Hydrodynamic Journal Bearing Optimization Based on Multidisciplinary Analysis of the Rotor-Bearing System for the Induction Motor

L.I. Moroz*, L.G. Romanenko*, R.E. Kochurov*, E.A. Kashtanov*, A. Nassar**
***SoftInWay Inc. 1500 District Ave, Burlington, MA 01803, USA**
****SoftInWay Turbomachinery Solutions Pvt. Ltd., Flat No. 301, Royal Manor, 70/10, Cunningham Road, Bangalore, India**

1.0 Introduction

High-performance machines usually operate at a high rotation speed and produce significant static and dynamic loads that acting on the bearings. Fluid film journal bearings play a significant role in machine overall reliability and rotor-bearing system vibration and performance characteristics. The increase of bearings complexity along with their applications severity make it challenging for the engineers to develop the optimal performance and reliable design. Bearing modeling should be based on accurate physical effects simulation. To ensure bearing reliable operation, the design should be performed based not only on simulation results for hydrodynamic bearing itself but also taking into the account rotor dynamics results for the particular rotor-bearing system, because the bearing characteristics significantly influence on the rotor vibration response.

Numbers of scientists and engineers have been involved in a journal bearing optimal design generation. A brief review of works dedicated to various aspects of bearing optimization is presented in [1]. Journal bearings optimizations based on genetic algorithms are considered in [2-5]. A numerical evolutionary strategy and an experimental optimization on a lab test rig were applied to get the optimal design of a tilting pad journal bearing for an integrally geared compressor in [6]. Optimal journal bearing design selection procedure for a large turbocharger is described in [7]. In this study power loss, rotor dynamics instability, manufacturing and economics restrictions are analyzed. To optimize the oil film thickness with satisfying the condition of maximizing the pressure in three lobes bearing, the multi-objective genetic algorithm was used in [8].

Conjugated optimization for the entire rotor-bearing system is a challenging task due to various conflicting design requirements, which should be fulfilled. In [9] parameters of rotor-bearing systems are optimized simultaneously. The design objective was the minimization of power loss in bearings with constraints on system stability, unbalance sensitivities, and bearing temperatures. Two heuristic optimization algorithms: genetic and particle-swarm optimization were employed in the automatic design process.

There are several objective functions that are considered by researchers to optimize bearing geometry, such as

- Optimum load carrying capacity [5];
- Minimum oil film thickness and bearing clearance optimization [1, 6, 8];
- Power losses minimization [6, 9];
- Rotor dynamics restrictions;
- Manufacturing, reliability and economics restrictions [7].

The most common design variables which are considered in reviewed works are clearance, bearing length, diameter, oil viscosity, film pressure, and many others.

Finding the optimal load carrying capacity or the minimum power loss together with the entire rotor-bearing system operating restrictions, require to employ optimization techniques, because accounting the effects from all considered parameters significantly enlarge the analysis process. The whole optimization process is an example of Multidisciplinary Optimization (MDO) with following disciplines involved: fluid flow, heat transfer, mechanics, elasticity, and tribology. Several numerical methods, such as finite difference and finite elements usually are employed to solve this complex problem and calculation process can sometimes be time-consuming and takes a large amount of computing capacity. Therefore, to leverage MDO of hydrodynamic bearings an efficient algorithms are needed.

In this study, optimal design of the hydrodynamic journal bearings for 13.5 MW induction motor is generated based on multidisciplinary analysis of the rotor-bearing system for the induction motor. This analysis consists of bearing hydrodynamic characteristics simulation/optimization and rotor dynamics analysis for a rotor-bearing system. The optimization procedure is based on the design of experiment methodology (DoE) and Random Best Successions (RBS) method.

2.0 Problem Formulation

The goal of the work is to optimize plain cylindrical hydrodynamic journal bearings for newly designed 13.5 MW induction motor (fig. 1). The motor nominal rotating speed is 1950 rpm. The mass of the rotor is 6500 kg.

