SEVENTH FRAMEWORK PROGRAMME

GreenDataNet

D3.1 – Analytical model of thermal behavior of DC

Status
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TABLE OF CONTENT

REVISION SHEET

Revision Number	Date	Brief summary of changes
Rev 0.1	31/01/2015	Baseline document
Rev 2.0	16/03/2016	Updated 3.3.3 based on consortium feedback

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1. INTRODUCTION

The main goal of the GreenDataNet project is to apply new technologies and tools to enhance the power efficiency of urban data centers, effectively decreasing their average Power Usage Effectiveness (PUE), and making them more sustainable in the future.

To optimize the operation of a data center, it is crucial to minimize both information technology (IT) and cooling energy consumptions. This deliverable presents a general GreenDataNet thermal model that contains first order approximation elements to enable the justification and optimization of green, virtualized data centers. By understanding the interaction of the outputs from the present model and integrating with the other detailed models developed within the project, a full overview of data center behavior can be quantified.

1.1 DOCUMENT PURPOSE

In this deliverable we assess the impact of operating conditions and data center configuration on the thermal performance of the data center, measured with PUE over a given time period. The model can be used to evaluate the potential of energy reuse or energy consumption reducing measures, such as free cooling applications detailed in (1). The concepts of energy reuse and exergy are discussed in detail in (1) with recommended KPIs and heat recovery strategies. Additionally a model can be used in conjunction with predicted conditions of IT load and weather to forecast and optimize the data center operation (2). This model can be expanded with other parts of the data center model that includes more accurate IT and networking loads (3).

The results in this document are presented for a small, pilot data center designed for 50kW maximum IT load and representing a typical urban data center (4). Basic parameters such as the operating inlet temperature to the IT racks, ambient outside temperature, the use of free cooling and the implementation of containment was evaluated (1).The model is however not limited to a 50kW IT load and a full set of operating parameters, constants and equipment efficiencies can be adapted for unique applications. A benchmark comparison is made against data provided by Credit Suisse (5).

1.2 DOCUMENT OVERVIEW

The rest of the document is organized as follows: Section 3 will explain the components for the thermal model and Section 4 will use an example and a benchmark to present results with varying conditions. Section 5 will list some conclusions.

2. THERMAL MODEL

A basic first order simplified model of a data center was programed to validate the sensitivity and configuration assumptions for a generic data center. The model is based on steady state operating conditions and most assumptions are aimed at maintaining a simple model as explained below with a very low computational cost.

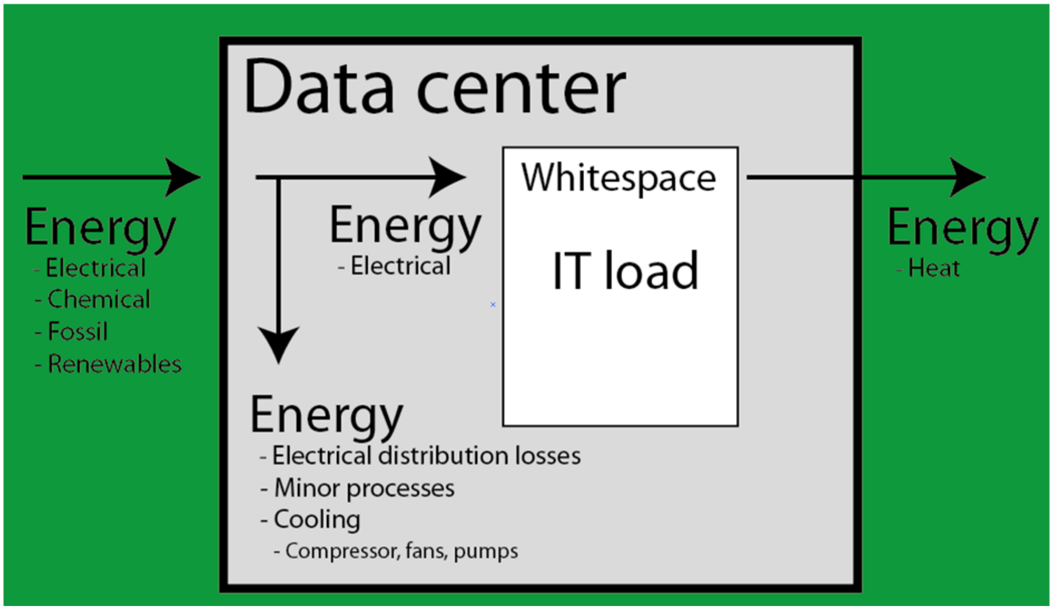


Figure 1 Thermal energy balance of a data center. PUE would be equal to the energy input divided by IT load. The primary energy form leaving the data center is heat.

Figure 1 presents a thermal energy balance for the data center. The input can include net grid power, renewable power (solar, wind, etc.) and non-renewable power (i.e. generators and fossil fuels). These elements are the actual power sources to run everything inside the data center control volume. Load shedding reduces the power input to the data center. If the data center is capable of injecting electricity back into the grid then electrical input energy becomes the net input, since it is safe to assume the data center will be a net energy consumer of electricity. Stored power in batteries and thermal reservoirs are not visible since they are inside the control volume. The data center is demanding input power to run the ICT load which is the main purpose of the data center. From a thermal perspective the only output is heat, which can be rejected to the environment or reused, if economical.

The operation and conditions will be explained with reference to a potential pilot project. The pilot is a 50 kW data center with 10 – 12 racks (6). The default test conditions are 40 kW IT load (80% of capacity) and 15°C average outside ambient temperature during the day. The inlet temperature to the racks is set to 20°C (range of 18 to 32°C suits GDN) and a temperature increase of 10°C is assumed for flow over the IT equipment. The airflow streams (Figure 4) from the computer room air handler (CRAH) is split into bypass air and the stream cooling the IT equipment. Similarly the IT exhaust air is split into recirculating flow (mixed with by-pass air and returned to the IT inlet) and the air stream returning to the CRAH unit. At each node the flow is assumed to become perfectly mixed to determine the temperature of the stream as it continues (Eq. 6).

The mass flow rates of air are determined from the heat load and sensible heating of air (Eq. 1). The power input from the fans and other prime movers are determined from simple pressure head assumptions (Eq. 7) and assuming efficiencies for the drive train and motor (Table 2).

Additional inputs to the model include electric power transmission via the UPS and transformer. The efficiency of an average UPS is 96% and for the transformer could be 93% (5). These two values are combined for the model to provide and overall efficiency of 90%. None of the other electrical losses are taken into account at the moment.

The model is used to run a series of steady state operating conditions over the period of time. During a 24 hour period ambient temperature varies and so does the operation of the data center. It would be feasible to run the model over an entire year, but this would average out the seasonal variations that can also be associated with the physical location (4). The main time variable inputs are ambient outside temperature, IT inlet set point temperature and IT load. A summary of the inputs and variables is provided in Table 1. The model does not deal with transient behavior.

