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PRESSURE MEASURMENTS IN THE BLADE PASSAGES OF A RADIAL INFLOW TURBINE

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ABSTRACT

Pressure measurements were carried out in an unshrouded radial inflow turbine in a closed circuit test facility, at nozzle inlet and exit, vis-a-vis rotor exit. Besides, blade surface pressures on the nozzle and rotor blades, wall static pressures on the back plate of nozzle and rotor blade passages, shroud static pressure variation along the casing from rotor inlet to exit were also measured for four different mass flows through the turbine. The increase of nozzle exit angle resulted in an increase of the stator loss coefficient at any mass flow rate. Rotor pressure measurements showed an adverse pressure gradient on the suction surface close to the hub region. Compared to the total pressure levels near the hub and shroud of the rotor passage, the pressure was higher at the mid height. The swirl at the rotor exit is somewhat stronger near the tip region as compared to the hub and the average loss is more towards the tip.

NOTATION

ch	:	chord
C	:	Velocity
C_p	:	Coefficient of Pressure
i	:	Incidence angle
INB	:	Instrumented Nozzle Blade
LE	:	Leading Edge
P	:	Planes at which measurements made
p	:	Static pressure
PS	:	Pressure surface
SS	:	Suction surface
STN	:	Measuring Locations chosen (STN1, STN2, STN3, STN4)
TE	:	Trailing Edge
x	:	Distance along the variable height
z	:	Number of blades/blade height
α	:	Fluid exit angle wrt to tangential direction at stator exit (deg).
θ	:	Angle about the casing Inlet

Subscripts

h	:	Hub
N	:	Nozzle
o	:	Turbine stator inlet
t	:	Tip
tot	:	total condition
1	:	stator exit or rotor inlet
2	:	turbine exit

INTRODUCTION

Radial inflow turbines have gained preference in certain applications over axial flow machines in the recent past, particularly when compact power sources are required. Nozzle exit angle is an important design parameter in radial inflow turbines and the flow angle at the nozzle exit varied with rotor speed and gas velocity. In the present work, experimental investigations were carried out on a radial inflow turbine to study, (i) the effect of the nozzle exit angle on the flow field in the inter space between the nozzle exit and rotor inlet, (ii) the pressure distribution on the nozzle and rotor blade surfaces, shroud static pressure variation along the casing from rotor inlet to exit, wall static pressure distribution on the back plate of nozzle and rotor blade passages and (iii) the flow condition at the rotor exit for different mass flows.

The earlier works of Hiett and Johnston (1963), Futral and Wasserbauer (1970), Nusbaum and Kofskey (1969), Kofskey and Nusbaum (1972), Futral and Holeski (1970), Mclallin and Haas (1980), Whitfield (1990), Eroglu et al (1990), Baines et al (1990, 1991), Hayami et al (1990), Zangench-Kazemi et al (1988) and Borges (1986) provided the designers with an invaluable understanding of how the basic design parameters affect the overall performance of the radial turbines. However, the presence of a relatively high loss region near the shroud at the exit plane of the turbine has not been very well resolved. The extent of the wake in the blade to blade plane is not clearly investigated. Most authors attribute this phenomenon to the presence of boundary layer separation on the shroud, or to the effect of secondary flows accumulating the low momentum fluid near the shroud. In the present experimental studies, an attempt is made to understand the characteristics of flow at the nozzle exit and the rotor exit and the rotor passages with the help of pressure measurements made inside the blade passages of nozzle and rotor.

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EXPERIMENTAL APPARATUS

An experimental facility for testing the radial inflow turbine was designed to operate in a closed circuit as illustrated in the schematic diagram, Fig. 1. The facility can be operated in a closed loop, using any suitable gas, Freon or air as the working fluid. However, for the present studies;, air was used as the working fluid. An orifice meter designed as per BS 1042 (1981) has been fitted with sufficient upstream and downstream straight pipe lengths for the purpose of measuring the volume flow through the turbine.

The turbine rotor is mounted on a shaft cantilevering inside the casing. To prevent leakage of working fluid from within the casing and the shaft, a mechanical seal is fitted. The assembly of the radial inflow turbine is shown in Fig. 2, comprising of the shaft, its supporting system, the casing, nozzle blade ring and the turbine rotor. The nozzle blades were instrumented for the measurement of surface static pressure distribution. One typical nozzle passage was chosen for this purpose and the static pressure tappings were provided on the back plate covering one nozzle passage. The static pressure signals from the tappings on the nozzle blade and back plate were transferred from inside to the static scanivalves through a pressure transfer plate, fixed on the turbine casing at a convenient location where it does not disturb the flow in the turbine. The nozzle casing and the rotor shroud casing were provided with wall static pressure tappings in the meridional plane, for measuring static pressure from inlet of the nozzles to the rotor exit. The clearance between the turbine rotor and the shroud casing was about 2 mm. Fig. 3 shows the details of the shroud casing while the turbine impeller with exducer is shown in Fig. 4. At the nozzle exit, provision was made for the measurement of flow condition by traversing a three hole wedge probe for obtaining the wake characteristics at the trailing plane of the nozzle blades, Fig. 5.

The rotor has been instrumented for the study of blade surface static pressures, rotor hub (back plate) surface static pressures and the total pressure within the blade passage from rotor inlet to rotor exit at different heights of the passage at chosen locations. To minimize imbalance, the above tappings were distributed suitably in the rotor blade passages. However, the instrumented rotor was dynamically balanced. All the pressure tappings were connected through hypodermic steel tubes to a rotating scanivalve, the output of which in turn is connected to a micro manometer for recording the pressures. Fig. 6(a) shows the disposition of static pressure tappings on the rotor blade surfaces while Fig. 6(b) shows the fixing of the total pressure probes (Kiel probes) within the rotor blade passage.

