DESIGN OF A SINGLE STAGE CENTRIFUGAL COMPRESSOR FOR REFRIGERANTS

A DISSERTATION

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CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in the dissertation entitled "Design of a single stage certrifugal compressor for Refrigerants" in partial fulfilment of the requirement for the award of the degree of Master of Engineering in Mecharical Engineering with specializtion in Thermal Engineering submitted in the Department of Mechanical & Industrial Engineering University of Roorkee, Roorkee is an authentic record of my own work carried out for a period of about four months from Mid November 1993 to Mid March 1994 under the supervision of Dr. Bhupinder singh Reader of Mechanical & Industrial Engineering Universtiy of Roorkee, Roorkee.

The matter embodied in this dissertation has not been submitted by me for the award of any other degree.

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This is to certify that the above statement made by the Candidate is correct to the best of my knowledge.

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CONTENTS

ABSTRACT

A centrifugal compressor, like a pump is a head or pressure producing device. Performance wise, the centrifugal compressor is less efficient than the axial type. However, a much higher pressure ratio compared to the axial type and wider range of stable operation are some of the attractive aspects of centrifugal compressor. It can handle a high volume flow rate of fluid. Owing to these attractive features, centrifugal compressors are used in large refrigeration plants.

To fulfil the requirement of a large capacity Ammonia plant, this work has been done. The design of a single stage centrifugal compressor is based on one dimensional axisymmetric flow of refrigerant. The mathematical model has been prepared in chapter 4 entitled 'DESIGN METHODOLOGY'.

It is clear from the results and performance curves that a single stage centrifugal compressor is suitable for Ammonia refrigerant when pressure ratio required is low. For higher pressure ratio, multistaging has to be done.

LIST OF FIGURES

NOTATIONS

AR : Area Ratio

B : Width of Diffuser/width of Impeller

b : Axial width

C_p : Specific heat at constant pressure

CP : Pressure Recovery

D_H : Hydraulic Radius

d : Diameter

h_o : Stagnation Enthalpy

H_{ad} : Compressor Adiabatic Head

h : Static Enthalpy

L : Diffuser Length measured along the centreline.

M : Abs. Mach Number

M_r : Relative Mach Number

m : Mass Flow Rate

N_d : No. of Diffuser Vanes

Δn : Axial Clearance in Radial part

Δp : Pressure Loss

Q : Discharge

q : Heat Transfer

R : Gas Constant

R_ϵ : Degree of Reaction

r : Radius

S : Constant depending upon the material of Disc

s : Entropy per unit Mass

T : Static Temperature

T_o : Stagnation Temperature

u : Tangential Velocity

V : Absolute velocity

V_u : Whirl Component

V_{rd} : Meridional component

V_r : Relative velocity

V_{di} : Absolute velocity at diffuser inlet

W : Diffuser Throat width

w : Work done

Z : No. of vanes in an Impeller~~s~~

Greek Symbols

α : Angle between Absolute Velocity with the tangent

β : Blade Angle

γ : Specific heat Ratio

η : Efficiency

θ : Diffuser wall Angle

μ : Slip factor

ξ : Loss coefficient

ρ : Density

ν : Coefficient of viscosity

ϵ : Fraction of Blade to Blade space occupied by wake

λ : Swirl parameter

ψ : Work Input Factor

ϕ : Pressure coefficient

ω : Angular Velocity

Subscript

0 : Stagnation Value

1 : Entry to the Impeller

1h: Inducer hub

1t: Inducer tip

2 : Exit from the Impeller

2': Entry to diffuser vanes

3 : Exit from the Diffuser

4 : Exit from Volute

eq,: Average Value

i : Eye inlet

j : Jet

w : wake

l : Loss

m : Mechanical

o : Eye tip

t : Total to Total

CHAPTER - 1

INTRODUCTION

In the past two decades, there has been a substantial progress in the development of centrifugal compressor for many applications. The efficiency and operating ranges of centrifugal compressors have been improved very much by using sophisticated design method and useful test data generated in the modern research facilities. As a result, today, centrifugal compressors are widely used in small gas turbine Engines, petrochemicals and allied industries, turbochargers and in refrigeration plants. It is important to note that the compressors used in these industries come in diverse specfications with regard to their pressure levels, capacity, number of stages and the medium handled. Also, their design configuration, internal blade shape and fabrication procedures are different for differeent types of applications. For example, a small gas turbine compressor, wholly incorporates a single stage 3-dimensional impeller with an inducer whereas industrial compressor unit will have multi stage impellers. The design of a centrifugal compressor is not universal and vary with pressure ratio and application. Thus the test data obtained in a particular machine can not be applied directly for the development of any other compressor used for different application. However, it forms a basis for such designs.

1.1 ELEMENTS OF A CENTRIFUGAL COMPRESSOR STAGE

A centrifugal compressor consists of the inlet duct, the Impeller, the Diffuser and the Collector as shown in the Fig. 1.

Inlet Duct

The purpose of the inlet duct is to deliver the flow to the impeller according to the design requirement. It guides and accelerates the fluid from the inlet of the stage to the impeller eye with a minimum loss in the pressure.

Impeller

The impeller receives flow in the axial direction and delivers in the radial direction. The rotation of the impeller causes a static pressure rise in the fluid flowing through the compressor along with an increase in kinetic energy of the flowing fluid. It can be made in one piece consisting of both the inducer section and a largely curved portion. The inducer receives the flow between the hub and tip diameters of the impeller eye and passes it on to the radial portion of the impeller below. The flow approaching the impeller may be with or without swirl. In the present day compressors also, inducers are made as a part of the impeller. Generally it is assumed that there is no change in radius of any streamline passing through the inducer.

Impeller (as a whole) essentially acts as an accelerating diffuser in the sense that it increases both the kinetic energy and the pressure energy of the incoming refrigerant vapour.

Diffuser

The fluid leaving the impeller of a centrifugal machine possesses a large amount of kinetic energy. It, therefore, becomes necessary to convert this kinetic eneregy into static pressure.

In the diffuser the swirl velocity is reduced by an increase in the flow area. The diffuser system is either vaneless or vaned depending upon the application. In the vaned diffuser, vanes are used to remove swirl at a higher rate than simple vaneless diffuser thus reducing the length of flow path and diameter.

In the case of a vaned diffuser the flow enters the semi vaneless space - this is the region between the leading edge and the throat section of the diffuser and a rapid adjustment rearranges the Iso-bar pattern from parallel to perpendicular to the flow direction. If the Mach number is higher than 1.0, a shock wave system will decelerate the flow in such a way that the throat section becomes subsonic.

In the divergent channel, a further decrease of the velocity is realised with a subsequent increase of static pressure. Sometimes, the boundary layers in this channel are so weak that separation occurs which limits the static pressure rise.

Casing or Collector

Casing covers the impeller all along the periphery. The casing receives the flow all along the periphery and supply it at one point called discharge section. Section of volute is varying to accommodate the increasing flow towards discharge section. volute casing is also called collector or recuperator.

The fluid leaving the last stage of the compressor is collected by a volute or collector. The purpose of the volute is to collect the fluid discharged form the last stage and ensure its delivery to the exit flange of the machine. Normally this serves only as a collector and in the most cases never utilised as a diffusing passage.

1.2 COMPRESSION PROCESSES

Various sections of a centrifugal compressor are shown in Fig. 2.

0 - 1 Inlet duct

1 - 2 Impeller

2 - 3 Diffuser

3 - 4 Casing

Inlet Duct

The section through which flow enters the impeller is called inlet duct. A small acceleration is provided in this section to supress the irregularities of the incoming flow.

$$P_{01} = P_0 + (0.5)\ \rho_o V_o{}^2$$

P_o is the static pressure corresponding to section "0". A small acceleration is provided in the inlet duct, so that $V_1 > V_0$

$$P_{01} = P_1 + (0.5)\ \rho_1 V_1^2$$

Since the stagnation enthalpy remains the same during the flow through inlet duct, hence

$$h_{00} = h_{01}$$

$$\text{i.e } h_0 + \frac{V_o^2}{2} = h_1 + \frac{V_1^2}{2}$$

Since $V_1 > V_0$ thus $h_1 < h_0$

In actual process, the flow takes place with losses

Therefore, $P_{01} < P_{00}$

Loss of stagnation pressure = $\Delta P_0 = P_{00} - P_{01}$

This loss of stagnation pressure depends upon change of entropy.

Impeller : 1 - 2

In the radial flow machines the flow remains in the radial plane for most of the time, although it enters the machine in the axial direction.

Applying steady flow energy equation between sections 1 and 2, we get :

$$h_1 + \frac{V_1^2}{2} + q = h_2 + \frac{V_2^2}{2} + w \text{ (for unit mass flow rate)}$$

$$\Rightarrow h_{01} + q = h_{02} + w$$

Assuming adiabatic process between 1 and 2, we get

$$h_{01} = h_{02} + w$$

Since work is done on the impeller, w is -ve

Finally

$$-w = h_{01} - h_{02}$$

i.e. $w = h_{02} - h_{01}$

Euler's equation of turbomachinery gives

$$w = 0.5\ (V_2^2 - V_1^2) + 0.5\ (u_2^2 - v_1^2) - 0.5\ (V_{r2}^2 - V_{r1}^2) \qquad \ldots(1A)$$

(a) (b) (c)

Term (b) will always be present as the radius varies from 1 to 2. In most of the cases $V_{r1} \# V_{r2}$. Thus all the three components contribute towards work transfer in a radial machine.

Work is imparted by the impeller to the fluid which raises its pressure from P_1 to P_2. Point 2S represents the isentropic process state whereas 2 represents the actual process state.

Kinetic Energy corresponding to state 2 is $V_2^2/2$

$$P_{02} = P_2 + (0.5)\ \rho_2 V_2^2$$

$(P_{02} - P_{01}) = \Delta P_{01-02}$ is the increase in the stagnation presssure. This is not equal to a loss as a work is done during process 1-2.

$(P_{02S} - P_{02})$ is the loss of stagnation pressure because of losses taking place in the actual process. This loss depends on the increase in entropy ΔS_{1-2}.

Diffuser : 2-3

This is a stationary element, no work transfer takes place here, thus

$$h_{02} = h_{03}$$

$$h_2 + (0.5)\ V_2^2 = h_3 + (0.5)\ V_3^2$$

In a diffuser $V_3 << V_2$ i.e. kinetic energy is converted into potential energy. This process taking place in a diffuser is only energy transformation. Hence, $h_3 > h_2$.

A Diffuser can be avaned diffuser or a vaneless diffuser depending upon the application.

Collector : 3-4

Volute casing can also be made to serve as an additional diffuser, Thus converting the kinetic energy into pressure. It is also a stationary element in which no energy transfer as a work takes place.

In cases where some kinetic energy is converted into pressure, then $V_4 < V_3$ which implies $h_4 > h_3$.

In the casing a large mixing of flow takes place as flow is collected all along the periphery, thus diffusion is generally avoided as diffusion with mixing of flow leads to greater fluid losses.

1.3 DEGREE OF REACTION OF A CENTRIFUGAL COMPRESSOR

The degree of reaction is a parameter which describes the relation between the energy transfer due to static pressure change and energy transfer due to dynamic pressure change. According to Shepherd[25], the degree of reaction R_ε is defined as the ratio of energy transfer by means of or resulting in a change of static pressure in the rotor to the total energy transfer in the rotor

$$R_\epsilon = \frac{(u_2^2 - u_1^2) - (V_{r2}^2 - V_{r1}^2)}{(v_2^2 - v_1^2) + (u_2^2 - u_1^2) - (V_{r2}^2 - V_{r1}^2)}$$

From Euler's equation of turbomachinery, we have

$$\left[\frac{V_2^2 - V_1^2}{2}\right] + \left[\frac{u_2^2 - u_1^2}{2}\right] - \left[\frac{V_{r2}^2 - V_{r1}^2}{2}\right] = u_2 V_{u_2} - V_1 V_{u_1}$$

$$\text{Therefore } R_\epsilon = \frac{\left[\frac{u_2^2 - v_1^2}{2}\right] - \left[\frac{v_{r2}^2 - v_{r1}^2}{2}\right]}{2(u_2 V_{u2} - u_1 v_{u1})} \qquad \ldots(1B)$$

CHAPTER - 2

LITERATURE REVIEW

A large volume of literature is available for the design and performance of industrial as well as gas turbine axial flow compressors. The centrifugal compressors, though used extensively are still less developed, most of the information available is in the form of correlations and potential theories. In certain cases, these theories do not match with the experimental results and fail to explain the flow phenomenon. In the recent years renewed interest in the gas turbine and the refrigeration fields has emphasized the need for higher efficiency and wider range of stable operation of centrifugal machines.

