CFD ANALYSIS OF AEROSTATIC BEARING FOR CRYOGENIC TURBO-EXPANDER

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CFD ANALYSIS OF AEROSTATIC BEARING FOR CRYOGENIC TURBO-EXPANDERS

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by

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CERTIFICATE

This is to certify that the thesis entitled **CFD analysis of aerostatic bearing for cryogenic turbo-expander**, submitted by **Ranjit Behera** to National Institute of Technology, Rourkela, is an authentic record of bona fide research work carried under my supervision and I consider it worthy of consideration for the award of the degree of Master's of Technology of the Institute.

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SELF DECLARATION

I, Ranjit Behera, Roll No. 214ME5442, student of M.Tech (2014-2016), Cryogenics and Vacuum Technology of Department of Mechanical Engineering, National Institute of Technology, Rourkela do hereby declare that I have not adopted any kind of unfair means and carried out the research work reported in this thesis work ethically to the best of my knowledge. If adoption of any kind of unfair means is found in this thesis work at a later stage, then appropriate action can be taken against me including withdrawal of this thesis work.

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This project would not be completed without all the help from my lab-mates. The time we spent together other than research work was truly refreshing. Any small technical talks that we had together were surely helpful for improving this thesis.

Date :
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NOMENCLATURE

b	Bearing radius
W	Load carrying capacity
a	Pocket radius
P_d	Discharge pressure at the exit of orifice
V	Velocity of the fluid
t	Time
k	Turbulent kinetic energy
P_a	Ambient pressure
T	Transpose
F	Surface tension

Greek symbols

ε	Turbulent dissipation
μ	Viscosity of fluid
ν	Kinematic viscosity of fluid
ρ	Density of fluid

Abstract

Expansion turbine plays the major role in any cryogenic plants like helium and hydrogen liquefier, air separation plants and low temperature refrigerator. These turbines run at speed around 50,000 to 5,00,000 rpm. At such a high speed no bearing is feasible to support the load except gas bearing. In gas bearing there is no physical contact between stator and rotor takes place so the friction between the stator and rotor is negligible. Because of this the rotor can move smoothly inside the stator. Hence aerostatic bearing used for achieving high accuracy and precession. Due to cleanness of aerostatic bearing it is also used in electronic and food processing industry.

In order to avoid complexities at the time of formulation analytical studies has been carried out. Governing variables are kept limited that helps in dynamic and performance conduct of aerostatic bearing. Finally a three dimensional model of aerostatic bearing was built in Solidworks and a two dimensional cross-section model of aerostatic bearing in ANSYS workbench 15. After doing rectangular meshing to the geometry made in workbench Computational fluid dynamics (CFD) analysis was done in ANSYS Fluent. The pressure and velocity distribution at the bearing clearance and at orifice had shown. It is found that there is rapid change in pressure and velocity at the exit of the orifice. Variation of Load carrying capacity with the change of inlet pressure was studied and compared with numerical value.

Keywords: Load carrying capacity, computational fluid dynamics analysis, helium and hydrogen liquefier

Chapter 1

Introduction

1.1 Use of turbo-expander in cryogenic processes:

Nature has provide us a bulk amount of gaseous raw materials in the form of mixture in atmosphere and also under the crust of the earth. Proper utilisation of these gases is the major challenge for the technologically advanced society. In order to use this gases properly we should have technology for the extraction, storage, and transportation. Liquefaction of the gases is the best way in order to achieve these applications. Low temperature distillation process is the most economical process for the liquefaction of gases. Liquefaction of gases can obtained through different processes. Linde and Heylandt cycles were used in air separation process and liquefaction of hydrogen and helium. Low pressure cycles are used in the cryogenic process plants. In this low pressure cycles an expansion turbine was used for the refrigeration process. The main advantages of turbine based plants are high thermal efficiency, good reliability, and can integrate easily with different system. Due to this reason expansion turbine considered as the heart of a cryogenic refrigeration system. Reciprocating expansion device can be used as a substitute for the expansion turbine. But on the view of good efficiency and reliability of turbine, use of reciprocating expansion devices is restricted for the cryogenic refrigeration process.

1.1.1 Application of cryogenic turbo-expander:

Several applications of cryogenic turbo-expanders are:

- Production of liquid cryogens
- Air conditioning in aeroplane by generating refrigeration
- Separation of hydro carbons and propane in the petrochemical industry.
- Helps in the generation of low temperature required for the recovery of ethane
- Condensation of impurities present in the gas stream

1.2 Anatomy of a cryogenic turbo-expander:

Basically turbo-expander consists of a turbine and a brake compressor fitted with a single shaft, which is supported by thrust and journal bearing. The basic components of a turbo-expander are

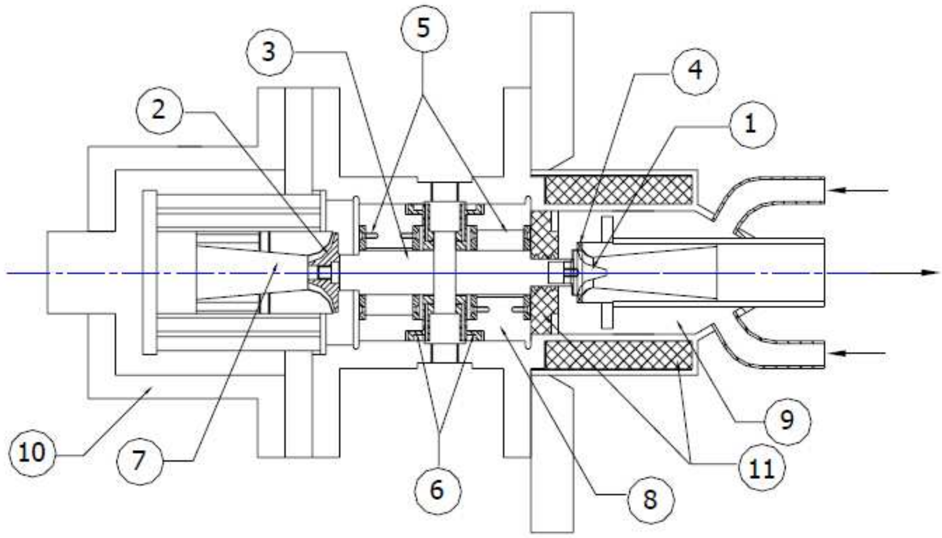


Figure 1.1: Anatomy of cryogenic turbo-expander [4]

1. Turbine wheel
2. Brake compressor
3. Shaft
4. Nozzle
5. Journal bearing
6. Thrust bearing
7. Diffuser
8. Bearing housing
9. Cold end housing
10. Warm end housing
11. Seals

For small and medium sized turbo-expander, rotors are installed vertically because in vertical orientation it is easy for installation and maintenance. It consists of a brake compressor and a turbine wheel fitted to a shaft at both ends.

High pressure gas enters the turbo-expander through the cold end housing than enters into the nozzle. As the gas enters into the nozzle velocity of the gas increases at the expense of decrease in pressure, because of the converging action of the nozzle. In order to avoid sudden change in the direction of flow and energy loss, nozzles and rotor blades are aligned up to some extent.

In the turbine wheel the fluid enters the wheel in radial direction and leaves in axial direction. Some work is produced because of the expansion process happens in the turbine. Due to the expansion process, temperature of the fluid decreases.

Shape of diffuser is diverging in nature. Because of this diverging shape there is gain of pressure energy at the expense of loss of kinetic energy. Therefore expansion ratio is increased by the production of cold.

For the extraction of the work output a loading device is essential. For small and medium turbine the brake compressor is assembled with the turbine in order to extract the work output of the turbine.

For large sized turbine, a electrical generator is fitted for the extraction of the work output.

In case of vertical orientation of the rotor, journal bearing supports only the radial load and thrust bearing support the weight of the component. For the horizontal orientation of the rotor, journal bearing supports both radial load and weight of the component. Thrust bearing supports only the axial load due to the pressure gradient.

1.3 Bearing for cryogenic turbo-expander:

The main challenging part of a cryogenic turbo-expander designer is the bearing part. Mass flow rate of cryogenic turbo-expander is much lower than the power generating turbine. rotors of a cryogenic turbo-expander rotates at speed around 50,000 to 5,00,000 rpm. At such a high speed no conventional bearing is capable for supporting the load, so cryogenic turbo-expander designer has the challenge for efficient bearing design for the turbo-expander. Because of wear and tear due to high friction use of ball bearing is restricted. Due to high rotating speed of the rotor, high temperature is generated between the mating surfaces of the bearing and oil lubricant will vaporise which restricts use of bearing having oil lubricant. So gas bearing is the only alternatives for supporting the load in cryogenic turbo-expander.

1.3.1 Ball and roller bearing:

A ball and roller bearing has three components those are inner ring, outer ring and a rolling element. This rolling element may be cylindrical, spherical, tapered shape. The inner ring is fitted to the rotor. Rotational speed of the ball and rollers is higher than the speed of the rotor, due to this fatigue failure occurs in the bearing surfaces and breakage of grease film occurs due to excessive heat generation. Another problem of ball and roller bearing is damping problem at critical speeds. In order to avoid this type of problem, flexible mounting is needed.

1.3.2 Oil lubricated bearing:

Oil lubricated bearing are still used in large turbo-expanders for air separation plants. There may be chances of mixing of oil lubricant with the working fluid of turbo-expander. Because of this reason proper sealing is required for the oil lubricated bearing, which is a very difficult task for the designer. For large rotors sealing may not be a problem, because large rotors rotate at relatively small speed. For small rotors sealing is the major problem because small rotors have high rotating speed. Due to high viscosity of oil lubricant, power loss at the bearing surface is more. This makes breakage of oil film lubricant. Due to these reasons use of oil lubricated bearing are restricted for the cryogenic turbo-expander.

