44TH TURBOMACHINERY & 31ST PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 14 – 17 2015
GEORGE R. BROWN CONVENTION CENTER

QUALIFICATION AND OPTIMIZATION OF SOLID POLYMER TILTING PAD BEARINGS FOR SUBSEA PUMP APPLICATIONS

Dieter Henssler
Engineering Manager
John Crane Bearing Technology GmbH
Göttingen, Germany

Simon Gassmann
Development Engineer Subsea Pumps
Sulzer Management AG
Winterthur, Switzerland

Lorenz Schneider
Development Engineer Subsea Pumps
Sulzer Management AG
Winterthur, Switzerland

Thomas Felix
Manager Subsea Pumps
Sulzer Management AG
Winterthur, Switzerland

Dr. Dieter Henssler *studied Mechanical Engineering at the University of Stuttgart where he also received his degree as a Dr.-Ing. in 1995, focusing on investigations of High Speed Journal Bearings. He joined Schiele PGW Turbomaschinen GmbH as Technical Director for industrial fans and compressors. Since 1999 he was working for Voith Turbo BHS in several positions as head of the research and development, calculations and design departments and finally as Director After Sales before joining John Crane Bearing Technology GmbH, Göttingen, Germany as Engineering Manager in 2013.*

Lorenz Schneider *started his career with Sulzer in 2007 as a Development Engineer where he is part of the "Subsea Pumps" team of the Product Development Unit in Winterthur, Switzerland.*
Lorenz graduated as a Master of Science in Mechanical Engineering at the Swiss Federal Institute of Technology in Zurich in 2007, focusing on computational fluid dynamics and system dynamics.

Simon Gassmann *started his career with Sulzer in 2008 as a Development Engineer where he is part of the "Subsea Pumps" team of the Product Development Unit in Winterthur, Switzerland.*
Simon graduated as a Master of Science in Mechanical Engineering at the Swiss Federal Institute of Technology in Zurich in 2008 and as a Master of Science in Subsea Engineering at the University of Aberdeen in 2013.

Thomas Felix *studied Mechanical Engineering at the University of Magdeburg, Germany where he received his MSc in Turbo Machinery in 1994. He joined Sulzer in Brazil in 1995 as Product Engineer Pumps. Since 2000 he is working for Sulzer in Winterthur, Switzerland in several positions in the pump development as mechanical analyst and hydraulic engineer in the area of multiphase pumps. In 2007 he became Manager Subsea Pumps, heading up Sulzer's subsea pump development team.*

ABSTRACT

Tilting pad bearings are used extensively in high power and/or high speed pump applications. Traditional white metal-lined also known as babbitted tilting pad bearings are used for oil lubricated bearings. Babbitted tilting pad bearings are not ideal for applications with low viscous lubricants, like water-based fluids. For these applications, polymer bearings offer many advantages over traditional bearing materials including:

- Low friction coefficient
- Higher flexibility
- Wide range of operational temperature
- High corrosion and chemical resistance
- High shock and impact resistance
- Tolerance against particle contamination
- Electrical insulation

To achieve the desired improvement in performance, the material specifics of the polymer have to be carefully considered in the design process.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

One option available is a polymer lined tilting pad bearing which combines the excellent tribological properties of polymers and the high structural stiffness of the steel back. But a polymer lined metal pad does not provide the same equalizing effect offered by the higher compressibility of a solid polymer pad. For example the problems of an edge carrier can be reduced for a tilting pad journal bearing and the self-equalizing mechanism can be avoided for a tilting pad thrust bearing while considering the higher flexibility of solid PEEK pads. An additional advantage of a solid polymer pad is the elimination of the bonding between the polymer and the steel, so that a possible source of a bearing failure can be avoided upfront already.

During a qualification test of a commercially available solid polymer tilting pad thrust bearing, the high Hertzian contact stress at the pivot let to indentations in the polymer. After a few operating hours these indentations are clearly visible and will finally lead to a complete loss of the advantages of a tilting pad bearing over a fixed geometry bearing.

This lecture describes the steps involved in the optimization of the solid polymer tilting pad bearing design, with the objective to increase the reliability of the bearing assembly. The limitations of the available solid polymer tilting pad thrust bearing have been overcome by the integration of a metal reinforced pivot as mentioned by Gassmann et al. (2013).

The results of the TEHD analysis presented form the base of the optimized journal and thrust bearing design. The new journal and thrust bearing design has been validated through testing of a prototype.

The data collected during tests with the optimized thrust and journal bearings are in excellent agreement with the simulations. With the optimized bearing design, the limitations of the original pad design have been mitigated. The successfully qualified journal and thrust bearing are one of the building blocks required for product lubricated pumps and will be used in subsea pump applications.

INTRODUCTION

Based on the advantages given by the use of solid polymer tilting pad bearings it was decided to utilize these bearings for pumps, handling clean or contaminated fluids. For these bearings a Contamination Class of -/15/12 of the lubricant in accordance to ISO 4406 is considered but tests have been performed up to a Contamination Class of -/19/16.

