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MITIGATION OF AXIAL VIBRATION AND THRUST BEARING FAILURES IN ROTATING MACHINERY

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ABSTRACT

The objectives of this tutorial are to familiarize rotating machinery engineers involved in detailed engineering and site operation on various aspects of axial vibration in rotating machinery. Among all rotating machines failure modes in various sectors of industries, issues of axial vibrations and thrust bearing failures are puzzling and difficult to solve. There are many factors involved with issue of high axial vibrations and thrust bearings failures which can be addressed right from design / selection (early engineering phase of a project). The prudent knowledge of various failure modes owing to high axial excursion of rotor / high axial loads at various operating points and proper selection of bearing and rotor setting can mitigate this type failures in great extent.
The proposed tutorial paper endeavors to provide readers various aspects and issues causing high axial thrust and mitigation actions in combination. This tutorial takes up few salient case studies as well. A sample thrust calculation included in this paper render a good understanding of predicted behavior of machine in terms of axial vibration and subsequent failure modes. Correct methods of sensor mounting, methods of proper setting of thrust bearing end play and voting are discussed as a part of on-site work carried out during pre-commissioning activity of mission critical turbomachinery.

INTRODUCTION

The objectives of this tutorial are to familiarize rotating machinery engineers involved in detailed engineering and site operation on various aspects of axial vibration in rotating machinery.
This tutorial does not serve as a design primer or selection guideline of machinery parts and bearings. This tutorial considered that primary audience are rotating machinery engineers involved in detailed engineering and plant machinery engineers involved in site operation
For a generic and quick understanding for the reader, the phenomena related to this topic is explained as below in a simple form –
A radial impeller rotating with an angular velocity in fluid, produces a centrifugal effect, the energy from the pump impeller is transferred to kinetic energy creating a hydraulic head proportional to peripheral velocity of rotating impeller. The pressure on back shroud of impeller is much higher than pressure at impeller front shroud and total thrust on each side is different in magnitude and direction .The axial thrust is the resultant force of all the axial forces (F) as vectorial sum of all forces acting on the shaft centerline of centrifugal machine .In a multistage straight through machine , axial thrust force increases with the number of stages and tries to move entire rotating assembly towards low pressure side. Chances of severe rubbing damage of rotating parts against stationary parts, failure of the thrust bearing and bending of shaft becomes high due to such high unbalanced axial forces. The axial or thrust position is one of the most

critical measurements in rotating machinery. If a thrust bearing fails, axial movement of the shaft is no longer constrained and uncontrolled axial movement leading to rotating and non-rotating elements to come in contact, resulting in disastrous consequences. Such unwarranted occurrences are financially devastating for the asset and can also be a serious safety risk to plant personnel.
Hence, it becomes imperative to protect the machine from such occurrence by proper design, installation, and monitoring methods. This tutorial deals with subject in perspective of centrifugal pumps and various types of compressors with phenomena, mitigation during design phase and mitigation during installation and commissioning phase.

Fig 1 Thrust bearing failures

PHENOMENA OF HIGH AXIAL THRUST WITH VARIOUS TYPES OF MACHINES

Centrifugal pumps being most widely used in industry and a good representation of other centrifugal type machines , example of axial thrust in centrifugal pump is shown for a good understanding [1]. All centrifugal machine in general follows the similar principle .
Axial thrust in centrifugal pumps

Axial forces acting on the rotor in the case of a single stage centrifugal pump is explained below for quick reference for readers. Vectorial sum of below forces results into net axial thrust force .

1. Thrust on the Front Shroud F1 due to suction (pointing away from direction of the driver)
2. Thrust on the Back Shroud F2 (pointing towards from direction of the driver)
3. Thrust Due to Momentum Change of Fluid Fm

The axial force (F1) is the difference between the axial forces on the discharge-side (F1) and suction-side (F2) impeller shrouds. Due to recirculation some of the pumped fluid exert additional force on the suction shroud F1.
Actual thrust force = F2 – F1
All components of axial thrust are normally shown with unit in N or daN.
Thrust due to rate of momentum change of Fluid (Fm) is a force which constantly acts on the fluid contained in a defined space (based on principle of conservation of momentum,). The principle of conservation of momentum or momentum theorem is a conservation law which states that all external forces acting on a fluid contained in an isolated environment (control volume) must be in equilibrium. It plays an important role in centrifugal pump engineering). The principle of conservation of momentum represents the integral form of the Navier-Stokes equation.
It is calculated as follows –

$$\mathrm{Fm} = \rho \cdot \mathrm{Q} \cdot \Delta \mathrm{Vax} \qquad (1)$$

where
Q is Flow rate,
ρ is density of the fluid handled
ΔVax is difference between the axial components of the absolute velocity at the impeller inlet and outlet.
To simplify further , when we multiply density and volume flow we get mass flow . When we multiply the mass flow with change of axial component of velocity we have change of momentum . However as the mass flow rate is measured in unit time such as kg per second so total expression becomes rate of change of momentum which is force [axial force] being subjected to rotor.

The axial thrust component (Fn + Fm) of closed impellers (i. e. with suction-side shrouds) which are not hydraulically balanced is:

$$(\mathrm{Fn} + \mathrm{Fm}) = \mathrm{kt} * \rho * \mathrm{g} * \mathrm{H} * \mathrm{Dm} * \Pi/2 \qquad (2)$$

Where

- Kt is axial thrust coefficient based on pump geometry and various dimensions.
- Dm is mean impeller diameter,
- ρ is density of fluid,
- g is gravitational constant.
- H is head developed by pump.

A short note on axial thrust coefficient Kt is as follows - Thrust measurement are normalized with flow which is a function of pump speed and geometry to produce a thrust coefficient Kt curve. The axial thrust coefficient Kt or sometimes shown as α is dependent on the specific speed (Ns) , type of impeller (radial flow / mixed flow),actual flow vs rated flow (ratio of flow coefficient) , clearance gap and leakage through side wall of casing .[2]

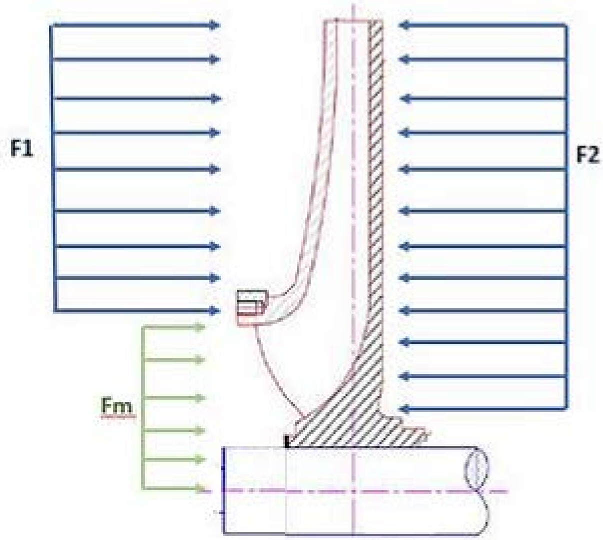


Fig 2 – Directional representation of actual thrust pressures in centrifugal pump section

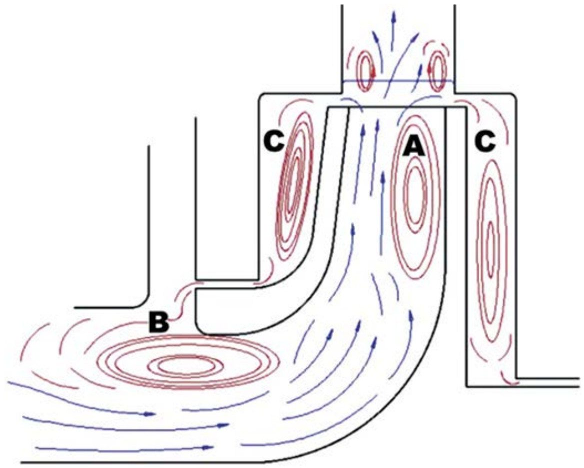
Axial forces due to shaft seal and flexible couplings are quite less in comparison and hence not included .
If the clearance gap width is doubled, axial thrust coefficient dependent on factor (Kt) increases by @ 7-9 %.
In the case of multistage pumps with diffusers (e. g. boiler feed pumps), the axial force (Fn) plays a pivotal role in impeller's axial position in relation to the diffuser.
Other axial forces such as the force of the rotor weight (Fw) on vertical centrifugal pumps or magnetic pull in the electric motor (Fmag), are also considered based on type of design and installation.
A unified formula which clarifies on vectorial addition of forces as net axial forces as below –

$$\overrightarrow{Fn} = \overrightarrow{F1} + \overrightarrow{Fm} + \overrightarrow{F2} + \overrightarrow{Fmag} + \overrightarrow{(Fw)} \tag{3}$$

We may argue that a double-suction impeller is in hydraulic axial balance with the pressures on one side counter-balancing the pressures on, the other side. In practice, this balance may not be achieved for the following reasons:

- The suction passages to the two suction eyes may not provide equal or uniform flows to the two sides.
- External conditions such as an elbow being too close to the pump suction nozzle may cause unequal flows to the suction eyes.
- The two sides of the discharge casing may not be symmetrical due to casting process, or the impeller may be located slightly off-center. These conditions will alter the flow characteristics between the impeller shrouds and casing, causing unequal pressures on the shrouds.

When these factors are combined, they create a definite axial force unbalance. To compensate for this, all centrifugal pumps, even those with double-suction impellers, incorporate a set of thrust bearings in form of single or double row / angular contact ball bearing to maintain rotor axial position. [3]
The unstable flow interactions identified as zone C in figure 3 with "Flow Inside the Sidewall Gaps "with the vortex that exists within the gap between the channel ring and impeller shroud. This vortex accounts for the localized thrust within this pocket. Any disruption to this vortex leads to a more unpredictable thrust regime.



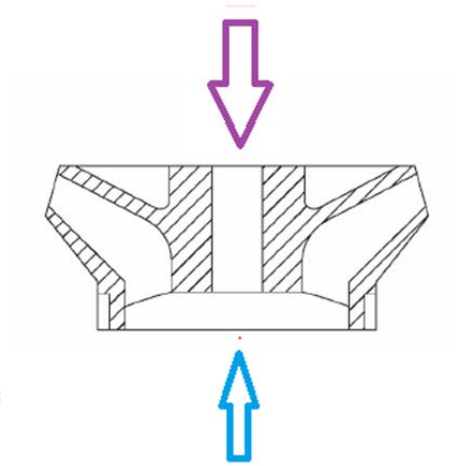


Fig 3– Flow recirculation in "side walls"

Fig 4– Up thrust (blue arrow) and down thrust (purple arrow)

Axial thrust Problems in vertical pumps

Vertical turbine pump impellers are subjected to hydraulic axial forces (thrust,) along with weight of pump rotor, which must be properly accounted for in pump and driver selection and design and pump operation. This is known as the down thrust load of a vertical pump

It is not advised to operate / start the pump against a closed valve, as this will result in very high down thrust loads. This can occur as part of start and stop procedure (pump operating in parallel, pumps on long pipelines). It may even occur during a field test of a pump.

