AALBORG UNIVERSITY

DENMARK

Modeling of Nonlinear Marine Cooling Systems with Closed Circuit Flow

Hansen, Michael; Stoustrup, Jakob; Bendtsen, Jan Dimon

Modeling of Nonlinear Marine Cooling Systems with Closed Circuit Flow

Michael Hansen [*] **Jakob Stoustrup** [**] **Jan Dimon Bendtsen** [**]

[*] *A.P. Møller - Mærsk A/S, Esplanaden 50, DK-1098, Copenhagen, Denmark, (e-mail: michael.hansen1@maersk.com), also affiliated with Aalborg University, Fredrik Bajers Vej 7, DK-9320, Aalborg, Denmark*
[**] *Aalborg University, Fredrik Bajers Vej 7, DK-9320, Aalborg, Denmark (e-mail: {jakob,dimon}@es.aau.dk)*

Abstract: We consider the problem of constructing a mathematical model for a specific type of marine cooling system. The system in question is used for cooling the main engine and auxiliary components, such as diesel generators, turbo chargers and main engine air coolers for certain classes of container ships. The purpose of the model is to describe the important dynamics of the system, such as nonlinearities, transport delays and closed circuit flow dynamics to enable the model to be used for control design and simulation. The control challenge is related to the highly non-standard type of step response, which requires more detailed modeling.

Keywords: Marine systems, modeling, transport delay, nonlinear models

1. INTRODUCTION

Maritime transportation is today considered to be the most energy efficient means of transportation when considering fuel consumption per ton goods (Rodrigue et al. (2006)). However, only in recent years have energy optimization of container ships and especially their subsystems gained the appropriate attention when building and modifying such ships. One of the subsystems that shows significant potential when it comes to energy optimization is the cooling system for the main engine and auxiliary components. Today, the cooling systems used onboard several classes of container ships are typically controlled manually with the assistance of a few simple controllers. This means that the pumps in this type of cooling system are used excessively, and that operating conditions for the consumers in the cooling system are not necessarily optimal. To deal with these shortcomings it is desired to introduce a control scheme that is not only optimal in terms of energy consumption, but also able to ensure optimal operating conditions for the consumers in the cooling system.

This paper concerns the construction of a model for the marine cooling system that can be used for controller design and simulation. Later work will deal with design and verification of controllers based on the models derived here.

Similar modeling have been carried out in (De Persis and Kallesøe (2009a)) and (De Persis and Kallesøe (2008)), where design of control laws for a system with hydraulic resemblance to the system in this work, is considered. However, since the objective in (De Persis and Kallesøe (2009a)) and (De Persis and Kallesøe (2008)) is to control the pressure at some end users, the constructed model and corresponding controller design only covers the hydraulic part of the system. In our context it is desired to include the thermodynamics of the system, as the temperature of the consumers in the cooling system is of great importance when it comes to set point optimization. Therefore the method from (De Persis and Kallesøe (2009a)) is adopted to cover the hydraulics, while the main contribution of this paper is the derivation of a thermodynamic model for the cooling system.

The size and structure of the cooling system results in significant transport delays that are dependent on the flow rates in the system. The introduction of energy optimizing control is likely to decrease the overall flow rates in the system, which means that the transport delays will increase. Furthermore, because the coolant is recirculated, the cooling system is subject to closed circuit flow dynamics, i.e., the response to any action performed on the system will repeat itself in some form. The result is that a classic control design may prove insufficient when dealing with both delays and closed circuit flow dynamics; this is best illustrated through a simple example. Let us assume that the system can be modeled as a linear first order system with a time delay as illustrated in Figure 1. The closed circuit flow dynamics is modeled by the red positive feedback path in Figure 1. The open loop unit
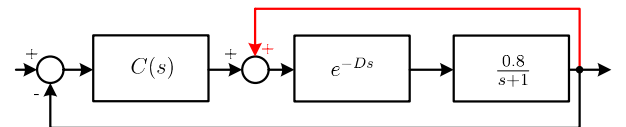

Fig. 1. Example system with time delay D and closed circuit dynamics modeled with positive feedback.

step response for the system in Figure 1 with $D = 20$ and $C(s) = 1$ is illustrated in the top plot of Figure 2.