Before these bearings are capable for a service in subsea pumps the material has to be selected and finally these bearings have to prove their performance in several component tests under realistic conditions. Therefore tilting pad journal and thrust bearings with solid polymer pads have been jointly developed and tested under realistic operating conditions, typical for a subsea pump application.

QUALIFICATION OF SOLID POLY-ETHER ETHER-KETONE (PEEK) TILTING PAD BEARINGS

Material selection

Taking in account the special demanding environmental and operational conditions of product or barrier fluid lubricated bearings in subsea pump applications, where a Mean Time Between Maintenance of up to five years and even a higher Mean Time Between Failure is required, only high performance polymers with excellent chemical and wear resistance can be taken into consideration. PEEK a semi-crystalline thermoplastic, offering a chemical and water resistance similar to PPS, was finally selected. The chemical structure of the PEEK monomer is shown in Figure 1. As Gerthsen (2008) mentions, the aromatic and heterocyclic carbon rings are influencing the stability of the polymer positively.

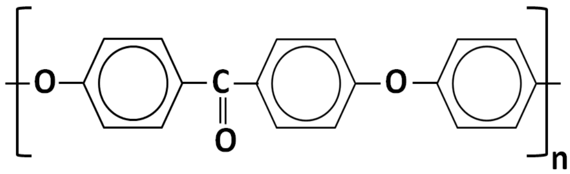

Figure 1: Monomer unit of PEEK.

PEEK is a good alternative for fluoropolymers for challenging environments. It offers a good wear and abrasion resistance, extremely low moisture absorption and excellent chemical resistance. A major factor for material selection was the fact that PEEK retains its superior mechanical and chemical resistance properties at high temperatures. PEEK has a glass transition temperature of 289°F (143°C) and melts at 649°F (343°C). In general the maximum operating temperature is limited to 482°F (250°C). For this bearing application it was decided to limit the maximum bearing temperature to 266°F (130°C). Passing the glass transition temperature should be avoided, as all thermal and mechanical properties of PEEK have a step change, while passing this temperature.

In general PEEK-lined steel pads are used for this type of bearings, combining the excellent tribological properties of a high performance polymer and the stability of the steel back. As shown in Table 1 the tensile strength of virgin PEEK is only 14 percent of the tensile strength of 16MnCr5, a standard steel used for manufacturing oil lubricated bearings. Its Young´s modulus is only 1.7 percent of the one of steel. Therefore virgin PEEK is not suitable for this special application. A FEM analysis showed that even a carbon fiber reinforced PEEK

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

compound with 30 percent of short, but randomly orientated carbon fibers is not able to cope with the stresses and leads to unacceptable high deflections. Only a PEEK composite with a carbon fiber content of 50 percent and compression molded out of PEEK impregnated fabric has enough strength to handle the stresses and reduce the deflection, so that it is suitable for this application. Due to the higher compressibility of the PEEK composite this material offers the advantage of better load equalization, especially for thrust bearings and thus allows higher specific loads. Therefore the PEEK composite was selected for manufacturing the pads.

	Direction	16MnCr5	Virgin PEEK	PEEK Compound	PEEK Composite
Fiber Content		-	-	30%	50%
Fiber Orientation		-	-	random	0/90°
Density		1	16,8%	18,2%	19,6%
Tensile Strenght		1	14,3%	19,1%	89,6%
Young´s Modulus		1	1,7%	5,2%	28,6%
Thermal expansion	x/y	1	3,92	1,28	40,0%
Thermal expansion	z	1	3,92	1,28	5,46
Thermal Conductivity		1	0,6%	2,3%	1,5%

Table 1: Comparison of several mechanical and thermal material properties of different materials.

As the carbon fibers of the composite are orientated in the x/y plane, the compression molded material is now anisotropic. Therefore the mechanical and thermal properties like yield strength, thermal elongation are dominated in this plane by the carbon fibers, whereas perpendicular to the fibers in the z direction, these properties are dominated by the PEEK matrix. Figure 2 shows the general material structure and the corresponding coordinate system. The now anisotropic material behavior has to be considered especially while predicting the bearing clearance under operating conditions and calculating the pad deflections due to temperature increase and load.

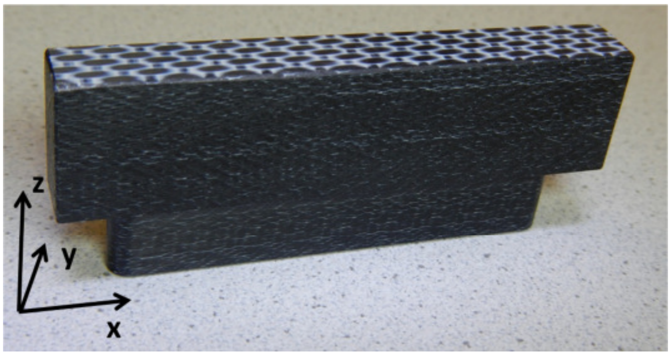

Figure 2: Material structure of the PEEK composite with corresponding coordinate system.

