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Influence of the excavator hydraulic system efficiency on the productivity

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Abstract: This paper presents the experimental research of factors influencing the UDS 214 excavator efficiency. The hydraulic oil flow rate is measured in hydraulic circuits for controlling the moving of the bucket of the UDS 214 excavator. From the measured values, the total power losses of the individual hydraulic circuits and their efficiencies are evaluated by calculations and measurements. Furthermore, the times of the excavator cycles during a soil excavation and loading of the transport vehicle were measured. From the measured operating cycle times of the excavator, the average value of the operating cycle time was evaluated and, from this average time, the theoretical performance and the operating performance of the given excavator in the given operational states were calculated. Then, at the end of the paper, the individual calculated power losses as well as the efficiencies of the hydraulic circuits for controlling the moves of the excavator are evaluated. According to the findings, the swing hydraulic circuit of the excavator, which has the second highest power loss of 5.926 kW and its percentage in the average tested cycle time of the excavator is 48%, seems to be a suitable hydraulic circuit for the innovation.

Keywords: flow rate; hydraulic circuit; power loss; pressure loss; pressure

In the current tense economic situation, improving the productivity of the excavation work and, consequently, building construction is a crucial issue (Lin et. al. 2010). At the same time, there is also an increased demand for construction machinery and especially for hydraulic excavators, which is reflected by the growing environmental concerns, energy savings and the resulting savings in financial fuel costs (Ge et al. 2017). Precisely for these reasons, hydraulic excavators have found a wide range of uses. There are millions of hydraulic excavators worldwide and they are a significant part of the machinery for most construction companies (Wang and Wang 2014). Hydraulic excavators are very popular with users for their performance, versatility, maintainability, easy manoeuvrability and also for the possibility of using a large number of attachments, which can be used on construction sites (Haga et al. 2001).

In practice, the working performance of a hydraulic excavator depends on many factors, such as the time of the working cycle, the type and volume of the working bucket, the soil loosening coefficient, the filling factor of the working bucket, the digging depth, the angle of rotation, the method of emptying the bucket, and also the quality of the cutting edge. It should be noted that the working cycle time affects the excavator performance the most. In the case of shovel excavators, the time required to fill

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the bucket to the maximum volume, raise the bucket to the dumping height, swing the machine with a full bucket, emptying the bucket, swinging the machine with an empty bucket and shifting the bucket down to its origin position is considered the working cycle (Jůza and Heřmánek 2020).

The UDS 214 telescopic excavator is used in agriculture for excavation work on land reclamation drainage canals, as well as for the creation and subsequent slope work in the creation of drainage ditches along roads, and for the construction and maintenance of field and forest roads. Furthermore, the UDS 214 telescopic excavator is also used for cleaning riverbeds and smaller watercourses, as well as for cleaning of ponds (POŽÁRY.cz 2013).

The main aim of this article is to determine a suitable hydraulic circuit, the innovation of which should increase the actual performance productivity of the UDS 214 telescopic excavator. For this analysis, the calculated total power losses of the individual hydraulic circuits for controlling the working moves of the UDS 214 telescopic excavator tool will be used depending on the calculated actual performance productivity of the UDS 214 excavator and with respect to individual percentage times of the individual work cycle phases in the total average measured excavator cycle time.

MATERIAL AND METHODS

Description of the UDS 214 telescopic excavator. The UDS 214 telescopic excavator (CSM Industry 2020), on a three-axle Tatra 815 truck chassis (Tatra Trucks, Czech Republic) is designed for finishing earthworks and, with the use of suitable additional equipment, for excavation and other earthwork (excavation of foundation trenches, canals, construction and maintenance of utility networks), which can be seen in Figure 1.

At present, the UDS 214 telescopic excavator is used for mining Class I and II soils according to the standard ČSN 73 6133:2010.

The excavator is also suitable for dealing with the consequences of emergencies such as floods, landslides, and also statically disturbed buildings. The UDS 214 telescopic excavator is used both in the private sector and in the armed forces and fire brigades, for example, in the Czech and Slovak Republics. The machine can also be equipped with a micro-travel gear (Jůza and Heřmánek 2020).

