EFFICIENT MODEL-BASED THERMAL SIMULATION METHOD DEMONSTRATED ON A 24-TON WHEEL LOADER

Eric Pohl[1]*, Jürgen Weber[1]

[1]*Institute of Mechatronic Engineering, Technische Universität Dresden, Helmholtzstrasse 7a, 01069 Dresden*

* Corresponding author: Tel.: +49 351 463-39081; E-mail address: eric.pohl2@tu-dresden.de

ABSTRACT

With the ongoing emission reduction and electrification of mobile machinery the efficiency of the hydraulics is becoming increasingly important. Here, the thermal behavior of the entire hydraulic drive system, including all cooling circuits, plays a decisive role. For a comprehensive quantification of the thermal behavior, a model-based analysis method is the most cost- and time-efficient option. In the recent past, thermo-hydraulic network models with lumped parameters have become established for this purpose. Due to their approach of solving coupled time-dependent differential equations, very small time steps and thus very long simulation times can occur for complex systems such as mobile machines. Thus, an efficient design process including parameter studies and optimization is highly uneconomical. In this paper, an efficient computational approach for the determination of local temperatures in thermal equilibrium is presented and used for the simulation of the thermal behavior of a 24 t wheel loader. The results are simulatively validated by a comparison with the classical transient calculation approach with exactly the same parameterization in order to prove the accuracy and the reduction of simulation time. The comparison shows that the presented simulation approach is able to reduce the simulation time by more than 4000 times with almost the same accuracy. This demonstrates the great potential of the simulation method for the design process including economic optimization and parameter studies.

Keywords: mobile machines, thermal, simulation, system design, lumped-parameter model, thermal equilibrium

1. INTRODUCTION

The thermal behavior of mobile machines has a substancial influence on their design, efficiency, operability and reliability. Efficiency in particular will become increasingly important in the near future, as the trend is towards low-emission and energy-efficient systems. For example, the efficiency of electrified battery-powered machines has a direct impact on machine costs due to the high battery costs. As of yet the thermal behavior is only taken into account to a limited extent during the development and design of the different machine or drive subsystems. Design adjustments are often carried out iteratively via costly prototype measurements or the systems are greatly oversized from a thermal point of view. In order to reduce development time and costs, an accurate prediction of the thermal behavior of the entire system is required as part of the design process. Due to the complex thermal interactions within and between different domains in hydraulic drive systems, efficient simulation methods are required to calculate the local operating temperatures within the thermal equilibrium. For stationary applications thermal lumped-parameter models have been successfully established as the most efficient simulation method in recent years [1] – [3]. A vivid example of a thermo-hydraulic network model is shown in **Figure 1**.

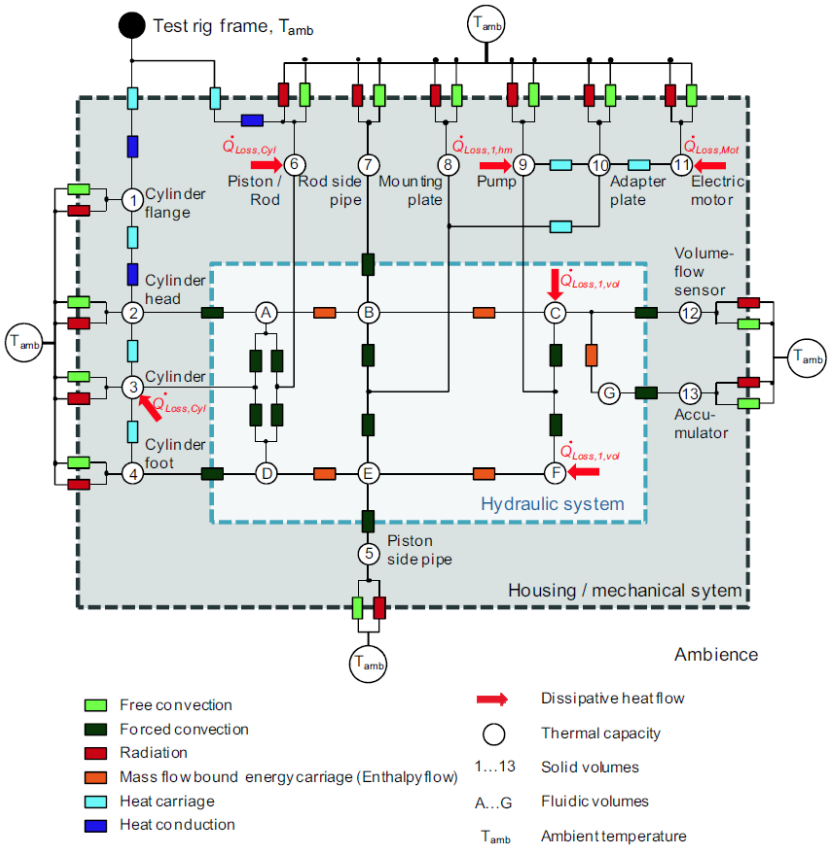

Figure 1: Exemplary thermal resistance network model of a compact drive [3]

