Improvement of the efficiency of hydraulic power steering

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Abstract. The purpose of the study is to increase the reliability of the steering system. The article reflects scientifically based recommendations for improving the design of steering. The research method is theoretical studies performed on the basis of the provisions, laws and methods of theoretical mechanics, hydraulics, heat engineering, power flow theory and mathematical analysis. The article presents the results of theoretical studies on the temperature regimes of a hydraulic power steering (HPS), the results of which establish the analytical relation between the temperature pressure of the working fluid and the efficiency coefficient of the HPS. As a result, knowing the temperature of the working fluid, it is possible to determine the efficiency of the HPS in any of its operating modes. The analysis shows that in order to increase the efficiency of the HPS it is necessary to lower the temperature of the HPS working fluid. To this end, a device is developed aimed at improving the efficiency of the cooling fluid in the power steering and allowing increasing the efficiency of the HPS.

1. Introduction
The achievement of significant economic success in the agro-industrial complex (AIC) of Russia is largely determined by the reliability of the agricultural equipment.

The main part of the AIC of RF is equipped with trucks with hydraulic power. One of the problems of the HPS design according to the data of L L Ginzburg (1972) is to increase its temperature regime as a result of a long action at the extreme positions of the steering wheel. Maintaining normal temperature regimes of the HPS significantly affects the reliability of the machine.

As it is shown by the analysis of materials on the steerability in the works of native and foreign scientists: E M Gonikberg and A A Golbreykh (1969); L L Ginzburg (1959); V V Osepchugov and A K Frumkin (1989), the process of heat generation in the HPS was not fully considered, no patterns of heat release in the HPS from structural and operational parameters were identified, and there were no uniform principles for choosing an economical and efficient HPS circuit [1-3].

Thus, the creation of new scientifically based decisions at the design stage will make it possible to predict the conformity of the steering controls to the requirements and take measures aimed at improving the design of the steering control [4].

2. Materials and methods
In the article, the design scheme is based on the hydraulic scheme of the HPS of Ural 4320-0010-31.

Analysis of the calculation methods showed that the HPS is a complex system in which transformations of mechanical, hydraulic and heat flows take place, each of which is calculated according to its own laws [5-7]. In this regard, to determine the uniform principles for calculating the HPS, the power flow theory developed by A S Antonov (1981). According to this theory, the HPS can be represented in the
form of separate power flows: mechanical, hydraulic and thermal.

For the formation of the design scheme, the moment \( M \) (Nm) is taken as the power factor of the mechanical flow, and the angular velocity of the shaft \( \omega \) (rad/s) is taken as the speed factor. The mass flow rate of the fluid \( q \) (kg/s) is taken as the power factor of the hydraulic flow, and the head \( H \) (m) is taken as the speed factor. The full heat capacity of the mass flow rate of a liquid \( W \) (W/°C) is taken as the power factor of the heat flow, and the temperature of the liquid \( T \) (°C) is taken as the speed factor of the heat flow.

The product of power and speed factors allows you to respectively obtain the power factor of the mechanical flow or the mechanical power \( N = M \omega \), the power factor of the hydraulic flow or the hydraulic power \( N^G = qH \), the power factor of the heat flow or the power of the heat flow \( Q = WT \).

According to the power flow theory, the whole variety of transforming devices included in the design scheme can be conditionally represented by three types of nodal points: branching, kinetic, and generalized. At the branching nodal point only force factors are transformed, and in kinetic nodal point speed factors are transformed. The generic nodal point is the union of kinetic and branching nodal points. It is simultaneously the transformation of power and speed factors.

The analytic model of the HPS (Figure 1) includes the following nodal points:

1) Kinetic: an oil pump tank, a radiator - a device for improving the efficiency of the HPS cooling fluid, a distributor, a filter;
2) Generic: an engine, a vane-type pump, a power cylinder, a resistor.

For each flow, we separately compose the equation of balance continuity at the nodal points. To compile the balance equations, we use the second and third principles of the power flow theory. According to the second principle of the power flow theory, the sum of the velocity factors of a closed power flow is zero.

Or in general terms:

\[
\sum_{k=1}^{n} U_k = 0, \quad (1)
\]

where \( U_k \) is the speed factor of K-flow; \( n \) is the number of flows.

