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Research Paper

Energy and exergy assessment in a perimeter cooled data center: The value of second law efficiency

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HIGHLIGHTS

- This work focusses on Data Center power consumption in the early years of operation.
- Maximum efficiency appears with all racks active in contained aisles.
- The Coefficient of Performance proves insufficient to address irreversibilities.
- The use of the second law efficiency in these Thermodynamic analyses is suggested.

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ABSTRACT

The Data Center Industry remains in steady growth with worldwide energy consumptions in the order of hundreds of TWh, and yearly growth rates higher than the growth of global electricity consumption. Electronic cooling represents an important part of a Data Center's energy demand, thus efficient thermal management practices have become critical in recent years. The Coefficient of Performance (COP) appears as the most commonly used metric when assessing the energy efficiency of a data center, although sometimes it proves insufficient to address irreversibilities in a system. In this work, the information provided by the COP is complemented with a second law efficiency analysis as a way to measure inefficiencies. The software Engineering Equation Solver (EES) is utilized to apply energy and exergy balances in each individual component and the global refrigeration system, simultaneously. The study considers variations in the number of active racks and the increase in hot air recirculation within the room, assuming that the approach closely represents the first years of operation. Maximum efficiencies are achieved when all racks are active and when aisle air containment prevents recirculation. Fewer active racks increase the exergy destruction, due to excessive power usage in a low demand situation. The chiller accounts for the highest power consumption with high density racks; whereas for low density racks, the cooling tower consumes most of the power. The second law analysis justifies the chiller power consumption based on its cooling requirements, and elucidates oversizing in the cooling tower power consumption. The COP increases with hot air recirculation, which could mislead the design towards inefficient decisions; the second law efficiency shows an inverse relationship with the recirculation, thus properly capturing the inefficiencies and better guiding the design. Appropriate energy efficiency analyses should consider both thermodynamic approaches (first and second law) to ensure proper use of working fluids cooling potential and maximize the system energy efficiency.

1. Introduction

The data center industry dramatically grew in the past decade, leading to a significant increase in its energy consumption. In 2010 data centers consumed approximately 1.3% of the total energy around the globe [1]. For this reason, many data providers have adopted new strategies, standards and/or certifications (ISO, Uptime Institute, ASHRAE) to optimize the energy use and enhance their corporate image.

Nearly 40% of a data center total energy consumption corresponds to the cooling system, rendering thermal efficiency and

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energy demand reduction as ways to reduce energy costs [2]. The challenge arises as best practices suggest operating around 22 °C server temperature [3], while new technologies keep increasing their heat dissipation requirements (tens of kW per rack). In this context, thermodynamic models appear as a suitable modeling technique to study the refrigeration system performance in data centers [4–10].

Demetriou et al. [5], through a thermodynamic simulation, evaluated the cooling energy consumption in a data center throughout an entire year. Monitoring variations in the Power Usage Effectiveness (PUE), they showed that adopting ASHRAE environmental guidelines increases the energy efficiency. Subsequent efforts addressed the effect of cooling air properties [6,7], the implementation of an economizer system [10], and the impact of aisle containment techniques [11] on the cooling efficiency. A literature review regarding data center thermal management can be found in the work of Fulpagare and Bhargav [12].

Most of the studies found in the literature focus on data center cooling energy consumption, which disregards irreversibilities and the working fluid cooling potential. This led current research toward second law analyses and the concept of exergy destruction as a way to identify and quantify inefficiencies.

Second law approaches can characterize the energy performance of different systems; for instance, Tu et al. [13] used the concept of exergy destruction to investigate the performance of ventilation systems in humid climates. Mohammadi et al. [14] employed a second law analysis to investigate the effect of working fluid properties on the performance of a renewable energy system, considering cooling, heating and power cycles simultaneously. They identified the conditions that lead to an increase in exergy destruction, and the components responsible for the highest exergy destruction within the system. Fang et al. [15] evaluated the operation performance of a Heating, Ventilation and Air Conditioning (HVAC) system under different control strategies, suggesting that this approach allows identifying the conditions that improve the system performance. Similarly, Singh and Das [16] developed an optimization model that identifies the operating parameters that minimize the exergy destruction within a cooling system.

The recirculation of hot air within the room increases the system irreversibilities, thus decreasing the cooling performance. Shah et al. [17] found that most of the exergy loss occurs in the vicinity of the racks where air mixing takes place. An estimation of the exergy destruction distribution within a perimeter cooled data center's air side, individualizing heat conduction and pressure drop effects, was presented by Silva-Llanca et al. [18]. Bhalerao et al. [19] developed a model that computes the airspace exergy destruction significantly faster than conventional CFD models, using a lumped parameter approach for a range of Computer Room Air Conditioner (CRAC) supply temperatures. Finally, Khalaj et al. [20] identified the energy wasted in a data center cooling system considering different locations within Australia. They aimed to quantify the energy wasted under different environmental conditions, emphasizing that a first law metric, namely PUE, cannot estimate the magnitude of wasteful practices in the system.

2. Motivation and goals

Most authors have used the second law of Thermodynamics to understand cooling energy consumption in data centers paying attention to the minimization of the exergy destruction due to hot/cold air mixing within the room and due to variations in local weather conditions. Insufficient or non-existing studies are conducted for data centers during the early years of operation, when few racks actually operate; indeed, most of the studies focus their attention in refrigeration systems for which all racks are assumed to be active.

This work intends to understand data center cooling energy consumption during the first years of operation, where oversized refrigeration systems work for only a few active racks, thus increasing operating costs. The system thermal efficiency is explored using both a first and second law approach.

The specific goals of the paper list as follows:

- Assessing the early years of operation effect upon the system by parametrically varying the number of racks and their heat dissipation
- Quantifying wasted energy (irreversibilities) using a second law approach
- Identifying the larger energy consuming and wasting components under the proposed conditions
- Studying the effect of room-air recirculation over the system thermal efficiency.

3. System description and modeling

3.1. Refrigeration system

In a typical data center room (Fig. 1), the cooling air that flows from the Computer Room Air Handler (CRAH) into the raised floor passes through perforated tiles towards the cold aisle, where it diverges and enters each rack. After absorbing heat from the rack, the warm air upwardly advects where it exhausts towards the CRAH, therein completing the cycle. Hot-to-cold aisle recirculation appears in most legacy data centers, where warm air "short-circuits" and enters the cold aisle, prematurely heating the cooling flow. This phenomenon emerges as one of the main sources of irreversibilities that manifests as exergy (available work) destruction in data center cooling, leading to energy—and ultimately—economic losses.

Fig. 2 depicts the proposed refrigeration system. Each component was selected based on real refrigeration systems requirements found in conventional data centers. Several CRAH units are employed to face variations in heat dissipation within the room.

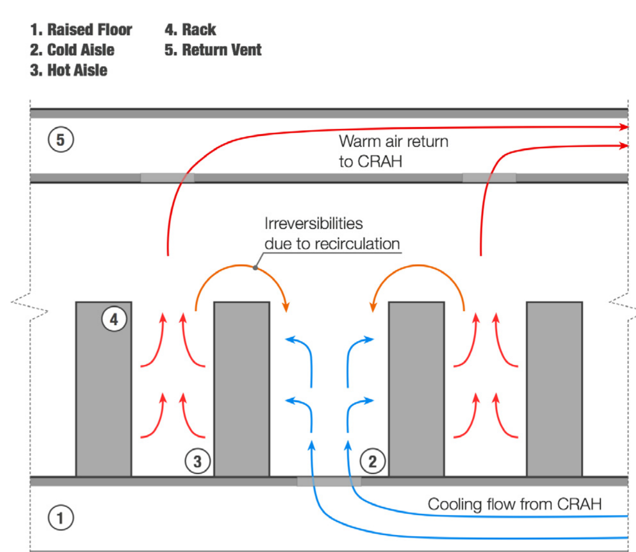

Fig. 1. Data center room schematic.