However, in the field of mobile machines the highly application- and operation-dependent varying environmental conditions and performance requirements make it difficult to collect comparable and repeatable data for parameterization and validation compared to stationary processes. The basic feasibility of functional thermal modeling based on lumped-parameter network models of a mobile machine has been demonstrated by Zimmerman and Busquets for a displacement-controlled mini-excavator [4] – [7] and Kwon et al. for the hydraulic drivetrain of a hybrid vehicle [8] – [10]. The aforementioned works have in common that they calculate the thermal behavior using coupled time-dependent differential equations to solve conservation of mass and energy, which result from lumped-parameter network models. Therefore, the entire heating process must be calculated over countless small-scale time steps in order to finally determine local temperatures in thermal equilibrium. For systems of low complexity, results can thus be obtained in a short to moderate time. However, with increasing complexity of the system, this computational approach can lead to simulation times of many hours up to days or weeks due to very small time steps. In addition, this is scaled by the stiffness of the system and the dynamics of the process which can drastically affect the stability of the thermal-hydraulic simulation and lead to even longer simulation times or crashes. For complex systems, the development and optimization with this calculation approach is therefore only possible with great simplifications or very high expenditure of time. However, manufacturers want to model their machines as accurately as possible and have short simulation times for extensive optimizations and adjustments. Therefore, for an economically efficient simulation of the thermal behavior of the entire machine, it is desirable to develop a calculation approach that significantly reduces the simulation time without losing accuracy.

The following paper presents an efficient simulation approach for thermal lumped parameter models with dynamic loads. It directly determines the local temperatures within the state of thermal equilibrium, taking into account the dynamic work process, without calculating the dynamic heating process. Following the statistical analysis of performance using an elementary hydraulic circuit in

[11], the methodology is now applied to a complex network model of a 24 t wheel loader. In order to demonstrate the potential of the approach, the simulation results and times are compared with the classic transient calculation approach.

2. SIMULATION APPROACH

For the design of mobile machines, knowledge of the transient behavior of component and fluid temperatures, in particular the heating process, is not necessary in most cases. One exception here is the improvement in cold start behavior, for example. To optimize efficiency, operability and reliability only temperatures at which the machine operates in the thermal equilibrium range are taken into account. Therefore, a computational approach for thermo-hydraulic network models has been developed to directly determine the temperatures in thermal equilibrium without time-consuming solution of time-dependent differential equations of energy conservation due to varying temperatures (**Figure 2**).

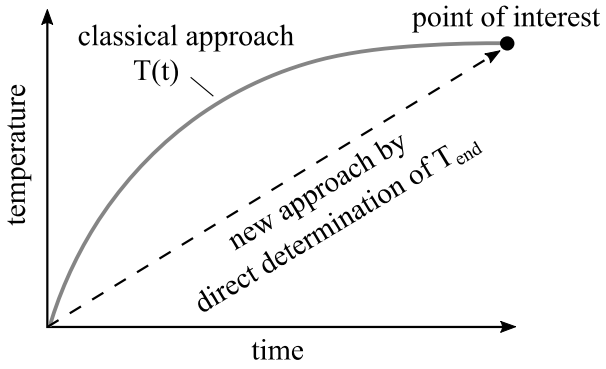

Figure 2: Developed steady-state approach compared to classical approach

In contrast to the classical transient simulation, the fluidic domain and thermal domain are decoupled. The fluidic domain is simulated using a functional-energetic network model, as is also used to calculate the kinematics of a mobile machine. The thermal domain is then calculated separately, using the same thermal parameterization as in classical coupled thermo-hydraulic network models. To determine process parameters and fluid properties, a considered duty cycle (dc) only needs to be simulated once instead of several times using the functional-energetic model (fluidic domain). This is carried out using the system simulation software SimulationX®. The time-dependent characteristics of parameters and properties are then used as boundary conditions for a steady-state calculation to determine the thermal equilibrium, which is realised with MATLAB®.

2.1. Derivation of the thermal model

The objective of the thermal model is to determine the steady-state operating temperatures of any system for an arbitrary repeating cycle. This state is called thermodynamic (or thermal) equilibrium and is defined by the 1st law of thermodynamics. The condition is reached when the internal energy U no longer changes over time:

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \sum_i \dot{Q}_i + \sum_j \dot{W}_j + \sum_k \dot{H}_{in,k} + \sum_l \dot{H}_{out,l} = 0 \qquad (1)$$

with $\dot{Q}$ as external heat flow, $\dot{W}$ as technical power and $\dot{H}$ as enthalpy flow into and out of the system. This also means that all temperatures within the system must be constant over the change in the time step dt. **Figure 3** illustrates an established state of equilibrium for an exemplary periodically repeated duty cycle with a defined cycle time t_{dc}. The state is reached after about 3000 s. Looking at the

behavior of the temperature curve for a single duty cycle within the equilibrium, it can be seen that the temperatures are still varying.

