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Design and Test of Transonic Compressor Rotor with Tandem Cascade

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ABSTRACT

The stage pressure ratio in axial flow compressors and fans can be considerably increased by using suitable tandem cascades. In the tandem cascade, the gap flow from the clearance between the front and the rear blades makes it possible to control the separation induced by shock. Therefore, the tandem cascade can obtain larger turning and higher pressure ratio without the increase in excessive loss. The three-dimensional viscous flow analysis was used to design the tandem cascade and the influence of the gap flow was investigated.

To confirm the concepts of the tandem cascade design, the single stage transonic compressor with tandem cascade was built and tested. The results of the compressor testing indicated that the pressure ratio of approximately 2.3 can be achieved in a single stage compressor. It was confirmed that with tandem cascades, the number of stages can be reduced compared to the conventional single cascades.

INTRODUCTION

As for the compressor which is one of the main components of an aero engine, high loading and high efficiency are constantly demanded for fuel consumption and weight reduction. By increasing the blade loading, it becomes possible to make the stage pressure ratio rise, and the number of stages of the compressor can be reduced. The typical methods of obtaining a high pressure ratio are increasing the rotating blade speed and enlarging the flow deflection in the rotor. As a result of increasing blade speed, the inlet Mach number of a rotor blade of modern axial compressor has reached transonic and the shock losses have increased. Therefore the control of the shock wave is becoming essential technology in maintaining compressor efficiency. On the other hand the technology of optimizing blade surface curvature to prevent the flow separation from suction surface is required to increase flow deflection in the rotor. The tandem cascade is proposed as one of the methods which aims at increasing the blade loading by increasing turning angle in the rotor blade. Bammert and Staude (1979) and Guochan et al (1985) applied it to the subsonic rotor of tandem cascade.

Figure 1 shows the schematic diagram of tandem cascade applied to the transonic compressor rotor. Conventional rotor has a single blade but tandem cascade consists of two blades, a front blade and a rear blades. In the transonic cascade, a shock wave occurs towards the suction surface of a blade from the leading edge of an adjacent blade, and the flow separation is caused by interfering with the boundary layer on the suction surface. However, the loading of a compressor rotor can be increased in the tandem cascade, since the flow from the gap between the front and the rear blades controls the separation. Adoption of the tandem cascade can improve significantly the loading limit on the conventional transonic cascade while maintaining rotor efficiency. The tandem cascade design was applied to the fan of an Air turbo Ram engine due to the strong demand of miniaturizing engine size (Hasegawa et al, 2003).

The aim of this study is to develop a transonic compressor rotor with tandem cascade. The single stage transonic compressor with tandem cascade was designed and aerodynamic rig test was carried out. Figure 2 shows the stage pressure ratio of axial compressors and fans plotted against circumferential blade speed at mean radius. The target of this study is shown in this figure. The targeted rotor loading is very high within the blade speed generally used.

In this paper the aerodynamic design concepts of the tandem cascade rotor blade are firstly described, and then test results of the single stage compressor rig is presented and discussed.