1.3.3 Gas bearing:

In case of small, high speed rotor gas lubricated bearing is the best option. This is because gases has low viscosity and low boiling point, for which gases are chemically stable for a long range of temperature. By using gas lubricant the device remains clean. Because of low viscosity of gas bearing, friction generated is low which results in less heat generation.

Low viscosity of gas bearing restricts the load carrying capacity of the bearing and damping properties of the bearing. Due to this reason gas bearing are unstable. This instability is due to half speed whirl. Gas bearing needs high mechanical precision so it needs very careful manufacturing. Due to these reasons design of gas bearing is a major challenge for the designer.

1.4 Objective

Already there are a lot of work done in so many journal and research papers, but these have little practical importance due to complexity in the analysis and formulation. Main objective is to develop high performance aerostatic bearing with low cost, making test facilities and to find out general tools for the analysis. In this analysis our main motto is to find the distribution of pressure and velocity at the bearing clearance and at the orifice by using computational fluid dynamics software. We will do our analysis for turbulent flow. We will calculate the load carrying capacity of the aerostatic bearing at different inlet pressure. Effect of eccentricity on the distribution of pressure and velocity also in the bearing clearance also shown.

Chapter 2

Literature Review

Discovery of new technology put demand on support system for different machine component. In order to face the challenge new methods are developing on the basis of combination of conventional designs or by approaching new techniques. Growing use of cryogenic liquefaction and low temperature refrigeration put demand on efficient turbo-expander. Efficient turbo-expander needs efficient bearing. For this purpose gas bearing achieves wide acceptance. In gas bearing contamination of bearing gas with the gas in turbo-expander is a major problem. To avoid this contamination no sealing is efficient, economic. So it is advised to use the same gas itself for the bearing where contamination of gases strictly not acceptable. Gas bearing divided into two categories (1) Aerostatic bearing which need pressurised gas for their operation (2) Aerodynamic bearing which create their own pressure required for the operation. In this chapter we will explore aerostatic gas bearing on reference for cryogenic turbo-expander.

2.1 Comparison of aerostatic and aerodynamic bearing:

In aerostatic bearing to produce the required pressure a compressor is required, but in aerodynamic bearing no such compressor required, the required pressure created by squeezing and shearing of gas within the bearing surfaces. Both aerostatic and aerodynamic bearing can axial, radial or both type of loads. A bearing can work as fully aerostatic or aerodynamic or can change from one mode to other mode with change of speed. In hybrid bearing combination of aerostatic and aerodynamic pressure generation takes place.

The main drawbacks of aerostatics bearing is that a pressure source and an pressure exit needed. Main advantage of aerostatic bearing is that in aerostatic bearing manufacturing tolerance achieved and can take load at low speed and also aerostatic bearings can take take stationary load. Externally pressurised bearing can work in dusty environment because the gas at the exit opposes the entry of solid particle of the environment to the bearing. Due to viscosity of gas lubricant aerodynamic bearing supported small load per unit area. Aerodynamic bearing needs watchful manufacturing and proper alignment. Aerodynamic bearing is suitable only when there is relative motion between the bearing surfaces. No auxiliary instrument necessary for generating pressure so discharge of exhaust gas not a major problem.

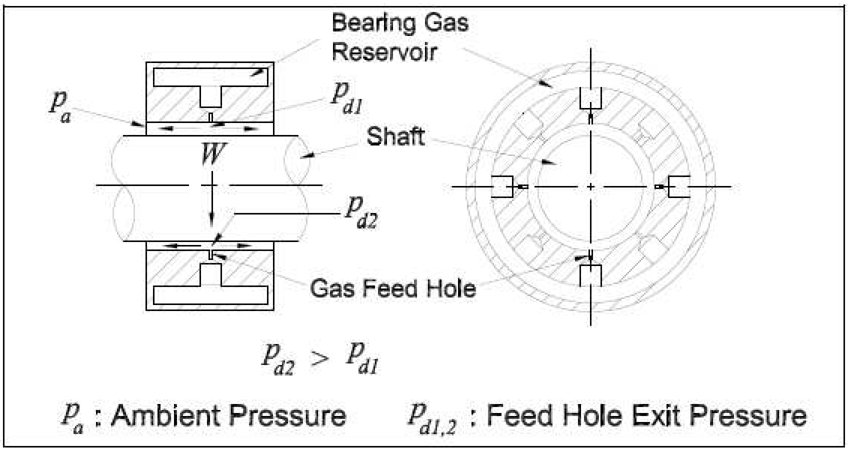


Figure 2.1: Aerostatic journal bearing [4]

2.2 Aerostatic bearing:

According to function of bearing, bearings are classified into two types i.e. journal bearing and thrust bearing . Journal bearing support the radial load so it has cylindrical geometry. Thrust bearing support the axial load. A thrust bearing has flat bearing surfaces. Aerostatic bearing can work combining as journal and thrust bearing of spherical or conical geometry. According to the type of flow restrictor aerostatic bearings are also classified .flow restrictor is the passage through which gas is supplied to the bearing clearance. Circular feed hole type restrictor used more widely, because circular shape is easy to produce. Some other type of restrictor used are of narrow slot type, capillary tube type. In case of an orifice having pocket at the outlet, smallest flow area generate at the bore of the fixed hole . throat area of the jet given by A = $\left(\frac{\pi}{4}\right) d_o^2$, where '$d_o$' is the feed hole diameter. In case of an orifice without pocket throat area A given by A = $\pi d_o h$, where h is the bearing clearance.

2.3 Gas bearing for cryogenic turbo-expander:

Working with various design of cryogenic turbo-expander suggests that gas bearing is the most suitable for supporting the shaft of cryogenic turbo-expander. Due to an instability factor half speed whirl, plain cylindrical bush type design of aerostatic bearing is not suitable. In order to avoid this half speed whirl type instability new geometry has developed. Early years aerostatic bearing have provided extra damping facility. Grooved and foil type of bearing used. In order to achieve high load carrying capacity and reliability no changes in the design of aerostatic thrust bearing done. In recent years grooved thrust plates gains more importance [4, 18, 23]. In 1934 for the first time cryogenic turbo-expander published by Linde work in Germany [4, 7]. For liquefaction of air an axial flow machine of single stage was used. In later stage the single stage axial flow machine was replaced by inward radial flow type impulse turbine. The shaft rotates at around 7000 rpm in horizontal position. Simple oil lubricated journal bearing of sleeve type used to support the shaft. Room temperature was maintained inside. In 1939 kapitza [7, 4] developed a much smaller turbine. This small rturbine used for production of liquid air at institute of physical problems, Moscow. This small turbine is radial inflow type having 50% reaction. This turbine was fitted with a flexible shaft that was balanced by two ball bearings. The orientation of the shaft is vertical. In order to stabilise the shaft from vibration damper was provided below the turbine.

In 1950 a lot of work was initiated for the development of small turbine at university of reading, England [4, 7]. The wheel diameter was taken 14.28 mm. Designed speed of turbine was 2,40,000 rpm and expected maximum speed is 3,00,000 rpm in room temperature. At the initial stage of turbine the shaft was supported by pivot type ball bearing. Applying kapitza's work on stability ball bearing was mounted flexibility. Damping was provided by a pair of oil dashpots situated perpendicularly. The bearing was lubricated by a small stream of oil continuously. This bearing works only for an hour [4, 19]. This problem gives fuel for the development of gas bearing.

2.4 Initial stage: aerostatic bearings

The experiment at university of reading gives foundation in the development of turbo-expander shaft supported by gas bearing. The first bearing developed couldn't last for a long. So ball bearing was replaced by gas bearing. In order to prevent the whirl instability the gas bearing was provided pneumatic dampers. Room temperature was maintained at the upper bearing while lower bearing was maintained at cryogenic temperature. The reliability of first bearing was not so much satisfactory. For improving the reliability shaft diameter increased up to diameter of rotor, and room temperature was maintained at both the bearing.

On the basis of experienced achieved at university of reading, development efforts was initiated by BOC corporation in 1959 [6]. At national bureau of standards, Boulder, USA Sixsmith and his team developed an expansion turbine for helium liquefaction purpose [3]. Rotor diameter taken 8 mm and designed speed is 6,00,000 rpm. The tested speed of machine is up to 7,20,000 rpm. At Los Amalton University this machine is currently in use for the refrigeration of cold neutron source. To support the rotor mounted in this turbine aerostatic thrust and aerostatic journal bearing was used. In order to eliminate half speed whirl type of instability a set of cavities and leak orifice are used as stabilising system for journal bearing [3].

In 1965 a high speed turbo-expander was developed at national bureau of standards , USA. This high speed turbo-expander was a part of a cold moderator refrigerator at Argonne National laboratory [22]. Here helium used as processed gas for the externally pressurised bearing. Birmingham, Sixsmith and Wilson [3] had introduced this bearing design. This bearing design helps in the analysis of stability at high speed. The bearing for the prototype turboexpander operated at extremely high pressure of 20 bar. Cold moderate bearing pressure was 6.5 bar. Up to speed of 3,00,000 rpm the operation was reliable.

