EFFECT OF TIP CLEARANCE ON PERFORMANCE OF A CENTRIFUGAL COMPRESSOR

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Abstract

The centrifugal compressor is to study the effect of tip clearance on the performance characteristics and the wall static pressure for a different flow co-efficient. The method of testing the compressor is run at a constant speed at 2000rpm. The tip clearance is varied by using spacers. The volume flow rate is varied with the help of throttling device to conduct the performance test. The performance characteristic of the centrifugal compressor showing the variation of discharge pressure with volume flow rate is plotted. Obtaining the performance characteristics showing the variation of discharge pressure with volume flow rate for different tip clearance, viz. □=2.2%, 4%, 6.1% and 7.9%. Measurement of periodic pressure at various tip clearance viz. □=2.2%, 4%, 6.1% and 7.9%. For each tip clearance pressure measured in radial location of impeller at 6 positions for different flow co efficient values. Five flow coefficients viz., ϕ =0.40, =0.34 (both above design flow), =0.28 (near design flow), =0.21=0.18 (both below design flow) and four values of non-dimensional tip clearance viz., □=2.2%, 4%, 6.1% and 7.9%, are chosen for experimental work. The objective of the research work is to measure the periodic variation static pressure on the casing over the rotor at different values of tip clearance and flow coefficients. With the availability of these data, it is possible to improve the tip clearance flow in centrifugal compressor.

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1. INTRODUCTION

Tip clearance in centrifugal compressor causes the leakage of high pressure fluid from pressure surface to suction surface of the impeller blade, making the flow field highly complex and affecting the performance. Hayami (1997) has found from his experiments that axial movement of the casing has better efficiency over the movement of casing in radial and axial directions. Radial movement of casing increases clearance at inducer, which reduces the operating range. The tip clearance studies are conducted to understand the flow behavior in order to minimise the effect of tip clearance. Pampreen (1973), Mashimo et al. (1979), Sitaram and Pandey (1990) have conducted experimental studies and suggested that by reducing the tip clearance gap size, the tip clearance effect can be minimised. The effect of tip leakage on flow behavior in rotating impeller passage was carried out by Hathaway et al. (1993). Farge et al. (1989) carried out the impeller passage measurements at five stations in a low speed centrifugal compressor using a five hole probe rotating with the impeller. The flow in the impeller is affected by many geometric and flow parameters. Due to its extreme complexity, the characteristics of the flow inside a rotating impeller are the least understood phenomenon in the turbo machinery aero dynamics. In centrifugal compressor, there are mainly of two types of impellers shrouded and unshrouded. In shrouded impeller, the shroud rotates with the impeller and so, there is no clearance between the shroud and the tip of the blades. In case of unshrouded impellers the front cover of the casing forms the stationary shroud and so, there is a narrow clearance between the tip of the blade and the shroud. While running, there is a leakage flow, through this tip clearance which deteriorates the performance of the machine. i.e. reduces the pressure ratio, increases the input power for decreases the efficiency. Now, the question is why one should go for an unshrouded impeller, while there is no tip clearance in case of a shrouded impeller. The reason being that in case of shrouded impellers, there are secondary flows from the pressure side of the blade to the suction side of the adjacent blade along the shroud. Over the last two decades or so there has been a requirement to measure time-varying pressures in turbo machinery applications to bandwidths of order 100 kHz and the silicon piezo resistive pressure sensor has been the device, which has been at the heart of many of the measurements. In case of unshrouded impellers the leakage flow through the tip clearance of blades deteriorates the performance of the machine. At the exit, since the specific volume of the gas is reduced considerably (pressure being high).i.e. tip clearance effect is more significant. The impeller geometry should be designed considering tip clearance effects. The leakage flow through tip clearance of the impeller blades also modifies the flow pattern in the impellers. Many investors have examined the details of the flow pattern, but the results have not been utilized to evaluate the pressure loss due to tip clearance of turbomachine.