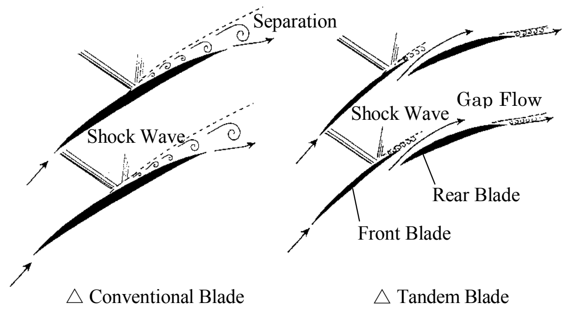

Fig. 1 Comparison of conventional blade and tandem blade

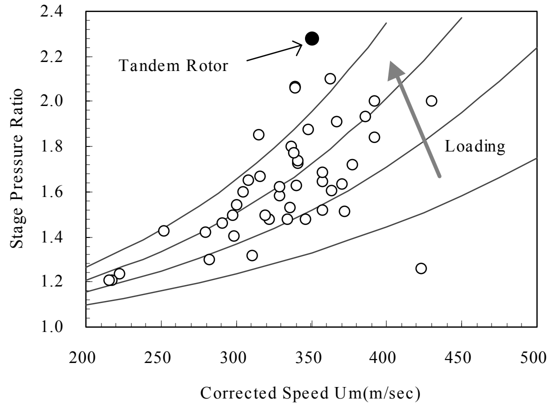

Fig. 2 Trend of stage pressure ratio and blade speed

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AERODINAMIC DESIGN

The aerodynamic design of the current compressor with the tandem cascade is similar to the design procedure of conventional compressors. The preliminary design was accomplished with axisymmetric, streamline-curvature-type and full radial equilibrium analysis method. In this process, it was considered that the front and the rear blades of the tandem cascade are independent. Losses were assumed to be equal to the sum of two components; losses due to diffusion factor, and losses due to a normal passage shock wave. Loss parameter used for tandem cascade was based on two dimensional cascade test data. To avoid excessive shock loss the inlet Mach number at rotor tip was restricted by approximately 1.4. Diffusion factors of the front and the rear blades of the tandem blade were kept at around 0.4 respectively. However, if a conventional blade is designed according to this velocity triangle, the diffusion factor will exceeds 0.6. Since the absolute exit flow angle of the tandem rotor was over 50 degrees and the inlet Mach number at stator hub was 0.98, the stage exit flow angle was set at 30 degrees in order to keep the aerodynamic loading level of conventional stator vane. The design parameters of single the stage compressor with tandem rotor cascades are shown in Table 1.

The blades were designed using multi-circular arc airfoil (MCA) for the front and the rear blades of tandem cascade and the stator vane. Each blade is separately designed based on the velocity triangle. The tandem configuration shown in figure 3 is defined by the relative gap normalized by blade pitch. The gap of tandem cascade is a displacement in circumferential direction between a trailing edge of the front blade and a leading edge of the rear blade at the hub. The three dimensional viscous flow analysis program of Dawes(1988) was used to optimize the detailed blade geometry. The tandem cascade had to be analyzed combining the front blade with the rear blade. The design was analyzed with the splitter type mesh and with no tip clearance. Each profile of the front and the rear blades were adjusted until the velocity distributions on the blade surface became suitable. In this process the relative gap at the hub was fixed at 0.12. The center of gravity of the profiles at each radial section was radially stacked in order to reduce bending stresses in the blade. Therefore, the relative gap varies in each radius location. The radial distribution of the relative gap is shown in Figure 4.

Table 1 Design specification of the transonic compressor

Corrected speed	29134 rpm
Rotor inlet radius	133.6 mm
Corrected flow rate	5.49 kg/s
Stage pressure ratio	2.22
Adiabatic efficiency	85.2 %

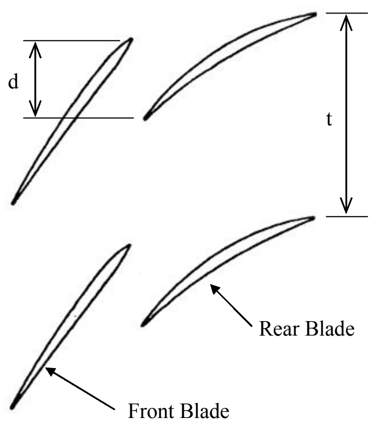


Fig. 3 Tandem rotor geometry: ***d*** gap width, ***t*** pitch

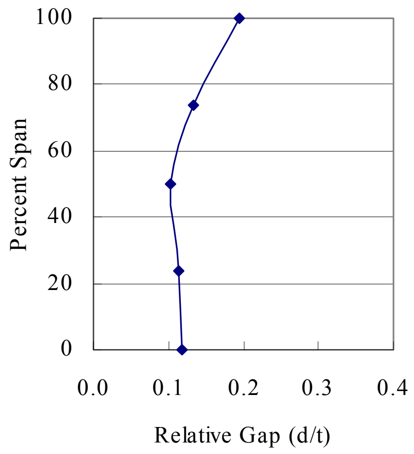


Fig. 4 Distribution of relative gap (d/t = 0.12 at hub)

In order to control the acceleration of the gap flow in the gap, blade profiles were adjusted so that the gap becomes large at the tip. After each blade profile was obtained, the gap was varied to adjust the strength of the gap flow which determines the performance of the tandem cascade. At this point, the blade geometry of the front and the rear blades had been fixed.