2.1 FLOW THROUGH THE INDUCER

Balje[4] has conducted experiments regarding maximum obtainable efficiency at different specific speeds. He concluded that the maximum obtainable efficiency drops rapidly towards lower specific speeds and more gradually towards higher specific speeds. The optimum hub ratio for maximum efficiency regions is about 0.6 and increases with decreasing specific speeds, but decreases with increasing specific speeds. The variation of the compressor adiabatic efficiency η_{ad} with the specific speed N_s is depicted in Fig. 4. Where

$$N_s = (N\sqrt{Q}) / (Had)^{0.75} \quad \& \quad Had = CpT_{01}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

Shephered[25] gave a general method of calculating the optimum inlet design applicable to the compressible flow in both axial and centrifugal compressors. He derived the following equation

$$\frac{4\pi\, \dot{m}\, N^2}{3600K\, P_{01}\, (\gamma R T_{01})^{1/2}} = \frac{M_{r1}^3 \cdot \sin^2 \beta_{1t} \cdot \cos \beta_{1t}}{\left\{1 + \frac{1}{2}(\gamma-1)\, M_{r1}^2 \cos^2 \beta_{1t}\right\}^{1/(\gamma-1) + \frac{3}{2}}} \qquad \ldots (2A)$$

From this equation it is clear that by specifying a particular value of M_{r1} as a limit, the optimum value of β_{1t} for the maximum mass flow can be found. He gave a comparison between blade angle at inlet and Non-dimensional mass flow parameter as shownin Fig. 5.

Mizuki, s, et al [21] examined the flow patterns and loss production mechanism within the centrifugal impeller channels at the design point conditions. They designed various impellers having different values of diffusion ratios of 0.6, 1.0 and 1.4. These impellers had twelve straight radial blades.

Dean[8] predicted that the maximum attainable diffusion ratio for sub-sonic inducers is around 1.7.

2.2 FLOW THROUGH THE IMPELLER

Dean and Senoo[7] worked extensively in the field of centrifugal compressors and developed various models linked with the experimental results. One such an approach which approximately simulate the real flow at impeller outlet is jet and wake model. Fig. 6 shows schemetically the jet and wake model. To find a relation between flow conditions at the separation point and the impeller exit, it is assumed by Dean that the jet flow is subjected to small shear stresses in the rotor and can be considered as an isentropic core. All the flow outside this core can be considered as belonging to the wake. This is even true when there is a tangential pressure gradient over the jet, due to the rotor work creating secondary flow that moves all the low energy fluid to the suction side. For a compressible flow, they showed that the mean relative Mach Number remains constant along the jet from the separation point to the impeller exit.

In developing Euler equation it is assumed velocities in turbomachines could be represented by single vectors which remains constant over the given cross-sectional area. This assumption is incorrect. The effect of discrepancy between the assumed and actual conditions is more noticeable in centrifugal machines, where energy transfer causes a pressure change across the Vane with a high pressure at the leading face and a low pressure at the trailing face. The lower pressure at the trailing face results in a higher speed of fluid flow compared with that at the leading face. As a result, the fluid leaves tangentially only at the high pressure face and nowhere else. At other points the direction of this velocity vector differs from that of the exit blade angle. The mean exit angle β_2', is thus less than the vane exit angle β_2 as shown in Fig. 7. The change in flow direction from one vane side to another may also be attributed to the action of coriolis forces. The reduction in whirl velocity V_{u_2} as compared with that in an ideal machine is called slip.

A number of empirical relations have been published on slip factor. The Fig·8depicts the flow model with slip. Assuggested by Stodola[27] for the slip factor μ, we have the following.

$$\mu = 1 - \frac{\pi}{z}\left[\frac{\sin \beta_2}{1 - \phi_2 \cot \beta_2}\right] \qquad \ldots(2B)$$

Weinsner[30] has given the following equation for the slip factor

$$\mu = \left[\frac{\sqrt{\sin \beta_2}}{(Z)^{0.7}}\right] \qquad \ldots(2C)$$

A comparison on μ by Weisner, Kearton[18] and Stodola is shown in Fig. 9. Whitefield[29] using the well established jet and wake model of Dean calculated overall slip factor for three impellers with different number of radial blades at varying flow rates. He compared the predicted results with the experimental results and found that the method accurately predicts the change of slip factor with the flow rate.

Eckardt[11] with the help of a LASER velocitymeter measured the velocity, its directions and fluctuating intensities in the internal flow field of a radial discharge centrifugal impeller.

He conducted experiments at the tip speeds upto 400 m/s, presented relative flow distributions for five measurement planes from inducer inlet to the impeller discharge. He concluded that the potential theory is fairly well suited to predict the undistributed flow pattern within the axial inducer and partially in the impeller.

2.3 FLOW THROUGH THE DIFFUSER

2.3.1 Flow in vaneless Diffuser :

A vaneless diffuser is an annular space usually of constant width that surrounds the impeller. Analysis of vaneless space shows that under the influence of friction the flow turned more towards the tangential direction with increasing radius to satisfy the vortex and the continuity reactions.

Dynamic head loss caused by the side wall frictionis dependent upon the path length up to the throat and the local velocity. The effect of loss can be expressed in terms of the total pressure recovery by the following relation

$$\text{Vaneless Space Recovery} = \frac{\text{Total pressure (Local)}}{\text{Total pressure at impeller tip}}$$

The relationship between total pressure recovery and vaneless space diameter ratio, Mach number and flow angle indicates that the loss can be minimised by designing for a small impeller discharge angle and diameter ratio. But for the small values of these parameters, the Mach number at vane leading edge will be higher and resulting shock system would be stronger. In addition, because the flow mixes as it leaves the impeller, a small diameter ratio should be avoided.

Dean and Senoo[7] analytically showed that the stagnation pressure loss beomes larger and the static pressure rise becomes smaller as the diffuser width reduces. They showed that the inlet flow distortion affects the flow behaviour of the vaneless diffuser. The magnitude of influence depends upon the impeller discharge angle, the wall friction and the geometry of blades, etc. They found that the total pressure loss in a diffuser may be less with inlet distortion than for the case with no inlet distortion. They also demonstrated that the frictional loss for asymmetric flow at the inlet of the diffuser was considerably larger than that for the equivalent symmetric flow. They also demonstrated that the asymmetric flow pattern becomes uniform by the reversible work exchange between the shear lines.

Johnson and Dean[16] developed two analytical methods based on a one-dimensional axisymmetric frictional loss analysis and a sudden expansion mixing technique to predict the loss in a vaneless diffuser. They compared their analysis which the experimental data and found good agreemnt between their predictions and actual losses. They found that the total pressure loss as predicted by Dean and Senoo[7] was equal to the sum of a simple mixing loss of an assymetric flow and the wall frictional loss of the equivalent symmetric flow.

$$\text{Mixing loss } (\Delta p_1)_m = \frac{1}{2}\rho_2\left(\overline{V_2}\right)^2 \left[\frac{1}{1+\lambda_2^2}\right]\left[\frac{(1-\epsilon) - B}{(1-\epsilon)}\right] \quad \ldots(2D)$$

$$\text{Frictional loss } (\Delta p_1)_f = \frac{1}{2}\rho_2(\bar{V}_2)^2 \left[1 - C_p - \left[V/\bar{V}_2\right]^2\right] \quad \ldots(2E)$$

Rustandler and Dean[22] showed that an increase in the area ratio of the diffuser resulted in an increase in pressure recovery. This situation is shown in the figure 10 where

$$AP = 1 + (2L/W_1) \quad \ldots(2F)$$

$$CP = \frac{P_e - p_t}{P_{ot} - p_t} \quad \ldots(2G)$$

2.3.2 Flow Through the Vaned Diffuser :

At high pressure a large amount of kinetic energy is present in the flow at the exit of impeller. To effectively convert this kinetic energy to pressure rise, vaned diffuser is a mandatory. A vanned diffuser guides the flow through the channels and imposes a torque on the flow by means of the vanes which are close to the impeller thereby reducing the angular momentum more rapidly than a vaneless diffuser

Sherztyuk and others [23] studied a wide variety of vane shapes and found that a wedge shaped diffuser vanes yield better efficiency in comparison to the other type of diffusers tested by them. Also the diffuser peak efficiency operating point for this type of diffuser was considerably away from the surge and choke point. They also found no efficiency gain by narrowing the diffuser passage.

Fisher[14] observed that a diffuser with 17 vanes was two to four percentage more efficient than a diffuser with 11 vanes. Fisher also found that a circular arc vaned diffuser gives a better performance in the case of high pressure ratio compressors than either the twisted vaned diffuser or the simple wedge vaned diffuser. However, for the case of low pressure ratio compressor, a simple wedge vaned diffuser seems to produce better performance.

The diffusing action on the fluid leaving the impeller is due to the small clearance space between the impeller and the surrounding diffuser vanes, Where velocity is reduced due to an increase in available flow area, though the angular momentum of the flowing fluid is constant here. This space should be so large that Mach number of the fluid entering the diffuser vanes is less than about 0.9 for the shockless entry. For this Lee, J.F.[20] suggested that the radius at the inlet to the diffuser vanes should be about 1.1 to 1.4 times that at the impeller exit.

Kenny[17] had pointed out from his experimental investigation that the boundary layer blockage in the throat section is the most important parameter for determining the diffuser channel behaviour.

2.4 LOSS MODELS

The calculation of the energy dissipated as the fluid passes through the impeller is based upon empirical loss correlations. The earlier work of Balje was further developed by the researchers to provide a general performance prediction procedure for centrifugal compressors.

Frictional Loss

Frictional losses within the impeller are dependent upon the absolute velocity along the shroud and the relative velocity along the blade hub.

Clearance Loss

Clearance loss result from flow leakage from the pressure to the suction side through the finite clearance between the stationary shroud and the rotating blades.

Exit Mixing Loss

Exit mixing loss is related to the separation inside the impeller passage. When the flow leaves the impeller tip and enters the vaneless space, the jet and wake mix. The mixing loss is evaluated as a sudden jump of the radial vector. The radial vector is influenced by the exit jet Mach number, the ratio of wake width to jet width at the impeller tip and the angle of the flow leaving the impeller. Because the exit mixing loss can be a large part of the total pressure loss. It is important in any impeller design to avoid early separation.

2.4.1 Losses in Impeller

Leiblein[19] derived a correlation for assessing the diffusion losses in axial compressors. This diffusion and blade loading loss was derived for a blade loading condition in which the relative velocity difference between suction and pressure surface increases linearly along the blade length. He concluded that the diffusion loss is a function of diffusion ratio. His empirical relationis given by the following equatiion.

$$W_D = \left[\frac{2\sigma}{\cos\beta_2}\right]\left[\frac{\cos\beta_1}{\cos\beta_2}\right] f\ (DF) \left[\frac{V_{r1}^2}{u_2^2}\right] \qquad \ldots(2H)$$

$$\text{where } \sigma = \text{solidity} = \left[\left[\frac{\pi d_m}{z}\right]\left[\cos\frac{\beta_{1m}}{2}\right]\right] / (R_2 - r_{1t} + b_2)$$

$$DF = \left[1 - \frac{\cos\beta_1}{\cos\beta_2}\right] + \left[\frac{\cos\beta_1}{2\sigma}\right] (\tan\beta_1 - \tan\beta_2)$$

It is assumed that the complete passage diffusion is achieved in the axial portion i.e. inducer itself. The above equation is valid for axial portion of the inducer only.

Dalley and Nece[9] formulated a relation for the disc friction loss which is given by

$$W_{DD} = \xi \left[\frac{2pr\ w^2\ d_2^4}{16}\right] (b_2 + 0.1\ d_2) \left[\frac{1}{m}\right] \qquad \ldots(2I)$$

$$\xi \quad = 5c/(4\pi\ R_{eN}^{\ 0.2}) \quad ; \ R_{eN} = (w\ d_2^2/4\nu)$$

$$C \quad = 0.102\ (2S/d_2)^{0.1}$$

Jansen[15] gave the following empirical relation for the back flow loss

$$W_{BF} = 0.02\ (DF)^2 \sqrt{\text{Tan}\alpha_2} \qquad \ldots(2J)$$

Jansen[15] developed the relationship for the clearance loss considering the flow through the clearance gap to undergo a sudden contraction followed by a sudden expansion.

$$W_{cl} = \frac{0.6\ \Delta n}{b_2} \frac{V_{u2eq}}{u_2} \left\{ \frac{2\pi\ (d_{1t}^2 - d_{1h}^2)}{b_2\ z\ (d_2 - d_{1t})} (1 + \frac{\rho_2}{\rho_1}) \frac{V_{u2eq}}{u_2} \frac{V_{m1}}{V_{u2}} \right\}$$

$$\ldots(2K)$$

Wher Δn is the axial clearance in the radial part. For the skin friction loss he gave the following empirical relation.

$$\Delta q_{st} = 4C_f\ L\ \bar{V}_r^2/2Du_2 \qquad \ldots(2L)$$

where $\bar{V}_r\ (V_{r1} + V_{r2})/2$

$$L = \text{Flow path length} = \frac{0.25\pi \left\{d_2 - 0.5\ (d_{1t} + d_{1h} - b_2 + 2L_{ax}\right\}}{\left[0.5\ (\cos\beta_{1t} + \cos\beta_{1h}) + \cos\beta_2\right]}$$

$$L_{ax} = 0.5\ (d_2 - d_{1t}) + b_2$$

$$C_f = 0.3164 \left[\frac{\bar{V}_r \cdot D_H}{\nu}\right]^{0.25}$$

$$\frac{1}{D_H} = \left\{\frac{\pi\ d_2 \cos \beta_2 + b_2 z}{4\ \pi\ d_2\ b_2 \cos \beta_2}\right\} + \left\{\frac{\pi\ (d_{1h} - d_{1t}) \cos \beta_{1m} + z\ (d_{1t} - d_{1h})}{2\pi\ (d_{1t}^2 - d_{1h}^2) \cos\ \beta_{1m}}\right\}$$

2.4.2 Losses in Vaneless Space

The mixing loss due to sudden expansion is given by Dean and Senoo(7) as

$$W_{mix} = 2(\Delta p_o)_{mix} / \rho\ \bar{V}_2^2 = 2\left[\frac{1}{1 + \lambda_2^2}\right] \left[\frac{1 - \epsilon - B}{1 - \epsilon}\right] \qquad \ldots (2M)$$

where B = width of Diffuser/width of Impeller

$$\lambda_2 = V_{u2}/V_{m2}$$

$$V_{u2} = \lambda\ (V_{u2})_w + (1 - \lambda)\ (V_{u2})_j$$

$$\bar{V}_m = \epsilon\ (V_{m2})_w + (1 - \epsilon)\ (V_{m2})_j$$

$$\bar{V}_2 = \bar{V}_{u2}^2 + \bar{V}_{m2}^2$$

similar equation has been predicted by them for the fricteonal loss as

$$W_{fv} = 2\ (\Delta p_o)_{fv} / p_2\ \bar{V}_2^2 = 1 - Cp - \left[\frac{V}{\bar{V}_2^2}\right]^2$$

CHAPTER 3

PROPOSED WORK

The Real flow through a centrifugal compressor is essentially three dimensional, viscous and unsteady. Till now, there is no mathematical model which predicts the flow in such a machine without compromising on some of the important flow phenomena, such as separation. Mach number influence boundary layer blockage, etc. Nevertheless, there is an urgent need for a design method which allows to predict compressor characteristics and to investigate the influence of various parameters on its performance because the centrifugal compressor is suitable for the refrigeration plants whose capacity exceeds 100 ton refrigeration.