The model does not take into account all losses in the power chain and represents a fairly ideal view of the sub components modelled in order to balance simplicity with the accuracy needed. For this reason PUE values will be lower than actual.

Table 1 Model variables that serve as input, design variables that need to be sized appropriately, calculated variables that are not necessarily reported and possible outputs.

Inputs	Design Inputs	Calculated	Outputs
IT power	Overall heat transfer coefficients	Mass flow rates for air, refrigerant and water	Active cooling power
White space IT inlet Temperature	Heat exchanger size	Temperatures of white space air at mixing points	Compressor work
Temperature rise over IT	Condenser or cooling tower temperature differences	Temperature of fluid in heat exchangers	White space air movement
Ambient outside air Temperature	Evaporator inlet vapor quality	Evaporator temperature	Total power
Airflow mixing ratios	Politropic coefficient	Condenser temperature	COP
Power supply losses (UPS and transformer)	Pressure head for fans and pumps	Saturation enthalpies of cooling cycle	PUE
		Condenser heat load	
		Compressor work	
		Work done by fans and pumps	
		Free cooling active/inactive	

2.1 MODEL COMPONENTS

The model components described in this section pertain to the heat loads (IT load, heat exchange, cooling cycle etc.), airflow (movement and mixing of air), prime movers (compressor, fans and pumps) and the environmental factors impacting the building. The component relations are shown in Figure 2. Different configurations of cooling equipment such as a water loop for the condenser and direct air economization can also be programed (7) (8).

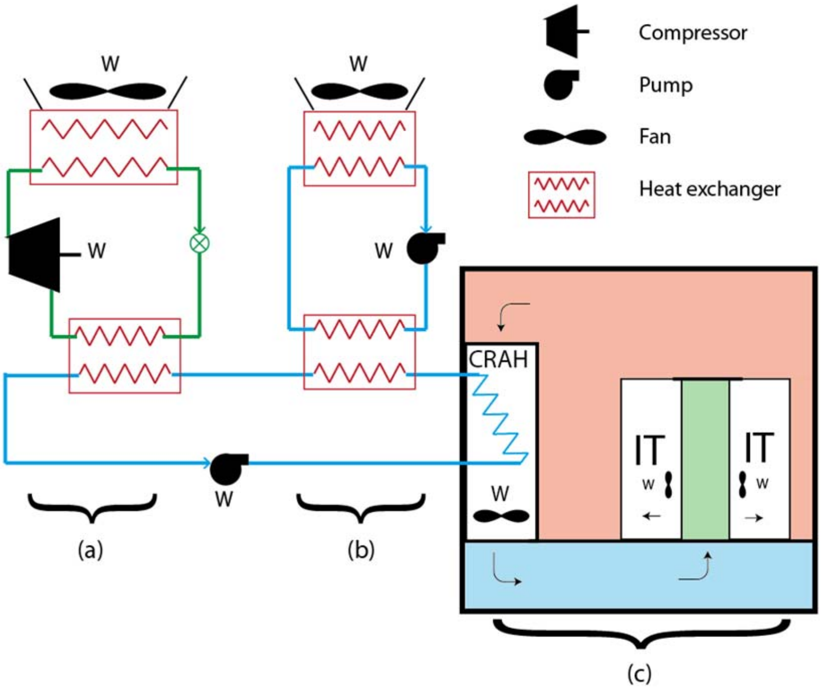


Figure 2 Diagram indicating the mechanical cooling (a), free cooling (b) and whitespace (c) of the data center. The whitespace indicated basic containment and a CRAH unit with a pumped water loop that is linked with heat exchangers to the cooling equipment.

2.1.1 HEAT LOADS

The IT load (kW) is a user input. This is used to determine the required air flow, m_{IT}, with:

$$Q_{IT} = m_{IT} c_p (T_{IT\ outlet} - T_{IT\ inlet}) \qquad \text{Eq. 1}$$

The CRAH and condenser/indirect free cooler heat exchangers are defined on their overall behavior with:

$$Q = U_o A (T_{Hot} - T_{Cold}) \quad \text{Eq. 2}$$

The heat exchangers are solved with the ε-NTU method and the refrigerant stream is assumed to be at constant temperature due to the phase change process (9). The overall heat transfer coefficient is assumed constant and representative of liquid to air, finned heat exchangers (10). The heat exchangers are assumed to be cross-flow with both streams unmixed. The area of each heat exchanger is determined to allow the cooling of the maximum load defined for the data center. This results in the following equations:

$$C_{min} = min\begin{cases} m_{air} c_{p\ air}, & air\ side \\ \infty, & isothermal\ side \end{cases}$$

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{Q}{C_{min}(T_{Hot\ inlet} - T_{Cold\ inlet})}$$

$$\varepsilon = 1 - e^{-NTU}\ or\ NTU = -\ln(1-\varepsilon)$$

$$NTU = \frac{U_o A}{C_{min}} \qquad \text{Eqs. 3}$$

Using the model of the heat exchangers an estimate of the refrigerant saturation temperature can be made for the evaporator and condenser (Eqs. 3). The vapor-compression cycle (11) is assumed to evaporate from a vapor quality of 10% until superheated conditions (the cooling side of the cycle, Eq. 4) and then the compressor does work on the fluid to compress it to the condensing pressure (the heat rejection side of the cycle) (Figure 3). The compressor is described in the following section.

$$Q_{IT} = m_{comp}(1 - x_{in})h_{lv} \qquad \text{Eq. 4}$$

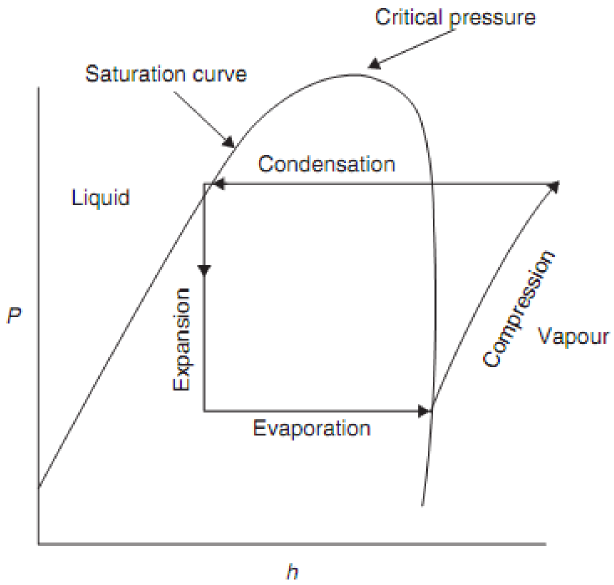


Figure 3 Pressure enthalpy diagram for an ideal vapor-compression cycle with the process indicated on the diagram.

