

FLOW STUDIES IN A CENTRIFUGAL COMPRESSOR STAGE

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ABSTRACT

This paper describes the behavior of steady and unsteady flow inside a centrifugal compressor stage. Time averaged and unsteady pressure measurements were carried out using high response pressure sensors to capture the flow behavior within the impeller channel. Experiments were carried out at five different speeds. At each speed the different operating points were obtained through a throttle control valve. From the measurements the critical regions in the stage which influence the performance has been identified. It was observed that the flow entering into the diffuser is non uniform and becomes uniform as the flow leaves the diffuser. At low flow rates there is a rise in static pressure at impeller inlet tip.

INTRODUCTION

The present trend in aero engine compressors is towards increased pressure ratios per stage; minimize the specific weight and specific fuel consumption. As the stage pressure ratio increases, the operating range reduces. Centrifugal compressor stage performance is dependent not only on the impeller aerodynamics, but also on the vane diffuser following the impeller. The vane diffuser influences the flow at impeller outlet. To extend the operating range of centrifugal compressor, it is necessary to develop a quantitative understanding of the basic flow phenomenon between impeller and diffuser. In a high pressure ratio centrifugal compressor a vane diffuser is provided after the impeller to convert large kinetic energy into useful static pressure head. This is accomplished through diffusion process.

Fink¹ Experimentally investigated system dynamics, unsteady effects in centrifugal impellers and detected flow instabilities. Haupt and Seidel² Experimentally investigated self excited flow oscillations in a centrifugal compressor with a straight channel radial vane less diffuser. The results obtained were compared with those of a previous study of the same compressor but with a cambered vane diffuser. The stability margin prior to surge and the operating regimes in which very intense pressure fluctuation found were different. Chu. et al.³ Experimentally investigated the unsteady flow in a radial vane less diffuser. It was shown that the circumferentially uniform discharge at the exit of the diffuser changes into periodic circumferentially fluctuating flow state. Kubu and Murata⁴ presented their studies made about the unsteady flow phenomenon in centrifugal fans. Ramamurthy and Murugesan⁵ carried out experimental studies about the

unsteady flow phenomenon in centrifugal impellers using hot wire anemometer system by measuring radial and tangential velocities at impeller outlet. Experimental investigations in a high-pressure ratio centrifugal compressor stage consisting of impeller and vane diffuser were carried out by detailed steady and unsteady measurements using conventional and high response miniature pressure transducers. The experiments were carried out at different speeds ranging from 15000 to 20000 rpm in a closed circuit centrifugal compressor test rig. Static pressure measurements were carried out on shroud wall from impeller inlet to diffuser exit to study the flow behavior in the compressor stage at different flow conditions. Static pressure measurements on the suction and pressure surface of the vane diffuser from diffuser leading edge to diffuser exit at different radius to study the variation of diffuser blade loading on the overall stage performance. Similarly unsteady flow measurements were carried out at impeller shroud and diffuser channel using miniature high response Kulite transducer and four channel simultaneous high speed data acquisition system to study the unsteady flow behavior at the compressor stage.

NOMENCLATURE

M – Mass flow rate (kg/s)
To – Total temperature (K)
Po – Total pressure (N/m²)
Ps – Static pressure (N/m²)
D – Pipe diameter (m)
MP – Mass flow parameter

Subscripts

1 – Impeller inlet
2 – Impeller outlet

Superscripts

1-4 – Reference serial number

EXPERIMENTAL SET UP

Test Facility

The schematic layout of the Closed Circuit Centrifugal Compressor Test Rig (CLOCTER) is shown in Fig.1. An electromechanically coupled twin DC motor system rotates the compressor at a desired speed. Thyristor control with feedback for the DC motors ensures maintenance of the speed to an accuracy of 1%. The compressor and DC motor were connected together with a step up gear box (20:1). An

electronic torque meter coupled in between the gear box and the compressor was used to measure compressor speed and input power. A gate valve provided in the closed circuit was used to vary the mass flow rate through the compressor. An orifice plate in the inlet duct was used for a mass flow measurement. A heat exchanger in the closed circuit was used to ensure steady inlet flow conditions. An external cooling tower system pumps cold water to this heat exchanger and to another heat exchanger, which cools the lubrication system. The orifice plate in the facility is used to monitor the flow rate, which can be varied by throttling the gate valve. This provides us to get wide range of operating points. Similarly a thyristor control through feed back circuit is useful for the variation of speed of the DC motor as this allows running the compressor to any desired speed.