In addition to the down thrust force (shown with purple arrow in Fig 4) which is caused by differences in pressure surrounding the impeller, there is also a force commonly known as upthrust. In the normal operating range of the pump, this upthrust (which is always present at any flow) is small in magnitude compared with the down thrust. However, at very high capacities, the actual value of upthrust force may be much greater in value than the down thrust. [4],[5]

For an impeller installed in a pump with a 90° turn in immediate discharge, the approximate value of this upthrust force (F)

$$F = (Ve^2 / 2g) \times (\text{net eye area}) \times \text{constant}. \qquad (4)$$

Where

- Ve = velocity in eye of impeller m/sec
- g is gravitational constant. m/ sec^2

The consequence of high upthrust are –

- Line shafts may bend (buckle) due to compression load, cause vibration, leading to rapid bearing wear and then impellers rub on top of bowl.,
- Driver radial bearings undergo upthrust loads and fail rapidly.
- Driver thrust bearings (such as angular contact) fail since they can take thrust in only one direction if they are taper roller or tandem angular contact ball bearings.
- Motor rotor may rub against stator causing severe electrical and mechanical damage.

When a pump is first turned on, the pump usually operates at very high capacities, because the motor gets up to speed in just a few seconds, and it may take somewhat longer for the head to build up due to system back pressure. The pump is therefore likely to operate in the very high-capacity range where upthrust occurs. In most installation, the head builds up almost immediately, so that the upthrust is only "momentary." However, even though it is momentary, equipment must be designed to take this upthrust.

If the motor is equipped with an anti-friction type thrust bearing (ball bearing, spherical roller), there should not be an issue of great concern, even if the thrust bearing is overloaded, because such bearings are capable of high overloads for a short duration without significantly affecting the life of the bearing. It is suggested that during detailed design review capability of the bearings must be understood. However, where the motor is equipped with a hydrodynamic thrust bearing, then the bearing thrust capacity must be designed for the shutoff thrust. Drivers must have @ 30% momentary upthrust capacity for vertical installation.

Adequacy of selected anti-friction bearings can be checked by pump designer using easily available online calculation tool for L10 life calculation

Axial thrust in Centrifugal Compressors

Thrust loads in compressors due to aerodynamic forces are affected by impeller geometry, pressure rise through the compressor, and internal leakage due to labyrinth clearances. The impeller thrust is calculated, using correction factors to account for internal leakage and a balance piston size selected to compensate for the impeller thrust load. In many cases, all the impellers (if multi-stage) are mounted in the same direction and after the last stage there is a balance drum. The balance drum will have the discharge pressure on one side and the suction pressure on the other side. The routing back to the suction side often accomplished by external piping. By proper sizing of the drum the differential pressure across the device generates a thrust opposing the thrust developed from movement of the process medium. It may not fully oppose the thrust, but the net thrust is what is used to size the thrust bearing. The drum have some extensive sealing arrangements since a pressure differential exists across the two sides. If those seals are worn then the drum/disk doesn't develop the intended thrust and the net thrust, that imposed on the thrust bearing increases. That line should be checked to insure it is open although it is always a good idea to verify OEM (Original equipment manufacturer) recommendations with respect to operating window of compressor Many installations use differential pressure gages to ensure that proper pressures exist on each side of the balance disk/drum. Installations also are sometimes equipped with a method of flow measurement of the process medium back to the suction side. If the flow becomes excessive that is an indicator that the drum sealing arrangement is malfunctioning. Under such conditions, thrust go up and can cause thrust bearing failure [6] . Proper balance drum sizing, operation and monitoring become critical factor in design. Standard API-617, most widely used standard for centrifugal compressors, suggests that a separate pressure tap connection may be provided to indicate the pressure in the balance chamber. It also suggests that the balance line shall be sized to handle balance piston labyrinth gas leakage at twice initial clearance without exceeding the load ratings of the thrust bearing and that thrust bearings for compressors should be selected at no more than 50% of the bearing manufacturer's rating [7]. During operation, the rotor of a centrifugal compressor is subject to an axial thrust F resulting from the sum of several components:

$$F = F m + F1 + Fb + Fc \qquad (5)$$

Where

- Fm, due to variation of gas momentum
- F1, due to differential pressure across the impellers
- Fb, due to differential pressure across the balance piston
- Fc, due to coupling pre-stretch

Fig 5 shows a generic sample calculation and direction of axial force in back-to-back impeller arrangement. A residual thrust calculation is a prudent activity to maintain a balance between thrust bearing dimension and balance drum dimensions. Thrust bearing capacity, oil consumption and mechanical losses also are key factors related to overall axial thrust.

One of key point in design is to use both sides of thrust bearings (active and inactive) when wide range of thrust inversion is anticipated for wider operating range of compressor.[8]

The generic procedure for estimating of net axial forces , an end user uses is to execute vectorial summation of forces acting on each stage and force on balance pistons caused by differential pressure across balance piston at various operating conditions [9]. Some inputs for calculations are taken from OEM data sheets and cross sectional drawings It is to be noted that for generic simple calculation ,front shroud curvature is not considered as it needs an specialized mathematical treatment using calculus (involving cyclic integrals)and polar coordinates. [10],[11]

stage -1 SUCTION PRESSURE	2.134	BAR A
DISCHARGE PRESSURE	16.37	BAR A
NUMBER OF IMPELLERS inline	5	
SHAFT DIA	18.5	cm
IMPELLER EYE DIA	54	cm
AXIAL THRUST FORCE BY IMPELLER TOWARDS SUCTION	29162.7472	kg
stage -2		
SUCTION PRESSURE	14.85	BAR A
DISCHARGE PRESSURE	40.55	BAR A
SHAFT DIA	25	cm
IMPELLER EYE DIA	58	cm
AXIAL THRUST FORCE BY IMPELLER TOWARDS SUCTION	56027.2428	kg
BALANCE DRUM DIA		cm
BALANCE DRUM AREA	927.23	CM^2
BALANCE PISTON THRUST TOWARDS DISCHARGE	37599.1765	kg
RESIDUAL THRUST FORCE	47590.8135	kg
TOTAL THURST FORCE	47590.8135	
SUCTION PRESSURE 2nd stage		
DISCHARGE PRESSURE final		
NUMBER OF IMPELLERS opposed		
AXIAL THURST TOWARDS SUCTION		
BALANCE PISTON DIA		
BALANCE PISTON THRUST TOWARDS DISCHARGE		
RESIDUAL THRUST		
THRUST BEARING AREA	155	cm^2
THRUST PRESSURE	47590.8135	
NO OF PADS		
PRESSURE ON THRUST BEARINGS	307.037507	KG/CM2
	4359.93259	PSI

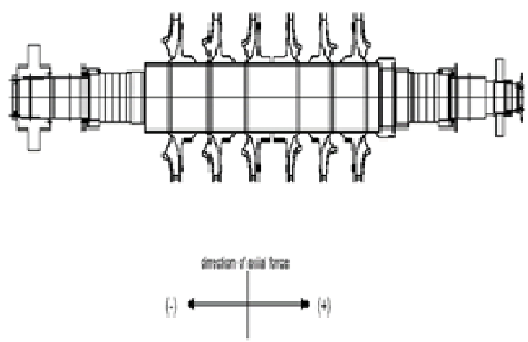


Fig 5 – Axial Load vectors (right) and thrust calculation (left) for rated condition

Axial thrust during centrifugal compressor surge and stonewall condition

For the accurate evaluation of the thrust in centrifugal compressors for medium and high-pressure applications with between bearing design, the paramount factors which influence the thrust and must be considered include the operating conditions (surge or choke), the different combinations of operating modes inside the compressor. For the design of a centrifugal compressor, the analysis of the axial rotor thrust during the pre-design phase has a decisive consequence on the final configuration of the rotor. Along with surge case scenario, it is necessary to consider the axial thrust in a choke case for proper sizing of the balance piston. There are some important technical papers on failure modes have been presented in turbomachinery symposia. [12],[13],[14]

During an event of surge, the pulsation of gases exerts axisymmetric force on rotor which causes axial excursion of rotor. The amplitude may be of very high order but at lower frequency. Below figure 6 shows as close correlation of pressure pulsation in psig/barg(yellow ochre) and compressor axial vibration in microns / mils (purple) during a surge event.

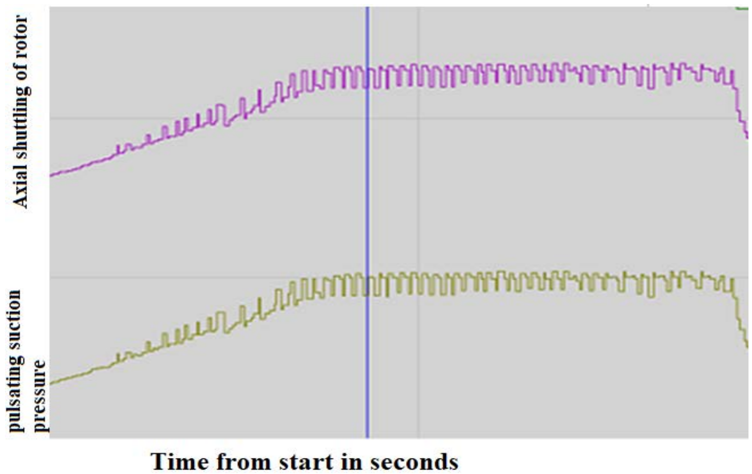


Fig 6 -Correlation between suction pressure pulsation and rotor axial shuttling

Electric Induction Motors

Many electric motors with sleeve bearing design (as shown in generic figure 7)have no thrust bearings and electromagnetic force do center the rotor to its geometric center. Hence the magnetic center of the motor rotor should be known, as well as the coupling float per OEM specifications. This is more important for a 2 Pole motor to prevent high axial thrust caused in this case, compared to a 4 Pole motor as 2 pole motor are comparatively lighter and hence axial excursion can be comparatively more.[15]

Fig 7 – Sleeve type hydrodynamic bearing for electric motor

Machines which are connected through a flexible coupling to equipment having an axially fixed shaft should have the running end play adjusted when the machine is uncoupled. When coupled and machine is running, the shaft position should be checked, and corrections should be made if a journal end is "drawn" to the bearing by the coupling. DBSE (distance between shaft end) is very important dimension to be maintained during coupling the driven and driver . Normally a margin of few millimeters is provided to compensate

thermal effects .
The below picture (figure 8) shows the coupling with DSBE at normal position when electric motor with sleeve bearing is in standstill The position of end face of coupling hub is shown in blue . However , when the motor turns and reaches its full speed ,the magnetic pull shall cause position of end face of coupling hub at new position shown in red which shall change the effective DBSE . This may cause operational anomalies of unit .

