Proceedings of
The Second National Congress on Fluids Engineering
August 22-24, 2002, Muju, Korea

Effect of tip-leakage flow on an isolated rotor of an axial compressor

Dongwook Yim*, N. A. Ahmed** Myeongho Lee*** and B. E. Milton****

축류압축기의 회전차에 관한 누설유동의 영향

임동욱*·N A. Ahmed**·이명호***, B. E. Milton****

Key words : *Tip-leakage(극간격누설), Axial compressor(축류압축기),Five-hole prove(오공프루브),rotor(회전차),*

Abstract

It has been recognized that the flow in the blade passage of an axial turbomachinery rotor is very complex and is influenced by various flow phenomena, of which the tip leakage flow passing through the gap between rotor blade tip and casing plays a significant role. The losses produced due to the existence of the clearance have been known to be a large contributor of the rotor overall losses. Despite several experimental studies on non-rotating blade in the cascade configuration, and on actual rotating blades, the detailed nature of the complex flow phenomena associated with tip leakage, however, remains largely unresolved. Thus, a single-stage compressor test rig was built and measurements were taken at upstream and downstream of the rotor of this compressor at the aerodynamics laboratory of University of New South Wales. A five-hole probe and a hot-wire probe were used to measure mean and fluctuating flow parameters. The results show that tip leakage losses rise rapidly beyond tip gap of 0.01. Furthermore, the present project also identifies the regions in the wake behind the rotor of the axial compressor where such losses are concentrated. These results should be useful in the better design of rotors for improved performance of axial compressor.

1. Introduction

The development of modern day civilization is heavily dependent on the continued improvement in the performance of turbomachine such as the axial flow compressor, which is used in various industries aircraft, land-based plant, and so forth. Designers of axial flow compressor are constantly striving toward higher stage loading and lower aspect ratios to increase their efficiencies [1,2,23]. This in turn requires a thorough understanding of the mixing effect and the secondary flow losses taking place inside this turbomachines, particularly in the blade passage. It has been recognized that the flow in the blade passage of an axial turbomachinery rotor is very complex and is influenced by various flow phenomena, of which the tip leakage flow passing through the gap between rotor blade tip and casing shroud plays a significant rate. [1, 2, 4] The losses produced due to the existence of the clearance have been known to be a large contributor of the rotor overall losses [1, 2]. The leakage flow arises due to a pressure difference between the two surfaces of the blade at the tip [2]. Apart from shrouded turbomachines, most turbomachinery rotor in general must have a small but finite clearance relative to its surrounding casing. [2] The clearance is typically about one percent of the blade span for the compressors and turbines used in gas turbine engines. However, the flow which leaks through this small gap has a surprisingly large effect in the aerodynamic performance of the machine. It may account for as much as one third of the losses in an axial turbomachine. Recently it has been realized that the tip-leakage flow also has a major influence on the initiation of stall in transonic fans and compressors [13]. In hydraulic turbomachines, the cavitation in the tip-leakage flow is also a significant concern [8]. Until now, much experimental studies [4, 5, 6, 7] on the turbomachinery have generally been carried out on non-rotating blade in the cascade configuration in which, however, a number of important turbomachinery effects arising from rotation are missing. Consequently, to obtain more realistic and accurate determination of flow pattern in the turbomachinery, experimental studies have also been attempted using test rigs with rotating blades. [2, 9, 10] However, due to the complex phenomena resulting from the interaction of rotating blades, the task of predicting accurately the tip leakage development through a rotor and its effect on the performance of an axial compressor has remained a difficult undertaking.

* University of New South Wales, dwyim@korea.com
** University of New South Wales, n.ahmed@unsw.edu.au
*** 서울산업대학교 기계공학과, myeong@snut.ac.kr
**** University of New South Wales, b.milton@unsw.edu.au

2. Literature review

2.1 General background

Axial flow compressors are designed for high volume, high efficiency and high reliability applications. These applications include gas turbines, air separation, sewage treatment, wind tunnels and so forth. The axial compressor imparts energy to a continuously flowing fluid by the dynamic action of moving blade rows. The rotor changes the stagnation enthalpy, stagnation pressure, and kinetic energy of the fluid. The flow in a blade passage of an axial compressor rotor, however, is influenced by various aerodynamic phenomena such as hub wall boundary layer, annulus wall boundary layer, separated flow, rotor wake and tip leakage flow.