Test Compressor

The compressor stage in vane diffuser configuration is shown in Fig.2 which consists of backswept impeller of 300 mm tip diameter having 19 blades and a vane diffuser having 17 blades. The diffuser inlet to impeller outlet diameter ratio was around 1.05, and the diffuser inlet to diffuser outlet radius ratio was around 1.555. The experiments were carried out by running the compressor at speeds ranging from 15000 to 20000 rpm.

Instrumentation

The time averaged parameters like total pressure, static pressure, total temperature, speed and power input to the compressor were measured using conventional probes and an on-line data acquisition system. The data acquisition system is connected to an industrial computer. Analog signals from the transducer and thermocouples were sequentially scanned and stored in the scanner. A 48 port scanivalve with a single Statham transducer of 50 PSIA range with sensitivity 146.6 $\mu\text{V}/\text{psi}$ was used for the measurement of pressure at various locations. The scanning rate of the scanivalve was controlled using a decoder. In the present case the system provides a scanning rate of 10 pressures per second, so that 48 pressures can be scanned in 4.8 seconds. The pressures were measured to an accuracy of 0.1% full scale.

The static pressure fluctuations at different flow coefficients were measured using high response miniature Kulite transducers at impeller shroud located at different positions in diffuser channel. The rated pressure range of the transducer is 25 PSID, with frequency response of 300 kHz and sensitivity is 3.371 mV/PSID. Separate 10 volts linear DC power supply was used to excite the Kulite transducers. The transducer signals were triggered using a once per revolution shaft pulse and acquired using high speed simultaneous data acquisition and processor card mounted in one of the slot of the computer. The transducer signals were conditioned using signal conditioner card mounted in another slot of the computer before feeding on to the data acquisition card. The block diagram of the instrumentation system for unsteady pressure measurement using Kulite transducer is shown in Fig.3. The data acquisition processor has an inbuilt 486 processor with A/D converter and 8MB on board memory.

This card is capable of acquiring four simultaneous signals with a throughput rate of 3.2 million samples per second. The transducer signals were acquired by triggering the high speed card through a Transistor Transistor Logic (TTL) pulse generated once per revolution using burst mode.

RESULTS AND DISCUSSIONS

The performance characteristics of the impeller and the compressor stage were obtained by running the compressor in closed circuit with air as working medium. Different operating points were obtained for speeds ranging from 15000 to 20000 rpm in steps of 1000 rpm by varying the mass flow rate using a main and an interconnecting gate valve. The pressure ratio and mass flow rates were estimated from the measured time averaged parameters. These estimated parameters were used to collaborate the unsteady static pressure measurements made on shroud casing. The pressure ratio across the impeller and stage has been plotted against mass flow parameter and is shown in Fig.4. The symbol “ \circ ” represents impeller pressure ratio and the symbol “ Δ ” represents the stage pressure ratio. It is observed from this figure that the impeller static pressure ratio is independent of mass flow parameter for a given speed. This is because the experiment has been conducted in closed circuit. For a given speed the decrement in mass flow parameter will reduce the inlet total pressure as well as impeller outlet static pressure, thereby the ratio is maintained same. As the speed increases, the impeller static pressure ratio increases at a given operating point. It is clearly seen that the diffuser becomes effective only after certain throttle setting. For example at 20000 rpm the diffuser provides recovery in static pressure for a mass flow parameter less than about 2.1. For mass flow parameter greater than 2.1, the flow inside the diffuser separates and the net results will be in the decrement in stage static pressure. As the speed increases, the effectiveness of the diffuser to provide increase in static pressure rise increases.

The compressor stage provides continuous increase in static pressure from inlet to outlet; these characteristics will depend on the operating point and the speed of the compressor. To study this, the static pressures were measured on the shroud from impeller inlet to diffuser exit. These measured static pressures are normalized with corresponding inlet static pressure at each mass flow parameter and are plotted for different operating points from 15000 to 20000 rpm. The static pressure variation at different location on the shroud and diffuser channel is shown in Fig-5. The location of the static pressure measurement from impeller inlet “A” to impeller outlet “B” and from diffuser inlet “C” to diffuser outlet “D” are indicated in the table by the side of Fig.5. In this figure, the static pressure variations from impeller leading edge to diffuser exit are shown for six speeds ranging from 15000 to 20000 rpm. In each speed there are sixteen operating points starting from large mass flow to mass flow close to the stall. The lower curve represented by “a” indicates larger mass flow, whereas the upper most curve represented by “b” indicated lower mass flow. It is observed at largest mass flow the static pressure peaks in the vane less portion and then rises constantly till diffuser trailing edge. This behavior is observed

