Thermal model to granular for our paper I think

DC Thermal Energy Flexibility Model for Waste Heat Reuse in Nearby Neighborhoods

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ABSTRACT

In this paper we address the Data Centers (DCs) energy efficiency problem from a thermal perspective by considering them as large producers of waste heat integrated with smart energy infrastructures and utilities, through which they can effectively exploit their thermal flexibility for nearby neighborhoods heating. We provide a mathematical formalism for modeling the thermodynamics of the processes within DCs equipped with heat reuse technology and proactive DCs operation control mechanisms that allow them to adapt their thermal response profile to meet various levels of hot water demand. Numerical simulation-based experiments are shown considering the hardware systems characteristics and operation of one server room from Engineering Pont Saint Martin (PSM) DC. The results show the potential of using workload delay tolerant time shifting and server room pre-cooling as flexibility mechanisms for adapting the DC thermal energy profile to meet the demand.

CCS CONCEPTS

• **Hardware** → Temperature optimization; Enterprise level and data centers power issues; • **Computing methodologies** → Modeling and simulation; • **Theory of computation** → Mathematical optimization;

KEYWORDS

Data Centers, Waste Heat Reuse, Thermal Energy Flexibility, Mathematical Models and Numerical Simulation.

1 INTRODUCTION AND RELATED WORK

Energy efficiency in Data Centers (DCs) is of utmost importance as they are among the largest consumers of energy at global scale and their energy consumption is rapidly increasing due to the rising digitization of human activities. One potential way to achieve cost-effectiveness and/or energy efficient solutions is the integration of DCs with the energy infrastructure and utilities, as a whole. As successfully investigated within the framework of the EU GEYSER project (http://www.geyser-project.eu/), DCs may become relevant, yet active stakeholders in a system-level energy value chain, and may gain significant benefits from cooperating under different loose or tighter modalities with smart energy grids (both smart electricity and heat grids). In this context of thermal energy reuse, DCs are large producers of waste heat, which can be effectively used either internally for Space Heating and/or Domestic or District Heating network operators.

Few approaches can be found in the state of the art literature addressing the thermal aspect of energy flexibility in relation with demand response programs and reuse of waste heat in nearby neighborhoods and buildings [1, 2]. The evolution of hardware in the last years leads to the continuous increase of the power density of chips and servers in DCs. As the IT server design improves for allowing them to operate at increasingly higher temperatures, and the server rooms' density will continue to rise, the DCs will be transformed in large producers of heat [3]. This generates an energy loss cascading effect in which the heat generated by servers must be dissipated and gets wasted while the cooling system needs to work at even higher capacity leading to even increased levels of energy consumption. There are two big issues with DC waste heat reuse in the local thermal grid: the relatively low temperatures in comparison with the ones needed to heat up a building and the difficulty of transporting heat over long distances [4]. Nowadays the DCs have started to use heat pumps to increase the temperature of waste heat, making the thermal energy more valuable, and marketable [5]. With the help of heat pumps, the heat generated by servers can be transferred to hot water heated to around 80 degrees Celsius, suited for longer distance transportation in the nearby houses.

A thorough review of cooling systems and waste heat recovery systems is presented in [6]. The authors discuss the DC layout and the management of classical cooling systems, emphasizing the problems that occur when the chip power density increases. Furthermore, the authors evaluate the main heat reuse technologies, such as plant or district heating, power plant co-location, absorption cooling, Rankine cycle, piezoelectric, thermoelectric and biomass. In [7] a study of ammonia heat-