The system is stable both with and without the red positive feedback path in Figure 1, though the responses
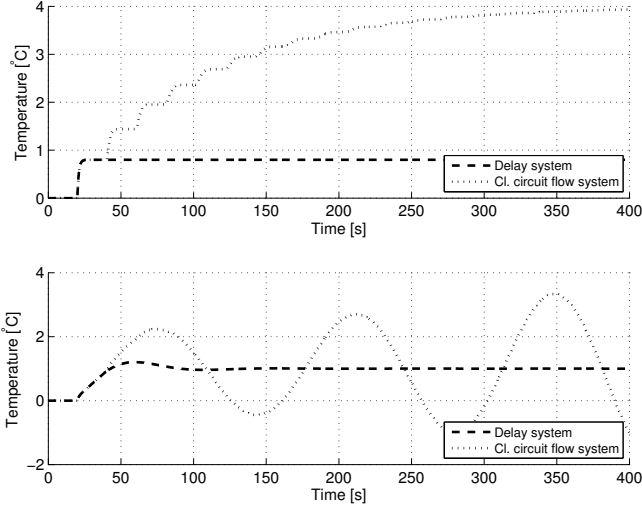
Fig. 2. Top plot: Simulated open loop step response for first order delay system with and without closed circuit dynamics. Bottom plot: Simulated closed loop step response for PI compensated first order delay system with and without closed circuit dynamics.

are very different. We now apply a regular PI controller on the form (Silva et al. (2001)):

$$C(s) = k_p + \frac{k_i}{s} \ , \tag{1}$$

where we set $k_p = 0.2$ and $k_i = \frac{1}{4}$. The resulting closed loop unit step response is illustrated in the bottom plot of Figure 2. The PI compensated system without the red feedback path from Figure 1 is clearly stable, while the system including the red feedback path is unstable. This example illustrates how it is possible to design a PI controller for a delay system such that the compensated system is stable, and how the same compensated system becomes unstable if it is subject to closed circuit flow dynamics. In future control design, which is not presented in this paper, theory from infinite dimensional systems will be applied for compensation of delays and of the closed circuit flow dynamics. The model derived in this paper is therefore structured to facilitate this control design approach.

In Section 2 the cooling system is outlined in order to provide an overview of its structure and function. Section 3 describes the derivation of the model, which is divided into a hydraulic and thermodynamic part. Section 4 provides verification of part of the derived model and a simulation example to illustrate the dynamics of the model. Finally, conclusions are given in Section 5.

2. SYSTEM DESCRIPTION

The cooling system consists of three circuits; a sea water (SW) circuit, a low temperature (LT) circuit and a high temperature (HT) circuit. This is illustrated in Figure 3 where q_{in} and q_{SW} are volumetric flows in the LT and SW circuits, while q_{HT} is the volumetric flow to the HT circuit.

As the name implies, the SW circuit pumps sea water through the cold side of the central coolers for lowering the temperature of the coolant in the LT and HT circuits. The
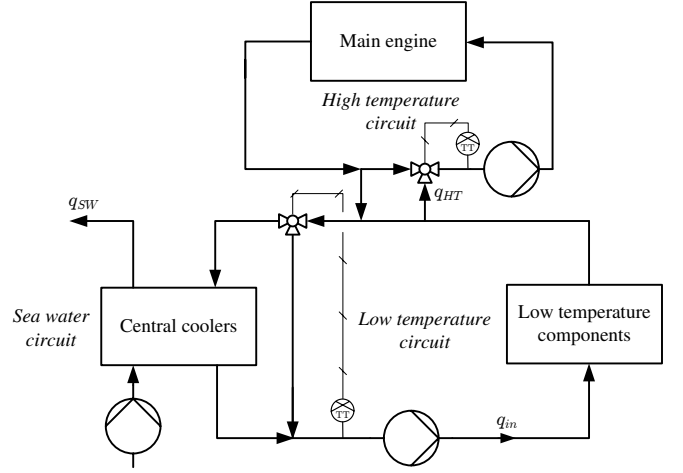

Fig. 3. Simplified system layout.

LT circuit contains all the auxiliary components that need cooling, such as diesel generators and turbochargers, all coupled in parallel. The HT circuit is only responsible for cooling the main engine of the ship, and since the cooling demand for the main engine is very strict there is little room for energy optimization in this part of the system. The main concern is therefore the LT circuit and the SW circuit.

