Experimental investigation of the influence of fluid viscosity on the efficiency of a crawler excavator

Deuster Sebastian, Schmitz Katharina*

Institute for Fluid Power Drives and Systems (ifas), RWTH Aachen, Germany
E-mail: sebastian.deuster@ifas.rwth-aachen.de, katharina.schmitz@ifas.rwth-aachen.de

Abstract

In view of decreasing energy resources and the rising problems associated with CO2 emissions and global warming, there is a strong interest in reducing the fuel consumption of machines in all sectors. Manufacturers of mobile machinery, such as hydraulic excavators, are also striving to develop increasingly efficient machines. Triggered by this development trend, the power density of hydraulic systems and their components continues to increase. This results in higher pressures, temperatures and lower oil volumes in the system. As a result, the hydraulic fluid used is subject to greater thermal stress and the systems thermal properties are becoming increasingly important. Further developments of tribological systems, for example in hydraulic displacement units, also create new demands on the hydraulic components and fluids. The fluid properties in particular are increasingly coming to the fore. For this reason, the content of this paper is to consider the influence of the hydraulic oil's viscosity on the efficiency of a crawler excavator and to present the viscosity dependent losses of valves and pipe/hose lines. To this purpose, experimental tests are discussed by regarding viscosity related energy losses of the hydraulic system of a crawler excavator. Therefore the results of experimental tests of a gravel cycle at different temperatures will be discussed. The results are divided into different types of energy losses. Finally, a hypothesis can be made about the dependence of the viscosity of the hydraulic fluid on the efficiency of the system.

Keywords: crawler excavator, temperature distribution, fluid viscosity, energy efficiency, energy losses

1 Introduction

In consideration of decreasing energy resources and the increasing problems associated with CO2 emissions and global warming, there is a growing interest in reducing the fuel consumption of machines in all sectors. Triggered by this trend, the power density of hydraulic systems and their components continues to increase. This results in higher pressures and temperatures as well as lower oil volumes in the system. As a result, the hydraulic fluid is subjected to higher thermal and mechanical loads. Due to this, knowing the thermal properties of the system becomes increasingly important. High temperatures in hydraulic systems favor the mechanisms of oil ageing, hence reducing the service life of hydraulic components and precision of the machine and the intervals of changing hydraulic oil. Additionally, an important aspect is, that the temperature of the fluid in various components of a hydraulic system influences its viscosity. This affects the power losses in the hydraulic system, including hydraulic-mechanical losses such as pipe friction and volumetric losses such as leakage losses. In a complex hydraulic system such as that of a mobile excavator, these types of losses only occur simultaneously.

To evaluate the economic efficiency of a machine, the total costs of ownership (TCO) are used. The TCO is divided into costs for the acquisition and costs for the operation of the machine. An energy-efficient hydraulic system has several advantages. For example, on the one hand, less energy is needed to operate the system, and on the other hand, less power loss has to be dissipated in the form of heat. The losses occurring in a hydraulic system can basically be divided into two types, component-dependent and system-related losses.

In mobile machinery, there is a variety of concepts for increasing functionality and energy efficiency. The component efficiencies and especially the systems architecture and its intelligent control have a significant influence on the overall efficiency of a drive system.

The concepts currently used and developed can be divided into two main categories. On the one hand, concepts that concern the systems structure and on the other hand those addressing intelligent controls. **Figure 1** shows several concepts used in hydraulic systems of mobile machinery. One example for intelligent control concepts is independent metering, where independent valves realize the inlet and outlet control edges. By decoupling the control edges, volume flow paths can be selected much more flexibly and recuperative operations can be realized.

Regarding the systems structure, in addition to the development of various hybrid structures, a trend toward the electrification and digitization of mobile machines can be observed. [1]

Furthermore, there are concepts combining multiple conceptions. The STEAM concept [2] for example uses a hydraulic system combining different concepts regarding the system structure to improve the energy efficiency of a mobile excavator.

Figure 1: Concepts for increasing efficiency

This paper discusses the potential of the influence of fluid viscosity on the efficiency of a mobile working machine. The temperature distribution of the hydraulic oil during operation in a mobile machine is within a wide temperature range. In addition to the temperature deviation between the different components, temperature variations over time occur in various working applications. This paper deals with the viscosity dependent losses of valves and pipe/hose lines of a crawler excavator. For this purpose, spatially resolved temperatures and pressures in the hydraulic system during the execution of a dig and dump cycle are experimentally obtained and analyzed with respect to their influence on the efficiency of the excavator. The dig and dump cycle represents the work task mainly performed during operational life of a crawler excavator.

2 Power dissipation of mobile machines

The energy conversion and conduction in components of hydraulic systems is usually accompanied by losses. Losses are caused by individual hydraulic system components. According to these component-dependency, the losses can be divided into volumetric and hydraulic-mechanical losses. These types of losses are influenced by the viscosity of the used pressure medium, moreover they are dependent on the operating point. The selection of the main circuit configuration of the hydraulic system determines the system dependent losses for example the throttling losses in a system with resistance control.