The essential element of a compressor is into blade because it is this component which impart the force and, more relevantly, the moment to the flow. Blade in an axial compressor has some features in common with aircraft wing but the situation in the axial compressor is more complicated. The flow varies very strongly along the span, both because of endwall boundary layer effects including tip leakage and because the radius and the blade speed change markedly.

3. Experiments

3.1 Experiment setup

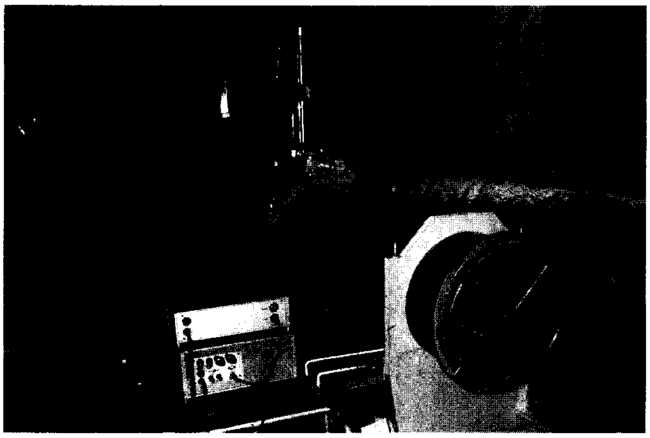

Fig. 3-1. Experiments setup

Figure 3-1 shows the experimental setup used in this project which was located in the aerodynamics laboratory of the University of New South Wales. The test rig was built specifically for this project to conduct on a single-stage axial compressor. The compressor consisted of an axial rotor housed in a 200mm long pipe of 295mm internal diameter. The compressor was sandwiched between two 1200mm long PVC pipes having 295mm, i.e. the same internal diameter as the compressor. A 74mm long conical inlet was attached at the upstream end of the pipe to ensure smooth flow in the compressor. The downstream pipe length was added to exhaust the flow smoothly to atmosphere. Air was drawn in the compressor using a 1.5kW three-phase AC motor (manufactured by CMG Australia) to produce a rotation range of 0~1,500 rpm. A speed controller was used to regulate the rotor rotation and hence the mass flow rate within the compressor.

Details of the dimensions of the test rig of this study are given in table 3-1.

	Upstream	Test section	Downstream	Conical inlet
Length	1200mm	200mm	1200mm	74mm
Inner Diameter	295mm	295mm	295mm	404mm

Table 3-1. Geometry of the test arrangement

Rotor was located in the center of the test section. A detail dimension of the blade arrangement is given in table 3-2.

Blade diameter	293mm	289mm	287mm
τ	1mm	3mm	4.5mm
τ/c	1.2%	3.6%	5.4%

Table 3-2. Test rotor dimensions

[c (Chord length) : 84mm, τ (tip clearance), hub length : 150mm]

3.2 Instrumentation

3.2.1 Five-hole probe

The main purpose of using a five-hole probe is to determine three-dimensional characteristics of the flow in complex flow fields. The five-hole probe is an instrument often used in low-speed wind tunnels to measure flow direction, static pressure, and total pressure in subsonic flow. Five-hole probe can be used either in the nulling or the non-nulling modes. Space limitations usually make nulling technique impractical. Hence five-hole probes in a non-nulling mode are, generally, employed for measurements in low speed, incompressible flows. [3]

The probe was manufactured by Flow Technology in Germany. The probe consisted of four direction-sensing ports plus a centre port which were precision bored into a sphere brass tip. This five-hole probe head was 6mm long and 2.8mm diameter. The measuring speed range of the five-hole probe was between 0.01~0.95 Mach. The expected Mach number range in this project was between -30°< α (pitch angle) <+30° and -30°< β (yaw angle) <+30°.

