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THERMODYNAMIC ANALYSIS OF A 5 MW DYNAMOMETER SET TO SIMULATE SHIP PROPULSION AND PROPULSION LOAD OF AN ALL-ELECTRIC SHIP

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Abstract. This study investigates a 5 MW dynamometer set which consists of two 2.5 MW variable speed induction motor drives on the same shaft. This dynamometer provides the capability to explore the dynamic performance of propulsion motor machines up to 2.5 MW. One dynamometer motor can simulate a 2.5 MW ship propulsion load. The second motor, also 2.5 MW, can operate as the ship propulsion responding to the propulsion load. Under the control of a Real Time Digital Simulator (RTDS), the dynamometer simulates the effects of the propeller, hull speed, shaft dynamics, and system inertia under different sea conditions. This paper focuses on the thermal behaviour of a water cooled propulsion motor responding to different simulated load conditions. The first law of thermodynamics and a simple exergetic accounting are presented. The results show that the thermal signature of the motor can be controlled by adjusting the coolant flow rate. Intelligent heat exchangers that dynamically adjust the flow based on the load and heat signature are needed for the next generation all electrical ships.

Keywords: *real time simulation, ship propulsion, motor cooling*

1. Introduction:

The US Navy has contracted to build a high power all electric warship. Due to the large number of sophisticated electronics and electrical weapons aboard, it has become imperative to design an effective thermal management system. This system must dissipate heat efficiently and minimize the thermal signature, which could give out critical operational information. To proficiently design an effective thermal management system, testing the ship motors in diverse sea conditions becomes critical. The Center for Advanced Power System (CAPS) at Florida State University, has a unique test bed facility where it can utilize two 2.5 MW dynamometers coupled together. (See Fig 1.) These two dynamometers are connected to a real time digital simulator, where one dynamometer motor is set to behave as the ship drive motor and the other as the propeller load. As different loads are applied to the propeller motor simulating different sea conditions the heat signature is monitored, recorded, and analyzed, giving an insight into the thermal management system performance.

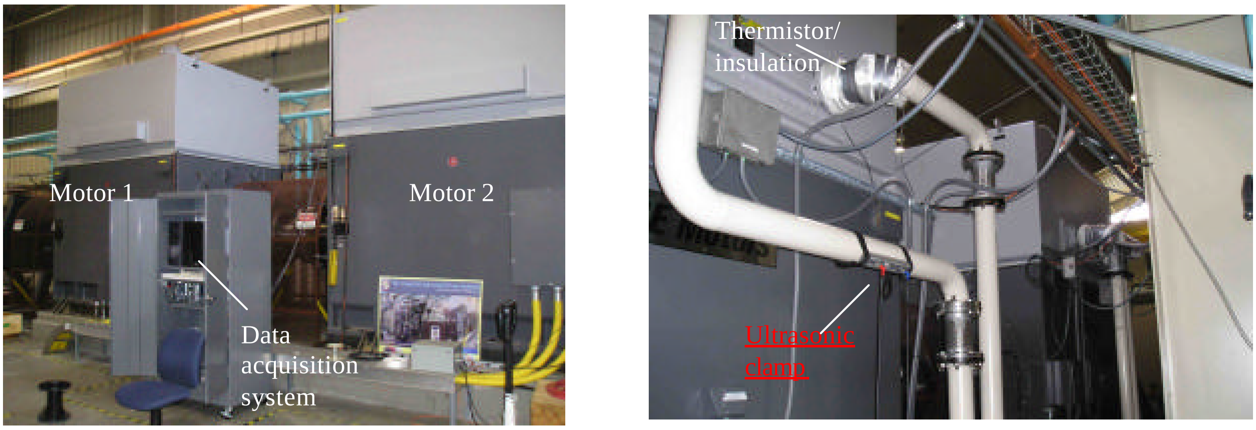

Figure 1 – 2.5 MW motor and 2.5 MW load (left). Instrumentation detail (right)

2. Experimental Setup:

The motors are cooled by means of a water heat exchanger and convection heat transfer through the walls. The water inlet and outlet temperature and the wall temperature of the motor is acquired using a multipurpose data acquisition system based on a DXI National Instruments card. The sampling interval used between readings is 11.8 seconds.

