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HYDRODYNAMIC ANALYSIS OF JOURNAL BEARINGS USING CFD AND FSI TECHNIQUES

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Abstract -Fluid mechanics journal bearings area unit vital power transmission elements for carrying high masses to extend power capability in varied machines. The precise operative parameters area unit essential for fluid mechanics journal bearings to style machines. Associate degree oil film pressure is one amongst the key operative parameter to describing the operative conditions in fluid mechanics journal bearings.

In this report, the CFD analysis, Static and Thermal analysis was done on liquid greased cylindrical bearing. so as to search out Pressure profile and temperature distribution within the bearing structure, satisfying the boundary conditions. To analyzed the geometric and operative parameters, like Journal Eccentricity quantitative relation (0.2, 0.4, 0.6 and 0.8.) and motion Speed (3000rpm) mistreatment Ansys.

In Journal bearings varied parameters were thought of for modeling like length and diameter (L/D) quantitative relation (0.8.) and eccentricity ratios (0.2, 0.4, 0.6 and 0.8.) and were sculptural in 3D modeling software system Pro/Engineer. The liquid stuffs SAE30 & forty oil was thought of as lubricant to see stress, strain and deformation of a bearing.

Journal bearing models was developed at speed of 3000 rate to review the interaction between the fluid and elastic behavior of the bearing. The speed is input for CFD analysis and also the temperature and pressure obtained from the CFD analysis was taken as input for thermal and static analysis. Computational fluid dynamics (CFD) and fluid thermal structural interaction was be tired Ansys.

Key Words: fluid mechanics bearing, fluid interaction technique, CFD.

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1.Introduction

The material pressure applied at intervals the bearing, in static analysis the deformation is not bearings square measure machine parts that enable parts to maneuver with association one another There ar 2 sorts of bearings, contact and non- contact. Contact kind bearings have mechanical contact between parts, that they embody lubricious, rolling, and flexural bearings. Mechanical contact means stiff- land ancient to the direction of motion ar planning to be terribly high, however wear or fatigue will limit their life.

Non-contact bearings embody externally controlled and mechanics fluid-film (liquid, air, mixed phase) and magnetic bearings. the dearth of mechanical contact suggests that static friction square measure attending to be eliminated, though viscous drag happens once fluids ar present; however, life square measure attending to be relating to infinite if the external power units required to manage them do not fail.

Non-Contact Bearings: Hydrodynamic

Mechanics bearings were altogether chance used for several years before human, World Health Organization was one all told the pioneers of fluid film analysis (as well as several completely totally different things!), introduced the thought of a step to assist build up the hydrodynamic wedge. In 1912, blue blood prince Kingsbury discovered that for large bearings, like those accustomed support turbines, it had been impractical to make a flat enough continuous thrust surface.

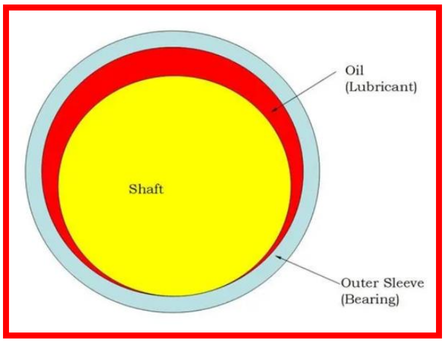


Fig 1.2: Non-contact bearing

He thus separated the surface into severally tiltable regions. The axis of tilt was positioned that the tilting pads would individuals mechanics wedge to resist thrust a whole bunch. His company, additionally as type of his original bearings, continues to be in business today! thus, mechanics bearings are generally used, e.g., in burning engines, thanks to their simplicity and intensely high load capability

2. Literature Review

Hydrodynamic journal bearings and varied, in general, fluid film bearings unit generally accustomed support rotors in stylish power plants (e.g., micro-GT cell hybrid systems) and crankshafts in automotive applications (e.g., rod bearing in combustion engines). Their rule is based on the interposition of a thin self controlled material material between the two opposing bearing surfaces that unit in relative motion. as degree example, forward that the relative radial clearance is one per thousandth (1/1000), since journal diameters of little GT bearings unit generally at intervals the vary 10–20mm, film thickness is relating to 10 microns [1].

