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Development of Lifetime Testing Methods for Increased Requirements of Coolant Pumps in Electrical Vehicles

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***Abstract* – Increasing market of electrical cars pushes part suppliers to fulfil new requirements and meet new challenges. This paper focuses to the increased lifetime requirements of parts in electrical vehicles. Component developers and suppliers are facing to difficulties of lifetime testing, as the increased periods are too long for practical tests. New accelerated endurance testing and mathematical estimation methods are required for reliable lifetime estimation. Research team of the University of Miskolc works on development of new lifetime estimation method for bush bearings of coolant pumps in electrical vehicles in cooperation with its industrial partners. Lifetime requirement for these parts is increased to 30.000 hours which makes no possible the traditional endurance testing at all. In the paper acceleration parameters and endurance test stands developed for defining practical abrasion curves are summarized. Mathematical model of lifetime estimation and two new methods for accelerated lifetime testing for industrial use are described.**

Keywords – lifetime test, bush bearing, electrical vehicle, endurance test.

I. INTRODUCTION

Lifetime requirements of components are increasing dramatically in electric cars. This increased lifetime either can be tested and verified using traditional testing and measuring methods with high expenses and high risk, either not feasible at all because the time period is too long. Endurance run must cover full life time request, but full test time cannot be realized because high risk if failure occurs in late test period.

University of Miskolc, Robert Bosch Energy and Body Systems Ltd. and Bay Zoltán Non-profit Ltd. for Applied Research work in cooperation on development of accelerated lifetime test of bush bearings with specification of 30.000 operating hours which are components in cooling pumps of electrical vehicles (Fig. 1.). Testing period in this case would last 3 and half years. Aim of the project is to develop new testing and estimation methods suitable for reliable determination of expected lifetime of bearings using results of 2000-3000 hours endurance tests.

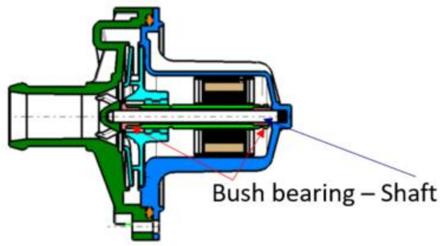

Figure 1. Electrical coolant pump (type Bosch PCE), lifetime requirement in EV is 30.000 hours [5]

II. DEVICE UNDER TEST (DUT) AND SETTING OF RESEARCH PLAN

Application device is an electric coolant pump with two sliding bearings. One of them is a simple cylindrical bearing and the other has a shoulder (Figure 2.)

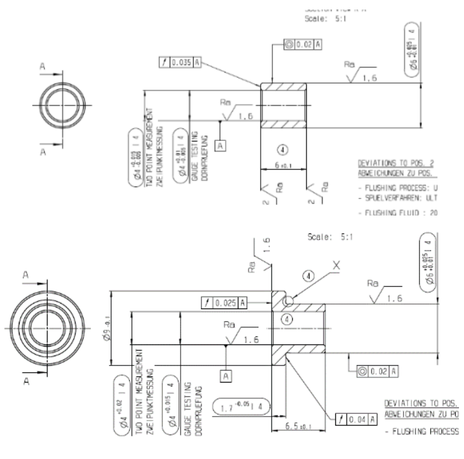

Figure 2. Bush bearings are units for testing

Main parameters of the cooling pump are included to Table 1.

Table 1. Main parameters of DUT.

Electrical data	
Nominal power / voltage	70 W / 12 V
Operating voltage	8.0 … 16.0 V
Current consumption	< 7.0 A
Electrical Interface	3-Pin (+ / S / -), Signal: PWM or LIN Interface
Hydraulic data	
Cooling fluid	Water glycol mixture
Hydraulic power	25 W
Nominal discharge pressure at 12 V	Δp = 85 kPa
Nominal flow rate at 12 V	1,000 l/h
Temperature conditions	
Temperature of cooling fluid	-40°C – 85°C
Ambient temperature	-40 °C - 125 °C (peak 140 °C)
Dimensions	
Diameter	80 mm
Length w/ port	89 mm
Weight	620 g
Durability	
Operating hours	> 8,000-30,000 h, depending on temp. load profile

The consortium team decided the following research phases:

1. Theoretical study of the wear and tear
2. Specification of the accelerating parameters for endurance tests
3. Develop and install endurance test stands for supporting mathematical model design by test results.
4. Analysis of practical wear curves and develop lifetime estimation model.
5. Develop accelerated lifetime test method for industrial use.

