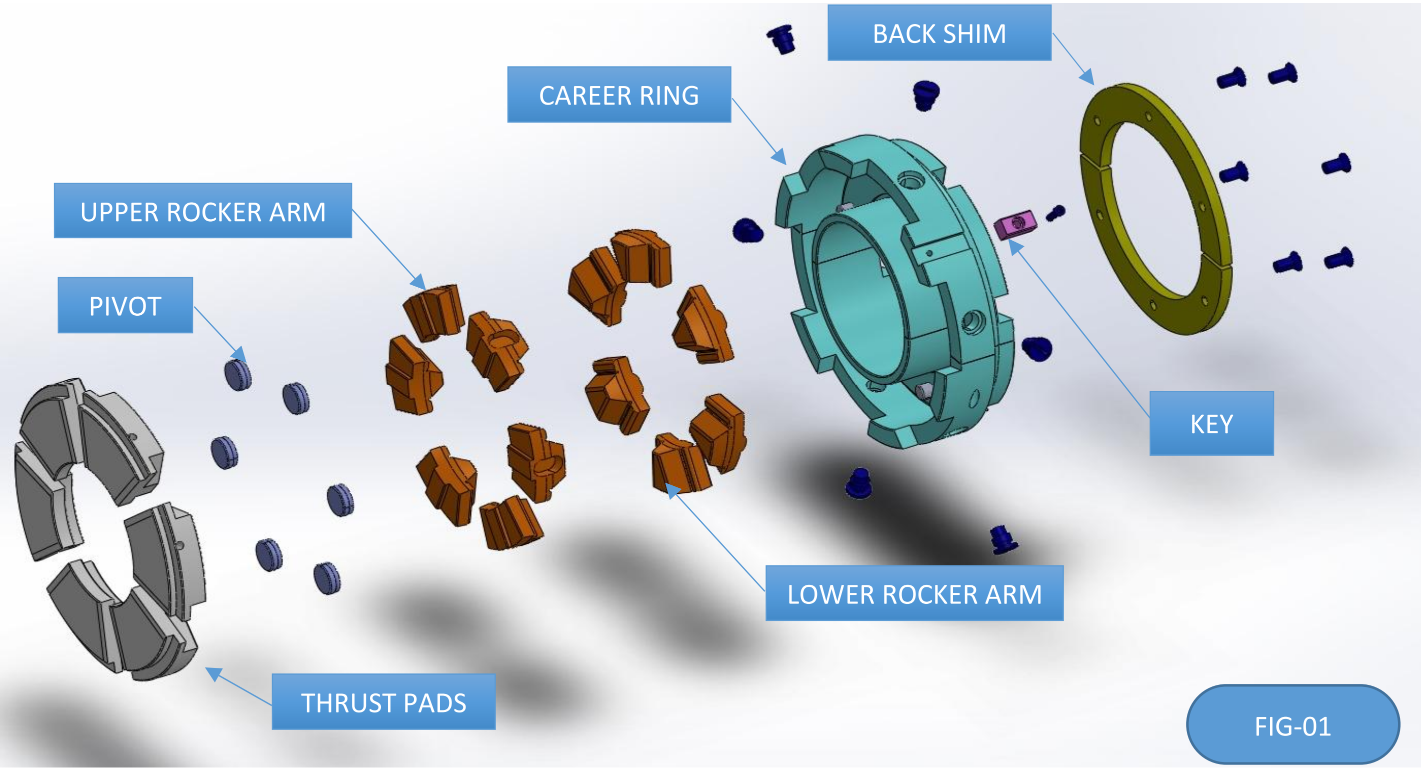
PRESSURE EQUALIZING THRUST BEARING ASSEMBLY – “SS” SERIES.
(DOUBLE BEARING WITHOUT COLLAR)
Ø100 TO Ø685 MM.

SUNTECH®
ENGINEERING CORPORATION
INNOVATION FOR THE NEXT

www.suntechbearings.com



The concept for the equalized thrust bearing is identical to the non-equalized thrust bearing except a series of Leveling plates or Rocker arm are used to equalize the axial load over the face of the bearing. This is required in high performance turbomachinery since mechanical and thermal effects can misalign the thrust bearing to the thrust collar. With the bearings discussed , any misalignment of the bearing to the thrust collar will act to decrease the load capacity of the bearing. This is because one sector of the bearing will be loaded higher than others resulting in a "weak link" that would be the first things to fail, quickly followed by the rest of the bearing. The bearing drawing in above FIG-01 defies some terminology used with thrust bearings of this Pressure Equalizing Thrust Bearing Assembly.

The SUNTECH® Thrust Bearing (Pressure Equalizing) offers many operating advantages, including:

- Excellent shock absorbing capacity.
- Superior damping characteristics.
- Life span equal to that of the machine.
- Versatility in application.
- Performance monitoring capability able to safely carry the highest axial loads at high speeds in turbomachinery.
- Best able to accommodate misalignment or deflection in supporting structure.
- Able to include special features, materials and instrumentation.

- Equal load capability for both directions of rotation with central pivot pads.
- Normally supplied split in two halves but offers the option of a one-piece ring.
- Pads are identical and interchangeable whether offset, either hand, or central pivot.
- Oil flow is controlled within the bearing using normal supply pressures.
- The widest range of optional sizes, materials and features.

