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MODELLING METHOD OF MECHANICAL PROPERTIES OF TILTING PAD BEARINGS

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Appendix 1: Example MATLAB code from the case A1 comparison

ABBREVIATIONS AND SYMBOLS

Abbreviations

BDC	Bearing Data Calculation, a MATLAB interpolation tool for RobeDyn
LBP	Load between pads
LOP	Load on pad
RoBeDyn	Rotor-Bearing Dynamics Toolbox for MATLAB
TPB	Tilting pad bearing

Roman symbols

B	nominal width of the bearing	$[m]$
$\frac{B}{D}$	geometric ratio	$[-]$
$c^*_{i,k}$	dimensionless stiffness coefficient	$[-]$
$c_{i,k}$	stiffness coefficient	$[\frac{N}{m}]$
$d^*_{i,k}$	dimensionless damping coefficient	$[-]$
$d_{i,k}$	damping coefficient	$[\frac{Ns}{m}]$
D	nominal diameter of bearing	$[m]$
D_J	nominal diameter of shaft	$[m]$
e_B	eccentricity	$[m]$
F	radial force	$[N]$
m	preload	$[-]$
$\frac{\Delta R_B}{C_R}$	profiling	$[-]$
So	Sommerfeld number	$[-]$
T	lubricant temperature	$[°C]$
VG	viscosity class of lubricant	$[-]$
Z	number of pads	$[-]$

Greek symbols

η_{eff}	lubricant dynamic viscosity	$[Pas]$
ρ	lubricant density	$[\frac{kg}{m^3}]$
$\varphi_{F,1}$	pivot angle	$[°]$
ψ_{eff}	relative clearance	$[\%.]$
Ω_F	pad bracket angle	$[°]$
Ω	pad central angle	$[°]$
$\frac{\Omega_F}{\Omega}$	pivot offset	$[-]$
ω_J	angular velocity	$[\frac{rad}{s}]$

1 INTRODUCTION

1.1 Background

Tilting pad bearings (TPBs) are commonly used in high speed applications. Their ability to endure heavy loads at high speeds makes them a better choice than conventional rolling element bearings to electrical or mechanical high speed machine. Basic structure of TPB is introduced in Fig 1. A main point in this thesis is a hydrodynamic plain journal bearing standard DIN 31657. This standard provides a number of bearing property tables under various types of loads. Using equations provided in this standard the dynamic properties of nearly every tilting pad bearing can be calculated.

Figure 1. Structure of radial tilting pad bearing [1]

1.2 Objectives and restrictions

In this thesis under investigation is a method of modelling mechanical properties of tilting pad bearings which is based on a theory of hydrodynamic fluid film bearings. A modelling tool is used to compare similarities between modelled and physical bearing properties. In order to make valid comparison to physical bearing properties, bearing coefficient tables must be interpolated. Therefore, a special interpolation tool is programmed to interpolate bearing characteristics provided by DIN 31657 standard. As results of this thesis, bearing properties such as stiffness and damping coefficients of tilting pad bearings are being calculated using specially designed MATLAB code. A bearing coefficient comparison to physical bearings is calculated using manufacturer bearing data to verify results.

Two separate bearing coefficient data sets are obtained by two major bearing manufacturers John Crane and Waukesha. Due to the limited data from these manufacturers the bearing property comparison in this thesis is restricted to four comparisons where rotational speed and shaft load is being varied.

1.3 Structure of the thesis

In section 2 is introduced a theory fluid film bearings and a modelling method for calculating mechanical properties of tilting pad bearings. Section 3 includes manufacturer bearing property tables and the results of bearing coefficient comparison. In section 4 is presented discussion of results and possible causes of variation in results.

2 TILTING PAD BEARINGS

2.1 Structure of tilting pad bearings

Technically tilting pad bearings are so called hydrodynamic fluid film bearings. They are most common bearings in rotating machinery. The load of shaft is carried by the fluid film pressure which is produced by speed difference between shaft surface and pad geometry of bearing. Because there is no contact between the shaft and the bearing fluid film bearings can have infinite life if the lubricant is kept clean and operation conditions are in safe dynamic range. Tilting pad bearings requires continuous lubrication, therefore a separate lubrication system is needed. This system works also as cooling system for the bearing. Commonly fluid film bearings are categorized as fixed geometry sleeve bearings and variable geometry tilting pad bearings. An example of variable geometry tilting pad bearing is shown in Fig 2. [2, 3]

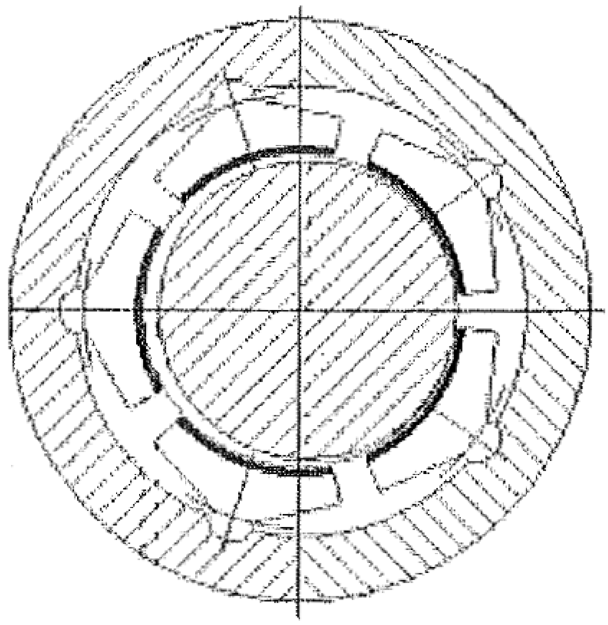


Figure 2. Variable geometry tilting pad bearing [2]

According to DIN 31657-3 standard tilting pad bearing consists of four or five pads. The pads can be flexible or rocker back type as shown in Fig 3. The number of pads and load direction can have an impact to the stiffness and damping coefficients. [3, 4]

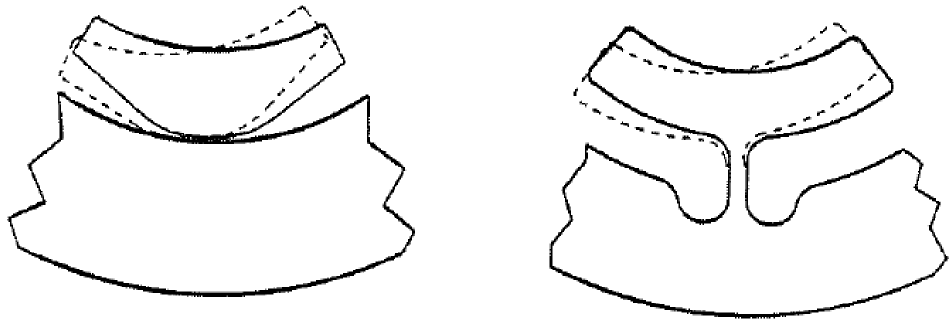


Figure 3. On the left a principle of rocker back tilting pad and on the right flexure pivot tilting pad [3]

2.2 Theory of tilting pad bearings

2.2.1 Influence of load direction

According to TPB geometry the shaft load can influence in different positions of bearing. Two typical situations are load on pad (LOP) and load between pads (LBP). The orientation of a tilting pad with respect to radial load is important since it directly affects to the stiffness and damping coefficients of the bearing.

In case of load on pad the bearing stiffness coefficients are asymmetry i.e. stiffness in direction of load is significantly larger than in the perpendicular direction to the load. In case of load between pads the horizontal and vertical stiffness coefficients are almost identical. Fig. 4 illustrates these two loading conditions.