III. THEORETICAL STUDY OF WEAR CURE AND FAILURE FORMS

According to the relevant literature, typical wear curve (Figure 3.) can be applied to our DUTs [1].

The wear curve consists of three different sections:

1st section is the starting wear which is degressive. In case of bush bearings, it means that reaming unevenness disappears. In this section wear is relatively fast. The 2nd phase is the normal operation phase, it is the longest in time and nearly horizontal, which means that practical wear is very small. In this phase the curve is quasi-static.

When the curve starts increasing after the normal running period, it is a signal that the lifetime is coming to the end soon. This phase is progressive. Our task is to determine exact quantitative curve for bush bearings of the coolant pump.

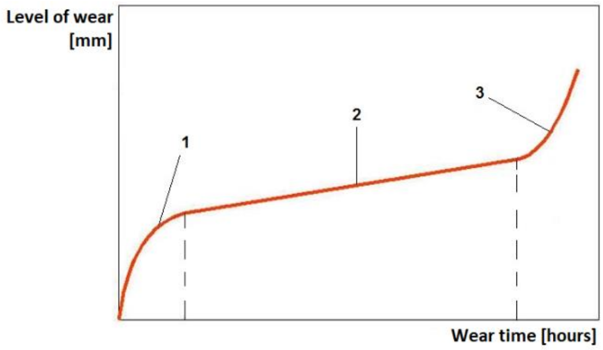

Figure 3. Typical wear and tear curve

Sliding or bush bearings have several types of usual wear which were also studied [2] [3]. These are summarized on Figure 4.

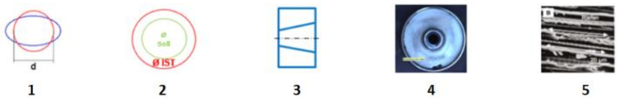

Figure 4. Typical failing forms of bush bearings

Types are:

1. Oval hole
2. Enlarged inner diameter
3. Tapered hole
4. Fractured edge
5. Grooved surface

In the first milestone of the research we aimed to define those effects which may cause the above abrasion. For this we analysed parameters which can serve as acceleration parameters in endurance tests.

IV. SPECIFICATION OF ACCELERATING PARAMETERS FOR ENDURANCE TESTS

In the pumps under test, coolant-lubricated bush bearings are used, which guide the impeller shaft. Lubrication is low and the coefficient of friction is high in the moment of start-up. Therefore higher wear of the bearing can be expected at the start-up. By increasing the speed, the bush bearing enters a mixed friction state, where the coefficient of friction and the degree of wear are much lower than in the case of boundary layer friction. As the speed increases further, the plain bearing enters a fluid friction state where there is almost no wear and the coefficient of friction is three orders lower than in the case of dry friction. [4]

Accepting the above described studies, the first accelerating parameter is the run with highly frequently repeated restarting and stopping. The reason is the missing lubrication in the moment of restarting. The only limitation in cycle time is the necessary time for settling down the shaft in the lubricant liquid.

According to the previous researches at Bosch, the second most promising accelerator parameter is the molding sand contamination in the coolant liquid. Also temperature change, speed of rotation change, mounting position or applying static load on bearings may cause faster wear in bearings. Analyzing matrix of the accelerator parameters can be seen in table 2.