According to [3], the power loss P_{loss} results from the drive power P_{input} and the efficiency η_{total} of the hydraulic system (Eq. (1)). It is assumed that the total power dissipation of the hydraulic system can be determined by the difference between the total power consumed by the hydraulic system and the total power used by the hydraulic system to perform work. The only heat that gets into the hydraulic oil comes from the power losses of the hydraulic system. [4]

$$P_{loss} = P_{input}(1 - \eta_{total}) \tag{1}$$

[5] presents the losses of a mobile excavator in relation to the input fuel energy. The results based on an experimental performance of a 90° digging cycle. In addition to the losses in relation to the total input power, the individual efficiencies of the various subsystems are also discussed. In **Figure 2** the results of the analysis are presented by means of a Sankey diagram. Based on the fuel energy fed into the system, only about 7.4 % is

converted into useful work for the digging cycle. Major losses occur when the chemical energy of the fuel is converted into mechanical work by the internal combustion engine (ICE). This mechanical energy is afterwards transferred to the hydraulic pump via a shaft, converting this mechanical power into hydraulic power. The control system in this case consists of valves, pipes and hoses. The work provided is implemented by the actuators. The efficiencies of the individual subsystems in relation to the energy supplied as well is shown. It is noticeable, that the individual efficiency of the control system is below the efficiency of the combustion engine. The losses occurring in the control system include, in addition to system-dependent losses, losses at valves and in piping systems, whose viscosity dependence is investigated in this paper. In the following, the viscosity dependence of volumetric and hydraulic-mechanical losses occurring in an excavator's hydraulic system will be discussed.

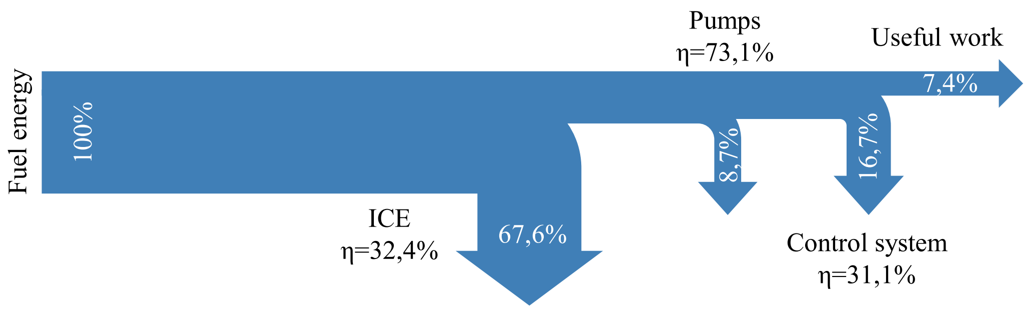

Figure 2: Energy losses of a mobile excavator [5]

2.1 Volumetric losses

Volumetric losses occur in form of internal leakage. In this case hydraulic fluid flows from the high-pressure side to the low-pressure side through function-related gaps between the components of the hydraulic unit that move relatively to each other. These gap losses occur, for example, at the ring surface of a piston-bushing contact of a pump or motor. Laminar flows through circular cross-sections are characterized by equation (2), the Hagen-Poiseuille equation. It describes the loss volume flow Q, r is the radius of the gap and l the respective length. η describes the dynamic viscosity of the fluid. Δp is the pressure difference occurring across the gap. [6]

$$Q = \frac{\pi \cdot r^4}{8 \cdot \eta \cdot l} \cdot \Delta p \tag{2}$$

At a high viscosity η, the volumetric losses decrease due decreasing gap volume flow. Conversely, filling losses occur because the pump sucks in the liquid less well due to the high friction losses. To avoid cavitation, the so-called permissible starting viscosity must not be exceeded. Cavitation can cause damage to the pump and leads to filling losses, which increase the volumetric losses. [6]

2.2 Hydraulic-mechanical losses

Friction losses in tribological contacts of hydraulic components and in flowing hydraulic fluids are called hydraulic-mechanical losses. A high viscosity, for example at low temperatures, leads to an increase in friction in the fluid. The hydraulic-mechanical losses occur in fluid-flow areas of the system, for example in hydraulic resistances. These flow losses cause pressure losses and a heating of the fluid. High flow velocities due to large volume flows and small flow cross sections increase the pressure losses. Furthermore, the flow losses depend on the geometry of the components by which fluid is leaded. It is taken into account by corresponding resistance coefficients and characteristic curves when a hydraulic system is designed.

At low relative velocities between two surfaces of tribological contacts shear stress occurs. As well as when the viscosity of the pressure fluid at high temperatures falls below the minimum required for the application, no load-bearing lubricating film can be generated and mixed friction occurs between the moving components. [7] Operation in the area of mixed friction can be minimized and the occurrence of mixed friction prevented. Thus, the viscosity influences the efficiency of the hydraulic system as well as the service life of the components and the availability of the entire hydraulic system.

2.3 Energy losses of a hydraulic system

Outgoing from the main circuit of the hydraulic system of the excavator, the power losses of the system can be divided to the different system components. **Figure 3** shows the various main loss mechanisms of a hydraulic system. Energy losses occur mainly in pumps and motors, valves and piping. Further losses are caused, for example, by filters and coolers. The energy losses shown in Figure 3 usually occur in the system in a superimposed form. The three main loss mechanism and their dependency on the fluid viscosity are explained in more detail below.

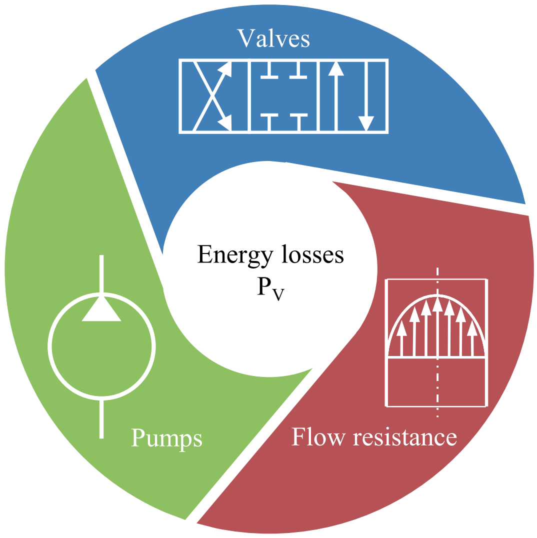

Figure 3: Energy losses

2.3.1 Pumps

Hydraulic displacement units can be characterized by volumetric, hydraulic-mechanical and total efficiency. All leakage losses in hydraulic units are summarized in the form of volumetric losses. Leakage losses are mainly caused by two fluid volumes with different pressure levels. This is the case, for example, in contacts with narrow gaps. In addition to leakage losses, hydraulic-mechanical losses occur. These losses result in the theoretical drive torque being smaller than the torque actually required. The mechanism include mixed friction, as well as pressure losses due to friction shear and throttling at cross-sectional changes. [6]

