

Energy-Aware Cooling for Hot-water Cooled Supercomputers

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Abstract—Hot-water liquid cooling is a key technology in future green supercomputers as it maximizes the cooling efficiency and energy reuse. However the cooling system still is responsible for a significant percentage of modern HPC power consumption. Standard design of liquid-cooling control relies on rules based on worst-case scenarios, or on CFD simulation of portion of the entire system, which cannot account for all the real supercomputer working conditions (workload and ambient temperature). In this work we first introduce an analytical model, based on lumped parameters, which can effectively describe the cooling components and dynamics, and can be used for analysis and control purposes. We then use it to design an energy-optimal control strategy which is capable to minimize the pump and chiller power consumption while, meeting the supercomputer cooling requirements. We validate the method with simulation tests, taking data from a real HPC cooling mechanism, and comparing the results with state-of-the-art commercial cooling system control strategies.

I. INTRODUCTION AND RELATED WORK

The power density of high performance computing (HPC) systems and datacenters has been rapidly increasing in the last decades, and it is expected to keep growing, with the arrival of recent ultra-dense many-core chips. As a consequence, removing the heat, produced by the computing equipment, is becoming a crucial issue. Liquid cooling solutions are widely adopted to address high power levels of for modern HPC facilities. However, the amount of electrical power drained to operate the liquid cooling devices (e.g. chillers, pumps) plays a significant role in the overall system consumption. Given the recent trend towards green computing, energy-efficient thermal management and cooling control strategies must be developed. This timely topic has attracted significant research activity. The challenge has been mainly addressed at small scale (at chip level). In [1], [2], [3], [4] energy efficient cooling control solutions, adjusting the liquid flow rate, are explored for novel technologies, based on 3D MPSoCs with inter-tier liquid cooling systems. In [5], convex optimization framework is exploited to adjust the speed of multiple processing elements under thermal constraints. To the best of the authors knowledge, few results consider the cooling system as a whole, trying to actively minimize its energy consumption from a large scale viewpoint. In [6], an insightful analysis and experimental-based power modeling of both the cooling and computing equipment is carried out for the *IBM BlueGene/Q* machine, with the purpose to analyze profitable conditions for

hot liquid free cooling regime, and in general, optimal liquid temperatures from an energy consumption standpoint. In [7], optimal rack displacement for cooling energy minimization in data centers is studied; considering hardware heterogeneity and probabilistic servers utilization function, an ILP problem is formulated. In [8] server fans speed is controlled to minimize the system total power, including a careful characterization of the leakage components as a function of the operating temperature. In [9] the rack inlet coolant temperature is dynamically controlled, regulating the circuit pump and heat exchanger to minimize a data center cooling power consumption.

The purpose of this paper falls in this category, and it is two fold; similarly to what in [6] the modeling of the overall cooling system mechanism is performed. Owing to system complexity, this is not a trivial task. If the focus is put at chip level optimization, numerical CFD methods [10], or complex nonlinear identification tools [11], can be used. However, it is impossible to scale up these algorithms to the overall system. Here, in a similar spirit to the approach presented in [12], we derive a simple, lumped parameters, analytical model of the overall cooling circuit, typically employed in supercomputers and datacenters. In this respect, Eurora, a supercomputer built by Eurotech and Cineca [13], is taken as benchmark system for modeling and parameters identification.

The model is able to describe the dominant dynamics, and to capture the overall system behavior, moreover it can be exploited to rigorously design power optimization strategies, which is the main contribution of this work. Based on the model, an optimization strategy minimizing the cooling system power under thermal constraint is formulated. The proposed algorithm allows to steer the temperatures of the liquid to the optimal values, automatically choosing when free cooling or hot liquid regimes are profitable. The considered approach shows similarities with the liquid cooling control described in [9], however, in that work a suitable heuristic approximation of the optimal energy path is exploited, considering the cooling devices (pumps, heat exchangers) as uncoupled systems (i.e. their effectiveness depends solely on their own control variables). Here we formally account for cooling system components interlaced action, optimizing their power status in one holistic formulation. On the other hand here steady state conditions are considered.