Therefore, really top of the range standards and tolerances unit required to supply this class of bearings, that have to be compelled to be compelled to be ready-made with high accuracy processes. The comparison of experimental and numerical results has been used at intervals the at intervals the on instability of mechanics journal bearings. as associate degree example, by considering all of the nonlinear terms in equations and victimization in-house developed take a glance at rigs, the effect of every journal and bearing defects on stability has been studied [2].

A lot of recently, experimental investigations have targeted on bearing friction. In [3], associate innovative take a glance at rig has been used thus on decide friction with high activity accuracy. The authors have administered whole totally utterly totally different tests variable load and slippery speed every in mixed and full film lubrication regimes. they have obtained finishes up inside the type of a Stribeck curve whose trend is in good agreement with literature findings. The static and dynamic performances of very little or no mechanics journal bearings (i.e., in laboratory applications) have already been investigated in several works by suggests that of every numerical ways that} within which inside which and RK-type experimental take a glance at rigs.

For instance, T˚umaandBiloˇs [4]have studied the firmness of rotor vibrations throughout an impression by suggests that of full analysis and a RK4 Rotor Kit device; they have finished that journal motion at intervals the bearing is dominated by a mix of equations of motion, which may be remarked as linear and nonlinear models.

Meruane and Pascual[5] have found that nonlinear dynamic coefficients can powerfully powerfully their stiffness and damping. they have used a CFD simulation model in ADINA and a Rotor Kit 2000 experimental rig.

T˚uma et al. [6] have disbursed every experimental and theoretical investigations to assess the influence of external excitation son rotor behaviour and stability: per the obtained results, they have noted that kinematic excitation can can amplitudes of vibration but not the firmness limit.

3. Theoretical Calculations: D=journal diameter

D b=bearing diameter

C=clearance

Min=minimum film thickness

h max=maximum film thickness

N= journal speed

L = length of journal

Ŋ = oil body

V m=peripheral rate of journal

ϵ = Eccentricity relation

W = safe operational load

ϕ = angle angle

ΔT = temperature rise

Ψ = relative clearance

Analytical calculation is created by exploitation vogue info book; we've got a bent to tend to assemble the planning info for bearing (given in Table 1) then we've got a bent to tend to used totally fully completely different formulas [6] for scheming safe most pressure, safe operative load and temperature rise.

Bearing pressure

General electrical industrial plant formula:- Pa = half-dozen.2*105 *

We know

V m = π D N/60

V m = 21.99 m/s

Then

Pa = 1.74*106 N/m2 or one.74 MPa [Average pressure]

Victor Tatarinoff‵s equation:-

P = 13.5*

P = 4.76MPa [safe most pressure]

H.F. Moore‟s equation for important pressure:- laptop = seven.23 *105

Pc = 3.4MPa

Safe oil film thickness

h min = 2.37*10-5 *Vm0.4 *A0.2

h min = 0.05067mm or fifty.67μm

Eccentricity

Now we all know that

h min = Cr(1-E)

ϵ = therefore eccentricity e = 2*10-4 m or 200μm

Attitude angle

ф = Tan-1

ф = 30.50

Ѳmax at Pmax = Cos-1 = 1620

Safe operational load

Victor Tatarinoff‟s equation for safe operational load

Hence W = 351.60 KN

Temperature rise

The temperature rise of the material film is because of heat generated that is to be frenzied by the material, are often found

Where

Q is [Volume of film that's *(D a pair of - d2)* length] per second

So,

Q = 2.31*10-5 m3/s

P = 10826.87N/m2

250N load of shaft for bearing portion as we've not set any load at the start i.e.

