Proceedings of the ASME 2010 3rd Joint US-European Fluids Engineering Summer Meeting and 8th International Conference on Nanochannels, Microchannels, and Minichannels
FEDSM-ICNMM2010
August 1-5, 2010, Montreal, Canada

FEDSM-ICNMM2010-30671

INLET DISTORTION OF A CENTRIFUGAL COMPRESSOR WITH A CIRCULAR-SECTIONED 90-DEGREE BEND AND ITS INFLUENCE ON THE PERFORMANCE

Kyung-Jun Kang
Energy mechanics center,
Korea institute of science and technology
, Republic of Korea
Kkj79@kist.re.kr

You-Hwan Shin
Energy mechanics center,
Korea institute of science and technology
, Republic of Korea
yhshin@kist.re.kr

Kwang-Ho Kim
Energy mechanics center,
Korea institute of science and technology
, Republic of Korea
khkim@kist.re.kr

Yoon-Pyo Lee
Energy mechanics center,
Korea institute of science and technology
, Republic of Korea
yplee@kist.re.kr

ABSTRACT

In this study, distortion due to inlet duct is experimentally analyzed and discussed in order to understand the performance characteristics of the compressor exposed to a curved inlet configuration. The flow from the bend inlet is not axisymmetric in the circumferential and radial distortion. In the combination including a centrifugal compressor, the bend inlet due to spatial limitation has been shown in a various applications such as industrial chiller. The performance of centrifugal compressors can be degraded by inlet flow distortion due to a 90 degree curved pipe. The bend have some curvature radius ratio, straight upstream and downstream pipes. The flow in bend inlet is sucked due to rotating of impeller and not axisymmetric in the circumferential and radial distortion. In general, primary flow and secondary flow are occurred in curved pipe. In this paper, flow at curved pipe was experimentally measured using by 5-hole probe and hot-wire probe. The 5-hole probe was used in observing the total pressure and velocity contour. Also, the measurement of hot-wire probe was carried out in order to obtain a streamwise velocity profile. The flow details are analyzed and discussed in order to understand the relation between inlet distortion and aerodynamic performance.

Keywords: Centrifugal compressor, 5-hole pneumatic probe, Inlet distortion, Curved pipe, Performance drop.

INTRODUCTION

Curved inlet duct exists in several engineering applications including a centrifugal compressor such as refrigeration system. This structure is largely because of a limitation of space. But, the curved inlet duct attached in front of the compressor is not desired in aspect of design. When designing of the compressor, the curved inlet duct is not considered for the design factor. In this case, the performance of centrifugal compressors can be seriously degraded by inlet flow distortions that results from an unsatisfactory inlet configuration.

Many studies have been done in order to understand the flow distortion resulting from the inlet duct. Williams et al.(1986) observed that the location of the maximum axial velocity moves toward the outer wall of a curved pipe. It is important to know the pressure drop in the developing and fully developed parts of the flow.

Sudo (1998) reported the measurement for the turbulent flow through a circular-sectioned 90° bend with a curvature radius ratio of 4.0. Cain (1999) and Rowe (1970) suggest that secondary flows appear whenever fluid flows in curved pipes or channels. Although such secondary motions can arise in a perfect inviscid fluid as a result of a nonuniform distribution of velocity at the entrance to the bend, throughout this review we shall consider real viscous fluids for which the secondary flow can be attributed principally to the effect of the centrifugal pressure gradient in the main flow acting on the relatively stagnant fluid in the wall boundary layer. The refrigeration compressors are quite similar to the process compressors except that their stage pressure ratio and flow conditions are defined by a unique thermodynamic cycle. The compressor stable operating range is seriously degraded in the present of inlet distortion. It is important to investigate the effect of inlet distortion to centrifugal compressor. A fully summary of inlet distortion and its effect on a turbo compressor performance, with an extensive reference list, is given by Williams (1986).

