = 0.78
REAR BRG at L/D = 0.75 e = 0.74

These results suggest that the bearings would operate satisfactorily at the design condition both in Freon 11 and in air. The eccentricities predicted for Freon 11 suggested that the bearings would operate with greater stability, however, take-off speed would be greater for Freon 11.

Variations in bearing eccentricity with rotational speed were estimated from data provided by Gross⁷⁴. Using the design point fluid properties for air the results of calculations, shown in Figure 4.11, indicate that bearing eccentricity would approach unity when the rotational speed was approximately 10,000 rpm. This represented the touch-down and take-off speed of both the forward and rear bearings. It was thought that 10,000 rpm was acceptable because under normal starting conditions the compressor would accelerate to this speed within a fraction of one revolution of the shaft, Hill⁵⁴.

Fractional speed whirl may occur resulting from interaction between the aerodynamic film and the whirling mass of the rotor. Gross⁷⁰ and MTI⁴⁴ conclude that the condition can be completely avoided by operating at a speed less than 1.6 times the lowest critical speed of the rotor-shaft. When this is not possible then an inherently stable bearing type, (such as a tilting pad type), should be selected.

Table 4.1 lists the estimated values of drive shaft critical speeds. It was concluded in Section 4.4 that the first rigid body critical speeds was 112 rps, (6720rpm). The critical operating speed in terms of fractional speed whirl instability was therefore 1792 rps, (10,752 rpm), which is approximately equal to the predicted take-off speed of the bearings. Supporting the bushes in 'O' seals to provide damping and the use of aerostatic lubrication when starting the compressor it was thought that whirl instability could be avoided. During later testing take-off speed was found to be 15,000 rpm and no indication of whirl instability was observed.

4.11 Augmented Take-Off

To prevent damage to bearing surfaces during testing it was decided to provide aerostatic pressure to both journal bushes. The design method adopted was due to Porritt⁷².

For design purposes it was assumed that there would be 8 feed jets. It was also assumed that the supply pressure would be 60 psig and the bearing clearance would be 0.035mm. To prevent bearing 'lock-up' and to give the optimum load capacity the optimum eccentricity should be less than or equal to 0.4, Porritt⁷². Charts provided indicate;

$$\frac{W}{(P_0 - P_{AMB})_{OPT.L/D}} = 0.144$$

For the present case the static load on the forward bearing is 20 N. The L/D ratio is 0.9. This gives a supply pressure of 1.98 bar.

The touch-down load represents the largest load which the bearing will take without scuffing. For this Porritt suggested the following criteria;

$$\frac{W}{(P_0 - P_{AMB})_{OPT.L/D}} = 0.225$$

In the present case the touch down load was estimated to be 31.2 N which was approximately the same as the anticipated load at the design speed.

From the charts supplied by Porritt⁷² the maximum bearing

stiffness occurs when;

$$\frac{S.C.}{(P_o - P_{AMB})_{OPT. L.D}} = 0.36$$

The maximum bearing stiffness for the present case was therefore 1.42×10^6 N/m.

For optimum stiffness Porrit⁷² gave the following relationship from which the orifice diameter may be determined.

$$\frac{n.d.(L/D)}{P_o.C^2} = 3000$$

where n = number of pressurization orifices

d = orifice diameter in inches

(Imperial units should be used in this case.)

Substituting known values into the above equation gave an orifice diameter, (d), of 0.035 inches or 0.88 mm.

Porrit⁷² gave the following expression for determining the volumetric air flow rate through a aerostatic journal bearing.

$$Q = 4.56 \times 10^5 C^2 (0.442 P_o^2 - P_{AMB}^2) D/L \text{ CFM}$$

(It should be noted that only Imperial values should be used in this expression)

Substituting values for the forward bearing gave a flow rate of 0.8 CFM.

Table 4.6 summarizes the design data resulting from the analysis of both the forward and rear bearings.

Gas jets should be perfectly circular and each should have the same pressure loss characteristics. Imperfections in the shape of the jets can result in a maldistribution of gas flow which may produce a pneumatic instability. Because of diameter of the jets, (0.88mm for the forward bearing and 0.96mm for the rear - see Table 4.6), it was decided to utilise 'Seitz', (watch makers), jewels which were available in the required sizes. Because of the physical size of the jewels used each was glued into a separate brass holder which in turn was fitted into a prepared hole in the bush.

This process was found to be unnecessary and in later variants the brass holders were dispensed with and the jewelled jets were fitted directly into the bush body.

4.12 Elastomer 'O' Seals

In addition to providing damping and a degree of compliance the 'O' seals provided sealing for a gas manifold around each bush. The bushes were also held in place by the frictional resistance of the 'O' seals within the bearing support assemblies. No other fixing device was originally thought necessary. The arrangement was found to be satisfactory during preliminary tests but failed during later test runs when liquid refrigerant is thought to have penetrated between the seals and the support assembly reducing the frictional coefficient which allowed the bushes to move. A spring clip was then used to secure each bearing in place.

It was important that the 'O' seals were made from a material which was compatible with Freon 11. Some elastomers are prone to swelling in the presence of fluorocarbons. Elastomers with less than 10% swell were thought to be suitable for use with Freon 11. These include,; Buna N, Hypalon 40, (Du Pont Co), Neoprene W, (du Pont Co), Vitron A, (Du Pont Co), PVA, Thiokol Fa, (Du Pont Product Technical Bulletin B-2).

For experimental purposes it was intended to mount the bearing bushes in, (size number 126), Vitol 'O' seals with a 65° Shore hardness. These seals have an internal diameter of 1.362 inches, (+/-0.006 inches) and a rubber diameter of 0.103 inches, (+/-0.003 inches).

Vitol is a fluoro-rubber. It was selected because of its resistance to solvents, including halogenated hydrocarbons such as Freon 11. It was also reported to have a high level of impermeability to gases; low temperature flexibility; resistance to creep; and good resistance compression set, (Du Pont Product Technical Bulletin B-2).

Absorption of liquids often causes softening of elastomers. Buick⁷³ suggested that Vitol elastomers soaked in Freon 11 experienced up to 10 % swelling. However when left to dry in air for 14 days it returns to the original dimension. No information in the literature was found on the variation in

the mechanical properties of elastomers which had been exposed to Freon 11 for extended periods. It was important, therefore, to gain some knowledge of this.

Experiments were carried out to determine the effect on Vitol 'O' seals, (Shore hardness 65'), from exposure to liquid Freon 11. The Youngs moduli of a sample set of six 'O' seals were measured before and after being soaked in Freon 11 for six months. Before soaking the Youngs modulus of each seal was measured using a standard tensiometer technique. An average value of 5.8 MPa was determined. This compares with a quoted value of 5.95 MPa. Care was taken to ensure that the samples were not over extended during testing and each sample was examined for surface defects before and after testing. After soaking in Freon 11 for six months the samples were re-tested and their Youngs moduli determined. It was found that the average Youngs modulus had fallen to 3.77 MPa, (65% $\pm$ 2% of the original value), which equates to a fall to 58^o Shore hardness. Softening of elastomers in the presence of solvents such as halogenated carbons is very common and was not unexpected.

The unextended length of six 'O' seals was measured before and after soaking and also after a period of drying following soaking. Each sample 'O' seal was cut perpendicular to its axis and its length measured. Before soaking the average length of the samples was 106.5mm. After soaking for 6 months the average length was found to be 113.2mm. This indicated an average linear elongation of 6.25% and a volumetric swelling of 20%. After drying in air for two weeks the average length had reduced by 2.3% to 104mm which represents a reduction in volume of 6.7%.

After four months soaking it was noticed that the Freon 11 became discoloured with what appeared to be a brownish substance. From this it was concluded that the refrigerant had washed some of the filler material from the elastomer. This was confirmed by the 6.7% volume shrinkage after the samples had been dried.

The Youngs modulus of elastomers depends on the filler content and they tend to soften as the quantity of filler is reduced. In the present case the stiffness had reduced after soaking as shown by the Young modulus results. It was thought that the softening of the 'O' seals may be aggravated by the cyclic loading and moderate heating.

During normal operation it was anticipated that the 'O' seals would infrequently be contaminated with liquid refrigerant. For the preliminary development of the present compressor it was felt that Vitol 'O' seals would provide adequate performance though it will be useful to examine the problem of softening in more detail as a separate study.

4.13 Bearing Material

Factors considered were; thermal expansion compatibility between the journal and bush, dimensional stability, corrosion resistance and sliding compatibility. Following the recommendations of Murray⁷⁴ it was decided to manufacture the bushes from the same material as the shaft, that is B17 M40 alloy steel. An important advantage of this was that the small radial clearances between bush and the journal would be maintained as the temperature of the bearing increased during operation. The material also has good corrosion resistance and dimensional stability.

A disadvantage of operating a steel journal in a steel bush is the problem of high sliding friction resulting in high wear rates. Under normal operation the journal would sit on a cushion of gas. This aerodynamic cushion does not produce sufficient lift below the take-off speed of the bearing. By providing external pressurization during the starting and shutdown this problem is overcome. High-speed touch down, (rubbing contact), is a common fault and the risk of seizure is still present particularly for steel-on-steel bearings. For this reason it was decided to add a solid lubricant to the bush bearing surface.

4.14 Solid Lubrication

There are a number of solid lubrication processes available. Of the polymer types available filled P.T.F.E. appeared to provide most promise. The P.T.F.E sprayed in a thin film onto the bush provides a surface which has a low friction and resistance to abrasion up to 150°C. It is reported to be suitable for low loads, (as in this case). A problem with P.T.F.E. is its low thermal conductivity and its tendency to ballup under conditions dependent on temperature and bearing surface pressure, Campbell⁷⁵

An alternative solid lubrication technique to polymer sprays are surface conversion processes such as the Sulfinuz. For

this the component is immersed in a salt bath at 540-600°C, Neale⁷⁴. The treatment introduces carbon, nitrogen and sulphur into the bearing surface. The Sulfinuz treatment improves scuffing wear resistance and fatigue resistance. Sliding resistance is reduced because sulphide in the surface layer acts as a solid lubricant and nitrogen in the salt produces a nitriding effect of the surface layers which improves surface hardness and consequently wear resistance. A disadvantage is that the process can cause distortion.

The use of hybrid bearings combined with a Sulfinuz anti-scuffing treatment ensured that bearing wear on start-up and shut-down would be minimal and scuffing resistance due to high speed rubbing contact would be greatly improved over that which could be expected for a steel-on-steel arrangement.

4.15 Thrust Bearing Design

The thrust bearing assembly, shown in Figure 1.1, consisted of a thrust washer, (item 11), and a pair of thrust plates, (item 13). The bearing assembly was 75mm in diameter with a 16 mm diameter central hole. The washer was located against a shoulder on the drive shaft by a spacer, (item 14), and nut, (item 15). An axial clearance of 0.003 inches between the thrust washers and the thrust plates was maintained by a spacer, (item 12). Compressed gas was fed to the thrust plates via radial holes and gas jets, (8 per thrust plate), equally spaced on a 1.9 inch P.C.D.. These are shown in Figure 4.3. There were eight 0.03 inches diameter gas jets in each thrust plate. Seitz 'watchmakers' jewels were used to ensure the geometrical precision of the jets. The thrust plates and spacer were secured to the internal forward face of the bearing support assembly, (item 8, in Figure 1.1), by eight 1/4 inch BSF cap-head screws.

4.16 Thrust Bearings Load

The maximum axial load from the impeller was thought to occur when the flow rate and compressor pressure ratio were both maximum. In practice these conditions never occur simultaneously due to the flow-pressure ratio characteristic of the compressor. Therefore a safety margin was provided in the design by this assumption.

In order to keep the diameter of the thrust bearing within

limits acceptable from a stressing point of view it was decided to balance some of the axial load with a labyrinth seal and circular nut, Figure 1.1, (items 15 and 18).

Referring to Figure 4.12 the axial thrust acting on the impeller was estimated from the following force balance;

$$F = m V_o + p_o A_o - p_1 A_2 + (p_1 - p_o) A_1$$

It was assumed that inlet static pressure also acts to the right of the dividing seal shown in Figure 4.12.

The design condition selected was for operation on Freon 11 with a compression ratio of 6. Values of mass flow, pressure ratio and compressor inlet temperature, from earlier research, Elder and Walters³ are listed in Table 4.9 with dimensionless flow parameter, (X), and values of total-to-static pressure and temperature ratios taken from isentropic tables. The flow velocity was calculated using the steady flow energy equation and flow area dimensions were taken from the compressor drawings.

Substituting the Freon and air values in Table 4.9 into the above equation and setting the resultant axial thrust, (F), to zero and solving for seal area, (A₁), gave seal diameters of 123mm for air, 37mm for Freon 11(i) and 20mm for the Freon 11(ii) case. Selecting the arithmetic mean of these values gave a seal diameter, (d), of 60mm. For operation on air the effective axial thrust was estimated to be 100 lbf with this a seal diameter of 60mm. Because this was the largest anticipated residual force it was decided to design the bearing to have a load capacity of 100 lbf.

4.17 Aerostatic Supply Pressure Ratio and Load Capacity

The standard design method was adopted, MTI⁴⁴. Table 4.7 lists the design data for the design of an aerostatic thrust bearing. For a double acting thrust bearing the load capacity is determined from:

$$W = 0.8.C.K$$

In the present case W = 100 lbf and therefore;

$$C.K = 125 \text{ lbf}$$

For inherently compensated bearings dimensionless bearing stiffness parameter is defined as;

$$K' = \frac{CK}{\frac{2}{3} \pi (R_o^2 - R_i^2) [P_s - P_{amb}]}$$

Substituting values listed in Table 4.7 gives:

$$K' (P_s - P_{amb}) = 31.7$$

MTI⁶⁶ suggest a maximum value of Dimensionless Load, (W'), is 0.4. In the present case a margin of safety was given and the Dimensionless Load was set at 0.25.

$$\text{Assuming } W' = 0.25$$

$$\text{then } W' = \frac{W}{\pi (R_o^2 - R_i^2) (P_s - P_{amb})}$$

Substituting values into this expression gives:

$$(P_s - P_{amb}) = 67 \text{ psi}$$

Therefore $K' = 0.47$ which from the design charts is acceptable and the bearing will be stable.

4.18 Dimension and number of pressure jets.

For an inherently compensated feed orifice M.T.I suggest the following expression which relates the number of jets, their

dimension and the bearing clearance.

$$n.d = \frac{P_s.C^2}{6.\mu.\sqrt{R.T}}$$

From the data listed in Table 4.9 $6.\mu.\sqrt{R.T} = 1.762 \times 10^{-4}$

Therefore $n.d = 6.69 \times 10^{-5}.C^2$

Assuming that $n = 16$

Then $d = 4.19 \times 10^{-4}.C^2$

Also $\xi = (1/2) \ln(r_o/R_i) = 0.49$

MTI⁴⁴ suggest that the pitch circle diameter of the feed jets must be lie within a range given by;

$$D = 2.\sqrt{(R_o.R_i)} \pm 20\%$$

In this case $D = 1.9$ inches

MTI⁴⁴ provide charts for selecting a minimum value of 'n' in consideration of 'd' which prevents bearing instabilities. The axes of these charts are given in terms of the groups $n.\xi$ and $d/\xi D$.

In this case $n\xi = 16 \times 0.49 = 7.8$

If this value is assumed to be the minimum value of n then from the charts provided;

$$d/\xi D > 0.03$$

assuming $d = 0.03$ inches:

$$d/\xi D = 0.03/(.49 \times 1.9) = 0.032 > 0.3.$$

From above;

$$d = 4.19 \times 10^4 \cdot C^2$$

Therefore $C = 0.00085$ inches.

A summary of design data for the aerostatic thrust bearing is listed in Table 4.8

4.19 Natural Frequency of the Thrust Bearing

Substituting known values into the equation given in Section 4.17 and solving for stiffness, (K), gave;

$$K = 1.15 \times 10^5 \text{ lbf/inch (20.1 MN/m)}$$

The mass of the rotor-shaft assembly is 7.3 lbs (3.31 kg)

The natural frequency of the bearing is therefore;

$$\begin{aligned} \omega_n &= \frac{\sqrt{K/M}}{2\pi} \\ &= 392 \text{ cps (23570 rpm)} \end{aligned}$$

Because the design operating speed was to have been 70,000 rpm this result suggested that the design was acceptable from a critical speed standpoint.

4.20 Aerodynamic Operation

According to Gross⁷⁰ a complete annular plane bearing can develop thrust in two ways. Either through a wedge effect caused by the inclination of the plane of the rotating disc to the thrust plate, or by a squeeze film effect. For parallel faced thrust bearings Fuller⁷¹ described a theory in which gas flowing through the bearing clearance experiences a temperature rise causing it to expand. The flow volume thus increases with constant clearance with the effect that the pressure within the bearing increases. For 'thick' film bearings the contribution of this effect to the load capacity of a bearing is small, Fuller⁷¹. As clearance reduces the heating effect increases and the contribution of the effect becomes more significant. In the present case the gas flow through the bearing is created by the centrifugal

acceleration within the fluid boundary layer of the rotating disc.

4.21 Thrust Bearing Material Selection

Because of the high operating speed, (70,000 rpm), and the need to have a large bearing area, (diameter), to support the axial load imposed by the impeller it was necessary that the material from which the thrust bearing was made to withstand the high centrifugal loads imposed on it and have good dimensional stability and wear resistance. It was decided to use 817 M40 alloy steel. This decision was consistent with the choice of material for other dynamic components.

Mild steel was selected for the static components of the thrust bearing. The advantages being cost and machinability. The disadvantage was susceptibility to corrosion, though this was controlled by specifying a good machine finish and carrying out the Sulfinuz treatment to the thrust plates.

4.22 Discussion

For the preliminary development it was decided to use the plane thrust bearing shown in Figure 4.3. A more complex bearing could have been used, such as stepped bearing faces, pad or sector type, described by Gross⁷⁰ or chevron type described by Whipple⁴² but for manufacturing purposes the relative simplicity of the plane bearing was an important consideration. Selecting the simplest design was thought to have the benefit of providing a datum for future studies.

Design calculations described in this section showed that it was realistic to select a plane, annular type, double-acting, aerostatic bearing. This choice simplified manufacture. The resulting bearing data are listed in Tables 4.7 and 4.8

Analysis predicted that the natural frequency of the bearing was approximately 23000 rpm. This was well below the operating speed of the compressor and therefore it was not expected to have any significant effect on the performance of the bearing system at the design operating speed.

4.23 Lubrication System

Gas pressure could not be allowed to build up in the compressor casing due to a lack of venting, therefore, an important consideration in the design of the bearing systems was the scavenge system for the lubricant. Figure 4.13 shows a schematic representation of the thrust bearing lubrication system. Gas enters the compressor under pressure at 'a' from where it flows into an annular chamber 'b' and to eight radial ports, 'c', in each thrust plate. These radial ports are seen in Figure 4.3. The gas is ducted to 'jewelled' feed jets, 'd', where it expands into the bearing clearance, 'f'. The gas flows outwards into an annular space at 'e' and inwards into volumes 'i' and 'j' and on via internal drillings 'g' and 'h' and into the motor casing where it is drawn along with the suction gas to the compressor inlet.

Figure 4.14 shows the gas path through the forward journal bearing. Gas enters at 'a' from where it is ducted under pressure to an annulus 'b' around the bearing bush, 'd', which distributes the gas evenly to eight jets, Figures 4.1 and 4.2. The distribution annulus is sealed by two 'O' seals, 'c'. The gas flows outward in the clearance between the journal and bush surfaces. Some of the gas flows directly out into the motor casing whilst the remainder flows into the annular space at 'e' where it mixes with gas from the thrust bearing and flows into the motor casing via internal drillings.

The gas supply and scavenge system for the rear bearing is similar to that of the forward bearing. Referring to Figure 4.15, gas enters the bearing housing at 'a' and is ducted to the bearing bush via an internal drilling. The gas is distributed evenly about the bush by an annular space, 'b'. Gas is fed to eight jets and on into the bearing clearance from where some gas flows directly into the motor casing at 'g' whilst the remainder flows into an annular space at 'e' and on via internal drillings at 'f' to the motor casing at 'g'.

4.24 Conclusions

The aim of this chapter was to evaluate the design of the compressor drive shaft assembly and bearing system. The drive shaft assembly and bearing system are shown in the general assembly drawing in Figure 1.1.

Potential sources of vibration were considered in Section 4.3. Sources from within the compressor unit include aerodynamic buffeting, out-of-balance magnetic and mechanical forces. It was concluded that aerodynamic buffeting would not be significant due to the symmetry and aerodynamic smoothness of the drive shaft. The use of close manufacturing and assembly tolerances would ensure accurate rotor alignment and would keep out-of-balance magnetic forces within acceptable limits. Out-of-balance mechanical forces would be kept to a minimum by dynamic balancing. The limit of acceptable out-of-balance moment was set at 3.9×10^{-4} N-m, Federn⁵⁷. During the application external sources of vibration would come from shunting manoeuvres when as much as 7g acceleration might be experienced, British Railways⁶⁷. For the purposes of this study external loading was neglected.

Accurate estimates of lateral and torsional critical speeds were required to ensure that they did not coincide with the operating speed of the compressor. A finite element program based on the M-P technique was used. Table 4.1 lists predictions for the first five critical speeds. It was concluded that the design operating speed, (70,000 rpm), would not coincide with any critical speed. The first critical torsional speed was estimated to be 76,440 rpm, Section 4.5. Some concern was felt because of the proximity of this speed to the operating design speed of 70,000 rpm. Recommendations were made for using a shorter, torsionally stiffer, shaft in future designs.

The maximum shear stress in the drive shaft was estimated to be 44.5 MPa which compared to a yield point value of 525 MPa for EN24T, the shaft material. It was concluded that the drive shaft would not fail due to over stressing.

Reasons for selecting gas bearings were discussed in Section 4.7. Gas bearings are preferred because no special provisions are required in the design of a refrigerator system for scavenging lubricant which simplifies the design and improves COP. Also, a reduced likelihood of chemical breakdown of the refrigerant gives a better high temperature operation of the compressor. Better thermal performance of heat exchangers is given because of the removal of the oil film on heat exchanger surfaces which also leads to improvements in system C.O.P.

Types of aerodynamic bearing were described in Section 4.8. The main purpose in modifying the plain bush type, for

example by introducing chevrons or herringbone marking into the bearing surface, is to make a bearing operate with more stability over a greater speed range. For this reason and for simplicity of design, it was decided to mount the journal bushes on compliant elastomer 'O' seals, a technique investigated by Kerr⁴⁵.

The properties of Viton 'O' seals were studied in Section 4.13. It was concluded that Viton seals, (65° Shore hardness), would be suitable. Tests showed that Viton is susceptible to swelling, (20%), and reduction in stiffness after prolonged soaking in Freon 11 liquid.

It was concluded that the bushes and thrust washer should be manufactured from EN24T, the same material as the shaft. This would provide thermal compatibility between the shaft and bearings which was important in view of the small clearances. Dimensional stability and some corrosion resistance would also be given. It was decided to use mild steel for the manufacture of the thrust plates which would be Sulfinuz treated to reduce scuffing wear and sticking in the event of high speed contact. The journal bearing bushes were similarly treated. Solid lubrication was considered in Section 4.14.

The remainder of this chapter was concerned with the detailed design of the journal and thrust bearings.

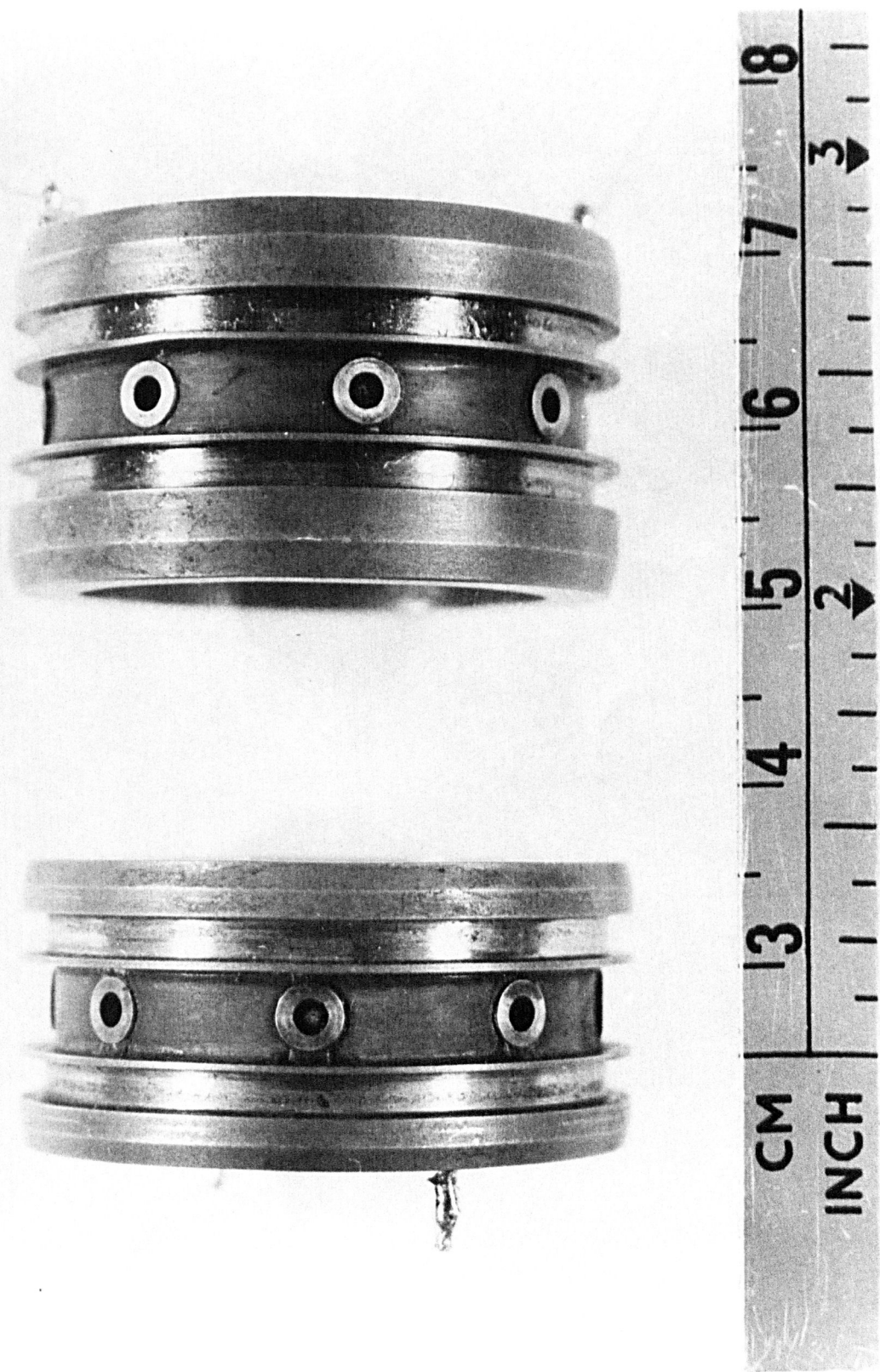


FIG. 4.1 JOURNAL BEARING BUSHES

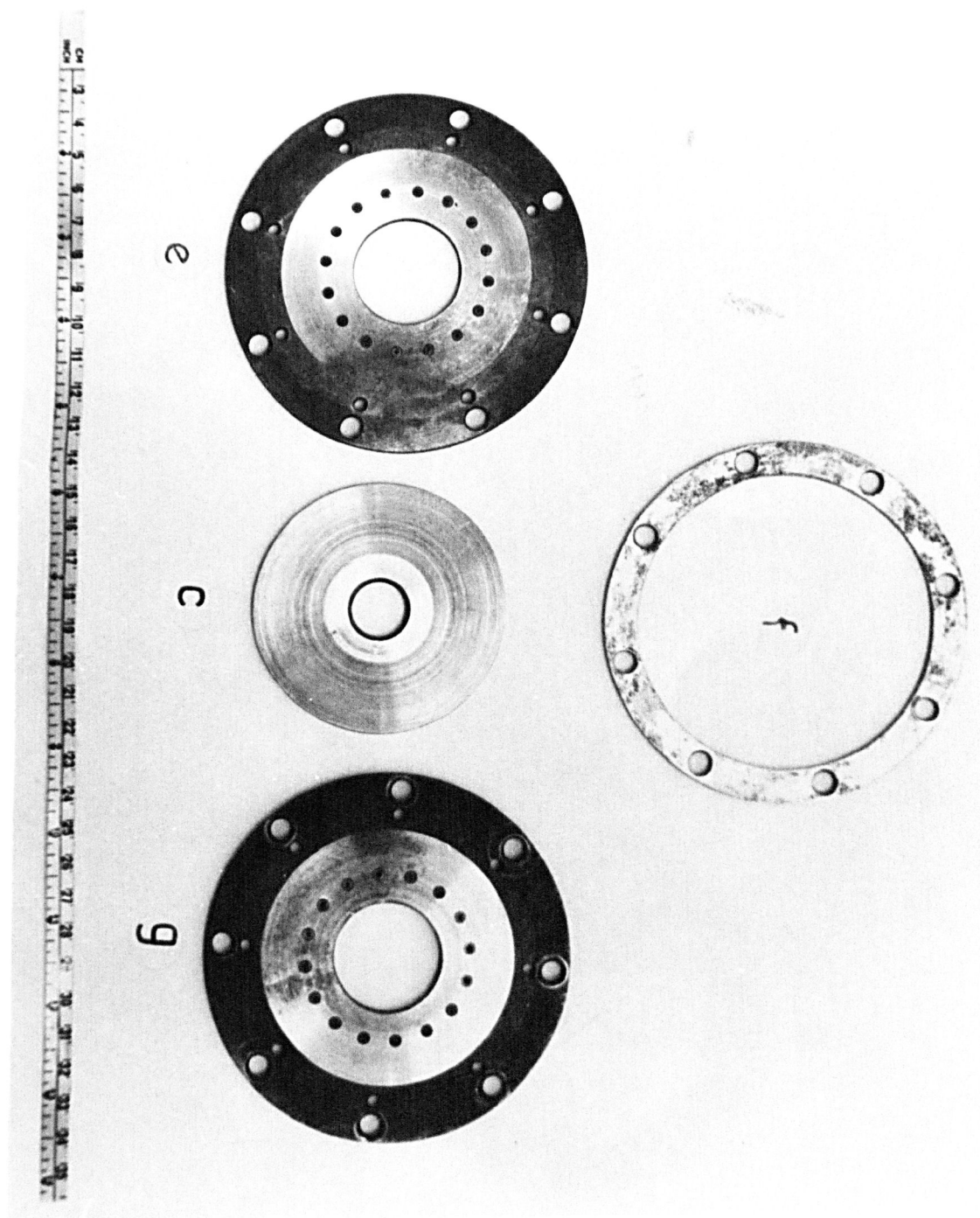


FIG. 4.2 ASSEMBLY

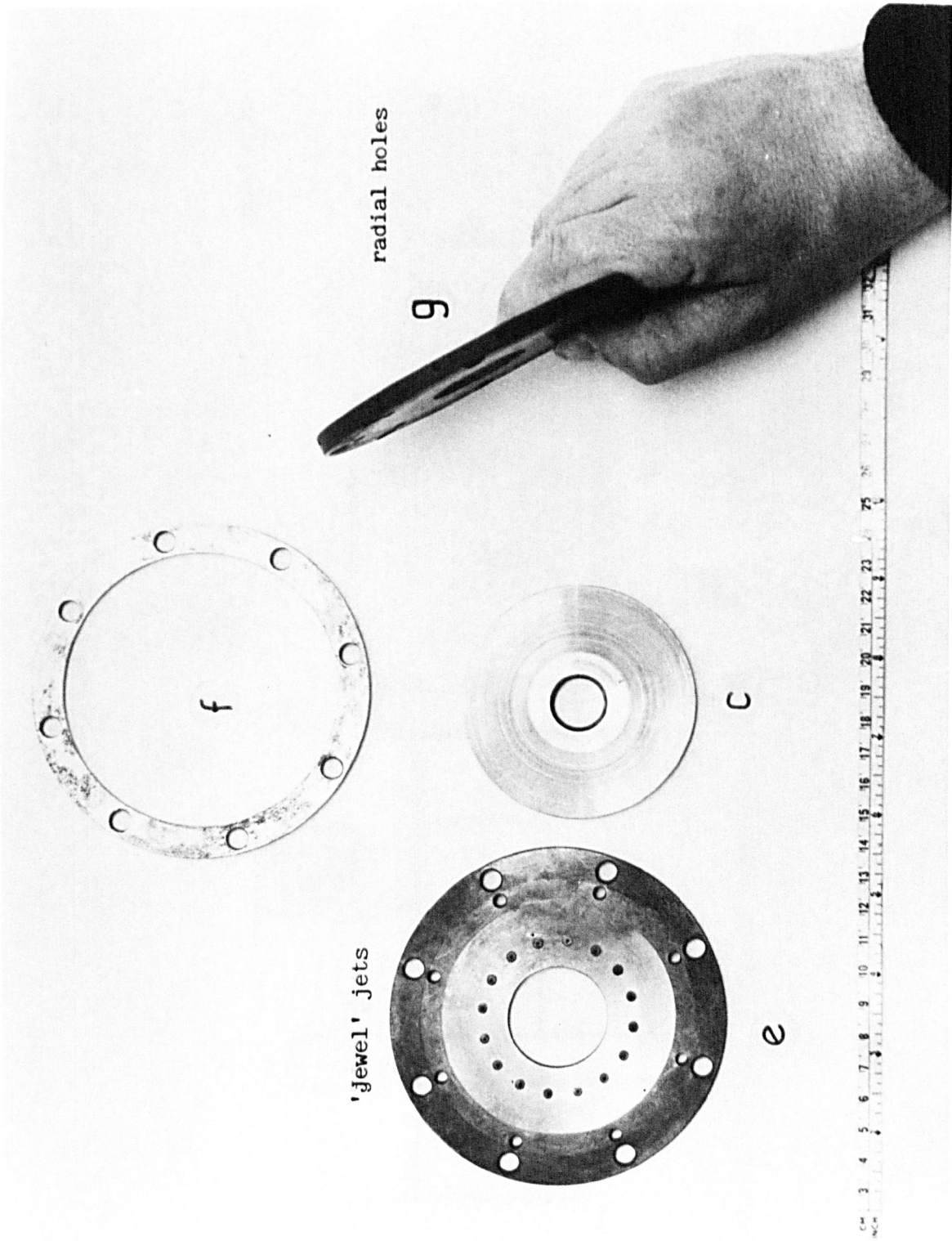


FIG. 4.3 THRUST BEARING ASSEMBLY

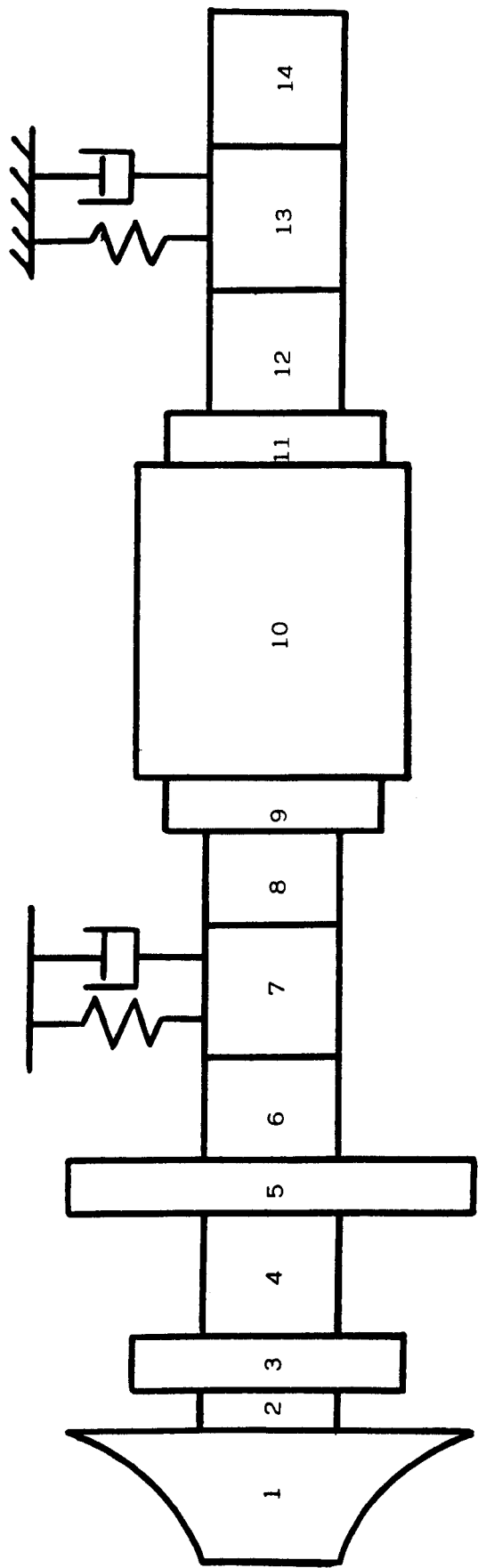


FIG. 4.4 14 REGIONS FOR F.E. MODEL OF SHAFT

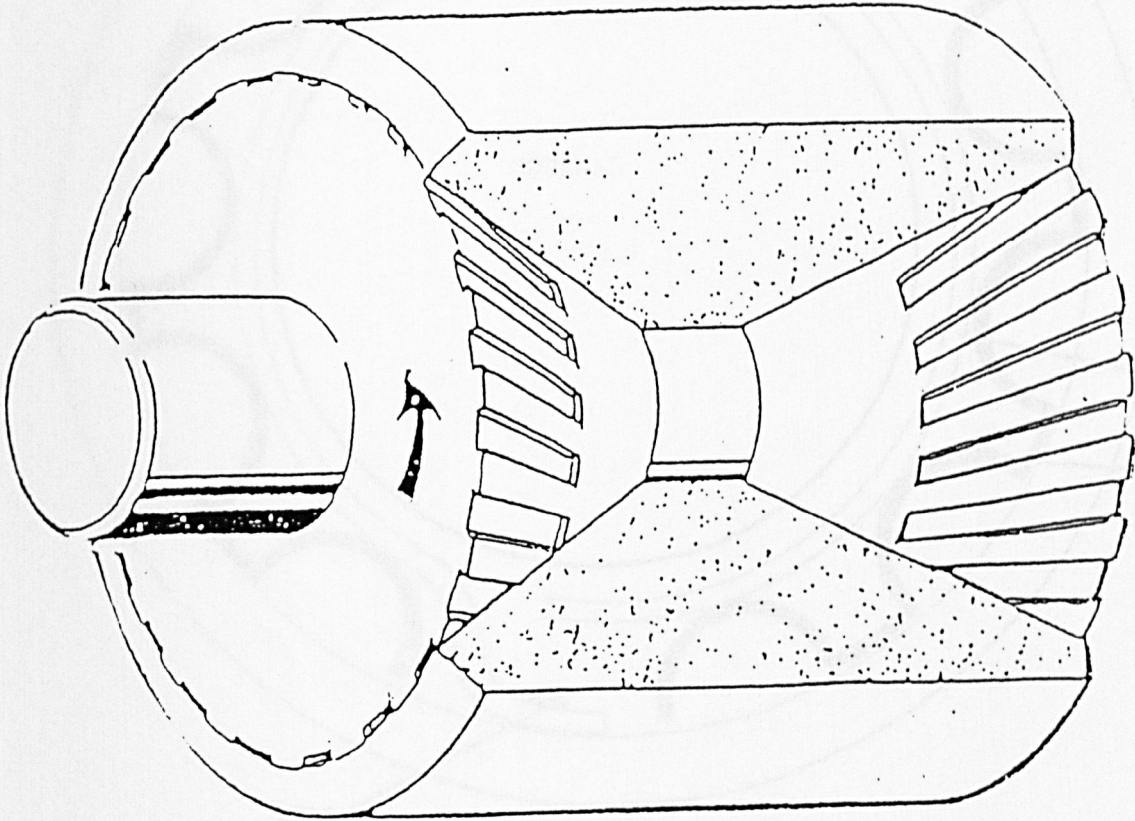


FIG. 4.5 SPIRAL GROOVE CONICAL BEARING

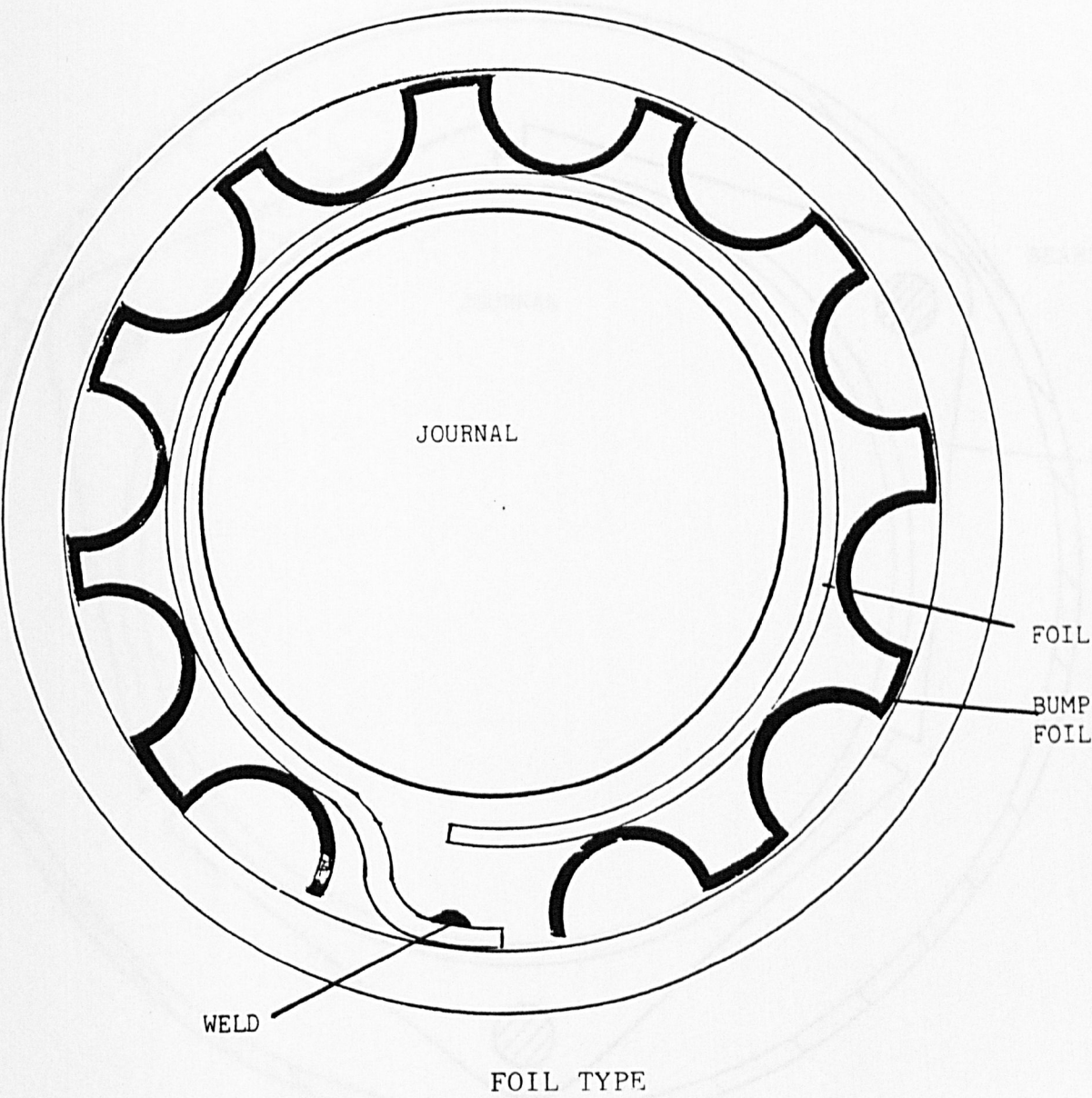


FIG. 4.6 JOURNAL BEARING

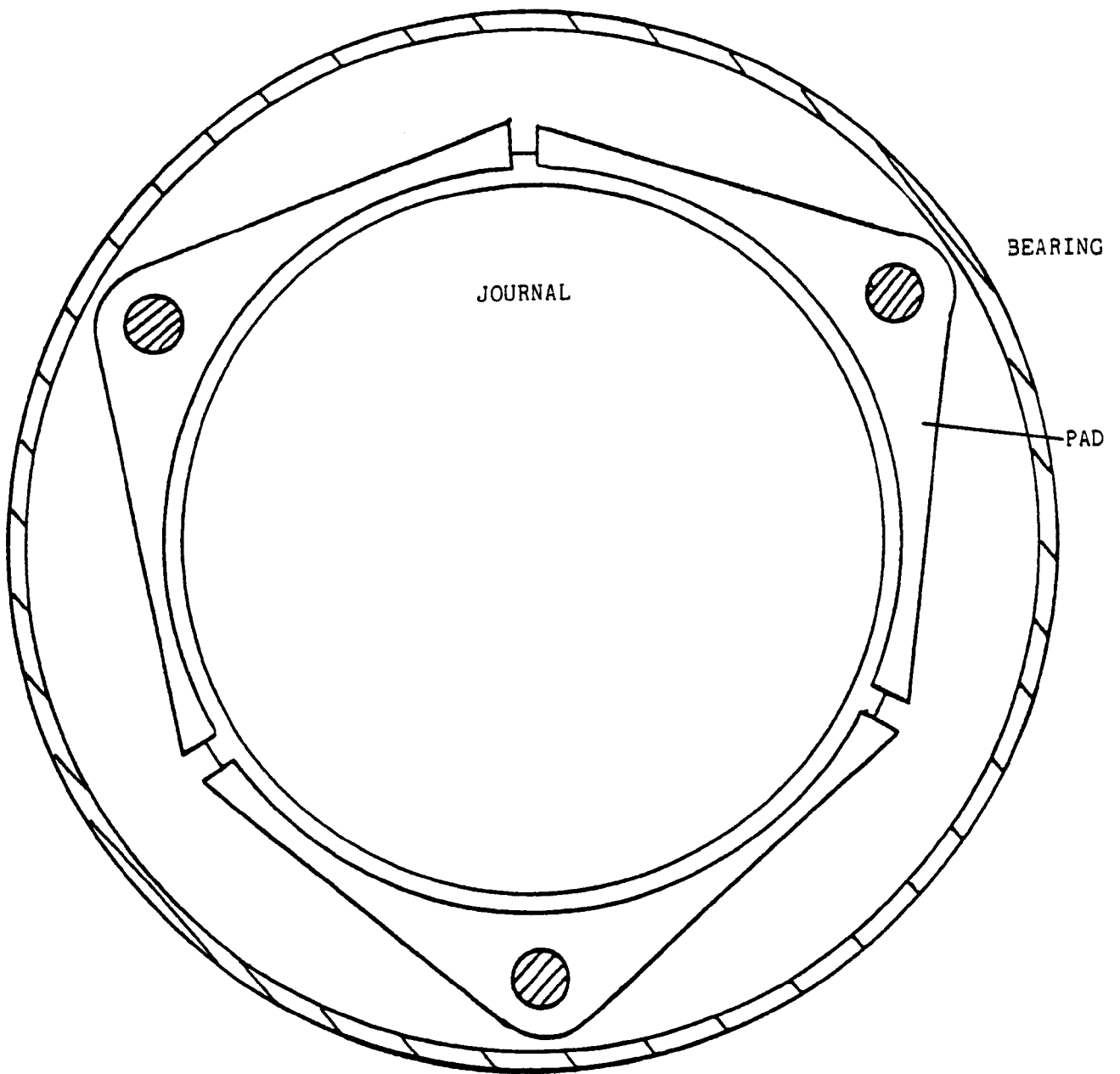
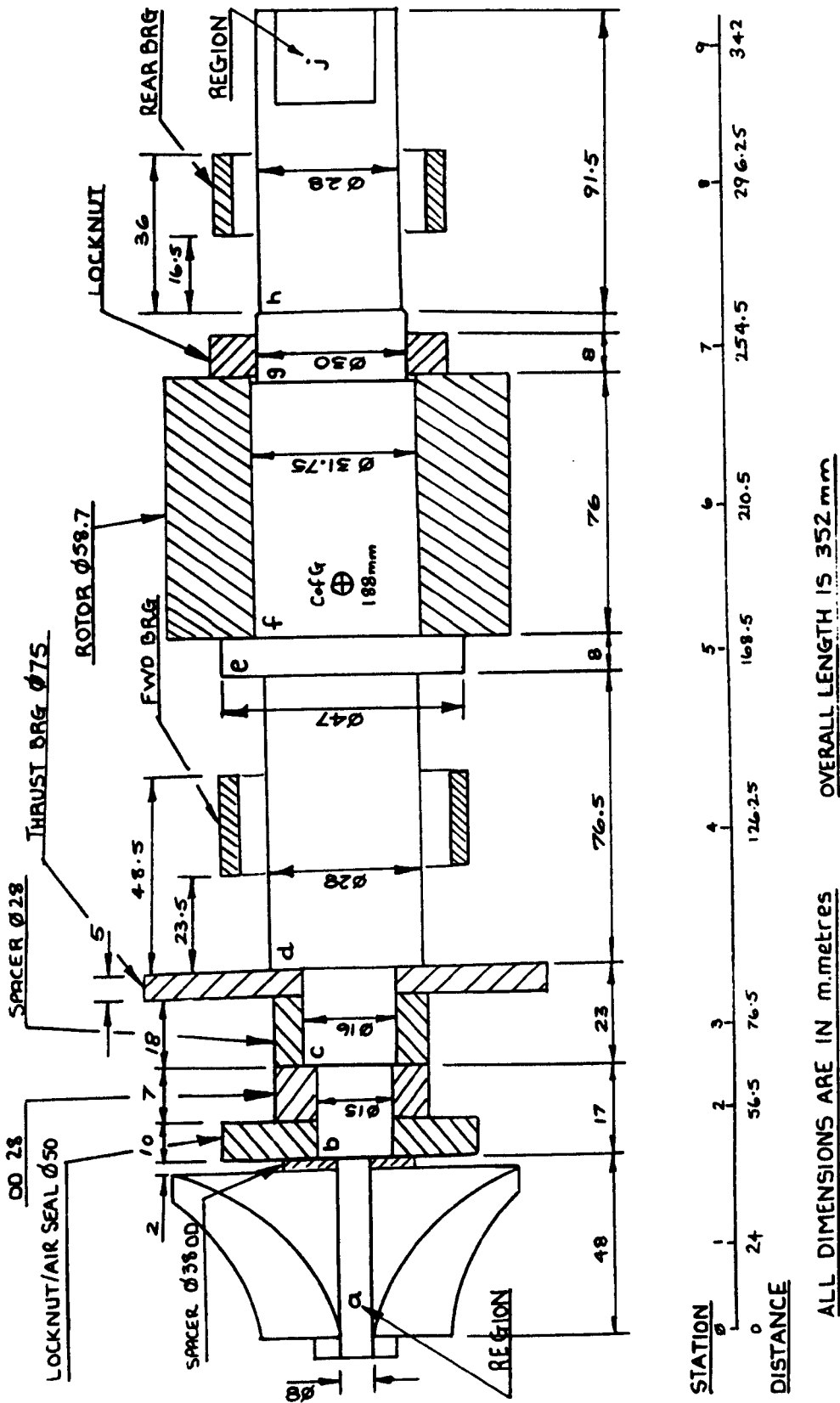


FIG. 4.7 TILTING PAD BEARING



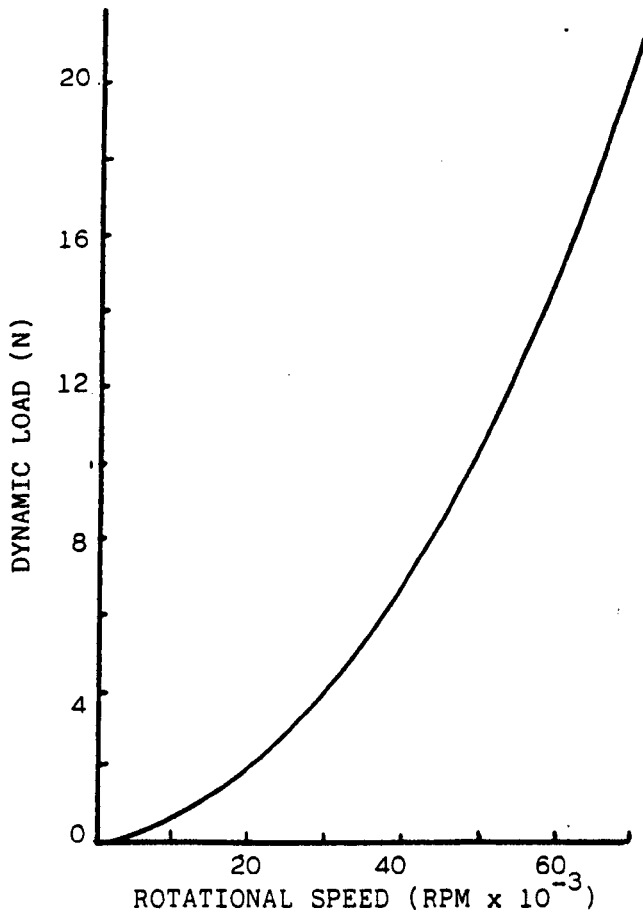


FIG. 4.9 VARIATION IN DYNAMIC LOAD WITH ROTATIONAL SPEED

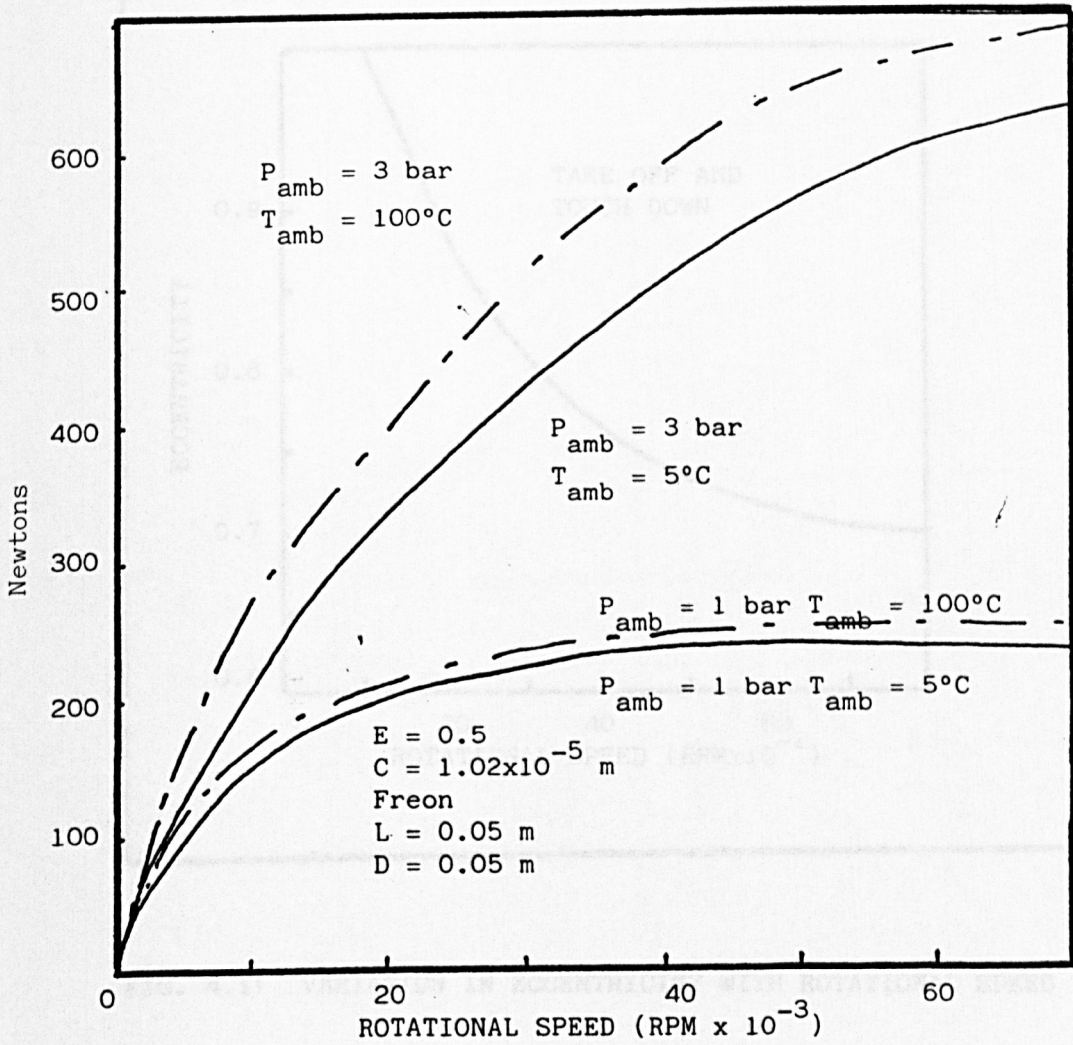


FIG. 4.10 VARIATION IN BEARING LOAD CAPACITY WITH ROTATIONAL SPEED, FLUID PRESSURE AND TEMPERATURE

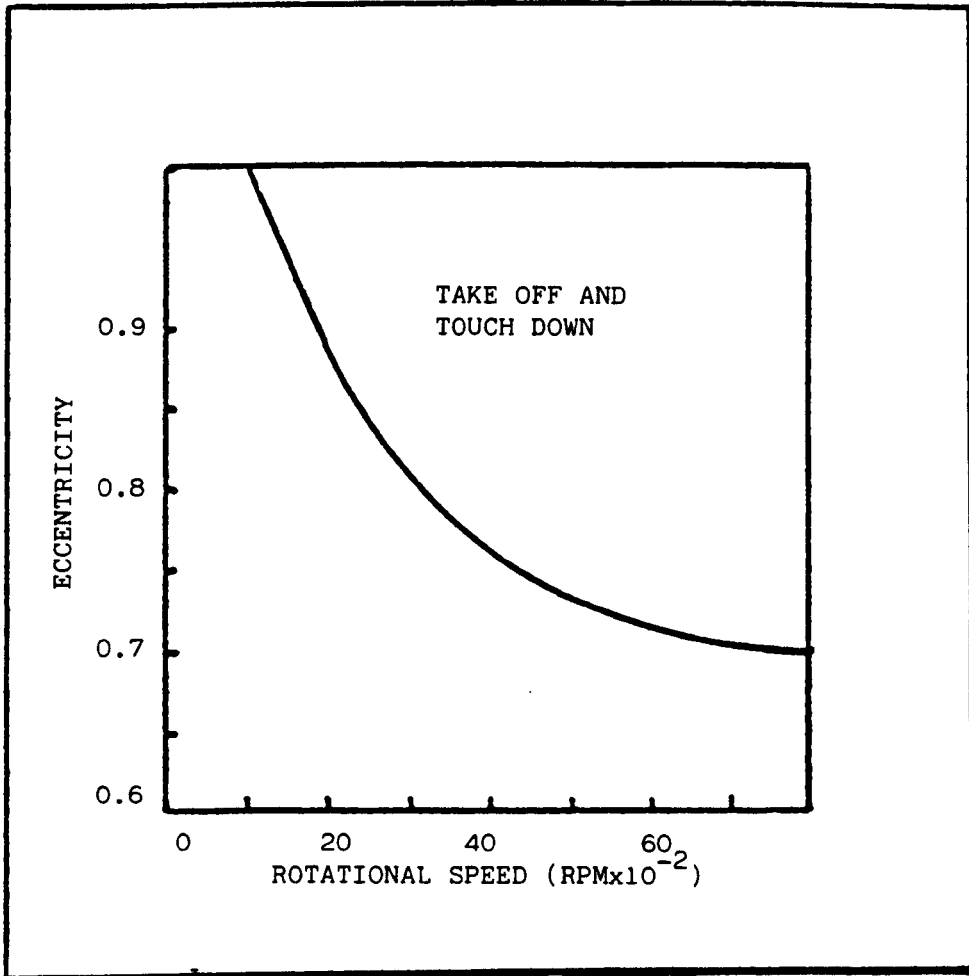


FIG. 4.11 VARIATION IN ECCENTRICITY WITH ROTATIONAL SPEED

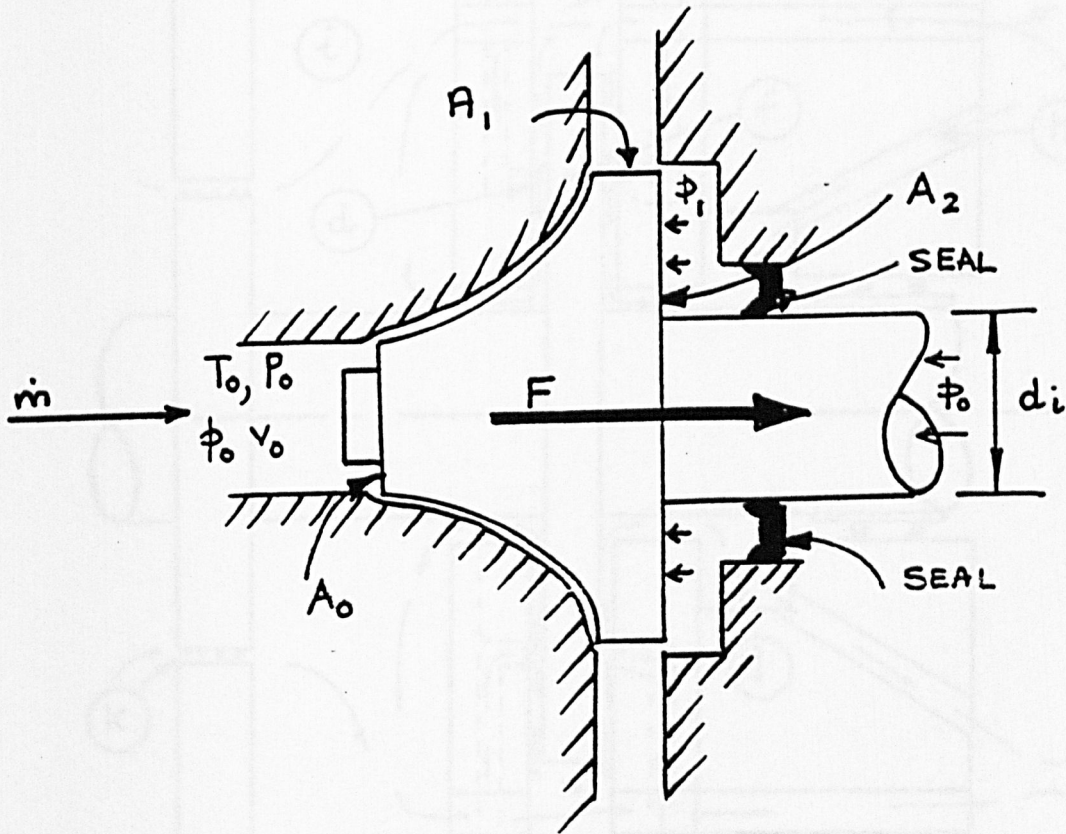


FIG. 4.12 CENTRIFUGAL COMPRESSOR THRUST MODEL

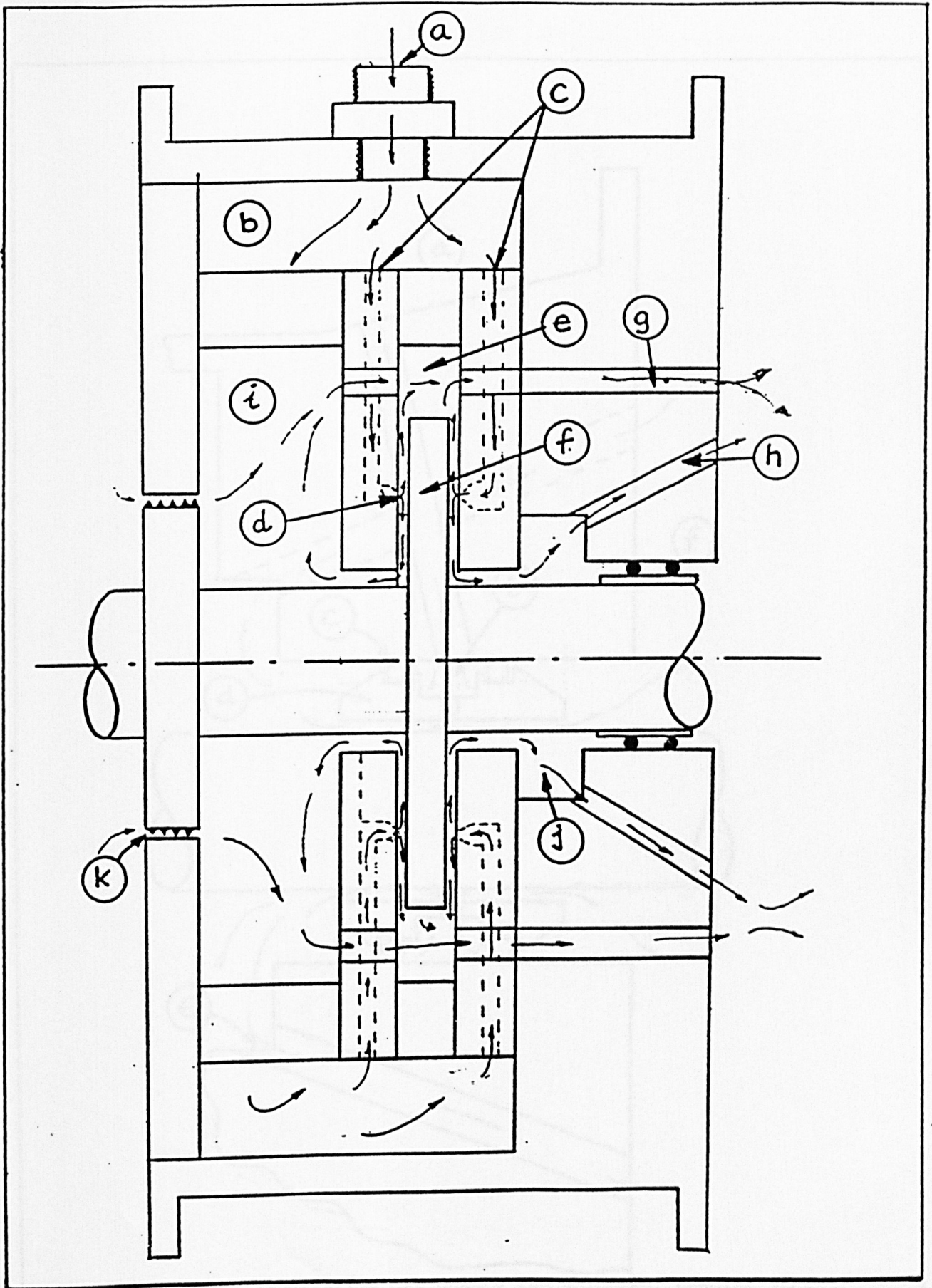


FIG. 4.13 GAS FLOW THROUGH FORWARD BEARING SUPPORT ASSEMBLY

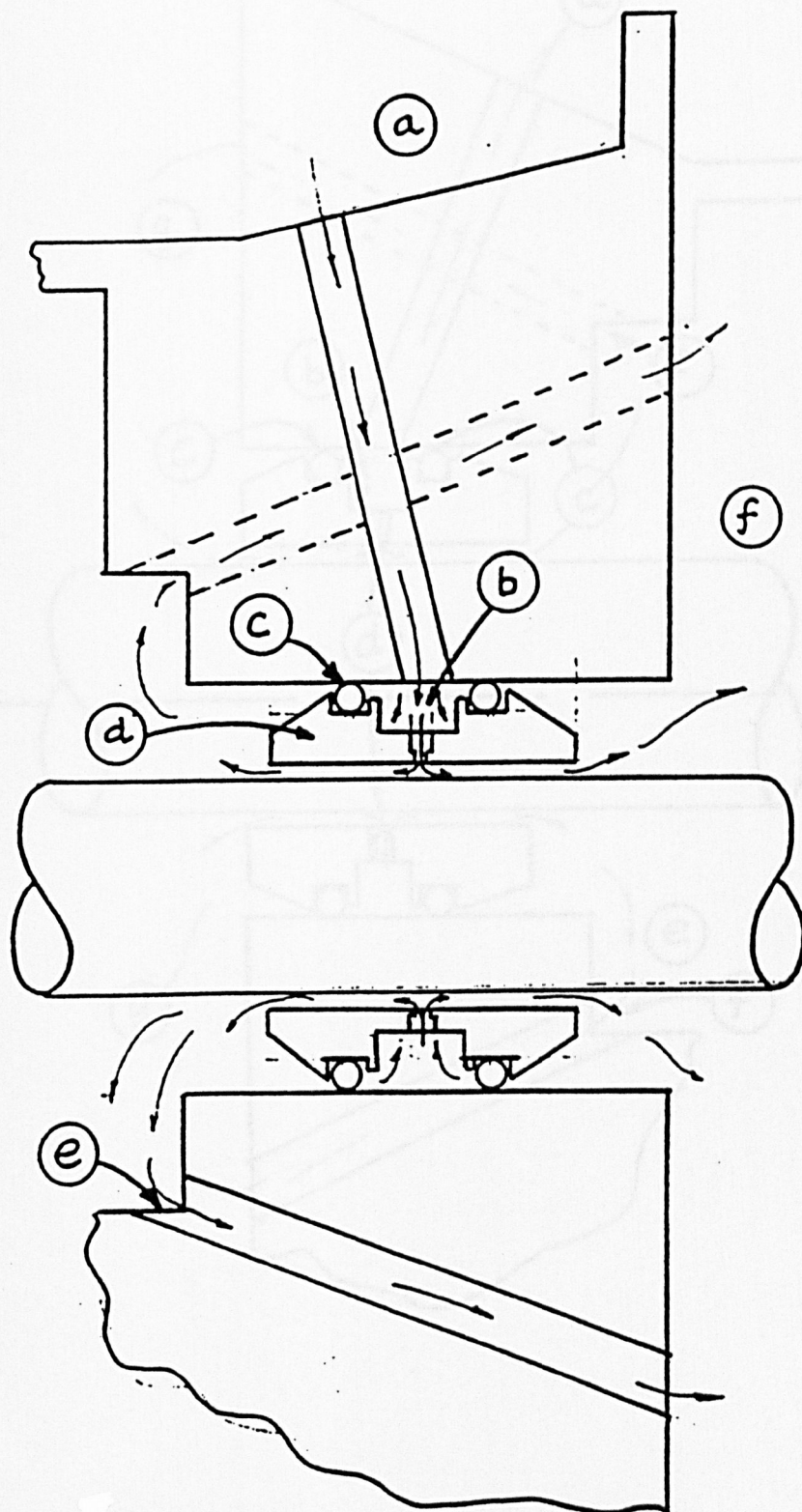


FIG. 4.14 PRESSURIZATION GAS SUPPLY AND SCAVENGE: FORWARD JOURNAL BEARING

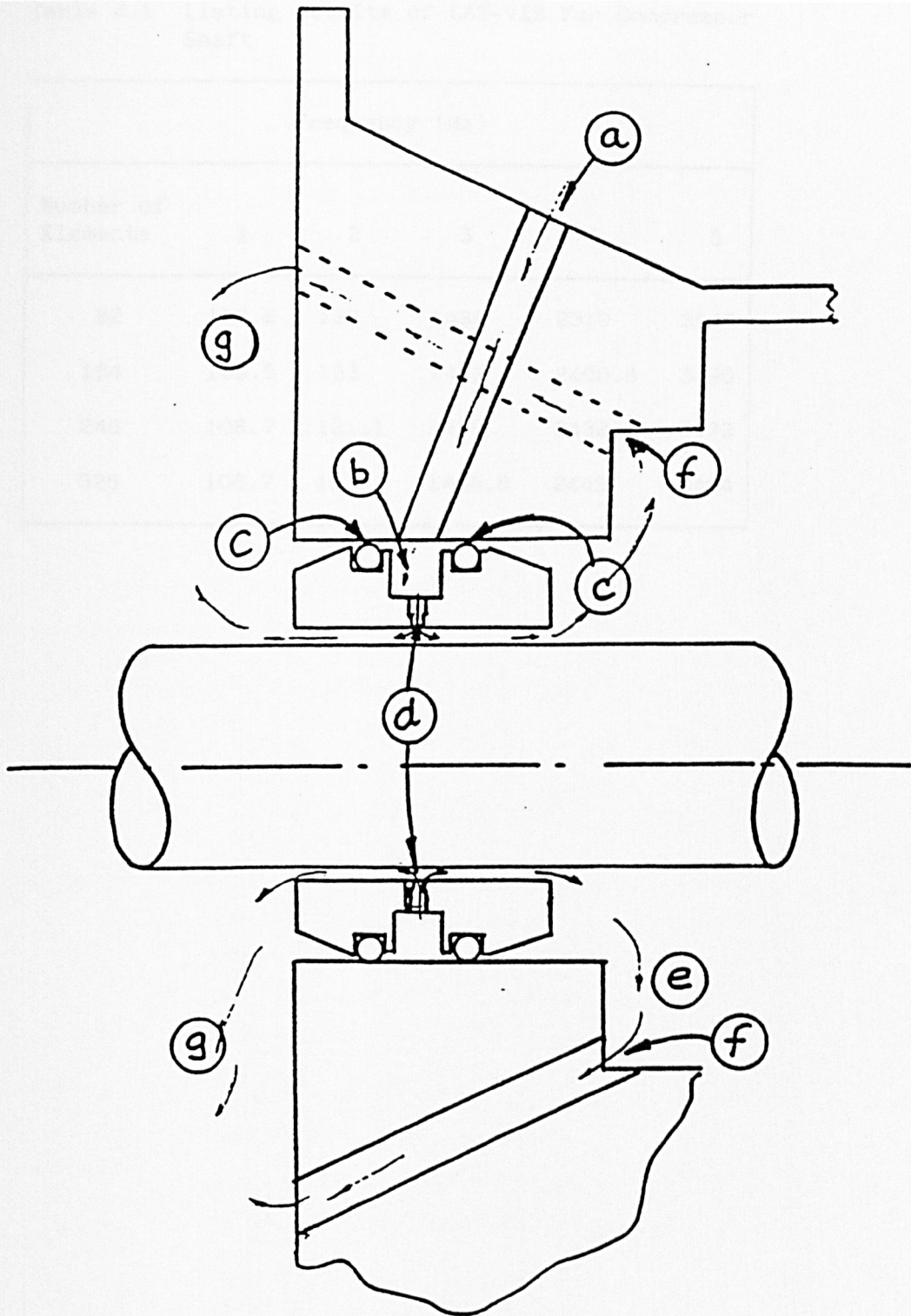


FIG. 4.15 PRESSURIZATION GAS SUPPLY AND SCAVENGE REAR JOURNAL BRG

Table 4.1 Listing Results of LAT-VIB For Compressor Shaft

Frequency (Hz)					
Number of Elements	1	2	3	4	5
82	112.6	138	1439	2310	3548
164	109.5	133	1451	2400.8	3490
246	108.7	131.1	1455	2432	3473
328	108.7	131	1456.8	2449	3464

Table 4.2 Weight and Balance Data for Shaft

Region	Weight x10(N)	Moment Arm (mm)	Moment about Stn. 0 (N-m) x10 ⁻²
a	1.753	22.5	0.394
b	2.33	56.5	1.315
c	3.58	76.5	2.742
d	36.5	126.25	46.1
e	10.75	168.5	18.12
f	46.63	210.5	98.16
g	6.57	254.5	16.73
h	34.12	296.25	101.1
j	7.872	342.0	26.92
Weight = 15.01 N		Moment = 3.124 N-m	

From Table 4.2:

- i) Centre of gravity = $M/W = 3.1161/15021 = 207.5$ mm
- ii) Mass of shaft = $W/9.81 = 1.5311$ kg (3.376 lbs)

Table 4.3 Weight and Balance Data for Shaft Assembly

Item	Weight (N)	Moment Arm (mm)	Moment (N-m)
Shaft	15.01	207.5	3.124
Impeller	0.809	22.5	0.018
Shim	0.102	49.0	0.005
Locknut/ Nut	1.6229	54.1	0.088
Spacer	0.5785	74.0	0.0428
Trust Brg.	1.8698	85.0	0.159
Rotor	11.573	212.0	2.453
Locknut	0.6374	255.5	0.163
Weight = 32.2		Moment = 6.052	

Centre of gravity = $M/W = 188$ mm
 Mass of shaft = $W/g = 3.3$ kg (7.239 lbs)

Table 4.4 Results of Journal Bearing Load Calculations

Rotational Speed $\times 10^3$	Fwd Brg. N	Rear Brg. N
0	20.0	12.3
20	21.2	13.0
40	24.3	15.0
60	29.7	18.3
70	33.2	20.4

Table 4.5 Journal Bearing Design Point Data

Journal Bearing Diameter	28 mm
Radial Clearance	0.035 mm
Bearing Eccentricity	0.7
Ambient Pressure	1 bar
Ambient Temperature	0°C; (thought to be the worst case)
Fluid Viscosity	1.75×10^{-5}
Rotational Speed	7330 rad/sec
Fwd Bearing Load	33.2 N
Rear Bearing Load	20.4 N

Table 4.6 Design Data for Aerostatic Journal Bearings

	FWD	REAR
Static Load	20 N	12.3 N
Optimum Bearing Pressure	2.98 bar	2.56 bar
Touch-down Load	31.2 N	19.3 N
Bearing Stiffness	1.42×10^6 N/m	8.8×10^5 N/m
Orifice Diameter	0.88 mm	0.96 mm
Compressed Air Consumption	0.8 CFM	0.5 CFM

Table 4.7 Design Data for Aerostatic Thrust Bearings

Gas	Air
Viscosity	2.5952×10^{-9} lbfs/square inch
Gas Constant (R)	2.47×10^5 in ² /sec ² °R
Temperature (T)	518.4°R
Outer Radius (R_o)	1.48 inches
Inner Radius (R_i)	0.55 inches
Bearing Load (W)	100 lbf

Table 4.8 Design Data for Aerostatic Thrust Bearing

Orifice Diameter	0.03 inch
Number of Orifices	16
P.C.D. of Orifices	1.8 inch
Axial Bearing Clearance	0.00085 inch
External Diameter	75 mm
Internal Diameter	32 mm

	AIR	$R_{11}(i)$	$R_{11}(ii)$
$\dot{m}$	0.1238	0.091	0.278
T_{oo}	288	273	293
T_{ol}	349	391	419
P_{oo}	1	0.4	0.73
P_r	1.5	6.0	6.0
P_{ol}	1.5	2.4	4.38
X_o	0.2209	0.183	0.3176
X_l	0.42	0.094	0.162
P_{oo}/P_{ol}	1.026	1.029	1.058
P_{ol}/P_l	1.11	39	19
T_{oo}/T_o	1.007	1.002	1.007
V_o	66.11	23.89	45.2
P_o	0.975	0.389	0.689
P_l	1.35	0.061	0.23

$$T_o/T = P_r^{\frac{\gamma-1}{\eta\gamma}}, \quad \gamma = \text{ratio of specific heats}$$

$\eta = \text{polytropic eff. assumed 60\%}$

$$X = \frac{\dot{m} \sqrt{RT}}{AP}$$

**TABLE 4.9 THERMODYNAMIC DATA FOR THRUST
BRG LOAD CALCS (3 CONDITIONS)**

CHAPTER FIVE

COMPRESSOR UNIT MANUFACTURE

5.1 Introduction

This chapter evaluates the manufacture of the compressor. Technical difficulties encountered are discussed with particular reference to surface finishes, machining tolerances and heat treatments.