The dependence of the volumetric and hydraulic mechanical efficiency of various pump designs on the fluid viscosity was investigated by [8]. In this context, various hydraulic oils with different base oils were measured with regard to their influence on efficiency of the displacement units. A dependence of the fluid viscosity on the efficiency was found, which led to an increase of up to 3 % of the overall efficiency compared to conventional HLP mineral oils. This behavior was observed for fluids with a high viscosity index. Due to the extensive investigations of this research work, these losses are not part of the considerations of this paper. The investigations of this paper focus on the viscosity-dependent parasitic losses in valves and pipe/hose lines.

2.3.2 Valves

Energy losses in directional control valves can be divided in terms of losses due to the separation of the flow from the control edge in the turbulent range and into losses due to wall friction. For a given nominal size, this pressure loss depends on the size of the volume flow, on the design and on the operating viscosity. Approximately, directional control valves can be considered as an orifice resistance. This approach of determine the losses caused by valves excludes the consideration of viscosity on the losses via the slide valve.

In the hydraulic system of a mobile machine test excavator, spool valves are used to direct the volume flow to the desired actuator. Each hydraulic consumer is controlled by its own spool valve, so that the power losses can be assigned to the individual actuators.

2.3.3 Flow resistance

On the one hand flow resistance in straight pipes is caused by friction of the fluid near the wall and on the other hand by the internal friction of the fluid itself. The pressure losses Δp_F in pipelines can be described with equation (3). The resistance coefficient λ is function of the Reynolds number, which has a viscosity dependence. A high viscosity at low temperatures leads to increasing friction in the fluid. Hydraulic-mechanical losses increase in all areas of the system where a volume flow exists, for example in hydraulic resistances. These flow losses cause pressure losses and heating of the fluid. High flow velocities v due to large volume flows and small flow cross-sections d increase pressure losses. Furthermore, the flow losses depend on the length l of the pipe. When designing a hydraulic system corresponding resistance coefficients and characteristic curves are taken into account.

$$\Delta p_F = \lambda \cdot \frac{l}{d} \cdot \frac{\rho}{2} \cdot v^2 \quad (3)$$

In the hydraulic system of mobile machines, flow resistances result from the routing of the fluid to the consumers, from deflections of the fluid flow in the system and from rapid changes of cross-sections.

3 Test setup

The dependence of viscosity on the energy losses of a mobile machine will be discussed in this paper. The following chapter presents the test setup for the corresponding measurements.

3.1 Test excavator

The table in **Figure 4** shows the specifications of the test excavator. The crawler excavator *EC18D* [9] manufactured by *Volvo CE* belongs to the class of compact excavators. It is driven by tracks. In addition to the linear actuators, which are required for digging, the excavator has actuators for raising and lowering the blade, as well as for adjusting the travel width. The test excavator is equipped with hydraulic-mechanical one-circuit load-sensing system, which is controlled by electro-hydraulic pilot valves instead of the usually implemented hydraulic joysticks. An electro-hydraulic prototype control allows for completely automated and reproducible digging cycles. In addition, joystick signals are processed electronically to allow for manual operation. The main pump supplies the respective hydraulic actuators. During operation, the highest load pressure in the system is detected and the displacement volume of the pump is adjusted accordingly. The total weight of the machine is 1790 kg. The gross power of the diesel engine is 12 kW. The nominal bucket filling level is 36 liters. The hydraulic tank consists of 15 liters and the hydraulic system consists of 21 liters. The hydraulic system is operated with a hydraulic oil of class HLP 46.

Specification	Value	Unit
Weight	1790	kg
Gross power	12	kW
Nominal bucket filling level	36	l
Hydraulic tank	15	l

Figure 4: Specifications Excavator [9]

3.2 Test Cycle

The typical construction site activity of an excavator consists of the sum of four different load cycles. These include “grading” with a ratio of 10 %, “driving” with 20 %, “idling” with 30 % and the load cycle “digging” with 40 % of the total task. [10] The executed test cycle is based on the test cycles defined by the Japan Construction Mechanization Association (JCMA) [11] for determining the fuel consumption and energy losses of hydraulic excavators, for example. In the test a 90 degree dig and dump cycle is performed. The dimensions and the execution of the cycle is shown in **Figure 5**. With the nominal filling quantity of 36 liters and the moved basalt chips (8-16 mm grain size), the weight per transshipment is about 54 kg. After unloading the bucket the upper carriage is swung back as well as the excavator arm is lowered to the starting position. The position of the excavator remains unchanged during the entire process. The tests are carried out in an air-conditioned laboratory so that the comparability of the measurements results is ensured. In addition, the ambient temperature is recorded.