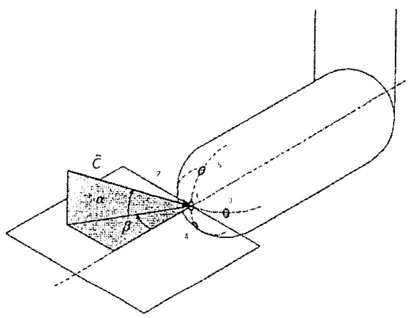

Fig. 3-2 Position of the pressure bores and orientation of the angle definition at the probe head

Figure 3-2 shows position of the pressure bores and orientation of the angle definition at the probe head.

The closed section open circuit wind tunnel used for five-hole calibration in this project which was located in aerodynamic laboratory of the University of New South Wales. This wind tunnel operates at subsonic speeds of between 0~30m/s and has a test cross sectional area of 18 inches by 18 inches. The motor drive on the tunnel consists of two DC motors, controlled by a thyristor controller.

The calibration of five-hole probe was performed following Treaster, and Yocum's method done. [3] The probe was first aligned in the wind tunnel and zero yaw angle was defined to be the angle where the pressures at ports 2 and 3 were equal. Then the probe was manually fixed at a known yaw angle and then rotated through the pitch plane. Measurements were obtained for both pitch and yaw angle between −30°~+30° in increments of 10°. In addition to the pressures from the probe's five holes, the total and static pressures were measured at the same stream wise position with a pitot-static probe.

The calibration coefficients are defined [3] as:

$$C_{P_{Yaw}} = \frac{P_2 - P_3}{P_1 - \overline{P}} \tag{1}$$

$$C_{P_{Pitch}} = \frac{P_4 - P_5}{P_1 - \overline{P}} \tag{2}$$

$$C_{P_{Total}} = \frac{\overline{P} - P_{Total}}{P_1 - \overline{P}} \tag{3}$$

$$C_{P_{Static}} = \frac{\overline{P} - P_{Static}}{P_1 - \overline{P}} \tag{4}$$

$$\overline{P} = \frac{P_2 + P_3 + P_4 + P_5}{4} \tag{5}$$

P_1, P_2, P_3, P_4 and P_5 refer to pressures from different ports. Each subscript represents the port number. This P_1 represents the pressure from the center port, while P_2 and P_3 are the pressure measured in the ports in the yaw plane and P_4 and P_5 are the pressure measured in the pitch plane. With these results, graphs of a total of five calibration curves were plotted.

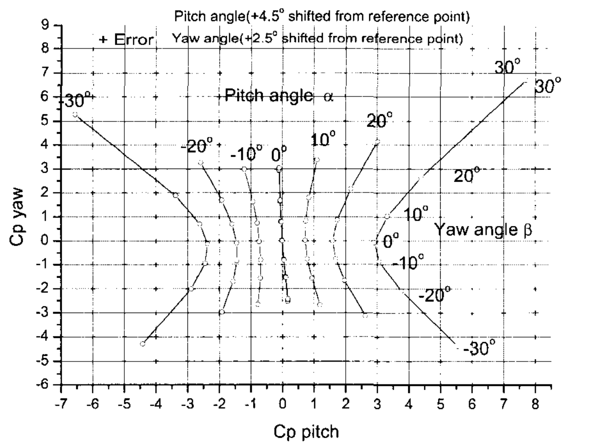

Fig. 3-3 Typical calibration data $C_{P_{yaw}}$ vs $C_{P_{pitch}}$

When the non-nulling method is used the probe can be placed in the flow and local total and static pressure, pitch angle, yaw angle and the three orthogonal velocity components can be calculated from the calibration data. Differential pressures (ΔP) were measured by five-hole probe at each measuring point, and then $C_{P_{Pitch}}$ and $C_{P_{yaw}}$ were calculated from formula (1)~(5). Interpolation program was used to evaluate the pitch and yaw angle at any position. From the α and β thus obtained, the total velocity vector and the other velocity components of the flow at the same point were obtained using the following formulae;