Engineering drawings of the compressor, in Appendix A, contain all necessary information for the manufacture of individual components. It is not intended in this Chapter to repeat the instructions contained therein. Neither is it appropriate to describe in detail the manufacture of every component; which consisted largely of elementary machining and fitting operations. This chapter seeks only to present an overview of the manufacturing process relating to each of the major sub-assemblies and to present the reasons for certain design choices where necessary. A description of the compressor assembly procedure is given in Appendix I.

Recommendations are made concerning the design of future prototype compressors.

5.2 Drive Shaft Manufacture

After preliminary machining the drive shaft, (Pt.No.PB15032/1 - NB part numbers refer to the drawing numbers contained in Appendix A), one millimeter was left on all dimensions for final machining after the material had been vacuum stress relieved at 700°C. The shaft was finished turned leaving 0.25 mm on all diameters for grinding. Following a hardening treatment at 840°C, gas quenching and tempering at 200°C the shaft was ground to size.

5.2.1 Manufacturing Tolerances

A general machining tolerance of 0.12 mm, (0.005 inches), was specified for non-critical dimensions. This tolerance is usual in most general machining work when greater accuracy cannot be economically justified.

The diametrical tolerance of the journal surfaces was set at

+0.0013 mm, -0.00 mm whilst the tolerance on the bush diameter was set at +0.00mm, -0.0013mm. This ensured a maximum error of 0.0026 mm, (0.0001 inches) of the bearing-bush clearance. This degree of tolerance was unrealistic because the capacity of precision cylindrical grinders is ± 0.0001 inches, (0.00254 mm), Hannan⁷⁷. A tolerance of 0.0013mm was placed on the concentricity of diameters.

Figure 5.1 shows the results of measurements taken of a finished drive shaft. Nomenclature used in Figure 5.1 are listed in Table 5.1. It can be seen from Figure 5.1 that the worst case of non-concentricity is when the journal bearing surfaces are used as the reference stations, (Z = 78, 83, 240, 250mm), is 0.00372mm which is more than the desired 0.0013mm value. The error occurs on the impeller end of the drive shaft and therefore did not Affect balancing. Referring to Figure 5.1 it is seen that the out-of-roundness, (E), is within the desired tolerance of the majority of the shaft except for the impeller end, which was thought to have been caused by an excessive bending during grinding. To alleviate this problem in the future fine cuts should be taken.

Figure 5.2 shows the variation in roundness of the forward and rear journal bearings, (at stations 78mm and 240mm), achieved during manufacture. It can be seen that the roundness of the bearings is within 0.0002mm which is considered to be more than satisfactory, Hannan⁷⁷.

5.2.2 Surface Finish

The surface finish of dynamic components has a direct effect on their fatigue life. The highest values of endurance stress are obtained when the surface of the component is perfectly smooth, polished and free from scratches and other stress raising features. For this reason it was proposed that all shaft diameters should be finished by grinding.

Maleev⁷⁸ suggested that slight scratches produced by grinding can reduce the endurance stress by as much as 12% and localized surface burning which sometimes occurs is thought to reduce this further. The precision grinding process should therefore involve shallow cutting depths, probably less than 0.0001 inches, with large quantities of cutting fluid and the correct selection of wheel is important.

The general surface finish of the shaft was selected at 0.2 micron CLA whilst the finish of the journal surfaces was set at 0.1 micron CLA. This is well within the capacity of the cylindrical grinding process, Hannan²². The finish of the journal faces was improved by lapping.

The surface finish specification was considered to be the minimum acceptable. In the event the surface finish of the journal faces was well within the required value. Figures 5.3 and 5.4 show the results of surface roughness measurements of the both journal faces. It is seen from the tables in the top right-hand corner of both Figures 5.3 and 5.4 that the measured finish was 0.045 and 0.048 micron CLA, (Ra), for the rear and forward journal surfaces respectively.

5.2.3 Drive Shaft Heat Treatment

In order that the machining stresses could be kept to a acceptable level, thus minimising distortion, cutting rates and feeds were kept down and the shafts were annealed after each machining stage and before grinding. To keep the distortion to a minimum during the heat treatment processes the shaft was hung vertically to ensure that the weight would be supported symmetrically about the longitudinal axis.

The preliminary heat treatment processing was carried out by D.P.Lush-Scientific Heat Treatment. The shaft was subjected to vacuum annealing at 700°C, (British Standards). The shaft was found to have distorted during the heat treatment and a thick oxide scale gave the shaft the appearance of being annealed in a conventional furnace. Fortunately the thickness of metal left after the initial turning was sufficient to allow the distortion to be machined out. Following the final turning operation the shaft was dispatched to Torvac Services Ltd where it was vacuum hardened at 840°C, gas quenched and tempered at 200°C.

Following the final heat treatment the shaft run-out was measured by supporting the shaft between centres and measuring the displacement of a the shaft halfway along its length using a dial-test-indicator. The shaft was found to 0.1mm out of true which was within the grinding allowance of 0.25 mm.

5.2.4 Balancing

The shaft assembly was balanced by Allen Gears. The best balance achieved was 3×10^{-4} oz-inches, (2.2×10^{-4} N-m in both planes of balance. This result exceeded the requirements of the design of 3.9×10^{-4} N-m, recommended by MTI⁴⁴. Figure 5.5 shows the balance sheet supplied by Allen Gears.

5.3 Journal Bearing Bushes

The bushes were manufactured from EN 24 steel. Following the initial turning the bushes were annealed at 700°C. After final turning 0.25 mm was left on all major diameters for grinding. Following the final grinding process they were dispatched to Abbey Heat Treatments of Earlsfield, London for Sulfinuz treatment.

It was found that the Sulfinuz treatment had caused slight distortion which producing some ovality. Figure 5.6 shows measurements of ovality taken by Rank-Taylor-Hobson. It is seen from Figure 5.6 that the maximum out-of-roundness of the forward and rear bushes is 7 micrometres, (0.0005 inches), and 10 micrometres, (0.0003 inches), respectively. The amount of distortion is significant when compared to the 70 micrometres diametrical clearance, (0.0028 inches). The effect would be to produce a more stable bearing although load capacity would be reduced.

A further disadvantage of the Sulfinuz treatment is that it produces a thick hard scale on surfaces. It was found that the oxide scale could be removed from the surface relatively easily with a knife but the bearing faces required cleaning and polishing afterwards. This was carried out using a fine grade of wet-and-dry emery paper and by lapping each bush on a ground bar using a fine grit paste. Surface roughness measurements showed that the resulting surface was not as good as that achieved for the journal faces. Figures 5.7 and 5.8 show the results of surface roughness measurements carried out by Rank-Taylor-Hobson. The results shown in Figure 5.8 are representative of the majority of the surface on both bushes which had a surface roughness measurement of 7.5 microinches CLA, (Ra). The data shown in Figure 5.7 is representative of the roughest areas found on both bushes which gives a surface roughness measurement of 25.3 micrometres CLA, (Ra).

In the initial bearing design the jewelled gas ports were held in place by cementing them first into brass holders and then into prepared holes in the bush body. The later design

did away with the brass holders and the jewels were cemented directly into prepared holes in the bush body.

5.4 Thrust Bearing

The thrust bearing, (PD15038/1), was manufacture from a EN24T steel. After the initial turning the disc was annealed at 700°C and then turned to size, leaving 0.004 inch allowance for grinding. Following heat treatment processes similar to those used for the shaft and described in Section 5.2.1, the disc was finished by cylindrical grinding. To enable this process to be carried out the disc was mounted on a specially prepared mandrel. The outer diameter of the disc was finished by grinding ensuring the concentricity of the outer diameter of the disc with the central hole.

The thrust bearing spacer, (PD15038/2), was made from mild steel bar stock. It was important for the spacer not to distort after manufacture. To minimise warping the spacer was initially turned to the finished dimension plus 1 mm and annealed before being finish turned and ground to size.

The two thrust plates, (PD15039/1), were manufactured from mild steel following the same processes as the spacer. After grinding the thrust plates were Sulfinuzed. When the scale was cleaned from the bearing surfaces it was found that both plates had suffered slight distortion. The gas port jewels were fitted to brass holders and cemented into prepared holes in the thrust plate face. These holes and their feeder passages were bored after the Sulfinuz treatment had been carried out because of the problem of surface oxidization.

5.5 Forward and Rear Housing

The bearing housings, (PB 15035 and PB 15033), were manufactured from a mild steel bar stock. Because of the amount of metal to be removed during manufacture it was necessary to fully anneal the housings after initial machining.

The principle aim was to ensure that the mounting face for the thrust bearing and the impeller shroud were perpendicular to the longitudinal axis of the housing. It was also important that the journal bearing bush seats were concentric with the longitudinal axis of the housings and in

alignment with each other. These requirements were important because of the degree of tolerance on individual components and the effect on bearing alignment.

The tolerances in alignment were achieved by machining the motor case mounting face and then mounting the housing on a prepared fixture which allowed the bearing bush seats to be turned and ground. This procedure allowed the thrust bearing mounting face and the forward journal bearing seat to be machined at one setting. Both housings were mounted on the same fixture during machining.

5.6 Motor Casing

The motor casing, (PB 15037/1), was manufactured from two 9 inch O/D and 6 inch I/D by 12mm thick mild steel blanks, (BS 4360 grade 43A), and an eight inch length of of HF seamless mild steel tube, (6.625 inch O/D and a 0.625 inch wall thickness).

After preliminary machining the flange blanks were welded to the tube section and the assembly was annealed to remove stresses and to minimise subsequent distortion. The casing was then finish turned and the flanged mounting faces were ground so that the two surfaces were parallel. The bearing housing fixing holes and other portings were then bored. The casing was then mounted on the same fixture used for the bearing housings and the motor stator seating was machined ensuring that it was perpendicular to the mounting faces and concentric with the longitudinal axis of the casing.

5.7 Conclusions

From the manufacture of the compressor unit it was concluded that the tolerances specified were achievable. The only exception to this was the concentricity of the impeller end of the drive shaft which had a maximum run-out of 3.7 micron compared with the 1.3 micron specified. This error is thought to result from excessive bending moment produced by grinding.

The fatigue life of components is affected by surface finish. Although fatigue failure of the drive shaft was unlikely, according to the stress calculations described in Chapter 4, experience has shown that the failure of lightly loaded components is possible. This is due to the effects of local stress raisers, such as scratches, surface defects and

localized burning caused by poor grinding technique. The overall surface finish of the drive shaft was specified at 0.2 micron CLA and 0.1 micron CLA in the region of the journal bearings. Surface roughness measurements following manufacture showed results of 0.045 micron and 0.048 micron CLA for the forward and rear journal bearing surfaces respectively.

The heat treatment of the drive shaft was discussed in Section 5.2.3. It was concluded that the drive shaft should be annealed after preliminary machining and again before grinding when harding and tempering were also carried out. The aim was to produce a shaft that which would not distort due to residual machining stresses. During heat treatment the shaft was hung vertically in the furnace to prevent distortion. Other major components were also annealed between preliminary and final machining to avoid subsequent distortion. It was important to keep distortion to a minimum in order to ensure alignment of bearings, drive shaft assembly and stator. The heat treatment of the bearing assemblies was discussed in Sections 5.3 and 5.4.

To prevent excessive wear during start-up and shut-down the bearing surfaces were Sulfinuz treated. The process, discussed in Chapter 4, impregnates a steel surface with sulphur which acts as a solid lubricant to prevent binding. The process is carried out at 540°C to 600°C and was found to produce some distortion of bearing surfaces, particularly the thrust plates, and to leave a thick oxide coating which was difficult to remove without spoiling the bearing surface. It was concluded that an alternative low temperature sulphur treatment should be used in future such as Sulf BT which is carried out at 180°C to 200°C, thereby minimising distortion, Neale⁷⁴.

The drive shaft assembly was balanced and the moment achieved was 2.2×10^{-6} N-m which exceeded the requirement of 3.9×10^{-6} recommended by MTI⁴⁴ discussed in Chapter 4. A balance report is included in Figure 5.5.

Following the manufacture of the compressor unit test facilities were constructed and test-runs carried out to prove the concept. These experiments are described and evaluated in Chapter 6.

Z	O	⊙	E	∠	ΔR
MM	UM	UM	UM	DEG	UM
* 78.0	0.70	0.00	0.00	0	0.13
* 83.0	0.45	0.00	0.00	0	-0.13
162.0	0.94	6.54	3.27	158	0.16
240.0	0.37	18.39	9.19	115	-0.32
250.0	0.57	20.19	10.10	115	-0.82
292.0	1.49	27.39	13.69	114	-4.55
343.0	3.72	33.73	16.86	109	-3.53

Z	O	⊙	E	∠	ΔR
MM	UM	UM	UM	DEG	UM
78.0	0.70	7.10	3.55	-51	1.19
83.0	0.45	7.70	3.85	-53	0.92
162.0	0.94	13.59	6.80	-84	1.21
240.0	0.37	9.48	4.74	-74	0.73
250.0	0.57	9.03	4.52	-76	0.23
292.0	1.49	7.40	3.70	-82	-3.50
343.0	3.72	7.47	3.73	-67	-2.48

Z	O	⊙	E	∠	ΔR
MM	UM	UM	UM	DEG	UM
* 78.0	0.70	0.00	0.00	0	0.13
* 83.0	0.45	0.00	0.00	0	-0.13
162.0	0.94	6.54	3.27	158	0.16
240.0	0.37	18.39	9.19	115	-0.32
250.0	0.57	20.19	10.10	115	-0.82
292.0	1.49	27.39	13.69	114	-4.55
343.0	3.72	33.73	16.86	109	-3.53

Z	O	⊙	E	∠	ΔR
MM	UM	UM	UM	DEG	UM
* 78.0	0.70	0.30	0.15	116	0.42
* 83.0	0.45	0.29	0.14	-65	0.15
162.0	0.94	6.47	3.23	-108	0.44
* 240.0	0.37	0.33	0.17	-50	-0.04
* 250.0	0.57	0.32	0.16	130	-0.53
292.0	1.49	2.60	1.30	107	-4.26
343.0	3.72	4.36	2.18	64	-3.25

Z	∠
MM	UM DEG
78.0	3.5 129
83.0	3.9 127
162.0	12.5 -80
240.0	17.5 -78
250.0	14.5 112
292.0	20.9 -78
343.0	24.2 -77

* Reference Stations

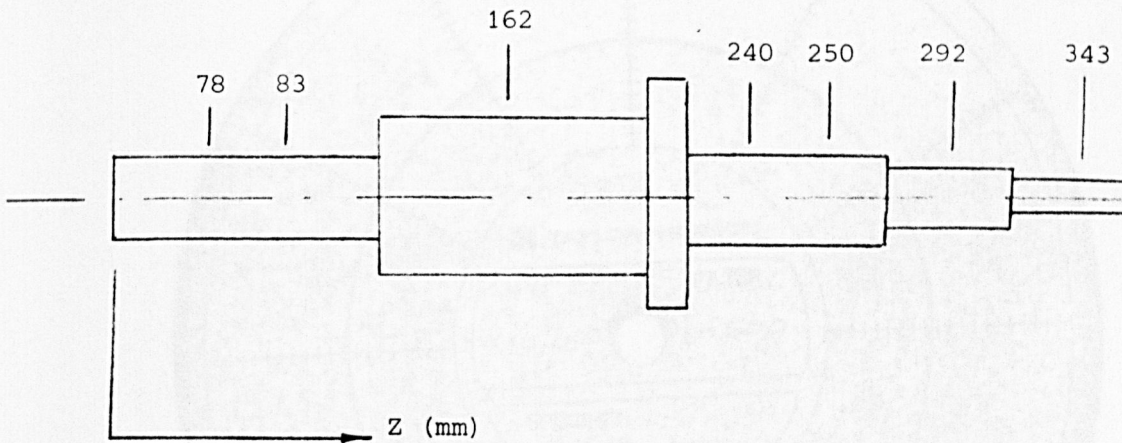


Fig. 5.1 Results of Geometrical Measurements of Drive Shaft Assembly taken using a Talycentre

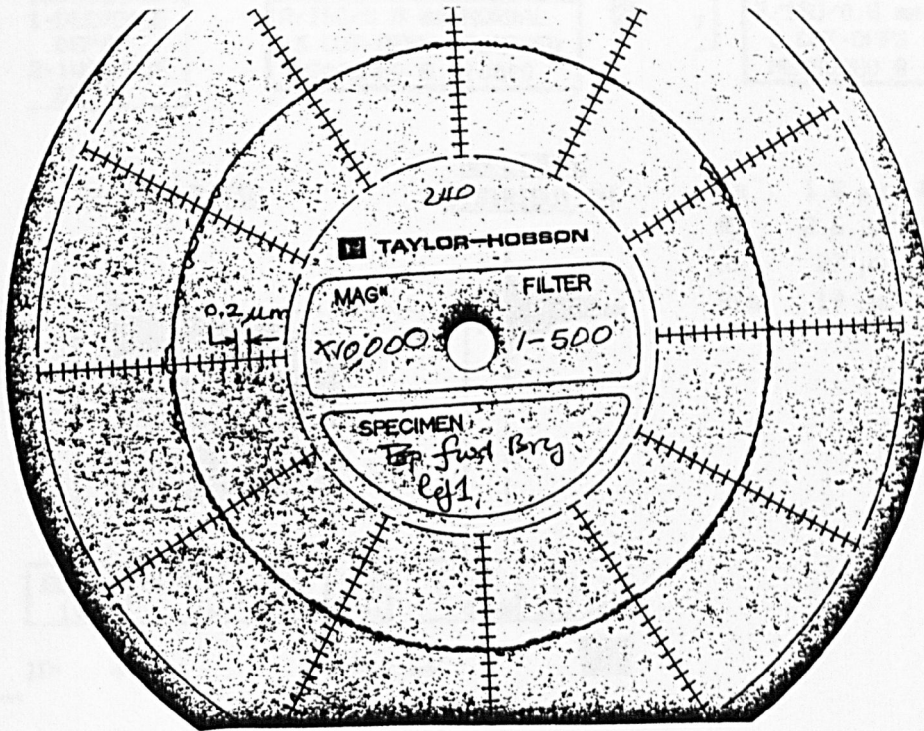
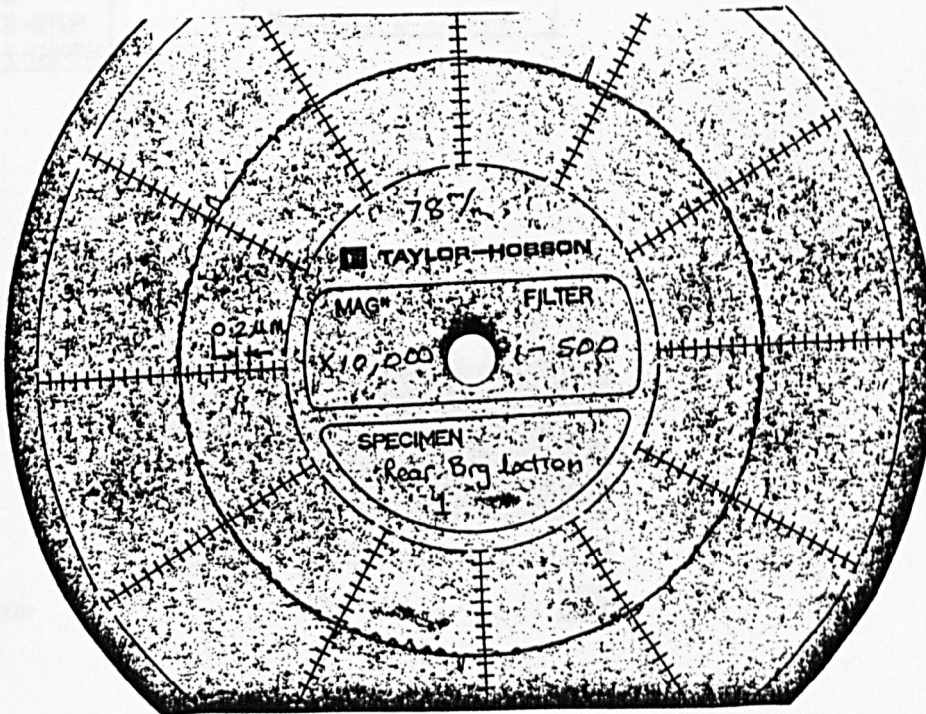


FIG. 5.2 JOURNAL ROUNDNESS

- (a) FORWARD BEARING
- (b) REAR BEARING

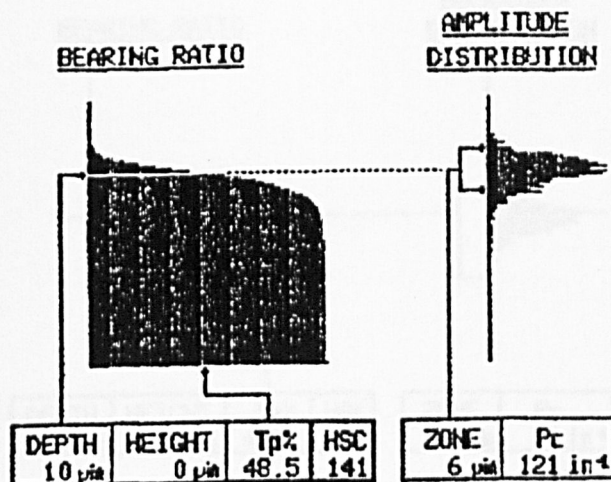
1-DECREASE
DEPTH
2-INCREASE
DEPTH

R/ISO/0.8 mm NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$

2

T

R/ISO/0.8 mm NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$



Ra	1.8 μ in	Rt ₁	14 μ in
Rq	2.6 μ in	Rt ₂	13 μ in
Rv	37 μ in	Rt ₃	17 μ in
Rtm	19 μ in	Rt ₄	37 μ in
		Rt ₅	16 μ in

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9-MAR-84

E-T

1-<<<
2->>>
3-STOP
4-GRAPH

R/ISO/0.8 mm NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$

3

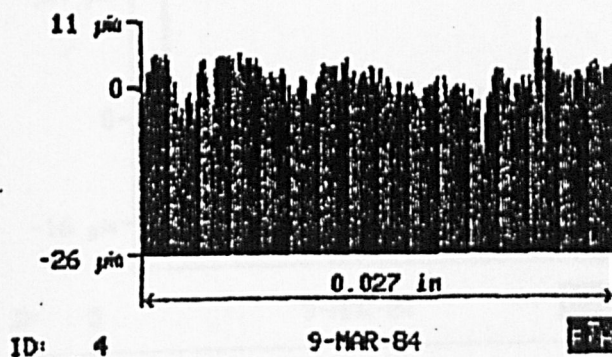


FIG. 5.3 REAR BEARING SURFACE ROUGHNESS ON SHAFT

1-DECREASE
DEPTH
2-INCREASE
DEPTH

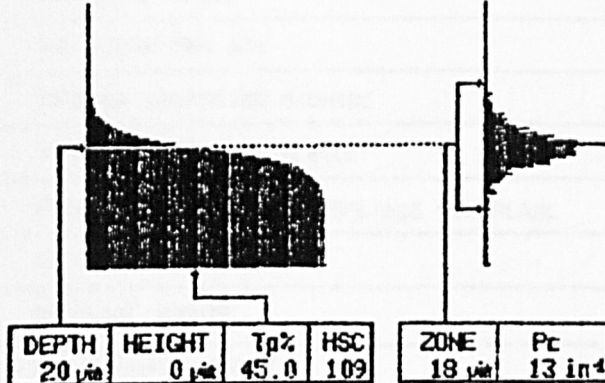
R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000

2

T

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000

BEARING RATIO

AMPLITUDE
DISTRIBUTION

Ra	1.9 μ m	Rt ₁	36 μ m
Rq	2.7 μ m	Rt ₂	15 μ m
Rv	36 μ m	Rt ₃	21 μ m
Rt _m	21 μ m	Rt ₄	13 μ m
		Rt ₅	22 μ m

ID: 5

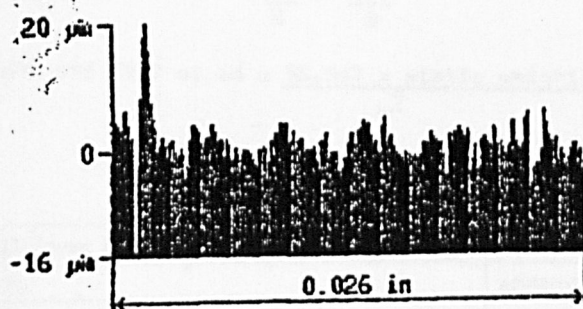
9-MAR-84

FTH

1-<<
2->>
3-STOP
4-GRAPH

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000

3



ID: 5

9-MAR-84

FTH

FIG. 5.4 FORWARD BEARING SURFACE ON SHAFT

**SCHENCK ELECTRODYNAMIC
BALANCING MACHINE REPORT**

WORKING NUMBER:	SET NO	P863972	
COMPONENT;		ROTOR SHAFT	
COMPONENT DRG NO;		PB 18033	
COMPONENT WEIGHT		3.4	kg
COMPONENT SPEED		70 000	RPM
BALANCING DRG NO:			
SCHENCK BALANCING MACHINE		H30 D	
TOTAL RESIDUAL UNBALANCE		50	gmm
ADMISSIBLE RESIDUAL UNBALANCE PER PLANE		308	gmm
LH PLANE RADIUS		22	mm
RH PLANE RADIUS		22	mm
MAX BALANCE WEIGHT		LH 0.014 RH 0.014	g
BALANCING SPEED		3346	RPM
<u>READING OF INITIAL UNBALANCE</u>			
LH PLANE		1	g
RH PLANE		0.25	g
<u>RESIDUAL UNBALANCE AFTER COMPENSATION</u>			
LH PLANE		0.01	g
RH PLANE		0.01	g
<u>REMARKS</u> State quality used and show calculations:- (BS 5265 Part 1 1979) (ISO 1940 1973) Grade G BS 3853 1966 $\frac{4W}{N}$ or $\frac{12W}{N}$ $AP1\ 613\ 1977\ oz\ in = \frac{56,347 \times static\ weight}{N^2}$		OPERATOR R.J. Lawton. INSPECTION <i>[Signature]</i> DATE 22/6/84	
PREPARED: (name & title) <i>G. L. [Signature]</i> J & I Sec		APPROVED: (name & title)	
APPROVED:		APPROVED:	
APPROVED:		O.C. CONCURRENCE: <i>[Signature]</i> Qual Engineer	
AMENDMENTS INCORPORATED: 1		SUPERSEDES:	
 ALLEN GEARS -APE LTD PERSHORE, WORCS.		ISSUE 3 DATE 1.11.93 SECTION: Fig 4 SHEET: OF 5	

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1.11.83

FIG. 5.5 SHAFT ASSEMBLY BALANCE SHEET

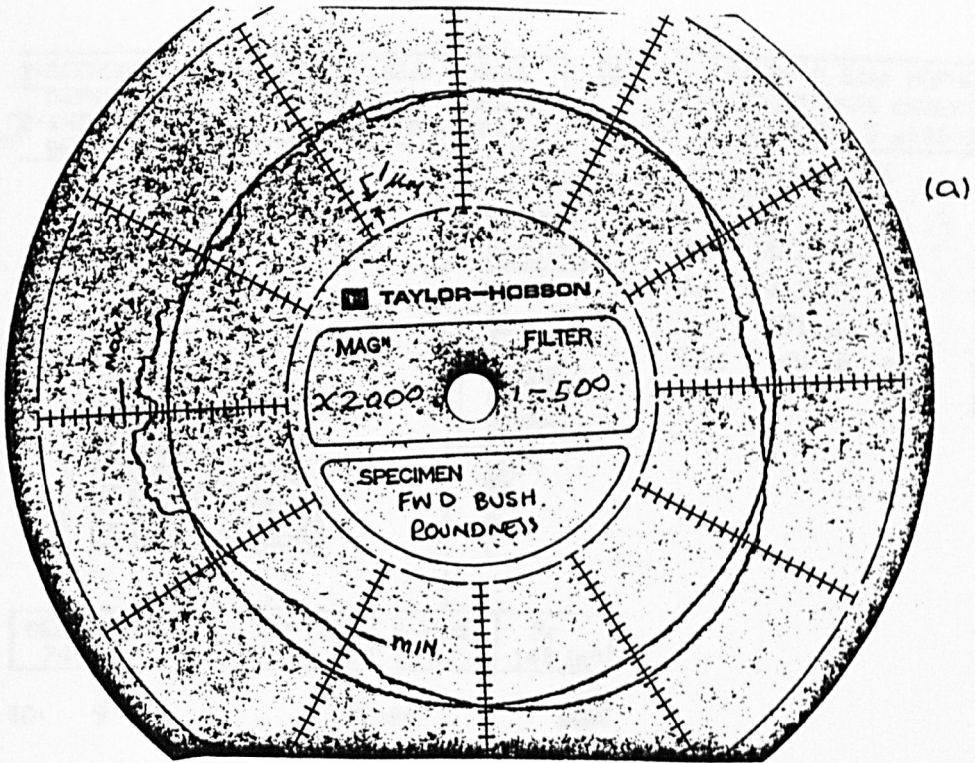
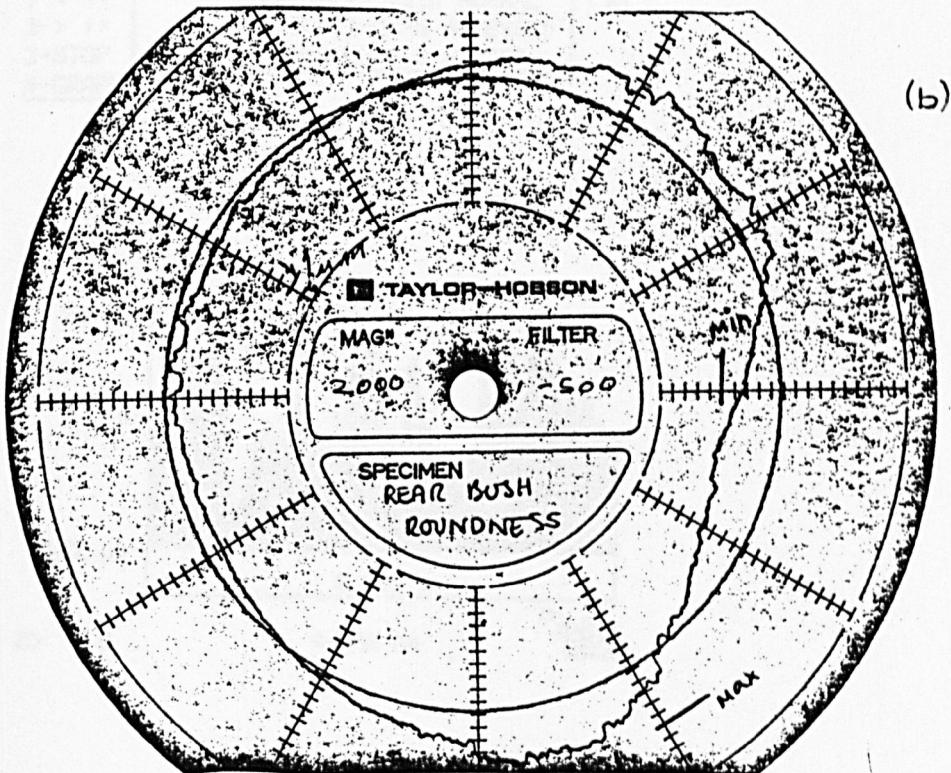


FIG. 5.6 BUSH ROUNDNESS

(a) FORWARD BUSH

(b) REAR BUSH

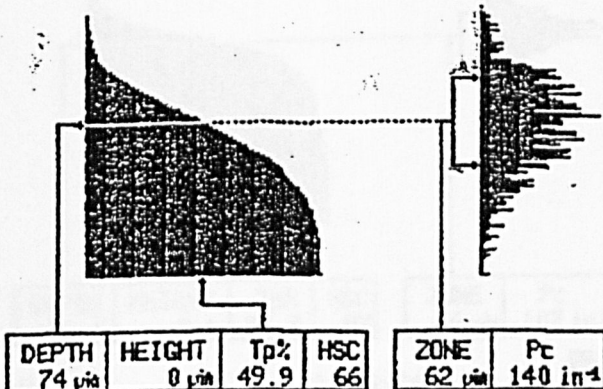
1-DECREASE
DEPTH
2-INCREASE
DEPTH

187
R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$

BEARING RATIO

AMPLITUDE
DISTRIBUTION



ID: 9

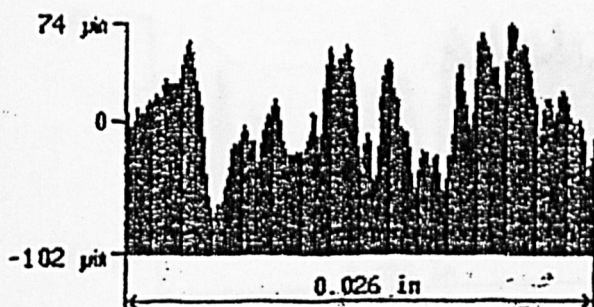
9-MAR-84

FIG

Ra	25.3 μ m	Rt ₁	128 μ m
Rq	31.2 μ m	Rt ₂	151 μ m
Rv	171 μ m	Rt ₃	145 μ m
Rtm	145 μ m	Rt ₄	128 μ m
		Rt ₅	171 μ m

1-<<
2->>
3-STOP
4-GRAPH

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ $\times 10000$



ID: 9

9-MAR-84

FIG

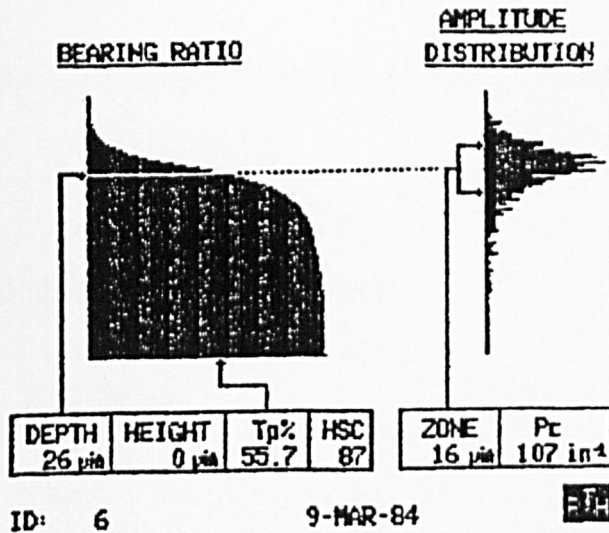
FIG. 5.7 SURFACE ROUGHNESS MEASUREMENTS FOR FORWARD BUSH MEASURED TOTALLY ON SULFUZ

1-DECREASE
DEPTH
2-INCREASE
DEPTH

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000

2

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000



Ra	7.5 µin	Rt ₁	72 µin
Rq	10.7 µin	Rt ₂	36 µin
Rv	79 µin	Rt ₃	79 µin
Rt _m	52 µin	Rt ₄	37 µin
		Rt ₅	37 µin

1-<<
2->>
3-STOP
4-GRAPH

R/ISO/0.03in NORMAL
5 CUT-OFFS ASSESSED
MEASURED @ x10000

3

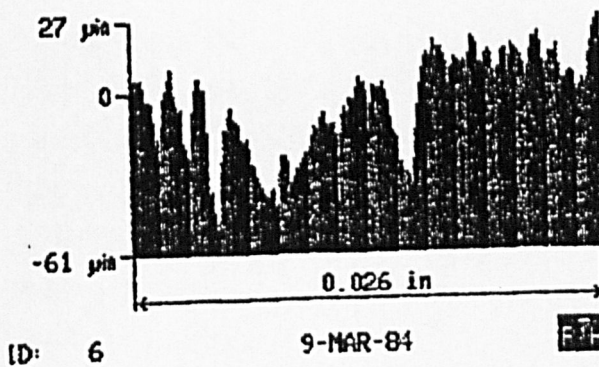


FIG. 5.8 SURFACE ROUGHNESS MEASUREMENTS, FORWARD BUSH, HALFWAY ALONG SURFACE

Nomenclature For Straightness

Z	vertical axis reference
O	roundness
⊙	concentricity with reference axis = 2x eccentricity (E)
E	eccentricity with reference axis
∠	Angular reference to machine (witness mark on shaft)
ΔR	maximum radial displacement with reference to reference axis
//	runout

Nomenclature for surface finish

Surface finish measurements carried out on Talysurf

Ra	Represents CLA value of surface roughness
Rq	Represents RMS value of surface (texture) roughness
Ry	Maximum peak to trough value over sample length
Rt _a	Mean peak to trough high over sample length
Rt ₁ -Rt ₅	Maximum peak to trough measurement in each cut off length
Cut of length = 0.03 in	
Sample length = 0.15 in	

**TABLE 5.1 STRAIGHTNESS MEASUREMENTS CARRIED OUT ON A TALYCENTRE.
NOMENCLATURE FOR STRAIGHTNESS**

CHAPTER SIX

TESTING THE COMPRESSOR UNIT

6.1 Introduction

This chapter describes and evaluates a series of experiments aimed at quantifying the mechanical and electrical performance of the prototype compressor design. Two series of experiments were carried out. The first providing information on the performance of the motor, drive shaft and bearing system. These tests were carried out with the compressor unit operating in air and showed short falls in design and resulted in some modifications. The second series of experiments examined the operation of the compressor in a refrigerant circuit. This chapter evaluates these experiments. Recommendations are made concerning the design of the compressor unit and the experimental apparatus with regard to the control and data aquisition.

6.2 Air Tests

The object was to prove the compressor would operate satisfactorily in an air environment prior to testing in a refrigeration system where access to instrumentation and components would be difficult. The aims of the preliminary experiments were:

- i) To prove the mechanical integrity of the drive shaft assembly.
- ii) To investigate the dynamic performance of the drive shaft and bearing assemblies.
- iii) To prove the electrical power system and the motor assembly.

6.3 Apparatus

Figure 6.1 shows a schematic layout of the test apparatus used for the preliminary experiments. The compressor, (item 1), was connected to the central test site compressed air supply, (item 13), via a control valve, (item 14). This air supply was used to drive the compressor wheel as a turbine.

Compressed air for the bearing system was supplied by a portable air compressor, (item 8 in Figure 6.1), with a

maximum pressure of 150 psia and a maximum flow rate of 8 cfm. The compressed air was stored in an accumulator where it was cooled and water condensate removed. The compressed air was then piped via a dehumidifier coil, (item 7), and filter, (item 6), to a manifold, (item 2), and distributed to each bearing.

The dehumidifier consisted of a tank filled with a water-glycol solution. The dehumidification coil was manufactured from copper tubing with a valve position at its lowest point for draining water condensate. The water in the tank was cooled using a vapour compression refrigerator consisting of a compressor, (item 10), an air cooled condenser, (item 11), a capillary tube, (item 12), and a plate type evaporator, (item 9).

Compressed air was fed to each bearing through 1/4 inch diameter copper pipe, (items 3,4 and 5), via a needle valve to control the flow coming from the manifold, (item 2). The pressure to each bearing was measured using Bourdon pressure gauges, (item 16).

Compressor speed was measured using a magnetic type pick-up, (item 7 in Figure 6.3), which was referenced to two flats machined into the drive shaft at the end opposite to impeller. The rotational frequency of the drive shaft was registered on a Gould 35 MHz Timer/Counter type TC320, (Figure 6.1,item 17).

Drive shaft displacement was measured using two eddy current type displacement transducers shown, (Figure 6.1,items 19 and 20). The output from the transducers was displayed on a Tektronix 544,4 channel oscilloscope.

Figure 6.2 shows a photograph of the experimental apparatus. Figure 6.3 shows the compressor mounted on the test-bed. Clearly seen in Figure 6.3 are the displacement probes located in the forward and rear bearing housing, (items 2 and 3), and the speed probe, (item 1). Also visible in Figure 6.3 are the additional location points for the displacement probes, (items 4 and 5).

Figure 6.4 shows an end view of the compressor with the end cap removed. The end of the drive shaft and the rear bearing bush and provision for gas venting are also shown along with the displacement probe, (item 1), and the speed probe, (item 2). Figure 6.5 shows a view of the compressor from the opposite end. Visible in Figure 6.5 are the compressor wheel

and the bearing gas supply pipes for the forward journal and thrust bearings.

An interesting feature of the apparatus was the design of a sensing system for detecting touchdown when a bearing contacted the drive shaft, ie when the lubrication was no longer effective. Figure 6.6 shows a schematic representation of the system. Each bearing had a low voltage lamp connected to it. The other side of the lamps were connected to the negative terminal of a 9 volt battery. The positive terminal was connected to the drive shaft via a carbon brush. When the shaft was rotating and the bearing system was operating satisfactorily the shaft was insulated by gas films from the bearing surface. Journal bearings were insulated from their supporting structure by the rubber 'O' seals on which they were mounted. The thrust bearing plates were not insulated from the body of the compressor. Contact between the drive shaft and a bearing surface caused the associated lamp to light, indicating that rubbing contact was taking place.

6.4 Instrument Calibration

6.4.1 Displacement Probes

Each probe was fitted so that the clearance between the probe tip and the drive shaft was 0.004 inches using a feeler gauge and with the journal bushes pressurized. The displacement probes were connected to an oscilloscope and the display was calibrated using feeler gauges.

Because only two displacement probes were available it was decided to locate one probe at each end of the shaft and to measure displacement in a single plane. It was felt that bearing instability would be detected and whirl amplitude could be measured in this way.

6.4.2 Speed Probe Calibration

Using a feeler gauge the clearance between the drive shaft and the magnetic type transducer was set to 0.010 inches. The speed of the drive shaft was accurately set at various speeds up to 6000 rpm using a stroboscope referenced to a white dot painted on the end of the shaft. The frequency counter reading was compared with the stroboscope reading.

The output of the frequency counter and stroboscope were found to be equal up to the limit of the stroboscope which was 6000 rpm.

6.5 Test Run Number 1

The aims of this experiment were;

- i) To confirm that the hydrostatic lubrication system operated satisfactorily.
- ii) To confirm the mechanical integrity of the compressor drive shaft assembly.

6.5.1 Method

The compressor was assembled, following the procedure described in Appendix I, and mounted on the test bed, (Figure 6.3). Air supplies and instrumentation leads to the compressor were connected and electrical continuity tests were carried out. The bearing air compressor was operated until the accumulator was charged and the dehumidifier was set running and water condensate drained from the coil and the air compressor accumulator.

Journal and thrust bearing pressures were set at 50 psig and the drive shaft was rotated by hand to ensure that there was no rubbing contact between the shaft and bearings. The touch-down indicator system was then checked by switching the flow of air to each bearing in turn observing that the associated lamp was illuminated. The drive-air valve was slowly opened causing the drive shaft to accelerate. The rotational speed was initially monitored using a stroboscope in order to calibrate the reading of the frequency counter, (see Section 6.4.2),. Once the preliminary checks were completed the drive air valve was opened and the speed of the compressor was allowed to increase to 20,000 rpm where it was stabilized and the vibration level was measured using the displacement probes. No vibration was detected at the forward bearing and there was slight signal interference from the rear bearing transducer.

Rotational speed was increased from 20,000 rpm to 70,000 rpm in 5,000 rpm increments over a period of 55 minutes by accelerating slowly between each stage and holding the speed constant for five minutes before taking readings of

presssures and vibration amplitude.

Deceleration was carried out over a period of 10 minutes by progressively closing the drive air valve. Bearing pressures and vibration amplitude were once again recorded at each stage.

The above procedure was repeated with bearing air pressure of 45 to 20 psig in 5 psig steps.

6.5.2 Discussion

Both shaft and bearings appeared to operate satisfactorily with air pressures ranging from 50psig to 20 psig in 5 psi increments.

During the experiments there were fluctuations in bearing air pressure, (+/- 5 psi), caused by intermittent compressor operation. Accelerating the drive shaft slowly from 20,000 rpm to 70,000 rpm and then decelerating from 70,000 rpm to zero rpm whilst monitoring the vibration of the assembly demonstrated that there were no critical vibrations within the operating speed range of the compressor.

The first time the compressor was assembled it was found that the drive shaft would not turn freely even with bearing pressures in excess of 100 psig. The reason for this was shown to be misalignment between the forward and rear journal bearing bushes. During assembly there was significant resistance between the 'O' seals and bearing housings which made positioning difficult. The fitting of the bushes became easier when the depth of the 'O' seal seating was increased by 0.005 inches, giving more clearance to the bushes in the housings. Freon 11 liquid was used to lubricate the 'O' seals during assembly but this did not alleviate the problem completely and as they were thought to be rolled up and sat askew within the seating, causing the bearings to be misaligned. A re-design of the bearing bush was necessary to overcome this problem.

A further problem encountered during this preliminary test was the signal instability from the rear bearing displacement transducer. It was thought that this interference was caused by positioning the probe close to the machined flats in the end of the shaft. It was decided to reduce the clearance between the probe and the shaft surface from 0.004 inches to 0.003 inches and to fit a shroud around the shaft between

the machined flats and the probe. This alleviated the problem.

No critical speeds were detected within the normal operating speed range of the compressor. The performance of the aerostatic journal and thrust bearings were found to be satisfactory.

This experiment had proved the mechanical integrity of the drive shaft and the satisfactory performance of the aerostatic bearing system. The total running time for the first set of experiments was approximately eight hours.

6.6 Test Run Number 2

The aim of this experiment was to determine the touch-down speed envelope of the forward and rear journal bearings and the thrust bearings

6.6.1 Method

The experimental apparatus, shown in Figure 6.2, was prepared as for the last group of experiments.

Once initial preparations had been made the air pressure to each bearing was slowly increased whilst observing the touch-down indicator lamps. As each lamp glowed the air pressure to that bearing was recorded. This experiment was repeated several times to determine the minimum bearing pressure required to support the shaft assembly.

The drive air valve was opened slowly and the drive shaft accelerated to 30,000 rpm where the speed was stabilized. The air valve controlling the flow to the forward journal bearing was slowly closed and the touch down lamp observed. When the bearing-air control valve was shut the drive-air valve was slowly closed allowing the drive shaft to decelerate until the touch down indicator lamp began to glow. At this point the rotational speed was recorded. Bearing air pressure was then re-established and the drive shaft speed increased to 30,000 rpm. The same test was carried out a number of times to ensure that the result was repeatable. The same process was repeated for rear journal bearing and the thrust bearing. Between each test the electrical continuity of the indicator system was checked.

6.6.2 Discussion

The thrust bearing was found to operate in the aerodynamic mode satisfactorily down to speeds of 3000 rpm without rubbing. Repeated trials confirmed this result. There was no increase in the vibration level when the air supply to the thrust bearing was turned off.

The forward bearing operated satisfactorily throughout the test. It was found that the touch-down speed of the forward bearing was approximately 15,000 rpm with a good degree of repeatability. There was a slight vibration when the air supply to the bearing was reduced from 10 psig to 5 psig. Once the air valve was shut there was no noticeable change in vibration from when the air valve was fully open.

It was not possible to get the rear bearing to enter the aerodynamic mode. The minimum bearing pressures which were achieved before the touch-down indicator lamp glowed are listed in Table 6.1. The aerodynamic capacity of the rear bearing increased with rotational speed as anticipated. This is shown in Table 6.1. The aerostatic pressure required to support the shaft falls off with increasing speed.

On the completion of the test run the prototype compressor was dismantled. Examination of the journal face of the rear bearing revealed a single scuff mark, 1mm in width and 1mm from the rear edge, running around the periphery of the bearing from an equivalent journal attitude angle of 30 degrees to 75 degrees from the vertical. The journal had light scuffing in the same point confirming that the rear bearing had been rubbing.

It was thought the reason for the rear bearing not 'self acting' was misalignment between the bush and journal surfaces when the aerostatic pressure was turned off. It was thought that the forces produced by the 'O' seals were not evenly distributed resulting in a twisting moment which skewed the bush. Although the aerostatic pressure corrected the resulting twisting moment the dynamic pressure was unable to do so. To confirm these results it was decided to repeat the experiment after careful assembly. In this case it was found that the performance of the rear bearing was much improved and after repeated trials the average touch-down speed of the rear bearing was found to be 14,100 rpm.

As with the previous test run the thrust bearing was found to operate satisfactorily throughout the experiment. In this

particular case the touch-down speed of the thrust bearing was found to be 6,125 rpm averaged over three trials.

During the second test run the forward bearing was found to produce insufficient aerodynamic lift below 60,000 rpm. It was found that the vibration level from the forward bearing increased noticeably as the air pressure was reduced below 10 psig. This vibration, which was detectable through the body of the compressor, disappeared once the bearing was self-supporting its lift. It was also found that by reducing the aerostatic pressure to the rear bearing in line with the pressure to the forward bearing, the rotational speed at which the forward bearing developed sufficient aerodynamic lift could be reduced. This suggested that bearing performance was very sensitive to the misalignment attitude of the drive shaft. A small degree of tilt appeared to have a detrimental effect on bearing performance. By progressively reducing all air flow to all bearings it was found that the drive shaft could be completely supported in aerodynamic bearings at all speeds above 40,000 rpm. Below this speed the forward bush rubbed against its journal.

With the forward bearing pressurized to 20 psig it was found that the thrust bearing and rear bearing would operate satisfactorily in the aerodynamic mode between 15,000 and 70,000 rpm. Rapid and slow acceleration tests were carried out which produced no unsatisfactory responses from the bearing system or drive shaft.

Above 40,000 rpm the drive shaft and bearing system operated satisfactorily. Acceleration and deceleration tests were carried out between 40,000 and 70,000 rpm. From these tests it was concluded that the drive shaft and bearing system was stable and there were no incipient whirl or aerodynamic instabilities within the normal operating speed of the compressor.

The total running time for this second test run was three hours and for the greater part of this time the bearing system was aerodynamically lubricated.

The results of this second test run were particularly encouraging because they appeared to confirm the conclusions of the previous test in that the failure of the rear journal bearing to generate sufficient lift was due to the misalignment between the bush and journal faces caused by

the 'O' seals. This view was supported by the observation that the forward bearing, which operated satisfactorily throughout the previous test, performed poorly during this later experiment.

It was found that the vibration level increased during the transition between aerostatic and aerodynamic operation. This was particularly noticeable when both journal bearings were depressurized at the same time. The vibration level fell again once the air pressure to the bearings was turned completely off. Nevertheless the vibration level was greater than when aerostatic lubrication alone was used.

On completion of the second set of tests the compressor was dismantled and examined. There was no additional damage sustained to the rear bush during the second test. The forward bearing had suffered a slight discolouration over a small area at the forward end of the bearing consistent with misalignment. There was evidence that the labyrinth seal had rubbed against the special nut. The thrust bearing surfaces were unmarked.

It was concluded that the journal bushes should be redesigned to reduce the misalignment forces produced by the 'O' seals. To achieve this it was decided to make the 'O' seal seating channels narrower and deeper thus reducing the radial force between the seal and the housing and making the bush easier to fit. This reduced the possibility of the 'O' seal becoming twisted as the bush was pushed into place. In addition to this the distance between the 'O' seals in each bush was shortened which reduced the effective twisting moment on the bush causing the misalignment. The length of the bearings was also increased and the bearing clearances adjusted to maintain journal eccentricity.

Following the redesign and manufacture of the new bearings the compressor was reassembled and tested.

6.7 Test Run Number 3

This section describes and evaluates the results of a further six hours of experimental testing which led to the satisfactory design of the bearing system.

The aim of this third set of tests was to establish the dynamic performance of the drive shaft and bearing assembly with the modified journal bearing design. The method adopted in this case was the same as that used in previous test runs

described in Sections 6.5 and 6.6.

6.7.1 Discussion

The first test run was abandoned after 20 minutes because the thrust bearing touch down lamp came on and could not be extinguished by increasing the air pressure. An examination of the compressor revealed that the labyrinth seal was scuffing against the 'special' nut. It was decided to increase the seal clearance to 0.006 inches. After this modification there was no further problem with the seal.

The second test run lasted 30 minutes, during which the touch down speeds of the forward and rear bearings were 10,000 rpm and 12,000 rpm respectively and the touch-down speed of the thrust bearing was 6,500 rpm. During a high speed acceleration trial with the all bearings operating in the aerodynamic mode the forward journal bearing touch-down lamp came on. The air pressure to this bearing was re-established but the lamp remained on. It was decided to abandon the test to prevent unnecessary damage to the shaft and forward bearing. Examination revealed that the forward bearing bush had moved inside the housing towards the thrust bearing end by about 1cm causing the touch-down indicator wire to break free from its attachment pin and brush against the drive shaft thus completing the electrical circuit and causing the forward touch-down lamp to glow. It was interesting that the bearing bush had moved out of alignment with the air pressure supply orifice and therefore it was not possible for aerostatic lubrication to the forward bearing to be established during the shutdown. This had caused no harm to the bearing surface. Although the problem of the bearing movement needed to be attended to this finding was encouraging as it suggested the possibility of starting the compressor with aerodynamic bearings.

It was thought that the forward bearing had moved during operation because the radial forces between the 'O' seal and the housing was insufficient. With the original journal bearing design it had been necessary to use some force to press the bushes into position and the resulting radial forces had held the bush firmly preventing axial movement. Subsequently the 'O' seal seating was modified to reduce the interference between the 'O' seals and the bearing housings. The clearances between the bush and housing are required to produce a light interference fit between them when mounted

in 'O' ring seals. This interference fit must simultaneously prevent axial movement of the bush and at the same time the compression of the 'O' seals must not be so great that any small misalignment cannot be corrected by the aerodynamic forces in the bearing. It was felt that it would be better to provide an alternative means of securing the bushes. Clips were manufactured from high density polyethylene and positioned on each side of the bushes holding them in place. Metal clips were not used because they would have formed an electrical contact between bush and housing spoiling the touch-down indicator system.

In the next test run the compressor operated satisfactorily between 13,000 rpm and 55,000 rpm with the bearings operating in the aerodynamic mode. The compressor speed was limited by a shortage of drive air. Forward and rear bearing touchdown speeds were found to be 11,500 and 13,000 rpm respectively whilst the thrust bearing was found to touch at 7,500 rpm. There was a slight increase in the vibration level of the shaft as the bearing pressure was removed but even at the lower speeds rapid switch on and off of air pressure to any of the bearings did not cause any instability or bearing surfaces to touch. The total running time for the third test run in this series was 3 hours. The test run was repeated a few days later but in this case the speed range of the test was increased to 70000 rpm. The results of this test run were the same as the previous test and with this encouraging result it was decided to fit the laminated rotor to the drive shaft in order to prove its mechanical integrity.

6.8 Test Run Number 4

For this test the laminated rotor was mounted onto a new drive shaft. The assembly was balanced and fitted into the compressor. The aim of this test was to prove the mechanical integrity of the laminated rotor at the design speed, (70,000 rpm).

The drive shaft was accelerated to 70,000 rpm and maintained for thirty minutes. During this time the vibration level of the drive shaft was monitored with the assistance of two displacement transducers and oscilloscope. The drive shaft speed was then lowered to 20000 rpm and the air flow to each bearing was turned off. The rotational speed was again increased to 70000 rpm. The drive shaft vibration level was again monitored for thirty minutes. Bearing air supply was

then re-established and the drive air was turned off allowing the drive shaft to brake under the windmilling action of the impeller.

The mechanical integrity of the laminated rotor was found to be satisfactory. The resulting vibration level was similar to that experienced with the solid rotor used in previous tests. Inadvertent operation at speeds between 90,000 and 100,000 rpm resulted in a slight movement of the laminations which caused an increase in the vibration level at normal speeds, (ie 70,000 rpm), but did not result in bearing touchdown. After rebalancing the drive shaft was found to function satisfactorily in all aspects.

6.8.1 Discussion

This particular test was repeated twice. During the first run the drive shaft vibration level when the bearings were pressurized was similar to that experienced in previous test runs when the solid rotor had been fitted.

During the first run, when the bearings were operating aerodynamically, the rotational speed was increased from 20,000 to 70,000 rpm measured using the Gould counter/timer. The vibration level steadily increased. The drive air valve was closed slightly causing the speed of the shaft to decrease and the vibration level to fall. Because of unusual driver-air flow setting it was decided to check the rotational speed of the shaft using an oscilloscope. It was found that the speed was 85,000 rpm which suggested that the speed had previously been in excess of 90,000 rpm. Air flow to the bearing was re-established and the drive air was turned off. Because the laminated rotor had only been stressed to a speed of 75,000 rpm it was decided that an examination of the compressor would be necessary before testing could continue.

During the subsequent examination of the compressor the surface of the laminated rotor was found to be stepped which indicated that individual laminations had shifted slightly. It was not considered worth regrinding the rotor, however, the drive shaft assembly was re-balanced. Examination of the bearings showed that they suffered no obvious damage and once repairs were completed the compressor unit was reassembled. The Gould counter/timer was replaced and the test run was repeated with satisfactory results. The total

running time for this set of tests was 1 hour 45 minutes.

It was concluded that from a mechanical strength standpoint the laminated rotor would operate safely at 70,000 rpm. It had also been shown that there was a good tolerance margin as the drive shaft had operated in excess of 85,000 rpm without catastrophic failure occurring.

6.9 Vapour-Loop Test Facility

A schematic representation of the Freon vapour loop compressor test facility is shown in Figure 6.7(b). The compressor, (item 32), draws Freon 11 vapour from a heat exchanger, (item 15), via valves, (item 14), which control the quantity of bypass gas used to cool the motor. Compressed vapour leaves the compressor, (item 32), and is ducted back to the heat exchanger where heat gained during both the compression process and from motor cooling is rejected to environmental air blown through the finned coil heat exchanger by a fan, (item 45). Excess refrigerant is stored in a receiver, (item 20).

On startup the compressor bearings were lubricated with pressurized Freon 11 vapour produced in a boiler, (item 2). The vapour produced was fed via pipes, through a control valve and non-return valve, (item 4), and refrigerant drier, (item 23), to the bearings via a further valve, (item 28). Once the compressor was running pressurized gas for the bearings was taken from stagnation tappings in the compressor discharge line.

The Freon 11 boiler was filled via a pipe which transferred Freon liquid from the receiver, (item 20), via a valve which was shut when the boiler was in operation. Excess pressure in the boiler was relieved by opening a shut-off valve, (item 22), in the pipe line between the boiler and the compressor inlet.

Figure 6.7(a) shows the general assembly drawing of the Freon 11 vapour loop compressor test facility. The layout of the test facility shown in Figure 6(a) follows closely that shown schematically in Figure 6.7(b) with the addition of pipe line heater tapes on the insulated bearing gas pressure

lines and the introduction of an orifice plate, (item 29), in the compressor suction line. A list of refrigeration components purchased for this project is included in Table 6.2.

Photographs of the completed test facility are included in Figure 6.8 to 6.11. Figure 6.8 shows the refrigerant receiver, (item 1), with dual sight glasses and charging valve on one end. To the left of the receiver can be seen the Freon 11 boiler charging valve, (item 18), and the insulated and heated pressure line, (item 17), from the boiler to the compressor. The Freon 11 boiler is not shown in the photograph as it was positioned in a shallow pit to the front of the test facility and is just below the picture in Figure 6.8. Item 16 in Figure 6.8 is a transformer for the heater tape used to superheat the bearing gas as it leaves the boiler.

The compressor is shown in Figure 6.8, (item 8). Vapour is drawn along the suction pipe, (item 6), via a standard orifice plate, (item 7), to the compressor. Pressurized vapour is fed via a 1.25 inch diameter copper pipe, (item 9), and valve, (item 10), to the heat exchanger, (item 3). The heat exchanger used was a Gould type 'Contardo' finned coil condenser which was estimated to have a capacity of 33Kw.

Air to cool the heat exchanger was ducted via a shroud, (Part Number PB 15065, item 2), from a fan, (not shown), supported an assembly, (PB 15064). Cooled refrigerant vapour was fed via a 2.625 inch diameter copper pipe, (item 11), to a splitter valve, (item 12), which proportioned the flow to the motor casing for cooling purposes and directly to the compressor inlet. The main purpose of this valve was to control the compressor suction pressure. Motor cooling gas rejoins the main flow just down stream of the splitter valve, (item 12). A second control valve, (item 14), is positioned in the motor cooling gas line to control the flow resistance through that circuit.

The electrical supply for the compressor was generated by an alternator shown in Figure 6.8, (item 4). The alternator was driven by a Mawdsley 40hp DC motor, (item 5), with a drive ratio of 3.67:1.

The alternator drive consisted of an endless 'Meteor' C6 type flat belt, (2013 mm length x 50mm width), and two cast iron pulleys; a plate type, (275mm o/d x 65mm width) and a solid type, (75mm o/d x 65mm width). The belt was tensioned

by adjusting the length of the 'tensioning link', (item 19 in Figure 6.8), positioned between the subframe assembly, (item 20 - PB 15050). The photograph in Figure 6.9 shows the Mawdsley DC motor, (item 1), mounted on its subframe with the shrouded belt drive at the left end of the photograph and the alternator in the background, (item 2).

The electrical supply from the alternator to the compressor motor was controlled by an ON/OFF lever on the switching box, (item 15, Figure 6.8). The supply current and voltage to the compressor were controlled by varying the alternator speed and field voltage.

6.10 Instrumentation

A schematic representation of the instrumentation system used in this set of experiments is shown in Figure 6.13. A photograph in Figure 6.10 is used to identify some of the equipment whilst Figure 6.11 provides an alternative aspect. The same numbering system is used in both Figure 6.10 and 6.11.

Type K thermocouples were used to measure the refrigerant temperature. Referring to Figure 6.13, refrigerant temperatures were measured upstream of the orifice plate, T1, (item 13, Figure 6.10), at the high pressure side of the compressor, T2, at the inlet to the motor and at the outlet from the motor casing. In addition the motor winding temperature was measured at six locations in the motor winding; three thermocouples equally spaced at each end of the stator. Direct reading of the thermometers was made using a Newport digital thermometer.

Refrigerant pressures were measured using Rabtherm compound pressure gauges.

The rotational speed of the compressor drive shaft was measured using a magnetic type pickup. The output from the magnetic pickup was recorded on a Gould type TC 320,35 Mhz, timer/counter, (Figure 6.10, item 3).

Displacement probes were used to measure the frequency and amplitude of drive shaft vibrations. The readings were recorded on a Tektronic 544 four channel oscilloscope, (Figure 6.10, item 2).

The rotary inverter consisted of a 40hp Mawdsley 600 VDC motor driving a 20hp alternator. A schematic representation of the rotary inverter system is shown in Figure 6.13.

The Mawdsley DC motor was controlled by an E.A.C. variable resistor unit capable of supplying 0 to 150 amps, (maximum), and 40hp,(continuous), at 600VDC, (Figure 6.10, item 15).

The alternator field winding voltage was controlled using a Zenith variable transformer type Y16HM with a 240 VAC 50-60Hz supply and 0 to 270 VAC, 8amp output, (Figure 6.10, item 14).

The power consumption of a single phase was measured using a Feed Back wattmeter type EW604. The supply current and voltage readings were also taken from the same phase allowing the power factor to be calculated. A standard AVO meter was used to measure the voltage and a clip-on type ammeter was used to measure the amperage.

6.11 Preliminary Air Tests

The aim of the experiment was to confirm the operation of the prototype electric motor and the rotary inverter system.

The compressor was prepared and mounted in on the test-bed, (Figure 6.10 and Figure 6.11). The Freon vapour loop was disconnected at the inlet to the the impeller and the bearing system was pressurized using air provided by a portable air compressor and accumulator. The coolant valve was set fully open and splitter valve shut to ensure that all flow was ducted through the motor casing to ensure maximum possible cooling effectiveness.

The following method was used to start the rotary inverter system and the prototype motor.

i) The starting lever on the switch box, (item 12) in Figure 6.10 was set to 'off' position.

ii) The alternator field voltage was set to 2 volts using the the Zenith transformer, (item 14).

iii) Compressor bearings were pressurized with air at 30 psig

iv) Following the operating instructions detailed on the 'Instruction Plate' on the front panel of the E.A.C motor controller, (item 15), the hand wheel, (item 10), was rotated until the voltage supply to the 40 hp Mawdsley DC motor was minimum. The control lever, (item 11), was then set and the Mawdsley motor started. The maximum motor amperage was recorded and when the starting sequence was completed the

control lever, (item 11), was set to the 'run' position.

v) The speed of the Mawdsley motor was adjusted to 1,250 rpm using the hand wheel, (item 10), to increase the voltage supply to the motor whilst the DC motor speed was measured using a hand held tachometer. A DC motor speed of 1,250 rpm was equivalent to an alternator supply frequency of 765 hz.

vi) The alternator field voltage was adjusted to produce 200 VAC across one phase. This caused a slight reduction in the DC drive motor speed which was adjusted causing the field voltage to require readjustment.

vii) The prototype motor starting lever , (item 12), was set to the 'ON' position.

viii) When the prototype compressor motor was engaged the DC motor current increased and its speed fell slightly. The speed of the DC motor was then reset by increasing the voltage supply.

ix) The prototype motor speed was then increased simultaneously with the voltage supplying the DC drive motor.

Once the operating speed of the prototype motor had been achieved, (ie 70,000 rpm), the air flow through the motor casing was reduced by allowing flow to bleed past the splitter valve and to atmosphere. This caused the motor to accelerate slightly. The pressure drop and motor winding temperatures were recorded.