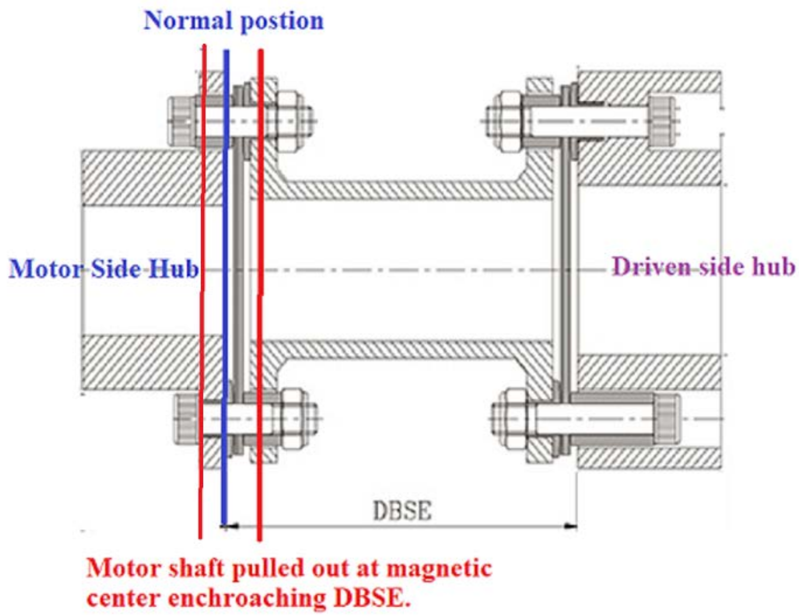


Fig 8 – Effect on DBSE due to electromagnetic pull during hunting

Magnetic center of an induction motor

Running a motor uncoupled will pull the motor rotor into a magnetic center, often marked on the shaft to ensure that the rotor is lined up on the magnetic center when it is coupled to the load. Axial misalignment of a coupling can occur if the coupling ends are not close enough when they are bolted together resulting in pulling the rotor out of magnetic center. When this happens, the rotor is constantly trying to pull itself through axial thrusting back to magnetic center and could result in uneven and or excessive bearing wear. Normally ,5th harmonic peak as an indicator of this magnetic center offset and axial movement which is single clean peak in the current spectrum. However, if axial thrusting is occurring the 5th harmonic will split into two smeared peaks.[16]
Magnetic center is a consideration only for horizontal sleeve bearing motors.
Typically, there is a means of showing the motor's magnetic center as determined by a factory or shop test run. The prudent commissioning / testing engineer will do a solo run with the motor in its final location to verify the real magnetic center.
The magnetic center is the driving parameter for alignment and coupling. Lacunae in considering magnetic center results in high axial loads. It is worth noting in conjunction the topic that all flexible couplings will have float, most of oil and gas industry use dry flexible coupling.
If the distance between shaft end (DBSE) is not within the specifications of the coupling, then the coupling must be modified or there will be no float or the distance between motor and driven must be adjusted (if possible). This implies that if magnetic center is lying A mm from mech center (outside motor). Then A mm is subtracted from DBSE and accordingly DBSE is set before startup of an equipment.

It is very important to check the magnetic center prior to assembly and alignment.
Another cause of the displacement of the magnetic center is the axial thrust caused by the cooling air fan for large motor. For a correct check, one should cover the suction of the fan.
Another important reason to have high axial vibration is due to End shield Resonance, identification, and mitigation of which at site takes a lot of time.
NEMA MG1-20.81 categorically tables the end play and rotor float for coupled sleeve bearing for horizontal machines which should be adhered at site.
Operating experience on horizontal sleeve bearing induction machines has shown that sufficient thrust to damage bearings may be transmitted to the induction machine through a flexible coupling. This can be avoided if the following limits are observed by the machine manufacturer and the assembler of the motor to driven equipment:
Most important requirements are to set so the motor won't run on a thrust shoulder continuously and is to allow it to run on magnetic center.
The standard procedure is

- Check the level of the motor which can confirm that shaft is levelled.

- Check and mark the magnetic center
- - Shift the stator or rotor to match physical center and the magnetic center
- Adjust the coupling gap leaving only the permitted float especially with respect to the thrust point

Case study 1

After a HI-HI vibration was reported on a heavy-duty motor NDE (non-drive end), restart attempts were unsuccessful with motor protection showing “stall protection” preventing the pump form running.

. The following damage was noted during dismantling and inspection.

- Damage to thrust section of DE and NDE bearing in the direction of the NDE (thrust from DE to NDE).
- Rotor shaft seized with NDE motor housing.

Although the measured DBSE of the pump and motor coupling hubs were measured and found to be out of tolerance, depending on motor shaft position (axial float) and at motor magnetic center it could not be concluded as root cause as damage was in NDE side. This means the rotor was not pulled towards DE side. It was found after talking to operating personnel that pump start up protocols were not followed during pump warm-up and was ramped -up to full capacity. The discoloration of the shaft suggested that once damaged, heat and increased friction continued to develop between the shaft and seal until eventually seizing once the pump shut down. Damage to the thrust portion of the bearings was a secondary factor caused by the shaft being pulled by the seal. The cause of overtemperature that has led to failure is a wrong axial alignment. Picture below shows damage axial face of DE bearing shell: which indicates that coupling of motor to pumps was done in wrong position and rotor shoulder was pushing against bearing thrust face. Normally, these type of bearing does not tackle axial load and shell faces have locating purpose only.

In addition, the OEM commented that the subject pump and motor assemblies did not complete a thermal cycle prior to being brought to full load. This was the major contributory factor in the seizing (sometimes named as “stall”).

Fig 9 – Rubbing NDE side of Electric Motor

Electric motor with Sleeve bearings [Perspective of axial movement]

The axial float of rotor is as standard ±8 mm from mechanical center. The running center locates within the float area and therefore operation is also allowed while motor is uncoupled, e.g., during test run. A pointer showing running center regarding end limits is available as standard. Continuous axial forces are not permitted and therefore limiting type coupling is needed to ensure rotor location within axial float range. On request to motor OEM, when special axial float is at least ±3 mm, magnetic running center can be adjusted within±2.4 mm from mechanical center (as option).

The prudent and best practice is- after marking the magnetic center and adjusting coupling side to make it as operating position of motor rotor. It is recommended that the top half of bearing housing is opened and optimum position (center position) is confirmed physically as well. As described above, similar failure cases of thrust bearing damages were reported due to not confirming the center position in the thrust bearing with respect to magnetic center.

Screw compressors

As one end of the rotors is at compressor discharge pressure and the other at compressor suction pressure an axial thrust is developed along the screw compressor rotors. As the pressure difference across the compressor is high, hence the thrust developed is correspondingly high. However, the high discharge pressure necessitates the rotor to casing axial clearance at the outlet end to be maintained at as small a value as possible to minimize gas leakage and thus ensure high compressor efficiency. This problem of possible rotor axial rub with casing is overcome in this type of compressor by incorporating a thrust balance piston to unload the angular contact

ball thrust bearings.
The oil is supplied at a higher pressure than the screw compressor discharge pressure, when the oil pressure drops, the compressor experiences axial thrust in the opposite direction leading to rubbing of rotors in the casing.[17],[18]
Normally, screw compressors don’t come with an installed axial thrust monitoring system, however even with axial protection, if issue on the lube oil supply to the compressor is not solved, we may still face the same result. With the axial trend in relation to the lube oil pressure trend, we can establish another protection i.e., trip on low oil pressure. Once we establish this, it will have good protection system in protecting the thrust bearing damage and rotor to end casing contact It is possible to trend the flow and an increase in the flow value will indicate a deteriorating condition of the balance piston and increased bearing clearances.[19],[20]

Case study 2
Axial rub during testing at OEM works not attributing to thrust bearing and rotor position is summarized in this case study. Repeated oil free screw compressor (for compressing propane) failures identified as due to excessive heat within the rotor chamber discharge end. Due to the excessive heat developed during the air test (as mandated in test procedure) caused local deflection of the casing which locally distorts and clashes with the rotors.
The casing distortion theory was confirmed as two axial probes fitted at same plane started drifting as the speed of compressor increased (Fig 10) caused due rise to high temperature at discharge end pocket.

