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EVALUATION AND THERMAL MODELING OF A HIGH-POWER GEARBOX

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Abstract. *This paper presents an evaluation of a high-power gearbox intended for driving belt conveyors in open-pit mining, with a particular focus on thermal modeling and stability under continuous heavy-duty operation. To improve thermal performance and ensure long-term reliability, thermal Finite Element Analysis (FEA) was conducted to optimize the geometry and cooling characteristics of the gearbox housing. Based on the simulation results, a prototype was manufactured and subjected to experimental validation on a test bench, simulating real operating conditions. Key performance parameters, including temperature distribution, structural integrity, and operational efficiency, were monitored and analyzed. The test results demonstrated that the optimized gearbox design successfully meets the thermal and mechanical requirements for demanding industrial applications. The study confirms the effectiveness of using thermal FEA in the early design phase and highlights the importance of integrated simulation and testing in the development of reliable power transmission systems.*

Key words: High-*power gearboxes, Heat calculation, Thermal analysis of gearbox, Testing of gearbox*

1. INTRODUCTION

Gear units are intended for transmission of power and torque from drives to working machines. High power gearboxes are used for belt conveyor drives in open-pit coal mines for the transfer of material from the excavator to the spreader.

They are designed for continuous operation at high speed, transmitting a lot of power, so unavoidable power losses generate significant amounts of heat, which is absorbed by oil and housing. Depending on the working conditions, the gearbox casing is cooled by passive or active convection. When natural airflow is insufficient, then forced ventilation is applied (using a fan). On the basis of the conducted thermal analysis of the gearbox housing, the required fan capacity is determined.

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A deep understanding of the interplay of machine elements in geared transmissions is fundamental for an optimal design. Except for selected space applications [2], oil is present in a gearbox. The interaction of rotating machine elements with fluid causes a fluid flow connected with no-load power loss. According to ISO/TR 14179-2, the gear-based no-load gearbox power loss includes power losses due to churning, squeezing, impulse, and ventilation effects.

In the past decade, research in the field of thermal analysis of high specific power gearboxes has significantly advanced, focusing on heat transfer optimization, loss reduction, and increased reliability under heavy load conditions.

With the rise in gearbox mechanical power, the thermal power limit is increasingly becoming a constraining factor. To achieve a balanced system, additional cooling of the gearbox is required. A common solution involves the use of a fan mounted on the high-speed input shaft.

The optimal design of this solution is the subject of ongoing research. Due to the complexity of the gearbox's thermal balance, analytical calculation methods are too inaccurate for precise evaluation of changes in the cooling system. In studies [1, 7], a numerical approach was applied to determine the optimal cooling concept involving a fan and an air guide.

Power losses in gearboxes result in frictional heating. Sufficient heat transfer from the gearbox to the environment is required for reliable operation. Heat dissipation from gears is linked to their interaction with fluids in the gearbox. Recent research has demonstrated the use of Computational Fluid Dynamics (CFD) to predict the gearbox fluid flow and no-load losses in an isothermal manner. Study [3] focuses on a numerical analysis of heat dissipation in a dry-lubricated gearbox under atmospheric conditions.

Testing on a proof stand is the final phase of the manufacturing process for any gearbox. This especially applies to high-power gear units, which are of special construction and have significant heat losses, thus a prototype is required to be tested first in factory conditions [4, 5].

The paper presents the development of a high-power gearbox intended for driving belt conveyors in open-pit mines, with a particular emphasis on thermal stability. In order to ensure reliable operation, thermal FEA (Finite Element Analysis) of the gearbox housing was carried out to optimize its shape. The design validation was performed by testing the gearbox on a test bench.

2. HIGH-POWER GEARBOXES

Powerful bevel-helical gearboxes with a flanged output shaft are used for high-capacity belt conveyor drives. The housing is considerably enlarged and completely ribbed. The fan is on the input shaft with a deflection hood that follows the contour of the housing.

Depending on the transmission ratio (usually in the range i=14...18), two or three transmission stages are applied. Three-stage gearboxes are used more often, because they have a larger surface of the housing (and better thermal capacity), increasing the possibility of combining and avoiding high partial transmission ratios, as well as offering a smaller size of the spur gear to the output stage.

To transfer the nominal power of 1000 kW (power EM), the GOŠA FOM company [9] has developed two basic types of three-stage gearbox, with the application factor

K_A=1.8 and 2, and the transmission ratio i=17.267 and i=17.198. The gear units are specially designed and manufactured according to specific customer requirements (see Table 1). Its long-standing experience was used in producing gear units for belts on different machines for surface mining, such as excavators, conveyors, and spreaders.

Table 1 Three-stage gearboxes with rated power 1000 kW

Type	Ratio i	Application factor K_A	Housing surface A_H [m^2]	Gearbox surface A_G [m^2]
1	17.267	1.8	30.4	34.7
2	17.198	2.0	38.5	44.6

For the dimensioning gear unit with high load capacity, it is necessary to make a calculation of toothed parts and bearings according to the customer requirements (application factor $K_{A\min}$= 1.8 at safety factors of toothing for contact and bending stress S_H=1.2 and S_F=1.4, and operating bearing lifetime min. 50,000h). The nominal working input speed of the gearbox is 1000 (range 600-1200) rpm.

Based on gearing and bearing calculations and thermal stability of the housing, workshop documentation is produced to manufacture the toothed parts and housing. The spiral bevel gear pair is made in HPG-quality by the Klingelnberg Cyclo-Paloid system. The gearing of helical gears is ground, with tooth profile correction and flank modification (so-called "barreling"). Premium materials are used (case-hardened steels), and tooth flanks to all gearings are finished.

The overview of the typical gearbox dimensions is given in Fig. 1.

During its operation, the gearbox continuously ("24/7") transmits high power, which generates certain power losses in all bearings and in each engaged gearing pair, in turn converting those losses into heat, so that the oil and the inside of the gearbox are heated, and the outer surface of the housing transfers that heat to the environment. These losses in a three-stage bevel-helical gear unit amount to about 7% of the rated power (about 1% in the spiral bevel gear pair and around 1.5% for the helical gears set, for a total of around 4%; losses in the bearings are about 0.1% per row, for a total of around 1.5%). In the case of high-power gearboxes, such losses of a few percent are converted into a considerable amount of heat (precisely about 70 kW), which has to be transferred to the environment over the outer surface of the housing.

Natural cooling of gear units is improved by increasing the outer surface of the housing with ribbing and elongation in the free area behind the output shaft. Fig. 2 shows the enlarged and ribbed gearbox housing with the main measures for K_A=1.8.

In such a housing the temperature in the interior of the gearbox during exploitation in the summer reaches about 120 °C.

When natural convection is not sufficient, forced cooling is applied (a fan on the input shaft), which must increase the speed of air flow around the gearbox, increasing the speed of air flow around the housing, and causing faster heat transfer from the inner to the outer housing surface, thus contributing to cooling the interior of the gear unit.

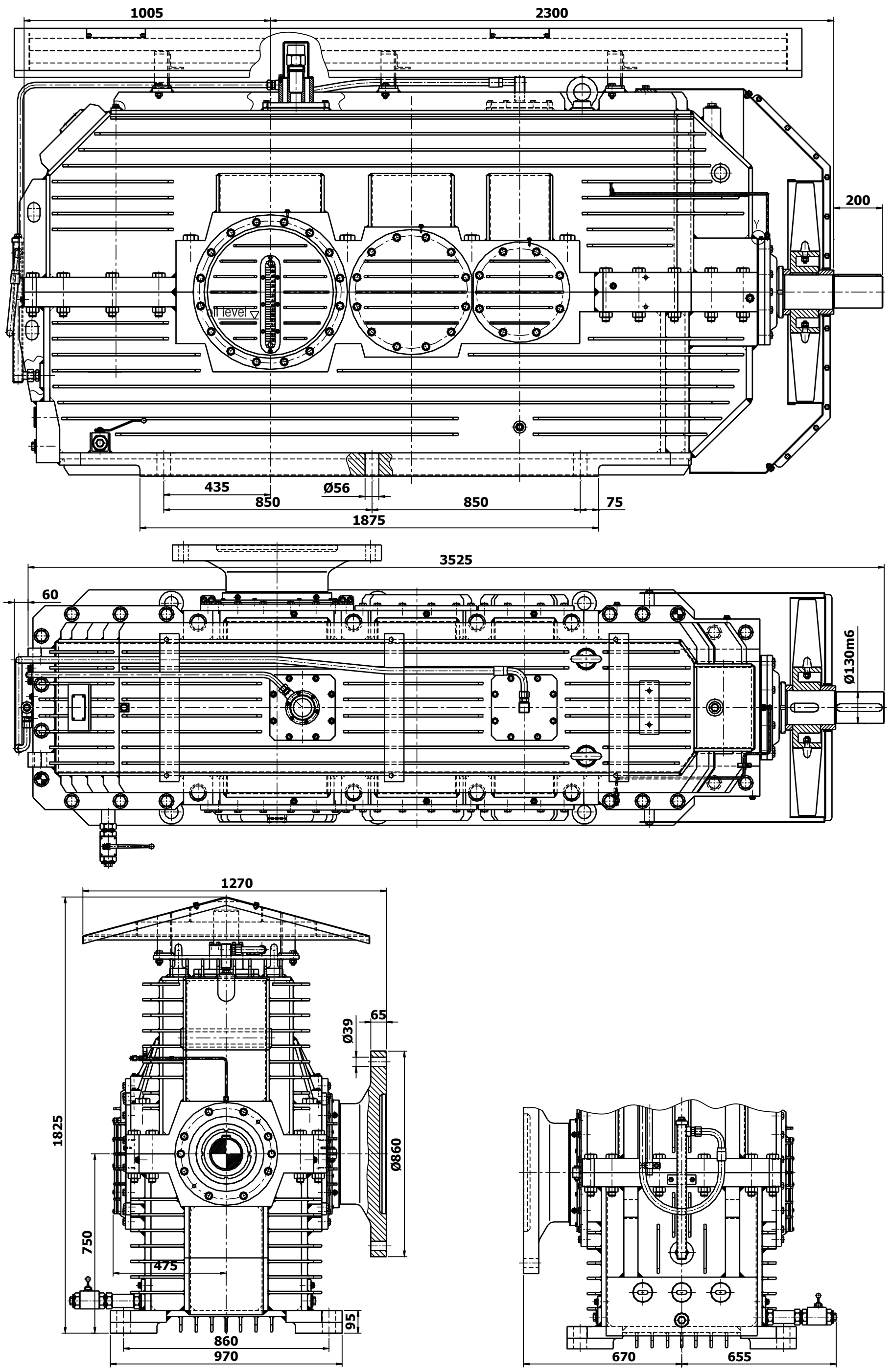

Fig. 1 Overview of the typical gearbox dimensions

The enlarged housing of the three-stage gearbox (using the fan) has a sufficient surface for the thermal stability of the gear unit below 80 (max. 90) ºC. For a gearbox in operation with mineral oil to have a working temperature below 85 °C, in all weather conditions, one should calculate the thermal behavior of the housing, i.e. conduct an analysis of the different airflow speeds around the casing.

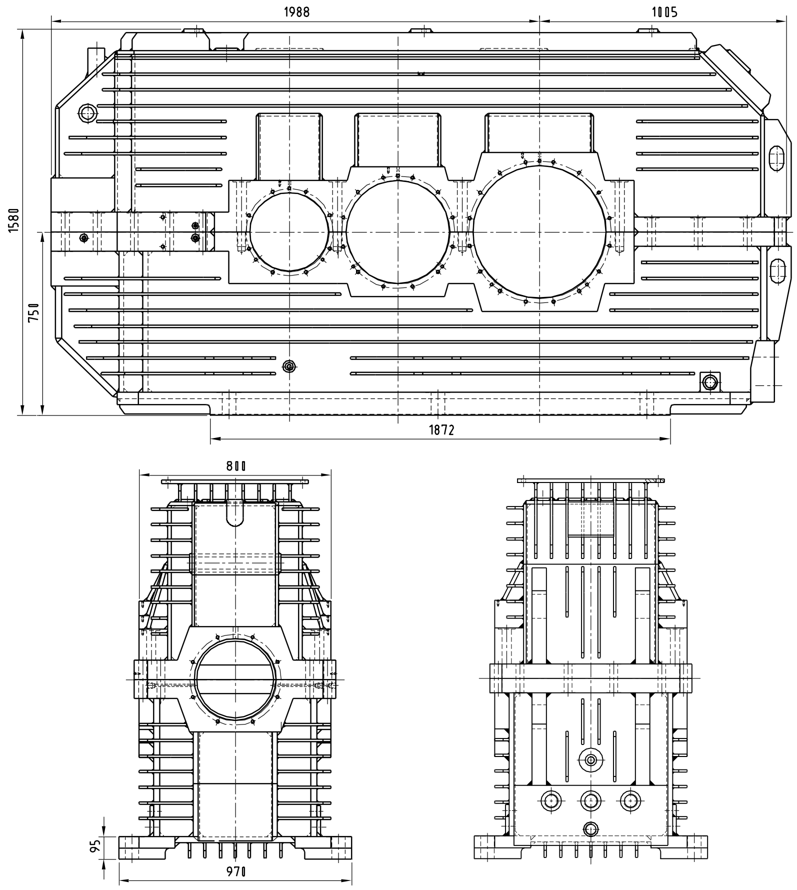

Fig. 2 Enlarged and ribbed gearbox housing

3. Heat Calculation and Thermal Analysis

Due to the power losses that are converted into heat, it is very important to implement thermal analysis of the gearbox or the housing in the design stage.

For thermal analysis of the gearbox and thermal stability of the housing basic calculation is used in line with the literature. Also, thermal analysis of the casing is performed using the finite element method (FEM). The basic heat calculation is used to obtain basic data on the required surface for housing sizing and thermal analysis is used to check the temperature condition of the outer and inner surfaces of the housing and the choice of fan capacity.

3.1 Heat calculation

The following basic equations are used for heat calculation [8]:

$$P_v = P_N \cdot (1-\eta) \cdot \frac{ED}{100} \quad (1)$$

$$\Delta t = \frac{P_v \cdot 10^3}{\alpha_k \cdot A_G} \quad (2)$$

$$t = t_u + \Delta t \quad (3)$$

where:
- Rating power: $P_N = 1000$ kW
- Max. periodic rating: $ED = 100\%$
- Total gearbox losses: $P_G = P_{Gt} + P_{Gb} + P_{Gs} + P_{Go}$
- Losses in the toothings: $P_{Gt} = P_{G\text{-}Ist} + P_{G\text{-}IIst} + P_{G\text{-}IIIst} = (1\% + 1.5\% + 1.5\%) \cdot P_N$
  $P_{Gt} = 4\% \cdot P_N$
- Losses in the bearings: $P_{Gb} = P_{GbShI} + P_{GbShII} + P_{GbShIII} + P_{GbShIV}$
  $P_{Gb} = [(0.3\% + 0.2\%) + 2 \times 0.2\% + 2 \times 0.1\% + 2 \times 0.2\%] \cdot P_N$
  $P_{Gb} = 1.5\% \cdot P_N$
- Losses in the seals: $P_{Gs} = P_{GSInput} + P_{GSOutput} = (0.3\% + 0.2\%) \cdot P_N = 0.5\% \cdot P_N$
- Other losses (including oil viscosity impact $P_{Go} = (0.5\%...1\%) \cdot P_N$= max. $1\% \cdot P_N$
- Losses in the gearbox: $P_G = P_{Gt} + P_{Gb} + P_{Gs} + P_{Go} = (4\% + 1.5\% + 0.5\% + 1\%) \cdot P_N$
  P_G = max. $7\% \cdot P_N = 0.07 \cdot P_N$
- Efficiency of the gearbox:
  $\eta = P_{izl}/P_{ul} = (P_{ul} - P_G)/P_{ul} = (1000 - 0.07 \cdot 1000)/1000 = 0.93$
- Loss of rating: $P_v = 70$ kW
- Heat transfer coeff. (without / with fan): $\alpha_k = 22 / 35$ W/(m^2·K)
- Gearbox surface: $A_G = 34.7$ m^2 (acc. to Fig. 1)
- Housing surface: $A_H = 30.4$ m^2 (acc. to Fig. 2)
- Ambient temperature: t_u = max. 40°C
- Temperature difference (without / with fan): $\Delta t = 91.7 / 57.6$ K
- Theoretic max. inside temperature: $t = 132 / 98$ °C
- Daily conveyor system capacity amount: $\varphi = 0.6 ... 0.75$
- Real max. inside temperature: $t^* = \varphi^{1/3} \cdot t$
  $t^* = 0.6^{1/3} \cdot 132 = 111$ °C / $0.75^{1/3} \times 98 = 89$ °C

Using equations (1) to (3) with a modification of parameters t, α_k and A_G, the below typical cases are obtained, which contain the following results for the required developed outer housing surface A_G, as well as the temperature inside the housing t:

1. Done: t=90 ºC, α_k=35 W/m^2/K Result: A_G=40 m^2
2. Done: A_G= 44.6 m^2, α_k=22 W/m^2/K Result: t=111.3 ºC
3. Done: A_G= 44.6 m^2, α_k=35 W/m^2/K Result: t=84.8 ºC

According to the literature [8], the convection factor without a fan is 20-25 W/(m^2·K), and with a fan is 30-40 W/(m^2·K) (for the calculation the mean empirical value of 22 and 35 W/(m^2·K) is taken). The temperature inside the gearbox is highly dependent on the ambient temperature. Said calculation takes the maximum ambient temperature in the summer to be 40 °C.

For the lubrication of the gearbox high-quality industrial gear oil is used, ISO viscosity grade 320 acc. to DIN 51517-3, e.g. 6 EP acc. to AGMA grade – CLP gear oils (kinematic viscosity ν_{40}=320 mm²/s at 40 °C, with a tolerance of ±10%). Mineral gearbox oil contains additives (corrosion protection, oxidation and foam resistance, extreme pressure (EP), anti wear (AW), VI improver).

The use of synthetic oil leads to the values lower by about 10-15 °C. However, these oils are very expensive (5-8 times the price of mineral oils), and are applied in up to 20-25% of cases in practice.

The external surface of the extended housing is increased by 35% in relation to the basic surface, by elongation in the free space behind the output shaft, as well as in the fins on all sides of the housing. Ribs must be set in the direction of airflow, thus in the longitudinal direction and horizontally. It should be noted that for the smaller gear units, where the fan is not in use, the ribs are placed vertically and transversely.

To reduce the temperature inside the casing, the final installation on site is performed above the gear unit set roof. The gearbox can be painted with a special paint that repels sunlight. These additional passive protection measures can reduce the temperature inside the gearbox housing by 5-8 °C.

3.2 Thermal analysis of the housing and the gearbox

The distribution of temperature in the bevel and helical gear transmission using FEM was done using the ANSYS Workbench software. The analysis was defined as a thermal analysis in the time domain that was used in the ANSYS module for transient thermal analysis. For generating the finite element mesh, the elements of a higher order or SOLID 226 were used. The mesh was generated with 236298 nodes which formed 45824 elements.

Figs. 3 and 4 show the 3D model of the gearbox and the simulation model of the gearbox housing with a generated finite element mesh.

4. Testing on a Proof-Stand

Testing on a proof-stand is the final part of the production process for each gear unit. Testing of high-power gearboxes is implemented in several stages. After they are manufactured, gear units are tested on the stand without and under partial load (around 200 kW, which is about 20% of the installed motor power). The load is increased gradually, usually in 2-3 steps. For single testing (with or without a fan), the input shaft is alternately turned right (clockwise) and left (counterclockwise). Paired gearboxes under load are tested only in one direction of rotation of the input shaft, so that each of the gear units has proper fan cooling.

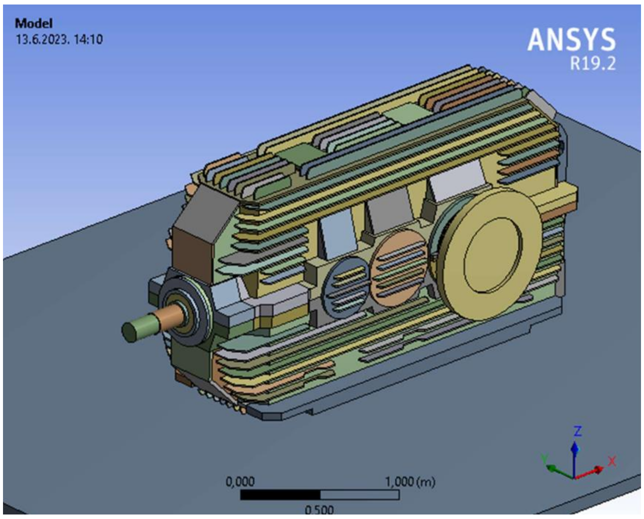

Fig. 3 3D-simulation model of the gearbox

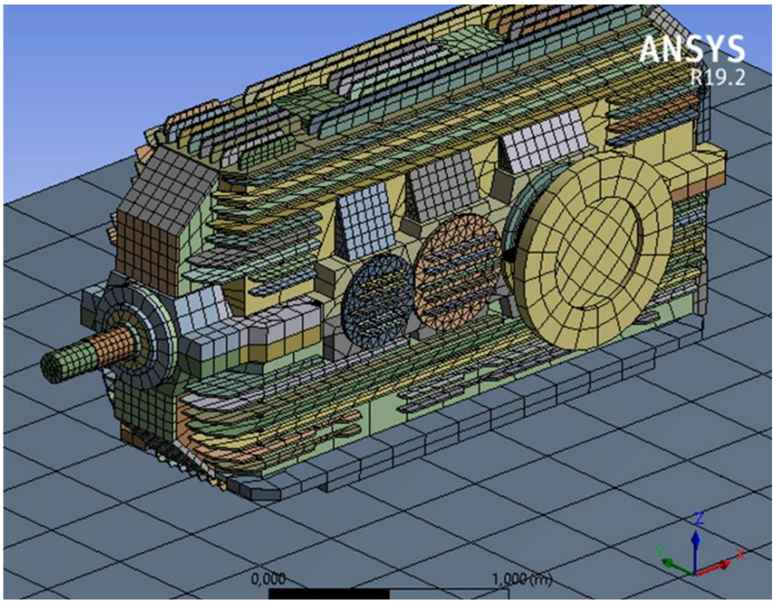

Fig. 4 Generated finite element mesh of the gearbox

Based on the analytically determined power losses in the gear pairs, bearings, and seals (Fig. 5), a simulation was performed to determine the temperature distribution of the gearbox housing (Fig. 6).

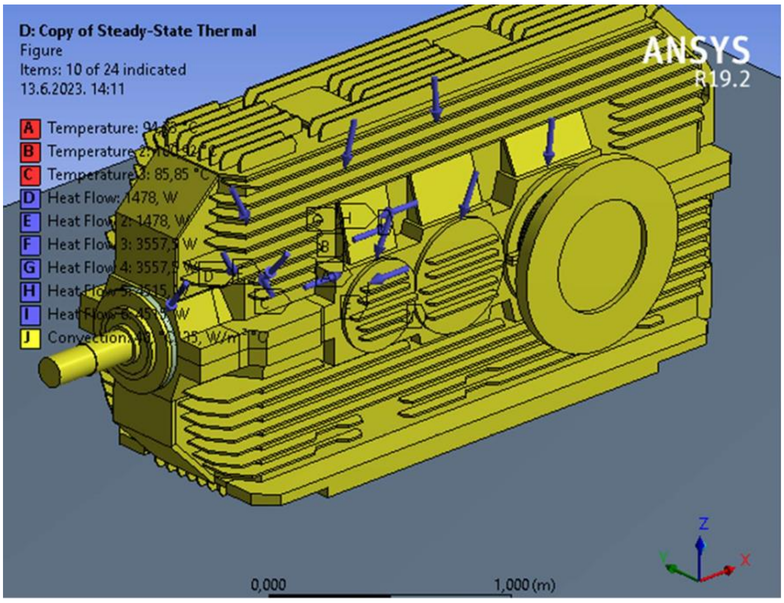

Fig. 5 Thermal load of the gearbox

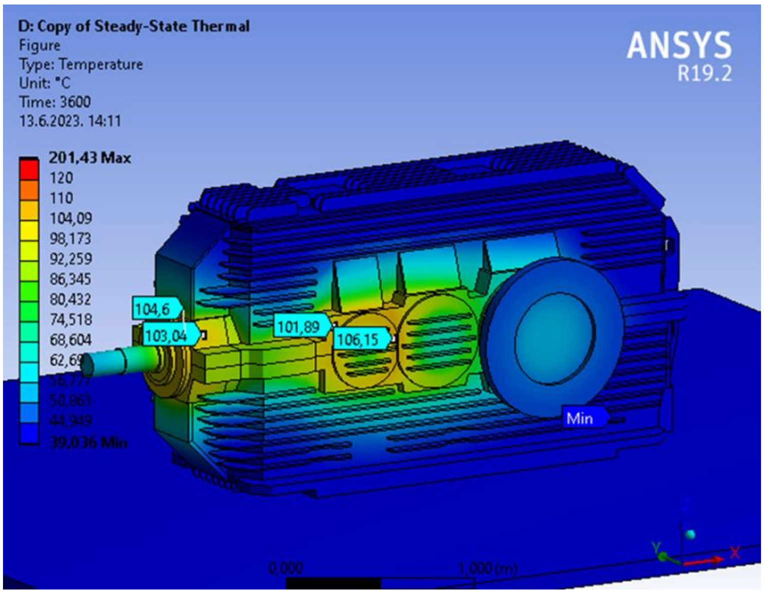

Fig. 6 Thermal analysis of the gearbox housing

Fig. 7 Gearboxes with a fan on the proof-stand

Only one reducer can be tested without load, first without and then with a fan, and finally with the assembled deflection hood. On the other hand, two gear units can be tested under partial load, connected between the flanged shafts through the cardan shaft. This is the so-called "back-to-back" principle of load. The proof stand has the capacity of max. 300 kW and the rotation speed is max. 1800 rpm. For practical purposes, the load up to 250 kW is applied, with a speed range of 600-1200 rpm. In this particular case, the testing parameters were: 200-220 kW load and 1000 rpm speed (short-time max. 1200 rpm).

The testing layout of the paired gearbox under load on the proof stand is shown in Fig. 8.

The testing procedure of the gear units on the proof-stand is presented in Tables 2 to 4.

Table 2 Full testing procedure of the gearbox on the proof-stand

Case	Gearbox No.	Fan	Load	Time (min)	Input shaft direction
1.	1	-	-	min. 120	CW / CCW
2.	1	+	-	min. 120	CW
3.	2	-	-	min. 120	CW / CCW
4.	2	+	-	min. 120	CCW
5.	1 + 2	+	+	min. 240	CW + CCW

Each gearbox test must last until reaching a stable (holding) temperature of max. 2K for 30 minutes for each measuring point (MM).

Table 3 Short testing procedure of the gearbox on the proof-stand

Case	Gearbox No.	Fan	Load	Time (min)	Input shaft direction
1.	1	-	-	min. 180	CW / CCW
2.	2	-	-	min. 180	CW / CCW
3.	1 + 2	+	+	min. 240	CW + CCW

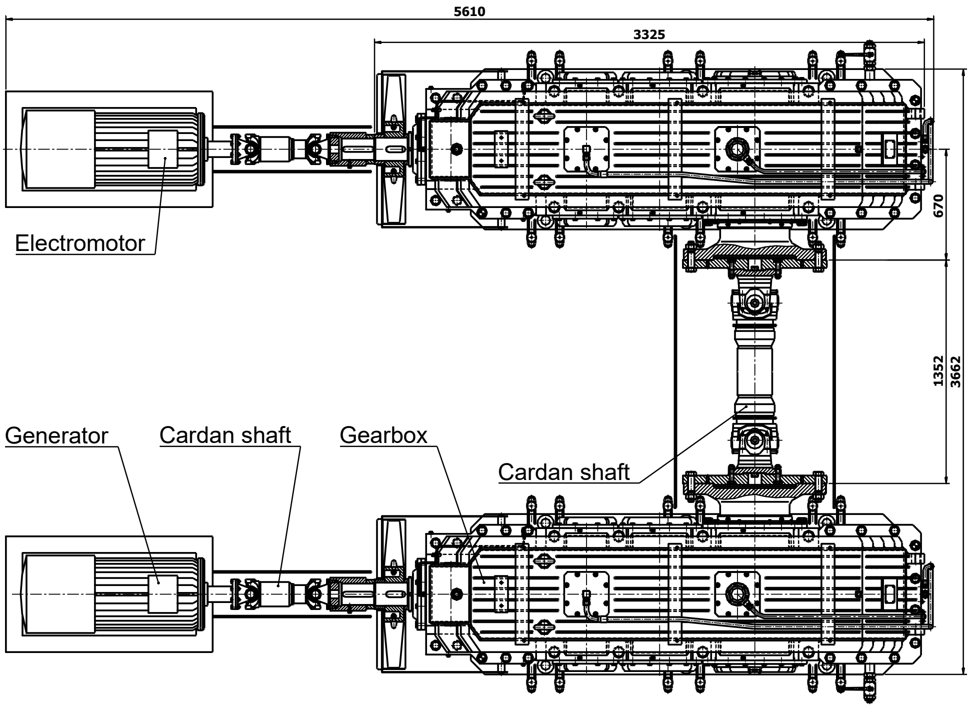

Fig. 8 Testing layout of the paired gearbox under load

Table 4 Testing regimes of the meshed gearboxes under load

Step	1.	2.	3.	4.
Load	-	5%	10%	20%
Time (min)	30 +30	30	30	120

5. Parameters Measuring Gearboxes

5.1 Equipment for monitoring gearboxes

The "monitoring" equipment of gearboxes is shown in Fig. 9.

The high-power gear unit is equipped with the following ("monitoring") equipment for measuring, condition monitoring and control of temperature, vibration and oil:

- resistance thermometers for measuring temperature of input shaft bearings,
- resistance thermometers for measuring temperature of oil sump,
- sensors for measuring vibration of the housing in the zone of the input shaft, with accessories,

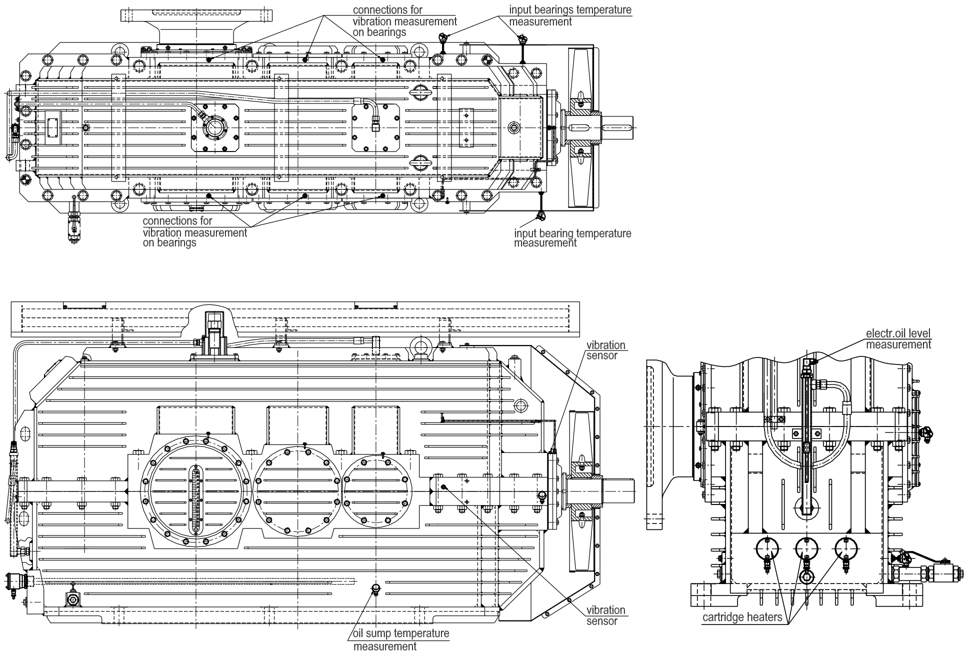

Fig. 9 Gearbox "monitoring" equipment

- connections for vibration casing measurements above others bearings, can be measured and shown by mobile device / equipment on site,
- oil level indicators - visually (numbered window) and an electronic device in a protective tube,
- oil heaters, for winter conditions.

During the testing, the temperature of all bearings and the vibrations of the housing near all bearings are measured on each gear unit. Temperatures of the bearings on the input shaft and oil sump are measured with resistance thermometers. Vibrations are measured on the housing near each bearing in 3 directions (vertical, horizontal and axial) with a mobile device.

An example of results for temperature measurement of the input shaft bearings (MM1 to MM3) and oil temperature (MM4) is shown in Fig. 10.

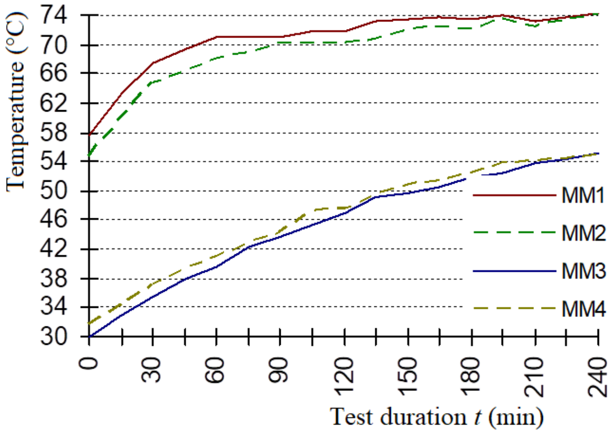

Fig. 10 Temperatures of bearings and oil (example)

Measuring point MM1 is the bearing close to the fan, MM2 is the inner bearing of the paired gearboxes, MM3 is the bearing near the gearing on the input shaft, and MM4 is the oil sump. The ambient temperature (in the hall) was 27 ºC.

The measuring of vibration velocity of the gearbox is shown in Fig. 11.

Fig. 11 Measuring vibration velocity (example)

The scheme of measuring points for vibration velocity of the gearbox housing is given in Fig. 12.

Examples of the results of measuring vibration velocity of the gearbox housing are given in Tables 5 and 6.

Table 5 Vibrations of gearbox 1 at measuring points [mm/s]

Measur. point	Ia	Ib	IIa	IIb	IIIa	IIIb	IVa	IVb
1. Vert.	0.97	1.23	0.66	0.69	0.8	0.72	0.57	1.04
2. Horiz.	-	0.79	0.71	0.76	0.9	0.66	0.71	0.52
3. Axial	-	-	1.29	1.87	1.72	1.85	1.88	1.64

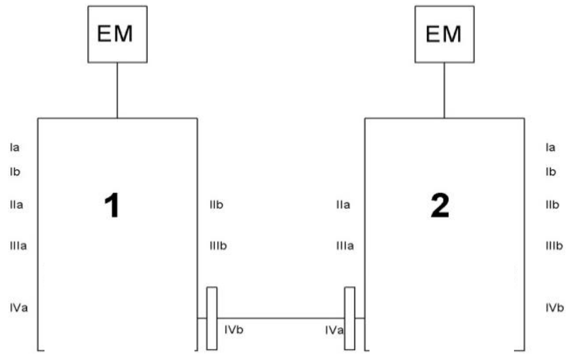

Fig. 12 Vibration measuring points on the gearboxes.

Table 6 Vibrations of gearbox 2 at measuring points [mm/s]

Measur. point	Ia	Ib	IIa	IIb	IIIa	IIIb	IVa	IVb
1. Vert.	1.24	1.26	0.61	0.58	0.72	0.61	1.02	0.98
2. Horiz.	-	0.7	0.63	0.59	0.85	0.72	0.83	0.67
3. Axial	-	-	1.34	1.74	1.69	1.75	1.73	1.89

The measured values are within the limits of zone B (terms "satisfactory", "acceptable" or "permissible") for machine group G (class IV), acc. VDI 2056 and DIN ISO 10816. The limit value is 3.5 mm/s for the criterion "rigid (strong)", i.e. 4.5 mm/s, for the criterion "flexible (soft)".

The gearbox noise was also measured and it did not exceed the value defined in the customer requirements. If not specified, the following value is taken as the authoritative maximum value for noise sound power level, acc. VDI 2159:

$$L_A = 0.8\cdot(71.7+15.9\cdot\log(P)) \tag{3}$$

P – rated gearbox power in kW
L_A – sound power level in dB.

6. CONCLUSIONS

Modern approaches to thermal analysis of gearboxes are based on the integration of numerical simulations, design optimization, and advanced modeling techniques. The aim of these methods is to achieve a higher level of cooling efficiency, reduce the risk of thermal overload, and increase the overall system reliability. Moreover, this approach allows for constructive improvements of gearbox components already at the design stage, significantly reducing the need for subsequent adjustments during operation.

This paper presents the development of a high-power gearbox intended for belt conveyors used in open-pit mining, with a particular emphasis on ensuring thermal stability under continuous operation conditions. Special attention was given to the design of the gearbox housing, as it plays a key role in heat transfer and dissipation. Finite Element Analysis (FEA) was employed to analyze temperature distribution and mechanical stresses, as well as to optimize the shape and dimensions of the housing to achieve an optimal balance between strength, mass, and heat dissipation capability.

The developed gearbox was then subjected to experimental verification on a test stand, confirming the accuracy of the numerical predictions and validating the overall approach. Based on the results obtained, it can be concluded that the application of such a methodology in the design of high specific power gear transmissions leads to a significant enhancement of performance. The most important effects are reflected in improved cooling efficiency, optimized gearbox housing design, and increased reliability under operational conditions.

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