Table 2. Matrix of accelerator parameters for lifetime tests

	Alternative 1	Alternative 2	Alternative 3	Alternative 4	Alternative 5	Alternative 6	Alternative 7	Alternative 8	Alternative 9	Alternative 10
Construction	Load - static	Load- dynamic	START/STOP	Temperature	[illegible]	[illegible]	Contamination	Speed of rotation	[illegible]	Mounting position
Detailed description	Increased load to bearings	Using vibromotor on bearing housing and/or using electromagnets and/or unbalanced blades / generate cavitation	Accelerated START/STOP process. Out of optimal surface friction during start process	Heating of the lubricant (or test equipment). Question is the cooling.	Changing volume of lubricant	Running without lubricant	Contamination in the lubricant can cause accelerated wear of the bearings due to the solid-solid contact.	Changing speed of rotation	Changing bushing clearance pressure conditions are changing, also load conditions may change	Mounting of pump can be horizontal or vertical. In vertical position the upper bearing is the "bull's-eye" because of possible bubbles.
Advantages	1. Wearing process is accelerated	1. Wearing process is accelerated, testing in application	1. Bearing has no optimal lubrication at start moment, which may cause higher abrasion. 2. Easy relization, good acceleration possibility.	1. Shaft/hole/bearings sizes may vary differently, also viscosity will be different, which can be useful for acceleration.	1. Different lubricant volumes creates different pressure ratio in the system, which may cause faster wear. 2.Easy application.	1. Expected very fast wearing.	1. Expected fast wearing.	1. Decreasing speed of rotation may cause out of optimal hydrodynamic friction conditions in bearings	1. Changing optimal clearance may cause accelerated wear.	1. Load of bearings is different in horizontal and vertical positions. START/STOP process can be more effective in horizontal mounting, while upper bearing probably can be weared faster in vertical position.
Disadvantages	1. Difference between loaded and normal wearing characteristics have to be analysed. Optimal load should be investigated. 2. Direct testing of bearings (not in application) would be useful	1. Difference between loaded and normal wearing characteristics have to be analysed. Conversion has to be developed.	1. Difference between wearing characteristics have to be analysed.	1. Difference between wearing characteristics have to be analysed. 2. Heating control and insulation have to be designed. (Cooling effect should be researched.) 3. Simulation results underline importance of insulation.	1. Difference between wearing characteristics have to be analysed. Intensive simulation is required before decision. High risk of usefulness.	1. Wearing characteristic expected very different from normal one. 2.Extended control development is required	1. Investigation and comparison of the wear characteristics is required	1. Investigation and comparison of the wear characteristics is required. 2. Speed control development is required, speed of rotation is constant is normal operation.	1. Investigation and comparison of the wear characteristics is required	1. Investigation and comparison of the wear characteristics is required. 2. Accelerating rate should be simulated and rated.
Remarks	Test bench will be equipped with constant radial load capacity, horizontal position of DUT, highly preferred	Cavitation is expected as an effective accelerating parameter, but conversion between functions should be developed	Good result expected - basic acceleration parameter	First tests are running at 110 °C	Not preferred in second year.	Not preferred in second year.	Sand in one test bench. Highly preferred.	In the third year can be used if necessary.	Temperature changes the clearance, further alteration is not preferred	Highly preferred

After the matrix was defined, endurance test sequences should be composed to develop the most effective endurance test stands for wear curve determination. Test sequences are described in detail in section VI. including current state of the test runs.

V. DEVELOPMENT OF ENDURANCE TEST STANDS FOR WEAR CURVE DETERMINATION

Basic requirement of the industrial partner is the testing in application, which means that test method should be validated without disassembling the pumps. Testing in application defines the main problem of the research and it is connected to the mass ratios. Several well-known methods are used for general bearing tests, including noise, vibration, temperature, current, etc. measurements. Bush bearings to be tested are made of special light mass composite carbon material and have 4 mm diameter, 6 mm height, one with shoulder and one without it. Bearings with above parameters practically cannot be tested in application by traditional bearing test methods. On the above described objectives research team decided to develop two test stands, one for application tests and one for individual part test, that is pumps are disassembled and rotors are tested individually and not in application.

Application test stand (Figure 5-6.) is suitable for testing 12 pumps at the same time. It includes two independent hydraulic cicuits, in the first one liquid is used on room temperature and in the second one the liquid is heated up to 110 °C. 12 pumps are divided into groups, 3 pumps in each group, in which different accelerator parameters are used. Running parameters and the present status of the tests are shown in tables 3-5.