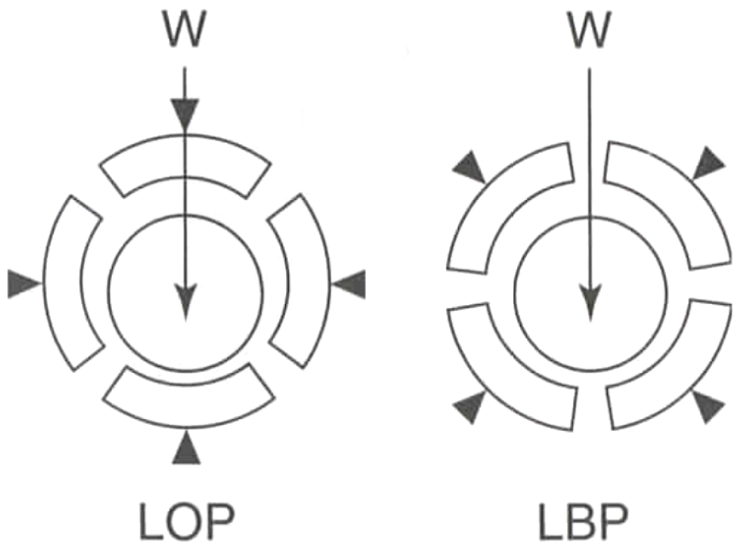


Figure 4. Two loading cases: load on pad and load between pads [3]

2.2.2 Influence of preload

The geometric preload in tilting pad bearings is an important parameter that affects to the magnitude of the bearings coefficients. Graphical representation of preload is shown in Fig. 5. Typical preload for the tilting pad bearing is from 0.15 – 0.75. Increase in preload will results to higher stiffness values especially at lower Sommerfeld numbers. Negative preload should be avoided since it will prevent oil entering the leading edge of the bearing and will cause flutter to the pad. Therefore, due to manufacturing and stack-up tolerance, a positive preload should be determined to avoid negative preload. Typically by increasing the bearing stiffness results to lower effective damping. [2]

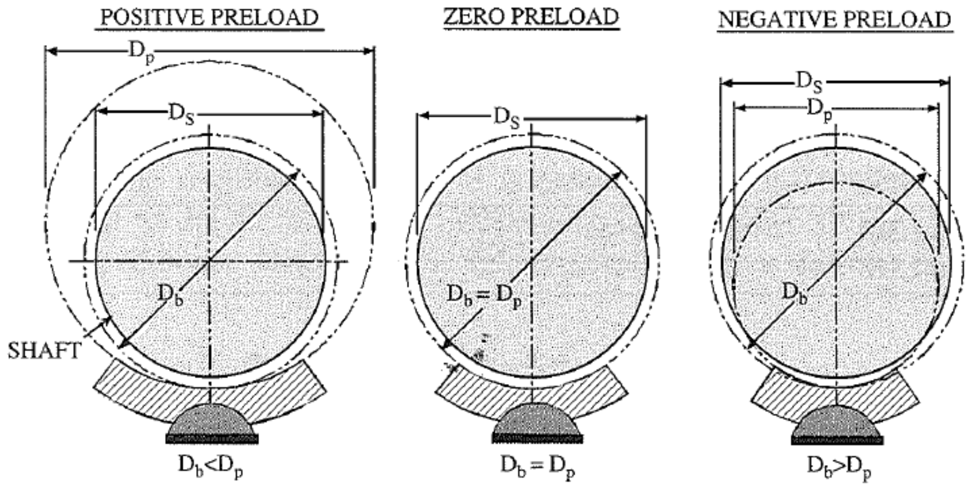


Figure 5. Schematic of determining preload factor [3]

2.2.3 Introduction to DIN 31657 standard

The first section of the DIN 31657-1 standard contains theory of calculating fluid film bearing properties. Reynold's equation is of which is the basis of calculating bearing properties is introduced in this part of the standard. Sections 2 and 3 of DIN 31657 contains calculated dimensionless coefficients of different types of bearings in various operating conditions. In table 1 is presented bearing coefficient data extracted from DIN 31657-3 Tablette 1. [4, 5, 6]

Table 1. Example of one DIN 31657-3 coefficient table [6]

So	c^*_{11}	c^*_{12}	c^*_{21}	c^*_{22}	d^*_{11}	d^*_{12}	d^*_{21}	d^*_{22}
0.000	0.283	0.000	0.000	0.283	0.335	0.000	0.000	0.335
0.021	0.286	0.000	0.000	0.286	0.337	0.000	0.000	0.337
0.042	0.295	0.000	0.000	0.295	0.342	0.000	0.000	0.342
0.086	0.330	0.000	0.000	0.330	0.362	0.000	0.000	0.362
0.136	0.397	0.000	0.000	0.397	0.399	0.000	0.000	0.399
0.197	0.505	0.000	0.000	0.505	0.456	0.000	0.000	0.456
0.273	0.678	0.000	0.000	0.678	0.546	0.000	0.000	0.546
0.374	0.954	0.000	0.000	0.954	0.686	0.000	0.000	0.686
0.512	1.403	0.000	0.000	1.403	0.905	0.000	0.000	0.905
0.713	2.167	0.000	0.000	2.167	1.253	0.000	0.000	1.253
1.023	3.562	0.000	0.000	3.562	1.841	0.000	0.000	1.841
1.245	4.707	0.000	0.000	4.707	2.299	0.000	0.000	2.299
1.541	6.389	0.000	0.000	6.389	2.931	0.000	0.000	2.931
1.943	8.951	0.000	0.000	8.951	3.848	0.000	0.000	3.848
2.512	13.077	0.000	0.000	13.077	5.241	0.000	0.000	5.241
3.358	20.191	0.000	0.000	20.191	7.500	0.000	0.000	7.500
4.708	33.852	0.000	0.000	33.852	11.308	0.000	0.000	11.308
7.086	63.748	0.000	0.000	63.748	19.069	0.000	0.000	19.069
12.031	148.095	0.000	0.000	148.095	37.650	0.000	0.000	37.650
26.371	535.824	0.000	0.000	535.824	107.479	0.000	0.000	107.478
58.756	2141.057	0.000	0.000	2141.055	306.671	0.000	0.000	306.670

$$So = \frac{F\psi^2_{eff}}{BD\eta_{eff}\omega_J} \tag{1}$$

where So is Sommerfeld number, F radial force, ψ_{eff} relative clearance, B nominal width of the bearing, D nominal diameter of the bearing, η_{eff} dynamic viscosity of the lubricant and ω_J angular velocity, respectively [4].

Eg. 1 describes Sommerfeld number which integrates dimensional stiffness and damping coefficients to the dimensionless coefficients presented in tables of the standard. Therefore Sommerfeld number has no dimension. In case of manufacturer data sometimes Sommmerfeld number can be pre-calculated from measured values. [4]

$$c^*_{i,k} = \frac{\psi^3_{eff}}{2B\eta_{eff}\omega_J} c_{i,k}, (i, k = 1, 2) \tag{2}$$

where $c^*_{i,k}$ is dimensionless stiffness coefficient, ψ_{eff} relative clearance, B nominal width of the bearing, η_{eff} dynamic viscosity of the lubricant, ω_J angular velocity and $c_{i,k}$ stiffness coefficient with dimension, respectively [4].

Eq. 2 describes connection between dimensionless and dimensional stiffness component. There are altogether four components: two axial and two cross-coupled components. In case of tilting pad bearings these cross-coupled components are usually so small that their effect can be neglected. [4]

$$d^*_{i,k} = \frac{\psi^3_{eff}}{2B\eta_{eff}} d_{i,k}, (i, k = 1, 2) \tag{3}$$

where $d^*_{i,k}$ is dimensionless damping coefficient, ψ_{eff} relative clearance, B nominal width of the bearing, η_{eff} dynamic viscosity of the lubricant and $d_{i,k}$ damping coefficient with dimension, respectively [4].

Eq. 3 describes connection between dimensionless and dimensional damping components. There are also four different components. [4]

$$\eta_{eff} = \eta_x e^{\left(\frac{159.56}{T + 95°C}^{-0.181913}\right) log \frac{\rho VG}{10^6 \eta_x}} \tag{4}$$

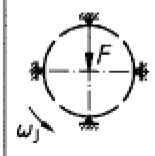
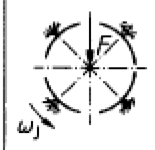
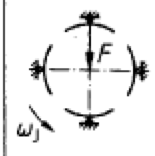
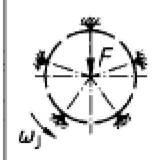
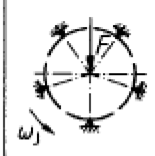
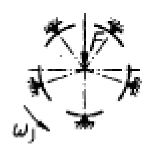
where $\eta_x = 0.18 * 10^{-3} Pas$, T lubricant temperature in celsius, ρ density of the lubricant and VG viscosity class of the lubricant, respectively [4].

