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Study on the Determination of Axial Force During Turbocharger Operation

Sol Song Pak,
Faculty of Dynamics, **Kim Il Sung** University, Taesong District, Pyongyang, DPR Korea

Abstract - The working stability of turbochargers supported by floating ring bearings depends on not only the structure of the rotors but also working characteristic of floating ring bearings.

These working characteristics are related to working conditions and design parameters of floating ring bearings.

The main goal of this research is to investigate the working characteristics of bearings by design parameters of floating ring bearings.

In order to study the working characteristics of bearings, turbocharger supported in FRBs are modeled as multi-body systems and the inner characteristics of FLBs are described by the Reynold's equation.

Numerical analysis about the interaction of the rotor-FRB system is taken by using time integral method and calculus of finite differences.

Analysis of the influence of design parameter is taken in the main working field.

Through this analysis, we can know that the tolerance ratio is the main factor in the floating ring bearing any other than the diameter ratio and length ratio and more convenient to regulate.

It is more helpful to improve the dynamical response of turbocharger that we analysis the influence of design parameters of FRBs on the working characteristics of bearings.

Improving the performance of piston motors as the main power supply is a very important issue in the enhancement and modernization of armament.

The turbocharger rotor is supported by a bearing system containing two radial and axial bearings.

The optimal design of axial bearings with floating bearings is very important. Proper design of axial bearings is essential to balance the forces in the turbine and compressor and to ensure normal operation of the supercharger. However, in the past, the axial bearings are designed using empirical values without considering axial forces, and the design of axial bearings is done by using them when they are not worn during the work, and redesigning them when friction occurs, but the axial forces are determined and the appropriate bearing performance and structural reliability are determined by the critical critical results of the design of the system for the design of the axial force and the design of the axial force is therefore critical for the analysis of the system.

***Keywords*: rotordynamics, turbocharger, journal bearing, floating ring bearing, axial force**

1. INTRODUCTION

Floating ring bearings(FRBs) are the most common type used for commercial turbocharger to minimize cost and power loss.

It is important to investigate the dynamics of high-speed turbocharger to control its vibration and guarantee safe operation.

It is very important to investigate the dynamics of bearings in improving the vibration characteristics of turbochargers and guaranting the stable working conditions.

Turbocharger supported in FRBs often operates at high speed under high temperature condition.

The stability of its rotor system is governed by not only the structure of the rotor but also by the nonlinear hydrodynamic force of two oil films.

The coupling of excessive oil self-excited vibration and unbalance forced response will generate the turbocharger dissonant noise, seriously result in turbine or the compressor impeller wear and then cause the rotor-bearing system failure, thus reduce the operation efficiency and the life of turbocharger.

The modeling of turbochargers consists of two main stages. The equations of motion of a standalone rotor (shaft with compressor and turbine wheel) should be created on one hand and the problem of forces transmitted by floating ring bearings on the other hand.

Lumped parameter models [1, 2, 3] and nowadays more finite element models [4, 5, 6] are employed for the rotor modeling.

Considering the rotor gyroscopic moments and shaft flexibility, LI, et al[5], presented a lumped parameter model to numerically investigate the self-excited response of a high-speed turbocharger rotor-FRB system and the nonlinear characteristics

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of limit cycles was obtained, which provided new theory standards for the design of the FRBs. Generally floating ring bearings have two strong bearing clearance, geometric ratios, that is, inner and outer length, external clearance, inner and outer diameter of FRBs.[4]

These six amounts are affected to the dynamic characteristics of rotor-floating ring bearing system.

The influence of the outer tolerance was investigated in [6] and [7] presented the method to decide the tolerance of FRBs on the basis of the measured response.

SCHWEIZER, et al[8], studied the influence of different operating conditions of oil supply pressure, oil supply temperature and rotor unbalance on the rotor oscillations and the system bifurcations.

[9]Investigated the influence of the unbalanced vibration on different working conditions in the supply pressure and temperature of oil.

[10]DAKEL, et al[10], solved rotor dynamic equations of motion by using the implicit Newmark-Beta time-step integration scheme.