Thomas [21] has designed a helium turbine. The working pressure limit of this helium turbine is 15 to 4.5 bar and flow rate 190 g/s. In this turbine externally pressurised gas used for both journal and thrust bearing. L/d ratio for the journal bearing was taken 1.5 and the shaft diameter was taken 25.4 mm. bearing clearance was taken 20 to 25.4 µm. The journal bearing consists of a centre section and two end caps. Machining of feed jets becomes easier due to this type of designing. The shaft material was nitride steel and the bearing material taken leaded bbronze. Leaded bronze has good antifriction properties. The bearing were tested for 40 hours (10 hours of 4 runs) at speed upto 95,000 rpm with no damage. This speed is beyond the maximum speed i.e. 85,000 rpm.

Mandar M. Kulkarni et al.[11] Explains the CFD simulation of bearings. They used CAD and CFD codes for the analysis of FSI (fluid structure interaction) problem. They observed that as shaft deviates from concentric position the fluid attempts to push the shaft towards the centre.

Khatait et al. [10] developed a simple design methodology for the design and selection process of bearing. They build a three dimensional model and had done CFD analysis in order to verify the theoretical results. For the development of future air bearing they designed a test setup and study the static and dynamic performance. They compared the experimental result with the analytical result.

Renn and Hsiao [17] made experimental and CFD study on the mass flow rate through the orifice in an aerostatic bearing. They made a number of simulation and experiment and find that mass flow characteristic through a nozzle is different from that of orifice. They suggested a new model for determining mass flow rate through orifice of an aerostatic bearing which helps in making a more precise design of an aerostatic gas bearing.

Chiang et al. [5] Studied the performance characteristics of journal bearing systems. They studied the dual effect of roughness on surface of journal bearing and lubricant blended with additives. Using the conjugate gradient method of iterations they numerically solved the film pressure distribution equation. He found that couple stress effect can increase the film pressure of the lubricant fluid, increase the load carrying capacity and decrease the friction parameter, particularly at high eccentricity ratio.

Nishio et al. [15] work on aerostatic annular thrust bearing having feed holes of diameter 0.05mm and investigate the dynamic characteristics both numerivally and experimentally. On the basis of CFD (computational fluid dynamics) software they calculate the discharge coefficient amd verified experimentally. They concluded that aerostatic bearing with small feed holes have large stiffness and damping coefficient as compare to bearing with compound restrictor.

Myatake and Yoshimoto [14] numerically investigated static and dynamic characteristic of aerostatic thrust bearing with small feed holes. For the numerical calculation they used computational fluid dynamics (CFD) software to determine discharge coefficient and finite difference method (FDM) for obtaining the bearing characteristics. They found that load capacity and static stiffness found by FDM gives nearly same result as obtained by CFD. They concluded that by reducing feed hole diameter higher damping coefficient can be obtained. They noticed that in aerostatic bearing having small feed holes, air flow is restricted at entrance of small feed holes and at the boundary. So two discharge coefficient needed in order to calculate mass flow rate of aerostatic bearing.

Bensouilah et al. [2] investigated theoretically the dynamic deformation of the foil on the dynamic performance characteristics of a self acting journal bearingoperated under small harmonic vibrations. Analysed the effect of dynamic deformation of bump foil of a self acting journal bearing. They used perturbation method for determining gas film stiffness and damping coefficient. By using Galerkin's finite element formulation they solved the nonlinear stationary Reynolds equation. They used finite difference method for solving the complex dynamic equations. They found that elastic deformation of bump foil affect the maximum value of dynamic pressure, and these effect are dominant when both dynamic and static deformation are considered.

Liang et al. [12] made analysis on aerostatic journal bearing with slot entry restrictor using computational fluid dynamics software. They showed the pressure distribution within the bearing model and showed under different eccentricity ratio. They showed the local pressure distribution and distribution of velocity around the slot entry restrictor. They calculated the bearing capacity and stiffness and show their curves. They compared their result with the result obtained by solving Reynolds equation using finite element method.

Majumdar [13] made theoretical and experimental study on externally pressurised gas bearing having multiple pressure inlet source. From this study he obtained solution for finite journal bearing considering axial and circular flow components. Validation of methods was done by using the experimental results. This method can be applied to bearing having any number of supply holes. This method can be used for the design of bearing under light and medium loads. From the experimental verification he predicted that the theoretical results obtained gives good bearing performances.

Fuller [9] made a review on the design of self acting gas bearings. He studied on the development in the basic phenomena of the bearing and technical design for this type of design. This study gives the fundamental idea about the gas bearing which will be helpful for the future researches.

Piekos [16] made simulation of journal bearing for the micro turbo-machines by using orbit method and a pseudo spectral technique for the treatment of Reynolds's equation. He published two stability chart and use that for the design of bearing for turbo-machines. Simulations for the instability, eccentric geometry, pressure gradient at the bearing ends are also taken into account during the designing. Flexibility of orbit method permits multiple options that can be applied to the model without extra formulation and the pseudo spectral treatment of Reynolds equation improves the computational efficiency in order to decrease the cost of simulation.

Belforte et al.[1] made theoretical investigation of fluid inertia effect and stability of self acting gas journal bearings. They numerically solved the Reynolds equation for compressible fluid film and develop a program. He validate the theoretical results with the help of the obtained experimental results. By varying the eccentricity he studied the effects of fluid inertia on the bearing. Effects of rotor imbalance on the stability of the bearing are studied. They concluded that numerical solution of the Reynolds's equation can be used for the prediction of static and dynamic characteristics of journal bearing.

Srinivasan and Prabhu [20] made analysis on the externally pressurised conical gas bearing. He solved the governing Reynolds equation in the clearance for the determination of pressure distribution. They find that conical bearings can be used to take both axial load and radial load at a time. For multiple supply holes this bearing design can give good results.

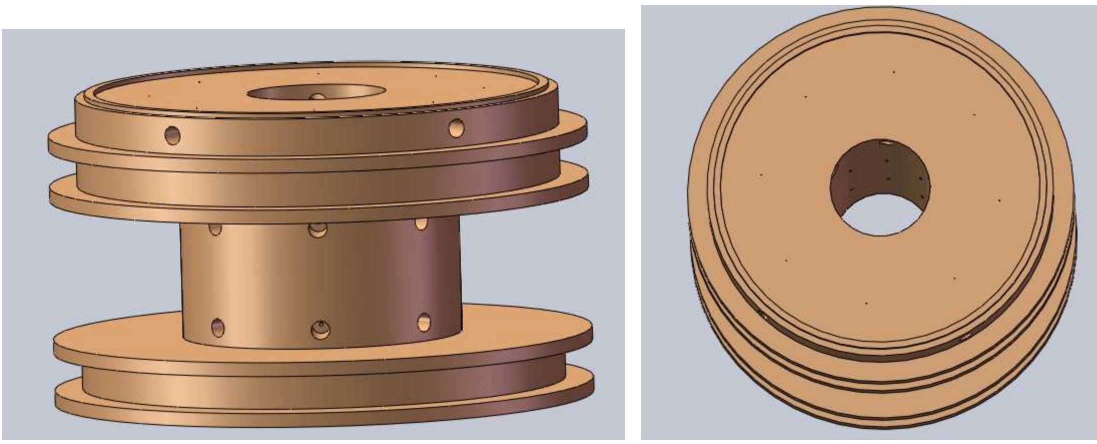
Denhard and Pan [8] application of gas lubricated bearing to different instrument. He discussed application of different type of gas bearing like self acting bearing, hydrostatic bearing and squeeze film bearing to different instrument. Self acting bearings are those which can generate its own pressure gradient required for the support of rotor.

Chapter 3

Numerical Simulation

3.1 Problem formulation:

In this work numerical simulation has been carried out on a two dimensional model of an aerostatic journal bearing using computational fluid dynamics software. The distribution of pressure and velocity in the bearing clearance and in the orifice were studied. Variation of load carrying capacity with respect to change in inlet pressure is analysed. In this study a three dimensional model and a two dimensional cross sectional model of aerostatic bearing used as shown in Fig. 3.2. Pressurised bearing gas enters the bearing clearance through the orifice. There are total eight number of orifice in each bearing. These eight orifice are situated in two rows. So in each row four number of orifice were located. In each row four bypass hole are situated. This four bypass hole and four orifice are arranged in a sequential manner. So there are total eight orifice and eight bypass hole are present in the aerostatic journal bearing. The high pressure bearing gas after doing its function exits to the environment through this bypass hole. Helium is taken as working fluid for this simulation in order to avoid problem due to contamination of bearing gas with the working fluid in the turbo expander, as the aerostatic bearing to be designed is for cryogenic helium turbo-expander. The high pressure gas supplied from each orifice inlet is sufficient to take the radial load. In order to take the axial load two



(a) Side view

(b) Top view

Figure 3.1: Three dimensional model of aerostatic bearing

aerostatic bearing placed in a position of one bearing inverted upon the other, so that from both side of the collar the high pressure gas can act. The diameter of orifice at the inlet is 2 mm. And near the clearance the diameter of orifice is 0.4 mm. Diameter of bypass hole is 0.6 mm. Shaft diameter is taken 15 mm. The bearing clearance is taken 50 µm. In order to simplify the simulation problem a two dimensional cross section model of aerostatic bearing was taken instead of the three dimensional model. Different cases are done by varying the inlet pressure. Load carrying capacity studied for different inlet pressure and plots are obtained. In this simulation problem analysis are made both for turbulent flow inside the bearing clearance. For the turbulent flow k-epsilon turbulent model was chosen.

The governing equations are given below.

