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FACULTY OF MECHANICAL ENGINEERING IN NIŠ

THE 2nd INTERNATIONAL CONFERENCE

MECHANICAL ENGINEERING IN XXI CENTURY

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PREFACE

Half a century of tradition, high standards in education of generations of students, modernly equipped classrooms, professional teaching and associate staff, their references and recognizability position the Faculty of Mechanical Engineering of the University of Niš as the leader in the field of engineering and technological sciences, not only on the territory of the Republic of Serbia, but also on the territory of the Western Balkans.

The proceedings of the Second International Conference **MECHANICAL ENGINEERING IN XXI CENTURY** appear in the year when the Faculty of Mechanical Engineering of the University of Niš celebrates its fifty third anniversary. The Department of Mechanical Engineering of the Faculty of Engineering in Niš was founded on the 18th of May, 1960, and it developed into the Faculty of Mechanical Engineering of the University of Niš in 1971. The Faculty of Mechanical Engineering grew intensely, thus becoming one of the most renowned scientific and educational institutions in the country.

The mission of the Faculty is to organize and conduct academic study programmes and to develop and realise scientific and professional work in the field of engineering and technological sciences. Its vision to be recognisable in the european and world academic environment in the area of mechanical engineering.

More than 90 teachers and associates, about 60 non-teaching workers, as well as numerous teachers and associates from other faculties and from the industry, are working hard every day to accomplish the mission and vision of the Faculty.

The Faculty of Mechanical Engineering of the University of Niš is accredited in compliance with the Law on Higher Education within the scientific and educational field of engineering and technological sciences. It conducts the academic studies of the first degree – undergraduate studies, the second degree – master academic studies, and the third degree – doctoral studies, within the scientific area of mechanical engineering and engineering management.

The Faculty of Mechanical Engineering is also a scientific research institution apart from being an educational one. There are 14 international scientific research projects within the frame of FP7, TEMPUS, CEEPUS, DAAD, bilateral and cross-border programmes, as well as 24 national scientific research projects, being realised at the Faculty in this year. The participation of teachers and associates from the Faculty in these projects is of utmost importance for their educational and research work and their further career.

The Second International Conference **MECHANICAL ENGINEERING IN XXI CENTURY** represents a forum for presentation of latest results, basic and development research and application within the topics of:

- Information Production Technologies and Industrial Management,
- Thermal and Process Engineering,
- Mechatronics and Control,
- Machine Constructions, Development and Engineering,
- Traffic Engineering, Transport and Logistic,
- Theoretical and Applied Mechanics,
- Applied Mathematics,
- Engineering Profession: State and Prospects,

as well as the Workshop in honor of Prof. dr Ljiljana Petković "Iterative methods in applied mechanics".

Eighty three papers, whose authors come from 11 countries, are published in these Proceedings. Papers present the research results of numerous projects financed by the Ministry of Education, Science and Technological Development of the Republic of Serbia, as well as the research results within international projects. There is no doubt that the papers published in these Proceedings will contribute to the development of a highly significant area – Mechanical Engineering.

As a Dean of the Faculty of Mechanical Engineering in Niš, I am honoured to greet all participants of the Conference and wish them successful work.

Dean of the Faculty of Mechanical Engineering,
University of Niš

Prof. dr Vlastimir Nikolić

Niš, June 2013.

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Experimental Tests on Bound Frequency of Axial Ball Bearings for Fixing the Ball Screws

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Abstract— In contemporary machine tools as well as processing centres, screw gears are getting increasingly used. On this occasion, the higher speed of work and high accuracy of guidance are expected. Regarding this, the issue of fixing is very delicate resulting in thrust ball bearings, which regarding their structure and performances, successfully meet the specified requirements. At greater work rates, there is heat load of these bearings. Due to that reason, in the designing phase, it is necessary to specify whether the temperature load shall be significant. If this is the case, the attention shall be paid to its thermal stability in the designing phase. In these papers, the experimental testing of thermal bound frequency of bearing type ZKLF, manufacturer "INA"- „Schaeffler Technologies“ GmbH & Co. KG

***Keywords*— Axial ball bearing, Experimental testing, Thermal stability, Revolution frequency**

I. BEARINGS FOR BALL SCREWS

Functioning of tool machines is not possible without ball screws. The basic requirements for tool machines are primarily high accuracy of precision. The ball screws of tool machines – main and auxiliary, are manufactured with high level of accuracy, and special attention is paid to fixing the ball screws regarding the rigidity, oscillation damping and thermal deformations.

