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An Investigation On Performance Characteristics of Radial In-Flow Turbo-Expander with Backswept Curved Rotor

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ABSTRACT
In this research, results of the designed radial inflow turbo-expander with back swept curve blade rotor have been presented. The spiral casing is designed considering constant mean flow velocity and the areas of cross-sections of the spiral casing are decreased gradually by reducing the width of the flow areas from inlet to end. The rotor is shrouded to cope up with centrifugal as well as bending load on the vanes of the rotor and avoid energy losses due to the tip clearance. The degree of reaction is selected as 0.25 in contrast to usual value of 0.5 at design point on the consideration of absolute velocity of air at nozzle exit needs to be higher than the rotor tip speed. The rotor shaft assembly is supported by externally pressured air bearing. The turbine with coupled brake centrifugal compressor, aerostatic radial and thrust bearing and test rig are developed for conducting experiments. The experiments are conducted with compressed cold air by varying mass flow rates and pressure ratios. At the pressure ratio of 1.6 the temperature drop of 12.5^{o}C of flowing air across turbo expander is achieved and 41% total to static efficiency is obtained. The findings of exhaustive experimentation with the developed turbo expander are illustrated in details in this paper.
Keywords: Air bearing, turbo expander, shrouded rotor, temperature drop

I. INTRODUCTION

In the recent years, a keen interest has been observed among the researchers in the field of high speed small turbo machines. A significant research work has also been carried out with single stage radial inflow turbo expanders (turbines). Certainly, radial inflow turbines dominate over axial flow turbines when the fields of interests are decentralized small power (less than 50 kW) generation, environment control of aircraft, solar thermal power generation, Organic Rankine Cycle, cryogenic refrigeration etc. This is because of machine’s simplicity in construction, compactness and high expansion ratio (~8:1) in single stage [1]. On the other hand, small turbo-expander running at very high speed creates a series of difficulties including design, fabrication, dynamic balancing of rotating system, supporting rotor assembly by bearing and certainly operation of the system [2].

The radial inflow turbines are mostly found with un-shrouded radial vane rotors owing to ease of manufacturing and considering the strength of vanes against centrifugal stress. The fluid flow through open impellers is strongly influenced by secondary flow phenomena of tip clearance space between rotor and casing [3]. It deteriorates the performance characteristics due to dissipation of energy by secondary flow phenomena arising from tip clearance.

The information on the design methodology of radial inflow turbine is available in open literature and in the textbook on turbo machines [4-6]. Design and estimation of performance characteristics of radial inflow turbines with radial bladed rotors have been reported in the literature [7–12]. The information on experimental findings of radial vane inflow turbines is also available in open literature. Hiett and Johnston [13] investigated experimentally the effect of major design parameters such as nozzle angle, tip width of the rotor, rotor size and shape on the performance of the turbine. Cho et al. [14] explored the feasibility of the utilization of small turbo expander as a substitute of expansion valves of refrigerator or air-conditioner and capturing the waste energy in throttling process. Doran et al. [15] experimentally correlated shroud radius and incidence angle at rotor entry for the radial turbine. Hayami et al. [16] investigated the effect of nozzle tip clearance on the performance of radial flow turbine. They found experimentally that efficiency decreases with the nozzle clearance and it is further magnified with increased blade loading due to less number of vanes in rotor.

The rotor is often designed in aforementioned radial inflow turbines such that specific work done by the fluid to the rotor is equal to square of the blade tip velocity (u). Consequently, blade tip velocity needs to be very high to deliver the higher specific work, but the induced centrifugal stress in blade materials imposes the limit of maximum operating speed. Moreover, the development of such high speed thrust as well as radial bearing remains a challenging issue for the designers and the users. Rotor, while running at such high speed, often gets locked in the bearings and creates abnormal vibration causing damage to the rotor shaft. To reduce such difficulties with high speed rotor, the design of radial in-flow turbine with rotor of backward curve vane is better option. However, the information on the performance characteristics of radial in-flow turbine with rotor of backward curve vane is hardly found in open literature.