[π /4*D2*L*density (898.9 kg/m3)] = 25kg approx. Or 250N

ΔT = = a hundred and twenty.08967 in degree Kelvin

4. Structural& Modal Analysis Of Cylindrical Journal Bearing Eccentricity-0.2

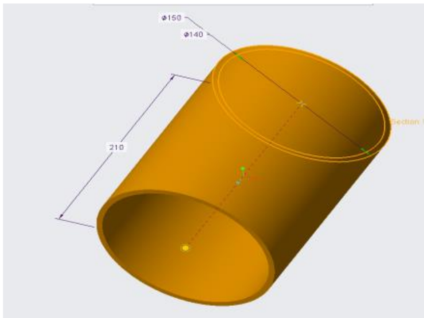


Fig4.1 : 3DModelling and Length For Eccentricity 0.2

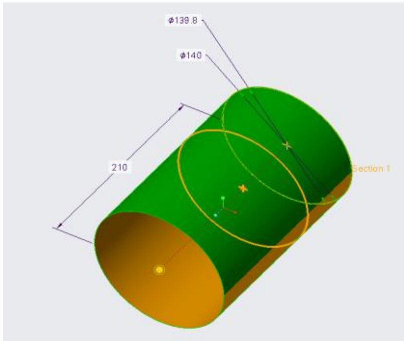


Fig 4.2: 3DModelling for Lubrication For Eccentricity 0.2

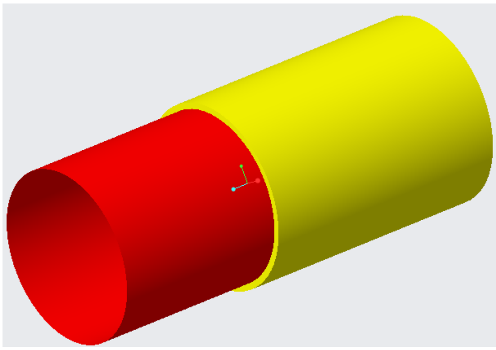
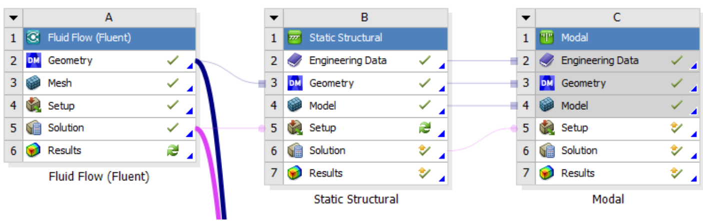


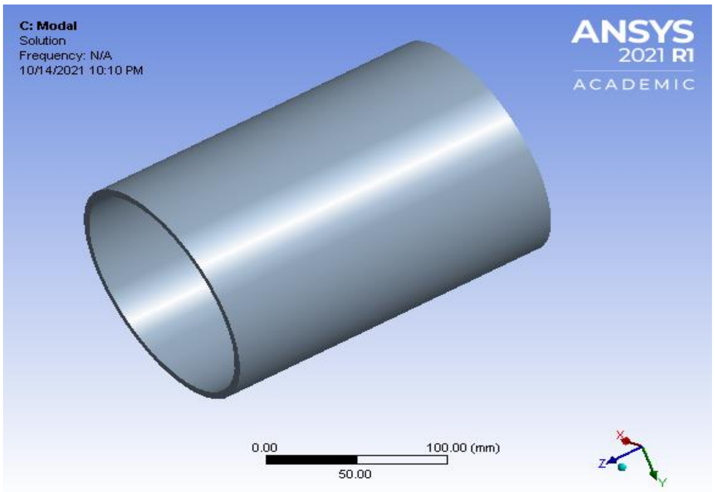
Fig 4.3 : Assembly of Bearing and Lubrication For Eccentricity 0.2

