ISSN (e): 2250 – 3005 || Volume, 09 || Issue, 05|| May – 2019 ||
International Journal of Computational Engineering Research (IJCER)

Axial Thrust Load Calculation for Turbocharger

C R KUMAR[1], BIDYUT PRAVA JENA[2], RABI SARANGI[3]
[1,3]Assistant Professor, Department of Mechanical Engineering, Gandhi Institute For Technology (GIFT), Bhubaneswar
[2] Assistant Professor, Department of Mechanical Engineering, Gandhi Engineering College, Bhubaneswar

Abstract: *- The axial loads generated by the rotating component of turbomachinary like turbocharger and turboexpander is calculated. Previously it was difficult and costly to generate spiral grooves but now it easily be developed with laser machining process. Current research target to calculate the axial thrust load generated due to rotation of turbomachinary. Later this calculated load can be used to design and develop an alternative aerodynamic grooved thrust bearing with spiral pattern to find pressure profile, load carrying capacity and friction coefficient etc.*
Keywords: - Turbocharger, Thrust Bearing.

I. INTRODUCTION:

Rotor-bearing system is an essential part in most of the micro turbo machines whose stability of functioning increases the efficiency of the machine. Therefore, it is very much important to discuss the stability of journal bearings. Plain journal bearings are inexpensive and easy in manufacturing but, it suffers with poor stability due to its low eccentricity.

In order to improve the stability the journal was pre-loaded by designing a geometrical shape in it such as dams, pockets, steps etc. In the as such design of plain journal bearings the lubricants present inside the geometrical shape creates some additional loads before running of the rotor and hence these additional loads are able to increase the stability of the rotor-bearing systems.

The spiral groove bearing performance depends on groove depth, groove width, groove angle, and grooves number on bearing surface. There are several types of spiral groove thrust bearings. However, we will explain only three important ones such as: -

I. Symmetrical spiral groove bearing.
II. Asymmetrical spiral groove bearing and
III. Partially grooved bearing.

II. LITERATURE REVIEW

It was Wipple [1] who successfully described the working of a spiral groove bearing by applying basic hydrodynamics theory. He however, considered the assumptions similar to that of analysing the smooth bearings like plain journal bearings.

Again it was Muijderman [2] who also successfully tried to solve the problems related to the lubricated spiral groove bearing. He assumed different initial parameters to analyse the bearing's load carrying capacity as well as power losses due to frictions. He had also reported the effect of groove parameters on the working performance of the bearing

However, it was Vohr and Chow [3] and separately by Hirs [4] who successfully forwarded a theoretical model to apply the Reynolds equationto spiral groove bearing for the first time and derived the required solutions for such bearings. They proposed that actual pressure profile is not needed always to find the load carrying capacity and stability of the spiral grooved bearing, instead, it is sufficient to identify the mean pressure across the width of a groove and land pair. According to them this proposed pressure profile is known as a smoothed pressure profile, is applicable only when there is infinite number of grooves either on the surface of the bearing or on the surface of the journal. But in real life it is not possible to generate infinite number of grooves either on the surface of a journal or on the surface of a bearing. Though, the proposed theoretical analysis is quite accurate when it is applied to bearings havingfinite number of grooves.

It was Hirs and Bootsma [5] who verified this proposed theory experimentally and opened the door for further study in this direction.

In another approach of finite difference method, Smally [6] solved the differential equation for the smoothed pressure profile. They assumed that the lubricant viscosity remains constant across the film. Basing pon this assumption they found results which were very close to the experimental one. On the other hand, Reinhoudt [7] used the method of finite element analysis to solve the problems related to the helical grooved bearings. In an extensive approach it was Yavelo [8] who tried to increase the load carrying capacity of a spiral groove bearing by optimizing several affecting parameters of the grooves.

It was Cunning and Fleming [9] who carried out an experimental stability analysis of the gas lubricated herringbone bearing. They were successfully attained a speed of 60000rpm without the application of any radial load with perfect stability. In addition they also found that the stability can be improved by reducing the bearing clearance.

Malanoski [10] also reported of attaining a speed of 60000 rpm without instability in an ultra-stable gas bearing of 40mm diameter and loaded eccentricity of 0.3.

Again, Molyneaux [11] also reported a stainless steel rotor running at a speed of 350000 rpm used in a 9mm spiral groove journal bearing which is suitably applied in industries. It is also reported that for the working of an expansion turbine helical groove journal bearing is used to support a rotor.