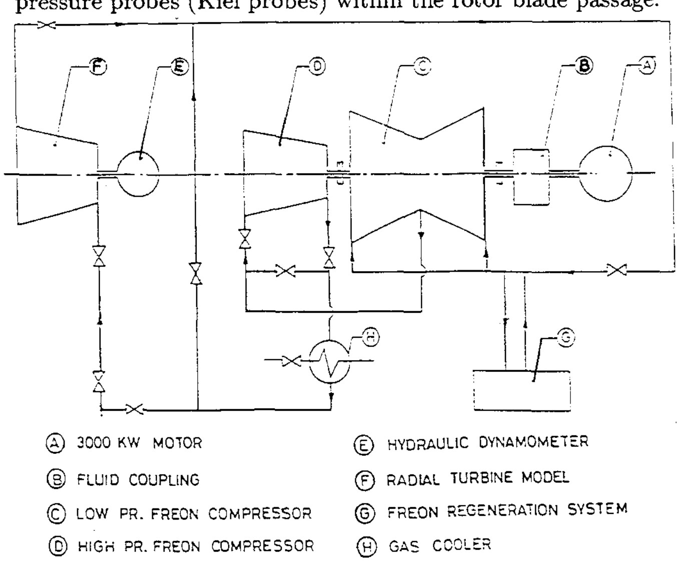

Fig 1 Schematic of Radial Turbine Testrig

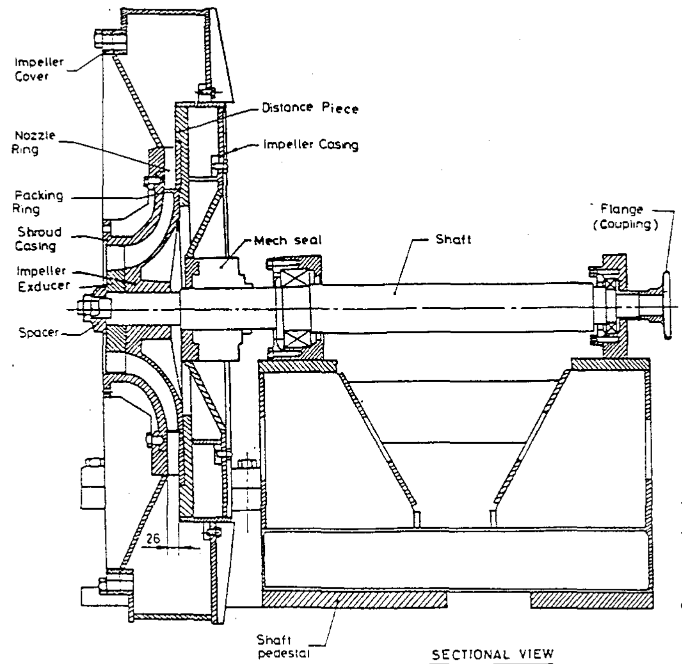

Fig 2 Radial Turbine Assembly

At the exit plane of the turbine rotor, a 5 hole probe was traversed from hub to tip, at one of the circumferential locations to determine the three dimensional flow pattern at the rotor exit for different mass flows. The turbine output power was measured using a hydraulic dynamometer at the test speed of the turbine.