The speed factor transformation (temperature) for the kinetic nodal points in a closed power circuit of an oil pump tank, a radiator – a device to improve the cooling efficiency of the HPS fluid, a distributor, a filter and generic nodal points of a vane-type pump and a main cylinder will be determined accordingly:

\[
T_p = T_1 - T_2, \quad (2)
\]
\[
T_d = T_3 - T_4, \quad (3)
\]
\[
T_{mc} = T_5 - T_6, \quad (4)
\]
\[
T_r = T_7 - T_8, \quad (5)
\]
\[
T_f = T_9 - T_{10} \quad (6)
\]
\[
T_{ot} = T_{11} - T_{12}, \quad (7)
\]

where \( T_p, T_d, T_{mc}, T_r, T_f, T_{ot} \) are temperature values of the pump, the distributor, the main cylinder, the radiator – the device to improve the cooling efficiency of the HPS fluid, the filter and the oil tank, correspondingly, °C. \( T_{1,2} \) are temperature values of the pump fluid of the hydraulic line between the pump outlet and the distributor inlet; \( T_{3,4} \) are temperature values of the distributor power fluid of the hydraulic line between the outlet of the distributor and the main cylinder inlet; \( T_{5,6} \) are temperature values of the main cylinder power fluid of the hydraulic line between the outlet of the distributor and the main cylinder inlet; \( T_{7,8} \) are temperature values of the radiator power fluid – the device to improve the cooling efficiency of the HPS fluid of the hydraulic line between of the hydraulic line between the radiator outlet and the filter inlet; \( T_{9,10} \) are temperature values of the filter power fluid of the hydraulic line between the filter outlet and the oil tank inlet; \( T_{11,12} \) are temperature values of the oil tank power fluid of the hydraulic line between the oil tank outlet and the pump inlet.
According to the third principle of the power flow theory, the sum of the powers of all the flows supplied and removed from the nodal point is equal to zero.

Or in general terms:

$$\sum_{i=1}^{n} N_i = 0,$$  \hspace{1cm} (8)

where $N_i$ is the power factor of $i$-flow; $n$ is the number of flows.

The third principle of the power flow theory is applied to each nodal point of the analytic model and, considering all output power fluids as negative, and all input ones as positive, we get:

1) For the distributor:

$$Q_2 - Q_1 - Q_d = 0,$$  \hspace{1cm} (9)

where $Q_1$ is the power of the heat flow of the power fluid of the hydraulic line between the distributor outlet and the inlet of the radiator - the device to improve the cooling efficiency of the HPS fluid, W; $Q_2$ is the power of the heat flow of the working fluid of the hydraulic line between the pump outlet and the distributor inlet, W; $Q_d$ is the heat transferred by the distributor to the environment, W.
2) For the radiator – the device to improve the cooling efficiency of the HPS fluid:

\[ Q_1 - Q_3 - Q_r = 0, \]  

where \( Q_1 \) is the power of the heat flow of the power fluid of the hydraulic line between the radiator outlet and the oil tank inlet, W; \( Q_r \) is heat dissipation capacity of the radiator, W.

3) For the oil tank:

\[ Q_3 - Q_4 - Q_{ot} = 0, \]  

where \( Q_3 \) is the power of the heat flow of the power fluid of the hydraulic line between the oil tank outlet and the pump inlet, W; \( Q_{ot} \) is the heat transferred by the oil tank to the environment, W.

4) For the pump:

\[ N_e - N^h_p - \Delta N_p = 0, \]  

where \( N_e \) is the mechanical power supplied from the engine, W; \( N^h_p \) is the hydraulic power removed from the pump, W; \( \Delta N_p \) is power loss, W.

According to the power flow theory of A.S. Antonov (Antonov, A.S., 1981) energy capabilities of the hydraulic drive are fully characterized by its power. The power balance can be represented as:

\[ N_{in} = N_{out} + \Delta N, \]  

where \( N_{in} \) is the input power, W; \( N_{out} \) is the power output, W; \( \Delta N \) is power loss in the hydraulic drive, W.