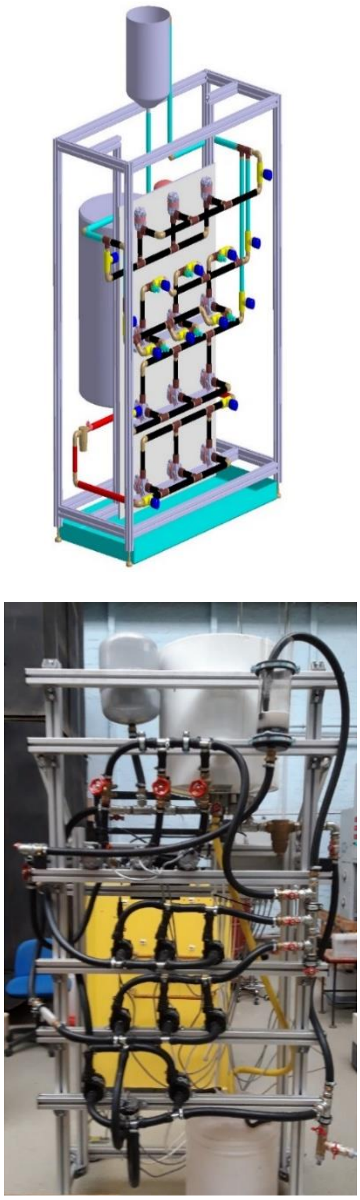

Figure 5-6. Application endurance test stand for 12 DUTs

Application test sequences and their status is summarized in tables 3 and 4.

Table 3. First runs of endurance tests

Test	Reference	A Test	B Test
Quantity of UT	3 pieces	3 pieces	3 pieces
Running time [hours]	8300	6700	6700
Sand	-	-	-
Temperature	115°C	115°C	115°C
Running method	Stop: 20 sec, Run: 10sec; Cycles: 50,000	Stop: 20 sec, Run: 10sec; Cycles: 50,000	Stop: 20 sec, Run: 10sec; Cycles: 50,000
Load	-	-	-
Cavitation	NO	NO	NO
Position	Horizontal	Horizontal	Vertical
Speed of rotation	100%	6,5%	100%
Running state	1 UT is running 2 UT is stopped after 5000 hours	1 UT is running 2 UT is stopped after 3300 hours	1 UT is running 2 UT is stopped after 3300 hours

Table 4. Endurance tests in cool liquid with sand contamination

Test	C Test	D Test	E Test
Quantity of UT	3 pieces	3 pieces	3 pieces
Running time [hours]	2500	3000	3000
Sand	Size max. 0.35 mm Sand quantity:1g/l	Size max. 0.35 mm Sand quantity:1g/l	Size max. 0.35 mm Sand quantity:1g/l
Temperature	25°C	25°C	25°C
Running method	Stop: 20 sec, Run: 10sec; Cycles: 50,000	Stop: 20 sec, Run: 10sec; Cycles: 50,000	Continuous Cycles: 500 hours
Load	-	-	-
Cavitation	NO	NO	YES
Position	Horizontal	Horizontal	Horizontal
Speed of rotation	100%	6,5%	100%
Running state	Still running	Still running	Still running

The first application tests were started at the end of 2018. year (Reference test - 3 reference pumps), and then in April 2019 tests A and B were also started with 3-3 pumps. Test stand was reconstructed by the end of 2019 and C, D and E tests were started at the very begin of 2020.

After analysing results of tests, the consortium research group concluded that practical wear curve hardly will be defined from application tests very fast. Therefore the group decided to develop a rotor tester (see Figure 7.), in which individual rotors disassembled from the pumps can be run with well-defined static load.