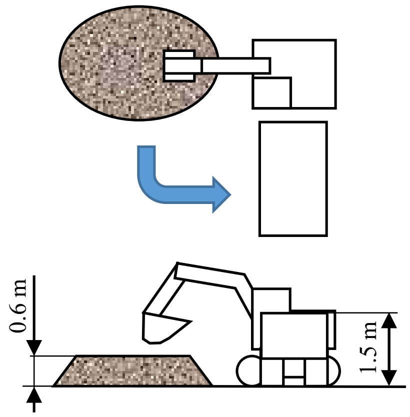

Figure 5: Dimensions dig and dump cycle

3.3 Measuring equipment

The test excavator is equipped with appropriate measurement technology to record the relevant data. **Figure 6** shows the main circuit of the hydraulic system with the installed sensors. The main circuit generally consists of the three linear actuators: The boom, stick and bucket cylinders. The swivel motor correspondingly drives the slewing gear for swiveling the upper carriage. The diesel engine drives the main hydraulic pump. Other actuators, such as the track motor, are not considered in this study because they are not activated during the executed test cycle.

3.3.1 Pressure

Pressure sensors are integrated in the system to identify the occurring losses. The main control valve consists of several proportional valves that control the volume flow to the individual actuators. Pressure losses that occur through the control edges of the valves are measured. Due to the system design of the load sensing system, the pump supplies the volume flow which provided the pressure of the major consumer in the system. This results in system-related losses at the other consumers due to the throttling of power in the direction of the tank.

The fluid is conducted to the respective consumers via hoses. Flow losses occur due to wall friction and deflections. To record the differential pressure, the test object is equipped with pressure sensors on the piston and rod side of the hydraulic cylinders. In case of the swivel motor, the differential pressure is not recorded because the hose connection from the main control valve to the motor is relatively short in comparison to the other hoses.

3.3.2 Temperature

Due to friction and throttling losses that occur in the hydraulic system of the excavator, energy is dissipated in the form of heat. This heat is transferred to the components of the system and to the hydraulic fluid, which effects a temperature increase as a result. The heating of the fluid is described by equation (4).

$$\phi = m \cdot c \cdot \Delta T \tag{4}$$

The fluid of mass m with heat capacity c heats up by temperature ΔT when it absorbs heat Φ. Part of the absorbed heat is dissipated to the environment via the hydraulic fluid and the surfaces of the tank and hydraulic lines, through heat conduction and convection. After reaching a steady-state temperature ($\Delta T = 0$), the entire power loss is dissipated to the environment in the form of heat.

The temperature is recorded at the throttling points. At these points in the system, high temperatures occur locally in the fluid, which are caused by friction. For this purpose, thermocouple is adapted into the center of the fluid flow. The identification of the temperature enables to infer the viscosity of the fluid, which is significant for the power calculation. According to Ubbelohde-Walther (Equation 5), the change in viscosity is expressed using an empirically determined equation. [12] The pressure dependence is neglected in the calculations due to the limited system pressure of 170 bar.

$$m = \frac{lglg(\upsilon_1 + 0{,}8) - lglg(\upsilon_2 + 0{,}8)}{lgT_2 - lgT_1} \tag{5}$$

Furthermore, the minimum viscosity in the hydraulic system is determined. Of particular importance is the viscosity of the hydraulic oil at the tribological contacts of the main pump. To record the leakage oil temperature, the temperature is recorded at the outlet of the leakage oil flow. If the viscosity of the hydraulic oil falls below a critical point, the parts moving relative to each other can no longer be sufficiently separated and thus there is no longer a lubricant film capable of bearing loads. The tribological system is thus operating in a state of mixed friction which leads to wear the component. In addition, the tank temperature and the ambient temperature are recorded.

3.3.3 Volume flow

The volume flow of the actuators is recorded in order to calculate the power loss occurring in the hydraulic system. The travel path of the linear cylinders is recorded by means of displacement transducers. The angular position of the upper carriage is recorded via a rotary encoder. The volume flow of the main pump is determined via a screw spindle counter.

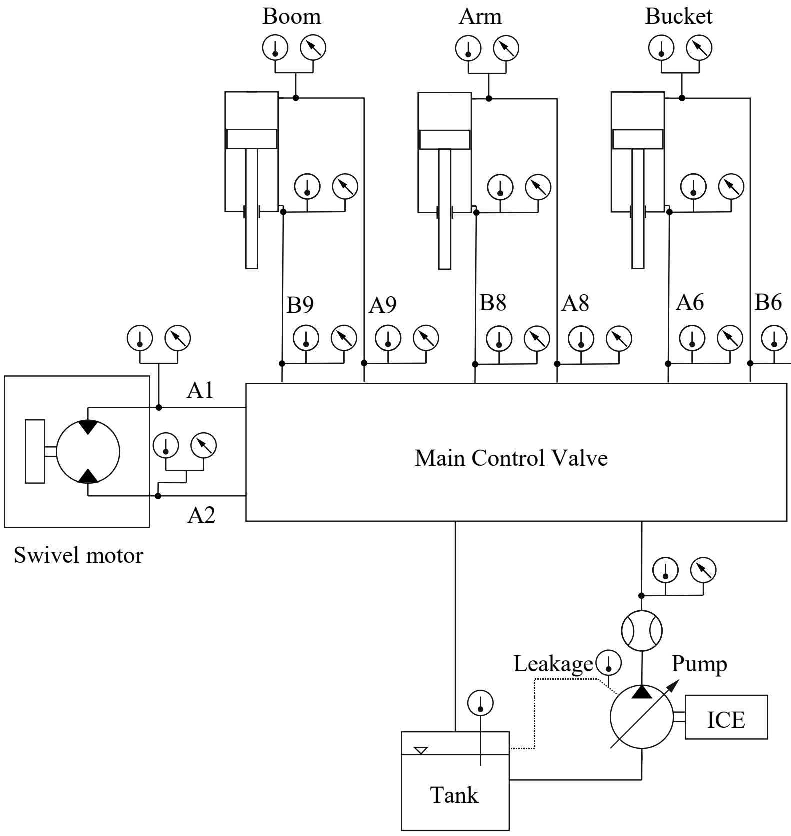

Figure 6: Main Hydraulic Circuit Excavator

4 Results

In this chapter, the results of experimental investigations will be regarded with regard to the test cycles, the pressure and temperature distribution and the dependency of the viscosity on the efficiency of the excavator. Efficiency considerations are based on energy losses of the parasitic resistances through the main control valve and the connecting lines to the actuators.

