

Research Article

The Theoretical Foundations of the Thermal Conductivity of Housing-Type Part Assemblies Restored with WEICON-TI Metal Polymer

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Abstract

This article explores the theoretical foundations for improving the durability of housing-type parts restored with WEICON-TI metal-polymer, focusing especially on bearing assemblies, where thermal conductivity is crucial. Modern mechanical systems rely heavily on the reliable performance of bearings, as they are constantly exposed to dynamic loads, friction, and varying operating temperatures. When the housing surfaces become worn or damaged, restoring them using polymer-metal composites like WEICON-TI provides an efficient and cost-effective alternative to traditional repair methods. One of the key factors affecting the performance of these restored assemblies is the material's ability to conduct heat away from contact surfaces. Proper thermal conductivity not only helps stabilize operating temperatures but also reduces the risk of localized overheating, which can lead to accelerated wear, microstructural damage, or even failure of the assembly. Therefore, understanding the principles of heat transfer in metal-polymer restored surfaces is essential for predicting service life and ensuring long-term reliability. This article systematically analyzes these theoretical aspects and shows how the thermal conductivity of WEICON-TI contributes to the enhanced load-bearing capacity, stability, and operational safety of restored bearing assemblies. By efficiently transferring heat, the material prevents excessive temperature rises, minimizes wear, and helps maintain the mechanical integrity of the system. As a result, parts restored with WEICON-TI last longer, operate more safely, and provide more stable performance under demanding conditions. Understanding these principles allows engineers to optimize repair processes and ensure that mechanical systems continue to function reliably over time, even in challenging thermal and mechanical environments.

Keywords

Reliability, Durability, Thermal Conductivity, Housing-type Parts, Bearing, Physico-mechanical Properties, Adhesive, Joint, Surface, Hub, WEICON-TI Metal Polymer, Restoration, Thermal Conditions

1. Introduction

The durability of housing-type parts with bearing assemblies depends on their physico-mechanical properties as well as the level of thermal conductivity.

The reliability of bearing assemblies is determined by their thermal conditions.

The transfer of heat to the housing is especially important

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during the start-up of the hub, since its temperature is much lower than that of the bearing. After the hub heats up, the transfer of heat to the housing stabilizes and decreases.

In steady operating mode of the hub, about 85-90% of the heat generated in the bearing is dissipated into the atmosphere through the lubricant, the bearing housing, and the hub housing, while 10-15% is transferred to the hub shaft [1, 2].

2. Manuscript Formatting

The heat balance equation of the bearings is expressed as follows:

$$\frac{N_1 - N_2}{J} = \rho \cdot c \cdot q \cdot \Delta t \quad (1)$$

Here:

N_1 - power loss due to the shearing of oil layers during the rotation of the hub shaft; N_2 - power spent on squeezing oil through the bearing; J - mechanical equivalent of heat; ρ - oil density; c - heat capacity of the oil; q - amount of oil in the bearing; $\Delta t = (T_m - T_{ex})$ - increase in the temperature of the oil layer; T_m - average temperature of the oil layer in the loaded part of the bearing; T_{ex} - oil temperature at the inlet part of the bearing [3, 4].

Power does not have a significant effect on the heat balance of the bearing. Therefore, in calculations it can be neglected, and the heat balance equation takes the following form:

$$\frac{N_1}{J} = \frac{\mu_o \cdot \omega_{np} \cdot l \cdot d^2}{J} = \rho \cdot c \cdot q \cdot \Delta t, \quad (2)$$

Here, μ_o the factors include the dynamic viscosity of the oil, ω_{np} the angular velocity of the shaft, l the working length of the bearing, and d the shaft diameter.

Under operating conditions, the bearing assemblies of housing-type parts must satisfy the following condition to operate stably in the hydrodynamic friction phase [5, 6].

$$Z = \frac{1}{J \cdot \rho \cdot c} \cdot \frac{P^2}{P_n \cdot \Delta t} \leq \frac{\pi \cdot m}{12} \cdot \left(\frac{d}{l} \right)^2 \frac{(1 - \frac{3}{2} \chi^2) \cdot \Phi_m}{\beta} \quad (3)$$

Here, Z is a dimensionless indicator describing the operating mode of the bearing; P_n the oil pressure in the bearing; m is the parabola exponent; and ε is χ the relative eccentricity. Solving equation (4) $P_n \cdot \Delta t$ and substituting the value of Δt , we obtain the following:

$$P_n (T_m - T_{ex}) \geq \frac{P^2}{J \cdot \rho \cdot c \cdot Z} \quad (4)$$

In the critical operating mode of the bearing, the average temperature of the oil layer is determined as follows:

$$T_m \leq T_n \left(\frac{\omega_{np} \cdot \Phi_m}{P \cdot \psi^2} \right)^{\frac{1}{\psi}} \quad (5)$$

Here, T_n is the allowable maximum temperature; ψ is the exponent, which ranges from 0.45 to 0.65 for plastic sliding oils within the temperature range of 50-100 °C.

Under the established thermal regime, the bearings, shaft, and various points of the oil heat up to certain temperatures, and the resulting excess heat is dissipated into the environment [7, 8].

Figure 1 shows a schematic view of the hub housing restored with WEICON-TI metal polymer, the shaft, and the inner ring of the bearing. In this case, the following heat flow scheme is assumed: heat is generated at the contact point between the bearing ring and the shaft, while the temperature of the shaft surface, T_v , is considered constant along both the radial and axial directions.

Thus, the temperature field around the bearing ring is not symmetric. For computational convenience, the heat removal scheme through the ring is converted into a symmetric form, and this process is represented using a special coefficient, K_φ .

$$K_\varphi = \frac{T_\varphi}{T_1} \quad (6)$$

T_φ - the average excess temperature of the bearing ring working surface.

The amount of heat Q generated on the rubbing surfaces per unit time is determined as follows [9, 10]:

$$Q = f \cdot P_a \cdot v \cdot d_1 \cdot l \quad (7)$$

Here: f - the coefficient of friction; P_a - the specific load acting on the ring; v - the sliding velocity; d_1 - the nominal diameter of the ring; l - the width of the ring.

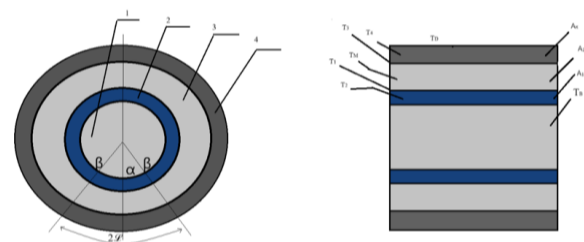


Figure 1. Calculation scheme of the bearing.

In *Figure 1*: $T_1, T_2, T_3, T_4, T_D, T_M, T_\theta$ - the ring, metal-polymer layer, inner and outer surfaces of the layer, $\alpha_1, \alpha_2, \alpha_k$ the environment, and the excess temperatures of the oil and shaft; - are denoted as heat transfer coefficients.

The heat balance is determined using the following formula [11, 12]:

$$Q = Q_\theta + Q_M + Q_\theta \quad (8)$$

Here, Q_θ, Q_M, Q_θ the heat fluxes dissipated per unit time through the housing, oil, and shaft, respectively, are expressed as follows:

$$Q_k = K_k \cdot T_1, \quad Q_M = K_M \cdot T_M, \quad Q_\theta = K_\theta \cdot T_\theta \quad (9)$$

Here, K_k, K_M, K_θ the heat transfer parameters characterize the amount of heat removed per unit time through the housing, oil, and shaft, respectively, when the surface is heated by 1 °C.

Additionally, T_M, T_θ - the excess temperatures of the oil and shaft working surfaces, respectively.

The heat flux dissipated through the housing is determined by the following formula:

$$Q_k = Q_{\alpha x} + Q_m + Q_{n.k.n.} \quad (10)$$

Here, $Q_{\alpha x}, Q_m$ and $Q_{n.k.n.}$ represent the heat fluxes dissipated through the housing, metal-polymer layer, and the outer ring of the bearing, respectively.

Based on the heat balance equation, the overall heat dissipation parameter K_t of the bearing assembly is determined using the following formula [13]:

$$K_m = K_k + K_M + K_\theta \quad (11)$$

The process of heat transfer from the housing to the environment is considered as heat conduction through a multilayer cylindrical wall (see *Figure 2*).

The specific parameters of heat dissipation through the housing, metal-polymer layer, and the outer ring of the bearing are expressed as follows [14]:

$$\left. \begin{aligned} B &= \sqrt{(\alpha_1 + \alpha_2) \cdot \lambda_c \cdot \ell \cdot (d_1 + t_1)^2}; \\ B_1 &= \frac{2\lambda_c}{\ell_n \cdot \frac{d_2}{d_1}}; \\ B_2 &= \frac{2\lambda_n}{\ell_n \cdot \frac{d_3}{d_2}}; \\ B_3 &= \frac{2\lambda_4}{\ell_n \cdot \frac{d_4}{d_3}}; \\ B_4 &= 2\alpha_x(d_4 \cdot t_2 + \frac{d_4^2 - d_3^2}{4}), \end{aligned} \right\} \quad (12)$$

Here, B, B_1, B_2, B_3, B_4 are the specific parameters of heat dissipation to the environment through the bearing inner ring, the metal-polymer layer, α_1 and α_2 the bearing outer ring, respectively; and are the heat transfer coefficients for the bearing inner ring, oil, and metal-polymer layer, respectively; α_x is the heat transfer coefficient of the bearing outer ring with the environment; λ_c, λ_n and λ_4 are the thermal conductivity coefficients of steel, metal-polymer material, and bronze, respectively; d_2, d_3 and d_4 are the diameters of the metal-polymer layer and the bearing ring, respectively; t_1 and t_2 are the thicknesses of the bearing inner ring, bearing outer ring, and housing wall, respectively.

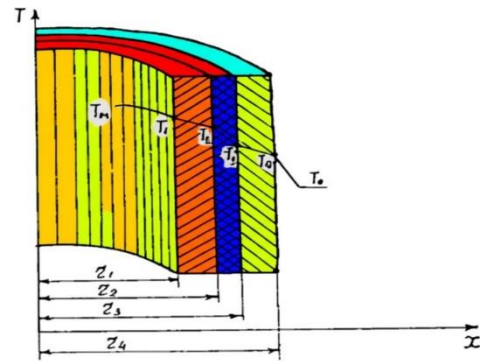


Figure 2. Heat transfer scheme through the main bearing housings.

In *Figure 2*, τ_1, τ_2, τ_3 and τ_4 denote the excess temperatures of the bearing inner ring, metal-polymer layer, bearing outer ring, and the environment, respectively.

The heat flow sequentially passes through the shaft journal (Q_{shaft}), the metal-polymer layer (Q_{layer}), and the bearing inner and outer rings (Q_{bearing}), and is finally dissipated into the environment (Q_{env}). Under the specified thermal regime, the amounts of heat passing through these components per unit time must be equal.

$$Q_{\text{BX}} = Q_{\text{II}} = Q_{\text{II.II}} = Q_0 \quad (13)$$

The heat flux for sequential heat removal is:

$$\left. \begin{aligned} Q &= \pi \sqrt{(\alpha_1 + \alpha_2) \cdot \lambda_c \cdot \ell \cdot (d_1 + t_1)^2} (K_\phi \cdot T_m - T_1); \\ Q &= \frac{\pi}{\frac{1}{2\lambda_c} \ell_n \frac{d_2}{d_1}} (T_1 - T_2); \\ Q &= \frac{\pi}{\frac{1}{2\lambda_n} \ell_n \frac{d_3}{d_2}} (T_2 - T_3); \\ Q &= \frac{\pi}{\frac{1}{2\lambda_4} \ell_n \frac{d_4}{d_3}} (T_3 - T_4); \\ Q &= 2\pi d_x (d_4 \cdot t_2 + \frac{d_4^2 - d_3^2}{4}). \end{aligned} \right\} \quad (14)$$

Here, T_1, T_2, T_3, T_4, T_m and T_o represent the excess temperatures of the bearing inner ring, the metal-polymer layer, the bearing outer ring surfaces, and the environment, respectively.

From the system of equations (15), we determine the partial differences of these temperatures [15]:

$$\left. \begin{aligned} K_\phi \cdot T_m - T_1 &= \frac{Q}{\pi} \frac{1}{\sqrt{(\alpha_1 + \alpha_2) \cdot \lambda_c \cdot \ell \cdot (d_1 + t_1)^2}}; \\ T_1 - T_2 &= \frac{Q}{\pi} \cdot \frac{1}{2\lambda_c} \cdot \ell_n \frac{d_2}{d_1}; \\ T_2 - T_3 &= \frac{Q}{\pi} \cdot \frac{1}{2\lambda_n} \cdot \ell_n \frac{d_3}{d_2}; \\ T_3 - T_4 &= \frac{Q}{\pi} \cdot \frac{1}{2\lambda_4} \cdot \ell_n \frac{d_4}{d_3}; \\ T_4 - T_o &= \frac{Q}{\pi} \cdot \frac{1}{2\alpha_x} \cdot (d_4 \cdot t_2 + \frac{d_4^2 - d_3^2}{4}). \end{aligned} \right\} \quad (15)$$

By adding the left and right sides of the system of equations, we obtain the following:

$$K_\phi \cdot T_m - T_o = \frac{Q}{\pi} \left[\frac{1}{\sqrt{(\alpha_1 + \alpha_2) \cdot \lambda_c \cdot \ell \cdot (d_1 + t_1)^2}} + \frac{1}{\frac{1}{2\lambda_c} \ell_n \frac{d_2}{d_1}} + \frac{1}{\frac{1}{2\lambda_n} \ell_n \frac{d_3}{d_2}} + \frac{1}{\frac{1}{2\lambda_4} \ell_n \frac{d_4}{d_3}} + \frac{1}{\frac{1}{2\alpha_x} (d_4 \cdot t_2 + \frac{d_4^2 - d_3^2}{4})} \right] \quad (16)$$

For computational simplicity, the excess temperatures of the bearing assembly surfaces are expressed in a relative form, i.e., they are referenced to the excess oil temperature:

$$K_o = \frac{T_o}{T_m}; \quad K_1 = \frac{T_1}{T_m}; \quad K_2 = \frac{T_2}{T_m};$$

$$K_3 = \frac{T_3}{T_m}; \quad K_4 = \frac{T_4}{T_m}; \quad (17)$$

By substituting these coefficients into equation (17), taking into account expressions (7) and (13), we obtain the equation for the heat dissipation parameter through the bearing layer when a metal-polymer layer is present.

$$K_n = \frac{K_\phi - K_o}{\frac{1}{B} + \frac{1}{B_1} + \frac{1}{B_2} + \frac{1}{B_3} + \frac{1}{B_4}} \quad (18)$$

The heat dissipation parameter through the bearing layer without the metal-polymer layer is expressed as follows:

$$K_n = \frac{K_\phi - K_o}{\frac{1}{B} + \frac{1}{B_1} + \frac{1}{B_3} + \frac{1}{B_4}} \quad (19)$$

3. Results

The temperature on the working surface of the bearing inner ring is determined as follows:

$$T_1 = K_\phi \cdot T_m - K_n (K_\phi \cdot T_m - T_o) \cdot \frac{1}{B} \quad (20)$$

Thus, the heat dissipation of the oil layer between the bearing inner and outer ring surfaces is determined by the heat dissipation parameters, which depend on factors such as the heat transfer coefficient between the bearing outer ring and the housing, the thermal conductivity of the materials, the cross-sectional area, and the temperature required to maintain the thermal balance of the housing.

4. Conclusions

The experimental results show that the operating surface temperature of a bearing installed in a housing with a metal-polymer coating is only 1.20 °C higher than that of a bearing without the coating under identical operating conditions. This slight temperature increase has a negligible effect on the operational performance and reliability of the bearing housing components, indicating that the application of the metal-polymer coating does not adversely influence the thermal behavior or functionality of the bearing assembly.

Abbreviations

μ_o	The Factors Include the Dynamic Viscosity of the Oil
ω_{pr}	The Angular Velocity of the Shaft

ℓ	The Working Length of the Bearing
d	The Shaft Diameter

Author Contributions

Polvonov Abduljalil Sattorovich: Conceptualization, Data curation, Formal Analysis, Methodology, Resources, Supervision, Visualization, Writing – original draft, Writing – review & editing

Abdusattorov Nodirjon Abduljalil o'g'li: Conceptualization, Data curation, Funding acquisition, Methodology, Software, Supervision, Validation, Writing – original draft, Writing – review & editing

Yunusxanov Doniyorbek Dilmurod o'g'li: Conceptualization, Data curation, Formal Analysis, Investigation, Methodology, Resources, Software, Validation, Writing – original draft, Writing – review & editing

Conflicts of Interest

The authors declare no conflicts of interest.

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Biography



Polvonov Abduljalil Sattorovich is a Candidate of Technical Sciences and Professor at the Department of Transport Engineering at Namangan State Technical University. Since May 1, 2023, he has held the position of Professor in this department. He was born on December 15, 1955, in Kosonsoy District, Namangan Region. He holds a higher education degree, having graduated in 1983 from the Tashkent Institute of Irrigation and Agricultural Mechanization Engineers (full-time program). His academic specialization is agricultural mechanization. He holds the academic degree of Candidate of Technical Sciences and the academic title of Professor.



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Research Field

Polvonov Abduljalil Sattorovich: Agricultural Mechanization, Transport Engineering, Irrigation and Agricultural Machinery, Design and Operation of Technological Machines, Mechanical Systems Reliability, and Efficiency of Machine Components.

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