This paper is concerned with the experimental investigation of the steady flow in a circular-sectioned 90° bend. The total pressure and axial velocity contour are measured by 5-hole probe and Hot-wire anemometer. Flow distortion region was verified in the curved inlet pipe.

NOMENCLATURE

C_p	Pressure coefficient.
D	Diameter.
p	Static pressure.
p_{ref}	Pressure at x/D = 3.3.
r	Radius.
α	Bend curvature angle.
θ	Circumferential coordinate.
V_m	Mean velocity at pipe inlet at x/D = 3.3.
x	Upstream coordinate of bend pipe.
y	Downstream coordinate of bend pipe.
ρ	Flow density

EXPERIMENTAL FACILITY

The test compressor has a single stage with an unshrouded radial impeller and a parallel wall vaneless diffuser. The experimental apparatus consists of centrifugal compressor and curved inlet pipe. The schematic and specification of test compressor present in Fig. 1(a). Atmospheric air is sucked into the compressor through inlet circular-cross-sectioned pipe. The inlet temperature and pressure are measured in the four points (θ = 0°, 90°, 180°, 270°) in the inlet pipe as shown in Fig. 1. The mass flow is measured with a venturi which is calibrated by a standard facility. The test compressor was driven by a 40 kW electric motor with a frequency inverter and operated as an open loop type. The flow rate was controlled by the throttle valve at the end of the discharge duct and measured using the orifice plate in the discharge duct.

Figure 1 (a) shows a meridional and front view of the centrifugal compressor. Total pressure, temperature and wall static pressure were measured at the impeller inlet and exit, diffuser, and discharge duct. To understand the flow mechanism in the curved pipe, flow field measurements were carried out using pneumatic five-hole probes (United Sensor DA - 187) at three cross-sectional planes.

For the pressure measurement a pressure scanner (PSI System 8400) was used. Static pressure tap hole was placed in r/r_2= 1.094, 1.191, 1.311, 1.431 and 1.738 to measure pressure inside vaneless diffuser. And, total pressure probe is fixed up in r/r_2= 1.094 and 1.738. Figure 1(b) presents the real image of test compressor with the curved inlet pipe. The circular-sectioned 90° bend pipe is shown in Fig. 1(c). The curved pipe had a radius of 297mm and a curvature radius ratio of with long, straight upstream and downstream pipe having 197mm diameter. Measurements of total pressure and mean velocity were obtained with 5-hole probe at the impeller upstream station. Streamwise mean velocity at station was used by Hot-wire anemometer. Measurement set up of 5-hole probe is shown in cross section plane of Fig.1(c). The 5-hole probe is inserted along the circumference of cross-sectioned plane and traversed toward radial direction.

RESULT AND DISCUSSIOIN

The influences of curved inlet pipe on the stage pressure ratio as well as on the axial distortion in the inlet pipe are quantitatively compared to that of straight inlet pipe. In addition, total pressure contours in inlet pipe and pressure distribution for the diffuser for three different flow rates (low flow rate, design point and high flow rate).

Figure 2 shows performance comparison of tested centrifugal compressor for each of inlet models (CP: Curved pipe, SP: Straight pipe). In the case with curved inlet pipe, deteriorating of pressure ratio compared to the case of SP appears in overall flow rate during rotational speed 14,000RPM. This tendency is increased as rotational speed increases. This result is similar to Ariga's (1989) suggestion that the performance degenerating effect due to the distortion grows as the rotational speed and the flow rate increases.

Figure 3 is the comparison of static pressure distribution at design point along circumferential angle inside diffuser. The measurement position is the radius ratio=1.094 corresponding to impeller exit and the radius ratio=1.738 corresponding to diffuser exit. From the figure, static pressure in case with the straight pipe is decreased compared to the case with the curved pipe. This pressure drop inside vaneless diffuser is similar to the result suggested by Engeda (2001).