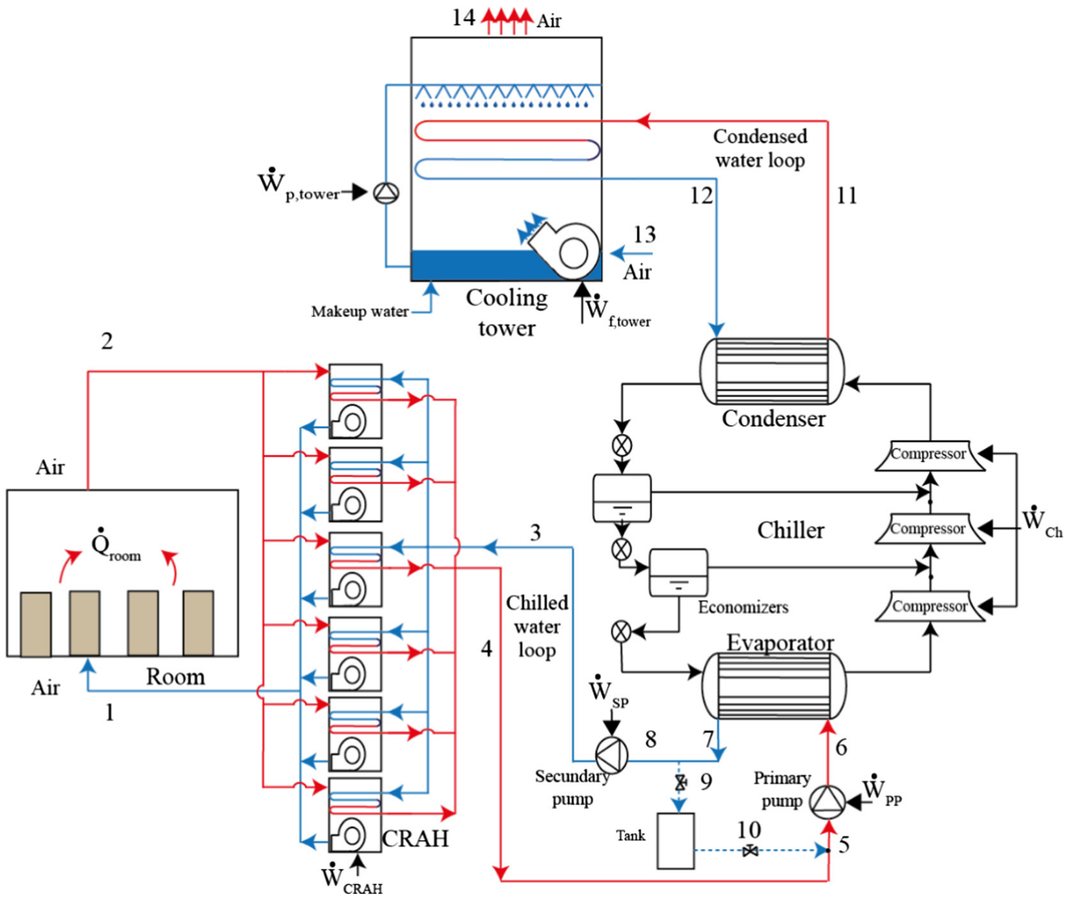

Fig. 2. Refrigeration system schematic.

The system goal is to provide cold air to a room with capacity for 42 racks, which can dissipate 2, 5, 10 and 15 kW per rack ($\dot{Q}_{rack}$). This rack load density range considers legacy data centers, designed to have loads of about 2 kW per rack [21]; whereas 15 kW considers the maximum cooling capacity allowed with the selected CRAH units when the 42 racks are active.

Each rack is provided with 0.3 m^3/s of air at T_1 = 20 °C and 50% relative humidity. Thus, $\dot{m}_1 = \dot{m}_2$ are functions of the number of active racks.

Cold air is obtained using one to six CRAH units (Uniflair, Leonardo Evolution TDCV 4300) in which heat is transferred to water at T_7 = 6 °C. Therefore, if one unit is not able to supply the required cold air, another unit activates. Each unit has a maximum water flow rate of 0.00473 m^3/s, maximum air flow rate of $\dot{\forall}_{air,CRAH,\max}$ = 8.33 m^3/s, and maximum heat flow dissipation capacity of $\dot{Q}_{CRAH,\max}$ = 118.6 kW. Water at 6 °C is provided by a chiller (Trane CVHE, refrigerant 123) with two economizers and three compression stages. This chiller has a nominal capacity of $\dot{Q}_n$ = 1055.06 kW and a minimum water flow rate requirement in the evaporator of 0.035 m^3/s. To control water flow rate going to the CRAH, a water tank is implemented.

A counter flow close circuit-cooling tower (Marley MCF7055) is used to remove heat from the chiller. Air and a water spray system are employed to provide cold water at T_{12} = 25 °C to the condenser. Hence, two centrifugal fan of nominal power $\dot{W}_{f,tower}$ = 15 kW each and one pump of $\dot{W}_{p,tower}$ = 5 kW are implemented.

The refrigeration system has two pumps to supply water to the chiller (primary pump, Grundfos NK 100-250/266, 50 Hz) and CRAH units (secondary pump, Grundfos NK 100-315/334, 50 Hz). Primary pump is employed to maintain the previously mentioned 0.035 m^3/s, whereas secondary pump is used to increase water pressure up to P_3 = 3 bar. As suggested by the manufacturer, 80.2% isentropic efficiency is assumed for both pumps. Table 1

Table 1
Refrigeration system operating parameters.

Parameter	Value
Number of racks	1–42
Rack load density (kW)	2; 5; 10; 15
Air flow per rack (m^3/s)	0.3
Processor temperature (°C)	65
CRAH, heat flow dissipation capacity (kW)	118.6
CRAH, fan nominal power consumption (kW)	5.44
CRAH, water temperature variation, T_4-T_3 (°C)	6
CRAH, power consumption associated to controllers and monitoring systems (kW)	3.26
CRAH, water line pressure drop (bar)	0.758
CRAH, maximum air flow rate (m^3/s)	8.33
CRAH, maximum water flow rate (m^3/s)	0.00473
Chilled water loop pressure, P_3 (bar)	3
Chiller, minimum water flow rate through the evaporator (m^3/s)	0.035
Chiller, cooling capacity (ton)	300
Chiller, minimum water flow rate through the condenser (m^3/s)	0.056
Cooling tower, fan power consumption (kW)	15
Cooling tower, fan nominal speed (RPM)	462
Cooling tower, fan diameter (mm)	660
Cooling tower, pump power consumption (kW)	5
Primary pump nominal power consumption (kW)	15
Secondary pump nominal power consumption (kW)	30
Isentropic efficiency of primary and secondary pumps (%)	80.2

summarizes the operating parameters used in the proposed refrigeration system.

3.2. Thermodynamic model

The thermodynamic analysis is carried out using the Engineering Equation Solver (EES) software [22]. This software is widely used by the scientific community dealing with macroscopic thermodynamic related simulations, allows the solution of algebraic equations governing the cycles, and also easily communicates with a thermos-physical properties database based on REFPROP-NIST. Hence, it allows to perform both energy and exergy efficiency analysis within the refrigeration system.