Eq. 4 describes how the lubricant viscosity can be calculated if it is not known. Eq. 1-4 are used to describe mechanical properties of tilting pad bearings.

2.2.4 Characteristic values describing properties of tilting pad bearings

According to DIN 31657 standard, six values are needed to identify a specific tilting pad bearing and a load configuration. Table 2 shows a categorized presentation of the first five characteristic values. According to DIN 31657-3 the sixth value, pivot offset, can have two different values. In other words, every configuration presented in Table 2 can have both pivot offset values. [6]

Table 2. Characteristic values of TPBs in DIN 31657-3 [6]

Z	Ω [°]	$\varphi_{F,1}$ [°]	$\Delta R_B/C_R$	B/D	Lastfall
4	80	45	2, 3, 5	0,5, 0,75, 1	
		0	3	0,75	
	60	45	2, 3, 5	0,5, 0,75	
		0	3	0,5	
5	60	36	2, 3, 5	0,5, 0,75	
		0	3	0,5	
	45	36	2, 3, 5	0,5	
		0	3	0,5	

The six characteristic values are:

- Z Number of pads. With tilting pad bearings standard provides tables for 4 to 5 pads.
- $\frac{B}{D}$ Geometric ratio of pad length and bearing inner diameter.
- Ω Geometrical central angle of pad. Product of central angle and number of pads is less than complete circle because there are gaps between pads.
- $\varphi_{F,1}$ Pivot angle of pad with respect to radial load. Direction of load affects to stiffness and damping coefficients.
- $\frac{\Delta R_B}{C_R}$ Profiling.

$$\frac{\Delta R_B}{C_R} = 1 + \frac{e_B}{\frac{D - D_J}{2}} \tag{5}$$

where e_B is eccentricity, D is nominal diameter of bearing and D_J is nominal diameter of shaft [4]. Profiling describes relative clearance of bearing and shaft caused by eccentricity between shaft and bearing. It can be calculated using Eq. 5 Fig. 6 illustrates geometrical variables used in equation.

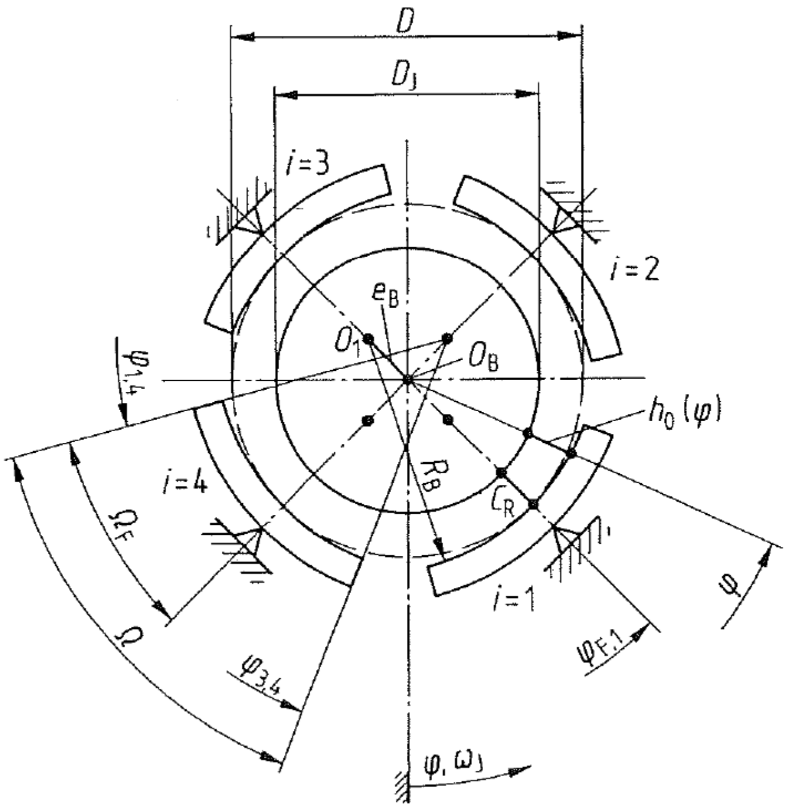


Figure 6. Geometrical variables of TPB [4]

$$\frac{\Delta R_B}{C_R} = \frac{1}{1-m} \tag{6}$$

where m is preload values [4]. Sometimes manufacturer announces a preload value instead of eccentricity. Therefore profiling can be calculated using Eq 6. In this case manufacturer usually calculates preload value using profiling.

- $\dfrac{\Omega_F}{\Omega}$ Pivot offset. Describes the position of pivot with respect to pad. Numerical value can be either 0.5, which means that bearing can operate both directions of rotation or otherwise 0.6 when rotating to both directions is restrained. Principle of pivot offset is introduced in Fig. 7.

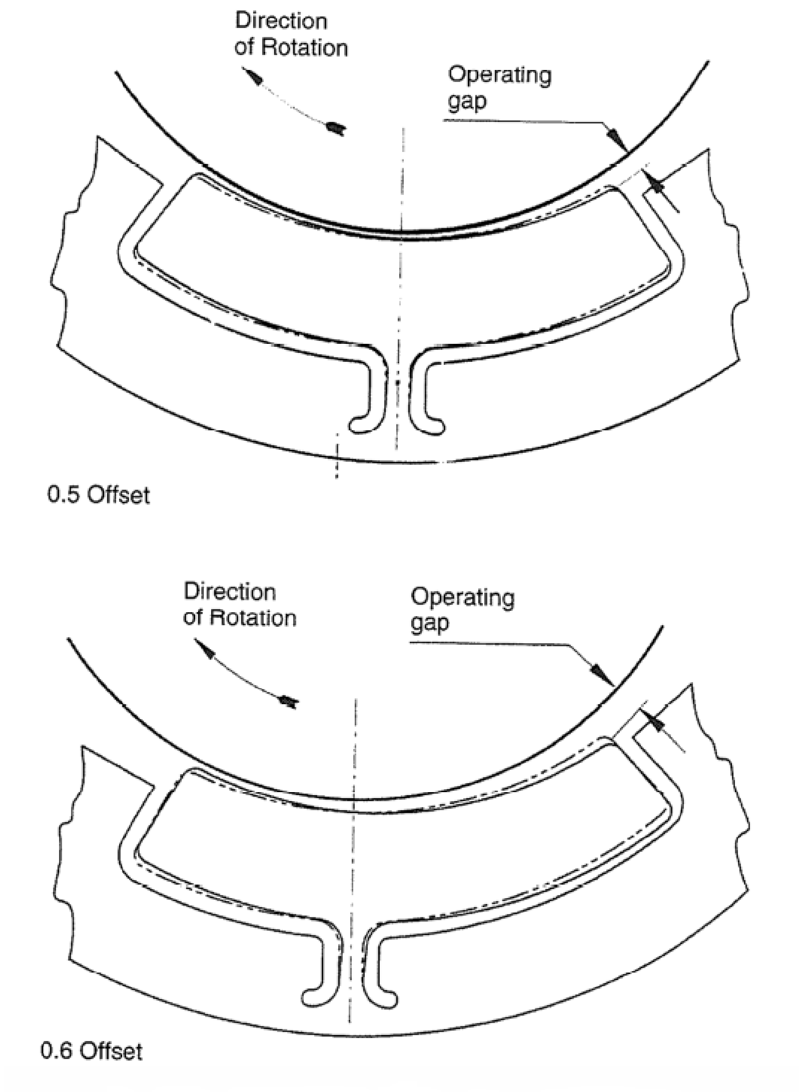


Figure 7. Pivot offset [3]

Using the six characteristic values and Eq. 1-6 the dynamic properties of the tilting pad bearings can be calculated. Variables that can have an impact to the dynamic properties are load, oil viscosity with respect to operationg temperature, relative clearance between the shaft and the bearing and angular velocity. Considering all characteristic values, oil viscosity can have greatest impact. [4, 6]

2.3 Introduction to the Bearing Data Calculation tool

2.3.1 Overview of bearing data interpolation package

Bearing Data Calculation (BDC) package operates in MATLAB as part of RoBeDyn toolbox for rotor dynamic analyses. Latest version of RoBeDyn is designed in Lappeenranta University of Technology by Professor Jussi Sopanen in the year 2009. BDC package provides interpolated dimensionless bearing stiffness and damping coefficients using requested bearing properties or so called characteristic values. Fig. 8 represents basic functionality of BDC. User requested bearing properties are entered using structural array in MATLAB. BDC algorithm defines suitable bearing coefficient tables that are provided in DIN 31657 standard for interpolation. Depending on requested bearing properties the interpolation phase can have multiple steps. One interpolation step for every deviant value with respect to original data tables provided in the standard. In the last step the data is organized directly usable form.