The paper is organized as follows. In Section II the benchmark system features are briefly recalled, in Section III the cooling system analytical model is discussed, while Section IV is devoted to present the optimal control solution. Section V validates the solution with simulation tests and comparison against experimental data aquired from Eurora. Section VI wrap up the paper with concluding remarks and future work directions.

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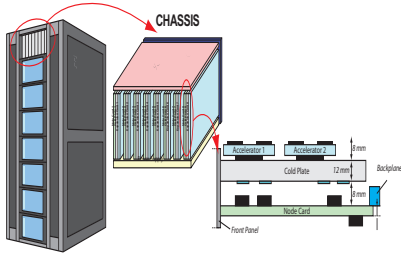


Fig. 1. Eurora Architecture.

II. EURORA ARCHITECTURE

The Eurora Supercomputer, developed by Eurotech and Cineca [14] is a high-end energy efficient supercomputer; in July 2013, it ranked first in the Green500 list. Hence, it is a perfect candidate to test energy efficient cooling solutions. Here the main features of this supercomputers are briefly recalled. Fig. II shows the system architecture; a single rack containing eight stacked chassis, each chassis host eight node cards and sixteen expansion slots. Each node card hosts 2 Intel Xeon E5 series (SandyBridge) processors and 2 expansion cards configured to host accelerator modules. Nodes are equally equipped with 8 cores 2.0 GHz E5-2658 processors (TDP=95 W), or 8 cores 3.1 GHz E5-2687W processors (TDP=150 W). The accelerator modules can be Nvidia Tesla (Kepler) (TDP=250 W), or, alternatively, Intel MIC KNC with (TDP=245 W). SMP CentOS Linux distribution version 6.3 is executed on each node of Eurora. A low-overhead monitoring infrastructure, capable of tracking in detail and in real-time the thermal and power characteristics of Eurora components with fine-grained resolution, has been implemented [13]. The information acquired with such infrastructure are key to implement sophisticated control and management solutions, as that presented in this work.

As regards the cooling solution, a liquid (water) based technology is adopted, the architecture and modeling are discussed in Section III, however it is important to remark that the machine is allowed to operate with hot coolant. However, due to its prototyping status, the current Eurora deployment, does not take advantage of this feature. This motivates the study carried out in this paper, which is mainly devoted to formally understand when free-cooling is not only feasible, but also optimal for what concerns the power consumptions.

III. COOLING SYSTEM MODELING

Modeling the cooling system is of outmost importance at design stage, and to run energy-aware strategies maximizing the efficiency. To obtain an accurate characterization of the cooling components of an HPC machine is a rather complex task; CFD numerical tools are typically exploited and, if a detailed analytical model is derived, it involves highly nonlinear, infinite dimensional dynamics, and some approximation to make it numerically tractable has to be performed [15]. Moreover, accurate cooling system description is typically carried out at small scale level (e.g. the single blade cold plate), since complexity would dramatically increase if larger machines sections were considered. However, this strategy does not provide a global description of the complete cooling system behavior. Hence significant energy contributions coming from unmodeled parts, or other large scale related phenomena, could be overlooked. The aim of this Section is to overcome such drawbacks, providing a lumped parameters

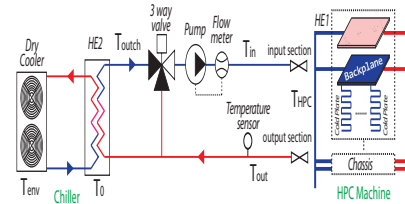


Fig. 2. HPC machine typical cooling system architecture.

model, characterizing the cooling system dominant dynamics, which can be exploited to design energy optimal cooling management strategies, in an analytical fashion. Fig. 2 shows the typical topology for a liquid-cooled HPC system considered in this paper. It can be reduced to three main components; a chiller, refrigerating the coolant (typically water), a variable speed pump pushing the liquid into the cooling lines, a three-way valve recirculating part of the HPC warmed return coolant, which is mixed with the chiller outlet water before re-entering the super computer. The model for this cooling circuit will be derived under the following hypothesis [16], [17]

- All flows are turbulent, and there are no laminar flow effects;
- There is no additional heat loss to surroundings, namely heat exchange takes place solely in the HPC machine or in the considered cooling components;
- The coolant does not change its phase during its cycle
- The coolant is assumed to be incompressible, and its density and specific heat are constant in the range of the considered temperature values;
- Forced convection over the coolant lines is assumed to be the dominant heat removal phenomenon.