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pumps used for simultaneous heating and cooling of buildings in Norway is conducted. The approach is analyzed from a thermodynamic point of view, computing the proper scaling with the building to be fitted in. Finally, a set of real-world deployments, in hospitals or research centers from Norway are discussed. The authors of [1] present an infrastructure created to reuse the heat generated by a set of servers running chemistry simulations in Condor to provide heat to a greenhouse located within the Notre Dame University. It consists of an air recirculation circuit that takes the hot air from the hot aisle containment and pumps it in the space needed to be heated. The main issue with this approach is that the heat is provided at the exhaust temperature of the servers is relatively low making it unsuited for long distance transportation. In [8] a study on the impact of a thermal battery attached to the heat pumps to increase the system's thermal inertia is conducted. The thermal battery consists of two water tanks, one for storing cold water near the evaporator of the heat pump, and one for storing hot water near the heat exchanger of the heat pump. The two tanks allow the heat pump to store either cool or heat. The authors conclude that such approach proves to be more economically efficient than traditional electrical energy storage devices, such as lithium-ion batteries. A similar heat pump based waste recovery system is proposed in [9] for Purdue University's DC which has an hourly consumption of about 1MWh. Their study shows that by equipping the DC with a CO_2 heat pump, 575m3/day of natural gas and 750kg/day of coal are saved, reducing the CO_2 emissions by almost 3000kg/day. In [10] an analytical model of a heat pump equipped with thermal tanks is defined and used to study the thermodynamics of the system within a building. The system is divided into subcomponents for achieving faster computation time while the transfer dynamics is modeled using partial-differential equations. The model was used to simulate the behavior of a building during a 24-hour operational day, analyzing the sizes of the storage tanks needed to satisfy the heating and cooling demands. The need for an intelligent hot water distribution system in the context of modern smart grids, high energy demand and renewable energy sources is discussed in [11]. Their work is intended to be a foundation for hot water ancillary services, and is based on evaluating a set of forecasting techniques on a set of water consumption patterns. Furthermore, an evaluation of hot water consumption by residential dwellings is conducted in [12] for UK and in [13] for US.

Starting from existing literature, in this paper we will address the DCs wasted heat recovery problem by proposing a technique for optimizing the operation of DCs equipped with heat reuse technology with the goal of exploiting and adapting DCs' internal yet latent thermal flexibility and feed hot water to the nearby offices and houses on demand. To achieve our goal, we bring the following contributions:

- Provide mathematical formalisms for modeling the thermodynamics of processes within DCs equipped with a heat pump and thermal batteries.
- Define a proactive DC operation control technique on top of thermal energy flexibility mechanisms that allows the DCs to adapt their thermal response profile to meet various levels of hot water demand of the nearby offices or houses.
- Use the provided model and control technique to simulate and estimate the wasted heat recovery potential of Engineering Pont Saint Martin (PSM) DC [16] in the context of different scenarios.

2 DC THERMAL ENERGY FLEXIBILITY

To model the thermal behavior of the DC as a complex system we have identified the relevant sub-systems and have represented them as interconnected building blocks. In our approach each sub-system is represented as Mealy Machine with infinite states (see Fig. 1) featuring: a set of inputs $I \in R^{N_I}$ of size N_I, a set of outputs $O \in R^{N_O}$ of size N_O, an internal state $S \in R^{N_S}$ of size N_S, a set of independent control variable and constraints $C \in R^{N_C}$ of size N_C and finally two transfer functions f and g.

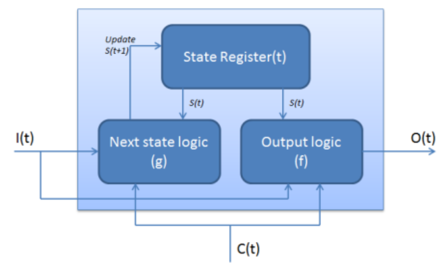

Figure 1: Sub-systems representation as Mealy Machine

Our modeling approach considers a **discrete time model** that involves having samples of the system and its components at equidistant timestamps $t \geq 0, t \in Z$.

