An efficient simulation approach to determine the thermal equilibrium of drive systems in mobile machinery

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Abstract

For the calculation of the thermal behavior of hydraulic systems, network models with lumped parameters have become established in the last decades. Here, the calculation of mass and energy conservation is carried out via time-dependent differential equations with very small time steps. For complex systems, such as the entire drive system of a mobile machine, this leads to very high simulation times, which are not suitable for parameter studies and optimizations. In this paper, an efficient computational approach to determine system temperatures in thermal equilibrium is presented, which significantly reduces the simulation time by up to a factor of 100 while maintaining almost the same accuracy. The new method is validated by a comparison with a classical, i.e. time-dependent, simulation using an elementary hydraulic circuit as an example. An extensive statistical analysis is made by varying the dynamic process conditions by different volume flow curves in order to prove the accuracy and the reduction of the simulation time.

Keywords: mobile machines, thermal simulation, lumped-parameter network model, calculation approach, thermal equilibrium, steady-state.

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1 Introduction

The thermal balance of mobile machines significantly influences their design, efficiency, operability and reliability. During the development and design of the different machine or drive subsystems, the thermal behavior is only taken into account to a limited extent. Currently, design adaptations are often carried out iteratively via costly prototype measurements. In order to reduce development time and costs, an accurate prediction of the thermal behavior of the entire system within the design process is necessary. Due to the complex thermal interactions within hydraulic drive systems, efficient simulation methods are needed to calculate the thermal equilibrium. In recent decades, thermal lumped-parameter models have been successfully established as the most efficient simulation option for stationary applications [1–3]. 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 the research of 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 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 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 due to very small time steps. Therefore, for an economically efficient simulation of the thermal equilibrium of the entire machine, it is desirable to develop a calculation approach that significantly reduces the simulation time without losing accuracy. In the following paper, an efficient simulation approach for thermal lumped-parameter network models with dynamic loads is presented, which directly determines the state of thermal equillibrium without calculating the dynamical heating process. Calculation results are discussed and compared to a classical time-dependent simulation. A statistical analysis was made due to the approximations inherent in the method. The results of the analysis and the potential scope of the method are shown and discussed.

2 Modelling Approach

In most cases, the design of mobile machines does not require knowledge of the transient behavior of component and fluid temperatures, especially the heating process. To optimize efficiency, operability and reliability only temperatures at which the machine operates in the thermal equilibrium range are useful. Therefore, a computational approach for lumped-parameter 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 (Fig. 1). For this purpose, the fluidic domain and thermal domain are decoupled.

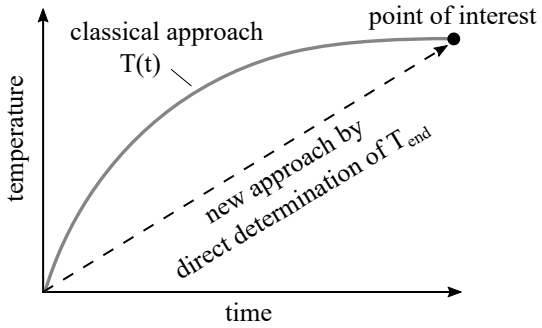

Figure 1 Developed steady-state approach compared to classical approach

For the determination of process parameters and fluid properties, a considered duty cycle (dc) has to be simulated only once instead of several times. The time-dependent characteristics of parameters and properties are then used as boundary conditions for a steady-state calculation to determine thermal equilibrium.

2.1 Derivation of thermal model

Defined by the 1st law of thermodynamics, thermal equilibrium of an arbitrary system is reached when the internal energy U no longer changes over time:

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

with $\dot{Q}$ as external heat flow, $\dot{W}$ as technical power and $\dot{H}$ as enthalpy flow into and out of the system. If U no longer change, all temperatures in the system have to be constant. An established state of equilibrium for an arbitrary