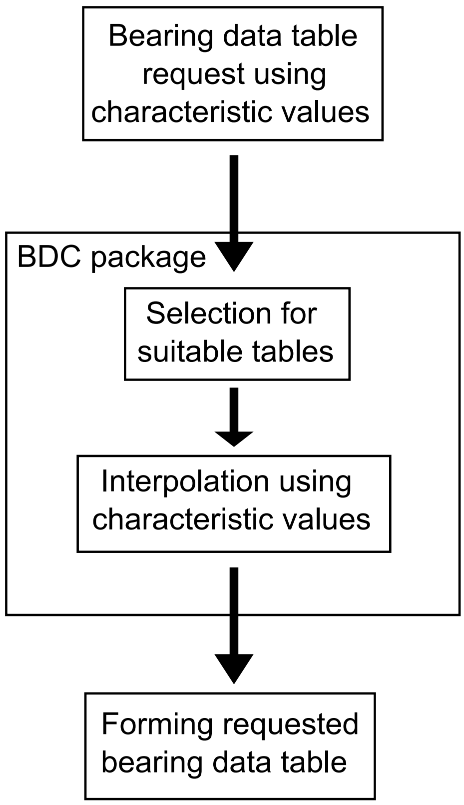


Figure 8. Operation flowchart of BDC package

2.3.2 Principle of bearing coefficient interpolation

In many cases the requested characteristic values bearing does not correspond the values of DIN 31657 tables. Therefore, use interpolation is required to get comparable data from bearing coefficient tables. Based on requested characteristic values the needed tables from standard can be selected for interpolation. Fig. 9 illustrates the principle how for four data tables can be interpolated to one comparable table using geometric ratio ($\frac{B}{D}$) and profiling ($\frac{\Delta R_B}{C_R}$) as characteristic values for interpolation. On the left side are tables selected from DIN 31657 standard. On center the original tables are interpolated with respect to profiling. On the right side two tables are interpolated second time with respect to geometric ratio thus getting dimensionless coefficients for further use. Basically every interpolation step combines two coefficient tables into one table. From mathematical point of view these tables are handled as matrices.

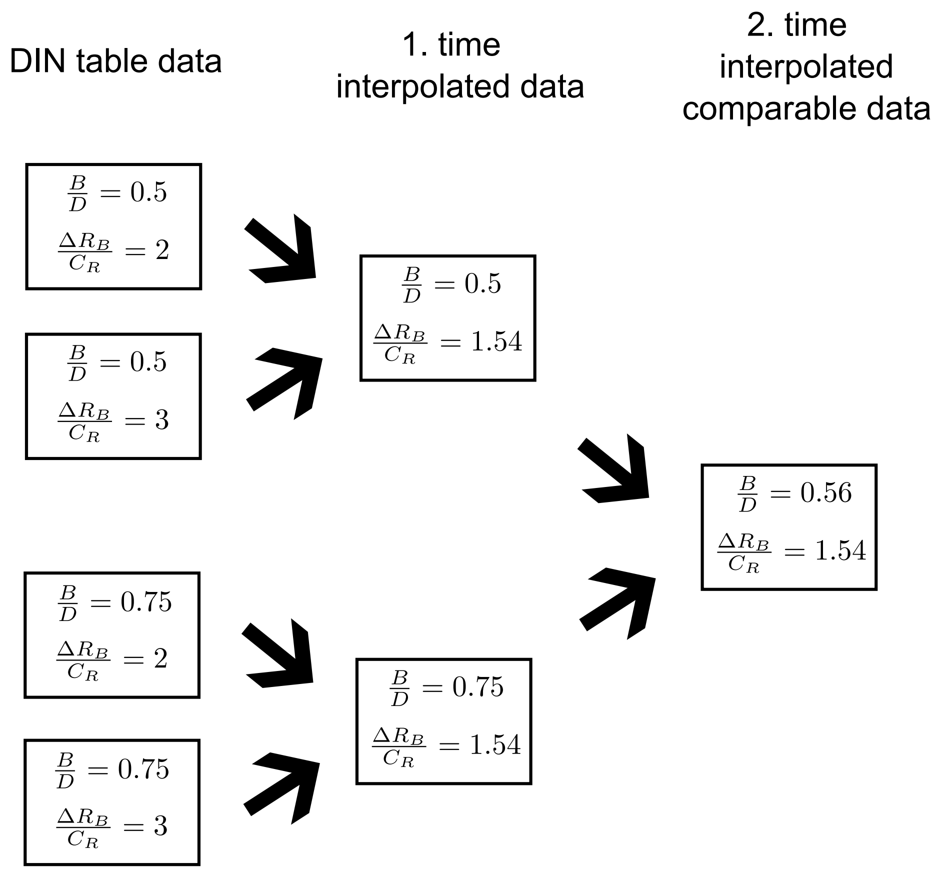
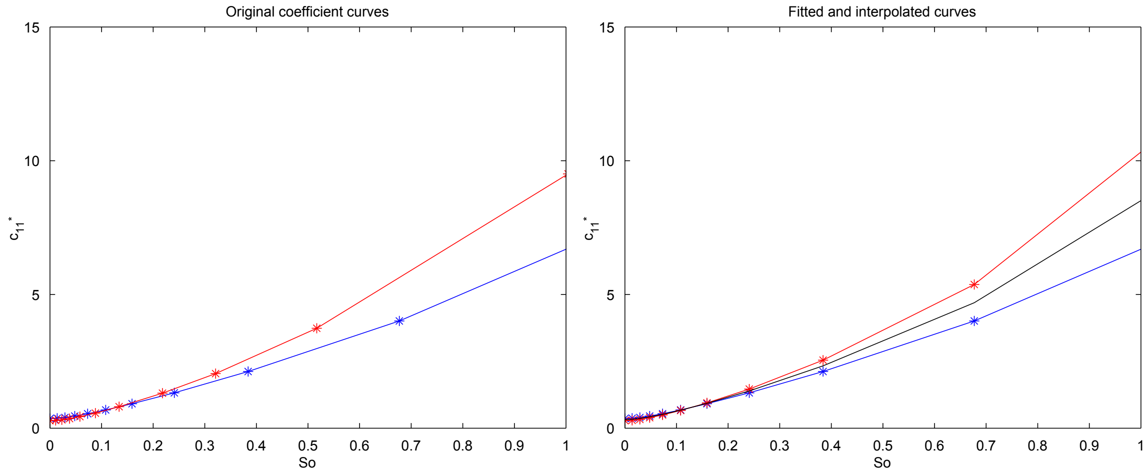


Figure 9. Interpolation principle of bearing data coefficient tables

Sommerfeld numbers which are the basis of the interpolation process. In most cases when comparing to different data tables these numbers do not match exactly as seen in Fig. 10a. The red and blue dots presents corresponding curves from two different coefficient tables. As can be seen in Fig. 10a the dots are not with same increment with respect to Sommerfeld number.

Therefore these two curves need to be fitted to correspond each others. This can be achieved using two step interpolation. Modified red and blue curves after first interpolation step are presented in Fig. 10b where Sommerfeld values are now with the same increment in both curves. Also in the same diagram in Fig. 10b the black plot is the result of second interpolation step. This procedure is done to all dimensionless stiffness and damping coefficients, altogether four to eight time depending on type of bearing. This fitting of data points regarding to Sommerfeld number is mandatory to perform actual bearing property interpolation properly.



(a) (b)

Figure 10. DIN 31657 table data preparation before actual interpolation. (a) Two original curves before and (b) after preparation including interpolated result

2.3.3 The use of BDC package

BDC package is an implementation of DIN 31657 standard. It generates bearing coefficient tables that the standard itself does not provide. Main idea is to interpolate between multiple coefficient tables that have most accurate characteristic values. Table 3 presents corresponding variable names that can be used in the interpolation tool in

MATLAB.

