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EXPERIMENTAL AND THEORETICAL INVESTIGATION OF LOSSES IN A RADIAL COMPRESSOR STAGE

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ABSTRACT

A model has been assumed to predict the hydraulic and disc friction losses in a radial compressor stage. A potential three dimensional flow field inside the impeller is solved, by using TSONICMR program. Then, the hydraulic and disc friction losses were predicted for the impeller, vaneless diffuser and the collector. Experimental investigation has been carried out for a radial compressor stage, in order to verify the validity of the assumed model. The pressure ratio and efficiency were measured for different mass flow rates, at different rotating speeds. The results showed satisfactory agreement with that predicted .

INTRODUCTION

The centrifugal compressors are today playing an essential role in the gas turbine applications. In small gas turbines, single stage radial compressor is usually replacing the multistage axial compressor. However, much work remains to be elaborated such that the flow inside the compressor is fully understood to predict it's behaviour.

The main part of the compressor is the impeller, as it delivers the energy to the fluid flowing through it. In the same time, it affects the flow upstream the diffuser stage and hence it's performance. The difficulties of predicting the flow inside the impeller may be partly attributed to the complex geometry of the flow passages. They are rotating curved passages from the axial to the radial direction. Also for open, or semi-open, impellers the tip leakage has an important effects on the flow field. Furthermore the viscosity and flow separation affect the build-up of the boundary layer and the deviation of the flow from potential behaviour.

Herein, the ideal potential compressor characteristics has been predicted, by using the modified version of the TSONICMR program, [1]. A loss model for both hydraulic and disc friction has been

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developed for the impeller, vaneless diffuser and for the collector. The losses have been assumed to occur due to incidence, viscous and mixing losses, as well as mechanical disc friction around the impeller. A computer program RISTIC has been written to evaluate the actual characteristics of the compressor stage, [5]. The obtained results have been verified experimentally for a model compressor stage of outer diameter of 100 mm.

I- POTENTIAL FLOW SOLUTION

It has been established earlier that the flow through stationary or rotating cascade cannot be axisymmetric if the flow has to exert a moment on the blades. It is for this reason an arbitrary flow through a cascade does not have stream surfaces of revolution or axisymmetric shape. Exceptions are the incompressible flow in radial cascades with stream surfaces of planes perpendicular to the axis, [2].

I.1- QUASI 3-DIMENSIONAL FLOW

Quasi 3-dimensional flow may be evaluated by combining the flow fields in the meridional plane and for the blade-to-blade surface of revolution. The meridional flow solution is firstly evaluated for the meridional streamlines. The stream sheets are then defined as that bound the streamlines. The blade-to-blade solution has been evaluated for each stream sheet by using the TSONICMR program, [1]. On the other hand the stream sheets geometries may be obtained by assuming equal flow areas.

For 3-dimensional potential flow, the flow equation may be written in a rotating cylindrical coordinates, (m, ϑ and r), as, [3] :

$$\frac{1}{R^2}\frac{\partial^2 \psi}{\partial \vartheta^2} + \frac{\partial^2 \psi}{\partial m^2} - \frac{1}{R^2}\frac{\partial (\ln R)}{\partial \vartheta}\frac{\partial \psi}{\partial \vartheta} + \left[\frac{\partial (\ln R)}{\partial m} - \frac{\partial (\ln b\rho)}{\partial m}\right]\frac{\partial \psi}{\partial m} = -2\, b\, \rho\, \omega \sin\alpha \qquad (1)$$

The stream function could be normalized by writing :

$$u = \psi / w \qquad (2)$$

where w is the mass flow rate per stream sheet. Then equation (1) could be reduced to :

$$\frac{1}{R^2}\frac{\partial^2 u}{\partial \vartheta^2} + \frac{\partial^2 u}{\partial m^2} - \frac{1}{R^2}\frac{1}{\rho}\frac{\partial \rho}{\partial \vartheta}\frac{\partial u}{\partial \vartheta} + \left[\frac{\sin\alpha}{R} - \frac{1}{b\rho}\frac{\partial (b\rho)}{\partial m}\right]\frac{\partial u}{\partial m} = -\frac{2\, b\, \rho\, \omega}{w}\sin\alpha \qquad (3)$$

If the flow is entirely subsonic, equation (3) is elliptic, while for supersonic flow it is hyperbolic.

The boundary conditions could be defined on the blade. The meridional flow channel, Fig.(1), is divided to 3 stream sheets.