It is observed that in an air lubricated spiral groove bearings the load carrying capacity are too small for most engineering applications. Therefore, it is very much important to increase the load carrying capacity of the bearing for which a high viscous substance should substitute air as a lubricant in the bearings. Grease as a lubricant can be a suitable alternative for air where a self-sealing action for the bearing is required. Dewar [12] and Muijdeman [13] have explained theoretically as well as experimentally the application of Grease as lubricant in spiral grooved bearing.

III. THRUST-LOAD CALCULATION OF THE ROTOR

To design the thrust bearing used for the high speed turbomachinery, the thrust load calculation must be resolute. Thrust load is generated by different pressure acting upon the compressor and turbine wheels as well as the impulsive force generated due to the flow in axial direction. Since the turbomachinery operates at various speeds therefore the thrust load depends on the speed of the rotor.

There are generally two ways to calculate the thrust load on the rotor: either using the CFD (computational fluid dynamics) or by simply applying the Newton's second law. The CFD method gives the precise result but needs huge computational exertion at all stages of turbomachinery, while using the Newton's second law is quite simple but needed some thermodynamics and turbomachinery knowledge. Though its analytical result is quite good as compared to the numerical result obtained by CFD. The result difference between both the methods is generally lower than the safety tolerance of the thrust load taken in the bearing design. Therefore the Newton's second law is applied to the control volume for the thrust load calculation on the rotor.

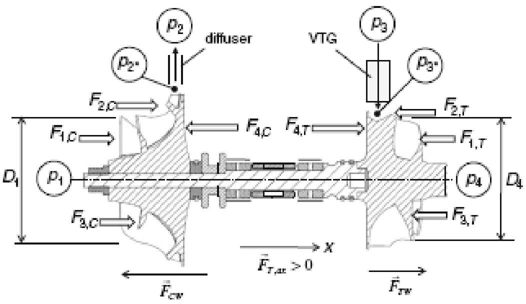

Figure 3.1: Forces acting on Turbocharger [14]

As the rotor is symmetric, the resultant force on the rotor acts along the axial direction (i.e., along x-direction), here it is represented as $F_{T,ax}$(Fig.3.1).

The $F_{T,ax}$is the resultant thrust load acting on the rotor, is the resultant of all forces acting on the compressor as well as turbine wheel as indicated in the Fig.3.1. Different forces acting on the compressor wheel (CW), which is in the left side of the diagram, $F_{1,C}$is the pressure force acting on the inlet surface; $F_{2,C}$ is the pressure force at the shroud surface; $F_{3,C}$ is the impulsive force acting on the CW and $F_{4,C}$ is the pressure force at the back face of CW. Similarly, different forces acting on the turbine wheel (TW, in left portion of the Fig.3.1) can be represented as $F_{1,T}$, $F_{2,T}$, $F_{3,T}$ and $F_{4,T}$.

From reference [14] for D_1 =15mm and p_1 =1.01x10^5 Pa pressure force $F_{1,C}$ can be calculated by using the equation

$$F_{1,c} = A_1 p_1$$

$$= \frac{\pi {D_1}^2}{4} p_1 = 17.848\ N --- (3.1)$$

Where D_1 is the diameter of compressor at inlet p_1 is the atmospheric air inlet pressure.

From the reference [30] for given value of blade height at inlet $b_1 = 3mm$ and at outlet $b_2 = 0.657mm$ one can calculate the mean blade height as

$$b_m = \frac{b_1 + b_2}{2} = 1.8285\ mm$$

Therefore the projected area of the shroud surface in the axial direction (here along X-direction) $A_{S,C}$ for given value of the outlet diameter of the compressor $D_2 = 33.7mm$, number of blades $z = 12$, and thickness of the blade $t = 0.75mm$ from the reference [14], can be calculated as

$$A_{s,c} = \frac{\pi(D_2^2 - D_1^2)}{4} - z \times b_m \times t$$
$$= 6.9878 \times 10^{-4} m^2 - - - (3.2)$$

The inlet and outlet pressure, temperature and fluid mass flow rate of the compressor cab be computed by using the turbomachinary processing. However, the value of the pressure p_2^* between the CW outlet and the diffuser are unknown. It is very difficult to calculate the above said pressures by means of measurement due to very narrow geometries of the gap between the wheel and their housings. Therefore, it is estimated in terms of reaction degrees of the compressor.