The compressor load was adjusted by simultaneously altering the coolant control and splitter valves settings, (items 14 and 12, Figure 6.8). The rotational speeds of the DC motor, alternator and compressor motor were then corrected by adjusting the field winding voltage to the alternator and the voltage supply to the Mawdsley DC motor.

Bearing aerodynamic lubrication was established at 70,000 rpm and the compressor was run continuously for forty five minutes at various speeds and loads. This allowed experience to be gained in controlling the rig and monitoring the instruments.

Before shutting down the compressor aerostatic lubrication was re-established. The prototype compressor was then shutdown by turning the control lever, (item 11, Figure

6.10), to the 'OFF' position. After this the alternator field voltage was turned off and the prototype motor control lever, (item 12, Figure 6.10), set to the 'OFF' position.

The compressor operated satisfactorily throughout the test. The bearing system performed well in the aerodynamic and aerostatic modes. No touch down was recorded and the vibration amplitude and frequency were similar to those experienced during previous tests.

On starting the maximum amperage to the DC motor was 70 amps and during the 60 minute test run the motor winding temperature slowly increased to 80°C. This temperature did not level out and appeared to have increased steadily throughout the test.

6.11.1 Discussion

The bearing system appeared to function satisfactorily even when the compressor was caused to surge, resulting in an increase in vibration level.

A particular problem was the control of the compressor speed. If when operating at a constant speed the flow rate was reduced, by closing the splitter valve, the speed of DC motor increased causing the alternator to accelerate. If the alternator field voltage was kept constant then the power output and frequency increased causing the prototype compressor to accelerate until a new balance point, (of the Mawdsley - alternator - compressor system), to be established. The compressor, alternator and motor speeds simultaneously increased as the flow rate was reduced. In order to maintain a constant speed it was necessary to adjust the alternator field voltage and DC motor voltage simultaneously to maintain the output frequency. The response to this technique was good when the compressor speed was high. However, at low speeds control became increasingly more difficult as the operating point of the Mawdsley DC motor moved towards the lower end of its power/speed characteristic.

Although the control of the compressor was not difficult at this time it indicated that control of the rig and particularly the prototype motor speed and output would be an important area for development in the future.

The system was stable at the higher speeds but became increasingly more unstable at lower speeds. Changes in flow could only be carried out slowly. When the flow rate was to be reduced the flow control valves would be closed by a small amount and then the speed and power output of the alternator would be corrected so that the compressor speed returned to its original value. The process would be repeated in small increments until the required flow rate was achieved. The instability of the system became apparent when the compressor was first accelerated to 70,000 rpm with a low mass flow. When the flow was first increased, by opening the flow control valves, the DC motor stalled and caused the test site DC supply circuit breaker to open. The rating of the circuit breaker was increased twice to prevent this. The manual control of the rig was hampered by the physical distance between control valves and levers.

To reduce the difficulty in controlling the power supply system Stone International designed and supplied a electronic control device which was to vary alternator field voltage as a function of alternator speed to maintain a constant winding voltage. With this device only manual control of the DC motor supply voltage would have been required once the system was running. This would have been of great value. The control device did not operate correctly and was subsequently returned to the factory for modification.

Once the compressor was running the coolant flow through the casing was turned off until the winding temperature reached 70°C, which took 12 minutes. Once the motor temperature reached 70°C the coolant flow was established and the temperature increased to 80°C over the remainder of the test at a rate of 0.2 °C/min.

The compressor operated satisfactorily in the manner for which it was designed. The test was repeated on four separate occasions with similar results.

6.12 Freon 11 Test Runs

The aim of these experiments was to determine whether the prototype compressor would operate satisfactorily in a Freon 11 environment. Of particular interest was the operation of the bearing system and the prototype motor. A further objective was to evaluate the performance of the test facility as a whole in order to make recommendation for its

future development.

The experimental method varied from the previous test runs only in the way in which the experimental apparatus was prepared and in the procedure for starting the compressor. In this case the compressor and vapour loop were hermetically sealed and charged with Freon 11. In all other aspects the preparation of the rig was the same as in the tests previously described.

Figure 6.14 shows a Mollier diagram of the vapour loop cycle consisting of a compression process, (1 to 2), a constant enthalpy pressure loss, (2 to 3), through the high pressure valve and a cooling process, (3 to 4). The flow was then split into two streams controlled by the coolant control valve, (item 14 in Figure 6.8), and splitter valve, (item 12). The coolant flow is heated as it passes through the motor casing, (process 4 to 5), after which it is remixed with the remaining flow in the suction line, to produce state point 1 in Figure 6.14, from where it returns to the compressor inlet.

To prepare the apparatus for testing all potential sources of leakage were sealed with a proprietary silicon sealant. The vapour loop system was then evacuated and charged with refrigerant until the liquid level in the reservoir was in the centre of the upper window, (item 1, Figure 6.8). At this point the charging valve was closed. The boiler was charged by gravity feed from the reservoir. It was estimated that the boiler would provide sufficient gas for a period of 15 minutes at a supply pressure of 100 psig.

Once the boiler was charged the valve between the reservoir and the boiler was closed. The boiler heater and pipe line heater tapes were switched on. It was important that the heater tapes were switched on well before starting the run in order to allow time for the pipework to reach a temperature above the saturation temperature of the refrigerant to prevent condensation of the vapour.

The Mawdsley DC motor and alternator were started following the sequence described in a previous section.

With the boiler pressure was 100 psi the supply valve to the bearings was opened. The supply pressure to each bearing was adjusted until all touch-down indicator lamps were extinguished. With the alternator supplying 200 volts at 765

Hz the control lever on the switch box between the alternator and the compressor was set to 'ON' causing the prototype compressor to accelerate. The voltage to the Mawdsley DC motor was increased with the alternator field voltage which caused the Mawdsley motor speed to recover and the prototype compressor to accelerate to 27,000 rpm. The maximum current drawn by the DC motor during starting was 100 amps.

Compressor speed was increased to 70,000 rpm in increments by alternately increasing the mass flow, alternator winding field current and Mawdsley DC motor speed. Care was needed particularly at lower speeds to prevent the DC motor speed from dropping which resulted in the main supply contact breaker becoming open circuited.

Having achieved a stable operating condition at 70,000 rpm the bearing gas supply was changed over from the boiler to the stagnation tapping in the pressure line, (coming from the compressor). The pressures fell from about 35 psig to about 10 psig in each case. The flow rate through the compressor was varied by altering the setting of the splitter and coolant flow valves.

Once the bearings were operating on the stagnation pressure created by the compressor itself the boiler and heater tapes were switched off.

The bearing gas supply pressure was further reduced by closing the stagnation pressure line valve until the bearings operated in an aerodynamic mode. When this condition was achieved the flow was adjusted and readings were taken.

On the completion of the test run bearing pressure was established, alternator field winding voltage was then adjusted to 2 volts and the DC motor voltage reduced to its 'STOP' position. The compressor switch box control lever was then turned to the 'OFF' position. A strip-down examination of the compressor was carried out after each test run.

6.12.1 Results

Six test runs were carried out, three were aborted because of an electrical supply failure and two failed due to a partial seizure of the bearing system.

Journal bearing static lift-off pressure was found to be 35

psig and 40 psig for the thrust bearing. Alternator supply voltage was reduced to 120 VAC before starting to reduce the DC motor current to 80 amps. The acceleration time was longer than it otherwise would have been but this prevented overloading the site DC circuit breaker. The DC motor speed was set at 1,200 rpm which was equivalent to an alternator output frequency of 765 Hz. When starting the compressor the DC motor speed fell and the alternator voltage reduced from 120 VAC to 20 VAC. The power output of the alternator was re-established by increasing the DC motor and alternator field winding voltages simultaneously.

The maximum motor winding temperature achieved throughout these tests was 90.9°C.

6.12.2 Discussion

The object of these tests was to prove the feasibility of the prototype compressor when operating on Freon 11. Several important areas for development were identified which related to the design of the test facility and the control and design of the compressor.

Difficulties were experienced in operating the experimental apparatus. Control was more difficult than previously when operating the compressor on air. The increase in compressor power consumption resulted in a greater degree of operating instability, (particularly at lower DC motor speeds). This resulted in the Test Area main DC supply circuit breaker failing on three occasions. This was overcome by reducing the starting load on the DC motor by changing the alternator field voltage from 200 to 120 volts during startup causing the DC motor starting current to be kept below 80 amps, which was within the surge limit of the modified circuit breaker fitted later.

To overcome the problem of control inherent in the test facility an outline scheme, shown in Figures 6.15 and 6.16, is proposed. In the proposed control system the primary independent variable would be compressor pressure ratio. The measured pressure ratio would be compared with the required value and the setting of a motorized splitter valve altered to correct the error.

Compressor speed would be controlled by the automatic

system, in Figure 6.16, which would operate as follows. Assuming that the compressor was operating at a predetermined speed and load an increase in pressure ratio or saturation pressure would cause compressor speed to fall. This would be sensed by a comparator which would be designed to automatically increase the DC supply voltage causing the DC motor to accelerate. This in turn would cause the alternator to accelerate increasing both supply frequency and voltage to the compressor. An increase in supply voltage would be sensed by a comparator and the alternator field voltage would be reduced until the output voltage reduced to the original setting. An increase in slip between the prototype motor and supply current would cause the compressor to accelerate to the set-point speed. It is envisaged that the operator would have some control over the alternator field voltage and the compressor speed.

It is recommended that a control system should include a facility for starting the test rig. This might take the form of a ramping device. In addition to the main switch, two safety relays would be included in the electrical supply lines, contactors 'A' and 'B' in Figure 6.16. The safety contactors would be connected in series with safety cut-outs to cause an open circuit if the compressor should seize, suffer excessive vibration, an electrical phase failure or if the motor winding temperature becomes excessive.

In addition to a control system, the test facility would be improved if the control and monitoring of the Freon boiler, heater tape and bearing pressures was centralised with other controls which would make the operation of testing more efficient.

The Freon 11 boiler performed satisfactorily throughout all test runs. It was estimated that the boiler would supply 6 minutes of vapour at the required pressure and flow. During the starting procedure boiler pressure fell from 104 psig to 35 psig and heater tape temperature fell from 107°C to 75°C, (on average). It is thought the reason for the rapid fall in temperature and pressure was the inability of the 2 kW heater to supply vapour at a rate to maintain the higher pressures. The system operated satisfactorily and sufficient capacity remained in the boiler to enable the bearings to be pressurized for shutdown.

Figure 6.17 shows the pressure lubrication process in T-S co-ordinates. The Freon leaving the boiler as a dry saturated vapour at 100°C and 118 psia. On the down stream

side of the bearing a gas control valve throttled the vapour to 55 psia and 68°C; indicating a slight increase in enthalpy due to the heating of the heater tape. It is interesting to note that the gas was dry saturated as it entered the bearing suggesting that the vapour was probably superheated within the bearings if constant enthalpy expansion is assumed.

When the vapour supply from the boiler and the stagnation tapping were turned off the bearing pressure fell to a value slightly greater than the pressure in the motor casing. Figure 6.18 shows a representation of the aerodynamic lubrication process in T-S co-ordinates. Process (1) to (2) represents a constant enthalpy process in which vapour is drawn into the low pressure side of the bearing. Process (2) to (3) represents the compression of the vapour through the aerodynamic film and process (3) to (4) represents a constant enthalpy expansion from the high pressure region within the bearing film into the casing cavity. Finally the vapour mixes with the vapour in the casing to return it to state point (1). In practice it is thought that the corners of the process diagram would be rounded and not so well defined as indicated in Figure 6.18. The possibility of the vapour being wet within the bearing was considered. Under normal conditions the temperature of the vapour within the casing will be close to the mean temperature of the casing which will be greater than the saturated temperature of the vapour. Condensation may occur during start-up, when the casing is cold. The effects of condensation and flash evaporation in bearing clearances may cause some bearing instability.

The fourth test run was aborted after fifty minutes because of bearing seizure. An examination of the compressor revealed that the rear journal bush had rotated in the housing which resulted in the partial disintegration of the supporting 'O' seals. What was particularly interesting was that the bush was still supported on the remains of the 'O' seals and free running on the journal. The rear bearing surfaces showed a light coating of corrosion which had probably formed during the delay between the previous air tests and completing the preparations for the Freon tests. It was thought that this corrosion had resulted in a momentary seizure of the bearing during startup. The damage to the bearing faces included light localized burning. The bearing faces were cleaned with fine emery paper and finished by honing.

The forward bearing was found to have very light superficial surface markings. The forward plastic retaining ring was found to have formed into a dense foam which had expanded to fill the space between the housing and shaft blocking the gas ventilation holes and impinging on the rear side of the thrust bearing. The reason the high density polyethylene clip foamed is not understood. Samples of the polyethylene material had been soaked in Freon 11 liquid for a six weeks and had suffered no significant loss in mechanical properties. The remaining three plastic clips had suffered no apparent damage. It can only be assumed at this stage that the clip had moved forward to rubbed against the thrust bearing; the resulting combination of high temperature, due to friction, and the presence of Freon 11 vapour had broken the polyethylene down. So vigorous was the expansion of the foam that it appeared to have acted as a brake on the shaft and thrust bearing and it was concluded that this had been the cause of the aborted test run. An examination of the thrust bearing showed that scoring had taken place particularly at the outer radius on the forward face of rotating disc as if the bearing had suffered excessive conical whirl. There was no resistance when dismantling the bearing suggesting that the damage had occurred with expansion of the foam.

The fifth test run was satisfactory. The compressor ran for ninety minutes without fault over a range of pressure ratios and speeds with the bearings operating in an aerodynamic mode. In this case the plastic bearing retaining clips were exchanged for metal clips. This meant that the touch down indicator could not be used because the clip would earth the bushes. As the bearings had been shown to operate satisfactorily up to this point without rubbing this was thought not to be a major disadvantage.

During the sixth test run the thrust bearing seized after fifty minutes. Figures 6.19 and 6.20 show the resulting damage to the forward and rear bearing faces. The damage to the forward bearing face is restricted to the periphery of the disc and thrust plate which suggested some form of conical whirl whilst damage to the rear face is found at both inner and outer radii. What was thought to have occurred was that the bearing picked up on a part of the disc which was damaged during test run 4. Nothing conclusive can be drawn from this result. Examination of the journal bearings revealed no significant wear to the bearing faces, Figure 6.21, and apart from the seizure of the thrust bearing test run 6 was satisfactory.

The motor cooling system had operated satisfactorily throughout the testing. The stable motor winding temperature was 90°C after ninety minutes operation at various speeds ranging from 30,000 to 70,000 rpm with pressure ratios up to 3:1.

No conclusions can be drawn from the test runs so far conducted in regard to the satisfactory running of the electric motor. Readings that were recorded tended to show that the electric motor was capable of producing the power output it was design to, ie 14 kW, but more data is required to enable the characteristic of the motor to be drawn.

6.13 Conclusions

In this chapter two groups of experiments were described and evaluated. The first group are covered in Sections 6.2 to 6.8 and were concerned with testing the prototype compressor unit in an air environment. These tests were carried out in order to prove the mechanical integrity of the drive shaft assembly and to develop and prove the bearing system. The second group of experiments, described and evaluated in Sections 6.9 to 6.12, showed that the prototype compressor unit would operate satisfactorily in a refrigerant environment.

There were a number of development problems encountered during the preliminary air tests. Misalignment of the journal bearings caused by poorly fitting 'O' ring seals repeatedly prevented the bearing system from operating in the aerodynamic mode. The problem was overcome by redesigning the 'O' seal seats in the bearing bushes. There was less interference between the 'O' seals and the bearing housing with the modified bushes. This resulted in the bushes moving within the housing during the operation of the compressor. To overcome this problem plastic clips were fitted on each side of the bushes to restrain axial movement. Later these plastic clips were exchanged for steel clips when the former were found to be susceptible to attack from Freon 11. The thrust bearing was shown to operate satisfactorily throughout.

During the preliminary tests the labyrinth seal was found to be rubbing. This was re-machined to increase the clearance thus eliminating the problem.

After 13 hours of testing the bearing system was shown to

operate in both the aerodynamic and aerostatic modes. The transition speed between the aerostatic and aerodynamic modes was found to be approximately 15,000 rpm. The shaft assembly with the laminated rotor was shown to withstand an overspeed condition of 90,000 rpm. Slight displacement of the laminations resulted in the shaft assembly being rebalanced.

A Freon vapour loop test facility was designed and built. A description of the facility is included in Section 6.9.1. A major problem in using the test facility was the control of the compressor speed because of the interaction between the compressor motor, alternator and D.C. motor. This difficulty had been foreseen and a speed controller had been designed to control the output of the alternator which would have made the control of the system far easier. Regrettably this device did not function properly and was returned to the manufacturer for modification and testing proceeded without it, (Section 6.11.1).

Six test runs were attempted with the prototype compressor operating on Freon 11. The first three were aborted when the main test site D.C. circuit breaker failed. The fourth was abandoned after 50 minutes when the bearing system failed, the fifth test run was successful and lasted 90 minutes and the final test was aborted after 50 minutes when the thrust bearing seized. The failure of the thrust bearing was put down to earlier damage resulting from the failure of the journal bearing bush plastic retaining clips during test run number four. This is discussed in Section 6.12.2.

Although the results of the 'Freon' tests were not conclusive, (ie., the drive motor had not been proved and the performance of the bearing system was not entirely satisfactory), they were encouraging and showed that the compressor concept was feasible.

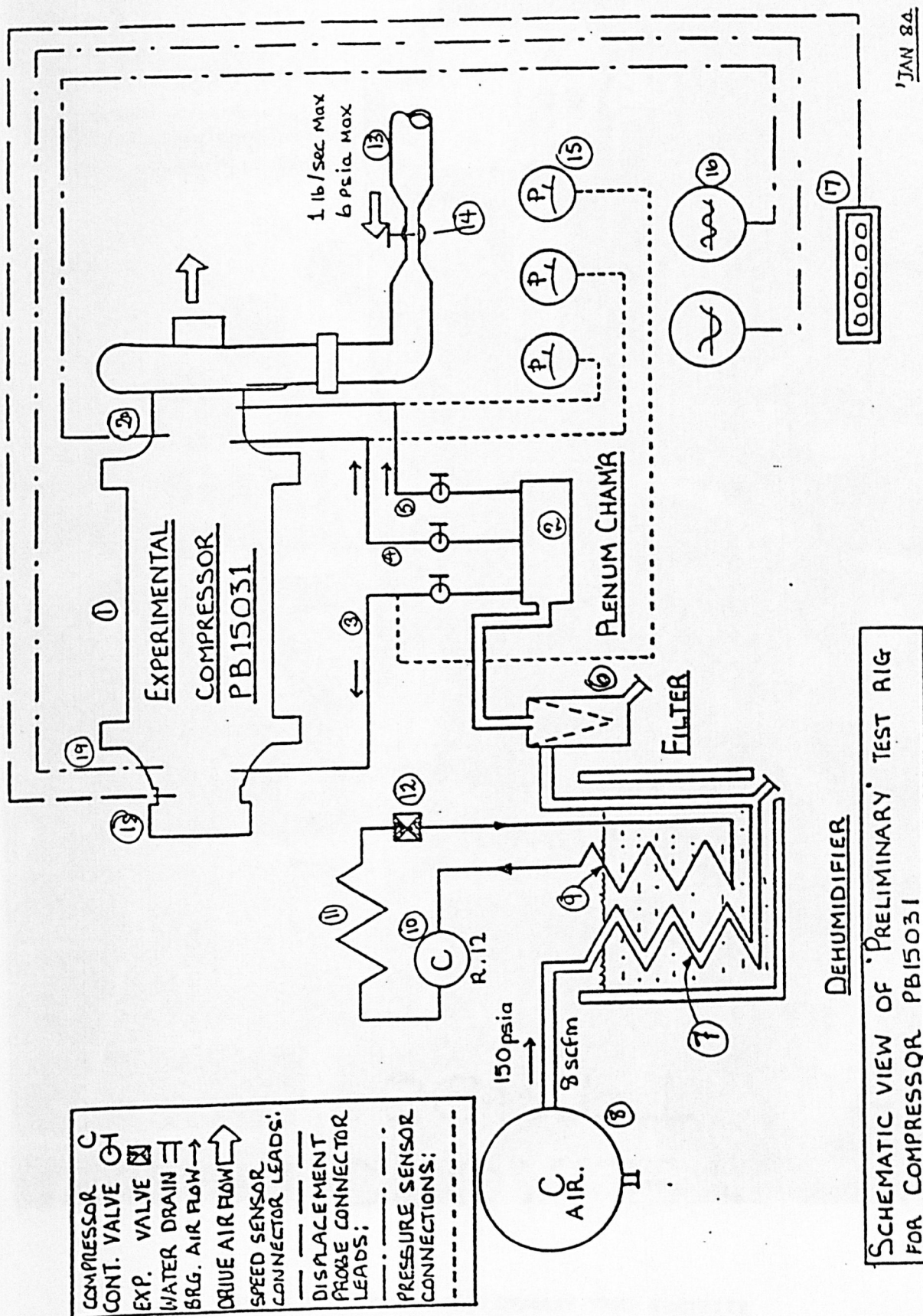


FIG. 6.1 SCHEMATIC VIEW OF PRELIMINARY TEST RIG FOR COMPRESSOR PB15031

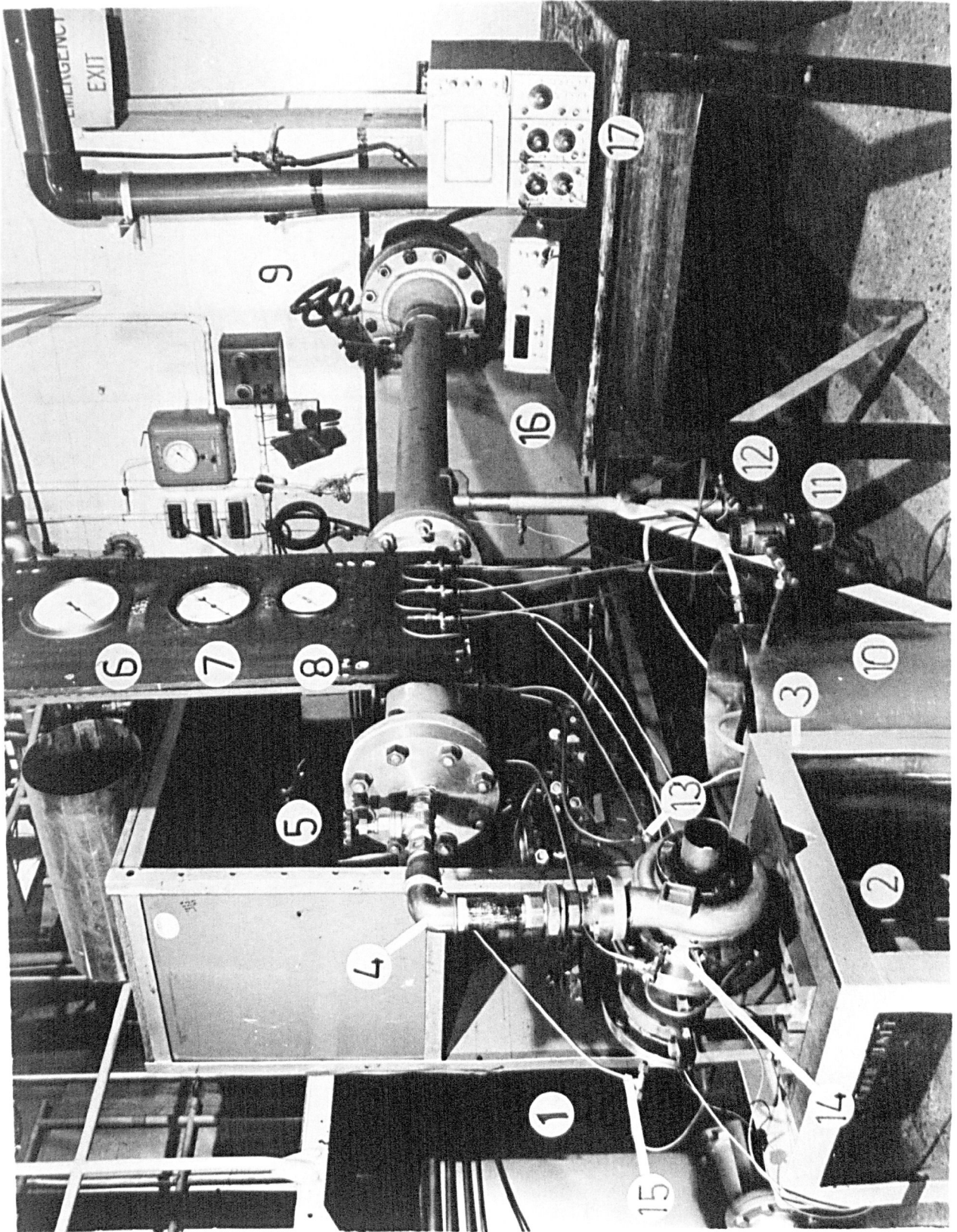


FIG. 6.2 VIEW OF PRELIMINARY TEST FACILITY

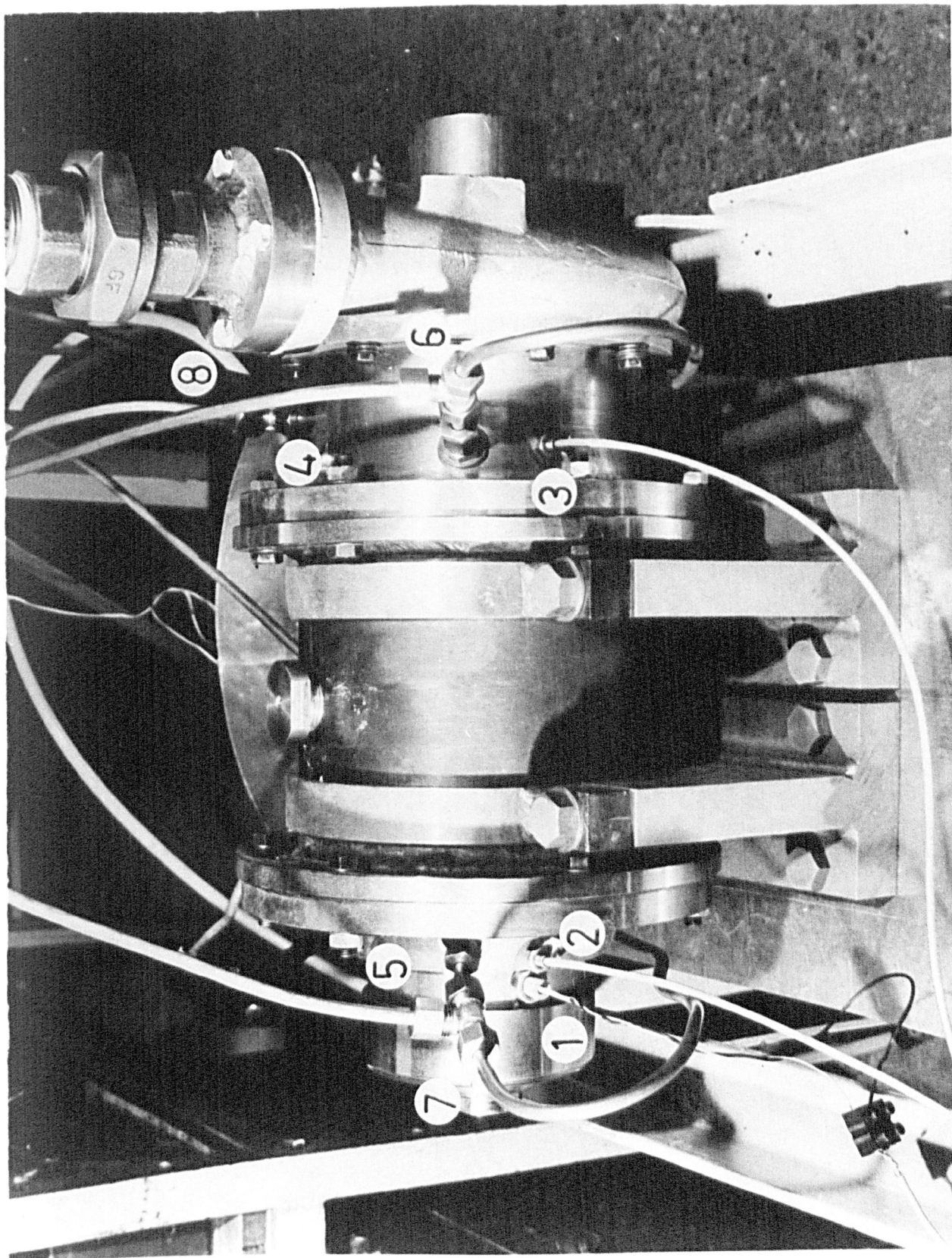


FIG. 6.3 VIEW OF PROTOTYPE COMPRESSOR MOUNTED ON TEST BED
FOR PRELIMINARY TESTING

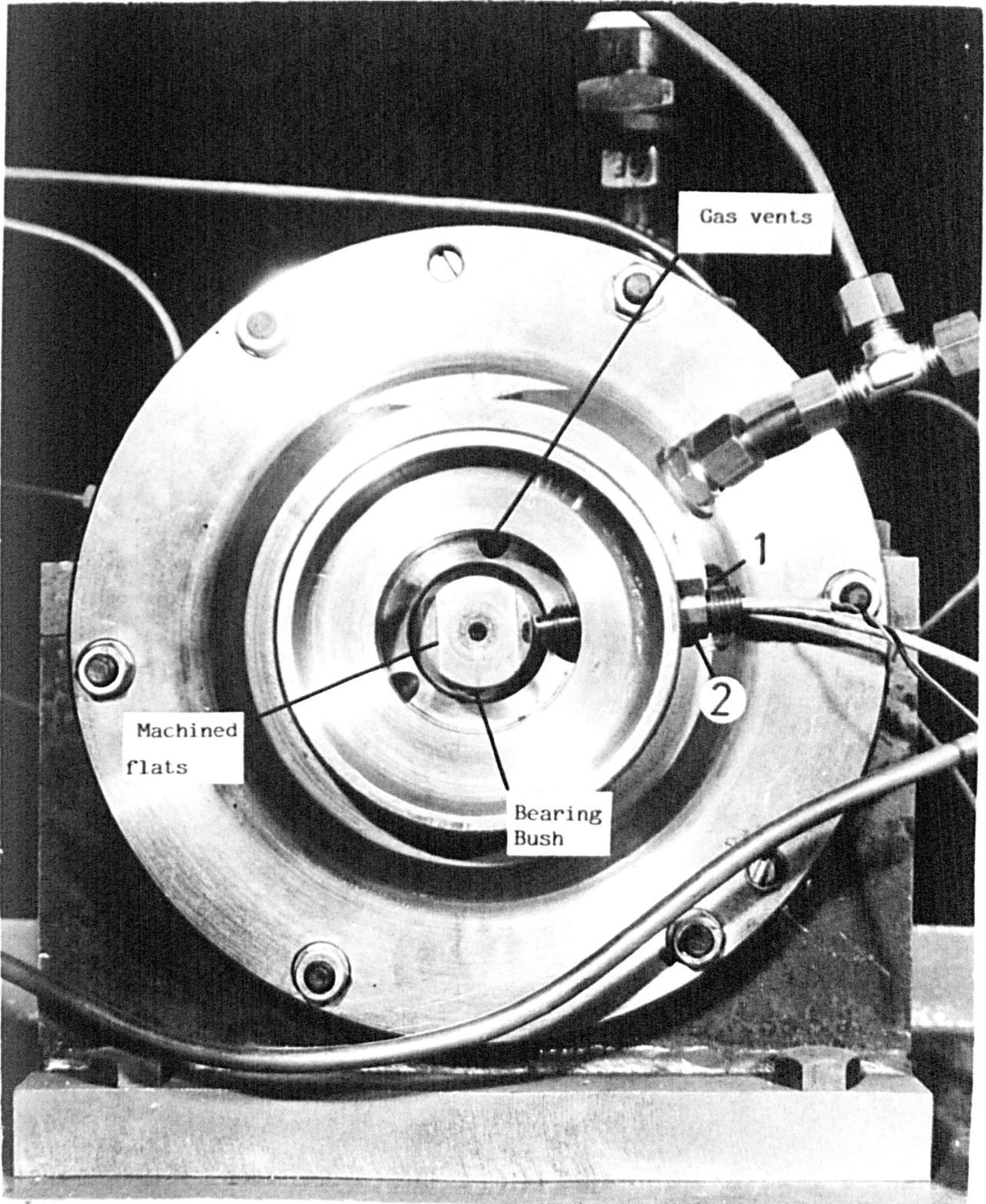


FIG. 6.4 VIEW OF EXPERIMENTAL COMPRESSOR FROM MOTOR END

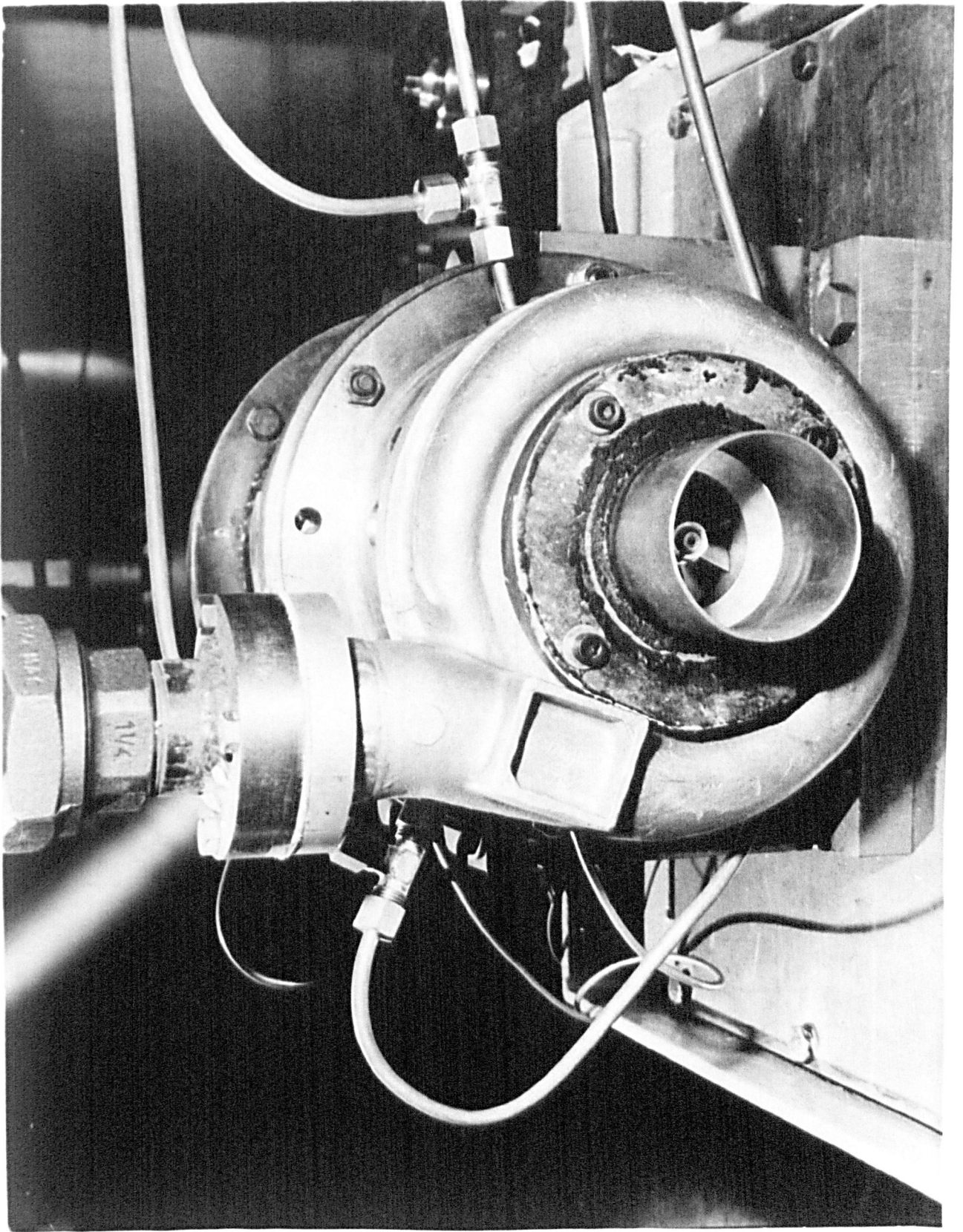


FIG. 6.5 VIEW OF PROTOTYPE COMPRESSOR SHOWING IMPELLER AND SHROUD

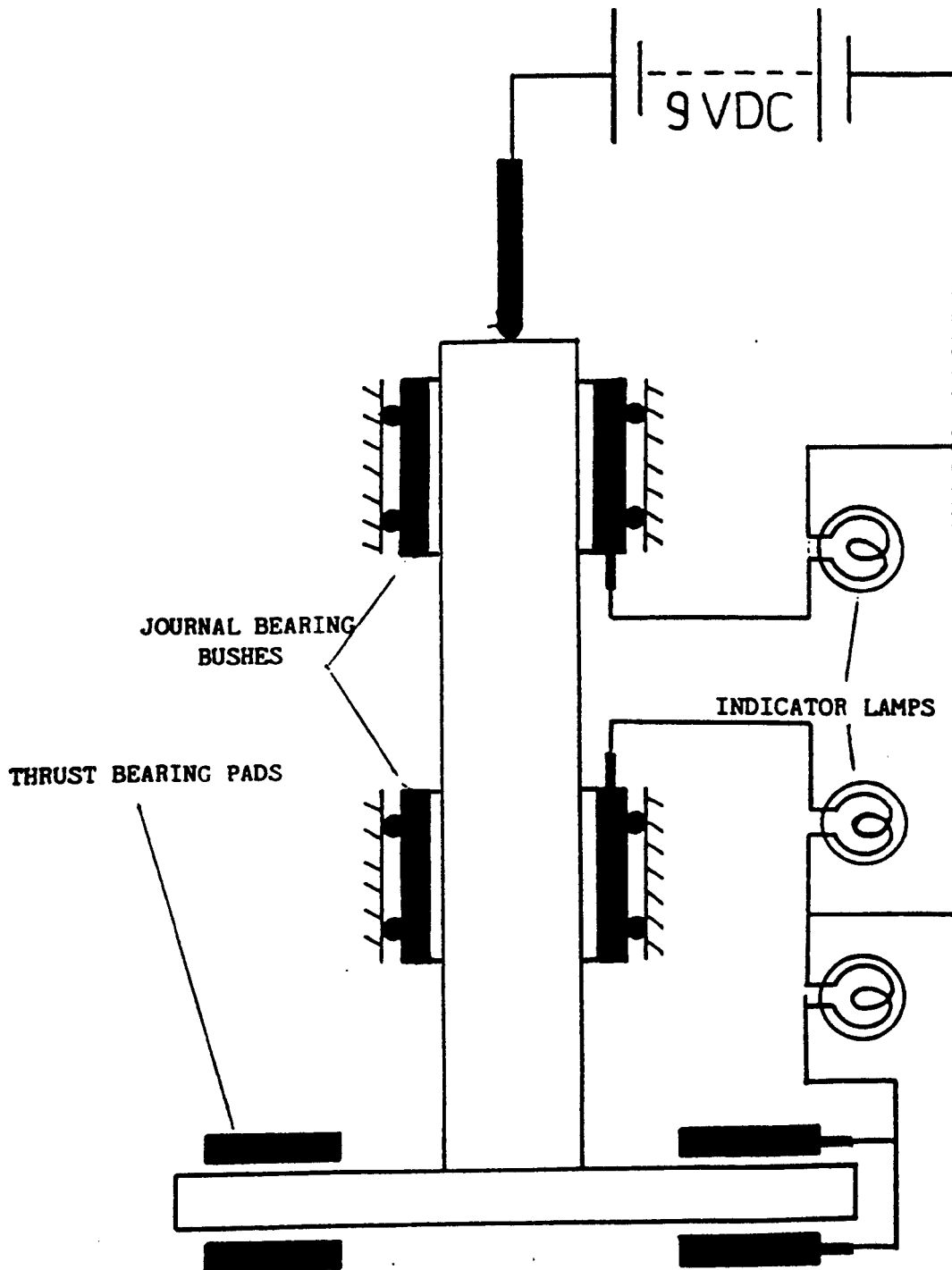


FIG. 6.6 SCHEMATIC REPRESENTATION
OF BEARING TOUCH DOWN INDICATOR SYSTEM

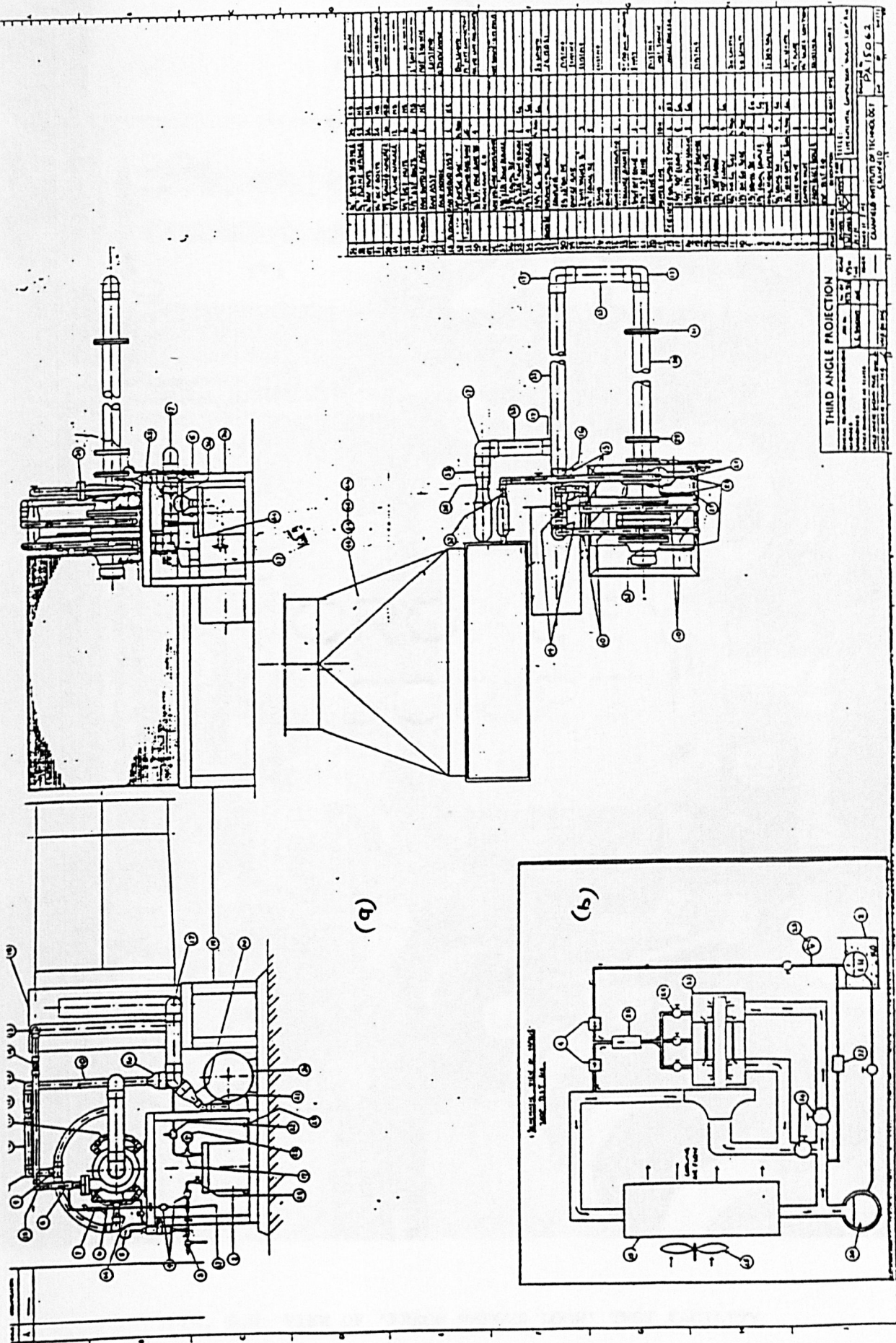


FIG.6.7(a) G.A. OF VAPOUR LOOP TEST RIG

FIG.6.7(b) SCHEMATIC REPRESENTATION OF SYSTEM

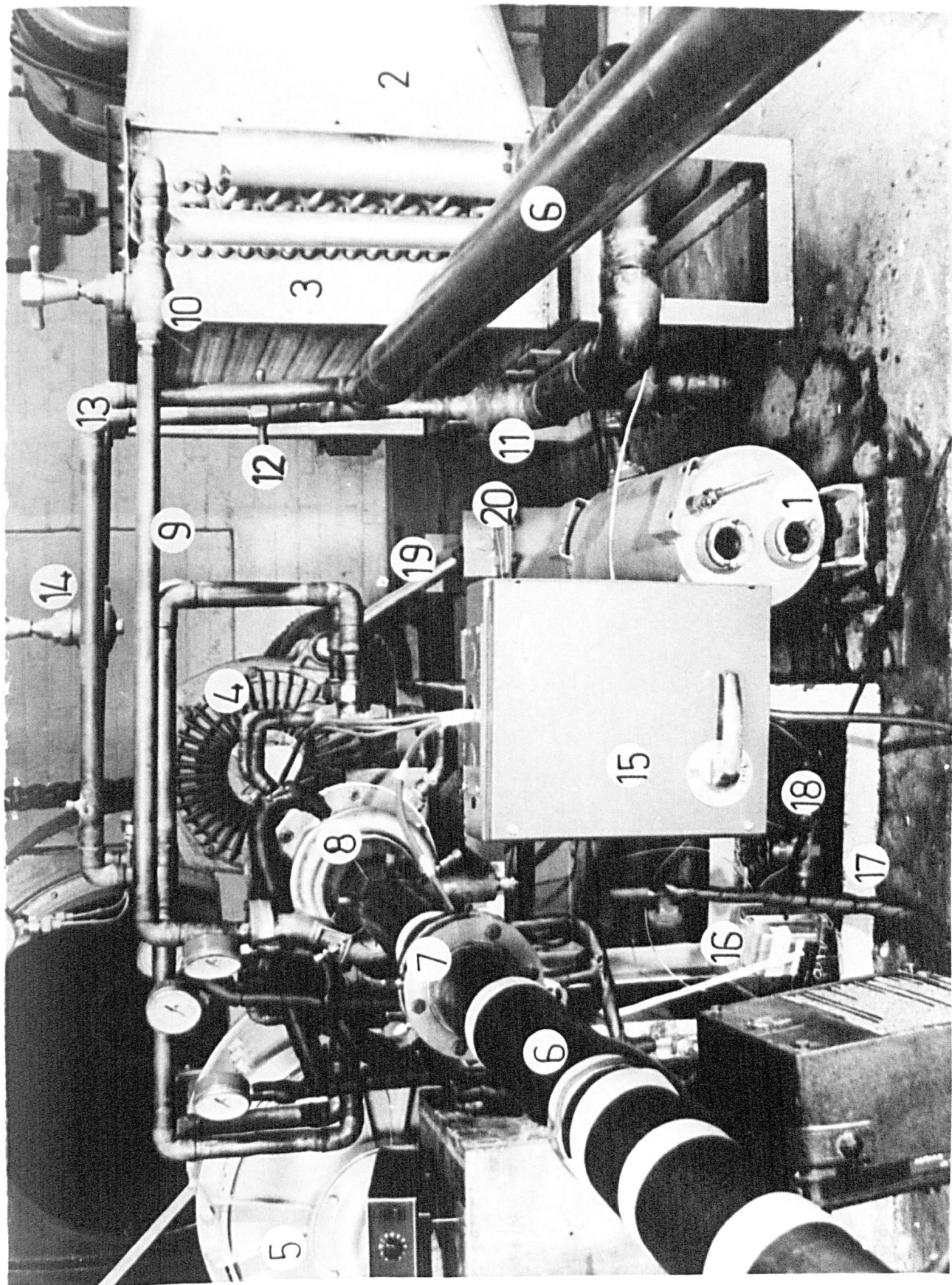


FIG. 6.8 VIEW OF 'FREON VAPOUR LOOP' TEST FACILITY

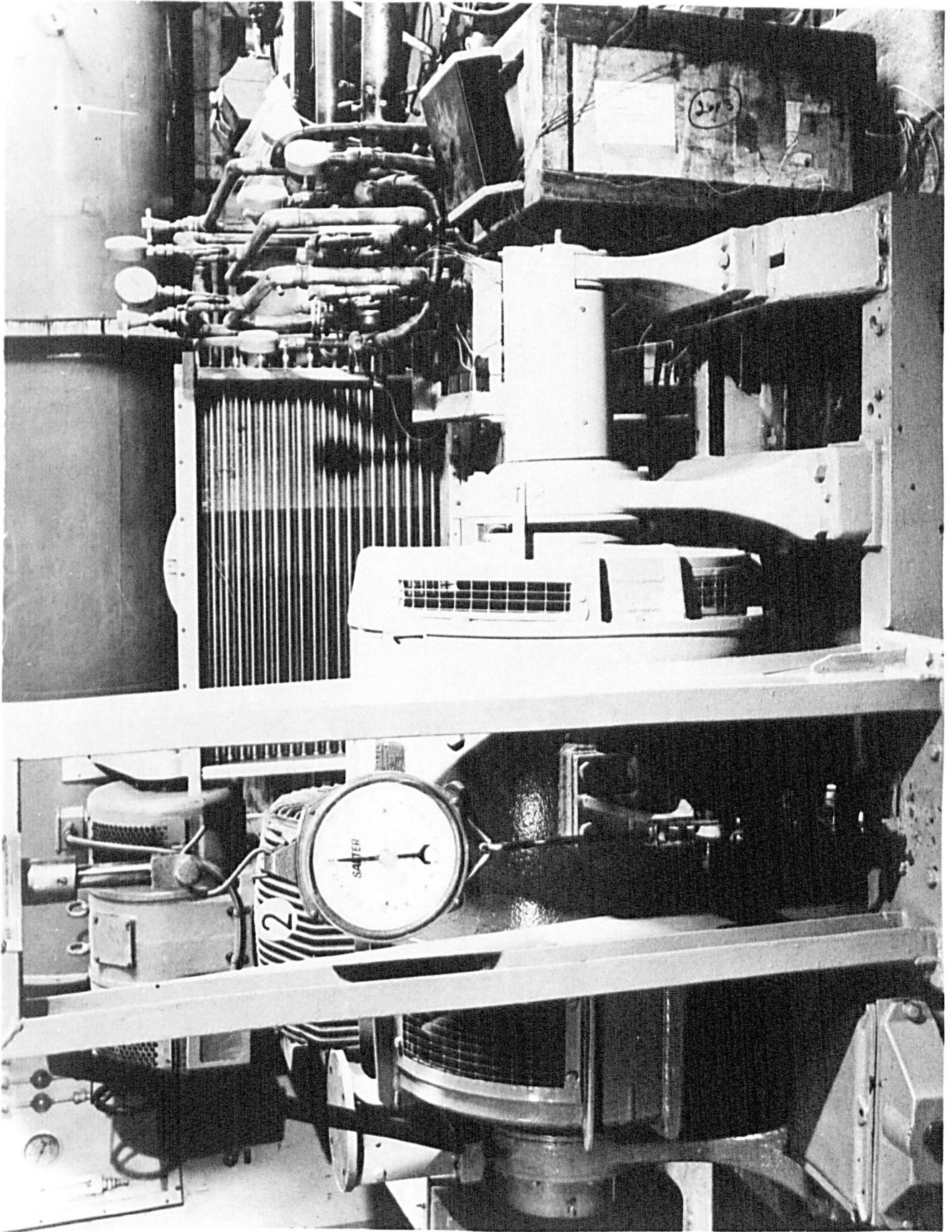


FIG. 6.9 VIEW OF 'FREON VAPOUR LOOP' TEST FACILITY
SHOWING MAWDSLEY DC MOTOR (1) AND ALTERNATOR (2)

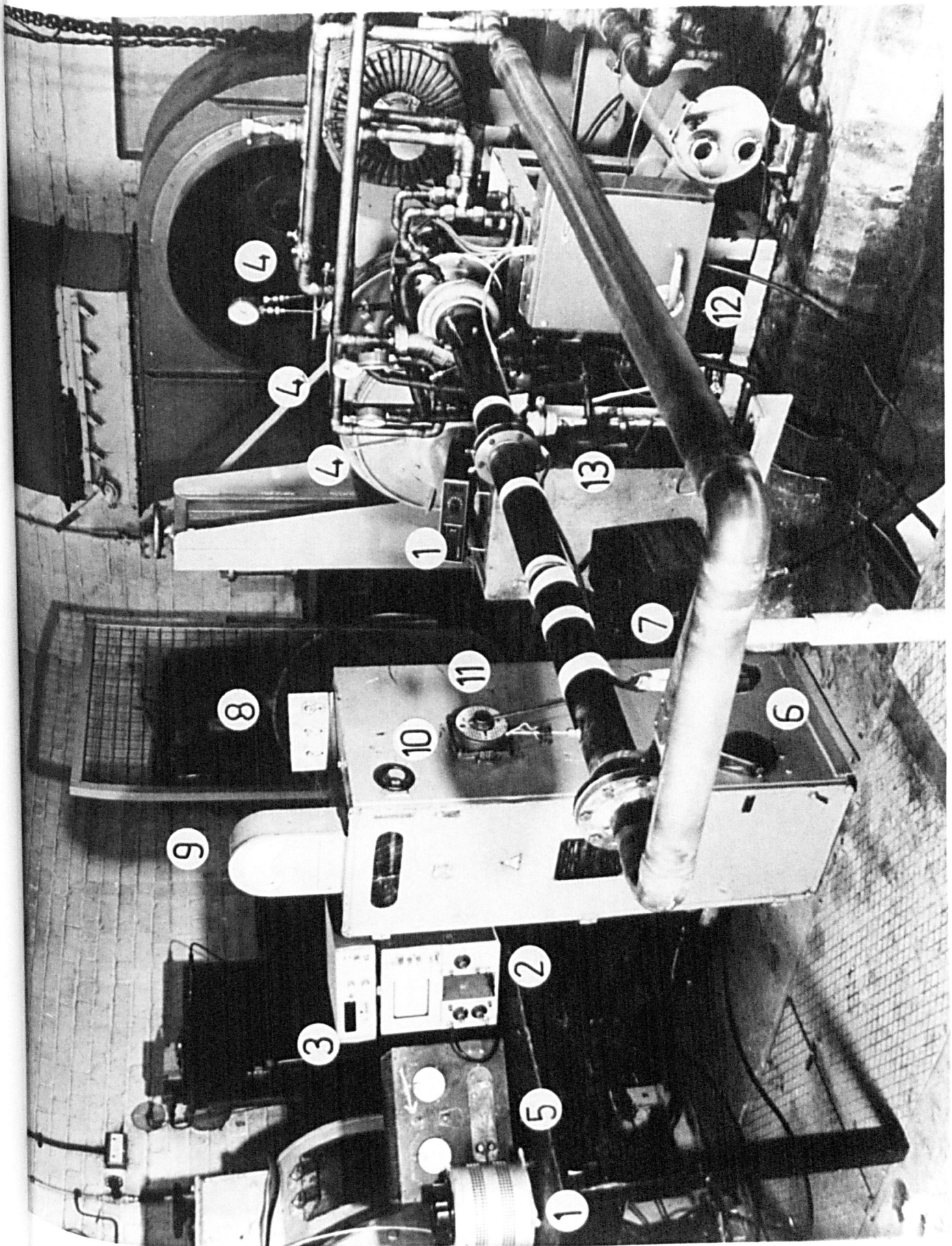


FIG. 6.10 SHOWING LAYOUT OF INSTRUMENTATION

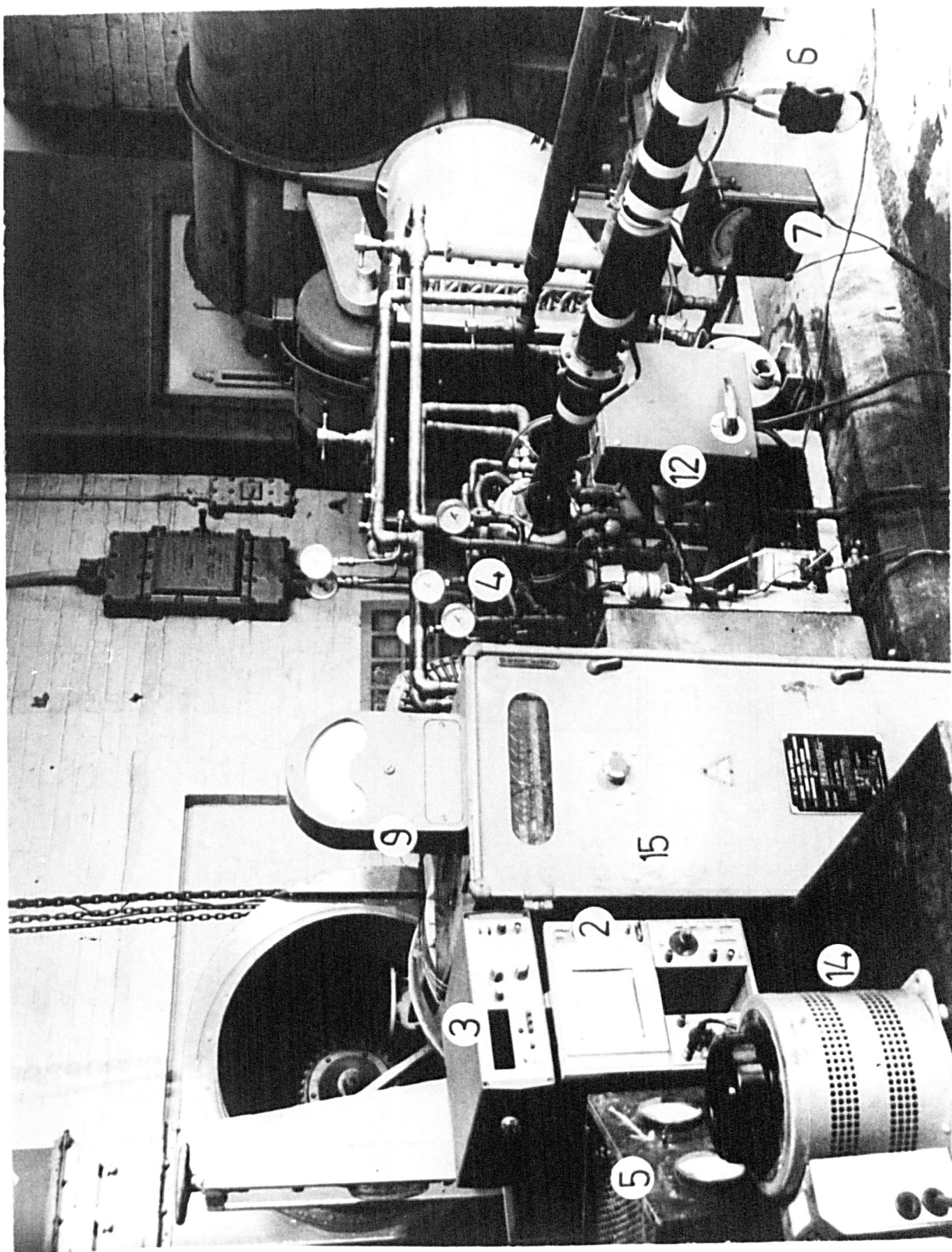


FIG. 6.11 A FURTHER VIEW OF TEST FACILITY
SHOWING LAYOUT OF INSTRUMENTATION

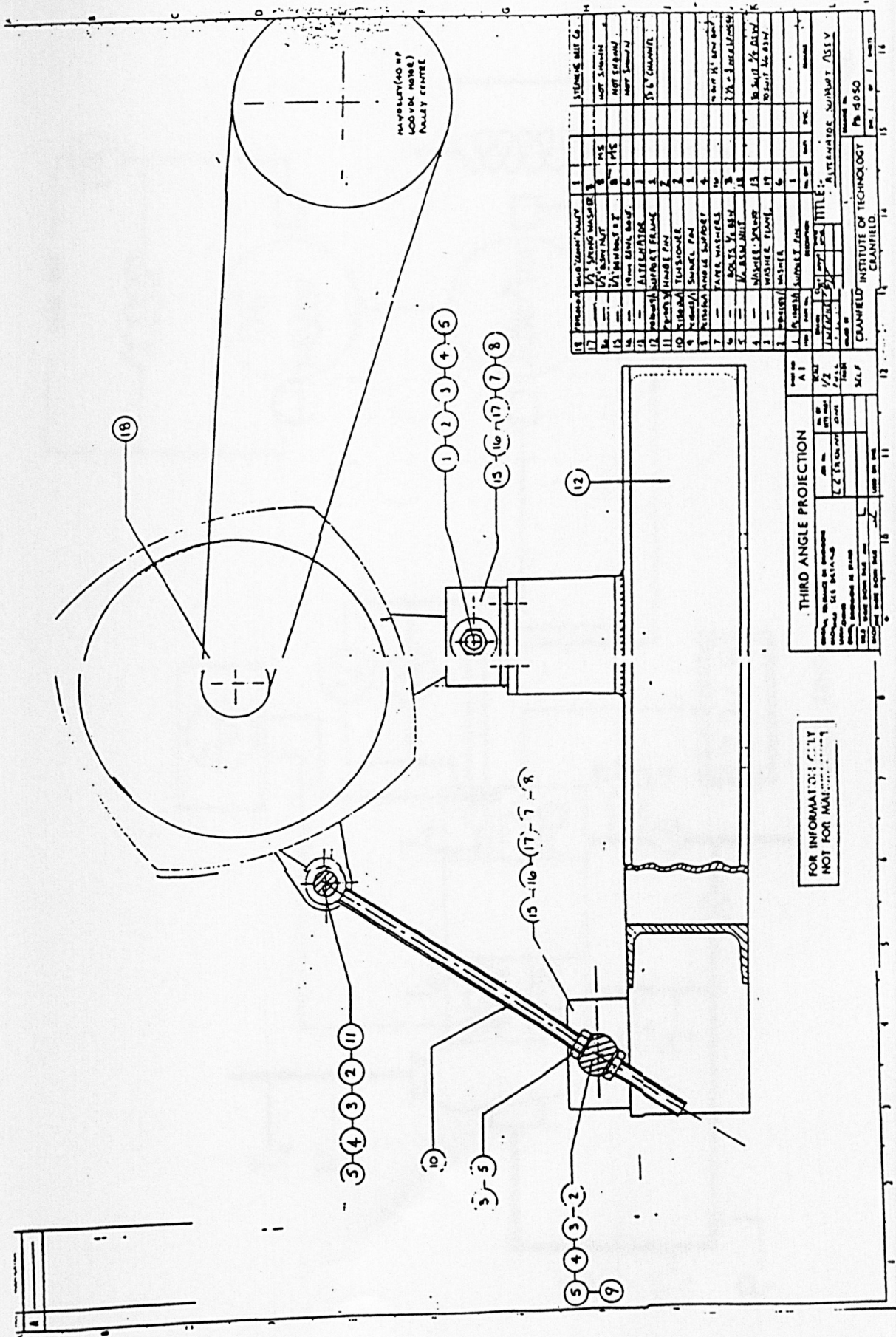


FIG. 6.12 MAWDSLEY/ALTERNATOR SUBFRAME

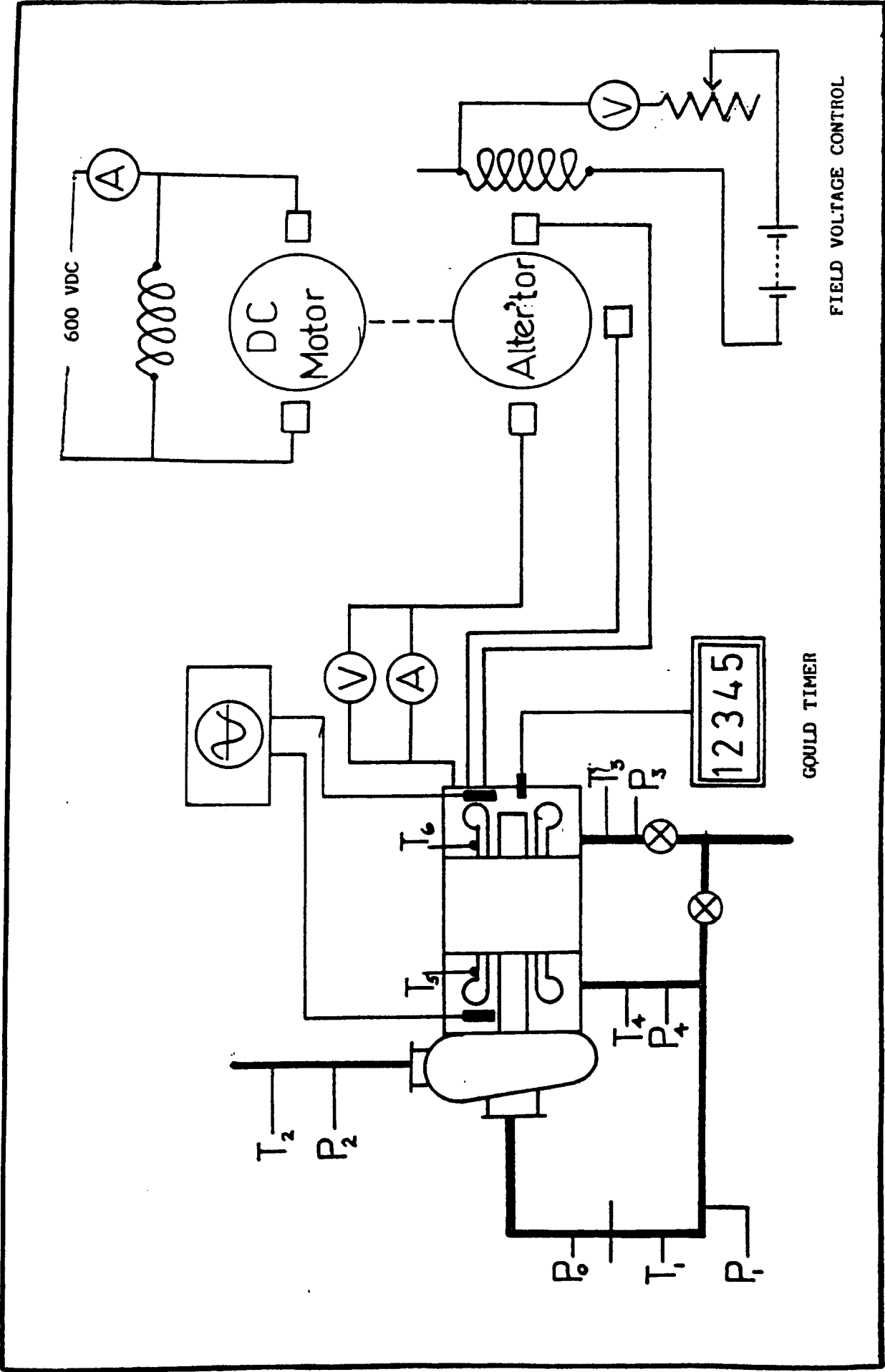
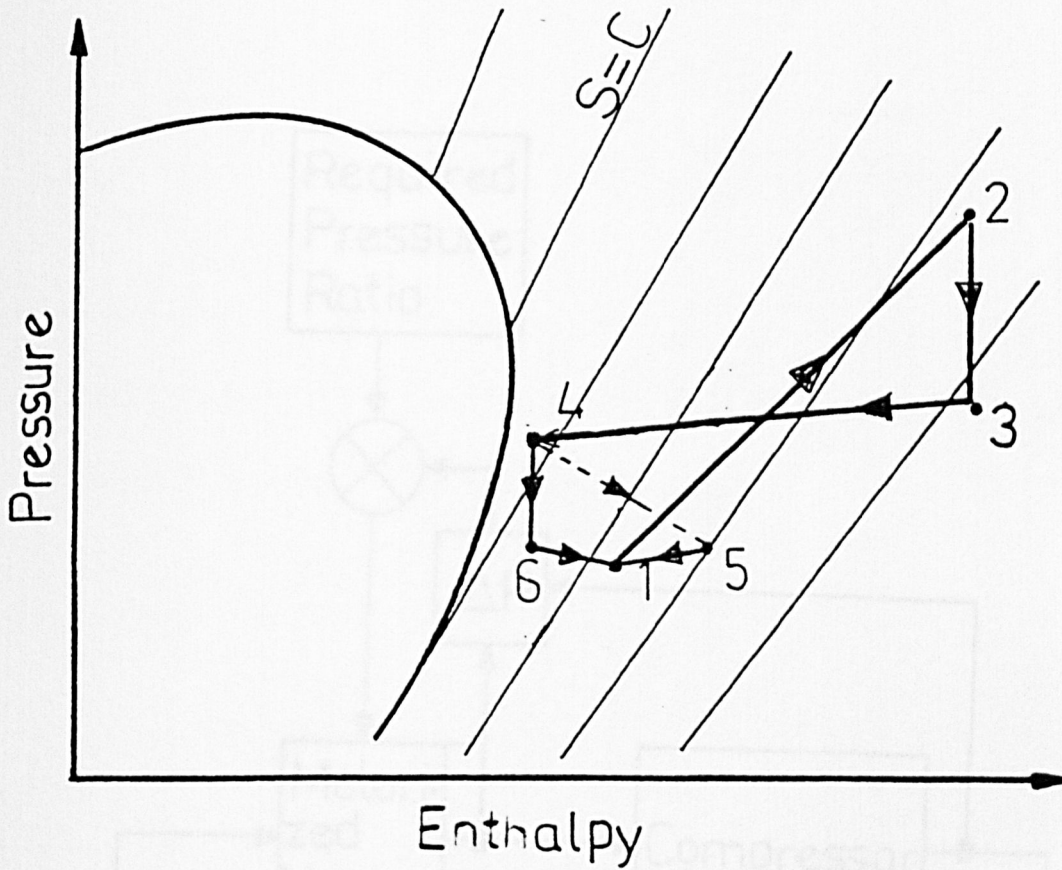


FIG 6.13 COMPRESSOR TEST RIG INSTRUMENTATION



- | | |
|---------|--|
| Process | 1 - 2 Compression |
| | 2 - 3 Discharge Line Throttle |
| | 3 - 6 Heat exchanger, (cooling) plus pressure loss |
| | 6 - 7 Splitter Valve - Throttle |
| | 6 - 5 Heat gain and Pressure Loss in Motor Casing |
| | 7 - 1) |
| | 5 - 1) Mixing down stream of splitter valve |