Lubricant: SAE-30

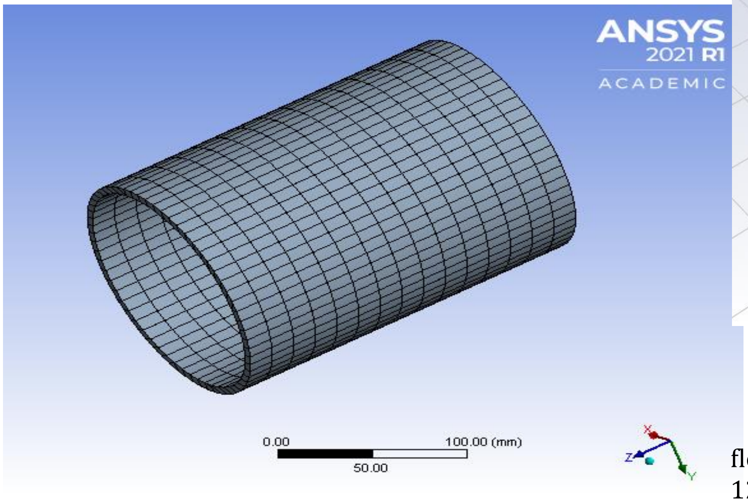
Saved from Model as .iges format



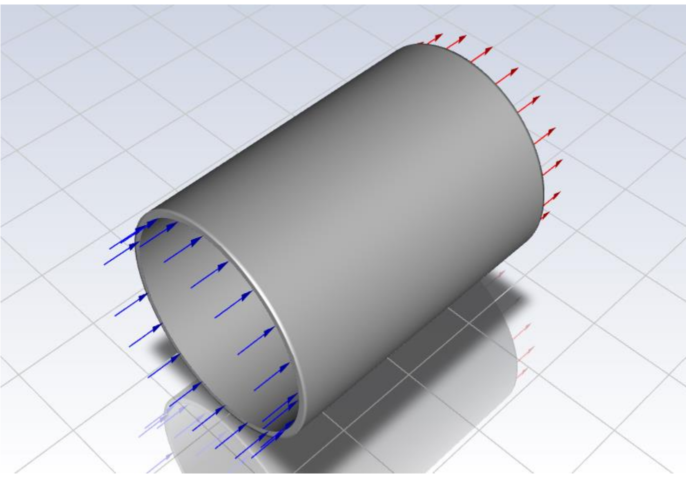
Import model



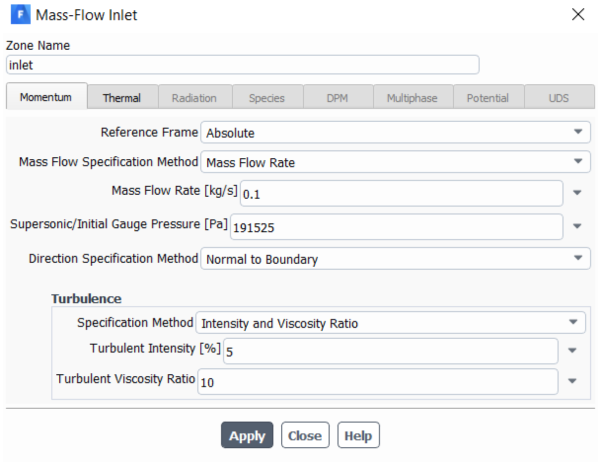
Meshed model



Inlet & outlet



Mass flow rate



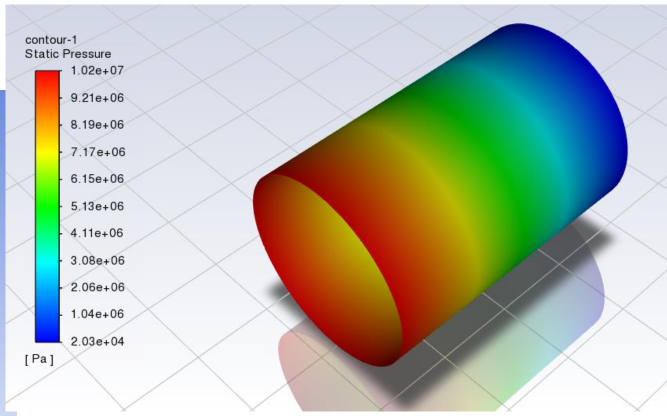


Fig4.4: Pressure

The lubricant (SAE30) is entering into certain mass flow rate (0.1 kg/sec) along with pressure of 1bar and also 120^{0}C temperature. The lubricant may change its properties due to surface friction and properties of the material. The maximum pressure (1.02e+7 Pa) is obtained at the entry of the journal bearing and minimum (2.03e+4 Pa) at exit.