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \nabla(\rho.\overrightarrow{V}) = 0 \cdots\cdots\cdots\cdots\cdots (i)$$

Momentum equation

$$\frac{\partial(\rho\overrightarrow{V})}{\partial t} + \nabla.(\rho\overrightarrow{V}\overrightarrow{V}) = -\nabla P + \nabla[\mu(\nabla\overrightarrow{V} + \nabla\overrightarrow{V}^T)] + \overrightarrow{F} \cdots\cdots\cdots\cdots\cdots (ii)$$

Where p is the pressure, t is the time, ρ is the density, μ is the dynamic viscosity, $\overrightarrow{V}$ is the velocity vector, F is the surface tension, T is the transpose.

3.2 CFD modelling:

The total numerical simulation was obtained by using ANSYS fluent 15 and. The simulation process goes on four stage, those are geometry creation, meshing, setup and solution. Each of the four stages are described below.

3.2.1 Geometry creation:

A three dimensional model of aerostatic bearing was created in SolidWorks. The three dimensional model has sixteen orifice placed in two rows. Each row has eight orifices. Above this two row of orifice another row of six bypass hole located. The diameter of orifice inlet is 2 mm and at the clearance the diameter of orifice is 0.4 mm. diameter of bypass hole is 2 mm. Shaft diameter taken 15 mm. The bearing clearance is 50 µm. In order to simplify the analysis a two dimensional cross-section model of nearly similar design of aerostatic bearing was drawn and CFD simulation was done upon it.

A two dimensional cross section model of aerostatic journal bearing was made in ANSYS workbench. This two dimensional cross section model of aerostatic journal bearing consist of eight orifice, eight bypass hole, and the bearing clearance. In order to create the geometry in ANSYS workbench, fluid flow fluent was selected from tool box. Geometry was created in the DESIGN MODELLER. Before building the geometry 2D must be selected in place of 3D in the analysis type.

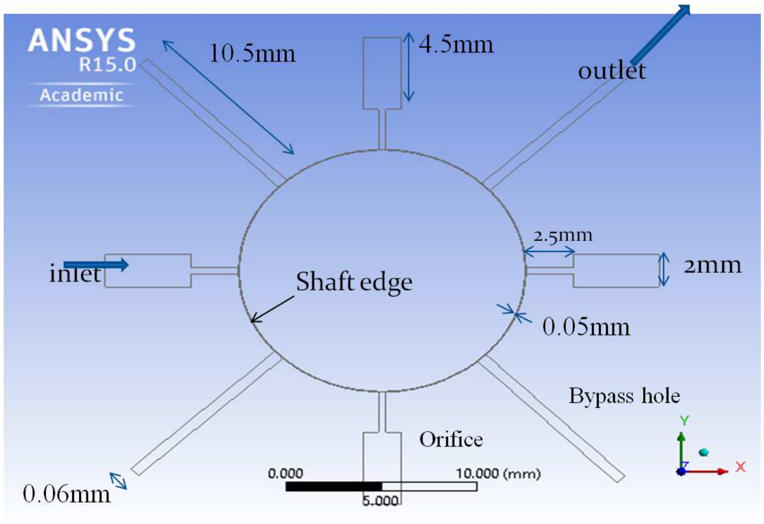


Figure 3.2: Two dimensional cross sectional model of aerostatic journal bearing

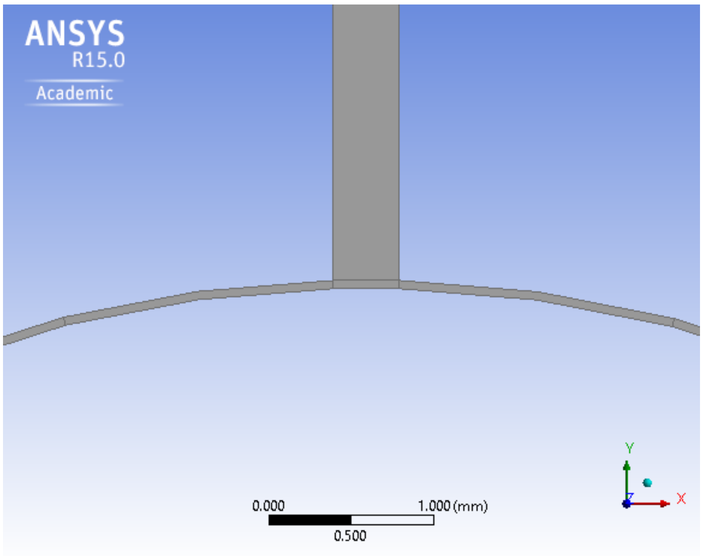
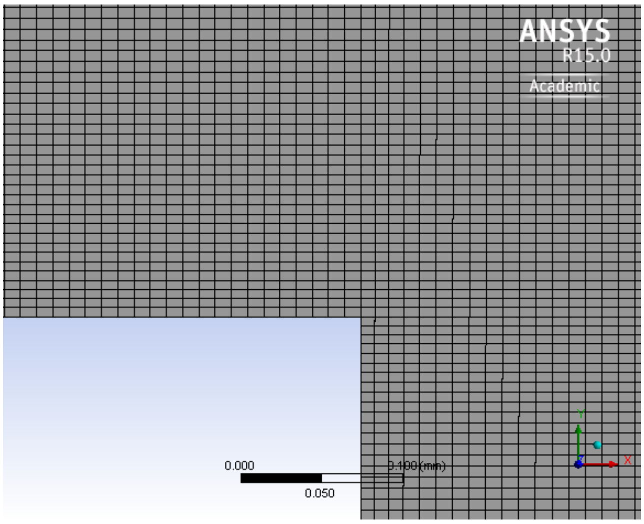
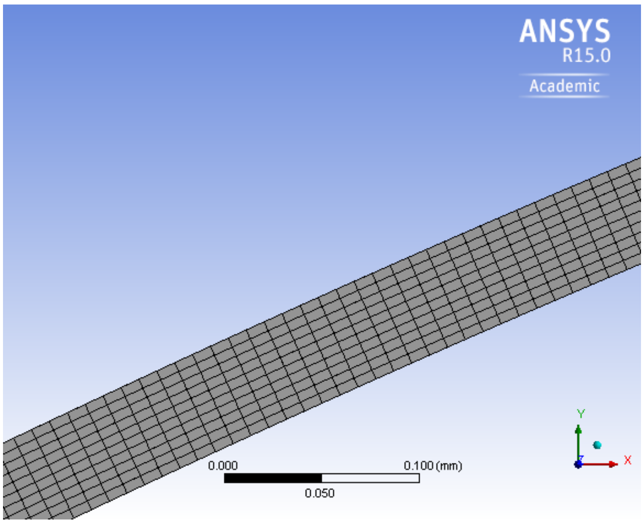


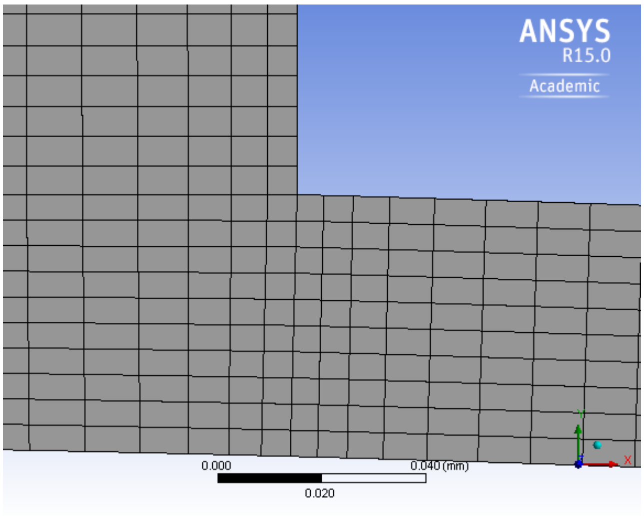
Figure 3.3: Orifice exit and bearing clearance



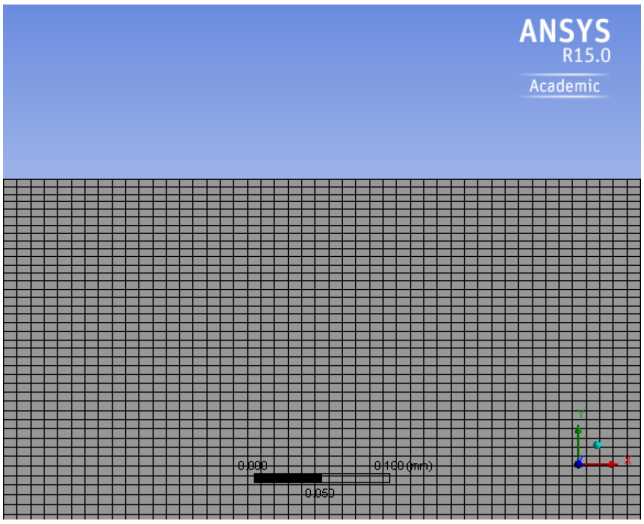
(a) orifice



(b) bearing clearance



(c) Exit of orifice and bearing clearance



(d) top of orifice

Figure 3.4: Meshing of two dimensional cross sectional model geometry

In order to sketch the geometry XY plane was selected and in unit mm was selected. After the parameter selection process sketch was drawn and the necessary dimension was given. After the creation of the sketches surface was generated. For the creation of surface clicked on concept than on surface from sketches than generate button. After surface generation face split was done, which will help in mesh creation. For the bearing clearance 40 face split was done and at the exit of each orifice a single face split was done. In this way a total of 48 face split was done. .After that geometry was saved.