4.1 Test execution

The test sequence described in chapter 3.2 is carried out in the laboratory at the *Institute for Fluid Power Drives and Systems* (*ifas) of RWTH University Aachen*. The position of the bucket tip x_{tip}; y_{tip}, the rotation angle φ and the cycle time t act as reference variables. The test cycle is repeated until comparable cycles regarding the reference variables can be identified. **Figure 7** shows the geometric coordinates of the excavator with respective designations. The first test is carried out after a cold start and assessed based on the power losses in the main circuit of the hydraulic system. This test is repeated with one cycle in the warmed-up condition and the respective results are compared. For reaching the warm cycled condition, the test cycle is repeated until the change in tank temperature before and after the cycle is stationary. This ensures that the maximum temperatures during the execution of the test cycle are reached.

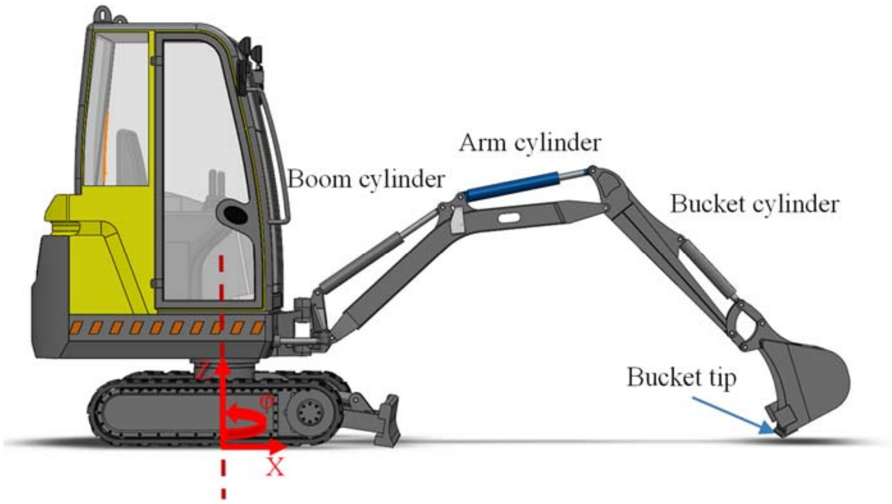

Figure 7: Geometrical description

4.2 Pressure distribution

The measurement of the pressure distribution over a dig and dump test cycle serves as the basis for the efficiency assessments of the two cycles. Thus, the energy loss at chokes and resistors can be calculated. The pressures are recorded according to the measuring points shown in Figure 6. The measured values are recorded over the entire cycle at a frequency of 1000 Hz. The cycle can be divided in four different sections shown in **Table 1**.The exact procedure of the test cycle is described in chapter 3.2. **Figure 8** shows the pressure distribution during the cycle recorded at the actuators. The pictograms illustrate the operations performed by the excavator during this period. The designations can be seen in Table 1.

Table 1: Division of the cycle

Section	Process	Time range
1	Digging	0 – 23 s
2	Rotation to unload	23 – 32 s
3	Unload	32 – 43 s
4	Rotation to the end	43 – 52 s

Before starting swiveling at a cycle time of approximately 17 seconds, a rise in the pump pressure p_Pump to a maximum of 170 bar can be seen, without a consumer requesting this pressure. This can be explained by the corresponding actuators moving at a low speed in the digging process. The variable displacement pump possess a minimum displacement degrees and thus continues to deliver a minimum volume flow of 20 l/min, which is, however, only used to a small extent by the actuators. There is a system-dependent power loss occurs in this section of the cycle noticeable.

The high pressures of the boom (A9/B9) or bucket (A6/B6) cylinder result from holding the excavator arm´s load and the bucket loaded with grit. The load acting on the arm cylinder (A8/B8) is to a large extent absorbed by the joints of the excavator and due to this, the pressure prevailing in the boom cylinder during load holding is considerably lower. At this load, the flow to the pump is correspondingly closed by the valve spool. There is a slight oil leakage from the pressure line to the tank. The significant sections of the cycle for energy loss calculation result from the corresponding work performed by the actuators and are explained further in the chapter **4.4.**