The reaction degree r_C of the compressor is defined as the ratio of the enthalpy increase in the CW to the enthalpy increase in the compressor stage. The value of the reaction degree of the compressor is normally between 55 to 60% for all operating conditions and it can be expressed as

$$r_c = \frac{\Delta h_c}{\Delta h_{st}} = \frac{1 - \left(\frac{p_2^*}{p_1}\right)^{\frac{k_a - 1}{k_a}}}{1 - \left(\frac{p_2}{p_1}\right)^{\frac{k_a - 1}{k_a}}} - - - -(3.3)$$

Where

k_a is the isentropic exponent of the air for compressor (here $k_a = 1.4$ from reference [14]);

p_1 is the atmospheric air inlet pressure to the CW

p_2 is the outlet pressure of the diffuser (here $p_2 = 4.45 \times 10^5 Pa$ from reference [14]);

The value of r_C from reference [14] is taken as $r_C = 0.55$. By solving the above equation the p_2^* is given by

$$p_2^* = p_1 \left[1 + r_c \left(\left(\frac{p_2}{p_1}\right)^{\frac{k_a - 1}{k_a}} - 1\right)\right]^{\frac{k_a}{k_a - 1}}$$
$$= 2.9344 \times 10^5 Pa - - - -(3.4)$$

The mean pressure of the compressor wheel on the basis of inlet pressure p_1 and the outlet pressure $p_2^* = 2.9344 \times 10^5 Pa$ can be calculated as

$$p_m = \left(\frac{p_1 + p_2^*}{2}\right) = 1.9722 \times 10^5 Pa$$

By using the above calculated value of the mean pressure, we can calculate $F_{2,C}$ as follows

$$F_{2,c} = A_{s,c} p_m = A_{s,c} \left(\frac{p_1 + p_2^*}{2}\right)$$
$$= 137.8161\ N - - - -(3.4)$$

Where

$A_{S,C}$ is the projected area of the shroud surface in the axial direction (here along x-direction);

p_1 is the CW inlet pressure;

p_2^* is the CW outlet pressure.

For the inlet pressure p_1 , inlet temperature $T_1 = 300K$ and inlet density $\rho_{in,c} = 1.176kgm^{-3}$ the mass flow rate $\dot{m}_c$ can be calculated as

$$\dot{m}_c = \rho_{in,c} \times A_1 \left(\frac{\pi \times D_1 \times N}{60}\right)$$

$$= 0.0163 \ kg/s$$

Where

N is the RPM of the CW.

For the given value of the RPM of CW, say N =100000 rpm.

The impulsive force $F_{3,C}$ can be calculated by using the momentum theorem and the perfect gas equation as follows

$$F_{3,c} = \dot{m}_c c_{m,1} = \dot{m}_c \left(\frac{\dot{m}_c}{\rho_1 A_1}\right) = \frac{\dot{m}_c^2 R_a T_1}{p_1 A_1}$$
$$= 1.2854 \ N ----(3.5)$$

Where

$\dot{m}_c$ is the mas flow rate of air through CW;

$C_{m,1}$ is the meridional component of the air velocity at compressor inlet;

R_a is the air characteristic gas constant;

T_1 is inlet temperature of air;

p_1 is the CW inlet pressure;

And A_{in} is cross-sectional area of the CW at the inlet.

According to the CFD result if the gap between bearing bushing and the back face of CW is more than 1mm then the pressure at the back face of CW is nearly remain unchanged. Hence, the pressure force $F_{4,C}$ corresponding to the pressure p_2^* can be calculated as

$$F_{4,C} = A_{bf,C} \, p_2^* = 90.0889N$$

Where

$A_{bf,C}$ is the CW back face surface area;

p_2^* is the CW outlet pressure.

The resulting force acting on the CW can be calculated as

$$F_{cw} = F_{1,c} + F_{2,c} + F_{3,c} - F_{4,c}$$
$$= 66.8608 \ N ----(3.6)$$

The inlet and outlet pressure, temperature and mass flow rate of the turbine cab be calculated by using the turbomachinary processing. Though, the value of the pressure p_3^* between the nozzle outlet and the vane less space of the turbine are unknown. It is very difficult to calculate the above said pressures by means of measurement due to very narrow geometries of the gap between the wheel and their housings. Hence, it is estimated in terms of reaction degrees of the turbine.