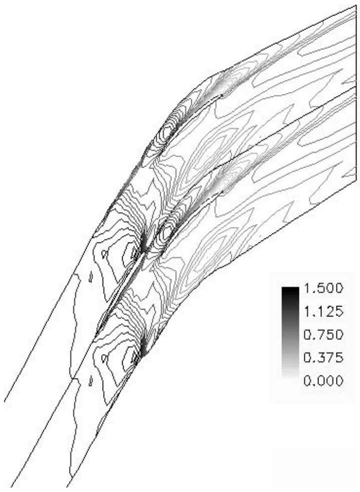
As an example of the influence of the relative gap to the tandem cascade performance, Mach number contours at 90 percent span height of the relative gap 0.06, 0.12 and 0.31 are shown in figure 5. In cases of the relative gap 0.06 and 0.12, there are no flow separation from the suction surface of the rear blade. But in the case of the relative gap 0.31, the wake from the front blade interfered with the rear blade and caused large separation. Fig. 6 shows the highest efficiency acquired in the analysis of the each gap. It was proved that rotor efficiency dropped rapidly with more than the relative gap 0.3. Consequently, the relative gap 0.06 and 0.12 were selected to obtain the expected effect of the tandem cascade. The blade parameter of rotor and stator designed for transonic compressor as a result is shown are Table 2.

Table 2 Design parameter of rotor and stator

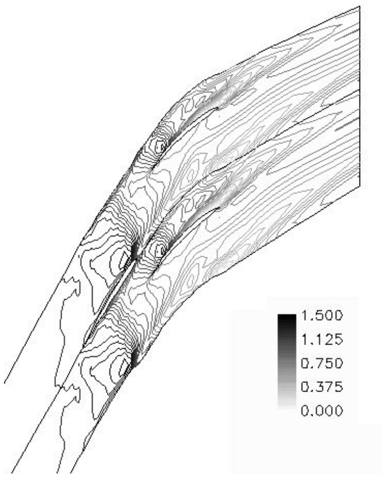
Parameter	
Rotor	
Number of blades	24
Inlet relative tip mach number	1.36
Mean blade speed	350 m/s
Inlet radius ratio	0.69
Aspect ratio (Front / Rear)	1.27 / 1.01
Mean Solidity (Front / Rear)	1.00 / 0.99
Stator	
Number of blades	31
Inlet hub mach number	0.98
Aspect ratio	0.63
Mean solidity	1.66

RIG TEST

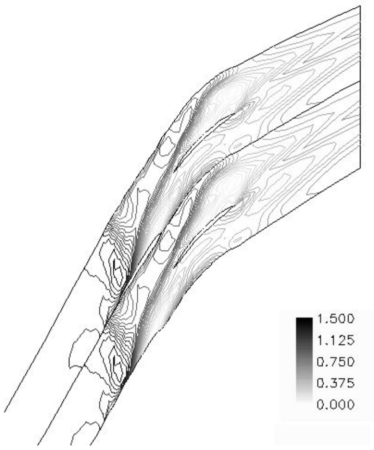
In order to investigate the aerodynamic characteristics of the compressor with the tandem cascade, the rotational rig tests were carried out. The tests were performed with two kinds of configurations, the rotor with tandem cascade only and the single stage. In the first step, the rotor tests were carried out with two cases of gaps which could expect the effect of the tandem cascade through analysis prediction. In the second step, the test of the single stage configuration which consists of the tandem rotor blades and the conventional stator vanes was carried out.



(a) Relative gap $(d/t)_{hub}$ = 0.06



(b) Relative gap $(d/t)_{hub}$ = 0.12



(c) Relative gap $(d/t)_{hub}$ = 0.31

Fig. 5 Contour of relative Mach number at 90%span

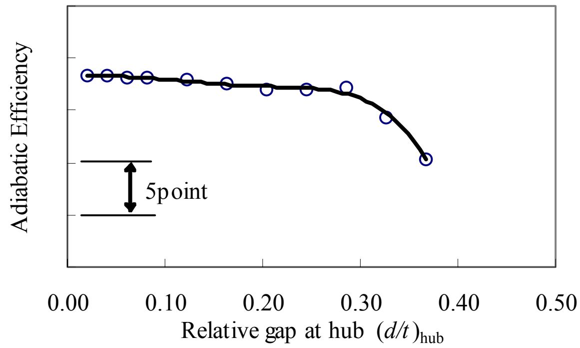


Fig. 6 Effect of gap to the performance of tandem rotor

Figure 7 shows the assembled compressor rotor of tandem cascade. The rotor was built from two separate integrally bladed disks, a front blade and a rear blade. Therefore it was possible to select the gap by adjusting the relative circumferential position of the two discs. Figure 8 shows the cross section of the test vehicle with the single stage configuration.

