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Mathematical modeling of energy loss reduction in gear boxes of oil and gas technological transport

I.M. Mykytiy

Ivano-Frankivsk National Technical University of Oil and Gas, Ukraine
E-mail: auto.ifntung@ukr.net

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The work is aimed at solving the problem of reducing energy losses in transmission units of hoisting machines for repairing wells. Method of rapid heating and maintaining the optimal temperature in the transmission units of lifting units by using the heat of the exhaust gases was proposed. Analysis of features of a design of transmissions of lifting installations for repair of wells is carried out. Studies of viscosity-temperature characteristics of modern transmission oils and temperature regime in transmission units have been performed. A mathematical model of energy release in transmission units during the operation of lifting units is proposed. Installed energy consumption for friction in the gears of the transmission units. Friction energy losses in bearings of transmission mechanisms of lifting units are determined. A method for reducing energy losses in transmission units of hoisting installations for well repair is proposed. Experimental studies of the implementation of the proposed method of reducing energy losses in transmission units. Dependence of power losses in the gearbox of lifting units depending on the temperature and grade of transmission oil is established. The dependence of power losses in the gearbox of the lifting unit of the UPA 60 / 80 A model depending on the temperature and grade of transmission oil is obtained. The results of calculations of fuel consumption in the gearbox of the lifting unit of the UPA 60 / 80A model with different power drives and at different temperatures of transmission oil are given.

Key words: oil and gas technological transport; hoisting installation for repair, gearbox; transmission unit; exhaust gases; heat utilization; power.

Introduction

The oil and gas complex of Ukraine and the world together with other structures includes numerous production units of oil and gas technological transport. Well-lifting installations for repairing wells have been widely used in the gas and oil industries. Mobile installations for current and overhaul of wells are widely used for repair of wells and downhill equipment and perform lifting and lowering of compressor and drill pipes, pump rods, pumps, ropes and current-carrying cable for electric pumps, gas lifts, etc.

The purpose of technological transport of oil and gas industry is to ensure the continuous operation of the main production by performing technological operations and transport work in the specified volumes, at the specified time and with minimal energy costs and costs. Therefore, the problem of reducing energy consumption for lifting plants, in particular reducing energy consumption in their transmissions, is an urgent task.

Analysis of modern foreign and domestic research and publications

Problem of improving energy efficiency is one of the main priorities of our country. And recently, under the influence of external factors, there are radical changes in approaches to the formation of state energy policy [1]: there is a transition from outdated models of the domestic energy sector, dominated by fossil fuels, to new models that maximize the use of alternative non-fossil fuels and the dominance of one type of energy production is minimized. At the same time, maximum priority is given to the use of energy from renewable sources and increase energy efficiency [2].

In order to achieve the goals set by the state energy policy, it is necessary to minimize the energy consumption of all facilities and equipment operated in the oil and gas industry of Ukraine, and in particular this fully applies to oil and gas technology transport as a whole and such an important its component as a lifting installation for repairing wells [3]. These units have energy-intensive, compared to the electric drive, diesel engines with transmissions, which requires, inter alia, the search for new ways to improve the energy performance of power drives of hoists.

For heavy-duty well repair units, standard ultra-heavy-duty automobile chassis or ultra-heavy-duty wheeled or tracked transport bases assembled from standard components - axles, wheels, transmissions, gearboxes, engines, but combined with a special frame - are used. Such wheelbases, depending on the required load capacity, have four or six axles, of which two or four are leading. The world has mastered the production of a large number of units for the repair of wells, with a capacity of 16 tons to 350 tons on wheels and tracks [4]. The most common in terms of frequency of application is the unit of current and overhaul of wells for work with open mouths. Lowering and lifting operations under pressure are performed much less often, so the units of current and overhaul of wells for this purpose are much smaller. In addition, due to the complexity of pressure operations, sometimes wells are blocked and repaired as usual. When using the same units that allow you to perform lowering and lifting operations under pressure, a complex process of silencing can be avoided [5].

Depending on the operations performed by mobile installations, their complete set can change. So, only for raising and lowering of pipes and rods the lifting installation is composed of the minimum number of knots. To use the hoisting rig for drilling, power is selected for the rotor, drilling pump and flushing system, which allows the use of the hoist for overhaul of wells. For lowering and raising electric pumps, the lift is completed with a drum for winding the cable, and for the suspension of the rods, the tower is equipped with a special gripper [6]. Hydraulic couplings and sometimes torque converters installed between the engine and the transmission are used to improve the performance and control efficiency of mobile well repair installations, as well as for more efficient pairing of drive motors in foreign-made hoists, especially large-capacity ones. In this case the transmission is called hydromechanical [7].

