Article citation info:
Vrublevskyi O, Skonieczna D. Optimisation-oriented verification of a plain bearing process model taking into account actual tolerances and measurement accuracy. Combustion Engines. 0000;XXX(X):xx-xx. https://doi.org/10.19206/CE-207652

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Optimisation-oriented verification of a plain bearing process model taking into account actual tolerances and measurement accuracy

ARTICLE INFO

Received: 12 May 2025
Revised: 25 June 2025
Accepted: 27 June 2025
Available online: 16 July 2025

This paper presents a methodology for verifying a numerical model of the plain bearing test process used to evaluate the characteristics of internal combustion engine components, in particular camshaft bearings. The developed approach is based on the use of optimisation methods under parametric uncertainty, which makes it possible to take into account the actual spread of technological and operational parameters.
The study uses a test rig that reflects the operating conditions of a bearing in an internal combustion engine, including a load simulated with an eccentric. SAE 15W40 grade oil, typical for engine lubrication systems, was used as the lubricant. The input parameter space includes geometrical features of the bearing (diameter, width, clearance, eccentricity), initial load force, shaft speed, and rheological properties of the oil.
The proposed approach to verification does not involve a direct comparison of computational and experimental data, but rather a search for the most probable solution within given tolerance limits and taking into account the measurement accuracy of the selected characteristics. The verification criteria are the measured values of oil and bearing surface temperature, load force, and friction torque in the oil film. Measurement uncertainty is also taken into account in the optimisation process.
The developed methodology makes it possible not only to assess the reliability of the numerical model, but also to analyse the sensitivity of the model to parameter variability and to determine the robustness of the friction node under study.

Key words: *camshaft, engine oil, wear, robustness, kinematic node*

1. Introduction

Nowadays, plain bearings are an important component of various technical objects [17], including in vehicle drive systems. The introduction of innovative control concepts with artificial intelligence elements, which are precisely adapted to changing operating conditions, requires the search for new or clarification of known bearing criteria (characteristics). Therefore, it is important to search for criteria that are able to combine tribological, thermodynamic, and hydrodynamic characteristics of the bearing processes. The success of this search is possible through the synergistic use of detailed simulation models describing bearing processes and experimental studies. It is worth noting that a number of research results have been published in recent years that present bearing calculation methods verified using empirical data.

For example, in [3], the authors present a highly effective computational method applied in the context of plain bearings in planetary gearboxes, validated using pressure measurement data in a wind turbine gearbox. The authors use the simulation results to analyse the influence of structural deformation and pressure distribution in the lubricating film. Fernández et al. [5] developed a bearing model using MSC software for multibody dynamics analysis. Adams performed its experimental validation. In addition, the authors use the presented model to minimise frictional losses in the bearing. König et al. [9] presented a method for predicting friction losses in plain bearings in start and stop modes. The paper [9] compared numerical models with experimental data, emphasising the importance of accurate simulation of mixed lubrication conditions. Machado and Cavalca, in their article [11], present an experimental validation of a bearing wear model using rotor frequency characteristics. The authors note good agreement between simulation and experiment. Goto et al. [7] developed a model for diagnosing plain bearing wear using the support vector method (SVM), validated using experimental data. The authors [7] indicate that high accuracy is achieved in wear diagnosis. Li et al. [4] developed a program to calculate friction in plain bearings and conducted experimental tests on a tribological test machine. Panara et al. [13] emphasise the importance of accurate simulation for high-speed and loaded bearings. The authors presented numerical models of fluid dynamics in bearings validated against experimental data.

Despite the fact that many contemporary works have proposed effective methods to validate plain bearing models using experimental data, in most cases a direct comparison of calculated and measured values at fixed parameters is used. This approach, although useful, does not take into account the actual uncertainty of the design parameters and test conditions. This paper proposes an alternative methodology in which the uncertainties in the input parameters and verification criteria are treated as an integral part of the problem, allowing for a more comprehensive and realistic assessment of model reliability.

One of the key challenges associated with the verification of engineering simulation models is the presence of unavoidable uncertainty in both the input parameters and the experimental results. Input data are characterised by fuzziness due to manufacturing tolerances, operational variability and limited parameter accuracy. On the other hand, the experimental verification criteria – temperature, load, and friction torque – are also subject to measurement errors. As a result, a solution space is created with undefined boundaries in both input and output parameters. Under these conditions, the verification problem should be considered not as a direct comparison of calculated and empirical values, but as a problem of finding such sets of input parameters in which the calculated values fall within an acceptable (fuzzy) experimental space. Such an approach also makes it possible to analyse the sensitivity of the model and assess its robustness (resistance to parametric bias).

In order to solve the described problem, different approaches can be used to explore the fuzzy parameter space. First of all, attention should be paid to optimisation algorithms. These algorithms provide for the generation of sample points [1, 12, 15], parameters located in the space, and the use of the multi-criteria principle [2, 16].

The aim of this paper is to develop and demonstrate a multi-criteria method for the verification of a simulation model of a plain bearing, taking into account parametric uncertainty and tolerances of experimental measurements. The proposed method combines experimental testing with optimisation parameter selection that ensures convergence of calculated and empirical characteristics, thus providing not only verification but also sensitivity and stability analysis of the model.

In order to solve the above problem of verification of the computational model created for an in-depth study of the processes accompanying the operation of a sliding bearing, the paper presents the methodology of experimental testing, basic information about the created model, the method of processing empirical data, the author's verification algorithm with elements of optimisation, as well as a sensitivity study of the tribosystem under study.

2. Subject of the study, features of the experiment, and measurement methods

The processes taking place in a plain bearing, the design of which is used in internal combustion engines, were investigated using a stand (Fig. 1). The set speed of the bearing shaft ω (Table 2) was provided by an electric motor and a manual gearbox. The stand is equipped with a load system and measuring channels for the bearing friction index M, the force acting on the bearing F_N, the oil temperature T_{oil}, and the surface temperature of the bearing bushing $T_{bearing}$. During the experiment, the plain bearing was immersed in SAE class 15W40 engine oil with a volume of 0.3 dm^3. Since one of the objectives of the experiment was to test the bearing from a tribological aspect and also to simulate the operation of the friction assembly under varying load conditions, which is typical for friction assemblies of internal combustion engines, the shaft had an off-centre mounting hole (eccentric ε). This provided a force variation during shaft rotation of ±20% of the nominal value, which will be shown below in the load force measurement results.

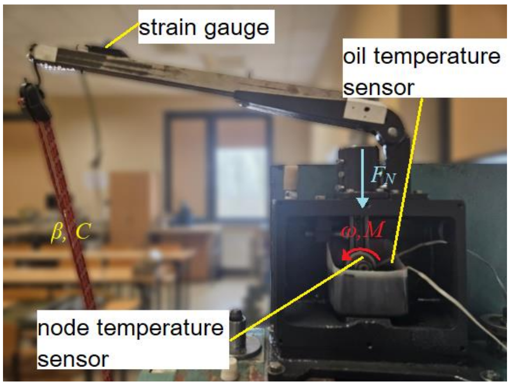


Fig. 1. Photo of the plain bearing and sensors on the test

Direct measurements of geometrical quantities were mainly used to determine the parameters of the plain bearing (Table 1). To determine the radial clearance h in the bearing, digital models of the shaft and sleeve (Fig. 2a, b), obtained by 3D scanning with the GOM® ATOS Core 80 scanner with post-processing in GOM Inspector, were used. For the central section, the clearance profile formed by the friction pair of the shaft-bushings (Fig. 2c) allowed the average value to be determined (Table 1). From the results presented, it can be seen that the bushing used in the test is close to the nominal diameter of 25 mm, with a deviation of no more than –0.02 mm. The bushing has a positive deviation of +0.03 mm. It should also be noted that the scanning accuracy is ±0.02 mm.

The method of conducting the experimental test was to load the bearing rotating at 300 rpm. The final experimental time was determined by reaching a temperature of 120°C on the surface of the bushing. For the tested bearing, the experimental time was approximately 5000 s.

As shown above (Fig. 1), the bench is equipped with channels for measuring load force F_N, friction torque M and temperatures (T_{oil}, $T_{bearing}$). A TorqueSensor Series 2300 from NCTE was used to measure the friction torque. The bearing load force was measured using a BF1K-3EB bridge strain gauge with a measuring base of 8.5×8.5 mm to ensure thermal compensation and linearity of characteristics and the temperature of the bushing surface (sensor position is shown in Fig. 1) using an Omega K-Type thermocouple. The measurement accuracy of the sensors used is shown in Table 2. A Picoscope 3000 series ADC and PicoScope 7 T&M® software were used for data collection and initial analysis.

Table 1. Plain bearing parameters

Parameter	Designation	Unit	Nominal value	Accuracy	Average	Deviation
Radial clearance	h	mm	0.020	±0.002	0.020	0.005
Stiffness of the load elastic element	C	$N \cdot m^{-1}$	100	±10	100	10
Damping coefficient of the load elastic element	β	$N \cdot (m \cdot s^{-1})^{-1}$	14	±1.4	14	1.4
Starting force	F_0	N	60	±6	60	6
Eccentricity	ε	mm	0.70	±0.07	0.70	0.07
Bearing width	B	mm	19	±0.19	19	0.19
Diameter of the sleeve	D	mm	25	±0.25	25	0.25

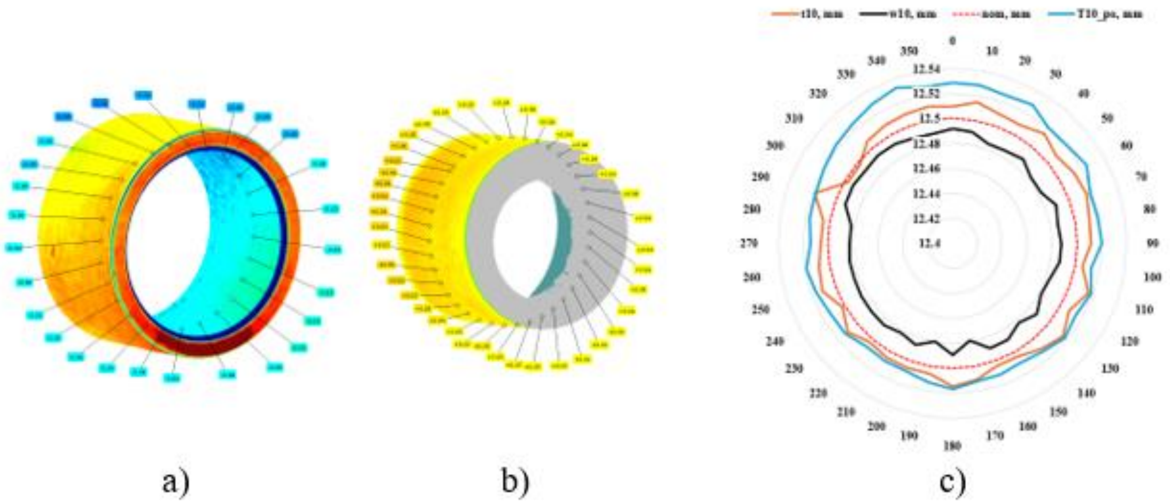


Fig. 2. Results of scanning the surfaces of the bushing (a), shaft (b) and clearance profile at the centre of the plain bearing (c)

Based on the measurements and tolerance analysis, it can therefore be concluded that the model input parameters vary within the ranges described in Table 1, and the measured characteristics vary to an accuracy of ± 7% (Table 2). These ranges were taken into account as part of the multi-criteria verification procedure.

3. Simulation model of a test stand with plain bearing

A simulation model of a sliding bearing test rig has been developed for the verification procedure, taking into account the identified parametric uncertainties. The model is built in the Amesim Simcenter environment and includes physical modules that describe friction, heat transfer and load kinematics.

In this study, the bearing model implemented in the simulation model of the test stand was used to increase the reliability of the modelling processes in the plain bearing and to develop a verification methodology with optimisation elements under fuzzy parameters and criteria. The computational scheme (Fig. 3) and the subsequent simulation were implemented in Amesim Simcenter space [14].

Table 2. Verification criteria and the accuracy of their determination

Name	Designation	Unit	Nominal value in the experiment	Accuracy
Verification criterion				
Oil temperature	T_{oil}	°C	48	±0.5°C
Bearing sleeve temperature	$T_{bearing}$	°C	82	±0.75%
Bearing load force	F_N	N	750	±7%
Friction torque	M	Nm	0.443	±5%
Friction power	P_{fr}	W	14	±3 W
Regulatory parameter				
Shaft speed	n, ω	rpm, $rad \cdot s^{-1}$	300, 31.4	±2.5%

The structural diagram (Fig. 3) is a set of interconnected modules that reflect the kinematic, tribological and thermodynamic processes in the friction unit. The key elements of the model are:

– Electric drive – sets a nominal shaft speed similar to that used in the experiment (300 rpm). Fluctuations in the shaft speed are taken into account in the simulation, which is ±2.5% of the nominal speed.
– Load mechanism – implemented by means of an eccentric, which allows the reproduction of a variable load force with an amplitude of ±20% of the mean value, as in the physical laboratory bench. In the loading mechanism, the stiffness and damping properties, as well as the preload force, are defined using a submodel of the elastic element. These parameters are also vague and are included in the verification process.
– The sliding bearing simulation unit is based on the solution of the Reynolds equation [8], which describes the pressure distribution in the lubricating layer during hydrodynamic lubrication, taking into account the non-stationarity of the loading and rotation conditions. In addition, the Goenki model [6, 10] is used to describe the thermal interaction in the tribosystem, which provides a calculation of heat dissipation depending on local friction parameters and lubricant viscosity.

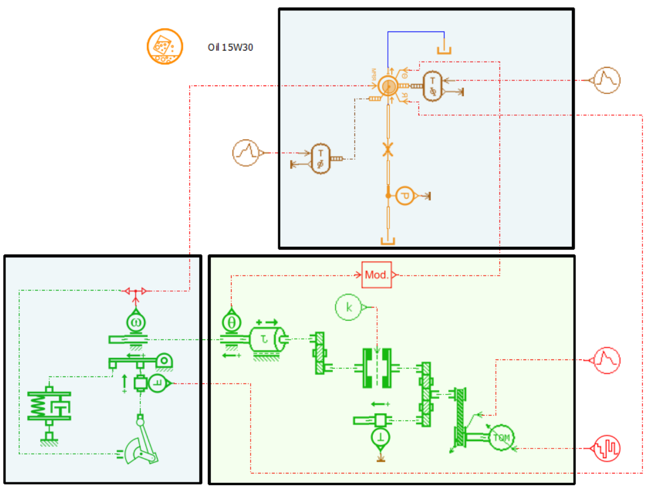


Fig. 3. Construction scheme of plain bearing test stand

– Heat transfer modules – simulate heat flows between bearing, shaft, and oil medium. This allows the change in temperature under load to be monitored, including self-heating and cooling.
– Sensors are virtual measurement units that correspond to the physical sensors (friction torque, temperature, load) used in the experiment. This allows comparison of modelled and empirical data for similar features.

In developing the model, a wizard was used to determine the physical and rheological properties of the oil. Empirical data obtained by the authors in the process of testing the engine oil samples used in the experiment were used to determine these properties.

The use of the Reynolds equation (1) allows the influence of geometric tolerances (for example, radial clearance) and lubricating film dynamics on the behaviour of the system to be taken into account, while the use of the Goenka model (2) allows an adequate assessment of the local thermal effects that occur under varying operating modes.

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left(\rho h^3\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\rho h^3\frac{\partial p}{\partial z}\right)= = 6\eta\left(\omega_j+\omega_b\right)\frac{\partial ph}{\partial\theta}+12\eta\frac{\partial ph}{\partial t} \quad (1)$$

$$\frac{d}{dt}=\frac{|F_N|\left(\frac{h}{r}\right)^2}{\eta BD}M\left(\varepsilon,\frac{B}{D}\right)+\omega\times e \quad (2)$$

where is p(z, θ, t) is the oil film pressure, ρ is the lubricant density, r is the bearing radius, h(z, θ, t) is the radial clearance in the bearing, η is the oil film viscosity, Θ is the bearing angle, and ω_j, ω_b are the journal angular velocity and bearing angular velocity.

Therefore, the model structure provides not only physical plausibility, but also high sensitivity to parametric changes, which is essential for multi-criteria verification.

4. Multi-criteria bearing model verification algorithm

A proprietary algorithm for multi-criteria verification of the mathematical model was proposed to account for uncertainties in input parameters and experimental measurement errors. Its structure is shown in the diagram (Fig. 4) and reflects the key steps in the integration of simulation modelling, experimental, and optimisation methods.

The method is based on the assumption that both the model input parameters and the experimentally obtained output characteristics are described not by point values, but by ranges resulting from technological and measurement uncertainties. The task of verification is to determine the parameter vector in which the simulated output values (criteria) are within the acceptable range defined by the experimental data, taking into account errors.

The algorithm involves the following main steps:

1. Creation of the parameter space. Nominal values and acceptable ranges of the model input parameters (for example, clearances, viscosity, eccentricity) are determined taking into account measurement errors, manufacturing tolerances, and operating conditions (Table 1).

2 Generating test points. Experimental design methods (e.g. Sobol nets, Monte Carlo method, Latin Hypercube) generate a set of test vectors of input parameters that uniformly cover the parameter space.

3 Running the simulation. For each test vector, the values of the verification criteria are calculated: oil temperature, friction torque, load, oil film temperature.

4. Definition of the correct range. Based on the experimental data and their errors, tolerance ranges are created for each criterion (Table 2). Together, these form a multidimensional parallelogram of acceptable solutions.

5. Comparison of calculated and experimental criteria. If at least one set of calculated outputs (vector) falls within the range of valid values, the model is considered verified. The parameters corresponding to these simulations form a set of valid solutions.

6. Sensitivity and robustness assessment. The sensitivity of the model to parameter changes is analysed, and the areas of stability of the friction unit, i.e., the parameter ranges at which stable operation of the plain bearing is maintained, are determined.

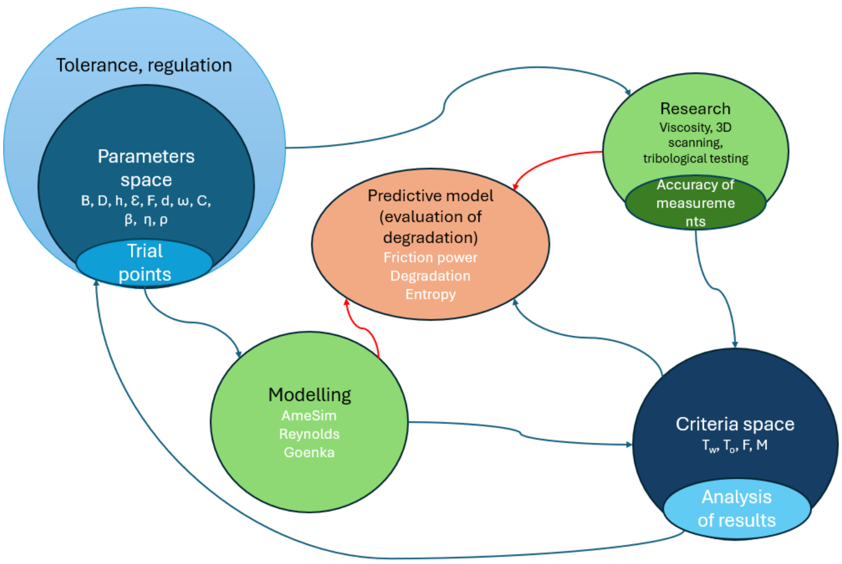


Fig. 4. Structure of the multi-criteria verification algorithm for the bearing model

Consequently, model verification takes the form of a vector parametric identification problem in the presence of fuzzy input and output data. The advantage of the proposed approach is its ability to account for real uncertainty at all levels, which is particularly important in the study of complex tribosystems operating under varying conditions.

5. Measurement results and analysis of verification criteria

5.1. Experimental observations

To verify the plain bearing model, a thermally stabilised mode of operation was used in which the system reached a steady state temperature. In this mode, the friction torque M and the load force F_N varied in time in a near sinusoidal manner due to the eccentricity (Fig. 5). Consequently, the friction power P_{fr}, calculated as the product of $M \cdot \omega$, where ω is the angular velocity of the shaft, also showed regular oscillations.

To ensure the validity of the comparison with model data, synchronisation of the measured signals is crucial. In this study, in order to synchronise the criteria M(t) and $F_N(t)$, an approach based on minimising the variance of the current friction coefficient μ_{fr}, calculated according to the formula: $\mu_{fr} = \frac{M}{F_N \cdot 0.5 \cdot D}$.

The optimum moment shift was ~0.016 s, ensuring the physical behaviour of $\mu_{fr}(t)$. After synchronisation, the coefficient of friction stabilised within $\mu_{fr} \approx 0.046 \pm 0.002$, corresponding to known values for mixed and hydrodynamic lubrication modes.

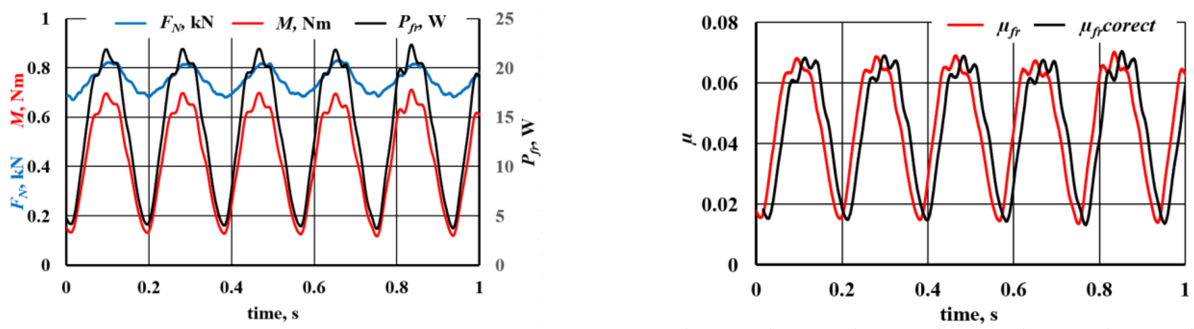
Analysis of the friction power showed stable oscillations between 4 and 22 W, without the presence of drift or instability. This confirms the correct operation of the test rig and sufficient repeatability.

The characteristics presented – friction torque, load force, and power – are used as key verification criteria. Their behaviour over time confirms that the measured data are physically interpretable, stable and suitable for further comparison with simulation results.

Thus, the data preprocessing, including synchronisation, smoothing, and evaluation of the stability of the criteria, provides a reliable basis for implementing the proposed multi-criteria verification algorithm.

5.2. Results of model verification

The stochastic generation of test points in the model parameter space was used to implement the developed multi-criteria verification methodology. For this purpose, a Monte Carlo method was used with a normal distribution of the input variables around nominal values. The use of Gaussian



a) b)

Fig. 5. Run area with stabilised plain bearing operation: a) measured load force F_N, bearing friction torque M and friction power P_{fr}, b) result of determining friction coefficient $\mu_{fr}(t)$ with correction of torque signal M

Table 3. Acceptable solution vectors

# point	B, mm	β, N·(m·s^{-1})$^{-1}$	C, N·m^{-1}	D, mm	F_0, N	h, mm	ε, mm	F_N, N	P_{fr}, W	T_{oil}, °C	T_{bearin}, °C
1	19.29	13.79	87.16	24.64	52.85	0.03	0.51	843.84	9.87	47.96	81.27
…											
3	18.97	12.26	95.92	25.15	50.59	0.02	0.39	799.49	11.21	47.96	81.33
…											
11	18.87	14.75	103.82	24.65	45.10	0.01	0.44	706.32	16.84	48.08	81.91
…											
24	19.02	10.23	92.50	25.20	44.69	0.02	0.40	702.76	15.52	48.02	81.64
25	18.99	15.93	86.91	25.01	46.83	0.02	0.40	741.29	15.62	48.02	81.64
…											
37	19.18	13.23	98.69	25.07	47.29	0.02	0.41	744.02	15.09	48.01	81.59
…											
40	19.21	14.82	97.16	25.09	47.47	0.02	0.36	746.41	13.82	47.99	81.49
…											
42	19.42	14.15	93.27	24.93	44.75	0.02	0.38	703.22	12.65	47.97	81.43
…											
46	18.97	14.74	108.51	25.19	46.08	0.02	0.49	722.01	15.32	48.01	81.61
…											
48	19.28	16.41	96.53	24.97	48.86	0.02	0.34	769.08	14.60	48.00	81.55
49	19.08	16.58	103.82	24.65	48.65	0.02	0.43	765.02	14.13	47.99	81.54
50	18.78	13.29	111.63	24.83	56.48	0.027	0.35	889.05	10.29	47.96	81.29

dispersion reflects the physical nature of variances such as process tolerances, inconsistent operating conditions and limited measurement accuracy.

5.3. Generation of input data and parameters

The calculations used 50 test points (Table 3), each representing a unique set of parameters: radial clearance h, stiffness C, and load system damping β, initial force F_0, eccentric ε, bushing width B, and shaft diameter D. Mean values and standard deviations are determined for each parameter, as shown in the Table 1.

5.4. Modelling and sampling

A numerical simulation of the plain bearing was carried out for 150 seconds for each test point, which was sufficient to achieve a quasi-stationary temperature regime. The calculations were carried out in the Amesim Simcenter simulation environment. The total execution time per calculation did not exceed 20 minutes. The output included the values of the key verification criteria: oil temperature, bushing surface temperature, load force and power.

5.5. Verification analysis in the criteria space

Test points were analysed in two-dimensional criteria spaces:

– Oil temperature – bushing temperature (Fig. 6)
– Friction force – load force (Fig. 7).

In the first diagram, most of the points were within the tolerance range set by the accuracy of the measurements. This is particularly true for points with smaller radial clearance values (e.g. positions 8, 27, 34, 41, 42, 49), confirming the sensitivity of the thermal behaviour of the model to the geometry.

In the second graph, the proportion of deviation points increases, indicating that the model is more sensitive to changes in loading parameters. Nevertheless, a compact cluster of trial solutions was identified (points 3, 11, 24, 25, 37, 40, 42, 46, 48, 49) that fell within the confidence interval for all criteria.

5.6. Interpretation of results

Analysis of the distribution of sample solutions allows the following conclusions to be drawn:

– The verification is considered successful because there are a number of points that meet all criteria within the experimental accuracy limits.
– A robust range of input parameters is revealed within which the model gives output characteristics consistent with experiment.
– At the same time, the model is found to have limited sensitivity within the current tolerances: most parameters have a weak effect, with the exception of the clearance h and load parameters.
– This provides a basis for using the model in engineering practice as stable or, if necessary, for refining the model structure in sensitive areas.

5.7. Verification

To quantify the effectiveness of the proposed verification methodology, ratios of simulation results falling within experimental tolerances were calculated according to the key criteria. A verified test point is one for which all calculated criterion values were simultaneously placed within the established uncertainty ranges obtained from the accuracy of the measurement systems.

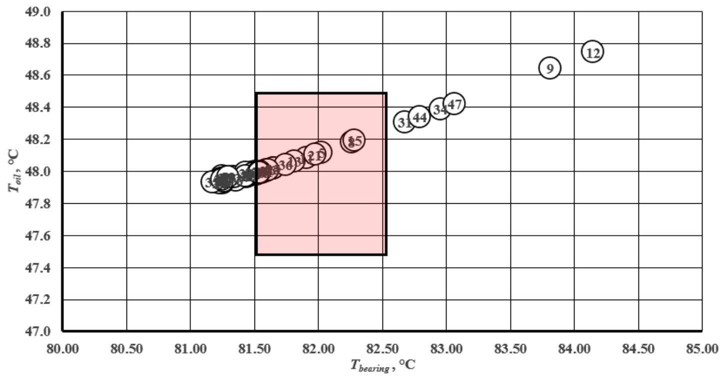


Fig. 6. Location of test points in the plane of the verification criteria oil temperature T_{oil} surface temperature $T_{bearing}$

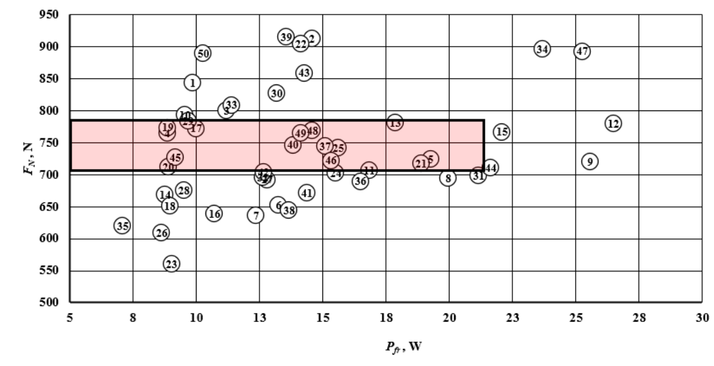
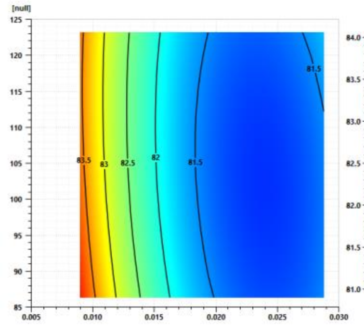
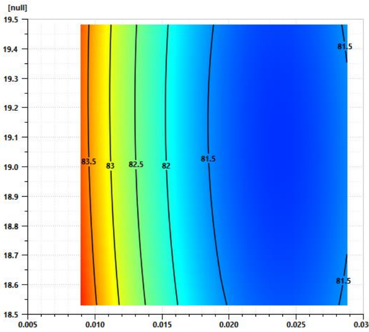


Fig. 7. Location of test points in the plane of the verification criteria, friction power P_{fr} and force F_N

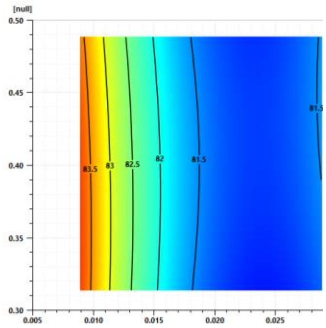
Table 4. Variation of the surface temperature of the bearing sleeve as a function of the input parameters



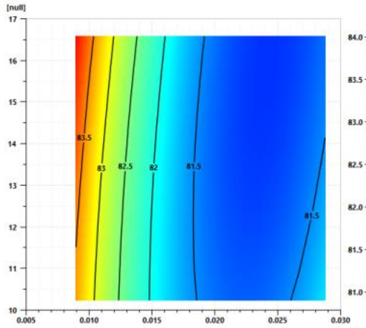
h, C



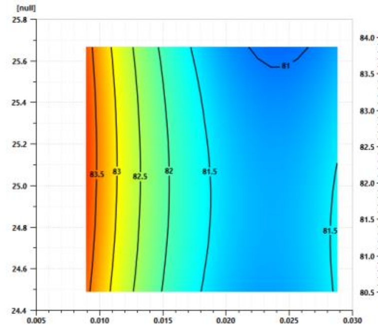
h, B



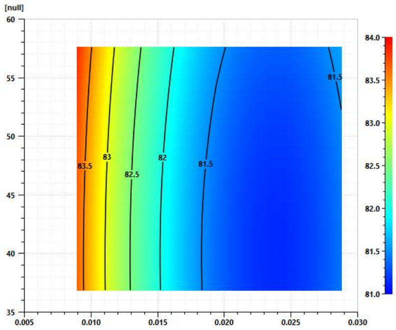
h, ε



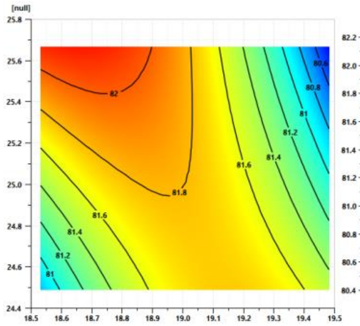
h, β



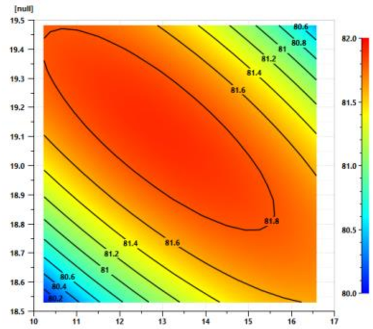
h, D



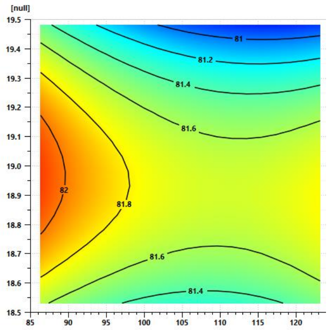
h, F_0



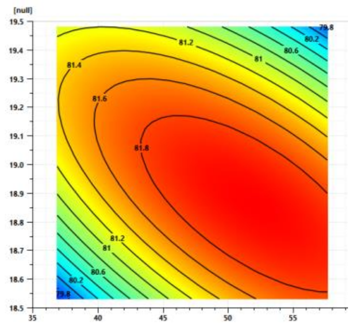
B, D



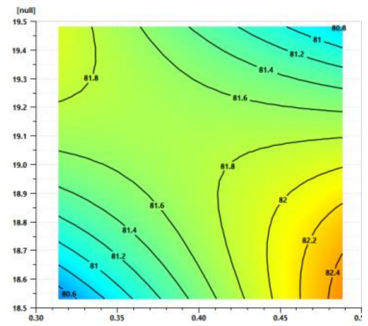
β, B



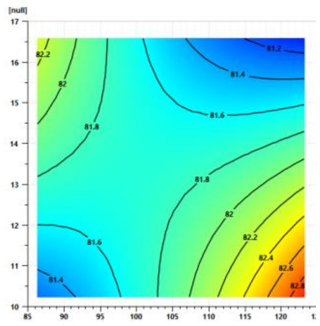
C, B



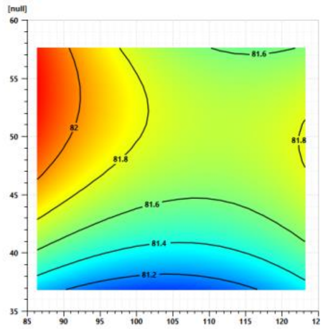
F_0, B



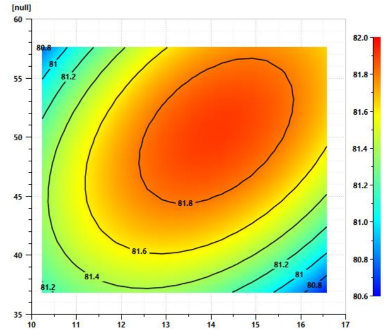
ε, B



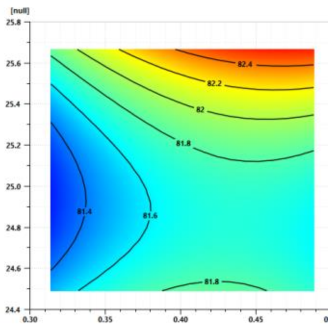
C, β



C, F_0



β, F_0



ε, D

The following criteria and acceptable deviations were selected for the verification assessment (Table 2).

As part of the verification step, 50 test points (Table 3) generated by the Monte Carlo method based on Gaussian distributions of the model parameters were analysed. Calculations were carried out in steady-state thermodynamic mode. The results obtained allowed the following indicators to be evaluated:

– number of points meeting the two temperature criteria: 37 out of 50
– number of points meeting both power and load criteria: 10 z 50
– number of fully verified points (according to all criteria): 10 out of 50
– total percentage of verification: 20%.

The percentage of verification obtained confirms the realism of the model, as in the case of parametric fuzziness and metrological uncertainty, absolute convergence is not required, but the existence of a domain of acceptable solutions is sufficient.

The identified cluster of verified solutions (10 points) indicates the existence of a parameter domain in which the model shows consistency with the experiment. This confirms the adequacy of the model structure, the physical consistency of the mechanisms for generating output values and the robustness of the model to acceptable parameter dispersion.

At the same time, the percentage of unverified points highlights the limited sensitivity of the model to certain parameter combinations, which may serve as a starting point for further refinement of the model.

6. Sustainability analysis

The final stage of the work was to assess the stability of the investigated plain bearing to changes in its parameters. As mentioned above, such changes are inevitable during the production, adjustment and operation of the bearing. Using the calculation results of the verified model (Fig. 6 and Fig. 7), it is possible to analyse the effect of selected parameter pairs on the surface temperature of the bearing sleeve (Table 4).

In spaces where one of the parameters is the radial clearance in the bearing h, the other parameter does not play a decisive role in the surface temperature level of the bearing bushing in the investigated steady state of operation. This indicates the dominant influence of the clearance on the thermal regime of the bearing and the low sensitivity of the system to other parameters at a constant value of h. The bearing shows local resistance to variations in the parameters C, B, ε, β, D, F_0 if the radial clearance is within tolerance. This also suggests that it is possible to simplify the model or control the system by focusing primarily on the clearance as the main factor affecting thermal behaviour.

In all other combinations (without h), the temperature field shows a clear extreme, i.e. a maximum or minimum temperature in the middle of the tested parameter range. In these zones, the system is sensitive to simultaneous changes in two parameters, which can result in thermal overload or, conversely, a mode with minimal losses. In this way, it is possible to identify unsafe parameter combinations that lead to overheating, and to locate optimal parameter combinations that ensure minimum temperatures (and probably minimum wear).

On the basis of a pairwise analysis, the following can be assumed:

A number of practical conclusions can be drawn from the analysis:

– A key factor is the radial clearance h
– Under variable parameter conditions without backlash control, the model shows local areas of high sensitivity, which can lead to unstable operation or overheating
– This technique allows the identification of zones of stability and instability, which is extremely important for design tasks, tolerances, diagnostics, and monitoring.

Table 5. The importance of the relationship between the parameters

C	It is sensitive in combination with other dynamic parameters (ε, F_0, β), influencing force transmission and thus losses.
β	It participates in several pairs with a clear extreme, influencing the stabilisation mode and amplitude of load fluctuations.
ε	A noticeable effect in combination with F_0, B, and D affects the change in cycle load and pressure distribution.
B	In combination with other geometrical parameters, it determines the contact area, but can both reduce and increase the temperature depending on the combination.
F_0	It affects pre-stress and can significantly alter temperature modes in combination with other power parameters.

7. Conclusions

The following conclusions can be drawn from the study:

1. A method for multi-criteria verification of a simulation model of a sliding bearing operating in a real experiment has been developed. The technique takes into account the parametric fuzziness of the input data and the errors of the measurement systems.

2. The verification was carried out by generating 50 test points using the Monte Carlo method, taking into account the statistical distributions of the parameters. Of these, 37 points met the temperature criteria, 10 met the load-power criteria and 10 met all criteria. This corresponds to an overall verification success rate of 20%, confirming that there is a range of valid design solutions.

3. Sensitivity analysis showed a dominant influence of radial clearance on the temperature behaviour of the system, with eccentricity and load parameters influencing the shape of the power and torque distributions. This highlights the key role of geometric tolerances in ensuring component stability.

4. This technique not only allows verification to be carried out, but also to identify areas of robustness – ranges of parameters at which the model remains within experimental tolerances. This is particularly important for engineering calculations related to tolerance selection and condition control.

5. The developed model and verification algorithm focus on plain bearings, typically used in internal combustion engines (crankshaft supports, camshafts, etc.), where evaluation of heating, friction, and loading is important under conditions of limited access to measurements.

6. The proposed method can be generalised to other types of bearings and contact assemblies where similar sources of uncertainty are observed.

Nomenclature

B	bearing width
C	stiffness of the load elastic element
D	diameter of the sleeve
F_o	starting force
F_N	bearing load force
h	radial clearance
M	friction torque
n	shaft speed
p	oil film pressure
P_{fr}	friction power
r	bearing radius
$T_{bearing}$	bearing sleeve temperature
T_{oil}	oil temperature
β	damping coefficient of the load elastic element
ε	eccentricity
η	oil film viscosity
Θ	bearing angle
μ_{fr}	current friction coefficient
ρ	lubricant density
ω	shaft speed
ω_b	bearing angular velocity
ω_j	journal angular velocity

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