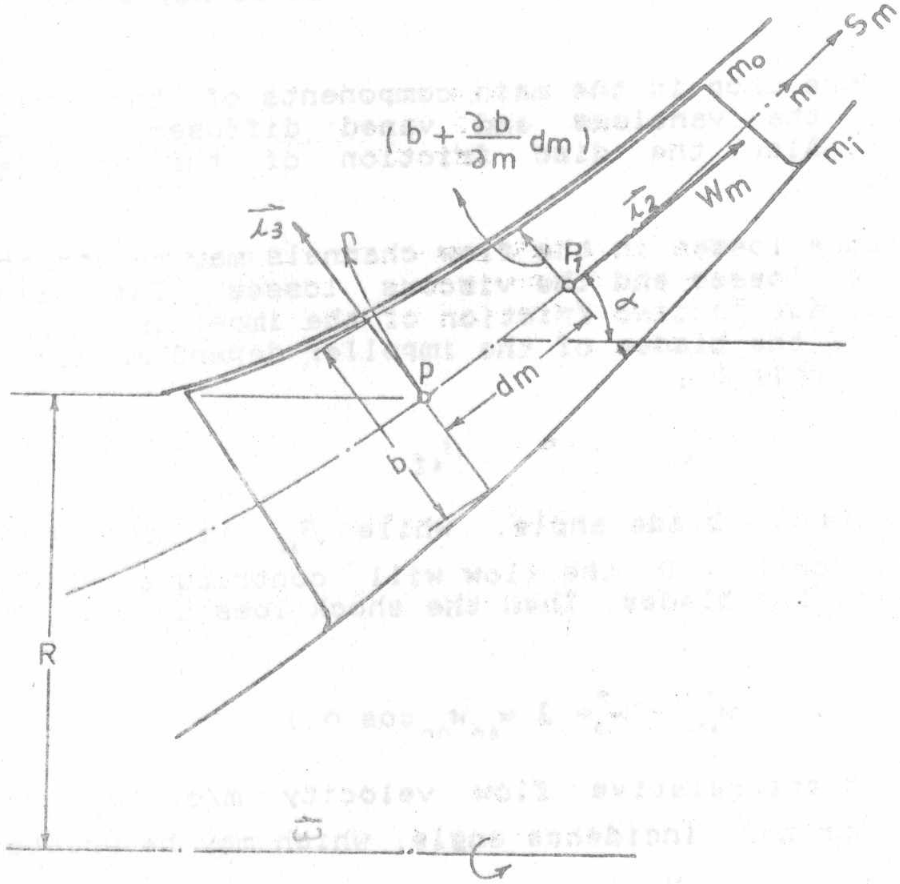

Fig.(1), Meridional Flow Channel.

The stream sheet center line will be defined by specifying several points in the (m - r) plane, while the blade geometries are defined in the (m -ϑ) plane.

I.2- TSONICMR PROGRAM INPUT/OUTPUT

The object of the program is to combine the finite difference method, which is used for subsonic flow, and the velocity gradient method, which is used for supersonic flow, so as to extend the range and cases which could be solved, [2].

The computer program requires as input the blade geometry in the (m- ϑ) plane, the geometry of the channel in the (m - r) plane, the gas constants, the operating conditions, the mass flow rate, the outlet and inlet blade angle as well as the rotating speed.

The output results are the velocity magnitude and direction at all interior mesh as well as the inlet and outlet average flow parameters.

II- PRESSURE AND DISC FRICTION LOSSES

The characteristics of the radial compressor stage may be predicted from the 3-dimensional flow model by evaluating the pressure and disc friction losses. These losses are estimated by considering the pressure drop in the main components of the compressor; the impeller, the vaneless and vaned diffuser as well as the collector. Also the disc friction of the impeller has been evaluated.

The pressure losses in the flow channels may be attributed to the inlet shock losses and the viscous losses. The main mechanical losses are due to disc friction of the impeller. The shock losses at inlet of the blades of the impeller depend mainly on the the incidence angle δ ;

$$\delta = \beta_{1b} - \beta_{1f} \tag{4}$$

where β_{1b} is the blade angle, while β_{1f} is the flow angle at inlet. So for δ 0, the flow will contribute shock losses at entrance of the blades. Then the shock loss L_s may be estimated as, [3, 4] ;

$$L_s = \rho \, (w_{1n}^2 + w_o^2 - 2 \, w_{1n} w_{on} \cos \delta \,) \tag{5}$$

where w_1 is the relative flow velocity m/s, w_{on} is the flow velocity for zero incidence angle, which may be written as :

$$w_{on} = \frac{u_n}{\cos\, (90 - \beta_{1b}\,)} \tag{6}$$

where u_n is the prephiral velocity of a stream sheet at center line.