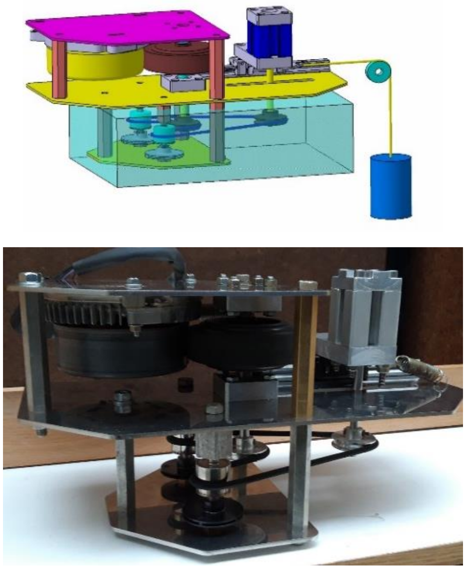

Figure 7. Rotor tester applying static load on bearings

4 x 2 DUT can be tested in parallel, drive method is friction, speed of rotation is 2500 rpm, measurements are performed after each 100 hours.

The first applied static load was 25 and 20N (tests F and G), but the rotors lifetime was less then and around of 50 hours, which is too low. The next test (H test) was run with 15N and the lifetime of bearings was around of 400hours. Now in test "I" 5N static load is applied and bearings wears are situated in phase 2 of the wear curve after 1200 hours. So this load is expected as the right one for accelerated to 2000-3000 hours lifetime test. Summary of test sequences is included into Table 5.

Table 5. Endurance tests performed with disassembled rotors

Test	F Test	G Test	H Test	I Test
Quantity of UT	2 pieces	2 pieces	8 pieces	8 pieces
Running time [hours]	<50	50	400	1200
Sand	-	-	-	-
Temperature	No liquid	25°C	25°C	25°C
Running method	Continuous Cycles: 50 hours	Continuous Cycles: 50 hours	Continuous Cycles: 50 hours	Continuous Cycles: 100 hours
Load	Static 25N	Static 20N	Static 15N	Static 5N
Cavitation	NO	NO	NO	NO
Position	Vertical	Vertical	Vertical	Vertical
Speed of rotation	80%	80%	80%	80%
Running state	Stopped	Stopped	Stopped	Still running

VI. 3D MEASUREMENTS OF BEARINGS AND RESULTS OF ANALYSIS

3D measurements of bearings are performed at the University of Miskolc by defined cycles shown in the tables 3-5. In certain defined intervals also Bosch laboratory performs validation tests.

Measuring sections of bearings is defined by Bosch and it is shown on Figure 8.

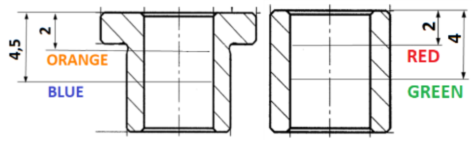

Figure 8. Measuring sections of bearings

Each bearing is measured in 2 sections, both bearings has a measuring section in 2 mm from the top surface and the lower section is placed to 4 or 4.5 mm from the upper surface depending on bearing type.

Graphs of the test results has 4 colours according to the measured sections shown on Figure 8.

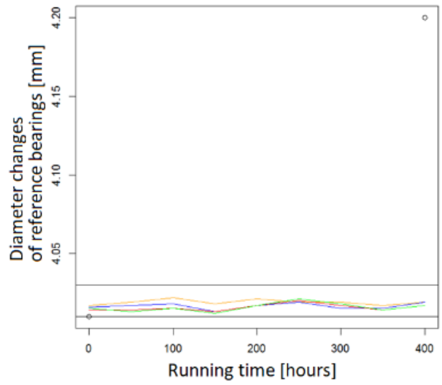

Figure 9. Reference measurement using not running bearings for eliminate environmental changes.

Nevertheless 3D measurement laboratory at the university has high quality environmental settings, constant temperature and humidity, in order to minimize measuring errors caused by environmental changes, 3D measurements always start with a reference bearing measurement (Figure 9.). These reference bearings are not running ones and used from the very first testing. Results of reference measurement are used to correct measured values. In this paper and in our research work relative diameter changes calculated with reference compensations are used for analysis.

In the framework of the present project more then 600 3D bearing measurements were performed so far including application and rotor tests.

A. Analysis of application test results

Application tests were started in first test period with 3 reference pumps and later 6 other pumps were stated. After 500.000 START-STOP cycles tests were stopped, 3 pumps were analysed by Bay Zoltán Institute, 3 pumps were returned to test and 3 were send for storage for future use if necessary. Finishing tests of some (6 out of 9) pumps was decided because all results of tests were consistent, practically no differences between pumps. Progressive phase of the wear curve is defined for all, but even after 1M START-STOP cycles pump's wearing in the quasi-static period and the end of the curve cannot be estimated.

Practical wear curve of one of the reference tests is shown on Figure 10. and summary of pumps wear for all tests are included into Table 6.

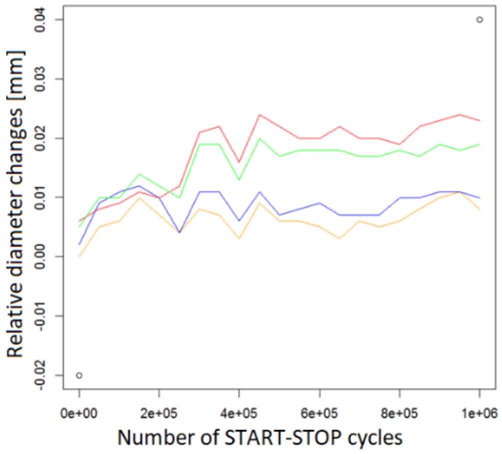

Figure 10. Results of bearing wear of reference pump after 1M START-STOP cycles

Table 6. Summary of application test results

Summary of application test status		
ID	Test	Wear description
28	Ref.	small asymmetrical wear
02	A	no wear
05	B	no wear
21	C	small asymmetrical wear
22	C	heavy asymmetrical wear
23	C	small asymmetrical wear
14	D	small asymmetrical wear
15	D	asymmetrical wear
17	D	small asymmetrical wear
18	E	average wear with small asymmetry
19	E	small asymmetrical wear
20	E	heavy asymmetrical wear

B. Analysis of rotor tests

First task in rotor tests was to define the optimal level of static load. In order to save time initial calculations resulted a maximum level of practical reality for running. This level was 30N, but tests was started with 25N (F test). Measurement results of "F test" can not be demonstrated, as all the bearings was totally worn-out within 50 hours.

The second step was 20N load (G test) when the total degradation happened around of 50 hours, and the next step is 15N load (H test), when the acceptable lifetime was around 350 hours. This wear curve is shown on Figure 11.

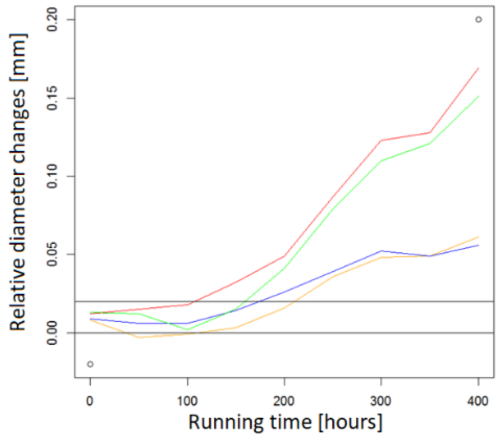

Figure 11. Rotor test results with 15N static load.

The 350-400 hours lifetime is still very short, not suitable for using as a real wear curve for lifetime estimation and do not meet the industrial partner's requirement of 2000-3000 testing time. Authors note that ratio of testing time and testing cost must be optimal, and it is set by the industrial partner. The very short testing time highly decreases reliability of lifetime estimation, increasing the time increases the reliability and also the costs.

