A Comparison of the Dynamic Temperature Responses of Two Different Heat Exchanger Modelling Approaches in Simulink Simscape for HVAC Applications

Samuel F. Fux, Babak Mohajer and Stefan Mischler
Belimo Automation AG, Brunnenbachstrasse 1, 8640 Hinwil, Switzerland

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Abstract: In the HVAC industry, the dynamic temperature response of water-to-air heat exchangers is of particular importance for control system design. In this paper, the dynamic temperature responses of two established thermal dynamic modelling approaches for heat exchangers, the single-segment modelling using the effectiveness-NTU method and the multi-segment modelling, are investigated. Both approaches are validated against experimental data recorded with two different heat exchangers used in HVAC systems. A quasi-static analysis reveals minor differences between the results of the two models considered. The dynamic analysis is performed with varying inlet conditions. First results show that the single-segment model may fail to properly reproduce the water outlet temperature dynamics of a heat exchanger under certain conditions. In the tests performed in this study, however, the multi-segment model captures the relevant dynamics. The influence of this difference in the dynamic behaviour of the single-segment model on the model-based development of control algorithms is subject of future studies.

1 INTRODUCTION

In heating, ventilation, and air conditioning (HVAC) systems in buildings, typically, water-to-air heat exchangers are used to condition the temperature of the supply air. To increase the occupant comfort in the building and to decrease the energy use of the HVAC system, optimal operation of the heat exchanger is crucial. The development and testing of new control algorithms for the operation of HVAC heat exchangers can be simplified and accelerated by using building simulation environments with an appropriate dynamic model of the heat exchanger (Zhou and Braun, 2004). An appropriate dynamic heat exchanger model must be able to accurately predict the water and air outlet temperatures not only during steady state operation but also during transients when the inlet conditions are changing. Furthermore, it is important that the model is accurate across the entire operating range and not only at full load.

In the past decades, authors presented different approaches for modelling the dynamic temperature behaviour of HVAC water-to-air heat exchangers for the purpose of control performance analysis. For instance, (Underwood, 1990) developed a heat exchanger model based on single energy balance equations for the water and the air side, respectively. The same lumped parameter approach is also employed by, e.g., (Zajic, Larkowski, Sumislawska, Burnham, and Hill, 2011) and (Afram and Janabi-Sharifi, 2015).

In (Zhou and Braun, 2004), a model is presented where the heat exchanger is divided into a series of basic elements. Each basic element represents a cross-flow finned tube. Then, a transient model for the basic element is introduced, which considers energy storage in the water and in the tube and fin material. This approach of discretising the heat exchanger into multiple smaller elements is adopted by (Jie and Braun, 2016).

A completely different approach is introduced by (Anderson, 2001), who proposes a linear model at an operating point. This model is formed by combining several first-order transfer functions and time delays.

The goal of this paper is to assess the suitability of two popular modelling approaches for testing control algorithms. For that purpose, two different heat exchanger models are compared with temperature measurement data from real heat exchangers. Both

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models are implemented in the MATLAB® Simulink® simulation environment using the Simscape modelling language (The MathWorks, Inc., 2023a).

The paper is structured as follows: In Section 2, an overview on the structure of HVAC heat exchangers is provided. Section 3 introduces the two models considered in this study. The comparison of the two models with measurement data is shown in Section 4. Finally, the conclusions of this study are summarized in Section 5.

2 STRUCTURE OF HVAC HEAT EXCHANGERS

Heat exchangers are usually characterized in one of the four major construction types: tubular, plate-type, extended surface, and regenerative heat exchangers (Shah and Sekulić, 2003).

One of the commonly used water-to-air heat exchanger types in the HVAC industry are the extended surface exchangers. To compensate the low heat transfer coefficient on the air side and to increase the efficiency, fins are being used to extend the surface area up to 5 – 12 times the primary surface area (Shah and Sekulić, 2003), see Figure 1.