KIRK, et al[11], predicted the stability and transient dynamics phenomenon of the rotor system for turbocharger, which found that the oil whirl frequency spectrum are similar to the experimental results.

[11]shows the relation between the rotating velocity and minimum tolerance of bearings.

TIAN, et al[12], discussed the effect of bearing outer clearance on the rotor dynamic characteristics by using the run-up and run-down simulation method.

SHI, et al[13], studied the effects of clearance ratio of the inner to outer oil film in FRBs on turbocharger rotor instability.

The results show that when clearance ratio is small, the inner oil whirl dominates the instability regions of the rotor system.

ANDRÉS, et al[14,15], showed further progress on the validation against measurements of linear and nonlinear rotordynamic models to predict shaft motions in the rotor-FRB system of an automotive turbochargers.

ZHAO, et al[16], presented the FE model of rotor-bearing system by using the software DyRoBeS and analyzed dynamical characteristics in different working conditions of bearing such as span, width, gap and so on.

The results show that influence of parameters on the system dynamic characteristics is different.

As you can see in real tests and investigations, the influences of inner and outer tolerance of FRBs are big deal.

The variety of inner and outer tolerance is very important in working characteristics of FRBs and it may spoil the inherent characters of FRBs.

The main parameters are inner and outer length, inner and outer tolerance and inner and outer diameter in the FRBs.

But investigations about the working characteristics of FRBs are not so much.

This paper analyzed the influence of main design parameters on working characteristics of FRBs and found the good design parameters and its field.

This investigation is helpful to control the combination of FRB tolerance clearance for high-speed turbocharger.

1. Calculation of the axial force of the rotor

To design the axial bearing used in automotive superchargers, the axial force must be determined first.

The axial force is caused by the different pressures acting on the compressor and turbine wheels, as well as the impact force produced by the flow inside the wheels in the axial direction.

Since the turbocharger changes the compressor and turbine operating conditions with the engine rotational speed and combustion process, the axial force depends on the rotational speed and its direction of action will change rather than act in one direction.

Usually, there are two methods for determining the axial force, one using CFD (CFD) and the other using Newton’s second law.

The first method gives accurate calculation results, but requires extensive computational effort, computational time and post processing of the calculation results in all working conditions of the supercharger, including the mesh generation of the whole supercharger.

Furthermore, it is laborious to use CFD methods under conditions encouraging conventional furnace simulations including blades and volutes to ensure accuracy of calculations, and it is time consuming to calculate the axial force under various conditions.

Conversely, applying Newton's second law to determine the axial force is quite straightforward.

The difference between them is less than 10% when the two methods are applied.

Also, using Newton’s second law requires much less computational time to calculate for all working conditions.

International Journal of Engineering Research & Technology (IJERT)
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Comparing the results between the two methods, it can be seen that the difference is less than the stable tolerance of the axial force in the bearing design.

Hence, Newton's second law is used to calculate the axial force.

1) Basic assumptions

- The gas in the compressor and turbine is ideal.
- The rotor is structurally symmetric.

2)Mathematical model

Because of the symmetry of the rotor, the resultant force of the rotor is the force acting on the axial x, called the axial force $F_{T,ax}$, as shown in Fig. 1.

Here, the turbine direction in the compressor is considered to be positive with x-axis.

The axial force $F_{T,ax}$ acting on the rotor is generated from all the forces of the compressor and turbine wheels shown in Fig. 1.

The force $F_{1,C}$ of the compressor wheel (left) is the pressure force at the compressor inlet surface, $F_{2,C}$ is the pressure force at the end surface, $F_{3,C}$ is the impact force of the compressor wheel, and $F_{4,C}$ is the pressure force at the compressor rear.

Similarly, the forces acting on the turbine wheel (right) are $F_{1,T}$, $F_{2,T}$, $F_{3,T}$ and $F_{4,T}$.