Models are constructed based on the assumptions that all flows are turbulent, and there are no laminar flow effects. There is also no heat loss to surroundings, i.e., heat exchange only takes place in the consumers or in the central coolers. There is no phase change of the coolant, and density as well as specific heat of the coolant is assumed to be constant in the temperature range of interest. Finally, the coolant in the system is assumed to be incompressible.

Each consumer in the LT system consists of a control valve in series with a heat exchanger and two pipe sections, as illustrated in Figure 4. From a control point of view it is
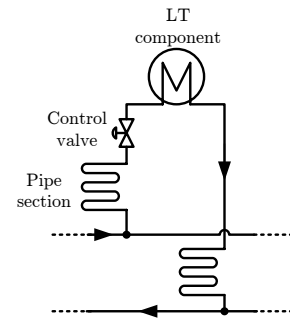
Fig. 4. Structure of the components in the LT circuit.

desired to adjust the temperature of the components in the LT circuit to an operational and energy-wise optimal set point. This should be achieved by the most energy efficient control inputs which are generated by the pumps and control valves. The model should therefore express how the control input and disturbances affect the temperature of the components as illustrated in Figure 5. In Figure 5, K_{cv}, h_p, and q denotes the position of the control valves, delivered pump head, and volumetric flows, while

Fig. 5. Block diagram of the model structure.
Q_{cc}, Q_{con} and Q_{HT} are heat transfer from central coolers, consumers, and the HT circuit, respectively.
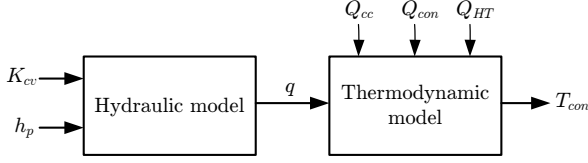
3. MODELING

The modeling is divided into two parts; a hydraulic part and a thermodynamic part. The reason for this division is that the thermodynamics in the system is assumed not to influence the hydraulics and it is therefore possible to consider the overall system as a thermodynamic part in cascade with a hydraulic part. In addition, there is a separation of time scales since the time constants of the heat dynamics are much slower than the hydraulic dynamics. The purpose of the hydraulic model is to describe the flow through the components in the three circuits as a function of the pump speeds and the position of the control valves. These flows are inputs to the thermodynamic model, which describes the temperature of the components in the system as a function of the sea water temperature, the heat generated by the components and of course the flows in the system.

3.1 Hydraulic model

The hydraulic model is separated into two parts; the SW circuit hydraulics and the LT circuit hydraulics, respectively. Due to the small number of components and the simple structure of the SW circuit, equations governing the flow for this system are derived directly using basic hydraulic laws. In the LT circuit however, the individual consumers are placed in parallel and are modeled as illustrated in Figure 4. This structure yields strong similarities with the system presented and modeled in (De Persis and Kallesøe (2009a)) and the model for the LT circuit hydraulics is therefore constructed by following the same method and notation. This means that valves are described by the relation:

$$h_i - h_j = K_v |q_v| q_v , \quad (2)$$

where $(h_i - h_j)$ is the pressure drop across the valve, K_v is a variable describing the valve position, i.e. the change in hydraulic resistance of the valve, and q_v is the volumetric flow through the valve. In a similar manner, pipe sections are modeled as:

$$J \frac{dq_p}{dt} = (h_i - h_j) - K_p |q_p| q_p , \quad (3)$$

where J and K_p are constant parameters for the pipe section, $(h_i - h_j)$ is the pressure drop along the pipe and q_p is the flow through the pipe. Finally, pumps are simply modeled as a pressure difference:

$$h_i - h_j = -\Delta h_p , \quad (4)$$

where $(h_i - h_j)$ is the pressure across the pump and Δh_p is the delivered pump head.

The SW circuit consists of two pumps in parallel, two pipe sections and a heat exchanger which is modeled as a valve

from a hydraulic point of view (De Persis and Kallesøe (2009b)). Since the circuit is not closed there should also be included a pressure drop due to the difference in height between the sea water circuit inlet and outlet. Since the sum of pressure drops in a closed loop must equal zero, the models given in (2), (3) and (4) can be combined to yield the following result:

$$\Delta h_{p,sw} = (J_{p1} + J_{p2}) \dot{q}_{sw}$$
$$+ (K_{p1} + K_{cc} + K_{p2}) |q_{sw}| q_{sw} + \Delta h_{io} , \quad (5)$$