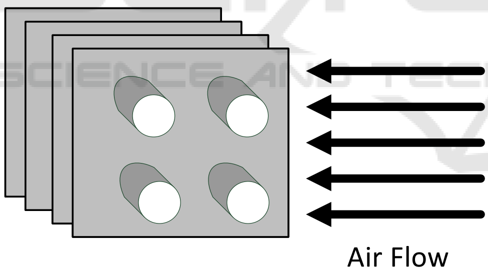

Figure 1: Finned tubular cross flow heat exchangers with both fluids unmixed.

This paper focuses on the modelling of fin and tube heat exchangers and their circuiting pattern. The circuiting pattern of the water tubes is one of the important aspects which influences the performance of the heat exchanger. The number of tube rows mark the depth and the number of tubes in each row mark the height of the heat exchanger. Furthermore, the number of circuits in a heat-exchanger is defined by the number of tubes that are fed by the supply header (Campbell Sevey, 2023). As an example, a quarter circuit heat exchanger with two feeds is shown in Figure 2.

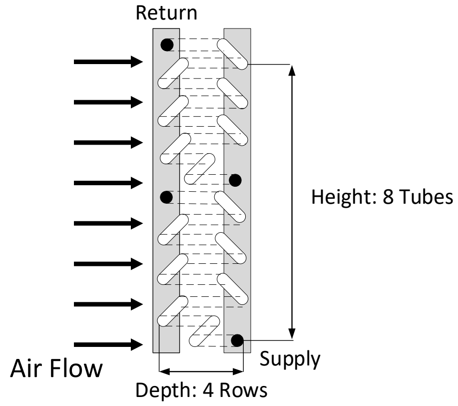

Figure 2: Quarter circuit heat-exchanger (adapted from (Campbell Sevey, 2023)).

Determining the proper number of circuits, the speed of the fluid in the tubes as well as the resulting pressure drop can be controlled. This both aspects have a direct impact on the heat transfer efficiency of the heat exchanger (Campbell Sevey, 2023).

While talking about the circuiting it is important to also consider how often the air crosses the inner tubes perpendicularly (number of passes).

Common flow arrangements of water-to-air heat exchangers are: (VDI Gesellschaft Verfahrenstechnik und Chemieingenieurwesen, 2019):

- multi-row, single-pass
- multi-row, multi-pass
- multi-row, two-pass

In Figure 3, a flow scenario with 4 rows and 4 passes is illustrated. In this flow arrangement the tube rows are connected in series with alternating flow directions in each row. The outside air crosses the tubes 4 times.

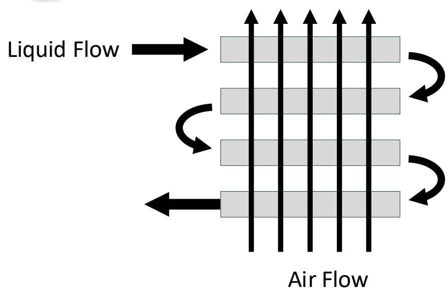

Figure 3: Crossflow definition for 4 rows and 4 passes (adapted from (VDI Gesellschaft Verfahrenstechnik und Chemie-ingenieurwesen, 2019)).

Considering the above-mentioned aspects, it is crucial to take the circuiting arrangement into account when modelling a heat exchanger.

3 HEAT EXCHANGER MODELS

As mentioned in the introduction, there exists a variety of dynamic models to predict the temperature behaviour of HVAC water-to-air heat exchangers. However, these models are usually based on one of the following basic approaches:

- Simple single-segment models, in which the water and air sides are entirely lumped into single thermal capacities each.
- Detailed multi-segment models, in which the heat exchanger is divided into a fixed number of smaller segments.

Furthermore, the various models differ in how detailed they compute the heat transfer between the water and the air side and if they consider the heat capacity of the tube and fin material or not.

In this paper, an example of a single-segment model and an example of a multi-segment model are evaluated using measurement data. The two models are presented in the following two subsections.

3.1 Single-Segment Model

The first model considered in this study is the heat exchanger model provided with the Simscape Fluids component library (The MathWorks, Inc., 2023b). Since only one energy conservation equation is formulated for each of the water and air sides, this model belongs to the category of single-segment models.