The main idea is to describe the HPC machine, the chiller, and the cooling circuit thermal behavior as the interaction between two main heat exchanger, one (HE_1) between the machine and the cooling circuit, the other at the evaporator side of the chiller (HE_2). In deriving the model we will assume that the energy absorbed/rejected by the heat exchangers depends on the mean value of the coolant inlet and outlet temperature at the corresponding heat exchange point [18], while the outlet temperatures will be taken as the variables describing the system *state*. Thus, exploiting the assumptions above, and using thermodynamics first principles (e.g. Young transportation theorem [19], [20]), the overall system dynamics can be expressed as

$$\begin{aligned}
 \dot{T}_{HPC}(t) &= \frac{1}{C_{thHPC}} \left(\dot{Q} - R_{HPC-HE1}^{-1} \left(T_{HPC}(t) - \frac{T_{out}(t) + T_{in}(t)}{2} \right) \right) \\
 \dot{T}_{out}(t) &= \frac{1}{C_{HE1}} \left(R_{HPC-HE1}^{-1} \left(T_{HPC} - \frac{T_{out}(t) + T_{in}(t)}{2} \right) - q_{cv}\rho(T_{out}(t) - T_{in}(t)) \right) \\
 \dot{T}_{outch}(t) &= \frac{1}{C_{HE2}} \left(q_{cv}\rho(T_{out}(t) - T_{in}(t)) - R_{HE2-CH}^{-1} \left(\frac{T_{out}(t-\tau_2) + T_{outch}(t)}{2} - T_0 \right) \right)
 \end{aligned} \tag{1}$$

where T_{HPC} is the temperature representing the HPC machine thermal status, i.e. the average temperature at the heat exchange surface with the coolant, C_{thHPC} models the machine thermal capacitance. Similarly, T_{in} , T_{out} , are the coolant inlet and outlet temperature, T_{outch} is the chilled coolant outlet temperature, R_{HE2-CH} , C_{HE2} are the thermal parameters of the heat exchanger at the chiller evaporator side, while $R_{HPC-HE1}$, C_{HE1} are the thermal resistance and capacitance of the heat exchanger modeling heat transfer from the super computer to the cooling mechanism. Finally, \dot{Q} is the thermal power produced by the HPC system, T_0 is the temperature at the chiller evaporator side, ρ and c_v are the coolant density and specific heat (at constant volume) respectively, while q is the coolant flow rate.

Remark 1: For the sake of completeness, the coolant outlet temperature in the third equation of (1) is delayed to account for propagation delays in the cooling pipes. Specifically, assuming temperatures T_{in} , T_{out} are measured at the machine side, the effects of $T_{out}(t)$ (evolving as the second equation in (1)) will affect the chiller heat exchanger after a time depending on the flow rate according to the law: $\tau_2 = \rho V_2 q^{-1}$ [16]. Where V_2 is the volume of the corresponding portion of the cooling pipe.

A crucial step for analysis and control is to express system (1) as a function of its own *state variables* T_{HPC} , T_{out} , T_{outch} . To this aim, by energy balance considerations, the following condition, relating the coolant temperatures entering/exiting the mixing valve can be obtained

$$T_{in}(t) = \alpha T_{outch}(t - \tau_1) + (1 - \alpha) T_{out}(t - \tau_1 - \tau_2) \quad (2)$$

where $\alpha \in [0, 1]$ denotes the position of the valve modulating the mixing between warm and chilled water. Again, considering the coolant cycle, and assuming the chilled coolant temperature is measured at the chiller output side, suitable delays have been added, considering the temperature of the chilled liquid is measured at the three-way valve side. $\tau_1 = \rho V_1 q^{-1}$ is the transport delay between the mixing point and the inlet coolant point at the HPC side, with V_1 the corresponding line total volume. Substituting into (1) yields