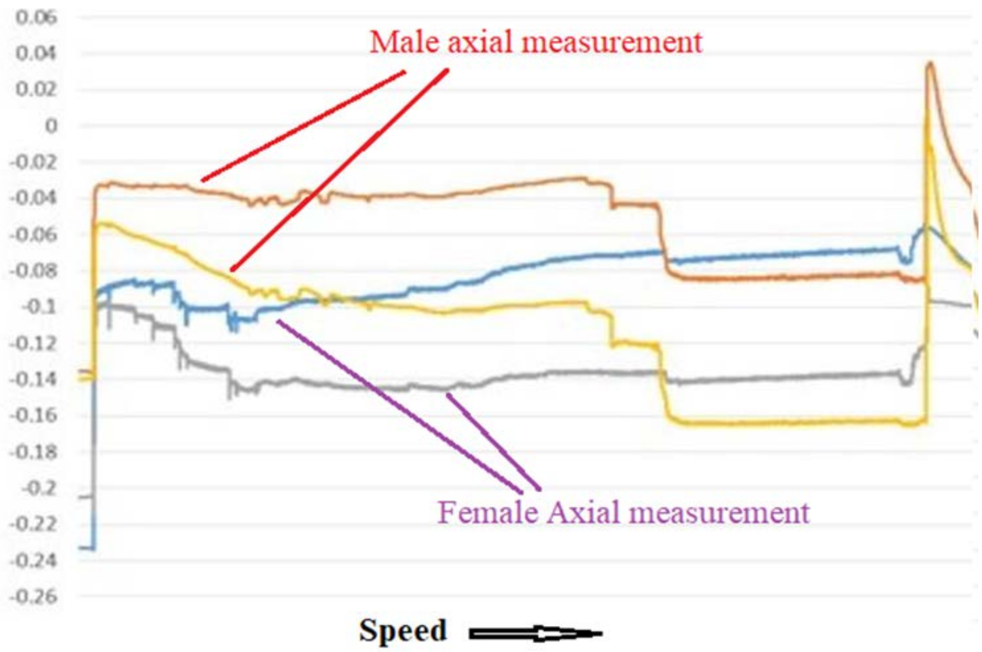


Fig 10 – Drift in axial measurements in screw compressor rotors

Integrally geared compressors with case study 3
In API 617 8th edition / API 672 5th edition compliant centrifugal blowers, axial thrust management is executed either by thrust bearing or by thrust collar inbuilt into HS shaft (pinion shaft). Thrust collar is applied as thrust bearing of each pinion shaft. In this system, the thrust force of the compressor is transmitted to the thrust bearing of LS shaft (bull gear), shaft by the pinion. [21],[22], [23]
The high-speed shaft of API 617 centrifugal blower compliant faced repeated failure of thrust collar with operating speed of 12000 RPM at OEM (Original equipment manufacturer) test bed.
During 1st full speed spin test, thrust collar of pinion shaft rubbed with bull gear, which was initially attributed to probable excitation of pinion lateral speed affecting axial movement by helical profile of mesh. In this arrangement there was no axial probe to protect machine with high axial thrust and this is common in single stage centrifugal blower/ compressor. With controlled start up on 2nd internal test thrust collar of pinion rubbed with bull gear again (however with lesser severity).

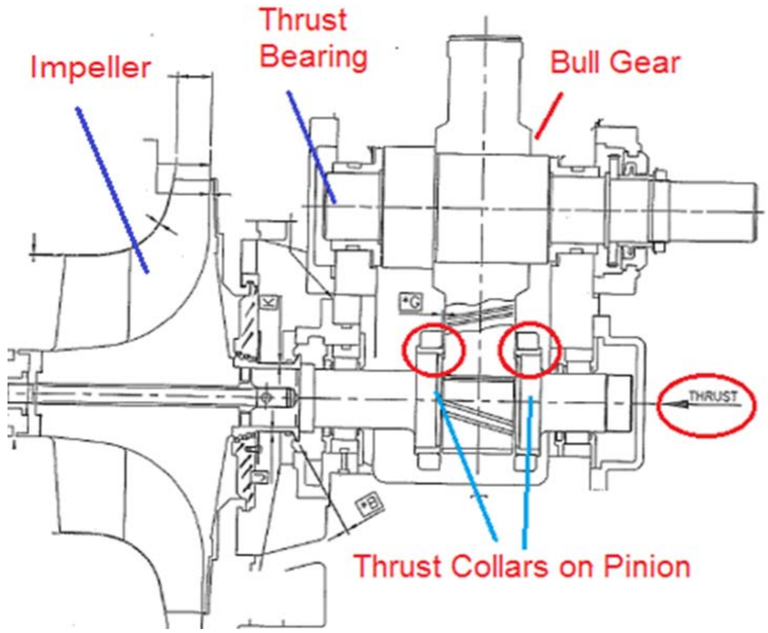
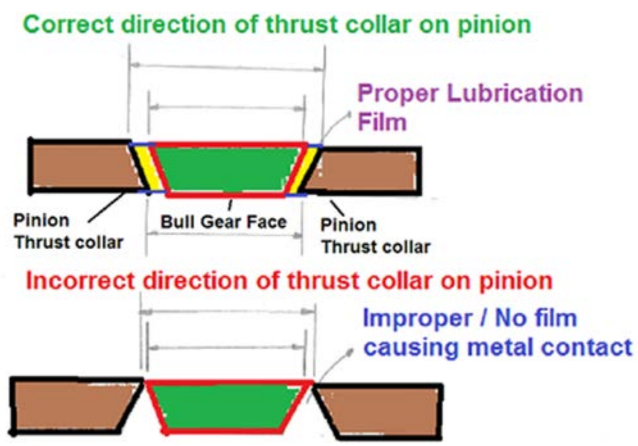


Fig 11 – Integrally geared compressor – Bull gear and pinion with thrust collars

Upon detailed investigation using Fish Bone chart including coupling pre stretch and LS (Low speed) and HS (High speed) shafts visual inspection, it was found that the taper of gear wheel and pinion thrust collar were reversed during manufacturing by gear unit manufacturer which was overlooked by OEM quality control department. The tapers are direction oriented to create an oil film wedge to enable lubrication between pinion and bull gear thrust collar faces. If the direction and orientation of taper is not matched, severe rubs can occur on thrust collars.



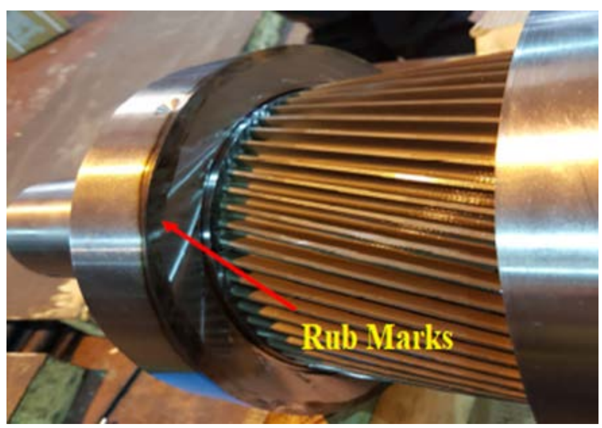


Fig 12– pinion thrust collar rubbing

Centrifugal Fans

Some cases were reported of high axial vibration in centrifugal fans not due to misalignment of pulleys or electric motor and fan but due to umbrella mode of impeller at operating frequency and /or flexible supports of bearing in axial direction. The fan frame may be comparatively weak in a particular direction which may cause a rocking effect on bearing housing causing different axial vibration readings on different places on a particular bearing.

Mitigation during design stage of machinery

During design stage all possible cases generating high axial thrusts to be jointly reviewed by driver and driven equipment supplier and suitable design options may be exercised as listed below -

• Complete absorption of the axial thrust via a thrust bearing (e. g. plain bearing, rolling element bearing) by checking axial thrust carrying capacity with respect to type of arrangement of angular contact bearings or preloading the taper roller bearings. A new type of

design known as Toroidal bearings are found to more effective than conventional spherical roller bearings in some installations.
• One of popular design now a days is to use offset pivot and LEG (leading edge groove) thrust bearing. They are known more thrust bearing capacity and maintaining better lubrication oil film during operation.[24],[25]

Fig 13- Leading Edge Groove (LEG) Thrust Bearing

• Axial thrust is generally high at the start and shut-off of the pump. During starting of the pump there is a sudden jerk of the whole impeller and shaft towards the driving end, as the liquid suddenly enters at impeller eye. Angular contact ball bearings can take an axial (thrust) load in only one direction, and therefore are almost always used in pairs in pumps to tackle axial thrust in both directions. In case of maloperation of a check valve (stuck in open condition), the reverse rotation of a high energy pump may cause axial thrust direction to reverse.
• Another type of bearing worth mentioning is toroidal roller bearing which accommodates misalignment and axial displacement within the bearing, without inducing internal axial loads with virtually no increase in friction. These types of bearings are normally used in non-locating bearing housing / Plummer blocks.[26]

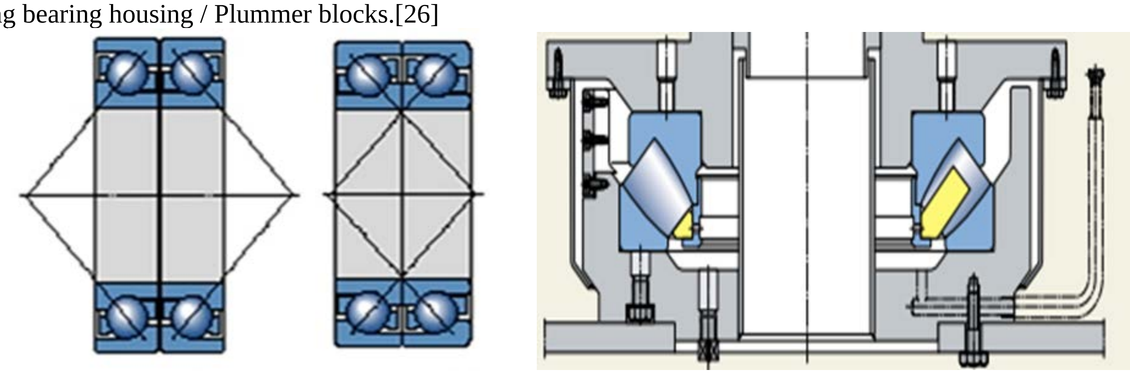


Fig 14-Arrangement of axial thrust bearings commonly used in pumps

• Balancing or reduction of the axial thrust on the individual impeller via balancing holes. This is the oldest method for balancing axial thrust and involves reducing the pressure in a chamber equipped with a throttling gap, usually down to the pressure level encountered at the impeller inlet. The pressure is balanced via balancing holes in the impeller.

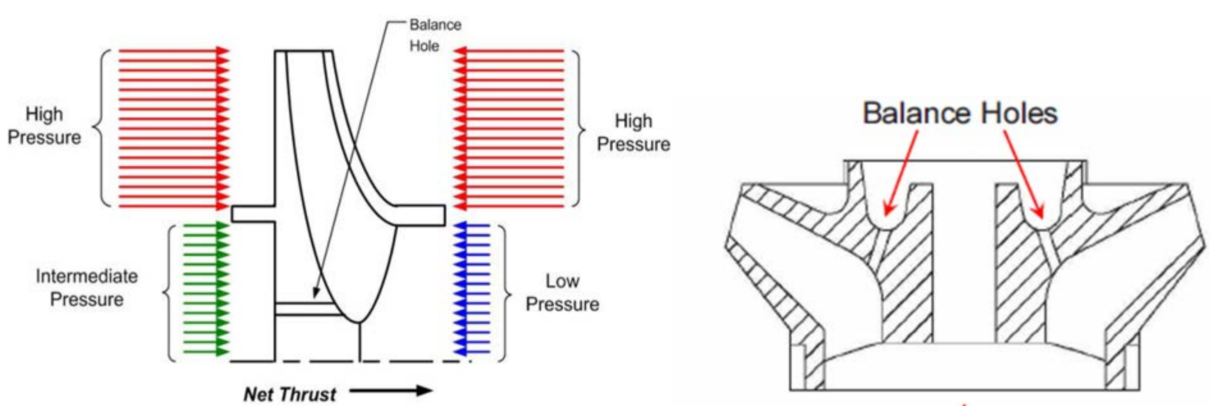


Fig 15- Thrust balancing with balancing holes

These balancing holes may lead to variations in axial thrust balancing because of varying inlet conditions like other arrangements. As a rule of thumb, from 10 to 25 per cent of the axial thrust always remains depending on the size of the holes. For ensuring a complete balance, the diameter of the wearing rings of the balancing chamber should be greater than that at the impeller eye. This requires OEM involvement during design stage.