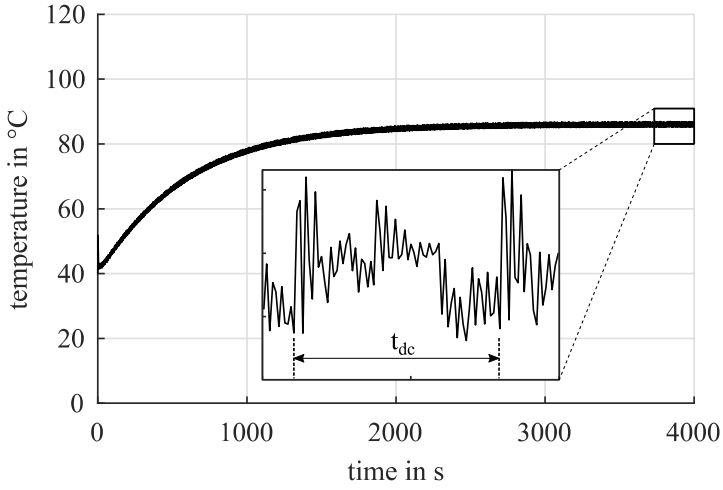

Figure 3: Dynamic temperature behaviour for an exemplary duty cycle

This is due to the fact that the state of equilibrium is never reached at any time t within the cycle, if the considered time step in (1) is $\mathrm{d}t \ll t_{dc}$. This leads to the conclusion that the thermal equilibrium is a cycle averaged consideration of temperatures and energy over several cycles with $\mathrm{D}t = t_{dc}$. Consequently, the 1st law of thermodynamics can be expressed by process averaged quantities.

$$\frac{\mathrm{D}\overline{U}}{\mathrm{D}t} = \sum_i \bar{\dot{Q}}_i + \sum_j \overline{\dot{W}_j} + \sum_k \bar{\dot{H}}_{in,k} + \sum_l \bar{\dot{H}}_{out,l} = 0 \tag{2}$$

The previously transient duty cycle is therefore considered as steady-state. This implies that there are process averaged temperatures (not time averaged) belonging to the quantities in (2). For a correct averaging of those quantities and solving of the local process averaged temperatures the differentiation of physical state variables have to be taken into account. They are divided into extensive and intensive quantities. Extensive quantities, like mass or energy, can be averaged directly over time due to the applicable conservation laws. Intensive quantities, like temperature or density, are always directly related to an extensive variable to calculate another extensive quantity. For a better understanding the heat term of the averaged enthalpy flow $\bar{\dot{H}}$ is given here as an example, which is calculated by the product of the constant specific heat capacity c_p, the mass flow $\dot{m}$ and the fluid temperature T_f.

$$\bar{\dot{H}} = \frac{1}{t_0 - t_{end}} \cdot \int_{t_0}^{t_{end}} c_p \cdot \dot{m}(t) \cdot T_f(t)\, \mathrm{d}t \tag{3}$$

Since $\dot{m}$ and T_f are directly related to each other and varying over time with $\mathrm{d}t \ll t_{end} - t_0$, only integration of the product is permissible according to the laws of physics. However, following the aforementioned assumption that there are process averaged temperatures, (3) can be simplified to

$$\bar{\dot{H}} = \frac{\tilde{T}_f}{t_0 - t_{end}} \cdot \int_{t_0}^{t_{end}} c_p \cdot \dot{m}(t)\, \mathrm{d}t \tag{4}$$

by introducing a process averaged fluid temperature $\tilde{T}_f$. For the presented calculation approach this assumption is only applied to temperatures. All other intensive quantities are not assumed as average

and are determined by the transient functional simulation model. A detailed description of the determination of all necessary variables can be found in [11]. Through the described procedure the thermal steady state can be represented by averaged variables depending on process-averaged temperatures.

Since any number of local temperatures and heat emissions can be defined, the sole application of (2) is not sufficient to determine them all, because it only describes the global thermodynamic behavior of the entire system. Therefore, it is necessary to define a set of equations which are describing the local behavior of any points in the system. The first two equations are derived by the local heat transfer at any given points in the system.

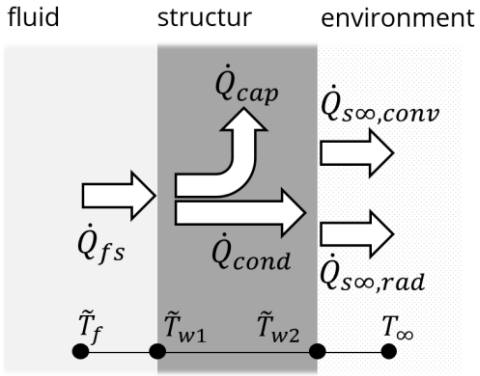

Figure 4: Local heat transfer out of the system

Figure 4 shows the heat transfer out of the fluid into the mechanical structure and out of this structure into the environment for a consideration with lumped parameters. In the heating phase the convective heat flow out of the fluid $\dot{Q}_{fs}$ divides into two parts, the structure heating flow $\dot{Q}_{cap}$ and the thermal conduction $\dot{Q}_{cond}$. When it comes to the emission into environment, the latter is divided into radiation and convection. In thermodynamic equilibrium $\dot{Q}_{cap}$ becomes zero in a process averaged consideration, as the average temperature of the structure is constant. Thus, the local heat transfer out of the system for any point i within the thermal steady state can be described by the following relationships.