3.2.2 Meshing:

Meshing is the important part of any numerical simulation. In order to get good accuracy fine meshing is desirable. Mesh was selected in the ANSYS workbench after the completion of the geometry. Mesh will be opened in a new window. By clicking on the generate mesh button mesh will be generated on the basis of bide fault value. For the generation of mesh different options are available in the mesh control. All quad was selected in meshing method. After that sizing was selected. In sizing edge sizing was chosen. In edge sizing the face split portion on the bearing clearance was selected and element size of 0.005 mm was given. So that the thickness of bearing clearance was divided into ten division. Grid type hard was selected for this. After that generate mesh button was clicked. For the curved portion of the bearing grid type soft was selected and element size of 0.01 mm was given. For the length part of the orifice, element size of 0.012 mm was given and grid type hard was considered. For this a bias factor of four was chosen. After edge sizing over mapped face meshing was done for all the face. Mapped face meshing was done to create uniform mesh structure. After that generate mesh option was clicked in order to create the mesh. After the mesh generation naming of different edges was done by selecting the edge and then create named selection. For this two dimensional simulation problem name 'inlet' was given to the eight bearing gas entry and name 'shaft edge' given to the intersection edge of shaft and bearing fluid. After the name selection was over mesh was exported to ANSYS fluent. In the figure given below mesh structure of different parts of the two dimensional model of aerostatic journal bearing was shown.

3.2.3 Setup in fluent:

Numerical fluid flow simulation problem was solved in ANSYS fluent. After the completion of the meshing the simulation work was done in fluent. The analysis type was chosen 2D in fluent. After fluent was opened exported mesh was read. After reading the exported mesh, mesh was checked. The parameter like aspect ratio, orthogonal quality etc. Are checked. Orthogonal quality should nearly equal to one. In the general setup menu pressure based type, steady time, absolute velocity formulation, planar 2D space was selected for the solver. Gravity has no role for this two dimensional simulation problem, so gravity was not selected in this problem.

In the model menu multiphase tab kept off because in our problem only one type of fluid was taken, mixing not take place. Energy tab kept off, as temperature and heat energy has no role for this problem. For the turbulent flow k-epsilon turbulent model was chosen. Where 'k' is the turbulent kinetic energy and $'\varepsilon'$ is turbulent dissipation. There are so many turbulent model like k-epsilon, k-omega, Reynolds stress model etc. Out of these turbulent model k-ε turbulent model was chosen because this is a validated turbulent model because of which it's application spreads from industrial to environmental fluid flow problem. k-ε turbulent gains popularity because of its validation and wide application. k-ε turbulent model is particularly for planar shear layer and recirculation flows. k-ε is a simple turbulent model where only initial and boundary condition need to be provided.

For turbulent kinetic energy k

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}[\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_j}]+2\mu_t E_{ij}E_{ij}-\rho\varepsilon$$

For turbulent dissipation ε

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho\varepsilon u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}[\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial\varepsilon}{\partial x_j}]+C_{1\varepsilon}\frac{\varepsilon}{k}2\mu_t E_{ij}E_{ij}-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}$$

where

$u_i$is velocity component in corresponding direction

$E_{ij}$is component of rate of deformation

$\mu_t$is eddy viscosity

$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$

By iteration and data fitting the values of adjustable constants obtained are

C_μ =0.09 , σ_k =1.00 , σ_ε =1.30 , $C_{1\varepsilon}$ = 1.44, $C_{2\varepsilon}$ =1.92

In the materials menu, bide fault air was there in fluid tab. Helium was selected from the fluent database as for this aerostatic bearing helium gas is the suitable working fluid. No need for selection of material in the solid menu as only the bearing fluid portion is considered for the present problem. In the cell zone condition menu, surface body type was selected as fluid and helium was selected in the fluid type. In the boundary condition menu, for the inlet and outlet bide fault velocity inlet was there. Velocity inlet was changed to pressure inlet and for the inlet gauge pressure of 6.8 bar was taken as absolute pressure inlet was 8 bar and ambient pressure was 1.2 bar. gauge pressure is the difference between absolute pressure and atmospheric pressure. As the outlet is to the environment, so gauge pressure for the outlet was 0 bar. Rotating speed of shaft was taken as 60,000 rpm along the origin (0,0) as the centre of the shaft lies in origin.

3.2.4 Solution:

In the solution menu SIMPLE scheme was selected for the pressure-velocity coupling. Least square cell based used for the gradient computation, 'second order' scheme was selected for the pressure descretisation, 'second order upwind' scheme was selected for the momentum descretisation, 'first order upwind scheme' was selected for the turbulent kinetic energy, 'first order upwind' was selected for the turbulent dissipation rate.

In the monitors menu convergence criteria of 0.001 was selected for continuity, x-velocity, y-velocity, turbulent kinetic energy (k) and turbulent dissipation (ε). In the solution initialisation menu 'standard initialisation' was selected by computing from all zones. After the initialisation was done case and data file was saved.

In the run calculation menu, 10^6 number of iteration was given. After the calculation was over, contours of total pressure, static pressure and dynamic pressure, velocity and vector diagram of velocity are extracted. Lines are created for the calculation of pressure at the exit of orifice, which is to be used for the calculation of load carrying capacity. The details are discussed in the chapter 'Results and discussion'.

Chapter 4

Results and Discussion

In this chapter the results of distribution of pressure and velocity in the bearing clearance are given. Fig. 4.1 describes the geometry of the aerostatic journal bearing. Thickness of bearing clearance taken is 50 µm.pressurised helium gas enters the orifice inlet at 8 bar pressure. The shaft rotates at 60,000 rpm. In the bearing clearance two types of pressure generated, one is static pressure which is due to inlet pressure and other is dynamic pressure which is due to the rotation of the shaft. Total pressure is the summation of static pressure and dynamic pressure.

In this work different cases are studied by varying the inlet pressure and bearing clearance. From these cases load carrying capacity of the bearing for different inlet pressure was studied. Contours of distribution of pressure and velocity at the bearing clearance and orifice are shown. Finally different plots are obtained.

4.1 Variation of total pressure at the exit of orifice and bearing clearance:

The contour of total pressure is shown in Fig. 4.2. It is noticed that inside the orifice there is not so much variation of total pressure. As the high pressure gas strikes the rotating shaft, decrease in total pressure is seen because of the obstacle. The pressure decreases along the bearing clearance. Presence of any obstacle in the direction of a flow path changes the dynamics of the flow and disturb the flow pattern and causes changes in drag characteristics. Because of the pressure drag, separation will occur. When the high pressure gas travels from orifice to clearance, velocity of the gas increases suddenly because the gas travels from a higher cross sectional area to a lower cross sectional area velocity of the gas increases from continuity equation. As velocity of the gas increases, so the pressure of the gas decreases as from Bernoulli's equation.

The distribution of total pressure at the exit of top orifice and bottom orifice are shown in Fig. 4.2 and Fig. 4.3 respectively. It is seen that the distribution of pressure at the exit of all the four orifice is same, due to which the shaft is balanced properly and can rotate smoothly in the bearing clearance.

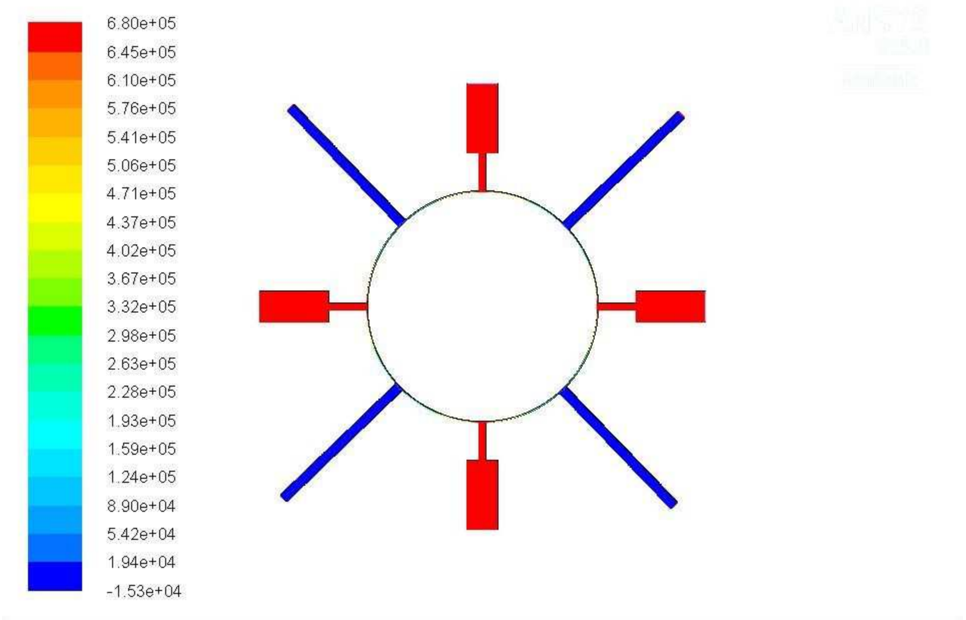


Figure 4.1: Contours of total pressure in the two dimensional cross-sectional model of aerostatic journal bearing