This model uses the effectiveness-NTU method to compute the heat transfer between the water and the air side. It can also consider water condensation on the air side. However, the thermal capacity of the tubes and fins are not considered. In addition to the temperature dynamics, the model also computes the pressure losses across the heat exchanger. Please refer to (The MathWorks, Inc., 2023c) for a detailed description of the model.

Important input parameters required by the model include the following: flow arrangement and geometries, length of the tubes, tube outer diameter, number of rows, number of tubes per row, longitudinal and transverse tube spacing, fouling factors, total fin surface area, and constant fin efficiency. The model does not require the input of parameter values which need to be identified using measurement data (e.g., convective heat transfers coefficients).

3.2 Multi-Segment Model

The second model used in this study is a multi-segment model developed by the authors.

The development of this model is based on the assumptions typical for this type of model (e.g., (Zhou and Braun 2004) or (Mathisen, Morari, and Skogestad, 1994)):

- all segments have the same size,
- ideal mixing of the water in each segment,
- ideal mixing of the air in each segment,
- the tube and fin material separating water and air side have a uniform temperature distribution in each segment,
- heat conduction between adjacent segments is neglected,
- heat losses from the heat exchanger to the surroundings are not considered.

The division of a heat exchanger into smaller segments is exemplified in Figure 4. The schematic of one segment of the multi-segment model is then depicted in Figure 5.

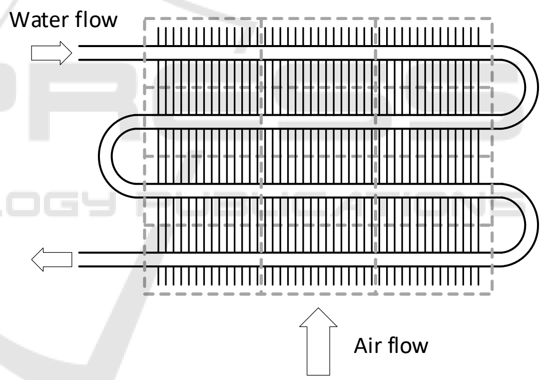

Figure 4: Example of the division of a heat exchanger into multiple smaller segments.

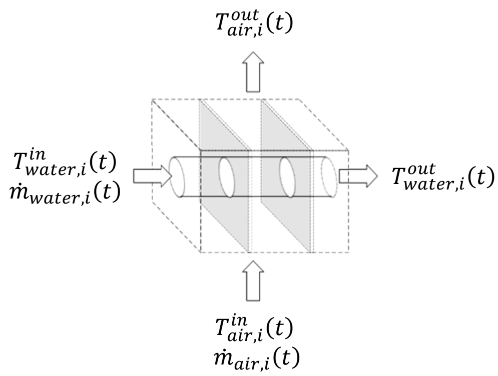

Figure 5: Schematic of the i-th segment of the multi-segment heat exchanger model.

In this model, energy conservation equations are formulated for the water side, the air side and for the metal mass. Obviously, the energy conservation equations account for the fluid flows through the segment and the convective heat transfers between the fluids and the metal surfaces.

The required geometric dimensions of the heat exchanger are computed in the model as presented in (Réz, 2004) and (Vetter, 2014). The fin efficiency of the continuous plate fins is calculated considering an equivalent fin radius as given in (Perrotin and Clodic, 2003).

Finally, the convective heat transfer coefficient on the air side is computed using the equivalent diameter D_e and the appropriate empiric correlations for the Nusselt number (Réz, 2004).

The multi-segment model is implemented in Simulink® using the Simscape modelling language, too. Except for the calculation of the convective heat transfer coefficient on the air side, the implementation of the model is based on component models available in the Simscape Foundation library. Consequently, the model also considers water condensation.

This model requires similar input parameters as the single-segment model presented above. However, it must be mentioned that the fin efficiency is not required since it is calculated online in the model.