**FIG. 6.14 MOLLIER DIAGRAM OF REFRIGERANT (FREON 11) VAPOUR LOOP
COMPRESSOR TEST FACILITY**

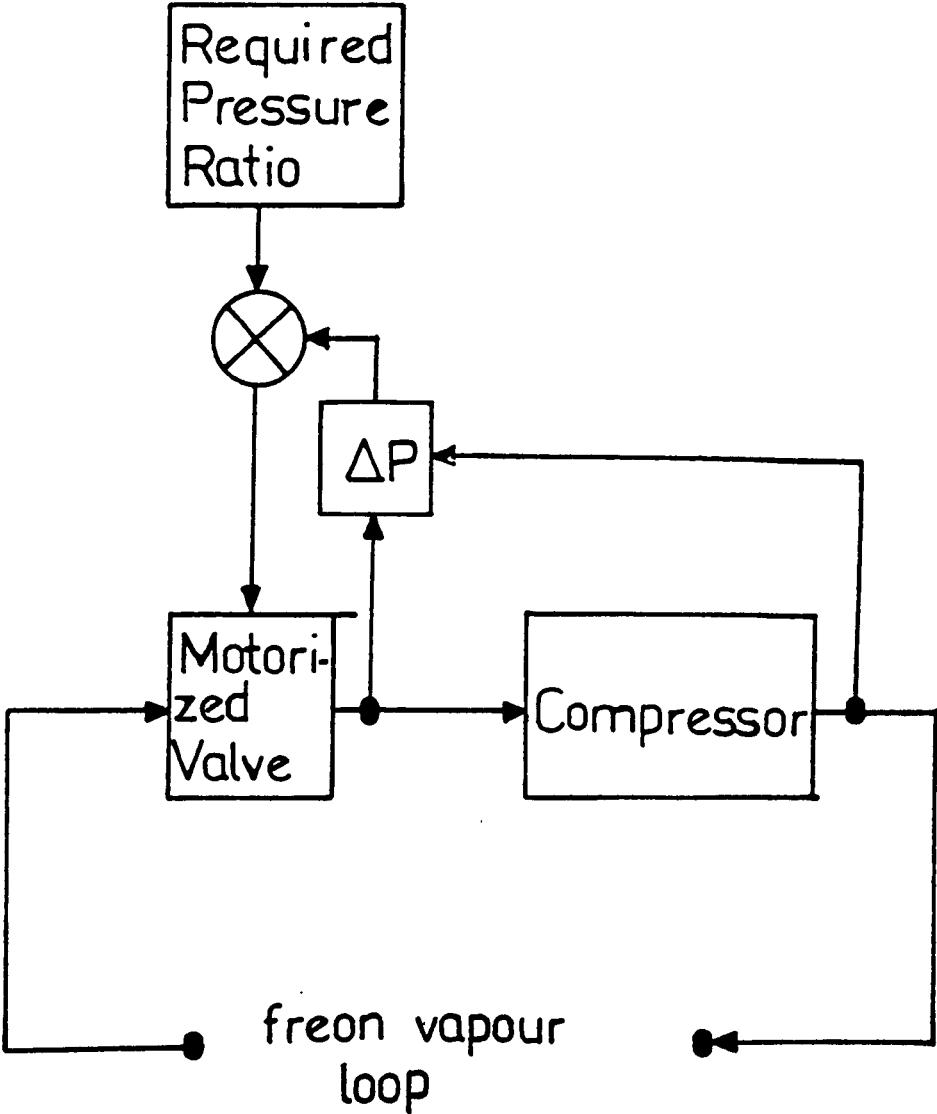


FIG. 6.15 PROPOSED COMPRESSOR RATIO CONTROL

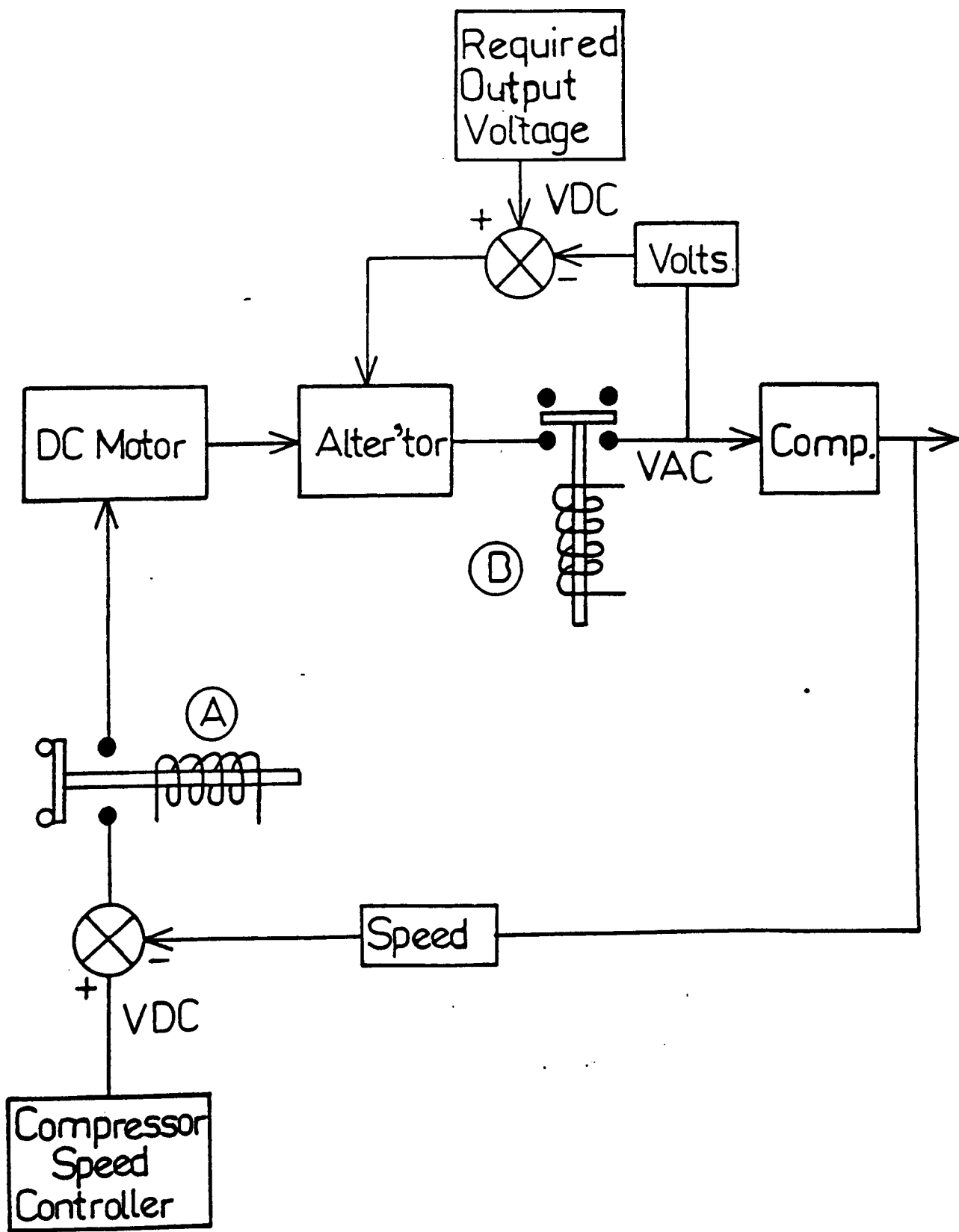
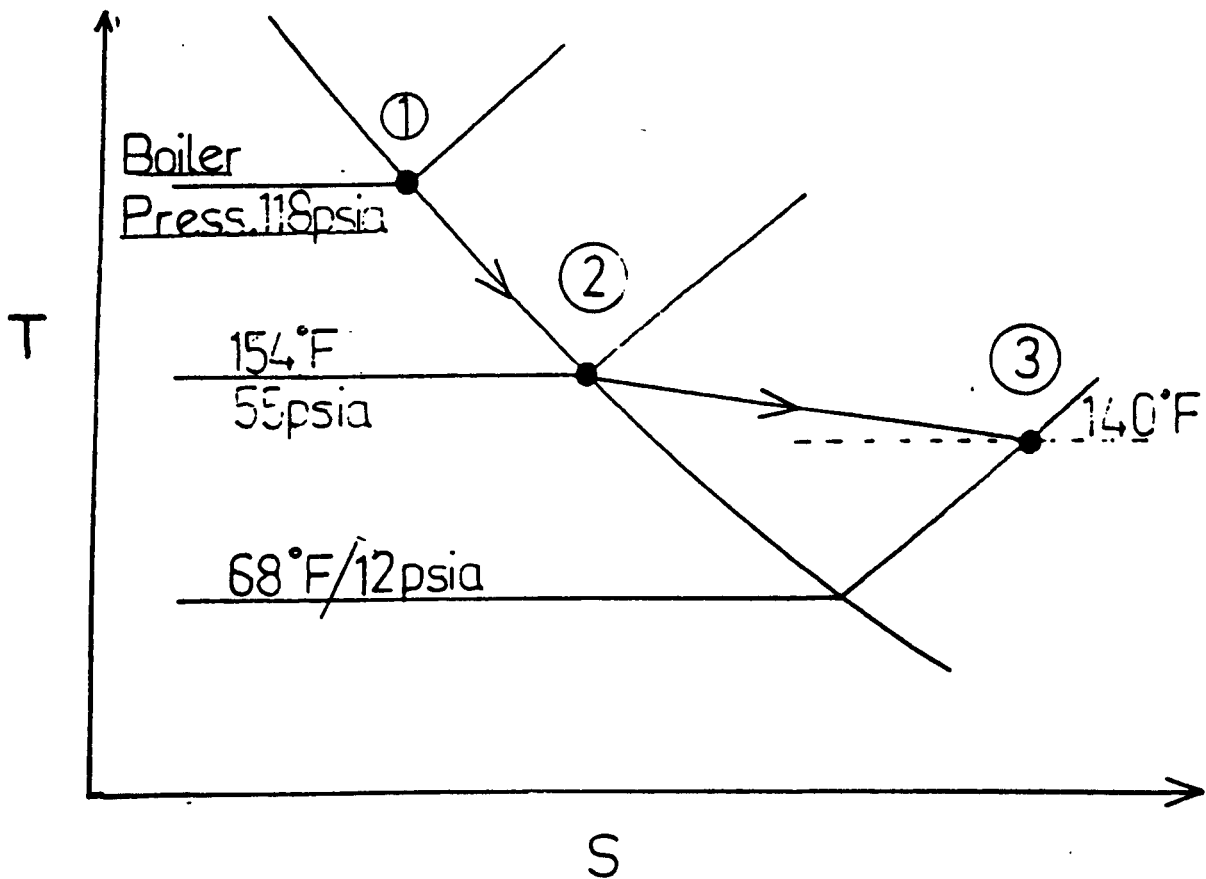


FIG 6.16 PROPOSED CLOSED LOOP CONTROL SYSTEM



Bearing Pressure = 55 PSIA

Bearing Gas Temperature = 154°F

$h_1 = 116.2$ BTU/lb

$h_2 = 110.1$ BTU/lb

$h_3 = 110.1$ BTU/lb

1 Vapour leaves boiler

1 - 2 Heater tape plus pressure losses

2 Vapour enters bearings

2 - 3 Vapour expands into casing

FIG. 6.17 SHOWING T/S DIAGRAM OF LUBRICATION GAS FLOWING FROM BOILER THROUGH BEARINGS INTO MOTOR CASING

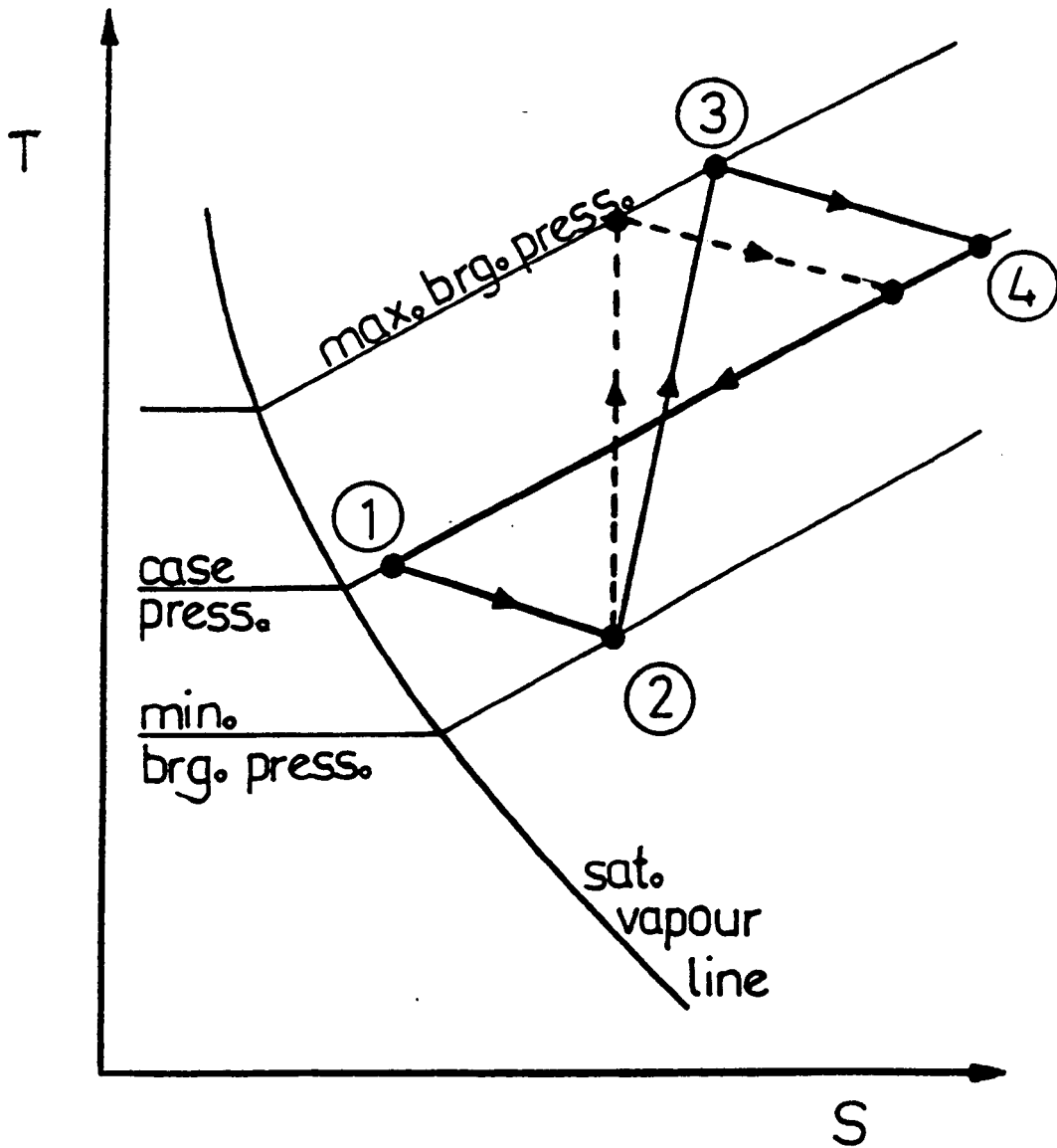


**FIG. 6.18 T-S DIAGRAM OF HYDRODYNAMIC LUBICATION USING A CONDENSABLE
VAPOUR**

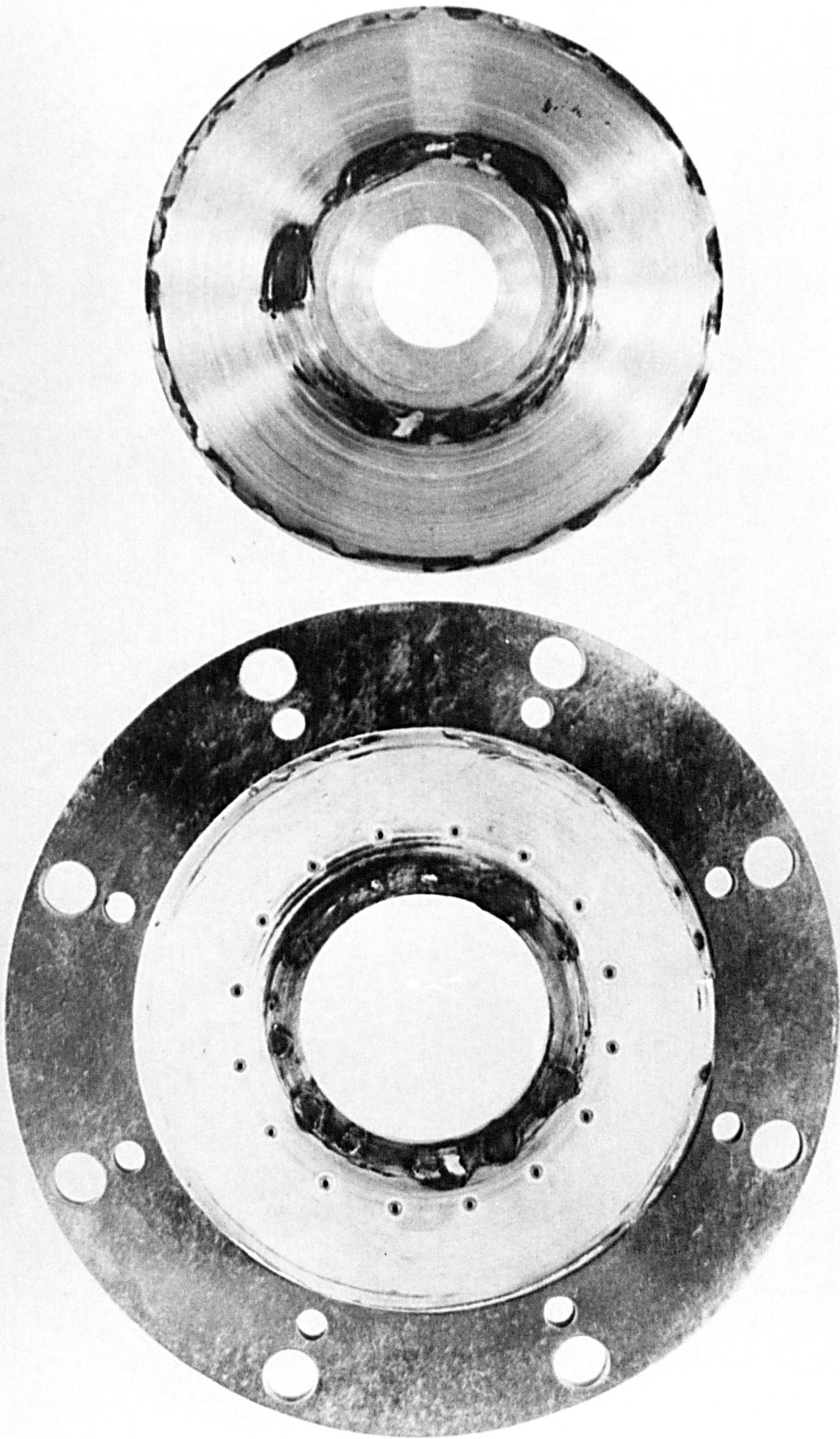


FIG. 6.19 THRUST BEARING REAR FACE SHOWING TOUCH-DOWN MARKINGS

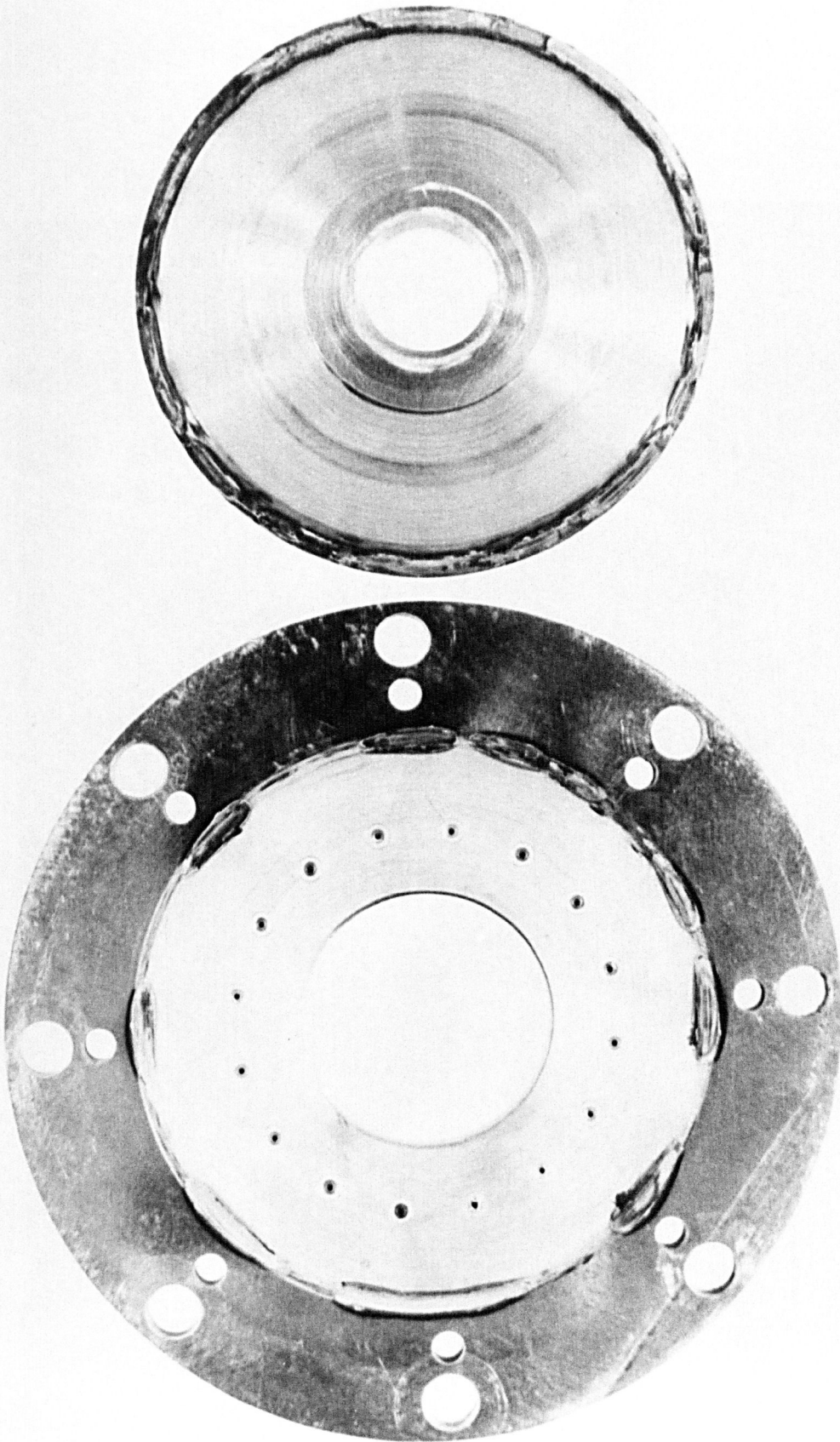


FIG. 6.20 THRUST BEARING FORWARD FACE SHOWING TOUCH-DOWN MARKINGS

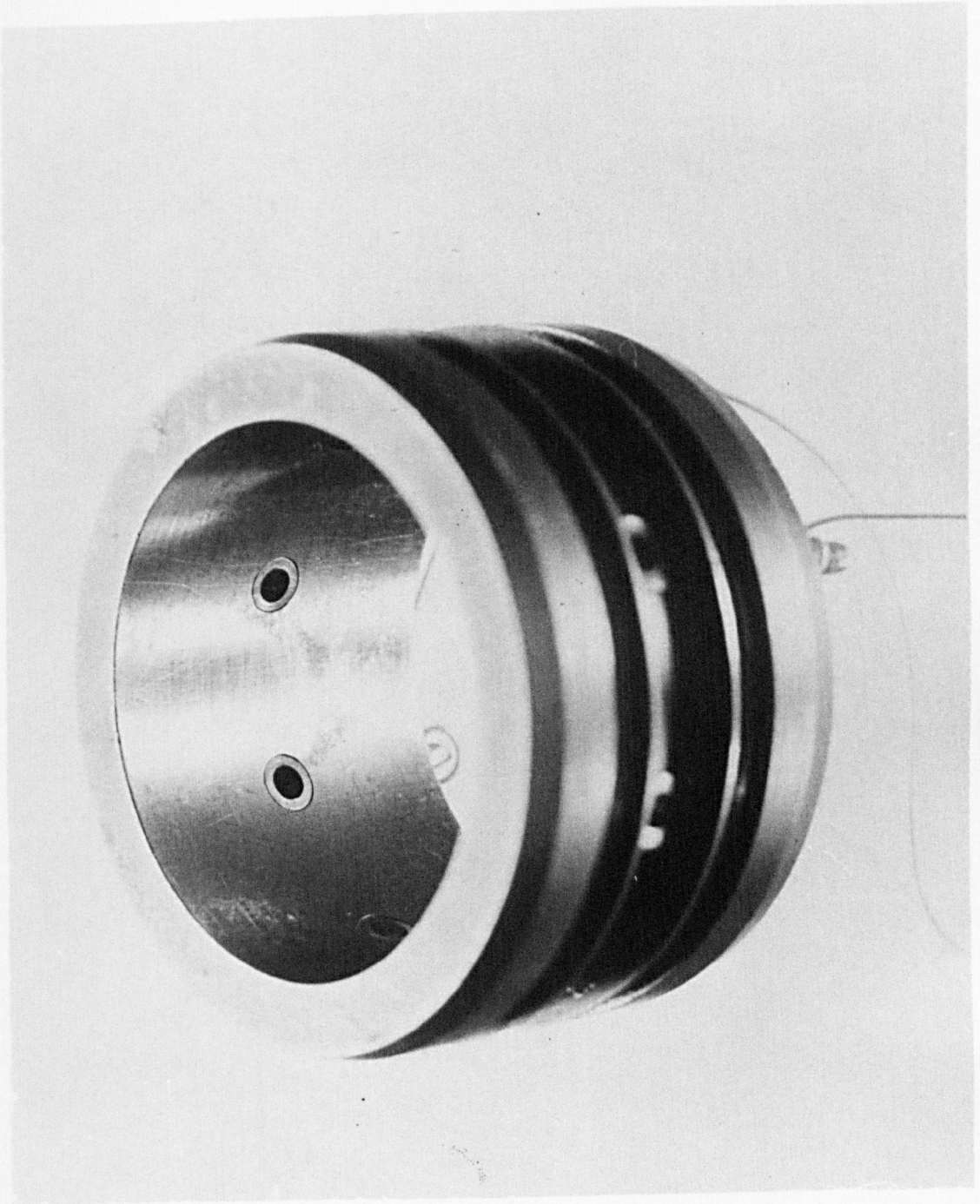


FIG. 6.21 BEARING BUSH

Speed (RPM)	Rear BRG Pressure (PSIG)
30	20
45	18
55	15
65	10
70	18

TABLE 6.1 VARIATION IN REAR BEARING PRESSURE WITH SPEED

ITEM	QTY	COST/ITEM	TOTAL COST	RABTHERM GUIDE 83 STOCK NO.	TYPE
Cu tub ¼" dia	15m		9.13	10014	
" 1½" dia	6m	13.18	26.36	10034	
" 2½" dia	3m	41.89	41.89	10038	
90° elbow 2½"	5 off	5.16	25.80	10498	9090
" 1½"	8 off	0.64	5.12	10494	"
Equal tee 2½"	1 off	10.11	10.11	10538	9130
" 1½"	3 off	1.73	5.19	10534	"
" ¼"	11 off	0.88	9.68	10527	"
45° bend	1 off	0.95	0.95	10508	9120
Equal coupler 1½"	1 off	0.68	0.68	10646	9270
Flow reducer 2½"x2½"	1 off	4.52	4.52	10633	9243
" 1½"x¾"	4 off	0.75	3.0	10592	9240
" 1½"x1½"	1 off	0.90	0.90	10598	"
" 2½"x1½"	2 off	4.52	9.04	10602	"
Capped valve ¼" soldered	1 off	5.51	5.51	11421	CVSZ-4
Check valve ¼" flare	2 off	5.14	10.28	11040	GBI-04F
2 way valve 1½"	3 off	23.18	69.54	10829	ILV-2
2½" compound press gauge	6 off	5.16	30.96	60394	M2-250
¾"x¾" equal tee	4 off	1.32	5.28	10531	9130
1½"x¾" reducer	4 off	0.76	3.04	10592	9240
Low voltage transformer	1 off	34.15	34.15	50183	180W
Heater tape	20M	8.00	8.00	50184	
Thermal insulation tape	100 ft	27.97	27.97	11188	
Drier	1 off	6.57	6.57	10901	C-052
Sleeves	6 off	0.10	0.60	50186	
Thimbles	6 off	0.10	0.60	50187	
¾"x¾" half union	8 off	1.26	10.08	10125	U1-12E
Male flarex pt ¼"x½"	6 off	0.40	2.40	10156	U3-49
Flare nuts long ¼"	12 off	0.30	3.60	10068	N4-4
¼"x¾" half union	4 off	0.43	1.72	10104	U1-4B
¾" dia Cu tube	6m	14.11	14.11	10024	
¾" equal flare tee	4 off	6.00	24.00	10190	
¾" short brass flare nut	20 off	0.71	14.20	10057	NS4-12
¾" equal flare connect ors.	8 off	1.72	13.76	10087	U2-12
Flaring tool	1 off	44.37	44.37	60020	103FS
					¾", ½", 1"
		TOTAL	£483.11		
			£416.60	inclusive of	VAT and
				25% discount	

TABLE 6.2 REFRIGERANT ITEMS FOR TEST RIG PA 15062

CHAPTER SEVEN

COMPUTER SIMULATION OF A REFRIGERATION SYSTEM INCORPORATING THE PROTOTYPE COMPRESSOR

7.1 Introduction

The prototype compressor was designed to replace existing units in packaged air conditioning systems of the type described in the Introduction to the thesis. The aim of this chapter is to describe a theoretical study into the part-load performance of the present compressor unit fitted in an air conditioning refrigerator system. Using a computer based mathematical model developed during this study it was possible to simulate the operation of a refrigerator over a range of ambient temperatures. The effect on the refrigerator performance resulting from changes in design point parameters including compressor speed, evaporator capacity, condenser capacity and compressor pressure ratio, were investigated. The effects of various control scenarios on the performance of both the compressor and refrigeration system were also studied.

The active components of a packaged air conditioner refrigerators are; a compressor, finned coil condenser and evaporator and an expansion device. Figure 7.1 shows a schematic representation of the refrigerator model used in the present study. The mathematical model assumed equilibrium operating conditions throughout. The air flow through both the condenser and evaporator were assumed constant throughout. Pressure losses in pipe work and heat exchangers were neglected.

Mathematical models for each element in Figure 7.1 are described in the Sections 7.3 to 7.9. A sequential method was devised to solve the simultaneous set of equations developed from the component models. This is described in Section 7.11. Results of calculations are discussed in Section 7.12. Conclusions were drawn and recommendations made for future design of a refrigeration system incorporating the present type of compressor.

Centrifugal compressors are susceptible to flow instabilities. Because of this they tend to have narrow operating regimes in terms of mass flow and pressure ratio. In the application of air conditioning systems there is a need to provide some method to control the operation of the compressor in order to maintain an adequate surge margin. An indication of the significance of this problem is given by considering the potential variation in refrigeration load imposed on an air conditioning system. The variation in cooling load for air conditioning systems is discussed by Jones^{7a}, Harris⁷⁷, Pita⁸⁰ and Stoeker and Jones^{2a}. The energy consumption of variable air volume systems of the type envisaged for packaged air conditioning system is discussed in detail by Jones^{7a}. The wide variation in operating conditions experienced by air conditioning plants require refrigerant compressors with equally wide operating maps; a characteristic which centrifugal compressors do not readily possess. Because of the importance of part-load control on the successful application of the prototype compressor this aspect of the design will be considered first.

7.2 Part-Load Control

A literature survey has suggested, (Table 7.1), the most widely used methods of controlling centrifugal compressors used in air conditioning systems.

Table 7.1 METHODS OF CONTROLLING CENTRIFUGAL COMPRESSORS

- (a) Compressor speed variation.
- (b) Compressor suction line throttling.
- (c) Hot gas bypass.
- (d) Variable geometry radial diffuser.
- (e) Variable inlet guide vanes.

7.2.2 Compressor Speed Variation

Hermetic and semi-hermetic compressors are most commonly driven by induction motors. Although variable frequency speed controls are available the additional cost inhibits

their general adoption. One technique uses switchable stator windings. This technique has been adopted in the EDPAC TECH80^{®1} system which utilizes a dual, two speed, hermetic reciprocating compressor unit giving four separate operating characteristics. A further example is given in reference 82.

The technique of pole switching is unsuitable for centrifugal compressors because the flow-speed relationship would result in an excessive reduction in both pressure ratio and flow, to be of value in air conditioning applications. Similarity laws for turbomachines show that switching from two pole to four pole operation would reduce the maximum operating pressure ratio by four times and halve the maximum volumetric flow.

Solid-state electronic speed controllers are available from manufacturers. For example Hellerman Electronic Components, Burns^{®3}, have quoted for a 25kW variable speed controller costing £5833. However this unit has a maximum frequency output of 100 Hz, which is too low for the present application which requires a supply of 14kW with frequency range of 400 Hz to 1,200 Hz. The cost of such a device at present would be uncompetitive, Plenty^{®4}, Embrey^{®5} and Williams^{®6}. Further research and some development is required in this area. At the present time neither variable speed nor constant speed solid state controllers are available from manufacturers with the desired characteristics.

The rotary-inverter type system used during the testing of the present prototype in its present form is also unsuitable because of cost and weight penalties.

7.2.3 Compressor Inlet-Line Throttling

Throttling the inlet supply line causes the compressor to act as though the evaporator temperature had fallen. As flow throttling proceeds the operating point will move from 'A' to 'B' on the constant speed characteristic shown in Figure 7.2.

In comparison with variable speed control, inlet throttling tends to be less efficient because it forces the compressor to operate within narrow choke and surge margins in regions of low isentropic efficiency.

An inlet throttling device can be relatively inexpensive to install and may consist of a simple butterfly valve and control linkage. It is thought that such a throttle should be positioned well upstream of the impeller, possibly with an intervening flow straightener to reduce compressor inlet flow distortion which otherwise may result in a reduction of compressor surge margin.

Because of the simplicity of this type of control device it was decided to incorporate it into the general system model in order to simulate the effect on system performance.

7.2.4 Variable Inlet Guide Vanes

The effect of variable inlet guide vanes on the performance of a compressor was described by Carrier²⁷.

Cost and additional manufacturing complexity are the main disadvantages to the use of variable guide vanes. In this particular case a difficulty of adopting this option is the physical size of the vanes required, particularly as the ducting at the impeller entry is a only 47mm in diameter with an impeller vane height of 16mm.

7.2.5 Variable Radial Diffuser Vanes

Diffuser flow area usually limits the flow range of a compressor because it is in this region that the flow usually limits the high flow but also the low flow because of diffuser vane stall. By varying the incidence angle of the diffuser vanes it is possible to vary the throat area of the diffuser and thereby widen the operating margins of the compressor. This effect, shown schematically in Figure 7.3, has been investigated by Elder²¹.

Disadvantages of using variable diffuser vanes are manufacturing complexity and cost. As with variable inlet guide vanes the physical size and complexity of variable diffuser vanes suggests that this approach may be unsuitable in the design of present compressor.

7.2.6 Variable (Vaneless) Diffuser Flow Area

This method produces similar results as the variable vane technique. An arrangement used by Carrier²⁷ is to make one wall of the diffuser movable. As the flow rate increases the position of the movable wall alters to give an increased

flow area and in doing so delays choking. As with other variable geometry techniques the main disadvantages are thought to be manufacturing complexity and cost.

7.2.7 Hot-gas Bypass

Hot-gas Bypass is a technique in which compressor discharge gas is directed, via a suitable control valve, back to the compressor inlet causing the refrigeration effect of the system to fall. The overall effect is to reduce the system COP.

When applied to systems with centrifugal compressors the effect of using the hot gas bypass technique is to move the operating point away from the surge point at low evaporator pressures and/or high condenser pressures. Hot-gas bypassing is not a completely satisfactory solution to the part-load because the power consumption of the compressor is likely to be greater than it would otherwise be due to the artificial load created by the bypass gas. However, an advantage of the technique is that it is relatively simple to implement and for this reason is often included in capacity control systems.

7.2.8 Capacity Modulation of the Carrier 19DM Centrifugal Chiller

The 19DM centrifugal chiller unit, manufactured by the Carrier Corporation provides an example which uses three separate techniques of capacity control. The system uses variable I.G.V.'s, variable diffuser passage height and hot gas bypass.

I.G.V.'s are activated by electrical actuators controlled by a central electronic control system which constantly monitors evaporator temperature. Diffuser wall position is activated by a hydraulic servo system. Hot gas bypass valve is controlled by a proportioning flow valve activated by an electrical actuator controlled by a central electronic control system.

7.3 Evaporator Model

The thermodynamic characteristic of a finned tube evaporator can be described by Equation 7.1 taken from Stoeker and Jones²⁹.

$$\dot{Q}_e = (\dot{m}.C_p)_a . \epsilon . (T_a - T_1) \quad (7.1)$$

where T_a = temperature of air entering the evaporator and,

T_1 = saturation temperature of the refrigerant.

ϵ = heat exchanger effectiveness

The effectiveness of an evaporator is determined from Equation 7.2.

$$\epsilon = 1 - \exp\left(\frac{-U.A}{\dot{m}.C_p}\right) \quad (7.2)$$

If the air flow rate across the evaporator coils is maintained at the design value then effectiveness becomes a function of the convective heat transfer coefficient between the refrigerant and the inside surface of the evaporator tubes, (h_i), which may be determined from Equation 7.3.

$$h_i = A_1 . \dot{m}_r^{-n} \quad (7.3)$$

where A_1 = constant

Assuming that the refrigerant viscosity and density remain constant then the reported values of 'n' in Equation 7.3 range from 0.5 to 0.87 depending upon the conditions of the vapour, Kreith³⁰. Trotter³¹ suggested a relationship of the form shown in Equation 7.4. Assuming that the refrigerant flow rate is approximately proportional to the heat transfer rate then the value of 'n' can be assumed to be 0.55, Equations 7.4.

$$h_i = A_1 . \dot{m}_r^{0.55} \quad (7.4)$$

The air side of the tubes produces the most significant thermal resistance. For this reason the performance of the evaporator with constant air flow can be described by Equation 7.1 in which $(m.C_p)_{\text{air}}$ is constant and so evaporator capacity, Q_{ev} , becomes a linear function of ambient air to refrigerant temperature difference only. This is known as the basic rating technique.

Trott²⁷ suggested that the basic rating technique can be modified to account for small variations in capacity which occur in evaporators due to changes in refrigerant flow. Stoecker and Jones suggested that part-load performance can be estimated using Equation 7.5 in which 'G' is a function of the temperature difference, Equation 7.6. This makes Q_{ev} a quadratic function in $(T_{\text{air}} - T_1)$.

$$\dot{Q}_{\text{ev}} = G.(T_{\text{air}} - T_1) \quad (7.5)$$

$$G = a + b.(T_{\text{air}} - T_1) \quad (7.6)$$

Following Stoecker and Jones²⁸, Equation 7.7 was derived which allows the part-load performance of an arbitrary evaporator to be determined if both the capacity and air to refrigerant temperature difference of the evaporator are known at the design condition.

$$\frac{\dot{Q}_{\text{OD}}}{\dot{Q}_{\text{DP}}} = \frac{((a + b.(T_{\text{air}} - T_1)).(T_{\text{air}} - T_1))_{\text{OD}}}{((a + b.(T_{\text{air}} - T_1)).(T_{\text{air}} - T_1))_{\text{DP}}} \quad (7.7)$$

where Q_{OD} and Q_{DP} = Off-Design and Design-Point capacities respectively.

Also a and b are dependent on heat exchanger effectiveness which for a particular evaporator are assumed to be constants.

In the present case the values of the constants ' a ' and ' b ' were determined from empirical data taken from an ACR Heat Exchanger catalogue²⁹;

$$\begin{aligned} a &= 0.0245 \\ b &= 4.733 \times 10^{-4} \end{aligned}$$

Figure 7.4 shows the variation in part-load performance of an evaporator predicted using Equation 7.7. The

characteristic upward curve of the graph is typical of finned tube evaporators.

7.4 Condenser Model

Because of the low thermal resistance on the refrigerant side of condenser coils the part-load performance of condensers is approximately independent of the refrigerant flow rate and the basic rating G , can be assumed constant and the capacity of a condenser is described by Equation 7.8.

$$\dot{Q}_{C,DP} = G \cdot (T_{\varphi} - T_a) \quad (7.8)$$

From which;

$$G = \dot{Q}_{C,DP} / (T_{\varphi} - T_a)_{DP} \quad (7.9)$$

where T_{φ} = saturation temperature of refrigerant
in the condenser and,

T_a = temperature of air entering the condenser

At part-load conditions the performance of a condenser can be described by Equation 7.10.

$$\dot{Q}_{C,OD} = G \cdot (T_{\varphi} - T_a) \quad (7.10)$$

Substituting Equation 7.9 for 'G' in Equation 7.10 gives;

$$\frac{\dot{Q}_{C,OD}}{\dot{Q}_{C,DP}} = \frac{(T_{\varphi} - T_a)_{OD}}{(T_{\varphi} - T_a)_{DP}} \quad (7.11)$$

Equation 7.11 characterises the part-load performance of refrigerant condensers.

7.5 Compressor Model

The modelling of catalogue data for reciprocating compressor characteristics was discussed by Stoecker and Jones²⁰ who suggested that power and refrigeration capacity characteristics can be approximated using Equations 7.12(a)

and (b).

$$\dot{Q}_e = (a_1 + b_1 T_1 + c_1 T_1^2)(a_2 + b_2 T_q + c_2 T^2) \quad 7.12a$$

$$\dot{P}_c = (d_1 + e_1 T_1 + f_1 T_1^2)(d_2 + e_2 T_q + f_2 T^2) \quad 7.12b$$

where $a_1, a_2, \dots, f_2$ are constants

Trott²⁷ suggested a shortened form of polynomial in Equation 7.13.

$$\dot{Q}_e = a + b.T_q + c.T_1 + d.T_1^2 \quad (7.13)$$

Experience has shown that Equation 7.13 provides sufficient accuracy for modelling compressor data. The performance characteristics of the present compressor for operation on Freon 11 were determined by Elder and Walters³. The efficiency/flow and pressure ratio/flow characteristics are reproduced in Figure 7.5 and 7.6. From the original data the corrected flow rate is defined as:

$$\dot{V} = \frac{60.\dot{m}_r \cdot \sqrt{T_s/288}}{(P_s/14.7) \cdot \rho_s} \quad (7.14)$$

where $\dot{m}_r$ = refrigerant flow rate in lbs/s
 P_s = compressor inlet pressure in psia
 T_s = compressor inlet temperature
 ρ_s = density of refrigerant at standard temperature and pressure taken as 14.7 psia and 288K

In the original data refrigerant density was determined from the ideal gas equation. In order to allow the data to be converted to its uncorrected values a standard density of 0.3629 lbs/ft³ was used determined from the ideal gas law at

the standard temperature and pressure, (288K and 14.7 psia).

The rotational speed, N , has been corrected with respect to standard temperature, (288K). This relationship is given in Equation 7.15.

$$\bar{N} = \frac{N}{\sqrt{(T_0/288)}} \quad (7.15)$$

The characterization of Figures 7.5 and 7.6 were undertaken using polynomial equations. In the case of the efficiency characteristic shown in Figure 7.5 it was assumed that;

$$\eta_c = (a_3 + b_3 \cdot \dot{V} + c_3 \cdot \dot{V}^2) \cdot (a_4 + b_4 \cdot \dot{N} + c_4 \cdot \dot{N}^2) \quad (7.16)$$

The values of the constants in Equation 7.16 are listed below in Table 7.2.

Table 7.2 CONSTANTS FOR COMPRESSOR EFFICIENCY MODEL

$a_3 = -0.8446$	$a_4 = -7.659$
$b_3 = 0.6808$	$b_4 = 17.291$
$c_3 = -8.904 \times 10^{-2}$	$c_4 = -8.5425$

Where $\bar{N} = \frac{N}{6 \times 10^4 \sqrt{(T_0/288)}}$

In Equation 7.16 the shape of the characteristic was modelled using the data available at the corrected rotational speed of 60,000 rpm. The other characteristics are scaled from this using a scaling factor which is dependent on the ratio between the corrected speed of interest and the reference speed, (60000 rpm in this case).

Because of the complexity of the pressure ratio characteristic it was found that no single quadratic equation would adequately model its form. Two separate functions, F_1 and F_2 were used to describe the pressure ratio characteristic at the reference speed, (66000 rpm), Equations 7.17(a) and 7.17(b).

$$F_1 = (a_5 + b_5 \cdot \dot{V} + c_5 \cdot \dot{V}^2 + d_5 \cdot \dot{V}^3) \quad (7.17a)$$

$$F_2 = (a_6 + b_6 \cdot \dot{V} + c_6 \cdot \dot{V}^2) \quad (7.17b)$$

Due to the complexity of the pressure ratio characteristic shown in Figure 7.6 it was necessary to apply separate scale factors to both (corrected) volumetric flow rate, (VSF) and pressure ratio, (PSF). These scaling functions are given in Equations 7.18a and 7.18b

$$VSF = a_7 + b_7 \cdot N + c_7 \cdot N^2 \quad (7.18a)$$

$$PSF = a_8 + b_8 \cdot N + c_8 \cdot N^2 \quad (7.18b)$$

In this case;

$$N = \frac{N}{66 \times 10^3 \cdot \sqrt{(T_0/288)}}$$

The values of the constants in Equations 7.17a to 7.18b are listed in Table 7.3.

Table 7.3 CONSTANTS FOR PRESSURE RATIO MODEL

$a_5 = 917.5$ $b_5 = -571$ $c_5 = 119$ $d_5 = -8.288$	$a_6 = -8.42$ $b_6 = 5.944$ $c_6 = -0.693$
$a_7 = 2.51$ $b_7 = -3.52$ $c_7 = 2.05$	$a_8 = -8.38$ $b_8 = 16.2$ $c_8 = -6.47$

The compressor characteristic equations were written into a computer program, COMPERF, which provides design point performance solutions. This program is listed in Appendix B.

Figure 7.7 shows the variation in refrigeration effect for the present compressor. The parameters used in Figure 7.7 are the same as those adopted by manufactures when presenting refrigeration compressor capacity data.

The COMPERF program assumes that there are no system restrictions placed on the compressor. The solution domain of COMPERF is shown in Figure 7.8. Outside the solution

domain the program predicts that the compressor either chokes or surges.

Assuming that the preferred operating characteristic provides equal surge and choke margins then referring to Figure 7.8 the equation describing this characteristic in terms of condenser and evaporator temperatures is;

$$T_2 = 1.3636 T_1 + 49.091$$

7.6 Electric Motor Characteristic

It was decided to include a drive motor into the system model which would allow the effects of motor cooling by the suction gas cooling on the performance of the compressor to be investigated and also give an indication of the electrical power consumption of the compressor.

Figure 7.9 shows a typical induction motor characteristic. It would be unusual to operate a motor in the region to the left of the peak torque value because operating in this part of the characteristic would be unstable. Because of this it was only necessary to model that part of the characteristic to the right of the peak torque value.

Fitting a polynomial to the right-hand side of the torque characteristic in Figure 7.9 yields:

$$\frac{\tau_{od}}{\tau_{dp}} = -19 + 57.33.\phi - 38.33\phi^2 \quad (7.19)$$

$$\text{for } 0.748 < \phi \leq 1$$

Equation 7.20 describes the variation in motor efficiency for $0.748 < \phi \leq 0.96$. For slippage factor values greater than 0.96 it was assumed that the motor efficiency remains constant at its optimum values.

$$\frac{\eta_{m,od}}{\eta_{m,dp}} = 0.08084 + 0.2155.\phi + 0.77288 \phi^2$$

$$\text{for } 0.748 < \phi \leq 0.96 \quad (7.20a)$$

and,

$$\frac{\eta_{mod}}{\eta_{mdp}} = 1$$

for $\phi > 0.96$ (7.20b)

To solve Equations 7.19 and 7.20 it is necessary to determine the torque absorbed by the compressor. Neglecting mechanical friction and windage losses;

$$\dot{W}_C = 2 \cdot \pi \cdot \phi \cdot f \cdot T \quad (7.21)$$

therefore: $T_{DP} = \frac{W_C}{2 \cdot \pi \cdot \phi_{DP} \cdot f} \quad (7.22)$

Assuming that $\phi_{DP} = 0.96$, (the assumed value for maximum efficiency),

Then,
$$\begin{aligned} T_{od}/T_{dp} &= 0.96 \cdot W_{C,od} / (\phi \dot{W}_{C,dp}) \\ &= -19 + 57.33\phi - 38.33\phi^2 \end{aligned}$$

Thus;

$$\dot{W}_{C,od} = (-19.79\phi + 59.72\phi^2 - 39.92\phi^3) \quad (7.23)$$

for $0.747 < \phi \leq 1$

After solving Equation 7.23 for ϕ , the efficiency of the motor is determined from Equation 7.20 given the value of the motor efficiency at the design point.

Electrical power consumption by the compressor was determined from Equation 7.24.

$$P_E = \dot{W}_C / \eta_m \quad (7.24)$$

$$(h_3 - h_2) = (\dot{W}_C / \dot{m}_r) \cdot (1 / \eta_m - 1) \quad (7.25)$$

The heat absorbed by the refrigerant as it passes through the compressor motor casing, (cooling the motor), was estimated from Equation 7.25 which assumes that the motor casing is adiabatic.

Pressure losses would occur in the motor casing. To enable this effect to be included in the model it was assumed that the pressure loss would be proportional to the dynamic pressure at the inlet to the motor casing. By specifying the design point value of pressure loss the effect at part-load conditions was investigated by using the ratio described in Equation 7.26

$$\frac{(P_2 - P_3)_{OD}}{(P_2 - P_3)_{DP}} = \frac{(\dot{m}_r)_{OD}^2}{(\dot{m}_r)_{DP}^2} \frac{\rho_{DP}}{\rho_{OD}} \quad (7.26)$$

7.7 Suction Line Throttle, (SLT)

It was assumed that the pressure reduction across a suction line throttle, (S.L.T.), would occur at constant enthalpy. Given a set of part-load conditions the aim of a control system, (via the SLT), would be to modify the compressor inlet pressure and enthalpy so that the compressor would operate within a specified pressure ratio-flow region.

The system conditions set by the S.L.T. are mathematically:

$$h_3 = h_4 \quad (7.27)$$

$$P_3 \geq P_4 \quad (7.28)$$

7.8 Hot Gas Bypass Valve

The hot-gas bypass valve maintains the pressure difference

across the compressor whilst allowing a proportion of refrigerant gas to be bled from the compressor discharge to inlet. This would cause the refrigerating capacity to change and move the operating point of the compressor to a more satisfactory region. The conditions imposed by the hot gas bypass valve are described in Equations 7.29 and 7.30.

$$h_5 = \frac{\dot{m}_{r3} \cdot h_6 + \dot{m}_{r4} \cdot h_4}{\dot{m}_{r3} + \dot{m}_{r4}} \quad (7.30)$$

$$\dot{m}_{r6} = \dot{m}_{r3} = \dot{m}_{r3} + \dot{m}_{r4} \quad (7.31)$$

$$\dot{m}_{r7} = \dot{m}_{r4}$$

7.9 Expansion Valve

It was assumed that the expansion valve is of the liquid bulb thermostatic type designed to maintain a constant suction line super heat temperature.

Thermodynamically expansion valves are constant enthalpy throttle devices. This constraint is described by Equation 7.31. In addition the model assumes a constant difference between the saturated vapour temperature, T_1 , in the evaporator and the temperature at the inlet to the motor casing, T_2 ; by virtue of the thermostatic expansion valve operation. The value of this temperature difference is selected at the design point. Typical values are in the range of 5 to 8°C.

$$h_9 - h_1 = 0 \quad (7.31)$$

$$(T_2 - T_1) = \text{design point value} = \text{constant} \quad (7.32)$$

The transient behaviour of thermostatic valves was not considered. It was assumed that the performance of the expansion valve matched precisely that required of it by the system over the entire operating regime investigated as suggested by Stoecker and Jones²⁰.

7.10 Refrigerant Data

It was considered important that the computer model should incorporate accurate thermodynamic property data for the process fluid. For the present study the property correlations devised by Downing⁷¹ based on the work of Martin and Hou⁷² were used. The former provides an excellent source of data for estimating the thermodynamic properties of twelve fluorocarbons whilst the latter provides a detailed description of the theory underlying the well documented Martin-Hou equation of state.

A computer program was written to determine the thermodynamic properties of Freon 11. A listing of the program is included in the system computer models in Appendix B and C.

7.11 Method of Solution

An iterative method was adopted to solve the set of simultaneous equations described in Sections 7.3 to 7.10. This process required the setting up a system of guesses and checks to enable a convergent solution to be made.

Two programs were written. The first to carry out design point calculations and the second to undertake the part-load calculations. The structure of these programs have many common features including the refrigeration property data sub-routines and compressor data sub-routine. A description of COMP-OD, the part-load program, will serve to explain the structure of both programs.

A flow diagram of COMP-OD is given in Figure 7.10. This diagram provides an over view of the program. The program itself, listed in Appendix C, is well documented in order to enable the logic to be accessible if the program is read in conjunction with Figure 7.10.

The nested arrangement of 'guesses' and 'checks' is shown in Figure 7.10, in the 'Control Program' part of COMP-OD, (see Appendix C, lines 240 to 1040). The program is made up from twenty four sub-routines. The purpose of the 'control program' is to direct the order in which sub-routines are operated. The flow diagram in Figure 7.10 reflects this.

7.12 Operation

There are two parts to the operation of the computer model. The first requires the selection of the design point conditions and the calculation of design point data, and the second, the determination of the part-load performance of the compressor and refrigerator over a selected range of ambient temperatures.

Design point calculations require the specification of ambient temperature, evaporator and condenser temperature, motor efficiency and the pressure loss through the motor casing. From this data design point power, refrigerant mass flow, condenser and evaporator heat transfer, compressor efficiency are calculated using the COMPERF program listed in Appendix B.

The part-load calculations are carried out using COMP-OD program listed in Appendix C. This program uses the design point data calculated by the COMPERF program. Access to the program is given directly from the computer keyboard and automatically via the COMPERF program.

The structure of the Model is shown in the flow diagram in Figure 7.11. The COMPERF program allows the user to modify the design-point data until a satisfactory operating point is predicted. Once this is achieved the COMPERF program OUTPUTs the data via a printer and VDU. A data file is then OPENed and COMP-OD is CHAINED. The COMP-OD program asks the user to select the range of part-load ambient conditions over which data is required. The program calculates the required data at each specified data point and provides a printed copy of the results. The simulation of hot-gas bypass and throttle line system controls are automatically invoked when required by the program to maintain unspecified choke and surge margins.

In order to reduce the running time of the COMP-OD program, a G-Code image of the BASIC program was produced. This cut the program operating time by thirty percent.

7.13 Discussion

Design-point calculations were carried out using the computer program COMPERF. Table 7.4 lists three conditions of interest including an assumed design condition based on

information provided by Graham² presented in Table I.1 . The results of the design-point calculations were transferred to the COMP-OD program which was used to carry out part-load performance calculations. Effects of condenser and evaporator heat exchange area, motor supply frequency and two separate control scenarios were studied. In the first case a floating point action was simulated which allowed the compressor operating point to 'float' between a high and low characteristics representing surge and choke lines, Figure 7.6. In the second case the compressor was constrained to operate along the linear characteristic in Figure 7.6, making the refrigerant flow rate proportional to the compressor pressure ratio. The effectiveness of hot-gas-bypass and suction line throttle controls were also investigated. Table 7.5 lists typical output data from the COMP-OD program and Figures 7.12 to 7.24 show some results of these analyses.

7.13.1 Effect of Changes in Evaporator Capacity

Variations in evaporator heat transfer area were simulated by altering the design point temperature difference between the ambient air and the refrigerant. By making this temperature difference smaller, for a given heat transfer rate, had the effect of simulating a large capacity evaporator. For system 'a' in Table 7.4 the temperature difference between the ambient air and the refrigerant was approximately 20°C whilst that of 'c' is 10°C, thus making evaporator 'c' the larger. Condenser size is made the same by making the temperature difference between the condenser refrigerant and ambient air approximately the same in both cases.

An effect of increasing evaporator size is seen in Figures 7.14 which shows that the capacity of the system with the larger evaporator, 'c', is greater than that of the smaller unit, 'a'. Also the coefficient of performance, (C.O.P.), of the system with the larger evaporator is slightly greater than that with the smaller unit. This effect is shown in Figure 7.15.

For refrigeration systems using conventional reciprocating compressors, Stoeker and Jones²⁹ have shown, by a sensitivity analysis, that a ten percent increase in evaporator capacity

can produce a two percent improvement in C.O.P. In the present case system 'c', with the larger evaporator, was found only to have a C.O.P. value approximately 5% greater than that of system 'a' over the performance range examined, even though the evaporator in 'c' had a basic rating, (G), equal to 1.76 times that of system 'a', (estimated from the design point values given in Table 7.4). Based on this value an improvement of the order of 14% in C.O.P. could have been expected, Stoeker and Jones²⁹. The reason for this shortfall, (and for the rather flat C.O.P. characteristics shown in Figure 7.15), is the inability of the compressor to take advantage of the fall in condenser/evaporator pressure ratio as the size of the evaporator is increased. The action of a suction line throttle, (S.L.T.), forces a rise in compressor pressure ratio by reducing the inlet pressure. This was necessary to maintain the compressor operating point within the region between the high and low limits in Figure 7.6. If the suction line throttle had not been included then the compressor would have tended to operate in a region of very low isentropic efficiency due to the very low pressure ratio. Operating the compressor at a lower speed would also have produced low efficiencies. However in this case the flow rate would have been insufficient to meet the desired refrigeration rate. The effect of pressure ratio on C.O.P. can be explained by examining the definition of C.O.P. given in the following equation.

$$\text{C.O.P.} = \frac{\text{Refrigeration Effect}}{\text{Compressor Power}}$$

Which may be approximated to,

$$\text{C.O.P.} = \frac{h_{fg}}{C_p T_c (Pr^{(n-1)/(n-1)})} - 1 \quad 7.33$$

Where h_{fg} = latent enthalpy of evaporation
in the condenser
 C_p = specific heat
 T_c = compressor inlet temperature
 Pr = compressor pressure ratio
 n = polytropic index

Assuming constant condenser conditions, values for n and C_p , an increase in evaporator temperature would result in a decrease in compressor pressure ratio producing in a rise in C.O.P. In this case the action of the suction line throttle caused the compressor to operate at an almost constant

pressure ratio throughout almost regardless of the conditions in the evaporator because the pressure ratio between the condenser and evaporator was almost always less than the minimum value required by the simulated control system to keep the compressor operating point within selected region in Figure 7.6. The control action tended to cause the compressor to follow a characteristic line close to the choke line and in a region of low efficiency, as shown in Figure 7.18. By forcing the compressor to operate at a higher efficiency, by increasing the pressure ratio would have further reduced the C.O.P., from an examination of the previous equation. Simply not including a suction line throttle may not have improved the performance of the system. Although a fall in compressor pressure ratio would have resulted in a rise in C.O.P. this would have caused a decrease in isentropic efficiency. Values of C.O.P. resulting from the combined effects of both falling pressure ratio and efficiency can only be guessed because the required compressor operating points lay outside the range of available experimental data. Any extrapolation of data would probably produce erroneous conclusions.

Some practical considerations of increasing evaporator size, (capacity), are now considered. The practical implications of design options 'a' and 'c' can be assessed by considering the effectiveness of the two evaporators.

The ratio of heat exchanger effectiveness can be determined from Equation 7.35, derived from Equation 7.1, assuming the same refrigeration effect.

$$\frac{\epsilon_a}{\epsilon_c} = \frac{\Delta T_c}{\Delta T_a} \quad 7.34$$

In the present case $\Delta T_c = 10^\circ\text{C}$ and $\Delta T_a = 20.5^\circ\text{C}$

Therefore; $\epsilon_a = 0.488$

Also ϵ_a and $\epsilon_c < 1$ and the limiting value of $\epsilon_c = 1$.

By assuming a limiting value of 0.99 for ϵ_c then from Equation 7.2;

$$\begin{aligned} \left[\frac{U.A}{m.C_P} \right]_c &= -\ln(1 - \epsilon_c) \\ &= 4.605 \end{aligned} \quad 7.35$$

From Equations 7.34 and 7.2;

$$\begin{aligned} \frac{U.A}{m.C_P} &= -\ln\left(1 - \frac{\epsilon_c \cdot \Delta T_c}{\Delta T_m}\right) \\ &= -\ln(1 - 0.99 \times 0.488) \\ &= 0.660 \end{aligned} \quad 7.36$$

Because the value of $(m.C_P)$ are the same in both cases then;

$$\frac{(U.A)_c}{(U.A)_a} = \frac{4.605}{0.660} = 6.98$$

Assuming that the overall heat transfer coefficient, (U) , is the same in both cases then the area of the evaporator in case 'c' required to reduce the temperature difference to 10°C is 6.98 times greater than that of design 'a'. If the effectiveness of the evaporator used in case 'c' was reduced to a more realistic design value of 0.9, then the area required for case 'c' reduces to 3.7 times that required for 'a'. This is a significant increase in area.

To investigate the effect of an increase in evaporator area by 6.98 times would mean in practice, an evaporator was sized using data provided by a heat exchanger manufacturer; A.C.R 90. For case (a) in Table 7.4, assuming a relative humidity of 75%. It was estimated that to transfer 47kW of heat a 5 row coil with a heat exchanger face dimensions of 1.83m x 0.61m would be required. By keeping the same number of rows and increasing the face aspect ratio would require a heat exchanger for case 'c' of 1.61m x 4.84, (5.28ft x 15.88ft). This is impracticable because of the lack of physical space needed to fit such a heat exchanger on a railway carriage. The additional weight would require a stronger and heavier support structure, neglecting additional cost.

7.13.2 Effect of Condenser Capacity

To simulate the effects of changes in condenser capacity on the performance of the system and compressor it was necessary only to alter the design point temperature difference between the ambient air and the condenser refrigerant. The smaller this temperature difference the larger the condenser capacity for the same heat transfer rate. In the present case the variation in refrigeration effect resulting from a change in condenser size can be seen by comparing characteristics 'a' and 'b' in Figures 7.14.

The design conditions of systems 'a' and 'b' are listed in Table 7.4 which shows the design point temperature difference between the condenser and the ambient air approximately is 17°C for system 'a' and 30°C in 'b' making the condenser 'b' smaller than 'a'. Assuming that the heat exchanger effectiveness 0.99 for 'a', calculations have shown it to be 5.6 times larger in area than condenser 'b'. The area ratio reduces to 3.3 if the effectiveness of 'a' is reduced to a more practical value of 0.9. From design data provided by A.C.R.²⁰ it was estimated that the face area of the condenser in case 'a' would need to be 1.83m x 0.915m, (6ft x 3ft), for a 5 row coil with 14 fins per inch, flat surfaced ripple edge fins with a 1 inch x 0.866 inch staggered tube pattern with 3/8 inch diameter tubing. If case 'b' was to retain the same face aspect ratio and number of tube rows then the dimensions would be 1.0m x 0.5m. These dimensions could be increased by reducing the number of rows. However, the relationship between row numbers and face area is non-linear.

The variation in C.O.P. with ambient temperature in Figure 7.15 shows that a fall in C.O.P. can be expected to result from a reduction in condenser heat transfer area. From the data in Table 7.4, a 15% increase in C.O.P. results from a doubling in condenser capacity. From Stoecker and Jones²⁰ this increase is not unexpected. Of particular interest is the increase in refrigeration effect resulting from a decrease in condenser capacity. This was unexpected and is contrary to the usual trend found with systems using reciprocating compressor. The reason for this apparently unusual behaviour is the increase in refrigerant flow rate resulting from an increase in vapour density at the compressor inlet.

Results in Figure 7.12 show that the evaporator temperature in system 'a' was greater than that of case 'b' within the range of ambient temperatures investigated. At an ambient temperature of 16°C, for example, the temperature of the evaporator in system 'a' was approximately 4°C whilst that of 'b' was -1°C. From Figure 7.17, at the same ambient temperature the compressor inlet pressure in system 'a' was 0.257 bar whilst that in 'b' was 0.387 bar. The effect of this pressure and temperature difference resulted in a compressor inlet density in system 'b' of approximately 1.5 that in system 'a'. With little difference between the volume flow rate through the compressors in both cases the effect was a greater refrigerant flow through system 'b' which increases the refrigeration effect. The computer model assumed that no subcooling had taken place in the condenser of either system which tended to reduce the refrigeration effect of system 'b' compared with 'a'. Overall there was 28% increase in refrigeration effect resulting from a 50% reduction in condenser capacity.

In the simulation of system 'a' the suction line throttle routine was invoked in order to keep the operating point of the compressor within the limits in Figure 7.6. This forced a low inlet pressure onto the compressor and consequently a low mass flow rate compared with that in system 'b'. The capacity of the condenser in system 'b' was such that in order to transfer heat to the environment at an adequate rate the refrigerant temperature and pressure had to be greater than in case 'a'. This result showed that there was no need for the suction line throttle routine to be invoked in order to ensure that the compressor operated within the set operating limits.

Continued reductions in condenser size are unlikely to further increase refrigeration effect. If this was done the evaporator temperature would continue to fall and eventually the compressor inlet density would equal that of case 'a' and with it the mass flow rates would also become more equal thus system 'b' would lose its advantage. In addition the compressor pressure ratio would also increase along with condenser temperature.

The effect of using a smaller condenser has implications on the design of the refrigerator. If a high C.O.P. is required then a large capacity condenser should be selected. If a large refrigeration effect is desired then the application

of a smaller condenser should be considered. This result is applicable to centrifugal refrigeration compressors which use only SLT capacity control.

7.13.3 Effect of Motor Supply Frequency

Altering the supply frequency changes the speed and capacity of a compressor. To study this effect on refrigerator performance the simulation program was run with a range of supply frequencies. The results showed refrigeration effect increases with compressor speed. This was not unexpected because compressor flow rate also increases with speed, Figure 7.6.

The increase in refrigeration effect with compressor speed is indicated in Figure 7.14 by comparing the characteristics of system 'd', which had a simulated motor supply frequency of 1100 hz, with those of systems 'a', 'b' and 'c', which had supply frequencies of 1200 hz. An exception is where the refrigeration effect of system 'a' falls below that of system 'd' at the lower ambient temperatures. This is due to the action of the suction line throttle in system 'a' which reduces the compressor inlet pressure, (and density), to keep the operating point of the compressor within the limits shown in Figure 7.6. The effect of the suction line throttle was to reduce the mass flow rate and thereby refrigeration effect.

The pressure ratio potential of centrifugal compressors falls approximately with the square of rotational speed. By reducing the speed of the compressor it is possible to support lower condenser pressures, (and temperatures), without resorting to the introduction of a suction-line throttle. This effect is seen in Figure 7.19 by comparing characteristic 'a' and 'd' which show the predicted values of pressure ratio required across the suction-line throttle to maintain the compressor operating point within the specified pressure ratio limits.

An additional benefit from lowering the compressor speed is the predicted gain in system C.O.P.. This effect is shown in Figure 7.15 which also shows that the C.O.P. improvement is lost as the ambient temperature increases. These effects can be explained using the similarity laws for rotor dynamic machines from which;

$$\text{C.O.P} \propto 1/N^2$$

The results of C.O.P calculations shown in Figure 7.15 appear to agree with this result at low ambient temperature. From Figure 7.15, at 10°C the C.O.P. of system 'd' was 3.2 whilst that of system 'a' was 2.53, therefore;

$$\frac{\text{C.O.P.}_a}{\text{C.O.P.}_d} = \frac{2.53}{3.20} = 0.79$$

Whilst the ratio of rotational speeds gives;

$$\frac{N_d^{0.2}}{N_a^{0.2}} = \frac{1100}{1200} = 0.84$$

The 6% difference in the two results above may be explained by the differences in compressor efficiency ;shown in Figure 7.18 and the predicted differences in condenser pressures which effects the dryness fraction of the refrigerant as it enters the evaporator.

The performance of system 'd', measured in terms of C.O.P., was predicted to fall with increasing ambient temperature until at approximately 30°C the C.O.P. values of both system 'a' and 'd' were equal. This may be explained by the gradual increase in pressure ratio experienced by system 'd' whereas the pressure ratio of the compressor in system 'a' remains approximately constant throughout due to the action of the suction-line throttle. The effect of pressure ratio on C.O.P. is described by Equation 7.33.

Of interest was the effect of motor supply frequency on the isentropic efficiency of the compressor. An effect of reducing the rotational speed of the compressor has been to increase its isentropic efficiency. This improvement has a direct effect on the values of C.O.P.. However, as described so far the overall effect has been a reduction in C.O.P. with increasing ambient temperature. In the next section it is shown that by constraining the compressor to operate on a fixed pressure-flow characteristic, with relatively high isentropic efficiencies, this problem is overcome and high C.O.P. values are possible along with high compressor efficiencies.

Motor supply frequency is a basic design feature of a refrigerator system and once selected for a particular application is fixed, assuming that variable speed capacity modulation is discounted on cost grounds. From the discussion so far presented it is concluded that the effect

of reducing the present compressor speed by a few percent from the design value of 1200 Hz results in a short fall in refrigeration effect at the design point ambient temperature of 28°C, however if a small reduction in refrigeration effect is allowed then there are some benefits to be gained from reducing the compressor speed; particularly in terms of compressor efficiency and C.O.P. values at ambient temperature less than 30°C.

7.13.4 Controller Function

The part-load performance of a compressor within a refrigerator system depends upon the operational constraints placed upon it by amongst other things the type of control system. Using the present model it was not possible to simulate transient behaviour of the compressor and refrigeration system to rapid variations in cooling load. In practice, however, it is probable that the time constant of the refrigeration load, (in this case an air conditioned railway carriage), would be greater than that of the refrigeration system. This suggested that it was possible to use the present model to investigate the behaviour of a refrigeration system operating within boundaries defined by certain control scenarios. Two types of control scenario were considered; 'floating-point' and a simple proportional type action.

The floating point control system envisaged, and simulated by COMP-OD, was designed so that as the compressor operating point moved across the surge line, in Figure 7.6, a hot-gas-bypass valve would open off-loading the condenser and thereby reducing the compressor pressure ratio and bringing the operating point back within the 'high limit' shown in Figure 7.6. At this point the hot-gas-bypass setting would be held constant until the operating point moved back across the surge line, when it would be opened still further, or else until the operating point moved across the 'lower limit', at which point the valve would begin to close. This could cause hunting problems particularly at low compressor speeds when the operating band, (the difference between the surge line and choke line pressure ratios), is small. If the compressor persisted in operating below the lower limit with the hot-gas-bypass valve shut then the suction-line throttle valve would operate causing the compressor pressure ratio to increase which would return the

operating point across the lower limit and back into the selected operating region in Figure 7.6. Hunting would again be possible if pressure ratio difference between the choke and surge lines was small.

The computer simulation showed that only the suction-line throttle was required to maintain the compressor operating point within the specified region. The reason for this was that the condenser temperatures selected for the design condition never allowed the pressure ratio to exceed the surge value at the design point rotational speed.

From Figure 7.18 the design point efficiency of the compressor never exceeded 47%. In fact the compressor has a potential efficiency of 60%. The control system simulation was therefore not making best use of the compressor. A modified system was envisaged in which the setting of the suction-line throttle, (the controlling element), would be adjusted until the compressor was operating at the maximum achievable efficiency. This action would reduce mass flow rate if the throttle was closed. This would result in a fall in refrigeration effect. The system C.O.P. would tend to increase due to the improvement in compressor efficiency.

The potential benefits of this alternative system were investigated using the simulation program COMP-OD. In order to simulate the effects of operating the compressor at its highest efficiency, a 'single-line' characteristic was selected in Figure 7.6 which ran parallel to the surge line but with 10% greater flow rate; surge margin. It is well known that the region of highest isentropic efficiency for the present compressor type lies within a few percent of its surge line. This is clearly seen in Figure 7.5 in which the proposed operating line in Figure 7.6 has been transposed in terms of compressor efficiency. A corrected speed value of 66,000 rpm at the design-point was tested and the evaporator and condenser capacities were set the same as in the previous system 'a' in order that comparisons could be made.

The results of this simulation are plotted in Figures 7.20 to 7.22. In this case the independent co-ordinates chosen were; ambient air temperature entering the condenser and the temperature of the air entering the evaporator. These were thought to provide a more realistic comparison with existing systems.

Figure 7.20 shows the variation in compressor efficiency.

Comparing the results in Figure 7.20 with those presented in Figure 7.18 it is shown that a considerable improvement in efficiency is predicted by choosing the 'single-line' characteristic instead of the 'floating-point' approach previously discussed. A maximum value of 59% was predicted whereas in system 'a', (the representative design case in Figure 7.18), a maximum value of only 47% was achieved. A similar result was found for C.O.P. in Figure 7.21 which shows a maximum value of 2.76 with an ambient temperature of 32°C and an evaporator air entry temperature of 24°C. Comparing the results shown in Figure 7.21 with the C.O.P. results shown in Figure 7.15 it is seen that there is a gradual increase in C.O.P. with temperature in the former case, which would normally be expected, whereas there is a gradual fall-off in C.O.P. values with increasing ambient temperature in the latter case for reasons previously given.

Figure 7.22 shows a particularly interesting result in which the refrigeration effect is apparently independent of the temperature of the air entering the evaporator. The result is due to the action of the suction-line throttle, closure of which causes the operating point on the characteristic in Figure 7.22 to move downwards as the ambient temperature falls. The greatest refrigeration effect for a particular evaporator air entry temperature occurs at the point where the throttle is fully open.

The results of the fixed running line calculations in Figure 7.22 indicate that the present compressor would meet the refrigeration capacity requirements for both the RP 40 and UP 31 systems of 40 kW and 24.4 kW respectively with air temperatures as indicated in Table I.1. The C.O.P. values at the respective design points for the RP 40 and UP 31 systems are 2.92 and 2.87. These values compare with predicted results of 2.9 and 2.8 at the same ambient and evaporator air entry temperatures shown in Figure 7.21. This result is encouraging. It is thought that a ten percentage point improvement in maximum isentropic efficiency would increase the C.O.P. values up to 3.4 and 3.3 respectively thus making the development of a production variant of the compressor a viable proposition.