STRUCTURAL & MODAL ANALYSIS OF

CYLINDRICAL JOURNAL BEARING

ESEENTRICITY-0.2

SAE-30

Material properties

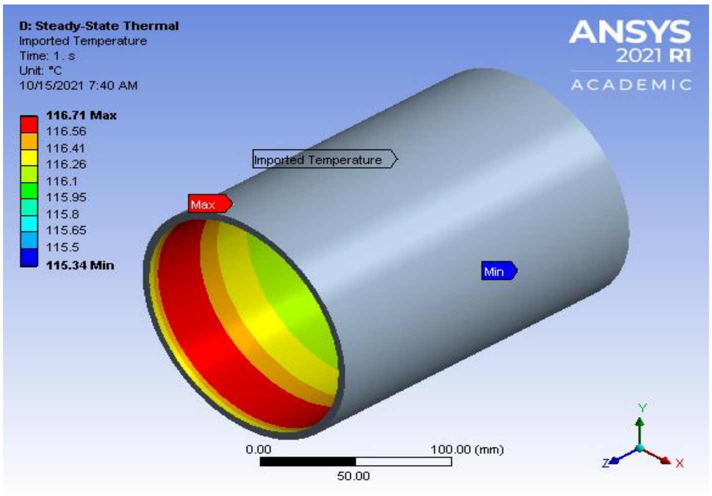


Fig4.5: Imported Pressure

Above figure represents the imported pressure from CFD Simulation. The pressure is contact with inner surface of the bearing.

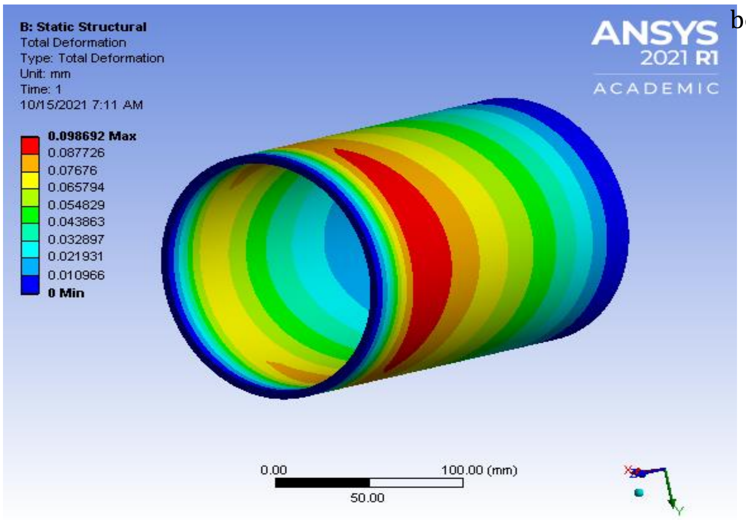


Fig4.6: Deformation

The imported pressure is contact with inner surface of the bearing then there may get deformation in bearings. The maximum deformation is 0.098692 mm and minimum is 0mm

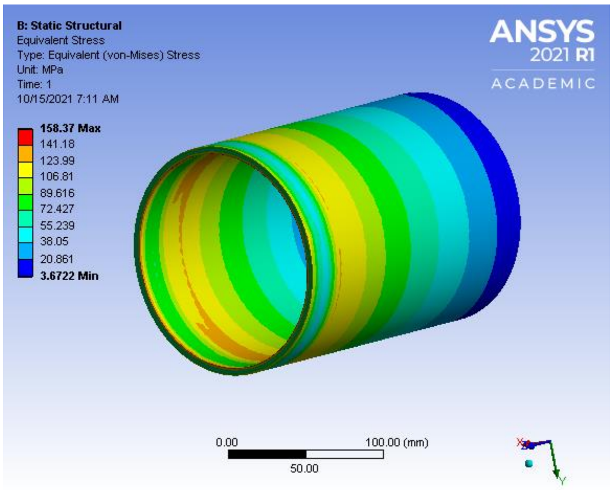


Fig4.7: Stress

The imported pressure is applied on the inner surface of the bearing therefore the internal stresses will develop. The maximum stress is 158.37MPa at entry of the lubricant and therefore gradually reduces at exit of the bearing (minimum stress is 3.6722MPa)