Fixing of ball screws is done by special bearings requiring high level of performances, which they successfully meet.

It is well known that fixing of ball screws is one of the most demanding and challenging tasks, because in these cases the high level of performance accuracy is required, whereas there are increased loads in radial and axial direction.

A long engineering experience resulted in real solutions for this type of bearings. Regarding the fact that there is a high number of revolutions (especially with drill machines, grinding machines, etc.), with the high axial forces, the rolled members are most frequently thrust bearings convenient for high number of revolutions. Since there are also axial loads, the reception, distribution and compensation issues of axial loads had to be solved. It was solved by thrust ball bearing with angular ball contact with rolling paths, as well as bias load in the mere bearing. Bias load is realized by two-piece internal ring planning the certain distance between the parts of rings acting as damper of axial forces occurring in the bearing. In the Fig. 1 there are two bearing structures for ball screws.

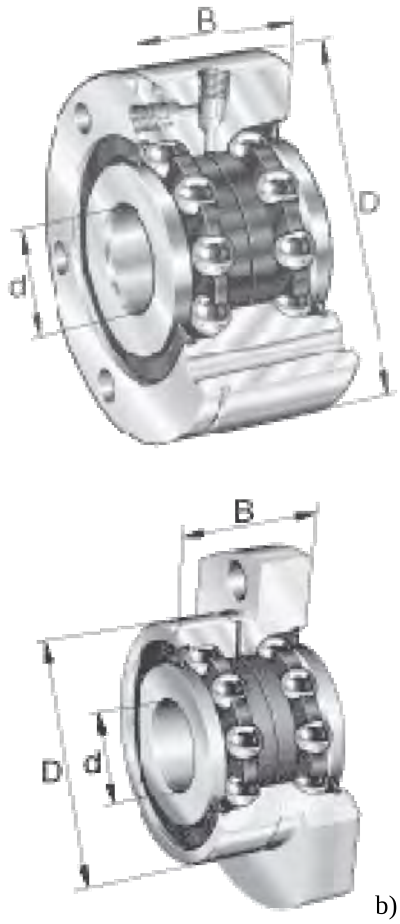


Fig. 1 Bearings for ball screws: a) Bearing of series ZKLF ...-2RS, ZKLF...-2Z; b) Bearing of series ZKLFA...-2RS, ZKLFA...2Z; d- internal diameter of bearing; D- outside bearing diameter; B- total bearing width

In some cases it is possible to couple two bearings so that there is a pair of coupled bearings providing higher bearing capacity as well as higher rigidity (Fig. 2).

Principally, the bearings for fixing of ball screws are specified as bearings of high precision. It refers mainly to the manufacture quality of all its integral parts.

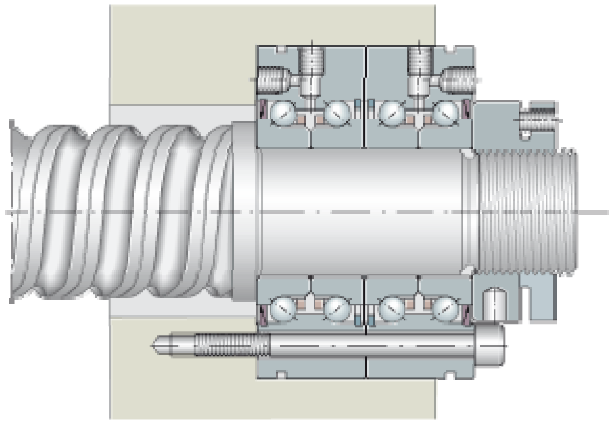


Fig. 2 Coupled ZKLF...-2RS-2AP bearings

Depending on the grade of bearing measurements, the thrust ball bearings for ball screws can receive both great radial and axial forces which make them more specific related to other bearing types. Internal ring, crest with balls and outside ring are placed on each other making one adjusted unit. Using such structure, it is possible to precisely axially load these bearings.

The protection against impurities and moisture is made by rubber sealing rings. In case of higher speeds, sealing by metal rings is recommended.