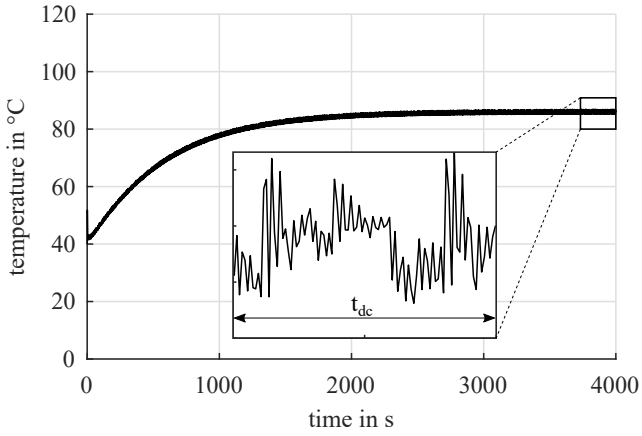

Figure 2 Exemplary dynamic temperature behavior

duty cycle with $t_{dc} = 10s$, which is periodically repeated, is exemplarily shown in Fig. 2. Looking at the behavior of the curve within the thermal equilibrium ($t > 3800s$) it can be seen that temperatures are still varying within the duty cycle. This is due to the fact that the state of equilibrium in (1) is never reached at any time t within the cycle, if the considered timestep in (1) is $dt < t_{dc}$:

$$\frac{dU}{dt} \neq 0 \tag{2}$$

Thus, thermal equilibrium does not establish locally within the cycle, but is an averaged consideration of temperatures and energy over several cycles ($dt = t_{dc}$):

$$\frac{d\bar{U}}{dt_{dc}} = 0 \tag{3}$$

This consideration leads to the assumption that the state of equilibrium can be described by process-averaged quantities:

$$\frac{d\bar{U}}{dt} = \sum_i \bar{\dot{Q}}_i + \sum_j \bar{\dot{W}}_j + \sum_k \bar{\dot{H}}_{in,k} - \sum_l \bar{\dot{H}}_{out,l} = 0 \tag{4}$$

Related to an arbitrary duty cycle, this means that the cycle is considered as a steady-state and not as a transient calculation case. Since conservation of energy applies, the energy E of a process is determined by integrating the

power P over time.

$$E = \int_{t_0}^{t_{dc}} P(t)\,\mathrm{d}t \tag{5}$$

Due to the applicable laws of physics, only conservation quantities, i.e. extensive quantities, may be averaged over time. In terms of energy this leads to the time-averaged power $\bar{P}$.

$$\bar{P} = \frac{E}{t_{dc} - t_0} = \frac{1}{t_{dc} - t_0} \cdot \int_{t_0}^{t_{dc}} P(t)\,\mathrm{d}t \tag{6}$$

This averaging is allowed for heat Q, technical work W and enthalpy H, since they are all forms of energy. This means that it is technically possible to reduce any duty cycle to a steady state case via energy or average power.

In case of technical work W the technical power $\dot{W}$ can be obtained by the functional process behavior of the considered system via volume flow rate $\dot{V}$, differential pressure Δp and total efficiency η_{tot}.

$$\bar{\dot{W}} = \frac{1}{t_{dc} - t_0} \cdot \int_{t_0}^{t_{dc}} \dot{V}(t) \cdot \Delta p(t) \cdot \eta_{tot}\,\mathrm{d}t \tag{7}$$

The map of total efficiency $\eta_{tot} = f(\dot{V}(t), \Delta p(t))$ is considered to be known.

Besides $\dot{V}$ and p enthalpy H additionally directly depends on fluid temperature T_f. Since temperature is an intensive quantity, the average value does not necessarily correspond to the time-averaged temperature of the cycle. To determine the extensive quantity H, the fluid temperature is related to the extensive quantity mass m or, in case of enthalpy flow $\dot{H}$, to the mass flow $\dot{m}$.

$$\begin{aligned} \bar{\dot{H}} = {} & \frac{1}{t_{dc} - t_0} \cdot \int_{t_0}^{t_{dc}} c_p \cdot \dot{m}(t) \cdot T_f(t)\,\mathrm{d}t \\ + & \frac{1}{t_{dc} - t_0} \cdot \int_{t_0}^{t_{dc}} \frac{\dot{m}(t)}{\rho(t)} \cdot (1 - \gamma(t) \cdot T_f(t)) \cdot p(t)\,\mathrm{d}t \end{aligned} \tag{8}$$