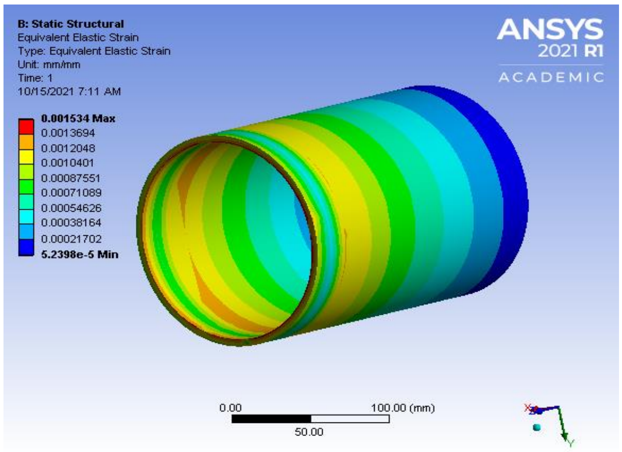


Fig4.8: Strain

The above figure represents the strain (change in length to original length). After passing the lubricant, the maximum strain is 0.001534 and minimum is 5.2398e-5.

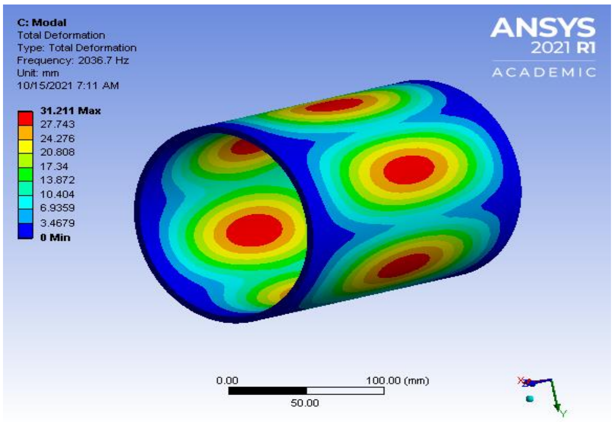


Fig4.9: Mode1

The above figure represents the mode1, the frequencies develops under the pressure is 2036.7Hz and maximum deformation due to frequency is 31.211mm and minimum is 0mm

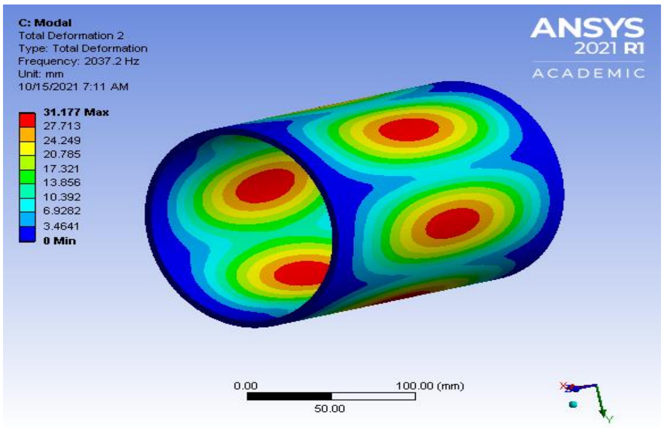


Fig4.10: Mode2

The above figure represents the mode2, the frequencies develops under the pressure is 2037.2Hz and maximum deformation due to frequency is 31.177mm and minimum is 0mm