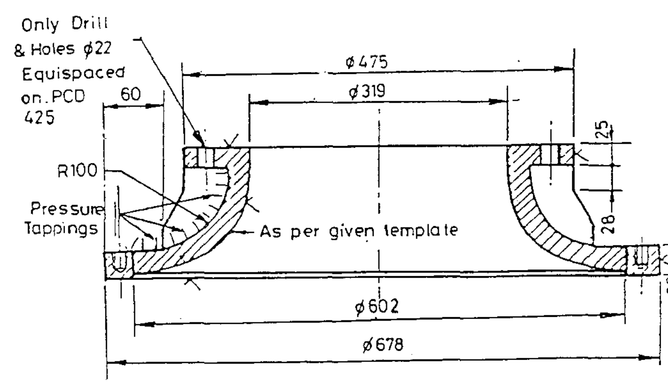

Fig 3 Shroud Casing

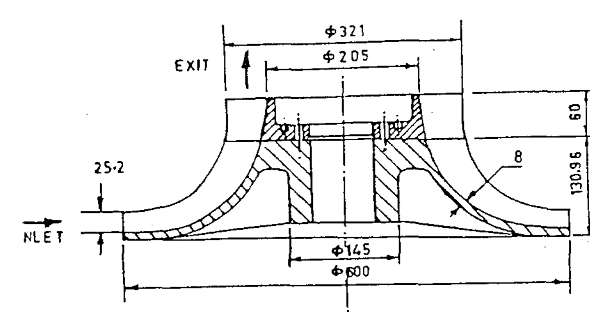

Fig 4 Turbine Impeller with Exducer

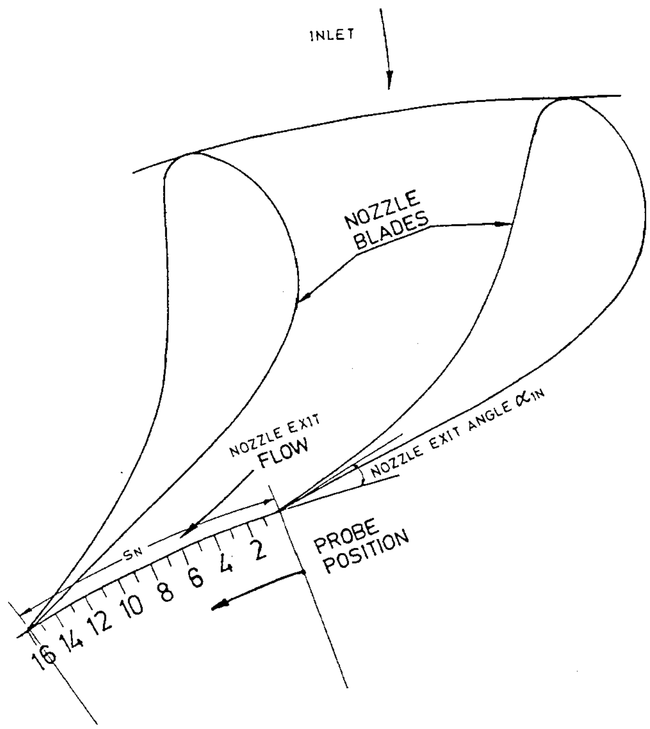

Fig 5 Wake Traverse Measurment at Nozzle Exit

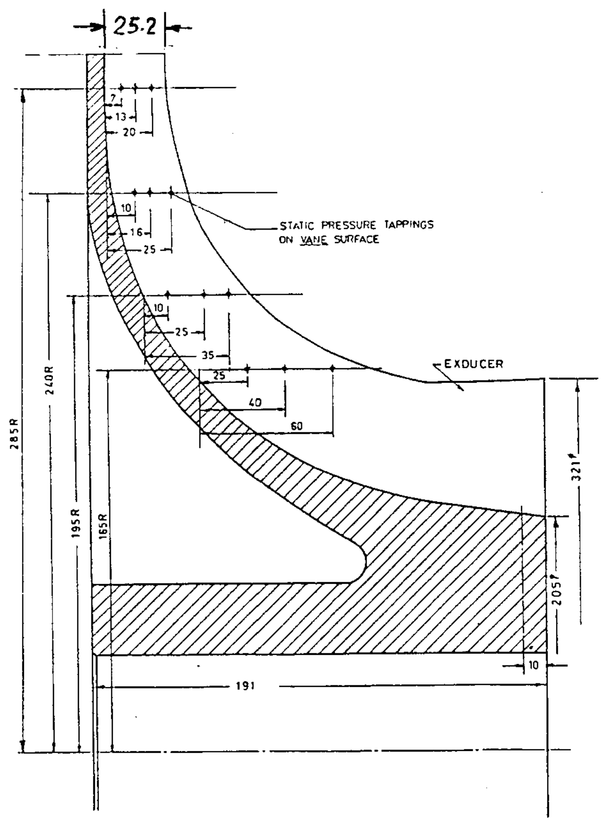

Fig 6(a) Disposition of the Static Pressure Tappings on the Vane Surfaces of the Rotor

Geometric details of the radial inflow turbine:

Number of Nozzle Blades	23
Nozzle exit angle, α_{1N}	14^0
Rotor inlet diameter	600 mm
Blade channel width at inlet	25 mm
Rotor inlet blade angle	90^0
Rotor exit hub diameter	205 mm
Rotor exit Tip diameter	321 mm
Number of Rotor blades	19
Rotor Shroud Running Clearance, τ_R	2 mm

(Angles are with respect to tangential direction)

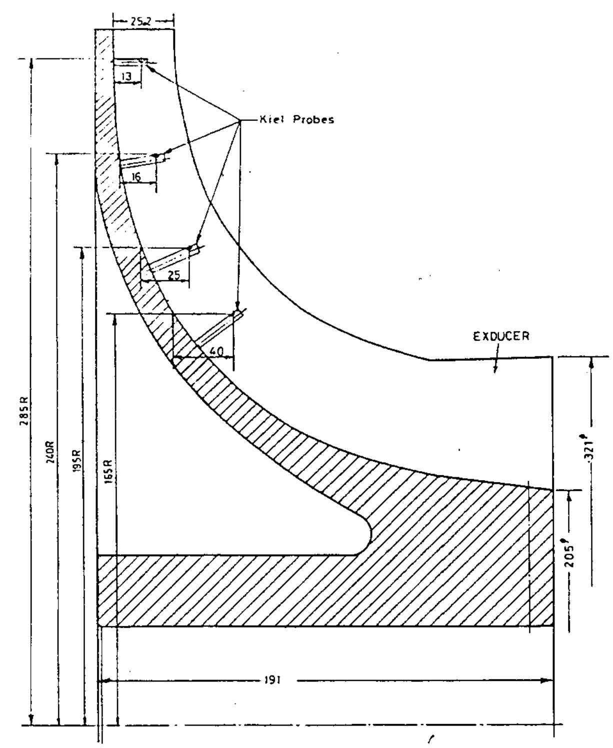

Fig 6(b) Typical Disposition of Kiel Probes in the Blade Passage (Meridional Plane)

RESULTS AND DISCUSSION

Velocity Variation at Nozzle Inlet:

To study the nature of flow at the nozzle leading edge plane, measurements of total pressure, static pressure and flow angle were made using a three hole wedge probe at the inlet of nozzle ring. Fig. 7 shows schematically the variation of incidence at nozzle inlet around the casing and the variation of inlet velocity along the circumference from top to bottom of the casing is shown in Fig. 8. The variation of the velocity indicates that the flow rate is not very much uniform in all the nozzle passages and hence the turbine performance is likely to be affected. Benisek et al (1990) who did experiments on a radial turbine inlet casing also stated that in spite of relatively large housing upstream of the nozzles, there was a significant variation of static pressure around the circumference. However, this variation was found to get very much reduced downstream