where $\Delta h_{p,sw}$ is the delivered pump head, J_{p1}, J_{p2}, K_{p1} and K_{p2} are pipe section parameters, K_{cc} describes the hydraulic resistance in the central cooler and Δh_{io} is the pressure difference due to difference in height between SW inlet and outlet. Rearranging and combining constants such that $J_{sw} = J_{p1} + J_{p2}$ and $K_{sw} = K_{p1} + K_{cc} + K_{p2}$ yields:

$$J_{sw} \dot{q}_{sw} = - K_{sw} |q_{sw}| q_{sw} - \Delta h_{io} + \Delta h_{p,sw} . \quad (6)$$

The hydraulics of the LT circuit is assumed to have the structure shown in Figure 6. Compared to the simplified diagram from Figure 3 it is seen that the shunt at the central cooler is not included in the hydraulic structure. This is justified by the observation that in the final control scheme it is desired to control the inlet water temperature in the LT circuit using the pumps in the SW circuit, such that the shunt valve is closed at all times, and thereby does not influence the hydraulics of the LT system. Using the notation from (De Persis and Kallesøe (2009a)) the hydraulic model for the LT system can be constructed as:

$$BJB^T \dot{q} = - B\lambda(K_p, B^T q) - B\mu(K_v, B^T q)$$
$$- B\mu_{cv}(K_{cv}, B^T q) + B\Delta h_p , \quad (7)$$

where B is the fundamental loop matrix, see (De Persis and Kallesøe (2009a)) for details. Furthermore, we have that:

$$\Delta h = [\Delta h_{p1}, \ldots, \Delta h_{pk}]^T ,$$
$$J = diag\{J_1, \ldots, J_k\} ,$$
$$\lambda(K_p, q) = [\lambda_1(K_{p1}, q_1), \ldots, \lambda_k(K_{pk}, q_k)]^T , \quad (8)$$
$$\mu(K_v, q) = [\mu_1(K_{v1}, q_1), \ldots, \mu_k(K_{vk}, q_k)]^T ,$$
$$\mu_c(K_{cv}, q) = [\mu_{c1}(K_{cv1}, q_1), \ldots, \mu_{ck}(K_{cvk}, q_k)]^T ,$$

where k is the number of components in the hydraulic network. The index $\{cv\}$ in (7) indicates contributions from controllable valves, as these generates inputs to the hydraulic system and should be distinguished from other valve types.

3.2 Thermodynamic model

The thermodynamic model is derived using basic thermodynamic laws, and consists of three parts; one part describing the temperature change in the components and two parts describing the transport phenomenon between the central coolers and the components in the LT circuit. All consumers are modeled as heat exchangers, so by use of the first law of thermodynamics and Reynolds Transport Theorem (Young et al. (2007)) it is possible to derive the following equation:
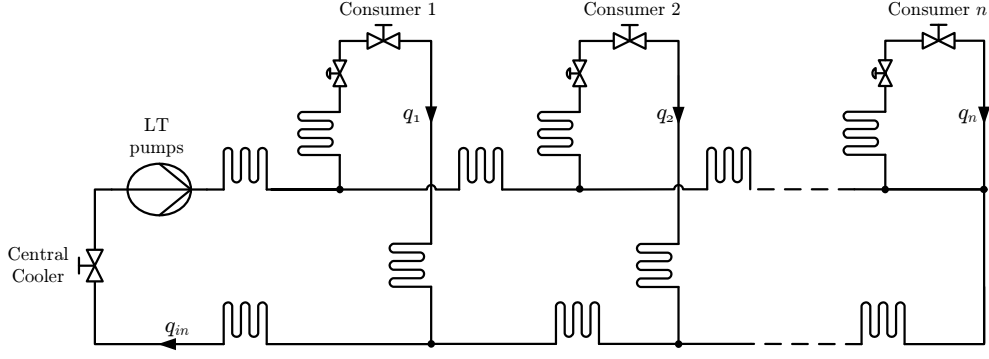
Fig. 6. Simplified hydraulic structure of the LT circuit where valves model the pressure losses in the heat exchangers.

$$\sum \dot{Q} - \sum \dot{W} = \frac{d}{dt} \int_{\Omega} e\rho \, dV$$
$$+ \sum e_{out}\rho_{out}A_{out}v_{out}$$
$$- \sum e_{in}\rho_{in}A_{in}v_{in} \; , \qquad (9)$$

where $\dot{Q}$ is heat transfer rate in or out of the system, $\dot{W}$ is the rate of work transfer in or out of the system, e is the specific energy of the system and ρ is the density of the fluid elements in the system. Also, Ω is the control volume, v is the average velocity of the flow at in- or outlet, and A is the control volume cross section area at the in- or outlet.