- Balancing of the complete rotating assembly via a balancing device with automatic balancing (e. g. balance disc and balance disc seat) or partial balancing via a balance drum.
- Reduction at the individual impeller by back vanes / pump-out vanes (dynamic effect) [27]
- Using pump out vanes / annular ribs on back shroud. The back pump-out vanes simply act to break down that discharge pressure to a value between suction pressure and discharge pressure. By reducing the pressure on the back side of the impeller, the thrust force is reduced. This produces an extended life for the thrust bearing because of the reduced axial load.

The angular velocity has a dynamic influence on the magnitude of the axial thrust. An increase in angular velocity is mostly achieved by back vanes which are radially arranged on the rear side of the impeller. In this method, radial ribs are used on the back shroud to reduce the pressure in the space between the impeller, and the pump casing. Figure 16 shows the schematic of an impeller housed inside pump casing. The radial back vanes provided at the back shroud of the impellers acts as auxiliary impeller which restricts the entry of liquid into the clearances between impeller back shroud and casing cover. This results in a lower axial force.

This type of device is commonly used in single suction impellers of large fan / blower to lower down effective thrust generated during operation.

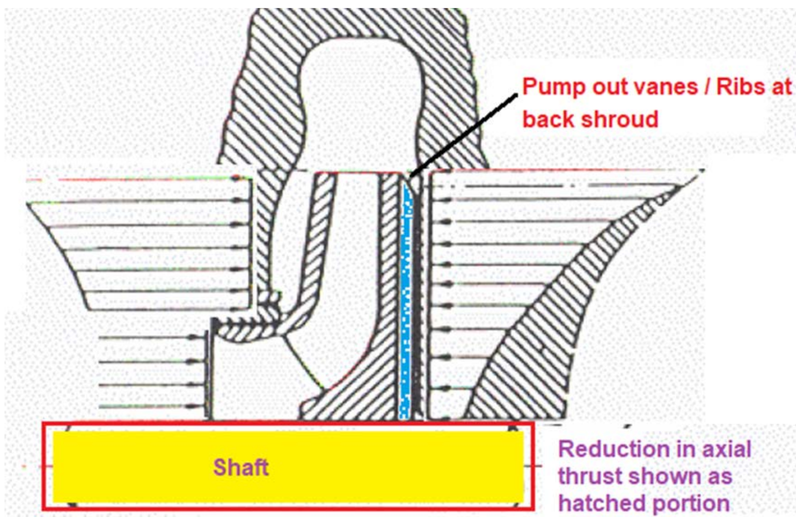


Fig 16– Thrust profile using pump out blades / vanes

- The number of impellers of multistage centrifugal pump are tried to be made even in number. The suction of half of the impeller is kept on one side and the suction of remainder half of impeller is kept from other side so that total axial thrust exerted on each side will neutralize each other. It is like concept of double suction impeller. The system is shown in figure 17.

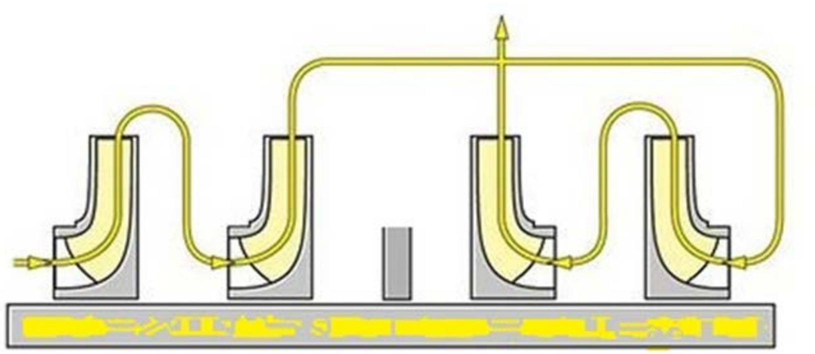


Fig 17 – Back-to-back arrangement of even numbers of Impellers to lower axial thrust courtesy KSB

The purpose of the balance drum is to limit the load on the thrust bearing, by taking a large differential pressure (close to the full differential across the machine) across it. The pressure on one side of the balance drum is total discharge pressure, the balance drum has a tight clearance labyrinth seal on the out diameter and a port on the other side that goes back to suction of the machine to equalize the pressure on the other side very close to suction pressure. This large differential pressure acts on the cross-sectional area of the balance drum to push the rotor in the opposite direction of normal thrust in the machine.[1]

The differential pressure is measured across the balance line. The key parameter is to have a baseline of the balance line ΔP and if this

ΔP increases over time, it indicates that the labyrinth seal has worn around the balance drum. If this happens, we may see an increase in axial position toward the active pads and a corresponding increase in pad
temperature in that direction. If the balance drum labyrinth is worn, this requires opening the case to fix, while replacing just the thrust bearing will not do anything. It is highly recommended to have a ΔP transmitter installed in order to have this parameter brought into the DCS and trended in real time

Axial thrust balancing with balancing devices – balancing drum, discs - The balancing device on centrifugal machines is designed to fully or partially compensate axial thrust generated by the pump rotor. Designs incorporating a balance drum require a thrust bearing to absorb the residual thrust.

When the machine is in operation, the balancing device requires a certain amount of balancing flow through the clearance gap between the balancing device's rotating and non-rotating parts. The balance flow is subjected to considerable throttling on its way through the gap This pressure loss results in an axial force acting upon the balancing device which counteracts the impeller's axial thrust and effects the required balancing. The balancing flow is the volume flow required to operate the balancing device of a centrifugal pump. Although it increases the clearance gap losses, it still constitutes an efficient and cost-saving design for axial thrust balancing. Labyrinth-type gap seals are fitted to minimize the high gap flow in drum balancing devices.[28]

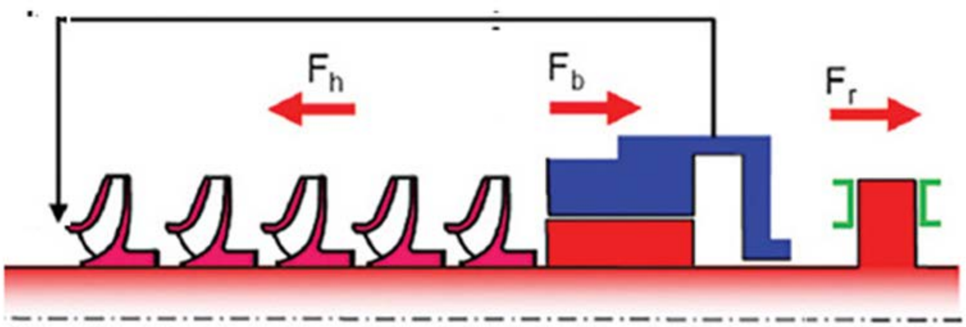


Fig 18- Thrust balancing device with balance line

The Effect of Wear Ring, balance drum and center bushing clearance

An axial thrust in two-stage or multistage pumps with back-to-back impellers can be caused by larger clearance at the center bushing which allows more leakage, resulting in higher thrust. The tooth on balance drum (as shown in fig 19) and its gap with stator part plays an important role in minimizing the axial thrust. At every stage of tooth with designed clearance, the pressure drops can be visualized as Δp and at end pressure becomes quite less and then an axial thrust is generated in opposite direction which lowers down the overall thrust as Fr = (Fh-Fb).

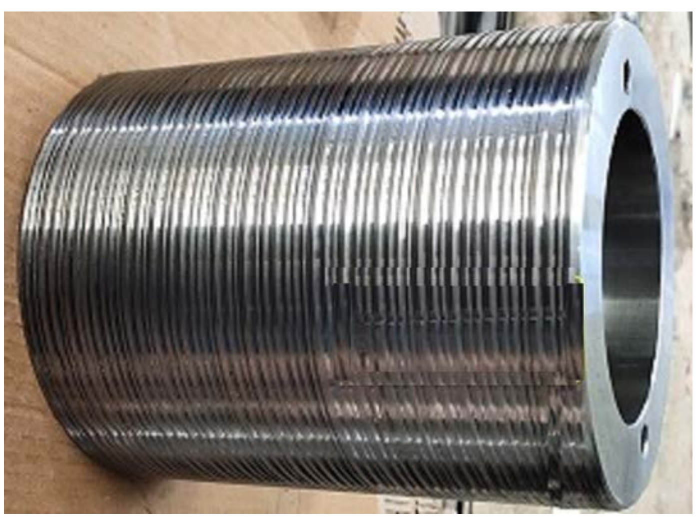


Fig 19– Toothed balance drum for multistage pump

Allowing the clearance to increase on the balance drum and center bushings cause short lives of thrust bearings. The difference between a balance disk and a balance drum is whether the primary pressure drop is taken across a radial clearance (drum) or an axial face (disk). Some pumps use a combination drum/disk that incorporates both radial and axial clearances.

