OPTIMIZATION OF DATA CENTER CHILLED WATER COOLING SYSTEM ACCORDING TO ANNUAL POWER CONSUMPTION CRITERION

Piotr Kowalski[1], Mieczysław Porowski[2]

[1]Poznan University of Technology
Piotrowo 5
60-695 Poznan, Poland
piotr.kowalski@emerson.com

[2]Poznan University of Technology
Piotrowo 5
60-695 Poznan, Poland
mieczyslaw.porowski@put.poznan.pl

ABSTRACT

The paper presents optimization model of the chilled water based data center cooling system. The optimization procedure includes system technological and mathematical model, limiting conditions and optimization criterion, which in this case is annual power consumption minimum. The cooling system model is defined by constant parameters and decision variables and consists of aircooled chiller, independent external freecooling heat exchanger (drycooler), computer room air handling unit (CRAH) and constant flow chilled water system with circulation pump. Influence of the server racks architecture (open or closed rack aisle), server inlet temperature and chilled water regime on the annual power consumption of the cooling system has been shown. Case study calculation based on the described model has been presented, including optimum variant designation and power usage effectiveness ratio of mechanical system calculation.

KEYWORDS

data center, efficiency, optimization, precision cooling, chilled water

NOMENCLATURE

c_p – specific heat, [kJ/kgK]
CRAH – computer room air handler
E_{CR} – CRAH unit fan power consumption, [kWh]
E_{CWP} – chilled water pump power consumption, [kWh]
E_{DX} – compressor power consumption, [kWh]
E_{FC} – freecooling coil fan power consumption, [kWh]
E_{IT} – server power consumption, [kWh]
E_K – condenser fan power consumption, [kWh]
E_{TOT} – cooling system power consumption, [kWh]
h_{CRS}, h_{CRR} – CRAH unit supply specific enthalpy, [kJ/kg]
h_{KA}, h_{Ke} – condensing coil supply/return specific enthalpy, [kJ/kg]
h_{FA}, h_{Fe} – freecooling coil supply/return specific enthalpy, [kJ/kg]
l – hydraulic system length, [m]
p – absolute air pressure, [Pa]
P_{CR} – CRAH unit fan power input, [kW]
P_{CWP} – chilled water pump power input, [kW]
P_{DX} – compressor power input, [kW]
P_{FC} – freecooling coil fan power input, [kW]
P_{IT} – IT systems power input, [kW]
P_K – condenser fan power input, [kW]
P_{TOT} – mechanical systems total power input, [kW]
PUE(m) – power usage effectiveness of mechanical systems, [-]
Q_{CW} – chilled water cooling capacity, [kW]
Q_E – evaporator cooling capacity, [kW]
Q_{FC} – freecooling coil cooling capacity, [kW]
Q_{IT} – server room cooling load, [kW]
Q_K – condenser capacity, [kW]
R – pressure loss coefficient, [Pa/m]
R_p – gas constant of air, 287,05 [J/kgK]
t_A – ambient temperature, [°C]
t_{Amax} – max amb. temperature for a given profile, [°C]
t_{Amin} – min amb. temperature for a given profile, [°C]
t_{CHR} – condensing coil return air temperature, [°C]

t_{CHS} – condensing coil supply air temperature, [°C]
t_{CRR} – CRAH unit return air temperature, [°C]
t_{CRS} – CRAH unit supply air temperature, [°C]
t_{CWE} – chilled water temperature before evaporator, [°C]
t_{CWR} – chilled water return temperature, [°C]
t_{CWS} – chilled water supply temperature, [°C]
t_E – evaporating temperature, [°C]
t_{FA}, t_{Fe} – freecooling coil supply/ return air temperature, [°C]
$t_{FC,0\%}$ – freecooling start ambient temperature, [°C]
$t_{FC,100\%}$ – full freecooling ambient temperature, [°C]
t_{FCR} – freecooling coil return air temperature, [°C]
t_{FCS} – freecooling coil supply air temperature, [°C]
t_{KA}, t_{Ke} – condensing coil supply/return air temperature, [°C]
T_E – evaporating temperature, [K]
T_K – condensing temperature, [K]
V_{CR} – CRAH unit airflow volume rate, [m^3/s]
V_{CW} – chilled water flow volume rate, [m^3/s]
V_{FC} – freecooling coil airflow volume rate, [m^3/s]
V_K – condensing coil airflow volume rate, [m^3/s]
Δp_{CR} – CRAH unit pressure drop, [kPa]
Δp_{CWP} – chilled water pump available pressure, [kPa]
Δp_E – evaporator pressure drop, [kPa]
Δp_{FC} – freecooling coil pressure drop, [kPa]
Δp_p – hydraulic system pressure drop, [kPa]
ΔQ_{FC} – freecooling coil capacity difference, [kW/°C]
Δt_{CR} – CRAH unit air temperature difference, [°C]
Δt_E – evaporating and chilled water supply temp. difference, [°C]
Δt_{EXV} – min. evaporating and condensing temp. difference, [°C]
$\Delta t_{FC,100\%}$ – 100% freecooling capacity temp. difference, [°C]
Δt_{IT} – server air temperature difference, [°C]
Δt_K – condensing and ambient temp. difference, [°C]
$\Delta\tau$ – duration of a given ambient temperature, [h]
η_{CR} – CRAH unit fan efficiency, [%]
η_{CWP} – chilled water pump efficiency, [%]
η_{DX} – compressor efficiency, [%]
η_{FC} – freecooling coil fan efficiency, [%]
η_K – condenser fan efficiency, [%]
ρ_{CRS} – mass density, CRAH unit supply air temperature [kg/m^3]

1 INTRODUCTION

Data Center cooling is one of the most energy consuming mechanical systems available. There are several factors that contribute to high power demand including continuous operation independently of ambient conditions (cooling load depends mostly on IT equipment and external heat gains have minor impact on it), high expenditures on cooling and movement of load carriers (especially air and water) and constant growth of heat gains and density of servers. Power consumption of mechanical systems has large impact on total cost of ownership and carbon footprint of data center class buildings, so many different researches to minimize this impact have been carried and the problem is still valid.

Application of optimization methods to design chilled water based HVAC systems have been presented by Lu (Lu, Cai, Chai, 2005; Lu, Cai, Xie, 2005), while Porowski (Porowski, 2011) presented composite strategy to select optimum ventilation and air-conditioning system based on energy consumption criterion. Research on data center cooling system optimization have been carried by i.e. Shah et al., (Breen, Walsh, Shah, 2010; Breen, Walsh, Punch, 2010; Breen, Walsh, Punch, Shah, