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Influence of Blade Tip Clearance on Performance of Multistage Compressors

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ABSTRACT: Multistage compressor is the most important component of gas turbines used in oil and gas industries, marine and aeronautical applications. Performance of gas turbines depends mainly on its multistage axial flow compressor. This paper mentions to a study on influence of tip clearance of rotor blades and the case to pressure ratio and performance of multistage axial compressors. The study is conducted by CFD (Computational Fluid Dynamics) method. CFD is conducted with changing the tip clearance of rotor blades and the case of a three stage axial air compressor. Result from the study shows that the smaller value of the tip clearance of the rotor blades and case, the greater of the pressure ratio generated and the higher the performance of the air compressor. However, in this study, the authors have not evaluated the vibration effect to the rotor tip baldes of the compressor when the tip clearance of the rotor blade and case is changed. The vibration could be considered as a factor that needs to be evaluated in designing and manufacturing air compressors. This factor may affects to the choice of materials and technology for air compressor manufacturing to avoid the impact of the tip of the rotor blades and the case while the compressor operates due to blade vibration in a small tip clearance.

KEYWORDS: Performance, axial flow compressor, tip clearance, CFD, pressure ratio.

INTRODUCTION

Gas turbines are widely used in oil, gas industries, marine and aeronautical aplications. In marine, gas turbines are used in specialized ships, especially military ships, to improve the maneuverability and powering of the ships. In addition, gas turbines are also used for unmanned underwater vehicles such as torpedoes, AUVs that moving at high speed. For a gas turbine, multistage axial compressor is one of the most important parts and it determines the efficiency of the turbines. Figure 1 shows blade level of a multistage axial air flow compressor in an inductrial gas turbine.

Figure 1. Axial flow compressor

There are four flow types of the air flows around the blades of axial compressors. They are including the flow at the end of the rotor blades at the hub; the flow at the end of the rotor blades at the case wall, the flow at the end of the stator blades at the rotor hub and the flow at the end of the stator blades at the case wall. Therefore, researches of the flows in axial flow compressors are almost study in one of the above four flows.

The loss of compressive power and energy consumption due to leakage of the flow through the tip clearance of rotor blades with the case wall is the most commonly studied problem in gas turbine flow studies. In addition, the tip clearance of rotor blades and the case wall is also caused of other disadvantages to the performance of the compressor. For example, it causes a reduction of the work of compressor because there is no work created by blades from the air flow in the tip clearance of rotor blades and case wall, and the pressure drop across the blade serves to increase the flow through the clearance, thus reducing the available fluid on which the blade can add work [1]. Another study by Hesselgreaves showed a correlation to predict the efficiency drop of axial flow turbo machines. The study mentioned that an increase in tip clearance will have two distinct effects. It reduces work output and reduces performance due to loss in kinetic energy through tip leakage flow [2]. Smith did a study on the flow phenomena in compressor casing treatment. In this study, he mentioned to the relationship between the tip clearance of the blades and case wall and the flow pressure. He found that there is 23% drop in maximum pressure rise and 15% 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 [3]. Wisler, has shown in a study that 1.5 point drop in efficiency of a low speed compressor when the tip clearance was doubled [4]. Other authors, Varpe has studied on investigation of the shear flow effect and tip clearance ona low speed axial flow compressor cascade, Chan has studied on interaction between inlet boundary layer, tip-leakage and secondary flows in a low-speed turbine cascade, Lee has studied on characteristics of tip-clearance flows of a compressor cascade and a propulsion pump [5-7].

This paper will conduct research influence of tip clearance on performance of multistage compressors through computational fluid dynamics conduction.

COMPUTATIONAL FLUID DYNAMIC PROCESS

The computational fluid dynamics is a modern research method, with the help of specialized computer software. In this study, a three stage axial air compressor was used to evaluate the performance by changing the tip clearance of the first floor (R1) and the case. Figure 2 shows the tip clearance of the first floor (R1) and the case of the axial compressor that will be modified in the study.

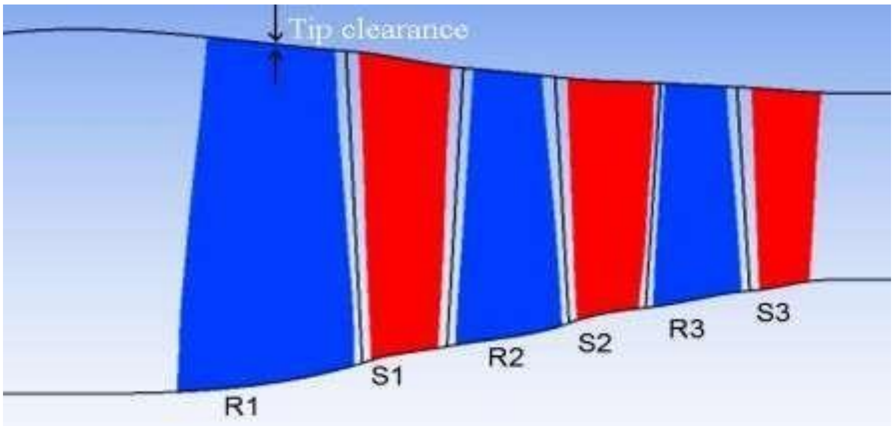


Figure 2. Tip clearance of rotor blades and compressor case

CFD was conducted for a three axial stage compressor with analysis and calculation via 3D-RAN method. The original design of tip clearance of the first floor rotor blades and the compressor case was 0.5 mm. The principle parameters of the compressor are shown in Table 1. The details of blade numbers of the rotor blades (R) and the stator blades (S) are shown in the Table 2. The construction of the floors of rotor baldes and stator blades are shown in Figure 3.

Table 1. Principle parameters of the compressor

Parameters	Value	Unit
Inlet total pressure	101.33	kPa
Inlet total temperature	288.15	K
Outlet static pressure	200 (variable)	kPa
RPM	8,500	rpm
Inlet Mass Flow Rate	85.91	[kg/s]
Inlet Volume Flow Rate	70.18	[m^3/s]
Total Pressure Ratio	2.39	
Total Temperature Ratio	1.36	
Total to total isentropic efficiency	79.18	%
Back pressure	200	kPa

Table 2. Rotor blades and stator blades in stages

Floor	R1	S1	R2	S2	R3	S3
Blade number	29	29	41	41	49	52
Reference Radius (mm)	328	328	328	328	332	332

where,

R1, R2, R3 are rotor blade number of floor 1, 2 and 3
S1, S2, S3 are stator blade number of floor 1, 2 and 3.

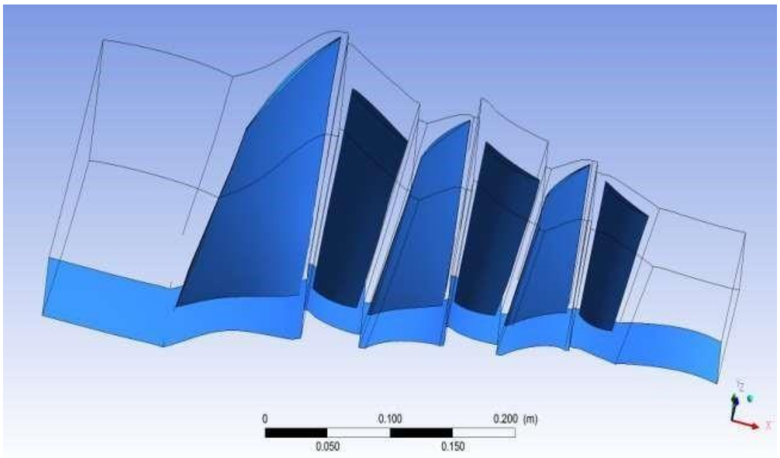


Figure 3. Construction of the stages of rotor and stator blades

CFD has conducted for four cases of different of tip clearance value of the first floor rotor blades and the compressor case of the three stage axial compressor, as shown in Table 3. Other parameters such as flow rate, pressure ratio and pressure efficiency are calculated to evaluate the performance of the three stage axial compressor when changing the tip clearance of the first floor rotor blales and the case of the compresor in the above mentioned studied cases.

Table 3. Studies cases of tip clearance of the first floor rotor blades

No.	Case 1	Case 2	Case 3	Case 4
Tip clearance (mm)	0.3	0.5	0.7	1.0

Domain and mesh

The domain for CFD process of the three stage air compressor is shown in Figure 4. A single periodic sector was modelled with periodicity interfaces in circumferential direction. Mesh was generated by using ANSYS Turbogrid software and analysis was carried out in ANSYS CFX software. All cases and boundary condition was set up in CFX pre-processor. 3D RANS simulation for steady state condition was performed at design operating speed. Around 550,000 hexahedral elements were used for analysis. Separate cases were generated to solve blade tip clearance variation of the first floor rotor blade and compressor case. CFX cases were setup in ANSYS workbench platform and performance curves were generated using parametric option available. Stage interfaces were provided between rotor and stator blades, where circumferentially averaged flow parameters were transferred from upstream domain to downstream domain.

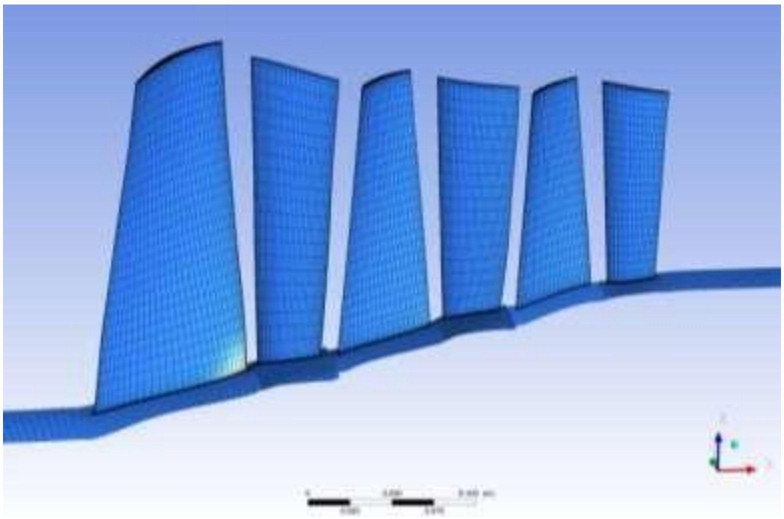


Figure 4. Domain for CFD process

Turbulent Model

To calculate the air flow surrounding the blades and case of the three stage axial compressor, a continuou equation of the flow, the Navier - Stokes equation, has been used with the k-ε turbulent model. The Navier-Stokes equation is written as follows [8]:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{1}{\rho}\left(\frac{\partial \tau_{xy}}{\partial y}+\frac{\partial \tau_{xz}}{\partial z}\right)+F_x \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{1}{\rho}\left(\frac{\partial \tau_{yx}}{\partial x}+\frac{\partial \tau_{yz}}{\partial z}\right)+F_y \tag{3}$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\frac{1}{\rho}\left(\frac{\partial \tau_{zx}}{\partial x}+\frac{\partial \tau_{zy}}{\partial y}\right)+F_z \tag{4}$$

where,

u, v, w: velocity in x, y and z axises
ρ: air density
p: pressure
τ: tangential stress

$$\tau_{xy} = \tau_{xy} \ \tau_{xz} = \tau_{zx} \ \ \tau_{yz} = \tau_{zy}$$

F_x, F_y, F_z: forces in x, y and z axises.

Boundary conditions

Boundary condition in CFD process are: inlet mass flow rate is 85.91 kg/s; inlet total pressure is 101.33 kPa; inlet total temperature is 288.15°K; outlet static pressure 200 (variable); shaft rotational speed is 8,500 rpm; total pressure ratio is 2.39; total temperature ratio is 1.36; back pressure is 200 kPa, density of air is 1.225 kg/m^3 and the viscosity of air is 1.7894× 10^{-5} kg/(ms).

RESULT AND DISCUSSION

After carrying out CFD process, the results of evaluating the working performance of the three stage compressor are shown in Figures 5-10 and graphs in Figures 11 and 12.

Figure 5 shows the velocity contour of stage 1, Figure 6 shows the velocity contour of stage 2 and Figure 7 shows the velocity contour of stage 3. These figures show the contours of stages relative velocity at 50% span.

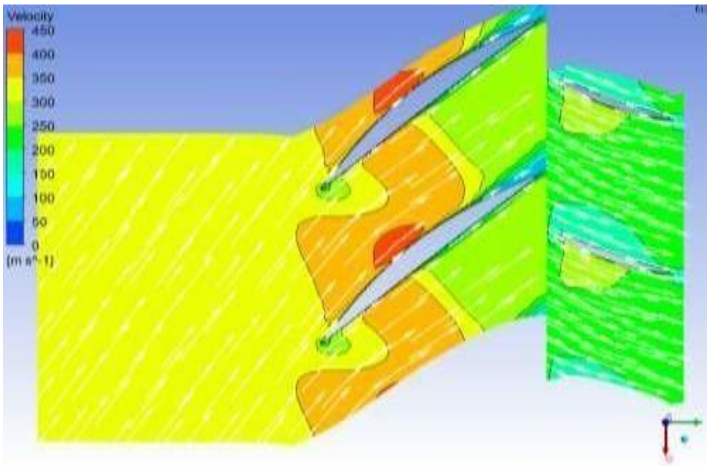


Figure 5. Velocity contours of stage 1

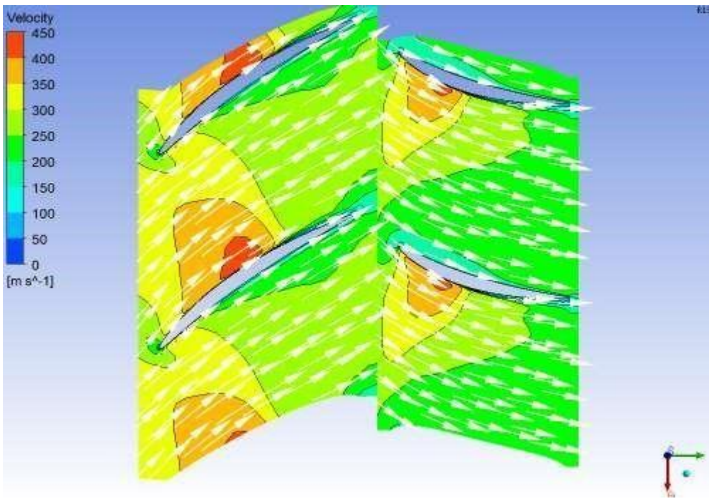


Figure 6. Velocity contours of Stage 2

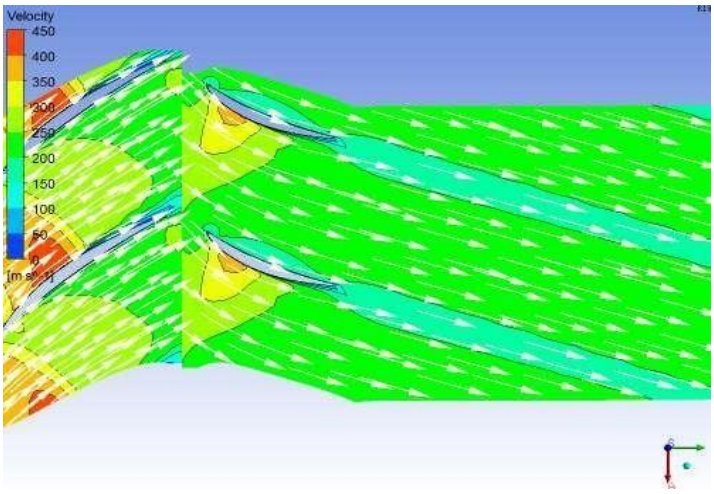
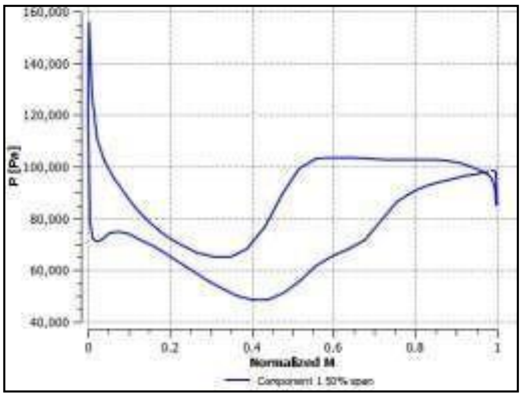


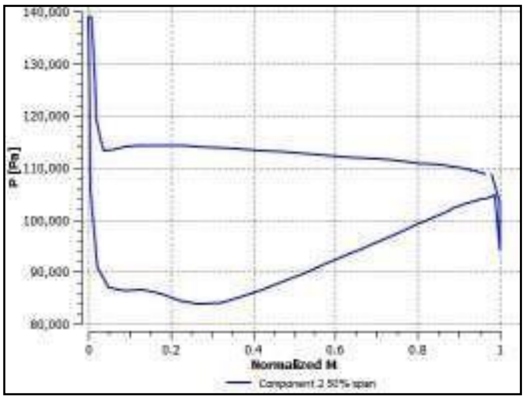
Figure 7. Velocity contours of Stage 3

Observations from Figure 5 to Figure 7 we can learn that there is a strong vortex field of the flow suroungding near the tip of the rotor blades and a weaker vortex field of the flow suroungding near the tip of the stator blades. Thus, may considere that the flow surounding the stator blades is more harmonious than the flow surounding the rotor blades.

Figure 8-10 show the loading chart of rotor and stator blades in three stages. Figure 8 shows the blade loading chart of R1 and S1. Figure 9 shows the blade loading chart of R2 and S2. Figure 10 shows the blade loading chart of R3 and S3.

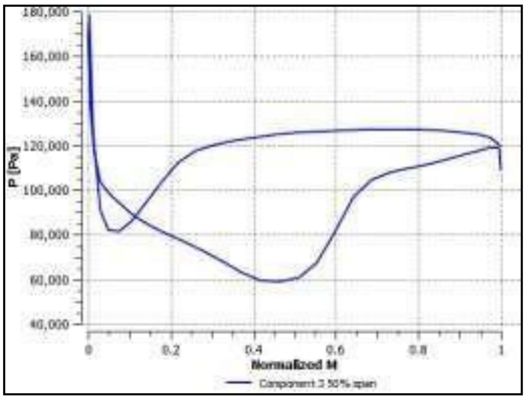


a. Blade loading chart of R1

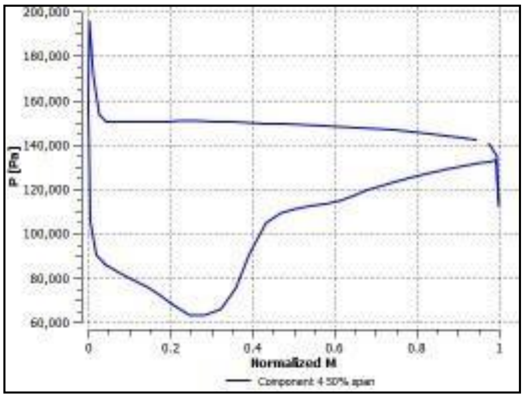


b. Blade loading chart of S1

Figure 8. Blade loading chart of the first stage

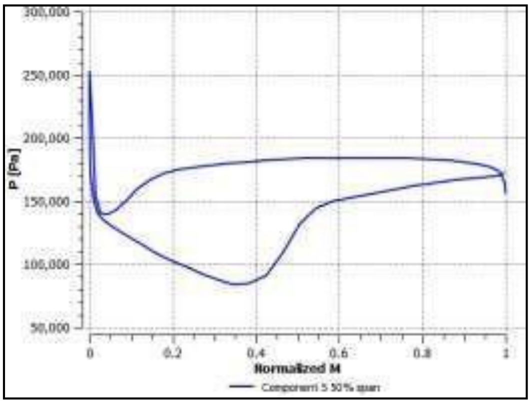


a. Blade loading chart of R2

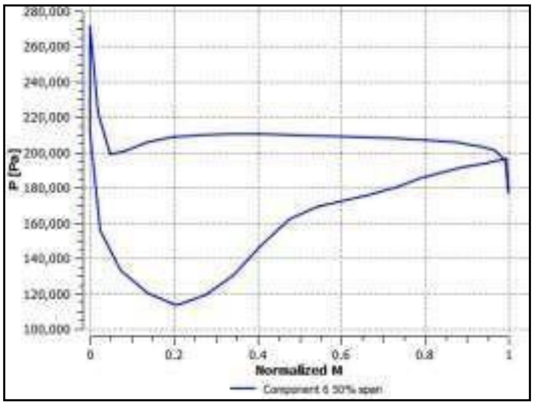


b. Blade loading chart of S2

Figure 9. Blade loading chart of the second stage



a. Blade loading chart of R3



b. Blade loading chart of S3

Figure 10. Blade loading chart of the third stage

After analysis the CFD process result, graphs of the influence of blade tip clearance on the pressure ratio and the performance of the three stage air compressor are drawn in Figuire 11 and Figure 12. These grahps are drawn with blade tip clearance of the first floor rotor blades and case are 0.3, 0.5, 0.7 and 1.0 mm.

In Figure 11 and 12, the horizontal axis is inlet mass flow rate Q (kg/s) of the three stage axial compressor. With Figure 11 the vertical axis is total pressure ratio of the compressor while in Figure 12, the vertical axis is the performance of the compressor.

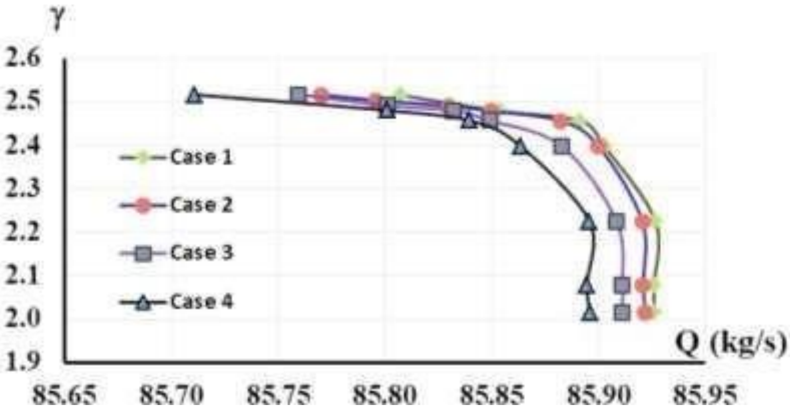


Figure 11. Graph of influence of blade tip clearance on the total pressure ratio

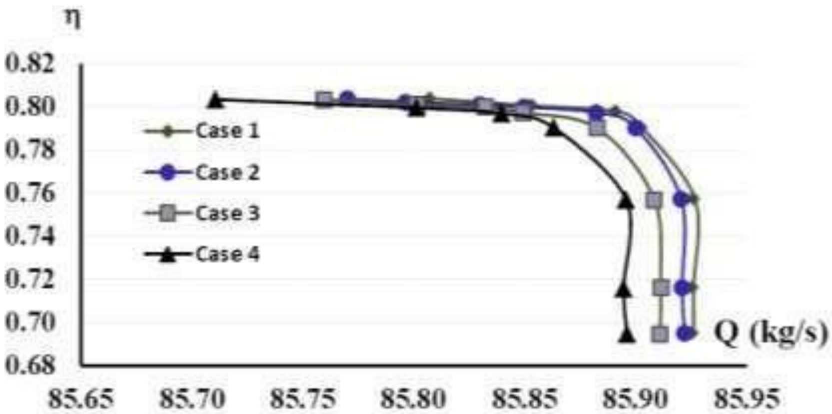


Figure 12. Graph of influence of blade tip clearance on the total pressure ratio

CONCLUSIONS

In this study, the authors used CFD method to evaluate the influence of the blade tip clearance of the rotor blade and case on pressure ratio and performance of a multistage axial compressor. It was a three stage axial air compressor. By using CFD research method, visual images of the velocity contours, pressure contours and vortex field of the air flow surrounding the rotor and stator blades, shaft and case of the axial air compressor could be obtained. These helps engineers to estimate the overall aerodynamic characteristics of the air flow in the compressor that may give the more suitable designs for the geometric dimensions of the blades of each floor in multistage compressors. Mutilstage air compressors are an important part of turbine engines, a powerful engine in the oil and gas, marine and aeronautical industries. For marine industry, turbine engines are used in specialized ships, especially in military ships, submarines, torpedoes and unmanned underwater vehicles.

Result from the study shows that the smaller value of the tip clearance of the rotor blades and case, the greater of the pressure ratio generated and the higher the performance of the air compressor. However, in this study, the authors have not evaluated the vibration effect to the rotor tip baldes of the compressor when the tip clearance of the rotor blade and case is changed. The vibration could be considered as a factor that needs to be evaluated in designing and manufacturing air compressors. This factor may affects to the choice of materials and technology for air compressor manufacturing to avoid the impact of the tip of the rotor blades and the case while the compressor operates due to blade vibration in a small tip clearance.

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