Hydraulic couplings or torque converters must be used for paired drive motors in foreign-made hoists of high capacity when operating on a single transmission. In this case, the total power transmitted to the transmission increases due to the self-regulation of each engine speed, which is impossible with their rigid mechanical connection. The torque converter allows you to smoothly change the speed while changing the torque. But with increasing transformation factor, the efficiency of this hydraulic machine decreases and decreases up to 0.60-0.65, which is a serious disadvantage of this method of regulation. To provide an even greater range of regulation, the drive with a torque converter of mobile well repair units can also be supplemented with a manual transmission [8].

Using of hoists for repairing wells in low temperatures entails difficulties in thermal preparation not only of the engine, pumping equipment, compressor equipment, hoisting system, but also transmission units, due to the high initial viscosity of the transmission oil in the cold period of a year. Many scientists have devoted their work to the problems of efficient operation of the transmission. Both mechanical transmissions based on automobile chassis [9] and mechanical transmissions based on tractor units [10], as well as automatic transmissions [11] were considered.

In a number of works [12, etc.], the use of forced oil supply to friction surfaces has been proposed as a way to reduce power losses in transmission units. Forced oil supply reduces losses depending on the number and depth of immersion of gears in oil, however, as shown by these studies, the greatest efficiency of oil application for rotating parts under pressure is observed from the moment of shift to stabilization of oil temperature, in addition, this method requires preheating olives. Studies by a number of authors [13, etc.] show that the temperature regime of car transmission units is one of the main factors influencing both power losses and the intensity of wear of transmission mechanisms. It is established that ensuring the optimal thermal regime of the transmission units will reduce additional fuel consumption by up to 10% and wear intensity up to 8 times. Studies [14] show that in a month with a low ambient temperature, there is an increase in failures of the ZF gearbox due to insufficient oil supply to the friction points.

For example, the number of failures of the front bearing of the secondary shaft increases by 33 %, the rear support bearing - by 20 %, and the number of failures of other parts of the gearbox increases by 12 %. Thus, the number of failures of the gearbox of this brand in general case increases by 65 %.

Based on the above data we can conclude that modern transmission oils with different viscosity-temperature characteristics do not provide torque transmission without power loss during heating of the transmission, and the optimal temperature of the transmission units is 303-313 K.

Coverage of previously unresolved parts of the overall problem

In general, the main energy losses during the operation of power drives for lifting wells are:

- internal losses in the transmission, determined by the efficiency of the mechanical or hydraulic transmission;
- internal losses in the hoist system, determined by the efficiency of the hoist system;
- internal losses in the drive engine, determined by the efficiency of the diesel internal combustion engine;
- losses due to suboptimal drilling process;

- losses due to suboptimal operation of drilling pumps;
- hydraulic losses in pipelines and fittings;
- internal losses in the winch, determined by the efficiency of the winch;
- internal losses in the rotor, determined by the efficiency of the rotor;
- losses due to the use of suboptimal fuels for the drive engine;
- losses associated with the lack of recovery of excess heat.

Transmission of the well repair unit consists of a set of couplings, shafts, chain gears, a winch and a hoist connected to the hook block. The number of speeds and their ratio are determined depending on the technology of lowering and lifting operations. In a car or tractor transport base, the standard gearbox of the transport base itself is often used as a torque converter from the engine. This scheme of lifting installation, built on mechanical transmissions, is currently the most common in Ukraine. However, comprehensive studies on reducing energy consumption in the transmission of hoists for repairing wells by using the heat of the exhaust gases to ensure rapid heating of the transmission units and maintain the optimal thermal regime has not yet been conducted.

Formulation of aims of the article

The costs of operation of technological transport account for a significant share in the cost of oil and gas products, so reducing energy consumption and cost of technological work during the operation of power drives for lifting wells - an urgent problem for oil and gas industry.

Therefore, the aim of this article is theoretical and experimental studies of ways to improve the energy efficiency of transmission units of hoisting installations for the repair of wells by using the heat of exhaust gases.

Mathematical model of energy release in transmission units during operation of lifting units

Analysis of kinematic diagrams of transmission units of hoists for repairing wells shows that they mainly consist of cylindrical and bevel gears, planetary gears, different types of bearings, seals and other elements in which there is internal mechanical and hydraulic friction and friction. rotating gears with oil bath.

Mechanical gearboxes in most cases gear less often chain with step speed control are mainly used as transmissions and power converters for hoists for repairing wells made in Ukraine, the former CIS and foreign production of small capacity.