Tilting Pad Journal Bearing

For the qualification tests a prototype bearing was developed, designed and manufactured as per specification shown in Table 2. A 5-pad rocker-pivot pad bearing in the LBP configuration was selected, as it´s dynamic coefficients suits the rotor dynamic requirements best. Additionally the LBP configuration gives lower bearing temperatures and higher fluid film thicknesses, resulting in a more robust bearing, than operating it in the LOP configuration. Although the bearing size was limited by the existing test rig, it was important that dimensions and operating conditions are realistic for an operation in a subsea pump.

Parameter	Value
Type of bearing	Solid polymer tilting pad, center pivot, 5 pads
Shaft orientation	Horizontal or vertical
Lubrication type	Flooded
Lubricant	Water
Lubricant inlet temperature	up to 122°F (50°C)
Viscosity lubricant	lower than ISO VG 5
Shaft diameter	3,6 inch (91,5 mm)
Shaft coating	Tungsten Carbide / CoCr HVOF coating
Operating speed range	1500 - 6000 rpm
Specific load	72,5 psi (0,5 MPa)
Ratio bearing width (pad width) to shaft diameter	0,5
Machined preload	0,3
Relative minimum radial clearance for a centered shaft	0,0014

Table 2: Design specification for the prototype tilting pad journal bearing.

Figure 3 shows the complete TPJB prior to testing. A single pad is shown in Figure 4. As the Hertzian contact stress between pad and carrier is very low due to the low load and an optimized geometry, a metal reinforced pivot is not necessary. Only the bore for the pad retention screw was reinforced by a metal bushing as protection against damaging this bore by dynamic loads.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

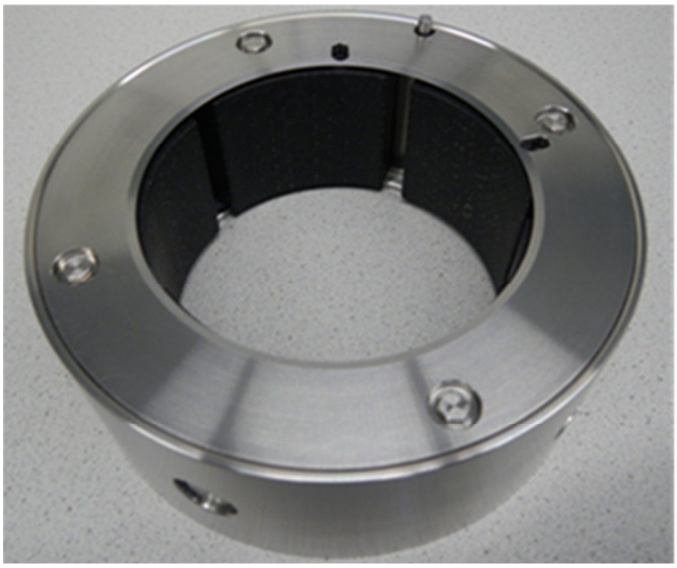

Figure 3: Tilting pad journal bearing equipped with solid PEEK pads.

As shown in Figure 4 the location of the temperature probe was selected according to API 670 requirements under consideration of the actual test rig capabilities.

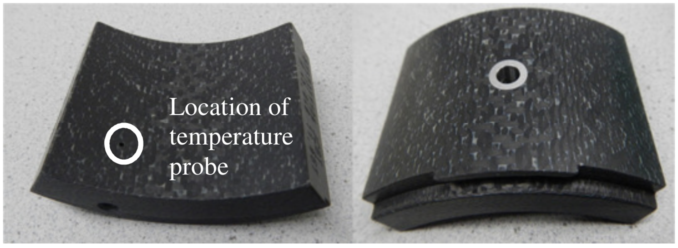

Figure 4: Active surface of the PEEK pad showing the provision for the fluid film temperature measurement and the pivot on the reverse side.

The running surface on the rotor is coated with a Tungsten Carbide / CoCr HVOF protective coating. The coating selected provides excellent wear resistance and reduces the risk of wire wooling caused by hard particles present in the lubricant which potentially embed into the PEEK pads.

Bearing tests

The journal bearing tests have been conducted on the component qualification test rig for journal bearings as shown in Figure 5. This test loop allows testing of journal bearings with water or water glycol mixtures as lubricant. In this case, the solid polymer TPJB has been tested in soft water in the LBP configuration.

The application specific lubrication setup of the journal bearing has been emulated during the qualification test. An external circulation pump gives full flexibility to adjust the lubricant flow rate supplied to the test bearing as required.

