Investigation of the Potential of Different Cooling System Structures for Machine Tools

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In the current cooling system structure of machine tools a central fixed pump provides a constant cooling volume flow to cool all the components of the machine tool. The provided cooling volume flow does not match the temperature development of each component. This may lead to some of the components heating up while the other components are simultaneously being cooled. Due to these temperature differences, a thermo-elastic deformation of the machine structure occurs. This deformation is responsible for the displacement of the Tool Centre Point (TCP) of the machine tools. Consequently, the machine’s accuracy during the production process is reduced.

The main goal of this paper is to analyse the thermal behaviour of the current cooling system structure of two demonstration machines and to present a simulative study of new cooling system structures under consideration. The investigation of this research will examine the effectiveness as well as the temperature characteristics of the components of the new structures under consideration comparing them to the current cooling system structure in order to ensure a uniform temperature distribution of the machine tool at minimal energy consumption.

The results show that the new concepts have great potential in respect to better thermal behaviour and lower hydraulic power compared to the current cooling system structure. The simulation results show a more stable temperature profile of the components as well as a lower energy consumption of the cooling system.

**Keywords:** machine tool, thermo-elastic deformation, cooling system, energy consumption, decentralized system

**Target audience:** Stationary Hydraulics, Manufacturing Industry, Machine Tool Producer

1 Introduction

Power losses of the machine tool caused by the manufacturing process are converted into thermal energy. Due to the temperature fluctuation, a thermo-elastic deformation of the machine structure occurs. This deformation is responsible for the displacement of the TCP of machine tools. Consequently, the accuracy of the machine during the production process is reduced. The warmed-up components such as the rotary table, tool holder, linear guide rails etc. need to be cooled. Therefore, cooling system is installed to reduce the temperature fluctuation of the components. In order to reduce the thermo-elastic deformations that occur and to enhance the production quality it is necessary to minimize the heat input. Previous research projects in this area mainly focused on reducing the energy demand of the machine tool and its main drives, reducing the energy consumption by developing more efficient components, and control strategies [1, 2, 3]. However, the analysis of the cooling systems and the investigation of their thermal behaviour has not yet been carried out in detail. Therefore, a detailed analysis of the existing cooling system structures, their thermal behaviour and their influence on the deformation of the machine structure is necessary in order to ensure a uniform temperature distribution of the machine tools at minimal energy consumption. Previous research activities of this project, which were carried out by the authors and focused on two demonstration machines, showed that sufficient cooling capacity in the cooling system is available but that the cooling is insufficiently adjusted to the process and to the individual demand of the machine components 4/5, 6/7. In order to address this deficit, it is necessary to consider and analyse the potential of new structures for cooling systems.

The main target of this study is to highlight the latest project activity regarding the investigation of the thermal behaviour of the current cooling system structure and to present a simulative study of new cooling system structures based on the validated simulation models of the current cooling system structures of two demonstration machines. The research will help obtain information concerning the effectivity of the new cooling system structures under consideration as well as the temperature characteristics of the components compared to the current cooling system structure.

2 Design of the cooling system structures of the demonstration machines

The main function of a cooling system is to provide the cooling media for the components or spots of the machine to dissipate the heat energy and to avoid high temperature fluctuations within the machine structure. This helps to reduce the thermo-elastic deformation and finally increases accuracy in the production process. Figure 1 illustrates the cooling system circuit of the demonstration machine type Scharmann DBF630 investigated first (machine 1). The cooling circuit consists of three components to be cooled, electrical cabinet 6, rotary table 4, and main spindle 7. Furthermore, a central fixed pump 14 provides the cooling medium (40 l/min at 5.5 bar) to the three components. The calculated hydraulic power ($Q \cdot \Delta p$) of the pump is about 370 W. Usually, the cooling medium used in cooling systems of machine tools is a mixture of water (60 % - 80 %) and Antifrogen® (40 % - 20 %). Moreover, the cooling unit 13 is placed directly into the return flow side and cools down the heated fluid to a set temperature. The function principle of the cooling unit (Counter-Clockwise Carnot Cycle Process or Clausius-Rankine-Process) is a two-point temperature-controlled refrigerator. It is turned on once the temperature overstrides the upper threshold of the set temperature and is turned off when the cooling medium is cooled to the lower temperature set. The cooling of the electrical cabinet is carried out by an air heat exchanger 12. The main spindle 7 and the rotary table 4 are cooled directly by the cooling medium that flows through integrated cooling channels.