$$\bar{\dot{Q}}_{fs,i}(\tilde{T}_{f,i}, \tilde{T}_{w1,i}) = \bar{\dot{Q}}_{cond,i}(\tilde{T}_{w1,i}, \tilde{T}_{w2,i}) \tag{5}$$

$$\bar{\dot{Q}}_{cond,i}(\tilde{T}_{w1,i}, \tilde{T}_{w2,i}) = \bar{\dot{Q}}_{s\infty,conv,i}(\tilde{T}_{w2,i}, T_\infty) + \bar{\dot{Q}}_{s\infty,rad,i}(\tilde{T}_{w2,i}, T_\infty) \tag{6}$$

This defines all energy flows out of the system and the local wall temperatures can be determined using the local fluid temperatures. However, the actual thermodynamic state of the latter is still undefined. In order to be able to calculate the local fluid temperatures, the causal relationships within the considered system must be known. These are described by the enthalpy flow, or rather by the change in enthalpy flow between two points under consideration (**Figure 5**).

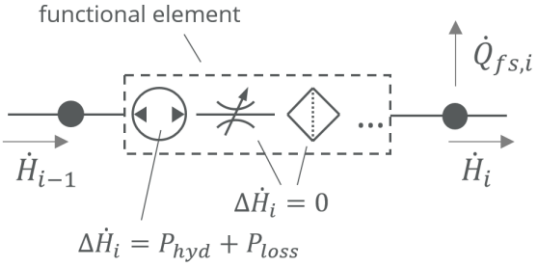

Figure 5: Change of state of the fluid between two considered points

In principle, the enthalpy only changes through the supply or removal of energy across the system boundary of the fluidic domain. As there are always functional elements such as pumps or valves between these points, there is always a change in enthalpy, as heat is dissipated into the mechanical structure. Pumps, for example, transfer the mechanical energy of the shaft into the fluid and thus feed energy into the fluidic domain. A pressure loss within the fluidic domain, as in a valve or a pipe, does not lead to a change in enthalpy. This merely shifts energy from the volume work to the heat portion of the enthalpy. Therefore, the causal relationship between two points under consideration is described by:

$$\bar{\dot{H}}_i(\tilde{T}_{f,i}) = \bar{\dot{H}}_{i-1}(\tilde{T}_{f,i-1}) + \Delta\bar{\dot{H}}_i - \bar{\dot{Q}}_{fs,i}(\tilde{T}_{f,i}, \tilde{T}_{w1,i}) \tag{7}$$

This means that the state of the previous point in the system must be known. Therefore, the determination of all local temperatures within the thermal equilibrium results in a matrix of all considered points, described by (5), (6) and (7), as well as the global condition in (2). For an example of a hydraulic circuit with i considered points and none enthalpy flow into or out of the system, the following set of equations results.

global:
$$\frac{\mathrm{D}\bar{U}}{\mathrm{D}t} = \sum_i \bar{\dot{Q}}_i + \sum_j \overline{\dot{W}_j} = 0$$

local:
$$\begin{pmatrix} \bar{\dot{Q}}_{fs,1} = \bar{\dot{Q}}_{cond,1} & \dots & \bar{\dot{Q}}_{fs,i} = \bar{\dot{Q}}_{cond,i} \\ \bar{\dot{Q}}_{cond,1} = \bar{\dot{Q}}_{s\infty,conv,1} + \bar{\dot{Q}}_{s\infty,rad,1} & \dots & \bar{\dot{Q}}_{cond,i} = \bar{\dot{Q}}_{s\infty,conv,i} + \bar{\dot{Q}}_{s\infty,rad,i} \\ \bar{\dot{H}}_1 = \bar{\dot{H}}_i + \Delta\bar{\dot{H}}_1 - \bar{\dot{Q}}_{fs,1} & \dots & \bar{\dot{H}}_i = \bar{\dot{H}}_{i-1} + \Delta\bar{\dot{H}}_i - \bar{\dot{Q}}_{fs,i} \end{pmatrix}$$

2.2. Determination of process parameters

In order to receive all necessary transient data to perform a physically correct process averaging to solve the resulting thermal matrix a functional-energetic simulation is needed. The characteristic of this model is basically the same as for kinematic simulations. There are no additional thermal elements necessary and a constant global fluid temperature is used for the calculation. If the considered system is exactly initialised only one duty cycle has to be calculated for the determination of the required process data. To avoid initialisation errors, the cycle can be simulated twice and the data extracted from the second cycle. The data determined in this way are purely functional parameters such as power $\vec{P}(t)$, pressure $\vec{p}(t)$ or mass flow $\vec{\dot{m}}(t)$. For the derivation of the actual fluid parameters, like the density $\vec{\rho}(t)$, a map of the fluid $\rho(p, T_f)$ is used. The actual benefit of this type of determination becomes apparent in the following section. In combination with the previous described thermal model the work flow of the developed approach shown in **Figure 6** results.