Both the propulsion motor and the load motor are water-cooled. The water flow to both motors is measured with an ultrasonic clamp-on flowmeter (FUJI PortaFlow-X). The water inlet and outlet temperature and the motor wall temperature are measured using high precision thermistors (YSI 44004) and additional wall temperature values of the motor wall temperature were obtained with an infrared thermometer with emissivity adjustment (Fluke 66). The motor rpm and torque were measured and controlled by means of a PLC system. Figure 1 (right) shows a detail of the flow measuring setup. Figure 2 illustrates the instrumentation setup for the evaluation of the motor thermal performance.

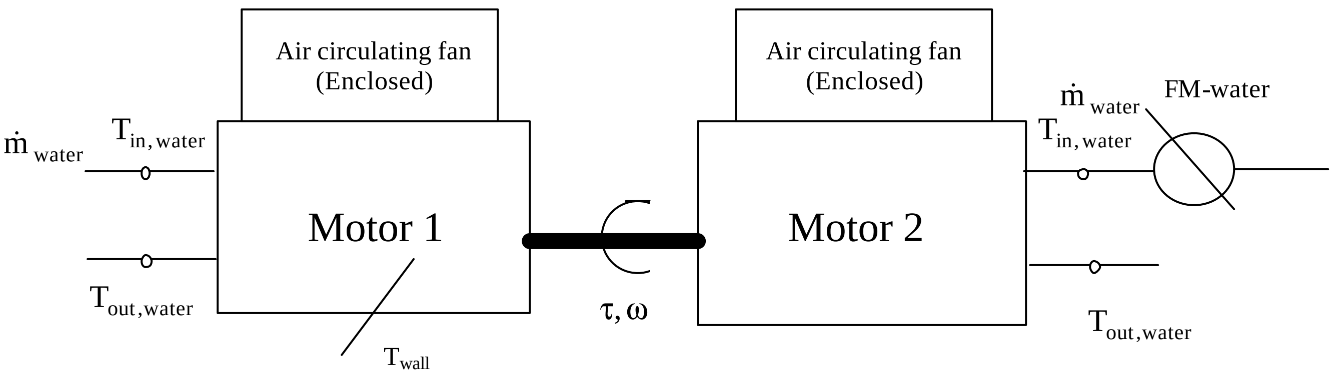

Figure 2 – Instrumentation setup

3. Test Procedure:

The test consisted of three stages:

Stage1: During the initial stage, motor 1 ran at 100% torque and 225 rpm, simulating a full speed sea condition. The heat exchanger of motor 1 was running at 138.2 gpm, and the heat exchanger of motor 2 at 301.5 gpm. After 285 minutes, when a steady state condition was reached, the torque was reduced to 50%, which corresponds to a cruising condition.

Stage 2: After running at 50% torque during the time interval 285 min – 348 min the 100% torque condition was reestablished at minute 348. The mass flow rate to the propulsion motor was kept constant and equal to 138.2gpm.

Stage 3: Running the motor at full torque (100%) and 225 rpm the flow rate to the propulsion motor heat exchanger was increased to 280.7 gpm at minute 478.

4. Results and Discussions:

Figure 3 illustrates the water temperature evolution in the heat exchanger of motor 1 during the three stages of the test.