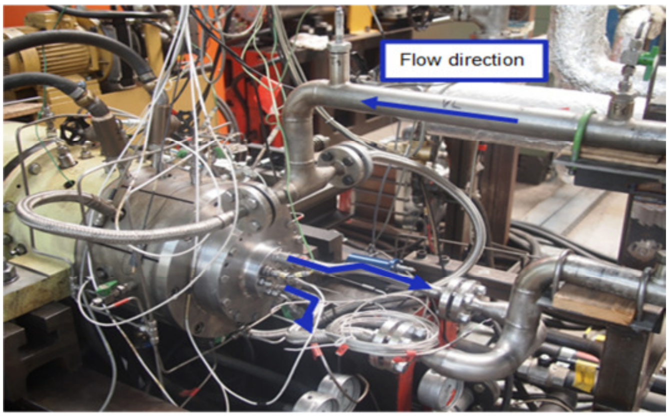

Figure 5: Journal bearing test rig for product lubricated bearings.

The instrumentation as per Figure 6 and Table 3 allows the measurement of the fluid film friction torque together with the local fluid film temperature on the two loaded pads. The none-contacting shaft position sensors installed, measure continuously the shaft position as function of the actual operating conditions of the bearing. The shaft position probes are installed 2 inch (50 mm) from the bearing center. Since the test rig has a short and stiff shaft, shaft deflections are negligible. Thus the probes provide accurate figures of the shaft position at bearing location, too.

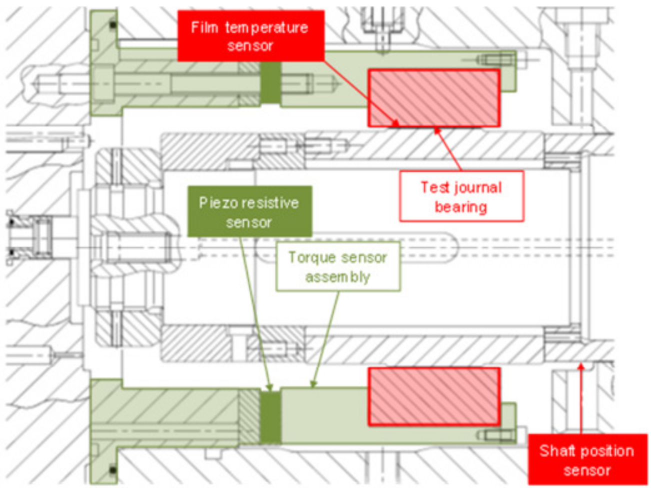

Figure 6: Cross sectional view of the journal bearing test rig, indicating the position of the various sensors.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Measured Value	Description
Torque	Friction torque bearing
Force	Bearing load
Flow rate	Lubricant flow to bearing
Temperature	Lubricant supply temperature
	Lubricant return temperature
	Fluid film temperature loaded pad
Displacement	Shaft position (x-axis)
	Shaft position (y-axis)
Keyphasor	Rotating speed shaft
Pressure	Lubricant supply pressure
	Lubricant return pressure

Table 3: Instrumentation of journal bearing test rig.

Figure 7 is an example of an orbit plot taken at 4000 rpm and a load of 450 lbf (2000 N). It proves, as all the other measurements taken over the whole operating range, that the bearing generates sufficient fluid film thickness in order to allow a safe operation. The minimum required film thickness was specified upfront to exceed 0,35 mils (9 µm). The clover shaped orbit is caused by a run out of approximately 0,2 mils (5 µm). The pentagon reflects the clearance limit of the TPJP with five pads.

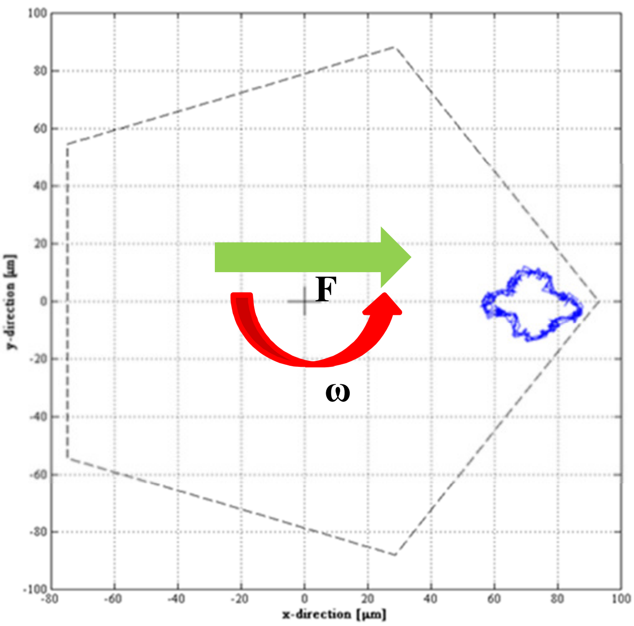

Figure 7: Orbit Plot taken at 4000 rpm and a specific load of 72,5 psi (0,5 MPa), showing an eccentricity of 2,75 mils (70 µm).

A variation of the lubricant flow from 1,6 gal/min (6 l/min) to 6,3 gal/min (24 l/min) showed a minor effect on the bearing temperatures only. Due to the low load the temperature increase of the lubricant and the bearing temperature itself was very low.