$$\overline{V} = \sqrt{\left(\frac{2}{\rho}\right)\left(\Delta P_1 - \Delta\overline{P}\right)\left(1 + C_{P_{static}} - C_{P_{total}}\right)} \tag{6}$$

$$V_X = \overline{V}\cos\beta\cos\alpha \tag{7}$$

$$V_R = \overline{V}\cos\beta\sin\alpha \tag{8}$$

$$V_\theta = \overline{V}\sin\beta \tag{9}$$

4. Results and Discussion

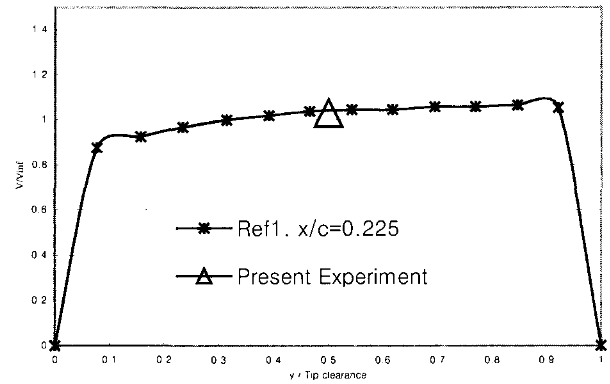

Fig. 4-1. Tip gap velocity at tip-clearance of 1mm

The mean velocity $\left(\frac{\overline{V}}{V_\infty}\right)$ at $\frac{y}{tip\ clearance(\tau)}$ =0.5 value was obtained from the mean of upstream and downstream of the rotor leakage passage velocities. This is shown in figure 4-1. To get some idea about the validity of results of this study, the published data from cascade tests of S. Kang et al. [4] were used for comparison.

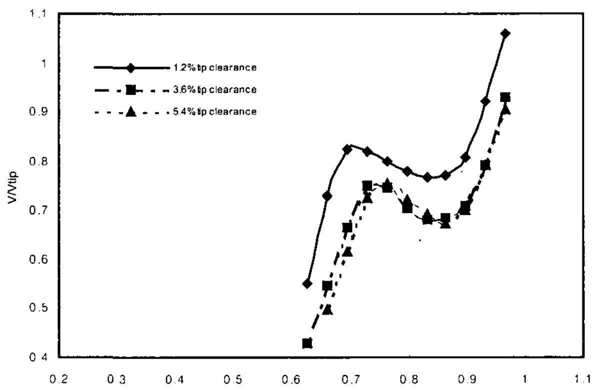

Fig. 4-2. Mean velocity $\left(\frac{\overline{V}}{V_{Tip}}\right)$ at $\frac{x}{c} = 0.02$

Figure 4-2 presents the mean velocity $\left(\frac{\overline{V}}{V_{Tip}}\right)$ at x/c=0.02 where is the pressure survey position. The trends of mean velocity are very similar at x/c=0.02, but the mean velocity values are different. The blade tip velocity (V_{tip}) was used to normalize the mean velocity. The mean velocity of τ/c=0.012 is higher than the other tip clearances. τ/c=0.012 is only 0.024 less than τ/c=0.036, however, the mean velocity of τ/c=0.012 is 13.02% larger than τ/c=0.036. τ/c=0.036 and 0.054 have very similar mean velocity values within error bend in range of R/R_0=0.76~0.97. Three different blade geometries were carried out same experimental condition.

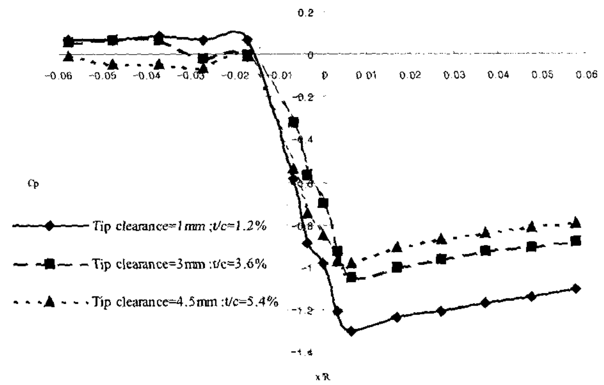

Fig. 4-3. Static pressure distribution on the casing surface

Figure 4-3 shows static pressure distribution on the compressor casing. Through the gap, the velocity increases, so the static pressure on the casing decreases, hence the C_p value becomes more negative within the gap between the upstream and downstream of the rotor. Behind the rotor, the total velocity decreases and the C_p value becomes less negative. With higher tip clearance, the drop in total velocity is more noticeable.