1. BASIC ELEMENTS OF SUNTECH® MAKE PRESSURE EQUALIZING THRUST BEARING.

1.1 CAREER RING.

Career Ring is basically main base frame of Pressure Equalizing Thrust Bearing Assembly, the Career Ring holds the Leveling Plates / Rocker Arm in their operating positions. An oil inlet hole, at the back of the Career Ring, distributes oil to radial slots in the ring's back face. This assembly uses the equalizing principle with the help of Rocker Arms which distribute the load equally over the bearing pads and transmit the load to the bearing housing.(FIG-02)

FIG-02

1.2 LEVELING PLATE / ROCKER ARM.

The equalizing mechanism of the SUNTECH® Pressure Equalizing Thrust Bearing enable each Thrust Pad to transmit an equal amount of the total thrust load. The special geometry of Rocker Arm reduces the chance of unequal load distribution over the entire Thrust Pads , combined with the spherical pad pivot support, also ensure that the thrust bearing face becomes perfectly aligned with the rotating thrust collar.(FIG-03)

FIG-03

FIG-04

1.3 THRUST PADS.

SUNTECH® Make Tilting Pads are designed to transfer maximum axial loads from rotating shafts with minimum power loss, while simplifying installation and maintenance. Thrust pads are the main portion of the assembly having Babbitt metal on its surface thickness of thrust pads and Babbitt have been selected to reduce the amount of thermal and elastic deformation. For a centrally pivoted thrust pad, a certain amount of thermal or elastic crowning is necessary for the thrust shoe to carry load, whereas excessive crowning reduces load-carrying capacity. Therefore, we have carefully optimized our designs so that the elastic or thermal crowning of a SUNTECH thrust pads comes with maximum load-carrying capacity.(FIG-04)

1.4 THRUST PADS PIVOT.

The Spherical Pivot on SUNTECH® Thrust Pads allows the pads to tilt not only in the direction of rotation but also in the radial direction, which enable to handle some misalignment between the thrust bearing face and the operating shaft flange. The ability of a pads to pivot as well as to tilt, increases load-carrying capability at all shaft speeds. The thrust pads pivot, or shoe support, and the upper leveling plate where the thrust shoe pivot makes contact, are both made of high carbon steel, heat treated to 450 to 550 HB, to prevent damage to the pivot contact areas.(FIG-05)

FIG-05

1.5 CAREER RING AND ROCKER ASSEMBLY.

Career Ring and Rocker Arms together make a proper arrangement which enables Pressure Equalizing Thrust Bearing Assembly to provide proper support as well as proper movement of Thrust Pads for alignment. This assembles basically a simple but significant arrangements of Career ring plus Lower rocker arm and upper rocker arm. Lower rocker arms are flexibly fixed with dowel pin attached in career ring. Upper rocker arms are flexibly fixed with locating screw attached in Career Ring.(FIG-06)

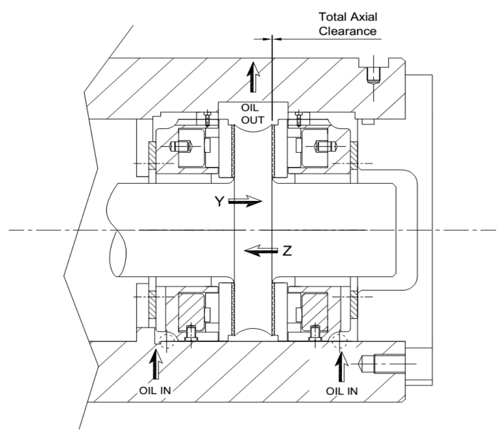
FIG-06

1.6 LUBRICATION.

For SUNTECH® Pressure Equalizing Thrust Bearings to operate safely and efficiently, continuous self-renewing oil films must be present between the thrust pads and collar. The oil supplied to the bearing should be cooled and filtered, so that the average particle size is less than the bearing's minimum film thickness. The typical oil flow path is shown below.(FIG-07)

1.7 AXIAL CLEARANCE.

To understand axial clearance, refer to the diagram of a double thrust bearing (one on each side of the collar). Axial clearance is the maximum distance between the thrust pad surface and shaft flange .That is the maximum gap in which thrust collar can be moved between the bearings during installation while applying load to either bearing. Axial clearance isn't an exact dimension. The shaft's maximum end play is limited to the smallest clearance between the stationary and rotating machine elements, while the shaft's minimum end play must be sufficient to prevent excessive power loss in the unloaded thrust bearing.