After the first set of tests the bearing was disassembled and visually inspected. All pads and the corresponding coated shaft protection sleeve as mating surface were in good condition, so that testing could be continued. Figure 8 shows the condition of one of the loaded pads after the first 10 operating hours at different speeds and loads and 10 loaded start-stops.

Figure 8: Condition of one of the loaded pads after the first 10 operating hours at different speeds and loads.

Additional tests for another 50 h continuous operation under maximum load and 250 consequent loaded start – stops to be performed for further verification of the long term bearing performance.

Comparison calculations – test results

During the design stage of the bearing, the complete range of operating data has been calculated with ALP3T as described by Fuchs, et al. (2005) and Hagemann (2010). In parallel COMBROS was used as a second bearing calculation code as described by Hagemann and Schwarze (2011). Both codes are solving the Energy and the Reynold´s Equation simultaneously considering a three dimensional temperature and viscosity distribution, a three dimensional lubricant film thickness and turbulence within the lubricant film.

The simulated journal bearing performance has been verified through testing at multiple test points throughout the relevant bearing envelop. The measured journal bearing performance is in excellent agreement with the simulations conducted.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

As shown in Figure 9, the measured shaft displacement indicates that the journal bearing is operating with an adequate hydrodynamic film thickness. The simulated eccentricity is larger than the measured one, which results finally in a higher film thickness, increasing the safety of the bearing tested. This finding has been verified after these tests by the visual examination of the pad running surface. No contact marks have been identified after 10 hours of testing and 10 loaded start-stops.

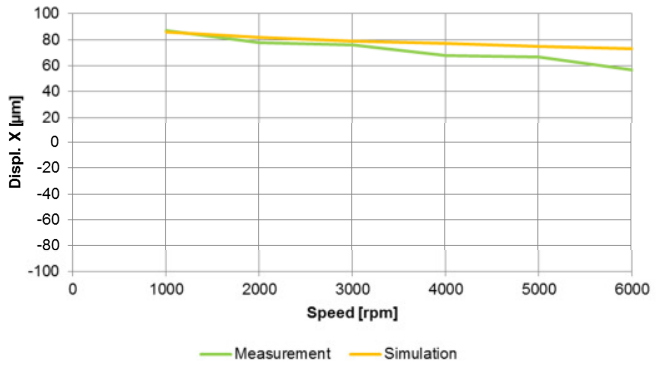

Figure 9: Comparison of measured shaft displacement at 72,5 psi (0,5 MPa) specific load against simulations with water as lubricant.

The simulated power losses of the bearing are consistent with the loss figures measured during the test, as shown in Figure 10. The discrepancy between the simulated and the measured power losses are within the uncertainty band of the torque sensor used.

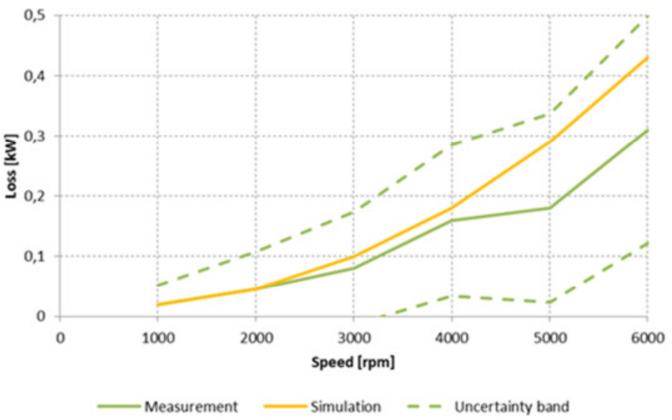

Figure 10: Comparison of measured power loss at 72,5 psi (0.5 MPa) specific load against simulations with water as lubricant.

Although the temperature increase of the fluid film is rather low, due to the low power losses generated with soft water as lubricant, the bearing temperatures measured are in good correlation with the predicted ones, as shown in Figure 11. This measurement was taken at a lubricant inlet temperature of 86 °F (30 °C).

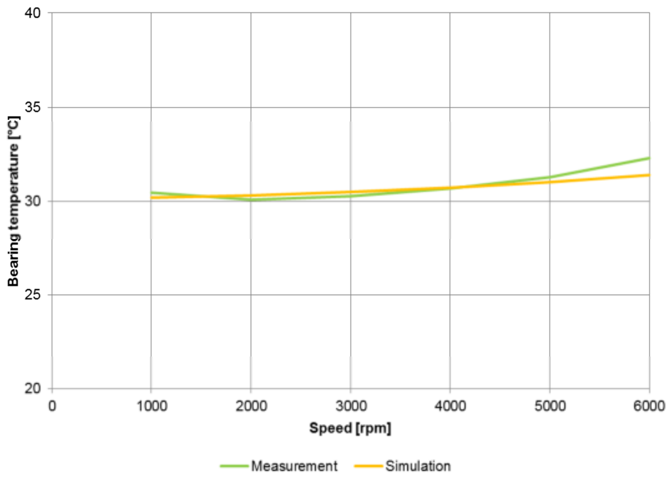

Figure 11: Comparison of measured journal bearing temperature at 72,5 psi (0,5 MPa) specific load against simulations with water as lubricant at an inlet temperature of 86°F (30 °C).