The impeller internal friction L_f is affected by the eddies formation and turbulence level of the flow in the blade-channel, as well as the skin friction on the blades. These losses are assumed to be function of the flow velocities as :

$$L_f = kl \; [\, \zeta_1 w_1^2 + \zeta \; c_{2m}^2] \tag{7}$$

where kl is an empirical factor to adapt the calculated to the the experimental results, [5], and it was assumed to be :

$$kl = 0.304 \; N_r \tag{8}$$

N_r is the speed ratio and is equal to N_c/ 25000.

N_c is compressor speed. The nominal speed is 25000 rpm.

w_1 is the average relative velocity at inlet of the blade.

c_{2m} is the outlet meridional velocity.

ζ_1 and ζ_2 are the inlet and outlet loss coefficient

The disc friction on the impeller L_{DF} may be assumed to depend mainly on the impeller and casing profile. It may be estimated as :

$$L_{DF} = k2 . K\ (\omega^2 R^2) \qquad (9)$$

where k_2 is an empirical factor and K is the disc friction factor. [6] :

$$K = \frac{\alpha . 10^3}{\pi} \frac{D_2}{n} \frac{u_2^2}{b_2 c_{2m}}$$

where α is a coefficient = 4.5 to 9 (large values for small sizes), D_2 is the impeller outlet radius, and ω is the angular velocity.

The vaneless diffuser losses L_{VD} may be estimated as, [7] :

$$L_{VD} = \zeta_f \frac{c_2'^2 - c_3^2}{b_2 \quad b_3} \frac{r_3 - r_2}{\sin(\alpha_2' + \alpha_2)/2} \qquad (10)$$

where c is the absolute velocity, b is the width, α is the absolute velocity angle and ζ_f is the loss coefficient (= 0.75 to 1.10^{-2}). The subscripts 2 is for the impeller outlet and 3 is for diffuser outlet.

The bladed diffuser losses may be assumed to be due to incidence and skin friction. The incidence loss L_i is evaluates as, [5] :

$$L_i = 0.5\ [\ c_3^2 + c_{o3}^2 - 2\ c_{o3}\ c_3 \cos \delta_3\ ] \qquad (11)$$

where δ is the incidence angle and the subscripts o3 is for zero incidence angle.

The skin friction L_{sF} may be given as, [5] :

$$L_{sF} = \zeta\ (\ c_4^2 - c_{33}^2)\ /\ 2 \qquad (12)$$

where the subscripts 33 is for the straight part of the diffuser and 4 is for diffuser outlet.

The collector losses L_c may given as :

$$L_c = \zeta\ (\ c_{4m} - c_5\ )^2 \qquad (13)$$

where the subscripts 4m is for the bladed diffuser outlet and 5 is for the inlet to the collector. The loss coefficient ζ is in the order of 0.45 , [5].

III- EXPERIMENTAL RESULTS

A test rig was built to verify the loss model, as well as finding the required empirical constants, [5]. The model compressor is driven by an air turbine of a maximum turbine speed of 30,000 rpm. This turbine is driven by means of the air flow delivered from a root compressor . The air flow rate was controlled by controlling the root compressor speed or by controlling the turbine flow rate.

The experimental measurements have been carried out to evaluate the compressor pressure ratio, power and efficiency for different flow rates and speeds. The volume flow rate was measured by means of orifice meter fitted at outlet of the compressor, while the pressure and pressure difference were measured by water inclined piezomanometers. The temperature was measured by a cupper-constantan thermo- couple probes. The input torque was measured by a friction brake, while the speed was measured by a hand tachometer, [5].

These measured parameters were used to evaluate the overall characteristics of the compressor stage. A sample of both the theoretical and experimental results is shown in figure ( ). The relative error was estimated to be about 0.4 to 2.5 % for the predicted total efficiency of the compressor stage.

CONCLUSION

Pressure and disc friction losses have been estimated for a radial compressor stage, by assuming a loss model. The real characteristics have been predicted, form the theoretical three dimensional potential flow and the loss model. The results were verified experimentally by for 100 mm external diameter model compressor. The results show that the agreement between the theoretical and experimental investigations are satisfactory, with an error of within 0.4 to 1.4 % for the pressure ratio, and within 0.4 to 2.5 % for the overall efficiency.