$$\begin{aligned} \dot{T}_{HPC}(t) &= \frac{1}{C_{thHPC}} \left(P_{th} - R_{HPC-HE1}^{-1} \left(T_{HPC}(t) - \frac{T_{out}(t) + \alpha T_{outch}(t - \tau_1) + (1 - \alpha) T_{out}(t - \tau_1 - \tau_2)}{2} \right) \right) \\ \dot{T}_{out}(t) &= \frac{1}{C_{HE1}} \left(R_{HPC-HE1}^{-1} \left(T_{HPC} - \frac{T_{out}(t) + \alpha T_{outch}(t - \tau_1) + (1 - \alpha) T_{out}(t - \tau_1 - \tau_2)}{2} \right) - \right. \\ &\quad \left. q c_v \rho (T_{out}(t) - T_{in}(t)) \right) \\ \dot{T}_{outch}(t) &= \frac{1}{C_{HE2}} \left(q c_v \rho (T_{out}(t) - T_{in}(t)) - \right. \\ &\quad \left. R_{HE2-CH}^{-1} \left(\frac{T_{out}(t - \tau_2) + T_{outch}(t)}{2} - T_0 \right) \right). \end{aligned} \quad (3)$$

The control knobs which can be moved to steer the above dynamics are the valve position α and the flow rate q , for what concerns the cooling circuit, while, as regards the chiller, temperature T_0 can be imposed.

IV. ENERGY EFFICIENT CONTROL STRATEGY

Based on the model derived in the previous Section, a suitable control solution, acting on the knobs α , q , T_0 can be implemented with the purpose of minimizing the cooling system energy consumption, while keeping the HPC machine in a proper thermal status. In the following, first the objective function and the system constraints are defined, then the energy-optimal control strategy is presented, underscoring, in a rigorous way, when the free-cooling operation (chiller is turned off) is energetically convenient.

A. Cost Function and Constraints Definition

In order to design an optimal cooling control strategy, a preliminary step is to define a cost function to be minimized by the algorithm. The primary issues in HPC machines are related to temperature limits and power consumption. Then it seems natural to associate the utility function, and the related constraints, to such variables. In particular, we choose to minimize the power terms absorbed by the pump pushing the coolant, and by the chiller to provide the required cooling load. Moreover, temperature constraints, along with control input physical limits are enforced.

The first step is to analyze the variables affecting the pump and chiller energy consumption. For what concerns the pump, it is well known that its consumption quadratically increases with the flow rate [1]. As regards the chiller, sophisticated models are generally needed to characterize its power efficiency as a function of the operating conditions [21], [22]. However, such models require accurate knowledge of several chiller internal variables, which are monitored if advanced control strategy have to be implemented to the chiller itself [23]. Here the focus is on the overall cooling system, thus, for the sake of simplicity, we assume T_0 as a control input, assuming a suitable chiller low level controller (see for instance [24]) is able to steer such variable to the required reference. For these reasons, the coefficient of performance (COP), defined as the ratio between the chiller cooling load and its power consumption, is approximated as the ideal *Carnot efficiency* of the refrigerating cycle, i.e. $COP_{ch} = \frac{T_0}{T_{env} - T_0}$ where T_{env} is the environment temperature at the condenser side, and absolute temperature values need to be considered. Beside being an ideal and quite rough approximation (no entropy augmentation is assumed in the refrigeration cycle), still, given a specific chiller (i.e. setting the cycle irreversibilities) the Carnot coefficient can be profitably used as a proxy for chiller efficiency. Furthermore, chillers thermal insulation cannot be ideal, thus power is drained also to keep T_0 at the prescribed value. Such additional dissipation term can be characterized by a thermal resistance R_{ins} , placed between the chiller evaporator side and the environment. Thus a power proportional to $(T_{env} - T_0)/R_{ins}$ has to be used in order to maintain T_0 , in face of T_{env} . Bearing in mind these considerations, the following cost function is considered

$$J = \gamma_1 k q(\tau)^2 + \gamma_2 \left(COP_{ch}^{-1}(T_{env}, T_0) \dot{Q} + \frac{(T_{env} - T_0)}{R_{ins}} \right) \quad (4)$$

where γ_1 , γ_2 are arbitrary positive weighting coefficients to be tuned for prioritizing the pump or chiller power minimization. k is a pump specific value to be multiplied by the required flow rate in order to compute the power consumption.