The specific heat capacity for constant pressure c_p is considered constant and the coefficient of thermal expansion γ is time-dependent. The assumption that the state of equilibrium can be described by process-averaged quantities implies that there have to be process-averaged temperatures belonging to those quantities. This leads to the fact that there exists a process-averaged

fluid temperature $\tilde{T}_f$, which is constant, so (8) can therefore be expressed as follows:

$$\begin{aligned}\bar{\dot{H}} = \frac{\tilde{T}_f}{t_{dc}-t_0} \cdot \int_{t_0}^{t_{dc}} \overbrace{c_p \cdot \dot{m}(t) \cdot -\frac{\dot{m}(t)}{\rho(t)} \cdot \gamma(t) \cdot p(t)}^{P_{H1}(t)} \, \mathrm{d}t \\ + \frac{1}{t_{dc}-t_0} \cdot \int_{t_0}^{t_{dc}} \underbrace{\frac{\dot{m}(t)}{\rho(t)} \cdot p(t)}_{P_{H2}(t)} \, \mathrm{d}t\end{aligned} \tag{9}$$

Because of the time dependence of $\dot{m}$ and the relationship between intensive and extensive quantities, $\tilde{T}_f$ is not equal to the time-averaged fluid temperature $\bar{T}_f$. As in (7) mass flow and pressure are known from the process. To determine ρ and γ, these are interpolated via $\tilde{T}_f$ and p using the known property map of the fluid. With all known quantities the average over time of the introduced auxiliary variables P_{H1} and P_{H2} can be build, which leads to

$$\bar{\dot{H}} = \tilde{T}_f \cdot \bar{P}_{H1} + \bar{P}_{H2} \tag{10}$$

With (10) the thermodynamic state of the fluid is describeable for any point in the system.

In general, heat extraction at a given point is determined by four heat flows: mainly convective transfer from fluid into the mechanical structure (fs), conduction through the mechanical structure (c), heating of the mechanical capacity (cap) and transfer from wall into the environment (s∞). The latter is divided into radiation (rad) and convection (conv). During the heating process, a part of the heat flow from the fluid into the wall is used to heat the mechanical structure until thermal equilibrium is reached.

$$\dot{Q}_{fs}(t) = \dot{Q}_c(t) + \dot{Q}_{cap}(t) \tag{11}$$

$$\dot{Q}_c(t) = \dot{Q}_{s\infty}(t) = \dot{Q}_{s\infty,conv}(t) + \dot{Q}_{s\infty,rad}(t) \tag{12}$$

with

$$\dot{Q}_{fs}(t) = \alpha_{fs}(t) \cdot A_i \cdot (T_f(t) - T_{w1}(t)), \tag{13}$$

$$\dot{Q}_c(t) = \lambda \cdot \frac{A}{d} \cdot (T_{w1}(t) - T_{w2}(t)), \tag{14}$$

$$\dot{Q}_{s\infty,conv}(t) = \alpha_{s\infty}(t) \cdot A_o \cdot (T_{w2}(t) - T_\infty), \tag{15}$$

and

$$\dot{Q}_{s\infty,rad}(t) = \sigma \cdot \epsilon \cdot A_o \cdot (T_{w2}^4(t) - T_\infty^4), \tag{16}$$

where A_i is the inner transfer surface with corresponding wall temperature T_{w1} and A_o is the outer transfer surface, i.e. to the environment, with corresponding wall temperature T_{w2}. λ is the heat conduction coefficient of the mechanical structure and the term $\frac{A}{d}$ depends on the considered geometry. Furthermore, the radiation is defined by Stefan-Boltzmann constant σ and the emission coefficient ϵ. As with the fluid temperature, it is assumed that there also exist process-averaged wall temperatures $\tilde{T}_{w1}$ and $\tilde{T}_{w2}$. Since λ, σ and ϵ are constants, the avegared heat conduction and radiation is defined by

$$\bar{\dot{Q}}_c = \lambda \cdot \frac{A}{d} \cdot (\tilde{T}_{w1} - \tilde{T}_{w2}) \tag{17}$$

$$\bar{\dot{Q}}_{s\infty,rad} = \sigma \cdot \epsilon \cdot A_o \cdot (\tilde{T}_{w2}^4 - T_\infty^4), \tag{18}$$