![Figure 1: Cooling system of DBF630 (machine 1)](image-url)

In contrast to machine 1, the cooling system of the second investigated demonstration machine type DUM60 eVo linear (machine 2) shown in Figure 2 simultaneously cools 13 components. Here, a fixed displacement pump 1 supplies the cooling medium (45 l/min at 4.5 bar) to the motor spindle 2, all the axes drives (3, 5, 7, 8), the housing of the B and C axes 4 as well as the rails of X, Y, and Z (6, 9-12). The calculated hydraulic power of the pump is approximately 340 W. The electrical cabinet is not cooled directly by the cooling system but by a separate cooling unit. Moreover, the cooling unit 13 of machine 2's cooling system is not integrated in the return flow as in machine 1; it is mounted directly to the tank. Furthermore, a three-way valve 14 is placed into the return flow side.
This valve is used as a diverting valve, so, with a defined setting, a part of the heated backflow is introduced directly to the inlet side of the pump, and the remaining fluid flows back to the tank! The controller of the three-way-valve adjusts the flow to the tank or to the inlet side of the pump so that the temperature on the pump inlet side always stays at 25°C.

Figure 2: Cooling system of DMU80 eVo linear (machine 2)

3 System model of the current cooling structures

Simulation models provide a flexible method for studying the behaviour of a system. In order to be able to give reliable statements about the simulation models, it is necessary to analyse the behaviour of the real system with the appropriate mathematical and physical approaches [6]. The methodology of the model development is based on thermo-hydraulic network modelling. For the hydraulic and thermodynamic domains, the laws of electrical engineering, such as rules of Kirchhoff’s circuit laws of series and the parallel connection of resistances can be used. The calculation of the hydraulic and the thermal resistance \( R_{ha} \) and \( R_{th} \) between two nodes is based on the following equations [6]:

\[
R_{ha} = \frac{\Delta \rho}{Q} \tag{1}
\]

\[
R_{th} = \frac{\rho}{V} \tag{2}
\]

In respect of the Kirchhoff’s node role

\[
T = \frac{1}{c_m} \sum_{i=1}^{n} Q_i \cdot dt \tag{3}
\]

\[
p = \frac{1}{c_m} \sum_{i=1}^{n} V_i \cdot dt \tag{4}
\]

Additionally, the convective heat transport through the cooling medium in the hydraulic hoses caused by the forced convection is taken into account. The corresponding heat transfer coefficient in Figure 3 is determined by the following equations [9]:

\[
Re = \frac{V_{av}}{\nu} \tag{5}
\]

\[
Pr = \frac{c_{fluid} \cdot \rho_c \cdot V}{\lambda_{fluid}} \tag{6}
\]

\[
Nu = 0.0235 \cdot (Re^{0.8} - 230) \cdot (1.8 \cdot Pr^{0.3} - 0.8) \cdot (1 + \left( \frac{d_i}{L} \right)^{0.25} \cdot \left( \frac{D_c}{D_i} \right)^{0.5})^{0.14} \tag{7}
\]

\[
\sigma_{outside} = \frac{\lambda_{fluid} \cdot Nu}{L} \tag{8}
\]

Applying exemplarily equations (5) to (8) to the hydraulic hose from the flow valve to the motor spindle (\( l = 5 \) m; \( d_i = 9 \) mm; \( d_o = 12 \) mm; volume flow \( V = 9.44 \) l/min, \( \lambda_{fluid} = 0.443 \) W/m K) of machine 2, a heat transfer number of \( \sigma_{outside} = 8.1 \) W/m² K is calculated. Furthermore, the free convection heat transfer at the outer surface (Figure 3) is considered. The related heat transfer coefficient is calculated by following equations [10]:

\[
\beta = \frac{1}{\lambda_{out}} \tag{9}
\]

\[
Gr = \beta \cdot \left( \frac{D_o^3 - (D_i - T_o)}{\nu^3} \right) \tag{10}
\]

\[
Ra = Gr \cdot Pr \tag{11}
\]

\[
Nu = \left[ 0.6 + \left( \frac{0.387 \cdot Ra^{1/6}}{1 + (0.559 / Pr)^{0.8}} \right)^2 \right] \text{ for } 10^{0.5} < Ra < 10^{12} \text{ and } 0 < Pr < \infty \tag{12}
\]

\[
\sigma_{outside} = \frac{\lambda_{out} \cdot Nu}{L} \tag{13}
\]