in all the speeds, whereas flow rates closer to stall there is a sharp increase in static pressure at all speeds. Magnitude of the peak increases with increase in speed. This is a common phenomenon observed in radial flow machines that at low flow rates there is a small separation region at impeller inlet tip. At low flow the incidence to the impeller inlet will be positive and there is a chance for the flow to separate from the pressure surface, whereas at high flow rates the incidence to the diffuser is negative and there is a chance that the flow can separate from the suction surface. Due to this behavior the two peaks at low and high flow rates were observed.

The vaned diffuser provides static pressure recovery in a compressor stage. This recovery depends on the incidence to the diffuser. To study this, the static pressures were measured at suction and pressure surface wall of the vaned diffuser. This static pressure is normalized with corresponding inlet static pressure at different operating points. Variation of the static pressure along the blade length is plotted for three different speeds of 16000, 18000 and 20000 rpm, which is shown in Fig.6. Suction surface pressures are indicated by the symbol "o" and the pressure surface pressures are indicated by the symbol "Δ". For each speed six operating points to cover the complete compressor range being considered. The static pressure variation on the diffuser blade is shown slightly away from the leading edge due to the difficulty in providing the pressure taps close to the leading edge. At high flow rates the incidence to the diffuser is positive and the flow might separate from the suction surface, due to this we see there is a small cross over in the pressure distribution and as the flow rate reduces this cross over vanishes and the diffuser provides the required loading which gives rise to increase in static pressure. Throughout the flow regime the pressure surface values are higher than the suction surface, except at high flow rates close to the leading edge. It is also seen very close to the trailing edge both the pressure s becomes equal. To see the effect of loading with the operating points, at each operating point the area within the pressure distribution were calculated and has been plotted against mass flow parameter for six different speeds and these are shown in Fig.7. For a given mass flow parameter the blade loading increases with increase in speed and at a given speed the area within the curve decreases with decrease in mass flow parameter, reaches a minimum point and then rises. The minimum point was very close to the optimum incidence to the diffuser. Further reduction in mass flow parameter the area within the curve increases. The location of the minimum point also changes with the operating speed. This may be due to the inlet flow characteristics like blockage, non uniform flow distribution, etc.

The flow inside the centrifugal impeller is highly unsteady and this unsteady flow is being carried out into the diffuser and affects the performance. To study this, static pressure measured on the shroud wall is made using high response transducers. For different transducers were mounted on the impeller shroud at locations marked as K1,K2,K3, and K4 to cover one blade channel from inlet to outlet. Similarly another four different transducers were mounted in diffuser

shroud wall to cover the diffuser channel which is marked in the figure as K5, K6, K7 and K8. The transducer signals for different operating points were collected for a given speed of 18000 rpm. These signals are ensembled to get an average signal over five blade channels. These signals were acquired by coinciding the impeller tip pressure surface with the shaft trigger signal to get the required reference. In Fig.8, the transducer signals for different mass flow parameter were shown along with locations. By the side of each figure, the numerical value which gives the normalized mean static pressure at that location. It is observed from this figure that the unsteady flow generated from the impeller moves towards the upstream and indicates a typical variation of static pressure generated to the blades and at this flow rate the impeller exit exhibit a small variation in static pressure across the blade and gets amplified at diffuser inlet. The amplified signal decays as it goes across the diffuser and becomes uniform at the exit. At low flow rates the symmetry in the channel is lost at inlet and the signal amplitudes are higher at impeller outlet, but the magnitude of the variation in pressure at diffuser inlet is larger, and it becomes uniform at diffuser outlet.

In general the diffuser faces a highly non uniform flow with amplitudes being higher at low mass flow parameter and as the flow reaches the exit it becomes uniform. In the impeller somewhere in between maximum and minimum mass flow parameter, impeller exit exhibits a large variation in static pressure. Hence it is observed that, the flow at the exit of the impeller and diffuser inlet is highly non uniform and unsteady.

CONCLUSIONS

From the performance characteristics, it is observed that the impeller static pressure ratio is independent of mass flow parameter for a given speed. As the speed increases, the impeller static pressure ratio increases at a given operating point. As the speed increases, the effectiveness of the diffuser to provide increase in static pressure rise increases. At largest mass flow the static pressure peaks in the vane less portion and then rises constantly till diffuser trailing edge. This behavior is observed in all the speeds, whereas flow rates closer to stall there is a sharp increase in static pressure at all speeds. Magnitude of the peak increases with increase in speed.