7.14 Conclusions

The following conclusions are drawn from the results of the

theoretical refrigeration plant model discussed in the last section.

1. The computer models described in this chapter operated satisfactorily over the desired range of ambient conditions. Some problems were experienced with the COMP-OD program when a motor frequency of less than 1000hz was specified. This was due to the selection of step values for the suction-line-throttle pressure ratio and the hot-gas-bypass mass flow used in the subroutines. This problem may be overcome by reducing these step values.

2. The computer based simulation model,COMP-OD,predicted that the present compressor would operate satisfactorily across the ambient temperature range considered when fitted into a refrigeration system with the same design point characteristics as the RD40 roof mounted unit,(described and discussed in the Introduction to this thesis) and also the smaller UP 31 system,meeting both capacity and performance requirements.

3. The model predicted that the refrigeration effect of a system would increase with evaporator and condenser capacities. This result confirms generally held assumptions.

4. A slight improvement in compressor operating efficiency was shown possible by increasing the area of the evaporator. This is shown in Figure 7.18 in which the efficiency of the compressor in system, 'a', at 30°C is 47% whilst that of system 'c', with the larger area evaporator, has an efficiency of 50%. These values of compressor efficiency are low because the refrigerator system pressure ratio was predicted to be low. The improvement in compressor efficiency falls off with ambient temperature.

Some improvement in compressor efficiency was shown to be possible by reducing the condenser heat exchange area. This result is shown in Figure 7.18 by comparing the characteristics of system 'a' and 'b' which has the smaller condenser area.

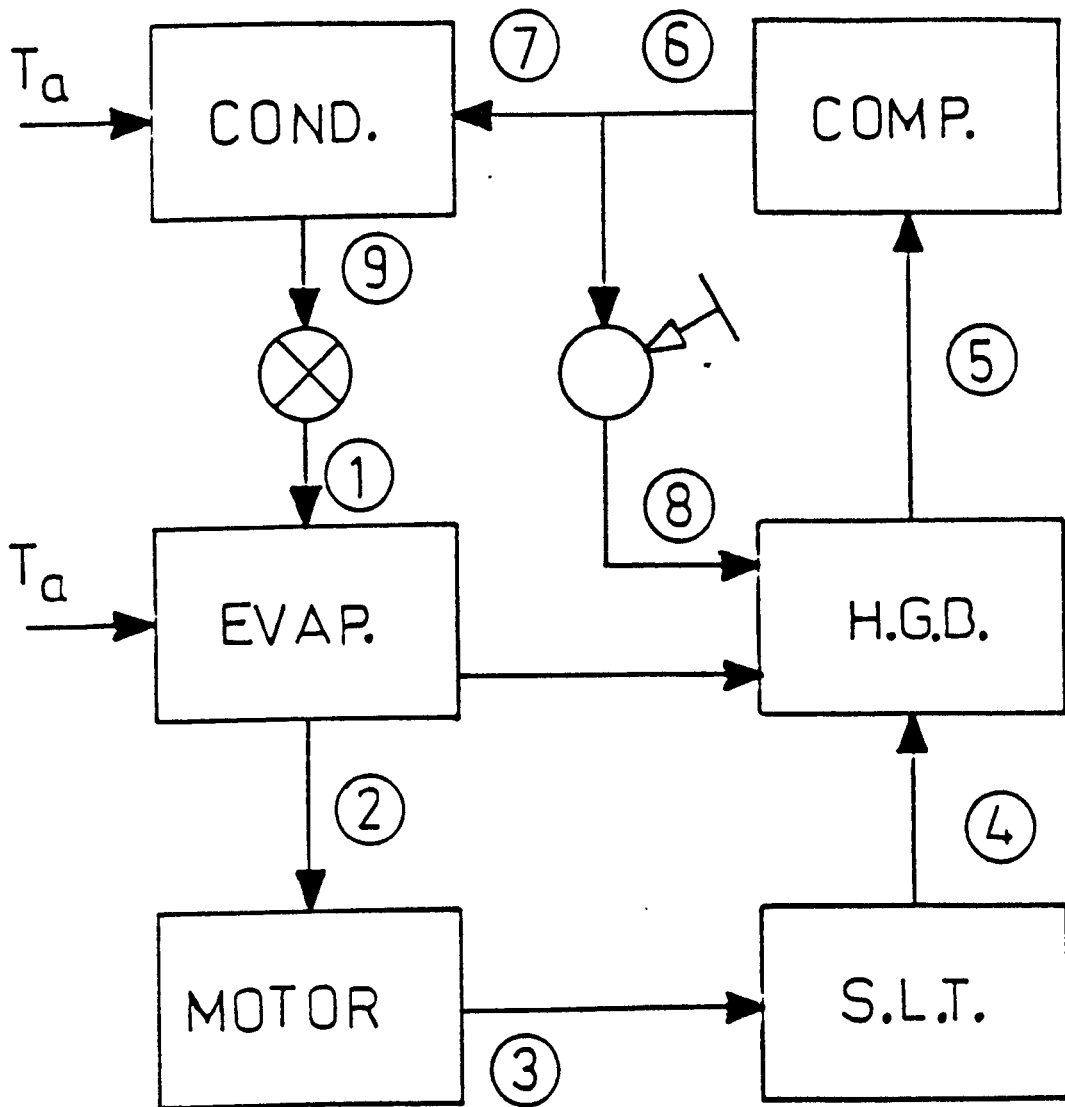
A single-line characteristic, in Figure 7.6, offers the potential for higher compressor efficiencies, as shown in Figure 7.20.

5. The disadvantage of larger capacity evaporators and condensers are cost and weight. The latter is particularly important because it affects the design of the supporting structure particularly for the railway carriage applications.

6. The simulation model showed that increasing the evaporator size does not significantly effect the system COP. Reducing the temperature difference between the ambient air and the evaporator at the design point from 20°C to 10°C increases the size of the evaporator required by approximately 3.7 times, (assuming an heat exchanger effectiveness of 0.9), to exchange the same quantity of heat transferred. A similar result was found when comparing large and small condensers.

7. It was concluded that the 'floating-point' control using the high and low limits shown in Figure 7.6 is unsatisfactory for the present system. The 'single-line' approach described in Section 7.12.4 provided a much more precise control of compressor performance. In both cases a suction-line throttle was the only active element of the control system. Future development should include an investigation into the application of microprocessor based proportional type of controller to modulate the suction-line-throttle via a stepper motor actuator. This system should also include close control over the system expansion valve.

8. Values of C.O.P. compare favourably with commercial units and with what must be achieved for competitive performance. Backsweeping of the compressor impeller would raise compressor isentropic efficiency and move maximum efficiency further from the stall condition, (see Chapter 2).



H.G.B. - HOT GAS BYPASS

S.L.T. - SUCTION LINE THROTTLE

FIG. 7.1 SCHEMATIC OF AIRCONDITIONER MODEL SHOWING NOTATION

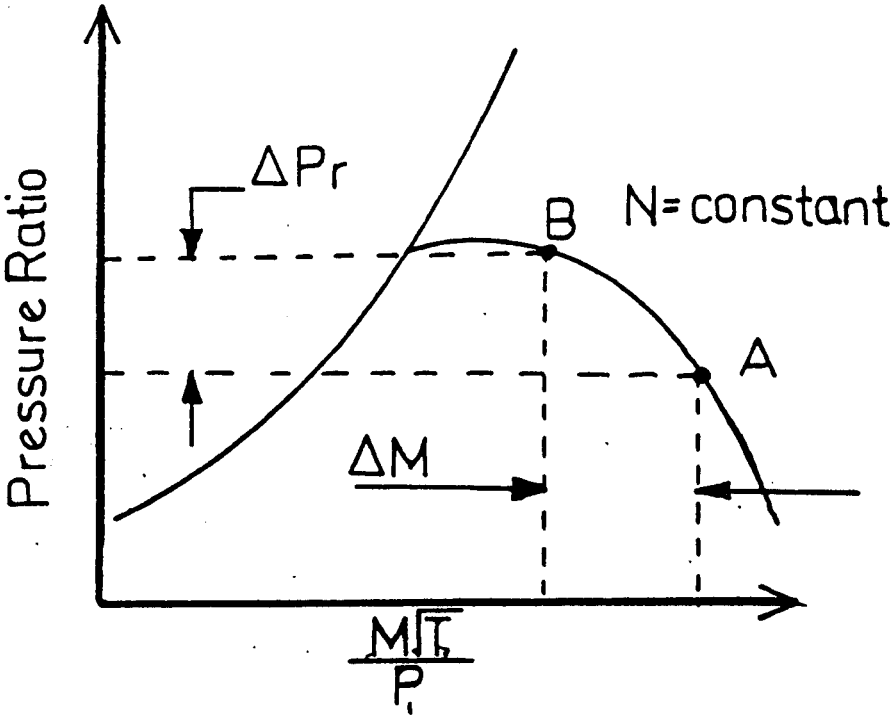


FIG. 7.2 VARIATION IN OPERATING POINT CAUSED BY
PRESSURE LOSSES IN SUCTION LINE

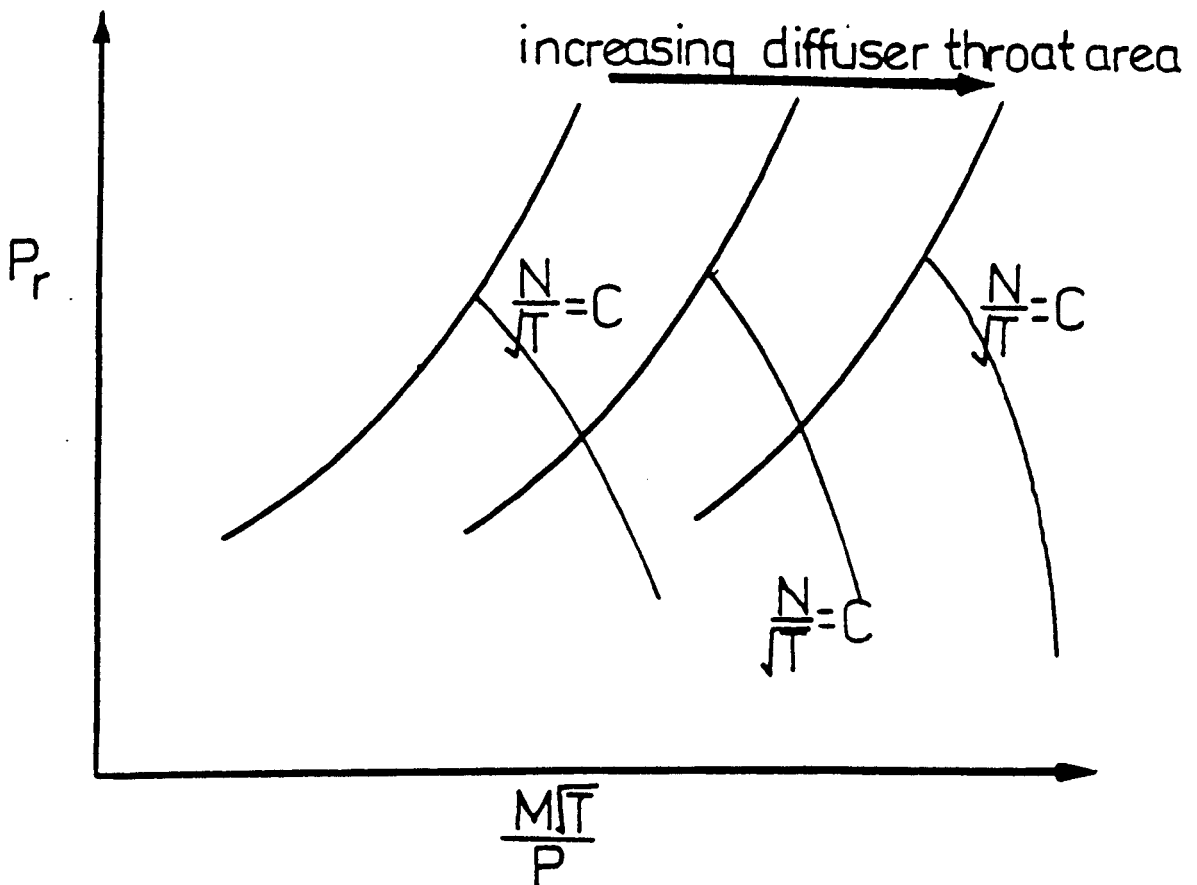


**FIG. 7.3. VARIATIONS IN A COMPRESSOR CHARACTERISTIC WITH CHANGES
IN DIFFUSER THROAT AREA**

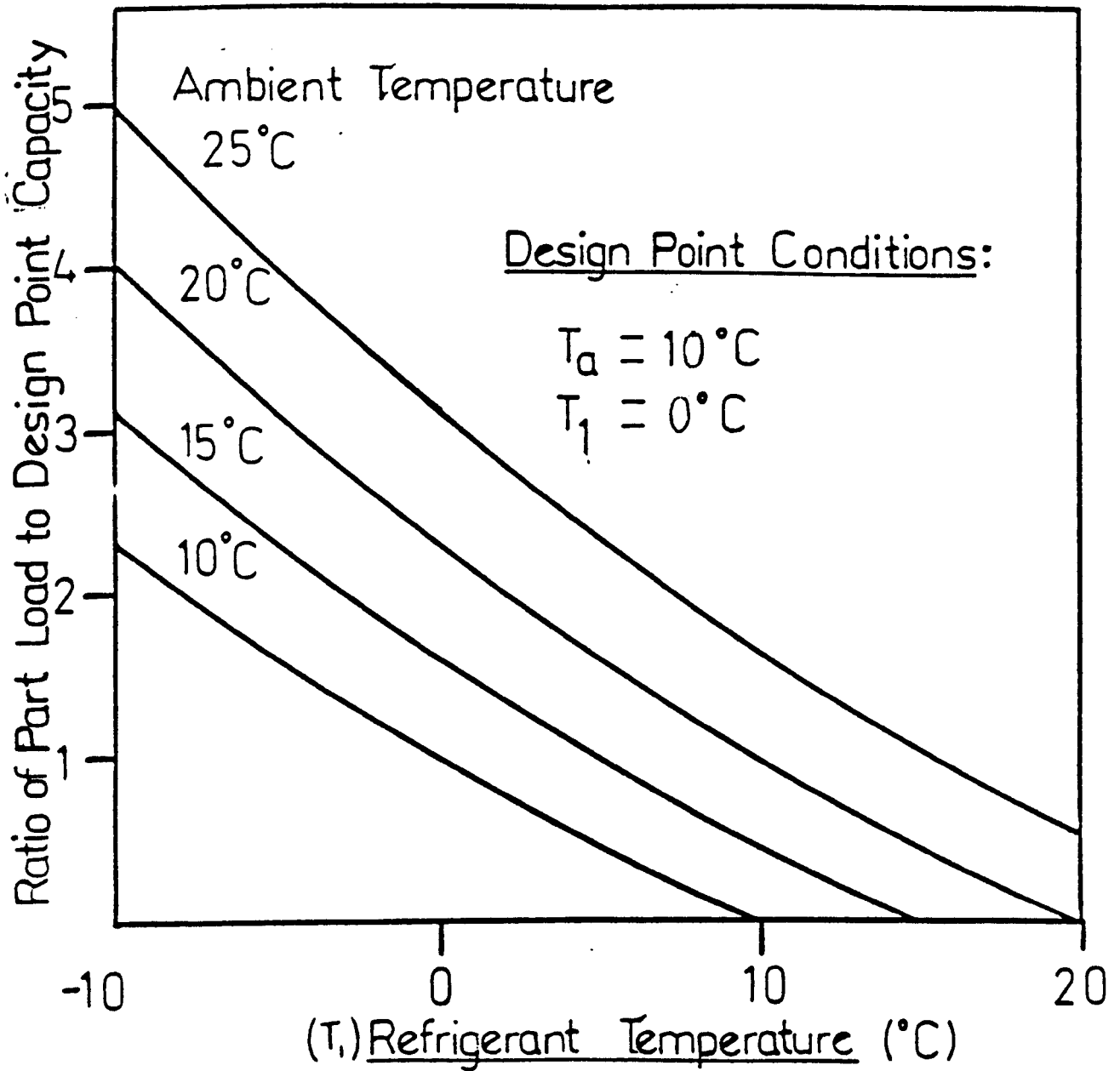


**FIG. 7.4 MATHEMATICAL PREDICTION OF THE PART-LOAD
LOAD PERFORMANCE OF A FINNED-TUBE REFRIGERANT EVAPORATOR**

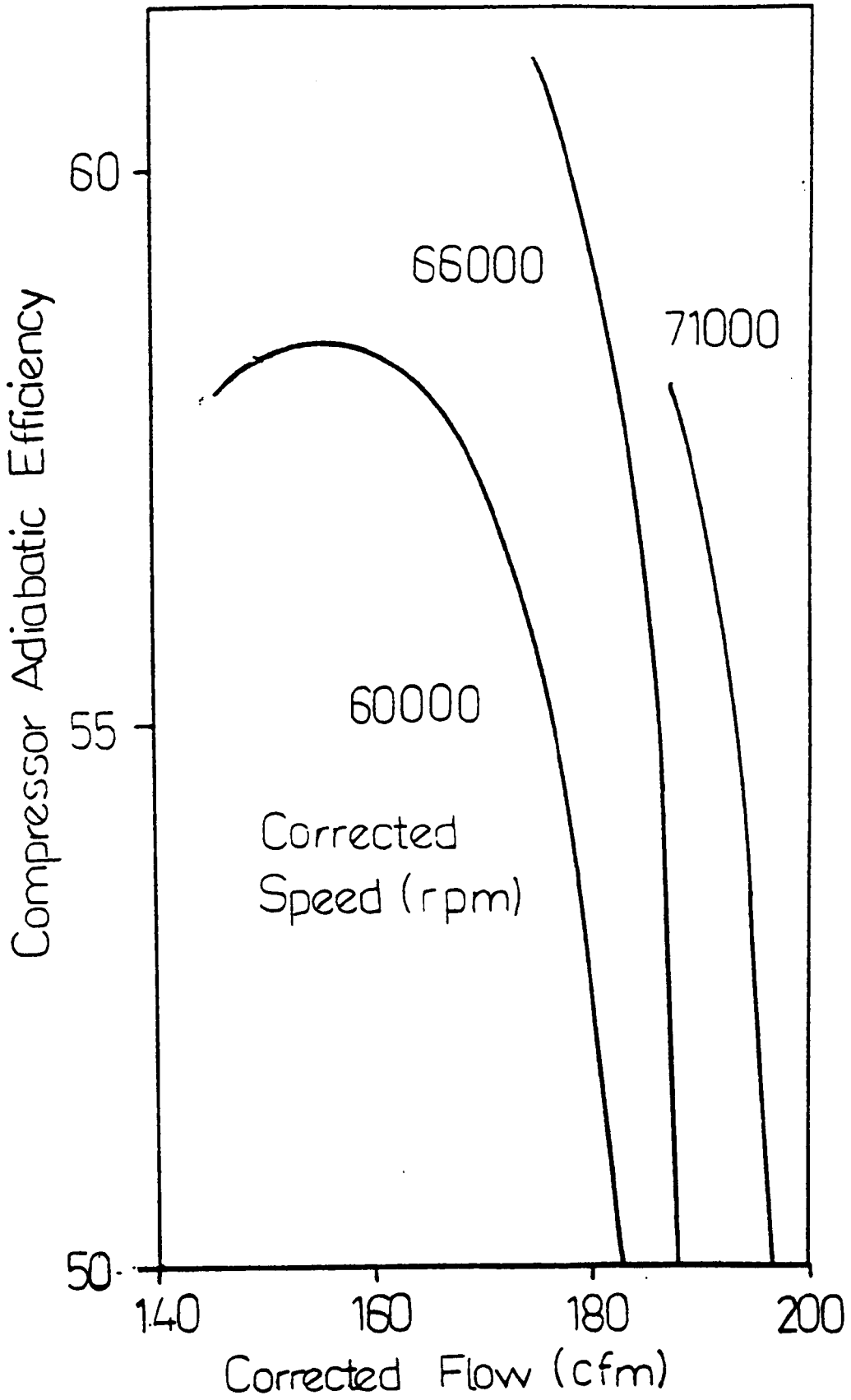


Fig. 7.5 (FREON 11) COMPRESSOR EFFICIENCY CHARACTERISTIC

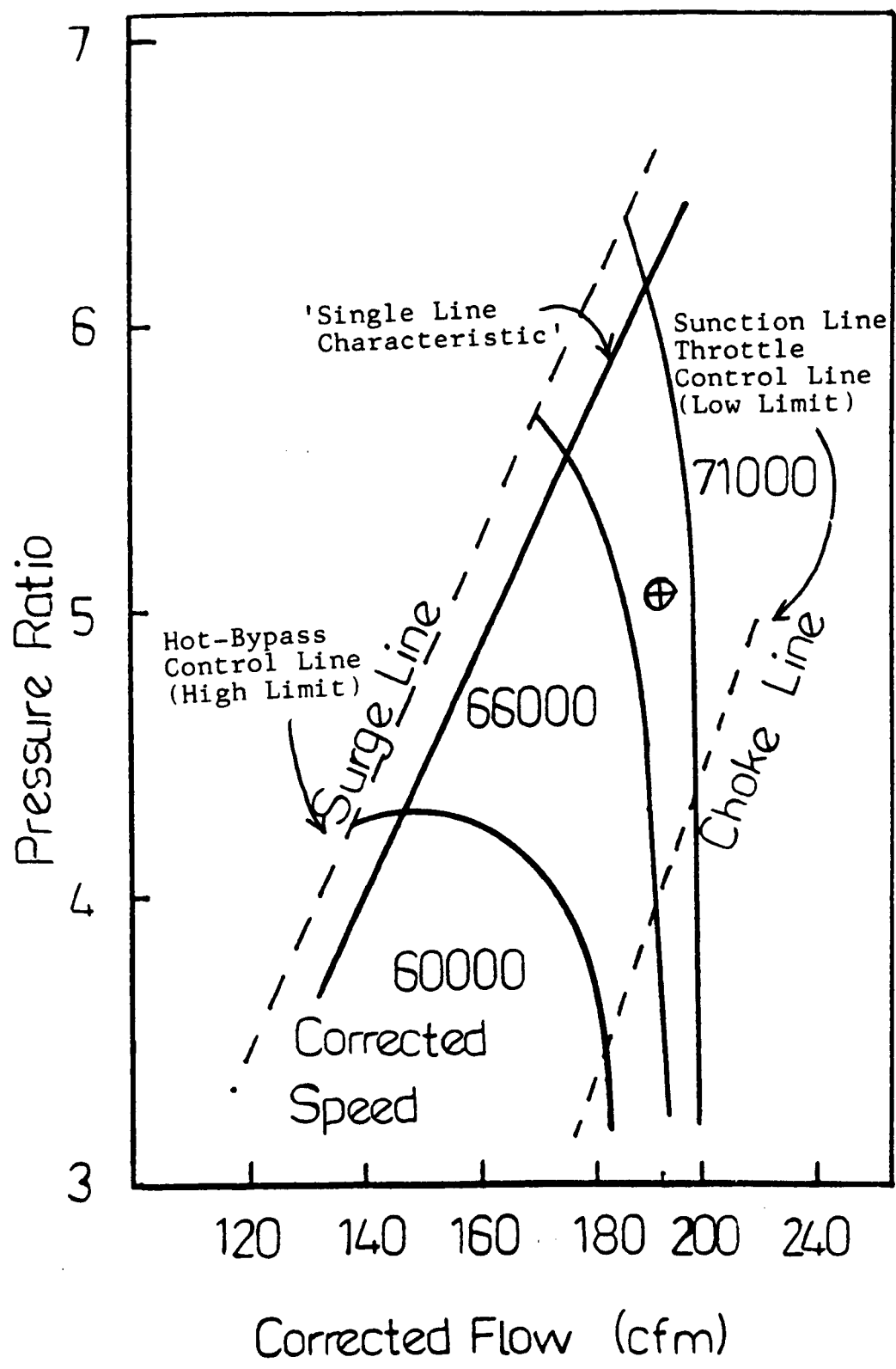


Fig. 7.6 (FREON 11) COMPRESSOR PRESSURE RATIO CHARACTERISTIC

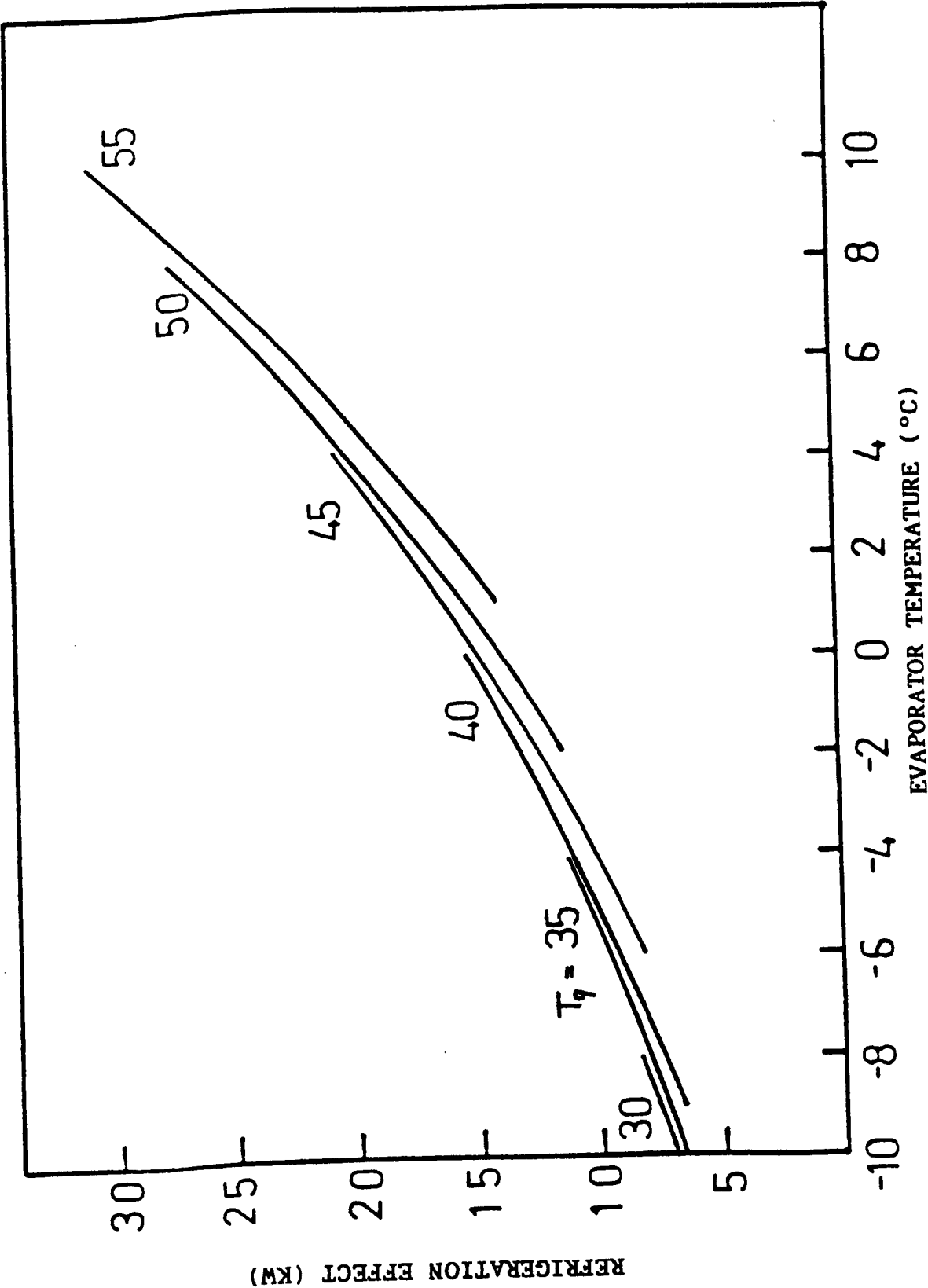


Fig. 7.7 PERFORMANCE OF CENTRIFUGAL COMPRESSOR PREDICTED BY COMP, ASSUMING 0°C SUPERHEATING

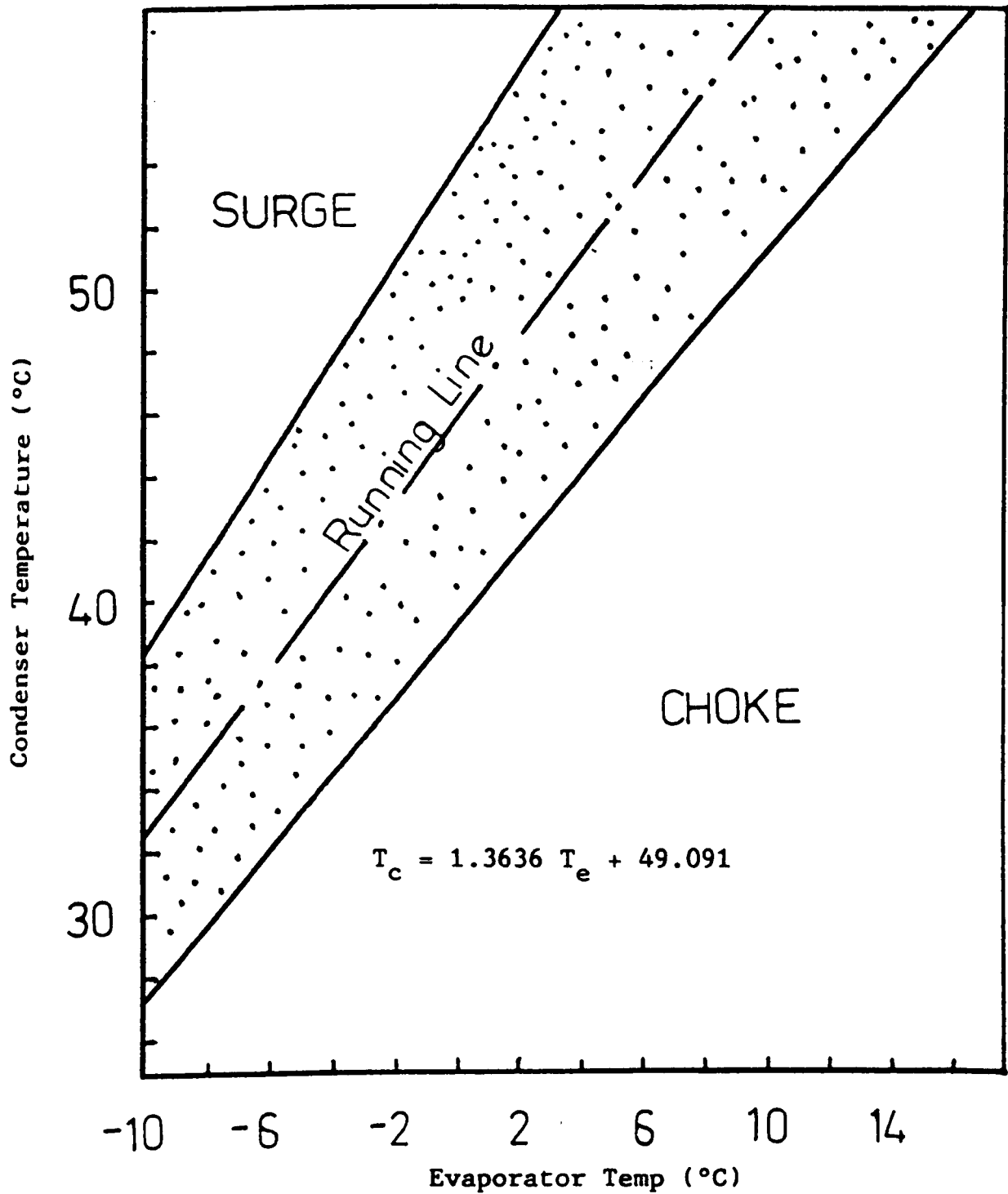


Fig. 7.8 SOLUTION REGIME OF COMPERF WITH 0°C SUPERHEATING AT COMPRESSOR INLET

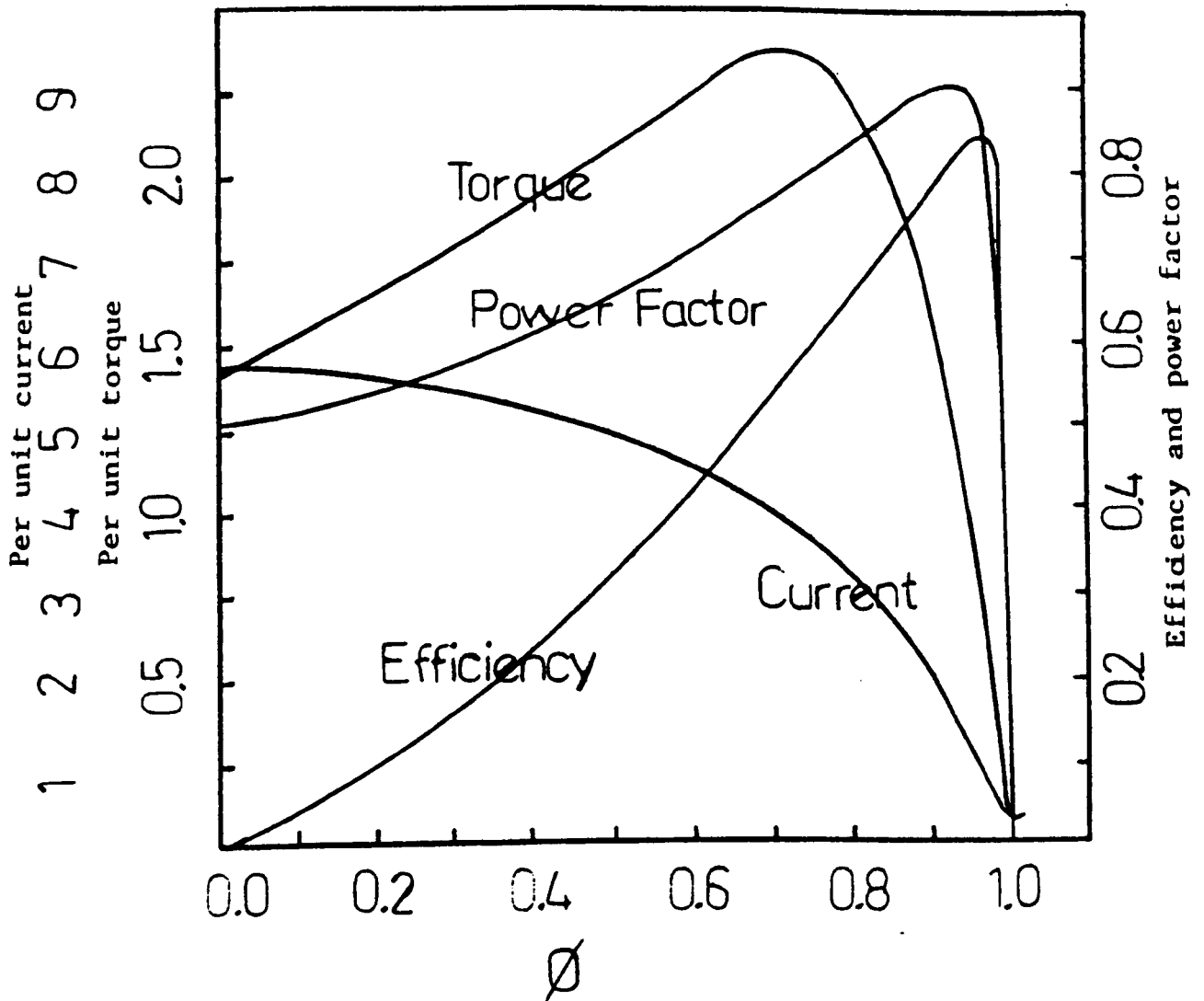


Fig. 7.9 PERFORMANCE CURVES OF TYPICAL POLYPHASE INDUCTION MOTOR (REF. 69)

Part-Load Performance Calculations: COMP-OD Flow Chart

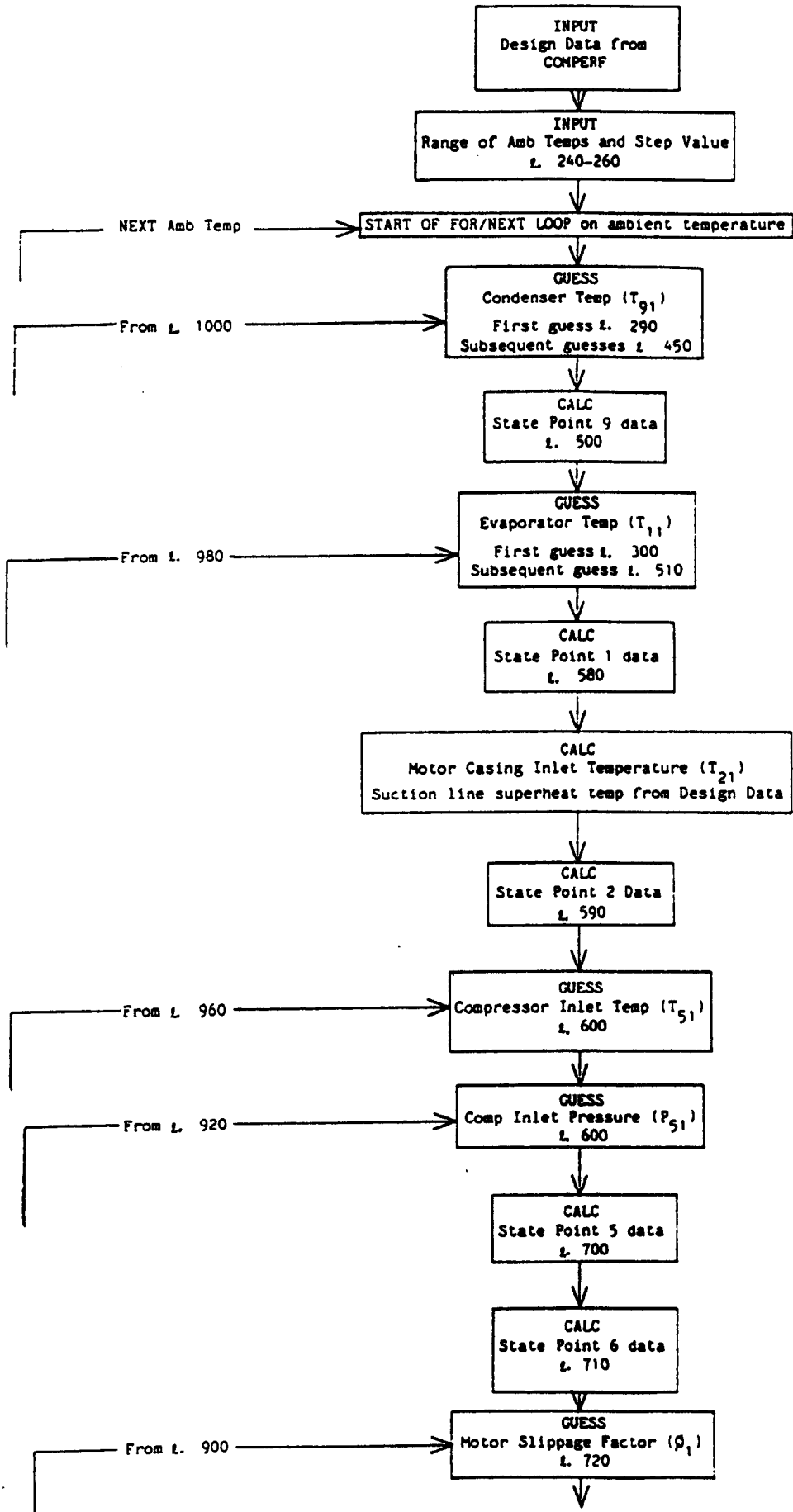


FIG. 7.10a

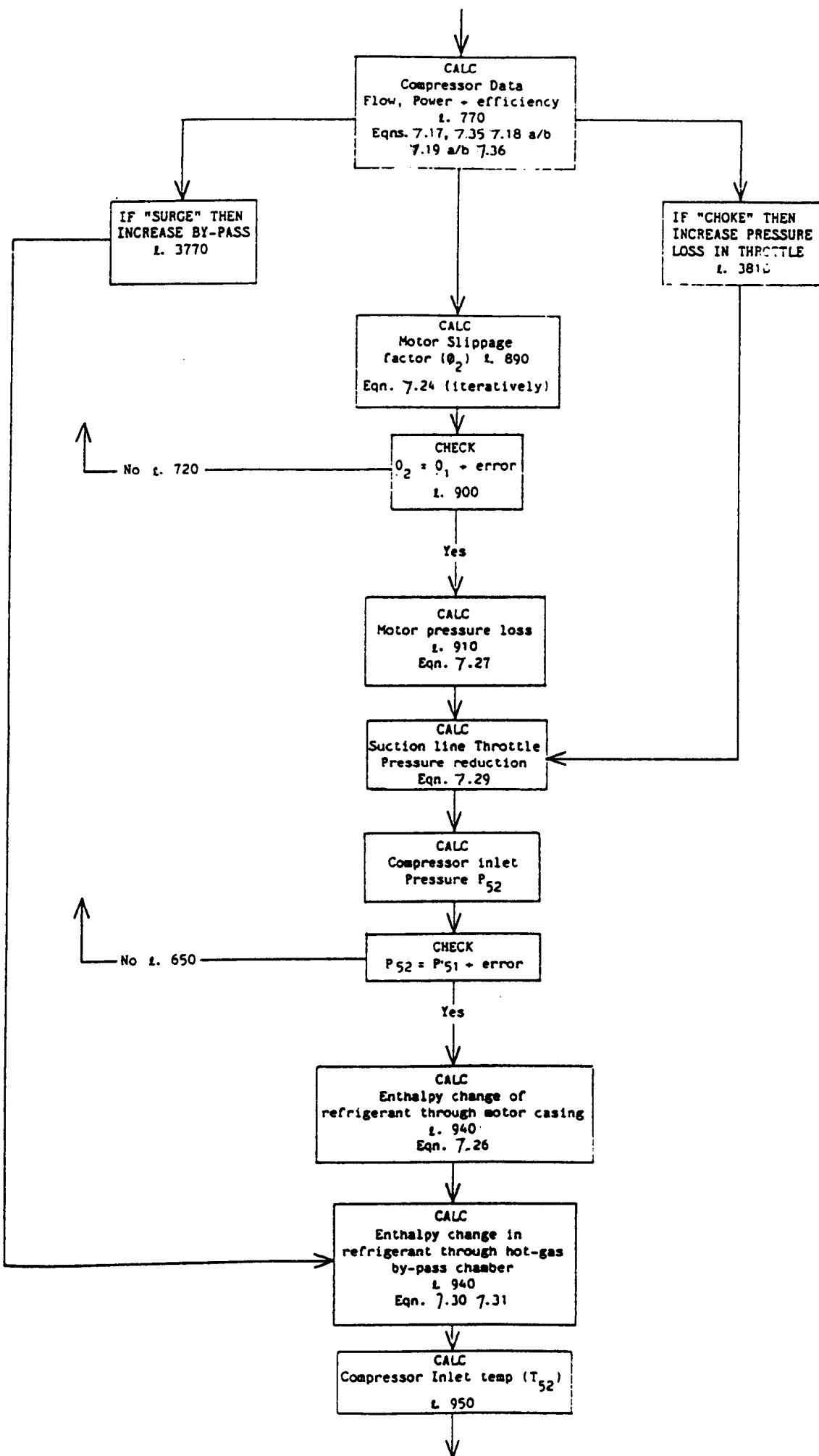


FIG. 7.10b

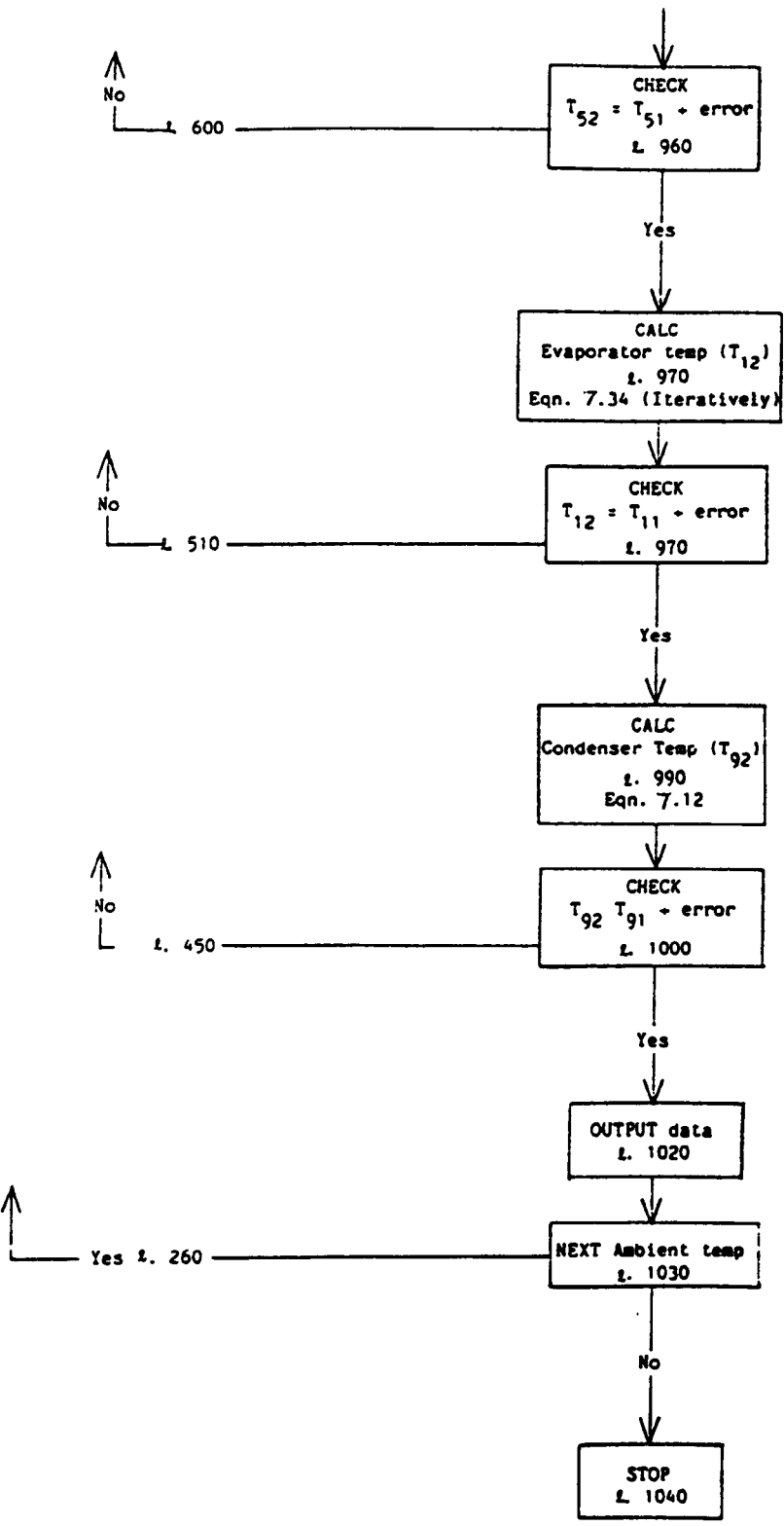


FIG. 7.10c

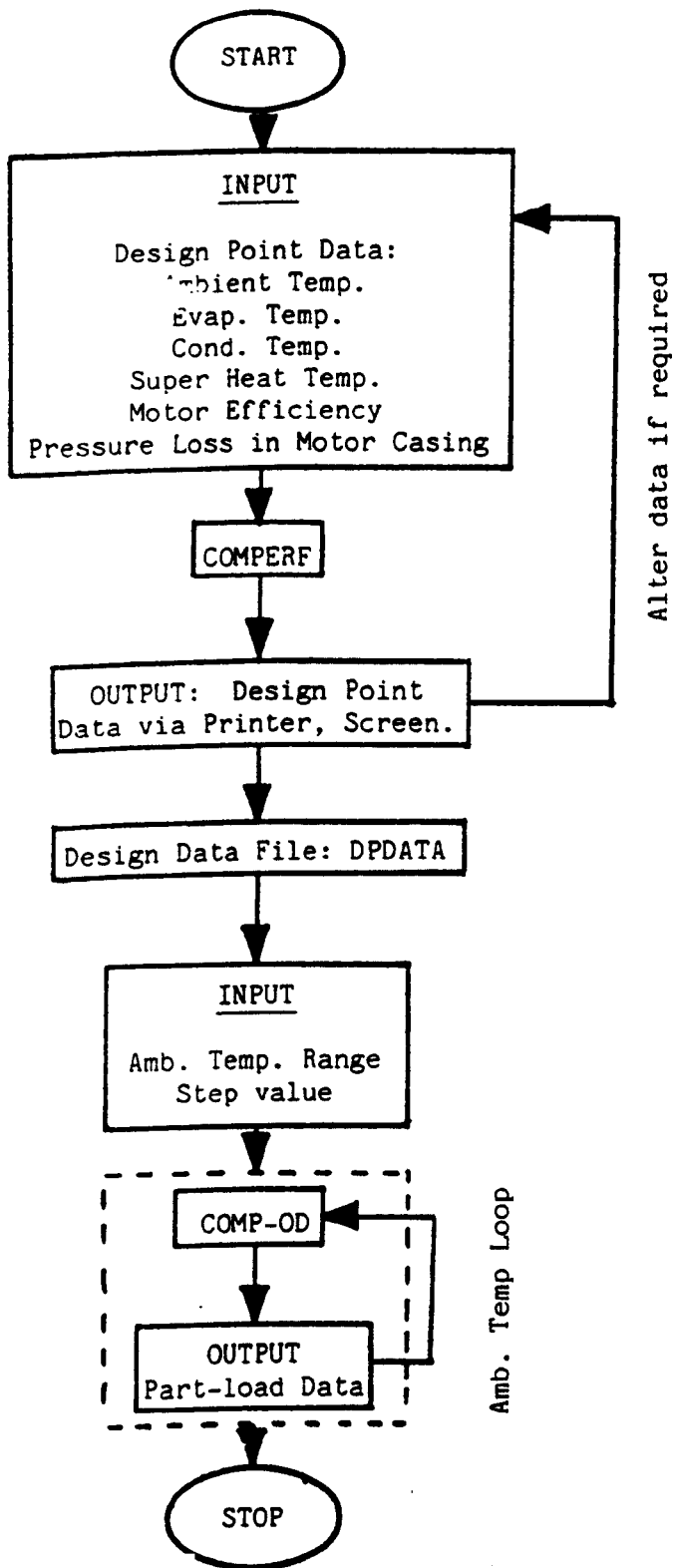
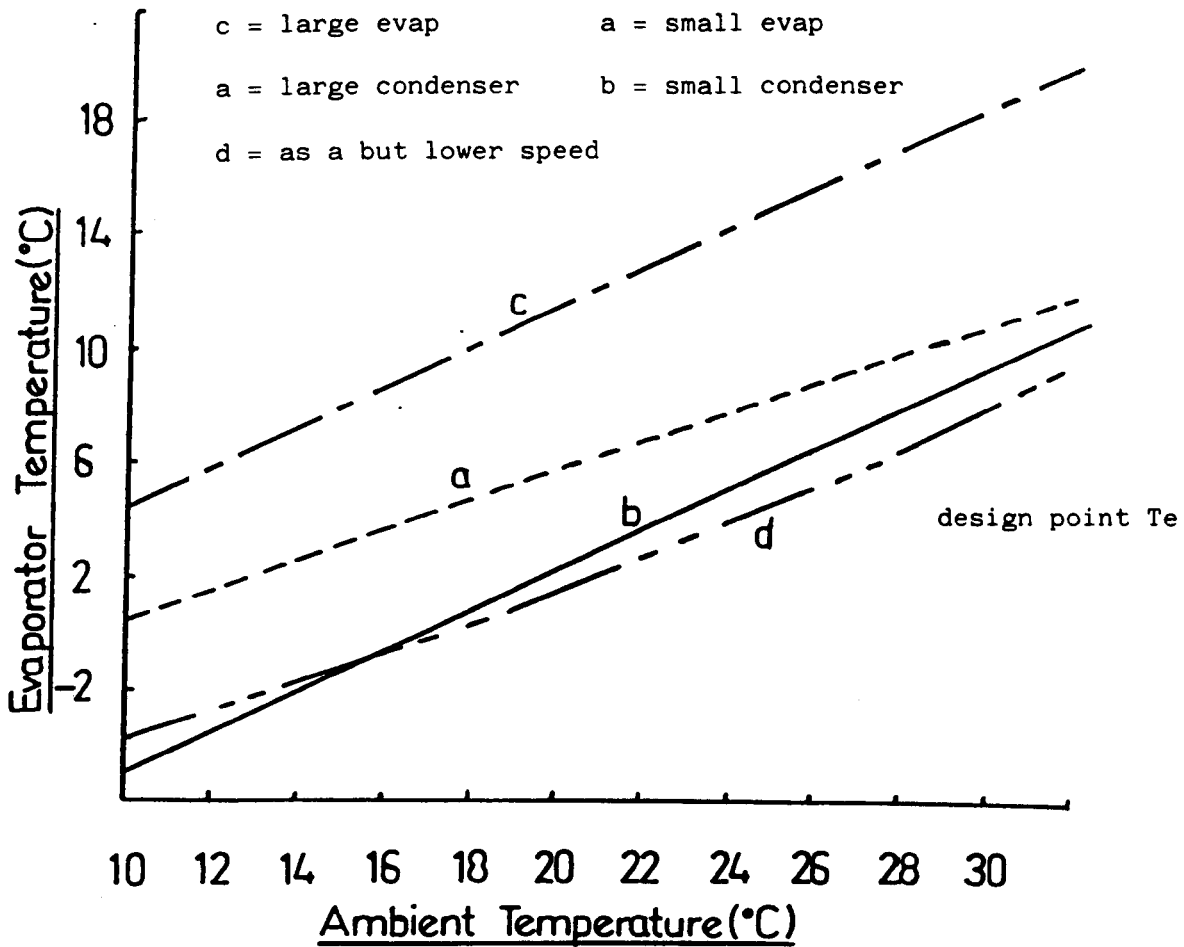


FIG. 7.11 FLOW DIAGRAM OF MODEL



- | | | |
|------------------------------------|---|--------------|
| a = small evap and large condenser |) | |
| b = small evap and condenser |) | 1200 hz comp |
| c = large evap and condenser |) | |
| d = small evap, large condenser |) | 1100 hz comp |

FIG. 7.12 **VARIATION IN EVAPORATOR TEMPERATURE WITH**
AMBIENT TEMPERATURE PREDICTED USING COMP-OP

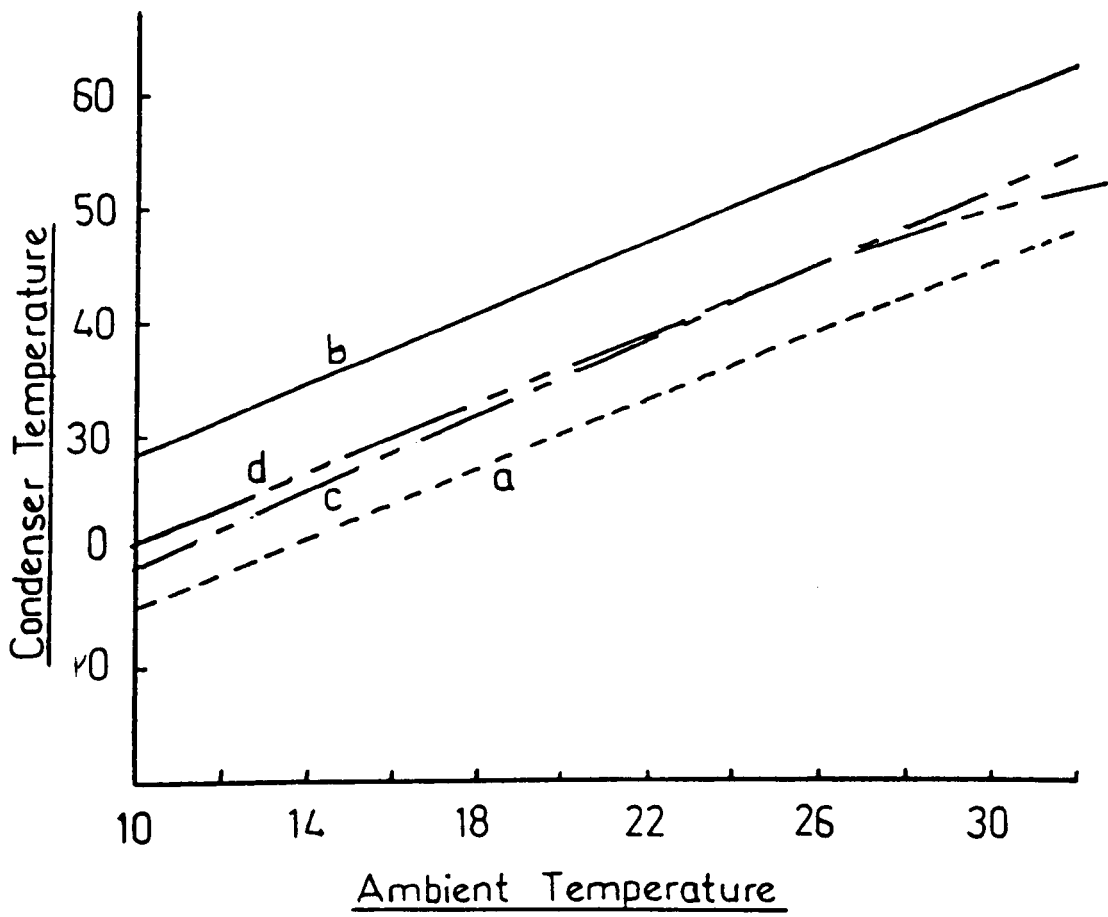


FIG. 7.13. VARIATION IN CONDENSER TEMPERATURE WITH
AMBIENT TEMPERATURE PREDICTED USING COMP-OD

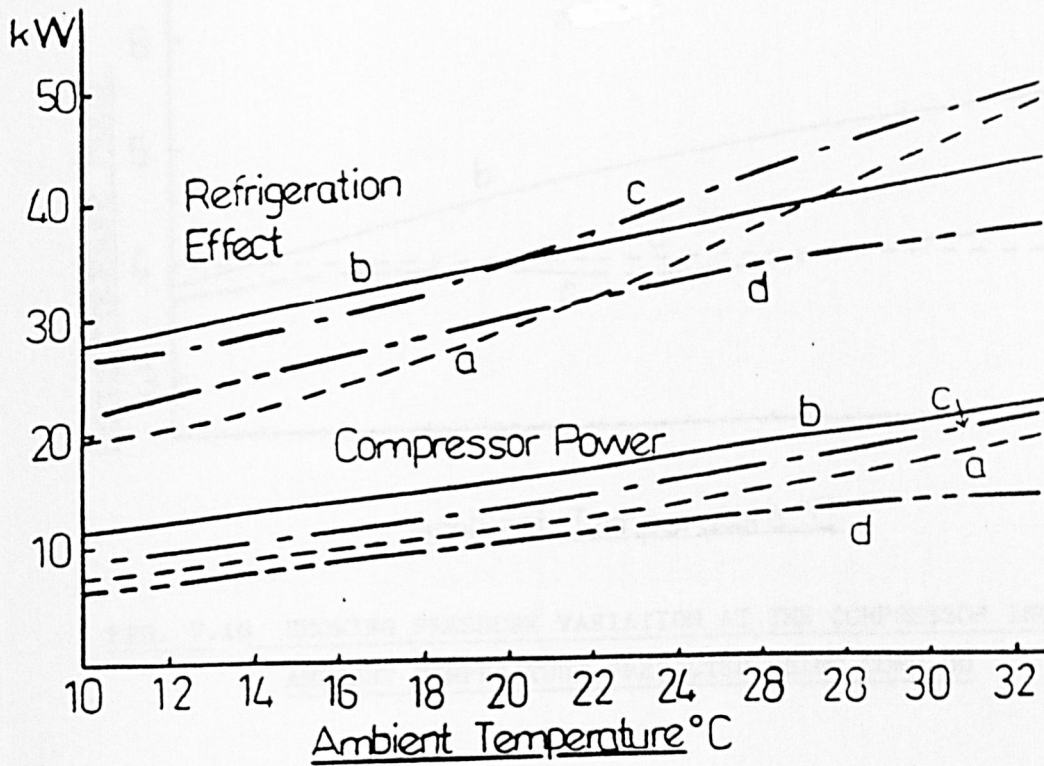


FIG. 7.14 SHOWING THE VARIATION IN COP WITH AMBIENT TEMPERATURE
PREDICTED USING COMP-OD

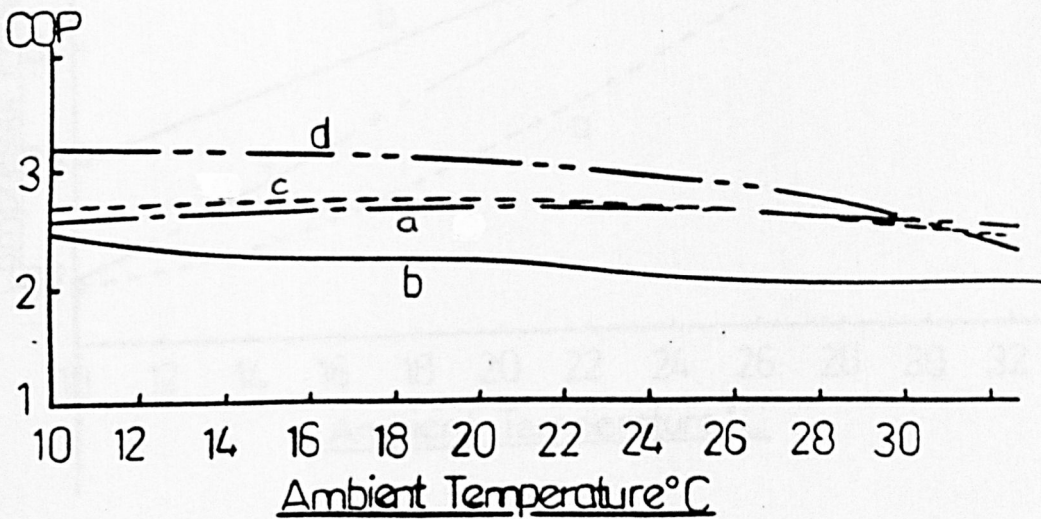


FIG. 7.15 THE VARIATION IN REFRIGERATION EFFECT AND
COMPRESSOR POWER WITH AMBIENT TEMPERATURE PREDICTED USING
COMP-OD

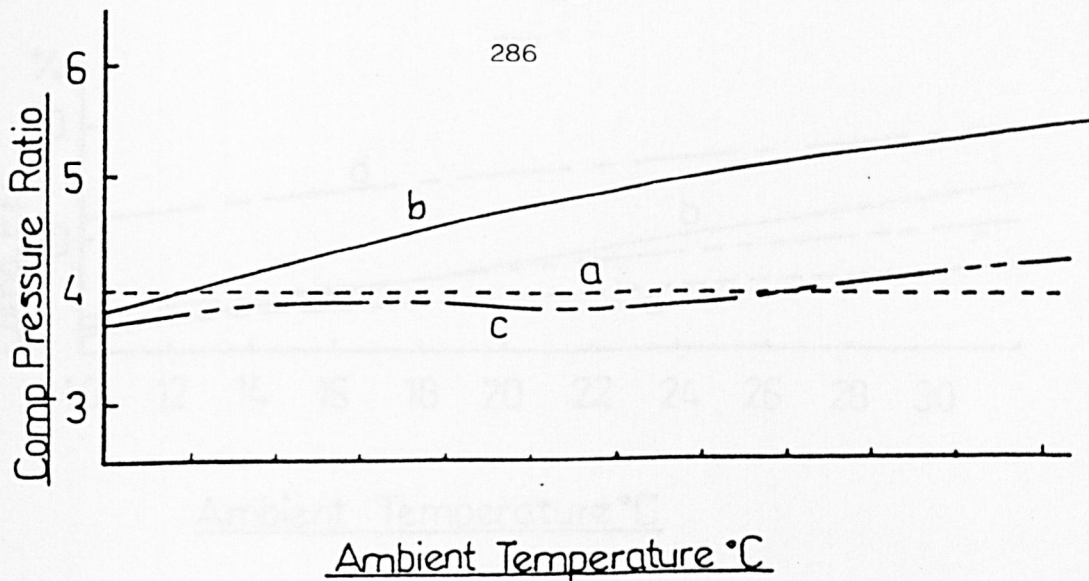


FIG. 7.16 SHOWING PRESSURE VARIATION AT THE COMPRESSOR INLET WITH
AMBIENT TEMPERATURE PREDICTED USING COMP-OD

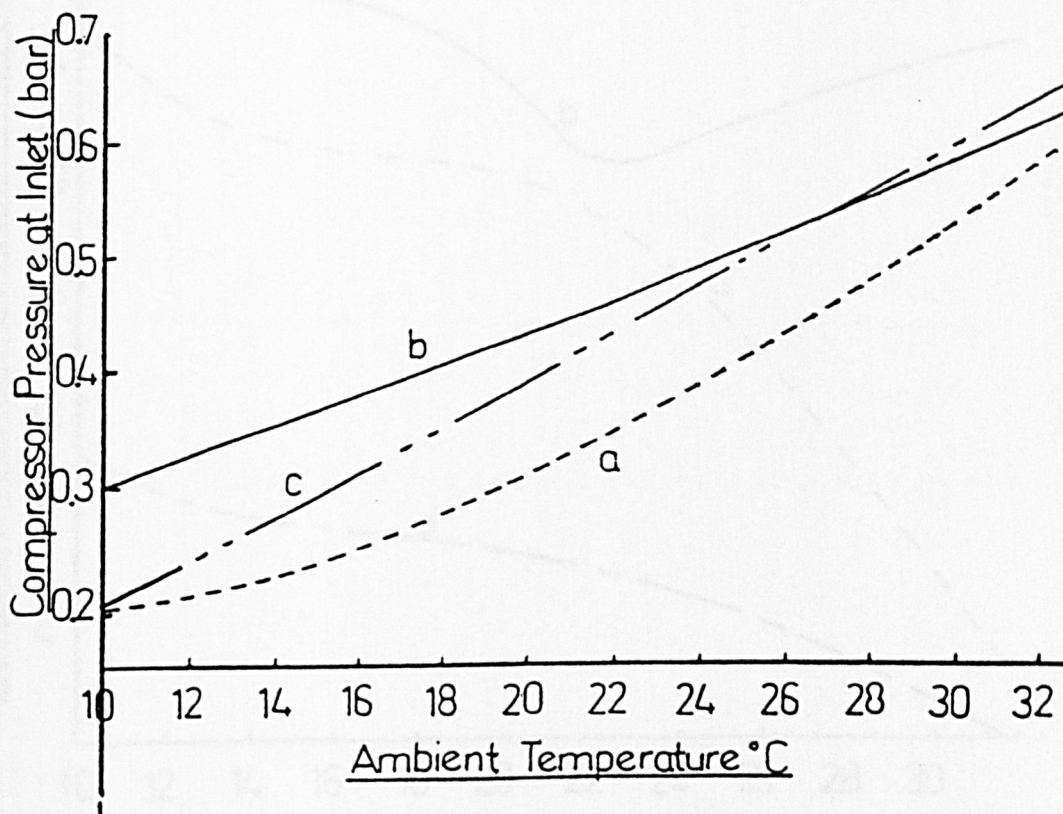


FIG. 7.17 SHOWING THE VARIATION IN COMPRESSOR PRESSURE RATIO WITH
AMBIENT TEMPERATURE PREDICTED USING COMP-OD

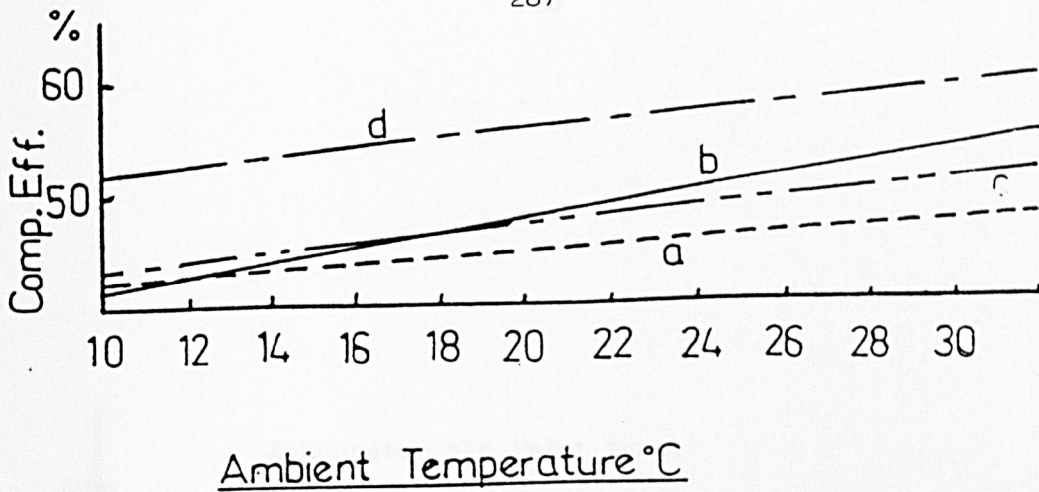


FIG. 7.18 SHOWING THE VARIATION IN COMPRESSOR EFFICIENCY WITH AMBIENT TEMPERATURE PREDICTED USING COMP-OD

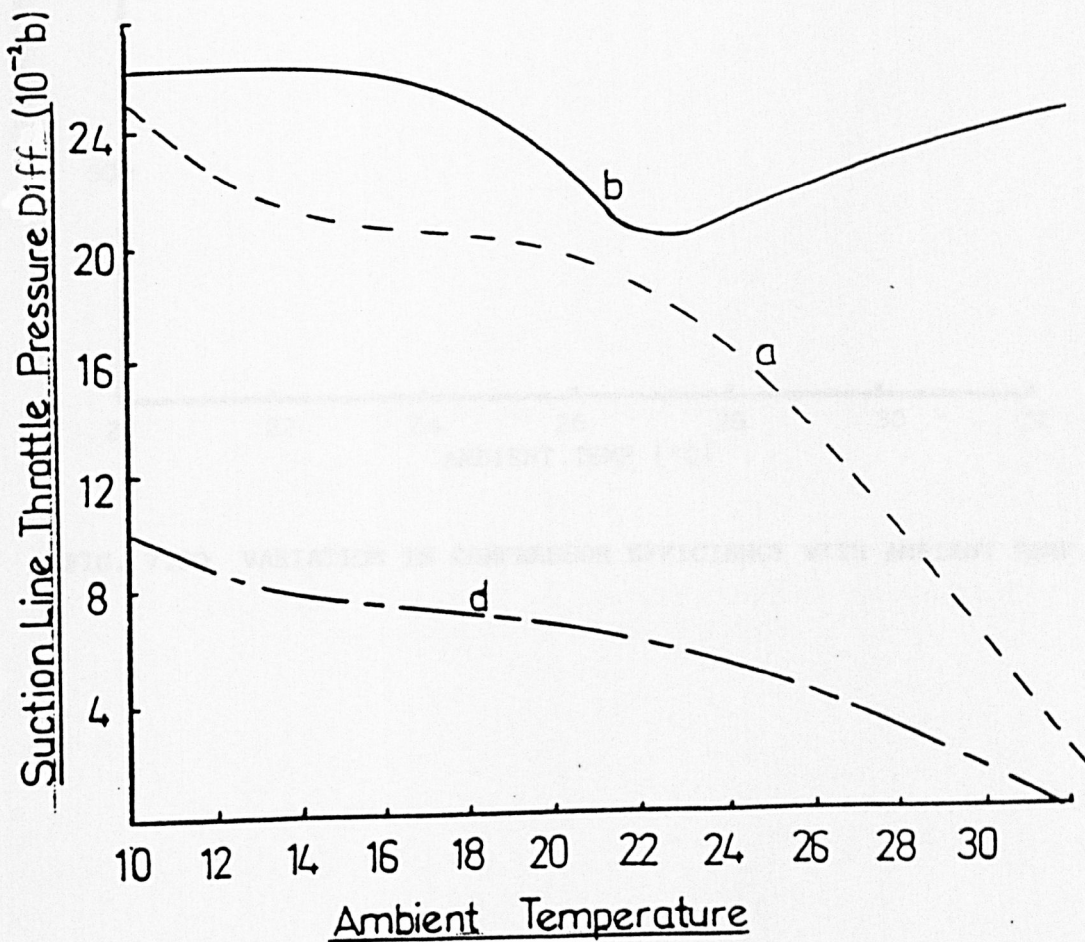


FIG. 7.19 SHOWING THE VARIATION IN SUCTION-LINE THROTTLE PRESSURE RATIO WITH AMBIENT TEMPERATURE

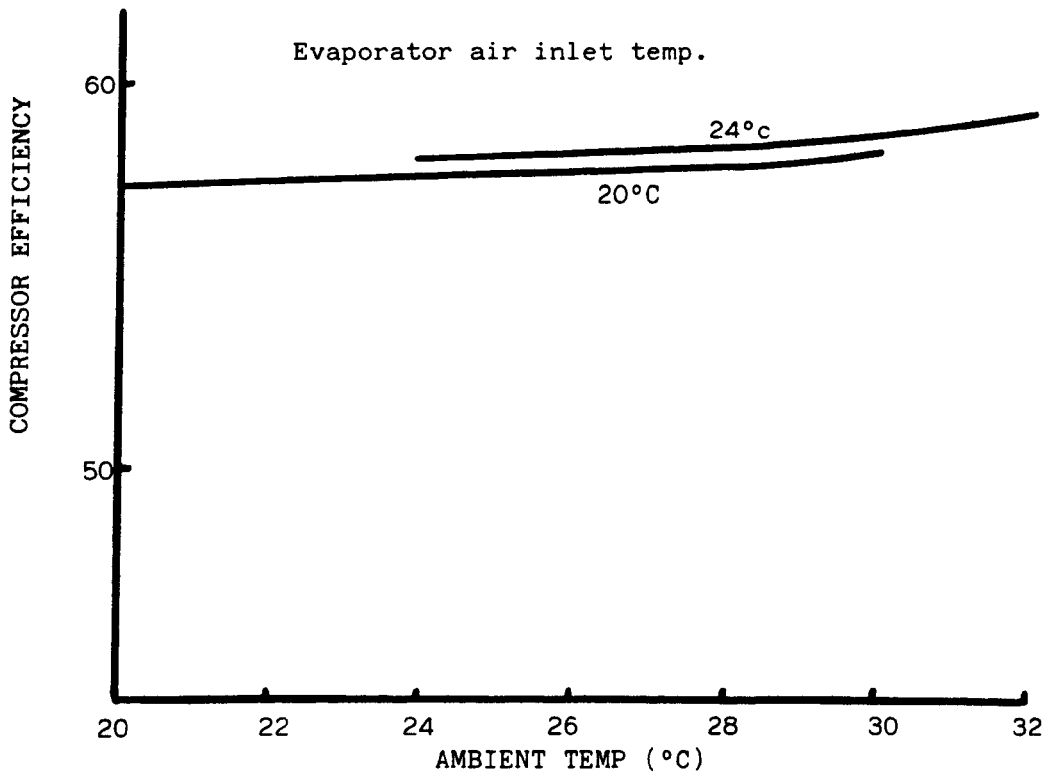


FIG. 7.20 VARIATION IN COMPRESSOR EFFICIENCY WITH AMBIENT TEMP.