Tilting Pad Thrust Bearing

As subsea pumps consist of two journal and one double acting thrust bearing, thrust bearings have to be tested for this qualification, too. Table 4 shows the specification for the tilting pad thrust bearing.

Parameter	Value
Type of bearing	Double acting solid polymer tilting pad, offset pivot
Lubrication type	Flooded
Lubricant	Water glycol mixture
Lubricant inlet temperature	up to 122°F (50°C)
Viscosity lubricant	ISO VG 5
OD thrust bearing	11,7 inch (297 mm)
Thrust collar coating	Tungsten Carbide / CoCr HVOF coating
Operating speed range	1500 - 6000 rpm
Specific load	up to 551 psi (3,8 MPa)

Table 4: Design specification of the tilting pad thrust bearing.

The prototype of the tilting pad thrust bearing is shown in Figure 12. One thrust pad is shown in Figure 13. A metal pivot was selected for the thrust bearing as the Hertzian contact stress between a PEEK composite pad and the metal carrier of this high loaded thrust bearing would cause unacceptable high

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

deformations, as mentioned by Gassmann et al. (2013). The contact surface between metal and PEEK is now increased, thus reducing the Hertzian contact stresses in the PEEK significantly. The metal pivot therefore mitigates the risk of creeping at the pivot in applications with high contact stresses.

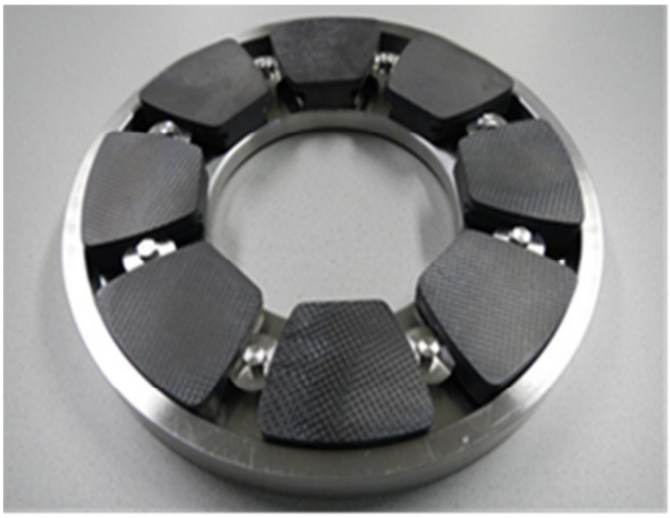

Figure 12: Thrust bearing equipped with solid PEEK pads.

Figure 13: Solid PEEK pad, showing the provision for the fluid film temperature measurement and the metal reinforced pivot.

Figure 13 shows as well the location of the temperature probe. It was selected according to the requirements of API 670 under consideration of the actual thrust bearing test rig capabilities.

Bearing tests

The thrust bearing tests have been conducted on the component qualification test rig for thrust bearings as shown in Figures 14 and 15. This test loop allows testing of thrust bearings with water glycol mixtures as lubricant.

The application specific lubrication setup of the thrust bearing has been emulated during the qualification test. An external circulation pump gives full flexibility to adjust the lubricant flow rate supplied to the test bearing as required.

Figure 14: Thrust bearing test rig for product lubricated bearings.

All tests of the tilting pad thrust bearing have been conducted with a water- MEG mixture of a viscosity similar to ISO VG 5. A variation of the volumetric lubricant flow rate from 16 gal/min (60 l/min) to 27 gal/min (102 l/min) showed only a minor effect on the bearing temperatures, even on the loaded bearing under maximum load condition.

The instrumention as per Figure 15 and Table 5 allows the measurement of the fluid film temperature of the loaded bearing, the load, the lubricant flow and the power loss. The power loss of the thrust bearing is determined by an energy balance across the bearing. The supply and return temperature of the lubricant together with its flow rate allow the determination of the heat generated inside the test rig. This signal is then corrected by the known drag loss of the rotating shaft of the test rig. In addition the motor terminal power serves as a plausibility check for the such determined bearing loss figures.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