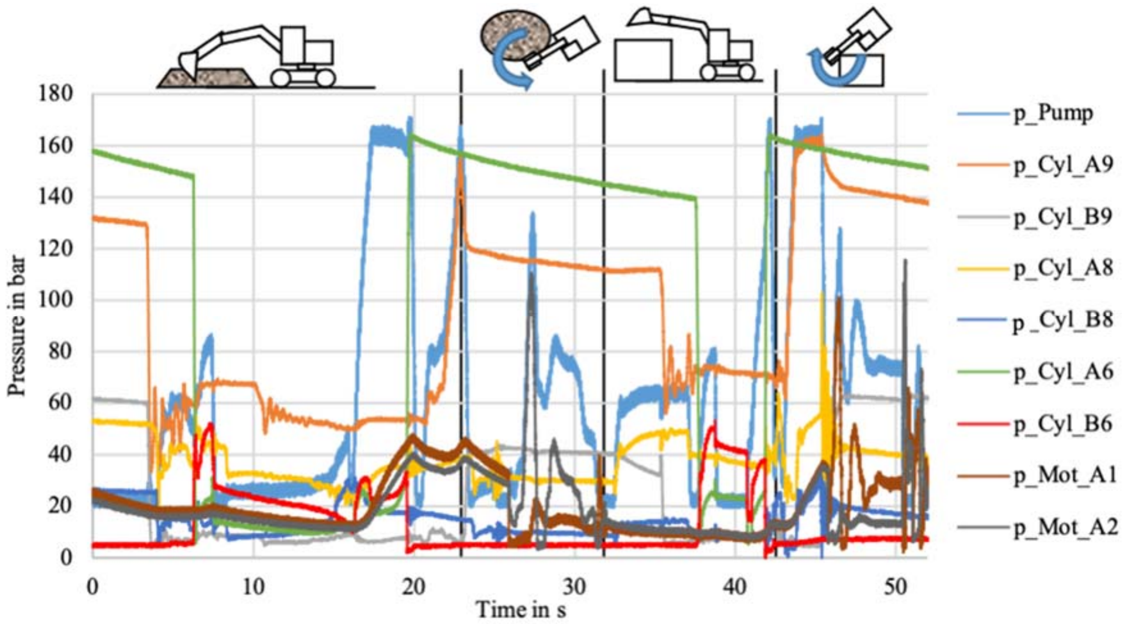

Figure 8: Pressure curves during the dig and dump cycle

4.3 Temperature distribution

The temperature distribution in the hydraulic system of the excavator during the cycle provides information about throttling points and relevant heat emissions in the hydraulic system. During the cycle, temperature distribution within the main circuit of the excavator was recorded. Based on this, the viscosity at relevant throttling points can be determined. The measuring points are located as shown in Figure 6.

An important measuring point is the tank´s temperature for determining the inlet viscosity. **Figure 9** shows maximum recorded temperatures at respective measuring points of the regarded hydraulic system. The red numbers represent temperatures of the warmed-up cycle and the blue numbers those of the cold started cycle. In both cases, an ambient temperature of 23 °C was measured. On average, the temperatures are 14.21 °C apart. It should be noted that these are maximum temperatures, which are shown in the figure regardless of the time of their occurrence during the cycle.

Maximum temperatures in the respective cycle occur in both cycles on the piston side of the arm cylinder. This occurs during the retraction of the arm after unloading the grit. In this case, due to the rapid acceleration of the cylinder, friction of the fluid occurs on guide rings and seals of the cylinder, leading to a corresponding increase in temperature in the fluid.

Furthermore, minimum viscosity of fluid during the cycle can be determined from the maximum fluid temperatures in the hydraulic circuit. A minimum viscosity of 24.2 mm^2/s results for the warm cycle and a viscosity of 38.9 mm^2/s for the cold one.

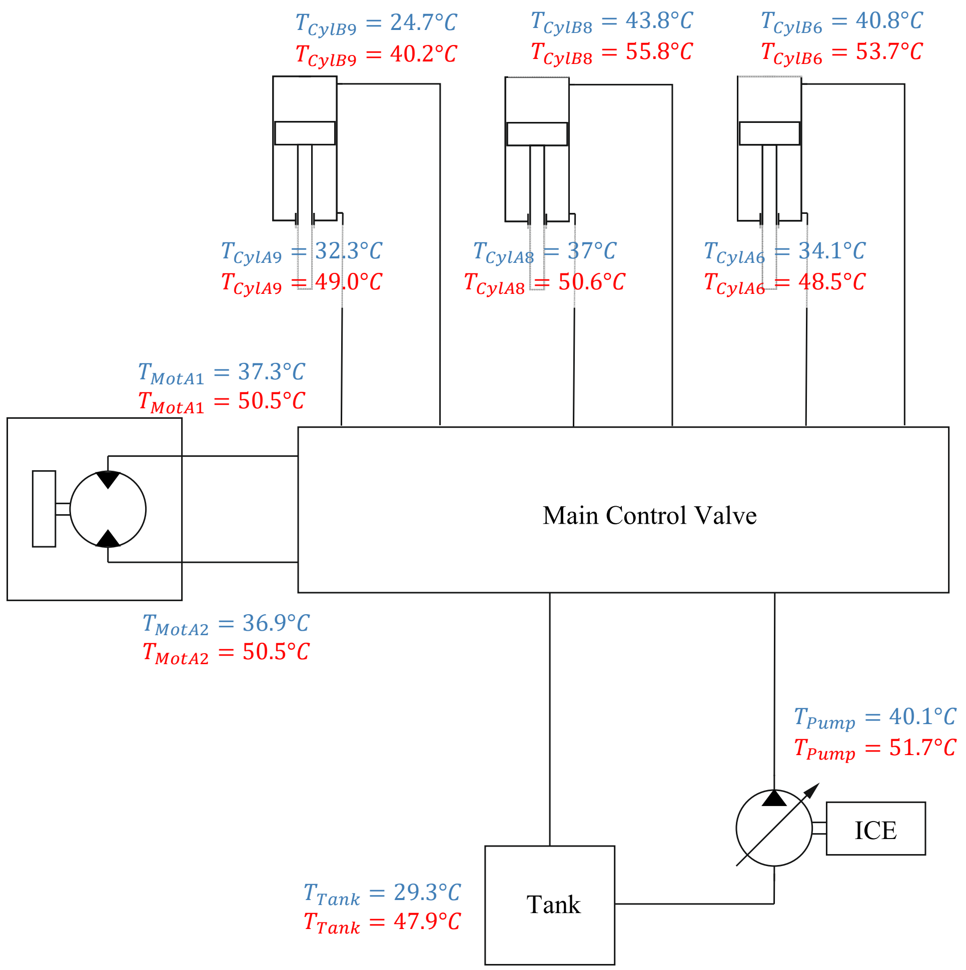

Figure 9: Max. Temperature during cycle

4.4 Efficiency

For efficiency analysis, various individual operation sectors of the cycles are considered in which the respective actuators are operated. These sectors are shown in **Table 2**. Only functions are considered in which the pump supplies the actuators with hydraulic power. The joystick is therefore deflected to its maximum so that the valve gate is fully open. No pressure is released via the load sensing system. At the operating points considered, the pump only supplies its controlled actuator. System-related losses are therefore not included in the calculations.