The energy analysis is performed by calculating the refrigeration system COP. The COP relates the dissipated heat to the power required during the dissipation process. Thus, the COP for the entire systems is calculated as

$$COP_{system} = \frac{\dot{Q}_{out}}{\dot{W}_{system}} = \frac{\dot{m}_2(h_2 - h_1)}{\dot{W}_{CRAH} + \dot{W}_{chiller} + \dot{W}_{tower} + \dot{W}_{pp} + \dot{W}_{sp}} \tag{1}$$

Properties at state 1 are assumed to be constant, however, if hot air recirculation increases T_2, exceeding the CRAH cooling capacity, h_1 is re-calculated using an energy balance in the CRAH.

$$h_1 = h_2 - \frac{n_{CRAH}\dot{Q}_{CRAH,\max}}{\dot{m}_2} \tag{2}$$

where n_{CRAH} is the number of the required CRAH units. In this equation, h_2 is calculated by an energy balance in the racks considering the total heat flow dissipated and the effect of hot air recirculation, as suggested in [23]. This implies that state 2 is calculated assuming the same thermodynamic properties at the rack exit. Thus, if hot air recirculation takes place, air exiting a rack re-enters it and mixes with the incoming cooling air. Then,

$$h_2 = h_1|_{20\,^\circ C} + \frac{n_{racks}\dot{Q}_{rack}}{(1-\gamma)\dot{m}_1} \tag{3}$$

where γ is the percentage of hot air that exits and re-enters the rack. This approximation neglects irreversibilities within the room due to rack size and configuration.

In Eq. (1) CRAH power consumption is calculated considering controllers, monitoring system and fan, as follows

$$\dot{W}_{CRAH} = \dot{W}_{c+m} + \dot{W}_{fan,CRAH} \tag{4}$$

where $\dot{W}_{c+m}$ is the power associated to controllers and monitoring (3.26 kW), and $\dot{W}_{fan,CRAH}$ is fan power given by fan law

$$\dot{W}_{fan,CRAH} = \dot{W}_{fan,CRAH,n}\left(\frac{\dot{\forall}_{air,CRAH}}{\dot{\forall}_{air,CRAH,\max}}\right)^3 \tag{5}$$

where $\dot{\forall}_{air,CRAH}$ is the air flow rate and $\dot{W}_{fan,CRAH,n}$ the CRAH nominal power consumption (5.44 kW).

As suggested by Meakins [24], and according to the manufacturer, for water at 6 °C, chiller power consumption can be described by the following equation

$$\dot{W}_{chiller} = 18.4257 + 31.6728(load) + 0.822349(load)^2 + 131.048(load)^3 \quad [\text{kW}] \tag{6}$$

In the above equation, *load* is given by

$$load = \frac{\dot{Q}_{eva}}{\dot{Q}_n} \tag{7}$$

where $\dot{Q}_{eva}$ is the heat flow dissipated by the evaporator, which is calculated by an energy balance.

The tower is assumed to operate at maximum capacity, therefore, $\dot{W}_{tower}$ = 35 kW. Primary pump power consumption ($\dot{W}_{pp}$) is calculated considering a nominal power of 15 kW, whereas secondary pump power consumption ($\dot{W}_{sp}$) considers 30 kW.

The exergy analysis is conducted by calculating the refrigeration system second law efficiency (or exergetic efficiency) as follows

$$\eta_{II,system} = 1 - \frac{\dot{\chi}_{dest,system}}{\dot{\chi}_{in}} = 1 - \frac{\dot{\chi}_{dest,system}}{\dot{W}_{system} + n_{racks}\dot{Q}_{rack}\left(1 - \frac{T_o}{T_p}\right)} \tag{8}$$

where $\dot{\chi}_{in}$ is the total exergy added to the refrigeration system, $\dot{\chi}_{dest,system}$ the refrigeration system exergy destruction, n_{racks} the number of active racks, and T_p the processor temperature (assumed to be at 65 °C). In the above equation the sub-index "o" represents the dead state quantities for which the effect of data center location is not considered, therefore, ambient temperature, pressure, and relative humidity are assumed to be constant (20 °C, 1 atm, and 60%, respectively).

The exergy destruction in Eq. (8) is calculated in each of the components to identify sources of irreversibility within the system. Thus,

$$\dot{\chi}_{dest,system} = \dot{\chi}_{dest,room} + \dot{\chi}_{dest,CRAH} + \dot{\chi}_{dest,chiller} + \dot{\chi}_{dest,tower} + \dot{\chi}_{dest,tank,loop} \tag{9}$$

An exergy balance is applied to each component as follows

$$\dot{\chi}_{dest} = \dot{W} + \dot{\chi}_{heat} + \sum \dot{m}_{in}\psi_{in} - \sum \dot{m}_{out}\psi_{out} \tag{10}$$

where $\dot{\chi}_{heat}$ is the exergy associated to heat transfer and ψ is the flow exergy, which is a function of the enthalpy h and entropy s such that

$$\dot{\chi}_{heat} = \dot{Q}_k\left(1 - \frac{T_o}{T_k}\right) \tag{11}$$

$$\psi = (h - h_o) - T_o(s - s_o) \tag{12}$$

Nonetheless, if the working fluid is wet air, the flow exergy is calculated considering both water vapor and air as ideal gases [25–27]. Thus,

$$\begin{aligned}\psi_{w,a} = {} & \left(c_{p_{air}} + \omega c_{p_{vapr}}\right)\left(T - T_o - T_o\ln\left(\frac{T}{T_o}\right)\right) \\ & + (1+1.608\omega)R_{air}T_o\ln\left(\frac{P}{P_o}\right) \\ & + R_{air}T_o\left((1+1.608\omega)\ln\left(\frac{1+1.608\omega_o}{1+1.608\omega}\right) + 1.608\omega\ln\left(\frac{\omega}{\omega_o}\right)\right)\end{aligned} \tag{13}$$

where c_p is specific heat, P is pressure, R_{air} is gas constant and ω is the ratio of vapor to dry air mass.

Thus, the exergy destruction in each component yields the following system of equations:

$$Room: \quad \dot{\chi}_{dest,room} = \dot{m}_1(\psi_1 - \psi_2) + n_{racks}\dot{Q}_{rack}\left(1 - \frac{T_o}{T_p}\right) \tag{14}$$

$$CRAH\ units: \quad \dot{\chi}_{dest,CRAH} = \dot{W}_{CRAH} + \dot{m}_1(\psi_2 - \psi_1) + \dot{m}_3(\psi_3 - \psi_4) \tag{15}$$

$$Chiller: \quad \dot{\chi}_{dest,chiller} = \dot{W}_{chiller} + \dot{m}_6(\psi_6 - \psi_7) + \dot{m}_{11}(\psi_{12} - \psi_{11}) \tag{16}$$

$$Cooling\ tower: \quad \dot{\chi}_{dest,tower} = \dot{W}_{tower} + \dot{m}_{11}(\psi_{11} - \psi_{12}) + \dot{m}_{13}(\psi_{13} - \psi_{14}) \tag{17}$$

Table 2
Thermodynamic states.

State	Fluid state	Temperature (°C)	Pressure (bar)	Relative humidity (%)
0	Dead state	20	1	60
1	Moist air	20	1	50
7	Subcooled water	6	1	–
8	Subcooled water	6	1	–
9	Subcooled water	6	1	–
10	Subcooled water	6	1	–
12	Saturated liquid	25	–	–
13	Moist air	20	1	60

$$\text{Tank loop}: \quad \dot{\chi}_{dest,\text{tank},loop} = \dot{m}_4(\psi_4 - \psi_8) + \dot{m}_5(\psi_7 - \psi_5) \tag{18}$$

The given thermodynamic properties in Eqs. (14)–(18) are shown in Table 2, whereas unknown properties are calculated employing an energy balance in the corresponding equipment and the operating parameters showed in Table 1. The flow exergy at state 14 is calculated using Eq. (13), for which an energy balance is first applied in the cooling tower assuming that saturated air exits the tower. Here, the percentage of water loss due to evaporation (E) and that must be supplied (makeup water) to the tower is calculated as

$$E = b\frac{\dot{m}_{11}(T_{11} - T_{12})}{\dot{m}_{p,tower}} \tag{19}$$

where b is a loss factor defined as function of the ambient temperature (T_o) and relative humidity ($r_{h,o}$) as follows

$$b = (113 - 8.417r_{h,o} + 1.6147T_o) \cdot 10^{-5} \tag{20}$$

Eq. (19) was validated in [28] considering the recommendations given by companies in the field of evaporative cooling devices such as Baltimore Aircoil and Graham Manufacturing. As suggested in [28], the water mass flow through the pump ($\dot{m}_{p,tower}$) is considered to be twice the air mass flow rate that enters the tower. Finally, water temperature variations through the cooling tower pump are neglected.