THRUST BEARING ASSEMBLY FLOODED LUBRICATION

FIG-07

1.8 BEARING GEOMETRY & DIMENSIONS.

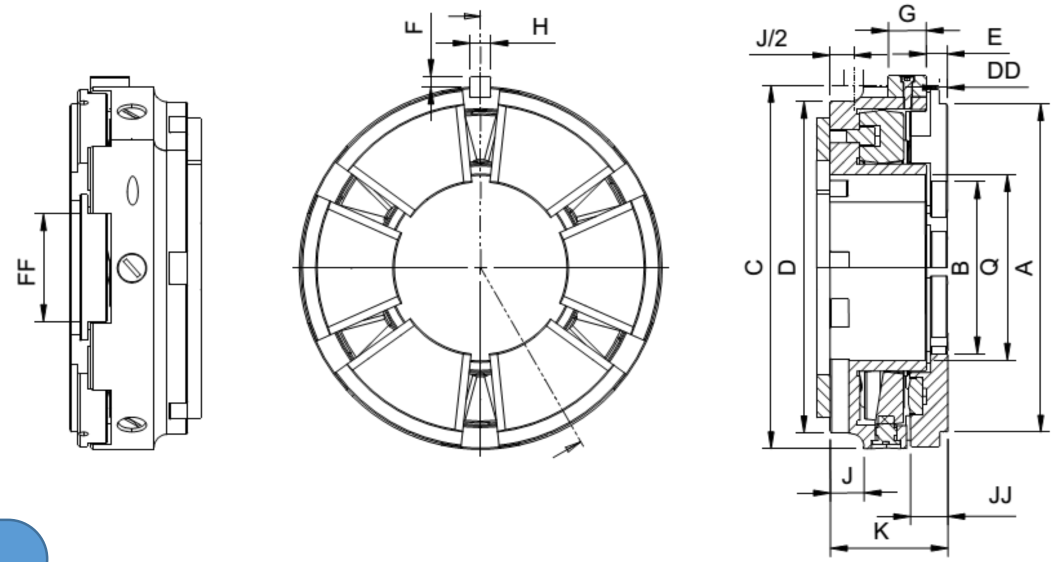


FIG-08

DIMENSIONS FOR "SS" SERIES PRESSURE EQUALIZING THRUST BEARING

Bearing Size	Area(mm sqr)	A - Babbitt O.D	B- Babbitt I.D	K - Bearing Thickness (SS)	K - Bearing Thickness (SB)*	C - Bearing O.D	Q - Base ring I.D.	D - Oil annulus dia	J - Oil annulus depth (J)
SS 100	5150	100	50.8	36.6	35.1	111.12	55.6	104.7	9.7
SS 125	8060	127	63.5	44.5	41.2	136.52	69.9	125.5	12.7
SS 150	11600	150	76.2	52.3	47.8	161.92	82.6	150.9	15
SS 175	15800	175	88.9	60.5	53.9	187.32	95.3	171.5	17.5
SS 200	20250	200	104.6	68.3	60.5	212.72	109.5	193.6	20.8
SS 225	26120	225	114.03	76.2	68.3	238.12	124	219	22.4
SS 265	35540	265	133.35	85.9	74.7	279.4	144.5	254	25.4
SS 305	46440	305	152.4	95.3	82.6	317.5	165.1	293.6	30.2

Bearing Size	G - Bearing key length	H - Bearing key , width	E - Collar to key	F - Key projection	M - Separate shaft dia	N – Integral shaft dia.	P – Max dia. over fillet	R – Dia. through base ring	S – Shaft lgth @ shoe I.D.
SS 100	9.7	6.4	7.1	3.1	44.5	41.2	46.5	49.3	12.7
SS 125	14.2	7.9	7.9	4.1	57.2	53.8	61.3	63.5	15.8
SS 150	16.8	9.7	9.7	4.8	69.9	66.6	74	76.2	19.1
SS 175	20.6	9.7	11.9	4.8	82.6	79.3	86.8	88.9	22.4
SS 200	23.9	11.2	12.7	4.8	95.3	92	99.3	101.6	25.4
SS 225	23.9	11.2	14.2	4.8	108	104.7	112.2	114.3	28.5
SS 265	28.5	12.7	15.8	5.6	124	120.7	130	133.4	31.8
SS 305	30.2	14.2	17.5	5.6	142.8	139.7	149.1	152.4	35.1

Bearing Size	X – Collar thickness	Y – Collar dia.	Z – Collar bore	T – Collar key depth	V – Collar key width	FF– traddle mill	JJ – Shoe thickness	DD – Shoe relief	W – Collar chamfer
SS 100	22.4	104.7	31.75	4.1	7.9	32.5	12.7	3.1	1.5
SS 125	22.4	130.1	44.45	4.8	9.7	40.5	15.88	4.1	1.5
SS 150	25.4	155.5	53.98	4.8	9.7	50	19.05	4.1	1.5
SS 175	31.8	180.8	63.5	6.4	12.7	59.5	22.23	4.8	1.5
SS 200	35.1	206.3	76.2	7.9	15.8	69.1	25.4	5.6	1.5
SS 225	38.1	231.7	88.9	7.9	15.8	77	28.58	7.9	1.5
SS 265	44.5	271.5	104.78	9.7	19.1	80.9	31.75	7.1	2.3
SS 305	50.8	309.6	120.65	9.7	19.1	100.8	34.93	8.6	2.3