Figure 4 and 5 indicate streamwise velocity profile and pressure coefficient measured at the θ = 0° (Inward wall) and θ = 180° (Outward wall) during operating point B. The static pressure at the inlet pipe wall is shown in the form of C_p in Fig. 4(a) for various *α*. C_p is the pressure coefficient, which is defined as $C_p=(p - p_{ref})/(\rho V_m^2/2)$. From the figure, difference of static pressure between outer wall and inner wall does not appear at pipe inlet in upstream station (x/D_{Inlet} = 3.2). However, in the section just before the inlet of the bend, that is, for x/D_{Inlet} = 0.23, the fluid is slightly accelerated near the inner wall. The acceleration of the fluid in this region leads to a weak secondary stream flowing from outer to inner wall over the cross section. When the fluid comes into the bend part, it experiences the centrifugal force and the static pressure in the fluid increases toward the outer wall. In the static pressure distribution at the bend part, the pressure gradient near the bend inlet gets favourable pressure gradient along the inner wall and unfavourable pressure gradient along the outer wall. The fluid near the inner wall is accelerated and the near the outer wall is decelerated as shown in Fig. 4(b). In the section just before the impeller (y/D_{Inlet} = 1), depletion of streamwise velocity occurs in inner wall side.

Figure 6 presents the total pressure and velocity contours obtained from 5-hole probe during the operating point B. The measurement positions are at α = 45° and 70° as shown in Fig. 1(c). In the α = 45°, favourable pressure gradient begins to turn adverse pressure gradient at inner wall as and shown in Fig. 5. In both cross-sectioned planes, pressure gradually increases from outer wall to inner wall. This pressure gradient can be formed by impinging on the outer wall of curved pipe and causes to secondary flow. Figure 7 shows the total pressure and velocity contours at upstream ($y/D_{Inlet} = 1$) of impeller as mass flow rate changes. In all of operating points, total pressure of inward wall side (θ= 0°) of circular-sectioned pipe is decreased because of the distortion effect due to the curved pipe. Difference of between maximum and minimum pressure is increased as close to high flow rate. The velocity distribution in Fig. 7(b) is consistent with streamwise velocity profile obtained from Hot-wire probe. In the velocity contours, streamwise velocity is decreased in inner wall and difference between maximum and minimum velocity is increased as mass flow rate increases.

CONCLUSION

This study is about influence of performance due to inlet distortion in centrifugal compressor.

1. Pressure ratio in the case with curved inlet pipe is decreased compared to the case with straight inlet pipe as rotational speed increase.
2. In the case with curved inlet pipe, distortion of streamwise velocity due to curved inlet pipe takes place inward wall side of pipe.
3. The reduction of total pressure due to curved inlet pipe appears inward wall side of pipe and increases as mass flow rate increases.

ACKNOWLEDGMENTS

This research was supported by the Korea Institute of Industrial Technology Evaluation and Planning (ITEP) grant funded by the Ministry of Knowledge Economy.

REFERENCES

Williams, D.D., 1986, "Review of Current Knowledge Response to Distorted Inlet Flow Conditions," AGARD Conference, Munich, AGARD CP-400.

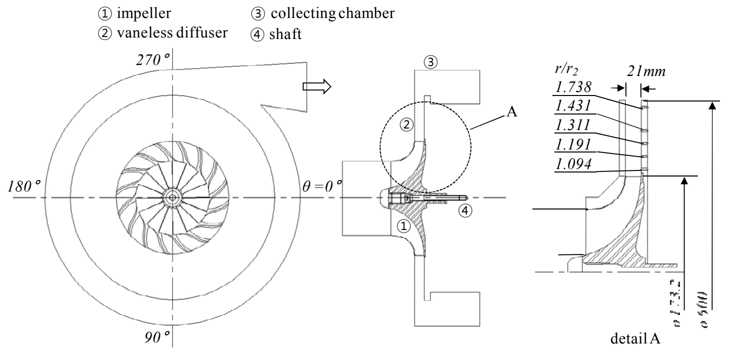
Ariga, I., Kasai, N., Masuda, S., Watanabe, Y., and Watanabe, I., 1982, "The Effect of Inlet Distortion on the Performance Characteristics of a Centrifugal Compressor," ASME PasperNo.82-Gt-92