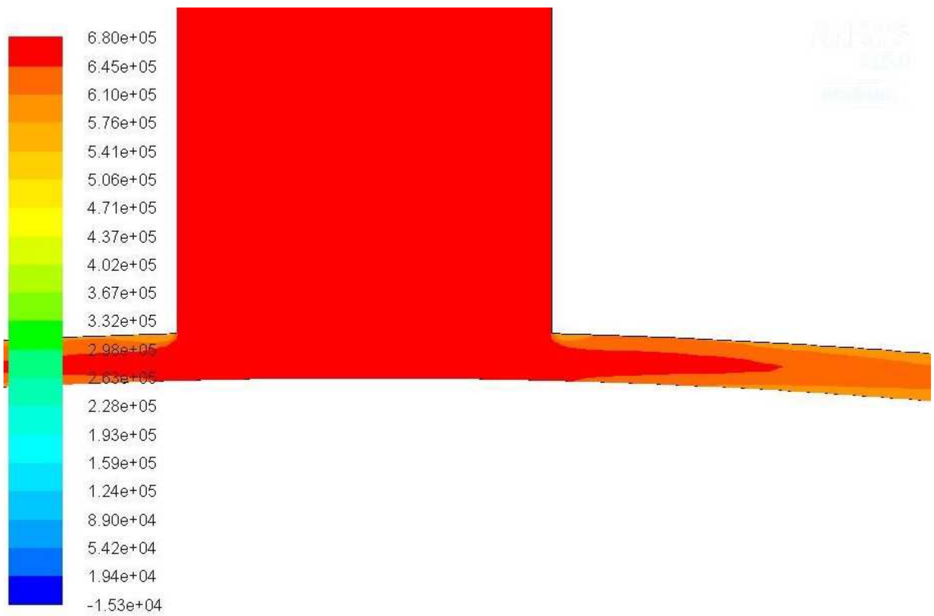


Figure 4.2: Contour of total pressure at the exit of top orifice

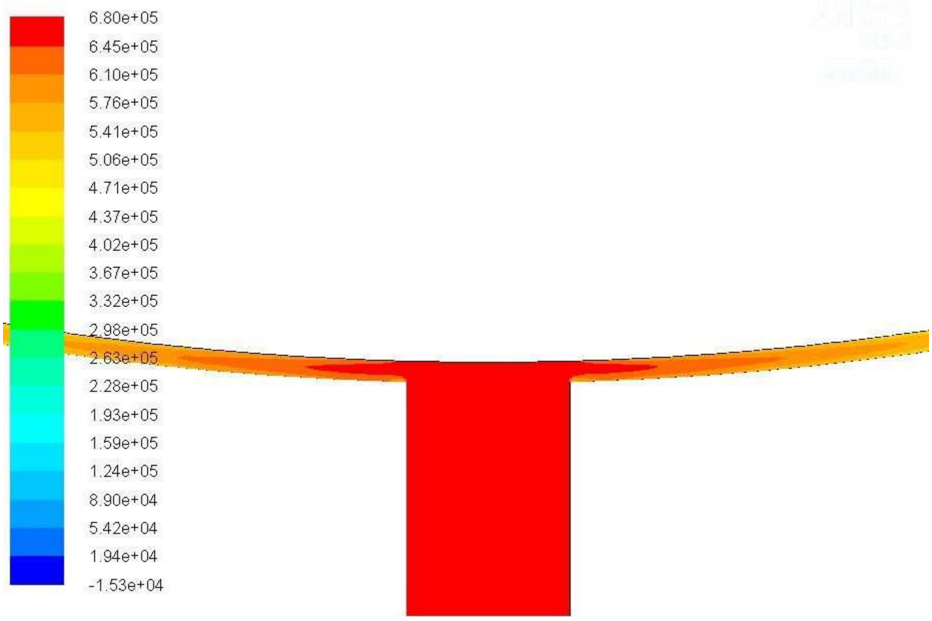


Figure 4.3: Contour of total pressure at the exit of bottom orifice

4.2 Contours of distribution of total pressure in the bypass hole:

The high pressure gas entered through the orifice after striking the shaft decreases its pressure. As the pressure of the gas decreases gradually along the clearance, hence the pressure of the gas when reaches the bypass hole a very low pressure zone formed. This low pressure gas exit through the bypass hole. There is no major variation of total pressure along the bypass hole. Noticeable variation of total pressure is seen only along the clearance, this is because of the variation of cross section from orifice to clearance and clearance to bypass hole. Total pressure is the summation of static pressure and dynamic pressure. From the total pressure contour in the bypass hole it is seen that as the medium pressure gas enters the bypass hole, up to some distance this medium pressure zone maintained than it converts to low pressure and travels through the bypass hole. Contours of dynamic pressure at the exit pressure and bearing clearance:

Dynamic pressure is the total kinetic energy per unit volume for a fluid element. In case of a fluid in motion dynamic pressure is derived from the conservation of energy equation. Dynamic pressure, static pressure and pressure due to change in height, are used in the Bernoulli's principle as energy balance in a closed system. These three terms are used in defining state of a closed system for an incompressible flow.

When dynamic pressure is divided by the product of fluid density and acceleration due to gravity than velocity is obtained. In a venturimeter pressure head differential can be used for the calculation of velocity head differential as shown in Fig. 4.5 Other name of velocity head is dynamic head.

From Fig. 4.6 it is seen that dynamic pressure in the bearing clearance is high while dynamic pressure in the orifice and bypass hole is less. The gas element in the bearing clearance rotates with the rotation of shaft but the gas element in the orifice and by pass hole are not affected with the rotation of the shaft. Due to this reason dynamic pressure in the bearing clearance is high and in the orifice and bypass hole is low.

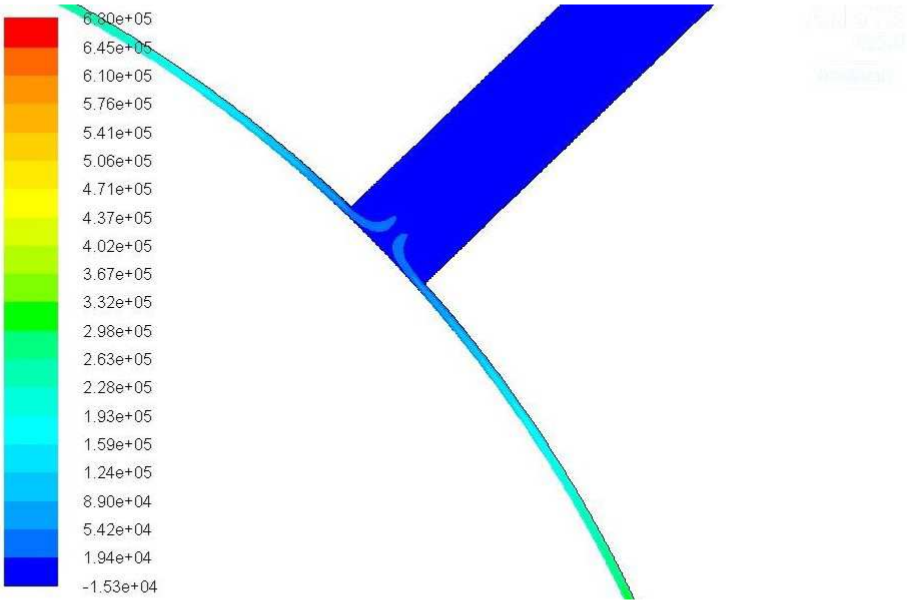


Figure 4.4: Contour of total pressure distribution in the bypass hole

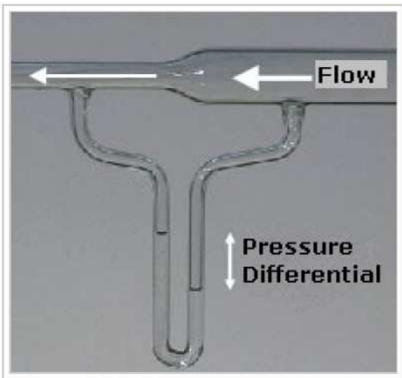
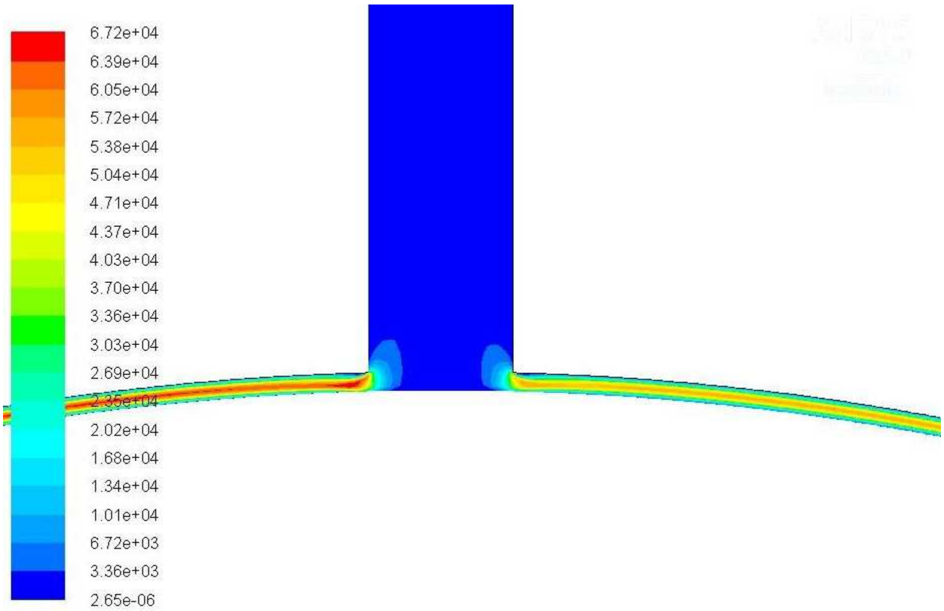
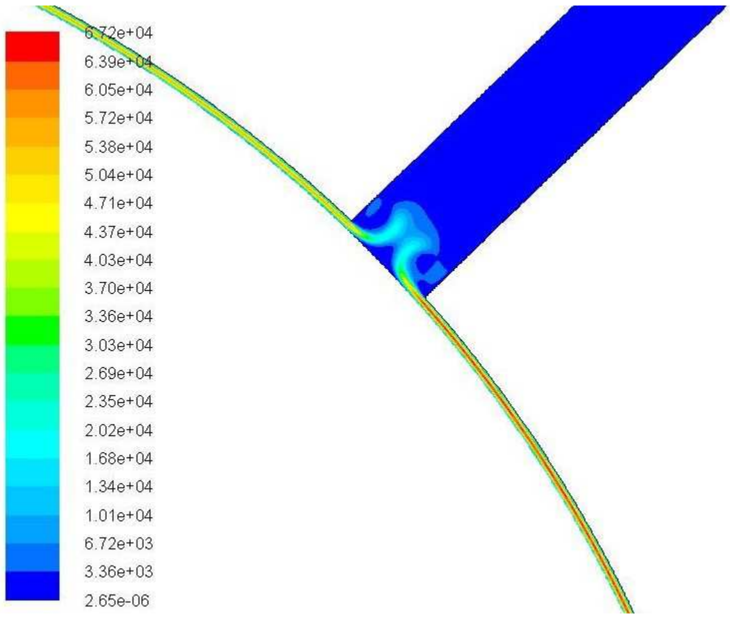