At low flow the incidence to the impeller inlet will be positive and there is a chance for the flow to separate from the pressure surface, whereas at high flow rates the incidence to the diffuser is negative and there is a chance that the flow can separate from the suction surface.

In case of vaned diffuser, for a given mass flow parameter the blade loading increases with increase in speed and at a given speed the area within the curve decreases with decrease in mass flow parameter, reaches a minimum point and then rises, the minimum point was very close to the optimum incidence to the diffuser.

Through unsteady flow measurements the vaned diffuser faces a highly non uniform flow with amplitudes being higher at low mass flow parameter and as the flow reaches the exit it becomes uniform. In the impeller somewhere in between maximum and minimum mass flow parameter, impeller exit exhibits a large variation in static pressure

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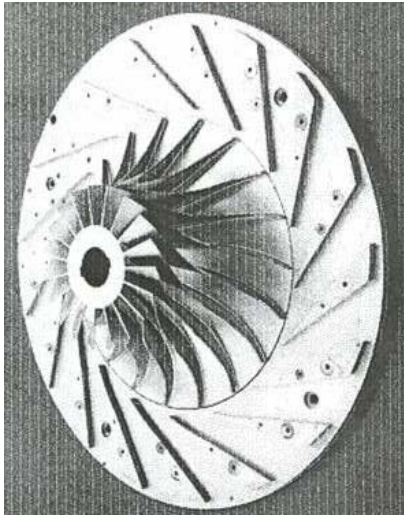


Fig.2 Centrifugal compressor stage

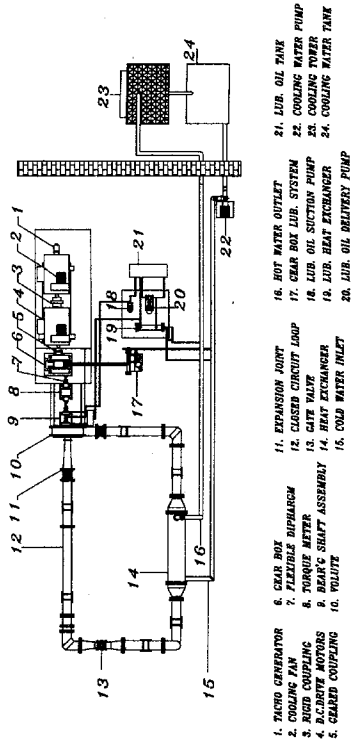


Fig.1 Schematic Layout of Closed Circuit Centrifugal Compressor Test Rig.