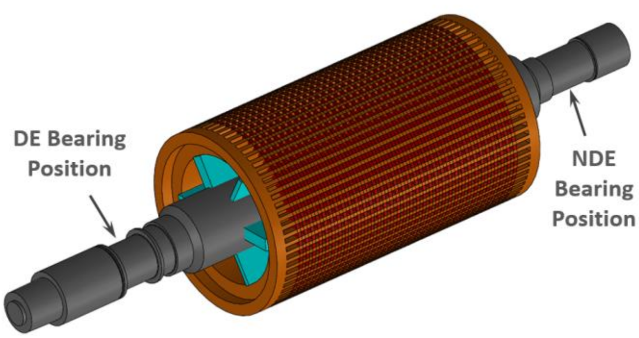
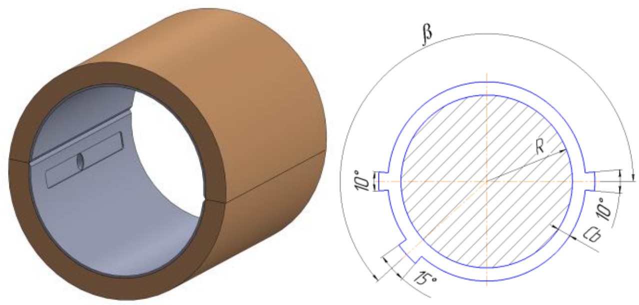


Fig.1 – Induction Motor Model

The initial design of the DE (drive end) bearing is presented in fig. 2. For initial design bearing loads are 35640 N for DE and 28060 N for NDE (non-drive end) bearing.



a) b)

Fig.2 – Induction Motor DE Bearing CAD Model (a) and Varying Parameters Scheme (b)

The bearing initial geometry parameters are listed in table 1 (see fig.2b).

Table 1 – Initial Bearing's Parameters

Parameter		Units	DE	NDE
Bearing Length	L	m	0.1404	0.1685
Shaft Radius	R	m	0.1	0.1125
Clearance	C_b	µm	304	340
Pocket position	β	deg	210	210
Pocket width	dβ	deg	15	15

The rotor-bearing system is supported by the pedestal, which was taking into the account in rotor dynamics analysis and modeled with equivalent characteristics – see calculation scheme in the fig. 3.

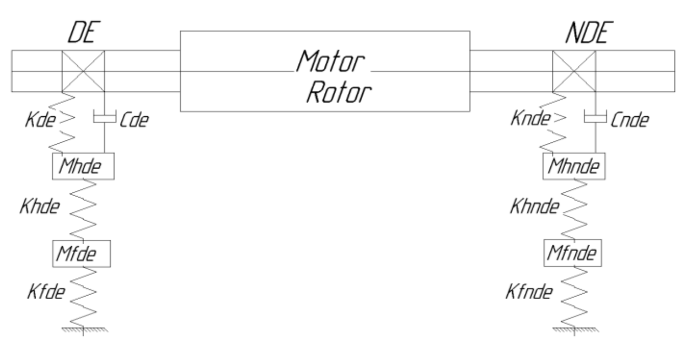


Fig. 3 – Rotor-Bearing System with Supports (Calculation Scheme)

Bearing housing and pedestal characteristics are presented in table 2 below.

Table 2 – Bearing housing and pedestal structure equivalent characteristic

Parameter	Mass, M	Kxx*e-6, K	Kyy*e-6, K
	kg	N/m	N/m
Mhde/Khde	425	2600	3250
Mfde/Kfde	8700	646	313
Mhnde/Khnde	230	1650	2150
Mfnde/ Kfnde	8700	646	313

In the table before:

Mh – mass for DE/NDE bearings housing;
Mf – mass for DE/NDE foundation;
Kh – stiffness for DE/NDE bearing housing;
Kh – stiffness for DE/NDE foundation;

Despite, the preliminary design for the induction motor is complete (fig.1), to optimize bearings, the following parameters for each bearing will be varied in some range:

1) Bearing clearance;
2) Bearing pocket positions (oil supply pocket);
3) Radii for bearing liner and the shaft;
4) Length for bearing liner and the shaft;

The goals of optimization are to increase reliability and efficiency of the rotor-bearing system. The following objective functions and constraints have to be considered for this task:

1) Minimal oil film thickness – the most critical parameter which influences on the bearing reliability. When its value drops below either the journal surface roughness or the value of the geometric distortions, the hydrodynamic lubrication changes to a mixed lubrication regime, which is characterized by metal-to-metal contact. In order to prevent undesirable metal-to-metal contact, the clearance should be designed to produce the maximum possible level of minimum oil film thickness;
2) Power loss – minimization of this function lead to machine efficiency increase;
3) Rotor-bearing system critical speeds – the lateral rotor dynamics constraint which should be taken into the account at the design phase to avoid possible vibration problems.

Finally, to confirm reliability and to ensure safe operation for the rotor-bearing system with optimized journal bearings, the full scope of rotor dynamics checking analyses should be performed.

3.0 Bearing Optimization Procedure

Bearing optimization procedure has been automated within supervisor program that organizes interaction between other modules for

- optimization (DoE based);
- bearing simulation
- rotor dynamics analyses.

The principal algorithm for optimization is shown in Fig. 4.

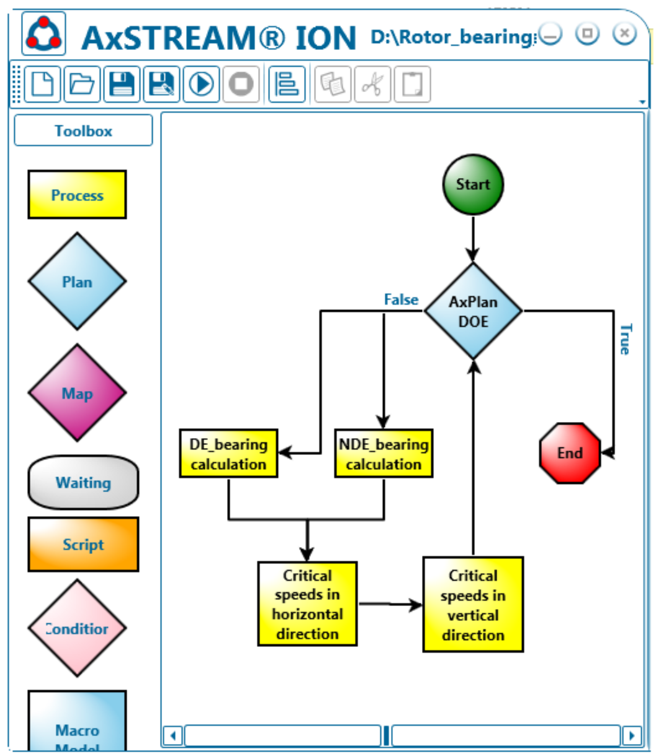


Fig.4 – Optimization Algorithm within Supervisor Program

At the first step of optimization, Rechtschaffner approximation plan [10] for 8 independent variables (45 computational points) was created and used for the numerical experiment.

For each point, the DE/NDE bearings and rotor dynamics results were calculated to fill the plan. Bearings simulation is performed based on finite difference method utilizing parametrical models shown in fig. 2b. Rotor dynamics FE model, which is participated in optimization process is shown in fig. 5 and created based on beam Timoshenko theory.

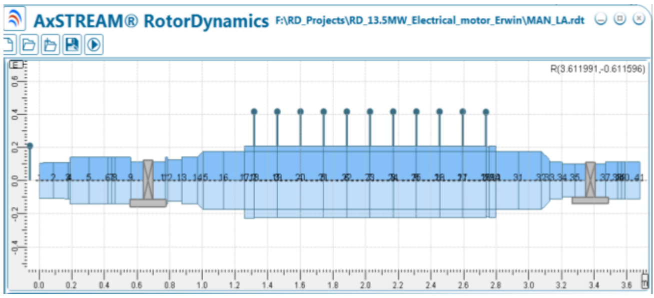


Fig.5 – Rotor Dynamics FE model

A good practice for safe rotor design typically involves the avoidance of any resonance situation at operating speeds with some margins including resonances by a reason of structural supports vibration. If the design can't be changed to satisfy frequency margins, the vibration amplitudes and forces acting on the bearings have to be measured and analyzed according to standards or company design practice [11].