4 TEST RESULTS

The two models presented above are evaluated using measurement data recorded with one heating and one cooling heat exchanger. Some properties of the two heat exchangers are given in Table 1 and Table 2, respectively.

The heating heat exchanger is installed in a test rig that is located in an indoor room. Therefore, the inlet air to the heat exchanger is the same as the room air. For this reason, the inlet air temperature is unrealistically high for an HVAC heating application. However, since the temperature dependency of the air properties are considered in both models, this is not deemed to be a problem for the validation of the models.

On the other hand, the cooling heat exchanger is installed in a real ventilation system that is used for the conditioning of a break room in an office building.

Both heat exchangers are equipped with temperature sensors at the inlet and outlet of the water side and the air side. For the air temperature sensors, it must be pointed out that they measure the air stream temperature only at one location in the air duct. Therefore, if the air stream does not have a uniform temperature distribution across the duct cross section, the accuracy of the temperature measurement must be challenged.

Table 1: Properties of the heating heat exchanger.

Property	Value
Height	0.398 m
Width	0.7 m
Number of rows	2
Number of tubes per row	16
Number of circuits	8

Table 2: Properties of the cooling heat exchanger.

Property	Value
Height	0.24 m
Width	0.369 m
Number of rows	4
Number of tubes per row	6
Number of circuits	1

Furthermore, there are water volume flow rate meters in the connecting water pipes of both heat exchangers. Additionally, measurement data of the air volume flow rate through the cooling heat exchanger is available. The heating heat exchanger is not equipped with a sensor measuring the air volume flow rate. Therefore, the air volume flow rate must be estimated based on an energy balance across the heat exchanger.

For the evaluation, various measurement data sequences are recorded with the heat exchangers. The recorded measurement data of the inlet conditions, i.e., water and air inlet temperatures and water and air volume flow rates, is then used as inlet conditions during the simulations with the two models. Finally, the simulated outlet temperatures are compared with the measured outlet temperatures.

In the so-called quasi-static tests, the water volume flow rate through the heat exchangers is slowly ramped up and down to avoid the excitation of dynamics in the heat exchangers. The other inlet conditions are kept as constant as possible during the recording of these measurement sequences. These quasi-static tests are used to assess the accuracy of the models across a wide range of operating points. For assessing the ability of the models to correctly capture the temperature dynamics, tests with fast varying inlet conditions are used. These tests are called dynamic tests in the following.

The comparison of the heat exchanger models with the measurement data of the quasi-static and the dynamic tests are shown in the following two subsections.

4.1 Quasi-Static Tests

The inlet conditions during the quasi-static test with the heating heat exchanger are shown in Figure 6.

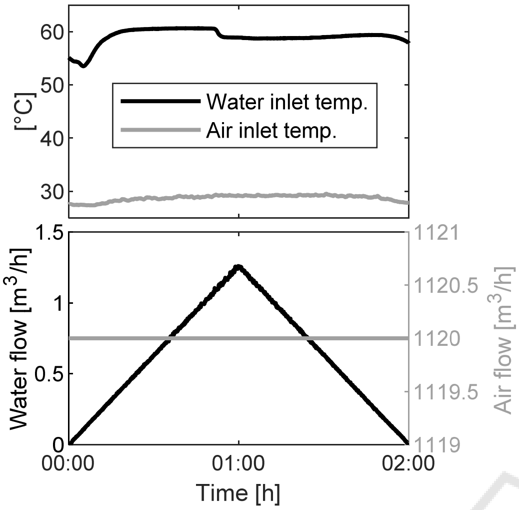

Figure 6: Inlet conditions for the quasi-static test with the heating heat exchanger.

The comparison of the two models with the measurement data of the water and air outlet temperatures is then plotted in Figure 7.