This means that the operator can drive the excavator around the construction site with the help of a John Deere 4045 excavator engine and the hydraulic system without the help of the combustion engine of the Tatra chassis. The operator can also control the direction of travel of the Tatra 815 chassis, control the parking brake, extend and retract the stabiliser supports of the excavator, all from the UDS 214 excavator's operator cab.

The power unit of the UDS 214 telescopic excavator is an inline four-stroke liquid-cooled four-cylinder John Deere 4045 diesel engine with direct fuel injection, which has the power of 94 kW at 2 200 rpm (Deere & Company 2021). The engine drives a Bosch Rexroth A8VO107 double axial piston pump (Bosch Rexroth AG 2021a), which sup-

Figure 1. UDS 214 telescopic excavator on a Tatra 815 truck chassis

1 – boom; 2 – inner telescopic arm; 3 – rotating head; 4 – five-teeth bucket with a 0.63 m^3 volume; 5 – lower frame; 6 – Tatra chassis 815; 7 – UDS 214 telescopic excavator

https://doi.org/10.17221/77/2021-RAE

plies pressurised hydraulic oil to the hydraulic circuits of the excavator's working movements via a seven-section Bosch Rexroth 7M8-22 mono-bloc manifold (Bosch Rexroth AG 2021b), allowing the UDS 214 telescopic excavator to perform the five basic movements of the working tool:

– extend and retract the inner telescopic arm,
– raise and lower the telescopic boom,
– turn the working tool (using the swivel head),
– open and close the working tool,
– swing the UDS 214 excavator (Jůza and Heřmánek 2020).

As the operator does not usually use the hydraulic circuit to turn the working tool during soil mining, which is used only for special operations, such as sloping, this movement is not included in the standard scheme of the excavator work cycle. Therefore, the hydraulic system to turn the working tool was not tested in this study (Jůza and Heřmánek 2020).

Hydraulic system of the UDS 214 telescopic excavator. The hydraulic system for controlling the working movements of the UDS 214 excavator consists of five main independent hydraulic circuits. The oil pressure source is provided by a Bosch Rexroth A8VO107 double axial piston pump (Bosch Rexroth, Germany). The double axial piston pump supplies hydraulic oil pressure via a Bosch Rexroth 7M8–22 seven-section mono-block manifold (Bosch Rexroth, Germany):

– into the linear hydraulic motor of the inner telescopic arm,
– into a pair of linear hydraulic motors for the boom stroke, whose hydraulic scheme is shown in Figure 2,
– into a pair of rotary hydraulic motors of the rotating head,
– into the linear hydraulic motor of the tool,
– into the rotary hydraulic motor of the swing body turning.

Methods of measurement. For each examined hydraulic circuit, measurements were realised firstly between the pump and the manifold and then between the manifold and the appliance. In this case, a linear hydraulic motor or a rotary hydraulic motor is meant as an appliance.

Two combined meters were used in the measurement, the first is a digital one from Hydrotechnik GmbH with the designation MultiSystem 5060 (Hydrotechnik GmbH 2012) and the second is an analogue one from the Owatonna Tool Company with the designation OTC H50, which can be seen in Figure 3 (Bosch Automotive Service Solutions 2021).