Remark 2: The chiller cooling load has been approximated in (4) with the thermal power \dot{Q} produced by the super computer. By model (3) it is easy to verify that this is true only in steady state conditions, however, since the ensuing discussion will be mainly devoted to these situations, this simplification has been adopted also in the objective function.

As mentioned, along with energy efficiency, system has to be kept in a nominal working region, where all the system temperatures lie within a safe range, and the required actuators effort meet the physical limits. Therefore, the following constraints have to be joined to the optimization problem minimizing J

$$\begin{aligned} T_{inMIN} \leq T_{in} \leq T_{inMAX}, T_{outMIN} \leq T_{out} \leq T_{outMAX} \\ T_{out} - T_{min} \leq \Delta T, q \leq q_{max}, 0 \leq \alpha \leq 1 \\ T_{HPC} \leq T_{HPCmax}, T_0 > T_{0min}. \end{aligned} \quad (5)$$

The minimum and maximum temperature, along with their difference limitation, for the inlet and outlet coolant, are imposed for computing device safety and performance criteria (e.g. avoid condensation close to the electronics, do not exceed core maximum temperature due to too hot inlet coolant). The same reasoning holds for the limit T_{HPCmax} , while the other inequalities regards feasibility of the control decision variables α, q, T_0 according to their bounds. T_{0min} is related to the evaporation temperature of the chiller refrigerant fluid, q_{max} depends on the pump and cooling pipes sizing, while the valve position clearly varies from fully closed to fully open.

B. Proposed Controller

Summarizing the reasoning of the previous paragraph, a constrained optimal control problem can be formulated. Based on model (3), inputs α, q, T_0 have to be selected to minimize objective function (4) under constraints (5).

Here steady-state situations under constant power values \dot{Q} are considered, namely $\dot{T}_{HPC}, \dot{T}_{out}, \dot{T}_{outc}$ are set to zero in (3). Despite not being comprehensive of all the possible scenarios, the steady-state analysis and optimization carried out in the remainder of the paper can cover significant working cases. This statement is backed-up by the assumption that supercomputers are typically endowed with job schedulers, optimized to balance the machine workload during its operating regimes, hence constant (or slowly varying w.r.t. cooling system dynamics) power levels \dot{Q} , can be assumed.

Bearing in mind these considerations, the following equations are derived from (3);

$$\begin{aligned} \dot{Q} = qc_v\rho(T_{out} - T_{in}) = R_{HPC-HE1}^{-1} \left(T_{HPC} - \frac{T_{in} + T_{out}}{2} \right) = \\ \alpha q_m c_v (T_{out} - T_{outc}) = R_{HE2-CH}^{-1} \left(\frac{T_{outc} + T_{out}}{2} - T_0 \right) \end{aligned} \quad (6)$$

equality above highlights how the quantity $(T_{outc} + T_{out})/2$ is affected only by the chiller control knob T_0 , while the temperature gradient $T_{out} - T_{in}$ across the HPC machine, and the difference $T_{out} - T_{outc}$ at the chiller side, can be steered by acting on the flow rate q and valve position α . Hence, T_{out}, T_{in} can be independently imposed; the corresponding steady-state inputs will be

$$q = \frac{\dot{Q}}{c_v\rho(T_{out} - T_{in})}, \alpha = \frac{\dot{Q}}{q c_v\rho(T_{out} - T_0 + \dot{Q} R_{thHE2-CH})}. \quad (7)$$