4. Number of racks and air mixing effect upon thermal efficiency

4.1. The effect of the number of active racks

In the proposed system, each rack requires an air flow rate of 0.3 m^3/s at 20 °C and dissipates the same amount of heat in four different scenarios: 2, 5, 10 and 15 kW. When a cooling unit (CRAH) maximizes either its cooling capacity or its supply air flow, another unit activates; the system can add up to six CRAH units. In this part of the analysis, the system neglects recirculation, as found in data centers with contained aisles.

Fig. 3 shows the CRAH power consumption as function of the number of racks, where the discontinuities observed in the data

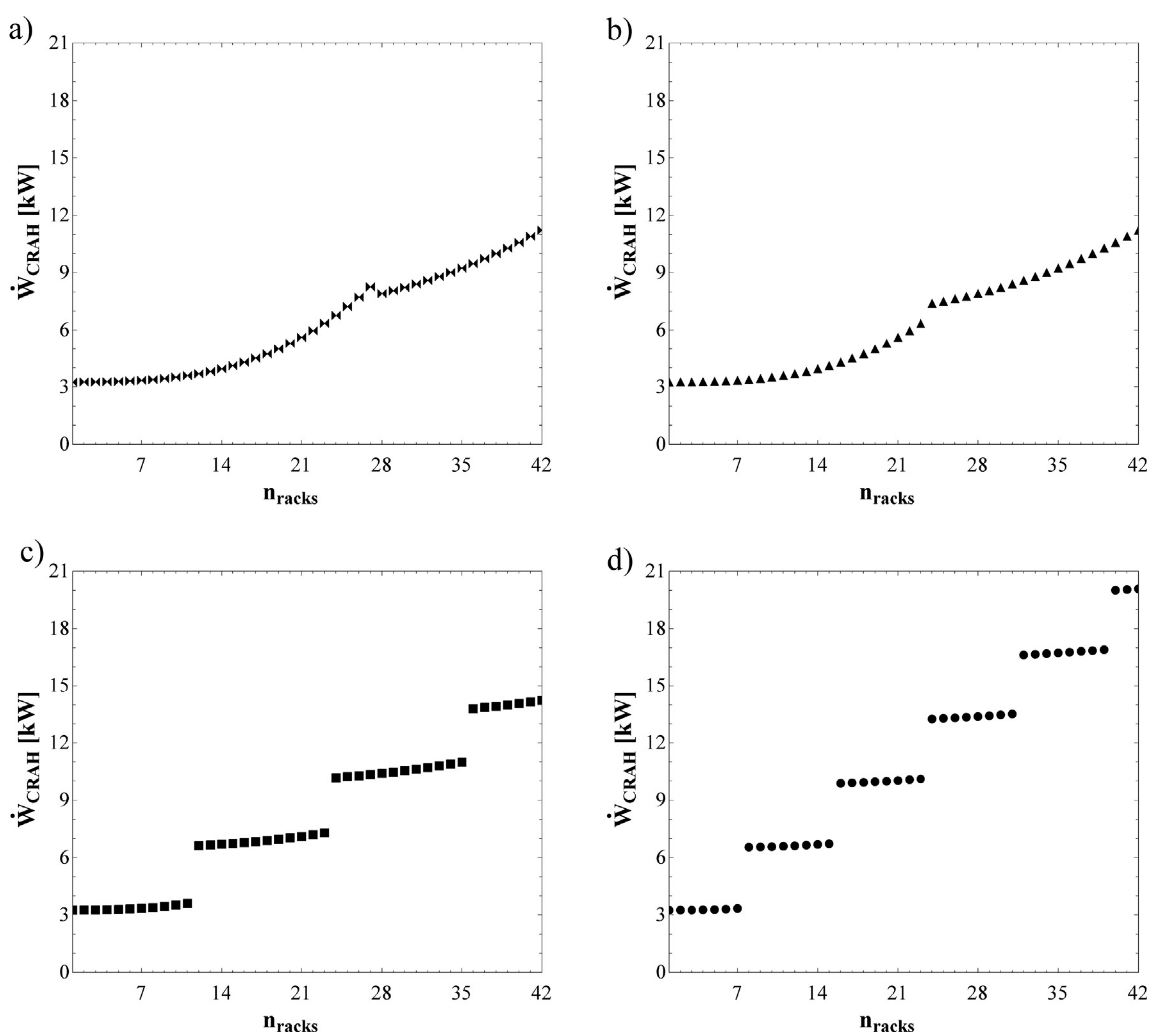

Fig. 3. CRAH power consumption as function of the number of active racks for (a) 2 kW, (b) 5 kW, (c) 10 kW, and (d) 15 kW.

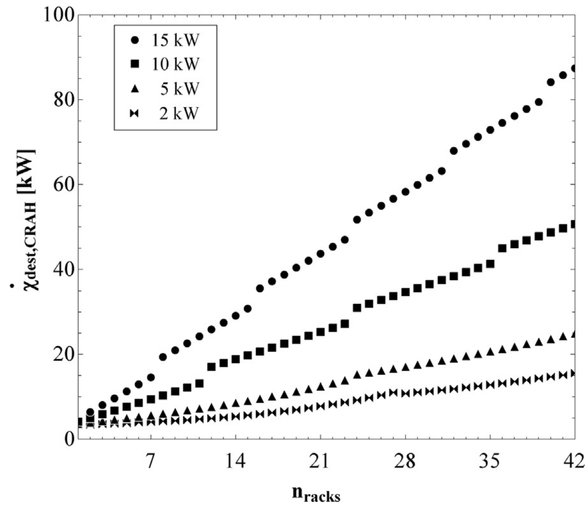

Fig. 4. Exergy destruction within CRAH units as function of the number of active racks.

represent the activation of an idle CRAH unit. For the lower load densities (Fig. 3a and b), two units are sufficient to meet the cooling demand, whereas the number increases for the highest loads (Fig. 3c and d). High load density systems (10 and 15 kW) require units with higher heat dissipation capabilities when operating at full capacity (42 racks). The power consumed by each unit decreases, however, the overall consumption increases. This is because more heat needs to be dissipated, whereas the air flow rate must remain constant. In other words, the flow rate at each CRAH unit decreases as load density increases, as well as its cooling potential.

Fig. 4 shows the CRAH exergy destruction as function of the number of racks. The exergy destruction decreases for low density racks, since power consumption decreases and lower air temperature differences within the room are achieved. On the other hand, the chiller destroys less exergy when the load density increases (Fig. 5). For 15 kW racks, the chiller finds its optimum operation when $n_{racks} = 21$, as its exergy destruction minimizes; similarly, for 10 kW dissipation, the optimum number of racks is 31. For 2 and 5 kW, due to the low amount of heat that needs to be dissipated, the chiller becomes more efficient when all racks are active.