Table 3. Characteristic values and corresponding ASCII style variables used in MATLAB functions of DIN 31657 bearing data table interpolation

Symbol	ASCII style	Definition
Z	Z	Number of pads
$\frac{B}{D}$	B_D	Geometric ratio of pad length and bearing inner diameter
Ω	Omega	Angle of pad
$\varphi_{F,1}$	fiiP1	Pivot angle of pad with respect to radial load
$\frac{\Delta R_B}{C_R}$	dRB_CR	Profiling
$\frac{\Omega_F}{\Omega}$	OmegaF_Omega	Pivot offset

Variation of characteristic values are compiled according to sections 2 and 3 in DIN 31657 standard. The following list expresses possible variation in different characteristic values. [5, 6]

- *Req.DIN* Reference number of section of DIN 31657 standard. Use *DIN=2* for multilobe bearings and *DIN=3* for tilting pad bearings.
- *Req.Z* Number of lobes or pads depending on type of bearing. With multilobe bearings *Z={2, 3, 4}* and with tilting pad bearings *Z={4, 5}*.
- *Req.B_D* Geometric ratio of lobe or pad length depending on type of bearing with respect to inner diameter of bearing. This value can be interpolated. Recommended range is *B_D={0.5 ... 1.0}*, although extrapolating out of recommended range is possible.
- *Req.Omega* Central angle of lobe or pad depending on type of bearing. With tilting pad bearings *Omega={45, 60, 80}* depending on number of pads, see Table 2. In

case of tilting pad bearings this value can be interpolated using recommend range *Omega={60 ... 80}* for 4-pad bearings and *Omega={45 ... 60}* for 5-pad bearings.

- *Req.fiiP1* Angle of which radial force is applied with respect to lobe or pivot depending on type of bearing. With multilobe bearings *fiiP1={180, 240, 270, 300}* for lemon bore bearings, *fiiP1={240, 300}* for three lobe bearings and *fiiP1={270, 315}* for four lobe bearings.

- *Req.dRB_CR* Profiling. This value can be interpolated. Recommended range is *dRB_CR={2 ... 5}*. This value is calculated from bearing geometry.

- *Req.OmegaF_Omega* Pivot angle, possible values are *OmegaF_Omega={0.5, 0.6}*, but it is advisable to use *OmegaF_Omega=0.5*. This value is entered only in case of tilting pad bearing.

Fig. 11 represents function diagram of BDC package. *Req* presents user defined request structural array for coefficient interpolation. Inside dashed area are all operation what are under *getBearingData* function. Because of this kind of structure of the code, notice that following the function code can be cumbersome. Dotted area presents interpolation part which is happening under *dataInterpolation* function. As returning value of *getBearingData* function is an n by 9 matrix containing dimensionless stiffness and damping coefficients with respect to Sommerfeld number.

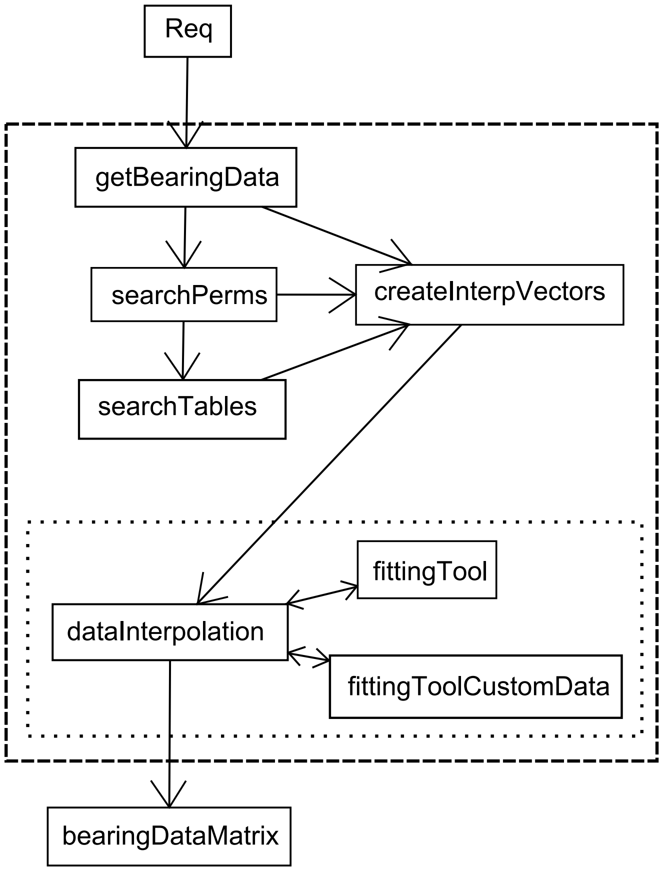


Figure 11. Function diagram of BDC package

2.3.4 Currently possible interpolation variables

For tilting pad bearings possible interpolation parameters are

- $\frac{B}{D}$ (Geometric ratio)
- Ω (Angle of pad)
- $\frac{\Delta R_B}{C_R}$ (Profiling)

The selection algorithm is based on the Table 2 which presents available bearing properties provided in DIN 31657. In case of tilting pad bearing with pivot angle as 0° the standard provides only one coefficient table. In these kinds of cases table selection algorithm can replace profiling ($\frac{\Delta R_B}{C_R}$) and geometric ratio ($\frac{B}{D}$) variables to correspond variables provided in DIN 31657. In other words if using pivot angle as 0° the outcome is always the same. Therefore, it is practical to get to know the standard and the variety of different combinations in these characteristic values. After table selection is interpolation phase and data reorganization back to table form. In this case table form means matrix which can be directly used in RoBeDyn model file or in this case directly in MATLAB script that provides bearing coefficient comparison. An example code is provided as attachment 1.

3 RESULTS

3.1 Usage of bearing coefficient data from manufacturers

Two major bearing manufacturers are John Crane and Waukesha, which bearing data is used in this thesis. John Crane bearing modelling is based on DIN 31657 [1]. These two manufacturers are referred further on as case A and case B. The bearing types used in the comparison are following:

- Case A: John Crane _K_T A6 80/160x70 5 x 52. deg/ between pads
- Case B: Waukesha TJ090-036/2D 5 Pads: centre pivot/steel backed. Load On Pad

Tables 4 and 5 presents characteristic values of tilting pad bearings in case A and case B that are being used in bearing coefficient comparison. The bearing coefficient comparison is based on two data sets from both manufacturers. The bearing properties are the same on both measurements, the only variable is applied shaft load. The lubricant viscosity in every comparison is calculated from measured temperate using Eq. 4. In addition, in case A the profiling value is calculated from preload value Eq. 6.

Table 4. Characteristic values of case A bearing

Z	5
B	45 mm
D	80 mm
$\frac{B}{D}$	0.56
ψ_{eff}	2.226 %.
Ω	52°
m	0.311
$\frac{\Delta R_B}{C_R}$	1.45
$\frac{\Omega_F}{\Omega}$	0.5
$\varphi_{F,1}$	36°

Table 5. Characteristic values of case B bearing

Z	5
B	36 mm
D	90 mm
$\frac{B}{D}$	0.4
ψ_{eff}	1.647 %.
Ω	57° (estimation)
$\frac{\Delta R_B}{C_R}$	2.4 (estimation)
$\frac{\Omega_F}{\Omega}$	0.5
$\varphi_{F,1}$	0°

3.1.1 Case A data tables

Table 6 presents measured bearing data when the shaft load is 1500 N. In Table 7 is presented similar measurement only in case when the shaft load is 3000 N. These values are used in actual bearing coefficient comparison.