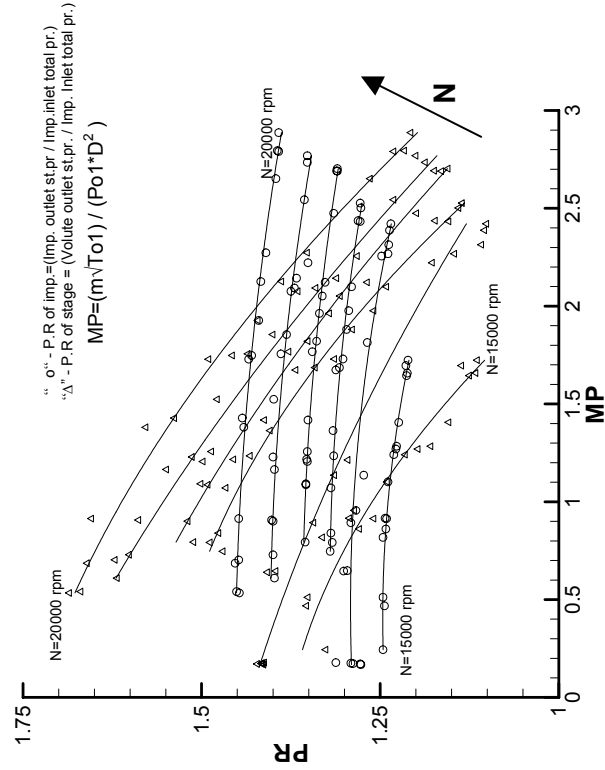


Fig.4 Performance characteristics of the compressor and stage

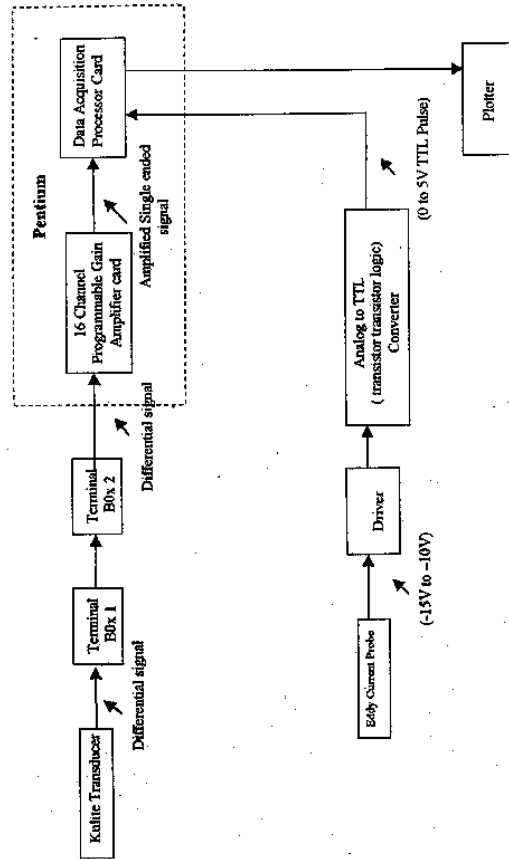


Fig.3 Schematic Layout of high speed data acquisition system

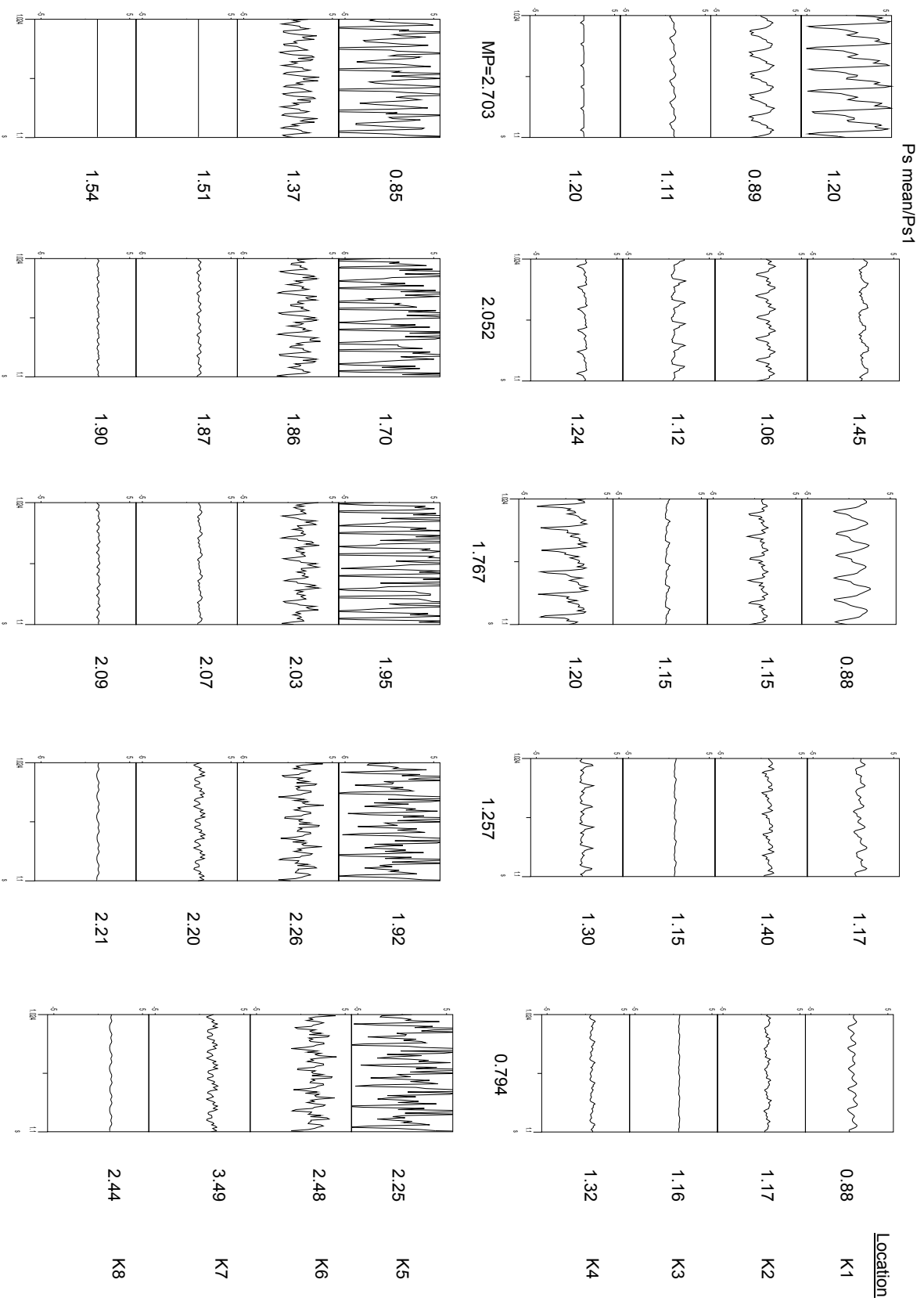
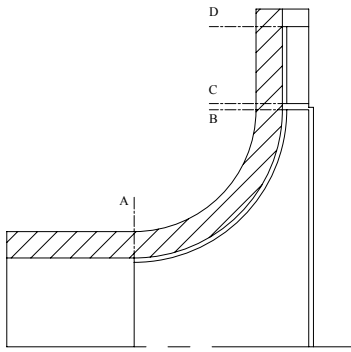