$$D_T = \{t_k | (t_{k+1} - t_k) = constant, \forall k \geq 0\} \quad (1)$$

The function f computes at each timestamp t the output of the system $O(t)$ considering the current input set $I(t)$, the control variable set $C(t)$ and its internal state $S(t)$ at the timestamp t:

$$f: R^{N_I} * R^{N_C} * R^{N_S} \rightarrow R^{N_O},\ \ O(t) = f\big(I(t), C(t), S(t)\big) \quad (2)$$

At the same time function g updates the internal state of the system, preparing it for the next timestamp $t + 1$:

$$g: R^{N_I} * R^{N_C} * R^{N_S} \rightarrow R^{N_S}, S(t+1) = g\big(I(t), C(t), S(t)\big) \quad (3)$$

Fig. 2 presents the DC equipped with heat reuse technology viewed as energy interconnected sub-systems. Each block represents an energy storage or transformation node from the DC, while the connections represent energy transfer links.

The first sub-system modeled is IT Servers System. The IT Servers have as input $I(t)$ the DC clients' workload that need to be executed which can be classified as real-time E_R and delay-tolerant E_D. For simplicity, we consider that all the electrical energy consumed by the servers is converted into heat (i.e. the output $O(t)$) and must be eliminated to keep the temperature constant (similar assumption is also made in [14] [15]).

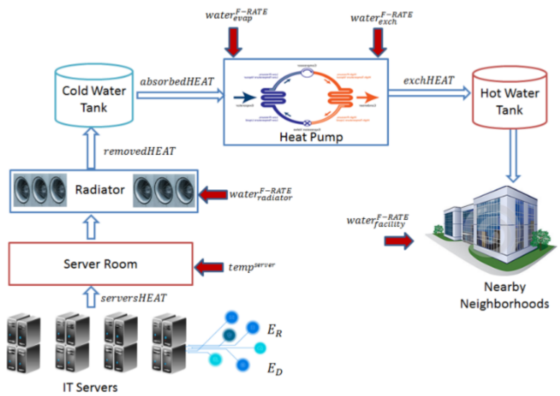

Figure 2: DC modeled as a set of energy linked sub-systems

The thermal energy flexibility is achieved by shifting the delay tolerant workload for future execution on a window of length T:

$$serversHEAT(t) = E_R(t) + E_D(t) \quad (4)$$

where the delay tolerant workload will be composed from the original baseline scheduled a timestamp t and the workload shifted for execution at t from previous timestamps $t-k$ while meeting the constraints defined for workload execution:

$$E_D(t) = E_D^{Baseline}(t) + E_D^{Shifted}(t) \quad (5)$$

$$\text{C1: } E_R(t) + E_D(t) < Servers_{Total\ Power} \quad (6)$$

The heat exhaust from the servers accumulates inside the server room (i.e. the physical containment that holds the IT servers) and leads to a temperature increase (i.e. heat accumulation). When the temperature inside the room exceeds a predefined set point (i.e. threshold), the system enters a critical state (it should be avoided at all cost):

$$\text{C2: } temp^{server}(t) \le threshold_{server}^{MAX} \quad (7)$$

Consequently, the heat removed from the server room should balance the heat gain leading to a constant temperature. Equation 8 allows for estimating the amount of heat that needs to be removed (i.e. $removedHeat$) for the server room over a time interval $[t_0, t]$ as a difference between the server generated heat, $serversHEAT$ and the heat accumulated during that interval.

$$\int_{t_0}^{t} removedHEAT(x)dx = \int_{t_0}^{t} serversHEAT(x)dx - accumulatedHEAT(t) \quad (8)$$

The heat capacity formula defined [13] can be used to determine the amount of heat accumulated in the server room in relation with the actual temperature value inside the server room at moment t compared with an initial baseline reference temperature at t_0 where m and c are the mass, respectively the heat absorption capacity of the substance (i.e. air).

$$accumulatedHEAT(t) = m * c * (temp^{server}(t) - temp_{reference}^{server}(t_0)\ ) \quad (9)$$