Therefore it can be written like this.

$$F_{T,ax} = F_{1,C} + F_{2,C} + F_{3,C} + F_{4,C} + F_{1,T} + F_{2,T} + F_{3,T} + F_{4,T} \quad (1)$$

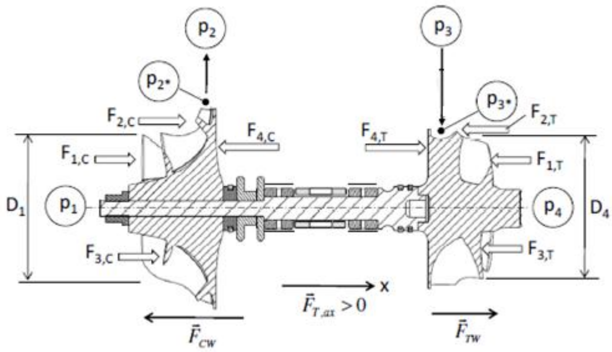

Fig. 1. Force acting on the rotor of the supercharger.

The pressure force is calculated as

$$F_{1,C} = A_1 p_1 = \frac{\pi D_1^2}{4} p_1 \quad (2)$$

Here, D_1-diameter of inlet side of compressor, p_1-inlet pressure

The pressure force r is calculated using the average pressure of the inlet and outlet pressures of the compressor wheel.

$$F_{2,C} = A_{sc} p_m = A_{sc}\left(\frac{p_1 + p_{2^*}}{2}\right) \quad (3)$$

$$A_{sc} = \frac{\pi(\mathrm{D}_2^2 - \mathrm{D}_1^2)}{4} - z \times b_m \times t \quad (4)$$

where b_m is calculated using the height of the inlet blade b_1 and the height of the outlet blade b_2.

$$b_m = \frac{b_1 + b_2}{2} \quad (5)$$

Generally, the pressure and temperature mass flow rates of the compressor inlet and outlet are calculated by computer using the computer-aided software turbomachinery processing.

However, the pressure value between the compressor wheel exit and diffuser is unknown.

Due to the very narrow geometry between the compressor wheel and its diffuser, it is very difficult to calculate the pressures presented above because of the disadvantages of the measurements.

Therefore, it is evaluated using the degree of reaction of the compressor.

The degree of compressor reaction is defined as the ratio of the enthalpy increase of the compressor stage to the enthalpy increase of the compressor wheel.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
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$$r_C = \frac{\Delta h_C}{\Delta h_{st}} = \frac{1-\left(\dfrac{p_{2^*}}{p_1}\right)^{\frac{k_a-1}{k_a}}}{1-\left(\dfrac{p_2}{p_1}\right)^{\frac{k_a-1}{k_a}}} \qquad (6)$$

Where

k_a - isentropic index of filling air, p_1 -compressor wheel inlet pressure, p_2 - diffuser outlet pressure

Solving the above equation, the pressure is given by

$$p_2^* = p_1\left[1+r_C\left(\left(\frac{p_2}{p_1}\right)^{\frac{k_a-1}{k_a}}-1\right)\right]^{\frac{k_a}{k_a\text{-}1}} \qquad (7)$$

Generally, the compressor reactivity is between 55% and 60% for all operating conditions.

Hence, we can calculate the degree of reaction of 0.55 and p_2^*.

Based on the inlet pressure p_1 and the outlet pressure p_2^*, the average pressure p_m of the compressor wheel is calculated as follows :

$$p_m = \left(\frac{p_1 + p_2^*}{2}\right) \qquad (8)$$

Using the average pressure calculated above, the pressure force $F_{2,C}$ is calculated.

The impact force $F_{3,C}$ is calculated using the momentum theorem and the ideal gas equation.

$$F_{3,C} = \dot{m}_C c_{m,1} = \dot{m}_C\left(\frac{\dot{m}_C}{\rho_1 A_{in}}\right) = \frac{\dot{m}_C{}^2 R_a T_1}{p_1 A_{in}} \qquad (9)$$

Where

$\dot{m}_C$ – air mass flow rate through compressor wheel

$c_{m,1}$ – meridional component of air velocity at compressor inlet

R_a – gas constant of air

T_1 – inlet air temperature

p_1 – inlet pressure of air

A_{in} – cross-sectional area at the inlet of compressor wheel

$$\dot{m}_C = \rho_{입c} \times A_1\left(\frac{\pi \times D_1 \times N}{60}\right) \qquad (10)$$

Here, N - number of revolutions per minute of compressor wheel.