Regarding to equations (9) to (13) to the mentioned hose at an ambient temperature of 25°C, a heat transfer number of \( \sigma_{outside} = 4.7 \) W/m² K is obtained. Moreover, the heat transfer through the heat conduction in the hose is determined by equation (14), for the exemplarily hose (\( \lambda_{out} = 0.42 \) W/m K) amounts \( \sigma_{con} = 243 \) W/m² K. For other hoses the same method is used to calculate the inner, outer, and conductive heat transfer numbers.

\[
\sigma_{con} = \frac{2 \cdot \lambda_{con} \cdot \ln \left( \frac{d_o}{d_i} \right)}{d_a \cdot \ln \left( \frac{d_o}{d_i} \right)} \text{ for cylinder shape} \tag{14}
\]

A simulation model of the current cooling system structure for the two machines is developed based on the modelling methods and machine documentations. The simulation model exemplified by machine 2 in Figure 4 consists mainly of a pump, flow valve, hydraulic hoses, a cooling unit, and the 13 components to be cooled as heat sources. The Table 1 gives an overview about the most important model parameters for the simulation model. Hydraulic connections or hoses are modelled by hydraulic volumes and hydraulic resistances. The geometrical parameters, such as length, inner and outer diameter are taken directly from the machine documentations.
drive B axis and spindle nut Z axis is higher than the outlet temperature. Other components, such as the motor spindle or the secondary part X axis, are cooled during the test process. In contrast to machine 1, the temperature characteristics of the components in machine 2 are not influenced directly by the state of the cooling unit, in spite of the two-point temperature control of the cooling unit. This can be traced back to the three-way-valve (Figure 2, pos.14) that is placed into the return flow side of the cooling system. The valve controls the suction flow to the pump so that the cooling medium always has a constant temperature of 25 °C ± 1 °C as shown in Figure 6 e.

![Diagram showing model development of cooling system in the simulation software exemplified by machine 2](image)

Table 1: Model parameter for the simulation model based on the model description in Figure 4

<table>
<thead>
<tr>
<th>No.</th>
<th>Element</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Thermal or hydraulic capacity</td>
<td>Geometry of the connections</td>
</tr>
<tr>
<td>2</td>
<td>Thermal resistance</td>
<td>$\varphi_{\text{inside}}, \varphi_{\text{outside}}, \varphi_{\text{cool}}$, fluid properties, pipes properties, ambient properties</td>
</tr>
<tr>
<td>3</td>
<td>Hydraulic resistance</td>
<td>Description form in the simulation software, $\alpha(Re)$, $\lambda(Re)$ laminar resistance, $(\Delta p, q)$ characteristic curve, reference measurement</td>
</tr>
<tr>
<td>4</td>
<td>Heat input</td>
<td>With aid of measurement is calculated by the equation (15)</td>
</tr>
<tr>
<td>5</td>
<td>Heat output</td>
<td>Calculated by forced convection, free convection and heat convection of the hoses, equations (5-14)</td>
</tr>
<tr>
<td>6</td>
<td>Flow valve</td>
<td>Characteristic curve of the valve</td>
</tr>
<tr>
<td>7</td>
<td>Pump</td>
<td>Flow rate and system pressure</td>
</tr>
<tr>
<td>8</td>
<td>Tank</td>
<td>Tank capacity e.g. 15 l</td>
</tr>
<tr>
<td>9</td>
<td>Cooling unit</td>
<td>Data sheets of the cooling unit, e.g. cooling capacity 4.5 kW</td>
</tr>
</tbody>
</table>

4 Model validation of the current cooling structures

For the experimental investigation of the cooling system of machines 1 and 2, several sensors, as shown in Figure 1 and Figure 2, are used to measure the temperature, the pressure, and the flow rate development. The measured process taken into consideration for the investigation is divided into four sub-processes: warm-up process, idle process, (variation of spindle speed, axis position while cooling system is active), setup process (tool/pa change