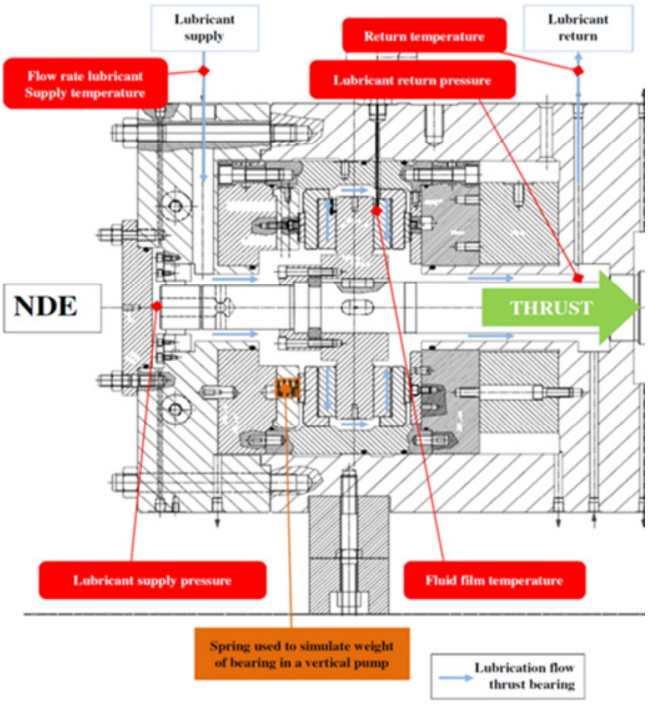

Figure 15: Cross sectional view of the thrust bearing test rig, indicating the position of the various sensors.

.

Measured Value	Description
Power	Motor terminal power
Force	Bearing load
Flow rate	Lubricant flow to bearing
Temperature	Lubricant supply temperature
	Lubricant return temperature
	Fluid film temperature loaded pad
Keyphasor	Rotating speed shaft
Pressure	Lubricant supply pressure
	Lubricant return pressure

Table 5: Instrumentation of the thrust bearing test rig.

Comparison calculations – test results

Similar to the TPJB the complete range of operating data was calculated for the TPTB prior to the testing with COBMROS-A, a bearing calculation code described by Kraft (2014). This code is specially developed for thrust bearing calculations considering the same features as for the journal bearings.

The temperature distribution at 4000 rpm at a specific load of 303 psi (2,09 MPa) and a lubricant inlet temperature of 86 °F (30 °C) can be seen in Figure 16. The bearing temperature measured at the probe location for the same operating condition is marked in the same figure. This proves the excellent correlation between simulation and measurement.

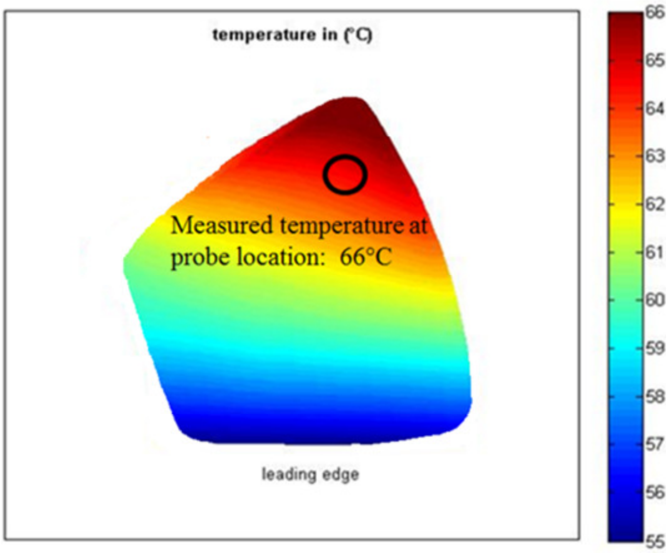

Figure 16: Calculated temperature distribution of one thrust pad for a rotational speed of 4000 rpm and a specific load of 303 psi (2,09 MPa) with water – glycol mixture as lubricant.

The measured and the calculated power loss of the complete thrust bearing assembly including the thrust collar are shown in Figure 17. This figure proves that the calculated power loss accurately reflects the measurement.

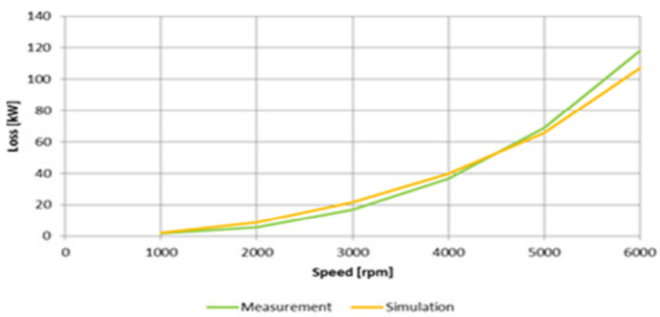

Figure 17: Comparison of measured power loss and the simulation of the complete thrust bearing assembly and the thrust collar at a specific load of 303 psi (2,09 MPa) with a water – glycol mixture as lubricant.

In order to verify the long term bearing performance the thrust bearing was operated for another 50 h at maximum load. Additionally the lubricant inlet temperature was increased to

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

122 °F (50°C) simulating realistic operating conditions for subsea pump applications.

As Zeidan et al. (1991) mentions, a fluid film bearing operating in the full hydrodynamic lubrication regime is expected to have an infinite life. Manufacturing and assembly flaws or an unsuitable bearing design are therefore detectable during the extended running hours and the several start-stops under load already.

After 250 consequent loaded start- stops the bearing was inspected. No signs of wear have been observed on the pads of the loaded and the unloaded TPTB and the thrust collar itself. Pad roughness measurements taken prior and after the tests are comparable. They proof that a proper fluid film was established during the whole test. A measurement of the pad thickness performed before and after the tests shows no deviations, proofing a proper hydrodynamic lubrication of the TPTB.