In this research exercise, turbine is developed with the incorporation of some new features differencing from conventional design of unshrouded radial vane inflow rotor. The rotor (tip diameter, 63.5 mm) is designed with back swept curved vanes to exploit the maximum available energy with a reducing effect on the rotational speed of shaft. The rotor is shrouded to cope up with centrifugal as well as bending load on vanes of the rotor and to avoid energy losses due to tip clearance. The degree of reaction is selected as 0.25 in contrast to usual value of 0.5 at design point on the consideration of absolute velocity (c_2) of air at nozzle exit needs to be higher than rotor tip speed (u_2). The rotor shaft assembly is supported by aerostatic bearing. The turbine, brake centrifugal compressor, aerostatic radial and thrust bearing and test rig are designed and developed completely utilizing institute's in-house facilities. A full fledge experimental work has been conducted on the developed turbo-expander with variation of inlet pressure and mass flow rate of supplied compressed air. The results of the experimentation are explained in this paper with a hope to provide some useful information to designers as well as users.

II. GENERAL DESIGN CONSIDERATIONS OF TURBO-EXPANDER

The closed (shrouded) type and curved bladed rotor interfacing with stator nozzles and spiral casing have been designed and fabricated with Aluminum alloy for the experimental exploration. The rotor vanes are provided with positive incidence angle at entry such that tangential velocity of the fluid is higher than blade tip speed. Thus, the blade tip speed of rotor i.e., rotational speed of the rotor with back swept curve vanes becomes less than that of rotor with radial vanes for the same expansion ratio. The degree of reaction is selected as 0.25 in contrast to usual value of 0.5 at design point on the consideration of absolute velocity (c_2) of air at nozzle exit needs to be higher than rotor tip speed (u_2). Externally pressurized air journal and thrust bearing are suitably utilized to support common shaft of turbine and compressor. The assembled view of turbine and brake compressor is shown in "Fig. 1". The spiral casing is developed considering constant mean flow velocity and the flow areas decreased by continual decrement of width of cross sections from inlet to the end. Thus the radial height of the flow passage of the spiral casing remains constant.

III. EXPERIMENTATION

A test rig has been developed in the institute to study the performance characteristics of the developed turbo expander. The rotor of the turbine interfacing with stationery nozzle is housed in the spiral casing. The impeller of the centrifugal compressor is also mounted on the other end of the turbine shaft. The impeller of the centrifugal compressor interfacing with radial diffuser is encased in the spiral volute. The common shaft is supported by externally pressurized air journal and thrust bearing. The brass pad type thrust bearings are provided both in compressor and turbine side to absorb the balance axial load arising from impeller of the compressor and rotor of the turbine. The radial clearance in the journal bearing is 24μm. A schematic arrangement is explained in "Fig. 1".

The schematic diagram of the test rig of turbo expander is presented in "Fig. 2" respectively. The detail of the turbine design parameters are listed in "Table 1".The screw compressor supplies pressurized air to the header of the test rig. The compressed air is distributed from the header pipe line to the turbo expander and as well as to the air bearings. The air flow rate and the pressure are controlled by manually operated ball valves provided in the respective inlet pipe lines of turbine and air bearings. The supplied air to the orifice area of the journal bearing is distributed to shaft and thrust pads.

Evaluation of performance characteristics of the developed turbo expander at varying inlet operating condition is carried out. During experimentation air static pressure at the entrance of the turbine is varied from 0.111 to 0.16 MPa. The pressure and temperature of inlet pipeline of diameter 25 mm (NB) for turbine are measured by respective transmitters. The exhaust from the turbine is diffused through an 80 mm (NB) diameter discharge pipe to the atmosphere. The pressure, temperature and mass flow rate in the exhaust pipe line are also measured. The pressure, temperature in the air supply line of the bearings is also recorded. The brake centrifugal compressor draws air from atmosphere through the suction pipe of diameter 80 mm (NB). The pressure, temperature and mass flow rate of inlet air to the compressor are measured by respective transmitters in suction line. The flow rate in the discharge pipe line is controlled by a manually operated ball valve. Pressure and temperature are also noted in the discharge line of the compressor. During experimentation on the turbo expander, the compressor is always operated in the safe zone avoiding surging.