The operation of the indirect free cooling is assumed to be a pumped water loop and a cooling tower in the outside air. The water pump is dealt with as a prime mover with a constant pressure head (Table 2). The cooling tower is only used to evacuate the heat from the IT load in free cooling mode. In comparison, to active mechanical cooling, a chiller unit has to evacuate the heat from the IT load and the chiller unit itself.

The operation of the indirect free cooling is determined by evaluating the potential heat that can be removed by the ambient air moving over the heat exchanger while maintaining the desired internal temperatures. Thus potential performance of the cycle is monitored and if the power it delivers is sufficient to cool the data center, the chiller unit is turned off and the free cooling can take over. The model does not allow partial load to be dealt with by the free cooling. Free cooling is common on larger application but there are examples of free cooling applications for three to four rack solutions (12).

2.1.2 AIR FLOW

The mixing of air is modeled based on a nodal model (Figure 4) where mixing of the air stream is calculated at each node (1 - 4) by using the conservation of mass law (Eqs. 5). In this model the fractions of air changing stream are defined as f_1 - f_4. The fractions are user inputs and can be set to simulate chaotic airflow or contained solutions and in-between.

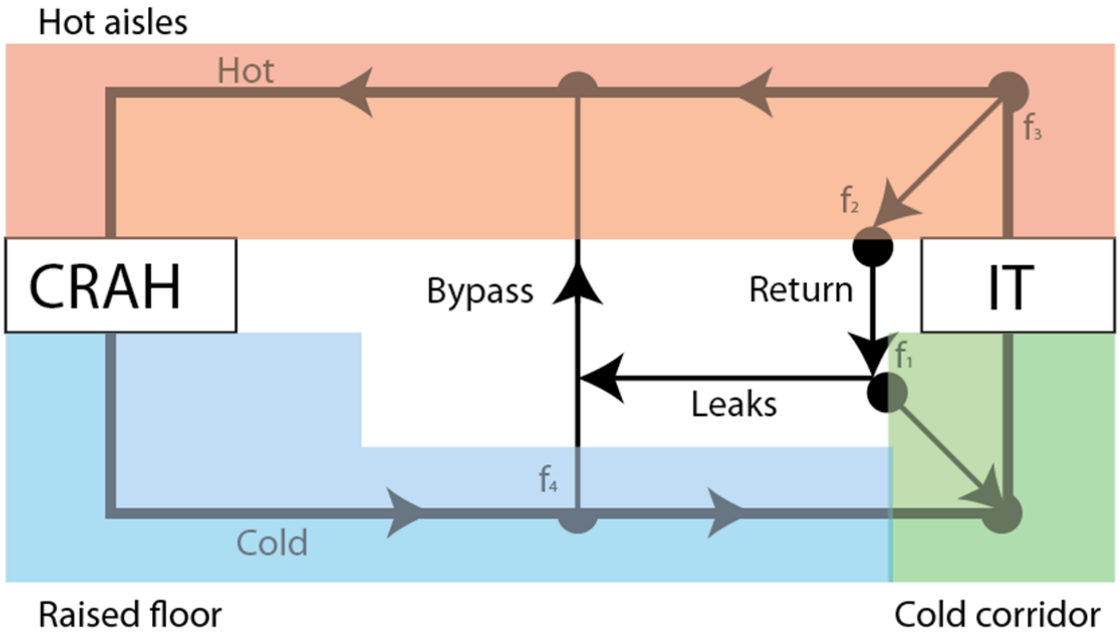


Figure 4 Representation of the nodal mixing model used for the airflow streams in the data center. There is the ideal stream around the outside, bypass flow and flow returning from the hot aisle to the IT equipment inlet.

$$\sum m_{inlet,i} - \sum m_{outlet,i} = 0 \qquad \text{Eqs. 5}$$

$$m_{IT} = (1 - f_4)m_{CRAH} + f_1 m_{return}$$

$$m_{bypass} = (1 - f_1)m_{return} + f_4 m_{CRAH}$$

$$m_{return} = (1 - f_2)m_{bypass} + f_3 m_{IT}$$

$$m_{CRAH} = (1 - f_3)m_{IT} + f_2 m_{bypass}$$

Uncontained solutions could allow up to 60% bypass air (13). Contained solutions from Eaton have been tested to a specification that no more than 3% of air will leak or bypass the IT equipment (14). See the benchmark test for examples of air stream volume flow rates for different containment solutions.

Since the temperature is known at the inlet of the server racks and directly at the outlet of the server racks the conservation of energy allows us to determine the temperature of the air for each stream.

$$\sum m_{inlet,i} c_p T_{inlet,i} - \sum m_{outlet,i} c_p T_{outlet,i} = 0 \qquad \text{Eq. 6}$$

The vapor-compression cycle is assumed to operate with R-134a refrigerant. R-134a is widely used as a refrigerant in refrigeration systems, heat pumps, air-conditioners, etc. originally replacing CFCs, R-12 and R-22 (15). The refrigerant properties determine the function of the vapor-compression cycle and the performance of the compressor. (16) provides the fluid properties. R-134a is a very common refrigerant in the cooling industry and was originally introduced as replacement for R-22 due to the Montreal protocol (15). In a wider context, new refrigerants have been introduced as a result of directives imposed by environmental concerns to minimize the damaging effects of refrigerants on the ozone layer or to limit the greenhouse effect based on information from scientific, environmental, technical and economic sectors (15). Today the industry is developing and working on the next generation low GWP and low ODP refrigerants (17).

2.1.3 PRIME MOVERS AND HEAT PUMP

Each prime mover is scaled for the maximum capacity of the facility. The efficiency of the device is a function of the duty of the facility and may vary slightly.

The power consumed is calculated by:

$$W_{fan} = \rho g h V \eta \quad \text{Eq. 7}$$

The compression is described by a politropic process to determine the specific work input from the compressor (Eqs. 8). The politropic process is considered as a process between adiabatic compression and isentropic compression. The specific heat ratio (n) of the refrigerant R-134a is 1.106 at 25 °C as an ideal gas.

$$\dot{W}_{Comp} = \frac{nR(T_{Condensor} - T_{Evaporator})}{n-1} \qquad \text{Eqs. 8}$$

$$W_{Compressor} = \dot{W}_{Comp} m_{comp} \eta$$

Table 2 Equipment efficiencies for various mechanical devices and an expected pressure head (15) (16)**.**

Equipment	Efficiency	Max. Head
Small server fan	20 - 25%	20 Pa
Large cooling tower fan	50 - 60%	20 Pa
Compressor	50 - 60%	R-134a model
Water pumps	70 - 80%	150 kPa

2.1.4 BUILDING AND ENVIRONMENTAL

The data center can be housed in a physical building and the geographical location of this building is influenced by various factors:

- Grid power sources
- Network connectivity
- Business proximity
- Safety and redundancy
- Environment

The environment can impact energy sustainability and the efficiency that can be accomplished (4). Major impacts from the outside environment include (20):