The scope of the present work is to develop a suitable mathematical model on the basis of one dimensional axisymmetric flow of the refrigerant through the compressor to know the influence of various parameters on its performance. An attempt will also be made to draw compressor characteristics and approximate shape of volute casing.

CHAPTER - 4

DESIGN METHODOLOGY

4.1 Imepller Design

Since no work is done on the refrigerant vapour in the diffuser the energy absorbed by the compressor will be determined by the conditions of the refrigerant vapour at the inlet and outlet of the impeller. The energy absorbed by the impeller is given by Euler's equation of the turbomachinery as

$$w = m\,(V_{u_2}\,u_2 - V_{u_1}\,u_1) \qquad \text{...(i)}$$

The equation (i) shows that the head developed in the impeller is reduced because of tangential prewhirl component of the velority V_{u_1} in the direction of rotation of the impeller at the inlet. With the selection of an appropriate inlet blade angle β_i the term V_{u_1} can be reduced to 0 i.e. the refrigerant vapour enters the impeller eye in axial direction. Taking this condition into account, we have

$$w = m\,(V_{u_2}\,u_2) \qquad \text{...(ii)}$$

The exit kinetic energy per unit Mass $(V_2^2/2)$ increases quite rapidly as β_2 increases, consequently an extremely efficient diffuser is needed to obtain a pressure rise utilizing the kinetic energy atthe impeller exit since most diffuser efficiencies are low because of adverse pressure gradient and the resulting thick boundary layer hence a complete utilization of the kinetic energy with a corresponding pressure rise is impossible. It is therefore to be expected that machines with forward curved vanes (large exit angle > 90^o) will be less efficient.

Further a problem is faced with the backward curved vane also. For a machine with a specified speed absorbing a specified amount of energy the diameter and V_{u_2} can be varied to maintain V_{u_2} u_2 at the required value. Hence if β_2 is large, V_{u_2} is small similarly if β_2 is small the diameter should be enlarged to give the same output. For these reasons, machines with backward curved blades are larger in size than those with radial and forward curved baldes of the same capacity and thus involving a huge expenditure of the material. Hence radial blades are used if the machine of small size is to be used. At the same time it is capable of developing a large pressure ratio.

For radial vanes at the exit $V_{u_2} = u_2$ if the effect of the slip is neglected However, due to inertia, the vapour trapped between the impller vanes is reluctant to move round with the impeller and this results in a higher static presssure on the leading face of a vane than on the trailing face. It also prevents the air from acquireng a whirl velocity equal to the impeller speed. This effect is known as slip. Howear the whirl velocity at the impeller tap falls short of the tip speed depends largely upon the number of vanes on the impeller. The greater the number of vanes, the smaller the slip i.e. the more nearly V_{u_2} aproaches u_2.

According to Weisner, F.[30] the slip faactor μ depends upon the exit vane angle and the number of blades Z. A broad review on the values of slip factor μ has been given in the form of the followling equation by him as

$$\mu = 1 - \sqrt{\frac{\sin\beta_2}{Z^{0.7}}} \qquad \text{...(iii)}$$

For the radial blades $\beta_2 = 90^0$ in equation (iii) , we get

$$\mu = \frac{(Z^{0.7}-1)}{Z^{0.7}} \qquad \text{....(iv)}$$

Putting $V_{u_2} = \mu u_2$ in (ii) and then subsequently putting the value of μ from (iv), we get the following expression for work w

$$W = m^{\cdot}\mu u_2^2 = m \frac{(Z^{0.7}-1)}{Z^{0.7}} u_2^2 \qquad \text{...(v)}$$

The static pressure rise caused by the centrifugal compressor is due partly to the action of the impeller and partly to the action of the diffuser while the stagnation pressure rise is caused entirely by the impeller reduced by losses in the diffuser. Hence on making an energy balance between the impeller inlet and diffuser outlet (Fig. 2), it is seen that

$$\Delta h_0 = (h_3 + \frac{V_3^2}{2}) - (h_1 + \frac{V_1^2}{2}) = h_{03} - h_{01}$$

$$\text{i.e. } \Delta h_0 = C_p(T_{03} - T_{01})$$

If the process were reverrible, the final stagnation state would be 03'. However the actual process is not isentropic due to the losess, the final state reached will be T_{03} (shown in Fig. 3). This fact is taken care by introducing compressor total to total isentropic efficiency η_t which is usually limited to 80%.

$$\eta_t = \frac{T_{03}' - T_{01}}{T_{03} - T_{01}}$$

$$\text{i.e. } T_{03} - T_{01} = \frac{T_{03}' - T_{01}}{\eta_t} = \frac{T_{01}(\frac{T_{03}'}{T_{01}} - 1)}{\eta_t} \quad \ldots\text{(vi)}$$

The overall stagnation pressure ratio is given by

$$\frac{P_{03}}{P_{01}} = (\frac{T_{03}'}{T_{01}})^{\gamma/\gamma-1}$$

$$\text{i.e.} \frac{T_{03}'}{T_{01}} = (\frac{P_{03}}{P_{01}})^{\gamma-1/\gamma} = (r_p)^{\gamma-1/\gamma} \quad \ldots\text{(vii)}$$

Putting the value of $\left(\frac{T_{03'}}{T_{01}}\right)$ from (vii) in (vi), we get

$$T_{03} - T_{01} = T_{01} \frac{[(r_p)^{\gamma-1/\gamma}-1]}{\eta_t}$$

$$\text{i.e.}\quad T_{03} = T_{01}\left[1 + \frac{[(r_p)^{\gamma-1/\gamma}-1]}{\eta_t}\right] \qquad \ldots(1)$$

Due to friction between the casing and the refrigerant vapour carried round by the vanes and other losses which have a braking effect such as disc friction or windage, the actual work input is greater than the theoretical value calculated from equation (v). A power input factor ψ can be introduced to take account of this, so that the actual work done on the vapour is

$$w = m^{\cdot}\left(\frac{Z^{0.7}-1}{Z^{0.7}}\right)\psi\, u_2^2 \qquad \ldots(\text{viii})$$

If $(T_{03} - T_{01})$ is the stagnation temperature rise across the whole compressor, them since no energy is added in the diffuser, this must be equal to the stagnation temperature rise $(T_{02} - T_{01})$ across the impeller alone. Therefore the following relation holds good

$$m^{\cdot}C_p(T_{03} - T_{01}) = m^{\cdot}\left(\frac{Z^{0.7}-1}{Z^{0.7}}\right)\psi\, u_2^2$$

$$\text{i.e.}\quad u_2 = \left[\frac{Z^{0.7}C_p}{(Z^{0.7}-1)\psi}(T_{03} - T_{01})\right]^{1/2} \qquad \ldots(2)$$

Where Cp is the mean specific heat over this temperature range. Typical values for the power input factor lie in the range of 1.035 - 1.04.

The rotational speed at which the compressor works is an important parameter in determining the safe stress level for its operation. It is usually set by the designer before determining the tip diameter of the impeller. A small diameter is helpful in producing a high rotational speed without undully increasing the hoop stress developed in the impeller material. The value of the hoop stress is maximum at the impeller tip.

The three parameters u_2, d_2 and N are related by the following equation

$$\text{i.e. } d_2 = \frac{60u_2}{\pi N} \qquad \ldots(3)$$

Where N is the rotations per minute of the impeller known The hoop stress developed in the impeller is

$$\sigma_h = 0.1\rho u_2^2 = 0.1\,\frac{\rho\pi^2 d_2^2 N^2}{3600} \qquad \ldots(4)$$

The equation (4) serves as a check equation for the calculated value of d_2 from the equation (3). If the hoop stress exceeds the permissible stress of the material the calculation should start from new values of pressure ratio and RPM.

The isentropic stagnation state is defined as the state a fluid in motion would reach if it were brought to rest isentropically in a steady-flow, adiabatic and zero work output proess. The stagnation enthalpy at state 1 is related to the static enthalpy at the same state by the following expression

$$h_{01} = h_1 + \frac{V_1^2}{2} \qquad \ldots(ix)$$

Puting $V_1 = M_1C_1$ where C_1 is acoustic velocity corosponding to state 1 and $h = C_pT$ we get

$$T_1 = \left[\frac{T_{01}}{1 + (\frac{\gamma-1}{2})M_1^2}\right] \qquad \ldots(5)$$

From equation (ix) $Cp(T_{01} - T_1) = \frac{V_1^2}{2}$

$$V_1 = \left\{2C_p(T_{01} - T_1)\right\} \qquad \ldots(6)$$

The static pressure P_1, the stagnation pressure P_{01}, The static temperature T_1 and the stagnation temperature T_{01} are related to one another by the equation

$$P_1 = P_{01}(\frac{T_1}{T_{01}})^{\gamma/\gamma-1} \qquad \ldots(x)$$

Applying the perfect gas equation at state 1, we get

$$P_1 = \rho_1 R T_1 \qquad \ldots(xi)$$

Putting the value of P_1 from (x) in (xi), we get

$$\rho_1 = \frac{P_{01}}{RT_1}\left(\frac{T_1}{T_{01}}\right)^{\gamma/\gamma-1} \qquad \ldots(7)$$

According to Lee J.F.,[20] it is necessary that (d_2/d_o) be greater than about 1.5 in order to provide a sufficient amount of space to accelerate the flow from the inlet to the exit of the impeller. Otherwise, too low efficiency will result. Values of (d_2/d_o) around 2 are considered as necessary for high efficiency.

Where, d_2 = Tip diameter of the impeller

d_o = Tip diameter of the eye

Hence $d_o = 0.5\, d_2 = d_2/2$ ... ... (8)

To find the inlet angle of the vanes it is necessary to determine the inlet diameter of eye. This can be determined by applying the continuity equation.

$$m^{\cdot} = \rho_1 A V_1 = \rho_1\ \pi/4(d_0^2 - d_i^2)V_1$$

$$\text{Hence } d_i = \left[d_0^2 - \frac{4m^{\cdot}}{\pi\rho_1 V_1}\right]^{1/2} \qquad \ldots(9)$$

Therefore the maximum tangential speed at the tip of the eye

$$u_{1t} = \frac{\pi d_o N}{60}$$

And the maximum relative velocity at the tip of the eye is

$$V_{r_{1t}} = (u_{1t}^2 + V_1^2)^{1/2} \qquad \ldots(10)$$

It is earlier mentioned that through axial entry of the refrigerant vapour i.e. $V_{u1} = 0$, the work capacity of the compressor is increased. This requires that the absolute velocity of the refrigerant vapour entering the compressor should vary from root to tip of the impeller. Then the tangential speed u becomes maximum at the tip of the impeller and it is quite possible that $V_{r_{1t}}$ becomes so large as to make the Inlet Mach Number beyond unity. This must be avoided while the compressors are dealing with the high molecular weight vapours like refrigerants. Even when this Mach Number is less than unity, it is possible that the local Mach Number can become large at some point within the passage and choke the flow. In order that no choking occurs, it is essential to maintain the flow Mach Number at the rotor inlet tip below 0.9.

$$\text{i.e. } M_{r1t} = \frac{V_{r_{1t}}}{\sqrt{\gamma R T_1}} \qquad \ldots(11)$$

should be less than 0.9. The equation (11) acts as a check equation for the calculated values of V_1, T_1, ρ_1, etc. If Mr_1 goes beyond 0.9, we must iterate by taking new values of N, r_p, M_1, etc.