To acquire the overall compressor performance, multi-head rakes for total pressure and total temperature were inserted at the compressor inlet station and the exit station. In order to investigate the detailed radial distribution of total pressure, total temperature and yaw angle, both three-hole probe and total temperature sensor were traversed at the compressor exit. Figure 9 shows measurement stations in two configurations.

In the tandem rotor tests, the static pressure contours on the rotor casing were also measured in order to directly compare with the computation results of the flow structure at the tip. The dynamics of static pressure were observed using sixteen high-response transducers mounted on the casing above the rotor tip referring to the way shown by Matsuoka et al (1995).

Fig. 7 Rotor of tandem cascade

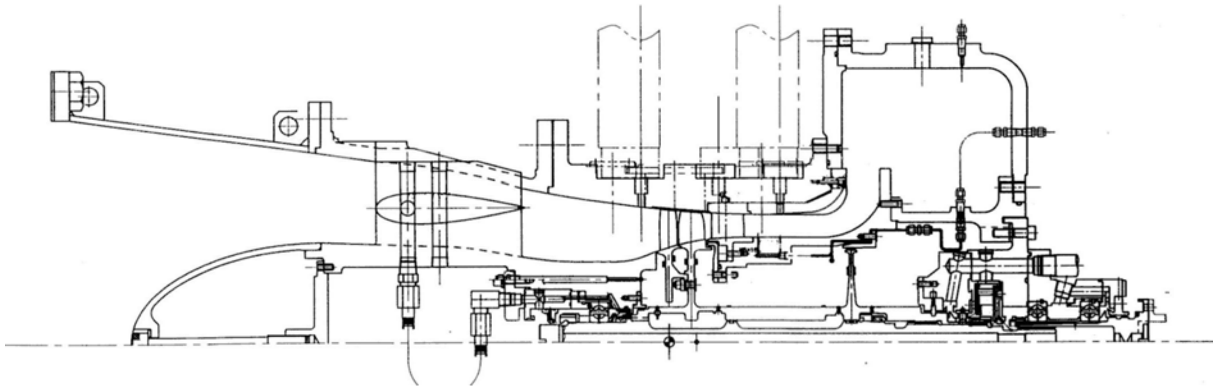
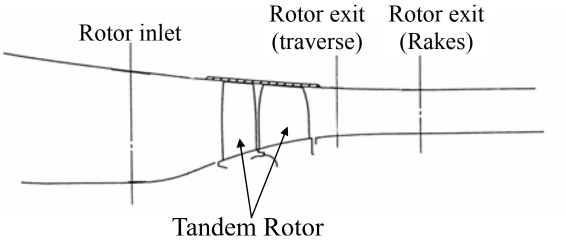
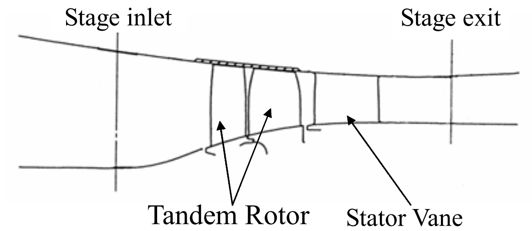


Fig. 8 Cross section of transonic compressor rig



(a) Tandem rotor test



(b) Stage test

Fig. 9 Measurement station

TEST RESULTS AND DISCUSSION

<u>Rotor test</u>

The rotor performance tests were carried out with two cases of tandem cascades, which predicted good performance in numerical analysis. They were Case1 with the relative gap 0.12 and Case2 with the relative gap 0.06. Figure 10 shows the comparison of measured aerodynamic characteristics between these two cases. In this figure, the pressure ratio and the corrected mass flow are normalized by one of the choke condition at design speed of Case1. Case1 shows better performance in pressure ratio and surge margin than Case2. Therefore Case1 was measured in more detail, by utilizing the traverse measurement at rotor exit and the static pressure measurement on the rotor casing.