2. EXPERIMENTAL INVESTIGATIONS

The effect of tip clearance on the performance of a centrifugal compressor has been studied by many persons. Ishida and

Senoo (1981) measured pressure distribution along the shroud at seven different tip clearances. They concluded that the pressure loss due to tip clearance is proportional to the pressure rise due to deceleration of relative velocity. They also observed that the change in input power due to a change of tip clearance is related to the effective blockage at the impeller exit. Ishida and Senoo (1986) modified their own theory on tip clearance loss of centrifugal impellers as Efficiency drop due to tip clearance of high pressure ratio compressors is less than that of low pressure ratio compressors, if the tip clearance ratios at the impeller exit in equal. The magnitude of clearance loss becomes smaller as the flow rate is reduced and also at a reduced shaft speed in cases of high pressure ratio compressors. Freeman (1985) suggested that 1% increase in tip clearance produced by increasing the casing diameter will give approximately 1.4% loss in efficiency Schumann et al. (1987) tested a centrifugal impeller with a vane less diffuser for four impeller area ratios from 2.322 to 2.345. The impeller was initially designed for a pressure ratio of approximately 5.5 and a mass flow rate of 0.959Kg/s. For each area ratio a series of impeller exit axial clearances was also tested. They found that impeller efficiency decreases by 0.4 point for every 1% increase in exit clearance. Wisler (1985) reported that for a low speed four stage compressor the effect of increasing rotor tip clearance from 1.6% to 3.4% was to reduce the efficiency by 1.5% and the peak pressure rise by 9.7% Smith and Cumpsty (1984) have shown a 23 percent drop in maximum pressure rise and a 15 percent increase in flow coefficient at stall in a large, low speed compressor as the tip clearance was increased from 1 to 6 percent of chord. Engeda and Rautenberg (1987) carried out experiments on five centrifugal pump impellers of different specific speeds ranging from 0.33 to 1.51 and studied the relative effect on tip clearance on the stage performance. They concluded that the tangential component of the absolute velocity is seen to be hardly influenced by the tip clearance variation. Leakage losses and so the overall efficiency decrement increased with increase in tip clearance due to increase in flow passage. Also, a clearer dependency of clearance effects on specific speed was not observed. They concluded that the problems tip losses could most likely be well understood from detailed impeller geometry, impeller relative velocity distribution measurements for secondary flow analysis.

3. COMPUTATIONAL STUDIES

With the advances in memory and speed of computers, the inviscid flow analysis for turbo machinery has progressed from the two-dimensional to the quasi-three dimensional flow analysis; computer time required is very large for it is not practical to use this analysis for the design of turbo machine. To reduce computer time, it is desirable to reduce the number of grid points in each cross section. A flow analysis between blades with the tip clearance requires many grid points because the grid must be considerably smaller than the clearance while the leakage flow rolls up, forming a vortex of a diameter about ten times of the tip clearance. Moore et al. (1984) developed a partially parabolic calculation procedure to calculate three dimensional viscous flows in a centrifugal impeller. The three dimensional pressure fields within the impeller was obtained by first performing a three dimensional inviscid flow calculation for then by adding a viscosity model and a viscous wall boundary condition to allow calculation of the three-dimensional viscous flow. The basic idea is that the clearance velocity field can be approximately decomposed in to independent through flow and cross flow, since chord wise pressure gradients are much smaller than normal pressure gradients in the clearance region. As in the slender body approximation in external aerodynamics, this description implies that the three dimensional, steady, clearance flows can be viewed as a two dimensional, unsteady flow. Using this approach, a similarity scaling for the cross flow in the clearance region is developed and a generalized description of the clearance vortex is derived. The predicted results are compared with the experimental results measures at the exit of the impeller using a two-hole probe with high-frequency response semiconductor pressure transducer developed by Goto (1988). Predicted internal flow, the interaction mechanism between passage vortices and tip leakage flow and the effects of increasing tip clearance were discussed.

4. EXPERIMENTAL FACILITY AND INSTRUMENTATION

The experimental set up of centrifugal compressor; driven by 5KW motor. The main components of the compressor are the nozzle at the inlet, the suction duct, the impeller, the vane less diffuser formed by the front and the rear walls of the casing and volute casing of circular cross section and a delivery duct with a throttle at its outlet. The D. C. Motor is directly coupled to the shaft carrying the impeller. A brief description of each of the component of the experimental set-up is given below. An inlet nozzle and a suction duct were provided at the inlet of the compressor. The purpose of the nozzle at the inlet of the suction duct is to provide uniform flow to the impeller by accelerating it. Besides, the inlet nozzle is used to measure the volume flow through the impeller. An inlet duct of 300 mm diameter and 1300 mm long is provided at the inlet to the centrifugal compressor. Since the inlet duct is long, the flow will settle by the time it approaches the impeller inlet. The front cover of the casing was designed such a way that its inner surface was similar to the shape of impeller blade shroud contour for its extension formed the front wall of the vaneless diffuser at the exit of the impeller. Six static pressure holes were made along the meriodional plane on the inner contour surface of the shroud from the inlet end to the exit end of the diffuser to measure the static pressures. All the tapping was normal to the inner contour of the shroud. An impeller of a centrifugal compressor is the most delicate part to be designed. It is to be ensured that the flow is ideally oriented in the channel passages. The energy transfer from the shaft to the fluid takes place in the rotating impeller. The existing impeller