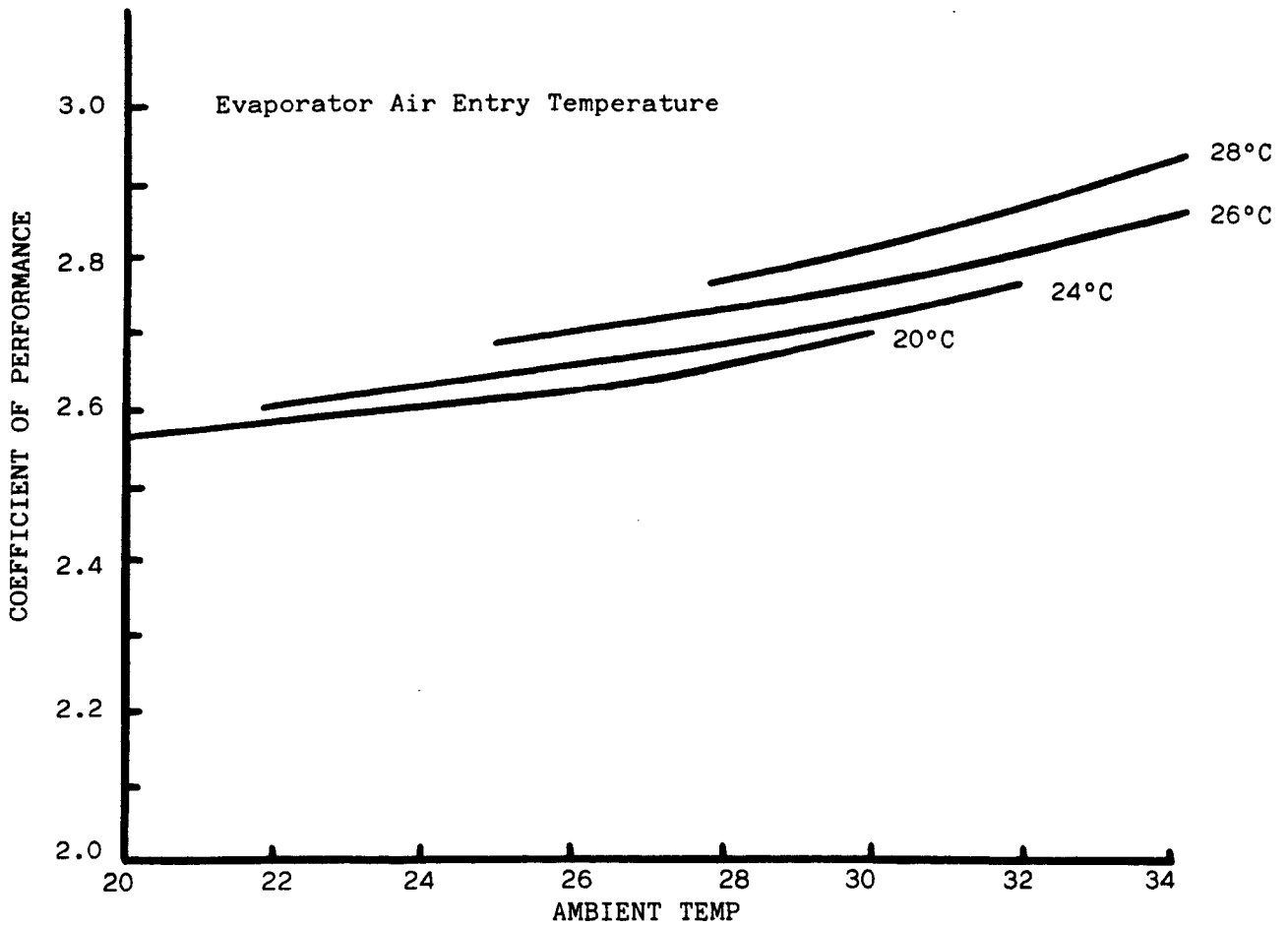


**FIG. 7.21 VARIATION IN COP WITH AMBIENT TEMPERATURE
(SINGLE-LINE) CHARACTERISTIC**

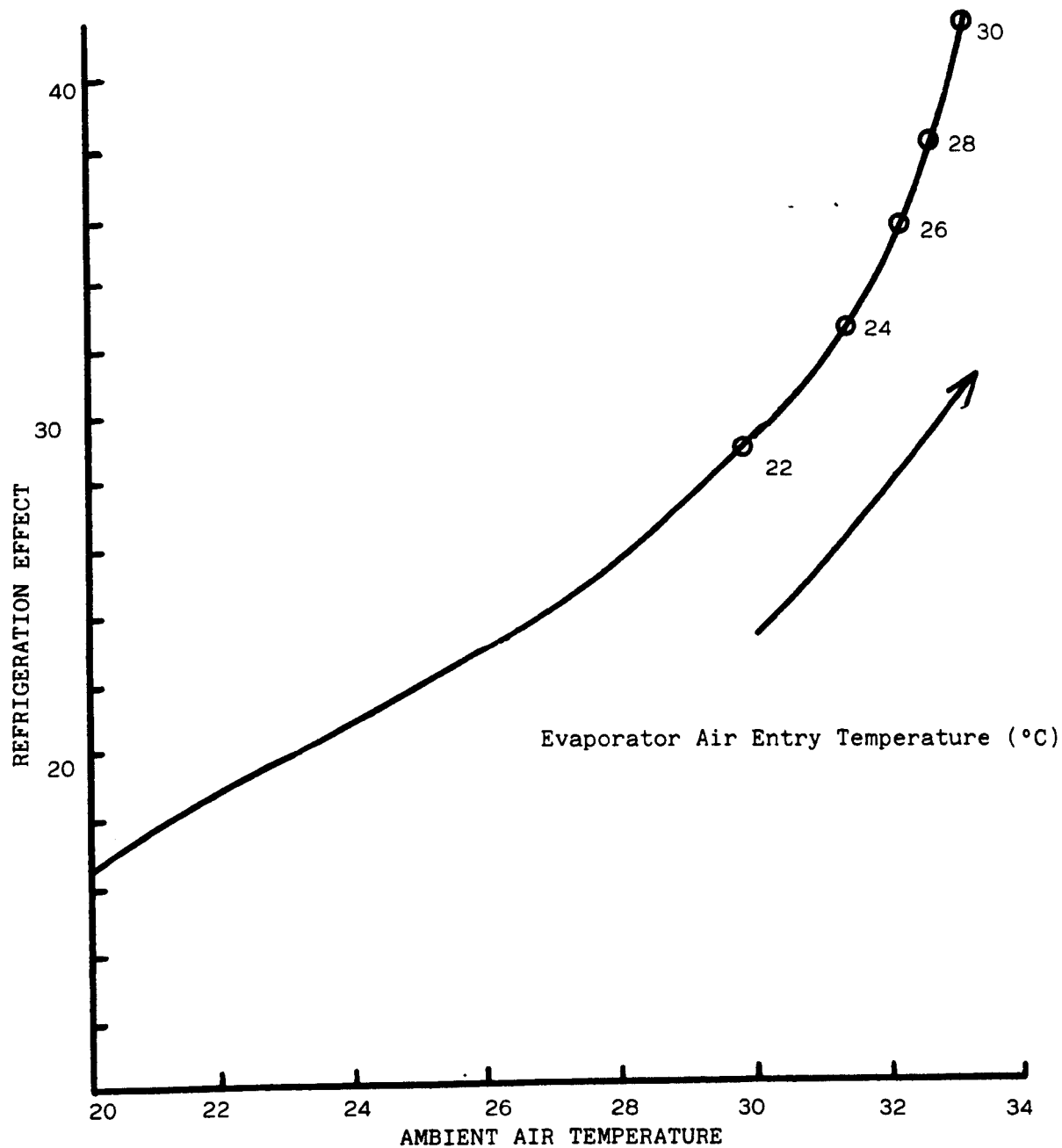


FIG. 7.22 VARIATION IN REFRIGERATION EFFECT WITH AMBIENT TEMPERATURE (SINGLE-LINE COMP CHARACTERISTIC)

	(a)	(b)	(c)
Rotational Speed	69120 rpm	69120	69120
Evap Temp	7.5°C	5°C	5°C
Evap Press	0.55 bar	0.495	0.495
Comp. Inlet Press.	0.54 bar	0.485	0.485
Comp. Inlet Temp.	21.6°C	21°C	21°C
Degrees of Superheat.	2°C	2°C	2°C
Cond Temp.	50°C	55°C	55°C
Cond Press.	2.35 bar	2.7 bar	2.7
Comp Efficiency	44.2%	45.8%	45.8%
Mass Flow Rate	363.8 gm/s	263 gm/s	263 gm/s
Comp Press Ratio	4.36	5.58	5.58
Power Output	18.6 KW	18.67 KW	18.67 KW
Refrigeration Effect	47. KW	40.4 KW	40.4 KW
C.O.P.	2.53	2.16	2.16
Motor Efficiency	90%	90%	90%
Electrical Supply Frequency	1200 H _z	1200 H _z	1200 H _z
Cooland Heating	2.1 KW	2.1 KW	2.1 KW
Power Supply to Motor	20.67 KW	20.74 KW	20.74 KW
Air Temp Entering Evap	28°C	25°C	15°C
Air Temp Entering Cond	33.3°C	25°C	35°C

**Table 7.4 Sample Design Point
Data (Results from COMPERF)**

PART-LOAD PERFORMANCE DATA

Evaporator Air-On Temperature	38 °C
Condenser Air-On Temperature	38 °C
COMPRESSOR DATA	
Rotational Speed	67931.9955
Comp. Inlet Press.	0.7198 bar
Comp Inlet Temp.	29.48 °C
Compressor Efficiency	50.26 %
Compressor Mass Flow Rate	372.4 gms/s
Compressor Pressure Ratio	3.968
Compressor Power Output	19.688 kw

MOTOR DATA

Motor Efficiency	0.874
Electrical power supply	22.502 kw
Slip Factor	0.9434
Motor Casing Pressure Loss	8.055E-3 bar

SYSTEM DATA

Condenser Heat Rejection	77.442 kw
Refrigeration Effect	54.952 kw
C.O.P. (Cooling)	2.442
Suction Line Throttle P-D	0 bar
Hot-Gas Bypass Percentage	0
Hot-Gas Bypass Mass Flow	0 gms
Evaporator Temp.	14.7857 °C
Evaporator Press.	0.7285 bar
Degrees of Superheat	2 °C
Condenser Temp.	56.897 °C
Condenser Press.	2.856 bar

TABLE 7.5 EXAMPLE OF OUTPUT DATA FROM COMP-OD

CHAPTER EIGHT

CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

The aim of this chapter is to summarise the findings of the study reported in this thesis.

Historically railway carriage air conditioning was restricted to countries with high ambient temperatures and high humidities, such as the Middle and Far East, and then only to the First Class compartments. For many years the only air conditioning on British Railways was provided by a rudimentary heating system and opening windows. More recently the application of air conditioning has increased significantly. Today high speed railway systems providing enhanced passenger comfort require railway carriages to have sealed windows and effective air conditioning.

The primary control variables affecting the cooling load on a railway carriage are solar radiation, ambient air temperature, train speed and number of passengers. Unlike conventional building systems railway carriages can have rapid variations in ventilation and cooling load. A railway carriage can be empty one moment and filled to capacity the next, carrying more than 200 passengers per coach during peak periods. Ambient load can also vary with the season. Because of the large window area of modern coaches solar gains can be significant. The installed cooling capacity can be as much as 23 kW in temperate climates to 93 kW in the tropics, Graham². A typical roof mounted system is shown in Figure 8.1 and shown installed in Figure 8.2. One potential application of the novel compressor studied in this project was to replace the existing unit in the type of packaged system shown in Figure 8.1.

Power supply for the compressor is an important factor. Railway carriages can be self powered from axle driven alternators combined with battery banks to store power for when the train is stationary. Power can also be made

available from the traction unit. A further alternative is to take electrical power supplied to the traction motor in the case of electric trains. In this case the voltage range is 600 to 1500 VDC, Graham².

Railway air conditioning systems often use open compressors because semi-hermetic and hermetic units are not considered to be sufficiently durable in the near industrial environment. The restrictions of limited voltage range and repetitive starts with frequent supply interruptions are reported to have severe effects on system reliability, Ford^{4a}.

Because the compressor can be roof mounted it is important that weight and physical size be kept to a minimum. The compressor must be robust, able to withstand a 7g loading at a peak frequency of 12 Hz during shunting tests according to the British Railways Board publications⁴⁷. In addition to dynamic loading being placed on the refrigeration system liquid carryover to the unit can occur which can harm valve gear and manifolds.

In the present project a novel refrigerant compressor concept was envisaged consisting of a 'small scale' centrifugal compressor driven by a high frequency induction motor. The motor and compressor impeller were mounted on a common drive shaft supported by aerodynamic bearings. This unique combination of 'state-of-the-art' compressor, bearing and motor technologies offers refrigeration system designers an advance compressor with improvements in system performance and reliability at lower capital and running costs than conventional compressors. The novel compressor would also offer significant reductions in weight and physical size making the concept particularly suitable for railway carriage applications where weight is an important factor.

The prototype compressor unit as designed was a semi-hermetic centrifugal type. Dynamic elements were supported in hybrid gas journal bearings, (mounted on compliant elastomer 'O' seals), and a plane hybrid double acting thrust bearing. The compressor impeller, (a highly modified vehicle turbocharger type), was driven by a three-phase squirrel-cage type induction motor. A general assembly drawing of the prototype compressor is shown in Figure 1.1 and a photograph of the compressor components is included in Figure 1.2. A drawing of the proposed compressor is shown in Figure 1.3. A detailed description of

the compressor is given in Section 1.2 and manufacturing drawings for individual components are included in Appendix A.

Comparisons between types of compressor are made in Section 1.3. Compressor types considered in Section 1.3 were; sliding vane, Buick¹⁹, Roots type blower, Screw, Laing¹² and O'Neill^{20,21}. The study concluded that positive displacement compressors tend to have lower specific speed values, reciprocating compressors having values less than 150. Centrifugal compressors, however, were shown to have specific speeds ranging from 40 to 350, Laing¹². In the present case it was concluded that the specific speed of the prototype unit was within the specific speed range for centrifugal compressors.

It was concluded from the study that the market position held by reciprocating machines in the refrigeration industry, (particularly the air conditioning industry), means that they offer the greatest challenge to the general application of the present novel compressor unit. A literature review described in Section 1.4, was undertaken to identify weaknesses in reciprocating compressor design in order that the advantages of the present concept may be more fully identified. It was concluded from this study that the main problem areas in the design and operation of reciprocating compressors are; bearing loads in reciprocating compressors increase with compression ratio thus bearing loads for air conditioning system compressors tend to be higher than those used in general refrigeration systems; coking of the valve gear resulting from oil breakdown leading to a fall off in compressor performance; special provisions in the design of refrigerators to ensure that lubricant carried over from the compressor is returned to the crankcase; pressure pulsations within suction and discharge manifolds due to the intermittent nature of flow can result in reductions in volumetric efficiency at part-load conditions; susceptibility to damage caused by liquid ingestion.

Unlike reciprocating units, centrifugal compressors carry the majority of the dynamic load symmetrically about the rotor so that journal bearing loads depend on the weight of the drive shaft and rotational speed rather than duty. The load on the thrust bearing will, however, increase with duty unless it is balanced, as in the present compressor unit which uses a special nut and labyrinth seal.

Centrifugal compressors are more suited to high pressure ratio and high compressor discharge applications. In the theoretical study of an air conditioner refrigeration system described in Chapter 7 it was concluded that some form of pressure ratio control would be required in order to keep the compressor operating point in a region of high efficiency. In this case a suction line throttle was proposed. The inclusion of such devices tend to reduce the mass flow rate through the system, due to the reduction in compressor inlet density caused by low inlet pressures produced by the throttling effect, resulting in a reduction in refrigeration effect.

Because the proposed compressor unit is oil free then all problems associated with oil breakdown at elevated compressor discharge temperatures are alleviated. Special provision in the design of a refrigerator for the recovery of lubricant is not required. Therefore, suction and discharge pipe pressure losses can be kept to a minimum by selecting larger diameter pipe lines than otherwise would be possible. Because of the low saturation pressures experienced by the chosen refrigerant, (Freon 11), pipe lines could be manufactured from a polymer type material.

The existing and well developed manufacturing technology for small scale turbocharger impellers and volutes makes the present compressor concept attractive. However, as was found in this study there remains a significant requirement for research into the isentropic performance of the present compressor unit. Compared with conventional machines the novel compressor unit would have lower weight, would be physically smaller, produce less noise and vibration, have low torque start and low maintenance costs.

For ease of development a semi-hermetic design was proposed for the prototype compressor. An hermetic design is proposed for any future production variant because being a factory sealed unit avoids tampering by inexperienced service engineers.

The compressor power supply was considered in Section 1.5 where it was concluded that solid state inverter and speed controllers with the required power and frequency range were not currently available. Although technically feasible the cost implications would outweigh all other advantages of the compressor unit. It was concluded that the use of pole switching would also be unsuitable as the reduction in flow and pressure ratio imposed by halving the speed of the unit

would be too great. For simplicity and cost reasons a rotary inverter system was designed to power the prototype compressor during the testing stages of the project.

During preliminary studies it was concluded that five key areas of technology required further research to ensure the success of the project. These were shown to be; compressor stage performance, electric motor cooling, stress analysis of the laminated motor rotor, drive shaft critical speed analysis, bearing system design study and computer simulation of refrigerator system incorporating the proposed compressor unit.

A study into the performance of centrifugal compressors was evaluated in Chapter 2. The aim was to identify causes of inefficiency in centrifugal compressors and to make recommendations for future modifications of the prototype unit which would improve the maximum isentropic efficiency from the current value of approximately 60% to something over 70% making the application of the compressor cost competitive with existing reciprocating machines. Typical distribution of losses in a compressor stage are listed in Table 2.1. which shows that losses are greatest in the impeller. From a study of factors affecting isentropic efficiency it was concluded that improvements in efficiency are possible by reducing the vane exit angle, increasing the Stage Reaction, (increasing the proportion of diffusion with the impeller), and by reducing the inlet relative Mach number.

The design of the electric motor was evaluated in Chapter 3. Assistance with the electrical design was given by Ron Hill²⁷. The mechanical integrity of the laminated rotor and motor cooling system effectiveness were identified as the priority research areas.

The high rotational speed of the motor and the relatively low yield stress of the lamination material, (Transil 315 - 370 MPa), restricted the physical dimensions of the rotor. A plane stress model was used to determine the outer and inner diameters of the rotor laminations. The overall length of the rotor was determined from a volume constraint imposed by electro-magnetic considerations. A more complex stressing model showed that the existance of the conductor rods within the rotor pack would have increased the hoop stress at the inner radius of the rotor beyond the yield stress of the material. This resulted in the proposed motor dimensions being modified.

The study concluded that the rotor lamination pre-load was important because it ensured the dimensional stability of the rotor. The effects of thermal and mechanical stresses during operation were sufficient to cause the laminations to separate and move relative to each other thus disturbing the balance of the rotor and drive shaft assembly. It was found necessary to provide an amount of axial pre-load during the assembly of the drive shaft. Analysis showed that the conductor rods alone were insufficient to provide the required pre-load, Sections 3.4 and 3.5. It was concluded that an assembly pre-load of 211 kN would be provided by a torque loaded nut which would clamp the laminated rotor against a machined shoulder on the drive shaft. It was estimated that the nut would need to be torque loaded to 386 N-m. A schematic representation of the rotor assembly positioned on the drive shaft is shown in Figure 3.4. Analysis also described in Chapter 3 indicated the rotor busting speed to be not less than 89,870 rpm. During subsequent testing the rotor operated at 90,000 rpm without bursting but required re-balancing afterwards because the laminations were found to have moved.

The study concluded that motor cooling was a critical area of study due to the high power density of the machine. Analysis described in Section 3.9 showed that both natural and forced convective heat transfer from the surface of the motor casing with finning was insufficient to maintain the winding temperatures below an acceptable limit for Class A insulation, (160°C), used in the construction. Internal cooling techniques were investigated. These included passing compressor discharge gas through the motor casing, using suction gas in the same way and by spraying liquid condensate directly onto the stator and windings. The latter method proved preferable, (from a performance standpoint), and should be considered for further study. Because a vapour loop was to be used during testing making the use of a condenser unnecessary it was decided to use the suction gas technique. It was concluded that the motor would be cooled by a combination of internal methods and natural convection from the external surfaces of the motor casing. Cooling passages running through the stator were proposed in order to improve heat transfer performance. These are shown in Figures 3.9 and 3.14. The design of these passages required a compromise between heat transfer performance and pressure, (pumping), losses. The off-design performance of the cooling system was also investigated and found to be satisfactory.

The design of the drive shaft and bearing assemblies is

evaluated in Chapter 4. Because of the slenderness of the drive shaft assembly and the distance between the support bearings it was important to ensure that the design operating speed of the compressor did not coincide with a critical lateral or torsional speed. Accurate estimates of undamped lateral modes of vibration were obtained using a finite element program based on the M-P technique. This analysis is described in Appendix G and the results are listed in Table 4.1. The natural torsional modes of vibration were estimated using the 'equivalent shaft' method. This analysis is described in Section 4.5. Potential sources of machine vibration were identified. External sources were not considered in the design of the prototype compressor unit. Potential sources of vibration were identified as; aerodynamic buffeting, imbalance of magnetic forces caused by the misalignment between the rotor and stator and out-of-balance mechanical forces. It was concluded that aerodynamic buffeting forces would be negligible due to the symmetry and 'smoothness' of the drive shaft assembly and was discounted. Magnetic out-of-balance forces were also considered to be negligible because of the use of close tolerance bearings which ensured alignment between the rotor and stator. Following the recommendations of Reiger⁴⁴ the maximum residual mechanical out-of-balance was set at 3.9×10^{-4} N-m, which was found to be easily achievable. Maximum stress in the drive shaft at 105% of the design speed were estimated to be within the yield stress limit for the material, (EN24 steel).

The design of the journal bearings is described and evaluated in Section 4.11. A comparison of aerodynamic and aerostatic bearings is included in Section 4.8. A hybrid bearing system incorporating the best that both systems offer was selected for the prototype design which provided aerostatic support for the shaft assembly during start-up and run-down, thus reducing bearing wear, and aerodynamic support during normal operation. In order to provide a compliant bearing mounting it was decided to support each bush in 'Vitol' elastomer 'O' ring type seals, a method which was investigated by Kerr⁴⁵. Tests were carried out to provide information on the compatibility of Vitol rubber with Freon 11. These tests are evaluated in Section 4.13. Static and dynamic bearing loads were estimated. This analysis is described in Section 4.10 and the results are listed in Table 4.4. Photographs of the bearing bushes are included in Figure 4.1. Because of the intrinsic simplicity of the aerodynamic bearing it was concluded that a future objective should be to develop such a bearing system for

later production compressor units. The design of the thrust bearings was evaluated in Section 4.16. A double acting hybrid system was selected which provided aerostatic support for the shaft assembly during startup and shutdown and aerodynamic support under normal operating conditions. A photograph of the bearing is included in Figure 4.2. The axial force on the drive shaft came from the action of the impeller. Some of this load was balanced by a labyrinth seal and the remainder was supported by the thrust bearing.

To prevent excessive wear during start-up and shut-down the bearing surfaces were treated with Sulfinuz, (a proprietary name). The process, discussed in Chapter 4, impregnates a steel surface with sulphur which acts as a solid lubricant to prevent binding and sticking of bearing surfaces. The process was found to produce distortion of bearing surfaces, particularly the thrust plates and to leave a thick oxide coating which was difficult to remove without spoiling the bearing surface. It was concluded from this study that a low temperature sulphur treatment should be used in future such as Sulf BT thereby minimising distortion, Neale⁷⁴.

The manufacture of the prototype compressor unit was described and evaluated in Chapter 5. It was concluded that the manufacturing tolerances specified in the detailed drawings in Appendix A were achievable.

Although fatigue failure of the drive shaft was unlikely experience has shown that the failure of lightly loaded components is possible due to the effects of local stress raisers, such as scratches, surface defects and localized burning caused by poor technique during grinding. The overall surface finish of the drive shaft was specified at 0.2 micron CLA and 0.1 micron CLA in the region of the journal bearings.

Heat treatment of the drive shaft was discussed in Section 5.2.3. It was concluded that the drive shaft should be annealed after preliminary machining and again before grinding when hardening and tempering were also carried out in accordance with the British Standard. During heat treatment the shaft was hung vertically in the furnace to prevent distortion. Other major components were also annealed between preliminary and final machining to avoid subsequent distortion. It was important to keep distortion to a minimum in order to ensure correct alignment of bearings, drive shaft assembly and stator. The heat treatment of the bearing

assemblies was discussed in Sections 5.3 and 5.4.

Following the manufacture of the compressor unit test facilities were constructed and test-runs carried out. The testing of the compressor unit was described and evaluated in Chapter 6. Two series of experiments were carried out. During the first the compressor unit was operated in an air environment. This allowed the bearing system to be developed, the mechanical integrity of the drive shaft assembly to be proved and the motor to be tested without the additional problems created by having to seal and charge a refrigeration system between modifications and rebuilds. The aim of the second series of experiments was prove the concept in a refrigeration system.

Bearing misalignment was overcome by redesigning the bearing bushes with less interference between the 'O' seals and the bearing housing. This modification resulted in the bushes moving within the housing during the operation of the compressor. To overcome this problem plastic clips were fitted on each side of the bushes to restrain axial movement. Later the plastic clips were exchanged for steel clips when the former were found to be susceptible to attack from Freon 11. In total there were thirteen hours of testing in the first series of experiments at the end of which it was concluded that the bearing system would operate in both the aerodynamic and aerostatic modes and the transition speed between the aerostatic and aerodynamic modes was approximately 15,000 rpm. The shaft assembly with the laminated rotor was shown to withstand an overspeed condition to 90,000 rpm.

A Freon vapour loop test facility was designed and built for the second series of experiments. A description of the facility is included in Section 6.9.1. A major development problem in using the test facility was the control of the compressor speed. This was found to be difficult due to the interaction of the compressor motor, alternator and D.C. motor.

Six test runs were carried out. The first three were aborted when the main test site D.C. circuit breaker failed. The fourth was abandoned after 50 minutes when the bearing system failed, the fifth test run was successful and lasted 90 minutes and the final test was aborted after 50 minutes when the thrust bearing seized. The failure of the thrust

bearing resulted from the failure of the journal bearing bush plastic retaining clips during test run number four. This was discussed in Section 6.12.2.

Although the results of the 'Freon' tests were not conclusive they were encouraging and showed that the compressor concept was feasible.

A computer based simulation model of a refrigeration system incorporating the prototype compressor is described and evaluated in Chapter 7. Two primary programs were written, COMPERF and COMP-OD. The first carried out design point calculations and determined constants for heat exchanger and electric motor models. A listing of COMPERF is included in Appendix B. The second program, COMP-OD, was designed to undertake part-load calculations for the simulated refrigeration system. Factors taken into account included changes in heat exchanger performance with ambient temperature, variation in motor efficiency, pressure losses through the motor casing, operation of a suction-line-throttle valve to control the operating point of the compressor and variations in compressor performance, (pressure ratio, flow, isentropic efficiency). A listing of COMP-OD is included in Appendix C.

Both programs operated satisfactorily over the desired range of ambient conditions. However, there were convergence problems with the COMP-OD program. From the use of COMP-OD it was possible to predict that the present compressor would operate satisfactorily across the ambient temperature range considered. The values of C.O.P. compare favourably with commercial units and with what must be achieved for competitive performance. The model also predicted that the refrigeration effect of a system would increase with evaporator and condenser capacities. Improvements in compressor operating efficiency were shown to be possible by increasing the area of the evaporator. Compressor efficiency was shown to fall with ambient temperature. Some improvement in compressor efficiency was found to be possible by reducing the condenser heat exchange area. Although the performance of a refrigerator can be improved by increasing the area of the evaporator and condenser there are cost, space and weight disadvantages.

It was concluded that the 'floating-point' is unsatisfactory for the present system. The 'single-line' approach provides

a more precise control of compressor performance. In both cases however a suction-line throttle was the only active element of the control system. Future research should include an investigation into the application of microprocessor based proportional type of controller to modulate the suction-line-throttle via a stepper motor actuator. This system should also include close control over the system expansion valve.

The present study demonstrated the feasibility of the novel compressor unit concept. The results of this project laid foundations for development of a production machine which would have potential uses in process refrigeration, air conditioning and heatpump applications. In order to further the development of the compressor unit the following recommendations are made.

8.2 Recommendations for Future Research

Resulting from the present study the following research should be undertaken towards the development of a production compressor unit.

An investigation into improvements in compressor isentropic efficiency resulting from the introduction of the following into the impeller design; backsweep, increase in stage reaction and inlet swirl. These modifications were discussed in Chapter 2.

It is recommended that the development of the bearing system be continued. Alternative types of bearing should be tried, such as foil and tilting pad types. The possibility of mounting each journal bearing bush in a compliant diaphragm instead of in elastomer 'O' seals as in the present case should be investigated. The use of a spiral groove type thrust bearing should be considered, Whipple¹¹².

The use of Sulfnuz treatment on bearing surfaces should be abandoned. An alternative, low temperature treatment such as Sulf BT should be considered. This is discussed in Chapter 5.

From a performance standpoint liquid, (condensate), spray cooling of the motor windings would be an improvement on the 'suction gas' technique used in the design of the prototype unit. It is therefore recommended that further research be undertaken into this aspect.

The development of a solid state, three phase inverter and speed controller with the desired power output and frequency should be undertaken, (11 kW, 415 VAC, 900 to 1200 Hz). This would give the novel compressor more general application.

The development of a control system, for the 'vapour-loop' test facility as described in Section 6.9 is highly recommended.

Further research into the general design of the compressor unit is recommended with the aim of reducing physical size and weight. Areas which require further study are the motor casing. This could be made lighter by using aluminium or a thin walled steel vessel. The length of the drive shaft should be shortened. This would make the bearing support assemblies less massive by reducing their length. Manufacturing the support assemblies from aluminium alloy casings instead of mild steel as in the prototype machine would help in this.

For future research it is strongly recommended that separate test facilities are constructed for development testing of the bearing systems, (thrust and journal bearing separately), the drive motor and the impeller-volute assembly. The vapour loop test facility should be developed by introducing automatic data gathering techniques as well as automating the control system.

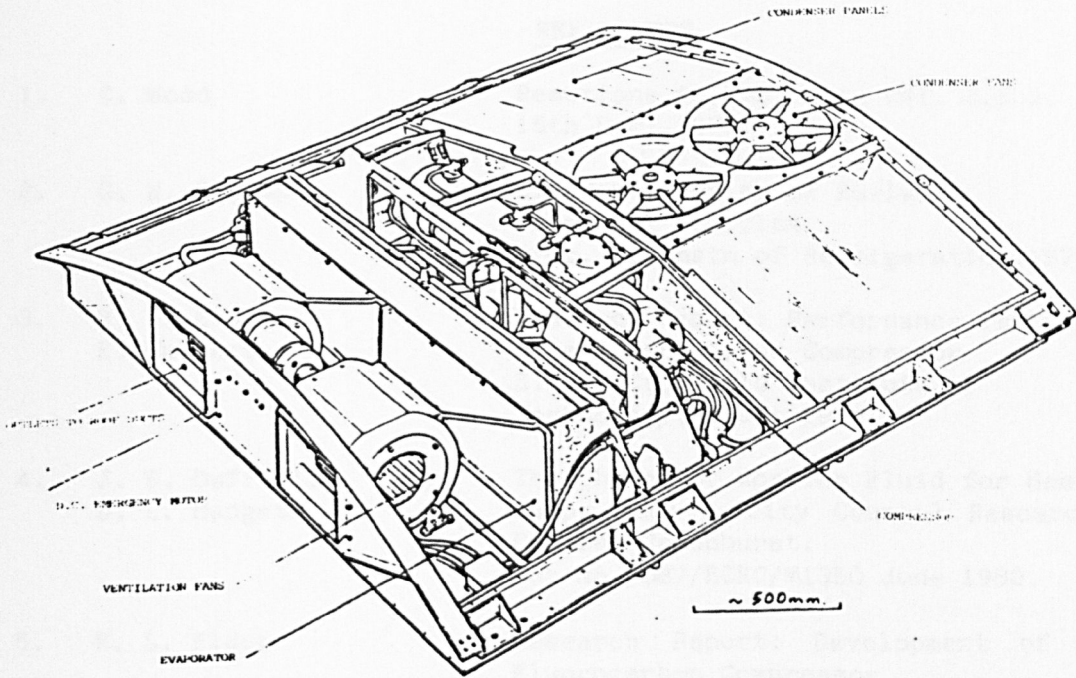


FIG. 8.1 STONE PLATT ROOF MOUNTED PACKAGED
AIR CONDITIONING UNIT TYPE RP40

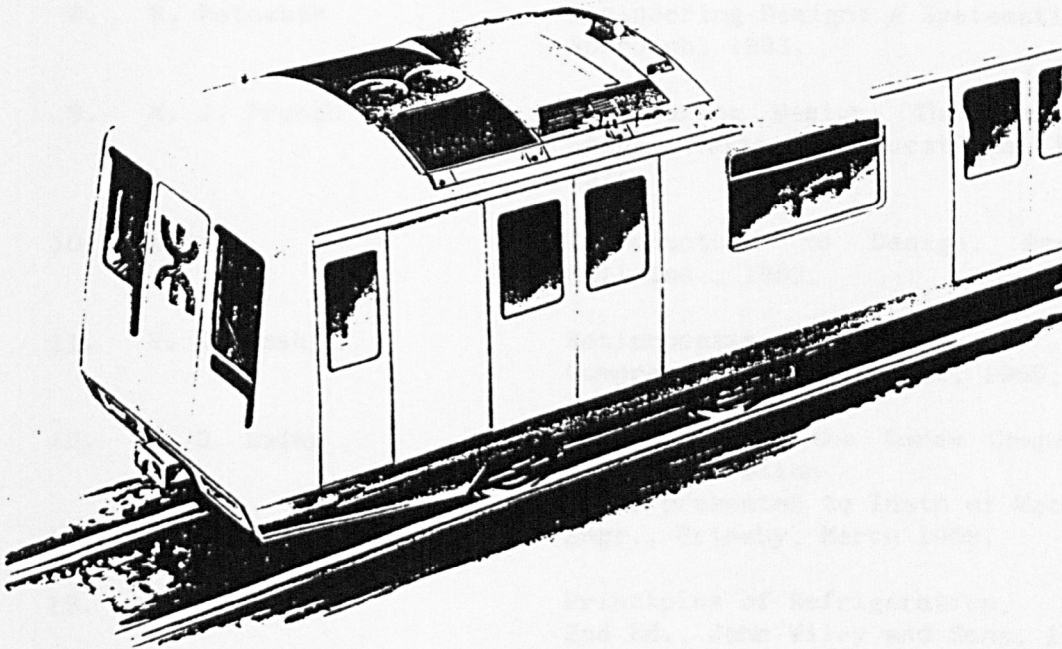


FIG. 8.2 RP40 UNIT AS MOUNTED IN COACH ROOF

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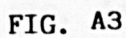
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APPENDIX A**MANUFACTURING DRAWINGS FOR PROTOTYPE COMPRESSOR UNIT**

<u>COMPONENT DESIGNATION</u>	<u>DRAWING NUMBER</u>	<u>FIGURE NUMBER</u>
Compressor wheel	Holset Drg	A1
Shaft	PB15032/1	A2
Support Fixture	PB15032/2	A2
Lockwasher	15032/3	A2
Slotted nut	PB15032/4	A2
Rear Brg.Support Assy.	PB15033/1	A3
End cap	PB15034/1	A4
Blanking Plug	PB15034/2	A4
Fwd.Brg.Support Assy	PB15035	A5
Dummy Rotor	PD15036	A6
Motor Case	PB15037	A7
Thrust Brg.	PB15038/1	A8
Spacer	PB15038/2	A8
Thrust Plate	PD15039/1	A9
Support	PD15039/2	A9
Spacer	PC15040/1	A10
Special Nut	PC15040/2	A10
Peg Spanner	PC15040/3	A10
Back Plate	PC15041/1	A11
Gas Seal	PC15042/1	A12
Spacer	PC15042/2	A12
Location Pin	PD15043/1	A13
Spacer	PD15043/2	A13
Comp Assy Tool	PD15044	A14
Foot Assy	PC15045	A15
Probe Support	PD15046	A16
Rear Brg Bush	PD15047/1	A17
Fwd Brg. Bush	PD15047/2	A17

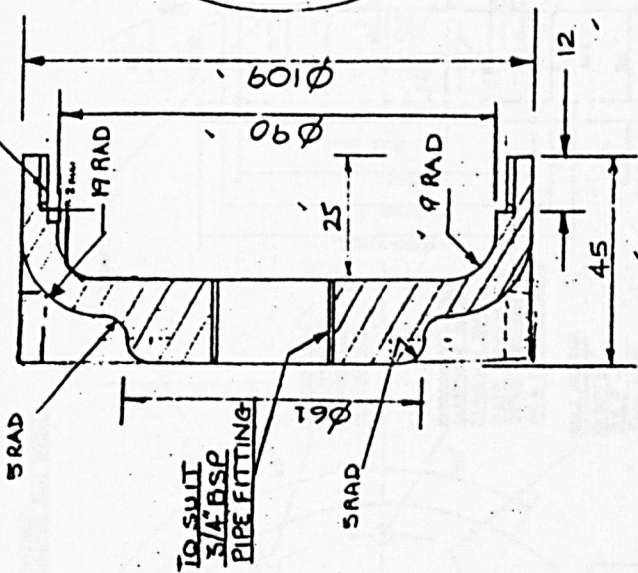
TABLE A1 SUMMARY OF DRAWING NUMBERS



ALL DIMENSIONS IN MILLIMETRES UNLESS OTHERWISE STATED.

DRAWING No.	
PD 15034	
ISSUE	MODIFICATION
A	—

20TPI 100 HAZ D/A
WHITWORTH FORM
TO SUIT ITEM PT NO PD15033/1



1/4"-28TPI
AN. FORM

BLANKING PLUG

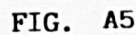
SCALE = 2 x FULL SIZE

END CAP
SCALE = FULL SIZE

THIRD ANGLE PROJECTION

GENERAL TOLERANCE ON DIMENSIONS ±0.12		JOB No.		No. OF SETS RECD		SHEET SIZE		ITEM		PART No.		DESCRIPTION		No. OFF		MATERIAL		SPEC.		REMARKS	
MACHINED ✓ ALLOWER		LONG CAUSE		1		A 3		1		PD15034		END CAP		ONE		M.S					
UNMACHINED		OTHER DIMENSIONS AS STATED				FULL UNLESS STATED		DRAWN		CHKD		APPR		STRESS APPVD							
						FINISH		ISSUED BY S M E													
WELD WHERE SHOWN THUS		USED ON: DRG PD15031				SELF		CRANFIELD INSTITUTE OF TECHNOLOGY												DRAWING No. PD15034	
FINE WHERE SHOWN THUS								CRANFIELD												SHT. 1 OF 1 SHEET	

FIG. A4

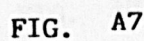


Technical drawing of a mechanical part showing a cross-section A-A. The part is a rectangular block with a central hole. The cross-section A-A shows a central hole with a diameter of 51.81 ± 0.012 and a wall thickness of 2.5. The block has a total width of 79 and a total height of 59. The top surface is chamfered at 45° . The drawing includes a top view and a side view.

THIRD. ANGLE PROJECTION

2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 91 92 93 94 95 96 97 98 99 100 101 102 103 104 105 106 107 108 109 110 111 112 113 114 115 116 117 118 119 120 121 122 123 124 125 126 127 128 129 130 131 132 133 134 135 136 137 138 139 140 141 142 143 144 145 146 147 148 149 150 151 152 153 154 155 156 157 158 159 160 161 162 163 164 165 166 167 168 169 170 171 172 173 174 175 176 177 178 179 180 181 182 183 184 185 186 187 188 189 190 191 192 193 194 195 196 197 198 199 200 201 202 203 204 205 206 207 208 209 210 211 212 213 214 215 216 217 218 219 220 221 222 223 224 225 226 227 228 229 230 231 232 233 234 235 236 237 238 239 240 241 242 243 244 245 246 247 248 249 250 251 252 253 254 255 256 257 258 259 260 261 262 263 264 265 266 267 268 269 270 271 272 273 274 275 276 277 278 279 280 281 282 283 284 285 286 287 288 289 290 291 292 293 294 295 296 297 298 299 300 301 302 303 304 305 306 307 308 309 310 311 312 313 314 315 316 317 318 319 320 321 322 323 324 325 326 327 328 329 330 331 332 333 334 335 336 337 338 339 340 341 342 343 344 345 346 347 348 349 350 351 352 353 354 355 356 357 358 359 360 361 362 363 364 365 366 367 368 369 370 371 372 373 374 375 376 377 378 379 380 381 382 383 384 385 386 387 388 389 390 391 392 393 394 395 396 397 398 399 400 401 402 403 404 405 406 407 408 409 410 411 412 413 414 415 416 417 418 419 420 421 422 423 424 425 426 427 428 429 430 431 432 433 434 435 436 437 438 439 440 441 442 443 444 445 446 447 448 449 450 451 452 453 454 455 456 457 458 459 460 461 462 463 464 465 466 467 468 469 470 471 472 473 474 475 476 477 478 479 480 481 482 483 484 485 486 487 488 489 490 491 492 493 494 495 496 497 498 499 500 501 502 503 504 505 506 507 508 509 510 511 512 513 514 515 516 517 518 519 520 521 522 523 524 525 526 527 528 529 530 531 532 533 534 535 536 537 538 539 540 541 542 543 544 545 546 547 548 549 550 551 552 553 554 555 556 557 558 559 560 561 562 563 564 565 566 567 568 569 570 571 572 573 574 575 576 577 578 579 580 581 582 583 584 585 586 587 588 589 590 591 592 593 594 595 596 597 598 599 600 601 602 603 604 605 606 607 608 609 610 611 612 613 614 615 616 617 618 619 620 621 622 623 624 625 626 627 628 629 630 631 632 633 634 635 636 637 638 639 640 641 642 643 644 645 646 647 648 649 650 651 652 653 654 655 656 657 658 659 660 661 662 663 664 665 666 667 668 669 670 671 672 673 674 675 676 677 678 679 680 681 682 683 684 685 686 687 688 689 690 691 692 693 694 695 696 697 698 699 700 701 702 703 704 705 706 707 708 709 710 711 712 713 714 715 716 717 718 719 720 721 722 723 724 725 726 727 728 729 730 731 732 733 734 735 736 737 738 739 740 741 742 743 744 745 746 747 748 749 750 751 752 753 754 755 756 757 758 759 760 761 762 763 764 765 766 767 768 769 770 771 772 773 774 775 776 777 778 779 780 781 782 783 784 785 786 787 788 789 790 791 792 793 794 795 796 797 798 799 800 801 802 803 804 805 806 807 808 809 810 811 812 813 814 815 816 817 818 819 820 821 822 823 824 825 826 827 828 829 830 831 832 833 834 835 836 837 838 839 840 841 842 843 844 845 846 847 848 849 850 851 852 853 854 855 856 857 858 859 860 861 862 863 864 865 866 867 868 869 870 871 872 873 874 875 876 877 878 879 880 881 882 883 884 885 886 887 888 889 890 891 892 893 894 895 896 897 898 899 900 901 902 903 904 905 906 907 908 909 910 911 912 913 914 915 916 917 918 919 920 921 922 923 924 925 926 927 928 929 930 931 932 933 934 935 936 937 938 939 940 941 942 943 944 945 946 947 948 949 950 951 952 953 954 955 956 957 958 959 960 961 962 963 964 965 966 967 968 969 970 971 972 973 974 975 976 977 978 979 980 981 982 983 984 985 986 987 988 989 990 991 992 993 994 995 996 997 998 999 1000 1001 1002 1003 1004 1005 1006 1007 1008 1009 1010 1011 1012 1013 1014 1015 1016 1017 1018 1019 1020 1021 1022 1023 1024 1025 1026 1027 1028 1029 1030 1031 1032 1033 1034 1035 1036 1037 1038 1039 1040 104

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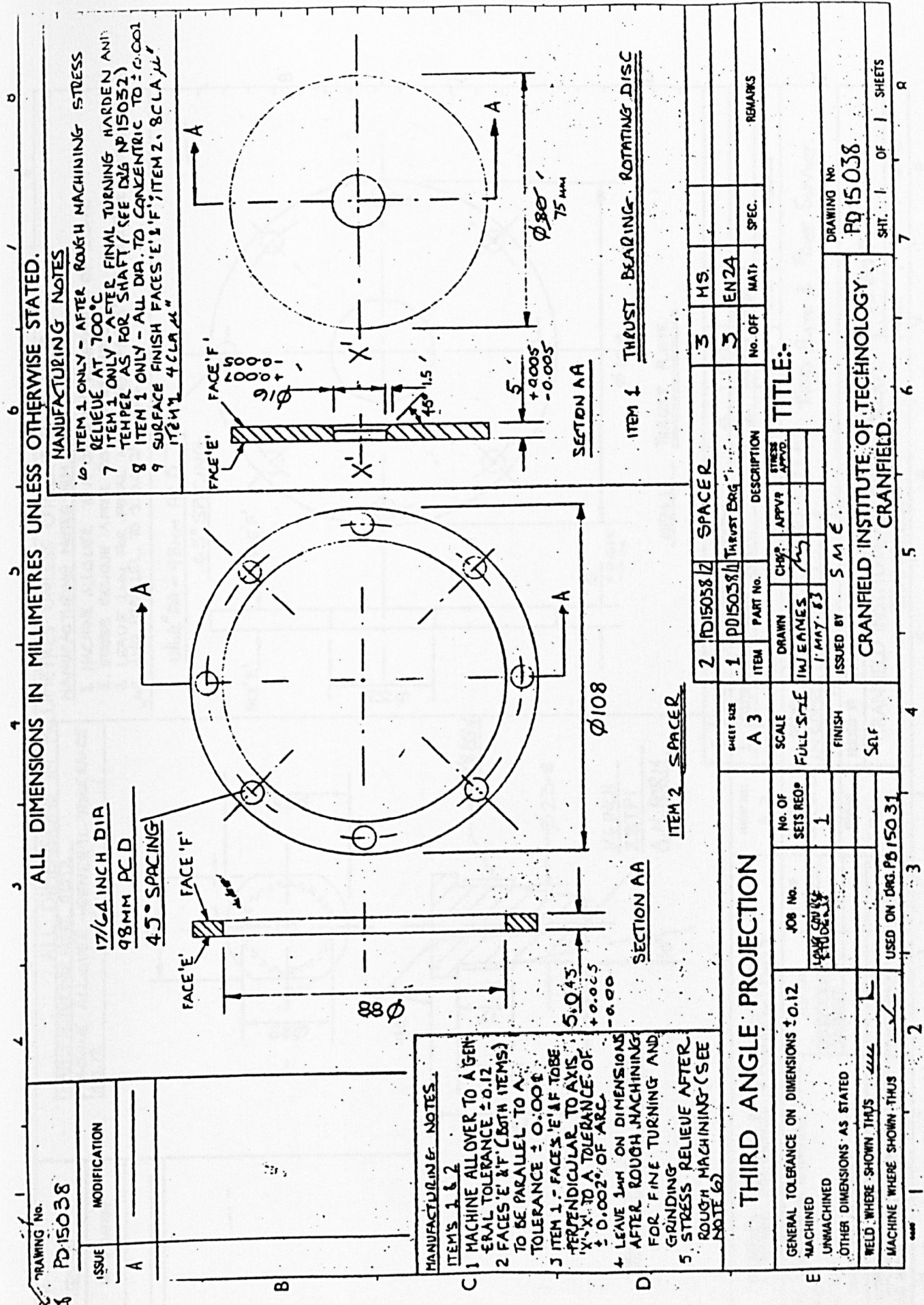
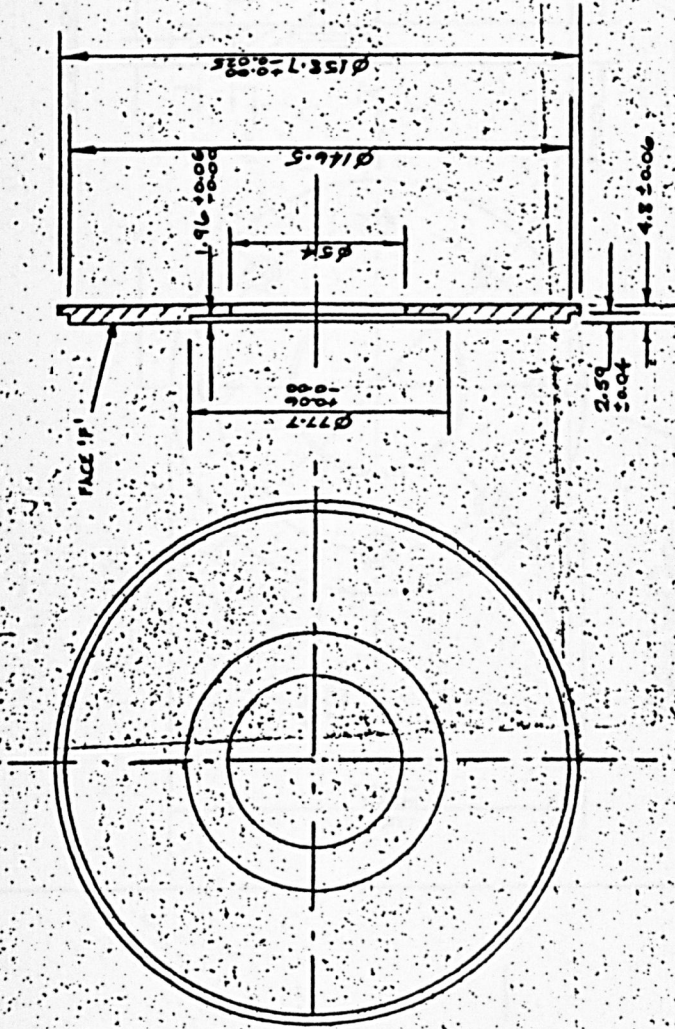


FIG. A8

FIG. A9

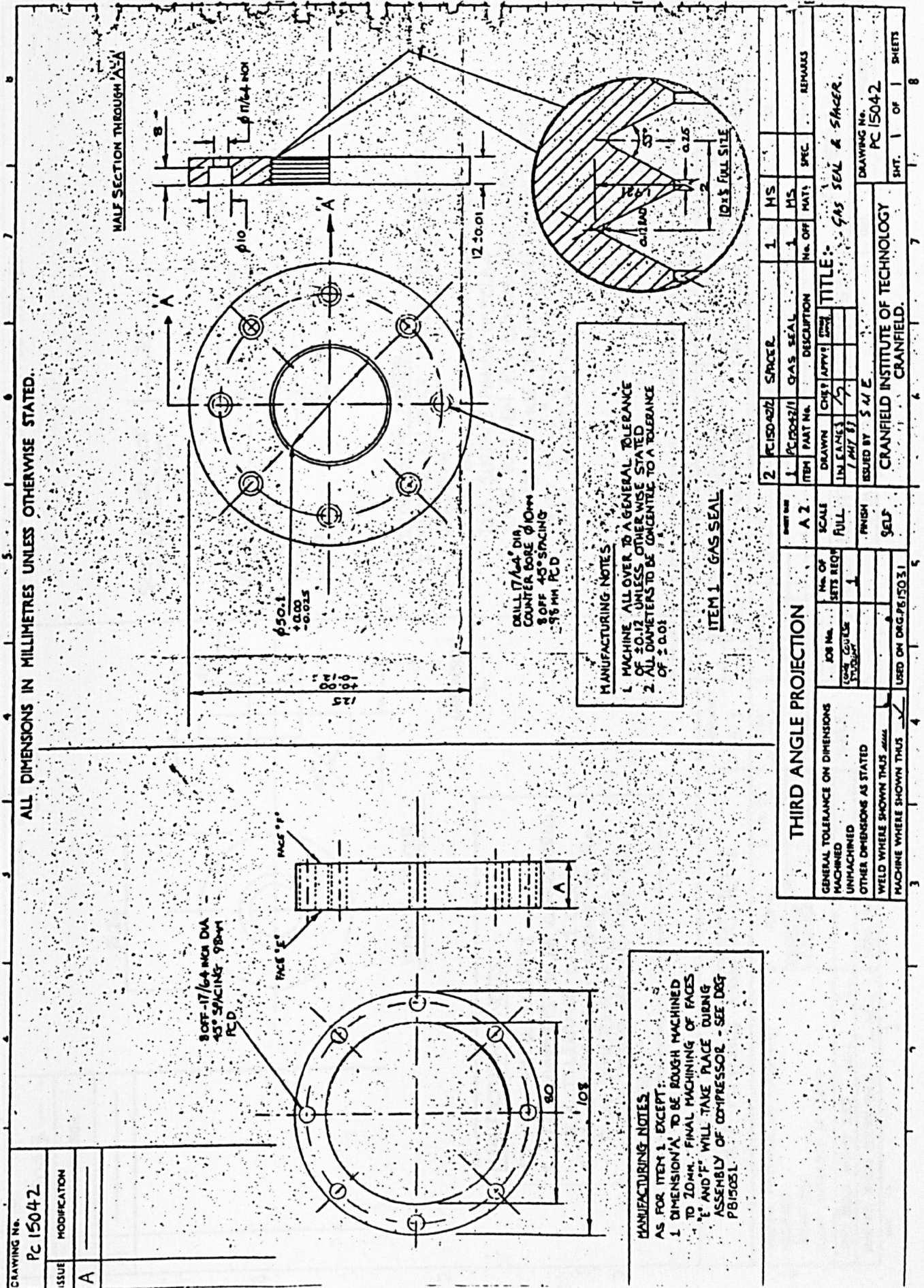
ALL DIMENSIONS IN MILLIMETRES UNLESS OTHERWISE STATED.



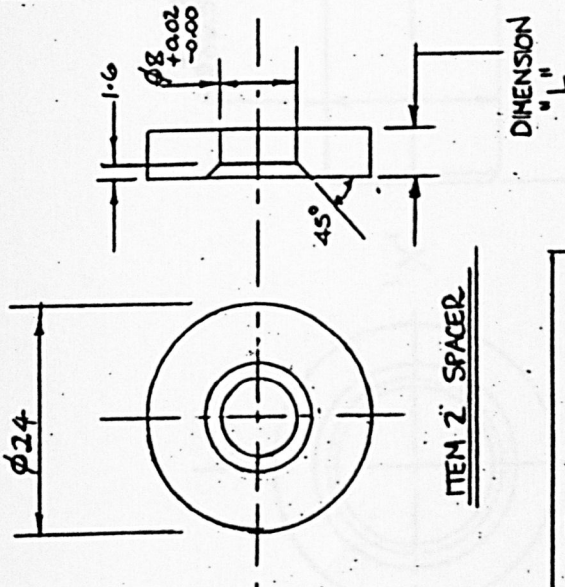
MANUFACTURING NOTES
 1 MACHINE ALL OVER TO A GENERAL TOLERANCE OF ± 0.12
 2 ALL DIAMETERS TO BE CONCENTRIC TO ± 0.05
 3 FACE 'P1' TO BE GRIND TO A SURFACE FINISH OF 8 CLA₄₀

DRAWING No. PC 1504		ISSUE A		MODIFICATION	
THIRD ANGLE PROJECTION		SHEET No. A 2		SCALE FULL	
GENERAL TOLERANCE ON DIMENSIONS MACHINED ± 0.12 UNMACHINED		JOB No. 1		N.B. OF SETS REQD. 1	
OTHER DIMENSIONS AS STATED		WELD WHERE SHOWN THIS		FINISH SELF	
MACHINE WHERE SHOWN THIS		USED ON Dwg. No. 15031		TITLE: BACK PLATE	
DESIGNER		CHECKED		APPROVED	
DRAWN		PART No.		MS	
DATE		DATE		DATE	
BY		BY		BY	
DATE		DATE		DATE	
DRAWING No. PC 1504		SHEET No. 1		TITLE: BACK PLATE	
CRANFIELD INSTITUTE OF TECHNOLOGY		CRANFIELD		CRANFIELD	

FIG. A11



DRAWING NO.	
PD 15043 --	
ISSUE	MODIFICATION
A	—



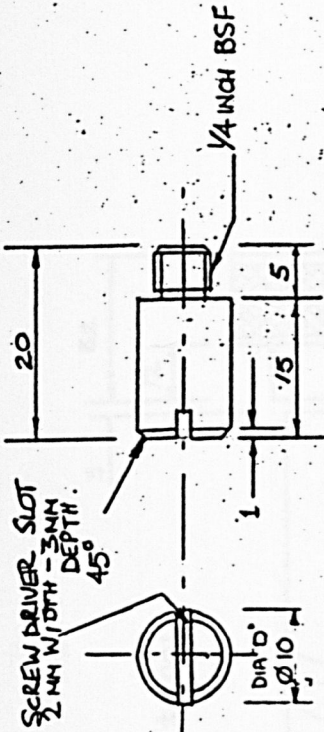
ITEM 2 SPACER

MANUFACTURING NOTES

AS FOR ITEM 1 EXCEPT:

- 1 ALL DIAMETERS TO BE CONCENTRIC
TO A TOLERANCE OF ± 0.005
2 LENGTH "L" ROUGH MACHINE TO 5MM
FINAL DIMENSION OF "L" WILL BE
FOUND DURING ASSEMBLY -SEE DRG
P815031

PB 15031



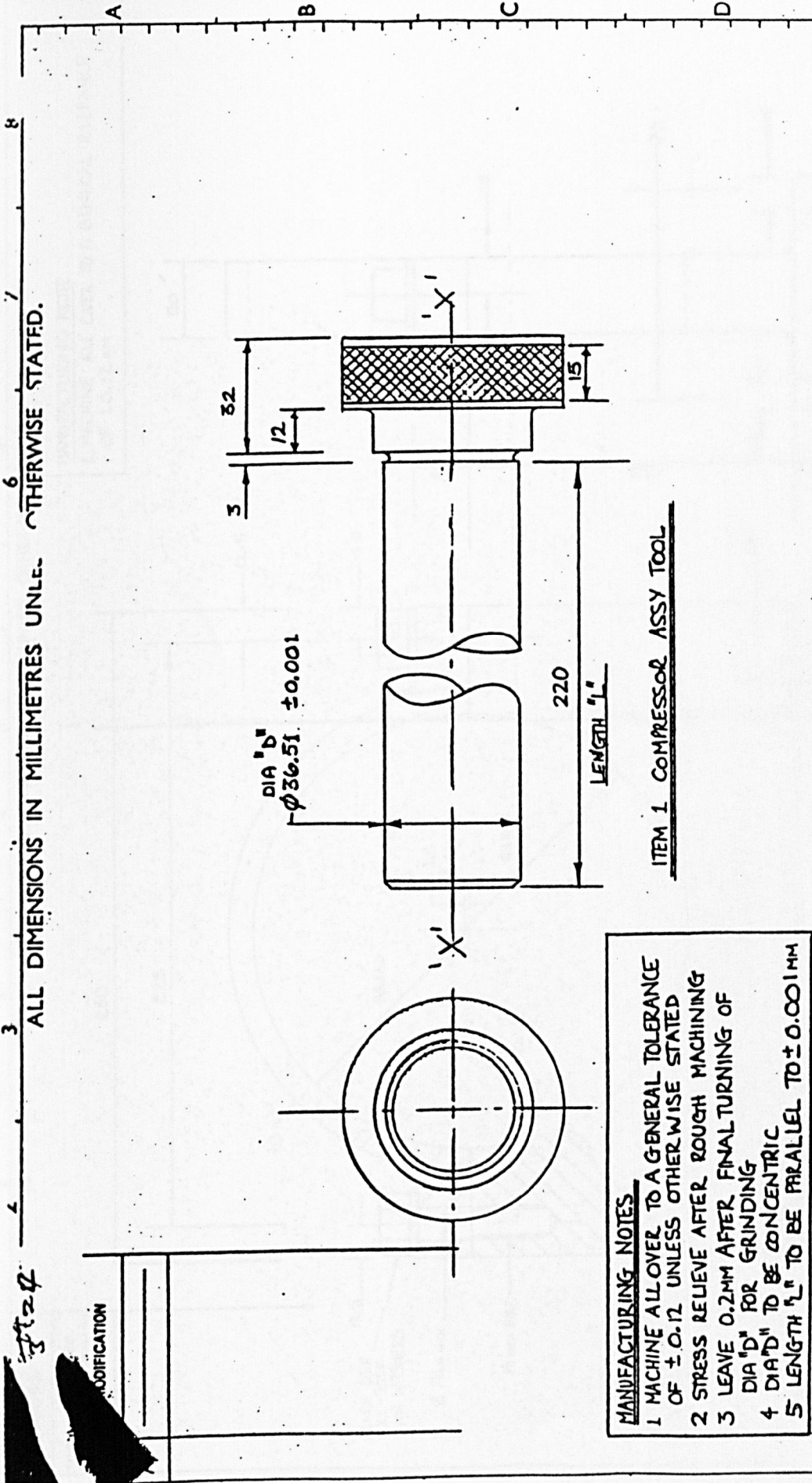
MANUFACTURING NOTES

1. MACHINE ALLOWER TO A GENERAL TOLERANCE OF ± 0.05 MM.
 2. LEAVE 0.08 MM ON DIA "D" FOR FINAL GRINDING TO A SURFACE FINISH OF 8CLA4"
 - 3 CASE HARDEN PIN BEFORE GRINDING.
 - 4 DIA "D" TO GIVE SLIGHT INTERFERENCE FIT WITH HOLE." SEE DRG PB 15031"
- ITEM 1 LOCATING PIN

THIRD ANGLE PROJECTION										SHEET SIZE A 3		2		PD150434		SPACER		1		MS											
GENERAL TOLERANCE ON DIMENSIONS MACHINED UNMACHINED OTHER DIMENSIONS AS STATED										JOB No.		NO. OF SETS REOP		SCALE		ITEM		PART No.		DESCRIPTION		No. OFF		MATI		REMARKS					
										LONG CRANFIELD		1		2 X FULL		1		DRAWN		CHKD		APPROV		STRESS APPROV							
										UNMACHINED								1 MAY 13													
WELD WHERE SHOWN THUS  MACHINE WHERE SHOWN THUS 										USED ON DRG. PD15031		FINISH		ISSUED BY		5 M1E		TITLE:-		SPACER & LOCATING PIN											
														SELF																	
										DRAWING No.		PD 150433																			
										CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD.																					
										SMT.		1		OF		1		SHEET		7		8		9							

FIG. A13

3 4 5 6 7 8
ALL DIMENSIONS IN MILLIMETRES UNLESS OTHERWISE STATED.