The HPS energy balance will be:

\[ N_{in} = N_{out} + \Delta p_{p} q + \Delta N_{p} + \Delta p_{r} q + \Delta p_{s} q + \Delta p_{mc} q + \Delta N_{mc}. \]

All hydraulic power losses are converted to heat.

Thus, the obtained calculation scheme reflects in general the processes of transformation of mechanical, hydraulic and heat flows of HPS, as their speed and power factors are interconnected by the second and third principles of the power flow theory.

3. Results

Direct determination of HPS efficiency is complicated by the fact that it is theoretically and practically very difficult to determine losses at all points of HPS characteristics [8-10]. The point of the characteristic that can be determined by calculation reliably is the point of maximum efficiency.

In general, the equation for the total power losses of the HPS is:

\[ \sum \Delta N = N_{in} - N_{out}, \]  

Since the value of \( N_{out} \) in formula (14) is unknown, we express this equation in terms of efficiency, thereby determining the analytical relationship between power loss and power efficiency factor of the HPS.

\[ \sum \Delta N = N_{in} (1 - \eta_{HPS}), \]  

from which

\[ \eta_{HPS} = \frac{N_{in} - \sum \Delta N}{N_{in}}. \]  

To further description of the mathematical model including parameter \( \sum \Delta N \) in formula (16), the whole variety of power losses is converted into heat (Prokofyev, V.N., 1960). That is, the article puts forward the hypothesis that the total losses in the steady-state modes of operation of the HPS are equal to the amount of heat that must be removed from the hydraulic booster

\[ \sum \Delta N = Q_{HPS}. \]  

Parameter \( Q_{HPS} \) does not include the amount of heat required to heat the working fluid and parts of the HPS, as well as the amount of convection discharged into the environment. That is, it is assumed that the percentage of \( Q_{HPS} \) is much larger than the listed components of heat dissipation.

Thus, the purpose of the mathematical model is to determine the amount of heat removed in steady-
state modes of the HPS work in order to determine the HPS efficiency more precisely later on.

The variable parameters of the mathematical model of the mechanical flow are: the angular velocity of the pump shaft $\omega_p$, the speed of the piston rod $V_r$, the input power $N_{in}$, and for the heat flow it is the amount of heat $Q_{HPS}$, withdrawn from the HPS [11,12].

In the deterministic process, when the velocity temperature fields of coolants do not change, there is no need to solve the heat balance equation in a differential form (Shukhman, S.B., et al., 2007).

Using the equations of continuity of heat fluxes at nodal points, we create the system of equations:

$$\begin{align*}
Q_1 - Q_2 - Q_d &= 0 \\
Q_1 - Q_3 - Q_r &= 0 \\
Q_3 - Q_4 - Q_{ot} &= 0
\end{align*}$$

(18)

We substitute the values of the power and speed factors of thermal hydraulic flows in the system of equations:

$$\begin{align*}
W_{wf} T_1 - W_{wf} T_6 - Q_d &= 0 \\
W_{wf} T_4 - W_{wf} T_7 - Q_r &= 0 \\
W_{wf} T_7 - W_{wf} T_{12} - Q_{ot} &= 0
\end{align*}$$

(19)

To determine $Q_d, Q_r, Q_{ot}$ according to the power flow theory, it is necessary to use the equation of characteristics. For $Q_d$ the equation has the following form:

$$Q_d = K_1 F_1 \left(T_d - T_{air}\right),$$

(20)

where $K_1$ is the heat transfer coefficient of the distributor, $W/m^2\cdot{^\circ}C$; $F_1$ is the distributor cooling surface, $m^2$; $T_{air}$ is the average temperature of the working fluid and air, respectively, °C;

$$Q_r = K_2 F_2 \left(T_r - T_{air}\right),$$

(21)

where $K_2$ is the heat transfer coefficient of the radiator, $W/m^2\cdot{^\circ}C$; $F_2$ is the radiator cooling surface, $m^2$; $T_{air}$ is the average temperature of the working fluid and air, respectively, °C;

$$Q_{ot} = K_3 F_3 \left(T_{ot} - T_{air}\right),$$

(22)

where $K_3$ is the heat transfer coefficient of the oil tank, $W/m^2\cdot{^\circ}C$; $F_3$ is the oil tank cooling surface, $m^2$; $T_{air}$ is the average temperature of the oil tank cooling surface, $m^3$; $T_{ot}$ is the average temperature of the working fluid and air, respectively, °C; The system of equations (19) has four unknown parameters, and the equation has three ones.