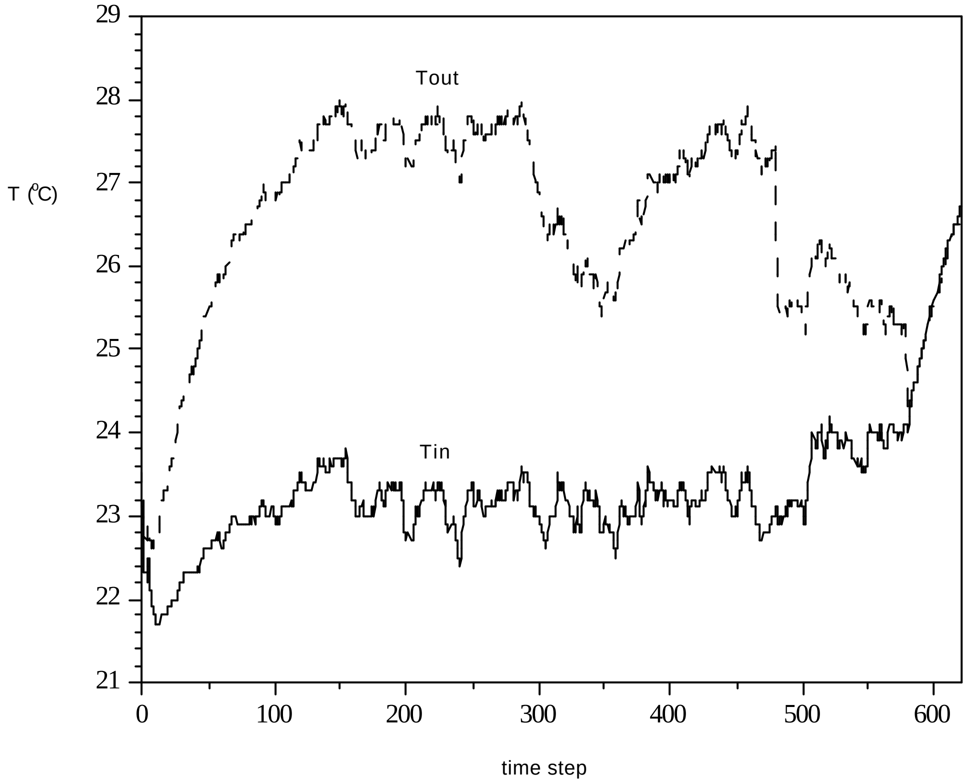

Figure 3 – Water inlet and outlet temperature for the propulsion motor heat exchanger

To facilitate the analysis we define the dimensionless torque as

$$\tilde{\tau} = \frac{\tau}{\tau_{max}}$$

and the dimensionless water mass flow rate,

$$\tilde{m} = \frac{\dot{m}}{\dot{m}_{max}}$$

In this way 100% torque corresponds to $t = 1$ and 50% torque to $t = 0.5$. The maximum flow rate was, $\dot{m}_{max} = 301.5$ gpm.

Figure 4 illustrates the temperature gain in the water stream, ΔT, as it passes through the propulsion motor heat exchanger, as well as the torque and mass flow rate variations during the three test stages.

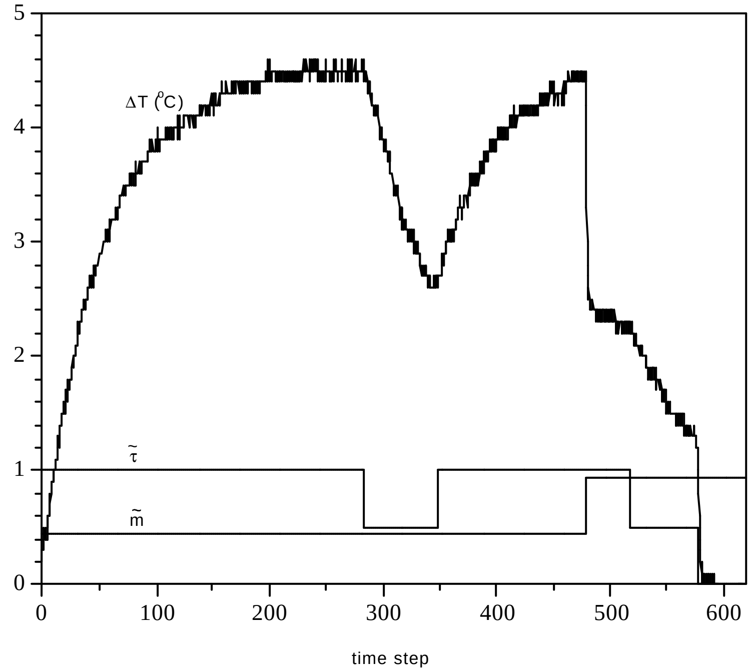

Figure 4 – Water temperature evolution in the driver motor

It can be observed that the temperature gain, ΔT, responds quickly to the load drop at minute 285, and almost immediately to the mass flow rate increase at minute 478.

Figure 5 illustrates the variation of the wall temperature.