It is now possible to find the blade angles at the hub and the tip of the eye.

$$\left.\begin{aligned} \text{At the hub, } \beta_{1i} &= \tan^{-1}\left(\frac{60V_1}{\pi d_i N}\right) \\ \text{At the tip, } \beta_{10} &= \tan^{-1}\left(\frac{60V_1}{\pi d_o N}\right) \end{aligned}\right\} \qquad (12)$$

The angle between the velocity V_2 at the impeller exit and the wheel tangent must be determined to fix the exit velocity triangle. This angle varies between 12^o and 25^o for the smooth shockless flow from the impeller exit to the diffuser vane inlet.

$$V_2 = V\mu_2/\text{Cos}\ \alpha_2 = \mu U_2/\text{Cos}\ \alpha_2 \qquad \ldots \quad (xii)$$

$$V_{f2} = V_{rd_2} = V_{u2} \tan\alpha_2 = \mu u_2 \tan\alpha_a \qquad \ldots \ (xiii)$$

To complete the impeller design, it is necessary to compute the axial width of the impeller. This can be done if the static temperature and pressure at the exit are known. Since $To_2 = To_3$, it is seen that

$$T_2 = T_{o2} - \frac{V_2^2}{2cp} \qquad \ldots (xiv)$$

putting the value of V_2 from equation (xii) in (xiv), we get

$$T_2 = T_{o2} - \frac{\mu^2 u_2^2}{2c_p \cos^2 \alpha_2}$$

But $T_{o2} = T_{o3}$, therefore $T_2 = T_{o3} - \dfrac{\mu^2 u_2^2}{2c_p \cos^2 \alpha_2}$ ...(13)

The computation of the static pressure at the exit of the impeller necessitates a knowledge of the isentropic static temperature T_3, at the diffuser exit. To find this we note that

$$T_3 = T_{o3} - \frac{V_1^2}{2c_p} \qquad \text{...(14)}$$

Further static pressure P_3, stagnation pressure P_{o3} are related to one another by the following equation

$$P_3 = P_{o3} \left(\frac{T_3}{T_{o3}}\right)^{\gamma/\gamma-1} = (r_p)\ P_{o1} \left(\frac{T_3}{T_{o3}}\right)^{\gamma/\gamma-1} \qquad \text{...(15)}$$

If T_3, be the actual static temperature at the exit of the diffuser, then we define the diffuser efficiency as

$$\text{Diffuser Efficiency } \eta_d = \frac{T_3' - T_1}{T_3 - T_1}$$

$$\text{i.e } T_3' = T_1 + \eta_d + (T_3 - T_1) \qquad \text{...(16)}$$

Experiments with subsonic diffusers of various divergence angles have shown that for the flow of liquid water, the highet diffuser efficiencies are obtained when the angle of divergence is about 6^o - 8^o. As the divergence angle increases, the diffuser efficiency drops, being highest for circular diffusers and lowest for the square shaped diffusers[14]. Usually circular diffusers have efficiencies around 60% - 70% though carefully designed diffusers can exhibit higher efficiency.

Then $P_2 = P_3\ (T_2/T_3')^{\gamma/\gamma-1}$

Applying perfect gas equation at the state 2, we get

$$\rho_2 = P_2/RT_2 = \left[\frac{P_2}{RT_2}\right]\left[\frac{T_2}{T_3'}\right]^{\gamma/\gamma-1} \qquad \ldots(17)$$

The axial width of the impeller can be calculated by Applying Continuity equation.

$$m^{\cdot} = \rho_2 A\ V_{f2} = (\pi\ b_2 d_2)\ \rho_2\ V_{f2}$$

$$\text{i.e } b_2 = \frac{m^{\cdot}}{\pi d_2\ V_{f2}\ \rho_2} \qquad \ldots(\mathrm{xv})$$

The actual impeller width has to be larger, since the rotor blades cause a reduction in flow area of about 7% to 10%. Considering this, the actual flow area = 0.90 $(\pi b_2 d_2)$. Taking this fact into account, we get, the following expression for the impeller width

$$b_2 = \frac{\dot{m}}{(0.90)\ \pi\ d_2\ v_{f2}\ \rho_2} \qquad \text{...(18)}$$

4.2 DIFFUSER DESIGN

The diffusing action on the fluid leaving the impeller is due to the small clearance space between the impeller and the surrounding diffuser vanes, where the velocity is reduced due to an increase in the available flow area Though the angular momentum of the flowing fluid is constant here. This space should be so large that the Mach Number of the fluid entering the diffuser vanes is less than about 0.9 for the shockless entry. It is usual to maintain the radius at the inlet to the diffuser vanes about 1.1 to 1.4 times that at the impeller exit.

$$d_2' = 1.2\ d_2 \qquad \text{...(19)}$$

The flow in the clearance space is free-vortex type i.e. $V_u r$ = constant. Therefore the tangential velocity of the fluid entering the vane is given by the boltowing expression

$$V_{u2}' = \frac{d_2\ V_{u2}}{d_2'} \qquad \text{...(20)}$$

The value of V_{u2}' should be such that the Mach Number at the diffuser entry must be less 0.9. Mach Number at the diffuser entry is

$$M_2' = \frac{V_{u2}'}{(\gamma R T_2)^{1/2}} \qquad \text{...(21)}$$

The equation (21) acts as a check equation for the calculated value of V_{u_2}' from equation (20). If M_2' exceeds 0.9, then calculation should, restart from equation (19) by taking new value of d_2'

In order to determine the inlet angle of diffuser vanes, it is necessary to know the velocities V_{u2}' and V_{rd_2}' at the diffuser entry. Vu_2' is known from equation (20). To find V'_{rd_2} an iteration is necessary involving the diffuser inlet static temperature and pressure. Absolute velocity at the diffuser inlet V_{d_i} is given by

$$V_{di}^2 = V_{u_2}^{2\prime} + V_{rd_2}^{\prime 2} \qquad \ldots(22)$$

A value of V_{rd_2}' is assumed to know V_{di}. With this the temperature at inlet of diffuser is

$$T_{di} = T_{o3} - \frac{V_{di}^2}{2c_p} \qquad \ldots (23)$$

$$\text{Therefore } p_{di} = p_2 \left(\frac{T_{di}}{T_2}\right)^{\gamma/\gamma-1} \qquad \ldots(24)$$

Hence, the density of the refrigerant vapour at inlet of the diffuser is

$$\rho_{di} = p_{di}/RT_{di} \qquad \ldots (25)$$

The axial width of diffuser b'_2 is usually about 6 mm greater than the axial width of the impeller exit.

$$b'_2 = (b_2 + 6 \times 10^{-3})m \qquad \ldots (26)$$

Applying continuity equation we get

$$V_{rd_2}' = \frac{\dot{m}}{\rho_{di} \pi d_2' b_2'} \qquad \ldots (27)$$

If the calculated value of V_{rd_2}' from equation (27) is not equal to the initially assumed value, Further iteration is necessary.

Hence inlet angle of the diffuser vanes is

$$\alpha_i = \tan^{-1} \frac{V_{rd_2}'}{V_{u2}'} \qquad \ldots (28)$$

The outer diameter at the diffuser exit d_3 is usually between 1.4 and 1.6 times the outer diameter of the impeller to get a reasonable area ratio for enhancing the diffusion process.

$$d_3 = 1.4\, d_2 \qquad \ldots (29)$$

Hence, relative velocity of the refrigerant vapour at the exit of diffuser is

$$V_{rd_3} = (\dot{m} RT_3)/(\rho_3 \pi d_3 b_2') \qquad \ldots (30)$$

Hence, the diffuser vane angle at the exit is

$$\alpha_e = \tan^{-1} \left(\frac{V_{rd_3}}{V_1}\right) \qquad \ldots (31)$$

Approximate throat width is

$$W = (\pi\ d_2'\ \cos\alpha_2)/N_D \qquad \ldots(32)$$

Where N_D = No. of diffuser vanes which should be taken greater than the impeller vanes (from mechanical considerations).

Finally power needed to drive the compressor is

$$\text{Power} = m^{\cdot} \cdot \left[\frac{Z^{0.7}-1}{Z^{0.7}}\right] \psi\ u_2^2\ .\ 1/\eta_m \qquad \ldots(33)$$

where η_m = Mechanical efficiency

4.3 Volute Casing Design

$$Q_\Theta = V_{rd_3}\ A_\Theta = \frac{\Theta}{360^o}\ Q$$

$$\text{Therefore } A_\Theta = \frac{\Theta}{360}\ \frac{Q}{V_{rd_3}} \qquad \ldots(xvi)$$

For a rectangular cross - section

$$A_\Theta = (r_4 - r_3)\ b_2' \qquad \ldots(xvii)$$

Putting the value of A_Θ from (xvii) into (xvi), we get -

But $r_3 = \frac{d_3}{2}$, we finally get

$$r_4 = \frac{d_3}{2} + \frac{m^{\cdot}\Theta}{360\ \rho_3\ V_{rd_3}\ b_2'} \qquad \ldots(34)$$

where θ is in degree

The volute radius for given values of r_3 and b_3 can be determined from eqn. (34).

And finally we define a non -dimensional term F by the following eqn.

$$F = \frac{m^{\cdot}\sqrt{h_1}}{p_1 d_2^2} \qquad \ldots(35)$$

Now a suitable computer program can be developed for these (35) equations.

The variables affecting the performance of centrifugal compressors are P_1, P_2, h_1, h_2, the mass flow rate m, the outer diameter of impeller d_2 and the speed. The pressure ratio may be expressed as

$$\frac{P_2}{P_1} = f\left[\frac{m^{\cdot}\sqrt{h_1}}{P_1 d_2^2}, \frac{d_2 N}{\sqrt{h_1}}\right]$$

For a given compressor we can drane a graph between P_2/P_1 and

$\frac{m.\sqrt{h_1}}{P_1 d_2^2}$ keeping speed canstant.

These two quantities are dimensionless. Another compressor characteristic which is of importance in the refrigeration plant is compressor cooling capacity Vs. pressure ratio. In the present design an attempt will be made to draw a graph between F and pressure ratio r_p and between compresser cooling capacity vs. pressure ratio r_p.

The compressor cooling capacity is $Q_o = m. (h_1 - h_\ell)$ ...(36)

Where h_ℓ = Enthalpy of saturated liquid at the condenser pressure

CHAPTER 5

RESULTS & DISCUSSION

A Computer program was developed for equations numbering from (1)to (35) of the chapter 4 entitled 'DESIGN METHODOLOGY'. The design was based on the iterative procedure with the consideration of the following points.

(i) The hoop stress at the impeller tip should not exceed the allowable stress of the material of the impeller.

(ii) The mach number at any point in the whole stage must be below o.g.

(iii) The exit temperature of the refrigerant from the compressor stage should be well below its critical temperature for high cop (coefficient of performance) of the plant.

When the required input data is fed, the computer program calculates the dimensions, property of the refrigerant at the exit of the compressor, power Input and Compressor Cooling Capacity.

Design Requirement : To design a single stage centrifngal compressor for Ammonia to handle a mass flow rate equal to 6kg/s and pressure ratio equal to 3.

Input data : These are Mass flow rate, pressure Ratio, specific heat Ratio, specific heat at constant pressure, Inlet temperature, compressor total the total adiabatic Efficiency, No. of Impeller Blades, No. of diffuser vanes, RPM, Evaporator pressure, power Input factor and evaporator pressure.

When Input data is fed, we get the following designe values

Mass Flow Rate	: 6kg/s
pressure Ration	: 3.0
specific Heat Ratio	:1.31
specific Heat at constant pressure	: 2066 J /kg-K
Refrigerant inlet Temperature	: 226K
Isentropic Efficiency	: 0.80
No. of Impeller Blades	: 20
No. of Diffuser Blades	: 17
Rotational Speed	: 15000 rpm
Inlet Mach Number	: 0.5
Evaporator pressure	: 1.18bar
power Input Factor	: 1.035
Impeller Tip Diameter	: 55.8cm
Eye Inlet Diameter	: 18.6cm
Eye outlet Diameter	: 27.94cm
Eye Inlet Angle	: 51.9^{o}
Eye outlet Angle	: 40.4^{o}
Impeller Axial width	: 1.50cm
Inlet Diameter of Diffuser	: 67cm
outlet Diameter of Diffuser	: 78.23cm
Diffuser Inlet vane Angle	: 12.146^{o}
Diffuser outlet vane Angle	: 58.53^{o}

Diffuser Axial width : 2.10cm

Throat width of Diffuser : 1.8cm

(B) Power needed to drive the compressor = 1127kw

(C) Exit Temperature of Refrigerant T_{o3} = 31ok

(D) Condenser pressure p_3 = 3bar

(E) Compressor colling capacity Q_o = 2217.8 ton.

To draw Compressor characteristic, speed is varied. Four Tables have bee shown in Appendix for refrigerant mass flow rate equal to 2kg/s and speeds equal to 12000 rpm, 13000 rpm, 14000 rpm and 15000 rpm keeping evaporator pressure equal to 1 bar and temperature at entrance to the compressor equal to 230k. Two graphs have been drawn. One between the compressor cooling capacity and the pressure rate and the other between the Non-dimensional parameter F and the pressure ratio at four speeds 12000RPM, 13000RPM 14000RPM and 15000RPM.

As the pressure ratio increases the compressor cooling capacity decreases. Similarly with an increase of pressure Ratio, Non-dimensional parameter (F) decreases.