The air pressure at the rear of the compressor wheel remains almost unchanged when the clearance between the bearing housing and the rear of the compressor wheel is about 1 mm, according to the CFD results.

Therefore, the pressure force $F_{4,C}$ at the rear side is calculated with the pressure force p_2^* as follows :

$$F_{4,C} = A_{bf,c}\, p_2^* \qquad (11)$$

Here

$A_{bf,c}$ – surface area of the rear face of compressor wheel

p_2^*-Output pressure in compressor wheel

The resultant force of the compressor wheel is calculated from Eq. (2) to Eq. (11) as follows :

$$F_{CW} = F_{1,C} + F_{2,C} + F_{3,C} - F_{4,C} \qquad (12)$$

International Journal of Engineering Research & Technology (IJERT)
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The turbine wheel $F_{1,T}$ is calculated from the turbine exit diameter as the pressure force at the turbine exit surface.

$$F_{1,T} = A_4 p_3^* = \frac{\pi D_4^{\ 2}}{4} p_3^* \tag{13}$$

Here

D_4 – diameter of exit surface of turbine

p_3^* – outlet pressure

The inlet and outlet pressures, temperatures and mass flow rates of the turbine are calculated by the operation of the supercharger engine.

However, the inlet pressure y of the turbine wheel is unknown.

It is very difficult to measure them.

Therefore, they are evaluated using the reaction rate of the turbine.

The reaction rate z of the turbine is also defined as the ratio of the increase in the enthalpy of the turbine stage to the decrease in the enthalpy of the turbine wheel.

The reaction rate of the turbine is in the range of 20% to 90%.

$$r_T = \frac{\Delta h_T}{\Delta h_{st}} = \frac{1-\left(\dfrac{p_4}{p_{3^*}}\right)^{\frac{k_g-1}{k_g}}}{1-\left(\dfrac{p_4}{p_3}\right)^{\frac{k_g-1}{k_g}}} \tag{14}$$

k_g – isentropic index of exhaust gas

p_3 – pre-turbine gas pressure

p_3^* – inlet pressure of turbine wheel

Hence, we find that

$$p_{3^*} = p_4\left[1+r_T\left(\left(\frac{p_3}{p_4}\right)^{-\frac{k_g-1}{k_g}}-1\right)\right]^{\frac{-k_g}{k_g-1}} \tag{15}$$

2. Computational examples and results analysis

The calculation conditions for calculating the axial force of a 1000 hp turbocharger are as follows :

Table 1 Computational conditions

Table 1. Computational conditions

	Compressor	Turbine
Inlet pressure, Pa	101325	325000
Outlet pressure, Pa	291874	130000
Inlet temperature, K	293	885
Inlet diameter, mm	156	119.5
Outlet diameter, mm	107	149
Inlet patch diameter, mm	30	30
Density, Kgm^{-3}	1.173	0.687

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Revolutions, rpm	55000	55000
Gas constant, $kmol \cdot K$	286.7	287.058
Enthalpy	0.5	0.5
isentropic index	1.4	1.34
Outlet blade height, mm	11.8	37.5
Inlet blade height , mm	37.5	26.2
Thickness, mm	0.7	2.1
Number of blades	12	12

Using the above-mentioned equations with the data given in Table 1, the axial force is obtained as follows :

Table 2. Force acting on compressor and turbine

	p_{2^*} ,Pa p_{3^*}	$F_{1,C}$,N $F_{1,T}$	$F_{2,C}$,N $F_{2,T}$	$F_{3,C}$,N $F_{3,T}$	$F_{4,C}$,N $F_{4,T}$	$F_{C,W}$,N $F_{T,W}$
Compressor	212163	1906.80	1553.2	3162.4	3903.2	2719.2
Turbine	227488	2571.52	957.96	2871.98	3803.88	-2597.57

Find the axial force as follows :

$$F_{T,ax} = F_{CW} + F_{TW} = 121.64\text{N}$$

Where "+" indicates that the axial force acts on the compressor toward the turbine, and if the calculated value is "-", the axial force acts on the turbine toward the compressor.
With the data shown in Table 1, the simulation results of compressor and turbine at 55000 rpm of 1000 hp supercharger are as follows.