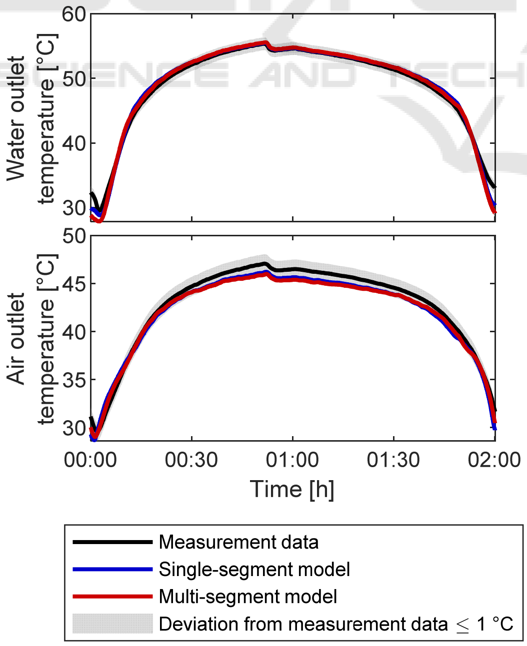

Figure 7: Comparison of both models with measurement data from the quasi-static test with the heating heat exchanger.

In addition, the root mean squared errors (RMSEs) between the measured and the simulated water and air outlet temperatures are listed in Table 3.

Table 3: Root mean squared errors of both models for the quasi-static tests.

Heat exchanger	Outlet temperature	Model: Single-segment	Model: Multi-segment
Heating	Water	0.68 °C	0.83 °C
	Air	0.77 °C	0.78 °C
Cooling	Water	0.25 °C	0.34 °C
	Air	0.71 °C	0.53 °C

Figure 8 shows then the inlet conditions for the quasi-static test with the cooling heat exchanger and the corresponding comparison of the two models with the measured outlet temperatures is shown in Figure 9. The RMSEs of both models for this test are given in Table 3, too.

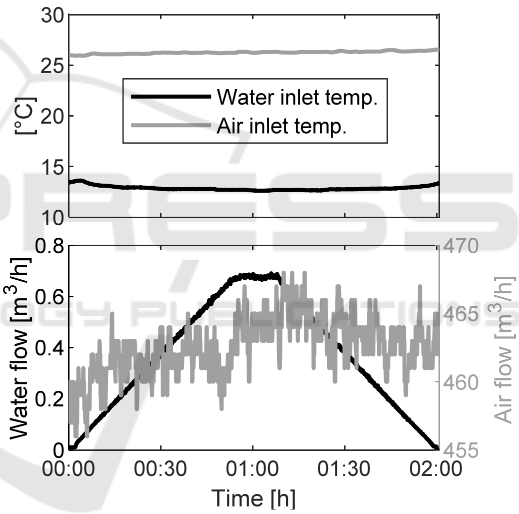

Figure 8: Inlet conditions for the quasi-static test with the cooling heat exchanger.

These results show that both models can accurately predict the water outlet temperature for quasi-static inlet conditions across a wide range of operating points. During most of the time, the deviation from the measured water outlet temperature is smaller than 1 °C for both models. Only at low water volume flow rates, where the water flow is modelled to be laminar, the deviation increases to values above 1 °C. In addition, it is assumed that at low water volume flow rates the heat losses to the surroundings are not negligible.

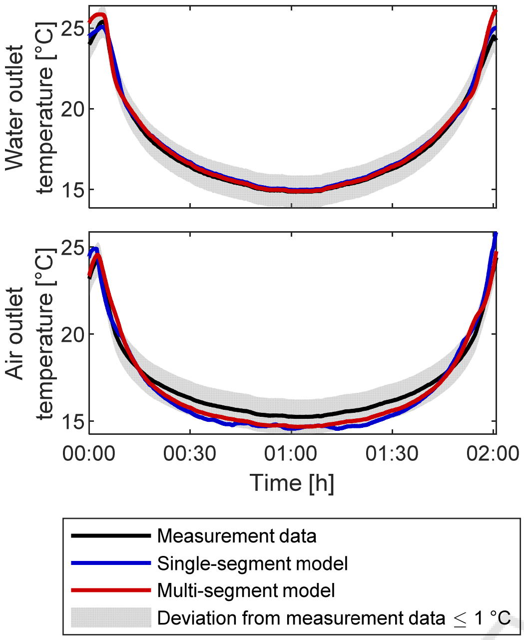

Figure 9: Comparison of both models with measurement data from the quasi-static test with the cooling heat exchanger.