1.9 SHAFT GEOMETRY & DIMENSIONS.

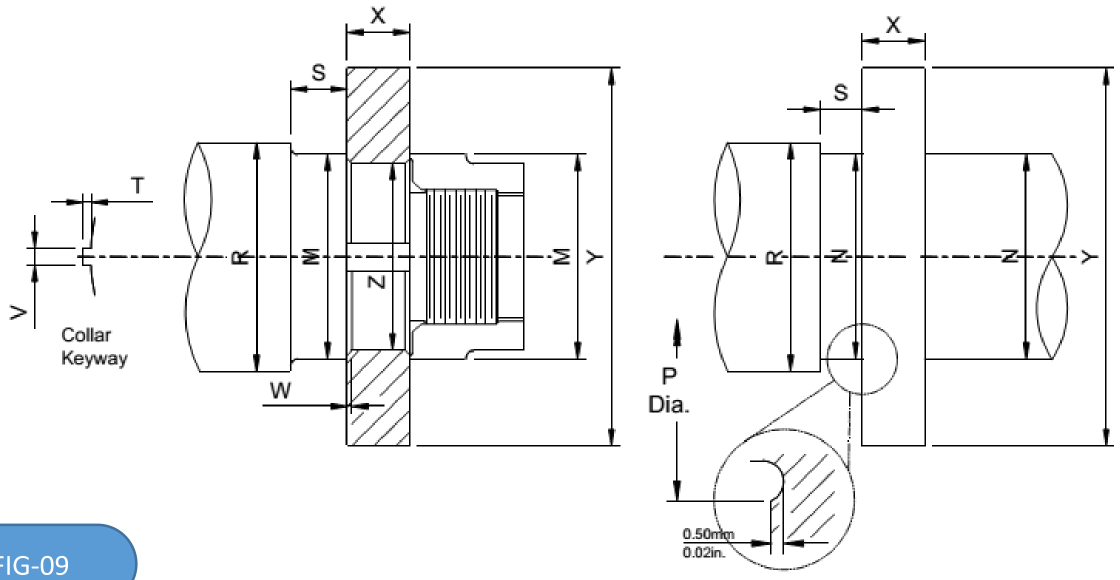


FIG-09

DIMENSIONS FOR "SS" SERIES PRESSURE EQUALIZING THRUST BEARING

Bearing Size	Area(mm sqr)	A - Babbitt O.D	B – Babbitt I.D	K - Bearing Thickness (SS)	K - Bearing Thickness (SB)*	C – Bearing O.D.	Q – Base ring I.D	D – Oil annulus dia	J – Oil annulus depth (J)
SS 342	58770	342	171.45	108	90.4	355.6	185.7	330.2	36.6
SS 380	72570	380	190.5	117.4	98.6	393.7	206.3	368.3	35.1
SS 430	93220	430	215.9	133.4	111.3	447.68	233.4	419.1	46
SS 480	116440	480	241.3		120.7	514.35	269.8	469.9	
SS 530	142250	530	266.7		133.4	565.15	298.5	514.4	
SS 585	170640	585	292.1		144.5	622.3	323.9	568.5	
SS 635	201600	635	317.5		157.2	673.1	355.6	622.3	
SS 685	235050	685.8	342.9		169.9	730.25	400	673.1	

Bearing Size	G - Bearing key length	H - Bearing key , width	E - Collar to key	F - Key projection	M - Separate shaft dia	N – Integral shaft dia.	P – Max dia. over fillet	R – Dia. through base ring	S – Shaft lgth @ shoe I.D.
SS 342	35.1	15.8	19.1	6.4	162.1	158.8	168.1	171.5	38.1
SS 380	38.1	17.5	20.6	7.9	177.8	174.8	186	190.5	41.2
SS 430	41.2	19.1	23.9	7.9	203.2	200.2	211.3	215.9	44.5
SS 480	44.5	22.4	25.4	8.6	225.6	222.3	235.6	247.7	50.8
SS 530	44.5	25.4	28.5	9.7	251	247.7	261	273.1	57.2
SS 585	53.9	25.4	33.3	9.7	273.1	266.7	283.7	298.5	60.5
SS 635	57.2	31.8	35.1	12.7	298.5	292.1	309.2	327.2	63.5
SS 685	60.5	31.8	36.6	12.7	320.6	311.2	332	352.6	69.9

Bearing Size	X – Collar thickness	Y – Collar dia.	Z – Collar bore	T – Collar key depth	V – Collar key width	FF– traddle mill	JJ – Shoe thickness	DD – Shoe relief	W – Collar chamfer
SS 342	57.2	347.7	136.53	11.2	22.4	107.2	38.1	9.7	2.3
SS 380	63.5	385.8	152.4	12.7	25.4	129.4	41.28	3.1	2.3
SS 430	73.2	438.2	168.28	12.7	25.4	145.3	46.03	3.1	3.1
SS 480	82.6	489	190.5	14.2	28.5	151.6	50.8	9.7	3.1
SS 530	92	539.8	215.9	15.8	31.8	177	55.58	12.7	3.1
SS 585	98.6	590.6	238.13	15.8	31.8	195.3	60.33	12.7	4.1
SS 635	108	641.4	254	19.1	38.1	203.2	68.28	12.7	4.1
SS 685	117.4	692.2	279.4	19.1	38.1	211.1	69.85	12.7	4.1

2. DESIGN & OPERATIONAL BEAVER.

2.1 THRUST PAD TEMPERATURE & PRESSURE VARIATION

A temperature sensor mounted at the 75/75 position can provide the critical shoe temperature for that one, fixed location, but cannot indicate the temperature pattern over the entire shoe surface. Basically, this temperature zone is indicated at high pressure zone. The one single factor which most limits thrust bearing load capacities is the babbitt temperature. Hence, the one factor addressed in most thrust bearing upgrades is reducing the maximum bearing temperature. Oil film thickness, film pressure, and mechanical factors also need to be addressed, but they are usually well within required design constraints (especially if the pad temperature is within acceptable levels).(FIG-09)