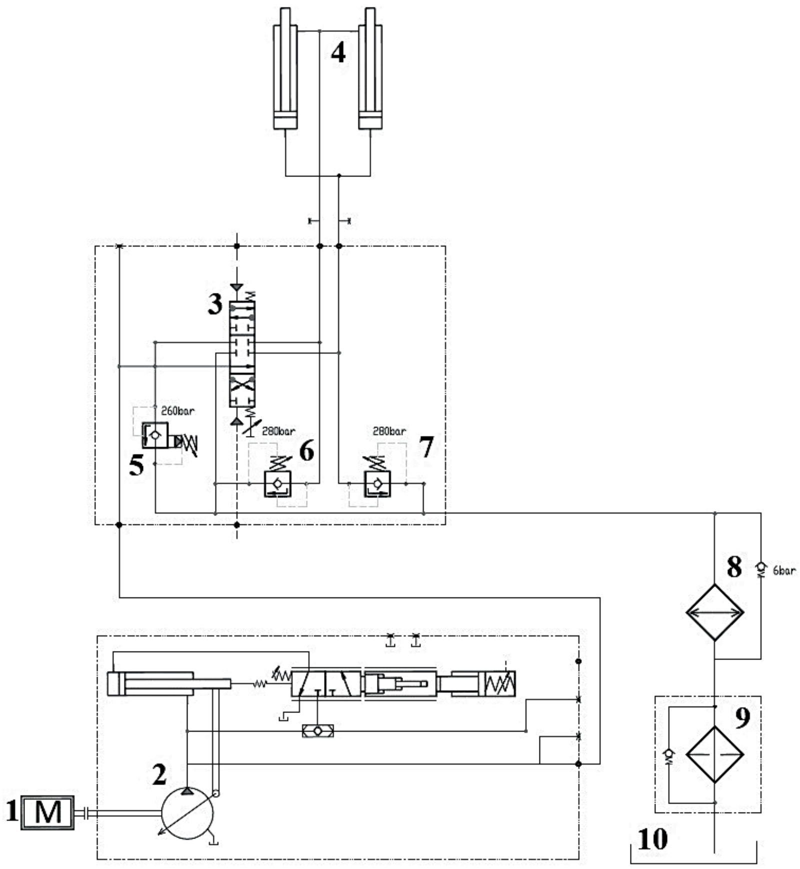

Figure 2. Hydraulic scheme of the telescopic boom stroke
1 – John Deere 4045 engine; 2 – right part of the Bosch Rexroth A8VO107 axial piston pump; 3 – lift section of the Bosch Rexroth 7M8-22 manifold; 4 – two linear hydraulic motors of the telescopic boom stroke; 5 – pressure relief valve; 6 – pressure relief valve; 7 – pressure relief valve; 8 – hydraulic cooler; 9 – hydraulic oil cleaner; 10 – hydraulic reservoir

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Both combined meters allow one to measure the pressure (0–40 MPa), the flow rate (0–200 $dm^3 \cdot s^{-1}$) and the temperature (0–120 °C) of the hydraulic oil in a given hydraulic circuit whose connection to the measured hydraulic circuit is shown in Figure 4. After connecting the combined meters to the measured hydraulic circuit, the testing was performed as follows:

– Setting the John Deere 4045 combustion engine speed to 1 600 rpm.

– Using the joystick in the cab to set the manifold slider to the working position for the appropriate movement of the given hydraulic motor.

– Loading the hydraulic circuit by the throttle valve on the OTC H50 analogue meter to the predetermined pressure.

– Using the Hydrotechnik MultiSystem 5060 digital meter to record the values of the hydraulic oil pressure and flow rate to the meter's memory. Writing the current hydraulic oil temperature and the engine speed into the prepared tables.

When testing the hydraulic circuits between the pump and the switchboard, the final value of the pressure setting was 30 MPa and when testing the hydraulic circuits between the manifold and the appliance, this value was 26 MPa, as the pressure relief valves in all the measured hydraulic circuits are adjusted to this pressure. When calculating the hydraulic circuits between the pump and the manifold, the measured value of the hydraulic oil flow rate at a pressure of 27 MPa was taken into account, as it was assumed that the hydraulic oil pressure at the pump discharge increases during the testing and simulated load at the end of the hydraulic circuit at a maximum pressure of 26 MPa.

Measurement of the hydraulic circuit between the pump and the manifold. For all the measured hydraulic circuits, the source of hydraulic oil pressure in the UDS 214 telescopic excavator is an axial piston double pump. When measuring the hydraulic circuits of the swing body turning and the extension of the telescopic boom, the intake hydraulic hose to both combined meters was connected to the discharge of the left side of the axial piston pump. The output hydraulic hose from the OTC H50 analogue meter was connected to the left inlet to the hydraulic manifold. When measuring the hydraulic circuits of the boom stroke and the opening and closing of the tool, the hydraulic intake hose to both combined meters was connected to the discharge of the right side of the axial piston double pump. The output hydraulic hose from the OTC H50 analogue meter was connected to the right inlet to the hydraulic manifold. During these measurements, after starting the John Deere 4045 engine, it is necessary to place the control joystick into the working position for the appropriate movement of the excavator work tool, so that the swash plate of the tested pump is displaced and, thus, the maximum hydraulic oil supply is achieved. The output of this measurement is the dependence of the pressure on the hydraulic oil flow rate in the measured hydraulic circuit. This dependence is plotted using the Hydrotechnik software and is shown in Figure 5. An example of measured values is given in Table 1.