- Heat loads from operations (IT load and utilities)
- Solar gains (Thermal radiation)
- Convection from the atmosphere (Air movement over the building)
- Casual gains from occupation of the building
- Infiltration (hot and cold flows into the building)

A chart with the potential hours of free cooling for locations over Europe was provided in (4). The hours of local atmospheric conditions, mapped out on a psychrometric chart against the recommended ASHRAE temperature range (21) gives a few insights (Figure 5) into how operations may vary with conditions. This simplified view indicates how many hours of use a certain location can expect for the following categories:

- Outdoor air mixed with heated return air and humidification
- Outdoor air mixed with heated return air
- Refrigeration
- Direct outdoor air cooling

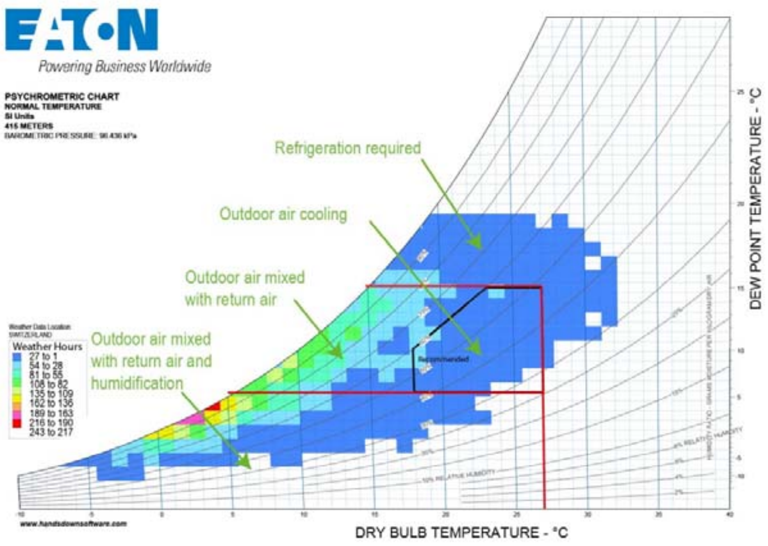


Figure 5 Psychrometric chart for atmospheric conditions in Switzerland indicating dry bulb temperature and relative humidity. The ASHRAE recommended zone for IT equipment is indicated and red lines delineate zones that require certain treatments.

Another potentially large environmental load can be the solar radiation. The solar radiation is a function of the location (mostly latitude and altitude) (Table 3) and the building characteristics (reflectivity and thermal inertia of the building). For a small data center the location within a larger building could mean that very little solar radiation is experienced. However, small data centers may also be located on the roof of a building in a very exposed location. The reflection coefficient (albedo) of the roof material could also determine how much heat is absorbed by the structure with significant impact even at northern latitudes. Reflection coefficient can vary from 0.03 for a tar and gravel roof (absorbing almost all incident radiation) to 0.7 for a highly reflective roof material and 0.8 - 0.9 for snow (22).

Table 3 Solar radiation on a flat surface at ground level as a function of the latitude (17).

Location	Latitude	Solar Gain (W/m^2)
North Africa	35 °	250
Spain	40°	228
Switzerland	46°	143
Netherland	50°	114
Nordics	60°	96

The convective heat transfer from a building due to the temperature difference between the inside and outside could also result in heat loss or gain. This is a function of the size of the building and its exposure to the outside. The size of a data center can be estimated by allowing 2.5 - 3m^2 for each rack and a factor of 2 - 3 for the size of the utility space relative to the white space depending on the tier level (23). Assuming a maximum exposure to solar radiation and convection the heat loads (in Watts) can be compared to the IT load in Figure 6. The radiation is 13% of the IT load, but this can be significantly reduced by the choice of building material. The convective component is comparatively negligible.

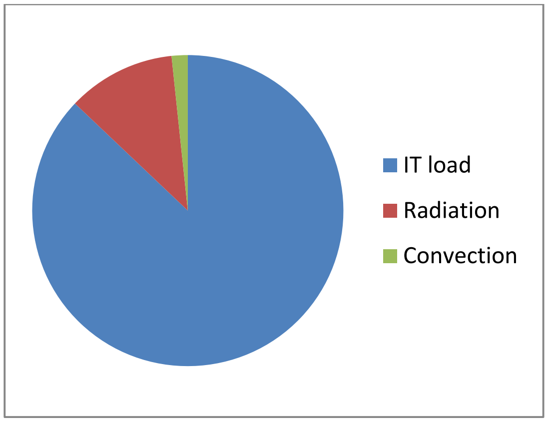


Figure 6 The IT load of a 50kW small data center with environmental heat load of solar radiation and convection for potential maximum exposure (zero reflection) conditions at latitudes corresponding to Switzerland.

2.2 METRICS AND KPI

For the data presentation PUE was chosen as the principle key performance indicator to communicate in this report. For relative comparison the actual energy used (kWh) is also presented in certain cases. The advantage of a numerical model is that all the data is available to compute any of the required KPIs if the project requires it ((4), Table 2-4).

3. RESULTS

3.1 MODEL PARAMETERS

Basic parameters such as the IT load, operating inlet air temperature to the IT racks, ambient outside temperature, the use of free cooling and the implementation of containment was evaluated. A benchmark evaluation was done on the Credit Suisse data center based on the available information presented in (5). Secondly, a sensitivity study was done on the pilot data center in order to evaluate the interaction of various parameters. The following section discusses the model parameters and the results.

3.2 BENCHMARK

The power consumption of the IT load was provided for intervals of 15 minutes (Figure 7). The ambient temperature conditions were gathered from hourly weather data for the Zurich area for the years 2010 - 2012 (24). The model was run for every hour of year 2010 and 2012. The IT load data was averaged for each hour to match the weather data. The cumulative energy consumption of IT load and the total data center can then be calculated to establish the PUE for the year. To show the seasonal variation in PUE the same calculation was done for one month intervals.

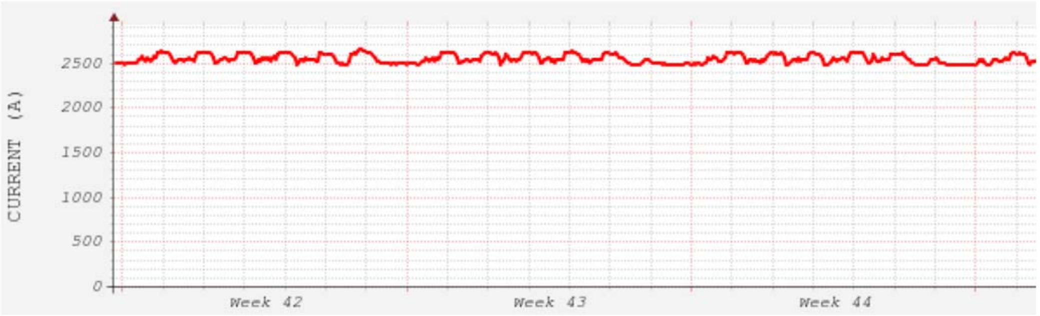


Figure 7 Total current consumed by the data center indicating very constant load conditions (<4% variation).