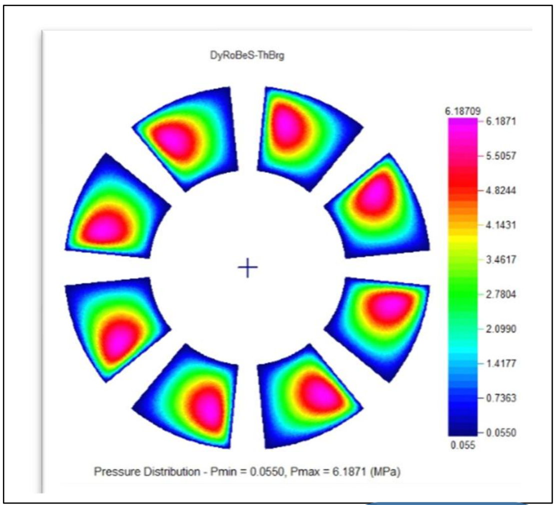


FIG-09

2.2 ALTERNATIVE MATERIAL FOR THRUST PADS.

One of the easiest ways to cool down a hot running bearing is by upgrading the pad backing material from steel to copper alloy. The high heat conduction properties of the copper pulls heat from the babbitt face where it can do the most damage. Several papers have been published on the impact of pad material on thrust bearing performance. One such paper includes test results that can be correlated with thrust. A 150 mm thrust bearing was tested with steel pads and copper alloy pads. The same conditions were analysed and the results plotted.(FIG-10)

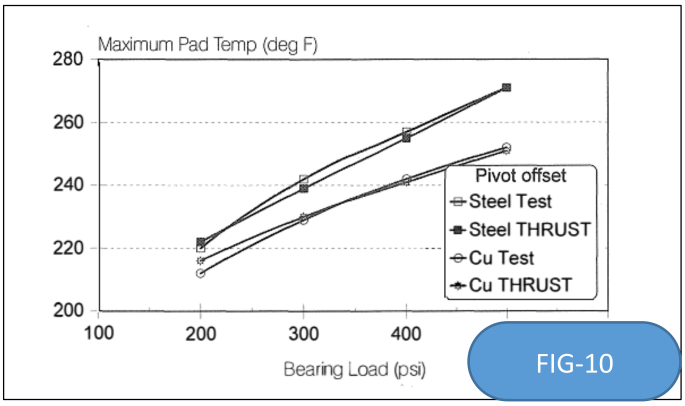


FIG-10

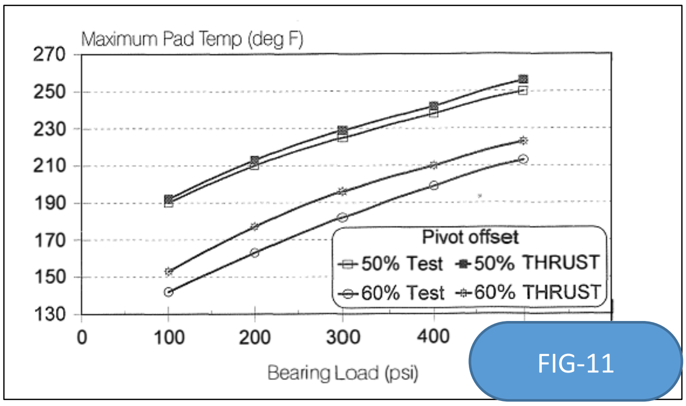


FIG-11

2.3 EFFECT OF OFFSET PIVOT ON BABBITT TEMPERATURE.

Another dramatic design variable that can be optimized is the circumferential pivot location. Most thrust bearings are supplied with pivots at a 50 percent offset (centered circumferentially). Moving the pivot towers the trailing edge forces the leading edge film thickness to be greater, thereby bringing more cooling oil onto the thrust face, resulting in lower operating temperatures.

The plot in Figure 11 is used to illustrate the correlation of the program with test data along with demonstrating the impact of the offset pivot configuration on performance. The top two lines are centre pivot (50 percent offset) results for the test and the analysis. Note the very good correlation between the test data and the analysis. The bottom two lines are results for the 60 percent offset pivot bearing. (FIG-11)

2.4 LOCATION OF TEMPERATURE SENSOR.

Thrust bearing temperature sensors shall be placed at 75 percent of the pad width radially out from the inside bearing bore and at 75 percent of the pad length from the leading edge.The sketch in Figure 12 illustrates thermocouple (TC) or RTD placement in tilting pad thrust bearings. Note that the sensors should be imbedded in the pad backing material, not the babbitt. Also note that the sensor should be 1.5 mm to 2.5 mm from the babbitt face but no closer than 0.76 mm from the bond line. As an example, for babbitt 1.5 mm thick, the sensor should be 2.25 to 2.5 mm from the babbitt face.