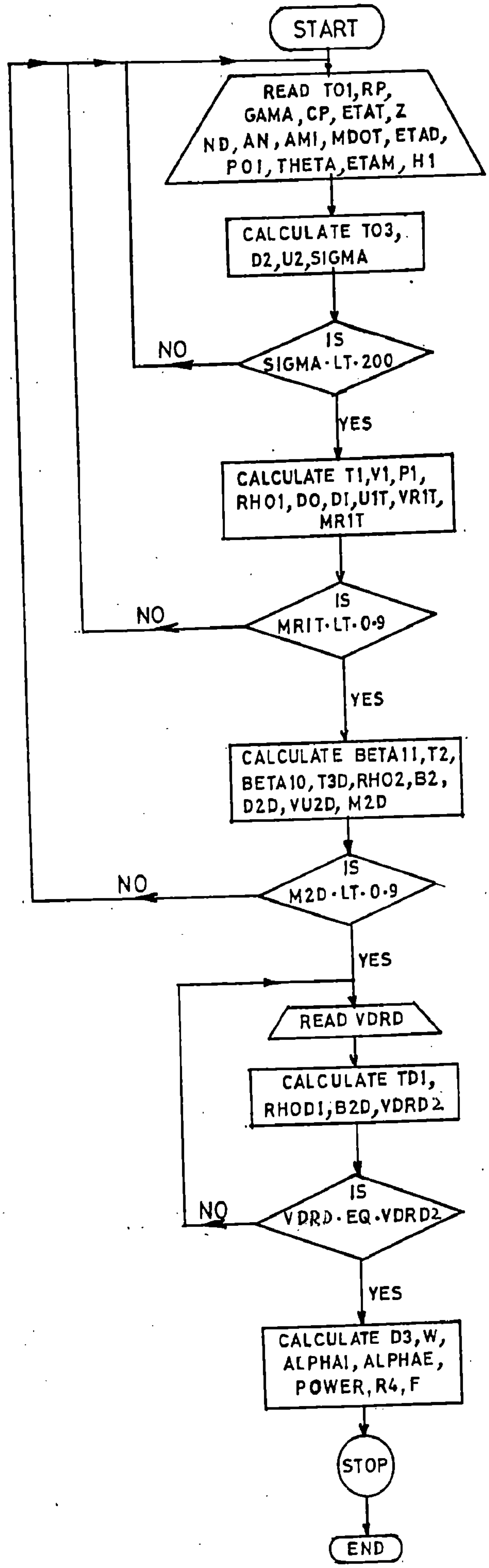
CHAPTER 6

CONCLUSIONS & FUTURE SCOPE

The method developed for the design of a single stage centrifugal compressor for refrigerants is based on one dimensional axisymmetric flow of the refrigerant. The mathematical model developed is suitable for low pressure ratio. As a first step, This method can be used as a preliminary tool for parametric study and performance evaluation of centrifugal compressors at low pressure ratio.

For the future work, The centrifugal compressor may be designed on the basis of the asymmetric flow of the refrigerant with suitable loss models on Dean's wake and jet model.

FLOW CHART


START
READ T01, RP, GAMA, CP, ETAT, Z, ND, AN, AM1, MDOT, ETAD, P01, THETA, ETAM, H1
CALCULATE T03, D2, U2, SIGMA
IS SIGMA·LT·200
NO
YES
CALCULATE T1, V1, P1, RHO1, DO, DI, U1T, VR1T, MR1T
IS MR1T·LT·0·9
NO
YES
CALCULATE BETA11, T2, BETA10, T3D, RHO2, B2, D2D, VU2D, M2D
IS M2D·LT·0·9
NO
YES
READ VDRD
CALCULATE TD1, RHOD1, B2D, VDRD2
IS VDRD·EQ·VDRD2
NO
YES
CALCULATE D3, W, ALPHAI, ALPHAE, POWER, R4, F
STOP
END

```
C
C                    IMPELLER DESIGN
C
C                          By - J. N. Chaudhary,ME(TH)
C**********************************************************************
C
C
      DIMENSION IT(300)
      COMMON  D2,Z,U2,T2,CP,R,GAMA,PI
      COMMON  T01,P2,TERM1,B2,MDOT,V1,T3
      OPEN(UNIT=5,FILE='IMP.OUT',STATUS='NEW')
      OPEN(UNIT=4,FILE='IMP.IN',STATUS='OLD')
      G=9.81
      PI=3.1415965
      R=489.076
      WRITE(*,*)'Pressure Ratio  : '
      READ(4,*)RP
      WRITE(5,*)'Pressure Ratio  : ',RP
      WRITE(*,*)'Specific Heat Ratio :'
      READ(4,*)GAMA
      WRITE(5,*)'Specific Heat Ratio :',GAMA
      WRITE(*,*)'Sp. Heat at Cons. Pressure : '
      READ(4,*)CP
      WRITE(5,*)'Sp. Heat at Cons. Pressure : ',CP
      WRITE(*,*)'Total Efficiancy  :'
      READ(4,*)ETAT
      WRITE(*,*)' ETAT=',ETAT
      WRITE(5,*)'Total Efficiancy  :',ETAT
      WRITE(*,*)'Inlet Temp.       :'
      READ(4,*)T01
      WRITE(5,*)'Inlet Temp.       :',T01

      WRITE(*,*)'GAMA='
      WRITE(*,*)GAMA
      TERM1=(GAMA-1)/GAMA
      READ(*,*)Z
      WRITE(*,*)'Z=',Z
      TERM2=(Z**0.7-1)/Z**0.7
      WRITE(*,*)'ETAT=',ETAT
      T03=T01*(1+1/ETAT*(RP**TERM1-1))
      WRITE(*,*)'Outlet Temp.       :',T03
      WRITE(5,*)'Outlet Temp.       :',T03
      WRITE(*,*)'No of Blades       :'
      WRITE(5,*)'NO OF Blades       :',Z
      WRITE(*,*)'TERM2=',TERM2
      U2=SQRT(CP*(T03-T01)/TERM2)
      WRITE(*,*)'Rotation Speed   :'
      READ(4,*)AN
      WRITE(5,*)'Rotation Speed   :',AN
      WRITE(*,*)'AN=',AN
      WRITE(*,*)'PI=',PI
      D2=60*U2/(PI*AN)
      SIGMA=7850.*U2**2
      WRITE(5,*)'Impeller tip dia.:',D2
```

```
WRITE(*,*)'Inlet Mach No.    :'
READ(4,*)AM1
WRITE(5,*)'Inlet Mach No.    :',AM1
T1=T01/(1+(GAMA-1)*0.5*AM1*AM1)
V1=SQRT(2*CP*(T01-T1))
WRITE(*,*)'Inlet TotaL Pressure :'
READ(4,*)P01
WRITE(5,*)'Inlet TotaL Pressure :',P01
WRITE(*,*)'T01=',T01
WRITE(*,*)'TERM1',TERM1
P1=P01*((T1/T01)**(1/TERM1))
WRITE(*,*)'R',R
WRITE(*,*)'T1=',T1
RHO1=P1/(R*T1)
D0=0.5*D2
WRITE(5,*)'Eye Outlet dia.     :',D0
WRITE(*,*)'Mass Flow Rate      :'
READ(4,*)MDOT
WRITE(5,*)'Mass Flow Rate      :',MDOT
WRITE(*,*)'RHO1=',RHO1
WRITE(*,*)'V1=',V1
DI=SQRT(D0*D0-4*MDOT/(PI*RHO1*V1))
WRITE(5,*)'Eye Inlet dia.      :',DI
U1T=PI*D0*AN/60.0
VR1T=SQRT(U1T*U1T+V1*V1)
AMR1=VR1T/SQRT(GAMA*R*T1)
IF(AMR1 .GT. 0.9)THEN
WRITE(*,*)'START FROM NEW VALUE OF N & RP'
GO TO 999
ENDIF
BETA1I=(180/PI)*ATAN(60*V1/(PI*DI*AN))
TERM3=BETA1I*180/PI
WRITE(5,*)'Eye Inlet Angle    :',TERM3
WRITE(*,*)'D0=',D0
BETA1O=(180/PI)*ATAN(60*V1/(PI*D0*AN))
TERM4=BETA1O*180/PI
WRITE(5,*)'Eye Outlet Angle   :',TERM4
WRITE(*,*)'Absolute angle at tip :'
READ(4,*)ALPHA2
WRITE(5,*)'Absolute angle at tip :',ALPHA2
V2=TERM2*U2/COS(ALPHA2*PI/180)
VRD2=TERM2*U2*TAN(ALPHA2*PI/180)
T2=T03-0.5*V2*V2/(CP*(COS(ALPHA2)**2))
T3=T03-0.5*V1*V1/CP
WRITE(*,*)'T03=',T03
WRITE(*,*)'TERM1=',TERM1
P3=P01*RP*((T3/T03)**(1/TERM1))
WRITE(*,*)'Diffuser Efficiancy :'
READ(4,*)ETAD
WRITE(5,*)'Diffuser Efficiancy :',ETAD
T3D=T2+ETAD*(T3-T2)
WRITE(*,*)'T3D=',T3D
P2=P3*((T2/T3D)**(1/TERM1))
RHO2=P2/(R*T2)
B2=MDOT/(PI*D2*VRD2*RHO2)
```

```
      WRITE(5,*)'Impeller Axial Width at Exit :',B2
      WRITE(*,*)'Relative Velocity at diffuser inlet :'
      READ(4,*)VDRD
      WRITE(5,*)'Relative Velocity at diffuser inlet :',VDRD
      CALL DIFFUSER(VDRD)
999   CLOSE(5)
      STOP
      END

      SUBROUTINE DIFFUSER(VDRD)
```

```
C       **************************************************************
C       *                    DESIGN OF DIFFUSER                      *
C       **************************************************************
C
        COMMON  D2,TERM2,U2,T2,CP,R,GAMA,PI
        COMMON  T03,P2,TERM1,B2,MDOT,V1,T3
        OPEN(6,FILE='DIF.OUT',STATUS='NEW')
        WRITE(6,100)
 100    FORMAT(////,10X,'DESIGN OF DIFFUSER'///)
        D2D=1.2*D2
        VU2D=D2*TERM2*U2/D2D
        AM2D=VU2D/SQRT(GAMA*R*T2)
110     VDRD=VDRD+0.1
        VDIS=VU2D*VU2D+VDRD*VDRD
        TDI=T03-VDIS*0.5/CP
        PDI=P2*((TDI/T2)**(1/TERM1))
        RHODI=PDI/(R*TDI)
        B2D=B2+0.006
        VDRD1=MDOT/(RHODI*PI*D2D*B2D)
        VDRD2=(VDRD1-VDRD)
        WRITE(*,*)VDRD1,VDRD,VDRD2
        IF( VDRD2.LT.0.01) GO TO 200
        GO TO 110
200     WRITE(6,*)'Relative Velocity',
     *  'at diffuser Inlet:',VDRD1
        ALPHAI=ATAN(VDRD/VU2D)
        D3=1.4*D2
        VRD3=MDOT*R*T3/(RHODI*PI*D3*B2D)
        ALPHAE=ATAN(VRD3/V1)
        POWER=MDOT*TERM2*1.035*U2**2/0.90
        WRITE(6,*)'Inlet dia. diffuser :',D2D
        WRITE(6,*)'Mach No. at diffuser Inlet :',AM2D
        WRITE(6,*)'Axial Width at diffuser Inlet:',B2D
        WRITE(6,*)'Diffuser Inlet Vane Angle :',ALPHAI
        WRITE(6,*)'Diffuser Outlet Vane Angle:',ALPHAE
        WRITE(6,*)'Diffuser Outlet dia.      :',D3
        WRITE(6,*)'Power   :',POWER
        CLOSE(6)
        RETURN
        END
```