A mathematical model has been developed to determine the energy $Q_{mp.a.}$ generated in transmission units during the operation of lifting units.

Calculation of the energy generated in the transmission units of lifting units is based on the determination of the efficiency of internal friction sources and taking into account the power transmitted through the transmission units. Based on the efficiency of the units and mechanisms that make up the unit, you can determine the amount of energy generated in the unit, according to the formula:

$$Q_{mp.a.} = \sum_i N_i \cdot t_i \cdot \left(1 - \eta_{b.mp.i}\right), \text{ J}, \tag{1}$$

where N_i – power transmitted by the i-th node of the transmission unit, Wt;

t_i – duration of operation of the i-th unit of the transmission unit, s;

$\eta_{b.mp.i}$ – efficiency of the i-th unit of the transmission unit.

Thus, the calculation of the internal energy release in the transmission is based on the functional relationship between the efficiency of the nodes of the transmission units and the power transmitted through the transmission nodes. Taking into account the fact that any transmission unit consists of elementary components and mechanisms, each of which has its own efficiency, the assessment of energy is based on the specific design features of the considered units.

The design of transmission units of hoisting installations for well repair is a set of cylindrical, bevel gears and chain gears of internal and external gearing. For example, the transmission of UPA-60/80 hoists on the KrAZ-63221 chassis (mechanical, two-band, eight-speed) consists of cylindrical spur gears (1st gear and reverse) and cylindrical helical gears (2 - 8 gears). Primary, secondary and intermediate shafts are installed in the crankcase sockets on ball and roller bearings.

Taking into account the design features, the total efficiency $\eta_{в.mp.кп}$ of the considered gearboxes is determined by the following expressions:

$$\eta_{в.mp.кп} = \eta_{zn}^{x} \cdot \eta_{gm}^{y} \cdot \eta_{mn}^{z}, \tag{2}$$

where η_{zn} – efficiency taking into account energy losses due to friction in cylindrical gears;

η_{gm} – efficiency taking into account energy losses due to hydraulic friction in the unit;

η_{mn} – efficiency taking into account energy losses due to friction in bearings;

x, y, z – the number of gears, gears in contact with the oil in the transmission unit, and bearings, respectively.

The front, middle and rear axles of UPA-60/80 hoists are a combination of one pair of bevel gears with helical teeth and one pair of cylindrical helical gears. The leading bevel gear of the front, rear and middle axles is mounted on two roller tapered roller bearings. The intermediate shaft of the axles is mounted on three tapered roller bearings.

Taking into account the design features, the total efficiency of the considered main gears in the bridges is determined by the following expressions:

$$\eta_{b.mp.gn} = \eta_{zn}^{x} \cdot \eta_{gm}^{y} \cdot \eta_{mn}^{z} \cdot \eta_{\kappa n}^{\chi}, \tag{3}$$

where $\eta_{\kappa n}$ – efficiency, which takes into account the losses in the gearing of bevel gears;

χ – the number of bevel gears in gear.

Friction energy losses in gear units of transmission unit

The value of efficiency of transmission units is determined by the viscosity of the oil used for lubrication, the gear ratios, the coefficients of friction of the gear teeth, the number of gear teeth and other factors. Typical values of the efficiency of gears at the optimum temperature are presented in table 1. It should be noted that at the optimum temperature the efficiency of gears reaches rather high values and provides effective work of transmission units with the minimum losses of energy. However, operating transmission mechanisms in conditions of negative temperatures, their efficiency becomes significantly lower. Thus, at typical winter temperatures of Ukraine (-15 … -10) °C at the beginning of the transmission efficiency is in the range of 0.55 … 0.60 [9]. In the process of moving the car as the self-heating units of the transmission, their efficiency increases after 30-40 minutes. after the beginning of the movement reaches values of 0.88 … 0.93 [9].

Table 1

Typical values of efficiency of gears at the optimum temperature

Transmission status	Average efficiency depending on the type of transmission	
	cylindrical, η_{zn}	conical, $\eta_{\kappa n}$
New	0,975	0,96
After running-in	0,98	0,97
Maximum earnings	0,99	0,98

The determining factor influencing the efficiency of transmission mechanisms is the viscosity of the transmission oil. Due to the fact that the dynamic viscosity of the transmission oil at negative temperatures varies widely, the value of the efficiency of the gears will also be variable. For gears whose dimensions are known, the efficiency can be determined by the formula [10]:

$$\eta_{зп} = 1 - 2{,}3 \cdot \mu_{mz} \cdot \gamma_{зм} \cdot \left(\frac{1}{z_1} \pm \frac{1}{z_2} \right), \tag{4}$$

where μ_{mz} – coefficient of friction in gearing;

$\gamma_{зм}$ – coefficient taken into account the displacement of the gear;

z_1, z_2 – the number of teeth of the master and slave gears accordingly.