In this case, the rate of the work transfer term can be described as the pressure forces acting on the inlet and outlet of the control volume, which can be written as

$$\sum \dot{W} = \dot{W}_{pf} = p_{in}A_{in}v_{in} - p_{out}A_{out}v_{out} \; . \qquad (10)$$

Since there is only a single flow from inlet to outlet it is possible to write:

$$\dot{m} = \rho_{in}A_{in}v_{in} = \rho_{out}A_{out}v_{out} \; . \qquad (11)$$

It is assumed that the change in potential and kinetic energy in the control volume can be neglected such that $e = u$ with u being the internal energy per mass unit. Also, $\sum \dot{Q} = \dot{Q}_{con}(t)$ which leads to the following result:

$$\frac{d}{dt} \int_{\Omega} e(t)\rho \, dV = \dot{m} \left(H_{in}(t) - H_{out}(t) \right) + \dot{Q}_{con}(t) \qquad (12)$$

where we also have exploited the fact that enthalpy is defined as $H = u + \frac{p}{\rho}$. Since there is no change of phase of the coolant in and out of the control volume, the enthalpy terms can be approximated by (Massoud (2005)):

$$\Delta H \approx c_p \Delta T \; , \qquad (13)$$

where c_p is the specific heat for the coolant and T is the temperature of the coolant. It is desired to have the model express the change in temperature rather than the change in stored energy. Preferably, the equation should express the change of energy in the control volume as a function of the outlet temperature. In order to keep the expression simple it is chosen to use the crude approximation:

$$\frac{d}{dt} \int_{\Omega} e(t)\rho \, dV \approx \rho c_p V_{CV} \frac{d \, T_{out}(t)}{dt} \; , \qquad (14)$$

where V_{CV} is the volumetric size of the control volume. This yields the result:

$$\rho c_p V_{CV} \frac{d \, T_{out}(t)}{dt} = \dot{m}(t)c_p(T_{in}(t) - T_{out}(t)) + \dot{Q}_{con}(t) \; . \qquad (15)$$

This means that in general for consumer $i = 1, \ldots, n$ we can write

$$\frac{d \, T_i(t)}{dt} = \frac{1}{\rho_w c_p V_i} \left(\dot{m}_i(t)c_p(T_{in,i}(t) - T_i(t)) + \dot{Q}_i(t) \right) \; , \qquad (16)$$

where $\dot{m}_i$ is the mass flow rate through the consumer, V_i is the internal volume of the consumer, Q_i is the energy transfer from the consumer, while $T_{in,i}$ and T_i are the temperatures of the coolant at the inlet and outlet, respectively, of the consumer.

Because of the distance from the central cooler to the consumers there is a transport delay in the temperature of the coolant at the outlet of the central cooler, to the inlet of the individual consumer. This delay is different for each consumer due to their spacing relative to the central cooler. To derive an expression for the inlet temperature for each consumer as a function of the central cooler outlet temperature and the corresponding transport delay, the structure in Figure 7 is considered.
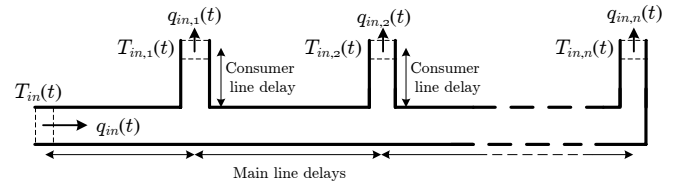

Fig. 7. The distribution of coolant from the central coolers to the consumers in the LT circuit.

The relation between the flow rate and the corresponding transport delay in the temperature of the water at the inlet of consumer i can be written as:

$$T_{in,i}(t) = T_{in,i-1}(t - D_{in,i}) \; , \qquad (17)$$

with the flow dependent delay $D_{in,i}$ given by:

$$D_{in,i} = a_{m,i} \underbrace{\left(\sum_{j=i}^{n} q_j \right)^{-1}}_{\text{Main line delay}} + \underbrace{a_{c,i}q_i^{-1}}_{\text{Consumer line delay}} \; , \qquad (18)$$

where $a_{m,i}$ and $a_{c,i}$ are system specific constants, and q_j is the flow to the j'th consumer.