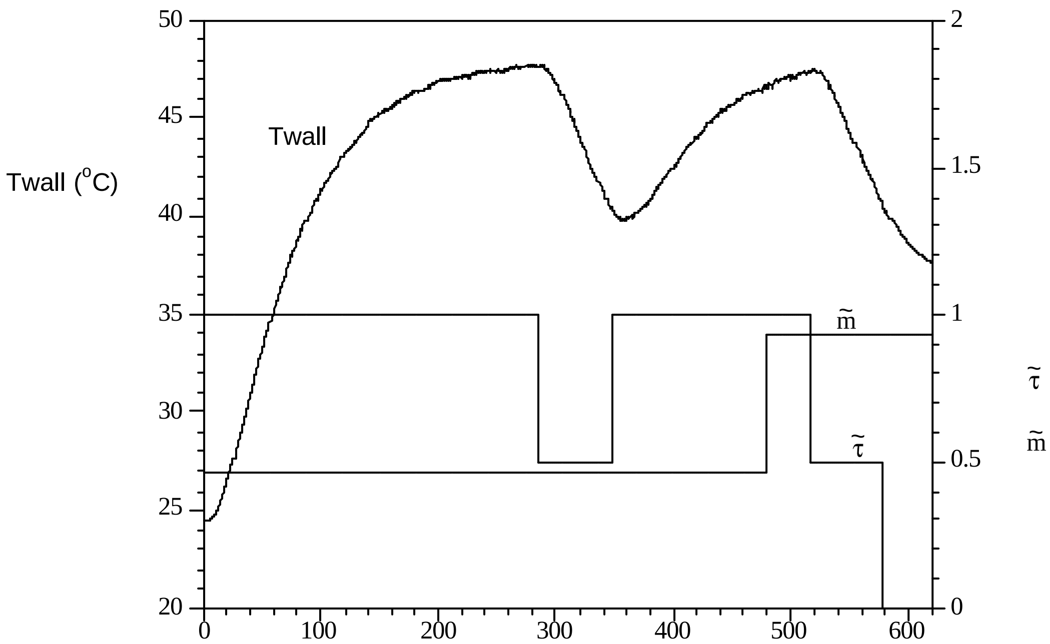

Figure 5 – Wall temperature evolution in the driver motor

5. Energy and Exergy interactions

As mentioned before the motor is cooled by a water heat exchanger and by convection to the surroundings. The convection heat transfer is given by

$$\dot{Q}_{wall} = hA(T_{wall} - T_{amb})$$

A typical value, $h = 10\,W/(m^2K)$ was used for the convection heat transfer coefficient. The area exposed to the environment was estimated from the motor geometry to be, A= 37.45m^2, and the average ambient temperature was, T_{amb} = 25^{o}C.

The energy extracted by the cooling water can be estimated from the enthalpy gain of the water stream:

$$\dot{Q}_{coolant} = \dot{m}\left(h_{water,out} - h_{water,in}\right)$$

The enthalpy of water was evaluated according to IAPWS and NIST (Lemmon et al., 2003). Figure 6 illustrates the magnitude of both dissipations during the motor testing.

The power output of the motor at full and cruise conditions are reported in Table 1.

Table 1. Power output from propulsion motor at different testing conditions

Test condition	Torque (N·m)	rpm	Output power (kW)
Full power (100%)	106052.08	225	2498.8
Cruise (50%)	53026.04	225	1249.4

Figure 7 illustrates a detail of the effect of the water mass flow rate increase on the heat exchanger performance. The stream-to-stream temperature difference reacts almost immediately to the flow rate increment.

The specific flow exergy is given by (Bejan, 1997)

$$e_x = h - h_0 - T_0(s - s_0)$$

where h_0 and s_0 represent the specific enthalpy and entropy of the substance at the restricted dead state (T_0, P_0). In this study we have adopted the customary restricted dead state (T_0, P_0) = (298.15K, 1atm).

Once the specific exergy is known, the flow exergy of each of the coolant streams can be obtained by,

$$\dot{E}_x = \dot{m}e_x$$

The exergy transfer associated with the heat loss through the motor walls can be estimated from

$$\dot{E}_Q = \dot{Q}_{wall}\left(1 - \frac{T_0}{T_{wall}}\right)$$

Table 2 reports the flow exergies and the exergy associated with the heat loss through the wall at the end of stage 1, when a steady state condition was reached, just before the torque was modified.