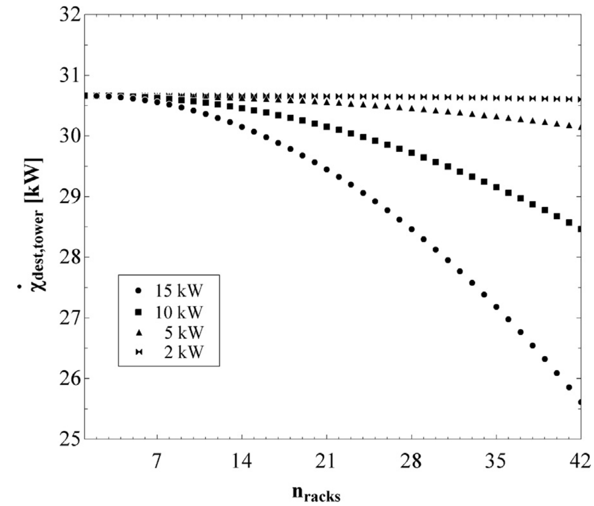

Fig. 6. Cooling tower exergy destruction as function of the number of active racks.

The exergy destruction in the cooling tower decreases if the number of active racks increases (Fig. 6), except for 2 kW where it is essentially constant. This occurs since the cooling tower is assumed to always operate at maximum capacity. Therefore, as the room requires higher dissipation, the cooling tower functions closer to its overall capacity. As observed, exergy destruction behaves differently among the various refrigeration system components.

4.2. The effect of hot/cold air mixing

As recommended by the ASHRAE [3], the rack inlet temperature must remain below 27 °C to prevent the internal electrical components from overheating; hence, air recirculation plays a detrimental role in this aspect. This section intends to investigate its effect on system efficiency (where all racks are considered in operation).

Hot air recirculation significantly increases the exergy destruction in the room, especially for high load densities (Fig. 7). A first law analysis shows that the rack inlet temperature increases monotonically with hot air recirculation (Fig. 8a). For the highest rack load (15 kW), the rack inlet temperature exceeds the ASHRAE operation limit when $\gamma > 28\%$. Furthermore, when $\gamma > 26\%$, the

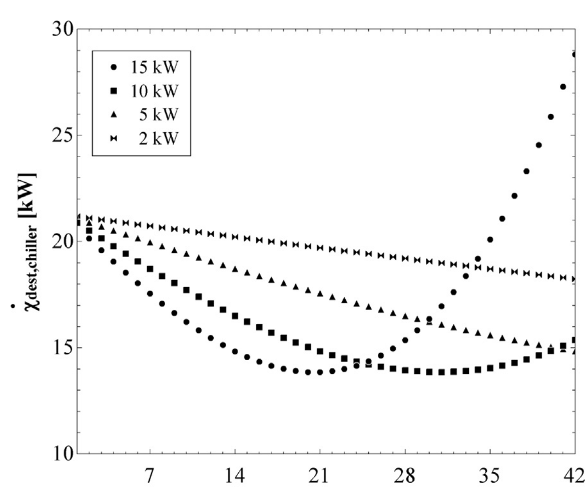

Fig. 5. Chiller exergy destruction as function of the number of active racks.

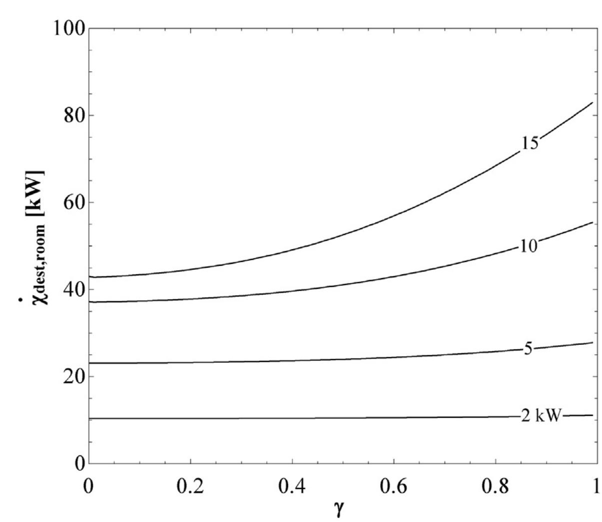

Fig. 7. Exergy destruction within the room as function of hot air recirculation.

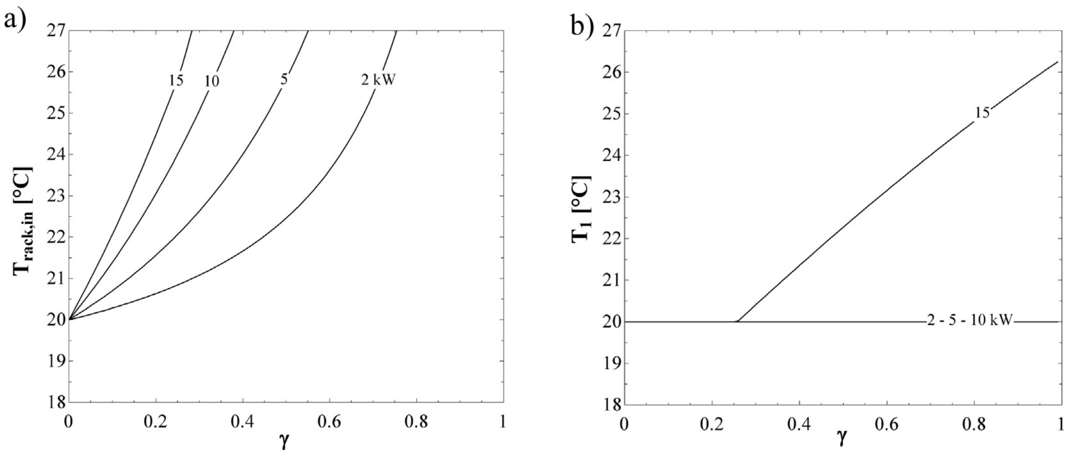

Fig. 8. Cold air temperature variation as function of hot air recirculation at (a) rack and (b) room inlet.

CRAH units reach their maximum cooling capacity and no longer supply air at 20 °C to the room, as shown in Fig. 8b.

When recirculation appears, more CRAH units must be activated to meet the cooling requirements in the intermediate scenarios (5 and 10 kW), increasing the energy consumption and exergy destruction (Fig. 9a and b, respectively). As previously shown (Fig. 3), in the absence of recirculation ($\gamma = 0$), two units are required for 2 and 5 kW, whereas four and six units are required for 10 and 15 kW, respectively. As the recirculation increases (Fig. 9a), one and two extra units must be activated for 5 kW and 10 kW load density racks, respectively. In the lowest dissipation case (2 kW), no additional units are required to satisfy the cooling demand. For 15 kW all racks and CRAH units are already in operation, and the energy consumption remains constant (Fig. 9a), but since the heat dissipation requirement surpasses the unit cooling capacity, the increase in exergy destruction is more significant (Fig. 9b).

The exergy destruction evidences an oversized chiller, especially for low density racks (Fig. 10). At 2 and 5 kW dissipation rates, the exergy destruction decreases with γ and the chiller runs more efficiently. Fig. 10 shows that $\dot{\chi}_{dest,chiller}$ is higher for 2 kW compared to 5 kW, because the refrigerant underuses its cooling potential due to the low amount of heat flow dissipated at 2 kW. For the higher dissipation cases, $\dot{\chi}_{dest,chiller}$ behaves in two ways: at 10 kW it increases monotonically, and for 15 kW it flattens when $\gamma > 26\%$, since the CRAH units have reached their maximum cooling capacity and no longer transfer heat to the chiller.

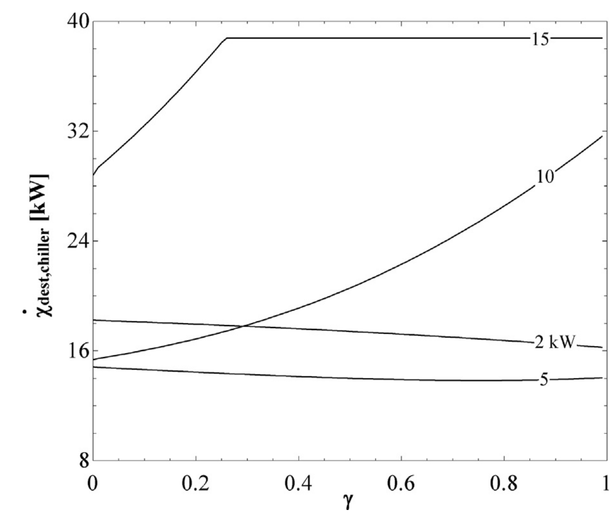

Fig. 10. Chiller exergy destruction as function of hot air recirculation.