Fig. 8 Static pressure fluctuations at impeller and vane diffuser at 18000 rpm



A – Impeller inlet
 B – Impeller outlet
 C – Diffuser inlet
 D – Diffuser outlet

Station Number	Distance from Impeller L.E
1	0.0000
2	6.6031
3	13.8666
4	20.7999
5	27.9660
6	34.5935
7	41.3676
8	47.9999
9	54.4461
10	60.6627
11	66.8890
12	71.6370
13	77.3310
14	82.4388
15	86.6820
16	90.9739
17	94.5080
18	98.0255
19	100.3930
20	102.2145

K1

K2

K3

K4

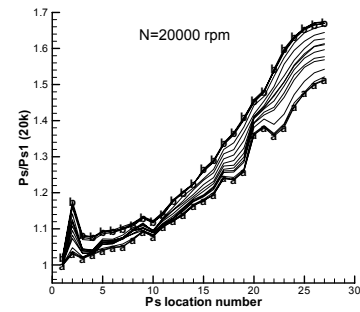
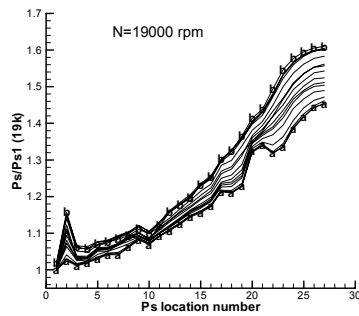
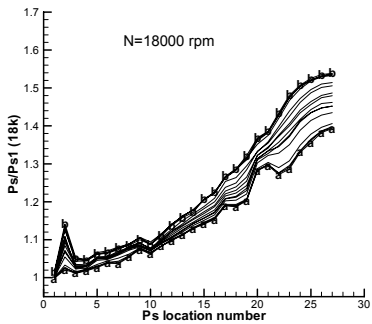
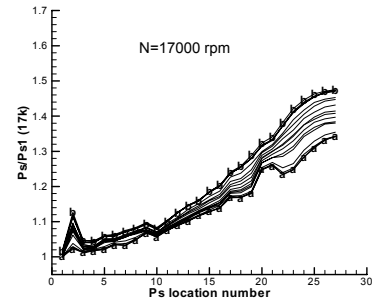
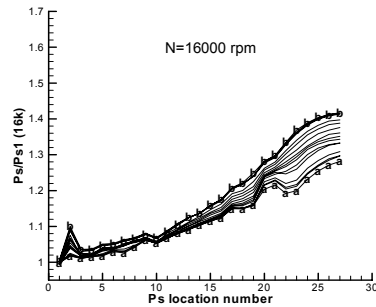
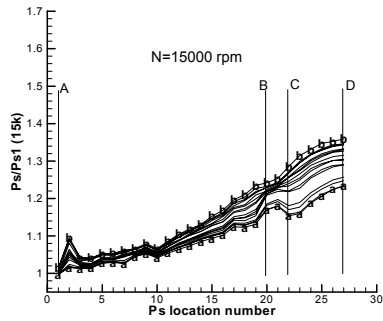
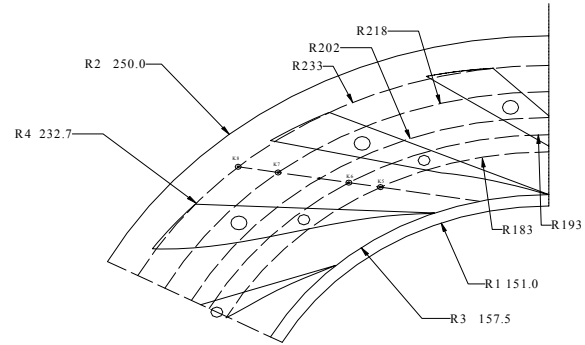
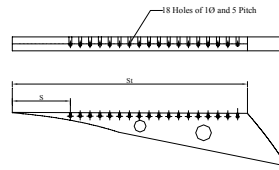
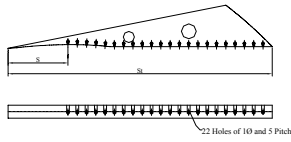