But since our task is to determine the difference between the temperatures of the HPS, we have two unknown parameters and replace them with the difference $\Delta T_{WF}^{HPS}$.

From the system of equations (19) we have

$$\Delta T_{WF}^{HPS} = \frac{T_1 - T_{12}}{W_{wf}} = \frac{Q_d + Q_r + Q_{ot}}{W_{wf}},$$

(23)

or

$$\Delta T_{WF}^{HPS} = \frac{K_1 F_1 \left(T_d - T_{air}\right) + K_2 F_2 \left(T_r - T_{air}\right) + K_3 F_3 \left(T_{ot} - T_{air}\right)}{C_{wf} q},$$

(24)

where $C_{wf}$ is the specific heat of the working fluid, $J/kg\cdot{^\circ}C$; $q$ is the working fluid consumption, kg/s.

Thus, using the systems of equations of thermal and hydraulic power balances as a result of transformations, a functional dependence of the steady temperature head of the working fluid of the HPS was obtained (24).

Next, we establish the analytical relationship between the temperature pressure of the working fluid and the efficiency of the HPS. To do this, we use the dependence (15), substituting the quantity of heat diverted from the HPS instead of $\Sigma \Delta N$. As a result, we get:

$$Q_{HPS} = N_{in} (1 - \eta_{HPS}),$$

(25)
Substituting in formula (25) the product of their speed and power factors instead of the power factors:

\[ W_{wf} \Delta T_{HPS} = M_p \omega_p \left( 1 - \eta_{HPS} \right), \]  

(26)

from which

\[ \eta_{HPS} = 1 - \frac{\Delta T_{HPS}^{WF} C_{wf} q}{M_p \omega_p}, \]  

(27)

4. Discussion

Analytical dependence (27) establishes the relationship between temperature and efficiency of the HPS. Knowing the temperature head of the working fluid, it is possible to determine the efficiency of the HPS in any of its operating modes. The analysis shows that increasing the temperature head of the working fluid leads to a decrease in the HPS efficiency. Therefore, to increase the efficiency of the HPS, it is necessary to lower the temperature head of the working fluid.

For this purpose, a device has been developed that is protected by a patent (Pat. No. 2665109, Russian Federation, MPK7V 62 D 5/06 Hydraulic Power Steering / Afinogenov, I.A., Byshov, N.V., Borychev, S.N. [and others] No. 2017108371; claimed 13.03.2017; published 28.08.2018, Bulletin No. 16), aimed at improving the efficiency of cooling the fluid in the hydraulic power steering (Figure 2).

The device operates as follows: during the operation of the HPS, working fluid 2 flows inside pipeline 1 and washes inserts 3. Due to the higher thermal conductivity of inserts 3 than that of the main material of pipeline 1, heat exchange between working fluid 2 and the surrounding air begins through inserts 3. In this case, heat exchange occurs along the entire length of pipeline 1, which hydraulically connects all elements of the HPS.

\[ \text{Figure 2. Device for the improvement of the efficiency of cooling the power steering fluid: 1 – Pipeline; 2 – working fluid (oil); 3 – inserts.} \]

When the temperature of fluid 2 increases, for example, due to the intensification of the HPS work, inserts 3 begin to play the role of local cooling radiators (heat sinks).

Thus, the proposed device allows increasing the cooling efficiency of the hydraulic power steering fluid and the HPS efficiency.
5. Conclusion

On the basis of the power flow theory, a process has been formed and a corresponding design scheme has been developed, which takes into account the flows converted in the HPS. Using the systems of equations of thermal and hydraulic power balances, a functional dependence of the temperature head of the working fluid of the HPS was obtained, allowing in turn the establishment of a specific analytical relationship between temperature and efficiency, bypassing the calculation of the power loss itself.

To increase the efficiency of the HPS, a device has been developed that improves the cooling efficiency of the hydraulic power steering fluid (protected by the RF patent No. 2665109).

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