The second sub-system modeled is the cooling system's radiators. They are used to chill the hot air from the server room by transferring the heat generated by servers from the server room to the water that flows through their pipes. We consider a simplified radiator model with a uniform dispersion of the water flow through the pipe system over a constant area. The heat transfer inside the radiator is modeled as:

$$water_{radiator}^{F-RATE}(t) = \frac{removedHEAT(t)}{water_{heat-capacity} * \Delta T_{radiator}} \quad (10)$$

where $removedHEAT$ is the instantaneous value of heat transferred by the radiator at timestamp t, measured in $[J/s]$, $\Delta T_{radiator}$ is the temperature difference gained by the water circulating through the radiator and $water_{radiator}^{F-RATE}(t)$ is the water flow rate through the radiator measured in $[L/s]$. Furthermore, we assume that the water flow is directly proportional to the heat that must be transferred, leading to a constant temperature rise $\Delta T_{radiator}$ of the water due to heat absorption process:

$$\text{C3: } \Delta T_{radiator} = const \quad (11)$$

Cold Water Tank is a specially insulated tank containing water at low temperatures (i.e. 5 degrees Celsius) which is circulated through the radiator pipe system to cool the radiator. It acts like a buffer of cold water that can balance the server room cooling process intensity by varying the flow rate of the water circulating through the radiator pipe system. We denote with $TES_{cold}(t)$ the amount of thermal energy stored inside the tank at timestamp t, capacity measured in $[kWh]$ or $[J]$. By circulating the water through the radiator pipe system, the overall thermal energy inside the water tank increases (i.e. the water becomes warmer), under the constraint that the thermal energy accumulated in the tank cannot be above a maximum limit over which the efficiency of the cooling system dramatically decades.

$$TES_{cold}(t) = TES_{cold}(t_0) + \int_{t_0}^{t} water_{radiator}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{radiator}\, dx \quad (12)$$

$$\text{C4: } TES_{cold}(t) < TES_{MAX} \quad (13)$$

The third sub-system modeled is the heat pump. Its goal is to transfer the thermal energy between the *Cold Water Tank* and *Hot Water Tank* by using a refrigerant cycle based on a compressor to increase/decrease the water temperature while consuming electrical energy. The Cold Water Tank has a secondary water pipe circuit through which the accumulated thermal energy is transferred to the heat pump to be reused and to decrease water temperature as result. By varying the rate of water ($water_{evap}^{F-RATE}$) circulated to the evaporator of the heat pump, the overall thermal energy inside the tank is controlled under the constraint that the thermal energy inside the tank cannot decrease under a limit:

$$TES_{cold}(t) = TES_{cold}(t_0) - \int_{t_0}^{t} water_{evap}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{evap}\, dx \quad (14)$$

$$\text{C5: } TES_{cold}(t) > TES_{MIN} \quad (15)$$

The *Hot Water Tank* is similar to the Cold Water Tank but it stores thermal energy for nearby neighborhood heating as hot water at around 70-80 degrees Celsius. The water is heated by the heat pump's compressor with the temperature ΔT_{exch} and the amount of energy stored in the tank is controlled by the flow rate of water through heat pump's hot water circuit ($water_{exch}^{F-RATE}$). Thus the amount of thermal energy stored inside the tank can be computed as:

$$TES_{hot}(t) = TES_{hot}(t_0) + \int_{t_0}^{t} water_{exch}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{exch}\, dx \quad (16)$$

The heat pump has two coefficients of performance indicators (COP), defined for the water cooling and for heating processes:

$$COP_{Cooling} = \frac{\int_{t_0}^{t}\left[\left[water_{evap}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{evap}\right]\right] dx}{E_{Compresor}} \quad (17)$$

$$COP_{Heating} = \frac{\int_{t_0}^{t}\left[water_{exch}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{exch}\right] dx}{E_{Compresor}} + 1 \quad (18)$$

where the numerator from (17) represents the thermal energy removed from the *Cold Water Tank*, the numerator from (18) represents is the thermal energy delivered to the *Hot Water Tank* and $E_{Compresor}$ is the energy consumed for the process. Using above COP coefficients, the amount of thermal energy transferred by the heat pump can be computed as:

$$exchHEAT(t) = \left(COP_{heating} - 1\right) * \frac{absorbedHEAT(t)}{COP_{cooling}} \quad (19)$$