The data in Fig. 11 demonstrates that the cooling tower decreases its exergy destruction proportional to the rack heat dissipation and air recirculation. For 15 kW racks, when the CRAH units achieve their heat flow dissipation limit, $\dot{\chi}_{dest,tower}$ becomes constant. The global behavior of the data suggests an oversized tower, which generally provides cooling in excess.

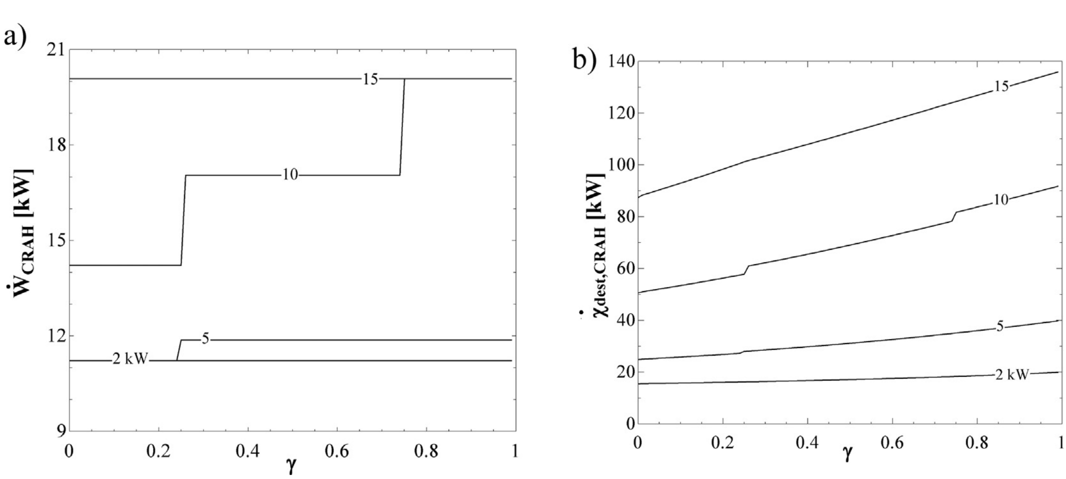

Fig. 9. CRAH (a) power consumption and (b) exergy destruction as function of hot air recirculation.

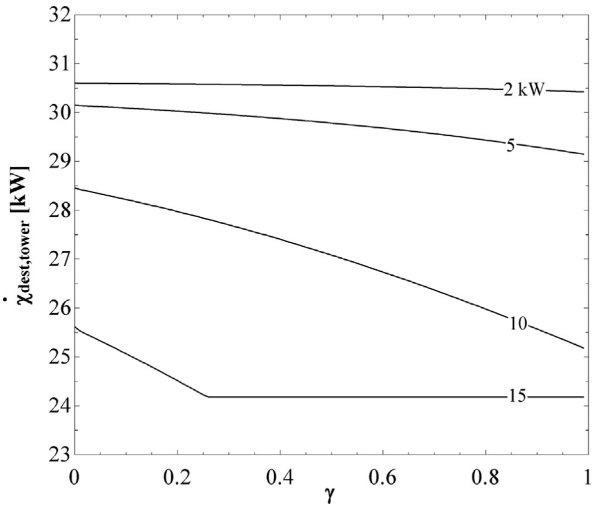

Fig. 11. Cooling tower exergy destruction as function of hot air recirculation.

5. Coefficient of performance versus second law efficiency

In some instances, first and second law efficiencies may bring the design reasoning into diverging paths. This section compares both approaches and presents a case where an exergy based efficiency definition better represents the refrigeration system behavior.

The system achieves optimum performance at its full capacity; i.e., all racks active and dissipating 15 kW. Fig. 12 compares the overall COP (Fig. 12a) and the second law efficiency (Fig. 12b). When all the components are integrated, both the COP and the exergetic efficiency increase as the number of active racks increases, except at 2 kW, where the exergetic efficiency decreases when the number of racks surpasses 12.

The data evidence an oversized refrigeration system for the lower load densities; in other words, for a larger number of racks, the ratio between exergy destroyed and supplied increases. According to first law, 15 kW racks maximize their efficiency with 39 racks operating, whereas the exergetic efficiency is optimized with 42 racks, although the difference between both optimum efficiencies remains below 2%.

The COP presents a counter-intuitive, and ultimately misleading, relationship with hot air recirculation (γ). Fig. 13 compares the COP and $\eta_{II,system}$ as they vary with the air recirculation, where symbols indicate the critical recirculation for which the rack inlet temperature becomes greater than 27 °C (ASHRAE limit). The COP indicates that the system improves when the recirculation increases (Fig. 13a), which seems contradictory. On the other hand, $\eta_{II,system}$ increases as the recirculation decreases, maximizing when $\gamma = 0$. In this case, the exergy destruction maximizes with the highest dissipation (except for the tower), and yet—interest

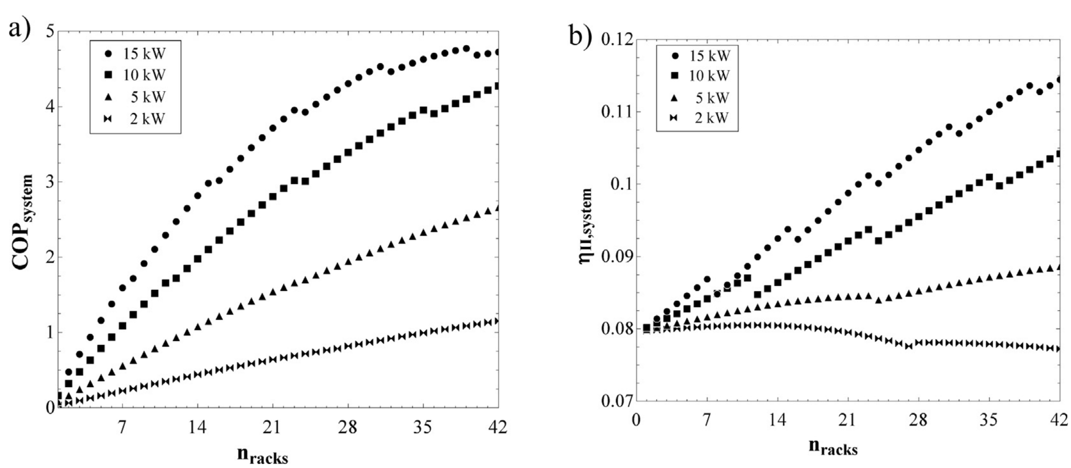

Fig. 12. Refrigeration system (a) Coefficient of Performance (COP) and (b) exergetic efficiency as function of the number of active racks.