Cain, S. A., and Padmanabhan, M., 1999, "Numerical Simulation of Flow Distortions and Swirl Due to a Combined Pipe Bend," ASME/JSME Joint Fluids Engineering Conference, Paper No. FEDSM99-7224.

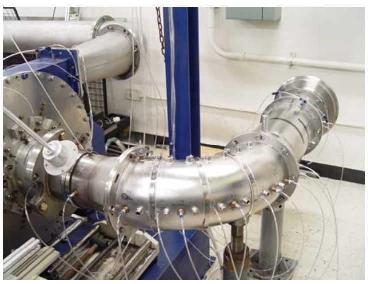
Kim, Y., Engeda, A., Aungier, R., and Direnzi, G., 2001, "The influence of Inlet flow distortion on the performance of a centrifugal compressor and the development of improved inlet using numerical simulations," IMechE Part A, Journal of Power and Energy, 215, pp.1-16.

Rowe M., 1970, "Measurements and computations of flow in pipe bends," Journal of Fluid Mech.43-4,pp.771-783.

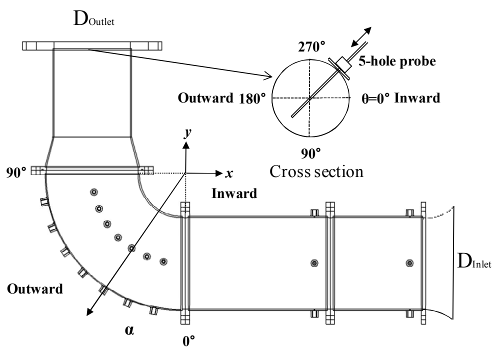
Sudo, K., Sumida, M., Hibara, H., 1998,"Experimental investigation on turbulent flow through a Circular-sectioned 90° bend," Experiments in Fluids,25, pp.42-49.



(a) Schematic of tested compressor



(b) Real image of test rig



(c) Curved inlet pipe

Figure. 1 Test rig with curved inlet pipe (a) Schematic of tested compressor (b) real image of test rig (c) curved inlet pipe