In the considered sectors, the respective considered consumer represents the highest consumer pressure in the system and the displacement volume of the pump is adjusted accordingly. In case of the boom cylinder, this involves functions that lift the excavator´s arm. In the case of the arm, this is an analogous situation when the arm is extended. In section of the bucket cylinder, the operation sector in which the bucket is opened against weight force is considered. And the case of the swivel motor, the rotations are the focus of the considerations.

Not considered are tasks when the arm segment is lowered. Here hydraulic power is throttled accordingly through the valve's gate towards the tank. In these cases, it is hydraulic power generated by weight force of the dredging arm and thus not provided by the main pump of the hydraulic system. Thermal power generated by the throttling transferring with the fluid into the tank.

Table 2: Operation sectors

Actuator	Sector 1	Sector 2
Boom (A/B 9)	Lifting after digging	Lifting after unload
Arm (A/B 8)	Extend to the digging	Extend for unloading
Bucket (A/B 6)	Open to digging	Open to unload
Swivel Motor (A1/A2)	Rotation to unload	Rotation to the end point

Figure 10 shows the position of energy losses considered. For each sector, energy loss from the main pump via spool of the main control valve ($E_{loss,M}$) and the energy losses via the hose line from main control valve to respective actuators are calculated ($E_{loss,H}$). For this purpose, the average value of power losses over time steps are identified and multiplied by duration of the respective sector.

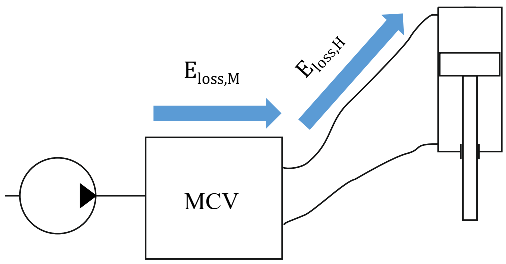

Figure 10: Position of energy losses

Table 3 shows results of the calculations for the cold and warm cycles. The column with index "*M*" (MCV) stands for losses from the pump via the respective spool of the main control valve. ($E_{loss,M}$ in Figure 10). The column with index "*H*" (Hose) represents losses from the main control valve to the respective actuator ($E_{loss,H}$ in Figure 10). The column with index "*H*" (Hose) represents losses from the main control valve to the respective actuator. Losses from the main control valve to the swivel motor and the main control valve to the boom cylinder were not taken into account in the calculation due to relatively short line lengths to those of the other actuators.

Table 3: Energy losses in J

	Boom				Arm				Bucket				Swivel motor			
	Sector 1		Sector 2		Sector 1		Sector 2		Sector 1		Sector 2		Sector 1		Sector 2	
	M	H	M	H	M	H	M	H	M	H	M	H	M	H	M	H
Cold cycle	116	--	25	--	529	48	531	49	554	324	241	81	282	--	273	--
Warm cycle	77	--	32	--	235	13	516	52	510	181	168	70	236	--	218	--

The data shows differences between the two regarded cycles. The largest losses are caused by the arm and bucket cylinders. This is due to relatively short travel length of the boom in relation to the other two linear actuators during the test cycle. The ratio is about 36 % for the boom and 30 % for the bucket cylinder. Furthermore, arm and bucket cylinders perform more dynamic tasks in the digging cycle than the boom cylinder, resulting in larger volume flows.

With exception of lifting the arm by retracting the arm cylinder after unloading the grit (arm/sector 2/H), the energy losses via respective hose lines (H) in the warmed-up cycle are lower than those of the cold cycle. On average, the energy losses are 46.5 J lower in the warmed-up cycle.

Losses via the main control valve (M) are also lower in warm condition of the hydraulic system for all actuators except for sector 2 of the boom cylinder (boom/sector 2/M). On average, the energy losses of the warmed-up cycle are 69.88 J below those of the cold cycle. A large deviation can be observed especially during the first retraction of the arm cylinder. This can be explained by the fact that the strokes of the two cycles differ by 50 mm in this case. If this factor is excluded from the analysis, the hot cycle is on average 37.86 J lower than the cold cycle regarding the energy losses from pump via the main control valve during the test cycle.

Table 4 shows the energy provided by the pump for both cycles considered, as well as the ratio of the sum of the losses from Table 3 to total energy of the pump during the cycle. The indices of Table 3 are also used in this table. The energy provided by the pump is 7.6 % higher in the cold cycle than in the warmed-up cycle. The amount of percentage differences in the ration of losses regarded to total energy provided by the pump point out the influence of fluid viscosity on the energy consumption of the hydraulic system of the mobile machine.

Table 4: Contribution of the energy losses to the total energy

Cycle	Pump	M	H
Cold	174340 J	1.46 %	0.29 %
Warm	161056 J	1.24 %	0.20 %

The percentage share of losses considered in total energy provided by the pump can partly be explained by inefficient control of the pump. The pump delivers a minimum volume flow of 20 l/min at minimum deflection (Chapter 4.2). This means that a not inconsiderable amount of energy is throttled back to tank due to the system structure. In addition, only individual operations of the cycle are included in the considerations. Furthermore, it is a cycle which is carried out at a relatively slow speed. With higher converted volume flows of the consumers, higher losses and a proportionally greater viscosity dependency of losses can be expected.