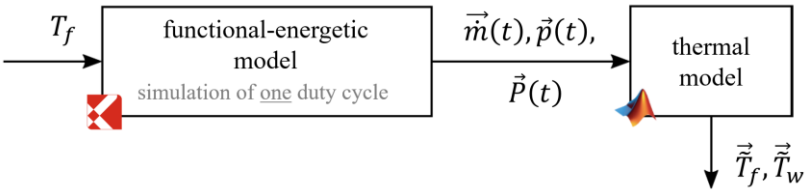

Figure 6: Initial model by combining the functional and thermal model

As the combined model can be initialised with any fluid temperature, the resulting process averaged temperatures within the thermal equilibrium are calculated by the process quantities corresponding

to this initial temperature. This leads to incorrect power values and power losses in the system and thus to incorrect temperatures calculated by the thermal model. In fact they have to be determined by the process parameters corresponding to the resulting temperatures. Therefore, a correction loop needs to be implemented. Thereby, the fluid temperature calculated by the thermal model is set as the new initial temperature for the next step, i.e. the following functional-energetic simulation of the single duty cycle. The loop ends when a certain relative deviation δ between the initial and newly determined fluid temperature is fallen short of, for example $\delta < 0.1$ %. In summary, the developed calculation approach works as shown in **Figure 7**.

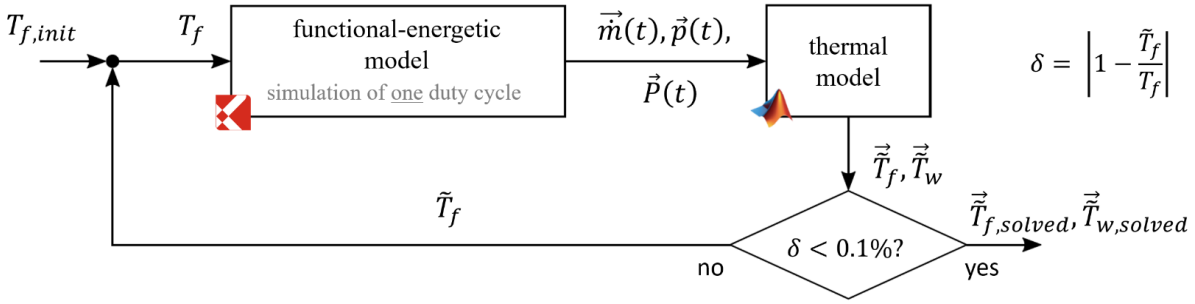

Figure 7: Resulting model by correct determination of fluid and process parameters

As already mentioned, the fluid parameters are derived by a map of the fluid $f(p, T_f)$ for each point in the system under consideration. When using the initial combined model, this is obsolete as only the initial fluid temperature can be used. The correction loop now uses the simulated fluid temperatures $\vec{T}_f$ to calculate the fluid parameters in the following run of the loop. This guarantees a solid solution of these quantities if the deviation δ is small enough.

3. SETUP DEMONSTRATION SYSTEM

The realisation and testing of the methodology just described is carried out on a 24 t wheel loader. This machine was prototypically build as part of the TEAM-project [12] with the objective to analyse and adapt the system structure at a functional level in order to improve the overall efficiency of the machine. The schematic system structure of the wheel loader is shown in **Figure 8**.