These bearings are manufactured with and without the strengthening openings (implying the bolt connection) on the outside ring. The bearings with strengthening opening on the outside ring can be strengthened to machine using the bolts at the place planned for fixing. The strengthening openings are clean industrial solution because they decrease the expenses of lid and machine opening processing.

With machines used in timber industry or means of transportation with ball screws, bearings of less precision are used, thus there are bearings with higher tolerance values.

II. Axial ball bearings with angular contact for ball screws

The requirements for fixing of ball screws are laid so that the bearings with their structure cannot use them at their optimum. That is why "INA“ and "FAG“ proposed a great palette of thrust ball bearings with angular contact which by its structure can meet the requirements regarding bearing capacity, rigidity, slight friction at highly dynamically loaded plant of ball screws. This range of products successfully meets all industrial requirements set by fixing of ball screws.

Axial ball bearings are manufactured as single line, two lines or three lines. They include the outside ring of high thickness, ball crest and single line or two line internal rings.

In its manufacture program, INA provides the axial ball bearings of series ZKLN, ZKLF, ZKLFA and DKLFA, for fixing of ball screws with two line internal ring.

Series in the manufacture program of "INA“ have, with many number of types, openings for direct strengthening to the machine where ball screws fixing is done. The rings with these bearings are made using the precise nuts for adjustment of bias load in the mere bearing.

It is important to specify that these bearings due to pressure angle (grazing angle) of 60º besides the radial one can also receive the high axial load.

A. *Axial ball bearings with angular contact of series "ZKLF“*

Axial ball bearing with angular contact of series ZKLF is directly connected to the machine using the bolts (Fig. 3).

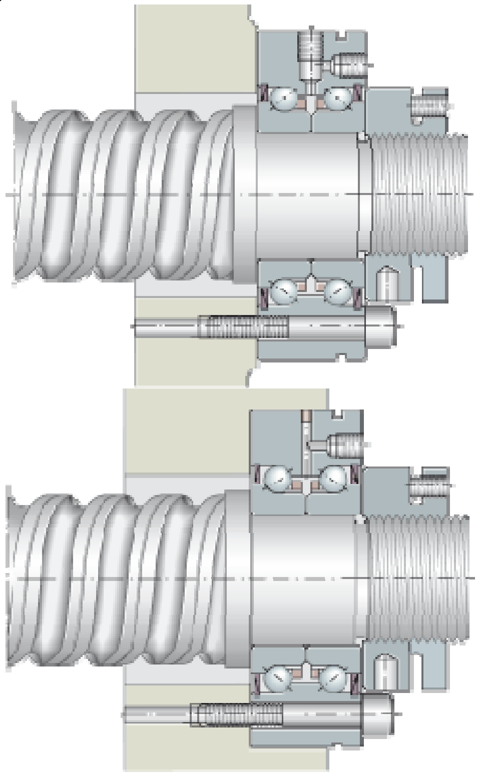


Fig. 3 Outside bearing ring is fixed to flat machine front and bias loaded by precise nut

Axial ball bearings with angular contact of series ZKLF...-2RS-PE match the bearing series ZKLF but they differ in tolerance and quality. Their precision grade is P5 in comparison to DIN 620 and therefore the diameter tolerance values are wider. The label "PE“ directly specifies that the quality if of lower precision grade [4]. They are applied when the precision of positioning can be lower. Then also the requirements for other structural members are lower regarding precision.

Axial ball bearings with angular contact of series ZKLF...- 2RS and ZKLF...-2Z are also available in greater dimensions. It is clear that higher cross sections imply greater bearing capacity.

The bearings of series ZKLF-...2RS-2AP, include two coupled bearings ZKLF...- 2RS, (Fig. 2). The coupled bearings at outside surface of the external ring in this case are labeled with arrow, assisting in defining the exact arrangement of coupled bearings. With regular assembly and arrangement of coupled bearings, the sealants should be directed outwards (Fig. 2). During the package mounting, both bearings must have their axes completely mated with the ball screw axis. As it is shown in Fig. 2, the coupled bearings are strengthened by bolt to machine case, whereas bias load is adjusted by precise nut.

The bearings with suffix "2RS“ are mutually sealed by rubber sealants, convenient for mounting at the positions with fine dust or liquid, whereas the bearings with suffix

"2Z" are mutually sealed with metal sealants which are convenient in cases of high number of revolutions.