5 Summary and Conclusion

Current research and development is aimed at developing new concepts to increase the efficiency of mobile machines. Despite innovative developments in systems design and intelligent controls, a significant portion of the energy entering the hydraulic system is dissipated. A proportion of losses are due to volumetric and hydraulic mechanical losses. These losses depend, in part, on the viscosity of the hydraulic fluid used. In this paper, the influence of viscosity-dependent losses of a crawler excavator is considered. A central aspect of the considerations are the viscosity-dependent losses via the main control valve and the hose lines to the actuators. The joystick is therefore deflected to maximum so that the valve gate is fully open. No pressure is released via the load sensing system. At operating points considered; the pump only supplies the controlled actuator. System-related losses are therefore not included in these calculations. The focus of the considerations are two dig and dump cycles at different fluid temperatures. For this purpose, the cycle was considered once at a cold start and in a warmed-up condition. Energy considerations refer to working operations of the machine, in which the pump supplies the respective consumers. Relevant operations were considered for each actuator. Energy consumption for the operations was then determined for both cycles. This enables showing the dependence of the viscosity on energy losses of the operations. The shares of the considered losses in the respective total power are nearly identical. The percentage of the considered energy losses (MCV and hose) on the total energy input of the pump is 0.31 % (=$(M_{Cold} + H_{Cold}) - (M_{Warm} + H_{Warm})$) higher in the cold cycle than in the hot cycle (Table 4). The determined values show a dependence of viscosity on the power loss of the hydraulic system of mobile working machine of approx. 1.6 % in average. The considerations show the relevance of the allowance of the fluid properties in the design and in the efficiency of hydraulic machines. Furthermore, the observations indicate that the viscosity dependence of the parasitic losses considered is present to a relatively small extent. Former research works showed that the influence of fluid viscosity on pump efficiency has an influence of up to 3 %. This indicates that, in case of efficiency savings through adjustment of the fluid viscosity, the pump in particular shows an important role, in addition to the losses in valves and pipe or hose lines. An approach of the targeted adaptation of the fluid viscosity to the conditions of the individual hydraulic system thus offers potential for further considerations. Further researches of the influence of fluid properties on the energy losses of hydraulic systems are planned with regard to the viscosity index. In this context, bio based hydraulic oils, which naturally have a higher viscosity index than mineral oils, are to be considered.

Nomenclature

Designation	Denotation	Unit
c	Heat capacity	J/(kg·K)
d	Diameter	m
$E_{loss,H}$	Energy loss hose	J
$E_{loss,M}$	Energy loss main control valve	J
η	Dynamic viscosity	Pa·s
η_{total}	Overall efficiency	-
l	Length	m
λ	Friction coefficient	--
m	Fluid mass	kg
m	Directional constant	-
υ	Kinematic viscosity	mm^2/s
p	Pressure	Pa
Δp_F	Pressure difference friction	Pa
P_{loss}	Power loss	W
P_{input}	Input power	W
ϕ	Heat	W·s
Q	Volume flow	l/min
r	Radius	m
ρ	Density	kg/m^2
T	Temperature	K
v	Velocity	m/s
x_{tip}	Position bucket tip on the x-axis	mm
x_{tip}	Position bucket tip on the y-axis	mm

6 References

[1] BAUMA 2020. Will electric and driverless versions of construction machines soon be available? https://www.bauma.de/en/trade-fair/industry-trends/efficiency/construction-machines-driverless/ (Accessed January 6, 2021).

[2] Vukovic, M. Hydraulic hybrid systems for excavators. Dissertation, RWTH Aachen; Shaker Verlag GmbH, **2017**.

[3] Hlawitschka, E. Zum Temperaturverhalten von Hydraulikanlagen. *Zeitschrift Agrartechnik*, **1979**, *29,* 86–88.

[4] *Proceedings of the 52nd National Conference on Fluid Power*.

[5] Sturm, C. *Bewertung der Energieeffizienz von Antriebssystemen mobiler Arbeitsmaschinen am Beispiel Bagger;* KIT Scientific Publ: Karlsruhe, Baden, **2015**.

[6] Schmitz, K.; Murrenhoff, H. *Grundlagen der Fluidtechnik // Hydraulik: Teil 1: Hydraulik;* Shaker Verlag: Aachen, **2018**.

[7] Jacobs, G., Ed. *Maschinengestaltung: Band 1;* Mainz: Aachen, **2016**.

[8] Otto, N. *Erprobung nachhaltiger Hydraulikfluide auf Ester- und Wasserbasis,* 1st ed.; Shaker: Herzogenrath, **2019**.

[9] Volvo Construction Equipment. EC15D, EC18D, ED20D: Volvo Compact Excavators 1.54-1.95 t 16.3 hp. https://www.volvoce.com/-/media/volvoce/global/products/excavators/compact-excavators/brochures/brochure_ec15d_ec18d_ec20d_t4f_en_21_20045575_c.pdf?v=6nxHPw (Accessed January 6, 2021).

[10] Helmus, M.; Fecke, M. *Standardisierung definierter Lastzyklen und Messmethoden zur Energieverbrauchsermittlung von Baumaschinen: Schlussbericht zum Forschungsvorhaben*, **2015**.

[11] JCMA - Japan Construction Mechanization Association Standards Commitee. JCMAS H 020:2007 - (Japan Construction Mechanization Association Standard): Earth-moving machinery – Fuel consumption on hydraulic excavator – Test procedures. http://www.cema.or.jp/general/news/pdf/17th/17thJapanese (Accessed December 18, 2020).

[12] DIN Deutsches Institut für Normung e.V. *Prüfung von Mineralölen und verwandten Stoffen – Bestimmung des Viskosität-Temperatur-Verhaltens: Richtungskonstante m;* Beuth Verlag: Berlin, **04.2011**.