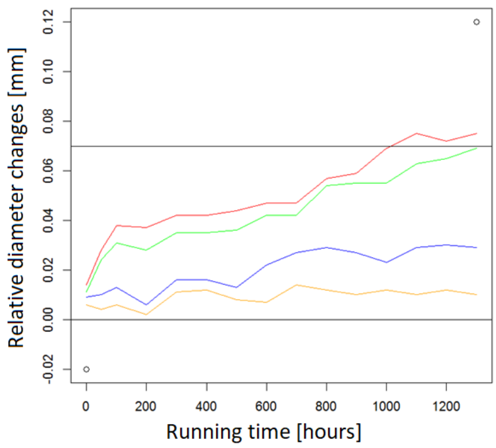

Figure 12. Rotor test results with 5N static load

The next step is to decrease static load to 5N and run new test sequences (I test). Tests with 5 N are still running and the first 1200 hours show promising results shown on Figure 12. The final phase of wear curve is not yet experienced which is correct and it is expected between 2000 and 3000 running hours.

VII. DEVELOPMENT OF ACCELERATED LIFETIME TESTING METHODS FOR APPLICATION TESTING

Results of application tests proved that no START-STOP cycles, no sand contamination, no speed or liquid temperature changes, no cavitation control will result the necessarily accelerated testing. The only static load is the parameter resulting effective acceleration. In the research period static load was used not in application but in disassembled form, which means that rotor was the DUT.

The final task is to develop and design such methods which are suitable for applying static load to the bearings in application.

Two methods will be designed for solving the above problem, one is a mechanical solution and one is electromagnetic solution. Electromagnetic solution will be worked out in next phase of the research.

A. Mechanical solution of applying static load in application

Radial load exists during normal usage in car, for example at acceleration or turning. For example in a car travelling by 80 km/h speed on a 80 radius arc, radial load of the rotor is 0.5N. Also during breaking a car and decelerating it by 6,2 m/s2 radial load will be the same 0.5N.

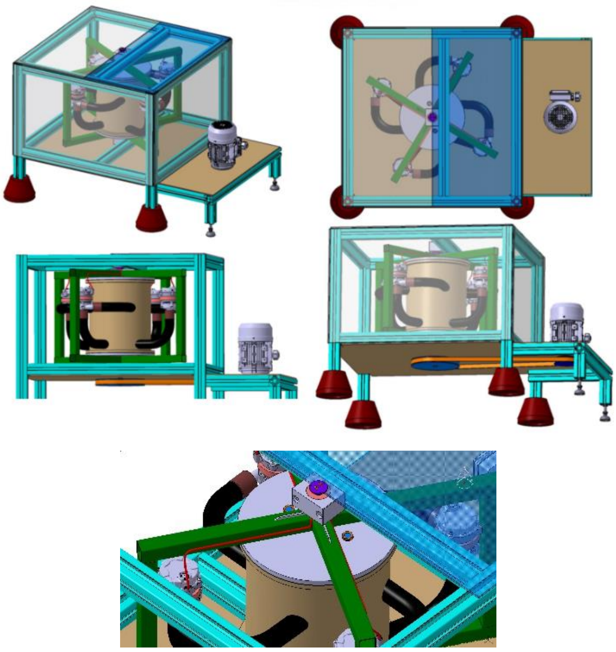

Figure 13. Mechanical loading method for accelerated endurance test

Conceptional design of the test stand is based on the above described idea. The present design includes 4 pumps as DUT (Figure 13.). Static radial load is applied by rotation. Geometry and number of pumps can be changed by changing the construction sizes of the stand. Speed of rotation is controlled according to the necessary load level.

In the example:

Mass of rotating part:	82 g
Diameter of rotation:	500 mm
Speed of rotation:	150 rpm
Radial force:	5,05 N

Depending on acceleration level, the load can be increased.

VIII. CONCLUSIONS AND OUTLOOK

Endurance tests show that cylindrical bearings wear is higher than wear of bearings with shoulder, which refer to the normal practical usage of bearings in the coolant pump DUT.

Accelerating parameters are analysed and tested, results did not meet expected efficiency of parameters, neither applying START-STOP cycles, not the sand contamination. Only the effective accelerator parameter is the static load which should be refined.

Design and development of test stands for application testing has started in two directions: one direction is a mechanical rotating table; the second is applying electromagnetic field. Board models for both methods will be performed and analyzed in near future.

IX. ACKNOWLEDGMENTS

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