![Graphs showing comparison of temperature development simulation and measurement in the idle process of machine 2](image)

Figure 6: Comparison of temperature development simulation and measurement in the idle process of machine 2

- a) Motor spindle
- b) Drive B axis
- c) Spindle nut Z axis
- d) Secondary part X axis
- e) Pump inlet temp.
- f) Tank

The simulation models developed for machines 1 and 2 show a high accuracy of the thermal and hydraulic quantities of the components /13/. The simulation model is validated and thus used for the improvement of the current structure of the cooling system as well as for the development of process- and demand-oriented control strategies. The considered heat transport in Figure 5 by enforced and free convection as well as by heat conduction of the total hydraulic hoses amount to 45 W as shown in Figure 7. Compared to the performance of cooling units 4.5 kW, the heat transfer through the hydraulic pipes is low, about 1%. It could be neglected in the future. The investigation of two demonstration machines shows that the cooling system of each machine in the idle process requires about 12 % (machine 1) and 26 % (machine 2) of the total energy consumption of the machine tool /4, 12/. The proportion of calculated hydraulic energy of the pump according to equation (16) is 37.2 % of 12 % (machine 1) and 18.5 % of 26 % (machine 2), the remaining energy 72.8 % of 12 % and 81.5 % of 26 % is consumed by cooling unit, electrical motor of the hydraulic pump and the pump efficiency. The investigation also depicts that these cooling systems as currently structured do not cool the warmed-up component based on their
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![Figure 6: Comparison of temperature development simulation and measurement in the idle process of machine 2](image)

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\[ E_{Phy} = \int_0^t \dot{E}_{Phy} \, dt \]

(16)

![Figure 7: Total heat transport through the hydraulic hoses of the machine 2](image)

5 Potential analysis of different cooling system structures

In relation to the shown deficits of the current cooling system structures of machines 1 and 2, the focus is developing new structures for the cooling system to optimize its thermal behaviour and its effectiveness according to the goal of a uniform temperature distribution at minimal energy consumption. Figure 8 shows three new structures of a cooling system that can be applied for a demand-oriented supply, in the subchapter the new structures under consideration are described in detail. The effectiveness of the new structures will be evaluated firstly in regard to a constant temperature behaviour at the components, a minimal pressure loss, and a minimal hydraulic power of the pumps. The new structure show also the possibility of the degree of the individualization for new developed cooling system structure.

![Figure 8: New structures for a cooling system](image)

5.1 Structure 1: Central, variable speed drive unit with proportional valves

The cooling concept considered with a central variable speed drive unit with proportional valves (structure 1) is presented in Figure 9 by three components, electrical cabinet, rotary table, and motor spindle (main drive). The components are cooled individually so that the system control of the cooling system compares the actual and set temperature of the individual components, and, on this basis, adjusts the proportional valves as well as the central variable speed drive unit. The temperature detection is carried out via sensors in the components. For this purpose,
a suitable sensor concept is necessary. If the temperature development of a component does not exceed a predefined threshold, the associated proportional valve will stay closed. With regard to the control strategy, Figure 9 shows that the cooling structure under consideration has three control variables (component’s temperatures) and four control elements (three proportional valves and a variable pump). This makes the system with actual concept over-determined. To solve this problem, three approaches can be taken into account [13]:

- definition of a constraint
- removal of a control element from the active control loop and
- definition of an additional control variable.

For the potential analysis of new cooling structures as well as current cooling structure, three set temperatures are defined for the components: 26 °C for the electrical cabinet, 27 °C for the rotary table and 28 °C for the motor spindle. Based on the measurement, an average equivalent heat flow in the idle process for each component is calculated with equation (15). It is 150 W for the electrical cabinet and rotary table and 1500 W for the motor spindle. Only static operating points of the cooling system are considered in the simulation so that the thermal capacity of the components is not required. Furthermore, the system inlet temperature on the suction side of the pump is considered in the simulation at 25 °C. The cooling unit stays in the two-point temperature control as a bypass cooling and refers to the mixing temperature of the tank.