All bearings are factory lubricated with grease based on lithium soaps according to GA28, but they have the possibility of additional lubrication made by the lubrication opening located on the external ring. In most cases, the factory lubrication is sufficient for entire working life of bearing [3].

III. Experimental testing

The goal of experimental testing of axial ball "ZKLF2575-2Z" and "ZKLF50115-2Z" bearings for fixing of roller screws is specifying the following parameters:

- Revolution frequency
- External ring temperature
- Surrounding temperature
- Friction force, i.e. friction moment
- Preload axial forces

B. Defining the testing protocol

The testing protocol stipulates that four units of bearings of both types shall undertake the specified metering. The sample selection shall be done using the method of "random sample".

Regarding the fact that it is the specific type of bearing, it is necessary to perform certain preparations of sample in order to do the regular mounting and setting the sample at the test bench.

The test bench comprises the shaft simulating the ball screw and by which from the engine the required revolution frequency is obtained on the internal ring of bearing, whereas the external ring remains free and it does not receive and load – it remains stationary during entire testing time.

During metering, also the surrounding temperature shall be measured, i.e. room temperature, which should be 20°C.

Testing every bearing sample shall be done in three phases as follows:

- Testing with sealing rings lubricated with factory grease,
- Testing without sealing rings lubricated with factory grease,
- Testing without sealing rings which was previously thoroughly cleaned from factory grease and lightly lubricated.

The idea is to do comparison of results in order to obtain the average values after every performed metering series of every sample.

Every sample after setting on the testing desk shaft must be prestressed by certain axial force of prestress stipulated by Schaeffler amounting:

for bearing ZKLF2575-2Z F_x= 1945 N

for bearing ZKLF50115-2Z F_x= 3148 N

Prestress is done by precise nut for prestress of subject bearings also manufactured by Schaeffler (AM25 and AM50). The specified prestress must remain at the constant level prior to, during and after the completed testing. Regarding the fact that there is no case to which the sample would be additionally strengthened by bolted connection, due to maintenance of constant prestress, the set of plate springs is inserted into the entire compound functioning as additional strengthening for the sample. The test bench with the sample is shown in Fig. 4, and the simplified diagram of test bench in Fig. 5.

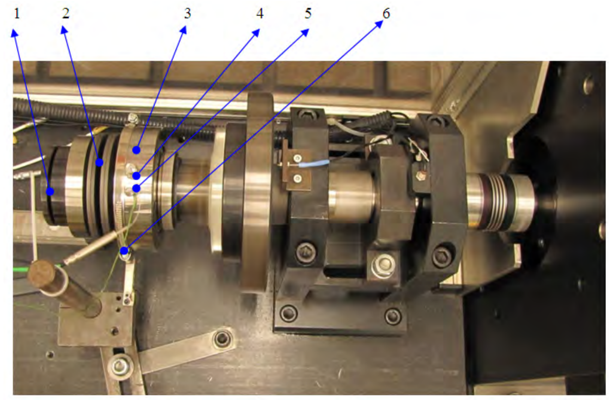


Fig. 4 View of test bench with the sample: 1- Precise nut for prestress; 2- Set of plate springs; 3- Bearing being tested; 4- Temperature sensor (lubrication opening) for temperature metering of internal ring; 5- Temperature sensor - temperature metering of external ring; 6- Friction force metering sensor

C. Testing program

The testing comprises:

7. Prestress metering for cold bearing,
8. Grease distribution control within the bearing,
9. Loads of tested sample with various levels of revolution frequency,
10. Prestress metering for warm bearing.

In order to obtain the precise results, it is necessary to make the equal distribution of grease within the bearing (factory filling). For both types of bearing, the lubrication L192 was used with the following quantities:

- for bearing ZKLF2575-2Z 1,9 g

- for bearing ZKLF50115-2Z 4,7 g

In order to distribute the grease equally, at the beginning of every metering, the operation of both types of sample is foreseen with two levels of revolution frequency, 500 min^{-1} and 1000 min^{-1} during 15 minutes. Afterwards, it is foreseen that the lubricant is equally distributed within the bearing.

When the lubricant it distributed at the bearing, then every sample is loaded until the planned maximum revolution frequency (8000 i.e. 6000 min^{-1}) for five minutes continuously.