1	Turbine rotor	9	Diffuser
2	Nozzles	10	Discharge Volute
3	Spiral Casing	11	Compressor Discharge Flange
4	Turbine Inlet flange	12	Turbine Discharge Diffuser
5	Thrust Bearings	13	Turbine Discharge Flange
6	Radial Bearing	14	Compressor Suction pipe
7	Pressurized Gas Inlet Port	15	Suction Flange
8	Compressor impeller	16	Supports.

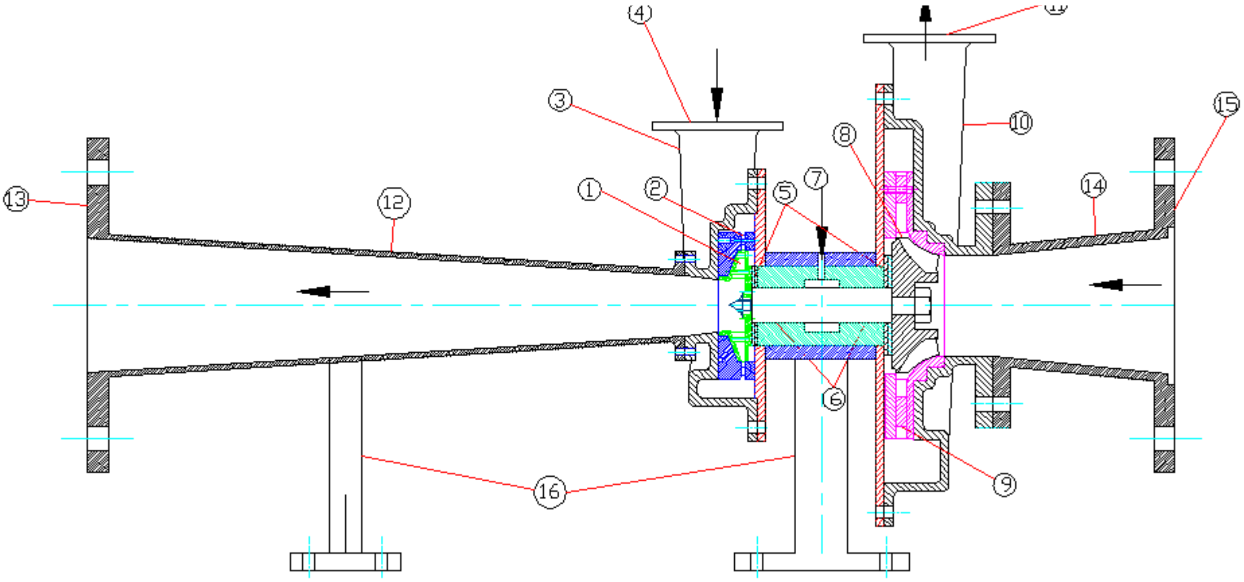

Fig.1 General Assembly view of Turbo expander and Brake Compressor

Legend

PF: Primary Filter
SC : Screw Compressor
NRV: Non Return Valve
D: Drier
(Transmitter)

R: Air Receiver
S: Secondary Filter
PI: Pressure Indicator
TI : Temperature Indicator (Transmitter)
DPI: Differential Pressure Indicator