The plus sign in expression (4) is valid for gears with external gearing, and the minus sign is valid for gears with internal gearing.

The main difficulty in using formula (4) is the analytical determination of the coefficient of friction in the gearing due to the influence of several factors simultaneously. Thus, in [10, 11] the following regularities of change of the coefficient of friction are established μ_{mz}:

- the coefficient of friction depends little on the material of the gears and the magnitude of the contact stress in the gearing;

- the coefficient of friction decreases with increasing sliding speed and rolling speed;
- the coefficient of friction increases with increasing surface temperature of the friction pairs;
- the coefficient of friction decreases with increasing viscosity of the oil.

Thus, currently the determination of the coefficient of friction in the gearing of cylindrical and bevel gears is carried out only on the basis of experimental dependences. A large amount of experimental studies with transmission oils [9, 11] allowed to derive an empirical formula for determining the coefficient of friction in gears:

$$\mu_{mz} = \mu_0 - 0{,}026\,\lg \cdot E_0, \qquad (5)$$

where μ_{mz} – the coefficient of friction of the teeth at a viscosity of 1 ° E. For transmission oil is accepted μ_{mz} = 0,11 [8];

E_0 – conditional viscosity of transmission oil in degrees Engler, E.

The translation of the non-systemic unit of viscosity of oils into the unit of kinematic viscosity ν is performed by the empirical formula [7]:

$$\nu = 0{,}073E_0 - \frac{0{,}063}{E_0},\ \text{St}. \qquad (6)$$

Given the fact that in the transmission units it is necessary to operate with the dynamic viscosity of the transmission oil, it is necessary to make the transition to the appropriate units Pa · s. The solution of this problem is carried out through the relationship of dynamic and kinematic viscosity. The ratio of dynamic η_{ol} and kinematic ν_{ol} viscosity of oil is characterized by expression:

$$\eta_{ol} = \nu_{ол} \cdot \rho_{ol} \cdot 10^{-4},\ \text{Pa} \cdot \text{s}, \qquad (7)$$

where ρ_{ol} – density of transmission oil, kg/m^3.

Taking into account the formula (6), expression (7) will take the following form:

$$\eta_{ol} = \left(0{,}073 \cdot E_0 - \frac{0{,}063}{E_0}\right) \cdot \rho_{ol} \cdot 10^{-4},\ \text{Pa} \cdot \text{s}. \qquad (8)$$

As a result of simple mathematical transformations (8) we obtain: $\left(0{,}073 \cdot E_0^{\,2} - 0{,}063\right) \cdot \rho_{ol} \cdot 10^{-4} - \eta_{ол} \cdot E_0 = 0$ and solve the quadratic equation.

$$E_0 = \frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ол}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\ . \qquad (9)$$

Substituting (9) into formula (5) we obtain the expression for calculating the coefficient of friction in the gear:

$$\mu_{mz} = \mu_0 - 0{,}026\lg\left(\frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ol}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\right). \qquad (10)$$

Then the expression for determining the efficiency of the cylindrical transmission mechanism as a function of the dynamic viscosity of the oil will be as follows:

$$\eta_{зп} = 1 - 2{,}3 \cdot \mu_0 - 0{,}026\lg\left(\frac{\eta_{ol} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ol}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\right) \cdot \gamma_{zм} \cdot \left(\frac{1}{z_1} \pm \frac{1}{z_2}\right). \qquad (11)$$

The efficiency of bevel gears is determined by the formula [14]:

$$\eta_{кп} = 1 - \frac{\pi \cdot \eta_{mz} \cdot \zeta}{2} \cdot \left(\frac{1}{z_{np1}} \pm \frac{1}{z_{np2}} \right), \tag{12}$$

where ζ – the coefficient of the duration of the gear teeth. For bevel gears it is recommended to accept ε = 5,5 [11];

z_{np1} and z_{np2} – the number of consolidated cogs of the leading and driven gear.

The consolidated number of gear cogs is determined from the expression:

$$z_{np.i} = \frac{z_i}{\cos \varphi_i},$$

where φ_i – the angle between the generator and the axis of the initial cone of the *i*-th gear.