The following step is loading every sample with various levels of revolution frequency. The scope of revolution frequency is:

- for bearing ZKLF2575-2Z 0-8000 min^{-1}

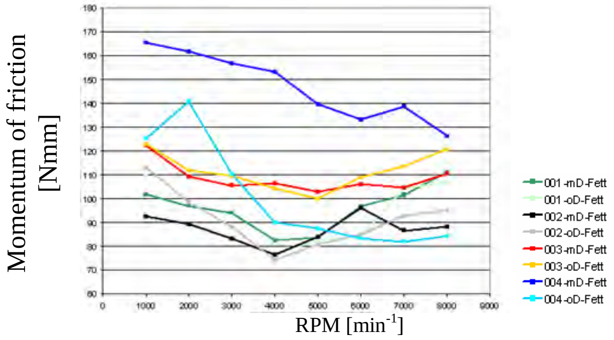
- for bearing ZKLF50115-2Z 0-6000 min^{-1}

The revolution frequency level is 1000^{th} part of maximum revolution frequency. The load duration of every revolution frequency level is 30 minutes. After the last load level, the prestress value is necessary to be metered again. In the following phase, from all samples, the sealing rings were removed and the load procedure is repeated based on grades and afterwards the prestress value was metered again.

In the last step, there are two bearing samples ZKLF2575-2Z completely cleaned – degreased, lubricated VG68 and repeated testing for the same load diagram. Prestress metering was necessary at the end of load.

D. Comparison of friction moment

ZKLF2575-2Z



ZKLF50115-2Z

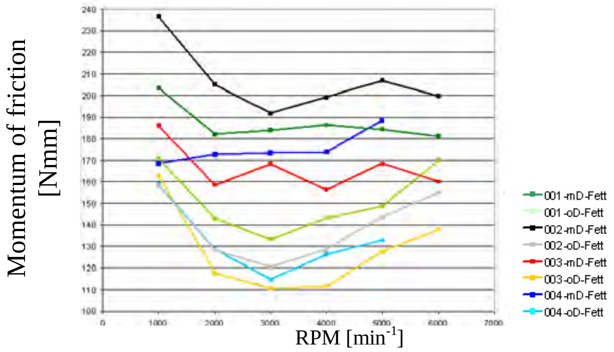


Fig. 7 Friction moment diagrams (mD- Fett – without sealings and fat, oD- Fett – with sealings and fat)

E. Comparative view of analytic and experimental results for axial ball bearings of type ZKLF

This chapter shows the comparative view of analytic and experimental results for axial ball bearings of type ZKLF.

For instant defining the bound revolution frequency, so-called fast operation characteristic nd_m, can be used which is at the same time also the basic guideline and internal recommendation of Schaeffler. This characteristic for subject bearings clearly provides analytical result showingt hat with the specified bearing it is possible to increase the bound revolution frequency (table 1).

Regarding the fact that the subject of paper is testing the bound revolution frequency in relation to thermal stability, in the further testing the analytic analysis of temperature load was done resulting from friction in the bearing. The following iteration was experimental testing and identification of effective parametres.

Due to easier comparison in Fig. 8, there are comparative diagrams obtained by analytical procedure in the program "Bearinx" and experiment.

TABLE 1 VALUES OF BOUND REVOLUTION FREQUENCY OBTAINED BASED ON SIMPLIFIED FAST OPERATION FORMULA

Bearing types		ZKLF2575-2Z	ZKLF50115-2Z
ZKLF-2Z	Based on formula (1)	11527	6435
	Catalogue	4700	3000
ZKLF..-2RS	Vmax=12m/s	5582	3140
	Catalogue	2600	1500

$$n_g = \frac{n \cdot d_m}{d_m} \qquad (1)$$

For

$$d_m < d_G \Rightarrow$$

$$nd_m = 750000 \left(\frac{d_m}{d_G}\right)^{0,2} \left(1 + \frac{2D_w \cos\alpha}{d_m}\right)^{-0,8} f_S f_M \qquad (2)$$