In Figure 10, the simulation results of the components’ temperature development and the needed cooling medium volume flow of structure 1 in comparison to the current structure of machine 1 are represented. The diagrams (a–c) show the temperature profile of the component and diagram (d) depicts the volume flow development. From the comparison of the simulation results is possible to derive that the thermal behaviour of the component in structure 1 is more stable than in the current structure of machine 1. The components’ temperature in structure 1 keeps constant at the set temperature (26 °C, 27 °C and 28 °C) despite the increase of heat input. In the current structure, the component temperature rises with an increase in the heat input. For this reason, the actual temperature of the components is dependent on the heat input to the component. Looking more closely at the volume flow profiles of the proportional valve of each component, it should be noticed that the cooling volume flow in structure 1 increases based on the temperature development (rising heat inputs) of the components. In contrast, in the current cooling structure the supplied volume flow to the component is constant. Finally, it is possible to achieve a demand-oriented supply with the considered structure and its controlling. Dependent on the heat input and the temperature development of the component, the proportional valve regulates the required volume flow. It can be ascertained that the volume flow control based on the temperature development is a means for designing the system in a more energy-efficient way based on each individual component’s demand.

The total hydraulic power of the pump in the current structure and structure 1 for different heat input is pointed out in Figure 11. With the variable central displacement pump (structure 1), the total hydraulic power is approximately 160 W at a maximal heat input. Compared to this, the hydraulic power of the fixed displacement pump (current structure) of machine 1 amounts to 370 W (40 l/min at 5.5 bar) and of machine 2 to 340 W (45 l/min at 4.5 bar). A significant energy savings of 56.7 % to 53 % in contrast to the current structures of machines 1 and 2 are possible. The new cooling structure under consideration is significantly more energy-efficient compared to the current structures of machines 1 and 2 with a continuous cooling volume flow.

5.2 Structure 2: Decentralized, variable speed drive units without flow control valves

Figure 12 shows the second optimization structure of the cooling system structures under consideration. In structure 2, the components are cooled with individual variable speed pumps that are connected to a common tank and a cooling unit. This kind of cooling structure design does not require flow control valves to distribute the volume flow to the components. Additionally, the system control of the cooling system compares the actual and the set temperature of the individual components and on this basis adjusts the variable speed drive units. So, each pump supplies a different demand-oriented cooling volume flow. If the temperature development of a component
does not exceed predefined threshold, the pump remains inactive. As well as in the structure 1, the cooling unit stays in the two-point temperature control as a bypass cooling and refers to the mixing temperature of the tank. The system boundary condition of structure 1, in regard to the components’ set temperature, heat input, system inlet temperature and static operating points of the system, applies also to structure 2.

Figure 12: Control strategy of structure 2

Similar to structure 1, Figure 13 shows the simulation results of the components’ temperature development and the required cooling volume flow of structure 2 in comparison to the current cooling system structure of machine 1. Since the components’ temperature remains constant at the set temperature despite an increase in heat input, structure 2 also evidences very good thermal behaviour of the components. A demand-oriented supply of the cooling medium to the components can be realized with this structure. In Figure 14, the total hydraulic power of the variable speed displacement pumps is about 130 W at maximal heat input. An essential energy savings of 67 % to 64.7 % in contrast to the current structures of machines 1 (370 W) and 2 (340 W) can be shown. So the new cooling structure under consideration is more energy-efficient compared to the current cooling structures.

Figure 13: Simulation results of cooling structure 2 under consideration in comparison of the current cooling structure a) electrical cabinet (EC) b) rotary table (RT) c) motor spindle (MS) d) volume flow profile

Figure 14: Comparison of the pump performance in the current system and in system structure 2

6 Summary and outlook

The TCP-displacement of machine tools influences the accuracy in the production process. In order to minimize the deformation of machine tools structure, it is necessary to reduce the thermo-elastic deformation of the machine components with the aid of fluidic systems such as a cooling system. Therefore the requirements on cooling systems in a machine tool with regard to their effectiveness (targeted cooling) and efficiency (lower energy consumption), is too high.

The experimental investigation of a cooling system for two different machine tools has been instrumental in determining the effectiveness as well as the energy consumption of the cooling system /4, 6/. It could be shown that the machine components are not cooled specifically with the current cooling structures, and that the cooling is adjusted insufficiently in reference to the component demand and process requirement. Therefore, the investigation and evaluation of new cooling concepts, both simulative (network models) and experimentally (test rig), is of great importance.