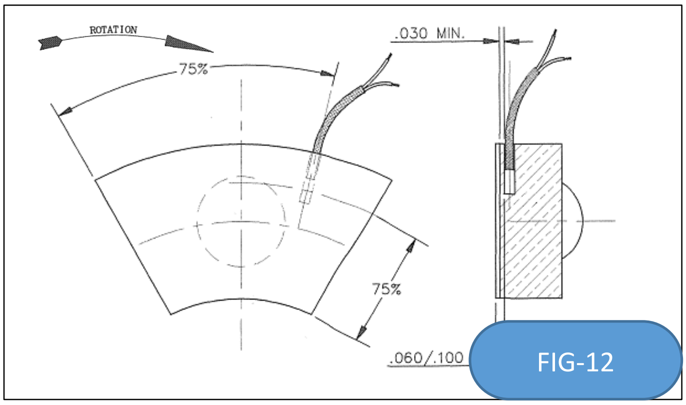


FIG-12

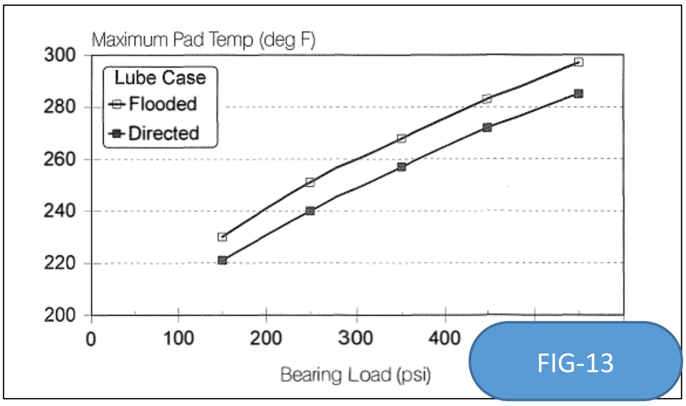


FIG-13

2.5 DIRECT LUBRICATION

The means of lubricating the bearing does not have as pronounced effect on bearing metal temperature as does pad material and pivot offset. However, with hot running bearings, having operating temperature of 92.5 deg.C or so drop in bearing operating temperature can be achieved, by upgrading from a standard flooded lube configuration to a directed lubrication.

In the conventional flooded lube design oil is introduced at the bore of the bearing, travels up the face of the thrust collar, and is discharged though a drain port at the top of the bearing chamber, at the collar OD. In this case, the entire chamber is flooded with oil and churning of the oil by the collar introduces heat to the bearing beyond the shearing of the oil film. In fact, the amount of heat generated by churning is often on the same order of magnitude as the heat generated by the shearing of the oil film. As a result of this, power savings can be realized by upgrading to directed lube bearings. Indeed with large utility sized bearings power savings of several hundred horsepower can be realized. A directed lube bearing minimizes this churning by introducing the oil through a series of orificed nozzle blocks located between pads. These blocks supply the cool inlet oil directly to the leading edge of each pad thereby eliminating the need to flood the chamber with oil. In this case the chamber has generous drain provisions at the bottom, located at the collar OD.

The effect of directed lubrication was analysed with very good results. New summarized results of a series of tests run with a 150 mm Bearing having result as per Figure 13.

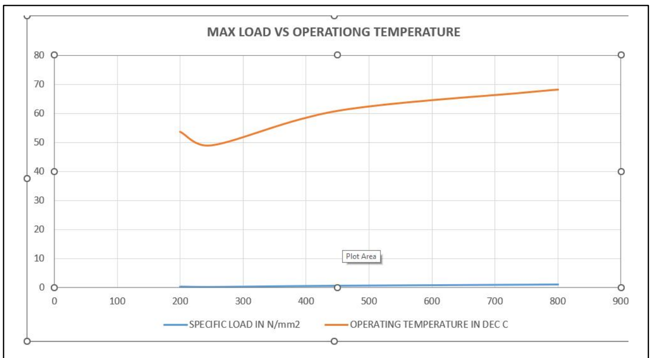


FIG-14

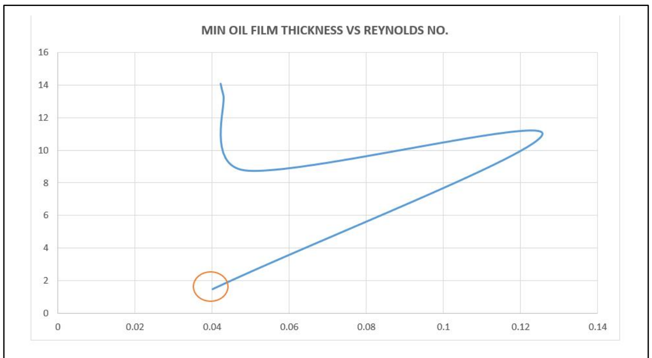


FIG-15

2.6 NUMERICAL SIMULATION OF LEVELING PLATE (BASED ON PREVIOUS PROJECT SUPPLIED TO JRC MARINE).

To have a start point of the designing of a new self-equalizing bearing, the numerical analyses of the levers are very important.
The presented numerical analysis was done for the bearing with 4 segments of thrust pads and 8 segments of rocker arm/ leveling plates. Sizes & external diameter of segments 102.10 mm and internal diameter of pad segments 54.55 mm using the software Autodesk simulation version. The numerical model has been defined of the non-linear type considering big deformations with contact elements defined in joins.
bearing was set-up based on the real values designed by “SUNTECH® Engineering Corporation” (INDIA) for “JRC Marine” application that was measured in real practice for propulsion system from which result that the bearing can be standardly loaded with an axial force of 15 kN. The basic dimensions of a rocker arm have been considered according to Fig. 16 they have wing thickness = 7.17 mm; effective width = 2.03 mm and length = 20.6 mm.

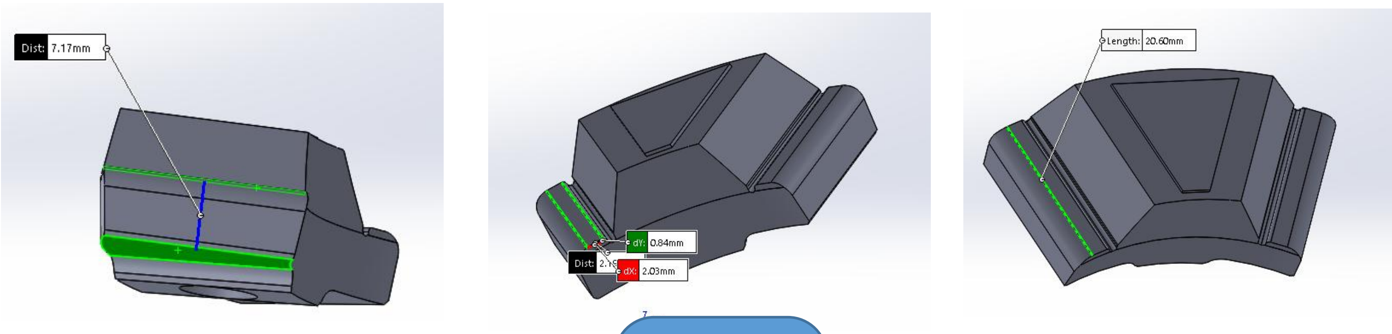


FIG-16

The boundary conditions were applied so to prevent the movement (collapse) of the system and at the same time use them to influence the stress distribution in the rocker arm as little as possible.