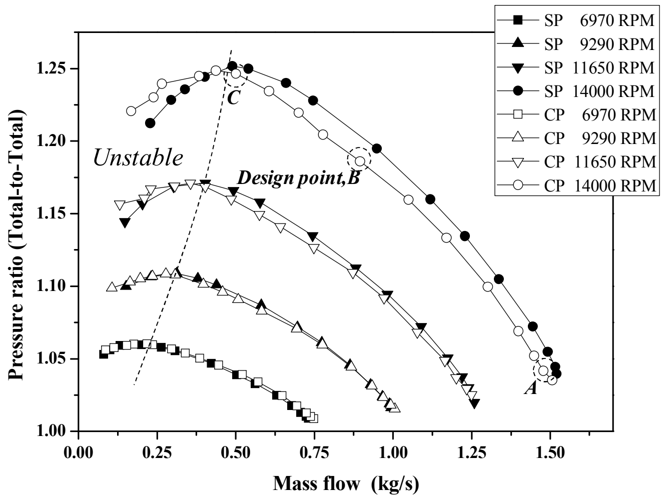


Figure. 2 Performance of a tested centrifugal compressor

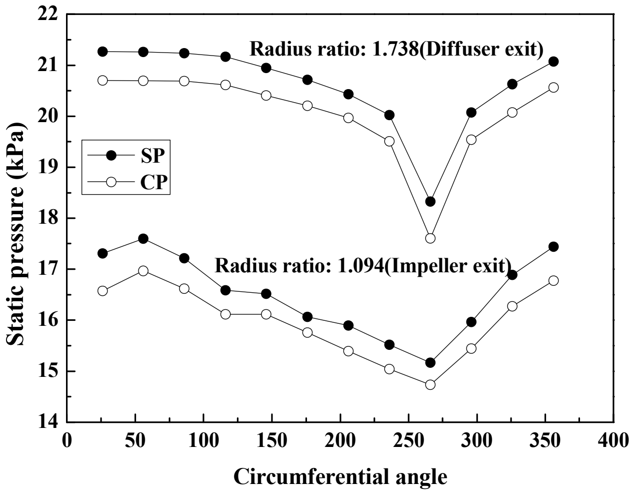


Figure. 3 Static pressure distribution in vaneless diffuser