[6] (Sankaran, 1986) is a special kind, designed to consider the effect of the outlet edges of the vanes of a centrifugal impeller on its performance and flow characteristics. The impeller was designed on the power and speed of the available drive motor. The shape number was chosen was 0.1. The impeller is unshrouded one, with an inducer at the entry. The inducer is provided to give a smooth guidance to the flow turning from axial to radial direction inside the inlet region of the impeller. The ratio of the inducer tip diameter to the impeller outer diameter, d1t/d2t, is 0.6 to minimize the frictional losses. The blade inlet angle β2 at the impeller tip is 350. The meridional component of velocity is uniform at the inlet of the impeller. It is nearly constant from the inlet to the exit of the impeller. The number of blades used in the impeller is 16. The selection of the number of blades is based on Pfleiderer's equation, which gives the required number corresponding to the impeller geometry. If an excessive number of blades are used, lesser frictional and blockage would be high reducing the efficiency. Also, if lesser number of blades were used there would not have been proper guidance of the fluid in the impeller resulting in high slip and lowering of the work addition to the fluid. With the lesser number of blades, the blade loading would also be higher. The impeller has twisted blades at the exit region. The impeller blade edges have a maximum twist angle of 150 at the hub and shroud ends. This makes the blade angles at the exit of the impeller to uniformly vary from 750 from the hub end to 900 at the mid span and then to 1050 at the shroud end. All angles are measured with respect to the tangential direction. The rear cover of the casing called as back shroud forms the hub wall and is attached to the volute casing by means of studs. A minimum clearance between the impeller and the rear cover is provided to minimize the leakage. The cover of the casing is extended to form the back wall of the vane less diffuser. The vaneless diffuser is a simple device consisting of an annulus in which the radial velocity component is reduced by an increase in the area and the tangential velocity by the requirement of constant fluid angular momentum (free vortex) .The extended part of the parallel walls of front cover and the rear cover formed vaneless diffuser. The width of vaneless diffuser was kept same to direct the flow from, impeller to the annular passage without lateral expansion. The function of the volute casing is to collect the air from the vane less diffuser and discharges it to the delivery duct. The volute casing was of circular cross section. The width at entry to the volute casing is also kept the same as that of the vane less diffuser exit width, so that the surfaces at the two sides of the volute at entry forms a continuous portion of the vaneless diffuser. It is firmly fixed to the foundation. The volute casing is designed for a constant angular momentum for the air stream. The cross-section of the volute is circular. It has a base circle of 630 mm diameter, a volute angle of 13o and cut-off angle of 23o. The passage width at the entry to the volute cross-section at the exit is 200 mm in diameter and is directly connected to a delivery duct of the same diameter through a 90o bend. The delivery duct is 200 mm in diameter and 5 m in length. The throttle cone is fitted at the end of the delivery duct to regulate the volume flow through the impeller. The average delivery pressure is measured through the static pressure tapping provided on delivery duct wall at a distance of about 1 m from the 90o bend following the volute casing. The throttle cone could be moved forward or backward manually inside the delivery duct to regulate the volume flow through the compressor. On the shroud adapters, for the pressure sensor, are provided at six different radial locations from inlet, just behind the leading edge, to exit, after the trailing edge of the impeller (Fig. 1). The sensor is passed inside a brass holder of 2 mm internal diameter and with external threading (M4). The sensor is glued to this brass holder using Fevikwik. The brass holder is screwed into these adapters at different locations for pressure measurement. Spacers were used to maintain the tip clearance accurately. Tip clearance is varied by moving the front shroud axially. Thereby the distance between the front and rear cover would alter. Provisions are made on front cover to fix the spacers in position. Four spacers were used for each tip clearance were fixed at 90 degrees. The experimental investigations involve study for four tip clearance .so hence four set of spacers were made. To measure the suction, delivery and wall static pressures along the shroud and scanning box (Furnace Control Model no. FCO 91-3) with a micro manometer (Model FCO12) manufactured by M/s Furness Control Ltd., Bexhill, UK is used. The micro manometer has a sensitive differential pressure measuring device capable of reading air pressures up to ± 1999 mm WG. It would respond to pressure inputs up to 5 kHz. But the time constant potentiometer could be used to dampen response the fluctuating signal. It is useful in damping flow fluctuations. The pressure can be measured to an accuracy of ± 0.1mmWG. The scanning box contains 20 valves so we can measure up to 20 pressures. The speed of the centrifugal compressor is measured using a non-contact digital type tachometer. The speed measurement is based upon the evaluation of reflecting pulses from the rotating shaft. It has a range of 0-9999 and its accuracy is ± 1 rpm.