The reaction degree r_T of the turbine is defined as the ratio of the enthalpy decrease in the TW to the enthalpy decrease in the expansion stage. Where the reaction degree of the turbine is varied from 20% to 90% depending on the position of the west gate and it can be expressed as

$$r_c = \frac{\Delta h_T}{\Delta h_{st}} = \frac{1 - \left(\frac{p_4}{p_3^*}\right)^{\frac{k_g - 1}{k_g}}}{1 - \left(\frac{p_4}{p_3}\right)^{\frac{k_g - 1}{k_g}}} ----(3.7)$$

Where

k_g is the isentropic exponent of exhaust gas (from reference [14] $k_g = 1.34$)

p_3 is the pressure of air in the vane less space of the turbine

Here, the value of p_3 from reference [14] is $p_3 = 4.319 \times 10^5 \, Pa$

p_4 is the pressure at the outlet of the turbine (from reference [14] $p_4 = 1.02 \times 10^5 \, Pa$)

The above expression can be solved to calculate pressure p_3^* which is the inlet pressure of TW as

$$p_3^* = p_4\left[1 - r_T\left(\left(\frac{p_3}{p_4}\right)^{-\left(\frac{k_g-1}{k_g}\right)} - 1\right)\right]^{\frac{-k_a}{k_a-1}}$$
$$= 2.6395 \times 10^5 Pa - - - -(3.8)$$

Now, the pressure force acting on the inlet surface of the turbine $F_{1,T}$ can be calculated for the given value of inlet diameter of the turbine from reference [14] $D_3 = 29.6mm$ as

$$F_{1,T} = A_3 p_3^* = \frac{\pi D_3^2}{4} p_3^*$$
$$= 181.6305\ N - - - (3.9)$$

From the reference [14] for the given value of blade height at inlet $b_3 = 0.709mm$ and at outlet $b_4 = 4.45mm$ of the TW one can calculate the mean blade height as

$$b_{m,T} = \frac{b_3 + b_4}{2} = 2.5795\ mm$$
$$= 2.5795 \times 10^{-3} m$$

Therefore the projected area in the axial direction (here along x-direction) of the shroud surface $A_{S,T}$ for given value of the tip and hub diameter of the turbine $D_{tip} = 17.8mm = 17.8 \times 10^{-3} m$, $D_{hub} = 8.9mm = 8.9 \times 10^{-3} m$ number of blades $z = 10$, and thickness of the blade $t = 0.6mm = 0.6 \times 10^{-3} m$ from the reference [14], can be calculated as

$$A_{S,T} = \frac{\pi\left(D_{tip}^2 - D_{hub}^2\right)}{4} - z \times b_{m,T} \times t$$
$$= 5.3268 \times 10^{-4} m^2$$

The mean pressure of the turbine wheel on the basis of inlet pressure p_3^* and the outlet pressure p_4 can be calculated as

$$p_{m,T} = \left(\frac{p_3^* + p_4}{2}\right) = 1.9197 \times 10^5 Pa$$

By using the mean pressure $p_{m,T}$, of the inlet and outlet pressure of the TW, the pressure force $F_{2,T}$ can be calculated as

$$F_{2,T} = A_{S,T} p_{m,T} = A_{S,T}\left(\frac{p_3^* + p_4}{2}\right)$$
$$= 102.2606\ N - - - - - (3.10)$$

Where

$A_{S,T}$ is the projected area in the axial direction (here along x-direction) of the shroud surface

p_3^* is the inlet pressure of the TW

p_4 is the outlet pressure of the TW.

For the reference [14] inlet pressure p_3^*, inlet temperature $T_3 = 104.57K$ and inlet density $\rho_{in,T} = 7.175kgm^{-3}$ the mass flow rate $\dot{m}_T$ can be calculated as

$$\dot{m}_T = \rho_{in,T} \times A_3\left(\frac{\pi \times D_3 \times N}{60}\right) = 0.765kgm^{-3}$$

Where

N is the RPM of the TW.

For the given value of the RPM of TW, say N =100000 rpm.