The choice of T_{in}, T_{out} is obviously not fully arbitrary, but subject to what reported in (5). Constraints on the minimum/maximum or difference value are trivially formulated,

while, after some computation, the maximum machine temperature limit T_{HPCmax} , can be expressed as

$$\dot{Q} R_{HE1-HPC} + \frac{T_{in} + T_{out}}{2} \leq T_{HPCmax}. \quad (8)$$

this bound is typically accounted in a conservative way. Usually cooling system control strategy is tuned on the worst case scenario; the reference values for T_{in}, T_{out} , are set to meet (8) under the machine maximum power \dot{Q}_{max} . However this condition is hardly reached during the system everyday activity, thus, adjusting the temperatures values depending on the working point and the environmental conditions, could significantly improve the energy efficiency. This is exactly the purpose of our cooling management strategy, which, summarizing all the previous considerations, can be cast into the following constrained optimization problem

$$\begin{aligned} \min_{(q, \alpha, T_0, T_{HPC}, T_{in}, T_{out}, T_{outc})} \quad & \gamma_1 k q^2 + \gamma_2 \left(\dot{Q} \frac{T_{env} - T_0}{T_0} + \frac{(T_{env} - T_0)}{R_{ins}} \right) \\ T_{inMIN} \leq T_{in} \leq T_{inMAX}, T_{outMIN} \leq T_{out} \leq T_{outMAX} \\ T_{out} - T_{in} \leq \Delta T, 0 \leq q \leq q_{max}, 0 \leq \alpha \leq 1 \\ T_{HPC} \leq T_{HPCmax}, T_0 > T_{0min} \\ \dot{Q} R_{HE1-HPC} + \frac{T_{in} + T_{out}}{2} = T_{HPC}, \\ T_{in} = \alpha T_{outc} + (1 - \alpha) T_{out}, Q = c_v \rho q (T_{out} - T_{in}) \end{aligned} \quad (9)$$

clearly (9) is an *implicit* formulation, eliminating the equality constraints related to the derived model an *explicit* formulation in the decision variables q, α, T_0 could be derived. Since the two formulations are equivalent from a theoretical standpoint (while numerical efficiency can vary depending on the problem) this manipulation is not shown, owing to space limitation. It is worth to remark that the optimization strategy (9) can rigorously provide a crucial information from an energy aware standpoint, i.e when it is profitable to go free cooling. Indeed, an optimal value $T_0^* = T_{env}$, means the heat can be safely removed without turning on the chiller, saving its energy cost. This property of the algorithm, along with its effectiveness, will be assessed in the ensuing Section, where insightful applications of the optimization strategy to the considered benchmark supercomputer are reported.

V. SIMULATION RESULTS AND COMPARISON WITH EXPERIMENTAL DATA

The proposed control strategy has been tuned and applied taking Eurora as a benchmark green supercomputer solutions. The system parameters are reported in Tab. V. In order to validate our solution, the algorithm (9) has been run under a rather exhaustive set of the system possible operating conditions. Specifically, the complete range of admissible power scenarios, from the idle conditions $\dot{Q}_{min} \approx 1.9kW$, to the maximum rated power $\dot{Q}_{max} = \sum_{node=1}^{64} TDP_{node} \approx 47kW$, is swept during the test, and several environment scenarios ($T_{env} = [0 \ 10 \ 20 \ 25 \ 30 \ 40]^\circ C$), covering the seasonal average temperature variations have been reproduced. The obtained results are shown in Figs. 3, 4. Optimization was run with equal unit weights γ_1, γ_2 for the pump and chiller power terms.