Table 2. Energy and exe rgy interactions at the end of test stage 1 (minute 285)

Location/ Interaction	Exergy (kW)	Energy transfer (kW)
Wall heat loss	0.596	8.46
Coolant input	2.253	852.26
Coolant output	2.601	1015.78
Net gain by coolant	0.348	163.52
Shaft work	2498.8	2498.8

At the steady state reached during stage 1, the energy transferred to the cooling water represents 6.5% of the shaft work and the wall heat losses represent 0.33%. The flow exergy of the coolant is small because of its proximity to the restricted dead state (T_0, P_0).

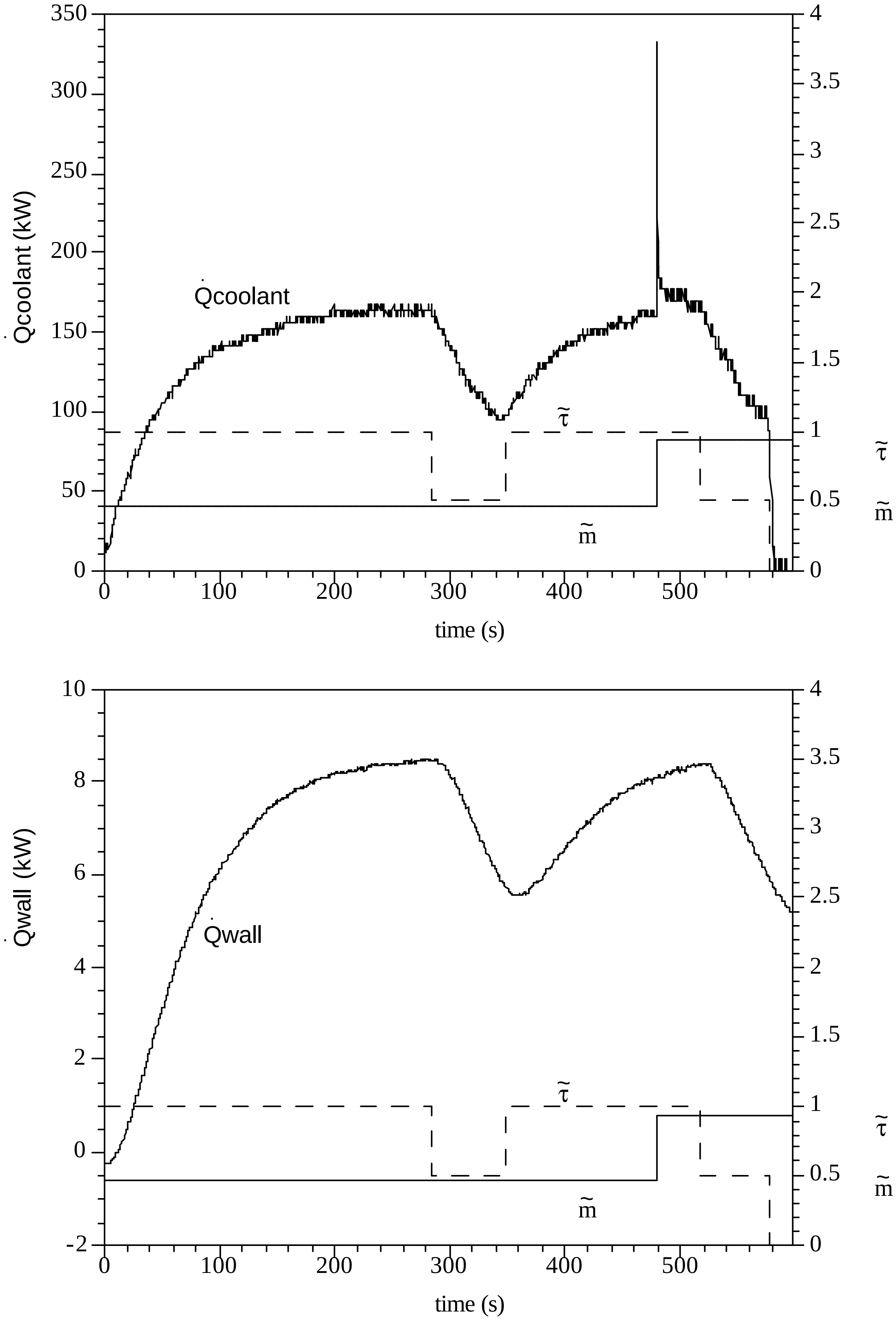

Figure 6- Rates of heat dissipation from the propulsion motor to the cooling water (top) and to the ambient through the walls (bottom).