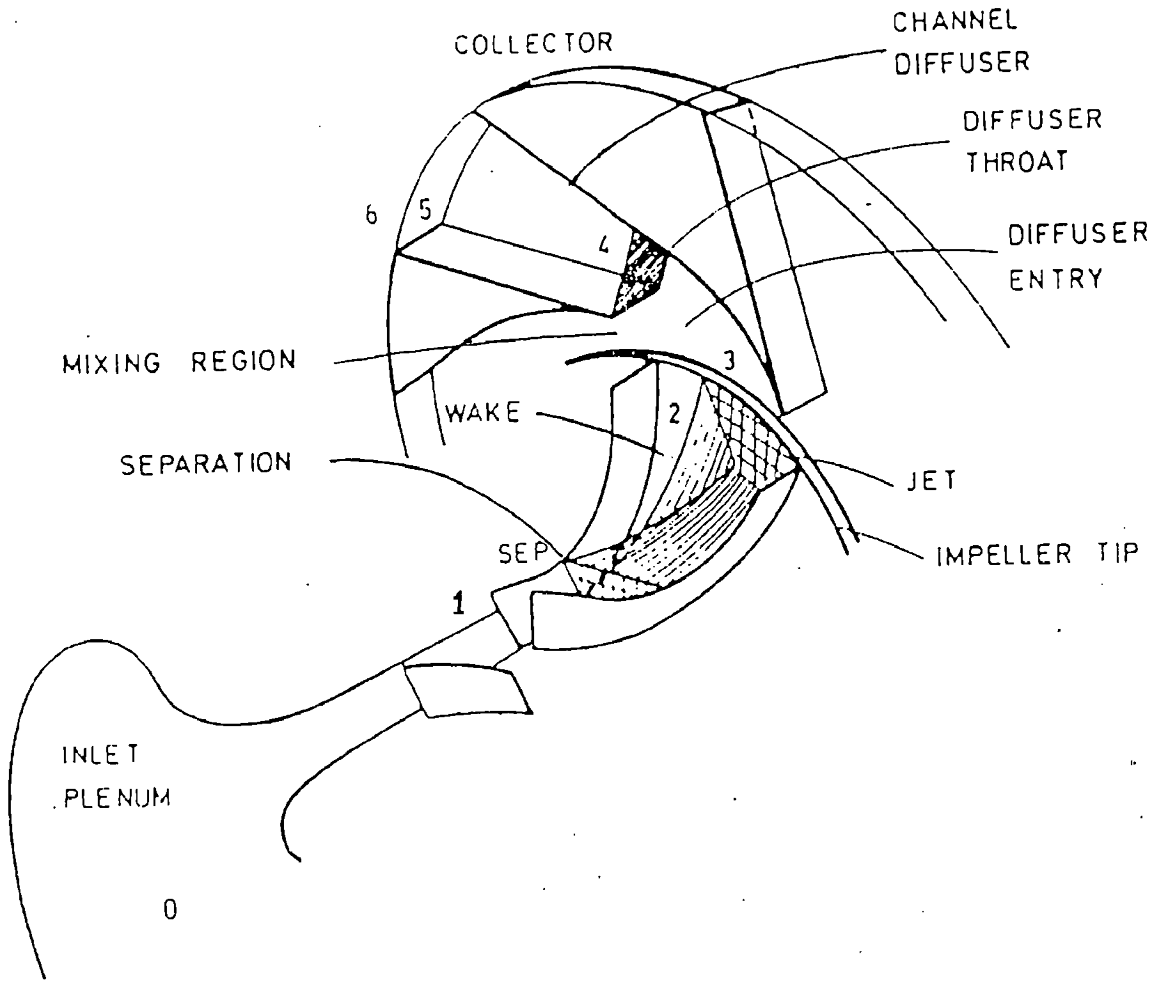


Fig. 1 : Element of a centrifugal compressor stage

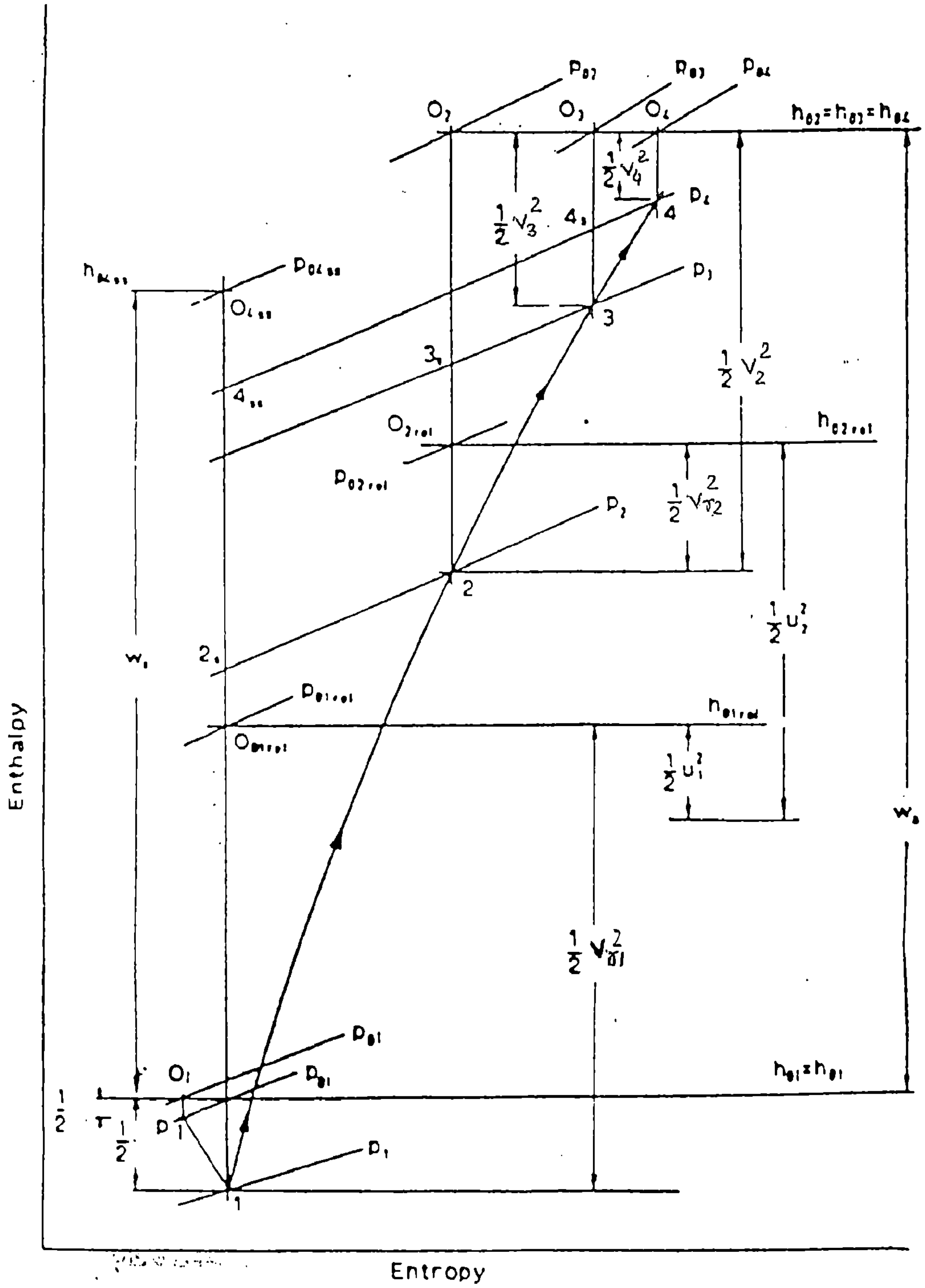


Fig. 2 : H-s Diagram

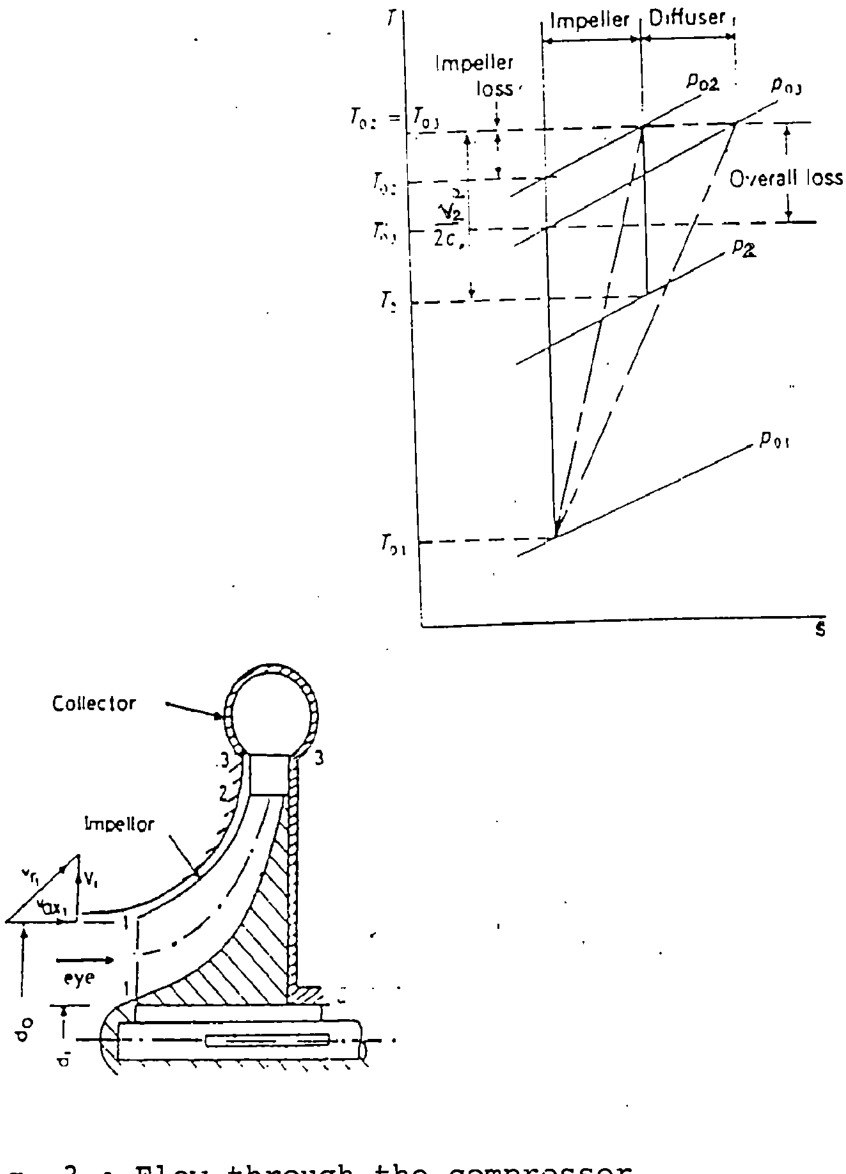


Fig. 3 : Flow through the compressor

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$$N_s = \frac{N\sqrt{Q}}{H_{ad}^{\frac{3}{4}}}$$

$$Q = \frac{\dot{m}}{g\rho_{01}}$$

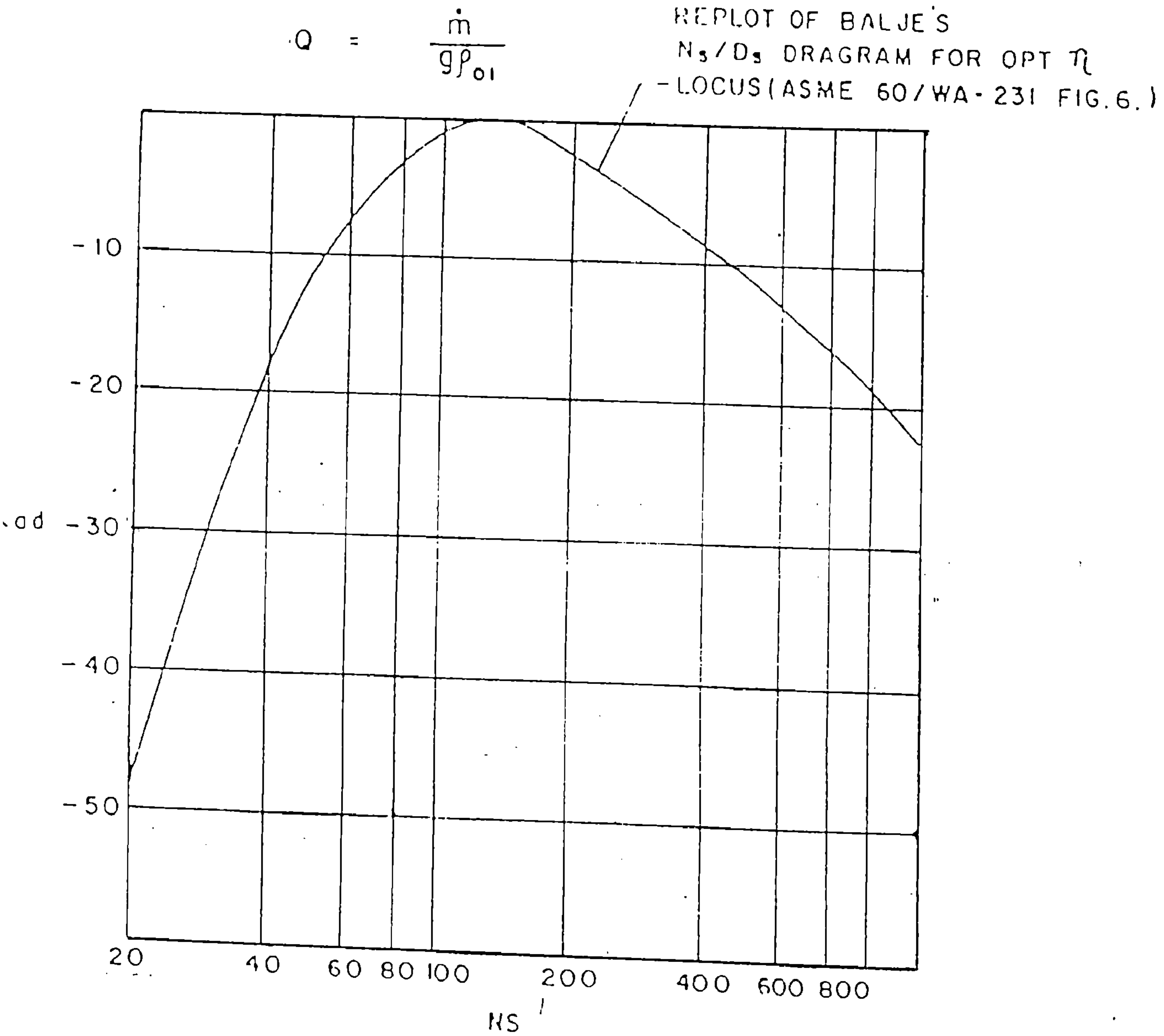

Fig. 4 : Variation of adiabatic efficiency with specific speed

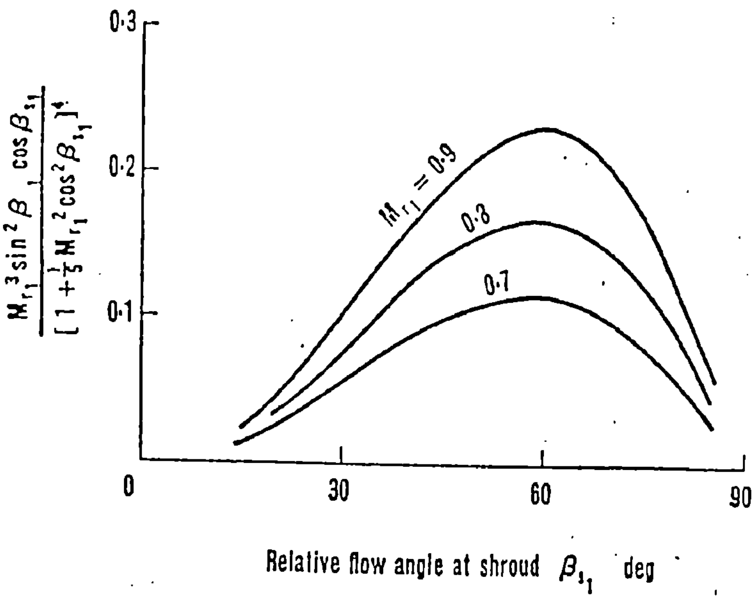

Fig.5: Mass flow function for a centrifugal compressor with zero entry swirl (adapted from Shepherd) .