For the air outlet temperature, both models seem to be a bit less accurate. However, it must be remembered that the accuracy of the air outlet temperature measurement must be questioned as explained above.

4.2 Dynamic Tests

To validate the dynamic behaviour of the heat exchanger models, water volume flow rate step changes are applied to the heat exchangers. In the following, the results of these dynamic tests are presented.

The inlet conditions during a segment of the dynamic test with the heating heat exchanger and the corresponding comparison of the two models with the measured outlet temperatures are shown in Figure 10 and Figure 11, respectively. The RMSEs of the simulated water and air outlet temperatures for the complete dynamic test are given in Table 4. Please note that the dynamics of the temperature sensors are considered in the simulations.

These results suggest that for this heating heat exchanger even the simple single-segment model can capture the relevant dynamics of both the water and the air outlet temperatures.

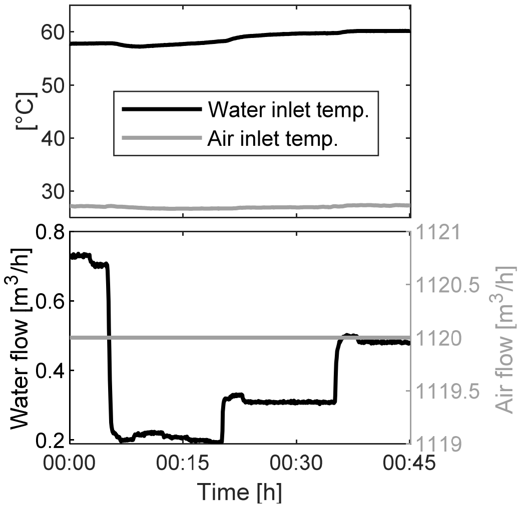

Figure 10: Inlet conditions during a segment of the dynamic test with the heating heat exchanger.

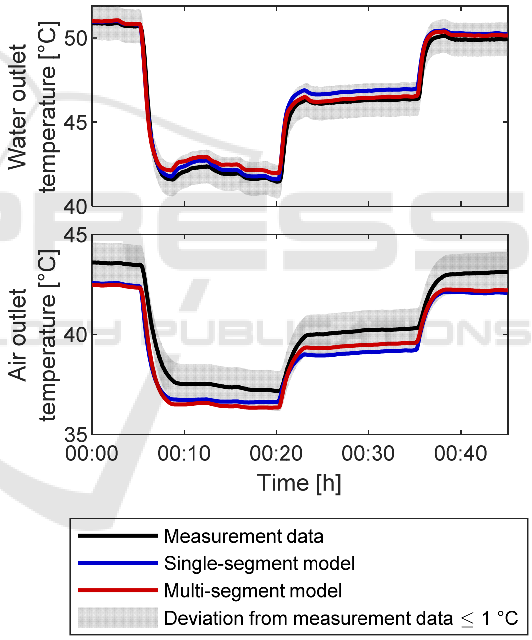

Figure 11: Comparison of both models with measurement data from the dynamic test with the heating heat exchanger.

Table 4: Root mean squared errors of both models for the dynamic tests.

Heat exchanger	Outlet temperature	Model	
		Single-segment	Multi-segment
Heating	Water	1.12 °C	1.18 °C
	Air	0.91 °C	0.85 °C
Cooling	Water	0.34 °C	0.35 °C
	Air	0.46 °C	0.47 °C

Figure 12 shows then the inlet conditions during a segment of the dynamic test with the cooling heat exchanger. The corresponding comparison of the two models with the measured outlet temperatures is plotted in Figure 13. The RMSEs of both models for the complete dynamic test are also provided in Table 4. These numbers let assume that both models are equally able to capture the relevant dynamics of the outlet temperatures. However, as can be seen in Figure 12, the water inlet temperature is subject to oscillating disturbances during the test because the water inlet temperature controller failed to keep a constant temperature. These disturbances reveal that the simple single-segment heat exchanger model fails in this case to always reproduce the dynamics of the water outlet temperature properly. The comparison of the simulated water outlet temperature of the single-segment model with the measurement data in Figure 13 indicates a time shift of the simulated response with respect to the measured one for certain inlet conditions. The multi-segment model, on the other hand, can capture these dynamics.