Fig.5 Normalized static pressure variation from impeller inlet to diffuser outlet



SUCTION SURFACE

PRESSURE SURFACE

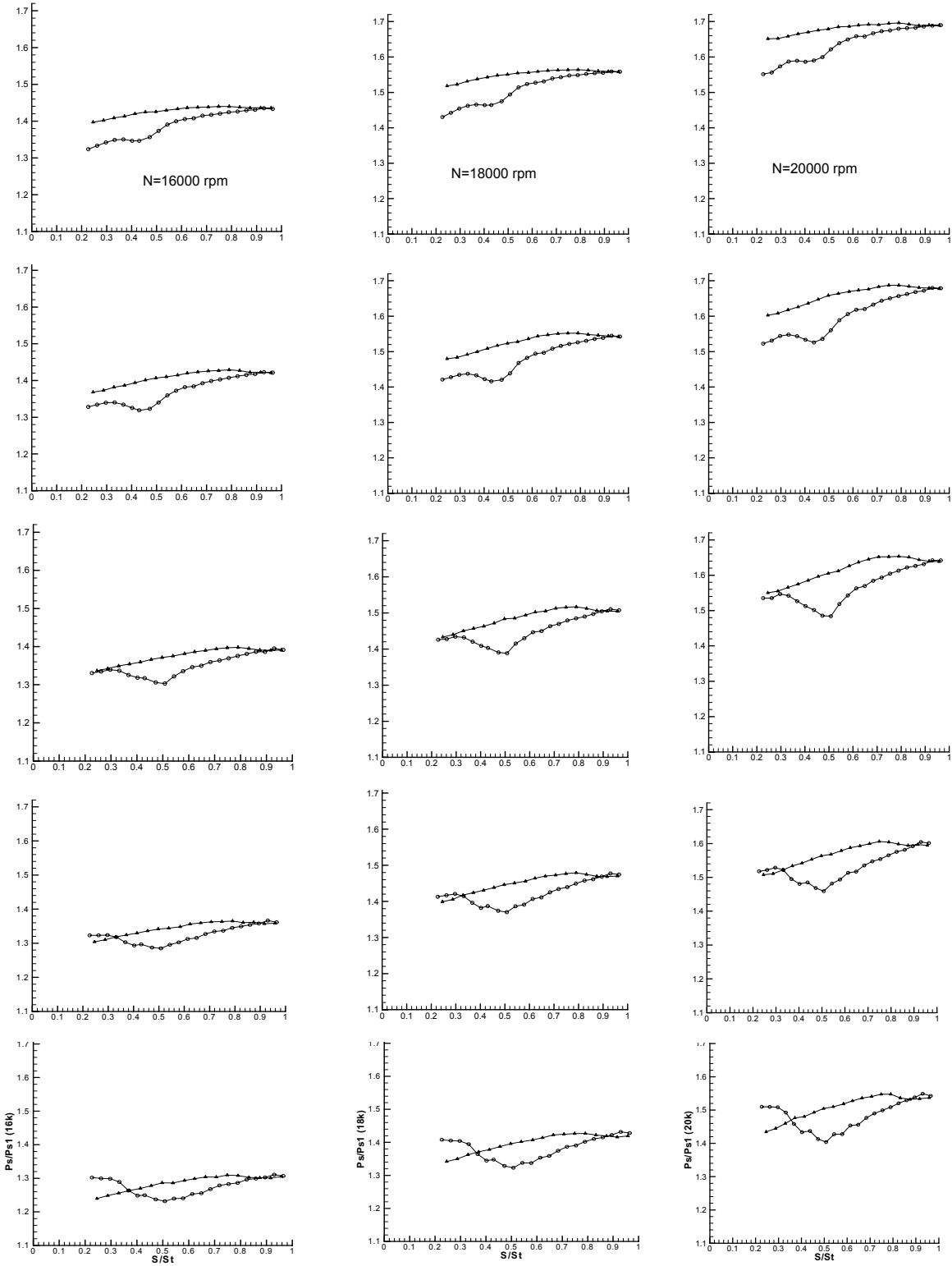