OP: Orifice Plate
T: Turbo Expander
C: Brake Compressor
HP: Header Pipe

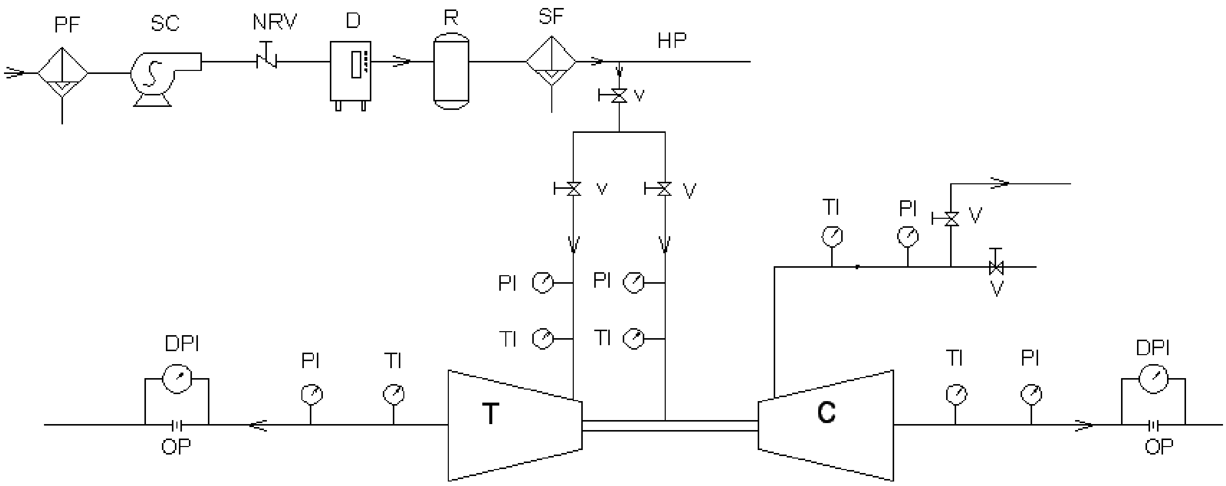

Fig. 2 Schematic diagram of test rig for turbo expander

Table 1 Details of the Turbo Expander

Sl	Design Parameters	
1	Fluid	air
2	Mass flow rate	0.05 kg/s
3	Expansion ratio	1:3
4	Power(cold test)	2.0 kW
5	Rotor tip dia	63.5 mm
6	Rotor exit dia	31.7 mm
7	Blade width at inlet	2 mm
8	Blade width at exit	7.6 mm
9	Inlet blade angle (w.r.t. tangent)	25^o
10	Exit Blade angle	30^o
11	No. of vanes in rotor	20
12	No. of vanes in nozzle ring	27
13	Nozzle angle w.r.t tangent	10^o
14	Shaft diameter	20 mm
15	Journal and bearing radial clearance	24 μm
16	Suction pipe	25 mm (NB)
17	Discharge pipe	80 mm (NB)

IV. INSTRUMENTATIONS

The test rig is instrumented at the relevant points to measures air mass flow rate, temperature, pressure and rotational speed of the turbine. The schematic arrangement of the test rig along with instrumentation is shown in "Fig. 2" and details of the instruments are listed in "Table 2. The in-line piezoresistive pressures sensors are used to measure pressure at inlet and discharge pipe of turbine. Temperatures are measure by temperature sensors of Resistance Temperature Detectors (RTD) type at the relevant points of the test rig. The temperature and pressure measured by this rig are static only.

The flow rate is measured by recording the differential pressure by piezoresistive differentiation pressures transmitter across the orifice plate. The rotational speed of the turbine rotor is recorded by infra red speed sensor and all the data are stored in a computer.

Table 2 Details of Instrumentation

Sl	**Specification**	
1	**Pressure Indicator (Transmitter)**	
	Make	Honeywell
	Model	STG 14L
	Range	0 to 0.6 MPa
	Accuracy	0.0375 %
2	**Differential Pressure Indicator (Transmitter)**	
	Make	Honeywell
	Model	STD 120
	Range	0 – 1000 mmWC
	Accuracy	0.0375 %
3	**Temperature Indicator (Transmitter)**	
	Make	Honeywell
	Model	STT25M
	Range	-200 to 400^oC
	Accuracy	0.15C
4	**Digital Panel Meter**	
	Make	Monarch Instrument
	Model	ACT-3
	Sensor	Infrared IRS 5W
	Range	5 to 99999 rpm
	Accuracy	0.0015%

V. RESULTS & DISCUSSION

In order to evaluate the thermodynamic performance behavior of the turbo expander, a series of tests with varying input of compressed air were conducted and the recorded data are analyzed.