CONCLUSIONS

The excellent correlation between the simulated results and the measurements of the TPJB and TPTB gives a strong indication that solid polymer tilting pad bearings are, in general, suitable for subsea pumps or other applications with low viscous and lightly contaminated lubricants. A sufficient lubricant film thickness is generated by the bearing, allowing a safe operation within the whole operating range tested.

For a verification of the bearing performance due to variations of the lubricant supply temperature, long term tests at elevated lubrication inlet temperatures have been conducted with the TPTB. Additionally the bearings have run for another 50 operating hours in order to verify the long term performance of the solid polymer tilting thrust pad bearings.

Finally 250 start-stops under maximum load have been conducted, simulating a typical number of starts-stops the bearings are expected to be subjected to during a targeted MTBM of 5 years in the field. Both bearings, the loaded and the unloaded one, are in excellent condition after the tests. No wear was found on the pads. The TPTB tests show that these bearings are working as expected under the given conditions with respect to load, bearing temperature, lubricant flow, fluid film thickness and power loss.

The first tests performed with the TPJB show promising results. In order to fully qualify solid PEEK tilting pad bearings, further tests with a variation of lubricant inlet temperatures to be performed in order to cover all possible operating conditions, followed by long term tests and the 250 consequent start stops.

Once the solid polymer tilting pad bearings have passed the complete test program successfully, the bearings are considered qualified and can be used in pumps with low viscous, clean or lightly contaminated lubricants. The qualification test program is based on the requirements of the relevant applications and end-user specifications. It followed the majority of the bearing qualification tests conducted in the past.

NOMENCLATURE

API	= American Petroleum Institute	
F	= Bearing Load	[lbf]
HVOF	= High Velocity Oxygen Fuel	
ISO VG	= ISO Viscosity Class	
LBP	= Load Between Pad	
LOP	= Load on Pad	
MEG	= Mono Ethylene Glycol	
MTBF	= Mean Time between Failure	
MTBM	= Mean Time between Maintenance	
PEEK	= Poly Ether Ether-Ketone	
PPS	= PolyPhenylene Sulfide	
TEHD	= Thermo-Elasto-Hydrodynamic	
TPJB	= Tilting Pad Journal Bearing	
TPTB	= Tilting Pad Thrust Bearing	
x,y,z	= Coordinates	
ω	= Angular Velocity	[1/s]

REFERENCES

API Standard 610, “Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries”, 11th Edition, American Petroleum Institute, Washington, D.C.

API Standard 670, “Machinery Protection Systems”, 5th Edition, American Petroleum Institute, Washington, D.C.

Gassmann, S., Meuter, P., Felix, T. 2013, Patent Application EP2653736 A1.

Gerthsen, T., 2008, „Chemie für den Maschinenbau 2, Organische Chemie für Kraft- und Schmierstoffe Polymerchemie für Polymerwerkstoffe“, Universitäts-verlag Karlsruhe.

Fuchs, A. Glienicke, J., Schlums, H., 2005, „Programm-dokumentation zu ALP3T, Version 4.2, FVV Forschungs-vorhaben Nr. 662 Robuste Lagerungen“, Forschungs-vereinigung Verbrennungskraftmaschinen, Frankfurt am Main.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Hagemann, T., 2010, „Anlage zu ALP3T 4.2 Anleitung zur Dokumentation der erweiterten Berechnung- und Ausgabeoptionen des ALP3T 4.3.1“, TU Clausthal, Clausthal Zellerfeld.

Hagemann, T., Schwarze, H., 2011, „Verbesserte Radialgleitlagerberechnung – stationär und instationär hoch belasteter Radialgleitlager für schnell laufende Rotoren bei Berücksichtigung der Lagerdeformationen, FVA Forschungsheft 996“, Frankfurt am Main.

ISO Standard 4406, “Hydraulic fluid power – Fluids – Methods for coding the level of contamination by solid Particles”, 2nd Edition, 1999, International Organization for Standardization, Geneva

Kraft, C., 2014, „Anleitung zur Handhabung des Programms COMBROS-A“, TU Clausthal, Clausthal Zellerfeld.

Vance, J., Zeidan, F., Murphy, B., 2010, “Machinery vibration and rotordynamics”, John Wiley & Sons, Hoboken, New Jersey.

Zeidan, F., Herbage, S., 1991, „FLUID FILM BEARING FUNDAMENTALS AND FAILURE ANALYSIS”, Proceedings of the 20th Turbomachinery Symposium, Houston Texas.

ACKNOWLEDGEMENTS

The authors thank Frank Licht and Martin Swierczek of Greene Tweed for their continuous support especially during the design stage, Paul Meuter and Sjors Huijbers of Sulzer, Uwe Klein, Heike Müller, Horst Schackmann and John Whalen of John Crane for supporting this project.

Copyright© 2015 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station