Rotor dynamics analysis for the initial design shows that first two modes (bending and conical) have critical speeds with separation margins 29.4% and 16.8% from nominal operating speed (see fig. 6).

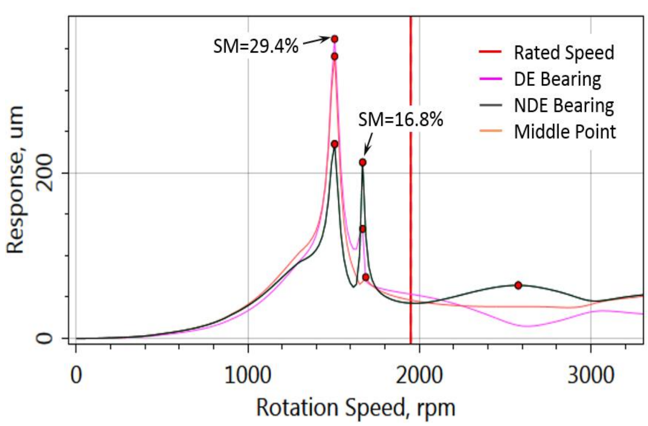
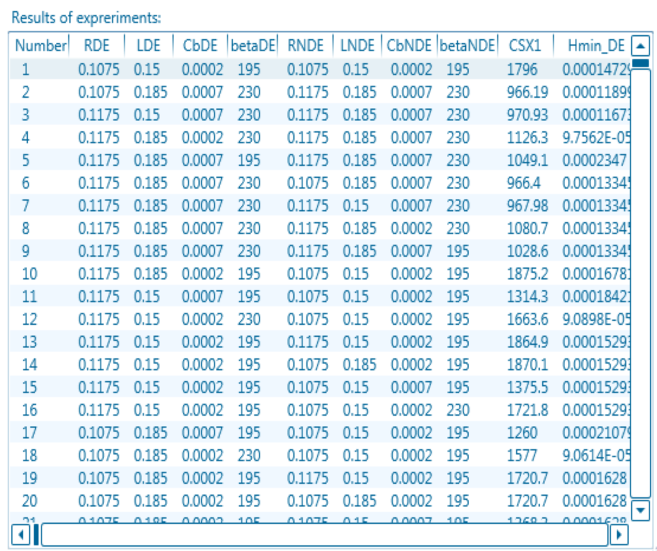


Fig. 6 – Frequency Response Characteristics for Initial Rotor-Bearing System Design

To eliminate potential resonance vibration problems the allowable range for critical speeds were imposed to provide separation margin higher than 16%. It is significant that restrictions were applied for critical speeds calculated in each plane – vibrations in horizontal and vertical directions.

Following the procedure presented in fig. 4, DE and NDE bearings were calculated separately within optimization algorithm and the bearings' characteristics then exported in RotorDynamics solver. Critical speeds simulation was performed in the RotorDynamics solver based on calculated bearings' characteristics.

The results of simulations for each point of DoE plan are presented in fig. 7.



Results of exprriments:

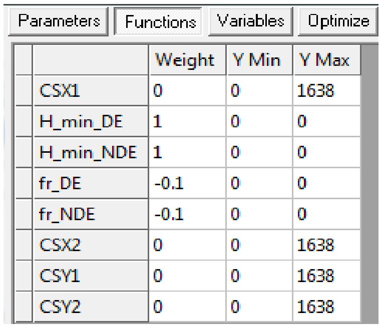
Number	RDE	LDE	CbDE	betaDE	RNDE	LNDE	CbNDE	betaNDE	CSX1	Hmin_DE
1	0.1075	0.15	0.0002	195	0.1075	0.15	0.0002	195	1796	0.0001472
2	0.1075	0.185	0.0007	230	0.1175	0.185	0.0007	230	966.19	0.0001189
3	0.1175	0.15	0.0007	230	0.1175	0.185	0.0007	230	970.93	0.0001167
4	0.1175	0.185	0.0002	230	0.1175	0.185	0.0007	230	1126.3	9.7562E-05
5	0.1175	0.185	0.0007	195	0.1175	0.185	0.0007	230	1049.1	0.0002347
6	0.1175	0.185	0.0007	230	0.1075	0.185	0.0007	230	966.4	0.0001334
7	0.1175	0.185	0.0007	230	0.1175	0.15	0.0007	230	967.98	0.0001334
8	0.1175	0.185	0.0007	230	0.1175	0.185	0.0002	230	1080.7	0.0001334
9	0.1175	0.185	0.0007	230	0.1175	0.185	0.0007	195	1028.6	0.0001334
10	0.1175	0.185	0.0002	195	0.1075	0.15	0.0002	195	1875.2	0.0001678
11	0.1175	0.15	0.0007	195	0.1075	0.15	0.0002	195	1314.3	0.0001842
12	0.1175	0.15	0.0002	230	0.1075	0.15	0.0002	195	1663.6	9.0898E-05
13	0.1175	0.15	0.0002	195	0.1175	0.15	0.0002	195	1864.9	0.0001529
14	0.1175	0.15	0.0002	195	0.1075	0.185	0.0002	195	1870.1	0.0001529
15	0.1175	0.15	0.0002	195	0.1075	0.15	0.0007	195	1375.5	0.0001529
16	0.1175	0.15	0.0002	195	0.1075	0.15	0.0002	230	1721.8	0.0001529
17	0.1075	0.185	0.0007	195	0.1075	0.15	0.0002	195	1260	0.0002107
18	0.1075	0.185	0.0002	230	0.1075	0.15	0.0002	195	1577	9.0614E-05
19	0.1075	0.185	0.0002	195	0.1175	0.15	0.0002	195	1720.7	0.0001628
20	0.1075	0.185	0.0002	195	0.1075	0.185	0.0002	195	1720.7	0.0001628

Fig. 7 – Parameters and Computation Results

Optimization with macro models assumes searching for a maximum of response functions linear combination with weight coefficients assigned. If the latter equal zero, the constraints are imposed on that function's range of variation. The constraints are also imposed on the ranges of dependent variables variation. At that, it is recommended that those ranges will remain within the region of the design of the experiment.

The engineer should make a decision what is more important – highest load capacity or lower friction in bearing construction.

For the current optimization task, weight coefficients were chosen to maximize minimal oil film thickness with high priority for DE and NDE bearings (weight coefficient is equal 1 – see table in fig. 8). Friction losses optimization was performed with lower priority (weight coefficient is equal 0.1).

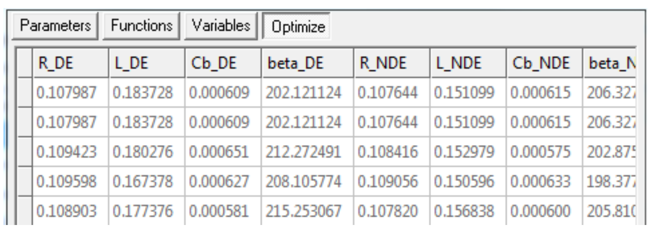


Parameters | Functions | Variables | Optimize

	Weight	Y Min	Y Max
CSX1	0	0	1638
H_min_DE	1	0	0
H_min_NDE	1	0	0
fr_DE	-0.1	0	0
fr_NDE	-0.1	0	0
CSX2	0	0	1638
CSY1	0	0	1638
CSY2	0	0	1638

Fig. 8 – Weight Coefficients and Constraints

Searching for extremum in the region of constraints is conducted using a variant of quasi-random search that features high uniformity of computational points' distribution in the variables space, so called Random Best Succession (RBS) search. Five best solutions for the optimization task are presented in the fig. 9.