Table 6. Case A1 data

rpm min^{-1}	Load N	T_{So} °C	So	C_{xx} N/mm	C_{yy} N/mm	D_{xx} Ns/mm	D_{yy} Ns/mm
1000	1500	50.8	0.834	44902	84382	266.8	473.3
2000	1500	52	0.43	35551	65757	141	233.8
3000	1500	53.3	0.295	32575	58839	103	161
4000	1500	55	0.231	31142	54602	83.3	121.7
5000	1500	56.8	0.193	30514	51373	71.7	98.6
6000	1500	58.8	0.168	30607	50146	63.9	86.1
7000	1500	60.4	0.15	31227	49816	58.7	78.5
8000	1500	62.2	0.136	32086	49987	54.7	72.1
9000	1500	64.4	0.127	33122	50692	51.4	66.6
10000	1500	66.3	0.118	34313	51554	48.9	62.3
11000	1500	69.4	0.114	35720	52960	46.3	58.5
12000	1500	71.3	0.108	37055	53905	44.5	55.5
13000	1500	73.2	0.103	38550	54925	43	53
14000	1500	75.6	0.099	40110	56179	41.7	50.8
15000	1500	77.2	0.094	41244	56925	40.6	48.8

Table 7. Case A2 data

rpm min^{-1}	Load N	T_{So} °C	So	C_{xx} N/mm	C_{yy} N/mm	D_{xx} Ns/mm	D_{yy} Ns/mm
1000	3000	51.1	1.67	118448	223940	533.2	979.6
2000	3000	52.4	0.865	92398	173776	269.5	479.4
3000	3000	54.1	0.602	81204	151516	183.7	317.6
4000	3000	56.1	0.474	75874	140444	142.7	239.6
5000	3000	58.1	0.398	72009	132163	117.5	190
6000	3000	59.8	0.345	70788	128722	102.3	163.7
7000	3000	62.1	0.311	69693	125493	90.6	141.1
8000	3000	64.1	0.283	69422	124208	82.4	126.7
9000	3000	66.3	0.262	69911	124211	76.1	115.4
10000	3000	68.3	0.244	70676	124374	70.9	106.5
11000	3000	70.9	0.232	71114	123503	66	98.3
12000	3000	73.1	0.221	70922	121345	61.8	91
13000	3000	75.4	0.211	71309	120610	58.6	85.4
14000	3000	77.8	0.203	72655	122027	56.2	81.3
15000	3000	78.8	0.191	73151	121688	54.1	77.4

3.1.2 Case B data tables

Table 8 presents measured data when the shaft load is 1300 N. Respectively, in Table 9 is presented similar measurement when the shaft load is 2800 N. It is noticeable that the manufacturer B provided more limited rotational speed data. In addition, the Sommerfeld number in both cases of B data sets is calculated from provided temperate values.

Table 8. Case B1 data

rpm min^{-1}	Load N	T_{So} °C	C_{xx} N/mm	C_{yy} N/mm	D_{xx} Ns/mm	D_{yy} Ns/mm
5000	1300	62	30100	63000	60.7	91.1
7000	1300	62	36100	61900	53.6	71.4
9000	1300	62	40500	62400	47.8	59.8
11000	1300	62	43600	63200	42.9	51.9
13000	1300	62	45900	63700	38.9	45.9
13800	1300	62	46700	63900	37.5	43.9

Table 9. Case B2 data

rpm min^{-1}	Load N	T_{So} °C	C_{xx} N/mm	C_{yy} N/mm	D_{xx} Ns/mm	D_{yy} Ns/mm
5000	2800	62	36800	179000	68.3	178
7000	2800	62	43700	161000	59.7	127
9000	2800	62	48500	151000	52.8	100
11000	2800	62	51900	145000	47.2	82.9
13000	2800	62	54200	140000	42.5	71
13800	2800	62	54900	138000	40.9	67.2

3.2 Bearing data comparison

3.2.1 The problem definition

The comparison process starts by defining characteristic values that are given in Tables 4 and 5. These parameters are entered to a MATLAB code that uses BDC package to interpolate corresponding dimensionless coefficients of stiffness and damping values. An example of this process is presented in MATLAB script in attachment 1.

The comparison is done mostly with using the BDC package. Stiffness and damping coefficients with dimensional values are calculated afterwards using Eq. 1-3. These calculated dimensional values are then compared to values provided by manufacturers.

3.2.2 Comparison limitations

Generally interpolation with respect to geometric ratio and profiling is a typical operation. The interpolation is also performed with respect to angle of pad. Interpolation with respect to pivot angle is not supported because using $0°$ as pivot angle the standard provides usually only one corresponding table. This may cause problems in further on when calculating bearing coefficients.

It is noticeable that DIN 31657 provides limited reliable interpolation possibilities. In some cases rational choices of characteristic values for the interpolation may cause more reliable results. Furthermore, BDC package does not support arbitrary pivot angles ($\varphi_{F,1}$). This is due to lack of provided coefficient tables with $0°$ as pivot angle.

3.3 Results of comparison

Results of comparisons are provided in a from of tables. In every case the main axial coefficients are presented in diagrams. Generally tilting pad bearings have relatively small cross-coupled stiffness and damping coefficients. Manufacturers usually do not release this information. Therefore, those coefficients are left of in these comparisons. In addition, every figure of results contains a mentioning of mean absolute difference in curve values. This information can be used to evaluate the accuracy of the comparison.

3.3.1 Comparison results of case A bearing data

Fig. 12 presents manufacturer A comparison when the shaft load is 1500 N. Two upper diagrams present difference in main axial stiffness coefficients. Respectively, two lower diagrams presents difference in damping coefficients.

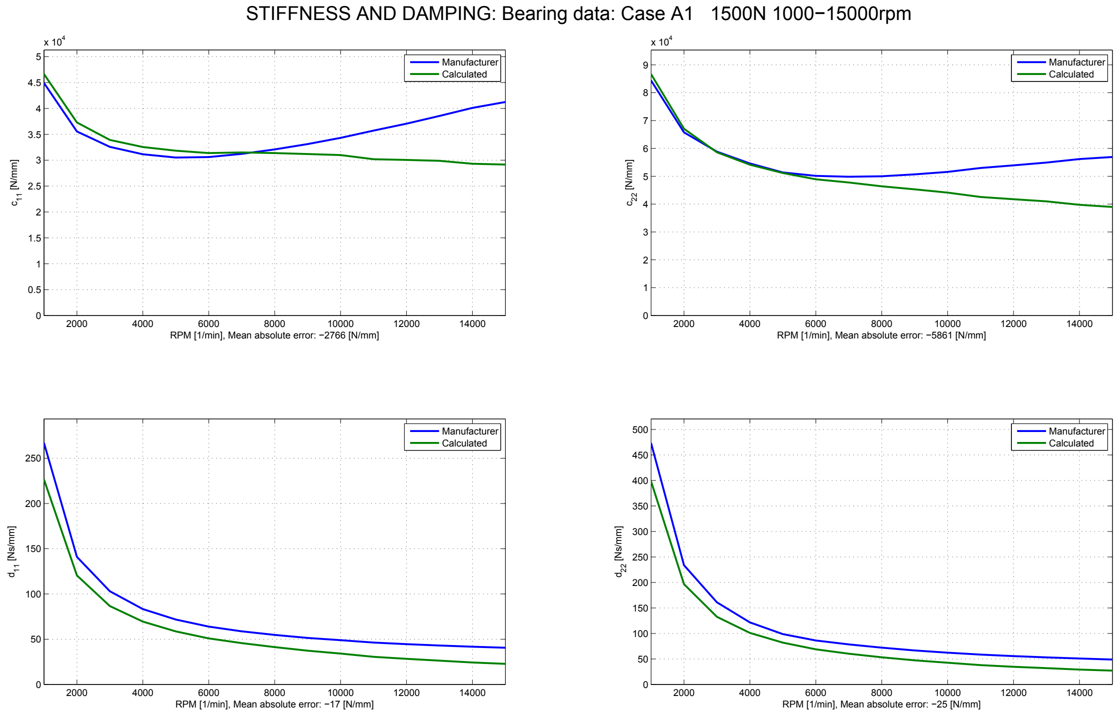


Figure 12. Results of case A1 bearing data comparison

In Fig. 13 can be seen corresponding results of comparison when the shaft load is 3000 N.