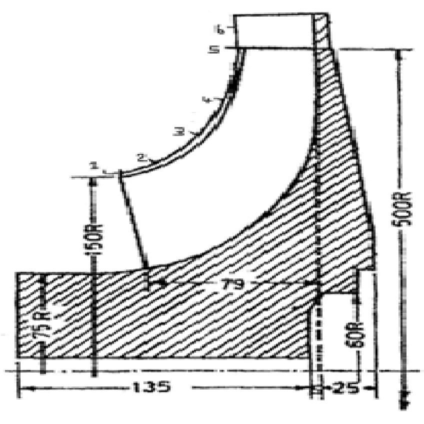

Fig. 1 Meridional view of impeller showing static pressure measurement

Table 1 Design Details of the Rotor

Total pressure rise, Δp:	300 mm WG
Volume flow rate, V:	1.12 m3/s
Speed of rotation, N:	2000 rpm
Shape number, Nsh:	0.092
No. of rotor blades, Z:	16
Inducer hub diameter, d1h:	160 mm
Inducer tip diameter, d1t:	300 mm
Rotor tip diameter, d2:	500 mm
Blade height at the exit, h2:	34.74 mm
Blade angle at inducer tip, β1t:	35°
Blade angle inducer hub, β1h:	53°
Blade angle at exit, β2: (a) At hub: 75° (b) At mean section: 90° (c) At tip: 105°	
All the angles are measured w. r. t. tangential direction	

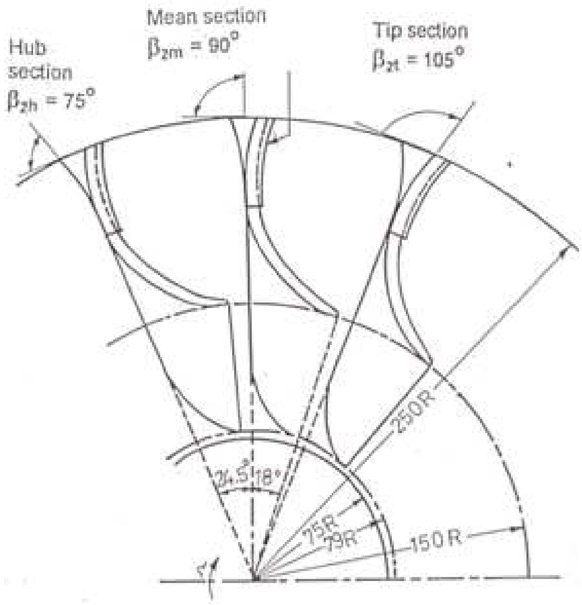

Fig. 2 Details of Configurations Tested

The non-dimensional parameters are defined as follows:

Flow coefficient, $\phi = \dfrac{V}{\pi d_2 b_2 U_2} = \dfrac{c_{2r}}{U_2}$

Where V = Volume flow (m^3/s)

d_2 = Rotor tip diameter (m)

b_2 = Rotor blade width at exit (m)

c_{2r} = Radial velocity at rotor exit (m/s)

U_2 = Rotor tip speed (m/s)

Energy coefficient, $\psi = \dfrac{2\,W}{U_2^2}$

Where W = Specific work (m^2/s^2)

$$= \frac{p_d - p_s}{\rho} + \frac{c_d^2 - c_s^2}{2} + g\Delta Z$$

Power coefficient $\gamma = \dfrac{2\,N_c}{\rho A U_2^3} = \dfrac{2\eta_m EI}{\rho A U_2^3}$

where N_c = Coupling power (Watts)