For calculating the temperature of the coolant in the return path, i.e. the temperature of the coolant to the HT circuit, the structure in Figure 8 is considered.
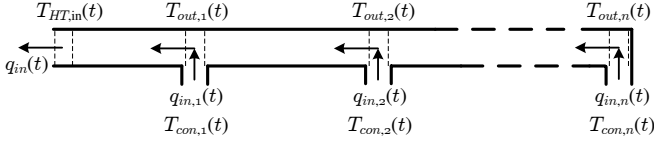
Fig. 8. The combination of coolant flows from the consumers in the LT circuit to the HT circuit inlet.

According to Figure 8 the temperature at the inlet to the HT circuit can be written as:

$$T_{HT,in}(t) = T_{out,1}(t - D_{out,1}), \qquad (19)$$

with the delay $D_{out,i}$ given by:

$$D_{out,i} = b_{m,i} \left(\sum_{j=i}^{n} q_j \right)^{-1} , . \qquad (20)$$

where $b_{m,i}$ is a system specific constant, and q_j is the flow from the j'th consumer. The temperature $T_{out,1}$ can be calculated in a recursive manner by solving the following equation iteratively for $j = n, n-1, \ldots, 1$:

$$T_{out,i}(t) = T_{con,i}(t - D_{con,i}) \frac{q_{in,i}(t)}{\sum_{j=i}^{n} q_{in,j}(t)} \qquad (21)$$
$$+ T_{out,n+1}(t - D_{out,n+1}) \frac{\sum_{j=i+1}^{n} q_{in,j}(t))}{\sum_{j=i}^{n} q_{in,j}(t)} ,$$

where the delay $D_{con,i}$ is given by:

$$D_{con,i} = b_{c,i} q_i^{-1} . \qquad (22)$$

Equation (21) builds on the assumption that the temperature of the coolant at consumer outlet no. i can be described by the temperature of the coolant from consumer i plus the temperature of the coolant at consumer outlet $i+1$ delayed by the time it takes the coolant to travel from consumer $i+1$ to i.

Since it is not desired to modify the control of the HT system, the HT circuit's impact on the temperature in the LT circuit is added to the model as a measured disturbance. It is assumed that the flow to the HT circuit equals the flow from the HT circuit. Denoting this flow by q_{HT} and the temperature of the water from the HT circuit by $T_{HT,out}$ the inlet temperature to the central coolers can be written as:

$$T_{CC,in} = T_{HT,in}(t - D_{HT,out} - D_{HT,in}) \cdot \frac{(q_{in} - q_{HT})}{q_{in}}$$
$$+ T_{HT,out}(t - D_{HT,out}) \frac{q_{HT}}{q_{in}} , \qquad (23)$$

where

$$D_{HT,out} = b_{HT,out} q_{in}^{-1} \qquad (24)$$
$$D_{HT,in} = b_{HT,in}(q_{in} - q_{HT})^{-1} . \qquad (25)$$

To close the circuit all we need is to describe the relation between the temperature into the central cooler, $T_{CC,in}$ and the temperature out of the central cooler, which has previously been defined as the inlet temperature T_{in}. This relation is modeled using Equation (16) where the heat transfer to the SW circuit is considered to be steady state, which yields the result:

$$\frac{d\,T_{in}(t)}{dt} = \frac{1}{\rho_w c_p V_{CC}} \left[\dot{m}_{in}(t) c_p (T_{CC,in}(t) - T_{in}(t)) \right. \qquad (26)$$
$$\left. + \dot{m}_{SW}(t) c_{p,sw} (T_{SW,in}(t) - T_{SW,out}(t)) \right] .$$

where $c_{p,sw}$ is the specific heat of the sea water.