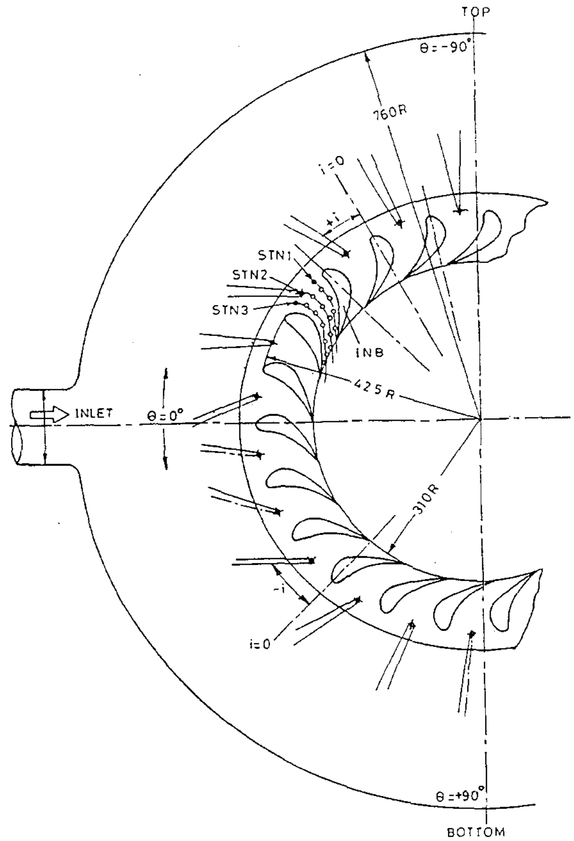

Fig 7 Variation of Incidence at Nozzle Inlet Around the Nozzle Ring

of the nozzle ring, thereby indicating a smaller influence on the performance of the turbine rotor due to this non uniform flow at nozzle inlet.

Using Glassman's (1976) design analysis program, the effect of change of incidence on the nozzle blade performance was studied. It was found that the variation of the inlet angle up to 10 degrees in the neighbourhood of zero incidence had minimal effect on the nozzle loss coefficient while at high incidence angles around 50^0 to 60^0 the loss coefficient increased by about 8% suggesting that the incidence angles are to be maintained low.

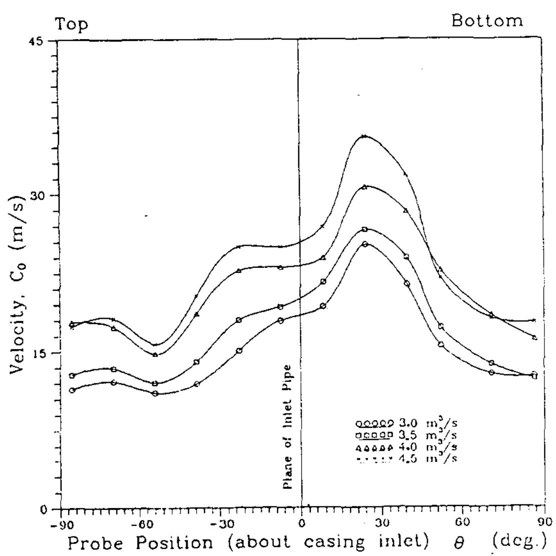

Fig 8 Variation of Inlet Velocity Along the Circumference of the Casing

Pressure distribution on the Nozzle Blade Surfaces:

Fig. 9 shows the variation of coefficient of static pressure on the nozzle surface for the nozzle exit angle α_{1N} of 14^0 in the plane P_1 close to the back plate for four different volume flows. The blade loading is found to increase on the nozzle blading with an increase in flow volume through the turbine.

Considering similar distribution of static pressure coefficient in the mid plane P_2 and close to the front plate, plane P_3 (not shown here), it is observed that there is a sudden deceleration of the flow upto 20% of the blade chord on the pressure