A = Suction duct area (m^2)

E = Motor voltage (Volts)

I = Motor current (Amps)

η_m = Motor efficiency

Compressor efficiency, $\eta_c = \dfrac{\rho V W}{N_c}$

COLOUR	DESIGNATION
RED	+ INPUT TO SENSOR
BLACK	- INPUT TO SENSOR
GREEN	+ OUTPUT FROM
WHITE	- OUTPUT FROM

Table 1.1 Colour designations of sensor wires

5. RESULTS AND DISCUSSION

The results of experiments conducted on the centrifugal impeller at four values of tip clearances i.e., =2.2%, 4%, 6.1% and 7.9% for f0ur different flow coefficients i.e., = 0.18, 0.21, 0.28, 0.34 and 0.40 are presented and discussed in this chapter. The performance characteristics of the impeller are presented first and then results of wall static pressure measured from the front shroud to impeller exit are presented.

6. PERFORMANCE CHARACTERESTICS

Performance test is carried out at a constant speed at varying throttling positions from low flow rate (in surge region) to high volume flow rate (near choking). Five flow coefficients viz., =0.40, =0.34 (both above design flow), =0.28 (near design flow), =0.21=0.18 (both below design flow) and four values of non-dimensional tip clearance viz., =2.2%, 4%, 6.1% and 7.9% are chosen for experimental work. The performance of the compressor in terms of energy coefficient vs. flow coefficient is presented in Fig. 5.1. The range of flow coefficient for which the compressor operates stably is =0.08 to =0.48.The result of the experimental investigations revealed that, energy coefficient across the compressor is reduced as the tip clearance is increased. The flow coefficient, where the energy coefficient is maximum decreases with the increase in tip clearance. Similar results are observed in a low speed radial tipped centrifugal compressor (Sitaram and Pandey, 1990). However the operating range of the compressor is decreased slightly as the maximum volume flow passing through the compressor is reduced with tip clearance. The performance curves shown include losses not only the in the impeller, but also in the following vaneless diffuser, volute casing and delivery duct. The static pressure distribution measured at the casing static pressure holes is shown in Fig. 5.2, at 2.2% tip clearance for different flow coefficients. On the shroud, static pressure taps were provided from the inducer leading edge to the impeller exit. The shroud has its inner shape contoured to match the impeller blade tip profile from the inlet to the exit. The static pressure measurements on the shroud indicate variation in the static pressure coefficient, which matches with the blade passage clearly. From the figure it is seen that the distribution is high for $\phi= 0.28$, as it is nearer to design point operation. The pressure initially decreases due to suction and then uniformly increases, indicating that there are no dead zones or eddies near the shroud region inside the impeller and energy transfer occurs smoothly to the fluid near the shroud. This can be inferred from the Fig 5.1. It is lesser in the case of other flow coefficients. The first two static pressure tapping shows a negative static pressure coefficient. Very close to the third hole position the leading edge of the impeller starts. Since then, the pressure coefficient took an increasing trend. This is due to the development of low pressure zone behind the blade in the suction side.

The compressor characteristic curve against flow coefficient at different tip clearances is shown in Fig. 5.3. Using the micro manometer the static pressure difference between the entry and exit of the compressor is measured, at a particular tip clearance. The figure depicts a negligible difference in efficiency for flow coefficients up to 0.30, for all tip clearances. At flow rates higher than this, the difference in efficiency is appreciable, the peak efficiency being lower for higher tip clearance and vice versa. This can be attributed to the increase in tip losses with the increase in the clearance. The reduction in efficiency is due to the reduction in energy transfer for higher tip clearances, since the input power remains more or less the same.

Figure 5.4 shows the effect of volume flow rate on pressure ratio P2/P1 across the compressor. Pressure ratio shows an inverse variation with increase of volume flow rate, for all tip clearances. As the tip clearance increases, the space available for the flow to expand is comparatively more than a lower tip clearance; the pressure drops more and is clearly seen in the figure. Figure 5.5 shows the effect of volume flow rate on pressure rise (P2-P1). From the figure it is clearly seen that when volume flow rate increases the pressure difference is reduced. For the same tip clearance, as the volume flow rate is increased the pressure at the exit increases for hence the pressure difference is having a decreasing trend.