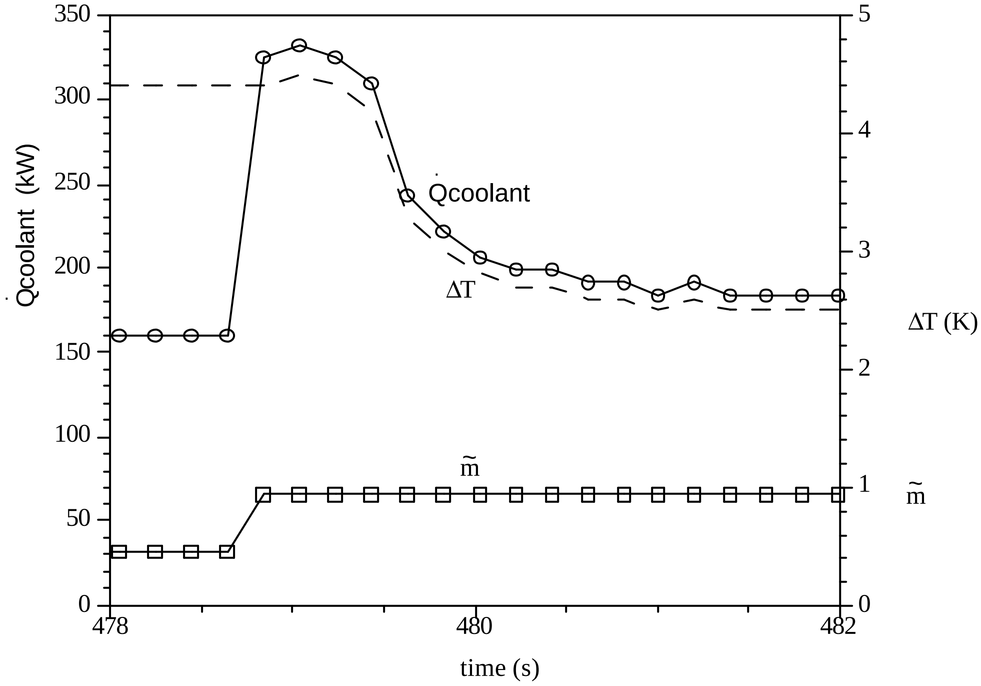

Figure 7- Heat exchanger response to a water mass flow rate increase.

5. Conclusions

In an all-electric ship, the propulsion motor is only one component of heat generation. Significant power in the megawatt range is generated when the powerful radars and electrical weapons are activated. A larger heat exchanger could be built to dissipate the heat, but is there an optimal heat exchanger size? A heat exchanger that is not properly designed will leave a heat signature that could be monitored to reveal critical operations data of the ship. In contrast a well designed heat exchanger could control or even mask the heat signature.

In this study, the thermal response of a 2.5MW ship propulsion motor was studied for different load conditions and different coolant flow rates. The results indicate that fast changes in the thermal signature can be obtained by controlling the coolant flow to the motor. The response to an increase of the flow rate from 138.2 gpm to 280.7 produced an almost instantaneous response in the thermal signature (Figure 7). This response is faster than the response of the motor wall temperature and coolant temperature gain, ΔT, to a load variation from 100% torque to 50% torque and from 50% torque to 100% torque. This suggests that the coolant mass flow rate is an effective mechanism to control and mimic thermal signature changes associated to load variations.

Intelligent heat exchangers that dynamically adjust the flow rate based on the load and heat signature are needed for the next generation all electrical ships. Future work will expand the existing testing unit to include other electrical components such as the motor drives, as part of the hardware-in-the-loop experiment. The work will be targeted towards the design of a dynamically reconfigurable heat exchanger.

6. Acknowledgments

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7. References

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