To sum up the model, the dynamics of the hydraulic part is given by (7), while the dynamics of the thermodynamic part is governed by equations (16) and (26). The inlet temperatures for the LT system, the HT system, and the central coolers are described by relations (17), (19), (21) and (23) with delays given by (18), (20), (22), (24) and (25). Descriptively we can state the combined hydraulic and thermodynamic model as:

$$\dot{x} = f(x, y, u, w)$$
$$y = g(x, y, D) \qquad (27)$$
$$D = h(x)$$

where:

$$x = [q_{SW}, q_{in}, q_1, \ldots, q_n, T_{in}, T_1, \ldots, T_n] \in \mathbb{R}^{2n+3}$$
$$y = [T_{CC,in}, T_{HT,in}, T_{in,1}, \ldots, T_{in,n},$$
$$\quad T_{out,1}, \ldots, T_{out,n}] \in \mathbb{R}^{2n+2}$$
$$u = [h_{p,SW}, h_{p,LT}, K_{cv,1}, \ldots, K_{cv,n}] \in \mathbb{R}^{n+2}$$
$$w = [T_{SW,in}, T_{SW,out}, T_{HT,out}, q_{HT}, \dot{Q}_1, \ldots, \dot{Q}_n] \in \mathbb{R}^{n+4}$$
$$D = [D_{HT,in}, D_{HT,out}, D_{in,1}, \ldots, D_{in,n},$$
$$\quad D_{out,1}, \ldots, D_{out,n}, D_{con,1}, \ldots, D_{con,n}] \in \mathbb{R}^{3n+2} .$$

4. MODEL VERIFICATION

Measurement data from the M-class vessel "Margrethe Maersk" have been obtained and is used for model verification. Since the implemented control on "Margrethe Maersk" only requires a small number of measurements, the available data for model verification are very sparse. The sampling time of the data is 5 minutes, and the data only covers a few key temperatures as well as the main engine (ME) load and the speed of the pumps. The consequence is that it is not possible to verify the hydraulic model as only the flow through the pumps can be estimated from the measurements and not the flow through each individual branch in the LT system. Also, the low sampling rate means that it is not possible to observe any transport phenomena in the measured temperature due to aliasing. However, it is possible to verify the model for the largest consumer in the LT circuit, namely the main engine scavenge air coolers, as both the inlet and outlet temperatures for this consumer are measured, and the flow through it can be reasonably estimated. Furthermore, the heat generated by the ME scavenge air coolers is a function of the ME load and can be approximated as well. The measurement data covers a period of 68 days, and because model parameters for the ME scavenge air coolers are unknown, part of the measurement data is used for parameter estimation, while another part is used for model verification. The simulated and measured temperature responses for one of the three ME scavenge air coolers aboard "Margrethe Maersk" are illustrated in Figure 9(a) for a period of 6 days. The data used for parameter estimation are from a time period 34 days earlier, also covering a period of 6 days. Figure 9(a) shows how the dynamics of the simulated response matches well with measured response. The top plot of Figure 9(b) illustrates the LT inlet temperature while the bottom plot shows the ME load during the test period.

To illustrate the models ability to represent the closed circuit flow dynamics, we construct an example where the model is applied to a system with two identical consumers
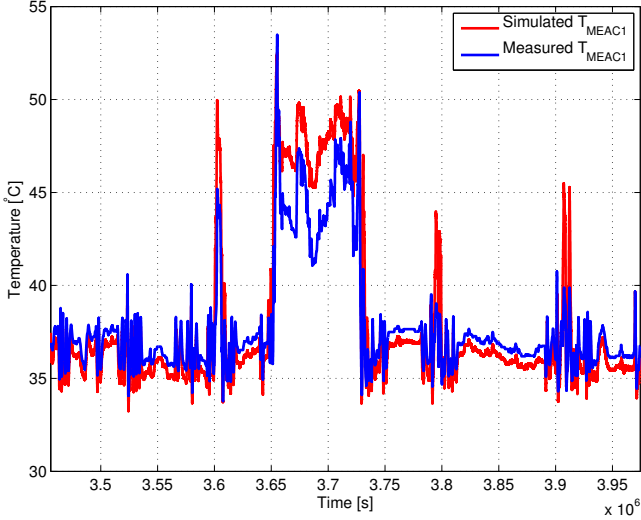
(a) Simulated and measured outlet temperature of ME scavenge air cooler 1
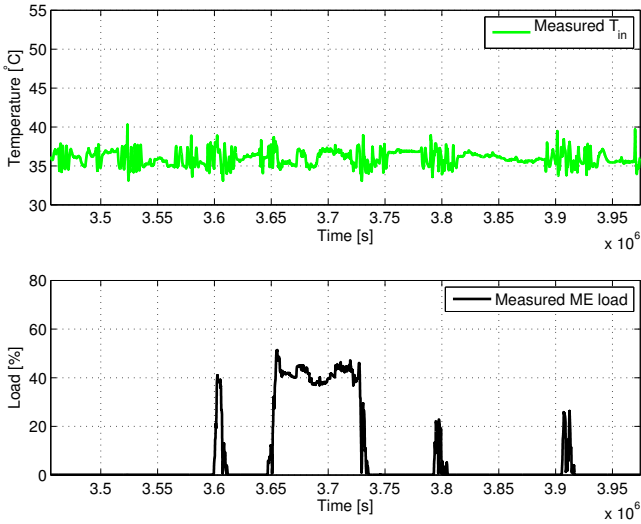
(b) Top plot: Inlet temperature for ME scavenge air cooler 1. Bottom plot: ME load percentage.