CONCLUSIONS

The present experimental investigations were conducted to study the effect of tip clearance on the performance and wall static pressure in a centrifugal compressor. Performance tests were conducted at five flow coefficient with four different tip clearances for each flow coefficient. Based on the experimental investigations, following conclusions are drawn

1. As the tip clearance is increased, there is a reduction in static pressure rise across the compressor, which causes reduction in energy coefficient.
2. The decrease in energy coefficient is more at higher value of flow coefficient.
3. The operating range slightly decreases with the increase in tip clearance.

The wall static pressure is high for lower tip clearance for all the flow coefficients. For high flow coefficient, reduction in static pressure is more near the inducer, which indicates more deceleration of flow at higher flow coefficient. The tip clearance is more pronounced at higher flow coefficients. From these graphs it shows that the maximum pressure is achieved for 2.2% tip clearance, as compared to other tip clearance, for all flow coefficients. Similarly for a flow coefficient of 0.28, the static pressure observed is higher than other flow coefficients for all tip clearances.

NOMENCLATURE

b : Distance between the shroud and hub at the rotor exit (m)
C_d: Velocity in delivery duct (m/s)
C_s: Velocityinsuctionduct(m/s)
d: Rotor diameter (m)
N: Rotational speed of rotor (rpm)
N_c: Coupling power (Watt)
N_{sh}: Shapenumber=$N\sqrt{V}/W^{3/4}$
P_d: Delivery pressure (Pa)
P_s: Suction pressure (Pa)
t : Rotor blade clearance (m)
U: Rotor tip speed = (πdn/60) (m/s)
V: Volumeflowrate(m^3/s)
W: Specific work (m^2/s^2)
φ: Flowcoefficient (defined in the text)
γ: Powercoefficient (defined in the text)
η: Efficiency (defined in the text)
ψ: Energycoefficient (defined in the text)

ρ: Densityofair(kg/m3)
τ: Tip clearance as a percentage of rotor blade
height at exit = (t/b_2) × 100.
Subscript : Tip.

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C_u Tangential velocity
m Non-dimensional meridional distance
P Static pressure
P_{atm} Atmospheric pressure
P_o Total pressure
R Non-dimensional radius
u_2 Blade tip speed
x Non-dimensional axial distance
φ Flow coefficient
ρ Density
τ Tip clearance
ψ_s Static pressure coefficient
ψ_o Total pressure coefficient

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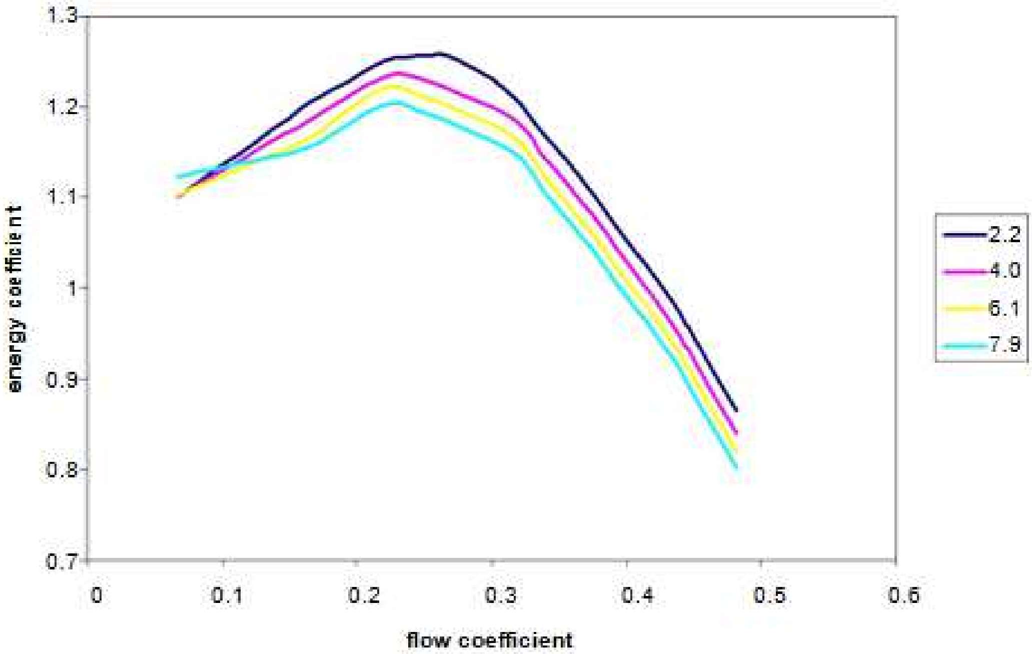