Axial thrust management in Turbo expander re-compressor

In addition to supporting the weight of the rotor, bearings must overcome axial forces generated in the wheels due to the difference in

pressures between the front and back of each wheel. The axial force of one wheel is intended to be as close as possible to the other, and the forces are pointed in opposite directions. This is not always possible due to fluctuations in gas operating conditions; therefore, excursions in thrust load (in either direction) are to be expected.
Turboexpanders supplied in the oil and gas industry are commonly equipped with an automatic thrust balancing (ATB) system, also known as an automatic thrust equalization (ATE) system. This system controls a valve that connects the compressor inlet to the cavity behind the compressor wheel. By opening or closing, this valve can increase or decrease the pressure in this cavity, resulting in an increase or decrease in the thrust force in the direction of the compressor. Oil-bearing machines actuate the ATB valve by means of a piston-cylinder device. Tubing lines from the cylinder on either side of the cylinder are connected to the thrust bearings. This cylinder actuates the ATB valve when there is a difference in oil pressure between the two thrust bearings

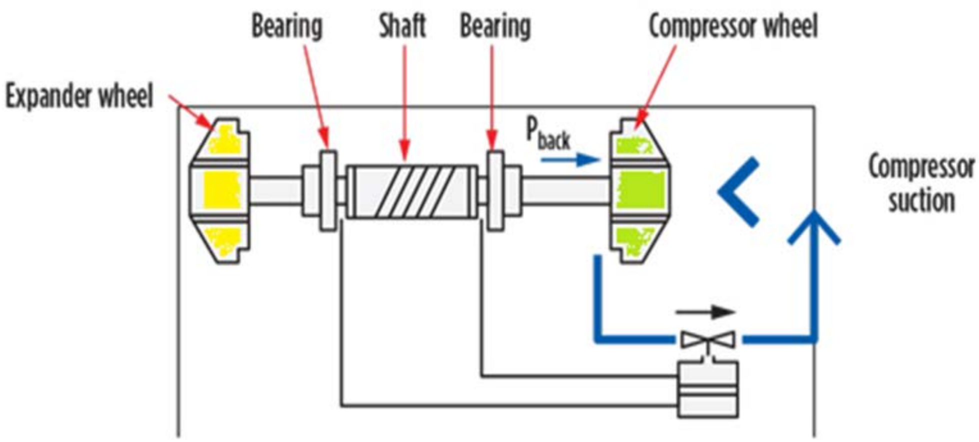


Fig 20- Schematic of a typical ATB system on an oil-bearing machine

Magnetic bearing-equipped machines use a valve actuator controlled by the main programmable logic controller (PLC). The PLC recognizes a thrust imbalance based on the difference in electric current in each thrust bearing and actuates the ATB valve to minimize the thrust difference.

Notes on Hydrodynamic thrust bearing

Based on API limitation on thrust bearing loading, stricter requirements are drafted in some operating companies as mandates. These requirements must be discussed much before by OEM with operating companies to highlight the possible issues of very high oil consumption and power loss. In addition, a larger thrust collar may affect lateral rotor-dynamic stability of centrifugal compressor.
While reviewing the thrust bearing capability an OEM should investigate the limits of surface speeds, units' specific loads Wu, temperature limit for babbitt metal, minimum film thickness and possible coking / varnishing property of lubricants.[29]

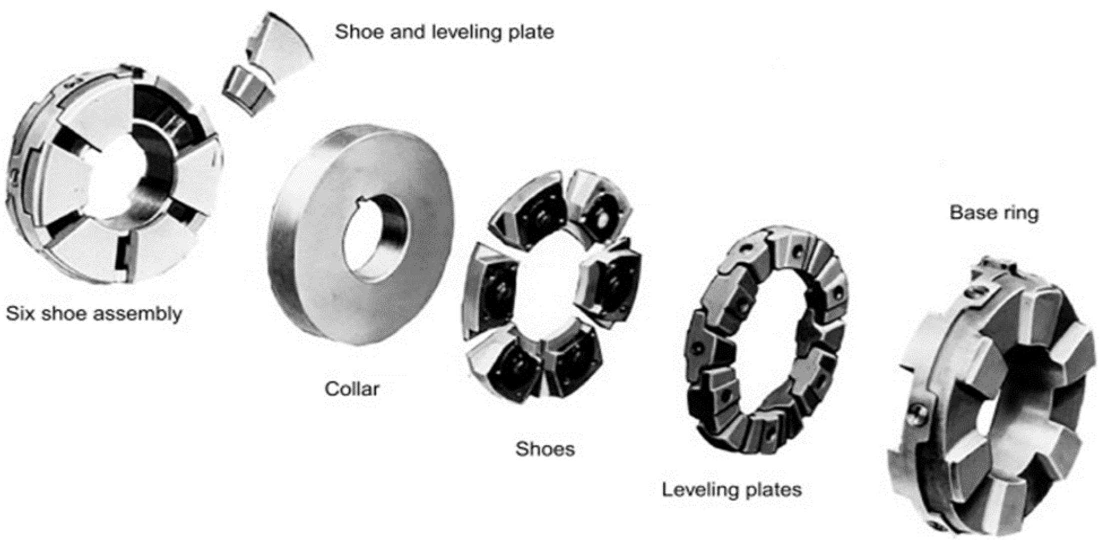


Fig 21 – Hydrodynamic thrust bearing   Courtesy – Kingsbury Inc.

The size of a thrust bearing and loading magnitude is often expressed as the specific or unit load given by the following:

$$Wu = Fa / NP \times AP \qquad (6)$$

where
- Fa is the axial load in N (lbf),
- NP is the number of pads and
- AP is the area of each pad in mm2 (in2).

Therefore, Wu is the load relative to the bearing area that is available to carry it and has the unit of MPa (psi).

To handle thrust load in both directions, there are usually two thrust bearings on both sides of the thrust collar. Based on normal movement direction one is named as active side and other is named as inactive side. Active and Inactive is the base in which direction of the shaft will be moved during the operation. The direction of thrust in a rotary machinery is always from HP (high pressure) side to LP (low pressure) side during rated operating condition.

For centrifugal compressor, the pressure of inlet gases is lower than that of outlet gases, thus the inlet side becomes LP and discharge side becomes HP. Therefore, thrust will act towards inlet (LP). Thus, it will require a thrust bearing on inlet side.

For gas turbine, the pressure of inlet gases is higher than that of outlet gases, thus the inlet side becomes HP and exhaust side becomes LP. Thus, thrust will act towards the exhaust side. With that understanding it will require a thrust bearing on outlet side. The inactive and active thrust bearing on any rotating shaft is located on adjacent faces of the thrust plate. Sometimes based on large fleet experiences, gas turbine inactive side bearing area is kept a bit smaller than active side.

It is to be noted the taper land thrust bearing cannot support any significant load if the rotational direction is reversed for a considerable period. In that case, central pivot bearings with copper backed pad should be used in applications where reverse rotation with appreciable load is expected.

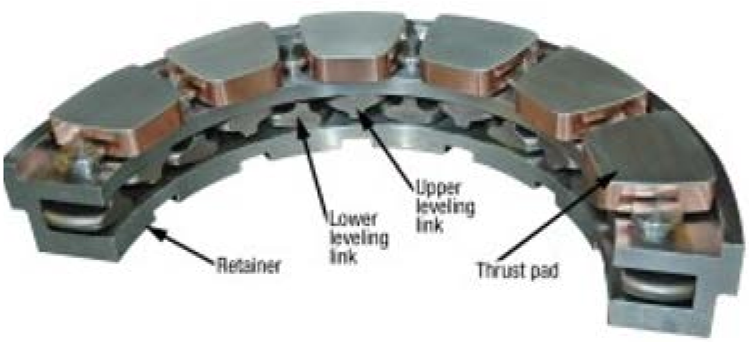


Fig 22- A self-leveling tilt pad thrust bearing

A thrust bearing has two operating limits: film thickness and temperature as basic principle of tribology. The film thickness limit is related to the surface roughness and bearing size. The temperature limit is related to oil oxidation. and/or babbitt creep and subsequent wiping out of white metal as shown in figure 23 below

Fig 23 –Localized burn mark and white metal damage on thrust pad

For compressors with an overall pressure difference lower than 70 bar, the current API rule (maximum permitted axial load lower than 50% of axial bearing capacity) is applied for the design point only at different clearances (nominal and twice maximum clearance).

For compressors with pressure difference higher than 70 bar, a comprehensive calculation shall be applied with consideration of the entire performance map (including choke, stability limit), clearances, roughness, and their combinations. The API rule of 50% capacity is still recommended. However, if this criterion cannot be fulfilled, a measurement of the thrust during a full load full speed (or Type 1/ type III) test should be performed to verify the calculations. The maximum calculated thrust, corrected by the measurements, should not exceed 67% (OEM can advise a more conservative value) of the bearing capacity.
It is worth noting that project specification may supersede the API mandate. In that case a clarification meeting must be conducted for joint understanding of purchaser, EPC contractor and OEM.

Using a coupling with limited end float
A motor shaft supported on sleeve bearings is allowed more axial movement which is attributable to the structural design of the bearings. In the following, we only refer to electric motors that are equipped with sleeve bearings, but do not have an additional axial bearing.
In this configuration, rotors can move axially by several millimeters. The coil has been wound in such a way that, in the center of the motor, the sum of all generated magnetic forces exclusively acts in the direction of rotation, and thereby contributes to maximum torque build-up.
As the rotor has more freedom to move axially, the drive line must be aligned such that the rotor remains at its magnetic center. The correct way to align the shaft is indicated by a mark provided between the shaft and the stationary motor housing.

Coupling with Limited end float
For limiting the end float or end play, a fixed bearing of another shaft is required which will axially locate the rotor shaft. The fixed bearing is arranged on the opposite side of the drive – at the non-drive end: This may, for instance, be the bearing of a gear shaft in a gearbox or the shaft bearing of a powerful compressor.
As mentioned in the beginning, a standard coupling allows for the compensation of axial misalignment of two shafts. However, this feature of the coupling prevents a rotor shaft supported in sleeve bearings from being properly aligned to the specified values. It is, therefore, necessary to use a coupling with limited end float.
For motors with sleeve bearings, we should always use limited end float couplings so that the magnetic center of the rotor can be aligned to a fixed bearing arranged opposite to the coupling, and that the rotor remains at its magnetic center during acceleration and deceleration of the driving machine. If this is not the case, undesirable magnetic forces would develop in the axial direction which can adversely affect the mechanical operating performance.

Monitoring and protection

Axial thrust position is the measurement of the relative position of the thrust collar to the hydrodynamic thrust bearing. The primary purpose of axial thrust position monitoring is to prevent an axial rub between the rotor and the stator. For thrust position, a perfectly smooth finish is not required as the system will average the DC signal and that's why API limit for axial runout is higher than radial slow roll runout.

Location of Axial position Transducer
Transducer location is very important for a proper thrust position monitoring system. The objective of the system is to measure actual thrust position. Thus, care needs to be taken that the system is not observing items such as thermal growth of the shaft.
It is recommended that the eddy current transducers used to monitor thrust position be located within two (2) shaft diameters of the thrust bearing. This assures that the eddy Current system is not adversely affected by shaft thermal growth. In some cases, this is not possible, and the engineer needs to be aware of the thermal growth expected and plan accordingly.
It is important to make a note that the eddy current transducers should observe an integral part of the shaft, as it possible for a nonintegral part to loosen creating a false reading. As an example, externally fitted thrust collars can become loose, showing normal thrust position readings while the machine is experiencing a situation of "about to catastrophic failure."