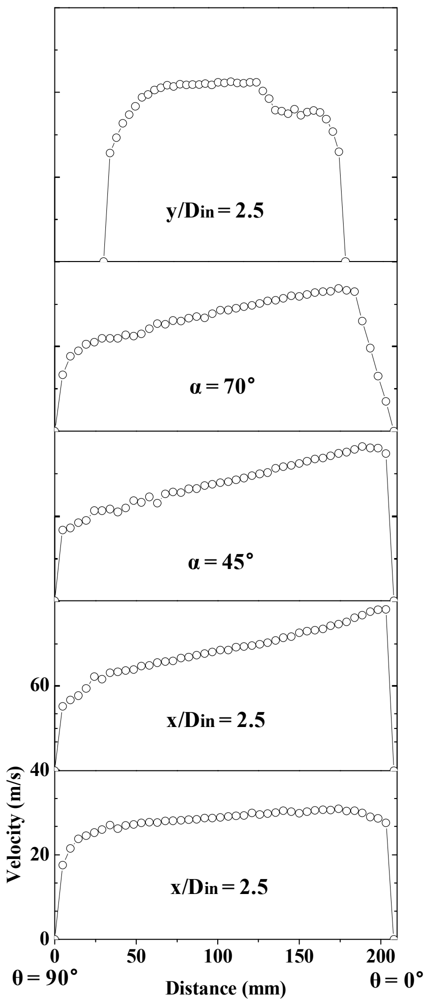


Figure. 4 Streamwise velocity profile at design point B

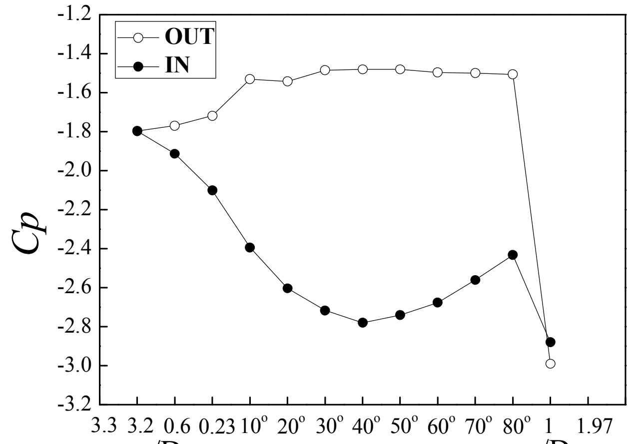
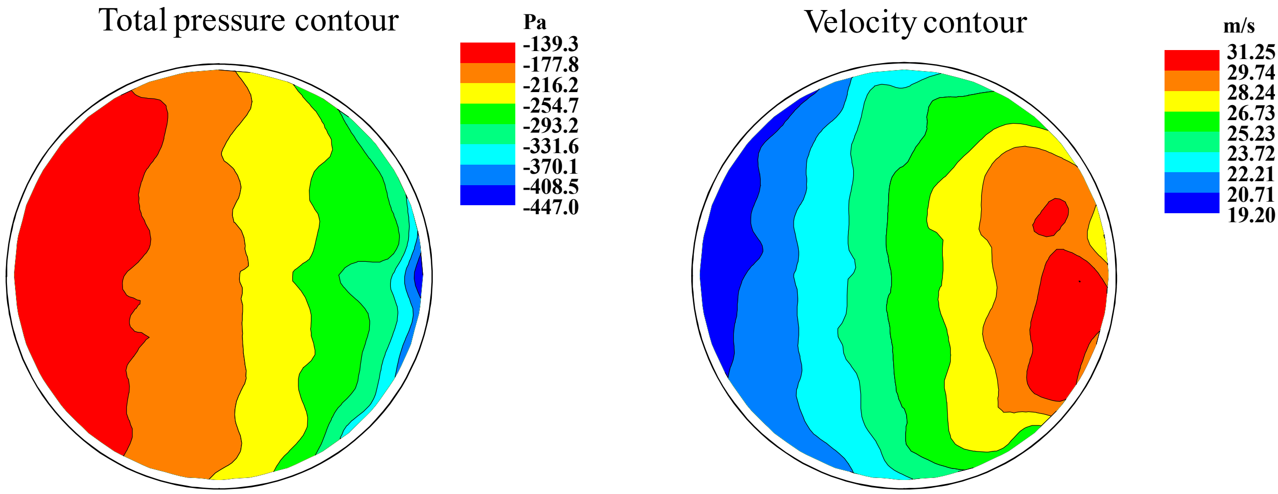
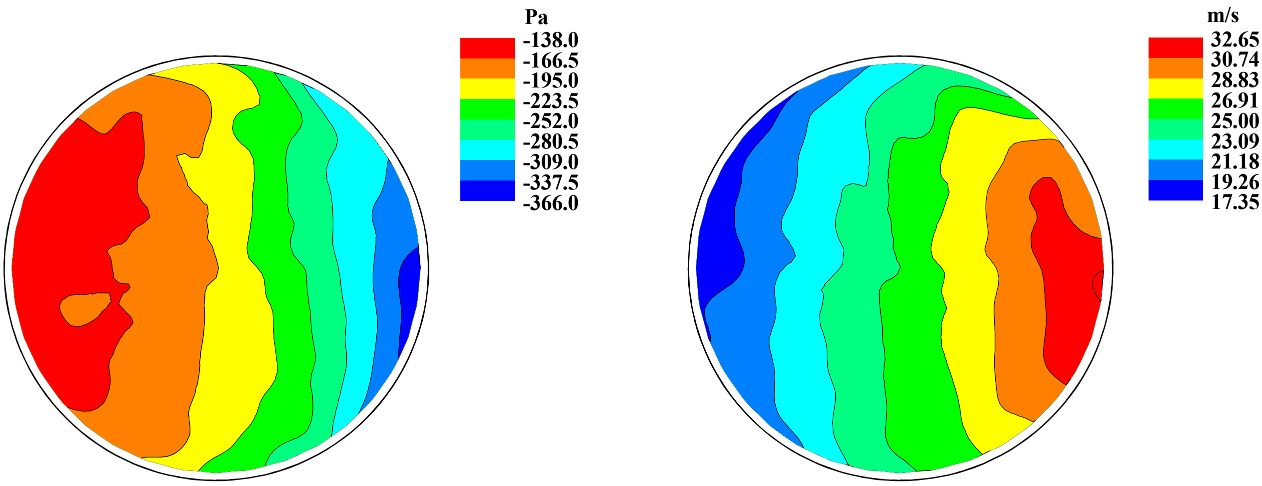