The power supply through the transformer and UPS is considered according to the conditions described in (5). The UPS load (40 - 50% @ 93.2% efficiency) and transformer efficiency (96%) together results in 90% efficiency of the power chain.

The flow rates of air are indicated (Table 4) for uncontained - chaos cooling, contained solution and an estimate of the 80% contained solution at Credit Suisse (5). Note the additional air volume moved by the CRAH in the uncontained solution in comparison to the contained option. The air streams returning to the IT equipment inlet (recirculating) and bypassing the IT equipment altogether are also significantly reduced by the containment. Air flow rates are calculated for each time step.

Table 4 Airflow estimates for ~2.7MW IT load at a temperature difference on 10°C and implementing the mixing model described above in Section 2.

	Airflow streams			
Containment condition	**IT load of 2.7MW (m^3/h)**	**CRAH (m^3/h)**	**Bypass returning to IT inlet (m^3/h)**	**Bypass returning to CRAH (m^3/h)**
Chaos	829285	1368312 (**165% of IT load**)	290258 (21% of CRAH)	829285 (61% of CRAH)
Contained	829285	847238 (102% of IT load)	24878 (3% of CRAH)	42832 (5% of CRAH)
Credit Suisse	829285	1036605 (125% of IT load)	82931 (8% of CRAH)	290250 (28% of CRAH)

The results for 2010 indicate a typical seasonal variation where the PUE is lower during the colder months of the year due to the use of free cooling and favorable condensation conditions (Figure 8). The annual PUE value from the model is 1.33 compared to 1.65 based on the actual data. This can be explained by the lack of accurately modeled mechanical and electrical losses in the data center ranging from machine efficiencies, generators, lights, PDU power losses, comfort cooling for the occupants and services running in the building. The exact power consumption measurement locations in the Credit Suisse data center are not mimicked in the model either. These elements account for a 25 - 30% difference between the model and the real data. Most importantly the trends in the model follow the real data trends accurately and behave in a predictable manner.

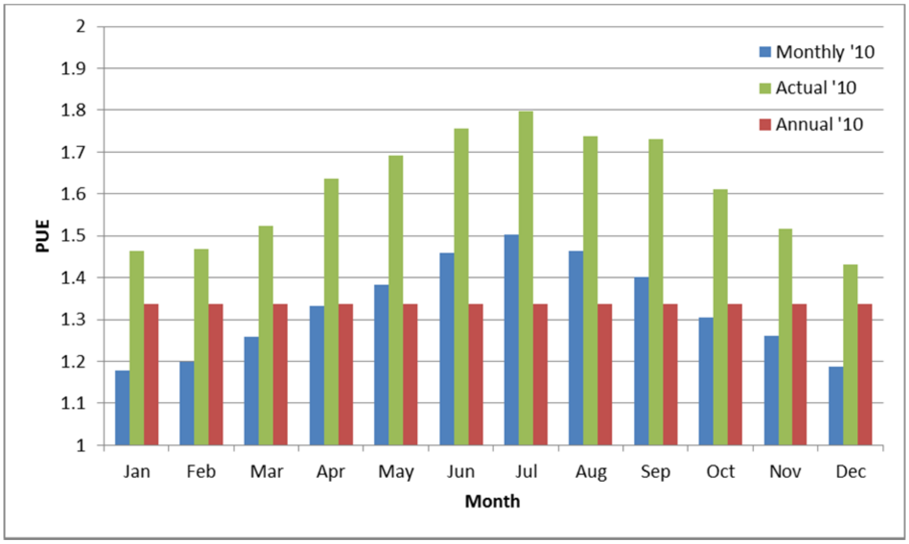


Figure 8 Annual data comparison between the thermal model and real data for 2010 indicating the seasonal variation in monthly reported values.

The results for 2012 are slightly more interesting since the seasonal variation is perturbed by an exceptionally warm month of March and a colder July (Figure 9). This particular trend also showed up in the modelled data indicating the capability of the model to accurately capture the trends due to external temperature. Overall the model is under predicting the PUE by the same amount as in 2010. The actual PUE for the year is 1.57 and the modelled PUE is 1.28, both lower than 2010. The average temperature during the year in 2010 was slightly lower than 2012; 8.89 and 9.93°C respectively (Table 5). This indicates that daily temperature variations are not well represented by the averages and that regardless of the higher average temperature in 2012 the year had a better PUE. The year 2010 had 12% more hours above 25°C compared to 2012 and both years spent roughly 40% of the time in free cooling mode. Thus, locations with more hours at exceptionally high temperature may affect performance significantly. Credit Suisse have indicated that they change the IT inlet air temperature set point from time to time, but this was not implemented.

Table 5 Summary of the PUE of the data center and the model with average temperature.

	PUE actual -	PUE model -	Temperature Average °C
2010	1.61	1.34	8.89
2012	1.57	1.28	9.93

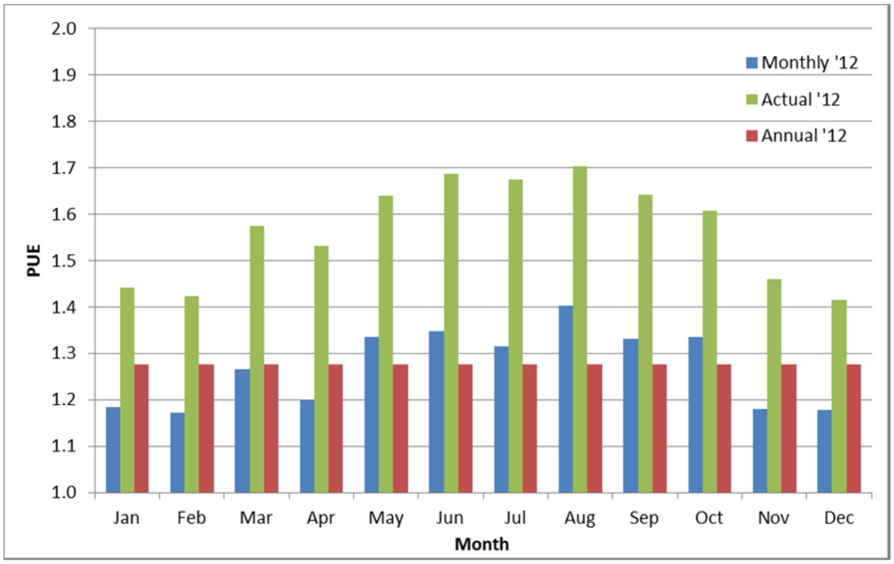


Figure 9 Annual data comparison between the thermal model and real data for 2012 indicating the seasonal variation in monthly reported values.