Figure 4.5: Pressure head differential in a venturimeter



(a) At the exit of top orifice



(b) At the bypass hole

Figure 4.6: Contour of dynamic pressure

4.3 Contours of static pressure:

The pressure of the fluid where the motion of the fluid is ignored is called as static pressure in fluid dynamics. In fluid dynamics where only pressure term is written it is referred as static pressure. In other words static pressure is the result of change in elevation.

From Fig. 4.7 no significant variation of static pressure is seen along the bearing clearance as the motion of fluid particle is not considered for the static pressure. The high pressure zone is maintained from orifice inlet to orifice exit and low pressure zone is maintained along the bypass hole.

4.4 Contours of velocity:

From Fig. 4.8 it is seen that velocity of gas when travels through the orifice is low. A sudden increase in velocity is noticed as the gas particle exits from orifice and enters the bearing clearance. This increase in velocity at the bearing clearance is due to sudden reduction in cross sectional area at the bearing clearance. So from continuity equation the velocity of the gas increases because of the decrease in cross sectional area.

In the Fig. 4.9 it is seen that when the bearing gas exits through the bypass hole reduction in velocity is seen, because of the sudden increase in cross sectional area from clearance to bypass hole.

4.5 Velocity vector:

From the velocity vector diagram Fig.4.10 it is seen that the bearing gas particle travels from orifice to bearing clearance at higher velocity due to the reduction in cross sectional area.

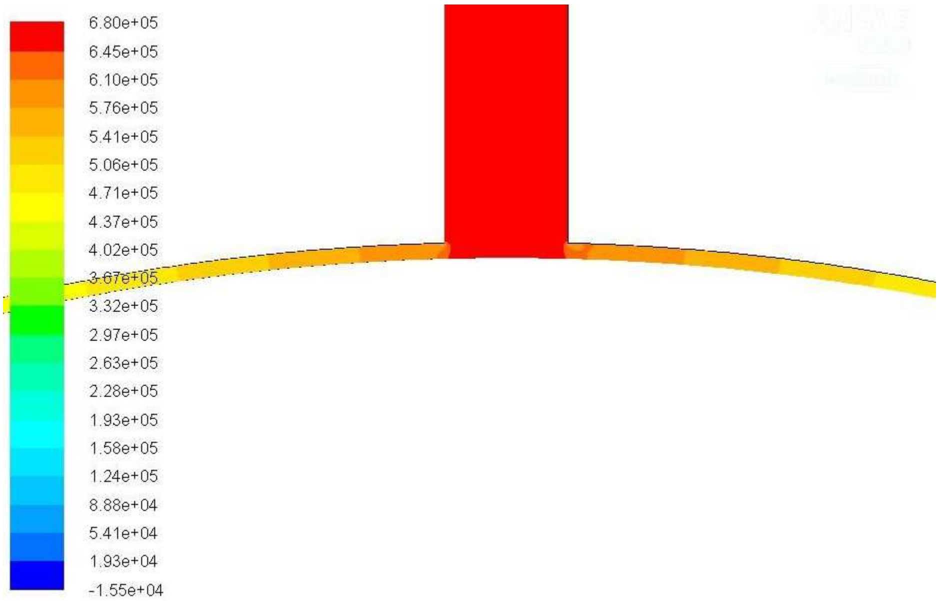
From the vector diagram Fig. 4.11 it is seen that bearing gas particle travels from bearing clearance to bypass hole at lower velocity due to the increase in cross sectional area.

4.6 Contours of turbulent kinetic energy:

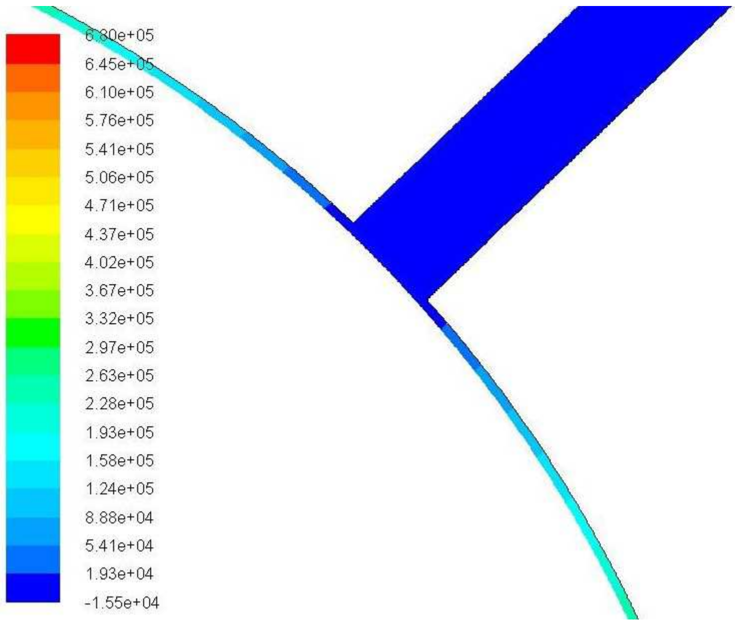
From Fig. 4.12 it is seen that turbulence of the gas increases when bearing gases from the orifice enters to the clearance. The increase in turbulence at the entrance to the bearing clearance is due to the sudden increase in velocity in this region.

4.7 Load carrying capacity:

Small turbines rotating at high speed generally designed for rotor in vertical position. This type of design removes radial load acted upon the journal bearing which have low load carrying capacity. As the rotors are small and light weight, therefore pressure and gravity load are absorbed by thrust bearings. For large size rotors, rotating speed is less because of which journal bearing with significant load carrying capacity can be made. In this design it is difficult to absorb the high weight of journal bearing. Because of which horizontal configuration are preferred for large turbine. For application



(a) At the exit of top orifice



(b) At the bypass hole

Figure 4.7: Contours of static pressure

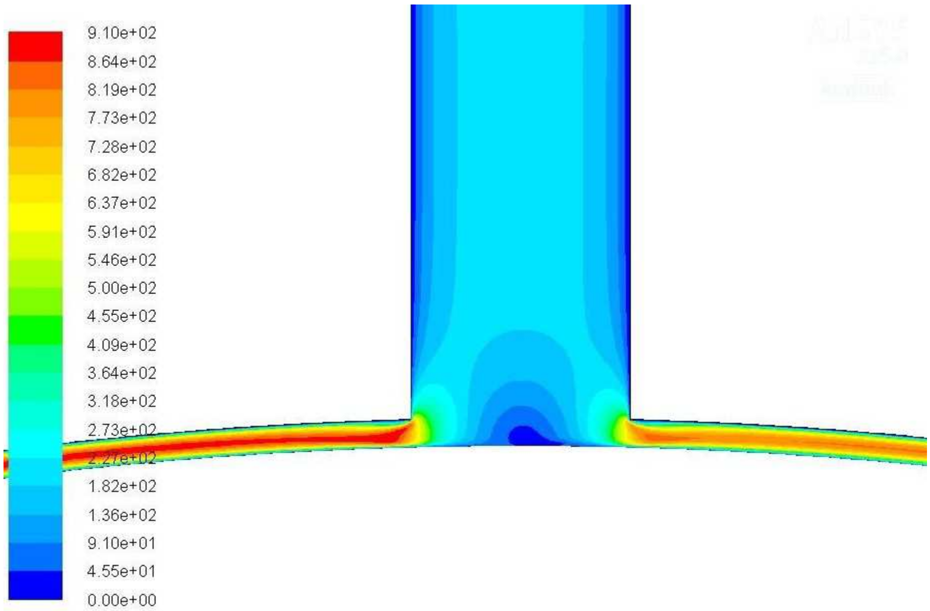


Figure 4.8: Contours of velocity at the exit of top orifice and bearing clearance