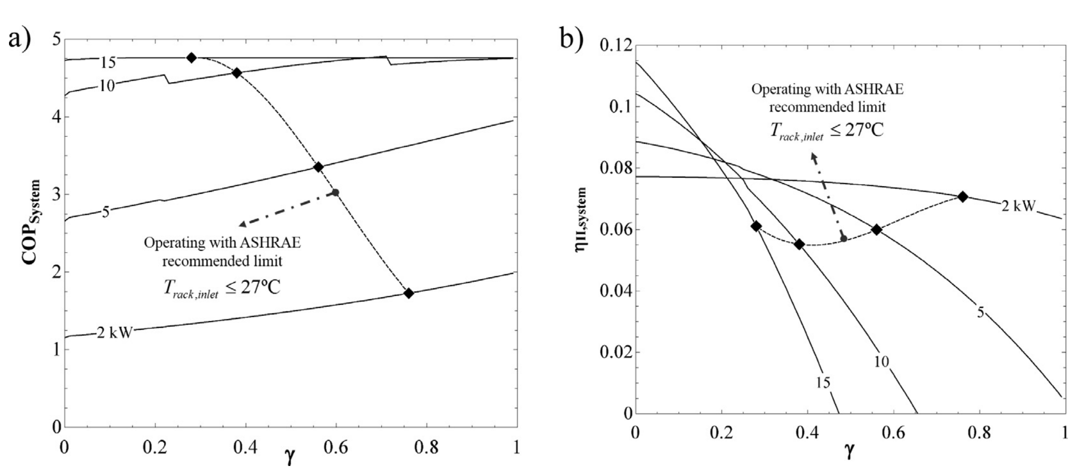

Fig. 13. Refrigeration system (a) Coefficient of Performance (COP) and (b) exergetic efficiency as function of hot air recirculation.

ingly—the system operates more efficiently under this condition (15 kW). This can also be inferred from Eq. (8), where the second term in the denominator decreases $\eta_{II,system}$ as the dissipation $\dot{Q}_{rack}$ increases. The second law efficiency captures the irreversibilities generated by the recirculation and better guides the design towards more energy efficient conditions.

6. Data center overall exergy and energy consumption

Figs. 14–16 show a comparison between the cooling power consumption and the total exergy destruction in the system. Since the cooling tower is assumed to operate at maximum capacity regardless of rack dissipation, the cooling tower overcools for low load density racks, consuming most of the cooling power and destroying more exergy within the system. The largest power consumer switches from the cooling tower at the lower densities to the chiller at the higher densities, which is better appreciated when more racks are active (Figs. 15a and 16a). The CRAH units consume roughly 15% of the total cooling power, independent on the load density and number of active racks, because of the implementation of modular components to overcome variations in heat flow dissipation.

Even when the chiller consumes the most power at the higher rack densities (Figs. 15a and 16a), its exergy destruction represents a small fraction (<15%) compared to the CRAH units (Figs. 15b and 16b). This indicates a more appropriate use of the refrigerant's cooling potential in the chiller refrigerant when the racks dissipate more heat. On the other hand, the CRAH units always increase their exergy destruction at higher load densities. This is mostly due to the inherent irreversibilities in the heat exchange between the cold water and hot air. It is important to remark that most engineering thermal processes inevitably destroy exergy; therefore when optimizing a system, one should aim to minimize $\dot{\chi}_{dest}$, rather than bring it to zero.

Finally, Fig. 17 shows the system second law efficiency as function of both the number of active racks and hot air recirculation. These results allow identifying the operating conditions that better

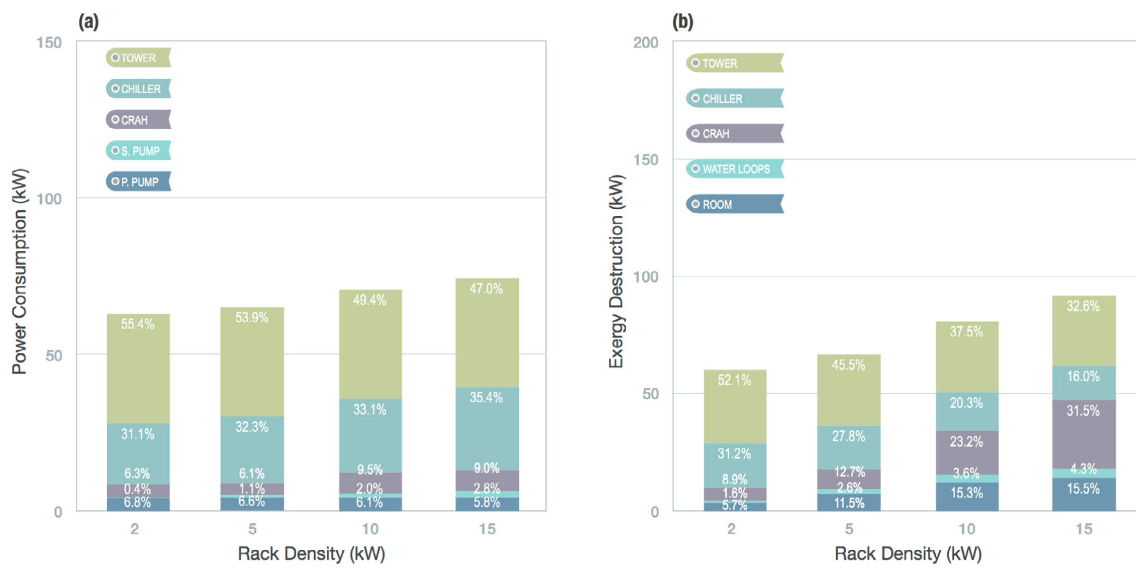

Fig. 14. Comparison between (a) cooling power consumption and (b) exergy destruction distribution for 14 active racks.

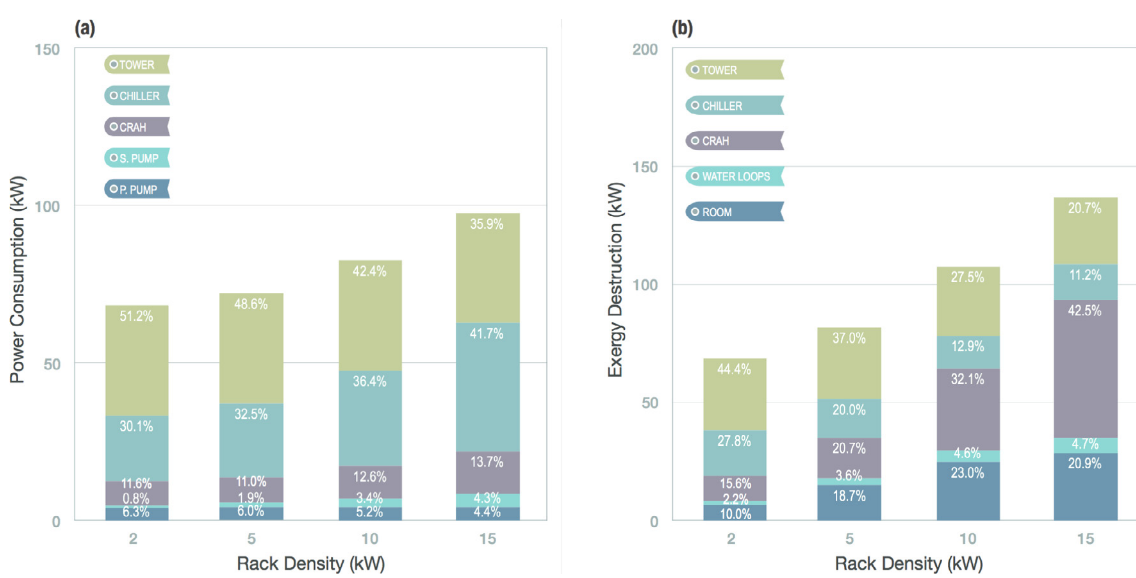

Fig. 15. Comparison between (a) cooling power consumption and b) exergy destruction distribution for 28 active racks.