Following the rule of averaging as in (9) and substituting the time-dependent temperatures in (13) and (15), the averaged convective heat flow is given by

$$\bar{\dot{Q}}_{fs} = \bar{\alpha}_{fs} \cdot A_i \cdot (\tilde{T}_f - \tilde{T}_{w1}), \tag{19}$$

$$\bar{\dot{Q}}_{s\infty,conv} = \bar{\alpha}_{s\infty} \cdot A_o \cdot (\tilde{T}_{w2} - T_\infty), \tag{20}$$

2.2 Boundary conditions

Since $\tilde{T}_f$, $\tilde{T}_{w1}$ and $\tilde{T}_{w2}$ are the process-averaged temperatures to be calculated, all other variables within the thermal model must be known. To determine the time-dependent process variables within a duty cycle a functional-energetic simulation of the cycle has to be made by using a lumped-parameter network model. In contrast to previous procedures, which simulate the considered duty cycle many times solving the time-dependent differential equations of mass and energy conservation untill thermal equilibrium is reached, only one duty cycle has to be calculated. In addition, this functional-energetic simulation of the single duty cycle is calculated with a constant fluid temperature, which makes it even faster because it greatly simplifies the calculation of energy conservation. Thus, the time-dependent process variables $\dot{m}$ and p are given. Pressure p und an initial fluid temperature $\tilde{T}_f$ are used to interpolate fluid properties ρ and γ using the known property map of the fluid.

The thermal equilibrium of any system is a state which is defined by the 1st law of thermodynamics. If this state is reached the inner energy U is not changing anymore. This means that all temperatures within the system have to be considered as constant. This leads to the fact that the heat flow into the mechanical structure $\bar{\dot{Q}}_{cap}$, described in (11), gets zero. Thus, all the heat flow from the fluid into the mechanical structure is also emitted from the outer wall to the environment. Hence, it follows that $\bar{\dot{Q}}_{fs}$, $\bar{\dot{Q}}_{c}$ and $\bar{\dot{Q}}_{s\infty}$ are equal.

$$\bar{\dot{Q}}_{fs} = \bar{\dot{Q}}_{c} = \bar{\dot{Q}}_{s\infty} \tag{21}$$

At this point, two of three necessary equations to solve the three unknown process-averaged temperatures are given. However, the information about the causal correlations is still unknown, i.e. the energy supply and removal within the system which determine the local state of fluid and structure.

In order to be able to calculate the causal correlations required to determine the correct thermal state of any point in the system, the enthalpy flow (10) is used. This is uniquely determined by the current state of the system and, considered in isolation, is independent of the system's previous history. Thus, in order to derive the causal relationship, the change of state from a defined point to the point under consideration must be considered.

$$\Delta \bar{\dot{H}}_{ab} = \bar{\dot{H}}_{a} - \bar{\dot{H}}_{b} \tag{22}$$

The change of enthalpy flow $\Delta \bar{\dot{H}}_{ab}$ is caused by the increase or decrease of technical work and by heat flows supplied and removed across the system boundary between state a and b. One of the possible heat flows must be the emitted heat flow in the point to be calculated (in this case point b). Losses within the system boundary do not lead to a change in enthalpy flow. The change in enthalpy flow can be expressed as follows:

$$\Delta \bar{\dot{H}}_{ab} = \bar{\dot{H}}_{a} - \bar{\dot{H}}_{b} = \sum_{i} \bar{\dot{W}}_{ab,i} + \sum_{j} \bar{\dot{Q}}_{ab,j} - \bar{\dot{Q}}_{fs,b} \tag{23}$$

Now, (21) and (23) form a system of equations, which can be solved by using (10), (17), (18), (19) and (20) to determine $\tilde{T}_{f}$, $\tilde{T}_{w1}$ and $\tilde{T}_{w2}$ at point b. If the state $\bar{\dot{H}}_{a}$ is known, for example by constant inflow conditions in an open hydraulic system, the state $\bar{\dot{H}}_{b}$, i.e. the temperatures at this point, can be calculated analytically. If the state $\bar{\dot{H}}_{a}$ is unknown, as in a closed hydraulic system, the temperatures must be calculated iteratively using an additional set

of (21) and (23) for state a. Therefore, all missing informations in terms of energy within the closed system have to be considered. This means that any number of points within the system, i.e. states, can be determined as long as all energy inputs and outputs within the system are taken into account.