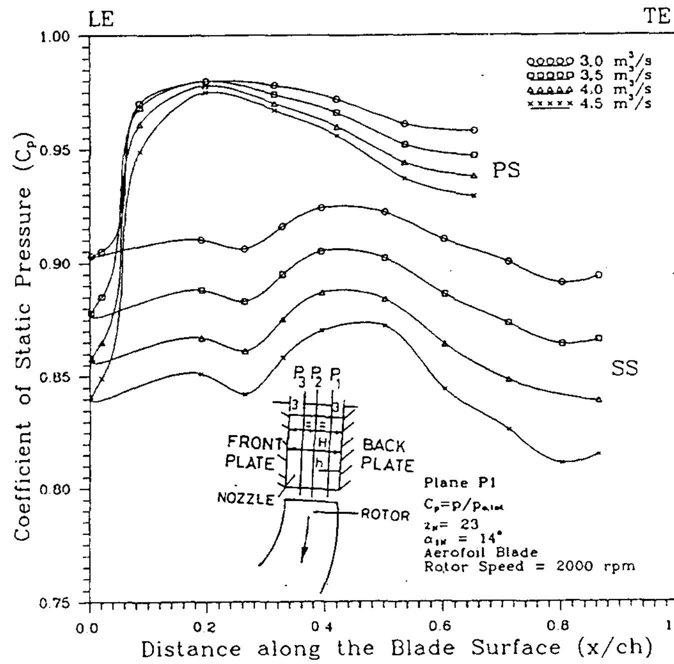

Fig 9 Variation of Coefficient of Static Pressure on the Nozzle Surface

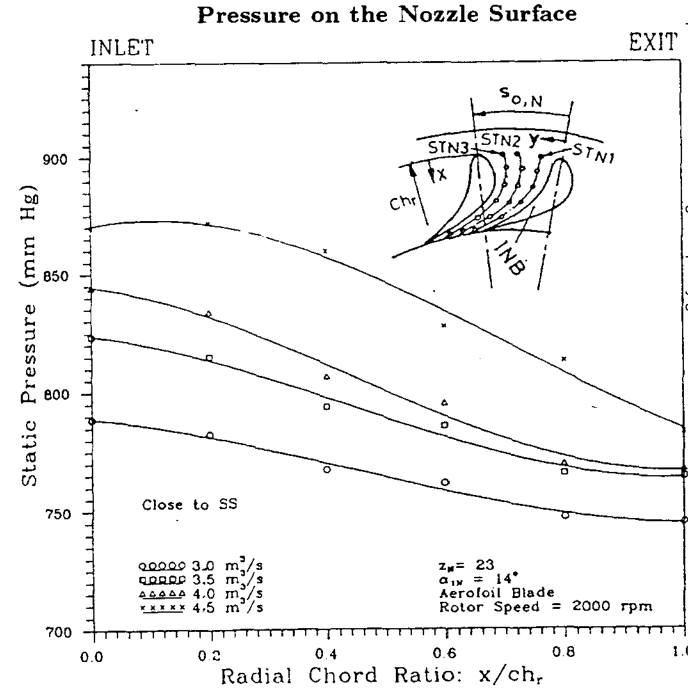

Fig 10 Variation of Static Pressure Along the Passage on the Back Plate

surface and accelerates later towards the trailing edge. But on the suction surface very small deceleration was obserbed upto 50% of the chord before accelerating rapidly towards the nozzle trailing edge/exit.

Close to the nozzle blade suction surface, the variation of static pressure on the back plate is shown in Fig. 10 for four volume flows. It is observed that static pressure decreases gradually from inlet to exit accelerating the flow. However, at mid passage the decrease of pressure is less steep initially. But close to the pressure surface, the static pressure remained nearly the same for almost upto 30% of the passage from inlet before falling steeply.

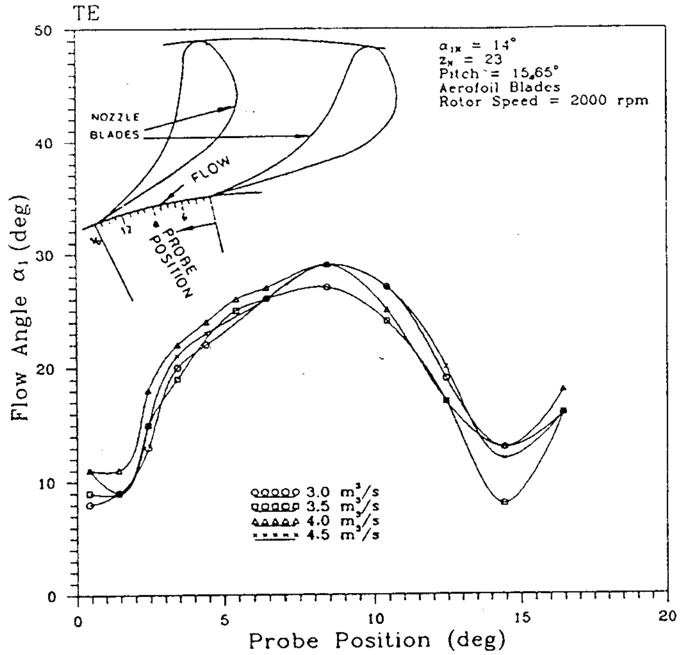

Fig 11 Variation of the Flow Angle Over a Pitch in the Wake of a Nozzle Blade

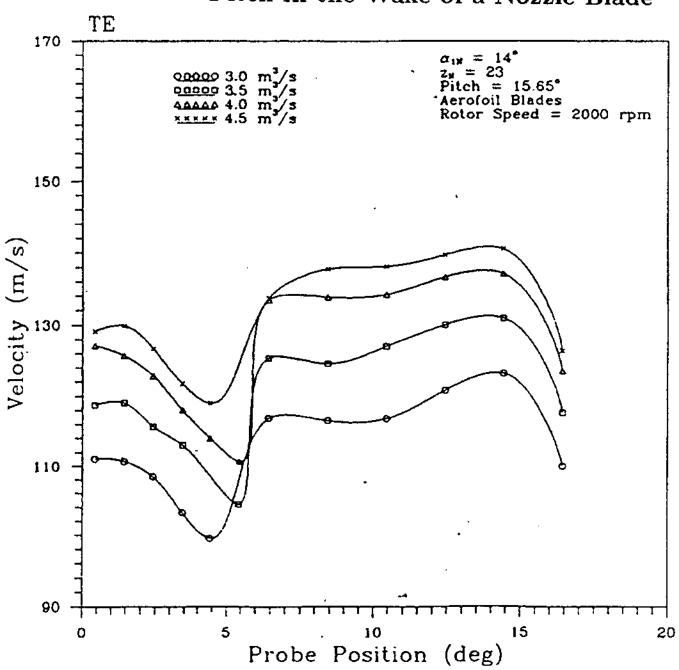

Fig 12 Variation of Velocity Over a Pitch in the Wake of a Nozzle Blade

Nozzle Exit Flow:

The flow condition in the interspace between the nozzle trailing edge and the rotor inlet was surveyed by traversing a 3 hole wedge probe from the back plate to the front plate of the nozzle blade ring and also in the circumferential direction a little over one blade pitch for three different volume flows. The variation of the nozzle exit flow angles obtained at the wake of the nozzles set at 14^0 exit angle is shown in Fig. 11. The maximum flow angle was 29^0 and the minimum was 10^0 while the nozzle exit angle was 14^0. The variation of the measured mean velocity along the circumferential direction is shown in Fig. 12. The small differences in velocities between the wakes and the free stream imply rapid mixing, also noticed by Khalil et al (1976). Lower velocity regions indicate the existence of considerable amount of cross flows carrying the low energy boundary layer fluid to the suction side corner.