Then expression (12) will take the following form:

$$\eta_{кп} = 1 - \frac{\pi \cdot \eta_{тз} \cdot \zeta}{2} \cdot \left(\frac{\cos \varphi_1}{z_1} \pm \frac{\cos \phi_2}{z_2} \right). \tag{13}$$

Taking into account the dependence of the coefficient of friction in the gearing on the dynamic viscosity of the transmission oil we obtain the equation for determining the efficiency of the bevel transmission of the transmission mechanism:

$$\eta_{кп} = 1 - \left(\mu_0 - 0{,}026 \lg \left(\frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018 \rho_{ол}^2 \cdot 10^{-8} l}}{0{,}15 \rho_{ol} \cdot 10^{-4}} \right) \right) \frac{\pi \cdot \zeta}{2} \cdot \left(\frac{1}{z_{np1}} \pm \frac{1}{z_{np2}} \right). \tag{14}$$

Friction energy losses in bearings of transmission mechanisms

To determine the efficiency of rolling bearings, it is first necessary to analyze the design of the transmission unit. For example, we will analyze the design of the gearbox of hoisting installations for the repair of wells UPA-60/80. The analysis shows that the design uses ball and roller bearings with cylindrical and tapered rollers. The efficiency of such bearings is due to rolling friction losses. The main factors influencing the efficiency of rolling bearings include the nature of the load, the viscosity of the oil and speed.

Analytical calculation of the efficiency of rolling bearings η_{ns} can be performed according to the formula:

$$\eta_{ns} = 1 - \frac{N_{bn}}{N_{bb}}, \tag{15}$$

where N_{bb} – power supplied to the drive shaft of the transmission unit on which the rolling bearing is installed, Wt;

N_{bn} – power loss in the bearing, Wt.

The power loss in the bearing is determined from a known expression:

$$N_{bn} = \frac{2\pi \cdot n_b \cdot M_{mn}}{60}, \tag{16}$$

where M_{mn} – moment of friction in the bearing, N × m;

n_b — the speed of rotation of the shaft on which the bearing is mounted, s^{-1}.

The moment of friction in rolling bearings is determined from the expression [14]:

$$M_{mn} = d_b \cdot P_n \cdot \mu_{np}, \text{ N×m}, \tag{17}$$

where d_b – diameter of a neck of a shaft under the bearing, m;

P_n – current load on the bearing, N;

μ_{np} – consolidated coefficient of friction.

The values of the consolidated coefficients of friction μ_{np} for different types of bearings that can be in the transmission units of lifting units for the repair of wells UPA-60/80 are given in table. 2.

Table 2

The values of consolidated coefficients of friction of different types of bearings

Bearing	Type	Consolidated coefficient of friction μ_{np}
Ball	Radial	0,0015…0,0025
	Radially stopping	0,002…0,004
	Stopping	0,003…0,005
Rolling	Cylindrical	0,004…0,010
	Conical	0,005…0,015

Taking into account the formulas (15 - 17), the expression for calculating the efficiency of the rolling bearing will take the following form:

$$\eta_{ns} = 1 - \frac{\pi \cdot n_b \cdot d_b \cdot P_n \cdot \mu_{np}}{30 N_{bb}} . \tag{18}$$

Then, for example, for the gearbox of UPA-60/80 hoists on the KrAZ-63221 with ball radial, cylindrical roller, and roller tapered roller bearings, which takes into account the energy loss due to friction in the bearings depending on the speed of the transmission shafts and power which is transmitted to consumers (winch, rotor, etc.) will be determined by the formula:

$$\eta_{mn} = \left(1 - \frac{0{,}001\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^a \cdot \left(1 - \frac{0{,}007\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^b \cdot \left(1 - \frac{0{,}01\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^c . \tag{19}$$

Experimental studies of implementation of the proposed method of increasing fuel energy

Experimental verification of the adequacy of the developed mathematical model was carried out in the laboratory of heat engines of the Department of Road Transport of Ivano-Frankivsk National Technical University of Oil and Gas based on the power drive of the diesel engine D21A1 (Fig. 1), which includes measuring equipment and gearbox stand. Given that the mechanical transmission has physical, geometric and thermal similarity, the results of the research can be extended to the gearboxes of cars and hoists.

Fig. 1. Appearance of experimental stand based on the diesel engine D21A1:
1 – converted to diesel engine D21A1;
2 – transmission;
3 – compressor K-5M;
4 – air heating device

Measuring complex based on a personal computer, an eight-channel motor tester, and chromel-copel thermocouples was used to record the temperature of exhaust gases and transmission oil. One of the sensors was installed at the outlet of the exhaust manifold (to the recuperator), the other - in the transmission units, immersed in oil. The sensitive elements of the temperature sensors (chromel-droplets) were located in the centre of the cross section of the exhaust pipe. The readings of the exhaust gas and transmission oil temperature sensors were recorded continuously.