Measurement of the hydraulic circuit between the manifold and the appliance. When measuring in the hydraulic circuit between the hydraulic mani-

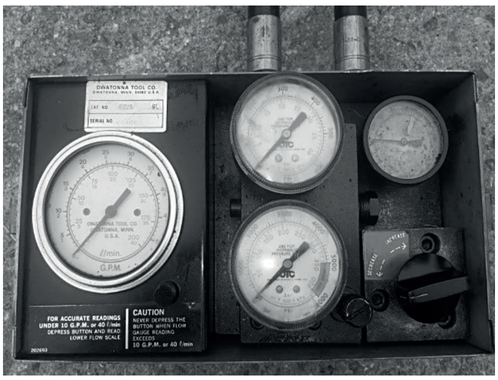

Figure 3. Combined OTC H50 analogue meter for the measuring pressure, flow rate and temperature of the hydraulic oil

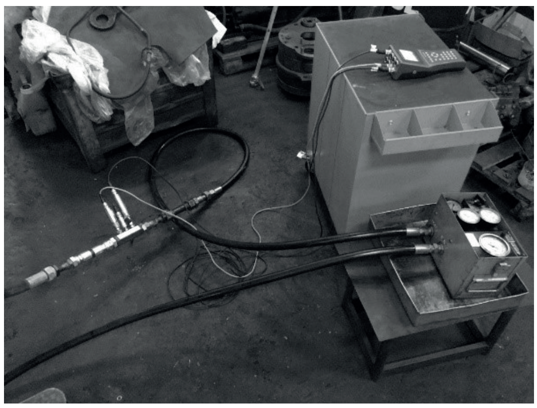

Figure 4. Connection of the OTC H50 analogue meter and the Hydrotechnik digital meter into the examined hydraulic circuit (testing at the beginning of the hydraulic circuit)

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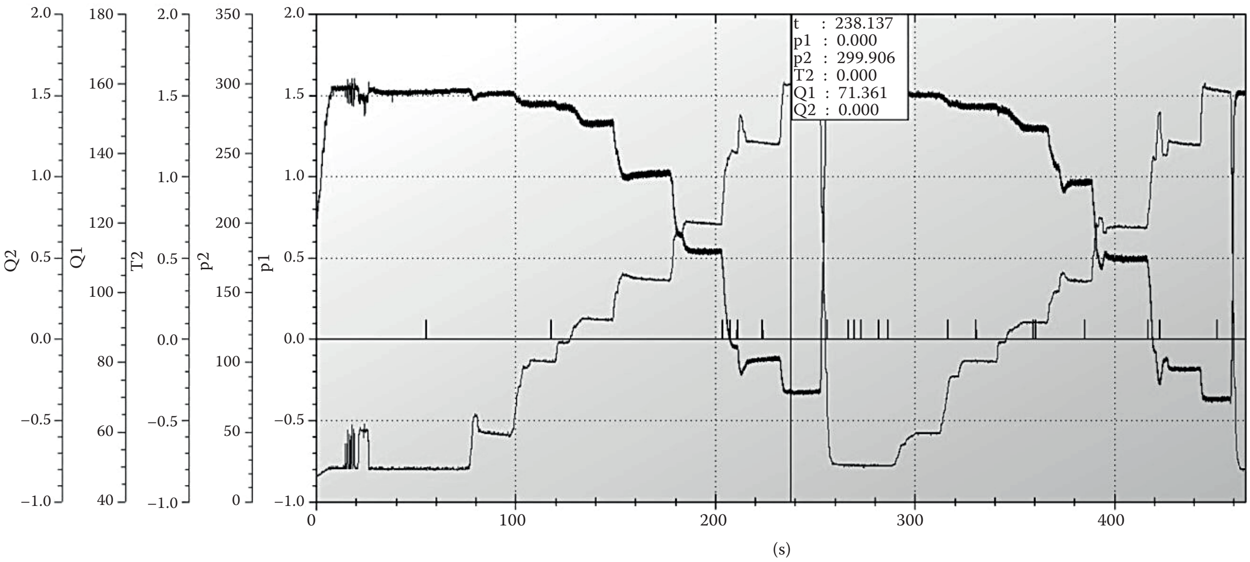

Figure 5. Time dependency graph of the flow rate and pressure of the hydraulic oil in the hydraulic circuit for the telescopic boom stroke (measured on the Bosch Rexroth A8VO107double axial piston pump) using the Hydrotechnik software

fold and the appliance, the combined meters were connected to the hydraulic circuit instead of the appliance instead of the linear or rotary hydraulic motors. The intake hose of the combined meters was connected to the intake line of the appliance and the output hydraulic hose from the analogue meter was connected to the return line from the appliance.