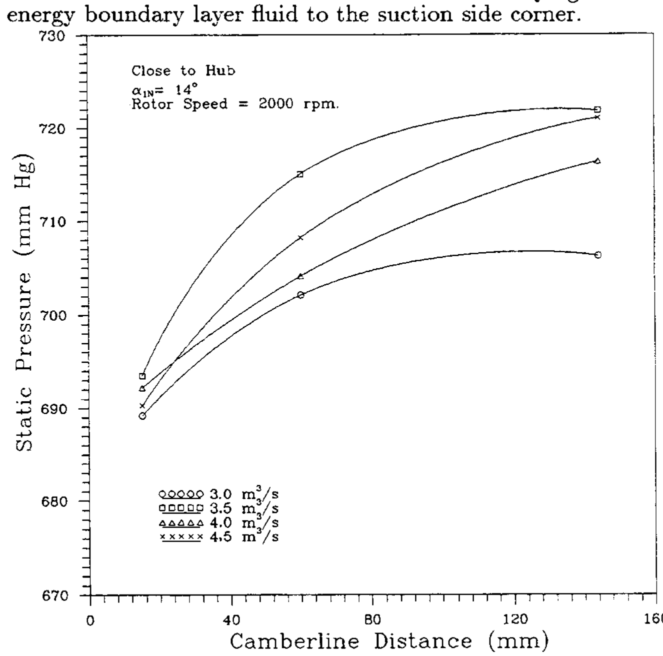

Fig 13(a) Variation of Static Pressure Along the Camberline on the Suction Surface

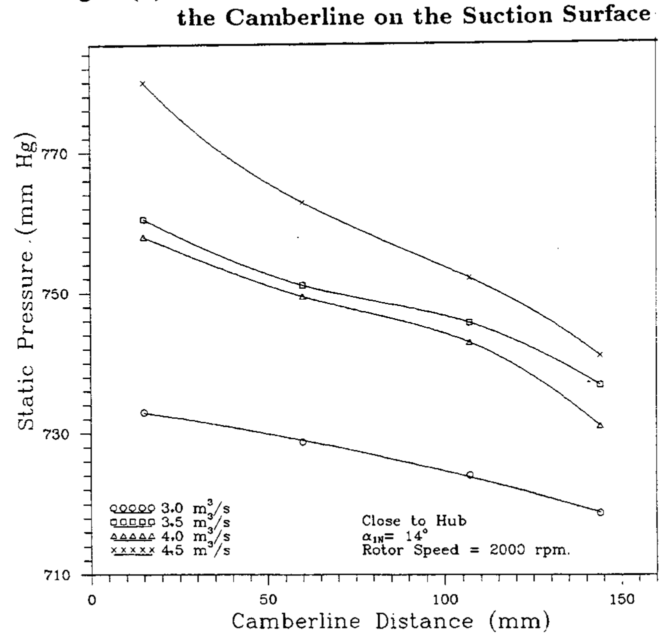

Fig 13(b) Variation of Static Pressure Along the Camberline on Pressure Surface

Pressure Distribution In Rotor Passages:

Static pressure distribution on the rotor blade and hub surfaces and the total pressure distribution along the mid passage from inlet to outlet was measured with the help of a rotating scanivalve and a Furness Control micro-manometer for four different volume flows. Fig. 13(a) and (b) show the variation of static pressure on the suction and pressure surfaces respectively from inlet to the turbine exit, close to hub surface. An adverse pressure gradient is noticed on the suction surface, which could be causing flow separation on the suction surface. From the pressure measurement, it is also observed that the static pressure on the rotor blade at mid height was higher compared to that near the hub and shroud surfaces. The wall static pressures had very little difference at the rotor inlet on the suction surface.

Fig. 13b shows that the variation of static pressure along the pressure surface close to the hub surface. As the flow increased, the static pressure decrease was noticed very steeply. From the static pressure readings obtained from the blade surface, it was found that the variation of static pressure was gradual along the blade. The lower pressures close to the hub and shroud are due to the presence of boundary layers near the wall surface. Close to the leading edge, there is a large low momentum region. Static pressure decreases from pressure side to the suction side of the blading and the isobaric lines lie in the radial direction in the case of zero incidence flow.

The flow separation regions are reduced when the flow rate increases at a constant tip speed. The relative velocities decrease linearly from suction side to pressure side of the channel. In the case of constant tip speed of the impeller, the flow deviation in the direction opposite to the rotation is apparent as the flow rate decreases, yielding larger separation. Flow behaviour is also found to depend upon the ratio of tip speed to flow rate. For larger flow rates and tip speeds, the isobaric lines are inclined a little in the direction of rotation, and the maximum static pressure region is shifted to the vicinity of blade pressure side near impeller inlet. Nearly uniform static pressures having maximum values within the passage are found in the vicinity of impeller inlet and the low pressure region exists in the interior of the passage near the suction side of the blades. In the interior of the passage, however the flow tends gradually to the radial direction and thus the static pressures decrease from the pressure side to the suction side of the blades.