Two temperature sensors were installed in the gearbox housing to study the temperature of the transmission oil. Sensor № 1 was installed in the lower layer of the gears of the first gear and reverse, sensor № 2 - in the upper layer of oil near the gears of the 5th gear. The drive of the primary shaft of the transmission was carried out from the diesel engine D21A1. The crankshaft of the diesel engine rotated by means of the electric motor of a direct current. To do this, the standard starter of the diesel engine D21A1 was replaced by a special DC gear motor. The motor was powered by a 14 V and 500 A power supply.

Measurement of power losses in the gearbox was performed by measuring the voltage and current on the drive motor. The study of power losses with a manual transmission included the determination of total losses (hydraulic and mechanical) to overcome the forces of resistance to rotation depending on the temperature of the transmission oil. Losses in the gearbox of the car are similar to the cost of power consumed by the motor of the installation, taking into account the power of mechanical losses lost in the drive motor. Two brands of the most widely used transmission oils were used for research: mineral TAP-15V SAE 80W-90 API GL-3 and semi-synthetic TM-5-18 SAE 75W90 API GL-5. The first oil is used in the oil and gas industry of Ukraine for transmissions of hoists manufactured in the former CIS countries, the second oil - for transmissions of hoists manufactured in the USA, Canada and European countries.

As a result of the study it was found that the power required to scroll the gearbox at an ambient temperature of 263 K at the time of starting the engine for mineral oil TAP-15V was 902 W for semi-synthetic TM-5-18 - 625 W (Fig. 2). Further scrolling of the transmission at 273 K led to a reduction in power consumption, reaching 720 W for oil TAP-15V and 540 W for TM-5-18. At an oil temperature of 303 K power losses when using oils of different grades were almost equal and amounted to 448 W and 425 W for mineral and semi-synthetic oils respectively.

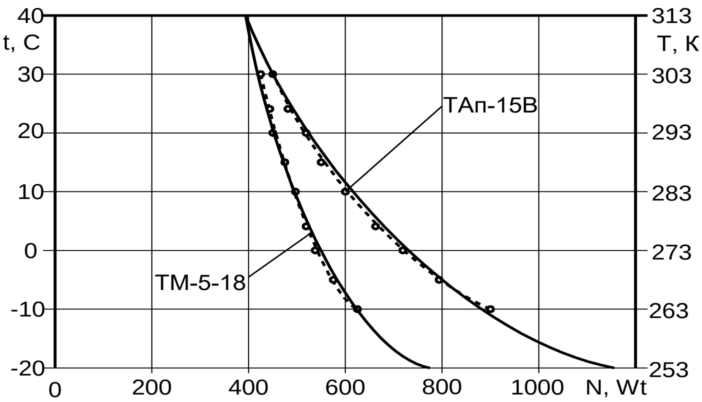

Fig. 2. Dependence of power losses in the gearbox depending on the temperature and grade of transmission oil

To verify the adequacy of the obtained analytical model in Fig. 3 were placed the theoretical dependences of the change in power loss in the gearbox from the temperature of the transmission oil. The combination of theoretical and experimental dependences showed that the maximum difference in the range of changes in power losses from temperature does not exceed 6 %. This indicates a satisfactory adequacy of the obtained mathematical model.

On the basis of the developed and confirmed mathematical model for the lifting installation for repair of UPA 60 / 80A wells on the KrAZ-63221-04 chassis calculations for power losses in a gearbox were carried out. The UPA 60 / 80A lifting unit on the KrAZ-63221-04 chassis can be equipped with a 176 kW (240 hp) engine or a 220 kW (300 hp) engine. As a result of the study it was found (Fig. 3) that the power required to scroll the gearbox at an ambient temperature of 253 K at the time of starting the engine for mineral oil TAP-15V is 14.20 kW, for semi-synthetic TM-5-18 - 9.50 kW. Further scrolling of the transmission at 273 K leads to a reduction in power consumption, reaching 9.30 kW for oil TAP-15V and 6.95 kW for TM-5-18. At an oil temperature of 313 K, power losses when using oils of different grades are almost equal and are for mineral and semi-synthetic oil respectively 4.85 and 4.80 kW.