The heat absorbed and the heat produced can be related to the flow of water to and from the pump, with the constraint that the circuits to the *Cold Water Tank* and the one to the *Hot Water Tank* have a constant temperature difference:

$$\text{C6: } \Delta T_{evap} = const \text{ and } \Delta T_{exch} = const \quad (20)$$

From the Hot Water Tank, the thermal energy is pumped and circulated through the nearby neighborhood proportionally with the demand curve of the water flow rate through the water grid ($water_{facility}^{F-RATE}$):

$$TES_{hot}(t) = TES_{hot}(t_0) - \int_{t_0}^{t} water_{facility}^{F-RATE}(x) * water_{heat-capacity} * \Delta T_{facility}\, dx \quad (21)$$

Table 1: DC Thermal Flexibility Actions and Model Variables

DC Sub System	Thermal Flexibility Action	Variables
IT Servers	Change DC temperature set point. Precooling the servers room	$temp^{server}$
	Shift delay tolerant workload	$E_D^{Shifted}$
Cooling System (Radiators and Cold Water Tank)	Increase/decrease water flow rate between radiators & cold water tank	$water_{radiato}^{F-RAT}$
Heat Pump with Cold and Hot Water Tanks	Increase/decrease water flow rate between the heat pump and the cold & hot water tanks	$water_{evap}^{F-RAT}$ $water_{exch}^{F-RAT}$

Our goal is to schedule the DC operation to meet and follow the hot water demand curve of the nearby offices and buildings. To accomplish this, on top of the above presented mathematical model we have defined a set of thermal energy flexibility actions that allow the DC to adjust and shape its thermal response by changing the value of model control variables and constraints. The thermal flexibility actions and their mapping on model control variables are presented in Table 1.

3 RESULTS AND DISCUSION

3.1 Experimental Setup

For estimating the potential thermal energy flexibility of DCs in relation with the nearby neighbourhoods' heat demand we have conducted numerical simulation-based experiments considering the hardware systems' characteristics and operation of one server room of Engineering PSM DC (see Table 2) [15].

Table 2: Simulation Parameters Configuration

Parameter		Value
IT Servers	No. Servers	1400
	$Servers_{Total\ Power}$	750 KWh
	$threshold_{server}^{MAX}$	25 °C
	Thermal inertia servers	90000 [kJ]
	$temperature_{reference}^{server}$	20 °C
Radiators	$water_{heat-capacity}$	4.185 $\frac{j}{g*°C}$
	$\Delta T_{radiator}$	3 °C
	MAX $water_{radiator}^{F-RATE}$	60 $\frac{L}{s}$
Heat Pump	MAX $water_{evap}^{F-RATE}$	60 $\frac{L}{s}$
	ΔT_{evap}	3 °C
	$COP_{cooling}$	3.8
	$COP_{heating}$	2.3
	ΔT_{exch}	10 °C
	MAX $water_{exch}^{F-RATE}$	3 $\frac{L}{s}$
Cold Water Tank	Capacity	40000 L
	Temperature	5 °C
Hot Water Tank	Capacity	10000 L
	Temperature	80 °C
Neigh. Demand	$\Delta T_{facility}$	10 °C
	$water_{facility}^{F-RATE}$	3 $\frac{L}{s}$

We have constructed a DC flexibility simulation environment which is based on Lindo Software [17] for solving nonlinear programing optimizations of our mathematical model. The workload energy demand was synthetically generated from the IT power consumption logs of the DC considering normalized samples acquired every five minutes. The technical characteristics of the heat pump system for waste heat reuse considered in our experiments were taken from [9]. The thermal inertia of the server room was determined experimentally. The cooling units of the server rooms were turned off for specific time intervals letting the heat accumulate and we have measured the increase of water temperature on the circuit between the server room radiator and the cold water tank. The result presented in Fig. 3 shows a temperature increase ($\Delta T = 6℃$) in around 10 minutes while the servers where operating at the 750 kWh load. The heat exhausted by the servers in 10 minutes is about $\Delta Q = 450.000\ kJ$.