Measurements of the flow field at inlet and exit of the rotor of the radial inflow turbine have added new data and produced new insights into the fluid dynamic processes of these machines, Baines et al (1991). Under equal admission conditions, the flow at inlet to the rotor showed some variation in velocity in the span wise direction due to the wake shed by the upstream nozzle blades and some distortion in the inlet casings, but in other respects the pattern is reasonably uniform. From the experimental and computational work of Dawes (1988), Kitson et al (1990) and Benisek et al (1990) concerning the rotor inlet flow field, the rotor flow processes could be understood better. These papers showed a strong evidence of the three dimensional vortex in the inlet region of the rotor, centered near the pressure surface, which caused the fluid to migrate from hub to tip on the pressure surface, and some motion from suction to pressure surface, particularly near the shroud, Fig. 14. This is caused by a pressure field due to the curvature in the meridional plane and the effects of rotationality.

The rotor passage vortex has its origin at the hub surface and is most strongly felt at the shroud. Benson et al (1971) described numerical solutions of the flow within a rotating impeller based on potential theory using a direct method of solution instead of by relaxation. There were differences between theory and experiment which could be accounted for by friction effects in the passage associated with boundary layer growth. Since the fluid particles adhere to the surface of the rotor blades and walls, one must consider the rotor boundary layers which are subject to the centrifugal and Coriolis forces. Benson et al (1968) using Stanitz's (1952) method for calculating inviscid flow through a rotor passage, show that flow reversals may take place on the driving face together with eddies in the blade passage.

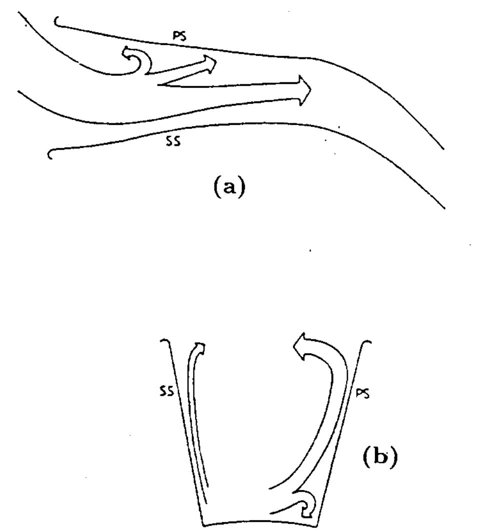

Fig 14 Secondary Flow in Rotor Inlet
(a) Blade-to-Blade Plane Near Hub
(b) Normal Plane Near Inlet

Flow Surveys at Rotor Exit:

The results of the flow surveys made at the rotor exit station are presented in Figs. 15 to 17. The turbine exit flow angle α_2 is measured with respect to axial direction and the exit flow angle is considered positive if the swirl is in the direction of impeller rotation and negative flow angle indicates that the exit swirl is in a direction opposite to the impeller rotation. Fig. 15 shows the variation of the exit flow angle from hub to tip for four different volume flows.

The flow angle is close to zero (flow nearly axial) from the hub to about 20% of the annulus at the exit and increases towards the blade tip to higher negative angles, $\alpha_2 = 20^0$, indicative of increased swirl velocities. The absolute flow angle variation is such that swirl velocities are slightly reduced in magnitude with increase of volume flow. The swirl at the rotor exit is some what stronger near the tip than near the hub. The increased intensity of swirl near the tips may be attributed to the accumulation of boundary layer near the tips and secondary flow in the flow passage of the rotor.

The region of high swirl near the tip may skew the exit flow. At the trailing edge, one can observe a region of high loss occupying a large part of the shroud and the shroud suction surface corner. Correlating experimental results and theoretical predictions, Zangeneh-Kazemi et al (1988), infer that the streamline curvature and Coriolis accelerations influenced the generation of secondary flows while the centrifugal accelerations could be partly responsible for the direction towards which secondary flows move the low momentum fluids.

In the radial part of the blade passage, near the inlet, the tangential component of the Coriolis acceleration is very large and is actually responsible for most of the blade loading in this section. In the radial to axial bend, the secondary flow is both due to both curvature and rotational effects; resulting in a complicated three dimensional flow. In the exducer the secondary flow is again due to both curvature and rotational effects. The rotational effect is due to the radial component of Coriolis acceleration which is in the direction of vorticity due to the blade surface boundary layers. This should result in a secondary flow which will try to move low momentum fluids from the hub towards the shroud. At the same time, the acceleration due to exducer curvature should result in a secondary flow moving the low momentum fluids from the pressure towards the suction surface.

Many investigators have observed the region of high loss near the tip at the exit plane. This has been of much importance in the study of turbine losses. Nusbaum et al (1969) by traversing a three hole probe at the turbine exit, stated that this loss region existed in design and as well in off-design operating conditions. It was noticed in the present case that the tangential component of velocity C_{2u} was marginal near the hub and later increased gradually towards the tip. It is opined that the boundary layer accumulation near the tip and the secondary flow in the flow passage of the rotor were the causes for the swirl velocity to increase towards the tip.