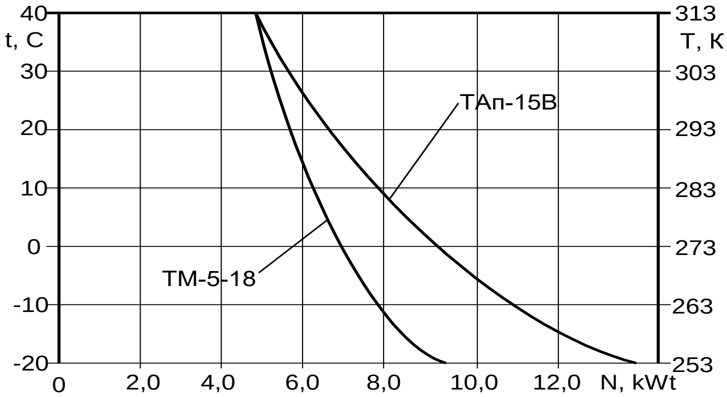

Fig. 3. Dependence of power losses in the gearbox of the lifting unit model UPA 60 / 80A depending on the temperature and grade of transmission oil

We will calculate the fuel consumption in the gearbox of the lifting unit UPA 60 / 80A with different power drives and at different temperatures of the transmission oil (Table 3).

Table 3

Results of calculations of fuel consumption in the gearbox of the lifting unit model UPA 60 / 80A with different power drives and at different temperatures of transmission oil

Temperature, K	Power consumption, kW		Engine / values of the minimum effective specific fuel consumption, g/(kWh)		Fuel consumption per gearbox drive, semi-synthetic oil, kg/h	Fuel consumption per gearbox drive, mineral oil, g/h
	Semi-synthetic oil	Mineral oil	YaMZ-238BE2	YaMZ-238VM		
313	4,80	4,85	195	214	0,93 - 1,02	0,94 - 1,04
273	6,95	9,30			1,36 - 1,49	1,81 - 1,99
253	9,50	14,20			1,85 - 2,03	2,77 - 3,03

On the main technological modes connected with drilling and repair of wells it is possible to accept average specific fuel consumption of 220 g/(kWh). Values of the minimum effective specific fuel consumption of atmospheric engines YaMZ-238VM with a capacity of 176 kW at the speed of the crankshaft of the engine 1300 min^{-1} amounted to 214 g/(kWh); nominal effective specific fuel consumption of atmospheric engines YaMZ-238VM at the speed of the crankshaft of the engine 2100 min^{-1} – 259 g/(kWh). The values of the minimum effective specific fuel consumption of supercharged engines YaMZ-238BE2 with a capacity of 220 kW at the speed of the crankshaft of the engine 1400 min^{-1} were equal to 195 g/(kWh); nominal effective specific fuel consumption of supercharged engines YaMZ-238BE2 at the speed of the engine crankshaft 2100 min^{-1} – 238 g/(kWh).

Conclusions

Studies have shown that reducing energy consumption in the transmission of hoists for repairing wells by using the heat of the exhaust gases to ensure rapid heating of the transmission units and maintain the optimal thermal regime is quite profitable.

As a result of calculations it was established (tab. 4) that the overuse of fuel necessary for scrolling of a transmission of the lifting installation of the UPA 60 / 80A model with various power drives at ambient temperature of 253 K at the moment of start for TAp-15B mineral oil makes 1, 83-1,99 kg for semi-synthetic TM-5-18 - 0,92-1,01 kg in comparison with temperature of 313 K. Further scrolling of a transmission at 273 K leads to decrease in fuel consumption having reached 0,87-0, 95 kg for oil TAp-15B and 0.43-0.47 kg for TM-5-18 compared to a temperature of 313 K.

Therefore, the obtained data showed that the high efficiency of heat transfer to the transmission units and the reduction of energy consumption in the transmission can be achieved due to the differences in the temperatures of the exhaust gases and the transmission oil. A significant stimulus for the further development of such systems is that they determine the possibility of cumulative improvement of the characteristics of the vehicle on a set of indicators. Their implementation on vehicles allows to utilize waste thermal energy and reduce fuel consumption by oil and gas technological transport.

References

1. Denisyuk S.P. Formation of the policy for the promotion of energy efficiency - the current policy of the European organization. Power engineering: economy, technology, ecology. 2013. No. 2. P. 7–22

2. Energy strategy of Ukraine for the period up to 2030 fate / Skvalena to the order of the CMU 15 March 2006 r. No. 145. - Access mode: http://zakon2.rada.gov.ua/laws/show/145-2006-r

3. Energy saving. Systems of energetic management. Vimogi ta nastanova shodo vikoristannya: DSTU ISO 50001: 2014. - K .: Ministry of Economic Development of Ukraine, 2014.26 p.