The variation of rotational speed of the turbine versus pressure ratio is as shown in "Fig. 3". The rotational speed of the turbine is observed to increase at slower rate from 12000 to 13500 rpm, very sharply from 13500 to 38000 rpm and afterwards speed increases again at slower rate with the pressure ratio

Experimental data shows that temperature drop between turbine inlet and outlet increases with increase of pressure ratio. The temperature drop against the pressure ratio is plotted in "Fig. 4". The pressure of exhaust air was maintained at atmospheric (0.1 MPa) and air temperature of 26.2^oC was maintained at inlet of the turbine during experimentation. Air pressure of 0.112 MPa at turbine inlet causes a temperature drop of 3 ^oC and temperature drop of 12.5^oC is attained at the inlet air pressure of 0.16 MPa. The total temperature, pressure and total to static efficiency were calculated as follows.

$$T_0 = T + \frac{c^2}{2c_p}, \quad P_0 = P + \frac{c^2}{2} \text{ and } \rho = \frac{P}{RT}$$

$$\eta_{ts} = \frac{T_{01} - T_{02}}{T_{01} - T_{2s}}$$

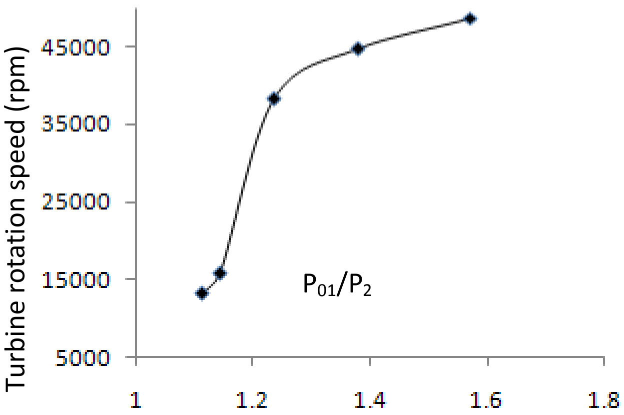

Fig. 3 Turbine Speed variation with pressure ratio

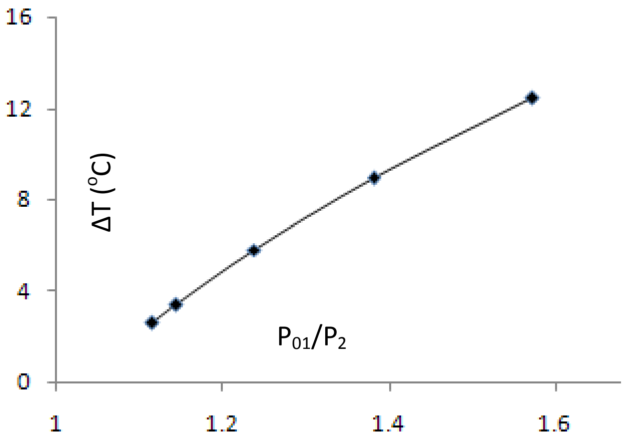

Fig. 4 Temperature drop across turbine with pressure ratio

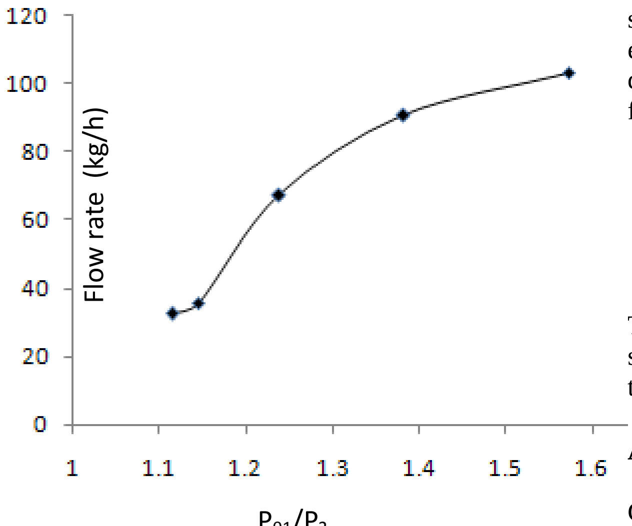

Fig. 5 Turbine Mass h pressure ratio

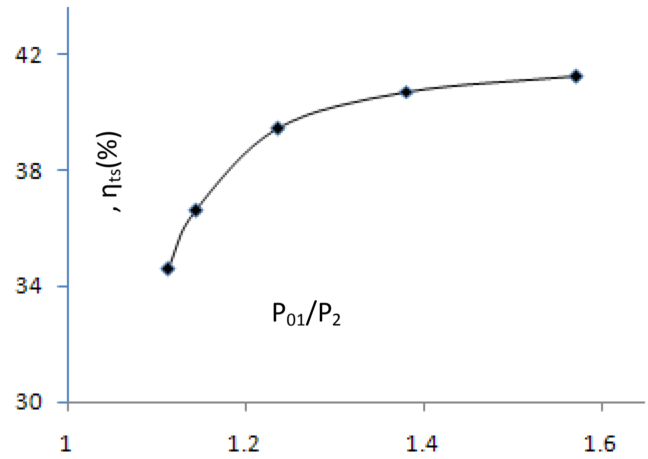

Fig. 6 Turbine Efficiency with pressure ratio

The mass flow rate against pressure ratio is plotted in "Fig. 5." Certainly the mass flow rate through the turbine increases with pressure ratio of air across the turbine until the choked flow condition in the nozzle. In the choked condition mass flow rate through the turbo expander remains insensitive with increase of inlet pressure for the specified configuration of nozzle geometry and critical pressure ratio. The total to static efficiency against pressure ratio is also drawn in "Fig. 6". It is observed that efficiency increased with the pressure ratio and it attains a value of 41% at a pressure ratio of 1.6