Fig. 5.1 Performance characteristics

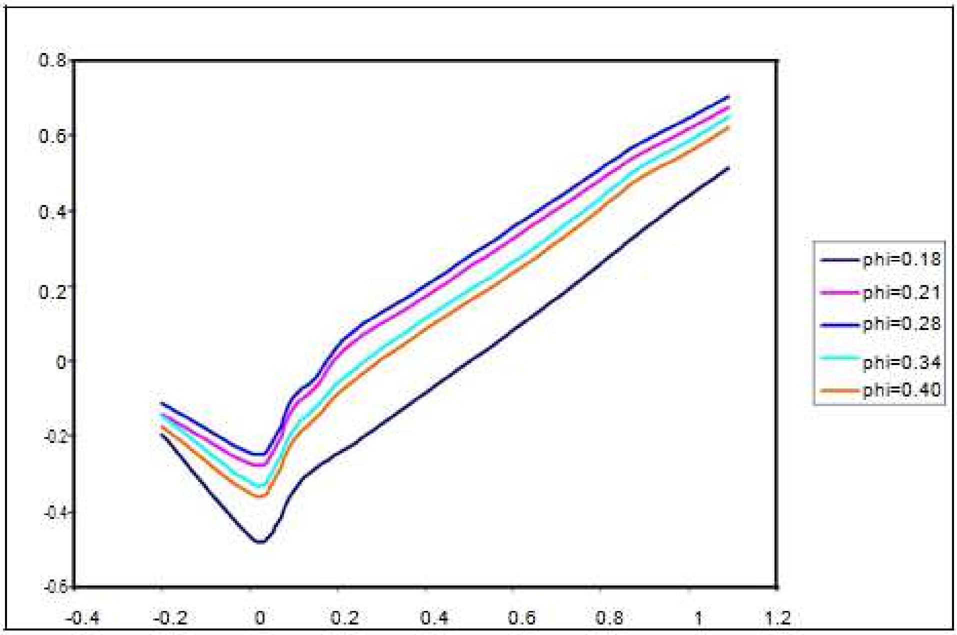

Fig. 5.2 Variation of time averaged static pressure along the casing

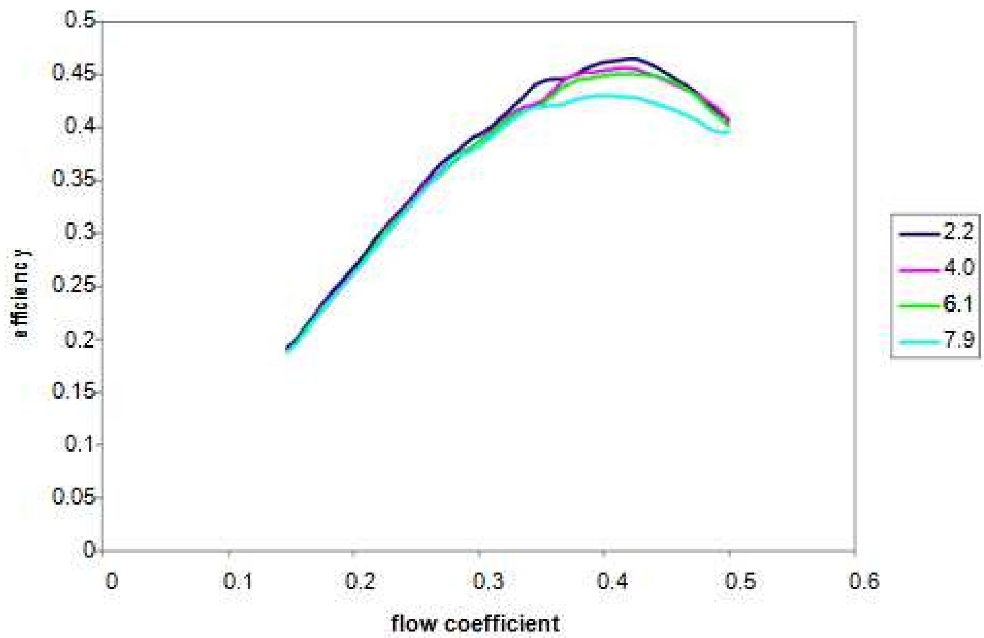