Measurement of the working cycle average time of the UDS 214 excavator. The measurement of the average working cycle time of the UDS 214 excavator was performed for the Class II soil according to the standard ČSN 73 6133:2010. Due to the enormous time utilisation of the provided UDS 214 telescopic excavator required to measure the working cycle, twenty cycles were measured and the average value was calculated from these values. Furthermore, a video recording of this measurement was made for the sake of control and to measure the times of the individual operations of the excavator during the working cycle. When measuring the average time of the working cycle, the procedure was followed exactly according to standard ČSN 27 7003:2012 for the theoretical working cycle of the excavator. Unfortunately, this standard is no longer valid in the Czech Republic and was repealed without any adequate substitution. However, since there is no other theoretical basis for measuring the average duty cycle time of the excavator, it was necessary to proceed with the measurements according to this standard:

– the initial position (the bucket is tilted to the maximum tilt angle and is leaned towards the middle of the depth range by the teeth or the cutting edge on the opposite side of the mining pit),
– digging and rock picking (the bucket must be filled to a nominal volume),
– raising the bucket to the dumping height (minimum 3.5 m),

Table 1. Measured data for the hydraulic circuit of the telescopic boom stroke (measured on the Bosch Rexroth A8VO107 double axial piston pump)

p (MPa)	Q ($dm^3 \cdot min^{-1}$)	t (°C)	n (min^{-1})
0	158	33.6	1 600
5	156	35.3	1 590
10	153.3	36.7	1 585
13	147	38.6	1 578
16	131.5	40.4	1 572
20	111.7	42	1 566
27	77.7	43.5	1 560
30	71.3	44.7	1 556

p – adjusted pressure of the hydraulic oil by the throttle valve of the OTC H50 analogue meter; Q – measured hydraulic oil flow rate by the Hydrotechnik MultiSystem 5060 digital meter; t – measured hydraulic oil temperature by the Hydrotechnik MultiSystem 5060 digital meter; n – measured speed of the John Deere 4045 internal combustion engine by the digital tachometer in the operator's cabin

https://doi.org/10.17221/77/2021-RAE

– swinging of the rotating upper (with a full bucket by 90°),
– soil dumping (at a dump angle of 45°),
– swinging of the rotating upper (back by 90°),
– lowering (setting the bucket back to the initial position) (ČSN 27 7003:2012).

Specified values of the used ISO VG 46 hydraulic oil are:
The density at 15 °C: ρ = 866 kg·m^{-3}
The kinematic viscosity at 40 °C: υ = 45.92 mm^2·s^{-1}

Specified values of the Bosch Rexroth A8VO107 axial piston double pump are:
The pressure efficiency: η_p = 0.94
The volumetric efficiency: η_v = 0.94
The maximum geometric volume: V_g = 2 × 0.107 dm^3
The double pump input power consumption: P = 87 kW

The calculation of the geometric volume of one half of the axial piston double pump is as follows: the calculation is performed for each tested hydraulic circuit and at the beginning and at the end of the given hydraulic circuit (Zhang 2008).

$$V_g = \frac{1000 \times Q}{n \times \eta_v} \tag{1}$$

where: V_g – the geometric volume of the half of the Bosch Rexroth A8VO107 axial piston double pump (dm^3); Q – the measured hydraulic oil flow rate with the Hydrotechnik MultiSystem 5060 digital meter (dm^3·min^{-1}); n – the measured speed of the John Deere 4045 internal combustion engine (min^{-1}); η_v – the volumetric efficiency of the Bosch Rexroth A8VO107 axial piston double pump (%).

The calculation of the theoretical flow rate of the axial piston double pump is as follows (Akers et al. 2006).

$$Q_t = V_g \times n \tag{2}$$

where: Q_t – the theoretical flow rate of the half of the Bosch Rexroth A8VO107 axial piston double pump (dm^3·min^{-1}).