VI. Conclusion

An inward radial flow turbo expnader with bakswept vaned rotor was designed and developed in house with its full experimental set up. A series of experiments was conducted by varying mass flow rates and pressures ratios of the supplied compressed air. The developed turbo-expander exhibited smooth operation and the details of the major findings are summarized as follow:

1. The rotor of the turbo-expander with aerostatic bearings rotates with the speed of 48000 rpm at the pressure ratio of 1.6.
2. The temperature drop of 12.5°C was attained at pressure ratio of 1.6 for the devloped turboexpander.

The experimental findings are certainly the stair steps for further improvement of the turboexpanders.

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NOMENCLATURE

A	flow area, m^2
Cp	constant pressure Specific heat, J/kg/K
c	absolute velocity ,m/s
c_θ	tangential component of absolute velocity ,m/s
c_m	meridional velocity ,m/s
k	specific heat ratio
K	constant
NB	nominal bore
$\dot{m}$	mass flow rate, kg/s
P	Pressure, Pa
T	temperature, K
Tq	torque, N-m
Tq_B	Balance torque, N-m
R	characteristics gas constant, J/kg/
Re	degree of reaction
u	peripheral velocity, m/s
w	relative velocity, m/s
ρ	density, kg/m^3
ϕ	flow coefficient
α	flow angle
β	blade angle
σ_s	slip factor
r	radius, m
ω	angular speed, rad/s
η	efficiency (%)

Subscripts

1	nozzle inlet, turbine, impeller inlet, compressor
2	rotor inlet, turbine impeller outlet, compressor
3	rotor exit, turbine Discharge volute exit, compressor
t	turbine
c	compressor
s	isentropic
ts	total to static

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