Fig. 9. Simulation results for ME scavenge air cooler 1.

that are subject to different flows, and have different spacing relative to the central coolers. Due to the limited space it is not possible to state the model parameters for the example here, but it is possible to choose a set of parameters such that the model generates the dynamics shown in this example. The model is subjected to a positive step in the ME load, which increases the heat generated by the consumers. This step occurs after 5000 s and the top plot of Figure 10 shows how the closed circuit flow dynamics influences the outlet temperature responses for the two consumers. The effect of the delays is best observed from the shift in the inlet temperatures in the bottom plot of Figure 10.

5. CONCLUSIONS

We have presented a model of a marine cooling system with application to control design and simulation. The small number of available measurements restricted the verification of the model to only include part of the ther-
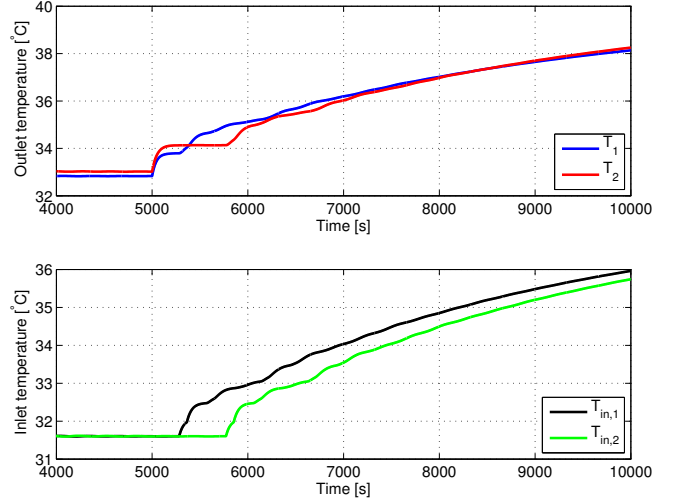

Fig. 10. Top plot: Outlet temperature responses for the two consumers in the example. Bottom plot: inlet temperature for the two consumers.

modynamics. The verification showed that when applying the thermodynamic consumer model to the ME scavenge air coolers in the LT circuit, the model response was very close to the measured response. An example showed that the model includes the dynamics necessary for adequately representing the behavior of the system. Future work includes verifying the remaining model and using the model for control design; first for developing a base line control for performance comparison, and later for designing energy and set point optimizing control laws.

REFERENCES

De Persis, C. and Kallesøe, C.S. (2008). Proportional and proportional-integral controllers for a nonlinear hydraulic network. *Proceedings of the 17th IFAC World Congress.*

De Persis, C. and Kallesøe, C.S. (2009a). Pressure regulation in nonlinear hydraulic networks by positive controls. *European Control Conference.*

De Persis, C. and Kallesøe, C.S. (2009b). Quantized controllers distributed over a network: An industrial case study. *Mediterranean Conference on Control and Automation.*

Massoud, M. (2005). *Engineering Thermofluids: Thermodynamics, Fluid Mechanics, and Heat Transfer.* Springer.

Rodrigue, J.P., Comtois, C., and Slack, B. (2006). *The Geography of Transport Systems.* Routledge.

Silva, G.J., Datta, A., and Bhattacharyya, S.P. (2001). Pi stabilization of first-order systems with time delay. *Automatica*, 37, 2025–2031.

Young, D.F., Munson, B.R., Okiishi, T.H., and Huebsch, W.W. (2007). *A Brief Introduction to Fluid Mechanics.* John Wiley & Sons Ltd.