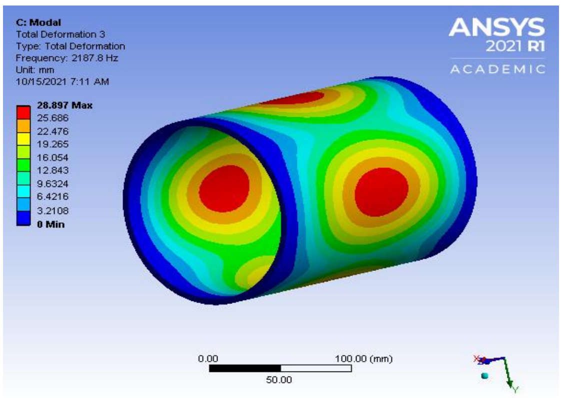
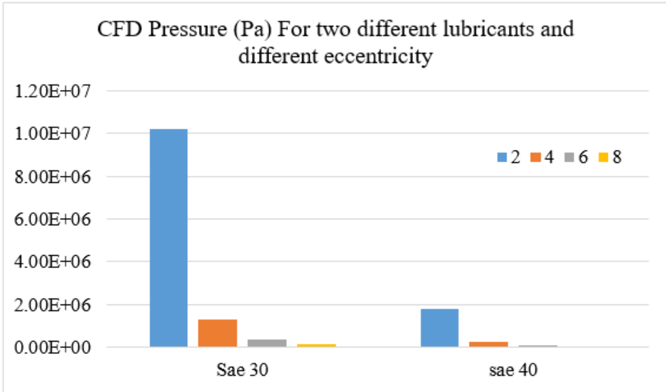


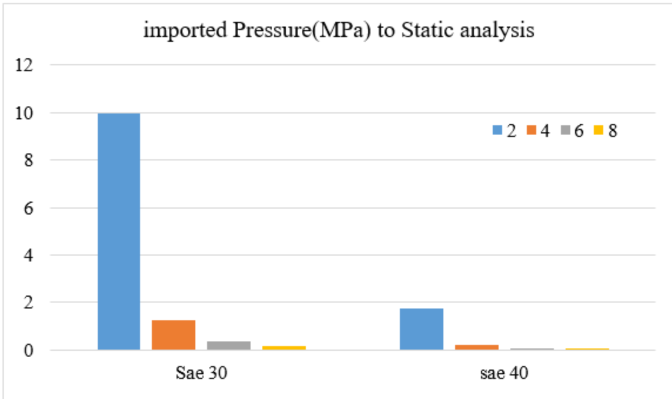
Fig4.11 Mode3

The above figure represents the mode3, the frequencies develops under the pressure is 2187.8Hz and maximum deformation due to frequency is 28.897mm and minimum is 0mm

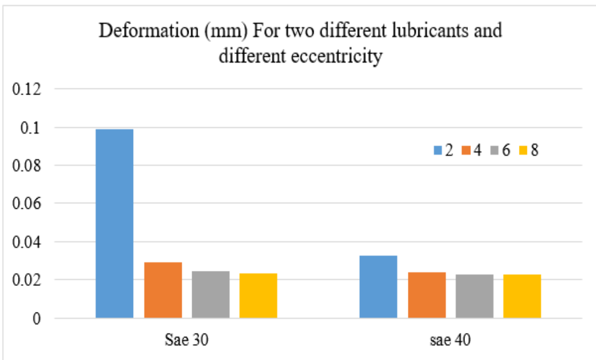
5. Results and discussions



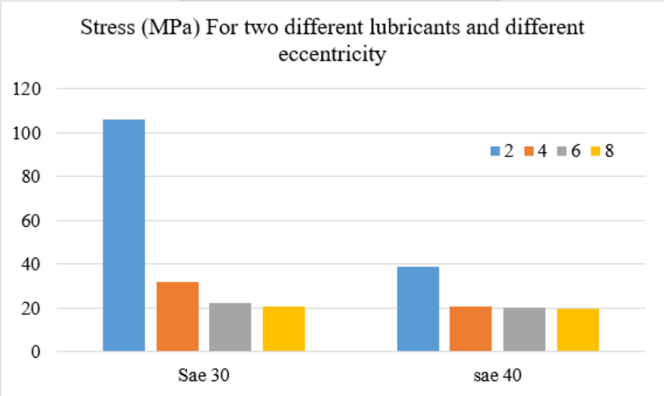
In above graph represents that CFD Pressure for two different lubricants for different eccentricities. In this, SAE30 have high pressure when compared with SAE40 oil in all eccentricities.