Flow angles and total pressures at the rotor exit station were measured by traversing three-hole probe. The loss coefficients from test results were computed by quasi-three dimensional through-flow calculation so that flow angles and total pressures will be satisfactory to the test result. Figure 11 shows the radial distribution of total pressure loss coefficients from the test result and from the design. The loss used in design was based on two dimensional cascade test data. The test result showed lower loss than design at mean and nearly equal to design at the tip. However it showed higher than design at the hub. The increase of loss at the hub often occurs due to the secondary flow on the hub region. However it can be considered that the narrowness of gap at the hub also influenced this pressure loss. Therefore the tandem cascade Case2 with a smaller gap than Case1 could not perform as good as predicted by numerical analysis in the rotor test.

Figure 12 shows the measurement result of blade to blade static pressure distribution on shroud surface at 90 percent design speed and numerical prediction at design speed. A measured passage shock is located near the trailing edge of the front blade and is in good accordance with numerical prediction. The pressure distribution at design point could not be measured unfortunately due to sensor module trouble. However, it is predicted that the structure of a shock wave is similar with the structure of measured 90 percent design rotation speed.

<u>Stage test</u>

The test of the transonic stage which consists of the tandem cascade Case1 and the conventional stator vane was carried out. Figure 13 shows the aerodynamic characteristics of the single stage compressor at 60 to 100 percent corrected speed. In this figure, the solid line shows the actual point where the surge occurred. Summary of measured stage performance at peak efficiency in design speed is shown in Table 3. The measured pressure ratio was 2.28 and more than the design target. Adiabatic efficiency was 84.9% and corrected mass flow was 5.58kg/s.

Figure 14 shows a comparison of the traverse measurement result of the flow angle and the pressure ratio at the stage exit and the numerical analysis. The distribution of the measured flow angle is close to that of the design except for the hub region where higher turning occurs. The decline of the pressure ratio near the tip and the hub is not due to excessive loss in the stator but to the secondary flow and tip clearance in the rotor.

Table 3 Test result measured at peak efficiency

Corrected speed	29134 rpm
Corrected flow rate	5.58 kg/s
Stage pressure ratio	2.28
Adiabatic efficiency	84.9 %

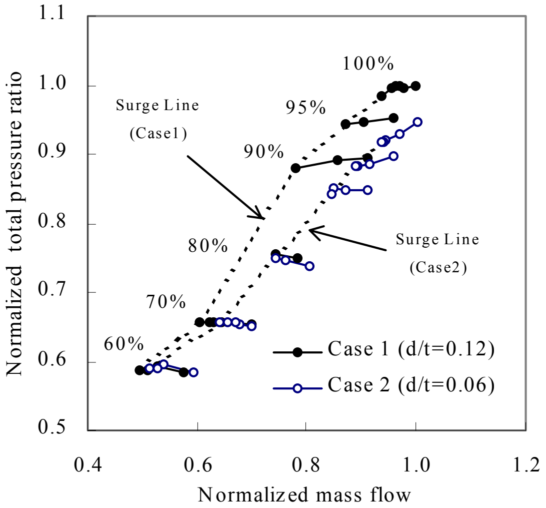


Fig. 10 Characteristics of the tandem rotor

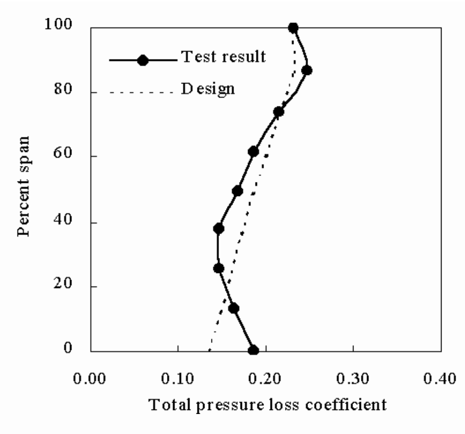
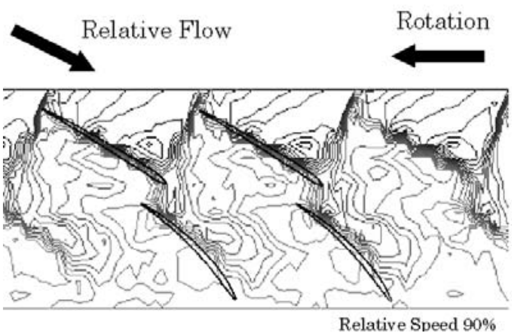
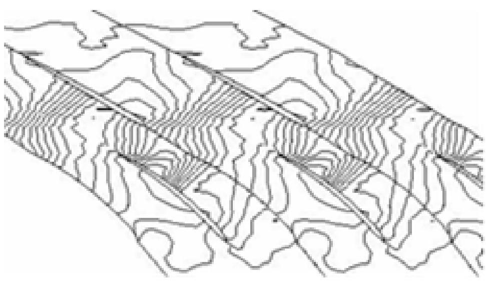