As expected the optimal power consumption monotonically increases with \dot{Q}, T_{env} (see Fig. 3 (a)). It is further to highlight the trade-off between the pump and chiller contributions; the kinks in the optimal objective functions noted for all the environment temperatures but $0^\circ C$, correspond to the point where the chiller is actively exploited, i.e. the free cooling

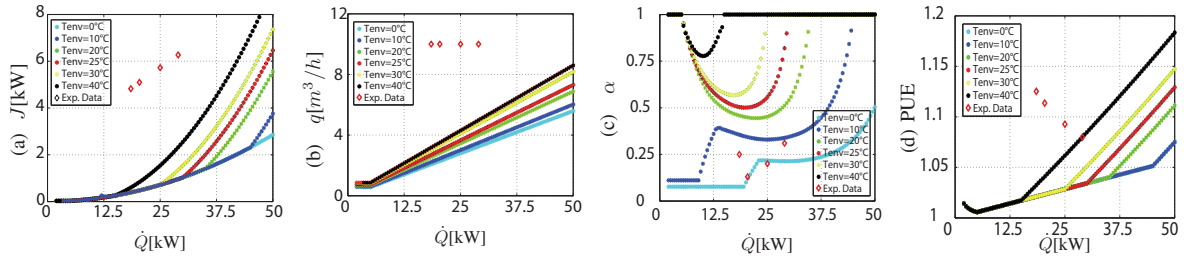


Fig. 3. Cost function (a) and variables q (b), α (c), and PUE (d) under different environmental and workload conditions (square points with different colors: proposed solution at different T_{env} ; red diamond points: Eurora experimental data at $T_{env}=25^\circ\text{C}$ with its native strategy).

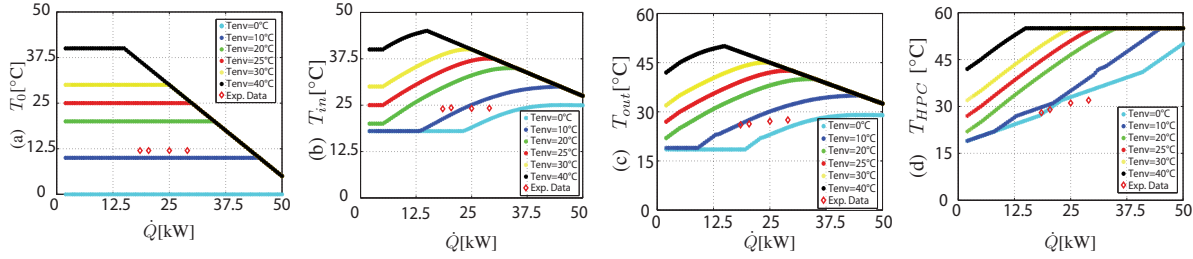


Fig. 4. Machine thermal power (a) inlet water temp. (b), outlet water temp. (c), and machine temp. (d) under different environmental and workload conditions (square points with different colors: proposed solution at different T_{env} ; red diamond points: Eurora experimental data at $T_{env}=25^\circ\text{C}$ with its native strategy).

condition is no longer feasible or convenient, and second term in the function (4) becomes not null. This is confirmed by the plots regarding T_0 (Fig. 4 (a)); as expected, until the chiller is not needed, free cooling condition $T_0 = T_{env}$ is enforced, while, when a critical amount of thermal power, depending on the environmental temperature, is reached, the chiller has to actively exploited to cool the system. When the chiller is on, still T_0 is maximized in order to keep a good COP. As far as the other decision variables are concerned, clearly q is minimized, and augmented to comply with constraints when the thermal load increases and the environment condition are worsened. The valve position α , shown in Fig. 3 (c), has a more peculiar behavior. For high power values and warm/hot environment temperatures, it is fully open, i.e. all the warm water flowing from the system is let into the chiller, so that temperature constraints can be met.

The same happens during free cooling, since T_0 is high. During intermediate situations, the valve is modulated accordingly to minimize the flow rate. Differently, for low T_{env} and low thermal power, the valve is kept almost fully closed, i.e. the chiller is almost completely bypassed, to keep the coolant temperatures above T_{inMIN} , T_{outMIN} . When thermal load increases then it is opened to exploit the environment temperature, if free-cooling is possible, or T_0 , to chill the outlet water. The coolant inlet and outlet temperatures, along with the machine working temperature are reported in 4 (plots (b) (c) (d)). Not surprisingly, during free cooling at low machine power, warm coolant temperatures are reached, then the chiller actively regulate their values to meet T_{HPCmax} limit and avoid a sudden flow rate increase.