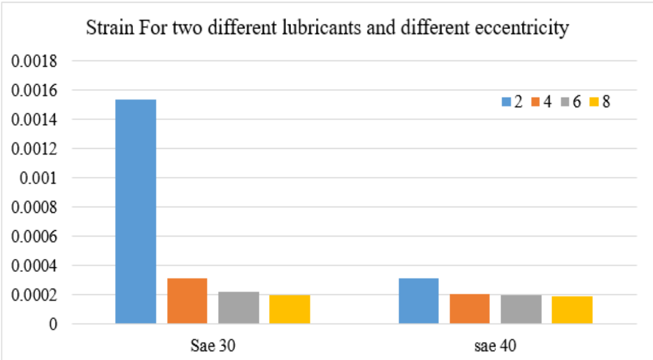
In higher than graph, the CFD pressure is foreign to the static analysis



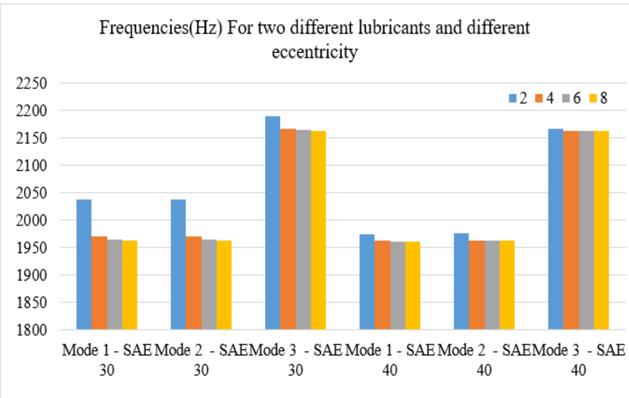
The material pressure applied within the bearing,in static analysis the deformation is noted for 2 lubricants and totally different eccentricities. In this, less deformation is determined once the material is SAE40 oil were used.



The lubricant pressure applied inside of the bearing, static analysis the strain is noted for 2 lubricants and totally different eccentricities. In this, less stress is determined once the material is SAE40 oil were used.



The lubricant pressure applied inside of the bearing,



The material pressure applied within the bearing, in static analysis the deformation is noted for 2 lubricants and totally different eccentricities. In this, less deformation is determined once the material is SAE40 oil were used.

Conclusions:

Journal bearing models will be developed for speed of 3000 rpm to study the interaction between the fluid and elastic behavior of the bearing. The speed is the input for CFD analysis and the temperature and pressure obtained from the CFD analysis will be taken as input for thermal and static analysis.The observations are followed by using ANSYS In CFD, the maximum pressure (1.02e+7pa) is obtained for sae30 oil when eccentricity is at 2 and minimum pressure (2.98e+4 Pa) is obtained for eccentricity – 8 for sae40 oil. In Static analysis, the maximum Deformation (0.09869mm) is obtained for sae30 oil when eccentricity is at 2 and minimum (0.0228mm) is obtained for eccentricity – 8 for sae40 oil.

The maximum stress (106MPa) is obtained for sae30 oil when eccentricity is at 2 and minimum (19.738MPa) is obtained for eccentricity – 8 for sae40 oil. The maximum strain (0.001534) is obtained for sae30 oil when eccentricity is at 2 and minimum (0.00019) is obtained for eccentricity – 8 for sae40 oil. The frequency of journal bearing under load was observed as maximum at mode3 at eccentricity 8 when sae 30 oil utilized and minimum for sae40 oil. From all above observation, in this study under 3000rpm the SAE40 OIL is preferred

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