ITEM 1 COMPRESSOR ASSY TOOL

- MANUFACTURING NOTES**
- 1 MACHINE ALLOWED TO A GENERAL TOLERANCE OF ± 0.12 UNLESS OTHERWISE STATED
 - 2 STRESS RELIEVE AFTER ROUGH MACHINING
 - 3 LEAVE 0.2MM AFTER FINAL TURNING OF DIA "D" FOR GRINDING
 - 4 DIA "D" TO BE CONCENTRIC
 - 5 LENGTH "L" TO BE PARALLEL TO ± 0.001 MM

THIRD ANGLE PROJECTION				SHEET SIZE A 3				ITEM 1 PD15044/COMP. ASSY TOOL				L MS.				REMARKS			
GENERAL TOLERANCE ON DIMENSIONS				SCALE FULL				FINISH QLF				TITLE: Assy Tool				DRAWING No. PC 15044			
MACHINED				NO. OF SETS REQD 1				STRESS APPROV.				No. OFF MATS				SPEC.			
UNMACHINED				JOB No. LONG COURSE STUDENT				DRAWN				CHKD				APPRD			
OTHER DIMENSIONS AS STATED				USED ON DRG. PB15031				1 MAY 83				1 MAY 83				1 MAY 83			
WELD WHERE SHOWN THUS				USED ON DRG. PB15031				ISSUED BY S ATE.				CRANFIELD INSTITUTE OF TECHNOLOGY				CRANFIELD.			
MACHINE WHERE SHOWN THUS				USED ON DRG. PB15031				SHT. 1 OF 1 SHEETS				SHT. 1 OF 1 SHEETS				SHT. 1 OF 1 SHEETS			

FIG. A14

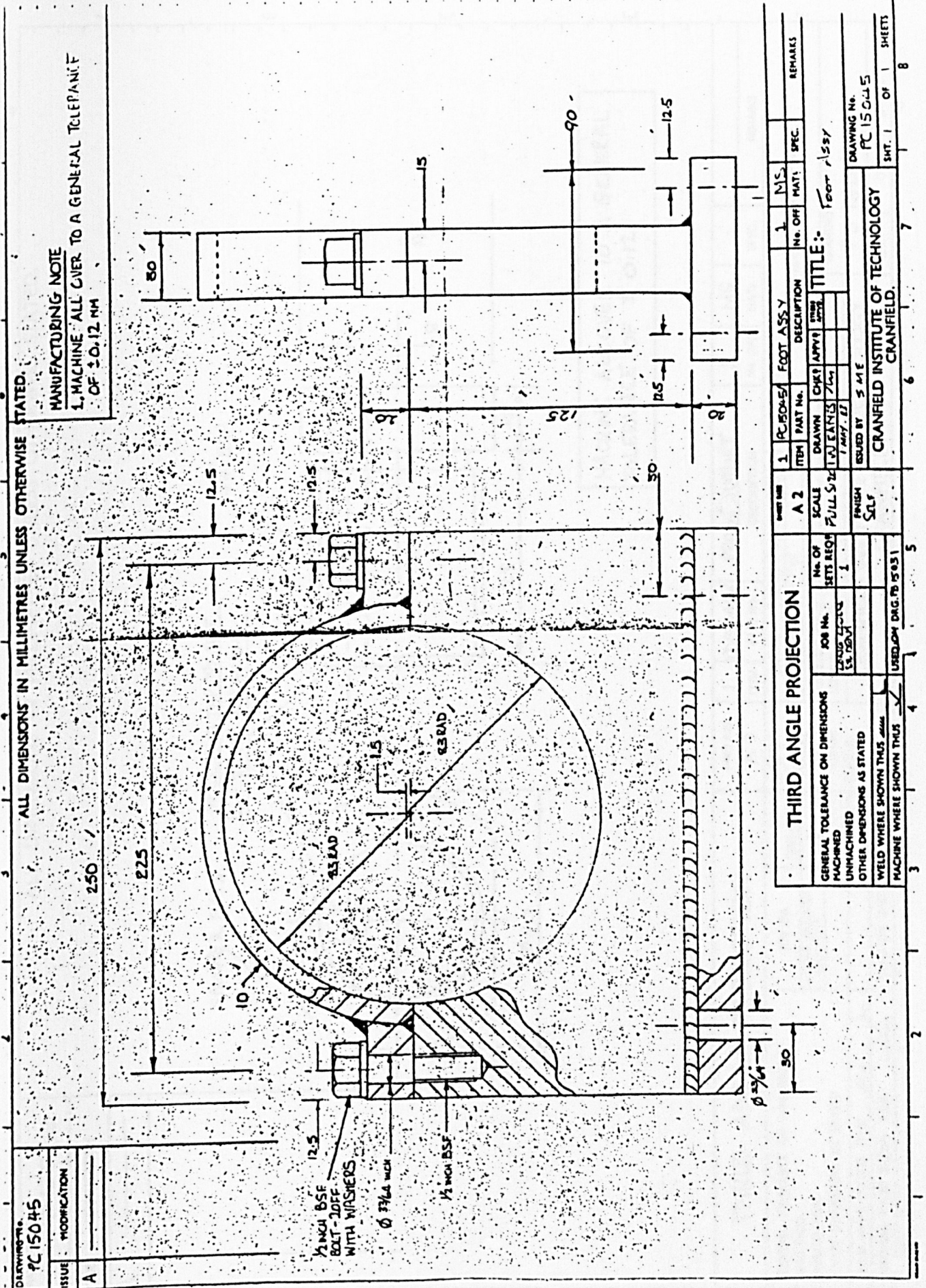
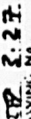


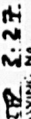
FIG. A16

3:27

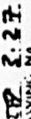
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3:27



3:27



3:27

3:27

- 3:27

3:27

APPENDIX B

COMPERF PROGRAM LISTING

```

L.  0 CLOSE#0
    122=0
    10 DIM K(40):DT12=0
    11 READ C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,C12,C13,C14,C15,C16,C17,C18,C19
    20 CLS:MODE3
    30 X=OPENIN ("HPDATA")
    40 FOR I=1 TO 39: INPUT EX,K(I):NEXT I: INPUT EX,XXX,YYY
    50 CLOSE#X
    60 PRINT SPC(200)
    70 PRINT;TAB(7); " -----"
    80 PRINT;TAB(7); "      Design Point Compressor Performance Prediction  "
    90 PRINT
    100 PRINT;TAB(7); "                      by"
    110 PRINT;TAB(7); "                      I.W.Eames  "
    120 PRINT;TAB(7); "                      1988  "
    130 PRINT
    140 PRINT;TAB(7); " -----"
    150 PRINT
    160 PRINT
    170 PRINT"GIVE DESIGN POINT DATA:"
    180 PRINT
    190 INPUT"Ambient Air Temperature";TADP1
    200 TADP2=TADP1
    210 INPUT"Evaporator refrigerant temperature ('C)";TDP1
    220 IF TDP1>TADP1 THEN PRINT"ERROR - Evaporator temperature too high - HAVE A
NOTHER GO!";GOTO 210
    230 INPUT"Degree of suctionline superheat ('C)";DSH
    240 IF DSH<0 THEN PRINT "ERROR - Super heat negative - HAVE ANOTHER GO!";GOTO
230
    250 INPUT"Condenser refrigerant temperature ('C)";TDP2
    260 IF TDP2<TADP2 THEN PRINT"ERROR - Condenser temperature too high - HAVE AN
OTHER GO!";GOTO250
    270 INPUT"Motor efficiency - eg:90 (%)";EFFMDP;EFFMDP=EFFMDP/100
    280 INPUT"Electrical supply frequency (hz)";FREQUENCY
    290 INPUT"Pressure loss in motor casing (bar)";DPMDF
    291 CLS
    300 PRINT;TAB(20,10);"Please wait - I'm thinking"
    350 READ C20,C21,C22,SDENSITY
    360 UN$="S"
    370 REM *****
    380 REM  STATE POINT ONE ROUTINES
    390 REM *****
    400 T1=TDP1*1.8+32+459.67;REM 'R
    410 TEMP=T1;GOSUB 1690;REM DETERMINE VAP.PRESS
    420 P1=VP;PRESS=P1;GOSUB 1760;REM DETERMINE V1
    430 SV1=VOL;GOSUB 1980;REM DETERMINE H1
    440 H1=ENTHALPY;GOSUB 2130
    450 S1=ENTROPY
    460 REM *****
    470 REM  STATE POINT TWO ROUTINES
    480 REM *****
    490 P2=P1-DPMDF*14.5
    500 SV2=SV1;S2=S1;H2=H1;REM FOR WHEN DSH=0 AND DELTAPRESSIN =0

```



```

510 T2=T1+DSH*1.8+DT12
520 TEMP=T2:PRESS=P2
530 IF DSH<=0 AND DPMDF<=0 THEN 580
540 GOSUB 1760:REM DET. SPECIFIC VOL
550 SV2=VOL:SVDP2=VOL:GOSUB 1980:REM ENTHALPY
560 H2=ENTHALPY:GOSUB 2130:REM ENTROP
570 S2=ENTROPY
580 REM *****
590 REM STATE POINT FOUR ROUTINES
600 REM *****
610 T4=TDP9*1.8+32+459.67:REM 'R
620 TEMP=T4:GOSUB 1690:REM DETERMINE VAP.PRESS
630 P4=VP:PRESS=P4:GOSUB 1760:REM DETERMINE V4
640 SV4=VOL:GOSUB 1980:REM DETERMINE H4
650 H4=ENTHALPY
660 GOSUB 1630
670 VROW=1/SV4:GOSUB1920
680 LAT4=LATHEAT
690 REM STATE POINT 3-2 DATA
700 T32=T2:P32=P2:REM FIRST GUESS AT T32
710 REM GUESS T32
720 TEMP=T32:PRESS=P32:GOSUB 1790
730 SV32=VOL:GOSUB 2000
740 H32=ENTHALPY:GOSUB 2130
750 S32=ENTROPY
760 REM *****
770 REM STATE POINT THREE ROUTINES
780 REM *****
790 P3=P4
800 T3=T2*(P3/P2)^(.118):REM A FIRST APPROXIMATION
810 TEMP=T3:PRESS=P3:GOSUB 1790
820 SV3=VOL
830 GOSUB 2130
840 S3=ENTROPY
850 REM ISENTROPIC COMPRESSION CALCULATION FOLLOWS
860 IF S3<S32-ER OR S3>S32+ER THEN T3=T3*S32/S3:GOTO 810
870 TEMP=T3:PRESS=P3:VOL=SV3:GOSUB2000
880 H3=ENTHALPY
881 ZZ=ZZ+1:PRINT;TAB(18+ZZ,12);" * ":IF ZZ>20 THEN ZZ=0
890 REM *****
900 REM COMPRESSOR CHARACTERISTIC
910 REM *****
920 TINC=T2:PRC=P3/P32:CN=0.96*FREQUENCY*60
930 GOSUB 2240
940 REM *****
950 REM REAL CYCLE PERFORMANCE
960 REM *****
970 FLOW=FLOW*(P2/14.5)/THETA/60
980 MASSR=FLOW*0.3629
990 COMPWORK=(H3-H2)/EFF
1000 WCDP=MASSR*(H3-H32)/EFF
1010 CH3=H2+COMPWORK
1020 QCDP=MASSR*(LAT4+CH3-H4)
1030 QEDP=QCDP-WCDP

```

```

1040 H32=H2+WCDP/MASSR*(1/EFFMDP-1)
1050 GOSUB 2830
1060 IF ABS(T31-T32)>1E-2 THEN T32=(T31+T32)/2:GOTO 710
1070 T1=T1-459.67:T2=T2-459.67:T3=T3-459.67:T4=T4-459.67:T32=T32-459.67
1080 GOSUB 1480
1085 CLS
1090 PRINT"CALCULATION IS NOW FINISHED - DO YOU WANT A HARD COPY Y/N"
1091 PRINT"PLEASE SWITCH PRINTER ON IF HARD COPY IS REQUIRED"
1100 IF GET$="Y" THEN VDU2 ELSE VDU3
1110 MODE 3
1120 PRINT
1130 PRINT " Design Point Performance Data for Experimental Compressor"
1140 PRINT " -----"
1160 PRINT " Rotational Speed ";TAB(40);CN
1170 PRINT " Evaporator Temp."TAB(40);T1" 'C"
1180 PRINT " Evaporator Press."TAB(40);INT(10000*P1)/10000" bar"
1190 PRINT " Comp. Inlet Press."TAB(40);INT(P2*10000)/10000" bar"
1200 PRINT " Comp Inlet Temp."TAB(40);INT(T32*100)/100" 'C"
1210 PRINT " Degrees of Superheat";TAB(40);DSH" 'C"
1220 PRINT " Condenser Temp."TAB(40);INT(1000*T4)/1000" 'C"
1230 PRINT " Condenser Press."TAB(40);INT(1000*P4)/1000" bar"
1240 PRINT " Compressor Efficiency";TAB(40);INT(10000*EFF)/100" %"
1250 PRINT " Mass Flow Rate";TAB(40);INT(10000*FLOW/SV2)/10" gms/s"
1260 PRINT " Compressor Pressure Ratio";TAB(40);INT(1000*PRC)/1000
1270 PRINT " Power Output";TAB(40);INT(1000*WCDP)/1000" kw"
1280 PRINT " Refrigeration Effect";TAB(40);INT(1000*QEDP)/1000" kw"
1290 PRINT " C.O.P.";TAB(40);INT(1000*QEDP/WCDP)/1000
1300 PRINT " Motor Efficiency";TAB(40);EFFMDP
1310 PRINT " Electrical Supply Frequency";TAB(40);FREQUENCY;" hz"
1320 PRINT " Coolant Heating";TAB(40);INT(MASSR*(H32-H2)*1000)/1000;" kw"
1330 PRINT " Power Supply to Motor";TAB(40);WCDP/EFFMDP;" kw"
1331 PRINT " Ambient Air Temperature";TAB(40);TADP1" 'C"
1340 VDU3
1350 PRINT" Another? Y/N"
1360 PRINT" *****"
1370 IF GET$="" THEN 1370
1380 IF GET$="Y" THEN 170
1390 PRINT" Do you want to carryout part load analysis on "
1400 PRINT" the present design point data? Y/N !"
1410 IF GET$="" THEN 1410
1420 IF GET$="Y" THEN 1430 ELSE 1470
1430 X=OPENOUT ("DPDATA")
1440 PRINTX,WCDP,QEDP,QCDP,TADP1,TADP2,TDP1,TDP9,EFFMDP,FREQUENCY,DPMDP,MASSR,
SV32,DSH
1450 CLOSEX
1460 CHAIN "COMP-OD"
1470 END
1480 REM*****
1490 REM CONVERT UNITS
1500 REM*****
1510 T1=(T1-32)/1.8:T2=(T2-32)/1.8:T3=(T3-32)/1.8:T4=(T4-32)/1.8:T32=(T32-32)/1

```

```

1520 P1=P1/14.5:P2=P2/14.5:P3=P3/14.5:P4=P4/14.5:P32=P32/14.5
1530 H1=H1*2.326:H2=H2*2.326:H3=H3*2.326:H4=H4*2.326:H32=H32*2.326
1540 S1=S1/.2388:S2=S2/.2388:S3=S3/.2388:S32=S32/.2388
1550 LAT4=LAT4/.4299
1560 SV1=SV1*0.06243:SV2=SV2*0.06243:SV3=SV3*0.06243:SV32=SV32*0.06243
1570 FLOW=FLOW/35.31
1580 WCDP=WCDP*3600/3412
1590 QCDP=QCDP*3600/3412
1600 MASSR=MASSR/2.205
1610 SVDP2=SVDP2/16.02
1620 RETURN
1630 REM*****
1640 REM SUB-ROUTINE LIQUID DENSITY
1650 REM*****
1660 FROW=(1-TEMP/K(8)):FROW=K(1)+K(2)*FROW^(1/3)+K(3)*FROW^(2/3)+K(4)*FROW+K(5)
) *FROW^(4/3)
1670 RETURN
1680 REM*****
1690 REM SUB-ROUTINE VAP-PRESS
1700 REM*****
1710 VP=K(9)+K(10)/TEMP+K(11)*LOG(TEMP)+K(12)*TEMP
1720 VP=VP+K(13)*((K(14)-TEMP)/TEMP)*LOG(K(14)-TEMP)
1730 VP=10^VP
1740 RETURN
1750 REM*****
1760 REM SUB-ROUTINE EQUATION OF STATE
1770 REM CALC.SPECIFIC VOL GIVE VP AND TEMP
1780 REM*****
1790 B=EXP(-K(32)*TEMP/K(8))
1800 D=K(15)*TEMP/PRESS:E=(K(17)+K(18)*TEMP+K(19)*B)/PRESS
1810 F=(K(20)+K(21)*TEMP+K(22)*B)/PRESS:G=(K(23)+K(24)*TEMP+K(25)*B)/PRESS
1820 H=(K(26)+K(27)*TEMP+K(28)*B)/PRESS:I=(K(29)+K(30)*TEMP+K(31)*B)/PRESS
1830 J=D+K(16):ER=1E-5;V0=J
1840 A=V0-K(16)
1850 C=1.65E38:REM SETS UPPER LIMIT FOR C
1860 IF K(33)*V0<88 THEN C=EXP(K(33)*V0)
1870 V1=J+E/A+F/A^2+G/A^3+H/A^4+I*A/(C*(1+K(34)*C))
1880 IF V1<V0-ER OR V1>V0+ER THEN V0=(V1+V0)/2:GOTO 1840
1890 VOL=V1:A=0:B=0:C=0:V0=0:V1=0:D=0:E=0:F=0:H=0:I=0:J=0
1900 RETURN
1910 REM*****
1920 REM SUB-ROUTINE LATENT HEAT
1930 REM*****
1940 A=VP*LN(10)*((-K(10)/TEMP^2)+K(11)/(TEMP*LN(10))+K(12)-K(13)*(LOG(EXP(1))
TEMP+K(14)*LOG(K(14)-TEMP)/TEMP^2))
1950 LATHEAT=.185053*TEMP*(1/VROW-1/FROW)*A
1960 RETURN
1970 REM*****
1980 REM SUB-ROUTINE ENTHALPY
1990 REM*****
2000 A=VOL-K(16):B=EXP(-K(32)*TEMP/K(8)):C=1.65E38:REM SETS UPPER LIMIT FOR C

```

```

2010 IF K(33)*VOL<88 THEN C=EXP(K(33)*VOL)
2020 D=K(17)/A+K(20)/(2*A^2)+K(23)/(3*A^3)+K(26)/(4*A^4)
2030 D=.185053*D
2040 E=K(19)/A+K(22)/(2*A^2)+K(25)/(3*A^3)+K(28)/(4*A^4)
2050 E=E+.185053*B*(1+K(32)*TEMP/K(8))
2060 F=K(35)*TEMP+K(36)*TEMP^2/2+K(37)*TEMP^3/3+K(38)*TEMP^4/4-K(39)/TEMP
2070 ENTHALPY=F+.185053*PRESS*VOL+D+E+XXX
2080 A=0:B=0:C=0:D=0:E=0:F=0
2090 RETURN
2100 REM*****
2110 REM SUB-ROUTINE ENTROPY
2120 REM*****
2130 A=VOL-K(16):B=EXP(-K(32)*TEMP/K(8)):C=1.65E38:REM SETS UPPER LIMIT FOR C
2140 IF K(33)*VOL<88 THEN C=EXP(K(33)*VOL)
2150 D=K(18)/A+K(21)/(2*A^2)+K(24)/(3*A^3)+K(27)/(4*A^4)
2160 D=D+K(30)/(VOL*K(33)^2)
2170 D=.185053*D
2180 E=K(19)/A+K(22)/(2*A^2)+K(25)/(3*A^3)+K(28)/(4*A^4)
2190 E=.185053*E*B*K(32)/K(8)
2200 F=K(35)*LN(10)*LOG(TEMP)+K(36)*TEMP+K(37)*(TEMP^2)/2+K(38)*(TEMP^3)/3
2210 F=F-K(39)/(2*TEMP^2)+.185053*K(15)*LN(10)*LOG(A)
2220 ENTROPY=F+E+YYY-D
2230 RETURN
2240 REM *****
2250 REM *      CENTRIFUGAL COMPRESSOR MODEL      *
2260 REM Input data required for this subroutine are
2270 REM inlet pressure (psia),inlet temperature ('R)
2280 REM making the units compatable with the refriger
2290 REM -ant data routines.
2300 REM Also required is design rotational speed in
2310 REM RPM.
2320 ER$="COVERGED"
2330 REM Calc. corrected speed
2340 THETA=SQR(TINC/(288*1.8))
2350 CORN=CN/THETA
2360 REM Now refer this to 66000rpm corrected
2370 NREF1=CORN/66E3
2380 NREF2=CORN/60E3
2390 REM Calc pressure ratio and flow scale factors
2400 PRSF=C1+C2*NREF1+C3*NREF1^2
2410 QSF1=C4+C5*NREF1+C6*NREF1^2
2420 EFSF=C17+C18*NREF2+C19*NREF2^2
2430 REM Calc corrected pressure ratio
2440 CPR=PRC/PRSF
2450 REM Calc corrected flow iteratively
2460 DIF=.0001
2470 CQ=170/35.31:DQ=1E-3
2480 CQ1=CQ:CQ2=CQ1+DQ
2490 GOSUB 2730
2500 CP1=RP:CQ=CQ2:GOSUB 2730
2510 CP2=RP:DQDF=(CQ2-CQ1)/(CP2-CP1):CQ3=CQ1+DQDF*(CPR-CP1):CQ=CQ3:GOSUB 2730
2520 CP1=RP:CQ2=CQ1:CQ1=CQ3

```

```

2530 SUM=SUM+1
2540 IF SUM>100 THEN ER$="DID NOT CONVERGE":GOTO 2570
2550 IF ABS(CP1-CPR)<=DIF THEN 2570
2560 GOTO 2480
2570 REM CAL EFFICENCY REFERRED TO 60000 AS A FUNCTION OF FLOW
2580 R=C14+C15*CQ1+C16*CQ1^2
2590 EFF=R*EFSF
2600 REM Calc True corrected flow rate
2610 FLOW=CQ1*QSF1
2620 REM CHECK SURGE MARGIN
2630 SURGEPRESS=C20+C21*FLOW*35.31
2640 WARN$=" "
2650 IF SURGEPRESS<PRC THEN WARN$=" WARNING ** SURGE ** WARNING "
2660 REM CHECK CHOKE MARGIN
2670 CHOKEPRESS=C22+C21*FLOW*35.31
2680 IF CHOKEPRESS>PRC THEN WARN$=" WARNING ** CHOKED ** WARNING"
2690 FLOW=FLOW*35.31
2700 PRINT;TAB(20,20);WARN$
2710 IF ER$="DID NOT CONVERGE" THEN PRINT;TAB(20,21);"CALCULATION "ER$:GOTO2950
2720 SUM=0
2730 IF CQ<=175/35.31 THEN 2750
2740 RP=C7+C8*CQ+C9*CQ^2+C10*CQ^3:GOTO2760
2750 RP=C11+C12*CQ+C13*CQ^2
2760 RETURN
2770 DATA -8.38000068,16.1980014,-6.46800069,2.51001197,-3.52102412
2780 DATA 2.04601216,917.569251,-571.015321,119.099585,-8.28827368
2790 DATA -8.41999655,5.94384842,-.69266432,-.844600214,.680813351
2800 DATA -8.09411425E-2,-7.65905766,17.2909108,-8.54254212
2810 DATA -1.80571429,4.37142857E-2,-4.86571429,.271276384
2820 REM *****
2830 REM CALC THERMO PROPERTIES OF STATE 5 REPEATED GUESS
2840 REM *****
2850 PRESS=P32:HB=H1:TB=T1:TA=0:HA=0
2860 TEMP=H32*TB/HB:GOTO 2880:REM FIRST GUESS
2870 TEMP=(TB-TA)*(H32-HB)/(HB-HA)+TB
2880 GOSUB 1760
2890 GOSUB 1980
2900 IF ABS(ENTHALPY-H32)<1E-2 THEN T31=TEMP:SV32=VOL:GOTO 2920
2910 TA=TB:TB=TEMP:HA=HB:HB=ENTHALPY:GOTO 2870
2920 GOSUB 2110
2930 S3=ENTROPY:H3=ENTHALPY:SV3=VOL:T31=TEMP
2940 RETURN
2950 FOR I=1 TO 1000:NEXT I
3000 CLS
3010 PRINT" PLEASE BEGIN AGAIN BY TYPING RUN AND PRESSING THE RETURN KEY. THE (
CONDENSER OR EVAPORATOR TEMPERATURE IS OUT OF RANGE":END
>

```

APPENDIX C **COMP-OD PROGRAM LISTING**

Table C1 COMP-OD Subroutines

<u>Subroutine</u>	<u>Line Number</u>
State Point 1	1060
State Point 2	1150
State Point 3	1280
State Point 4	1470
State Point 5	1690
State Point 6	1910
State Point 9	1800
Compressor Characteristic	2090
Output Data	2200
Liquid Density	2660
Vapour Pressure	2730
Equation of State	2820
Latent Heat	2980
Enthalpy	3070
Entropy	3220
Centrifugal Compressor Model	3370
Electric Motor	3980
Suction Line Throttle	4180
Evaporator Temperature	4290
Condenser Temperature	4400
Iterative Method - Properties State Point 5	4510
Hot Gas Bypass	4640
Suction Line Throttle	4730
VDU Data	4780

Table C2 COMP-OD Program Listing

```

>L.
10 *SHADOW
20CLS:MODE3
30 PRINT SPC(200)
40 DIM K(40)
50 PRINT "-----"
60 PRINT " Off-design performance prediction of"
70 PRINT " an experimental refrigeration compressor"
80 PRINT "-----"
90 X=OPENIN ("HPDATA")
100 FOR I=1TO39:INPUTEX,K(I):NEXT I:INPUTEX,XXX,YYY
110 CLOSEEX
120 X=OPENIN ("DPDATA")
130 INPUTEX,WCDP,QEDP,QCDP,TADP1,TADP2,TDP1,TDP9,EFFMDP,FREQUENCY,DFMDP,MR,SVD
P2,DSH
140 MRDP=MR
150 CLOSEEX
160 REM *****
170 REM COMPRESSOR CHARACTERISTIC CONSTANTS
180 REM *****
190 READ C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,C12,C13,C14,C15,C16,C17,C18,C19
200 READ C20,C21,C22,SDENSITY
210 REM *****
220 REM          CONTROL PROGRAM
230 REM *****
240 INPUT"Ambient Temperature Range - Lowest temperature first,('C)";TR1,TR2
250 INPUT"Step value through this range";TR3
260 FOR TAOD1=TR1 TO TR2 STEP TR3
270 TAOD2=TAOD1
280 T11=0:T91=0:T51=0:P51=0:PHI=0:DMR=0:TICK=0:DPSLT=0:TUCK=0:CLICK=0
290 T9=TDP9-TADP2+TAOD2
300 T1=(T9-49.091)/1.3636:REM FIRST GUESS AT EVAOPOR TEMP
310 GOSUB 1060:REM STATE POINT ONE DATA
320 T5=T1+15:REM FIRST GUESS
330 P5=P1:REM FIRST GUESS
340 PHI2=0.9417:REM FIRST GUESS
350 SLTHROTTLE$="OFF":HGBYPASS$="OFF"
360 GOSUB 4780
370 PRINT,TAB(15,10);ABS(T1*1E4)/1E4
380 PRINT,TAB(15,3);ABS(T9*1E4)/1E4
390 PRINT,TAB(54,10);ABS(T5*1E4)/1E4
400 PRINT,TAB(15,12);ABS(P1*1E4)/1E4
410 PRINT,TAB(54,12);ABS(P5*1E4)/1E4
420 PRINT,TAB(15,19);0.9417
430 PRINT,TAB(20,23);"      "
440 PRINT,TAB(20,23);TAOD1
450 REM GUESS T9
460 PRINTTAB(15,3);"      "
470 PRINTTAB(15,3);INT(ABS(T9*1E4))/1E4
480 PRINTTAB(15,4);"      "
490 PRINTTAB(15,4);INT(ABS((T9-T91)*1E5))/1E5

```

```

500 GOSUB 1800:REM DETERMINE STATE POINT 9 DATA
510 REM GUESS T1
520 PRINTTAB(15,10);"
530 PRINTTAB(15,10);INT(ABS(T1*1E4))/1E4
540 PRINTTAB(15,11);"
550 PRINTTAB(15,11);INT(ABS((T1-T11)*1E5))/1E5
560 PRINTTAB(15,12);"
570 PRINTTAB(15,12);INT(ABS(P1*1E4))/1E4
580 GOSUB 1060:REM STATE POINT ONE DATA
590 GOSUB 1150:REM STATE POINT TWO DATA
600 REM GUESS T5
610 PRINTTAB(54,11);"
620 PRINTTAB(54,11);INT(ABS((T5-T51)*1E4))/1E4
630 PRINTTAB(54,10);"
640 PRINTTAB(54,10);INT(ABS(T5*1E3))/1E3
650 REM GUESS P5
660 PRINTTAB(54,13);"
670 PRINTTAB(54,13);INT(ABS((P5-P51)*1E4))/1E4
680 PRINTTAB(54,12);"
690 PRINTTAB(54,12);INT(ABS(P5*1E4))/1E4
700 GOSUB 1690:REM STATE POINT 5 DATA
710 GOSUB 1910:REM STATE POINT 6 DATA
720 REM GUESS PHI2
730PRINTTAB(15,20)"
740PRINTTAB(15,20);INT(ABS((PHI2-PHI)*1E5))/1E5
750PRINTTAB(15,19)"
760PRINTTAB(15,19);INT(ABS(PHI2*1E4))/1E4
770 GOSUB 2090:REM COMPRESSOR DATA
780 PRINT,TAB(54,3);"
790 PRINT,TAB(54,3);INT(ABS(WCOD*1E4))/1E4
800 PRINT,TAB(54,4);"
810 PRINT,TAB(54,4);INT(ABS(MR*1E4))/1E4
820 PRINT,TAB(38,7);"
830 PRINT,TAB(38,7);INT(ABS(DMR*MR*1E4))/1E4
840 IF TICK=1 THEN GOSUB 4640:REM HOT-GAS BYPASS
850 IF TICK=1 THEN TICK=0:GOTO 1000
860 IF TUCK=1 THEN GOSUB 4730
870 IF TUCK=1 THEN TUCK=0:GOTO 920
880 PRINT,TAB(57,19);DPSLT
890 GOSUB 3980:REM CALC PHI
900 IF ABS(PHI2-PHI)>1E-2 THEN PHI2=(PHI2+PHI)/2:GOTO 720
910 GOSUB 4180:REM CALC PRESS LOSS IN MOTOR
920 IF ABS(P5-P51)>1E-3 THEN P5=(P5+P51)/2:GOTO 650
930 GOSUB 4140:REM CALC MOTOR EFF
940 GOSUB 4240:REM CALC H5
950 GOSUB 4510: REM CALC T5
960 IF ABS(T5-T51)>1E-1 THEN T5=(T5+T51)/2:GOTO 600
970 GOSUB 4290:REM CALC NEXT GUESS OF T1
980 IF ABS(T1-T11)>1E-1 THEN T1=(T11+T1)/2:GOTO 510
990 GOSUB 4440:REM CALC CONDENSER TEMPERATURE GUESS
1000 IF ABS(T9-T91)>1E-1 THEN T9=(T9+T91)/2:GOTO 450

```



```

1010 IF WARN$=" WARNING ** SURGE ** WARNING ":DMR=DMR+0.05: GOTO 990
1015 IF OVERPOWER=1 THEN GOTO 5035
1020 GOSUB 2220:REM OUTPUT DATA
1030 NEXT TAOD1
1040 STOP
1050 REM *****
1060 REM SUB-ROUTINE STATE POINT ONE DATA
1070 REM *****
1080 TEMP=T1:GOSUB 2730:REM DETERMINE VAP.PRESS
1090 P1=VP:PRESS=P1:GOSUB 2820:REM DETERMINE V1
1100 SV1=VOL:GOSUB 3070:REM DETERMINE H1
1110 H1=ENTHALPY:GOSUB 3220
1120 S1=ENTROPY
1130 RETURN
1140 REM *****
1150 REM SUB-ROUTINE STATE POINT TWO DATA
1160 REM *****
1170 P2=P1
1180 SV2=SV1:S2=S1:H2=H1:REM FOR WHEN DSH=0
1190 T2=T1+DSH
1200 TEMP=T2:PRESS=P2
1210 IF DSH<=0 THEN GOTO 1270
1220 GOSUB 2820:REM DET. SPECIFIC VOL
1230 SV2=VOL:GOSUB 3070:REM ENTHALPY
1240 H2=ENTHALPY:GOSUB 3220
1250 S2=ENTROPY
1260 RETURN
1270 REM *****
1280 REM SUB-ROUTINE STATE POINT THREE DATA
1290 REM ELECTRIC MOTOR CHARACTERISTICS
1300 REM *****
1310 IF START=0 THEN WCOD=WCOP
1320 GOSUB 3980:REM MOTOR CHARACTERISTIC
1330 H3=H2+DH23
1340 P3=P2-DP23
1350 REM CALC THERMO PROPERTIES AT STATE POINT THREE
1360 PRESS=P3:HB=H2:TB=T2+273:TA=0:HA=0
1370 TEMP=H3*TB/HB:TEMP=TEMP-273:TB=TB-273:PRINT"970 TEMP AND TB ="TEMP" "TB:G
1380 TEMP=(TB-TA)*(H3-HB)/(HB-HA)+TB
1390 GOSUB 2820
1400 GOSUB 3070
1410 IF ABS(ENTHALPY-H3)<1E-2 THEN T3=TEMP:SV3=VOL:GOTO 1430
1420 TA=TB:TB=TEMP:HA=HB:HB=ENTHALPY:GOTO 1380
1430 TA=0:TB=0:TEMP=T3:HB=0:HA=0:GOSUB 3220
1440 S3=ENTROPY
1450 RETURN
1460 REM *****
1470 REM SUB-ROUTINE STATE POINT FOUR DATA
1480 REM SUCTION LINE THROTTLE

```

```

1490 REM *****
1500 H4=H3
1510 IF SLTHROTTLE$="ON" THEN DP34=DP34 ELSE DP34=0
1520 P4=P3-DP34
1530 IF START=0 THEN 1660
1540 IF SLTHROTTLE$="OFF" THEN 1660
1550 REM DETERMINE THERMO PROPERTIES
1560 PRESS=P4:HB=H3:TB=T3+273:TA=0:HA=0
1570 TEMP=TB*P4/P3:TEMP=TEMP-273:GOTO 1590:REM FIRST GUESS
1580 TEMP=TB+(TB-TA)*(H3-HB)/(HB-HA)
1590 GOSUB 2820
1600 GOSUB 3070
1610 IF ABS(ENTHALPY-H4)<1E-2 THEN T4=TEMP:SV4=VOL:GOTO 1630
1620 TA=TB:TB=TEMP:HA=HB:HB=ENTHALPY:GOTO 1580
1630 GOSUB 3220
1640 S4=ENTROPY
1650 GOTO 1680
1660 S4=S3:T4=T3:SV4=SV3
1670 RETURN
1680 REM *****
1690 REM SUB-ROUTINE STATE POINT FIVE DATA
1700 REM *****
1710 TEMP=T5:PRESS=P5
1720 GOSUB 2820
1730 SV5=VOL
1740 GOSUB 3070
1750 H5=ENTHALPY
1760 GOSUB 3220
1770 S5=ENTROPY
1780 RETURN
1790 REM *****
1800 REM SUB-ROUTINE STATE POINT NINE DATA
1810 REM *****
1820 TEMP=T9:GOSUB 2730:REM DETERMINE VAP.PRESS
1830 P9=VP:PRESS=P9:GOSUB 2820:REM DETERMINE V9
1840 SV9=VOL:GOSUB 3070:REM DETERMINE H9
1850 H9=ENTHALPY
1860 GOSUB 2660
1870 VROW=1/SV9:GOSUB 2980
1880 LAT9=LATHEAT
1890 RETURN
1900 REM *****
1910 REM SUB-ROUTINE STATE POINT SIX DATA
1920 REM *****
1930 P6=P9
1940 S62=0:S61=0
1950 T5=T5+273
1960 T6=T5*(P6/P5)^(.113):REM A FIRST APPROXIMATION
1970 T5=T5-273:T6=T6-273:T62=T6:T61=0
1980 TEMP=T6:PRESS=P6:GOSUB 2820
1990 SV3=VOL
2000 GOSUB 3220

```

```

2010 S6=ENTROPY
2020 REM ISENTROPIC COMPRESSION CALCULATION FOLLOWS
2030 S62=S6
2040 IF S6<S5-ER OR S6>S5+ER THEN T6=T62+(S5-S62)/(S62-S61)*(T62-T61):S61=S62:T
61=T62:T62=T6:GOTO 1980
2050 GOSUB 3070
2060 H6=ENTHALPY:SV6=VOL
2070 RETURN
2080 REM *****
2090 REM SUB-ROUTINE COMPRESSOR CHARACTERISTIC DATA
2100 REM *****
2110 TINC=T5:PRC=P6/P5:CN=PHI2*FREQUENCY*60
2120 GOSUB 3370
2130 FLOW=FLOW*P2/THETA/60
2140 MR=FLOW/SV5
2150 COMPWORK=(H6-H5)/EFF
2160 H61=COMPWORK+H5
2170 WCOD=MR*(H61-H5)
2180 RETURN
2190 REM *****
2200 REM SUB-ROUTINE OUTPUT DATA
2210 REM *****
2220 VDU2
2230 MODE 3
2240 PRINT
2250 PRINT " PART-LOAD PERFORMANCE DATA"
2260 PRINT " -----"
2270 PRINT " PART-LOAD DATA"
2280 PRINT " -----"
2290 PRINT
2300 PRINT " Evaporator Air-On Temperature";TAB(40);TAOD1;" °C"
2310 PRINT " Condenser Air-On Temperature";TAB(40);TAOD2;" °C"
2320 PRINT " COMPRESSOR DATA"
2330 PRINT " -----"
2340 PRINT " Rotational Speed ";TAB(40);ABS(CN)
2350 PRINT " Comp. Inlet Press."TAB(40);INT(P5*10000)/10000" bar"
2360 PRINT " Comp Inlet Temp."TAB(40);INT(T5*100)/100" °C"
2370 PRINT " Compressor Efficiency";TAB(40);INT(10000*EFF)/100" %"
2380 PRINT " Compressor Mass Flow Rate";TAB(40);INT(10000*MR)/10" gms/s"
2390 PRINT " Compressor Pressure Ratio";TAB(40);INT(1000*P9/P5)/1000
2400 PRINT " Compressor Power Output";TAB(40);INT(1000*WCOD)/1000" kw"
2410 PRINT
2420 PRINT " MOTOR DATA"
2430 PRINT " -----"
2440 PRINT
2450 PRINT " Motor Efficiency";TAB(40);INT(1000*EFFMOD)/1000
2460 PRINT " Electrical power supply";TAB(40);INT(1000*WCOD/EFFMOD)/1000;" kw"
2470 PRINT " Slip Factor";TAB(40);PHI2
2480 PRINT " Motor Casing Pressure Loss";TAB(40);INT(1E6*DP23)/1E6" bar"
2490 PRINT
2500 PRINT " SYSTEM DATA"

```

```

2510 PRINT " -----"
2520 PRINT " Condenser Heat Rejection";TAB(40);INT(1000*QCOD1)/1000" kw"
2530 PRINT " Refrigeration Effect";TAB(40);INT(1000*QEOD1)/1000" kw"
2540 PRINT " C.O.P. (Cooling)";TAB(40);INT(1000*QEOD1*EFFMOD/WCOD)/1000" kw"
2550 PRINT " Suction Line Throttle P-D";TAB(40);DPSLT;" bar"
2560 PRINT " Hot-Gas Bypass Percentage";TAB(40);DMR
2570 PRINT " Hot-Gas Bypass Mass Flow";TAB(40);INT(1E7*DMR*MR)/1E4" gms"
2580 PRINT " Evaporator Temp."TAB(40);T1" °C"
2590 PRINT " Evaporator Press."TAB(40);INT(10000*P1)/10000" bar"
2600 PRINT " Degrees of Superheat";TAB(40);DSH" °C"
2610 PRINT " Condenser Temp."TAB(40);INT(1000*T9)/1000" °C"
2620 PRINT " Condenser Press."TAB(40);INT(1000*P9)/1000" bar"
2630 VDU3
2640 RETURN
2650 REM*****
2660 REM SUB-ROUTINE LIQUID DENSITY
2670 REM*****
2680 TEMP=TEMP*1.8+491.67
2690 FROW=(1-TEMP/K(8)):FROW=K(1)+K(2)*FROW^(1/3)+K(3)*FROW^(2/3)+K(4)*FROW+K(5)
) *FROW^(4/3)
2700 TEMP=(TEMP-491.67)/1.8:FROW=FROW/0.06243
2710 RETURN
2720 REM*****
2730 REM SUB-ROUTINE VAP-PRESS
2740 REM*****
2750 TEMP=TEMP*1.8+491.67
2760 VP=K(9)+K(10)/TEMP+K(11)*LOG(TEMP)+K(12)*TEMP
2770 VP=VP+K(13)*((K(14)-TEMP)/TEMP)*LOG(K(14)-TEMP)
2780 VP=10^VP
2790 TEMP=(TEMP-491.67)/1.8:VP=VP/14.5
2800 RETURN
2810 REM*****
2820 REM SUB-ROUTINE EQUATION OF STATE
2830 REM CALC.SPECIFIC VOL GIVE VP AND TEMP
2840 REM*****
2850 TEMP=TEMP*1.8+491.67:PRESS=PRESS*14.5
2860 B=EXP(-(K(32)*TEMP/K(8)))
2870 D=K(15)*TEMP/PRESS:E=(K(17)+K(18)*TEMP+K(19)*B)/PRESS
2880 F=(K(20)+K(21)*TEMP+K(22)*B)/PRESS:G=(K(23)+K(24)*TEMP+K(25)*B)/PRESS
2890 H=(K(26)+K(27)*TEMP+K(28)*B)/PRESS:I=(K(29)+K(30)*TEMP+K(31)*B)/PRESS
2900 J=D+K(16):ER=1E-5:V0=J
2910 A=V0-K(16)
2920 V1=J+E/A+F/A^2+G/A^3+H/A^4
2930 IFV1<V0-ER OR V1>V0+ER THEN V0=(V1+V0)/2:GOTO 2910
2940 VOL=V1:A=0:B=0:C=0:V0=0:V1=0:D=0:E=0:F=0:H=0:I=0:J=0
2950 TEMP=(TEMP-491.67)/1.8:PRESS=PRESS/14.5:VOL=VOL/16.02
2960 RETURN
2970 REM*****
2980 REM SUB-ROUTINE LATENT HEAT
2990 REM*****
3000 TEMP=TEMP*1.8+491.67:VP=VP*14.5:VROW=VROW*0.06243:FROW=FROW*0.06243
3010 A=VP*LN(10)*((-K(10)/TEMP^2)+K(11)/(TEMP*LN(10))+K(12)-K(13)*(LOG(EXP(1)))/

```

```

TEMP+K(14)*LOG(K(14)-TEMP)/TEMP^2))
3020 LATHEAT=.185053*TEMP*(1/VROW-1/FROW)*A
3030 TEMP=(TEMP-491.67)/1.8:VP=VP/14.5:VROW=VROW/0.06243:FROW=FROW/0.06243
3038 PRINT "MOTOR OVERPOWER"
3040 LATHEAT=LATHEAT/0.4299
3050 RETURN
3060 REM*****
3070 REM SUB-ROUTINE ENTHALPY
3080 REM*****
3090 TEMP=TEMP*1.8+491.67:VOL=VOL*16.02:PRESS=PRESS*14.5
3100 A=VOL-K(16):B=EXP(-K(32)*TEMP/K(8)):C=1.65E38:REM SETS UPPER LIMIT FOR C
3110 IF K(33)*VOL<88 THEN C=EXP(K(33)*VOL)
3120 D=K(17)/A+K(20)/(2*A^2)+K(23)/(3*A^3)+K(26)/(4*A^4)
3130 D=.185053*D
3140 E=K(19)/A+K(22)/(2*A^2)+K(25)/(3*A^3)+K(28)/(4*A^4)
3150 E=.185053*E*B*(1+K(32)*TEMP/K(8))
3160 F=K(35)*TEMP+K(36)*TEMP^2/2+K(37)*TEMP^3/3+K(38)*TEMP^4/4-K(39)/TEMP
3170 ENTHALPY=F+.185053*PRESS*VOL+D+E+XXX
3180 A=0:B=0:C=0:D=0:E=0:F=0
3190 TEMP=(TEMP-491.67)/1.8:VOL=VOL/16.02:PRESS=PRESS/14.5:ENTHALPY=ENTHALPY/0.
4229
3200 RETURN
3210 REM*****
3220 REM SUB-ROUTINE ENTROPY
3230 REM*****
3240 TEMP=TEMP*1.8+491.67:VOL=VOL*16.02:PRESS=PRESS*14.5
3250 A=VOL-K(16):B=EXP(-K(32)*TEMP/K(8)):C=1.65E38:REM SETS UPPER LIMIT FOR C
3260 IF K(33)*VOL<88 THEN C=EXP(K(33)*VOL)
3270 D=K(18)/A+K(21)/(2*A^2)+K(24)/(3*A^3)+K(27)/(4*A^4)
3280 D=D+K(30)/(VOL*K(33)^2)
3290 D=.185053*D
3300 E=K(19)/A+K(22)/(2*A^2)+K(25)/(3*A^3)+K(28)/(4*A^4)
3310 E=.185053*E*B*K(32)/K(8)
3320 F=K(35)*LN(10)*LOG(TEMP)+K(36)*TEMP+K(37)*(TEMP^2)/2+K(38)*(TEMP^3)/3
3330 F=F-K(39)/(2*TEMP^2)+.185053*K(15)*LN(10)*LOG(A)
3340 ENTROPY=F+E+YYY-D
3350 TEMP=(TEMP-491.67)/1.8:VOL=VOL/16.02:PRESS=PRESS/14.5:ENTROPY=ENTROPY/0.2
88
3360 RETURN
3370 REM *****
3380 REM * CENTRIFUGAL COMPRESSOR MODEL *
3390 REM Input data required for this subroutine are
3400 REM inlet pressure (psia),inlet temperature ('R)
3410 REM making the units compatible with the refriger
3420 REM -ant data routines.
3430 REM Also required is design rotational speed in
3440 REM RPM.
3450 ER$="CONVERGED"
3460 TINC=TINC*1.8+491.67
3470 REM Calc. corrected speed
3480 THETA=SQR(TINC/(288*1.8))
3490 CORN=CN/THETA
3500 REM Now refer this to 66000rpm corrected

```

```

3510 NREF1=CORN/66E3
3520 NREF2=CORN/60E3
3530 REM Calc pressure ratio and flow scale factors
3540 PRSF=C1+C2*NREF1+C3*NREF1^2
3550 QSF1=C4+C5*NREF1+C6*NREF1^2
3560 EFSF=C17+C18*NREF2+C19*NREF2^2
3570 REM Calc corrected pressure ratio
3580 CPR=PRC/PRSF
3590 REM Calc corrected flow iteratively
3600 DIF=.0001
3610 CQ=170/35.31:DQ=1E-3
3620 CQ1=CQ:CQ2=CQ1+DQ
3630 GOSUB 3880
3640 CP1=RP:CQ=CQ2:GOSUB 3880
3650 CP2=RP:DQDF=(CQ2-CQ1)/(CP2-CP1):CQ3=CQ1+DQDF*(CPR-CP1):CQ=CQ3:GOSUB 3880
3660 CP1=RP:CQ2=CQ1:CQ1=CQ3
3670 SUM=SUM+1
3680 IF SUM>100 THEN ER$="DID NOT CONVERGE":GOTO 3710
3690 IF ABS(CP1-CPR)<=DIF THEN 3710
3700 GOTO 3620
3710 REM CAL EFFICIENCY REFERRED TO 60000 AS A FUNCTION OF FLOW
3720 R=C14+C15*CQ1+C16*CQ1^2
3730 EFF=R*EFSF
3740 REM Calc True corrected flow rate
3750 FLOW=CQ1*QSF1
3760 REM CHECK SURGE MARGIN
3770 SURGEPRESS=C20+C21*FLOW*35.31
3780 WARN$=""
3790 IF SURGEPRESS<PRC THEN WARN$=" WARNING ** SURGE ** WARNING ":DMR=DMR+0.01:
:T1=T1+1:TICK=1:REM HOT-GAS BYPASS CONTROL
3800 REM CHECK CHOKE MARGIN
3810 CHOKEPRESS=C22+C21*FLOW*35.31
3820 IF CHOKEPRESS>PRC THEN WARN$=" WARNING ** CHOKED ** WARNING ":TUCK=1:GOTO 3
910:REM SUCTION LINE THROTTLE
3830 PRINT,TAB(20,23);"
3840 PRINT,TAB(20,23);WARN$
3850 PRINT,TAB(20,22);"
3860 PRINT,TAB(20,22);"CALCULATION "ER$
3870 SUM=0
3880 IF CQ<=175/35.31 THEN 3900
3890 RP=C7+C8*CQ+C9*CQ^2+C10*CQ^3:GOTO3910
3900 RP=C11+C12*CQ+C13*CQ^2
3910 RETURN
3920 DATA -8.38000068,16.1980014,-6.46800069,2.51001197,-3.52102412
3930 DATA 2.04601216,917.569251,-571.015321,119.099585,-8.28827368
3940 DATA -8.41999655,5.94384842,-.69266432,-.844600214,.680813351
3950 DATA -8.09411425E-2,-7.65905766,17.2909108,-8.54254212
3960 DATA -1.80571429,4.37142857E-2,-4.86571429,.271276384
3970 REM *****
3980 REM SUB-ROUTINE ELECTRIC MOTOR
3990 REM *****
4000 A=0.9415*WCD/WDPF

```



```

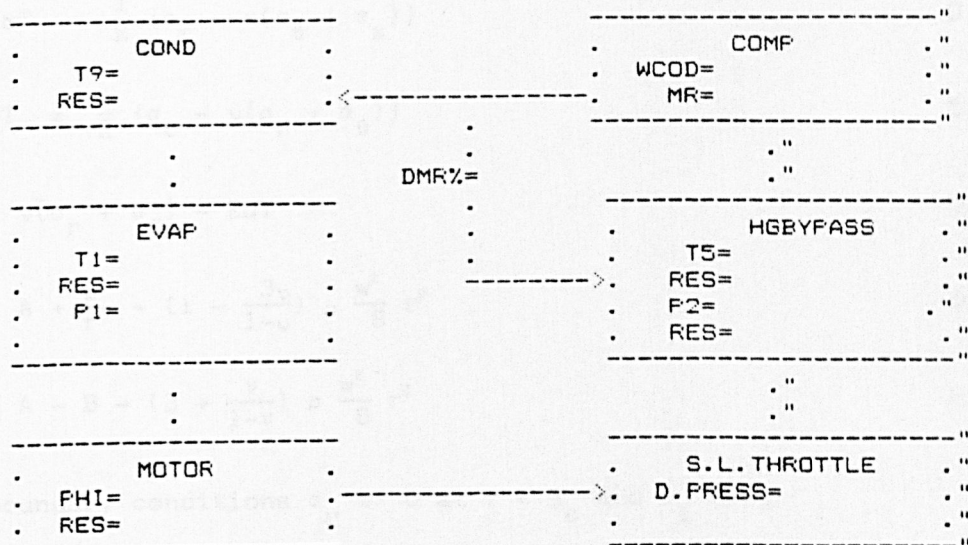
4010 IF A>1.97*0.9415 THEN OVERPOWER=1
4020 IF A<=1.97*0.9415 THEN OVERPOWER=0
4030 b=-19:c=57.33:d=-38.33:D1=0.6:D2=1:E1=0:PHI1=1:J=1:I=0
4040 A1=A/PHI1
4050 PHI=PHI1+(A1-(b+c*PHI1+d*PHI1^2))/(c+2*d*PHI1)
4060 E=A1-(b+c*PHI+d*PHI^2)
4070 PHI1=PHI
4080 IF ABS(E)<1E-5 THEN GOTO 4100
4090 GOTO 4050
4100 A2=b+c*PHI+d*PHI^2:A1=A/PHI1
4110 IF ABS(A1-A2)<1E-5 THEN 4130
4120 GOTO 4050
4130 RETURN
4140 EFFMOD=EFFMDP*(0.08084+0.2155*PHI2+0.77288*PHI2^2)
4150 IF PHI2>0.9417 THEN EFFMOD=EFFMDP
4160 RETURN
4170 REM *****
4180 REM MOTOR PRESSURE LOSS + SUCTION LINE THROTTLE PRESSURE LOSS
4190 REM *****
4200 DP23=DPMDP*((MR*(1-DMR)*SV2)/(MRDP*SVDP2))^2:REM MOTOR DP
4210 DP25=DP23+DPSLT:REM DPSLT = SUCTION LINE THROTTLE DP
4220 P51=P1-DP25
4230 RETURN
4240 REM ENTHALPY INCREASE
4250 DH23=WCOD*(1/EFFMOD-1)/MR
4260 H5=(1-DMR)*(H2+DH23)+DMR*H61
4270 RETURN
4280 REM *****
4290 REM SUB-ROUTINE EVAPORATOR TEMPERATURE (T1) USING ENERGY BALANCE
4300 REM *****
4310 DD=(0.0245+(TADF1-TDP1)*4.733E-4)*(TADF1-TDP1)
4320 QEOD1=MR*(1-DMR)*(H2-H9+LAT9)
4330 C=-DD*QEOD1/QEDF:B=0.0245:A=4.733E-4
4340 DET1=(-B+SQR(B^2-4*A*C))/2/A
4350 DET2=(-B-SQR(B^2-4*A*C))/2/A
4360 IF DET1>=0 AND DET2<0 THEN T11=TAOD1-DET1
4370 IF DET1<0 AND DET2>=0 THEN T11=TAOD1-DET2
4380 IF DET1>0 AND DET2>0 THEN GOTO 4400
4390 GOTO 4420
4400 IF (DET1-DET2)>=0 THEN T11=TAOD1-DET2:GOTO4420
4410 T11=TAOD1-DET1
4420 RETURN
4430 REM *****
4440 REM CONDENSER HEAT USING EQUATION 15
4450 REM *****
4460 QEOD1=MR*(1-DMR)*(H2-H9+LAT9)
4470 QCOD1=MR*(1-DMR)*((H61-H9)+LAT9)
4480 T91=((TDP9-TADF2)*QCOD1/QCDF)+TAOD2
4490 RETURN
4500 REM *****
4510 REM CALC THERMO PROPERTIES OF STATE 5 REPEATED GUESS
4520 REM *****

```

```

4530 PRESS=P5:HB=H1:TB=T1+273:TA=0:HA=0
4540 TEMP=H5*TB/HB:TEMP=TEMP-273:TB=TB-273:GOTO 4560:REM FIST GUESS
4550 TEMP=(TB-TA)*(H5-HB)/(HB-HA)+TB
4560 GOSUB 2820
4570 GOSUB 3070
4580 IF ABS(ENTHALPY-H5)<1E-2 THEN T51=TEMP:SV5=VOL:GOTO 4600
4590 TA=TB:TB=TEMP:HA=HB:HB=ENTHALPY:GOTO 4550
4600 GOSUB 3220
4610 S5=ENTROPY:H5=ENTHALPY:SV5=VOL:T51=TEMP
4620 RETURN
4630 REM *****
4640 REM SUB-ROUTINE HOT-GAS BYPASS
4650 REM *****
4660 IF DPSLT>0 THEN DPSLT=DPSLT-.01:GOTO 4720
4670 GOSUB 4440:REM CALC NEW T91
4680 GOTO 4720
4690 T9=T91
4700 P6=P9
4710 GOSUB 2090:REM NEW COMP DATA
4720 RETURN
4730 REM SUB-ROUTINE SUCTION LINE THROTTLE
4740 IF DMR>0 THEN DMR=DMR-0.01:T1=T1-1:REM REMOVE H/G BYPASS FIRST
4750 DPSLT=DPSLT+0.005
4760 GOSUB 4180
4770 RETURN
4780 CLS
4790 PRINT
4800 PRINT "
4810 PRINT "
4820 PRINT "
4830 PRINT "
4840 PRINT "
4850 PRINT "
4860 PRINT "
4870 PRINT "
4880 PRINT "
4890 PRINT "
4900 PRINT "
4910 PRINT "
4920 PRINT "
4930 PRINT "
4940 PRINT "
4950 PRINT "
4960 PRINT "
4970 PRINT "
4980 PRINT "
4990 PRINT "
5000 PRINT "
5010 PRINT
5020 PRINT " Ambient Temperature ="
5030 RETURN
5035 VDU2

```



```

5036 CLS
5037 PRINT
5038 PRINT "MOTOR OVERPOWER"
5040 PRINT
5050 PRINT "Evaporator Air-On Temperature";TAB(40);TAOD1" 'C"
5060 PRINT "Evaporator Air-On Temperature";TAB(40);TAOD1" 'C"
5070 PRINT "Condenser Air-On Temperature";TAB(40);TAOD2" 'C"
5080 VDU3
5090 GOTO 1030

```


APPENDIX D**ROTOR STRESS CALCULATIONS****D1 Plane Strain Model**

The plane strain condition implies that mechanical strain occurs in one plane only. In the case of a hollow thick-walled cylinder it is assumed that:

$$e_z = 0$$

The sign convention used in this analysis assumes all tensile forces and associated strains are assumed to be positive.

Stress and strain relationships for the Plane Strain condition are given as, (50):

$$e_r - \alpha T = \frac{1}{E} \{ \sigma_r - \nu(\sigma_\theta + \sigma_z) \} \quad D.1$$

$$e_\theta - \alpha T = \frac{1}{E} \{ \sigma_\theta - \nu(\sigma_r + \sigma_z) \} \quad D.2$$

$$\sigma_z = \nu(\sigma_r + \sigma_\theta) - E\alpha T \quad D.3$$

$$\sigma_\theta = A + \frac{B}{r^2} - \left(1 - \frac{3\nu}{1-\nu}\right) \rho \frac{w^2}{8} r^2 \quad D.4$$

$$\sigma_r = A - B - \left(3 + \frac{\nu}{1-\nu}\right) \rho \frac{w^2}{8} r^2 \quad D.5$$

Using the boundary conditions $\sigma_r = 0$ at $r = R_o$ and R_i then:

$$A = \rho \frac{w^2}{8} \left[3 + \frac{\nu}{1-\nu} \right] (R_o^2 + R_i^2) \quad D.6$$

$$B = \rho \frac{w^2}{8} \left[3 + \frac{\nu}{1-\nu} \right] R_o^2 R_i^2 \quad D.7$$

Equation D.3 applies to a restrained cylinder, of fixed length, or, where the deformation is small compared with the length of the cylinder. If the cylinder is unrestrained then:

$$\sigma_z = \frac{\nu \rho w^2}{4(1-\nu)} (R_o^2 + R_i^2 - 2r^2) \quad D.8, (89)$$

D2 A Yield Criterion

If the forces experienced by the rotor are assumed to be due only to the dynamic effects of rotation. In this case the principle stresses are equal to the tangential stress, the radial stress and the axial stress resulting from rotation.

For a steel rotor the yield criterion to be adopted will be the well documented Tresca criteria which states that a material will yield when:

$$\frac{\sigma_{\max}}{2} - \frac{\sigma_{\min}}{2} = \tau_{\max} = \frac{\sigma_{\text{yield}}}{2} \quad \text{D.9}$$

where $\sigma_{\max}$ and $\sigma_{\min}$ are the maximum and minimum principle stresses.

For a three dimensional case the roots of the following expression give the values of the principle stresses, (93).

$$\begin{aligned} \sigma^3 - (\sigma_x + \sigma_y + \sigma_z) \sigma^2 + (\sigma_x \sigma_y + \sigma_y \sigma_z + \sigma_x \sigma_z - \tau_{xy}^2 - \tau_{y2}^2 - \tau_{zx}^2) \sigma - \\ (\sigma_x \sigma_y \sigma_z + 2\tau_{xy} \tau_{zx} + \sigma_x \tau_{y2}^2 - \sigma_y \tau_{y2}^2 - \sigma_y \tau_{xz}^2 - \sigma_z \tau_{yx}^2) = 0 \end{aligned} \quad \text{D.10}$$

For the present case:

$$\sigma_{\max} = \sigma_{\theta} \text{ and } \sigma_{\min} = \sigma_z$$

$$\text{as } \sigma_r = 0 \text{ at } r = R_i$$

therefore:

$$\sigma_{\theta} - \sigma_z = \sigma_{\text{yield}} \quad \text{D.11}$$

D3 Plane Stress Model

An important assumption of the Plane Strain model, described in Section D2, is that the cylindrical rotor must be able to support an axial tensile stress. Although individual laminations may support such a stress the rotor-pack can not: Compression of laminations during assembly is neglected for the moment. For this reason a Plane Stress model may be more appropriate. This requires that all stresses are concentrated in one plane. When considering dynamic stresses in a rotating disc with a central hole it is usually assumed that:

$$\sigma_z = 0$$

The stress-strain relationships for the Plane Stress condition are quoted as, (56):

$$\sigma_r = A + B/r^2 - (3+\nu) \rho \omega^2 r^2 / 8 \quad \text{D.12}$$

$$\sigma_{\theta} = A + B/r^2 - (1-3\nu) \rho \omega^2 r^2 / 8 \quad \text{D.13}$$

and

$$e_z = -\frac{\nu}{E} (\sigma_\theta + \sigma_r) \quad D.14$$

applying the boundary condition

$\sigma_r = 0$ at $r = R_i$ and R_o gives:

$$A = \rho \frac{w^2}{8} (3+\nu) (R_o^2 + R_i^2) \quad D.15$$

$$B = \rho \frac{w^2}{8} (3+\nu) R_o^2 R_i^2$$

Therefore:

$$\sigma_{\max} = \rho \frac{w^2}{4} [(3+\nu)R_o^2 + (1-\nu)R_i^2] \quad D.16$$

In this case the principle stresses at the inner radius are σ_r and σ_θ .
From which:

$$\sigma_{\min} = \sigma_r \text{ and } \sigma_{\max} = \sigma_\theta$$

And:

$$\sigma_{\text{yield}} = \sigma_\theta \text{ at } r = R_i \quad D.17$$

From discussion with Hill, (57), the laminated rotor volume and L/D ratio were constrained such that;

$$R_i = \left[R_o^2 - \frac{V}{1.5\pi R_o} \right]^{\frac{1}{2}} \quad D.18$$

Substituting Equations 18 and 17 into 16 and rearranging gives:

$$R_o^2 = \sigma_{\text{yield}} / \rho w^2 + V(1-\nu) / 6\pi R_o \quad D.19$$

Substituting known values and solving Equation D.19 gives $R_o = 32.75$ mm
from which:

$$\begin{aligned} D &= 65.5 \text{ mm} \\ D_o &= 36.8 \text{ mm} \\ L_i &= 98.3 \text{ mm} \end{aligned}$$

D_o, D_i	outer, inner diameter (m)
e	strain
E	Youngs modulus
L	length (m)
r	radius
R_o, R_i	outer, inner radius
ΔT	change in temperature ($^{\circ}C$)
α	coefficient of thermal expansion ($1/^{\circ}C$)
ρ	density
w	rotational speed
σ	stress
τ	shear stress
σ_{yield}	yield stress
ν	Poissons ratio
sufficies	
θ, r, z	)
	)co-ordinate systems
x, y, z	)

APPENDIX E**ROTOR LAMINATION STRESS CALCULATIONS****E1 Introduction**

In the Appendix D the rotor laminations are treated as simple discs with a central hole. In fact this is not the case as each lamination has, in addition to a central hole, a number of small, equally spaced, holes set circumferentially about its periphery. This is shown in Figure 3.2. The inclusion of these holes will cause the stress distribution within the laminations to be different from that suggested by the calculations described in Appendix D.

This Appendix describes a mathematical 2-D model used to determine the stresses in the electric motor, rotor laminations.

E2 Model for Regions A and C

This model is based on equations D12 and D13. For ease of reference these are given again below:

$$\sigma_r = A + \frac{B}{r^2} - \frac{(3+\nu)}{8} \rho w^2 r^2 \quad \text{D12}$$

$$\sigma = A - B - \frac{(1+3\nu)}{8} \rho w^2 r^2 \quad \text{D13}$$

In this case the boundary conditions are assumed to be:

$$\sigma_r = Z_o \text{ at } r = R_o \quad \text{E1}$$

$$\sigma_r = Z_i \text{ at } r = R_i \quad \text{E2}$$

Substituting these conditions into equations D12 and D13 gives:

$$\begin{aligned} \sigma_R = & \rho w^2 \frac{(3+\nu)}{8} \left[R_o^2 + R_i^2 - \frac{R_i^2 R_o^2}{r} - r^2 \right] \\ & - \frac{R_i^2 R_o^2 (Z_o - Z_i)}{T^2 (R_o^2 R_i^2)} - \frac{(Z_i R_i^2 - Z_o R_o^2)}{(R_o^2 - R_i^2)} \end{aligned} \quad \text{E3}$$

$$\sigma_{\theta} = \rho \omega^2 \frac{(3+\nu)}{8} \left[R_o^2 + R_I^2 + \frac{R_o^2 R_I^2}{r^2} - \frac{(1+3\nu)}{(3+\nu)} r^2 \right] + \frac{R_I^2 R_o^2 (Z_o - Z_I)}{r^2 (R_o^2 - R_I^2)} - \frac{(Z_I R_I^2 - Z_o R_o^2)}{(R_o^2 - R_I^2)} \quad E4$$

Radial growth compatability is given by $e_r = \partial u / \partial r$

Thus,

$$u = \int_0^r e_r dr \quad E5$$

If two points, at similar radii from the axis of rotation, are to maintain the same angular displacement when they are strained equally in a radial direction it is necessary for them to suffer the same angular strain; which is given by:

$$e_{\theta} = \frac{u}{r} + \frac{1}{r} \frac{\partial V}{\partial \theta} \quad E6$$

where $\partial V / \partial \theta$ is a finite angular displacement. From equation D6 the total angular strain is given by:

$$r = \int_0^{2\pi} e_{\theta} d\theta = 2\pi u + \int_0^{2\pi} \left(\frac{\partial V}{\partial \theta} \right) d\theta \quad E7$$

Substituting equation D5 gives:

$$r = \int_0^{2\pi} e_{\theta} d\theta = 2\pi \int_0^r e_r dr + \int_0^{2\pi} \left(\frac{\partial V}{\partial \theta} \right) d\theta \quad E8$$

i.e. Change in Circumference = Increase due to Radial Strain + Increase in Circumference to Maintain Compatability.

In this case the radial growth at any radius 'r' with regions A and C is given by:

$$\Delta r = r e_{\theta} = r/E(\sigma_{\theta} - \nu \sigma_r) \quad E9$$

E3 Model for Region B

For the purpose of these calculations it is assumed that $\sigma_\theta = 0$ within Region B. This comes from an assumption that Region B suffers only radial stress; acting like a simple rod under tension with the addition of body forces due to rotation.

It can be shown that the centrifugal force at any radius within Region B, (Figure 3.2), due only to Region B, is given by:

$$F(r) = \int_r^d w^2 \rho r^2 \theta dr - \int_r^d w^2 2\rho r \sqrt{c^2 - (B-r)^2} \quad E10$$

From equation D10:

$$\sigma_r = \frac{F(r)}{r\theta - 2\sqrt{c^2 - (B-r)^2}} \quad E11$$

Also

$$\Delta r_B = \int_a^d e_r dr = \frac{1}{E} \int_a^d \sigma_r dr \quad E12$$

E5 The effect of Conductor Rods on Stress

It is thought that a reasonable approximation of the effect on stress, within region B, due to the centrifugal load of the conductor rods, will be given if it is assumed that the centrifugal load of each rod is resisted by a segment of Region B. This idea is shown schematically in Figure E3.

Centrifugal force acting on a conductor rod is given by:

$$F_2 = \frac{\pi}{4} c_2 \cdot B \cdot \rho \cdot w^2 t \quad E13$$

(per unit length of rod)

where t = lamination thickness

E6 Compatability Between Regions B and C

Under the action of centrifugal force the hoop like, Region C would expand to a diameter greater than if it were restrained by Region B. This imposes an additional load onto Region B causing it to extend radially. This force is defined as F_3 .

E7 Effect of Compatability and Conductor Rod Load on Equation E12

Applying the principle of Superposition the stress at any radius, r , within Region B is given by adding the additional forces identified in Sections E6 and E5 and the body force within the region, as defined in equation E11. Thus:

$$\sigma_r = \frac{F(r) + F2 + F3}{r\theta - 2\sqrt{c^2 - (B-r)^2}}$$

E8 Method of Solution

The following method was used to solve equations E3 and E4 for Regions A and C, and equation E14 for Region B; enabling the stress variation, throughout the rotor lamination, to be determined.

- (1) Guess $F3$.
- (2) Determine conductor rod load, ($F2$), from equation 13.
- (3) Determine the radial strain in Region B, (RB), by substituting equation E14 into E12 and solve iteratively.
- (4) Determine the radial stress in Region B. Using these results, determine boundary conditions for Regions A and C at their interface with Region B.
- (5) Determine radial and tangential stresses in Regions A and C using equations E3 and E4.
- (6) Determine radial strain in Regions A and C using equation E9.
- (7) Check strain compatability between regions, i.e.:

$$\Delta r_A + \Delta r_B = \Delta r_C$$

If this condition is not satisfied then return to (1) and reguess $F3$ and repeat steps 1 to 7.

- (8) Once strain compatability between regions is satisfied calculate the stress distribution throughout the three regions using equation E14, (iteratively), and equations E3 and E4.

The computer program was written to carry-out the above procedure.

Notation

In addition to the notation described in Figure E1 the following is used in this Appendix.

A,, B	constants in equations D12 and D13
E	Youngs modulus
e	strain
D	diameter
F(r)	radial force in Region B - a function of radius
R, r	radius
u, v	radial and angular displacement
Z	defined by equations E1 and E2
θ	angular displacement (radians)
ν	Poisson's ratio
w	rotational speed
σ	stress

subscripts

r	polar co-ordinate system
I, o	referring to inner and outer conditions of Region

APPENDIX FTHE STIFFNESS OF LAMINATED AND SOLID CYLINDERS IN TENSION

The aim is to show that a laminated rotor, pre-loaded by a tension bolt running through its centre, will have the same stiffness as a solid cylinder of the same length and diameter. To show this it is assumed that the laminated rotor is placed in tension by a force 'F' shown in the figure F3 causing an extension 'e'. It is shown that a plain cylinder subjected to the same force 'F' will produce the same extension 'e' and therefore the stiffness of the laminated rotor is the same as solid cylinder as long as the laminated rotor remains pre-loaded.

Before assembly the laminated pack has an unrestrained length L_L with CSA A_L and the tension bolt has a length L_B with CSA A_B as shown in Figure F1. After assembly the laminations are compressed by an amount e_L and the tension bolt is extended by an amount e_B . Internal stresses produce forces F_L and F_B . When assembled these forces are equal and opposite in direction as shown in Figure F2. One end of the laminated assembly is fixed to a base and a load F is applied to the upper end causing the pack to extend by an amount e , shown in Figure F3, such that $e < e_L$, i.e. the lamination remain pre-loaded.

A force balance gives;

$$F = F_B - F_L$$

$$\left(\begin{array}{l} \text{Force} = \text{strain} \times \text{Youngs Modulus} \times \text{Area} \\ = \frac{\text{change in length} \times \text{Youngs Modulus} \times \text{Area}}{\text{original length}} \end{array} \right)$$

$$F_B = \frac{E_B A_B}{L_B} (e_B + e) \quad \text{and} \quad F_L = \frac{E_L A_L}{L_L} (e_L - e)$$

$$\therefore F = \frac{E_B A_B}{L_B} (e_B + e) - \frac{E_L A_L}{L_L} (e_L - e)$$

This relationship is constrained by $|e| < |e_L|$ and has a boundary condition: $F = 0$ when $e = 0$, from which;

$$E_B \frac{e_B A_B}{L_B} = \frac{E_L e_L A_L}{L_L}$$

Substituting this condition and noting that $E_B = E_L$ for steel laminations and tension bolts gives;

$$F = \frac{EA_B e_B}{L_B} + \frac{EA_B e}{L_B} - \frac{EA_L e_L}{L_L} + \frac{EA_L e}{L_L}$$

$$\therefore F = eE \left[\frac{A_B}{L_B} + \frac{A_L}{L_L} \right]$$

The stiffness of the rotor assembly in tension is given by;

$$\frac{F}{e} = E \left[\frac{A_B}{L_B} + \frac{A_L}{L_L} \right] = E \left[\frac{A_B}{L - e_B} + \frac{A_L}{L + e_L} \right]$$

For strains within the elastic limit e_B and e_L are small compared with L . At the elastic limit $e_B/L \approx 0.01$ and $e_L/L \approx 0.01$ for a 1% proof stress.

$$\text{Therefore } \frac{F}{e} = \frac{E}{L} (A_B + A_L)$$

A solid cylinder with the same dimension as the laminated rotor assembly will have a cross sectional area, A , given by;

$$A = A_B + A_L$$

And stiffness;

$$\therefore \frac{F}{e} = \frac{EA}{L} = \frac{E(A_B + A_L)}{L}$$

This result shows that the stiffness of a laminated rotor assembly will have a stiffness equal to that of a solid cylinder of the same dimensions, as long as the tensile stress does not exceed the pre-load stress.

The analysis given here shows that the assumption, (used in the drive shaft dynamic model in Chapter 4), that the laminated rotor could be treated as a solid cylinder for the purpose of determining critical speeds is valid.

Nomenclature

A_B	cross sectional area of tension bolt (m^2)
A_L	cross sectional area of lamination (m^2)
E	Youngs Modulus (GN/m^2)
e	displacement or deflection (m)
F_B	Tensile force in bolt (N)
F_L	Compression force in laminations (N)
F	Applied load (N)

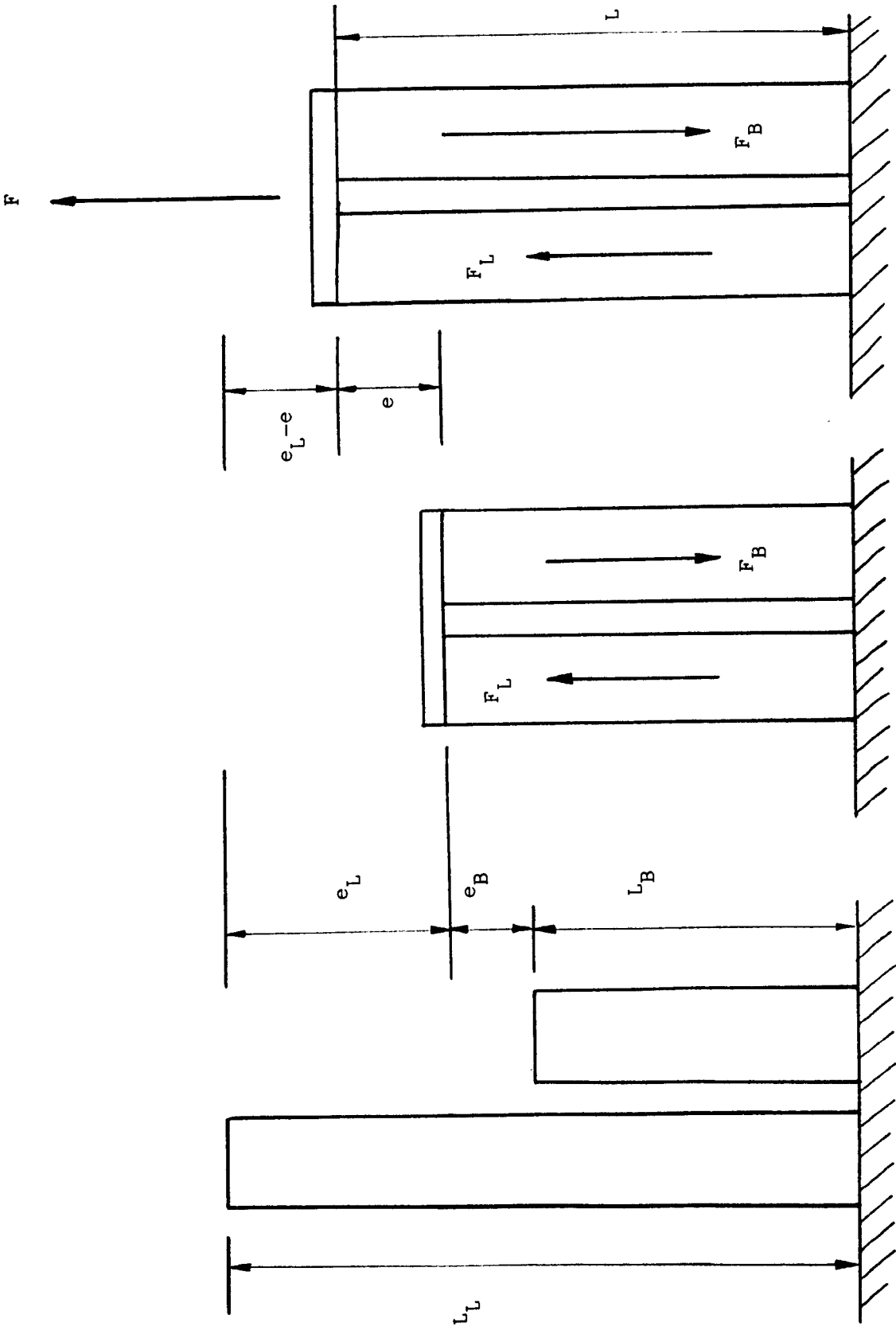


FIGURE F3

FIGURE F2

FIGURE F1

APPENDIX G

THE DYNAMIC PERFORMANCE OF THE PROPOSED ROTOR-SHAFT ASSEMBLY

Introduction

A finite element technique, for determining the critical lateral speeds, due to Myklestad and Prohl is described. A computer program entitled LAT-VIB was developed to set-up and solve the Myklestead-Prohl equation. This program was used to estimate the critical speeds of the rotor-shaft assembly up to and including the fifth critical speed. The LAT-VIB program is discussed and its application is described.

The fundamental torsional (critical) speed was estimated using a standard technique described and discussed in Section G4 along with the results of the analysis.