3.3 SENSITIVITY STUDY

Since PUE calculations over extended periods average out the short term variations as seen in the benchmark study a single standard day was used to demonstrate the model interaction with perturbations. The study was done on a 50 kW data center with standard test conditions of 40 kW IT load (80% of capacity) and 15°C average outside ambient temperature during the day. The inlet temperature to the racks is set to 20°C and a temperature increase of 10°C is assumed for flow over the IT equipment. The model is used to run a series of steady state operating conditions over the period of a single day. During the 24 hour period ambient temperature varies with a classic peak in the early (Figure 10). The temperature profile is perturbed by adding or subtracting a fixed temperature for each time step. The main parameters tested are:

- ambient outside temperature
- IT inlet air set point temperature
- IT load
- Free cooling
- Containment

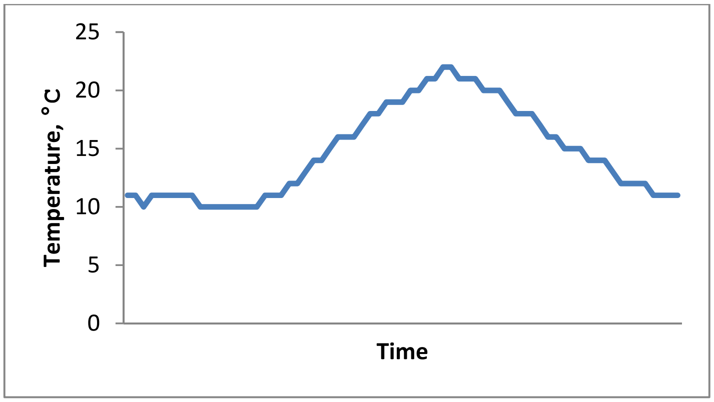


Figure 10 Ambient temperature variation during a single day.

The results are reported in Figure 11. The central block in the cross represents the standard case detailed in the previous paragraph. Each case is made of 4 sub conditions as described in the legend. Each sub condition is a possible combination of containment and free cooling.

For brevity we will discuss only the contained solution with free cooling activated (red text) since this option had the best performance. Reference will be made to the other sub conditions for the standard case only. The relative magnitude of the PUE values is more important than the actual value in this model.

3.3.1 STANDARD CASE

The best performer was the contained data center airflow with free cooling active at temperatures where sufficient cooling was possible with the heat exchangers (PUE = 1.17). The uncontained solution with active free cooling was not able to optimize IT energy use (PUE = 1.22) and performed slightly worse than a contained solution without free cooling (PUE = 1.21). This indicated that, for these specific conditions it was more advantageous to implement containment than free cooling. This result only changed when the ambient temperature decreased, thereby allowing additional free cooling hours. The worst performer was uncontained airflow without free cooling (PUE = 1.27).

3.3.2 CONTAINED WITH ACTIVE FREE COOLING

Reducing the IT inlet air temperature set point in the data center increases the work required by the cooling cycle and therefore increase the PUE value (Figure 12). Increasing the IT inlet air temperature reduces the PUE. This impact is not as advantageous, is this case, going from 20°C to 23°C as is was to increase from 17°C to 20°C. This is caused by the cooling cycle operation and average temperature difference between inside air and ambient outside air conditions.

Cooler ambient air temperatures result in better PUE values due to the double benefit of additional free cooling and the lower condensing temperatures in the vapor-compression cycle. Higher outside temperatures reduce the potential free cooling hours and rapidly increase the power required for the cooling cycle (Figure 13).

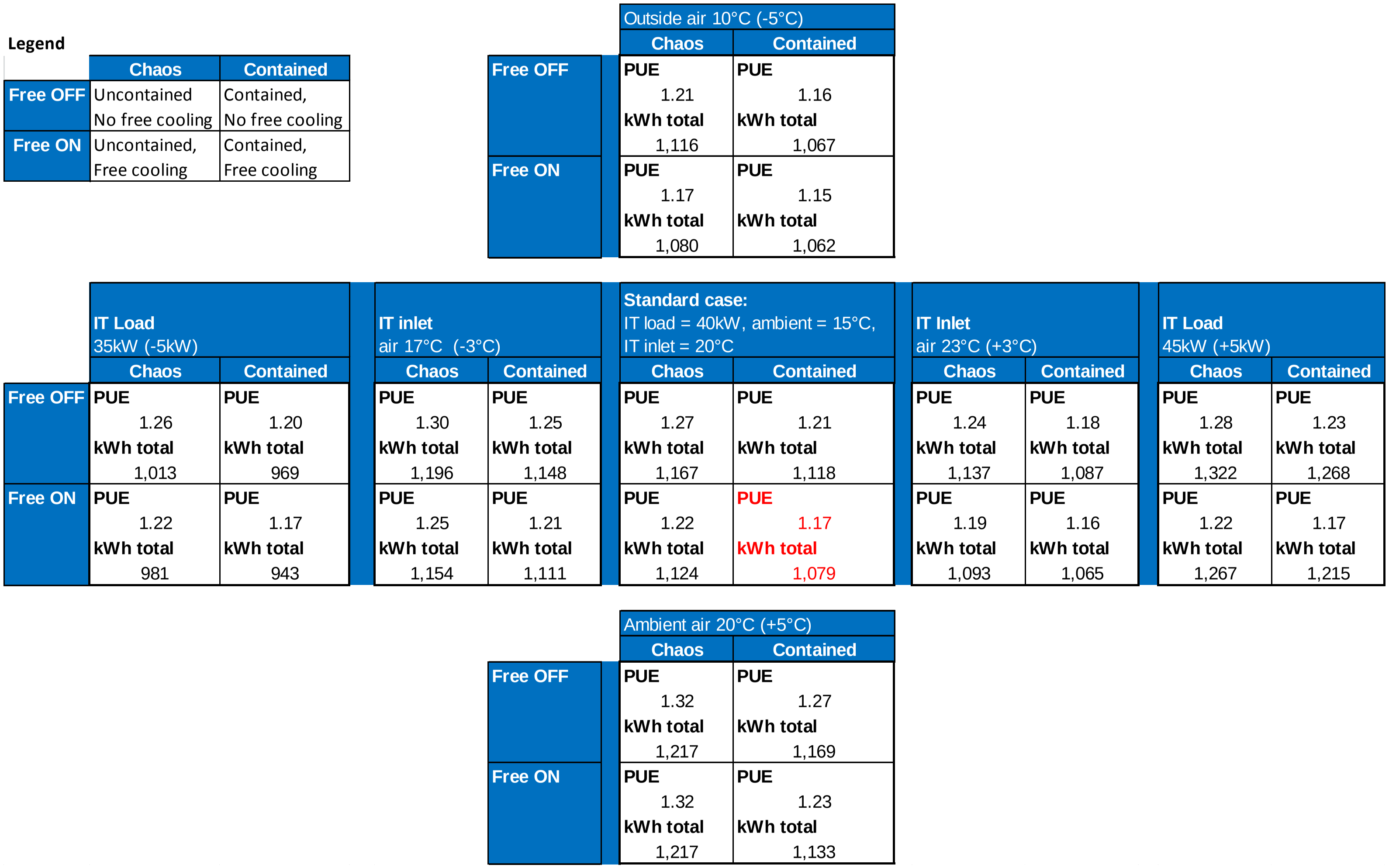
If the IT load in the data center reduces because of lower production requirements a well contained data center maintains a stable PUE. In the case of the contained data center with free cooling the PUE remained constant at 1.17 from 35 - 45kW. In all other cases the PUE performance decreased as the IT load increased.