Figure. 5 Distribution of wall static pressure at operating point B



(a) α = 45° at operating position B



(b) α = 70° at operating position B

Figure. 6 Total pressure and velocity contour at operating point B

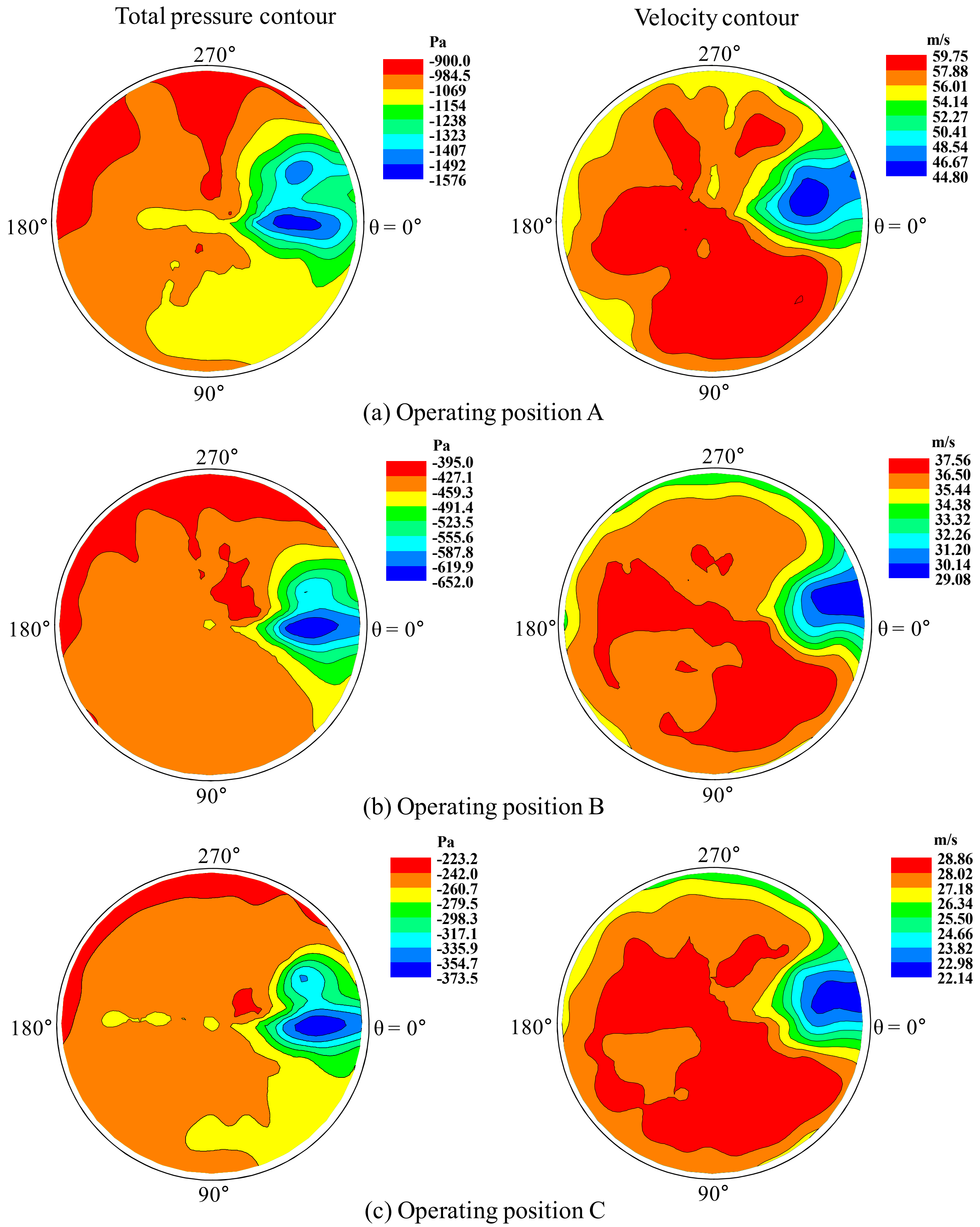


Figure. 7 Total pressure and velocity contour at different flow rate