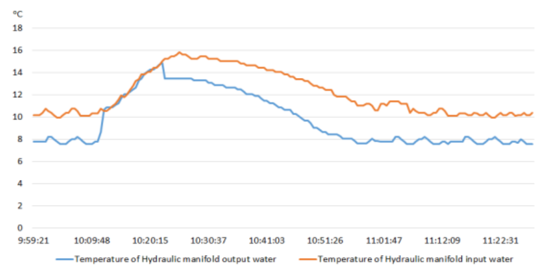

Figure 3. Water temperatures variation at the hydraulic manifold when cooling units are turned off (10:09'-10:20')

As it can be seen in Fig. 3 the temperature of hydraulic manifold output water (to the server room) and the temperature of hydraulic manifold input water (from the server room) have increased with the same lapse rate showing the high DC thermal inertia. At the end of the test, the two temperatures have reached to a steady state condition: 8°C for the hydraulic manifold output water and 10°C for hydraulic manifold input water. The transient state of water to server room finished at about 11:01', while the transient state of water from server room at about 11:12'.

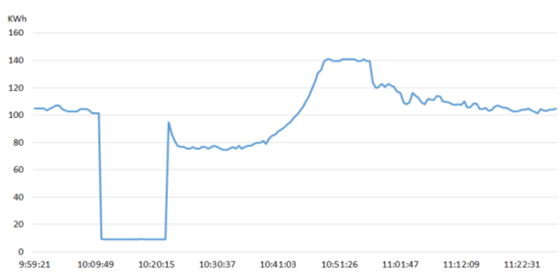

Figure 4. Power consumption variation of cooling unit

The energy consumption impact of cooling units being turned off is shown in Fig. 4. As it can be seen the transient state of power consumption finishes at about 11:22' thus lasting more than the transient's state of the thermal inertia.

3.2 DC Thermal Flexibility Evaluation

In this section, we conduct a set of simulation experiments to determine the degree in which the DC may adapt its thermal energy profile by proactively controlling its operation to meet a specific hot water demand. We have considered a 4-hour time window in which the heat demand from the nearby neighborhood varies in terms of hot water flow rate (i.e. on the circuit between hot water tank and thermal grid) and we have evaluated the impact of various thermal flexibility actions on adapting the DC thermal profile to follow the demand.

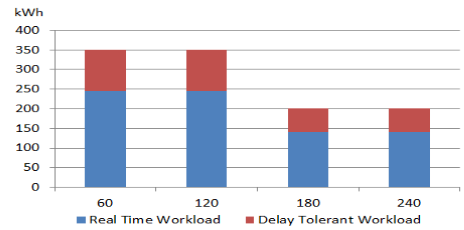

Figure 5: DC initial non-optimal workload scheduling

The initial situation is shown in Fig. 6, where due to non-optimal scheduling of delay tolerant workload execution (see Fig. 5) the servers' heat converted to hot water generates a profile that does not match the neighborhood demand.