2.3 Correction loop

As already mentioned, process quantities and fluid properties are determined by a functional-energetic simulation of a single duty cycle with constant fluid temperature. For this, an initial fluid temperature must be specified. Since this temperature can be arbitrary, it is obvious that the determined properties and quantities do not match those in the state of thermal equilibirum. This leads to incorrect power values and power losses in the system and thus to incorrect temperatures calculated by the thermal model. Therefore, a correction loop must be done. Thereby, the calculated fluid temperature calculated by the thermal model is set as the new initial temperature for 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 Fig. 3. The thermohydraulic system behavior is decoupled in a functional-energetic, i.e. only hydraulic, and a thermal lumped-parameter network model. The first mentioned model, which includes all mandatory power sources and sinks, is simulated over only one considered duty cycle, for example an y-cycle of a wheel loader ($t_{sim} = t_{cycle}$). This is done with an arbitrary initial fluid temperature. Since the functional-energetic model shall not take into account

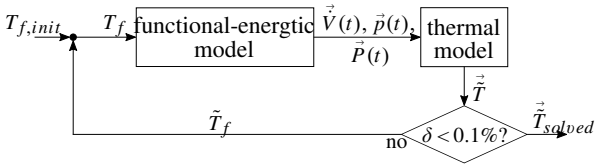

Figure 3 Correction loop within developed approach

the thermodynamic bevahior, the fluid temperature is constant over the whole cycle. The obtained process parameters are forwarded to the script based thermal model, where the averaged quantities are calculated in the first step. These are then used to iterate the temperatures within thermal equilibrium. Since the simulated power quantities belong to the initial fluid temperature, the determined fluid temperature $\tilde{T}_f$ is used as the new initial value. This

correction loop is performed until termination criterium is reached, i.e. the determined relative deviation is undercut.

3 Comparison with classical approach

3.1 Elementary hydraulic circuit

To validate the developed calculation approach, a comparison with the classical, i.e. time-dependent, approach is made. Therefore, an elementary hydraulic circuit was built. As mentioned above, the calculation of an open hydraulic system with known inflow conditions is just an analytical solution, since one state is defined by boundary conditions. Thus, the focus of the comparison is on a closed hydraulic system. The elementary circuit consists of a hydraulic pump with constant displacement, which is controlled by rotational speed via preset curves. Furthermore, a hydraulic motor with a constant moment is used to drain technical work from the system, which ensures same load conditions for varying volume flow curves. Pump and motor are connected by hoses, which are simplified modeled as a flow resistance and corresponding volume. On the low-pressure side, a pressure source is used to define a constant pressure of 10 bar and ensure robust calculation of initial values for any setting of initial conditions. To simplify the model, the behavior of the pump and motor is considered to be lossless. Nevertheless, a total loss is determined via a constant total efficiency and inserted as heat flow behind the pump and the motor, respectively. The resulting elementary circuit with external heat flows is shown in Fig. 4. The thermodynamical states, i.e. fluid and wall temperatures, are determined for the hoses. The heat transfer is modeled by convective heat flow from fluid into the mechanical structure, conduction within the structure and convection and radiation from the mechanical structure to the environment. The heat transfer coefficients required for convective heat transfer are determined using Nusselt correlations. Since these depend not only on geometric and fluidic parameters but also on the mass flow, they can be calculated time-dependently for arbitrary volume flow curves, which is necessary for correct averaging. For simplicity, the emitted heat flows of the mechanical structure of pump and motor are not included in this model, but can easily be added. There are no limits to the complexity of thermal discretization. In contrast to the developed calculation approach, in the classical calculation method all the above-mentioned functional and thermal aspects must be modeled in one simulation model in order to solve the mass and energy conservation correctly.