The examined new cooling structures in this paper, central, variable speed drive unit with proportional valves (structure 1) and decentralized, variable speed drive units without flow control valves (structure 2), show high accuracy with respect to the temperature control of the components compared to the current cooling structure. Apart from that, the hydraulic pump performance of the new structures is about 53 % to 67 % lower than the hydraulic pump of the current cooling structures.

The focus of further research of the projects will address, firstly, the evaluation of decentralized, variable speed drive units, tanks and cooling units (structure 3), compared to the current structure as well as to structure 1 and 2. Secondly, an energetic analysis is to be made of the overall system for each structure considered, the energy consumption of the electrical motor, frequency converter etc. Lastly, the new structures under consideration shall demonstrate their benefit practice and not only in simulation. To this end, a test rig is being developed, which will allow an experimentally sound statement about the structures regarding their effectiveness and efficiency.

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Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
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</table>
\( A \) Area of the hydraulic connection \[ \text{[m]} \]

\( c_{\text{fluid}} \) Specific heat capacity at constant pressure \[ \text{[J/kg\cdot K]} \]

\( C_{\text{th}} \) Thermal capacity \[ \text{[J/K]} \]

\( C_{\text{hyd}} \) Hydraulic capacity \[ \text{[m}^3/\text{Pa]} \]

\( D_H \) Hydraulic diameter \[ \text{[m]} \]

\( D \) Outside diameter of the hydraulic connection \[ \text{[m]} \]

\( d_o \) Outer diameter of the hydraulic connection \[ \text{[m]} \]

\( d_i \) Inner diameter of the hydraulic connection \[ \text{[m]} \]

\( g \) Gravity \[ \text{[m/s}^2\text{]} \]

\( Gr \) Grashof number \[ \text{[-]} \]

\( l \) Length \[ \text{[m]} \]

\( L \) Characteristic length \[ \text{[m]} \]

\( N_u \) Nusselt number \[ \text{[-]} \]

\( R_a \) Rayleigh number \[ \text{[-]} \]

\( R_e \) Reynolds number \[ \text{[-]} \]

\( p_{\text{outlet}} \) Component outlet pressure \[ \text{[Pa]} \]

\( p_{\text{inlet}} \) Component inlet pressure \[ \text{[Pa]} \]

\( \Delta p \) Pressure difference \[ \text{[Pa]} \]

\( Pr \) Prandtl number \[ \text{[-]} \]

\( \dot{Q}, \dot{Q}_{\text{heat}} \) Heat flow \[ \text{[W]} \]

\( \dot{V} \) Volume flow \[ \text{[m}^3/\text{s]} \]

\( t \) Time \[ \text{[s]} \]

\( a_{\text{inside}} \) convective heat transfer coefficient inside the hydraulic connection \[ \text{[W/m}^2\text{K]} \]

\( a_{\text{outside}} \) convective heat transfer coefficient outside the hydraulic connection \[ \text{[W/m}^2\text{K]} \]

\( a_{\text{con.}} \) convective heat transfer coefficient in the hydraulic connection \[ \text{[W/m}^2\text{K]} \]

\( \beta \) Coefficient of thermal expansion \[ \text{[1/K]} \]

\( T_{\text{outlet}} \) Component outlet temperature \[ \text{[K]} \]

\( T_{\text{inlet}} \) Component inlet temperature \[ \text{[K]} \]

\( T_w \) Wall temperature of hydraulic pipe \[ \text{[K]} \]

\( T_{\text{amb}} \) Ambient temperature \[ \text{[K]} \]

\( \Delta T \) Temperature difference \[ \text{[K]} \]

\( \lambda_{\text{fluid}} \) Thermal conductivity of the cooling medium \[ \text{[W/m\cdot K]} \]

\( \lambda_{\text{air}} \) Thermal conductivity of the air \[ \text{[W/m\cdot K]} \]

\( \lambda_{\text{con.}} \) Thermal conductivity of the hydraulic connection \[ \text{[W/m\cdot K]} \]

\[ \nu \] Kinematic viscosity \[ \text{[m}^2/\text{s]} \]

\[ \rho \] Density \[ \text{[kg/m}^3 \text{]} \]

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