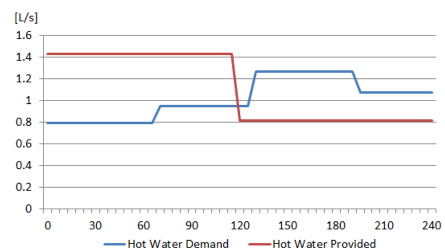

Figure 6: DC waste heat reuse initial situation

In the first evaluation case, we will determine the impact of delay tolerant workload shifting onto the DC thermal energy response in terms of hot water generation. The delay tolerant workload is shifted from time interval 0-120 minutes, when the DC thermal energy response exceeds the actual demand, to increase the server loads and usage in the time interval 180-240 minutes. Thus, more heat was generated in the server room and as result increasing the hot water generation profile (see Fig. 7).

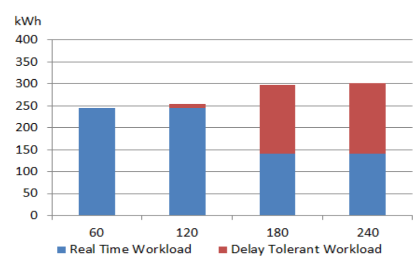

Figure 7: Optimal delay tolerant workload distribution

The extra heat generated by IT server is absorbed by the heat pump, and transferred to the hot water tank from where it is fed to the thermal energy grid of the nearby neighborhood (see Fig. 8). The two profiles show an aggregated match (i.e. curve following) of 86% over the time period, varying between 75-98%.

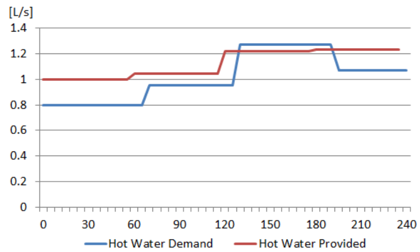

Figure 8: DC adapted profile due to workload time shifting

In the second evaluation case, we will determine the impact of server room pre/post cooling on the DC thermal energy profile. We will evaluate the thermal potential of the server room by changing the temperature set point and as a result letting the heat to accumulate until the temperature reaches the new allowed maximum (Fig. 9).

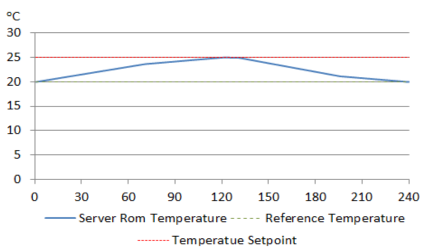

Figure 9: Temperature variation in the server room

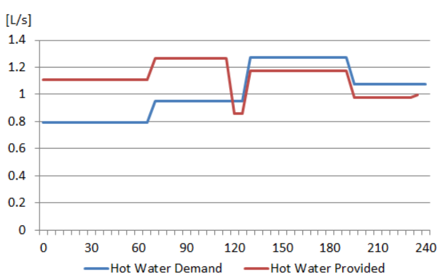

Figure 10: DC adapted thermal energy profile due to server room heat accumulation

Fig. 10 presents the adjusted DC thermal energy profile in terms of hot water generation in comparison with the demanded profile. As it can be seen, once the temperature in the server room reaches the new maximum set point (i.e. after minute 120') the DC thermal energy profile follows closely the requested profile with a degree of accuracy between 75% and 95%. Nevertheless, before reaching the set point the matching degree is quite low (i.e. between 35% and 45%) and this is due to the server room high thermal inertia. Thus, we obtain an aggregated matching degree between the adjusted thermal energy profile and the demanded profile of around 60% showing that the pre-cooling technique alone provides small thermal flexibility.

4 CONCLUSIONS

In summary, we have proposed a model for estimating and controlling the thermal energy flexibility of major sub-systems inside a DC and associated thermal inertia with the goal of optimizing the DC operation for reusing the waste heat in nearby neighborhoods. Simulation based experiments have been conducted using the PSM DC hardware configuration of a server room with the goal of determining the impact of delay tolerant workload relocation and pre / post cooling flexibility actions on the thermal energy profile adaptation. The results show the thermal flexibility potential of providing hot water for 120-360 homes at different levels of capacity usage considering the hourly household daily water consumption reported in literature [18].

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