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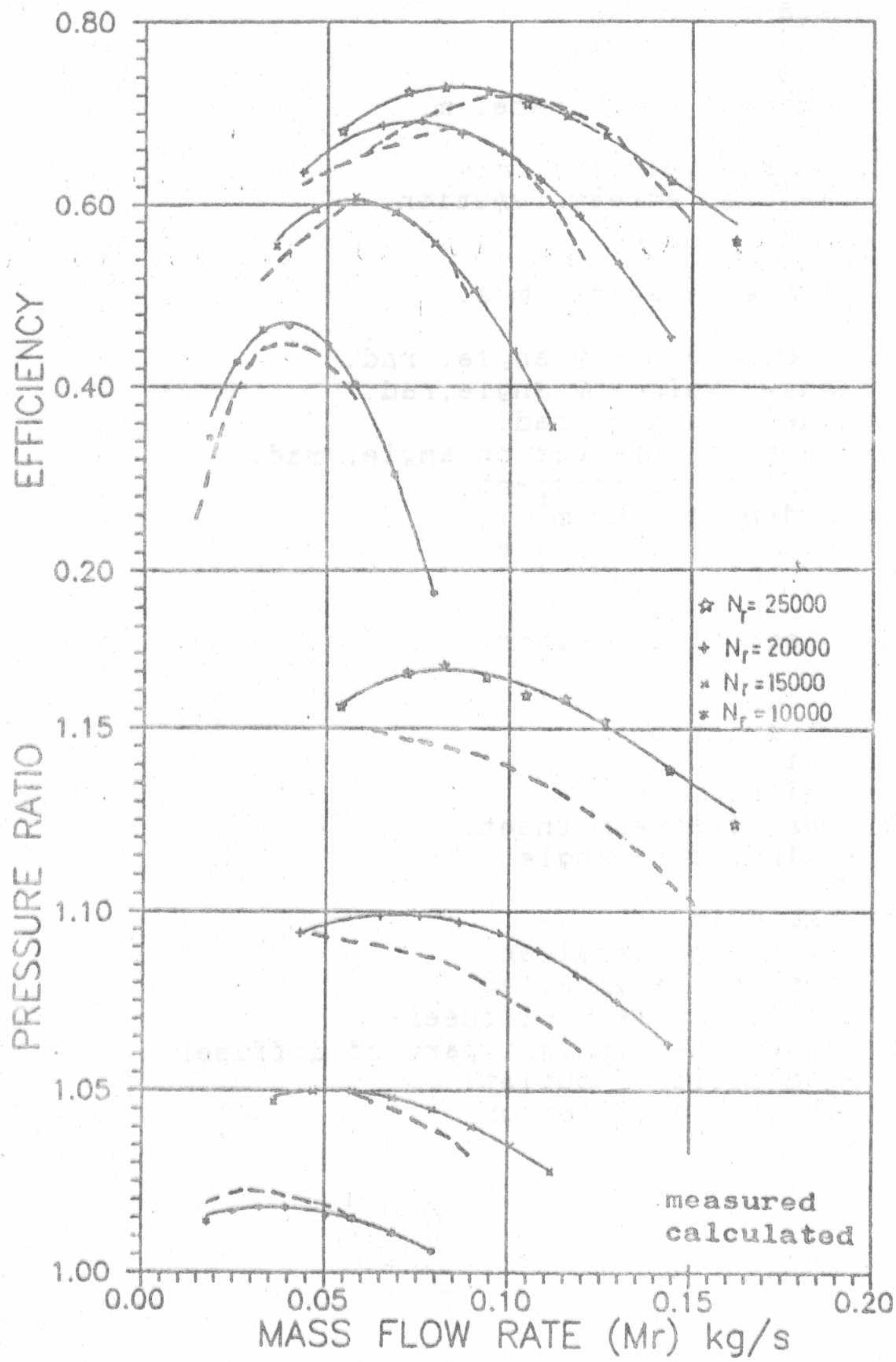

Fig.(2), Experimental and Theoretical Characteristics.

NOMENCLATURE

a	area, m^2.
α	coefficient.
b	channel width, m.
c	absolute velocity. m/s.
k	constant.
k	empirical factor.
L	losses.
m	meridional coordinate, m.
N	speed, 1/s.
u	prephiral velocity, m.
u	normalized stream function.
R	radius, m.
r	coordinate, m.
w	relative velocity, m/s.
α	absolute velocity angle, rad.
β	relative velocity angle,rad.
δ	incidence angle, rad.
ϑ	coordinate orientation angle, rad.
ζ	local loss coefficient.
ρ	mass density, kg/m^3.
ω	angular velocity, 1/s.
c	compressor, collector.
D	disc.
F	friction.
f	flow, friction.
i	incidence.
m	meridional.
n	number of stream sheet.
o	zero incidence angle
r	rated.
s	shock, skin.
1	inlet to the impeller.
2	outlet of the impeller.
3	inlet to vaneless diffuser.
33	inlet to the straight part of diffuser.
4	bladed diffuser outlet.
5	inlet to the collector.