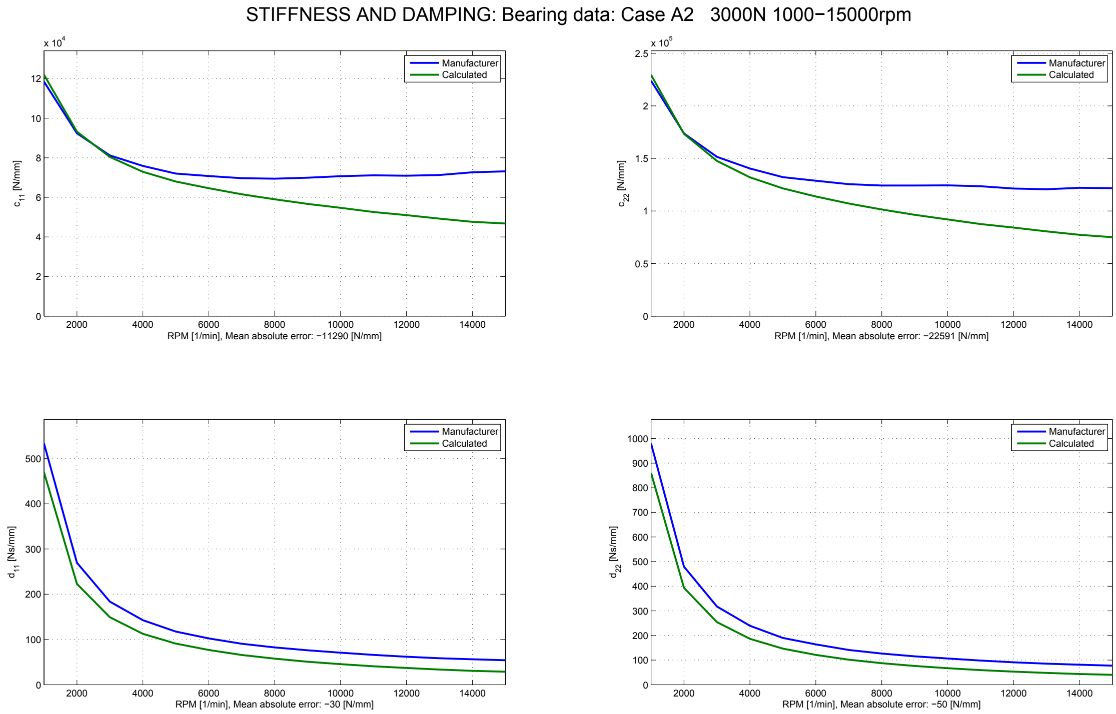


Figure 13. Results of case A2 bearing data comparison

3.3.2 Comparison results of case B bearing data

Bearing coefficients comparison accoding to case B data with the shaft load of 1300 N is presented in Fig. 14.

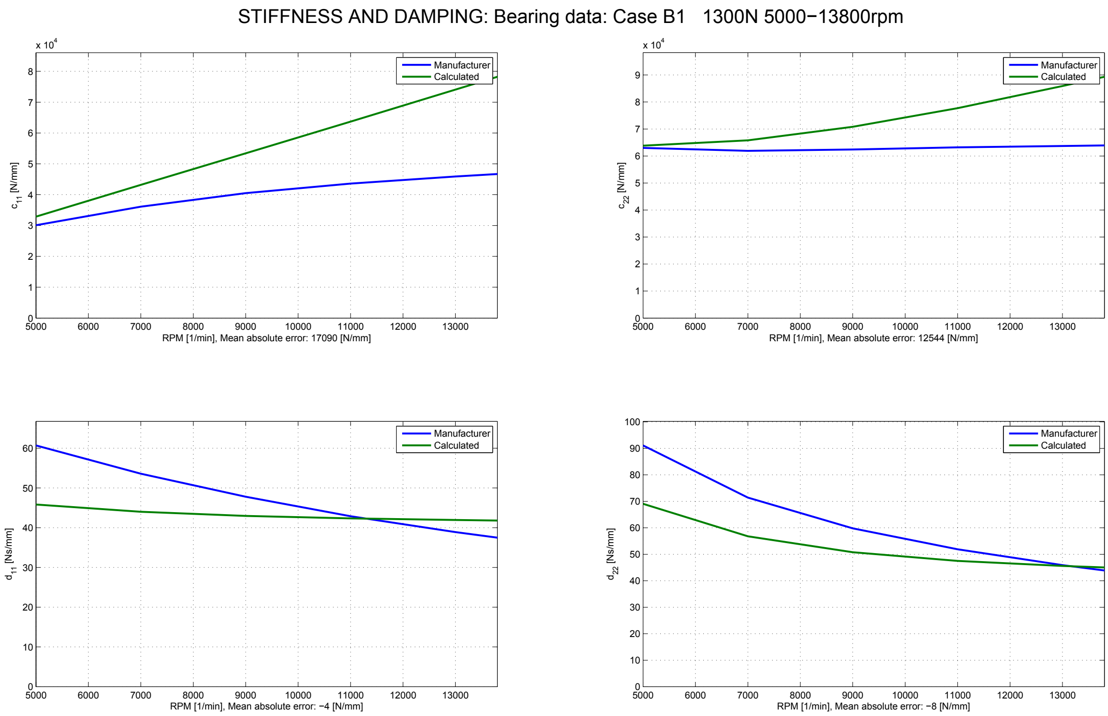


Figure 14. Results of case B1 bearing data comparison

In Fig. 15 can be seen similar case only with the shaft load of 2800 N. In both cases of manufacturer B comparison the speed range is more concise.

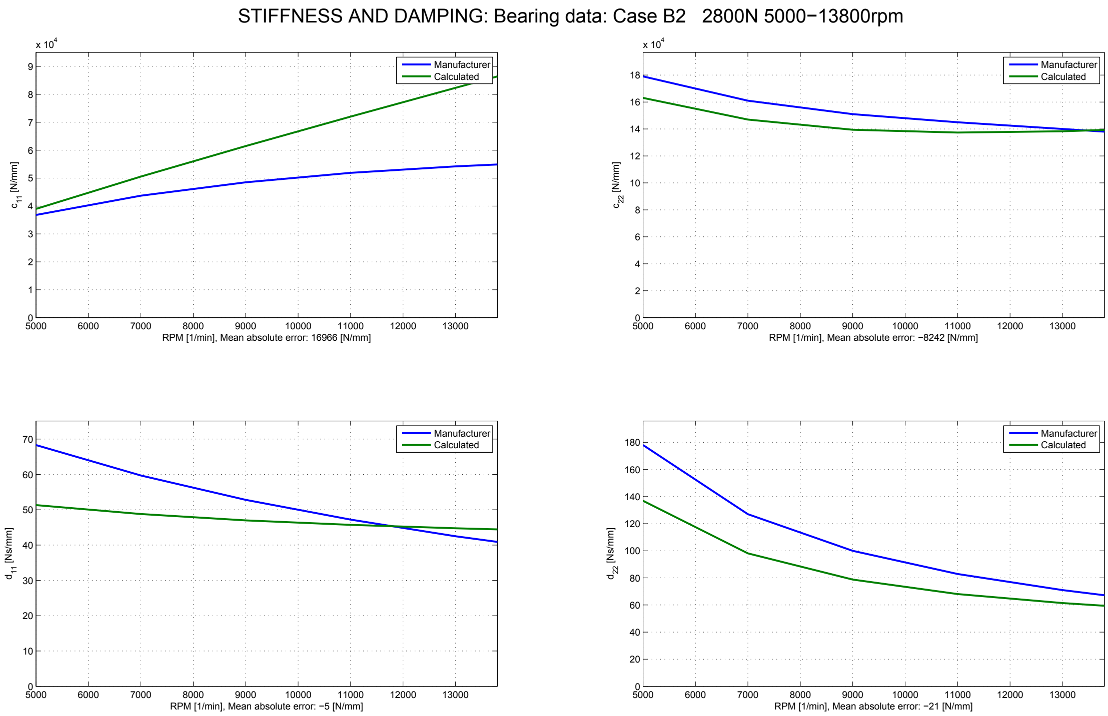


Figure 15. Results of case B2 bearing data comparison

4 DISCUSSION

4.1 Accuracy of bearing data comparison

Bearing data comparison is performed using two major manufacturer bearing data. For enabling the comparison the BDC package is used in MATLAB. Results are given separately in diagrams for two main axial components of stiffness and damping coefficients. Every speed variety diagram includes calculated average error of particular component for comparison purposes. In both cases A and B compared manufacturer data were given with same bearing but different on load. Therefore results are separated and organized by this load force variation.