Mounting of axial position sensors
External mounting is the preferable method of mounting and can be completed when the end of the shaft is accessible through a cover plate or end plate. Care must be taken to ensure that the thrust bearing is on the same end of the machine so that the measurement will not be affected by thermal growth of the shaft.
Advantages of External Mounting:
- one of the Eddy Current Probe replacements while machine is running.
- Usually, good viewing surface
- Gap may be changed while machine is running one by one.

Disadvantages of External Mounting:
- May not be close to thrust bearing.

Internal mounting is accomplished by installing the eddy current transducers either directly through the thrust bearing or with a custom designed bracket viewing the thrust collar directly or a nearby shoulder on the shaft. Care must be taken in locating and tying down the transducer cable(s) to prevent damage. If an existing exit hole from the case does not exist, one hole will need to be drilled and tapped above the oil line.

Advantages of Internal Mounting:

- Usually, good viewing surface
- Close to thrust bearing.

Disadvantages of Internal Mounting:

- No access to transducer while machine is running.
- Cables must be tied down due to "windage".
- Transducer cable exits must be provided.
  - Care must be taken to avoid oil leakage.

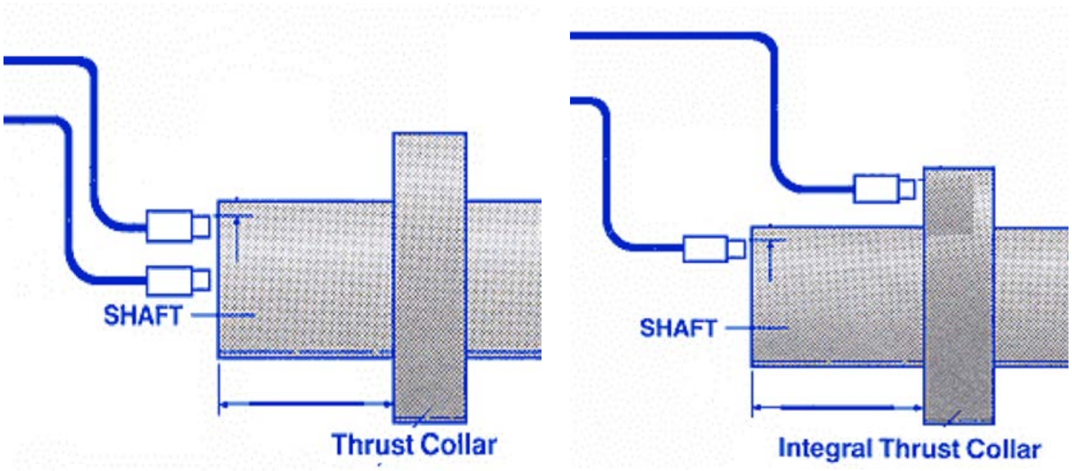


Fig 24 – Schematic of Axial position sensor arrangement courtesy GE Bently Nevada Tutorial / Orbit magazine

Number of transducers and thrust voting

When determining the number of transducers for monitoring thrust position on a machine, several factors should be considered. First, will the system be required to trip the machine if thrust failure is detected, or secondly what other means are available to verify thrust failure.

One of the rotating machinery instrumentation standards, API- 670 (American Petroleum Institute), specifies dual voting thrust position at each thrust bearing. This approach to thrust position measurement requires that two transducers be mounted at each thrust bearing. Their respective output signals are then compared to alarm and shutdown limits. Both output signals must exceed the shutdown limit before the machine is tripped. This method of thrust measurement increases the system's reliability and is recommended for shutdown operation.

A single Eddy Current transducer for thrust position measurement should only be used when the monitoring system is not required to shut down the machine, and other means are available to verify thrust failure.

It is very important to note that “Circuit OK and Fault Detection Circuits” are not to be used when monitoring thrust position, as they could affect the proper monitoring of this parameter. The reason for this is that a rapid thrust failure could cause fault detection circuits to operate inhibiting a valid shutdown alarm.

Role of temperature & pressure monitoring

Axial position and thrust bearing temperature measurements are perhaps two of the most important parameters to monitor on turbomachinery, as measuring a single variable at a single location may not ensure sufficient protection. Hence, both systems should be installed. To anticipate a thrust bearing failure due to overload soon enough to prevent damage to the thrust bearing itself, temperature sensors embedded in the bearing are required. With this method, an increasing load is seen as an increasing metal temperature, and it is a highly responsive indicator of an overload condition.

Upgrade from Steel-Backed Pads to Copper Alloy-Backed Pads to reduce thrust bearing pad temperature and associated fatigues is one of the important issues to be investigated during a detailed design phase.

By moving the pivot downstream circumferentially (from the leading edge), the pad is encouraged to tilt, opening the leading edge, and allowing more oil into the area between the pad and the thrust collar. This increase in pad flow cools the bearing.

Common design options to increase the load capacity of a thrust bearing that fits into the same envelope—so no other modifications are required. They include:

- Upgrade from steel-backed pads to copper alloy-backed pads
- Modify to an offset pivot design
- Upgrade to a direct lubricated design

Following parameters are advised to be monitor – a change by 20% of axial position backed by 20% increase in pad temperature must be investigated.

Balance line differential pressure may be monitored and investigated for coherence with pad temperature and axial position.

High Thrust Bearing Temperatures Due to Varnish
Incidents of rising pad temperatures and large differences in temperature readings over time are typical behaviors when varnish forms on thrust pads toward the trailing edge of rotation. [30] In a case study it was found that hydrodynamic bearings were not overloaded while a high temperature rise was reported. As identified upon inspection, the pad supports were in excellent condition and showed no signs of indentation or hard contact.
As heat is generated from the shearing of the oil film, the overheated varnish deposits start to "cook" and decompose.

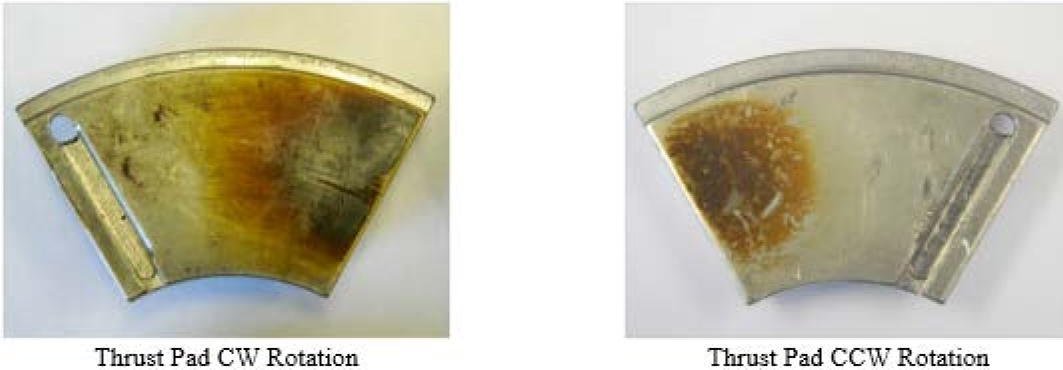


Fig 25 – Damage due to accumulation of varnish

There are cases of damage (washboard effect) thrust pads with stay current/ electromagnetic discharge taking through thrust bearing pads in view of close clearances (gap). The phenomena and tell-tale effect are shown in a generic picture as below

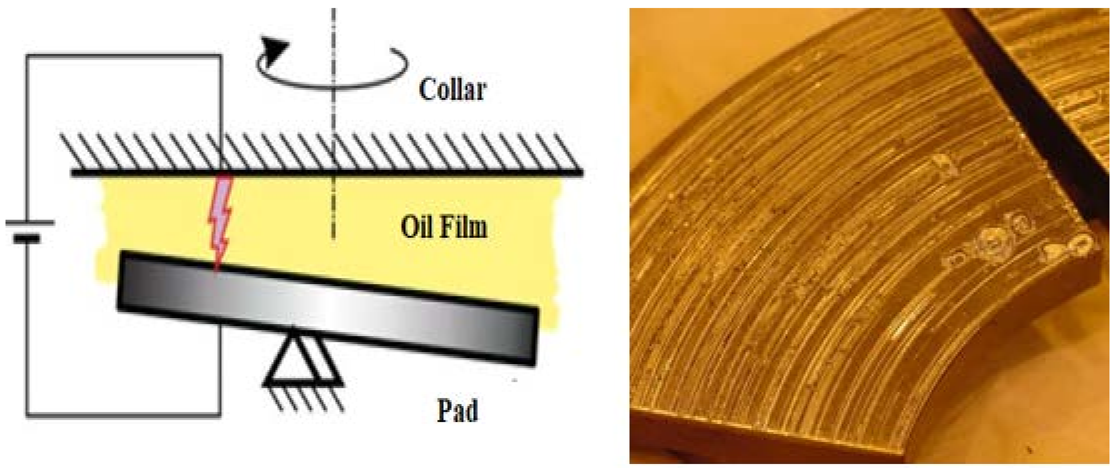


Fig 26- Stray current effect on white metal bearing

The symptoms at start of problem are frosting on Babbitt material of thrust bearing pads.
The most common root cause of magnetization of bearing housing due to welding carried on compressor balancing line / close proximity of bearing housing without local earthing of welding machine[31]. Another issue which may occur during prolonged operation of high energy machines with hydrodynamic thrust bearing where no correlation between the Thrust position and the Thrust bearing temperature values are found. Without a significant rise in temperature and normal operating condition, thrust position started showing high values than alarm limit. In that case after a deep study such as alignment reading, DBSE, complete bearing assembly to be inspected for possible pivot wear.

Mitigation during rotor assembly stage and at site adjustments

When a new pump is manufactured, during assembly of pump, a thrust clearance or end play is left. For example, on a typical horizontal single stage overhung pump (OH2 per API 610), the thrust ball bearings (normally angular contact ball bearing in pairs) hit the bearing housing on one side and hit the end cover of bearing housing on the other side, when the shaft is moved axially. The gap end play is based on the temperature of the fluid allowing for shaft expansion and too little gap will put too much load on the thrust bearing may cause it to overheat and lose lubrication on the rolling elements (anti-friction type) or score pad faces on thrust bearing, causing excessive wear. These gap specifications are suggested by the pump or bearing manufacturer. Too much gap will cause mechanical seal problems and other moving parts such as impeller rubbing stationary casing areas. Generally, we should set them "in the middle" of the float, as pumps can experience reverse thrusting if they operate far right on the curve. For multistage and vertical turbine pumps total measured float must be recorded and according to a washer is installed before installing bearings. The details of calculating thickness of such washers can be found in OEM installation and maintenance manual.