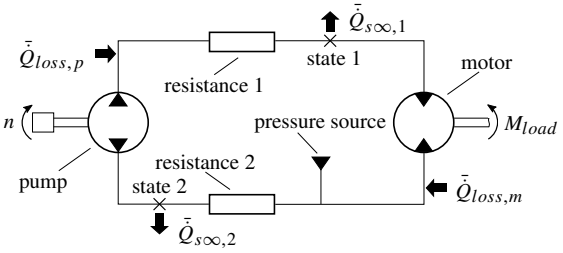

Figure 4 Elementary hydraulic circuit used for comparison

4 Statistical test cases

Due to the aforementioned assumptions, the uncertainty based on the averaged quantities and the non-linear dependence of the heat transfer coefficients on the volume flow rate, a deviation from the results of the classical calculation approach is to be expected. Therefore, test cases were generated using statistical methods in order to be able to provide a statement on the possible scope of the developed calculation approach. To cover a wide range of possible curve shapes within a process, a statistically representative number of 10^6 volume flow curves were randomly generated within a range of $0 - 400\frac{l}{min}$ and a process duration of $10s$, where a random volume flow rate value is assigned for each second. Since simulating 10^6 cases is too costly, a careful selection of 100 curves was made. Since these curves are intended to cover

Table 1 Volume flow rate related measures

measure	**description**	**equation**
D_1	mean value	$\frac{1}{t_{dc}-t_0} \cdot \int_{t_0}^{t_{dc}} \dot{V}(t)\,\mathrm{d}t$
D_2	maximum value / mean value	$\frac{\dot{m}_{max}}{D_1}$
D_3	dynamics	$\int_{t_0}^{t_{dc}} \lvert \frac{d}{dt}\dot{V}(t) \rvert\,\mathrm{d}t$
D_4	dynamics / mean value	$\frac{D_3}{D_1}$
D_5	mean absolute deviation	$\frac{1}{n} \cdot \sum_{i=1}^{n} \lvert \dot{V}_i - \bar{\dot{V}} \rvert$
D_6	mean abs. deviation / mean value	$\frac{D_5}{D_1}$

the entire spectrum of all curves, it is not expedient to randomly select 100 curves. Therefore, in a first step, six different measures related to the volume flow rate are introduced to define certain characteristics of the curves (Tab. 1). If the values of the measures are related to each other, they form a complex six-dimensional structure representing the entire spectrum of all curves. In order to achieve the best possible distribution within this structure, the latin hypercube sampling method is used in multiple steps. Therefore, measures are normalized to ensure equal weighting despite different magnitudes. This made it possible to select 100 curves that both cover the extrema in the boundary regions and have a good distribution within the entire spectrum. Fig. 5 shows exemplary volume flow curves with the maximum and minimum values of D_3 and D_5, respectively. The 100 volumetric flow curves, representing

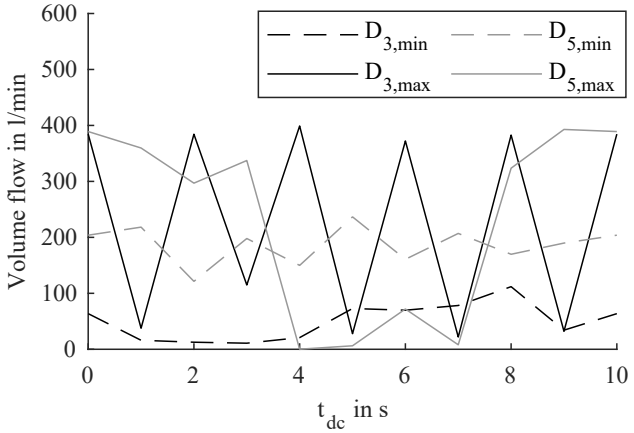

Figure 5 Exemplary volume flow curves for D_3 and D_5

the full spectrum of all randomly generated curves, are now used to compare the developed approach with the classical approach, allowing the potential scope to be estimated.

5 Results and validation

For the classical calculation of the elementary circuit and for the simulation of the functional-energetic model of the developed approach the multi-domain software SimulationX® is used. The thermal model of the presented approach is scripted and calculated in MATLAB® . The objective of the presented cal-

culation method is a significantly faster determination of temperatures within the thermal equilibirum of the complex system of a mobile machine. In the left graph of Fig. 6 the percentage of the computation time of the developed method from the simulation time of the classical method is shown versus absolute computation time of the classical method. It can be seen that the