Fig. 12 and 13 represent tilting pad bearing coefficients for manufacturer A with radial load of 1500 N and 3000 N. Stiffness coefficients in both cases seem to be very accurate at low speeds in Fig. 12 and 13. However, the high speed range the calculated coefficients are decreasing and absolute differences are becoming relatively high. Damping coefficients seem to be following manufacturer's data relatively well although in both cases these calculated values are lower. Average relative error in case on 1500 N is 19 % and with 3000 N is 24 %, respectively. Comparison is done using six bearing data tables from DIN 31657.

Fig. 14 and 15 represents bearing coefficients from manufacturer B with radial load of 1300 N and 2800 N. Stiffness coefficients in general are more accurate at low speeds whereas damping coefficients are more accurate at high speeds as seen in Fig. 14 and 15. Average relative error with 1300 N is 21 % and with load of 2800 N is 17 %.

In addition, it is noticeable that in both cases of manufacturer B bearing data comparison the radial load of bearing affects on pad. This corresponds to an unusual case, since DIN 31657 standard provides only one table for this particular case. Therefore actual bearing data interpolation was possible only with respect to angle of pad. Although, the calculated coefficients do not respond the properties of the bearing used in comparison. The amount of average absolute differences are within similar range than in case of manufacturer A tilting pad bearings. Comparisons are done using two bearing data tables from DIN 31657.

4.2 Discussion of error sources

Numerical errors were determined to not have an affect to the results by varying interpolation order of characteristic values. However, a number of used coefficient tables from standard may possibly have impact to the results. In case of manufacturer A data, the results were more accurate when the angle of pad was not used in the interpolation. On the other hand, the modelled and the real bearing properties were in that case different.

The bearings from manufacturer B had a load on pad in both cases. This property caused the usage of only two tables in interpolation phase. Also, the operational temperature data were not provided at every point of measurement. Only an average temperature corresponding to whole test was available. Viscosity was calculated using the average temperature as main variable. This may have been the main source of error in this study.

When considering the sensitivity of this kind of calculations used in these comparisons the relative clearance can have a major impact to the results. In the studied cases, the clearance was assumed to be a constant. Therefore, it could not have an impact to these results.

5 CONCLUSIONS

Tilting pad bearings are one of the most used hydrodynamic bearing types in the high speed machine industry. The purpose of this thesis was to study the accuracy of a modelling tool for mechanical properties of tilting pad bearings. This interpolation tool is based on DIN 31657 standard that provides a theoretical background for defining the mechanical stiffness and damping coefficients for tilting pad bearings.

In general, most of the cases the comparison were more accurate at lower speed ranges. Correspondingly, at high speeds relative errors were higher than average value. Altogether comparisons had approximately 20 % margin of error. Variation in a number of used coefficient tables from DIN 31657 standard in the interpolation phase can have direct impact to the differences in results.

Variables used in interpolation were geometric ratio, central angle of pad and profiling. One possible source of error may come from interpolation of profiling. In both cases of manufacturer A data the calculated profiling values were smaller than values provided in the standard. Actually, instead of interpolation, extrapolation had to be used in these cases. Therefore, this out of range extrapolation may cause some additional error.

Some preliminary results indicated that comparison results could be more accurate doing coefficients interpolation without using central angle of pad. Using central angle of pad as one interpolation variable can cause some additional error to comparison. In case of manufacturer A comparisons the interpolation of geometric ratio was not always possible when using also interpolation of central angle of pad at the same time.

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Appendix 1. Example MATLAB code from the case A1 comparison

```
clear, close all, clc

% source: LB7749g3_speed_variation.xls
% type='_K_T A6 80/160x70: 5 x 52. deg/ between pads'

% Request and manufacturer data input
data.Req.DIN=3;
data.Req.Z=5;
data.Req.Omega=52;
data.Req.OmegaF_Omega=0.5;
m=0.311;
data.Req.dRB_CR=1/(1-m);

man.D=80e-3;
man.B=45e-3;
data.Req.B_D=man.B/man.D;
data.Req.fiiP1=36;

% String input for plotting purposes
man.ID='5 x 52. deg/ between pads';
man.name='Case A1';
man.bearing='_K_T A6 80/160x70';

%% Manufacturer varibles

% Relative clearance [%.], [per mille]
man.psii_eff=2.226e-3;

% Temperature [C]
T=[50.8
52
53.3
55
56.8
58.8
60.4
62.2
64.4
66.3
69.4
71.3
73.2
75.6
77.2
];

```

(continues)

Appendix 1. (continued)

```
% Manually calculating viscosity reagarding to temperature
% Viscosity [Pa*s], [kg/(s*m)]
nx=0.18e-3;
VG=46; % estimation!
rhoo=860;

man.myy_eff=nx*exp( (159.56./(T+95)-0.181913) * log((rhoo*VG)/(1e6*nx)) );

% Force [N]
man.N=1500;

% Rotation speed [1/min]
man.n=[1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
];

% Angular velocity [rad/s]
man.omegaj=2*pi*man.n/60;

% Sommerfeld number [-]
man.So=[0.834
0.43
0.295
0.231
0.193
0.168
0.15
0.136
0.127
0.118
0.114
0.108
0.103
```

(continues)

Appendix 1. (continued)

```
0.099
0.094
];

% Calculated Sommerfeld number
% man.So=(man.N.*man.psii_eff.^2)./(man.B*man.D.*man.myy_eff.*man.omegaj);

% Stiffness coefficients [N/m]
man.C11=1e3*[44902
35551
32575
31142
30514
30607
31227
32086
33122
34313
35720
37055
38550
40110
41244
];

man.C12=0*man.C11;

man.C21=0*man.C11;

man.C22=1e3*[84382
65757
58839
54602
51373
50146
49816
49987
50692
51554
52960
53905
54925
56179
56925
];

```

(continues)

Appendix 1. (continued)

```
% Damping coefficients [Ns/m]
man.D11=1e3*[266.8
141
103
83.3
71.7
63.9
58.7
54.7
51.4
48.9
46.3
44.5
43
41.7
40.6
];

man.D12=0*man.C11;

man.D21=0*man.C11;

man.D22=1e3*[473.3
233.8
161
121.7
98.6
86.1
78.5
72.1
66.6
62.3
58.5
55.5
53
50.8
48.8
];


%%

% Common variables
% Req.B_D=60/80;  % Bearing width / Journal diameter [mm]
% Req.DIN=3;      % 2 for multilobe, 3 for tilting pad
% Inp.Bearing(jj).Req.Z=5;        % Number of lobe/pads
```

(continues)

Appendix 1. (continued)

```
% Inp.Bearing(jj).Req.Omega=60;    % Angle of lobe/pad
% Inp.Bearing(jj).Req.fiiP1=36;    % Lobe/pivot angle
%
% % NOTICE:
% % In DIN 31576-2 lobe angle symbol in ascii form is fiiP1
% % and in DIN 31576-2 pad angle symbol in ascii form is fiiF1
% % but workspaces of din tables are made using only fiiP1 ascii symbol
% % so this may cause slightly confusion.
%
% % Only multilobe variables
% %Inp.Bearing(jj).Req.h0max=-1;            % Relative gap
%
% % Only tilting pad variables
% Inp.Bearing(jj).Req.OmegaF_Omega=0.5;    % Pivot offset
% Inp.Bearing(jj).Req.dRB_CR=1.59;         % Profile

% You can comment unneeded multilobe or tilting pad variables


% Interpolating suitable data table
bearingDataMatrix = getBearingData(data);

So=bearingDataMatrix(:,1);
c11=bearingDataMatrix(:,2);
c12=bearingDataMatrix(:,3);
c21=bearingDataMatrix(:,4);
c22=bearingDataMatrix(:,5);
d11=bearingDataMatrix(:,6);
d12=bearingDataMatrix(:,7);
d21=bearingDataMatrix(:,8);
d22=bearingDataMatrix(:,9);


%%

% PLOTTING SETTINGS
% plotting=1; % normal 2x2 to stiffness and damping
% plotting=2; % 1x2 to stiffness and damping
% plotting=3; % 2x1 to stiffness and damping
plotting=4; % 2x2 combined stiffness and damping
% plotting=0; % no plots

calc_plot_comparison
```