Thrust Bearing End Play (Axial Clearance) Basics

Critical turbomachinery rotor axial clearance diagram shows the key axial clearances and reference dimensions found on a typical rotor clearance drawing. It is worthwhile to review these references individually and to understand their significance when setting rotor axial position.

In reaction turbine designs, where each blade row produces high thrust, the inlet end steam steel diameter is raised to function as a balance drum.

OEM balance drum calculations should be audited during the pre-bid project phase to ensure proper thrust balance under all load and steam conditions.

For high-speed turbomachines, purpose of thrust bearing end play is to provide axial clearance between the thrust collar and the thrust bearing assemblies. The end play allows room for the formation of an oil film, misalignment, and thermal expansion of the bearing components. End play is the total distance the shaft can move between the two thrust bearings and is sometimes called float, thrust bearing clearance or axial clearance.[32]

Initially setting end play requires measuring the available space between the bearing housing and the thrust collar. For equalizing bearings, the pad height must be checked using a flat plate, placed on the bearing shoes, or placed babbitt face down on a flat surface. With this information, the thickness of the shim or filler plates required behind the bearings can be determined.

Once the shims or filler plate thicknesses have been determined the bearing components can be assembled into the bearing housings with the cover assembled. The end play should be verified by checking the axial movement of the shaft. The shaft should be moved in each direction and loaded with a force equal to between 50 to 150 psi bearing unit load. This is important to make sure the bearing shoes and leveling plates are set in their correct positions and that shims and filler plates are flat.

If using a dial indicator to measure shaft movement, it should be located as close as possible to the bearing. Ideally, axial proximity probes should be used to measure the gap along with dial indicators to confirm the probes setup. It is advisable keeping the end play on the smaller side rather than on the larger side.

The most common recommendation is that proximity probes used to monitor thrust position be placed within two (2) shaft diameters of the thrust bearing (for example 4on a 4-inch diameter shaft the probes should be mounted no further than 8 inches from the thrust collar). This assures that the proximity probe system is not adversely affected by shaft thermal growth. In some cases, this is not possible, and the analysts needs to be aware of the thermal growth expected and compensate the readings accordingly.

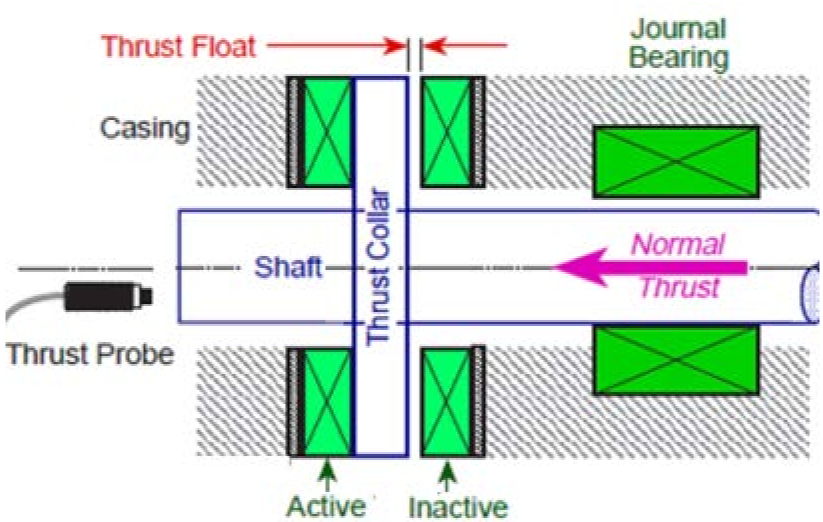


Fig 27 – Rotor axial setting with required thrust float

Where probes cannot be mounted directly monitoring the shaft, they can sometimes be mounted to observe the thrust collar or some other integral axial shaft surface. Once the probes are installed correctly, they must then be properly gapped. Extreme care must be taken when this step is performed. Improper gapping results in the permissible range of the thrust bearing falling outside of the probe's linear measurement range. To properly gap the probe, the shaft is mechanically barred against the active thrust shoe or other known position. The proximity probe can then be gapped, and the DC voltage documented. To ensure the proper placement of the probes, a worksheet incorporating the allowable shaft wear, float zone and probe parameters should be developed. This will help determine the optimum gap and that all alarms fall within the probe's measurement range.

When using the "Zero Active Shoe" method of gapping the eddy current transducer, the shaft is mechanically barred in the active or normal direction through the float zone until it is against the active thrust shoe. The eddy probe and monitoring System are then calibrated to Zero (0). This method of calibration provides more system range in the active direction, and when the machine is operating normally

with no wear of the thrust bearing the monitoring system will read "0".
When using the "Center of Float Zone" method, the shaft is mechanically barred to accurately measure the total float zone with the Eddy Probe or a dial indicator. The Eddy Current Transducer and monitoring system are then gapped and calibrated to read zero (0) when the thrust collar is in the center of the float zone. This method provides equal range in both active and inactive directions, and when the machine is running normally with no wear the monitoring system will display one half the float value (i.e.: +5 mils or 125 microns).
Method of Plus (+) Active (Normal) Direction -All thrust position monitoring systems are installed and calibrated so that wear on the active thrust bearing or normal direction produces a plus (+) or upscale reading. Minus (-) or down scale readings indicate motion towards the inactive thrust bearing. It is imperative to follow directive provided in OEM installation manual.

Note on Calibration of axial trust probe (axial float calibration)
Calibration is to be made with bearing housing / casing cup half fully assembled. Few times people calibrate with top half of housing opened. in case one must open the bearing housing again to (inspect bearing etc.), calibration has to be again performed.
It is experienced that axial float measurement varies with pressure put on crowbar used for pushing rotor to one end (typically in case of zero active shoe calibration method). Hence it is recommended not to use crowbar and instead use a fixture for pushing rotor at one end with screw / bolting arrangement (horizontal jacking).
Thrust bearing also fixes the rotor position with respect to diaphragm i.e., optimum desired position. Hence the thrust bearing position is to be carefully fixed so as to obtain desired optimum position of rotor (impellers) in the diaphragm. Otherwise, it may lead to undesired outcome like efficiency loss, vibrations / surge like phenomenon in centrifugal compressor.

A Short note on identification and Precaution to be taken as good engineering practice
Most practical vibration experience indicates that high axial vibration indicates coupling misalignment. Some use the rule that when the axial vibration reaches at least 50 percent of the radial vibration (horizontal or vertical), the source for the vibration is coupling misalignment. Coupling misalignment doesn't always create high axial vibration. There are many instances whereby coupling misalignment is the source of the problem, and axial vibration as low as only 20 to 30 percent of the radial vibration. It is supposed that the amount of axial vibration depends on the coupling type and the type of misalignment (parallel or angular). Resonance, particularly directional resonance or end shield resonance can magnify lower levels of axial vibration from other sources and then it looks like large vibration due to misalignment.
Precaution: When axial vibration is high, first check to see if there are symptoms for other sources for higher than usual axial vibration, such as:
1. Resonance of some part in the axial direction, such as a pedestal or pipe. When diaphragm (flexible disc-type) couplings are used, the disc or discs may be resonant in the axial direction.
2. The couple component of dynamic unbalance.
3. Bent shaft or resonant whirl.
4. Misaligned bearings (rare). Sometimes bearings can be misaligned by a frame twisted by improperly shimmed foot pads.
5. Motor with sleeve bearings, hunting for its magnetic center.
If the items in the list are not probable, then it can be fairly and safely assumed that the axial vibration is due to misalignment.
Current signature is an excellent method for identifying axial movement of the rotor. The current distortion caused by the axial movement causes a noticeable distortion at the fifth harmonic of the fundamental for 60 HZ that would be 300 HZ. The distortion causes a split peak at the fifth harmonic.
Confirmation of the axial movement is also a simple task. When the motor is de-energized, place a mark on the rotor shaft near the bearing housing. Start the motor. When the motor is running observe the mark that was placed on the shaft by monitoring with a strobe tachometer. The mark will be moving in and out if axial movement is occurring. If axial movement is present, the motor should be shutdown, uncoupled, magnetic center identified, followed by re-coupling and alignment based upon the correct location of magnetic center. A very slight movement, just a few millimeters is all that is necessary to cause this indication. Using a scribe pen while motor is running need extra safety precaution.

CONCLUSIONS

Failures due to high axial thrust or suddenly unbalanced thrust can lead to disastrous effects on mission critical rotating machines. The OEM, End User and EPC contractor should develop a common understanding right from start of technical evaluations in terms of design features and instrumentation. From operation point of view, it is very important to ensure cleanliness of operating fluids to prevent damage of balance drum labyrinths, partial choking of balance line and use a robust and tested surge protection system.
It is very important that a very experienced mill wright fitter is engaged along with rotating machinery engineer and electrical engineer from contractor side and OEM side.
With various phenomena causing high axial to thrust, accurate prediction of axial thrust due to fluidic interaction can be done using commercially available CFD codes such as 3D RANS equations with proper meshing for a prototype / mission critical equipment for un-spared and critical service [33],[34]. It is worth to have a proper analysis and design audit of such components, rather than spending more time on repairs, conducting failure analysis and rectification at site.
For un-spared installation, even a small change in axial shift is to be noticed, recorded and studied/ analyzed (it has been seen at few times operator allow this to be noticed till the values reach near alarm limits). In advent of AI based predictive analytics and digital

solution platform, even a small change in axial shift value is recorded and analyzed to prevent catastrophic damages. This approach can be used when a large number of similar machines are being supervised from a centralized location those are installed at remote area .

NOMENCLATURE

OEM = Original Equipment manufacturer
API = American Petroleum Institute
CFD = Computational Fluid Dynamics
DC = Direct Current
RANS = Reynolds averaged Navier stokes
AI = Artificial Intelligence
Kt = Axial Thrust coefficient (non-dimensional)
CW = Clockwise
CCW = Counterclockwise
DBSE = Distance between shaft ends
EPC = Engineering procurement and construction
ATB = Automatic Thrust balancing
ATE = automatic thrust equalization
G = Shear modulus of material of coupling / shaft
HP = Average power transmitted through coupling

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