The evaluation of the overall efficiency of the half of the axial piston double pump is as follows (Wood 2005).

$$\eta_{CHG} = \frac{Q}{Q_t} \times \eta_p \tag{3}$$

where: η_{CHG} – the overall efficiency of the axial piston double pump (%); η_p – the pressure efficiency of the axial piston double pump (%).

The calculation of power loss of half of the axial piston double pump is as follows (Gotz 1998).

$$P_{ZHG} = (1 - \eta_{CHG}) \times P \tag{4}$$

where: P_{ZHG} – the power loss of half of the axial piston double pump (W); P – double pump input power consumption (W).

The calculation of the theoretical performance productivity of the UDS 214 Excavator is as follows (ČSN 27 7003:2012).

$$Q_B = \frac{3600}{t} \times V \tag{5}$$

where: Q_B – the basic productivity (theoretical) (m^3·h^{-1}); t – the measured average time of the excavator working cycle (s); V – the bucket volume (m^3).

The calculation of the actual performance productivity of the UDS 214 Excavator is as follows (ČSN 27 7003:2012).

$$Q_A = Q_B \times f_F \times f_O \times f_A \times f_W \times f_B \tag{6}$$

where: Q_A – the actual productivity (m^3·h^{-1}); f_F – fill factor of the bucket according to the rock detachability class (–); f_O – operator qualification factor (–); f_A – swing angle factor (–); f_W – bucket wear factor (–); f_B – the factor of ratio of the UDS 214 excavator bucket volume to the volume of the dump truck load space (–).

RESULTS AND DISCUSSION

For the purpose of the calculations, it was first necessary to measure all the essential geometric dimensions of the given hydraulic circuit (inner diameter of the hoses, the pipes and the length of the lines) for the UDS 214 excavator on which the testing was performed. The individual elements arranged in series were determined from the schemes of the individual hydraulic circuits and after studying the given hydraulic circuits directly on the UDS 214 excavator on which the measurement took place. These elements were entered in a table belonging to a certain hydraulic circuit. Each table always states the name of the element in the hydraulic circuit, the number of pieces, the calculated value of the pressure loss p_z and finally the calculated power loss of the element P_z. It is necessary to determine the pressure losses of all the elements in the hydraulic circuit except for the pump. The pressure losses in the indi-

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vidual elements of the hydraulic circuit, multiplied by a given measured flow rate in a given section of the hydraulic circuit, means the power loss of the element in the hydraulic circuit. It is necessary to calculate the power loss directly for the pump. After summing these individual power losses, we obtained the total power loss of the hydraulic circuit. It was necessary to calculate the pressure losses in the direct line for hydraulic hoses and hydraulic pipes, according to the values of measured hydraulic oil flow rates in the hydraulic circuits of the UDS 214 excavator. Furthermore, it was necessary to calculate the pressure losses in the local resistances of the hydraulic circuits, such as a screw connection (junction of a hose and a steel pipe or a manifold neck) as well as a 90° elbow or a relief valve. For the subsequent calculations of the efficiencies of the individual circuits, it was first necessary to determine the values of the pressure efficiency, geometric volume and input power consumption of the pumps. Moreover, the pressure losses of the Bosch Rexroth 7M8-22 switchboard, which are stated by its manufacturer, were used. From the known values, the theoretical flow rates of the pumps, the overall efficiencies of pumps and, finally, the power losses of the pumps were calculated. When calculating the power losses of the elements, such as the fittings, 90° elbows, etc., which are included in the hydraulic circuit in front of the hydraulic manifold, the calculated pressure losses of these elements were multiplied by the actual measured hydraulic oil flow rate at the double axial piston pump. The pressure losses of the elements behind the manifold were multiplied by the actual measured flow rate at the end of the hydraulic circuit. The total power loss was then determined in the prepared tables for each hydraulic circuit.