STATIC ANALYSIS: LOAD DISTRIBUTION ON ROCKER ARM.

General objective and settings:

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	05-08-2022, 00:11
Model State	[Primary]
Design View	Default
Positional	[Primary]
Detect and Eliminate Rigid Body Modes	No
Separate Stresses Across Contact Surfaces	No
Motion Loads Analysis	No
Volume	648234 mm^3
Center of Gravity	x=-0.051152 mm y=-0.103196 mm z=-0.00108541 mm

Note: Physical values could be different from Physical values used by FEA reported below.

Status

Design Status	WorkInProgress

Physical

Mass	5.08864 kg
Area	169447 mm^2

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	No
Use part based measure for Assembly mesh	Yes

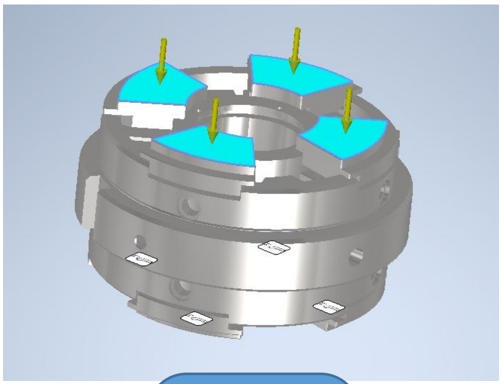
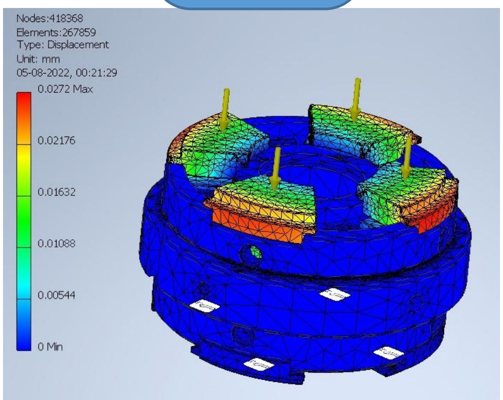


FIG-17



Material(s)

Name	Steel, Mild	
General	Mass Density	7.85 g/cm^3
	Yield Strength	207 MPa
	Ultimate Tensile Strength	345 MPa
Stress	Young's Modulus	220 GPa
	Poisson's Ratio	0.275 ul
	Shear Modulus	86.2745 GPa
Part Name(s)	rocker arm without hole.ipt Thrust Pad.ipt thrust pad 1.ipt rocker arm Part5.ipt	

Force:1

Load Type	Force
Magnitude	15000.000 N
Vector X	-0.000 N
Vector Y	0.000 N
Vector Z	-15000.000 N

Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	15000 N	0 N	0.757772 N m	0.580181 N m
		0 N		-0.487451 N m
		15000 N		0 N m

Result Summary

Name	Minimum	Maximum
Volume	648213 mm^3	
Mass	5.08847 kg	
Von Mises Stress	0.000283208 MPa	141.059 MPa
Displacement	0 mm	0.0271959 mm
Safety Factor	1.46747 ul	15 ul