3.3.3 HEAT REUSE

Using the ERF on the data center model with containment and free cooling, one can assume a constant heat reuse percentage (60%) similar to the example in (1). Figure 14 indicates how the ERF decreases as the ambient air temperature increase. This is due to the fact that the total energy expenditure increases and the reuse is a constant percentage of the IT load (also constant). If a Carnot cycle is implemented between the maximum and minimum ambient temperatures the available energy would decrease as the ambient temperature increases since the Carnot efficiency is a function of the difference in the two temperature reservoirs.

Similarly, the ERF increases with an increase in set IT inlet temperature (Figure 15). This result is due to the increase in reusable, available energy and the reduction of energy consumed by the entire system. Both the parameters mentioned are components of the ERF equation.

As seen in the trends and discussed in (1) the exergy available requires careful evaluation of the economic value of reuse heat and the investment. Employing a thermodynamic cycle limits the amount of useful energy due to the low temperatures. Better heat collection technology or directly using the low exergy heat in suitable processes are recommended. Such low quality heat reuse would avoid the use of a high quality heat source (ex. fossil fuel combustion) or reduce the amount of energy consumed (25). The reuse of year-round constant data center heat is also best achieved by linking processes that are continuous and constant, such as a feed water preheater for a conventional power station (25). Seasonal space heating is often easy to implement, but only limited to a fraction of the year and much energy is rejected to the atmosphere during the remaining part of the year.



Legend

	Chaos	Contained
Free OFF	Uncontained No free cooling	Contained, No free cooling
Free ON	Uncontained, Free cooling	Contained, Free cooling

Outside air 10°C (-5°C)	Chaos	Contained
Free OFF	PUE 1.21 kWh total 1,116	PUE 1.16 kWh total 1,067
Free ON	PUE 1.17 kWh total 1,080	PUE 1.15 kWh total 1,062

	IT Load 35kW (-5kW)		IT inlet air 17°C (-3°C)		Standard case: IT load = 40kW, ambient = 15°C, IT inlet = 20°C		IT Inlet air 23°C (+3°C)		IT Load 45kW (+5kW)	
	Chaos	Contained	Chaos	Contained	Chaos	Contained	Chaos	Contained	Chaos	Contained
Free OFF	PUE 1.26 kWh total 1,013	PUE 1.20 kWh total 969	PUE 1.30 kWh total 1,196	PUE 1.25 kWh total 1,148	PUE 1.27 kWh total 1,167	PUE 1.21 kWh total 1,118	PUE 1.24 kWh total 1,137	PUE 1.18 kWh total 1,087	PUE 1.28 kWh total 1,322	PUE 1.23 kWh total 1,268
Free ON	PUE 1.22 kWh total 981	PUE 1.17 kWh total 943	PUE 1.25 kWh total 1,154	PUE 1.21 kWh total 1,111	PUE 1.22 kWh total 1,124	PUE 1.17 kWh total 1,079	PUE 1.19 kWh total 1,093	PUE 1.16 kWh total 1,065	PUE 1.22 kWh total 1,267	PUE 1.17 kWh total 1,215

Ambient air 20°C (+5°C)	Chaos	Contained
Free OFF	PUE 1.32 kWh total 1,217	PUE 1.27 kWh total 1,169
Free ON	PUE 1.32 kWh total 1,217	PUE 1.23 kWh total 1,133

Figure 11 Results for the PUE and energy use for a single day and the perturbations in IT load, ambient temperature and IT inlet air temperature.

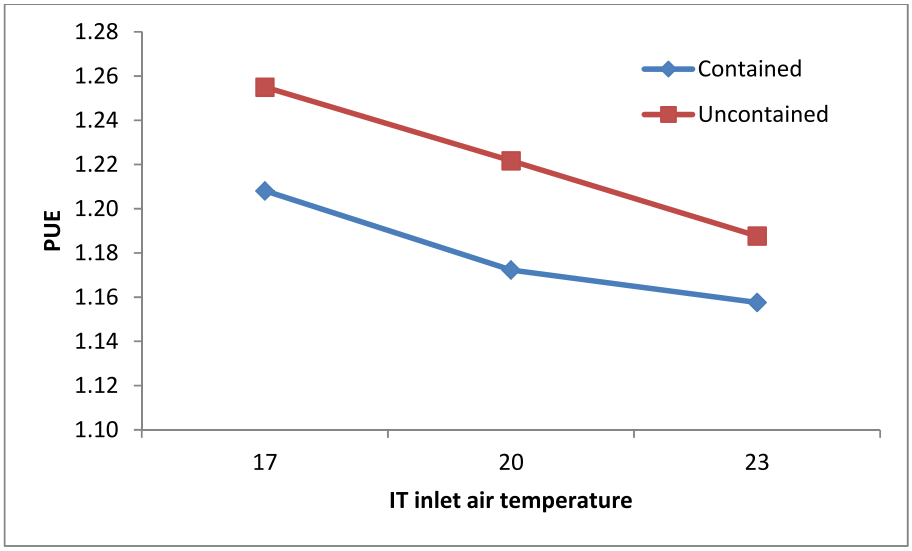


Figure 12 The PUE decreases as the IT inlet temperature increased and contained conditions fared better than uncontained. Free cooling was active.

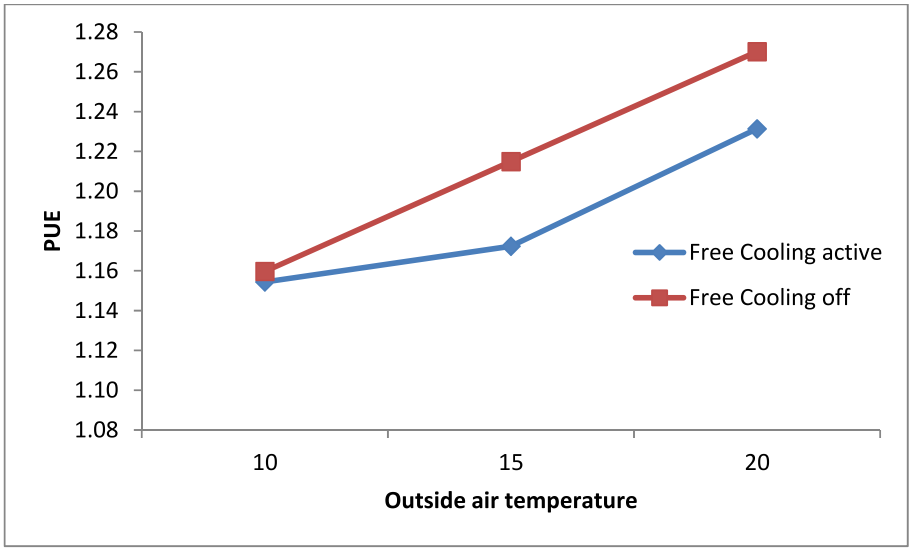


Figure 13 The PUE increases as the outside temperature increased and free cooling reduced the PUE. As the temperature decreases and free cooling is active 100% of the time the curves will coincide. Similar conditions will occur once no free cooling is possible. Results are for contained conditions.