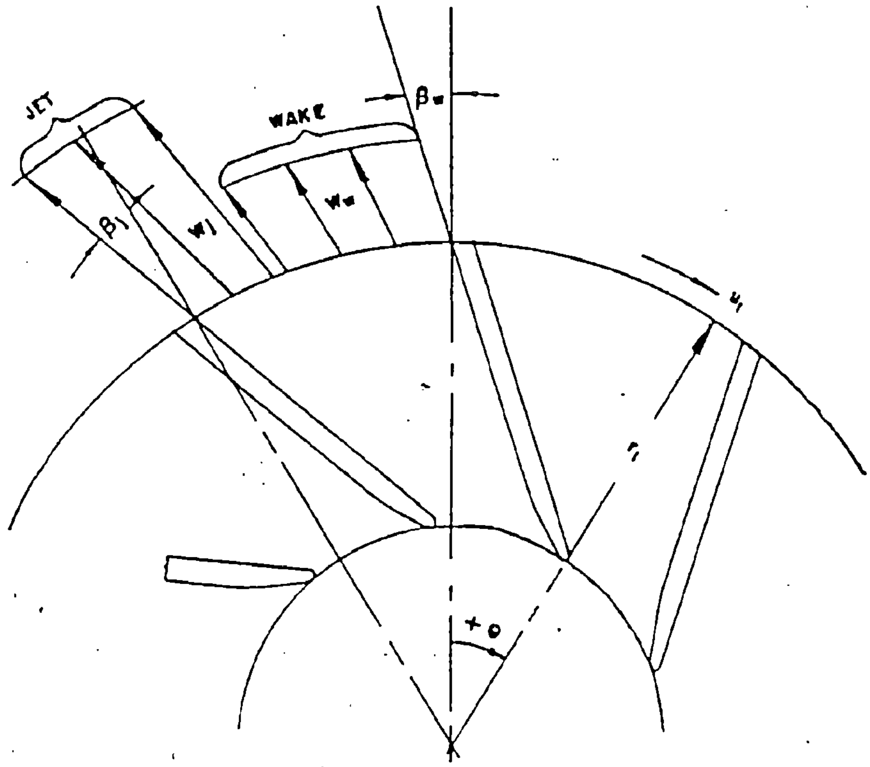

Fig. 6 : Wake and jet model

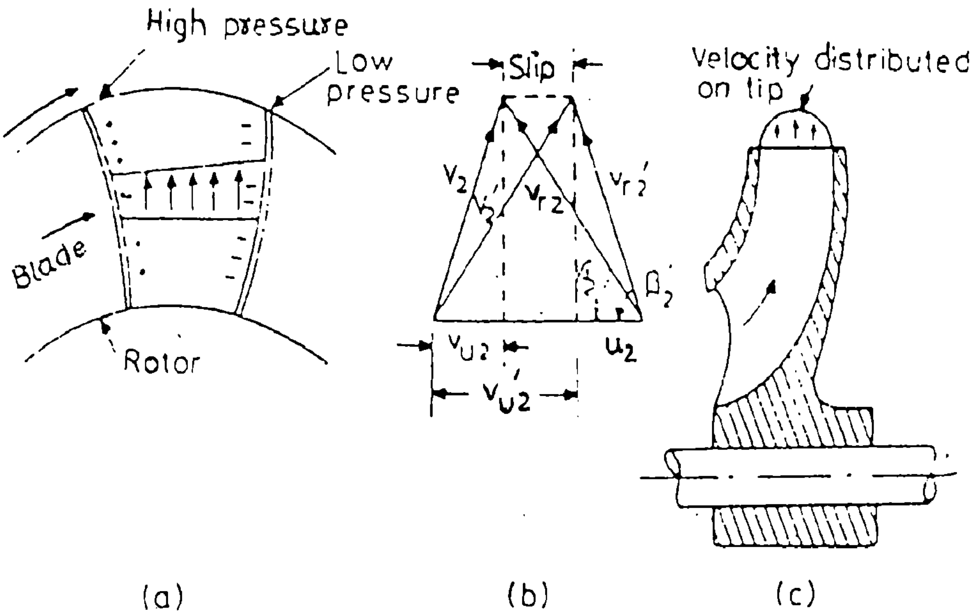

Fig. 7 : Effects of coriolis forces

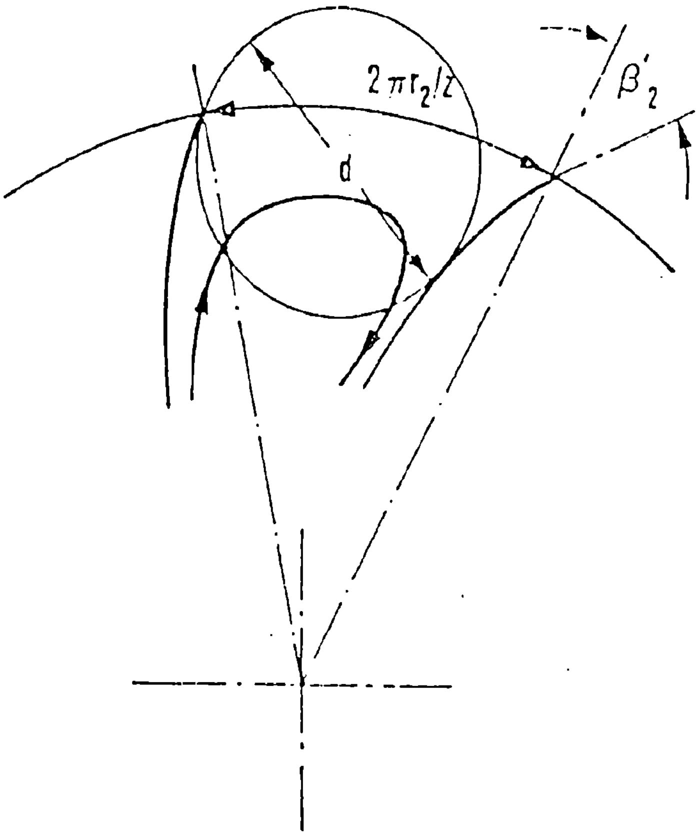

Fig. 8 : Stodola flow model with slip

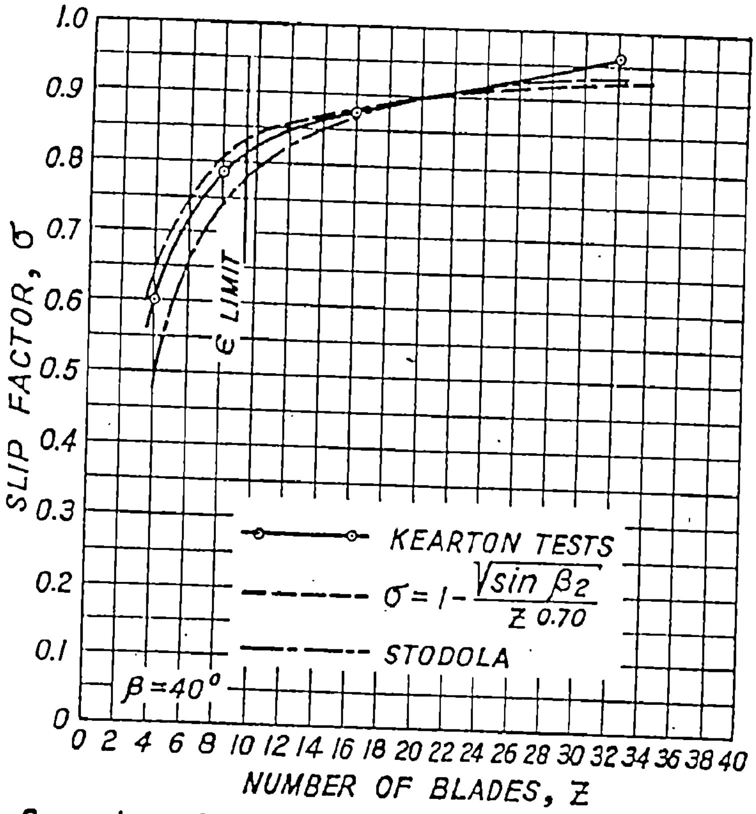

Fig. 9 Comparison of slip factors with Kearton's experimental results

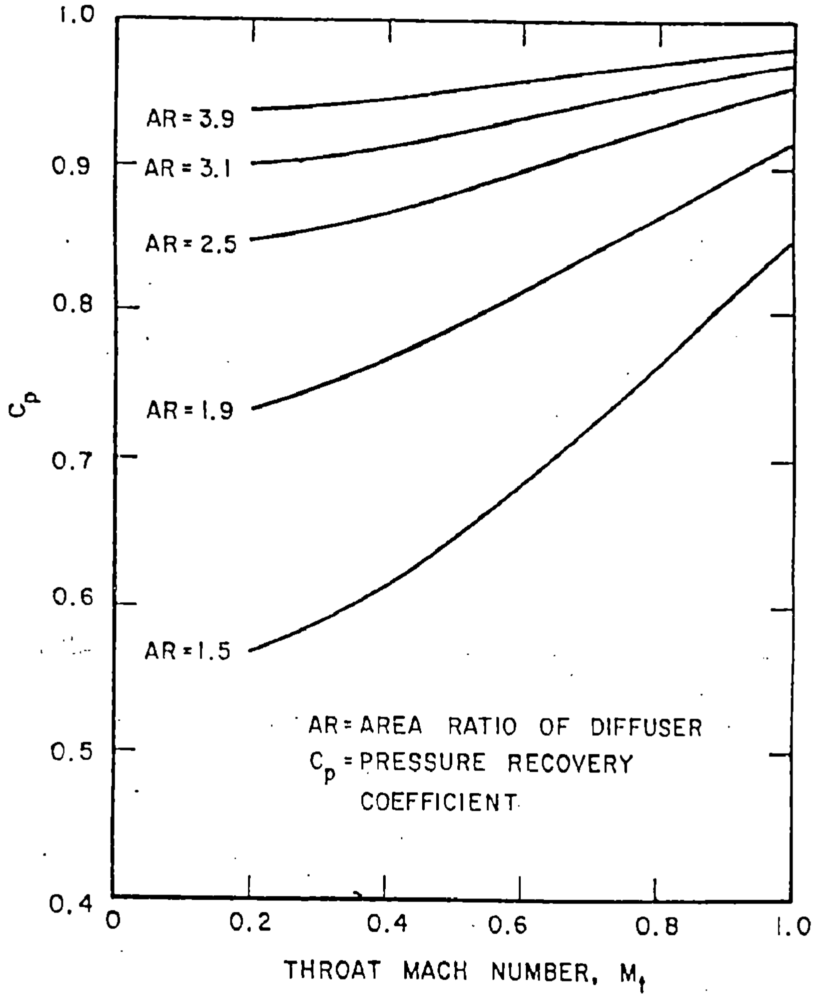

Fig. 10 Ideal pressure recovery versus throat Mach number with area ratio as a parameter