Overall losses were calculated total-to-total pressure at upstream and downstream. Overall losses are 12.3% at tip clearance ratio 1.2%, 28.95% at 3.6%, 25.02% at 5.4%. Tip leakage introduces losses in the overall performance of axial compressor. These losses increase rapidly beyond tip leakage gap of over 0.01. Losses increase at a grater rate with increase in $\frac{\tau}{c}$.

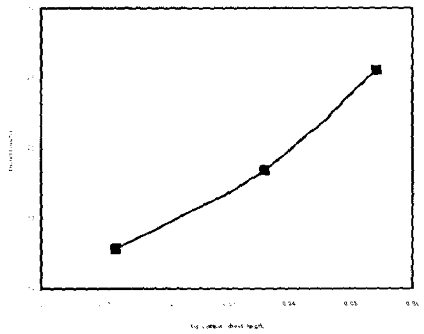

Fig. 4-3. Overall losses

5. Conclusions

The main conclusions of the investigation of tip leakage study carried out in this project on an isolated rotor of an axial compressor are summarized below:

These results show that tip leakage plays a significant role in the performance loss of an axial compressor as reported by works of other researchers both on cascade and rotating set-ups of axial compressor. This study, however, is more specific and quantifies accurately relationship between tip gap and performance loss of an axial compressor with an isolated rotor. The overall loss is found to increase beyond tip gap (τ/c) of over 0.01. The rate of increase, thereafter, is more rapid with further increases in $\frac{\tau}{c}$. The present study also shows that five-hole probe can be used effectively to determine the complex three dimensional nature of the flow field, particularly the mean flow characteristics associated with tip leakage flow. The five-hole probe is easy to use, but its application is limited to low angles of yaw and pitch. This study also demonstrates that this situation can be improved by rotating the probe to a different angular position such as the +50° of yaw angle, when high flow angularity is encountered.

References

[1] Von Karman Institute for Fluid Dynamics "Secondary and Tip-clearance flow in axial turbines" Lecture Series 1997

[2] Budugur Lakshminarayana, "Fluid Dynamics and Heat Transfer of Turbomachinery" 1996, Wiley-Interscience, First edition

[3] A.L. Treaster, A.M. Yocum , "The calibration and application of five-hole probes" ISA Transactions, Vol. 18(1979), pp23-34

[4] S. Kang, C. Hirsch "Tip leakage flow in linear compressor cascade" Journal of turbomachinery, Vol.116(1994), pp657~664

[5] J. A. Stoter, N. A. Cumpsty "Tip leakage flow in axial compressors" Journal of turbomachinery, Vol. 113 (1991), pp252-259

[6] J. P. Bindon, G. Morphis "The development of axial turbine leakage loss for two profiled tip geometries using linear cascade data" Journal of turbomachinery, Vol. 114(1992), pp198-203

[7] F. J. G. Heyes, H. P. Hodson, G. M. Dailey "The effect of blade tip geometry on the tip leakage flow in axial turbine cascades" Journal of turbomachinery, Vol. 114 (1992), pp643-651

[8] K. J. Farrell, M. J. Biller "A Correlation of leakage vortex cavitation in axial flow pumps" Journal of fluids engineering, Vol.116 (1994), pp551-55

[9] M. Inoue, M. Kuroumaru, and M. Fukuhara "Behavior of tip leakage flow behind an axial compressor rotor" Journal of engineering for gas turbines and power" Vol. 108 (1986), pp7-14

[10] I. H. Hunter, N. A. Cumpsty "Casing wall boundary -layer development through an isolated compressor rotor" Journal of engineering for power Vol. 104 (1982), pp805-818