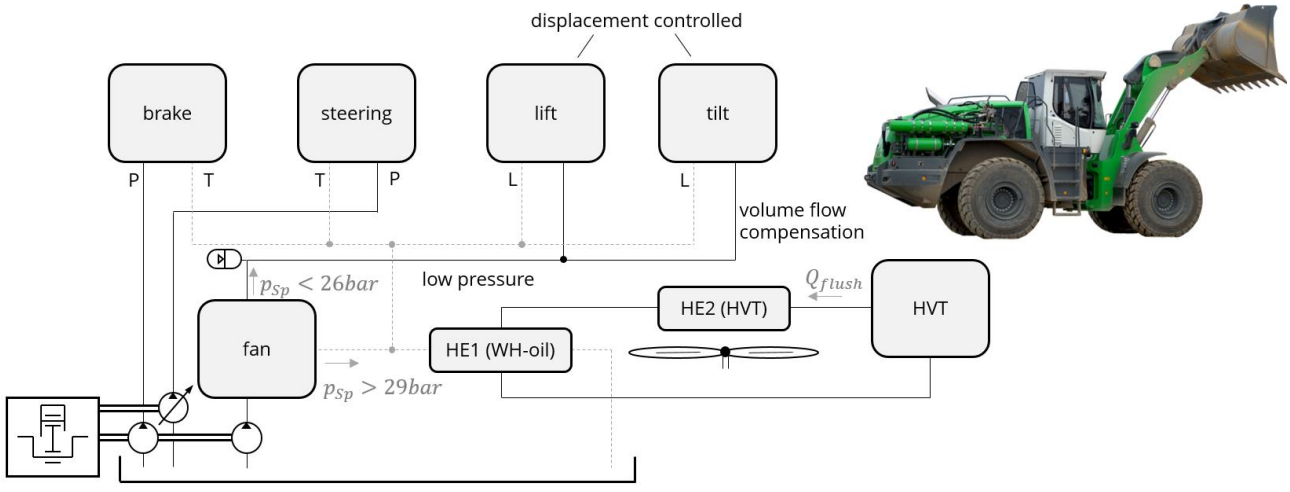

Figure 8: Demonstration object used for the methodology introduced

The test object has special functional features compared to conventional series machines. Firstly, the combustion engine has been optimised in terms of its efficiency for its operating range. The travel drive of the wheel loader is realised via a hydro-mechanical variable transmission (HVT). Furthermore, the working hydraulics (WH) is realised by a displacement controlled system. The

lifting and tilting functions are controlled by separate pumps. **Figure 8** also shows that there is a separate hydraulic circuit for the HVT and a separate hydraulic circuit for the working hydraulics and auxiliary units. The only connection between the circuits is through thermal interaction via the oil-oil heat exchanger HE1. Here, the oil from the working hydraulics is cooled by the oil from the transmission circuit. The transmission oil itself is previously cooled via the air-oil heat exchanger HE2 and circulates in a closed circuit. The axial fan used for this is driven by a hydraulic motor via the fan subsystem. The pumps of the brake, steering and fan subsystems draw hydraulic oil from the reservoir. In the fan subsystem, the entire pump volume flow is used to charge the hydraulic accumulator when the accumulator pressure falls below 26 bar. If the threshold value is exceeded, the volume flow is routed into the reservoir via HE1. The accumulator branch is used for the supply and removal of volume flow for the closed displacement-controlled circuits of the lifting and tilting subsystems. Only the leakage from the pump units is passed on to the reservoir via the heat exchanger. Due to the system complexity described, which includes both valve- and displacement-controlled structures and their complex interaction, this demonstrator is particularly suitable for demonstrating the simulation methodology developed.

4. MODELLING AND RESULTS

For a comprehensive comparison between the classical transient and the developed steady-state calculation method, the thermo-fluid network model was first set up for the classical approach. For this purpose, the existing functional-energetic model was adapted with regard to a thermo-fluidic simulation and extended by thermal elements.

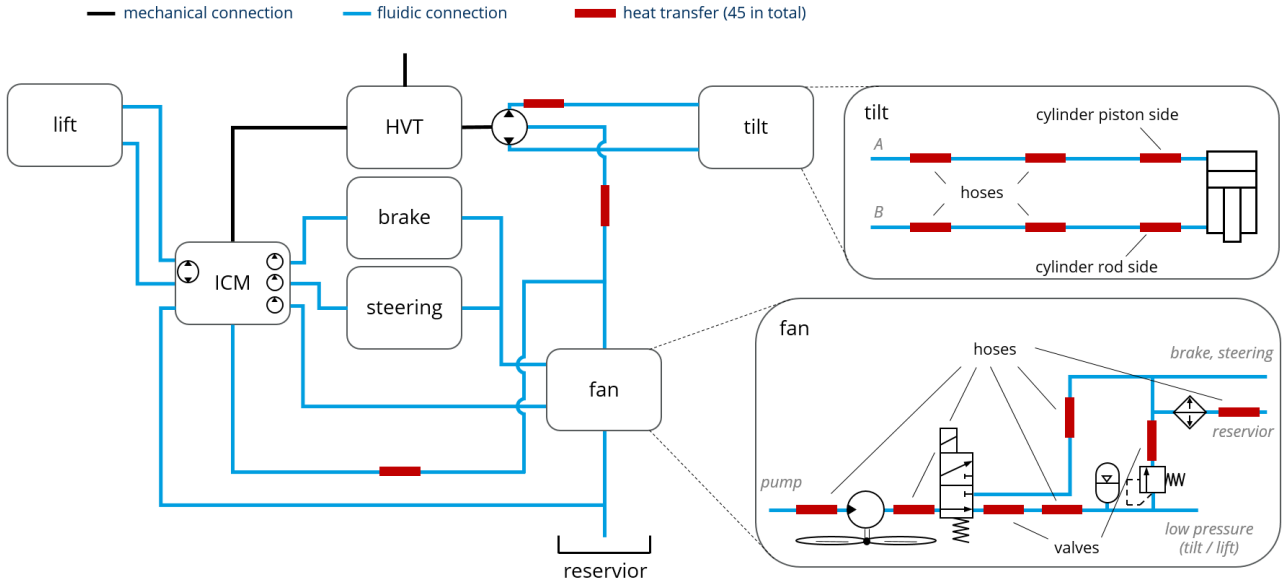

Figure 9: Simplified scheme of the thermo-fluidic model with exemplary subsystems