Parameters | Functions | Variables | Optimize

R_DE	L_DE	Cb_DE	beta_DE	R_NDE	L_NDE	Cb_NDE	beta_N
0.107987	0.183728	0.000609	202.121124	0.107644	0.151099	0.000615	206.327
0.107987	0.183728	0.000609	202.121124	0.107644	0.151099	0.000615	206.327
0.109423	0.180276	0.000651	212.272491	0.108416	0.152979	0.000575	202.875
0.109598	0.167378	0.000627	208.105774	0.109056	0.150596	0.000633	198.377
0.108903	0.177376	0.000581	215.253067	0.107820	0.156838	0.000600	205.81

Fig. 9 – Five Best Solutions

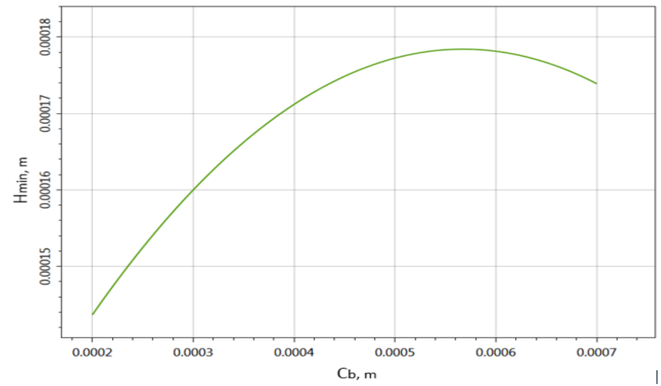
Dependence on the response functions and variable parameters can be displayed on charts. In fig. 10 we are able to see how minimum oil film thickness is changed vs. clearance increase for DE bearing for the point with the 1st solution from fig. 10a.

Friction coefficient vs. clearance plot for DE bearing is presented in fig. 10b for the 1st solution from fig. 9.

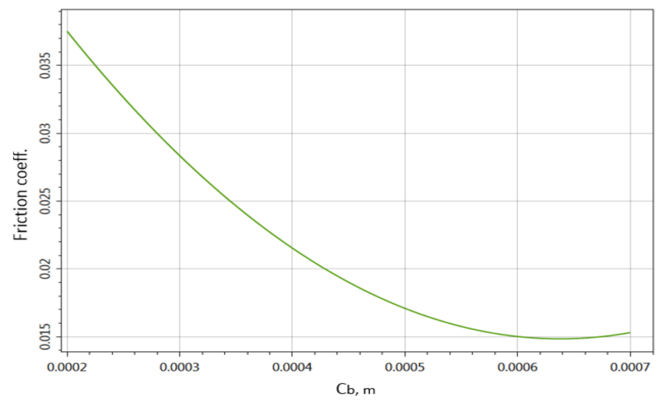
Verification reveals that function values at the optimum point obtained with macro models and original models have a good convergence – see table 3.

Table 3 – Results Comparison

	Bearing	Units	Macro Model	Bearing Module
Minimal Oil Film Thickness	DE	um	205	204
	NDE	um	190	187
Friction Coefficient	DE	-	0.0144	0.0151
	NDE	-	0.0132	0.0141



a)



b)

Fig. 10 – Objective Functions vs. Clearance:
(a) Minimum Oil Film Thickness,
(b) Friction Coefficient

4.0 Optimization Results Discussion

Initial and optimized bearings parameters are presented in table 4 for DE bearing and in table 5 for NDE bearing.

Table 4 – DE Bearing Designs Parameters

DE bearing		Initial	Optimized
Min oil film thickness	µm	**132**	**204**
Friction coefficient	-	**0.018**	**0.016**
Power loss	W	11861	11429
Radial clearance	µm	304	602
Shaft radius	mm	100	108
Bearing length	mm	140.4	184
Pocket angle	deg	210	202
Oil temp. increment	deg, C	96.3	29.7
Oil Consumption	m^3/h	7.7	1.14
Maximum oil pressure	MPa	4.2	2.80

Table 5 – NDE Bearing Designs Parameters

NDE bearing		Initial	Optimized
Min oil film thickness	µm	**188**	**187**
Friction coefficient	-	**0.029**	**0.017**
Power loss	W	17119	9631
Radial clearance	µm	340	615
Shaft radius	mm	112.5	108
Bearing length	mm	168.5	151
Pocket angle	deg	210	206
Oil temp. increment	deg, C	118.2	23.3
Oil Consumption	m^3/h	9.0	2.60
Maximum oil pressure	MPa	2.3	3.00

Comparison for initial and optimized bearings shows that friction power losses for DE and NDE bearings were decreased significantly (by 38% for the entire system). Minimal oil film thickness is increased for DE bearing and remain the same for NDE bearing.