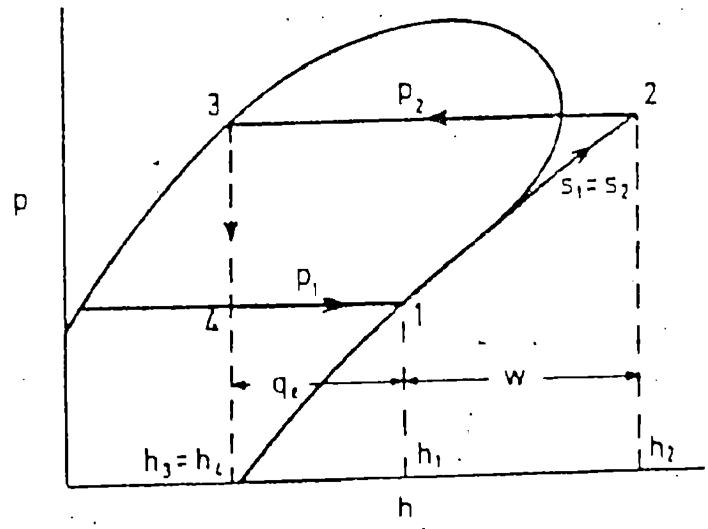

Fig. 11 : p-h diagram

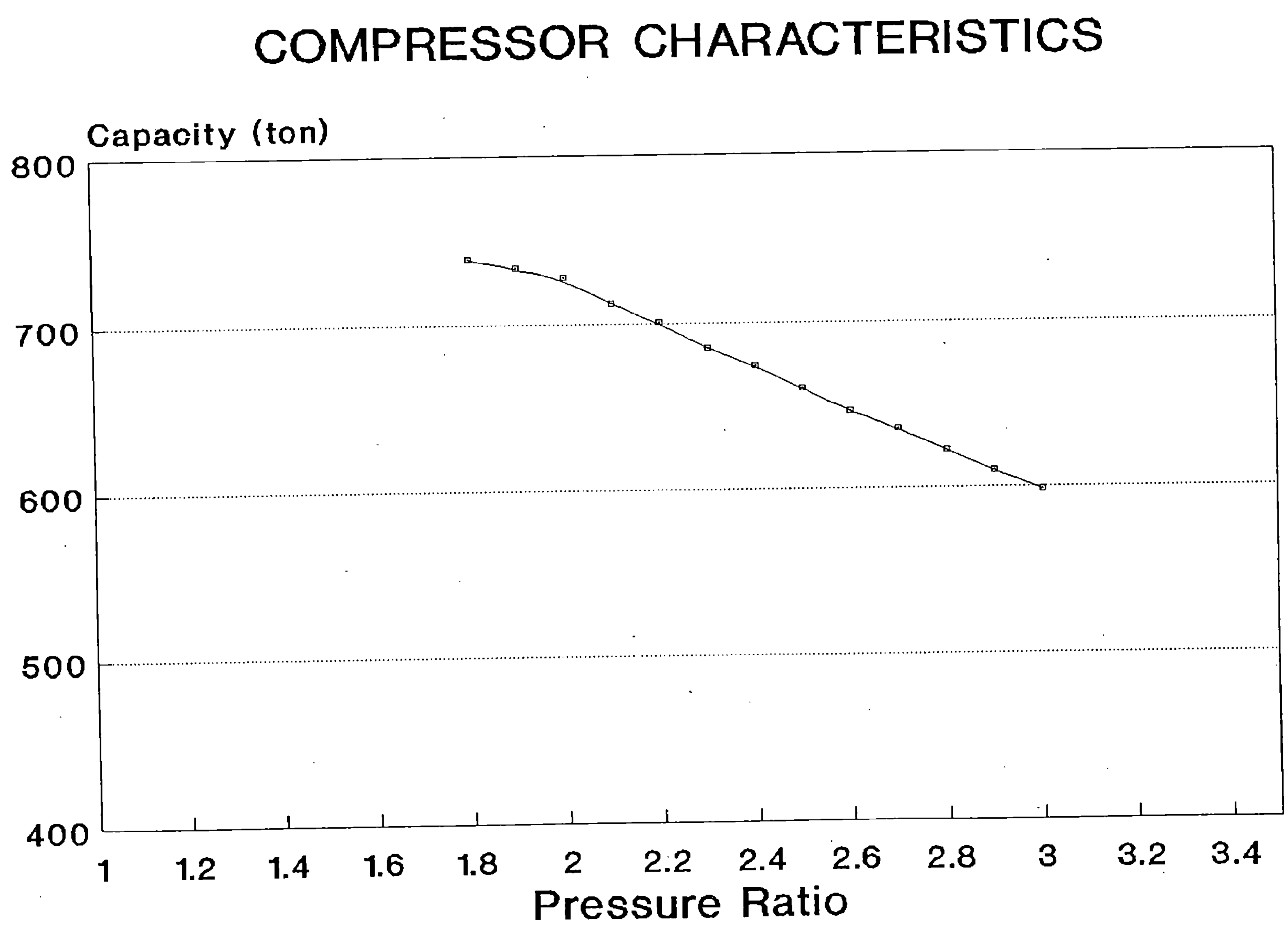

Fig. 13: Pressure Ratio Vs. Capacity

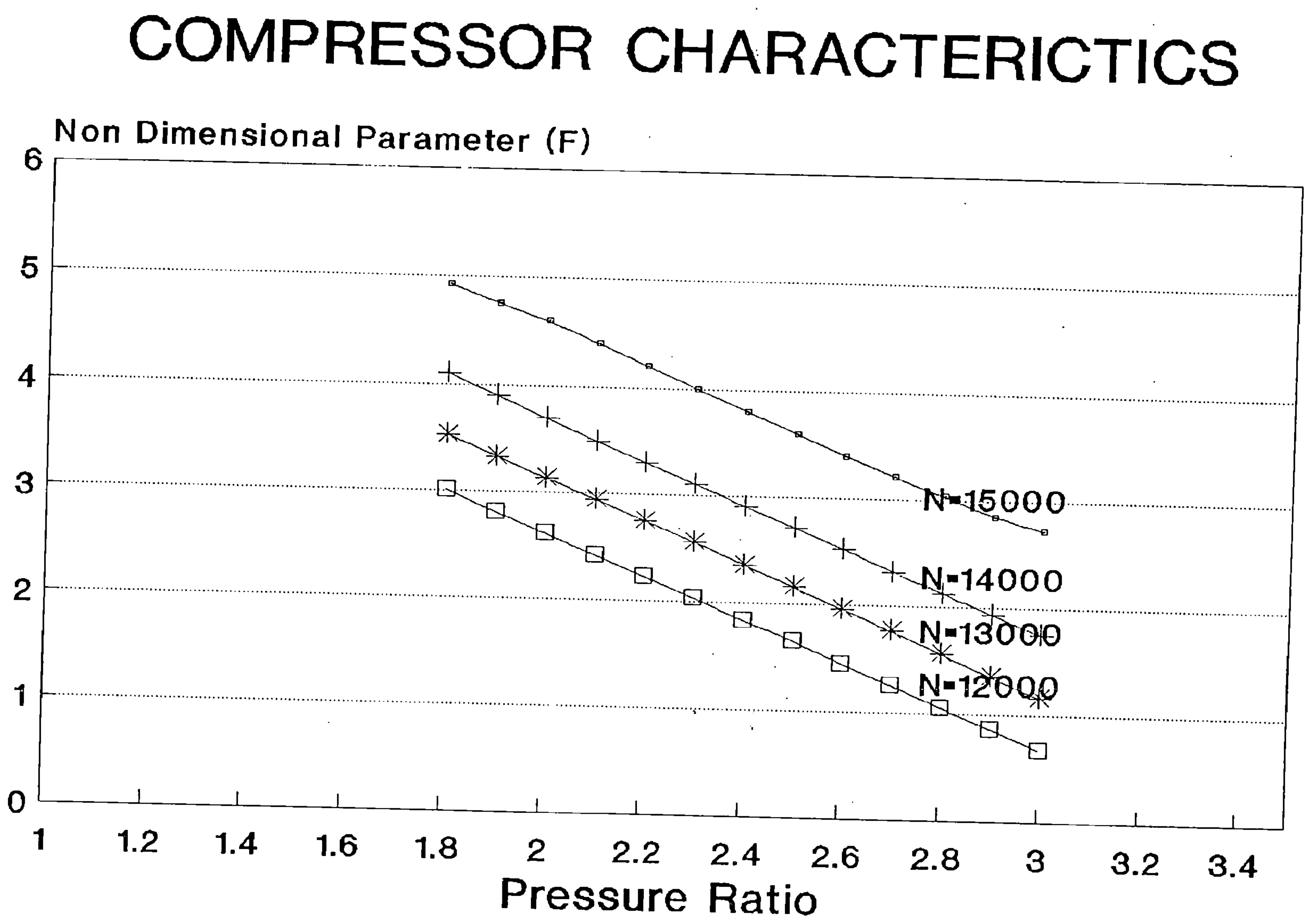

Fig. 14 :Pressure Ratio Vs. F

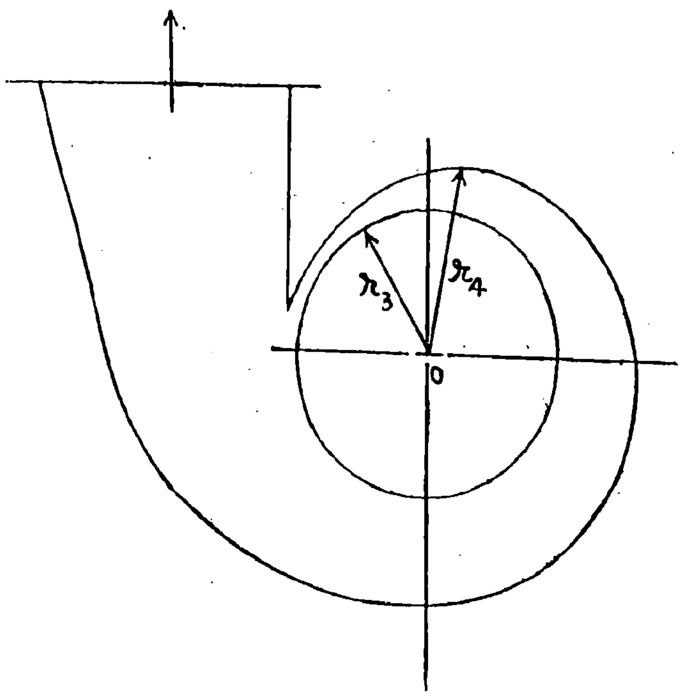

Fig. 15 : Approximate shape of volute

APPENDIX

TABLE No. 1

$P_1 = 10^5\ N/m^2$; $h_1 = 1390$ kJ/kg ; m = 2 kg/s ; $To_1 = 230$ k

N = 15000 rpm				N = 14000 rpm			
r_p	d_2 (m)	Q_o (Ton)	F	r_p	d_2 (m)	Q_o (Ton)	F
1.8	0.390	739.26	4.92	1.8	0.427	739.26	4.16
1.9	0.410	733.60	4.45	1.9	0.448	733.60	3.72
2.0	0.435	727.90	3.95	2.0	0.467	727.90	3.43
2.1	0.452	712.20	3.66	2.1	0.484	712.20	3.19
2.2	0.467	706.60	3.43	2.2	0.501	706.60	2.98
2.3	0.480	700.94	3.24	2.3	0.516	700.94	2.81
2.4	0.501	694.12	2.98	2.4	0.531	694.12	2.65
2.5	0.518	688.21	2.78	2.5	0.546	688.21	2.51
2.6	0.527	681.10	2.69	2.6	0.562	681.10	2.40
2.7	0.539	675.60	2.57	2.7	0.578	675.60	2.23
2.8	0.552	662.48	2.45	2.8	0.605	662.48	2.04
2.9	0.567	654.60	2.32	2.9	0.619	654.60	1.95
3.0	0.582	646.40	2.18	3.0	0.633	646.40	1.83

TABLE No. 2

$P_1 = 10^5\ N/m^2$; $h_1 = 1390$ kJ/kg ; $m = 2$ kg/s ; $To_1 = 230$ k

N = 15000 rpm				N = 14000 rpm			
r_p	d_2 (m)	Q_o (Ton)	F	r_p	d_2 (m)	Q_o (Ton)	F
1.8	0.460	739.26	3.53	1.8	0.498	739.26	3.01
1.9	0.482	733.60	3.22	1.9	0.522	733.60	2.74
2.0	0.503	727.90	2.95	2.0	0.544	727.90	2.52
2.1	0.522	712.20	2.74	2.1	0.565	712.20	2.34
2.2	0.540	706.60	2.56	2.2	0.584	706.60	2.19
2.3	0.556	700.94	2.42	2.3	0.602	700.94	2.06
2.4	0.572	694.12	2.28	2.4	0.620	694.12	1.95
2.5	0.588	688.21	2.16	2.5	0.637	688.21	1.84
2.6	0.605	681.10	2.04	2.6	0.655	681.10	1.74
2.7	0.622	675.60	1.93	2.7	0.674	675.60	1.65
2.8	0.636	662.48	1.85	2.8	0.689	662.48	1.57
2.9	0.652	654.60	1.76	2.9	0.706	654.60	1.50
3.0	0.644	646.40	1.66	3.0	0.721	646.40	1.44

TABLE No. 3

$V_{rd_3} = 70$ m/s

Θ (degree)	r_3	R_4 (cm)	Θ (degree)	r_3 (cm)	r_4 (cm)
0	36.05	36.00	180	36.05	48.42
20	36.05	37.43	200	36.05	49.80
40	36.05	38.80	220	36.05	51.17
60	36.05	40.17	240	36.05	52.55
80	36.05	41.55	260	36.05	53.92
100	36.05	42.92	280	36.05	55.30
120	36.05	44.30			
140	36.05	45.67			
160	36.05	47.05			

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