Figure 9 illustrates the simplified scheme of the complex system. For a better understanding, two exemplary subsystems with their thermal discretisation are shown. For each discretised point, i.e. thermal element, the fluid and wall temperatures are calculated. In order to reduce the size of the model and thus improve the performance, some elements are combined, e.g. hoses connected in series without thermally relevant elements in between. The entire system consists of a total of 45 thermal points that are analysed. In each point the heat transfer is modeled via analytical approaches and empirical correlations, which are commonly used in lumped parameter network models and described in detail in [13]. The heat transfers were validated by loading the individual subsystems separately until the local thermal equilibrium was reached. The results of the validation can be found in [14]. A complete validation and comparison to the measured results of the entire machine is not possible due

to the lack of loss maps for the pumps, which are absolutely essential for this. Furthermore, the machine cannot yet perform any cycles automatically, which is another constraint for correct validation. For this reason, only the two simulation methods are compared. For this purpose, the validated heat transfers and the use of constant pump efficiencies are sufficient.

For the calculation approach developed, the functional-energetic simulation model for determining the dynamic process quantities is the same as the model mentioned for the classical approach, with the exception of the thermal elements. These are modelled with exactly the same parameterisation in the decoupled thermal model.

To further reduce the calculation time of the transient model, a simplified cycle was used in which the vehicle is stationary and only the lifting and tilting functions are used. Nevertheless, all other subsystems also heat up due to the interconnected system structure.

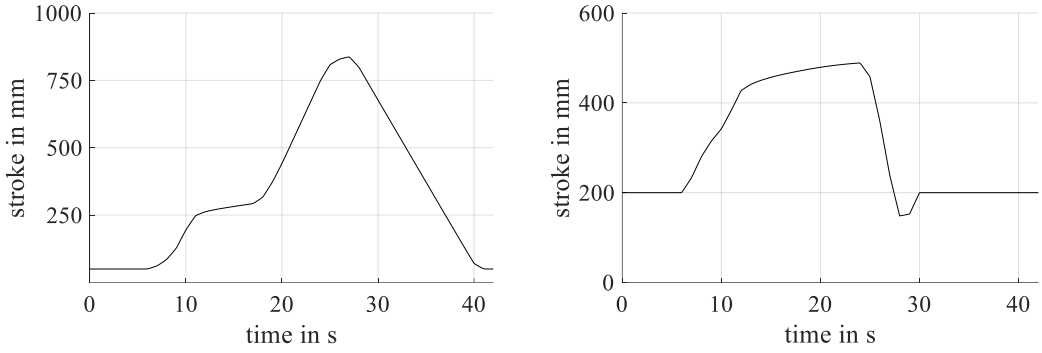

Figure 10: Trajectories of lifting (l.) and tilting (r.) cylinders of the duty cycle used

The simplified duty cycle used takes 42 s and represents a typical motion of the working hydraulics of a wheel loader, such as during a y-cycle (**Figure 10**). Using the cycle shown, the simulations are now carried out with both methods and the results are compared with each other.

Since the objective of the steady-state approach is the reduction of simulation time while maintaining accuracy, the latter must first be proven. Therefore, the fluid temperatures of all 45 thermal points indicated in **Figure 9** are compared for the two simulations methods (**Figure 11**). Looking at the quantitative differences, it can be seen that the absolute deviation between the temperatures calculated is less than 2 K in most cases. Only the displacement controlled subsystems (lift and tilt) show higher deviations of up to 4 K.

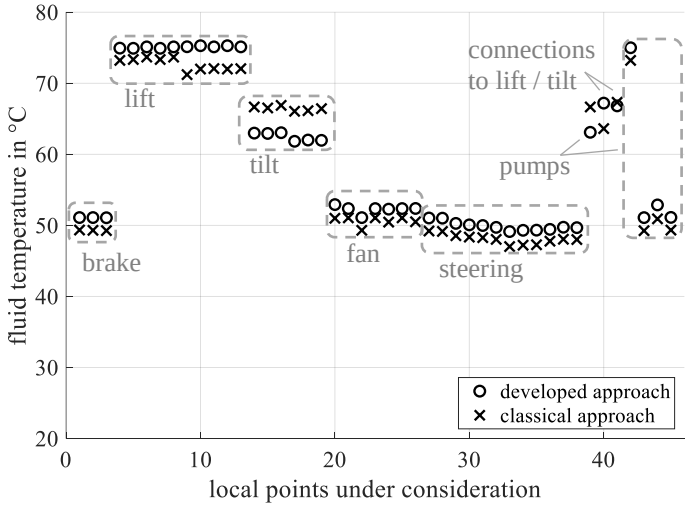

Figure 11: Comparison of calculated fluid temperatures for all points under consideration

In addition to the small quantitative deviations, the qualitative distribution of the temperatures also shows very good correlation. The only exception to this is the lift subsystem. This is due to the fact that the power losses of the pump in a displacement-controlled system vary in magnitude in the respective flow direction, which is not yet taken into account in the thermal model. The same effect occurs in the tilt subsystem, but is much smaller due to the lower difference in magnitude of the power losses and the much shorter process duration when tipping the bucket. To summarise, it can be said that both the quantitative and qualitative differences are acceptably small. However, there is still potential to improve the accuracy of the developed methodology.