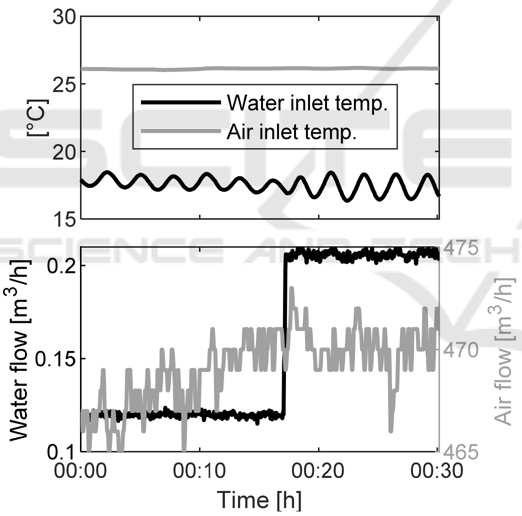

Figure 12: Inlet conditions during a segment of the dynamic test with the cooling heat exchanger.

5 CONCLUSION

The subject of this paper is to investigate two different approaches of modeling an HVAC water-to-air heat exchanger and to compare their dynamic temperature responses for variations in inlet conditions. For this assessment, measurement data recorded with one heating and one cooling heat exchanger are used.

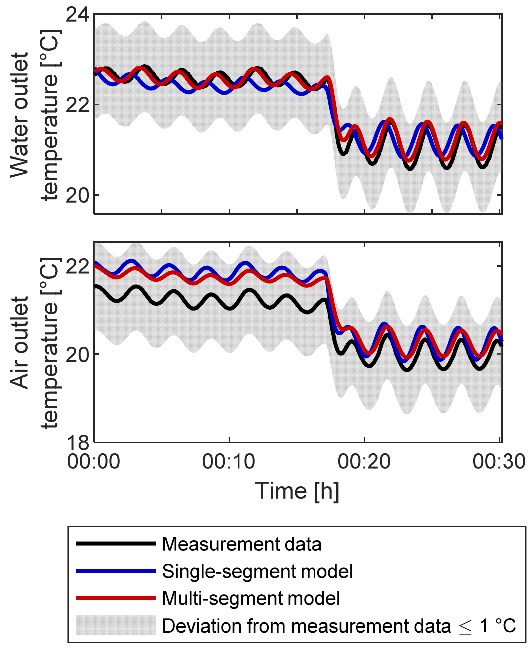

Figure 13: Comparison of both models with measurement data from the dynamic test with the cooling heat exchanger.

The single-segment model delivered by the Simscape Fluids component library and the detailed multi-segment model developed by the authors are showing overall good accuracy for quasi-static inlet conditions across a wide range of operating points.

For the heating heat exchanger, both models show comparable dynamic behavior. Hence, in this case, the effort for developing the very detailed multi-segment model and its additional computational burden during a simulation are not justified.

For the cooling heat exchanger, however, a noticeable time shift between the water outlet temperature response of the single-segment model and the measurement data is observed for certain variations in inlet conditions, while the multi-segment model remains accurate. Reasons for this could be that the multi-segment model additionally considers the thermal heat capacity of the tube and fin material and that it represents the transport delay in the tubes more accurately due to the discretization approach.

It must be emphasized that the analysis to substantiate this assumption is not yet complete. To confirm this initial hypothesis, tests with further variations of the inlet conditions must be performed.

Finally, so far, the models have been evaluated only with open loop tests. Additionally, closed loop studies need to be performed to examine the impact of the time shift of the single-segment model on control loop analysis.

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