Fig. 5.3 Compressor characteristic curve

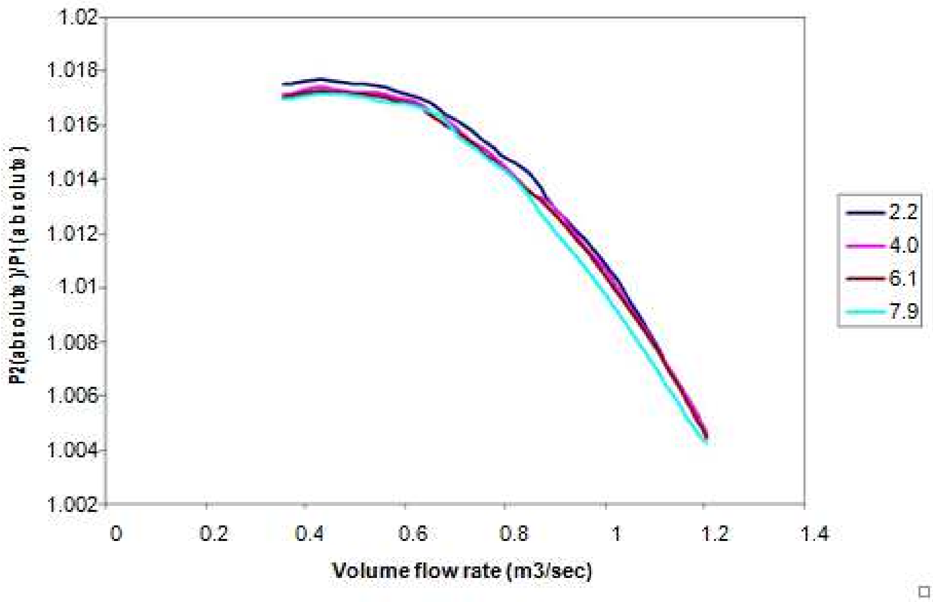

Fig. 5.4 Effect of volume flow rate on pressure ratio

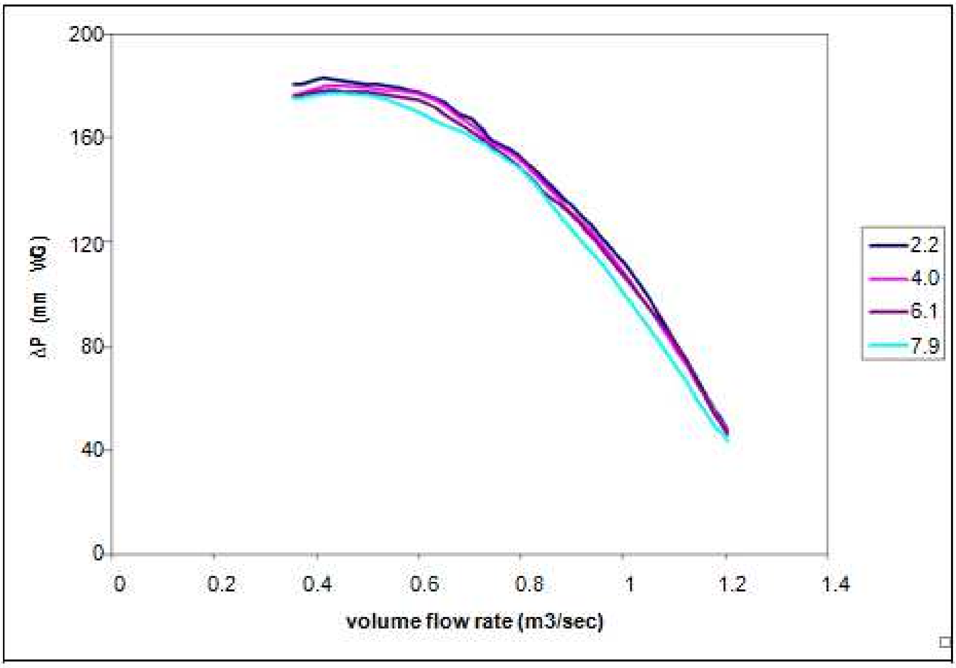

Fig. 5.5 Pressure difference across the compressor