In order to validate the results, a comparison with some experimental scenarios acquired from the real system, has been carried out. The data, (denoted as diamond shaped samples in Figs. 3, 4) were recorded during days in late spring when, for the geographical location of the machine, an average environment temperature of 25°C is estimated. Different loads

($\dot{Q} \approx [18.4 \ 20.5 \ 25 \ 29]\text{kW}$) have been considered. A state-of-the-art commercial cooling control solution is run on the system, with the purpose to keep the prescribed working conditions, but no specific penalties on power absorption. The flow rate is kept constant at a value ($10\text{m}^3/\text{h}$) close to the maximum, while the chiller is controlled to set $T_0 = 12^\circ\text{C}$ in any condition, and the valve is modulated to enforce $T_{out} - T_{in} < 5^\circ\text{C}$. In order to compare this approach against our solution, the cooling consumption under such strategy has been estimated by plugging the experimental data into (4). The obtained values are remarkably higher (3/4 times) w.r.t. what could be obtained by our strategy (see 3 (a)). This is expected, since two out of three decision variables are kept at fixed value in the commercial solution. On the other hand, the actual machine temperatures are way below the limits, since the chiller (which is always on), and the high flow rate, keep the coolant temperatures down. Similar reasoning can be made for the Power Usage Effectiveness (PUE), defined as $(\dot{Q} + J)/\dot{Q}$. This factor is portrayed in plot (d) of Fig. 3. It can be noted how, even under the worst case conditions, the values obtained are considerably lower than the experimental data, recorded at around 25kW and $T_{env} = 25^\circ\text{C}$.

Our solution outperforms the commercial strategy because it would have allowed free cooling under all the reported experimental scenarios. This confirms the chiller power plays a crucial role in the total system cost.

On the other hand, under the power optimal control inputs, the machine temperature limit would have been much closer to its limit with respect to the experimental case (see Fig. 3 (a) and (d)). However, to prevent premature wear and tear of the computing devices due to prolonged exposure to high temperatures, a tighter constraint can be imposed to keep a safe margin from the maximum, or a penalizing term can be added to the objective function.

Table I Eurora Cooling System Parameters

Parameter	Value
$R_{HPC-HE1}$	0.7 [mK/W]
R_{HE2-CH}	0.6 [mK/W]
C_{HE1}	225 [kJ/K]
C_{HE2}	450 [kJ/K]
C_{HPC}	150 [kJ/K]
τ_{1max}	30 sec.
τ_{2max}	30 sec.
T_e	3 [°C]
T_{inMIN}	18 [°C]
T_{outMAX}	50 [°C]
T_{outMIN}	18 [°C]
T_{HPCmax}	55 [°C]
q_{max}	12 [m^3/h]
R_{ins}	80 [K/W]
c_v	4186 [J/(K $g \times$ K)]
ρ	1000 [K g/m^3]

VI. CONCLUSIONS

In this paper an analytical model of the overall liquid cooling mechanism, typically exploited in super computers, has been presented, with the aim to obtain a “coarse grain” but significant representation which can be exploited for cooling energy optimization. Based on this description an optimal thermal management strategy has been designed, considering steady-state scenarios, to rigorously minimize the cooling parts (pumps and chiller) power consumption, and fulfill the machine temperature constraints. The approach has been tuned to the parameters of Eurora HPC, tested under various possible environmental and workload scenarios, and compared with experimental data acquired from the system which run state-of-the-art cooling regulation algorithms. Our solution shows promising opportunities to improve the cooling efficiency, which is already a significant parameter for green supercomputers, and it is expected to play even a bigger role in the next generation machines. The cooling system energy saving compared to the Eurora strategy is on average 3.5kW in the considered scenarios. Normalized to the machine nominal power (25kW), it means a 12% of energy reduction, which significantly affect the operating cost. Future works will be devoted to extend this preliminary results, by joining the optimizer with dynamic controllers able to bring the cooling system to its optimal point, while complying with constraints during the transient phases. The optimizer can also be improved and combined with other optimization strategies, e.g. energy aware job dispatcher, dynamic frequency control, running at other levels of the software pyramid.

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