Notation

The majority of notation used in this appendix is described in the text, however for convenience the following list is provided.

c	damping coefficient
$f_1 f_2$	distributed load
k_s, k	stiffness coefficients
l	length
m	mass
t	time
x, y)	coordinate systems
θ, r)	
ω_c	critical rotational speed
T_n	torsional stiffness
ρ	density
G_1, G_2	constants
D	diameter
E	Young's Modulus
F	Force
G	Modulus of rigidity
I	Second Moment of area
J	Polar moment of inertia
M	Moment
N_c	critical speed rpm
R_A, R_B	reaction at station A, B
V	shear force
$V_1 V_2$	Eigen Vectors
W	load per unit length

G1 Myklestad-Prohl Method

To apply this method the shaft assembly is divided into elements.

Each element has associated with it a transfer matrix, H_n , and a state matrix, $\{Z_n^R\}$, as follows:

$$\begin{bmatrix} Y \\ \theta \\ M \\ V \end{bmatrix}_n^R = \begin{bmatrix} 1 & L & \frac{L^2}{2EI} & \frac{-L^3}{6EI} \\ 0 & 1 & \frac{L}{EI} & \frac{-L^2}{2EI} \\ 0 & -\omega^2 J & \frac{1-\omega^2 J L}{EI} & \frac{-L+\omega^2 J L^2}{2EI} \\ -(\omega^2 m - k) & -(\omega^2 m - k)L & \frac{-(\omega^2 m - k)L^2}{2EI} & \frac{1+(\omega^2 m - k)L^3}{6EI} \end{bmatrix} \begin{bmatrix} Y \\ \theta \\ M \\ V \end{bmatrix}_{n-1} \quad G1$$

or,

$$\{Z_n^R\} = H_n \{Z_{n-1}^R\} \quad G2$$

By applying a recurrence relationship we have:

$$\begin{aligned} \{Z_n^R\} &= H_1 \cdot H_2 \cdot H_3 \dots H_n \{Z_n^R\} \\ &= H \cdot \{Z_n^R\} \end{aligned} \quad G3$$

In the matrix above:

$$\begin{aligned} I &= \pi D^4 / 64 \\ J &= \frac{m D^2}{16} - \frac{m L^2}{12} \\ D &= \text{element diameter} \\ m &= \text{element mass} \\ L &= \text{element length} \\ \omega &= \text{rotational speed in radians/sec} \\ K &= \text{bearing stiffness} \\ E &= \text{Youngs Modulus} \end{aligned}$$

From Equation G1 we have:

$$\begin{bmatrix} Y \\ \theta \\ M \\ V \end{bmatrix}_n = \begin{bmatrix} h_{11} & h_{12} & h_{13} & h_{14} \\ h_{21} & h_{22} & h_{23} & h_{24} \\ h_{31} & h_{32} & h_{33} & h_{34} \\ h_{41} & h_{42} & h_{43} & h_{44} \end{bmatrix} \begin{bmatrix} Y \\ \theta \\ M \\ V \end{bmatrix}_0 \quad G4$$

Natural frequencies of a rotating shaft can be found by defining the values of the boundary matrices $\{Z_n^R\}$ and $\{Z_0^R\}$.

To determine the natural frequencies it is necessary to specify whether a property, (such as Y, θ , M or V) is zero or has some unknown value.

Typical boundary conditions are:

Shaft type	Y	θ	M	V
Simply supported	0	1	0	1
Cantilever:				
Fixed end	0	0	1	1
Free end	1	1	0	0
Encastre support	0	0	1	1

As an example consider the case of a long shaft supported in long bearings. This arrangement can be likened to an encastre beam with the boundary state vectors of:

$$\begin{Bmatrix} Z_n^R \\ \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{Bmatrix} \text{ and } \begin{Bmatrix} Z_o^R \\ \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{Bmatrix}$$

Substituting these values into Equation G4 necessary conditions for solution are:

$$\begin{aligned} h_{13} + h_{14} &= 0 \\ \text{and } h_{23} + h_{24} &= 0 \end{aligned} \quad \text{G5}$$

For Equation G5 to have a non-trivial solution it is necessary for the determinant to be zero.

$$\text{i.e. } \begin{vmatrix} h_{13} & h_{14} \\ h_{23} & h_{24} \end{vmatrix} = 0$$

$$\text{or } h_{13}.h_{24} = h_{23}.h_{14}$$

The main problem in applying the Myklestad-Prohl method is the determination of the transfer matrix H, (in Equation G1), by repeated multiplication of individual transfer matrices, (i.e. $H_1 \cdot H_2 \cdot H_3 \dots H_n$). To overcome this problem a computer program (LAT-VIB) has been written. This is listed in Table G1. To assist future users a short description of the operation of the program is now given.

G2 A Computer Program - LAT-VIB

The program LAT-VIB was written for a BBC Master Series computer and as is written in BBC BASIC.

The method used in the program is the Prohl-Myklestad method which obtains solutions by trial and error. An advantage of this method is that it is unnecessary to assume a mode shape and higher natural frequencies can be obtained without difficulty. In addition an approximation to the mode shape is given.

The program listing, given in Table G1, is, for the most part, self explanatory. There is the minimal amount of data preparation required before using the program which makes the program easy to use.

G2.1 Application

Although outside the subject area of this thesis, a short example of applying LAT-VIB will demonstrate to potential users the simplicity of the program and also demonstrate its accuracy.

Consider a 1 metre long, 25 mm diameter shaft, supported in long bearings. It is usually assumed that the deflection and inclination at each end of the shaft are both be zero and the moment and shear force are both unknown.

The program user must divide their shaft into regions. A region may consist of any length of shaft of constant cross section and constant bearing stiffness. In the present case one region is assumed.

Next the user must decide on the number of, (finite), elements he wishes to use in each region. Any number may be chosen, (within the capacity of the computer). The accuracy of the result improves with increasing numbers of elements, particularly for higher order frequencies, but computing time takes longer.

Users are asked to define the geometry of each region by giving its length and diameter, and also the material properties of density and Youngs Modulus. Also required is the bearing stiffness for each region. Tables G2 to G6 list the input/output data for LAT-VIB used in this example. Only the first and second frequencies are determined.

Table G8 Summary of Results

Number of Elements	1st Frequency (Hz)	2nd Frequency (Hz)
3	115.1	260.2
5	114.4	325.5
10	112.8	315.2
20	112.4	310.8
40	112.3	309.7

It is concluded from Table G8 that:

- (i) The calculated frequency converges to a value with an increasing number of elements.
- (ii) The rate of conversion is greater for lower order frequencies.

It is noted that using Cases formula, (106), for long bearings i.e.:

$$\omega_c = \frac{9}{4} \frac{\pi^2}{l^2} \sqrt{\frac{gEI}{W}}$$

gives a result for the first frequency of 111.5 Hz. A better result is given when comparing Cases formula of a shaft with free ends, i.e.:

$$\omega_c = \frac{\pi^2}{l^2} \sqrt{\frac{gEI}{W}}$$

and the result calculated by LAT-VIB. Both give the same result of 311.5 radians per second: (10 elements used in the LAT-VIB program).

Fig. G1 shows the deflection curves predicted by LAT-VIB for the present example. It should be noted that values are relative to the deflection at station 0 and not absolute values.

Deflection curves for 1 m long 25 mm dia. shaft supported in long bearings.

G.3 Compressor

Shaft Critical Speed

The compressor shaft was divided into 14 regions. Each region was sub-divided into a number of elements - the longer the region the greater the number of elements in that region - so that the total number of elements was 82. Values of Youngs Modulus and material density used are those of steel and the bearing stiffnesses for the 'bearing regions' were determined using design charts, MIT.

The boundary conditions assumed for the LAT-VIB program, (Table G1), were;

$$\begin{Bmatrix} Z_o^R \\ 0 \end{Bmatrix} = \begin{Bmatrix} 1 \\ 1 \\ 0 \\ 0 \end{Bmatrix} \text{ and } \begin{Bmatrix} Z_{14}^R \\ 0 \end{Bmatrix} = \begin{Bmatrix} 1 \\ 1 \\ 0 \\ 0 \end{Bmatrix}$$

Using the above information the LAT-VIB program was run and the critical speed calculated.

The program was re-run for 164, 246 and 328 elements. Program input/output data for the final, (328 element), run are listed in Table G7. The results are summarized in Table 4.1.

The results of this analysis and the underlying assumptions are discussed in Chapter 4.

G.4 Torsional Vibration

In the following analysis it was assumed that the compressor shaft can be represented by a torsionally equivalent shaft of constant diameter.

Let τ_n be the torque required to produce a unit angle of deflection in region 'n' of the compressor shaft. If l is the length of the region and d its diameter then

$$\tau_n = \frac{GJ}{l} = \left(\frac{G\pi d^4}{32l} \right)_n$$

The twist of the element due to a unit of torque is given by.

$$\frac{1}{\tau_n} = \left[\frac{32l}{G\pi d^4} \right]_n$$

The total twist of the shaft due to an applied unit of torque is:

$$\text{Total twist (radians)} = \frac{32}{G\pi} \sum_{n=1}^{n=13} \left(\frac{l}{d^4} \right)_n$$

For the model shaft to have the same torsional equivalence it is necessary for it to deflect the same amount under a similar torque. From this we get:

$$\frac{l_m}{d_m^4} = \sum_{n=1}^{n=8} \left(\frac{l}{d^4} \right)_n$$

If it is assumed that the model has the same length as the compressor shaft then

$$l_m = \sum_{n=1}^{n=8} l_n$$

and

$$d_m = \left[\frac{\sum_{n=1}^{n=8} l_n}{\sum_{n=1}^{n=8} \left(\frac{l}{d} \right)_n} \right]^{0.25}$$

The dimensional data is listed in Table G9

Table G9

n	Length(mm)	diameter(mm)	(l/d^4)	$l.d^4/10^8$
1	10	50	1.6×10^{-6}	6.25
2	25	28	4.067×10^{-5}	1.536
3	5	80	1.22×10^{-7}	20.48
4	76.5	28	1.24×10^{-4}	4.702
5	8	47	1.6394×10^{-6}	3.903
6	79	59	6.519×10^{-6}	95.7
7	8	47	1.6394×10^{-6}	3.903
8	95.5	28	1.553×10^{-4}	5.87
<hr/>				
$\Sigma l =$	307	$\Sigma l/d^4 =$	3.3205×10^{-4}	$\Sigma l d^4 = 142.4$ m ⁵

From Table G9

$$\text{Model length} = 307 \text{ mm}$$

$$\begin{aligned} \text{Model diameter} &= \left[\frac{307}{3.3205 \times 10^{-4}} \right]^{0.25} \\ &= 31 \text{ mm} \end{aligned}$$

The mass moment of inertia of the shaft may be represented by a flywheel, at the free end of the shaft, having a mass moment of inertia, (I_p), equal to 1/3rd that of the shaft. The moment of inertia for the shaft is given by summing the moments for each region, i.e.;

$$I = m_1 \frac{d_1^2}{8} + m_2 \frac{d_2^2}{8} + \dots + m_n \frac{d_n^2}{8}$$

$$\text{note that } m_n = \left[\rho \pi \frac{d^2}{4} l \right]_n$$

$$\text{then } I = \frac{\rho \pi}{32} [d_1^4 l_1 + d_2^4 l_2 + \dots + d_n^4 l_n]$$

and the moment of inertia for an equivalent flywheel is given by

$$I_p = \frac{I}{3} = \frac{\rho \pi}{96} [d_1^4 l_1 + d_2^4 l_2 + \dots + d_n^4 l_n]$$

From Table G9, $\Sigma d^4 l = 1.4237 \times 10^{-6}$

Assuming $\rho = 7850 \text{ kg/m}^3$ we get;

$$\text{Thus } I_p = 3.657 \times 10^{-4} \text{ kg.m}^2$$

The equation of motion for this system is;

$$I_p \ddot{\theta} = -k\theta$$

$$\text{where } k = \frac{GJ}{l}$$

From which the fundamental frequency of oscillation is given by:

$$\omega_c = \sqrt{\frac{GJ}{I_p l}}$$

The modulus rigidity for steel, (G), = 79.3 GPa.

$$\begin{aligned}\text{For the model shaft } J &= \frac{\pi d^4}{32} \\ &= 9.066 \times 10^{-8}\end{aligned}$$

$$l = 0.307 \text{ m}$$

$$I_p = 3.657 \times 10^{-4} \text{ kg m}^2$$

$$\therefore \omega_c = \sqrt{\frac{79.3 \times 10^9 \times 9.066 \times 10^{-8}}{3.657 \times 10^{-4} \times 0.307}}$$

$$\omega_c = 8002 \text{ radians/sec.}$$

$$N_c \approx 76413 \text{ rpm}$$

The results of this analysis and the assumptions made in the model are discussed in chapter 4.

TABLE G1: LISTING OF LAT-VIB

```

>L.
OCA=0:CB=0:MOST=0:CC=1:ITER=0:NOD=0:E1=0:E0=0:W0=0:W1=0
1CLS: CNT=0
2MODE3
3 PRINT TAB(14,8)"*****"
4 PRINT TAB(14)"*          PROG. LAT-VIBS          *"
5 PRINT TAB(14)"*                                     *"
6 PRINT TAB(14)"*          BY          *"
7 PRINT TAB(14)"*                                     *"
8 PRINT TAB(14)"*          I.W.EAMES          *"
9 PRINT TAB(14)"*                                     *"
10 PRINT TAB(14)"*****"
11 PRINT
12 PRINT"Press ANY KEY to continue":
13 select=GET
14 CLS
15 PRINT:PRINT:PRINT
16 PRINT TAB(10)" This program uses the Myklestad-Prohl method to"
17 PRINT TAB(10)" calculate the natural frequencies and associated"
18 PRINT TAB(10)" deflection curves of a solid circular shaft vibrating "
19 PRINT TAB(10)" in a lateral mode."
20 PRINT:PRINT
21 PRINT TAB(10)" Make sure the printer is switched on ":PRINT:PRINT
22 PRINT TAB(10)" Users are asked to read the instructions in"
23 PRINT TAB(10)" the accompanying text before proceeding"
24 PRINT
25 PRINT"Press ANY KEY to continue"
26 select=GET:CLS
27 PRINT TAB(10,16)"Give Job description (60 characters maximum)"
28 INPUT"          ";JOB$:IF LEN(JOB$)>60 THEN CLS:PRINTTAB(10,15)"ERROR -
tle length":GOTO27
29 CLS
30 PRINT TAB(10,16)"Give date and time ":INPUT"          "DAT$
31 VDU2:PRINT TAB(5)JOB$
32 PRINT TAB(5)DAT$
33 VDU3
34 CLS
35 DIM B(4,4),H(4,4),G(4,4)
36 MODE135
37 ZAP=0
38 PRINT TAB(5,6)"Give number of regions":INPUT ZN:DIM EL(ZN),D(ZN,9)
39 CLS
40 PRINT TAB(5,6)"Beginning with the"
41 PRINT TAB(5,7)"right-hand region"
42 REPEAT
43 CA=CA+1:PRINT TAB(5,9)"give the number of"
44 PRINT TAB(5,10)"elements in region"CA:PRINT:INPUT"          ";EL(CA)
45 CLS
46 UNTIL CA=ZN
47 SUM=0
48 FOR P=1 TO ZN:SUM=SUM+EL(P):NEXT P
49 DIM Y(SUM+5),THETA(SUM+5),M(SUM+5),V(SUM+5)
50 REM *****
51 MODE3
52 PRINT:PRINT:PRINT
53 PRINT"Beginning with the right-hand region give the length of the region,
ts diameter,material density,Youngs modulus and bearing stiffness,(if any).
54 PRINT"If values for Youngs modulus,Bearing stiffness and Material Density
are not given then default values will be assumed. These values,give below,"
55 PRINT"are typical values for steel. To initiate default values press RETU
N key when asked to give values for specified regions."

```



```

56PRINT:PRINT
57 PRINT"DEFAULT VALUES;Youngs modulus = 200E9 Pa and Bearing stiffness=0"
58 PRINT"                               Material Density = 7850 kg/m^3"
59 PRINT"-----"
60 PRINT"PRESS ANY KEY TO CONTINUE"
61 CHOSEN=GET
62 MODE135
63 REPEAT
64 CB=CB+1:CLS
65 PRINT:PRINT TAB(2,2)"For region"CB:INPUT"Length";D(CB,1):INPUT"Outer diameter";D(CB,2):INPUT"Material density";D(CB,4):INPUT"Youngs modulus";D(CB,5):INPUT"Bearing stiffness";D(CB,6)
66 IF D(CB,5)=0 THEN D(CB,5)=200E9
67 IF D(CB,4)=0 THEN D(CB,4)=7850
68 D(CB,7)=PI*D(CB,2)^4/64:D(CB,8)=D(CB,4)*PI*D(CB,2)^2*D(CB,1)/4/EL(CB)
69 D(CB,1)=D(CB,1)/EL(CB):D(CB,9)=D(CB,8)*(D(CB,2)^2/16-D(CB,1)^2/12):D(CB,6)=D(CB,6)/EL(CB)
70 REM D(CB,7)= second moment of area,D(CB,8)=element mass,D(CB,1)=element length,D(CB,9)= element J
71 UNTIL CB=ZN
72 REM *****
73 CLS
74 PRINT TAB(2,2)"Give boundary conditions;for the program to run it is necessary to know two conditions at each boundary:please refer to notes in the thesis if you are unsure."
75 PRINT TAB(2,6)" Give boundary conditions for"
76 PRINT TAB(2,7)" right-hand end of shaft "
77 PRINT TAB(2,8)" Y(N),THETA(N),M(N),V(N)"
78 INPUT"                               ";S5,S6,S7,S8
79 PRINT TAB(0,9)" "
80 PRINT TAB(2,7)" left-hand end of shaft "
81 PRINT TAB(2,8)" Y(0),THETA(0),M(0),V(0)"
82 INPUT"                               ";S1,S2,S3,S4
83 CLS:VDU2
84 PRINTTAB(5)" Input Data;"
85 PRINT TAB(5)"Number of regions = "ZN
86PRINT
87PRINTTAB(5)"Boundary conditions Y          THETA          M          V"
88PRINTTAB(5)"Left-hand"TAB(17)S1,S2,S3,S4
89PRINTTAB(5)"Right-hand"TAB(17)S5,S6,S7,S8
90 FOR TT=1 TO ZN
91 PRINT:PRINTTAB(5)"Region"TAB(13)TT
92 PRINTTAB(5)"*****"
93 PRINTTAB(5)"Number of elements = "TAB(25)EL(TT)
94 PRINTTAB(5)"Length of region = "TAB(25)EL(TT)*D(TT,1)
95 PRINTTAB(5)"Diameter = "TAB(25)D(TT,2)
96 PRINTTAB(5)"Material Density = "TAB(25)D(TT,4)
97 PRINTTAB(5)"Youngs Modulus = "TAB(25)D(TT,5)
98 PRINTTAB(5)"Bearing stiffness = "TAB(25)D(TT,6)
99PRINT
100PRINTTAB(5)"*****"
101PRINT
102 NEXT TT
103 VDU3
104 Y(0)=S1:THETA(0)=S2:M(0)=S3:V(0)=S4
105 Y(SUM)=S5:THETA(SUM)=S6:M(SUM)=S7:V(SUM)=S8
106 GOSUB 199:REM DETERMINE BOUNDARY STATE MATRIX
107 CLS
108 REM *****
109 PRINT TAB(1,8)
110 INPUT" Starting value for search routine";W
111 IF W<=0 THEN CLS:PRINTTAB(2,8)"ERROR - W must be greater than zero":GOTO1:

```



```

112 INPUT " Preliminary incremental value";DW
113 IF DW<=0 THEN CLS:PRINTTAB(2,3)"ERROR - DW must be greater than zero":GOTO
112
114 CLS
115 REM *****
116 PRINTTAB(1,1)"Residual"TAB(25,1)"Frequency"
117 W=W-DW
118 W=W+DW
119 IF ITER=1 THEN GOSUB 154
120 CD=1:CC=1:REM COUNT TO SWITCH CC
121 GOSUB 147
122 FOR P=1 TO 4: FOR Q=1 TO 4:H(P,Q)=B(P,Q): NEXT Q:NEXT P
123 CD=2
124 GOSUB 147
125 REPEAT
126 FOR P=1 TO 4
127 FOR Q=1 TO 4
128 G(P,Q)=B(P,1)*H(1,Q)+B(P,2)*H(2,Q)+B(P,3)*H(3,Q)+B(P,4)*H(4,Q)
129 NEXT Q
130 NEXT P
131 FOR P=1 TO 4 :FOR Q=1 TO 4:H(P,Q)=G(P,Q):NEXT Q:NEXT P
132 IF CD=EL(CC) THEN CC=CC+1:CD=0
133 IF CC>ZN THEN 136
134 IF CD=0 THEN GOSUB 147
135 CD=CD+1
136 UNTIL CC>ZN
137 CC=1
138 ER=H(X1,X3)*H(X2,X4)-H(X1,X4)*H(X2,X3)
139 E0=E1:E1=ER:W0=W1:W1=W
140 IF E0=0 OR E1=0 THEN GOTO 142
141 IF E0/ABS(E0)+E1/ABS(E1)=0 THEN ITER=1
142 IF ABS(W0-W)<0.1 THEN GOSUB 161:REM DETERMINE DEFLECTION CURVE
143 CNT=CNT+1:IF CNT>15 THEN CNT=0: CLS:PRINT TAB(1,1)"Residual"TAB(25,1)"Fr
uency ":PRINT
144 PRINT TAB(1)ER;TAB(25)INT(W)
145 GOTO 118
146 END
147 REM *****
148 L=D(CC,1):M=D(CC,8):I=D(CC,7):J=D(CC,9):K=D(CC,6):E=D(CC,5)
149 B(1,1)=1:B(1,2)=L:B(1,3)=L^2/2/E/I:B(1,4)=-1*L^3/6/E/I
150 B(2,1)=0:B(2,2)=1:B(2,3)=L/E/I:B(2,4)=-1*L^2/2/E/I
151 B(3,1)=0:B(3,2)=-1*W^2*J:B(3,3)=1-1*W^2*J*L/E/I:B(3,4)=-L+W^2*J*L^2/2/E/I
152 B(4,1)=-1*(W^2*M-K):B(4,2)=-1*(W^2*M-K)*L:B(4,3)=-1*(W^2*M-K)*L^2/2/E/I:B
4,4)=1+(W^2*M-K)*L^3/6/E/I
153 RETURN
154 REM *****
155 IF E1-E0=0 THEN GOTO160
156 DET=W0^2*W1-W1^2*W0
157 AA=(E0*W1-W0*E1)/DET
158 BB=(W0^2*E1-W1^2*E0)/DET
159 W=-BB/AA
160 RETURN
161 REM *****
162 REM DATA OUTPUT AND DEFLECTION CURVE
163 CLS:PRINTTAB(2,20)"Natural Frequency found !":FOR QQ=1 TO 2.5E3:NEXT QQ
164 PRINTTAB(5,20)"
165 ITER=0:NOD=NOD+1
166 VDU2
167 PRINT TAB(5,20)"Natural Frequency = "INT(10*W/2/PI)/10" Hz"
168 ER=0:W0=0:W1=0:E0=0:E1=0
169 VDU3
170 REM *****
171 REM DEFLECTION CURVE

```

```

172 CC=1:CD=1:R=0
173 GOSUB 215
174 GOSUB 147
175 FOR R=0 TO SUM-1
176 Y(R+1)=B(1,1)*Y(R)+B(1,2)*THETA(R)+B(1,3)*M(R)+B(1,4)*V(R)
177 THETA(R+1)=B(2,1)*Y(R)+B(2,2)*THETA(R)+B(2,3)*M(R)+B(2,4)*V(R)
178 M(R+1)=B(3,1)*Y(R)+B(3,2)*THETA(R)+B(3,3)*M(R)+B(3,4)*V(R)
179 V(R+1)=B(4,1)*Y(R)+B(4,2)*THETA(R)+B(4,3)*M(R)+B(4,4)*V(R)
180 IF CC+1>ZN THEN 183
181 IF CD=EL(CC) THEN CC=CC+1:CD=0:GOSUB 147
182 CD=CD+1
183 NEXT R
184 VDU2
185 IF SUM<10 THEN MUS=1 ELSE MUS=INT(SUM/10)
186 PRINT TAB(5,21)"Deflection Curve;"
187 PRINTTAB(15,22)"Deflection"
188 PRINT
189 FOR P=0 TO SUM STEP MUS
190 PRINT TAB(15)Y(P)
191 NEXT P
192 PRINT:PRINTTAB(15)"*****"
193 VDU3
194 PRINT"SEARCHING FOR NEXT FREQUENCY": FOR QQ=1 TO 1E4:NEXT QQ:CLS
195 Y(0)=S1:THETA(0)=S2:M(0)=S3:V(0)=S4
196 Y(SUM)=S5:THETA(SUM)=S6:M(SUM)=S7:V(SUM)=S8
197 RETURN
198 REM *****
199 REM BOUNDARY MATRICIES
200 BOUND=16^Y(SUM)+8^THETA(SUM)+4^M(SUM)+2^V(SUM)
201 IF BOUND=26 THEN X1=3:X2=4
202 IF BOUND=22 THEN X1=2:X2=4
203 IF BOUND=20 THEN X1=2:X2=3
204 IF BOUND=14 THEN X1=1:X2=4
205 IF BOUND=12 THEN X1=1:X2=3
206 IF BOUND=8 THEN X1=1:X2=2
207 BOUND=16^Y(0)+8^THETA(0)+4^M(0)+2^V(0)
208 IF BOUND=26 THEN X3=1:X4=2
209 IF BOUND=22 THEN X3=1:X4=3
210 IF BOUND=20 THEN X3=1:X4=4
211 IF BOUND=14 THEN X3=2:X4=3
212 IF BOUND=12 THEN X3=2:X4=4
213 IF BOUND=8 THEN X3=3:X4=4
214 RETURN
215 REM *****
216 REM SET INITIAL DEFLECTION
217 IF Y(SUM)=1 THEN 225
218 IF BOUND=26 THEN Y(0)=H(X1,X4)/H(X1,X3):THETA=-1:M(0)=0:V(0)=0
219 IF BOUND=22 THEN Y(0)=H(X1,X4)/H(X1,X3):M(0)=-1:THETA(0)=0:V(0)=0
220 IF BOUND=20 THEN Y(0)=H(X1,X4)/H(X1,X3):V(0)=-1:THETA(0)=0:M(0)=0
221 IF BOUND=14 THEN THETA(0)=H(X1,X4)/H(X1,X3):M(0)=-1:Y(0)=0:V(0)=0
222 IF BOUND=12 THEN THETA(0)=H(X1,X4)/H(X1,X3):V(0)=-1
223 IF BOUND=8 THEN M(0)=H(X1,X4)/H(X1,X3):V(0)=-1
224 GOTO 231
225 IF BOUND=26 THEN Y(0)=(1+H(X1,X4))/H(X1,X3):THETA=-1
226 IF BOUND=22 THEN Y(0)=(1+H(X1,X4))/H(X1,X3):M(0)=-1
227 IF BOUND=20 THEN Y(0)=(1+H(X1,X4))/H(X1,X3):V(0)=-1
228 IF BOUND=14 THEN THETA(0)=(1+H(X1,X4))/H(X1,X3):M(0)=-1
229 IF BOUND=12 THEN THETA(0)=(1+H(X1,X4))/H(X1,X3):V(0)=-1
230 IF BOUND=8 THEN M(0)=(1+H(X1,X4))/H(X1,X3):V(0)=-1
231 RETURN

```

EXAMPLE 1 - INPUT AND OUTPUT DATA FOR LAT-VIB

SIMPLE SHAFT - LONG BEARINGS

SIMPLE SHAFT - LONG BEARING

31/7/87

Input Data;

Number of regions = 1

Boundary conditions	Y	THETA	M	V
Left-hand	0	0	1	1
Right-hand	0	0	1	1

Region 1

Number of elements = 3
Length of region = 1
Diameter = 2.5E-2
Material Density = 7850
Youngs Modulus = 2E11
Bearing stiffness = 0

Natural Frequency = 115.1 Hz
Deflection Curve;
Deflection

0
-1.48706519E-6
-1.48706518E-6
1.02140518E-14

Natural Frequency = 260.2 Hz
Deflection Curve;
Deflection

0
-6.51545536E-7
6.51545626E-7
6.17284002E-14

TABLE G3

EXAMPLE 2 - INPUT AND OUTPUT DATA FOR LAT-VIB

SIMPLE SHAFT - LONG BEARINGS

Input Data;

Number of regions = 1

Boundary conditions	Y	THETA	M	V
Left-hand	0	0	1	1
Right-hand	0	0	1	1

Region 1

Number of elements =

5

Length of region = 1

Diameter = 2.5E-2

Material Density = 7850

Youngs Modulus = 2E11

Bearing stiffness = 0

Natural Frequency = 114.4 Hz

Deflection Curve;

Deflection

0

-7.51109968E-7

-1.75736457E-6

-1.75736456E-6

-7.51109953E-7

1.19904087E-14

Natural Frequency = 325.5 Hz

Deflection Curve;

Deflection

0

-3.07283748E-7

-2.56742959E-7

2.56742579E-7

3.07283958E-7

-2.94098079E-13

TABLE G4 - EXAMPLE 3

INPUT AND OUTPUT DATA FOR LAT-VIB

SIMPLE SHAFT - LONG BEARINGS

Input Data;

Number of regions = 1

Boundary conditions	Y	THETA	M	V
Left-hand	0	0	1	1
Right-hand	0	0	1	1

Region 1

 Number of elements =

10

Length of region = 1
 Diameter = 2.5E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Natural Frequency = 112.8 Hz
 Deflection Curve;
 Deflection

0

-2.3516909E-7
 -7.69674701E-7
 -1.3612093E-6
 -1.80709086E-6
 -1.97165076E-6
 -1.80709085E-6
 -1.36120929E-6
 -7.69674684E-7
 -2.35169075E-7
 9.14546217E-15

Natural Frequency = 315.2 Hz
 Deflection Curve;
 Deflection

0

-1.19649167E-7
 -3.1568677E-7
 -3.92996998E-7
 -2.69745118E-7
 3.47846751E-14
 2.69745168E-7
 3.92997001E-7
 3.15686729E-7
 1.19649113E-7
 -1.49186219E-14

TABLE G5EXAMPLE 4 - INPUT AND OUTPUT DATA FOR LAT-VIB

SIMPLE SHAFT - LONG BEARINGS

Input Data;

Number of regions = 1

Boundary conditions	Y	THETA	M	V
Left-hand	0	0	1	1
Right-hand	0	0	1	1

Region 1

Number of elements =

20

Length of region = 1

Diameter = 2.5E-2

Material Density = 7850

Youngs Modulus = 2E11

Bearing stiffness = 0

Natural Frequency = 112.4 Hz

Deflection Curve;

Deflection

0

-2.36701634E-7
 -7.75171629E-7
 -1.37149851E-6
 -1.82118695E-6
 -1.98718666E-6
 -1.82118696E-6
 -1.37149853E-6
 -7.75171654E-7
 -2.36701667E-7
 -4.11511103E-14

Natural Frequency = 310.8 Hz

Deflection Curve;

Deflection

0

-1.2192041E-7
 -3.22616175E-7
 -4.02316207E-7
 -2.76401691E-7
 -2.57944707E-14
 2.76401651E-7
 4.0231619E-7
 3.2261617E-7
 1.21920377E-7
 -1.40328721E-13

TABLE G6

EXAMPLE 5 - INPUT AND OUTPUT DATA FOR LAT-VIB

SIMPLE SHAFT - LONG BEARINGS

Input Data;

Number of regions = 1

Boundary conditions	Y	THETA	M	V
Left-hand	0	0	1	1
Right-hand	0	0	1	1

Region 1

 Number of elements =

40

Length of region = 1
 Diameter = 2.5E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Natural Frequency = 112.3 Hz
 Deflection Curve;
 Deflection

0

-2.37094585E-7
 -7.76572766E-7
 -1.37411579E-6
 -1.82476938E-6
 -1.9911339E-6
 -1.82476914E-6
 -1.37411537E-6
 -7.76572305E-7
 -2.37094245E-7
 3.68997367E-14

Natural Frequency = 309.7 Hz
 Deflection Curve;
 Deflection

0

-1.22536304E-7
 -3.24446393E-7
 -4.04757447E-7
 -2.78139637E-7
 -1.57811047E-14
 2.78139611E-7
 4.04757433E-7
 3.24446382E-7
 1.22536266E-7
 -1.21183012E-13

TABLE G7

INPUT AND OUTPUT DATA FOR LAT-VIB
(COMPRESSOR SHAFT)

COMPRESSOR SHAFT

COMPRESSOR SHAFT

31/7/87

Input Data;

Number of regions = 14

Boundary conditions	Y	THETA	M	V
Left-hand	1	1	0	0
Right-hand	1	1	0	0

Region 1

Number of elements =

40

Length of region = 5.3E-2

Diameter = 2.8E-2

Material Density = 7850

Youngs Modulus = 2E11

Bearing stiffness = 0

Region 2

Number of elements =

32

Length of region = 2.2E-2

Diameter = 2.8E-2

Material Density = 7850

Youngs Modulus = 2E11

Bearing stiffness = 25937.5

Region 3

Number of elements =

24

Length of region = 1.65E-2

Diameter = 2.8E-2

Material Density = 7850

Youngs Modulus = 2E11

Bearing stiffness = 0

Region 4

 Number of elements =
 8
 Length of region = 8E-3
 Diameter = 4.7E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 5

 Number of elements =
 64
 Length of region = 7.9E-2
 Diameter = 5.87E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 6

 Number of elements =
 8
 Length of region = 8E-3
 Diameter = 4.7E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 7

 Number of elements =
 24
 Length of region = 2.8E-2
 Diameter = 2.8E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 8

 Number of elements =
 20
 Length of region = 2.5E-2
 Diameter = 2.8E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 41500

Region 9

 Number of elements =
 20
 Length of region = 2.35E-2
 Diameter = 2.8E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 10

 Number of elements =
 8
 Length of region = 5E-3
 Diameter = 8E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 11

 Number of elements =
 24
 Length of region = 2.5E-2
 Diameter = 2.8E-2
 Material Density = 7850
 Youngs Modulus = 2E11
 Bearing stiffness = 0

Region 12 382

Number of elements =
8
Length of region = 1E-2
Diameter = 5E-2
Material Density = 7850
Youngs Modulus = 2E11
Bearing stiffness = 0

Region 13

Number of elements =
8
Length of region = 2E-3
Diameter = 2.8E-2
Material Density = 7850
Youngs Modulus = 2E11
Bearing stiffness = 0

Region 14

Number of elements =
40
Length of region = 4.8E-2
Diameter = 8E-3
Material Density = 7850
Youngs Modulus = 2E11
Bearing stiffness = 0

Natural Frequency = 108.7 Hz
 Deflection Curve;
 Deflection

2.32423638E-2
 6.56436683E-2
 9.27531517E-2
 0.114740139
 0.152255749
 0.191603955
 0.227964709
 0.267002065
 0.299384642
 0.328174723
 0.327551503

Natural Frequency
 Deflection Curve
 Deflection

Natural Frequency = 131 Hz
 Deflection Curve;
 Deflection

0.200575828
 0.242989895
 0.27017911
 0.292271528
 0.329900595
 0.369335441
 0.405929221
 0.44707334
 0.483706955
 0.518473956
 1.01441266

Natural Frequency
 Deflection Curve
 Deflection

Natural Frequency = 1456.8 Hz
 Deflection Curve;
 Deflection

8.40104197E-2
 0.127159806
 0.159514714
 0.194660739
 0.267917486
 0.353425278
 0.536419392
 1.55018331
 3.40694593
 6.06567594
 310.915511

 Natural Frequency = 2449.4 Hz
 Deflection Curve;
 Deflection

5.96680679E-2
 0.103602452
 0.141829036
 0.192430823
 0.308786689
 0.453264468
 0.897122479
 3.90716975
 9.73246583
 18.6220245
 1557.15814

Natural Frequency = 3464.7 Hz
 Deflection Curve;
 Deflection

4.56404207E-2
 9.04372312E-2
 0.135504857
 0.20485673
 0.374712253
 0.597519249
 1.5086788
 8.44393742
 22.5063523
 45.5532042
 5801.36188

Natural Frequency = 6842.1 Hz
 Deflection Curve;
 Deflection

2.76990121E-2
 7.60833394E-2
 0.151934163
 0.311306028
 0.74977027
 1.44053467
 7.0200696
 58.6433718
 180.259699
 435.787969
 148279.972

Natural Frequency = 14184.4 Hz
Deflection Curve;
Deflection

1.85042324E-2
7.9142833E-2
0.270682973
0.817640603
2.54737585
6.71950699
79.0565462
915.963805
3636.30217
11863.2774
13624402.2

Natural Frequency = 14223.9 Hz
Deflection Curve;
Deflection

1.84774452E-2
7.91955566E-2
0.271528213
0.82138708
2.56142943
6.7677488
79.8931908
926.846921
3683.99413
12032.0243
13890871.9

Natural Frequency = 14259.8 Hz
Deflection Curve;
Deflection

1.84532498E-2
7.92435454E-2
0.27229615
0.82479344
2.57421619
6.81172436
80.6578802
936.806731
3727.6898
12186.7539
14136481.9

Natural Frequency = 14355.2 Hz
Deflection Curve;
Deflection

1.83891586E-2
7.93723568E-2
0.274351059
0.83392028
2.60851849
6.93008145
82.725539
963.797255
3846.33765
12607.4699
14810408

Natural Frequency = 14379.1 Hz
Deflection Curve;
Deflection

1.83732231E-2
7.94047675E-2
0.274866684
0.836213129
2.61714553
6.9599366
83.2492805
970.647767
3876.50561
12714.5739
14983384.6

Natural Frequency = 14436.7 Hz
Deflection Curve;
Deflection

1.83349275E-2
7.94832861E-2
0.276113552
0.841762123
2.63803999
7.03239141
84.5239579
987.343482
3950.11886
12976.1364
15408198.4

Natural Frequency = 14222.4 Hz
Deflection Curve;
Deflection

1.84784899E-2
7.91934924E-2
0.271495152
0.821240485
2.56087934
6.76585869
79.8603666
926.419671
3682.12076
12025.3932
13880373.3

Natural Frequency = 14425.3 Hz
Deflection Curve;
Deflection

1.83424922E-2
7.94677057E-2
0.275866388
0.840661651
2.63389444
7.01799958
84.2703596
984.019267
3935.452
12923.998
15323249.7

Natural Frequency = 14496.8 Hz
Deflection Curve;
Deflection

1.82952215E-2
7.95656391E-2
0.277417925
0.847573855
2.65994785
7.10858227
85.8698554
1005.0071
4028.13575
13253.6736
15862620.9

Natural Frequency = 14541.2 Hz
Deflection Curve;
Deflection

1.82660103E-2
7.96268418E-2
0.278385128
0.85188781
2.67622557
7.16533931
86.8760865
1018.2362
4086.65626
13462.0699
16206291.7

Natural Frequency = 15767 Hz
Deflection Curve;
Deflection

1.75040355E-2
8.14164446E-2
0.306023895
0.976805074
3.15327539
8.8825581
118.685616
1445.8618
6014.11661
20408.2233
28770833.7

Natural Frequency = 15979.1 Hz
Deflection Curve;
Deflection

1.73800498E-2
8.17446017E-2
0.310989097
0.999585275
3.24142291
9.21088466
125.046565
1533.37539
6415.85033
21871.5571
31670651

Natural Frequency = 16018.9 Hz
Deflection Curve;
Deflection

1.73570395E-2
8.18066975E-2
0.311925917
1.00389505
3.25813819
9.27352018
126.26952
1550.27104
6493.65358
22155.4381
32242676.7

Natural Frequency = 16119.9 Hz
Deflection Curve;
Deflection

1.72989294E-2
8.19651975E-2
0.314313466
1.01489556
3.30085863
9.43414196
129.419275
1593.8885
6694.85862
22890.2493
33737380

Natural Frequency = 19773.8 Hz
Deflection Curve;
Deflection

1.54591636E-2
8.83721476E-2
0.40848244
1.46805992
5.12371469
16.9482344
294.781208
4047.51338
18504.8014
66679.4115
152870261

APPENDIX EMOTOR COOLING ANALYSIS

To assess the possibility of natural convective cooling the motor-stator is assumed to be finite cylinder, with insulated ends, mounted horizontally and with internal heat generation. This arrangement is shown in Figure H1.

The conduction of heat from the core of the motor is given by:

$$Q = \frac{2\pi k_s L(T_1 - T_s)}{\ln(R_2/R_1)} \quad -H1$$

The natural convective heat transfer coefficient at the surface of a horizontal cylinder may be determined using the following relationship.

$$\bar{N}_{u_D} = 0.53 (Gr_D Pr)^{1/4} \quad -H2$$

$$\text{by definition } Gr = \rho^2 g \beta (T_s - T_w) \frac{D^3}{\mu^2} \quad -H3$$

For air at 20°C: $\rho = 1.2 \text{ kg/m}^3$

$$\beta = \frac{1}{293} \text{ 1/}^\circ\text{K}$$

$$\mu = 2.5 \times 10^{-5} \text{ Pa.s}$$

Also: $D = 0.16 \text{ m}$ and $g = 9.81 \text{ m/s}^2$

Substituting known values into equation 3 yields.

$$Gr = 315969 (T_s - T_w) \quad -H4$$

Assuming $Pr = 0.7$ and substituting Equation H4 into 2 gives:

$$\bar{N}_{u_D} = 11.5 (T_s - T_w)^{1/4} \quad -H5$$

$$\text{By definition: } \bar{N}_{u_D} = \frac{\bar{h}_0 D}{K_{AIR}} \quad -H6$$

Substituting known values into Equation H6 and the result into Equation H5 gives on rearrangement

$$\bar{h}_s = 1.795 \times 10^{-2} (T_s - T_w)^{1/4} \text{ kW/m}^2/\text{K} \quad -H7$$

Radiant heat emission from the surface of the motor casing may be determined from:

$$Q_R = \epsilon \sigma [T_s^4 - T_w^4] A \quad -H8$$

For blackened steel the emissivity, $[\epsilon]$, is given in tables as 0.8.

Stefan-Boltzman constant, $\sigma = 56.7 \times 10^{-12} \text{ kW/m}^2/\text{K}^4$.

Heat transfer from the surface of the cylinder is given by the sum of convective and radiant components, thus:

$$Q_s = Q_c + Q_r$$

$$\therefore Q_s = h_s A [T_s - T_w] + \epsilon \sigma [T_s^4 - T_w^4] A \quad -H9$$

The core temperature of the motor (T_i) is determined by solving Equation (9), (7) and (1) simultaneously. Results of analysis are plotted in Figure 3.5.

H3 Forced Convective Heat Transfer

From Krieth (88) the convective heat transfer coefficient for a horizontal cylinder may be determined from:

$$\bar{h}_s = \frac{k}{D} \left[0.4 Re_D^{0.5} + 0.06 Re_D^{0.67} \right] Pr^{0.4} \left[\frac{\mu_s}{\mu_\infty} \right]^{0.25} \quad -H10$$

where μ_s = viscosity at the surface of the cylinder

and μ_∞ = viscosity of air at a remote distance from the cylinder surface.

For $T > 200$ K

$$\mu = 1.329 \times 10^{-5} + 4.715 \times 10^{-6} (T-200) - 1.418 \times 10^{-11} (T-200)^2$$

where T is given in Kelvin.

For $T = T_w = 293$ K, $\mu = 1.755 \times 10^{-5}$ Pa.s.

By definition: $Re_D = \frac{\rho c D}{\mu}$

Also assuming: $Pr = 0.7$ and $\rho = 1.2$ kg/m³

To determine the core temperature Equations (H10), (H9) and (H1) are solved simultaneously over a range of Q , c values. Results of this analysis have been plotted in Fig. 3.6. From an examination of Fig. 3.6 it is concluded that forced convection at reasonable air velocities over the cylinder will be insufficient to produce the necessary heat transfer required to maintain adequate motor core temperatures. With heat rejection rates of 2.0 kW a core temperature of 500°C is predicted when a flow velocity across the cylinder is 41 m/s.

H4 The Addition of Fins

The addition of fins to the surface of the motor casing will augment the heat transfer. One way to investigate the effect of fins is to use a fin augmentation factor (α) which is defined as

$$\alpha = \frac{\text{Effective Heat Transfer Area}}{\text{Actual area without fins}}$$

Apply α to Equation (9) we get

$$Q_s = 2\pi R_0 \alpha (h_s (T_s - T_\infty) + \epsilon \sigma (T_s^4 - T_\infty^4))$$

Once again the core temperature is determined by solving Equations H10, H1 and H11 simultaneously over a range of α , Q values. The results of this analysis are plotted in Fig. 3.7.

Referring to Fig. H4 it is concluded that a Fin Augmentation Factor, (α), much greater than 5 is needed to reduce the core temperature of the motor to an acceptable level.

The performance of circumferential and axial finning was investigated using a computer program listed in Table H1. This program was written to enable various fin option to be investigated. It determines the performance of rectangular fins only.

The necessary Input Data for the program are:

- a) proposed fin height
- b) fin thickness
- c) material conductivity
- d) local heat transfer coefficient
- e) fin base temperature (T_s)

By assuming an heat flux of 2 kW and a motor core temperature (T_1) of 150°C, then from Equation H1 the surface temperature is given by:

$$T_s = T_1 - \frac{Q \ln (R_s/R_1)}{2\pi k_s l}$$

From which $T_s = 53^\circ\text{C}$.

From Equation H10, we get $h_s = 86 \text{ W/m}^2/\text{K}$.

Results of FINDRUM analysis are plotted in Fig. H2.

These results are based on circumferential fins of height 10 mm and 20 mm.

From Fig. H2 it is seen that the highest heat transfer rate is 358 W for 20 mm fins of 3 mm thickness.

In this case the effective fin area is:

$$\begin{aligned} A_f &= 2\eta_{fin} \cdot \pi [(R_0 + H)^2 - R_0^2] \cdot n \\ &= 0.018 \text{ m}^2 \end{aligned}$$

For 20 mm fins the spacing = 0.4×20
= 8 mm

For a motor length of 0.076 mm the number of fins is 7.

$$\therefore \text{Total effective fin area} = 7 \times 0.018 \\ = \underline{0.126 \text{ m}^2}$$

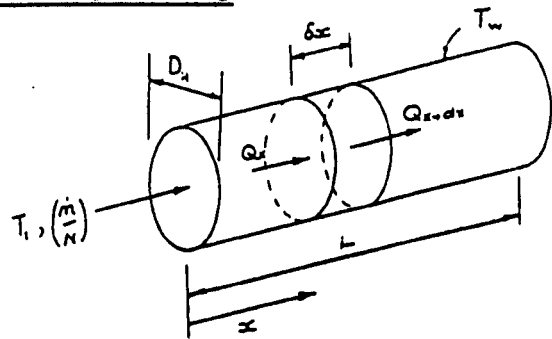
$$\text{Casing area with fins} = 7 \times 0.008 \times \pi \times 0.16 \\ = \underline{0.028 \text{ m}^2}$$

$$\text{Total Effective Heat Transfer area} = 0.028 + 0.126 \\ = \underline{0.154 \text{ m}^2}$$

$$\text{Casing area without fins} = \pi \times 0.16 \times 0.076 \\ = \underline{0.0382}$$

$$\therefore \alpha = \frac{0.154}{0.0382} = 4$$

H5.1 Stator Cooling Passages Design Theory



The above diagram represents a cooling passage running axially through the motor-stator laminations. The mass flow through each of N passages is equal to $\dot{m}/N$ where $\dot{m}$ is to total refrigerant flow rate. Refrigerant gas enters the cooling passage with a temperature T_1 and the stator laminations are assumed to be at a temperature T_w .

An energy balance taken across an elemental length, (dx) , of a cooling passage yields:

$$Q_{x+dx} - Q_x = hP (T_w - T_x) dx$$

where h is the heat transfer coefficient and P is the perimeter of the passage. Applying a Taylors-series expansion to Q_{x+dx} gives

$$Q_{x+dx} = Q_x + \frac{dQ}{dx} dx$$

substitution yields

$$\frac{dQ}{dx} = hP (T_w - T_x)$$

$$\text{Also } \frac{dQ}{dx} = \left(\frac{\dot{m}}{N} \right) C_p \frac{dT}{dx} = hP (T_w - T_x)$$

Integration yields:

$$\frac{T_w - T_1}{T_w - T_2} = \exp \left[\frac{NhPL}{\dot{m} C_p} \right]$$

From which: $T_2 = T_w - \frac{(T_w - T_1)}{\exp \left[\frac{NhPL}{\dot{m} Cp} \right]}$

The heat transfer to the cooling flow may be determined from:

$$Q = \dot{m} Cp (T_2 - T_1)$$

Substitution yields:

$$Q = \dot{m} Cp (T_w - T_1) \left[1 - \exp \left[- \frac{NhPL}{\dot{m} Cp} \right] \right] \quad -H12$$

The expression, $\left[1 - \exp \left[- \frac{NhPL}{\dot{m} Cp} \right] \right]$ is analagous to the effectiveness

(ϵ), of a heat exchanger. To maximise the heat transfer, (or cooling

efficiency) the design would need to maximise ϵ .

H5.2 Heat Transfer Coefficient (h)

For the purpose of analysis a mean value of heat transfer coefficient, (h), is assumed. Krieth (§§) suggests the follow (h) ing Nusselt modulus correlation for long ducts and passages.

$$\bar{N}_{DH} = 0.023 Re_{DH}^{0.8} \cdot Pr^{0.33} \quad -H13$$

By definition: $\bar{N}_{DH} = \frac{hD_H}{K}$, $Re_{DH} = \frac{\rho C D_H}{\mu}$, $Pr = \frac{Cp \mu}{K}$

Therefore: $h = 0.023 \left(\frac{K}{D_H} \right) Re_{DH}^{0.8} Pr^{0.33} \quad -H14$

For a rectangular passage of height (l) and breadth (b), the hydraulic diameter is given by:

$$D_H = \frac{2lb}{(l + b)} \quad -H15$$

Assuming the following property data for Freon 11 at 10°C:

Thermal Conductivity (K) = 0.007 W/m/K

Viscosity (μ) = 9.9×10^{-6} Pa.s.

Specific Heat (Cp) = 502 J/kg/

Density (ρ) = 3.6 kg/m³

Therefore $Pr = \frac{502 \times 9.9 \times 10^{-6}}{0.007} = 0.71 \quad -H16$

From mass flow continuity $\frac{\dot{m}}{N} = \rho Ac$

Therefore $c = \frac{\dot{m}}{N \rho lb}$

Substituting this result into the Reynolds number expression gives:

$$Re = \frac{2\dot{m}}{Nu (l+b)} \quad -H17$$

Substituting Equations H15.4, H16.5 and H17.6 into H14.3 and substituting property values yields:

$$H = 1.262 \frac{(l+b)^{0.2}}{lb} \left(\frac{\dot{m}}{N} \right)^{0.8} \quad -H18$$

$$\text{Let } \phi = \frac{NhPL}{\dot{m}C_p} \quad -H19$$

Substituting Equation H18.7 and noting that $P = 2(l+b)$ and $C_p = 502 \text{ J/kg/K}$ yields:

$$\phi = 5.03 \times 10^{-3} \frac{(l+b)^{1.2} L}{(lb)} \left(\frac{N}{\dot{m}} \right)^{0.2} \quad -H20$$

Letting $\lambda = \frac{b}{l}$, where $\lambda \in [0,1]$ and substituting into Equation H20 gives:

$$\phi = 5.03 \times 10^{-3} \frac{[1 + \lambda]^{1.2}}{l^{0.8} \lambda} L \left(\frac{N}{\dot{m}} \right)^{0.2} \quad -H21$$

An examination of Equation H21 shows that as λ decreases the value of ϕ increases, and with this the heat-transfer, as is shown from an examination of Equation H12. It is concluded that for an effective cooling passage its breadth should be many times greater than its height. There should also be a large number of passages (N).

The disadvantage of narrow cooling passages is that pressure losses tend to be high.

H5.3 Pressure Losses

The pressure loss in a cooling passage may be determined from:

$$\Delta P = f \left(\frac{\rho c^2}{2} \right) \left[\frac{L}{D_H} \right] \quad -H22$$

For turbulent flow in smooth pipes, ($Re > 2000$), the friction factor (f) may be determined from the approximate formula:

$$f = 0.2 Re^{-0.2} \quad -H23$$

Substituting Equation H23 into H22 and substituting property values yields

$$\Delta P = 0.028 \frac{c^{1.8} L}{D_H^{1.2}} \quad -H24$$

In this case: $c = \frac{\dot{m}}{N\rho lb}$ and $D_H = \frac{2lb}{(l+b)}$

On substitution into Equation H22 and substituting property values yields:

$$\Delta P = 1.215 \times 10^{-3} \frac{(l+b)^{1.2}}{(lb)^3} \left(\frac{\dot{m}}{N} \right)^{1.8} L$$

Substituting $b = f\lambda$ gives:

$$\Delta P = 1.215 \times 10^{-3} \frac{[1 + \lambda]^{1.2}}{f^{4.8} \lambda^5} L \left(\frac{\dot{m}}{N} \right)^{1.8} \quad -H25$$

H5.4 Selection of Cooling Passage Geometry

From the point of view of minimizing pressure loss selecting a large number circular passages would be best. For example 200 x 2.5 mm diameter passages would provide sufficient cooling and the parasitic power to overcome pressure losses would be 59 watts. From a manufacturing point of view a smaller number of larger diameter passages would be better.

After some discussion with the company manufacturing the stator lamination and repeated pressure loss and heat transfer calculation 40 x 1.5 mm x 1.8 mm were chosen.

Substituting Equation H21 into H12 gives:

$$Q = \dot{m} C_p (T_w - T_1) \left\{ 1 - \exp \left[\frac{-(1 + \lambda)^{1.2}}{199 f^{0.8} \lambda} L \left(\frac{N}{\dot{m}} \right)^{0.2} \right] \right\} \quad -H26$$

$$\text{Also } W_p \frac{\dot{m}}{N} \Delta P V = \frac{\dot{m}^{2.8} (1 + \lambda)^{1.2} L}{2963 f^{4.8} \lambda^5 N^{1.8}} \quad -H27$$

TABLE H 2

Design Data

$N = 40$	$T_w = 90^\circ\text{C}$
$\lambda = 1.5/8 = 0.1875$	$T_1 = 10^\circ\text{C}$
$m = 0.1433 \text{ kg/s}$	$C_p = 502 \text{ J/kg/K}$
$f = 0.008 \text{ m}$	
$L = 0.076$	

Substituting known and assumed values into Equation H.10, H.14, H.15 and H.16 yields

TABLE H 3

$\phi = 0.367$
$\epsilon = 0.31$
$Q = 1770 \text{ watts}$
$W_p = 315 \text{ watts}$
$\Delta p = 7918.6 \text{ Pa } 0.08 \text{ bar}$

The heat transfer of 1770 watts equates to an electrical efficiency of 80% if it is assumed that all waste heat is transported away in the refrigerant. It will not be of course and so the predicted value of Q is thought to be more than adequate.

The pressure loss, (ΔP), across the cooling passages is predicted to be 7918 Pa - which is relatively small - however the parasitic loss (W_p) of 315 watts is considered to be significant. This aspect of the design should be re-examined for future studies.

Nomenclature

A	Area m^2
b	cooling passage breadth m
C	velocity m/s
C_p	specific heat at constant pressure
D	Diameter m
H	Fin height m
h	convective heat transfer coefficient $W/m^2/K$
K	thermal conductivity $W/m/K$
l	passage height m
L	passage length or stator length m
N	Number of passages
m	mass flow rate
ΔP	Pressure loss Pa
P	Perimeter
Q	Heat transfer W or kW
R	Radius
T	Temperature K or $^{\circ}C$
x	co-ordinate m

Greek

α	Fin augmentation factor
β	Bulk modulus
ϵ	Emissivity or Effectiveness
η_{fin}	Fin efficiency
μ	Viscosity
ρ	density
σ	Stephen-Boltsman constant
ϕ	defined by equation H19

Dimensionless Moduli

Re	Reynolds
Pr	Prandtl
Gr	Grassof
Nu	Nusselt

Sufficies

s	surface
i	inside surface
∞	ambient
D	Based on diameter
o	outer surface
H	Hydraulic diameter

TABLE H-1 COMPUTER PROGRAM: FINDRUM

>L.

```

1 MODE3
4 PRINT"*****"
5 PRINT"  PROGRAM FINDRUM I.W.EAMES  25/11/84  "
6 PRINT"*****"
20 DIM A(10),B(10),C(10)
40 FOR I=1 TO 4
50 READ A(I),B(I),C(I)
60 NEXT I
70 REM PROG. DRUMFINS
80 REM THIS PROG USES FIG2-18
90 REM IN KREITH TO CALC. THE FIN
100 REM EFFICIENCY OF A GIVEN DESIGN
110 REM AND FROM THIS THE EXPECTED
120 REM HEAT TRANSFER
140 PRINT
150PRINT
160 PRINT" TO CALC HEAT TRANSFER"
170 PRINT" THROUGH A FINNED DRUM"
180 PRINT"*****"
190 PRINT" NECESSARY INPUT DATA:-"
200 PRINT" DRUM RADIUS      MM      "
210PRINT" FIN HEIGHT      MM      "
220PRINT" FIN THICKNESS    MM      "
230PRINT" HEAT TRANS COEF. W/M^2.K"
240PRINT" CONDUCTIVE COEF. W/M.K"
250PRINT" AMBIENT TEMP.    'C      "
260PRINT" DRUM SURFACE TEMP 'C"
270PRINT
280PRINT"NOW GIVE INITIAL DATA AND"
290PRINT"PRESS RETURN AFTER EACH ENTRY"
300PRINT"*****"
310 INPUT"DRUM RAD. MM";DRAD
320 DRAD=DRAD/1000
330 INPUT"FIN HEIGHT (MM)";FINH:FINH=FINH/1000
340 INPUT"FIN THICKNESS MM";FINT:FINT=FINT/1000
350 INPUT"CONVECTIVE HEAT TRANSFER COEF. W/M^2.K";HCOEF
360 INPUT"CONDUCTIVE COEF.";CCOEF
364 INPUT"AMBIENT TEMP 'C";AMB_T
365 INPUT"FIN BASE TEMP 'C";BAS_T
370REM*****
375 PRINT"IF AXIALLY STRAIGHT FIN THEN PRESS 'S' "
376 G$=GET$:IF G$=""THEN376 ELSE GOTO 380
380REM NOW DETERMINE FIN FACTOR
390 Z1=SQR(2+HCOEF/(CCOEF*FINT*FINH))
400 Z2=(FINH+FINT/2)^(3/2)
410 Z2=Z2*Z1
415 IF Z2>2.4 THEN PRINT" FIN FACTOR OUT OF RANGE; INCREASE CONDUCTIVITY OR THICKNESS":END
420REM*****
430 REM NOW CALC. FIN EFFICIENCY
440 FR=(DRAD+FINH+FINT/2)/DRAD

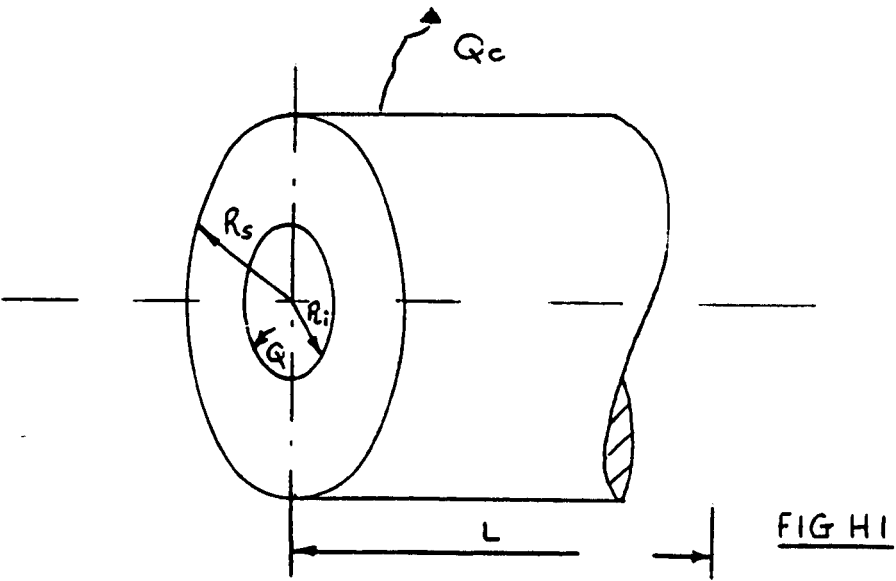
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Table H1 continued

```

445 IF G$="" THEN FR=1:MF=0
450 IF FR<1.25 AND FR>=1 THEN I=1:MF=(FR-1)/(1.25-1)
460 IF FR<1.5 AND FR>=1.25 THEN I=2:MF=(FR-1.25)/(1.5-1.25)
470 IF FR<=2 AND FR>=1.5 THEN I=3:MF=(FR-1.5)/.5
480 IF FR>2 THEN PRINT "FIN HEIGHT OUT OF RANGE;CHANGE FIN HEIGHT"
490 EA=A(I)+B(I)*Z2+C(I)*Z2^2
500 EB=A(I+1)+B(I+1)*Z2+C(I+1)*Z2^2
510 REM NOW INTERPOLATE BETWEEN EA AND EB
520 FIN_EFF=EA-(EA-EB)*MF
525 FIN_EFF=INT(FIN_EFF*10000)/10000
530 REM*****
540 REM NOW CALC. HEAT TRANS. PER FIN
550 REM*****
555 IF G$="" THEN INPUT "AXIAL LENGHT OF FIN IN MM";L3:L3=L3/1000:GOTO565
560 FRAD=DRAD+FINH:AREA=PI*(FRAD^2-DRAD^2)*2:GOTO 570
565 AREA=2*L3*FINH
570 Q1=HCOEF*AREA*(AMB-T-BAS-T)*FIN_EFF
580 Q1=ABS(Q1)
581 Q1=INT(Q1*1000)/1000
590 REM*****
600 REM          OUT PUT DATA
610 REM*****
620 VDU2
625 CLS:PRINT " FINDRUM RESULTS FOLLOW:-"
626 PRINT "-----"
627 PRINT
628 PRINT
630 PRINT "FIN HEIGHT=",FINH*1000 " MM"
640 PRINT "DRUM RADIUS=",DRAD*1000 " MM"
650 PRINT "FIN THICKNESS=",FINT*1000 " MM"
660 PRINT "'H' VALUE  =",HCOEF " W/M^2K"
670 PRINT "'K' VALUE  =",CCOEF " W/MK"
680 PRINT
690 PRINT
700 PRINT " HEAT TRANSFER PER FIN "TAB(30)=""Q1" WATTS"
710 PRINT " FIN EFFICIENCY          "TAB(30)=""FIN_EFF
720 PRINT "-----"
721 PRINT "-----"
725 VDU3
730 DATA 1.02280755,-.268275548,-1.57281154E-04
740 DATA 1.02742117,-.289254132,4.11502091E-03
750 DATA 1.03039235,-.317597334,1.04520824E-02
760 DATA 1.03876731,-.370703667,2.34841085E-02

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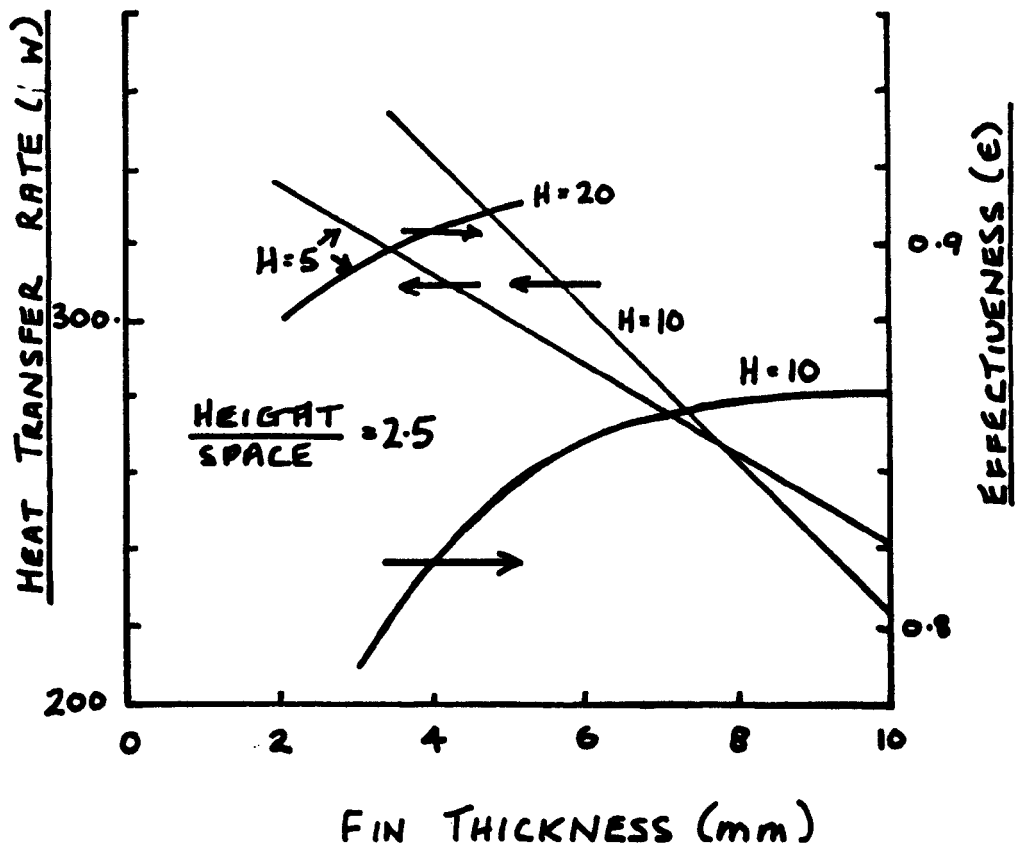


FIG H2 VARIATION IN HEAT TRANSFER
WITH FIN THICKNESS AND HEIGHT

APPENDIX IREFRIGERANT COMPRESSOR ASSEMBLY PROCEDURE (PB15031)

The following describes the procedure for assembling the experimental refrigerant compressor.

1. With the main casing supported vertically and the front face clean and dry, fit the large 'O' seal into the groove provided.
2. Locate the front bearing support housing on the front flanged face of the main casing using precision location pegs, (B1 and B).
3. Secure front bearing housing to casing using 6 x 5/16 BSF bolts, nuts and washers. It is advisable to carefully tighten each a little in turn, tightening diametrically opposite bolts in rotation. Finally torque load each bolt to 20 lbs ft.
4. Invert the casing so far assembled and clean the rear face in preparation to fitting the rear bearing assembly. Fit the large 'O' seal.
5. Locate the rear bearing support assembly on rear flanged face using the alignment tool (Pt No PC15044). It is advisable to lightly oil the alignment tool in order to prevent binding between metallic surfaces.
6. Following a similar procedure outlined in paragraph 3, secure the rear bear support assembly. Care must be taken to ensure accurate alignment of bearing housings. .
7. Re-invert the compressor so far assembled and remove the front bearing support assembly.
8. Fit bearing bushes into front and rear bearing support assemblies. With regard to the rear bearing, fitment is easiest with the motor casing laid on its side, but care must be taken to ensure that it does not roll off the bench. Care must also be taken to ensure that:
 - (i) Bushes, particularly the jewelled jets, are clean and undamaged.
 - (ii) 'O' seals are clean and undamaged and not twisted. Selective assembly of 'O' seals is necessary as the elastomer cord diameter set by the manufacturers is ± 0.003 ". It is important to select seals of similar diameter. Number 24 seals are used to support the bushes.
 - (iii) Accurate axial alignment of bushes is important, therefore a mandrel has been produced to assist this.
 - (iv) Ensure correct alignment between the lubricant inlet orifice and the annular chamber formed by the 'O' seals. The method used so far has been to measure the distance from the 3 mm dia orifice located in the support housing to some exterior reference plane, (an 'engineers' rule is quite suitable for this), push the bush into the housing a distance equal to this distance less half the overall length of the bush using the mandrel provided. Correct location of the bush can be checked by attaching a compressed air supply to the lubricant supply inlet and by placing a finger on the bush. Correct location is identified by the impingement on the finger of jets of air flowing from the jewelled orifices. Further checks are made by testing for air leakage past the

- 'O' seals by painting the outer rim (between the housing and the bush), with a quantity of any suitable volatile liquid such as 'Gramasol' degreasing agent. Leakage is easily identified by rapid 'blowing away' of the liquid in the region of the leak.
- (v) Fitment of bushes into housing is eased by wetting the 'O' seals with a suitable volatile liquid such as R.11 or 'Gramasol' degreasing agent. White spirit and similar fluids are unsuitable as they tend to leave oily deposits behind, on drying.
 - (vi) Bearing retaining clips are advisable. They will prevent axial movement of bushes when the motor is running. It is advisable to fit the rear bush forward retaining clip prior to fitting electric motor stator as it may be difficult to access the forward end of the bush afterwards.
9. Any instrumentation wiring required for the rear bearing assembly which must be fitted within the motor casing will need to be fitted at this point.
 10. The electric motor stator, and its supply cables and instrumentation wiring are now fitted in the following way.
 - (i) Fit the thermocouple leads attached to the rear half of the projecting stator windings through the holes provisioned for them at the rear of the motor casing at ten o'clock when viewed from the front. These leads should be long enough to ensure a clear working area within the motor casing unimpeded by the stator.
 - (ii) Slide the stator into the motor casing making sure that the rear-winding thermocouple leads are not trapped in the process. A step has been machined into the motor body to ensure correct axial location of the stator.
 - (ii) Rotate the stator until the power supply cables lie below the elongated slot in the motor casing. Using a pair of snipe-nosed pliers draw each cable into the slot whilst at the same time rotate the stator so that the ends of the cables are pushed through the slot and out of the motor casing.
 - (iv) Fit two 5/16 inch diameter bolts through flange holes which straddle the elongated slot. These would be impossible to fit when gas-tight terminal block is fitted.
 - (v) Fit gas-tight terminal block to motor casing using a suitable sealant between steel casing and terminal block (Hylamar has been used successfully). Fit electrical cables to brass conductors in terminal block and carry out an electrical insulation check.
 - (vi) Fit forward winding thermocouple leads through prepared holes in casing.
 11. Final check and cleaning of bearing faces is now carried out ensuring that no dust or small pieces of cleaning cloth are electrostatically adhered to the surfaces.
 12. With the motor casing so far assembled stand it vertically on a bench and with a small piece of cloth placed inside the rear housing, (so that it doesn't obstruct the bearing), lower the rotor shaft into the lower bush taking care not to damage the shaft on the stator laminations or displace the rear bush.
 13. With the shaft in position the front bearing support housing can

be fitted in a manner similar to that described in paragraph 3, taking care not to displace the bearing bush.

14. The thrust bearing is fitted into the front bearing support assembly in the following manner.
 - (i) Thoroughly clean the thrust bearing housing, thrust plates and disc ensuring that they have been degreased and that lubrication jets are clean and undamaged.
 - (ii) Assembly marks are used extensively on the thrust bearing assembly and care must be taken to ensure that it is assembled in the same way each time. Assembly marks etched onto rotating components are particularly important as they were put on prior to the rotating assembly being balanced.
 - (iii) Assembly is straightforward and should pose no additional problems. It is advised however that even tightening of the eight cap-head screws which secure the assembly to the bearing housing will avoid inadvertant misalignment of the thrust plates.
15. A large diameter circular nut fitted just behind the impeller spacer on the rotor shaft secures the thrust disc to the shaft and acts as a gas seal balancing some of the axial load on the impeller. A peg spanner has been supplied to assist in the removal and tightening of this nut and it is important to note that this nut has a "Left Hand Thread". The shaft may be prevented from rotating whilst tightening this thread by holding the shaft with a spanner on the 'flats' provided at the rear end of shaft.
16. Fitment of spacer, impeller and shroud are straightforward. Care is advised when tightening the shroud securing screws as the aluminium threads tend to fail if overtightened. Also note that the shroud back plate must be fitted before impeller.

General Comments

As gas bearings are used exclusively in the design of this compressor the need for cleanliness is emphasized, however the possibility of metal to metal seizure is highlighted by this and therefore it is recommended that those components likely to suffer from this problem are given a very light coating of grease (anti-seize) prior to fitment. One particular component which has given problems in this area is the rear bearing support assembly end cap - a light coating of grease on these threads will prevent future problems of this kind.