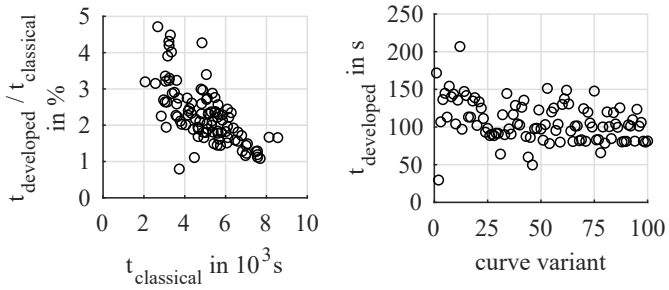

Figure 6 Simulation time of developed approach compared to classical approach

maximum proportion of simulation time is about 4.7% and minimum is about 0.8%, respectively. That means that the method calculates the state of thermal equilibrium 20 times up to 100 times faster for the elementary circuit. Furthermore, it can be seen that the proportion decreases with increased classical simulation time, i.e. with volume flow curves with higher dynamics and mean absolute deviations. Looking at the right graph of Fig. 6, where the simulation times of the developed method show a constant behavior across all simulated cases, it can be concluded that with an increase of the model complexity the proportion will behave similarly. To prove the potential scope of the developed calculation approach, the difference of simulated final temperatures to the classical approach are compared. Fig. 7 shows the absolute temperature

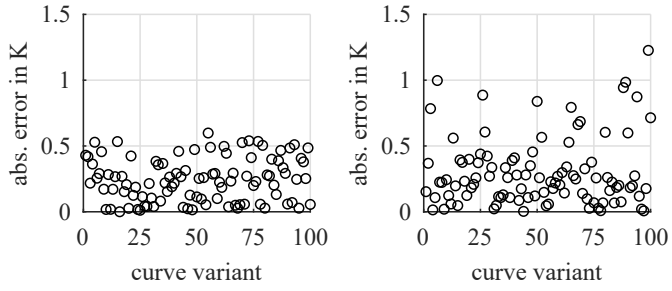

Figure 7 Abs. error compared to class. approach for state 1 (l.) and state 2 (r.)

differences for fluid temperatures of states 1 and 2 (range of absolute values is from approx. $30 - 130^{\circ}C$). It can be seen that the deviation of fluid temperatures across all analyzed curves is less then $0.6K$ in state 1 and less then $1.3K$ in state 2, respectively. In order to be able to interpret these differences correctly, they are shown as relative values in relation to the ambient temperature in Fig 8. The maximum relative deviation in both states across all curves is less then 2.2%. Hence, it can be said that the presented method

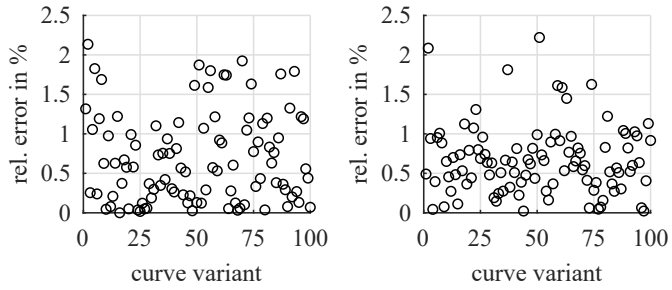

Figure 8 Rel. error compared to class. approach for state 1 (l.) and state 2 (r.)

not only significantly reduces the simulation time but also, in the case of the elementary circuit shown, can be used for arbitrary processes considering the small deviations of the final temperatures.

6 Conclusion

In this paper a calculation approach is presented, which reduces the dynamic heating process of any system to the computation of the steady-state final state using functional dynamic quantities as boundary conditions. This is based on the approach that the state of thermal equilibrium is a process-averaged consideration. Using the well known 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 the state of thermal equilibrium. A comparison with the classical approach, i.e. using time-dependent differential equations to solve mass and energy conservation, showed a significant reduction of simulation time by up to 100 times. Moreover, the small deviations from the classically calculated temperatures of maximum 2.2% using a carefully selected set of volume flow curves covering a wide range of process conditions showed that the method can be used for any processes, related to the considered elementary hydraulic circuit. Thus,

it can be assumed that the developed approach is applicable to any hydraulic system and, for systems with high complexity, such as the entire drive system of a mobile machine, it far outperforms the classical approach in terms of computational time, making it much more suitable for parameter studies and optimization.

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