Fig. 11 Total pressure loss coefficient



(a) Measured at 90% speed



(b) Computation at Design speed

Fig. 12 Blade to blade static pressure distribution

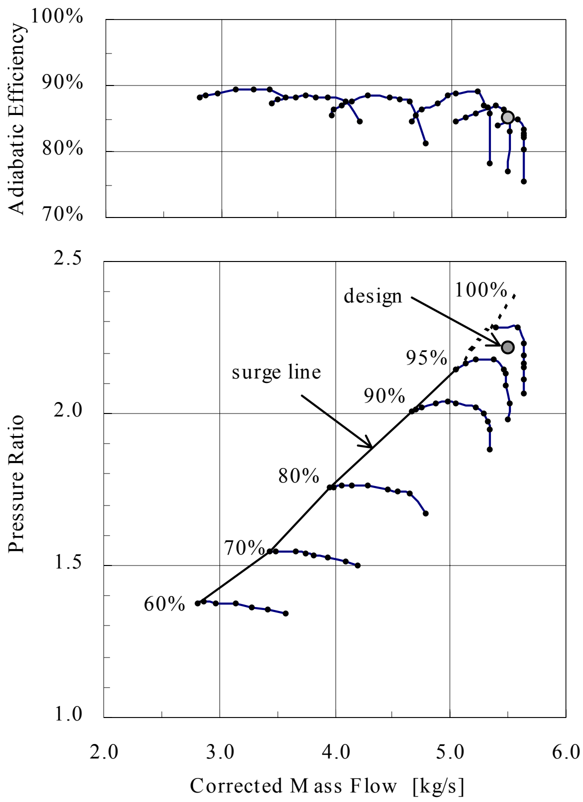
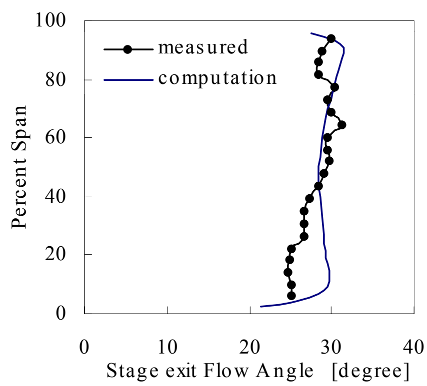
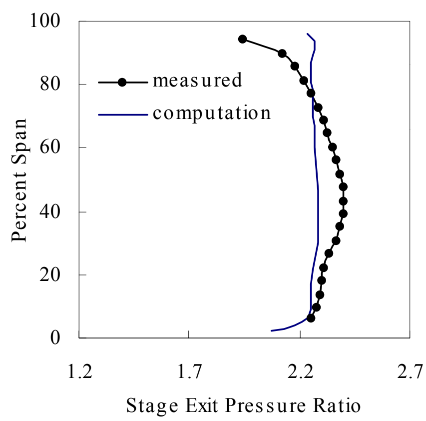


Fig. 13 Characteristics for single stage compressor



(a) flow angle



(b) Pressure ratio

Fig. 14 Radial distribution at stage exit

CONCLUSION

The tandem cascade was applied to the transonic compressor rotor. The tandem cascade was designed according to the design procedure of conventional cascade and optimized the gap between the front and the rear blades. The numerical and experimental results indicate that there is an optimum gap for the tandem cascade, and the relative gap 0.12 is recommended. The results of the compressor testing indicated that the pressure ratio of approximately 2.3 can be achieved in single stage compressor.

These tests have indicated many possibilities of improvement and further development. It is expected that with the tandem cascades the number of stages can be reduced to the conventional single cascade.

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