Fig.6 Normalized static pressures at suction and pressure surface of the vaned diffuser

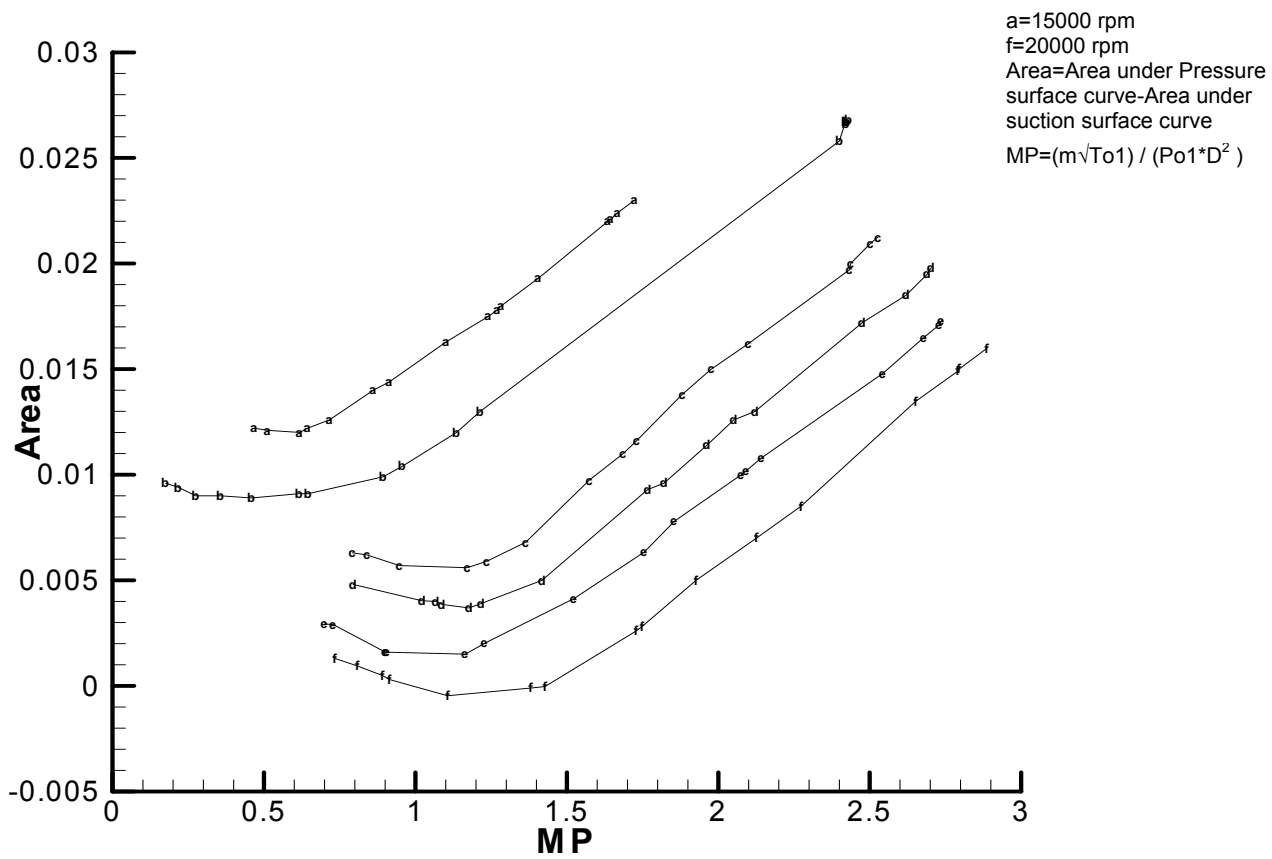


Fig.7 Variation of area at different speeds