For

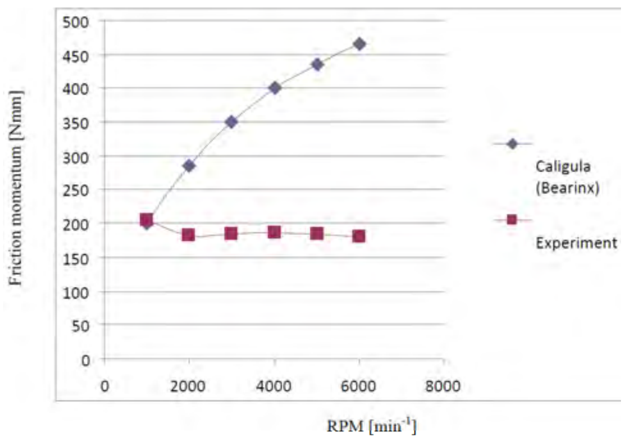
$$d_m \geq d_G \Rightarrow$$

$$nd_m = 750000 \left(\frac{d_m}{d_G}\right)^{-0,08} \left(1 + \frac{2D_w \cos\alpha}{d_m}\right)^{-0,8} f_S f_M \qquad (3)$$

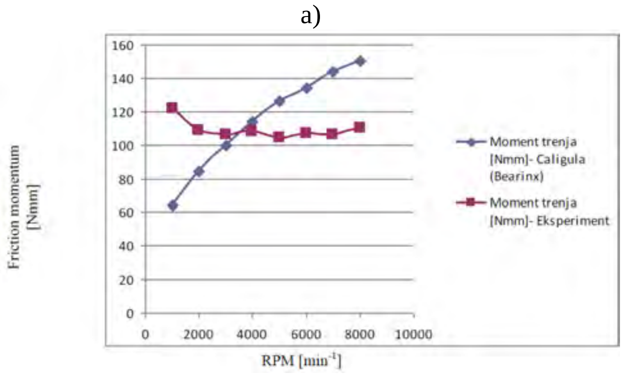
where:

d_m	-	Average diametre of bearing in mm
D_w	-	Ball diameter, i.e. rolling item in mm
f_S	-	Lubrication factor f_S = 0,75
f_M	-	Cage type factor (1,3 – for brass; 1 for steel cage)
α	-	Contact angle of rolling items
d_q	-	Bearing form factor

For bearings which are rubber sealed from both sides (2RS) the maximum revolution frequency is obtained from the condition that the maximum sliding speed at contact sealants is v=12 m/s [5].



a)



b)

Fig. 8 Comparative diagram of friction moment for bearing: a) ZKLF2575-2Z, b) ZKLF50115-2Z

The reasons for occured discrepancy are the following ones:

- Technical possibilities of the test bench are such that it was not possible to simulate the axial load

- Factors of lubrication and loads in program "Bearing" are not completely defined for this type of bearing

As it is known, the bearings of type ZKLF are axial ball bearings with angular contact for fixing of ball screws. Their main characteristic is possibility of transfer of great axial forces, i.e. loads with high guide precision. During experimental testing, the test bench was used whose technical characteristics could not have provided the load in axial direction. The only axial load which was present during the experimental testing is an axial force induced by precise nut for prestress and set of plate springs. The specified axial force in the bearing cannot be treated as real axial load because in practice the axial loads are significantly higher.

Due to the specified reason, the only conclusion to be made is that the bearing loads are only the consequence of revolution, thus certain descrepancies are found compared to analytical expectation.

In the analytical procedure performed in the program "Bearinx", the standard procedure for specification of thermal frequency of revolutions is used which is shown in code DIN 732 part 1 and 2. All coefficients necessary for calculation are automatically taken from the specified code. It should be specified that the code DIN 732 does not consider the effect of machine case to which the bearing is connected using the bolted connection nor the effect of mere bolted connection [1, 2, 4].

Regarding the fact that the aim of paper is testing the bound thermal revolution frequency for bearings of series ZKLF, as well as significant parametre identification, the occured discrepancy can be considered technically acceptable.

The second part of the task refers to the parameter identification affecting the bound thermal revolution frequency. It was noted that there are certain discrepancies between analytic and experimental results.

The obtained experimental results imply the main significant effects: grease distribution in bearing, bearing acceleration as well as effect of centrifugal force to cage and rolling items (balls).

For specifying additional parameters, the thorough analysis is required which should include a great number of experiments in order to define the grease quantity for lubrication, coefficient resulting from the type of bearing and lubrication for referential conditions, relation coefficient for number of revolutions, coefficient which shall take into consideration the mixed friction in the bearing at the acceleration as well as coefficient which shall define the effect of centrifugal force to the cage and rolling items.

IV. CONCLUSION

The actual task was research of bound thermal revolution frequency of axial ball bearings with angular contact for fixing of screw balls as well as identification of significant parametres.