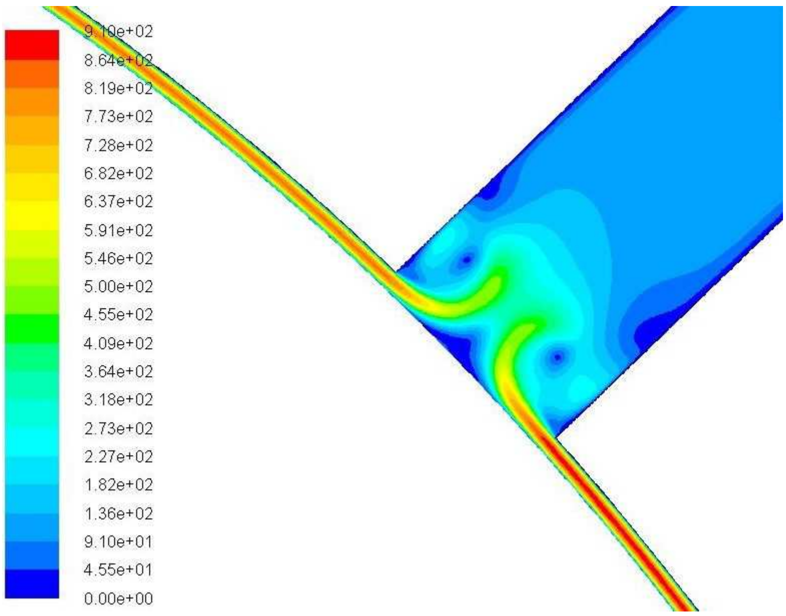


Figure 4.9: Contours of velocity at the bypass hole

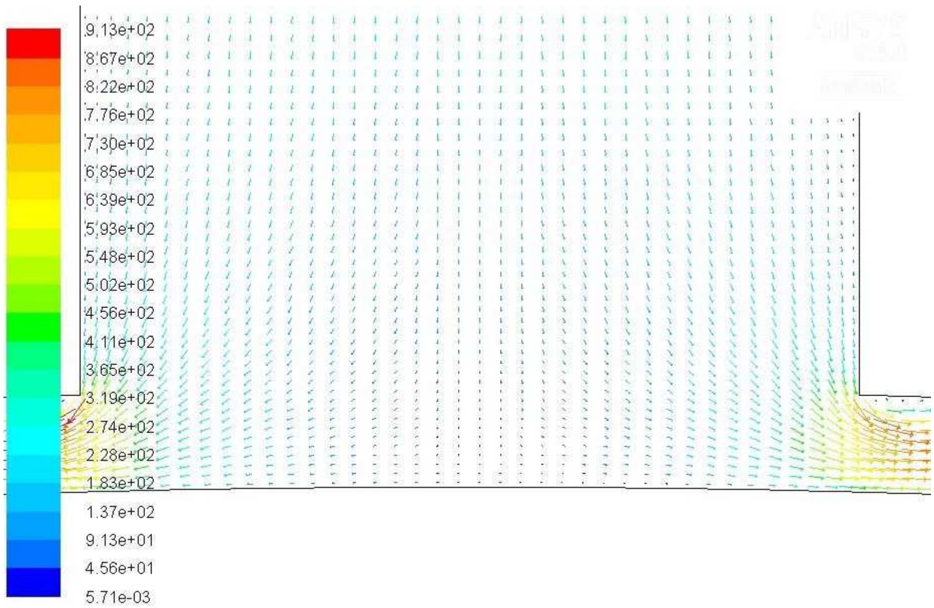


Figure 4.10: Velocity vector at the exit of top orifice and bearing clearance

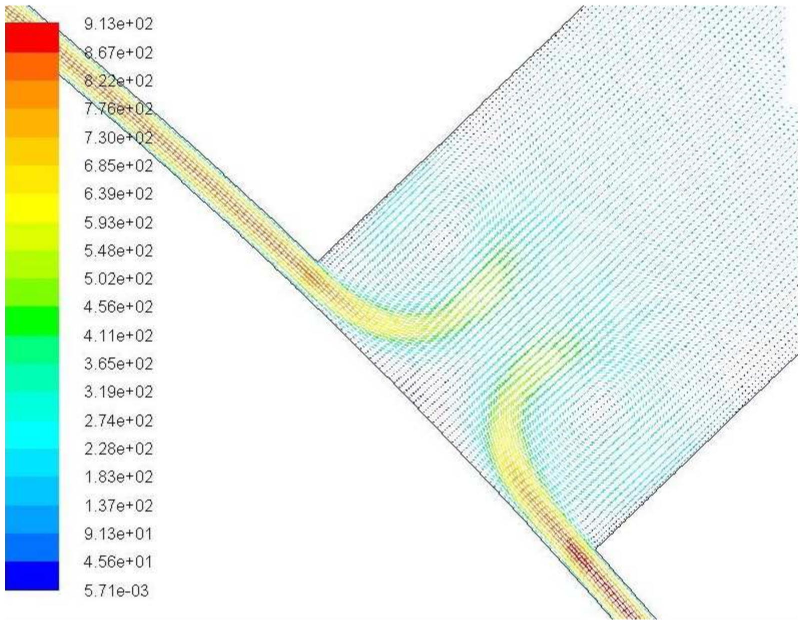


Figure 4.11: Velocity vector in the bypass hole

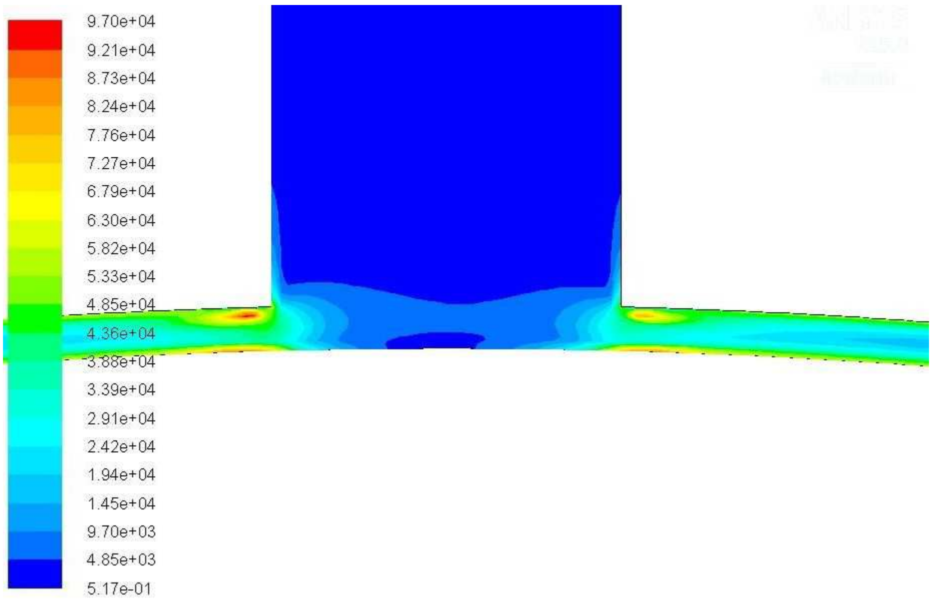


Figure 4.12: Contours of turbulent kenetic energy in the orifice exit

Inlet pressure (bar)	Load carrying capacity (N)
8	38.1483
9	43.7623
10	49.3717
11	54.9815
12	60.5898
13	66.1998

Table 4.1: Load carrying capacity with the change in inlet pressure

in critical condition, vertical configuration prevents heat leakage between cold and warm ends of the machine.

Load carrying capacity of a bearing means the maximum load taken by the bearing. It is the main design factor of a bearing. For the aerostatic bearing the load carrying capacity is given by

$$W = (P_d - P_a)\frac{\pi(b^2-a^2)}{2log_e(\frac{b}{a})}$$

Where

P_d is discharge pressure at the exit of orifice

P_a is ambient pressure

b is the bearing radius

a is the pocket radius

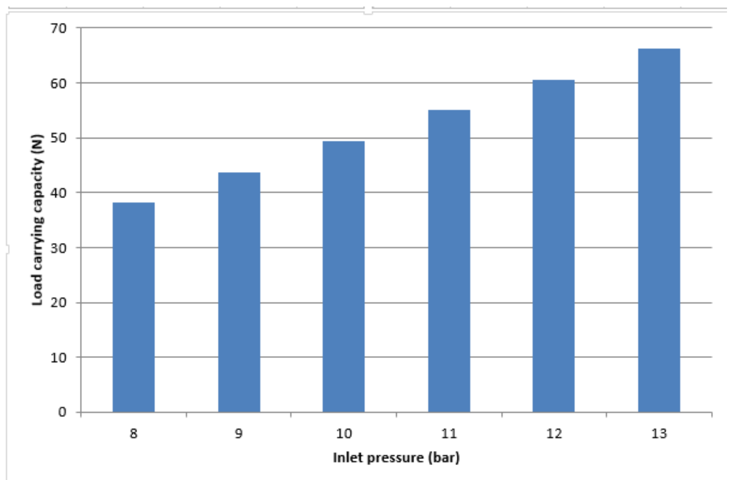


Figure 4.13: Variation of load carrying capacity with inlet pressure

Chapter 5

Conclusion:

Computational fluid dynamics analysis was done on the two dimensional cross-sectional model of aerostatic journal bearing. Results for distribution of total pressure, static pressure, dynamic pressure and velocity are obtained after the simulation was done in ANSYS Fluent. From the obtained results it is seen that the total pressure distribution at the exit of all the orifice is same, which helps in providing good support to the rotor and allows the shaft to run smoothly.

By changing the inlet pressure the discharge presure at the exit of the orifice also changes. The discharge pressure at the exit of orifice is the main driving factor for the support of the load. High discharge pressure has higher capacity to take the load. The magnitude of discharge pressure at the exit of orifice depends on the shape of orifice and it's dimension. In this CFD simulation problem shape and dimension of orifice are not changed. By changing the value of inlet pressure, the load carrying capacity is calculated and it is concluded that load carrying capacity increases linearly with the increase in inlet pressure.

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