4. Shevtsov V.M. Theoretically, the advancement of the infinitely variable transmission of the KrAZ 63221-02 vehicle for the repair of naftogaz verdorovyh. / O.O. Ostroverkh, A.P. Kozhushko, V.M. Shevtsov // Bulletin of the National Technical University "KhPI". - Kharkiv: NTU "KhPI". - 2017. - No. 6 (1228). - S. 45 - 51.

5. Virvinsky P. P., Khomenko V. L. Sverdlovin repair: Navchalny pos_bnik. - Dnipro ¬petrovsk: National State University, 2003 .-- 219 p.

6. Bukhalenko E.I. Oilfield equipment. Directory. Moscow: Nedra, 1990.

7. Samorodov V. B., Ostroverch A. O., Kozhushko A. P. Development and analysis of stepless two-flow hydrostatic-mechanical transmission by criterion of greatest efficiency for cars used to repair oil and gas wells. Visnyk Natsional'nogo tekhnichnogo universytetu "KhPI" [The bulletin of the National Technical University "KhPI"]. 2012, vol. 60, pp. 105-111.

8. Samorodov V. B., Ostroverch A. O., Kozhushko A. P. Analysis of the technological mode of operation of a continuous hydrovolume-mechanical transmission using native hydro-units of a car used to repair oil and gas wells]. Visnyk Natsional'nogo tekhnichnogo universytetu "KhPI" [The bulletin of the National Technical University "KhPI"]. 2013, vol. 30, pp. 23–31.

9. Razyapov M.M. Improving the performance of transmission units of automotive vehicles at low temperatures: abstract of Ph.D. dis. ... Cand. tech. sciences / M.M. Razyapov. - Ufa, 2013 .-- 16 p.

10. Pustozerov Yu.I. Investigation of the operating modes of the transmission of the DT-75 tractor in low temperature conditions: abstract of Ph.D. dis. Cand. tech. Sciences / Yu. I. Pustozerov - Novosibirsk, 2003 .-- 23p.

11. Selivanov N.I. Improving the efficiency of tractor units in the winter conditions of the agro-industrial complex of Eastern Siberia: author. dis. ... Dr. Tech. sciences / N.I. Selivanov. - Krasnoyarsk, 2006.

12. Karpov P. M. Improving the fuel efficiency of agricultural tractors in winter conditions by improving the lubrication modes of transmissions (for example, the DT-75M tractor): author. dis. ... Cand. tech. Sciences / P.M. Karpov. - Ulyanovsk, 1988 .-- 25 p.

13. Kutlin A.A. Investigation of the influence of the mode of movement of vehicles on the temperature of their main units and fuel consumption in winter operating conditions: author. dis. ... Cand. tech. nauk / A.A. Kutlin. - Kiev, 1981 .-- 20 p.

14. Razyapov M.M. Improving the performance of transmission units of automotive vehicles at low temperatures: author. dis. ... Cand. tech. sciences / M.M. Razyapov. - Ufa, 2013 .-- 16 p.

Микитій І.М. Математичне моделювання зниження втрат енергії в коробках передач технологічного транспорту нафти та газу.

Робота спрямована на вирішення проблеми зменшення втрат енергії в блоках передачі підйомних машин для ремонту свердловин. Запропоновано метод швидкого нагрівання та підтримки оптимальної температури в трансмісійних блоках підйомних агрегатів за допомогою тепла відпрацьованих газів. Проведено аналіз особливостей конструкції передач підйомних установок для ремонту свердловин. Проведено дослідження в'язко-температурних характеристик сучасних трансмісійних масел та температурного режиму в агрегатах трансмісії. Запропоновано математичну модель виділення енергії в блоках передачі під час роботи підйомних установок. Встановлені витрати енергії на тертя в шестернях трансмісійних агрегатів. Визначено втрати енергії тертя в підшипниках механізмів передачі підйомних агрегатів. Запропоновано метод зменшення втрат енергії в блоках передачі підйомних установок для ремонту свердловин. Експериментальні дослідження реалізації запропонованого методу зменшення втрат енергії в блоках передачі. Встановлено залежність втрат потужності в коробці передач підйомних агрегатів в залежності від температури та марки трансмісійного масла. Отримано залежність втрат потужності в коробці передач підйомного агрегату моделі UPA 60 / 80A в залежності від температури та марки трансмісійного масла. Наведено результати розрахунків витрати палива в коробці передач підйомного агрегату моделі UPA 60 / 80A з різними приводами потужності та при різних температурах трансмісійного масла.

Ключові слова: технологічний транспорт нафти і газу; підйомна установка для ремонту, коробка передач; блок передачі; вихлопні гази; використання тепла; потужність.