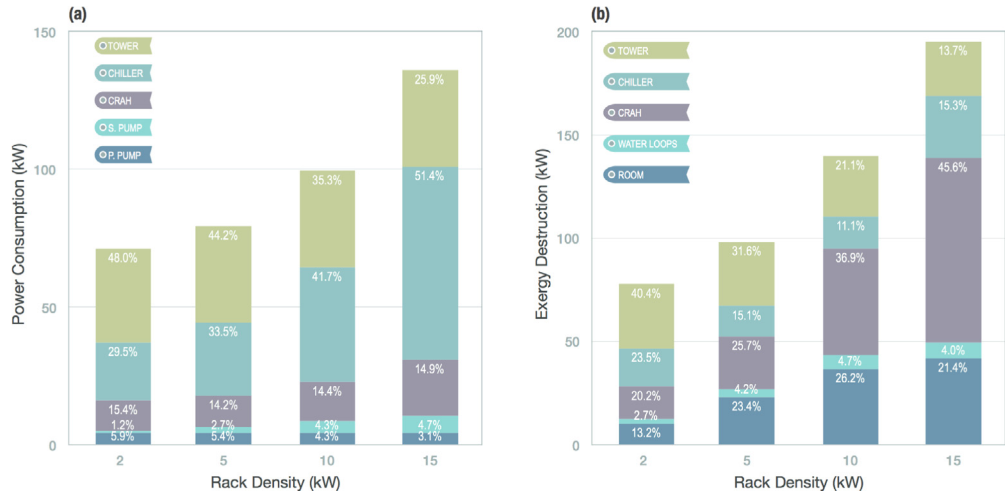

Fig. 16. Comparison between (a) cooling power consumption and (b) exergy destruction distribution for 42 active racks.

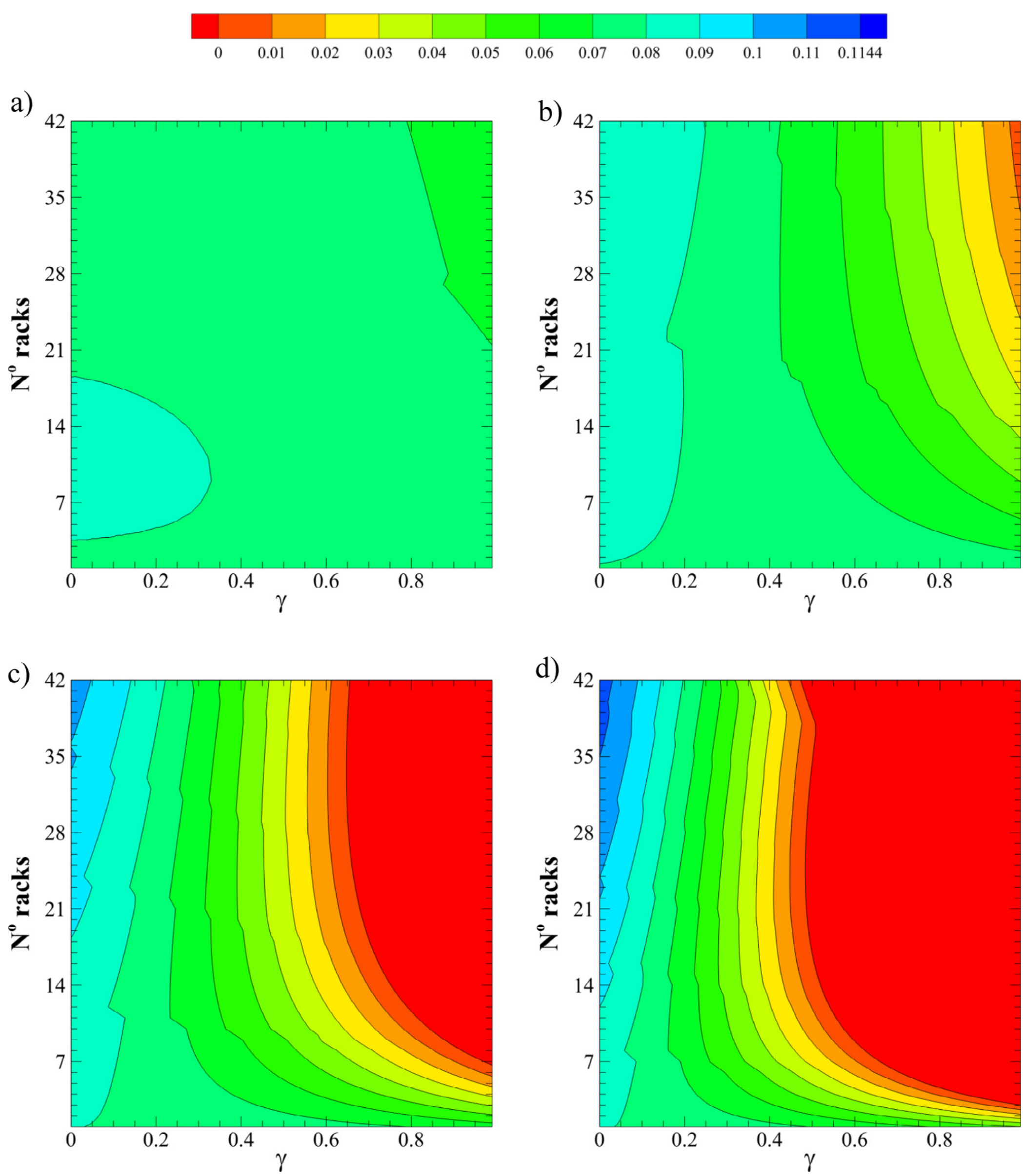

Fig. 17. Second law efficiency as function of both the number of active racks and hot air recirculation for a load density rack of (a) 2, (b) 5, (c) 10, and (d) 15 kW.

meet the data center thermal requirements. The impact of increasing hot air recirculation on the second law efficiency is always negative. Only when 2 kW racks are used (Fig. 17a) the effect of air recirculation is negligible, demonstrating that the proposed refrigeration system is oversized for this thermal requirement. Here, the increase in the number of active racks decreases the second law efficiency since more power is consumed by the CRAH units to provide the necessary air flow. Fig. 17b also suggests that the refrigeration system is oversized for 5 kW racks since a 20% of hot air recirculation, approximately, does not vary the system efficiency considerably. Fig. 17c and d shows that, for a moderate hot air recirculation (<20%), the system maximizes its efficiency when the number of active racks increases. The highest second law efficiency is achieved when the 42 racks are active and $\gamma = 0$. These results suggest that an aisle air containment technique must be employed when high load density racks are used to avoid increasing the rack inlet temperature above the ASHRAE operating limit. Thus, the proposed analysis can be employed to design future data center refrigeration systems avoiding overcooling and consuming unnecessary power.

7. Conclusions

The thermodynamic analysis of a data center refrigeration system presented in this work intended to simulate the early years of operation by parametrically varying the number of operating racks and their heat dissipation. The study included the effect of hot air recirculation inside the room over the overall system efficiency.

Rather than individualizing the analysis per component, this work focused in the overall system response to parametric variations. This strategy better identifies the conditions that optimize the entire system, which might be obscured when optimizing each component separately.

The largest consumers (CRAH units, chiller, and cooling tower) behaved differently depending on the system operating conditions. For high load density racks: (1) the chiller demonstrated an appropriate use of its cooling potential as its contribution to the system's exergy destruction remained below 16%, despite being the largest power consumer; and (2) the CRAH units, although consuming less power than the chiller (<15%), destroyed most of the exergy in the refrigeration system, leaving room for a re-design and potential energy consumption improvements. For low load density racks, the cooling tower - assumed to operate at maximum capacity throughout - consumed an excessive amount of power that manifested in its exergy destruction.

First and second law analyses showed that maximum efficiencies are achieved when all racks are active, provided the implementation of an aisle containment technique that avoids, or mitigates, hot/cold air mixing within the room. If hot air recirculation exists, the system exergetic efficiency always decreases. On the other hand, the system COP might (counterintuitively) increase if oversized components are selected, since more heat is dissipated without increasing considerably the power consumption.

This work demonstrates that a first law-based analysis is insufficient to find and quantify thermodynamic irreversibilities when optimizing data center energy performance. Complementing the study with a second law analysis advances the optimization process, from single component to overall system energy efficiency, allowing identifying both sources of high energy consumption and energy waste.

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