The geometric volumes of half of the Bosch Rexroth A8VO107 double axial piston pump were calculated by Formula (1). In this particular case, for the hydraulic circuit of the telescopic boom stroke,

Table 2. Results of the calculation of the total power loss and overall efficiencies for the boom stroke hydraulic circuit – before the innovation of the hydraulic system of the UDS 214 excavator

Element in the circuit	No. of elements	p_z (kPa)	P_Z (W)
Bosch Rexroth A8VO107 axial piston double pump	1	–	5 072
Hose Ø 25/660 mm	1	4.21	5.45
Fitting Ø 25 mm	1	0.31	0.40
Hose Ø 25/1 600 mm	1	10.21	13.22
Control spool	1	20.89	27.05
Bosch Rexroth 7M8-22 open centre control block	1	600	777
Pipe Ø 28/110 mm	1	0.16	0.09
90° elbows Ø 28 mm	5	2.5	1.44
Pipe Ø 28/140 mm	1	0.2	0.12
Pipe Ø 28/60 mm	1	0.09	0.05
Fittings Ø 25 mm	3	0.186	0.11
Hose Ø 25/410 mm	1	1.16	0.67
Pipe Ø 28/1 250 mm	1	1.8	1.04
Pipe Ø 28/110 mm	1	0.16	0.09
Hose Ø 25/1450 mm	1	4.11	2.36
90° elbow Ø 25 mm	1	0.78	0.45
Fitting Ø 25 mm	1	0.062	0.04
Hose Ø 25/1600 mm	1	4.53	2.60
Total power loss P_{ZC} (W)			5 904.28

P_{ZHG} = 5072.1 W: calculated η_{CV} = 86.43%; Q_1 = 1.295 $dm^3 \cdot s^{-1}$; measured η_{CN} = 86.44%; Q_2 = 0.575 $dm^3 \cdot s^{-1}$; p_z – calculated pressure loss of the individual elements of the hydraulic circuit (kPa); P_Z – calculated energy loss of the individual elements of the hydraulic circuit (W); P_{ZC} – total power loss of the given hydraulic circuit (W)

https://doi.org/10.17221/77/2021-RAE

Table 3. Comparison of the total power losses and overall efficiencies of the particular hydraulic circuits – before the innovation of the hydraulic system of the UDS 214excavator

	Before innovation		
	P_{ZC} (W)	calculated η_{CV}	measured η_{CN}
		(%)	
Telescopic boom stroke	5 904.28	86.38	86.37
Turning the swing body	5 926.80	86.38	86.37
Opening and closing the working tool	6 012.47	86.18	86.17
Extending the telescopic boom	4 575.74	89.48	89.49

P_{ZC} – total power loss of the given hydraulic circuit (W); η_{CV} – overall efficiency of the hydraulic systems obtained by calculation (%); η_{CN} – overall efficiency of the hydraulic systems obtained by measurement (%)

the geometric volume of the double axial piston pump was 0.053 dm^3.

The theoretical flow rates of half of the axial piston double pump in the individual hydraulic circuits of the UDS 214 excavator were calculated using Formula (2). For the case of the already mentioned hydraulic circuit of the telescopic boom stroke, the theoretical flow rate for the half of the axial piston double pump was 1.378 dm^3·s^{-1}. The overall efficiencies for half of the axial piston double pump in the excavator's individual hydraulic circuits were evaluated by Formula (3). For the hydraulic circuit of the telescopic boom stroke, the efficiency of half of the axial piston double pump was 88.34%. Finally, Formula (4) was applied to calculate the power losses of half of the axial piston double pump in the individual hydraulic circuits. The pump power loss for the hydraulic circuit of the telescopic boom stroke was 5.072 kW. The determination of the total power loss of the hydraulic system is given in the example of the hydraulic system of the excavator telescopic boom, see Table 2.

Then it was possible to evaluate the overall efficiency of the particular hydraulic circuits determined by the calculations and the measurements. The comparison of the total power losses of the individual hydraulic circuits with the evaluation of their efficiency is given in Table 3.

Subsequently, it was necessary to calculate the operating performance of the given UDS 214 telescopic excavator in its operational state.

Calculation of the performance productivity of the UDS 214 excavator - before the hydraulic system innovation

Specified and measured values. The average time of the working cycle of the UDS 214 excavator were obtained by measurement: t = 25 s with a bucket volume: V = 0.63 m^3. The total average work cycle time of the UDS 214 excavator is composed by the individual measured average operation times, as shown in Table 4. The percentage of the particular average operation times in the total average excavator working cycle time is shown in the pie chart in Figure 6.