To check rotor-bearing system separation margins for optimized design, unbalance response analysis was performed.

Frequency response characteristics for the optimized system are presented in fig. 11.

The minimal separation margin for the optimized case is 16.8%, which is acceptable [11]. It also can be observed that vibration amplitudes for the optimized case are decreased, which is lead to smoother operation for the motor.

Basing on the presented results we can make the conclusion that optimized design is preferable for the motor application.

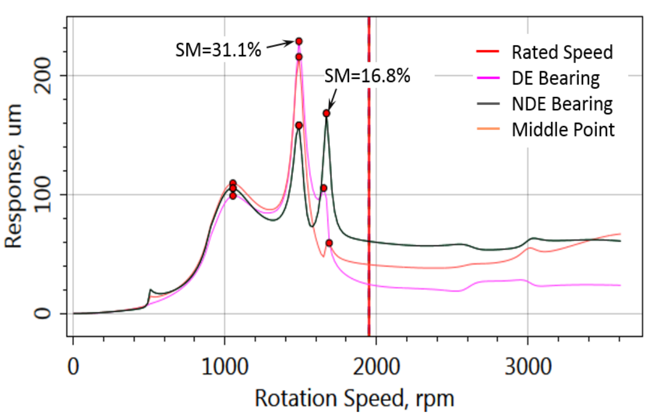


Fig. 11 – Frequency Response Characteristics for Optimized Rotor-Bearing System Design

5.0 Conclusions

As a result of the study, the approach, which allows to find optimal bearing design, taking into the account the effect of bearing geometry and rotor dynamics restrictions is developed.

The optimized DE and NDE bearings were generated for 13.5 MW induction motor and show a significant decrease of the friction power loss along with maximized oil film thickness.

REFERENCES

[1] Sutariya C., Tamboli K., George P. M., 2011, “Optimum Design of A Journal Bearing – A Review”, *National Conference on Recent Trends in Engineering & Technology*, India.

[2] Weißbacher, C., 2014, “Optimization of Journal Bearing Profiles With Respect to Stiffness and Load-Carrying Capacity”, Wiley-Verlag, New York, USA.

[3] Hirania H., Suhb N.P., 2004, “Journal bearing design using a multiobjective genetic algorithm and axiomatic design approaches”, *Tribology International*, United Kingdom.

[4] Saruhan H., Rouch K.E., Roso C.A, 2002 “Design Optimization of Tilting-Pad Journal Bearing using a Genetic Algorithm”, *International journal of rotating machinery,* London, United Kingdom.

[5] Amit S., Prakash P., Bhargirath P., 2014, ”Design Formulation and Optimum Load Carrying Capacity of Hydrodynamic Journal Bearing By using Genetic Algorithm”, *International journal for research in applied science and engineering technology*, Faridabad, India.

[6] Havlik N., Lutz M., 2015, “Optimization of tilting-pad journal bearings for integrally geared compressors”, *ASME Turbo Expo*, Montreal, Canada.

[7] Cao J., Dimond T., 2015, “Reduction of vibration and power loss in industrial turbochargers with improved tilting pad bearing design”, *ASME Turbo Expo*, Montreal, Canada.

[8] Biswas N., Chakraborti P., Dhar P., 2016, “Optimization of pressure and oil film

thickness in Mulilobe Bearing using Response Surface Methodology and Moga"

[9] Untaroiu C.D., Untaroiu A., 2010, "Constrained Design Optimization of Rotor-Tilting Pad Bearing Systems", *Journal of Engineering for Gas Turbines and Power.*

[10] Rechtschaffner R.L., Saturated fractions of 2n and 3n factorial designs, Technometrics, 1967, Vol.9, N°4, 569-575.

[11] API, 684, 2005. *API recommended practice*, American petroleum institute, Washington, USA.