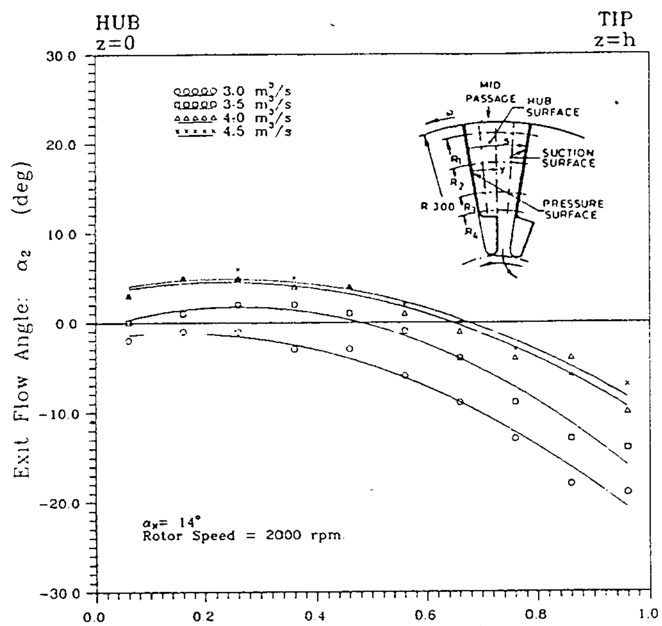

Fig 15 Variation of Absolute Flow Angle at the Turbine Rotor Exit

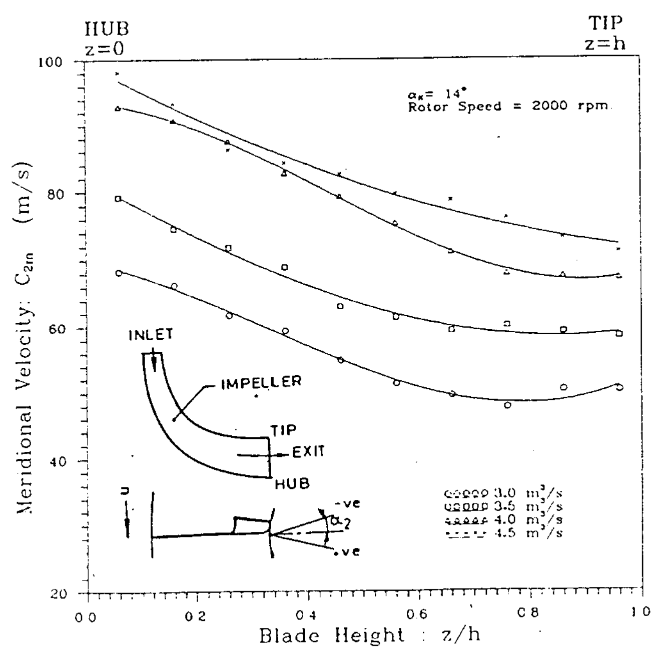

Fig 16 Variation of Meridional Velocity at the Turbine Rotor Exit

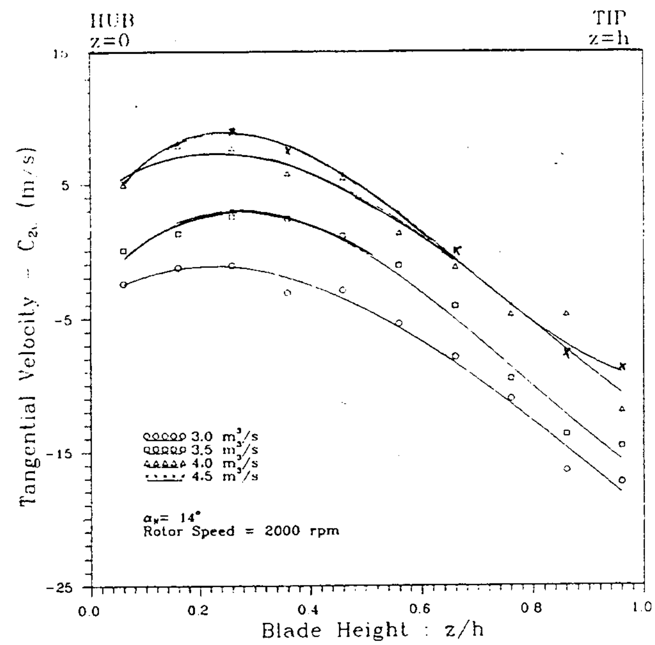

Fig 17 Variation of Tangential Velocity at the Turbine Rotor Exit

CONCLUSIONS

The aerofoil nozzle blades are found to give a high velocity coefficient with a low value of the total pressure loss coefficient. The flow field inside the nozzle blade passage is influenced by the upstream casing geometry and nozzle inlet flow.

Near the pressure surface the incidence is maximum at the mid height of the nozzle passage and near the front and back plates the incidence is minimum. The nozzle loss coefficient is found to be influenced by the incidence angle and the nozzle trailing edge angle.

The nozzle exit flow angle appeared to have very little scatter with respect to the volume flow and the inlet casing total pressure. The exit flow angle showed a variation from 10^0 to 29^0 while the nozzle exit setting angle is 14^0.

The increase of nozzle exit angle resulted in an increase of the stator total pressure loss coefficient at any mass flow rate and the loss coefficient remains nearly the same with the increase of mass flow. Rotor flow studies revealed that on the suction surface there was more adverse pressure gradient along the blade close to the hub region.

The absolute flow angle variation at the rotor exit is such that the swirl velocities are slightly reduced in magnitude with increase of volume flow. The swirl at the rotor exit is some what stronger near the tip than near the hub. The region of high swirl near the tip may skew the exit flow.

At the rotor exit, the trend of the total pressure variation was to decrease gradually from hub to tip. The average loss was more towards the tip meaning that there remains a high loss region towards the tip. The flow near the tip seemed to be much under turned. The increased intensity of swirl near the tips may be attributed to the accumulation of boundary layer near the tip and to the secondary flow in the flow passage of the rotor.

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