The theoretical performance productivity was calculated by Formula (5) which was 90.72 m^3·h^{-1}. The actual performance productivity was evaluated by Formula (6) and the UDS 214 excavator actual performance was found to be 83.81 m^3·h^{-1}. The individual coefficients were determined from the tables of the given coefficients for this calculation.

The UDS 214 telescopic excavator has a performance productivity of Q_A = 83.81 m^3·h^{-1} in the original state of the hydraulic system and in the given operational technical conditions.

Table 4. Average times of the particular actions during the mean working cycle of the UDS 214 excavator

Portion of the excavator working cycle	Measured average time (s)
Filling the bucket	4.01
Lifting the full bucket (at least 3.5 m)	2.21
Swinging with full bucket (swing angle 90°)	6.21
Dumping (dump angle 45°)	4.16
Swinging back with empty bucket (swing angle 90°)	5.76
Lowering (setting the bucket to the starting position)	2.65
Overall average time of the excavator working cycle	25

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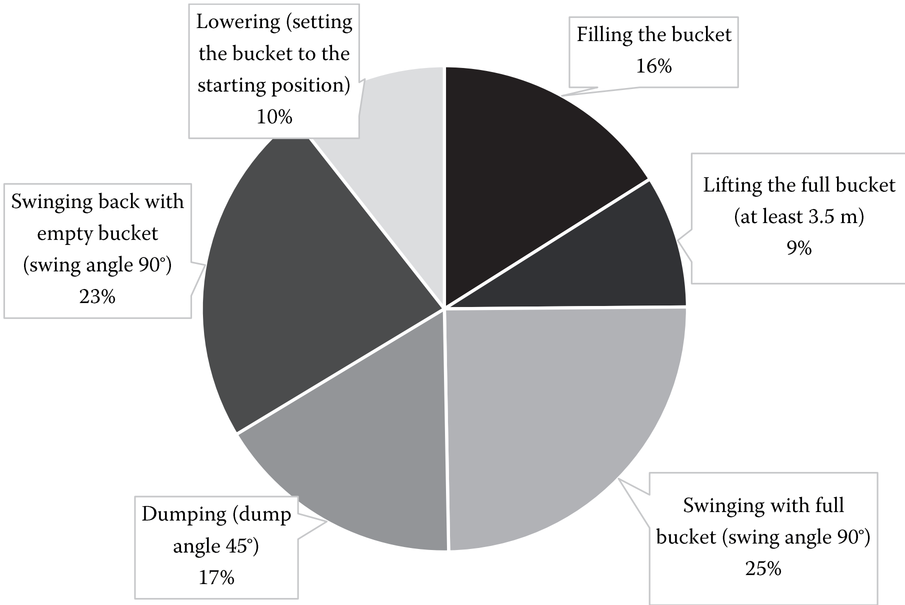

Figure 6. Percentage of the particular actions from the average working cycle time of the UDS 214 excavator

CONCLUSION

The measurement results showed that the hydraulic circuit had the highest total power loss of 6.012 kW with the lowest calculated and measured efficiency in the hydraulic circuit for the opening and closing the bucket. This fact is due to the enormous length of the hydraulic line between the manifold and the hydraulic cylinder to control the opening and closing of the bucket. Taking the fact that the percentage of time of using this hydraulic system is only 33% of the overall measured average cycle time of the UDS 214 into account, it follows that there is not much space for the possible innovation of this hydraulic system, which would lead to a significant increase in the performance productivity of the UDS 214 excavator. From this point of view, it is much more interesting considering the hydraulic circuit for swinging the body of the UDS 214 excavator, which has the second highest power loss, namely 5.926 kW, together with the hydraulic circuit for the stroke of the telescopic boom, which has the second lowest efficiency. As the percentage of using this hydraulic circuit is 48% from the overall average excavator cycle time, there is a great potential for the successful innovation of this hydraulic system which could reduce its total power loss. It would also reduce the average working cycle time of the UDS 214 excavator during soil mining and loading, thus creating a substantial increase in the performance productivity of the UDS 214 excavator. This assumption is followed by the fact that when measuring the average cycle time of the UDS 214 after the innovation of the hydraulic circuit for excavator swinging the body, the excavator must be operated by the same or a more experienced operator so that the measured excavator results would not be subjectively affected.

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