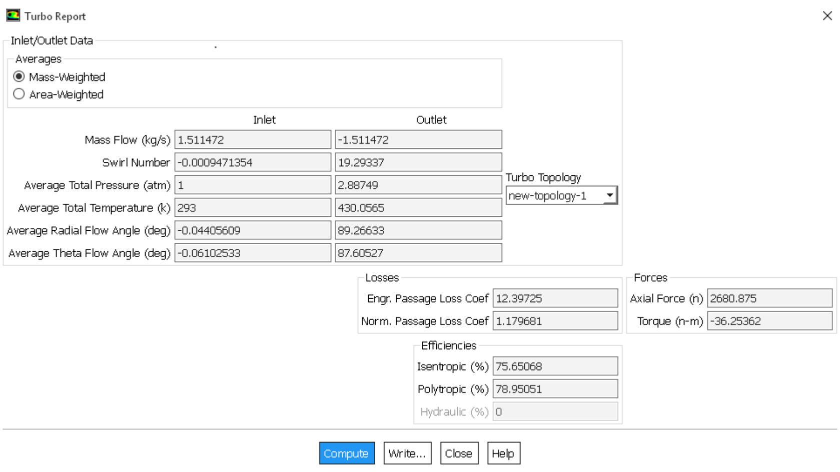

Fig. 2. Results of compressor analysis of a 1000 hp turbocharger.

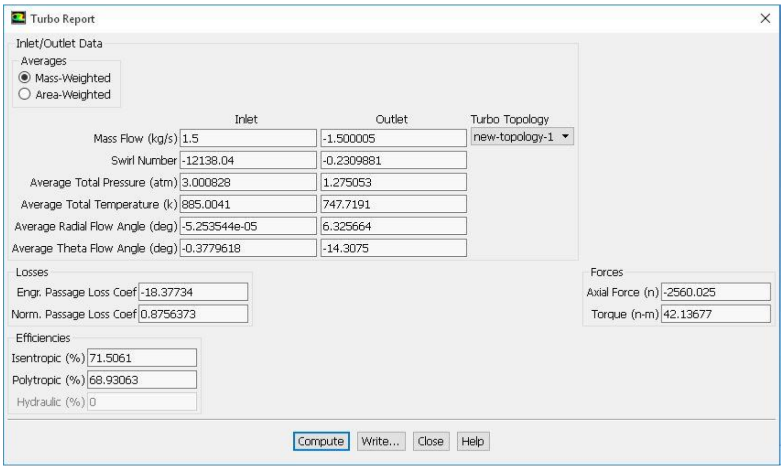

Fig. 3. Results of turbine analysis of a 1000 hp turbocharger.

Table 3. Simulation results

	Compressor,N	Turbine,N	Discharger
Axial force	2680	-2560	4.03

The error between the simulated results and the axial force calculated using the laws of dynamics is like this.

Table 3. Results analysis

Axial force ,N	Compressor ,N	Turbine ,N	Rotor ,N
1 dimensional results	2719.2	-2597.57	121.63
Simulation results	2680	-2560	120
Error	1.14%	1.44%	1.34%

5. CONCLUSION

This paper has reduced the computational time and computational efficiency by establishing an axial force calculation system for high speed rotary machines.

This has improved the reliability of the choice and use of axial bearings that are necessarily used in engines.

The supercharger must design an axial bearing that can withstand the force by calculating the axial force at that speed, in order to be able to work safely in a rotating regime.

The initial parameters of the supercharger must be taken into account in calculating the axial force, which thoroughly reflects the gas state given by the geometry of the supercharger and the operating characteristics of the engine, from the system calculations.

This is because the turbine and compressor are organically coupled to each other.

Thus, the turbine is driven by the total gas of the turbine, which has a lot of energy from high pressure and temperature, and hence the compressor is driven to compress the air.

This compressed air causes combustion and determines the turbine voltage and temperature throughout this combustion process. Therefore, the engine performance must be taken into account. Using the computational system established in the paper, the axial force at various rotational speeds of any supercharger can be easily calculated.

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