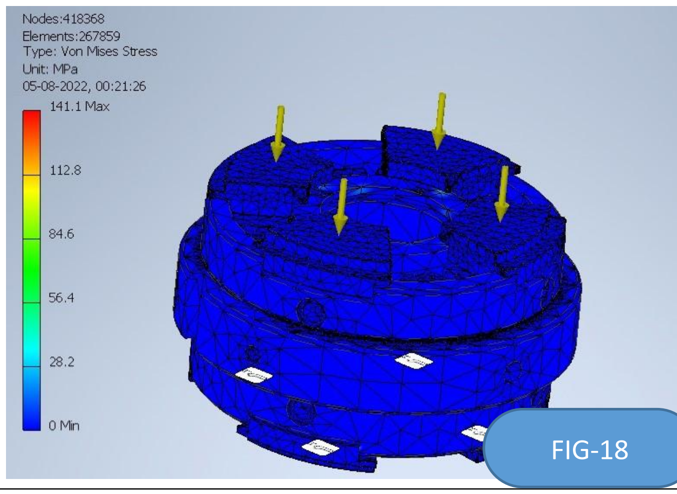


FIG-18

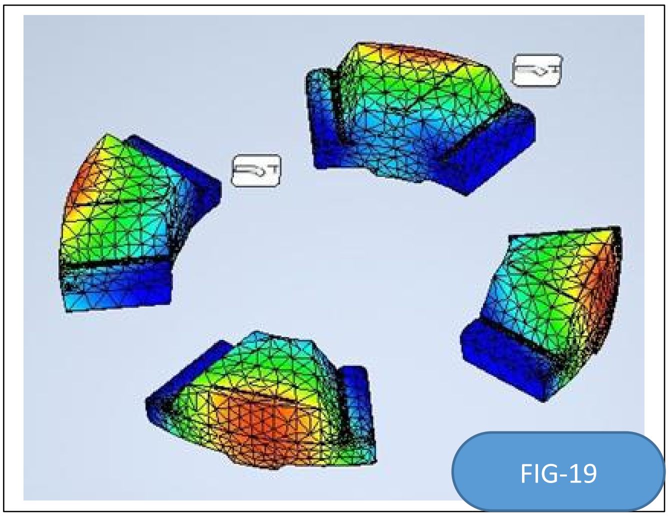


FIG-19

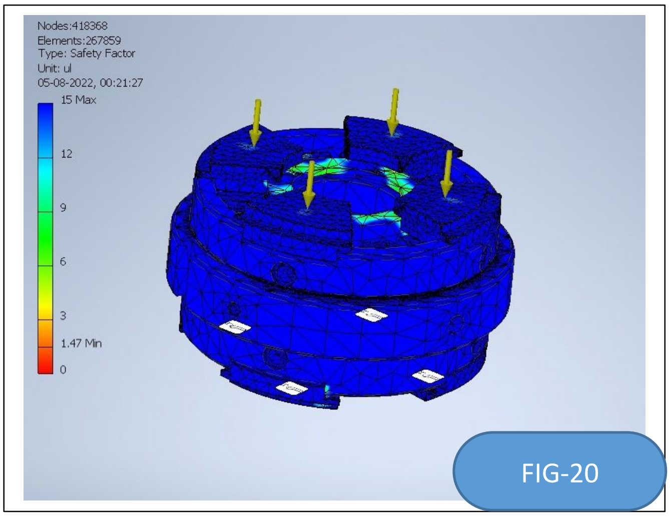


FIG-20

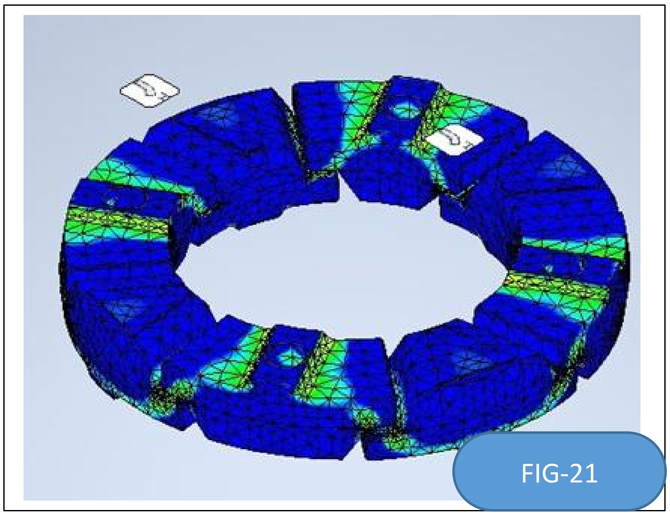


FIG-21

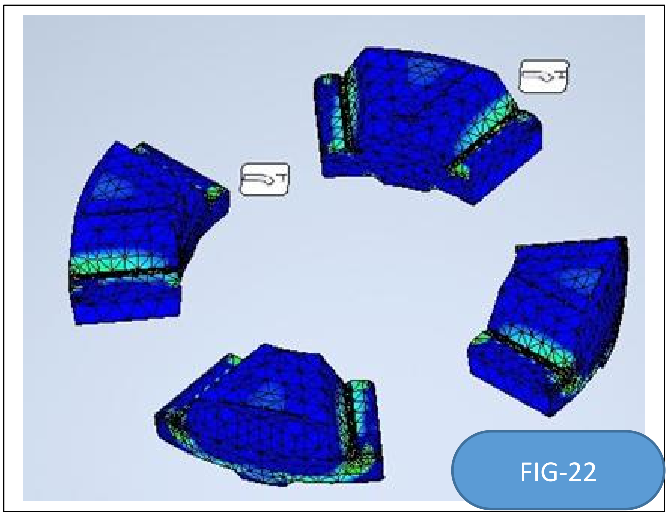


FIG-22

The maximum value of the vertical displacement reached 0.027 mm. A small inclination of the levers towards the outer circumference of the bearing is also observable here.

The maxima of the dominant vertical stress in the inner corners of the lever arms reach the values σz = 141.059 MPa in tension.

Conclusions:

IS 2062 has been used for the levers in a refined state with a yield strength of 207 MPa. Due to this, the stress values of the levers can be considered safe and factor of safety will be 1.47.
Observation :

1)Overall Bearing performance found ok as per Hydrodynamic calculation in every Speed Vs Load Parameter except only when the thrust Bearing goes under 2100000N load with 20 RPM. Minimum film thickness we found 0.024 mm where as permissible limit is 0.026 mm for the particular operating condition.

THRUST BEARING PURCHASE INPUT DATA SHEET.

sales@suntechbearings.com
For additional contact information, Contact SUNTECH® ENGINEERING CORPORATION.

Please confirm bearing selection and give estimated operating data for the following:

Name:____
Company:______________________________________
Contact E-mail:__________________________________
Project Ref:____________________________________
Date:____

1. General

Application:____

2. Thrust Load

Normal :____________
Max continuous: _______________
Minimum load :____________________________
At instant of start up:___________________________________
Max.momentary:____

3. Lubricant

Type:_______________
ISO viscosity grade:________________
Temperature at bearing inlet:____________________________
Pressure at bearing inlet:____

4. Shaft Speed

Normal :_________________
Max continuous :_______________________
Over speed :____
Bi- or uni-directional :____

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