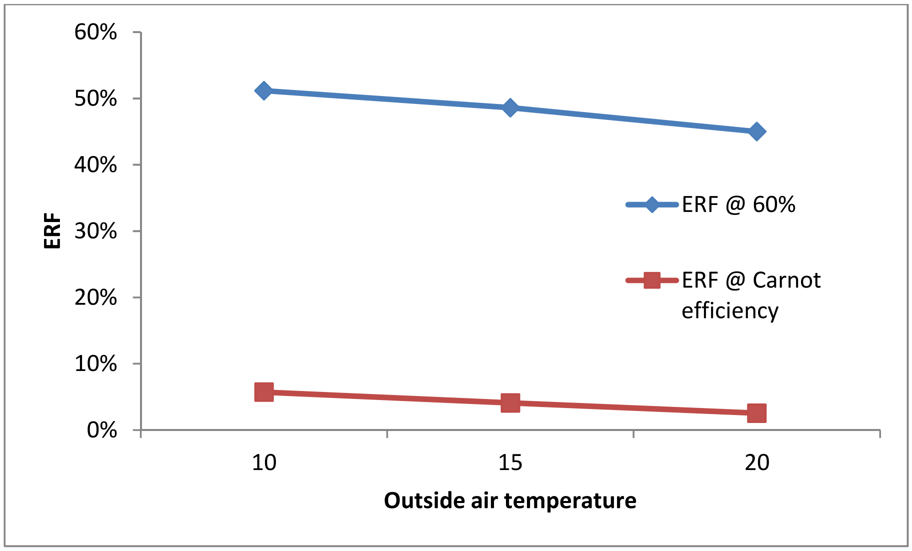


Figure 14 The ERF decreases as the outside air temperature increased and direct reuse of 60% of the heat achieved higher ERF that a Carnot cycle. Free cooling and containment was active.

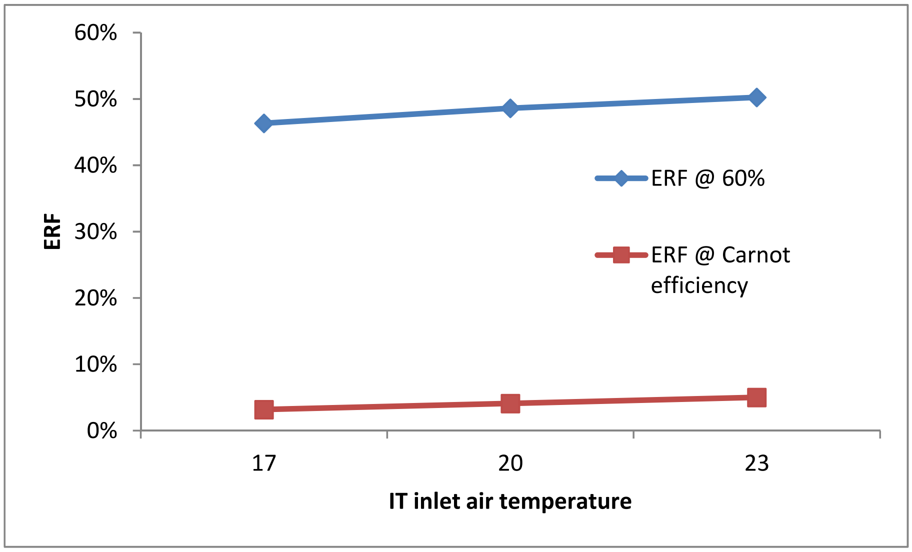


Figure 15 The ERF increases as the IT inlet temperature increased and direct reuse of 60% of the heat achieved higher ERF that a Carnot cycle. Free cooling and containment was active.

4. CONCLUSIONS

An analytical model was programmed to model the thermal behavior of a data center. The model uses ambient air temperature, IT load, IT inlet air temperature, containment behavior and free cooling operation to predict the power consumed by the data center utilities related to cooling.

One main factor that affects the data center is the ambient air temperature. Colder environments perform better. Additional environmental conditions that influence the data center include solar radiation, building construction and convective heat transfer. Locations with more hours at exceptionally high temperature may affect performance significantly even if the rest of the year is comparable with others and the hours of free cooling utilized is similar.

Implementing free cooling together with containment results in the best operating conditions. Within the optimal infrastructure control system set points and equipment configuration may also benefit the performance. For example, higher IT inlet air temperature will reduce energy consumption. Well contained data centers maintain a more constant PUE as the load changes.

NOMENCLATURE

Acronyms and abbreviations

ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineers
CRAH	Computer Room Air Handler
CUE	Carbon Usage Effectiveness
DC	Data Centre or Direct Current
ERF	Reuse energy/Total energy
ICT	Information and Communication Technology
IT	Information Technology
KPI	Key Performance Indicator
PDU	Power Distribution Unit
PUE	Power Usage Effectiveness
UPS	Uninterruptible Power Supply

Nomenclature

A	Area, [m^2]
$\mathbf{c_p}$	Isobaric specific heat, [Jkg/K]
$\mathbf{C_{min}}$	Specific heat rate, [W/K]
f	Fraction, [-]
g	Gravitational acceleration, [m/s^2]
h	Specific enthalpy, [J/kg] or height [m]
m	Mass flow-rate, [kg/s]
NTU	Number of transfer units, [-]
P	Pressure, [Pa]
Q	Heat, [W]
R	Specific gas constant, [J/kgK]
T	Temperature, [K]
$\mathbf{U_o}$	Overall heat transfer coefficient, [W/m^2K]
V	volume flow, [m^3/s]
W	Power, [kW]

x	Vapor quality, [-]

Greek

ρ	Density, [kg/m^3]
η	Efficiency, [-]
ε	Heat exchanger effectiveness, [-]

Subscripts

Air	Air conditions
Bypass	Bypass flow
Cold	Cold side
Comp	Compressor or refrigerant related
CRAH	Computer room air handler
Fan	Fan related flow or power
Hot	Hot side
Inlet	Inlet conditions
IT	Information technology related
lv	Latent heat of evaporation
Outlet	Outlet conditions
Min	Minimum value
Pump	Pump related conditions
return	Return flow

LIST OF FIGURES