Due to that reason, the task was done in two stages. In the first stage, the analytical procedure was carried out with the result of relation between friction moment and revolution frequency. In this part of research, the basic benchmark was the case of extreme revolution frequency and that is doubled bound value of revolution frequency which is specified in the brochure of Schaeffler. It should be noted that the analytical procedure is carried out in program module "Caligula" foreseen for the calculation of thermal load of bearings. The analytic procedure of this program module was taken from the existing code DIN 732 part 1 and 2, providing a corresponding algorithm for specifying the bound thermal revolution frequency. However, the specified code does not consider the additional impacts which are an integral part of a real mounting situation, which greatly change the existing analytical procedure. This refers mainly to the existing bolted connection as well as machine case to which bearing is additionally strengthened, as well as precise nut for bearing prestress, increased contact angle of rolling elements (in the subject case it is 60°) as well as necessary prestress in the mounted bearing. Due to that reason, it is necessary that the existing algorithm specified in the aforementioned code, is completed with the corresponding coefficients as well as identify the significant parameters for the subject type of bearing.

In the second stage of operations, the experimental research of subject type of bearing was carried out with set revolution frequency defined in the first stage.

As it was already specified, the basis of analytical method is specifying the referential surface on which the thermal energy transfer is done which is generated during the operations of bearing. In mathematical regard, it represents the sum of all contact areas over which the heat transfer is done. Regarding the fact that in the specified case the contact areas are different sizes and that various materials are used, the subject analysis gets more complex. Actually, since the analysis is based on thermal load of bearings, the thermal characteristics of materials making the referential surface are emphasized. This refers to the type of material and its thermal conductibility.

In order to specify the bound thermal revolution frequency of axial ball bearings, it is necessary to establish the thermal balance at the bearing-case system. Considering the code DIN 732 as well as observed real system, it is logical to define the heat transfer coefficient for the entire system. In this course, the analogy between heat flux and electricity is used.

The following iteration was identification of significant parameters. In this iteration there is a solution of obtained analytical results for the program module "Caligula" and experimental results. The certain discrepancy of results was noticed, which was used for identification of significant parameters. Regarding the fact that the aim of the operations is identification of significant parameters and research on bound thermal revolution frequency for the bearing of type ZKLF, the occurred discrepancy of results can be regarded technically acceptable.

The comparison of analytical and experimental results lead to the conclusion that the significant parameters for the observed case of mounting are: grease distribution in bearing effect, bearing acceleration effect as well as centrifugal force effect on cage and rolling elements. Regarding the fact that the significant parameters vary, the logical conclusion is made that multidisciplinary approach is required for their definition.

It should be pointed out that at the current level of knowledge, the research of revolution frequency related to the thermal stability with thrust bearings of type ZKLF should be done for each single case of mounting. Actually, every case of mounting includes various lengths of ball screws, various materials of shaft, various forms, sizes and case materials to which the bearings are strengthened, thus everything aforementioned changes the size of referential surface as well as thermal characteristics considering the level bearing-case system. Based on the aforementioned, it can be concluded that every single case of mounting has its own corresponding thermal balance directly affecting the thermal bound revolution frequency.

Eventually, it should be emphasized that the thermal bound revolution frequency is a very imposing parameter for analysis of bearings with the significant temperature load. However, also in those cases, no kinematic bearing performances should be neglected for the specified work conditions, since the bearing for the specified work conditions can reach the bound thermal revolution frequency without causing disturbing the bearing kinematics.

LITERATURE

[1] DIN 732 Teil 1 (1994) Thermische Bezugsdrehzahl

[2] DIN 732 Teil 2 (1994) Thermisch zulässige Betriebsdrehzahl

[3] Schaeffler Gruppe Industrie (2009) Lager für Gewindetriebe, Katalog TPI 123D-D: Schaeffler Gruppe

[4] Schaeffler Technologies GmbH&Co. KG: Bearinx program package: Schaeffler Technologies GmbH&Co. KG

[5] Krstić, V., Miltenović, A., Miltenović, Đ., Banić, M. (2011) Grenzdrehzahlermittlung an Axial-Schrägkugellager für Gewindetriebe, Zlatibor: Proceedings of the 7th International Conference “Testing and Development of Mechanical Elements and Systems” IRMES, pg.563-568

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