Once the acceptable accuracy has been proven, the focus can now be placed on the reduction of simulation time. Due to the complexity of the system and the high level of detail in terms of the large number of thermal heat transfer elements, the presented model is enormously computationally complex. In addition, due to stability problems with the classical transient method, some fluid capacities had to be greatly increased in order to be able to dampen dynamic effects sufficiently. Although this leads to a shorter computing time of a cycle, it also causes a delayed behavior of the temperatures for reaching thermal equilibrium. This means that a single cycle can be calculated faster, but significantly more cycles must be simulated to reach steady state. To ensure best possible comparability the fluid capacities were also adjusted in the functional-energetic model of the developed method.

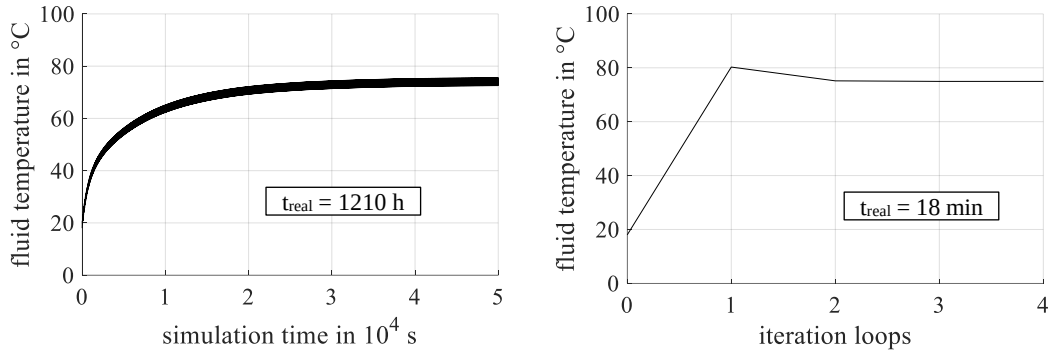

Figure 12: Comparison of computational times of the classical (l.) and the steady-state (r.) approach

The transient heating process of a point within the lifting subsystem calculated by the classical approach is shown in the left diagram of **Figure 12**. The thermal equilibrium is reached after approximately $5 \cdot 10^4$ s, which corresponds to a real time of around 1210 hours. In contrast, the iteration process for the same point calculated by the developed method is shown on the right diagram. The decoupled simulation requires 4 iterations, e.g. 3 correction loops, until the fluid temperature in the thermodynamic steady state is determined. The entire simulation with the decoupled steady-state approach only takes about 18 minutes. Compared to the classical approach, this means that it is more than 4000 times faster. This shows that the new methodology can drastically reduce the simulation time of complex hydraulic systems. This comparatively very short calculation time with acceptable accuracy opens up completely new possibilities in the model-based design and optimisation of mobile machines and furthermore for any hydraulic system.

5. CONCLUSION AND OUTLOOK

In this paper a simulation approach is presented, which reduces the dynamic heating process of a 24 ton wheel loader to the computation of the steady-state final state using functional dynamic quantities as boundary conditions. This is based on the approach that the thermodynamic equilibrium is a process-averaged consideration. Using the analytical and empirical equations of heat transfer together with state changes of enthalpy flow and functional dynamic process quantities allows an iterative calculation of process-averaged temperatures within this equilibrium. A comparison with the classical

approach, i.e. using time-dependent differential equations to solve mass and energy conservation, showed a substantial reduction in simulation time by a factor of more than 4000. The proof of only minor qualitative and quantitative deviations from the results of the classical approach underlines that the presented approach represents an efficient methodology for the design and optimisation of mobile machines. Furthermore, it can be applied to any hydraulic system as long as only operating temperatures are relevant. To further improve the approach shown, future work will focus on improving the accuracy of displacement-controlled systems.

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NOMENCLATURE

c_p	Specific heat capacity	J/(kg·K)
$\dot{H}$	Enthalpy flow	W
$\dot{m}$	Mass flow	kg/s
P	Power	W
p	Pressure	bar
$\dot{Q}$	Heat flow	W
T	Temperature	°C
t	Time	s
U	Internal Energy	J
$\dot{W}$	Technical power	W
δ	Relative threshold	
ρ	Density	kg/m³
HE	Heat exchanger	
HVT	Hydro-mechanical transmission	
ICM	Internal combustion machine	
WH	Working hydraulics	

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