The impulsive force $F_{3,C}$ can be calculated by using the momentum theorem and the perfect gas equation as follows

$$F_{3,T} = \dot{m}_T C_{m,3} = \dot{m}_T \left(\frac{\dot{m}_T}{\rho_{in,T} A_3} \right)$$
$$= \frac{\dot{m}_T^2 R_a T_3}{p_3^* A_3} = 96.7741\ N ----(3.11)$$

Where

$\dot{m}_T$ is the mas flow rate of air through TW;

$C_{m,3}$ is the meridional component of the air velocity at turbine inlet;

R_a is the air characteristic gas constant (i.e., $R_a = 287.058 Jkg^{-1}K^{-1}$);

T_3 is the TW inlet air temperature;

p_3^* is pressure of the TW at inlet;

And A_3 is the TW cross-sectional area at inlet.

According to the CFD result if the gap between bearing bushing and the back face of TW is more than 1mm then the pressure at the back face of TW is nearly remain unchanged. Hence, the pressure force $F_{4,T}$ corresponding to the pressure p_4 can be calculated as

$F_{4,T} = A_{bf,T} p_3 = A_3 p_3 = 181.6305N$ -- (3.12)

Where

$A_{bf,T}$ is the back face surface area of the TW;

p_3 is the inlet pressure of the TW.

The resulting force acting on the TW can be calculated as

$$F_{TW} = -F_{1,T} - F_{2,T} - F_{3,T} + F_{4,T}$$
$$= -199.0347\ N ------(3.13)$$

Hence the resultant force acting on the rotor is

$$F_{T,ax} = F_{CW} + F_{TW}$$
$$= -132.1739\ N ----(3.14)$$

Here –ve sign shows that the direction of the resultant thrust load is from turbine to compressor side.

IV. CONCLUSION

- On the basis of input parameters the axial thrust load is calculated with its direction.
- The calculated axial load can be utilised to design a spiral grooved thrust bearing.

V. FUTURE SCOPE

- Design of thrust bearing is required to sustain the generated thrust load due to rotation of turbocharger at high speed.
- Different profile of thrust bearing can be designed and tested for the above purpose.
- Parameters variation and testing can be done to optimize the thrust bearing performance.

REFERENCES: -

[1]. Whipple, R. T. P. (1949). Herringbone Pattern Thrust Bearing, AERE T/M 29.

[2]. Muijderman, E. A. (1965). New Possibilities for the Solution of Journal Bearing Problems by Mean of Spiral Groove Principle. *Proc. Inst. of Mech. Eng*, *1966*, 75.

[3]. Vohr, J. H., & Chow, C. Y. (1965). Characteristics of herringbone-grooved, gas-lubricated journal bearings. *Journal of Basic Engineering*, *87*(3), 568-576.

[4]. Hirs, G. G. (1965). The load capacity and stability characteristics of hydrodynamic grooved journal bearings. *ASLE transactions*, *8*(3), 296-305.

[5]. Bootsma, J. (1973). The gas liquid interface and the load capacity of helical grooved journal bearings. *Journal of Lubrication Technology*, *95*(1), 94-100.

[6]. Smalley, A. J. (1972). The narrow groove theory of spiral grooved gas bearings: Development and application of a generalized formulation for numerical solution. *Journal of Lubrication Technology*, *94*(1), 86-92.

[7]. Reinhoudt, J. P. (1972). *On the stability of rotor-and-bearing systems and on the calculation of sliding bearings* (Doctoral dissertation, Philips Research Laboratories).

[8]. Yavelov, I. S. (1983). On the Influence of Spiral Groove Depth on the Experimental Characteristic of Radial Hydrodynamic Bearings, TrenicIznos Vol. 4, No. 1, pp. 134-146.

[9]. Cunning, R. E. and Fleming, D. P. (1965) *Experimental Stability Studies of the Herringbone Grooved Gas Lubricated Journal Bearing*, Trans. ASME, Journal of Lubrications Technology.

[10]. Malanski, S. B. (1967). Experiments on an Ultrastable Gas Journal bearing. *Journal of Lubrication Technology*, *89*(4), 433-438.

[11]. Molyneaux, A. K., & Leonhard, M. (1989). The Use of Spiral Groove Gas Bearings in a 350 000 rpm Cryogenic Expander. *Tribology transactions*, *32*(2), 197-204.

[12]. Dewar, D. M. (1974). An Analysis of Grease and Oil Lubricated Spiral Grooved Bearings. *Journal of Lubrication Technology*, *96*(2), 275-283.

[13]. Muijderman, E. A. (1979). Grease-lubricated spiral groove bearings. *TRIBOLOGY international*, *12*(3), 131-137.

[14]. Hung Ping., Numerical Calculation of Lubrication, methods and Programs. Singapore, John Wiley, 2013.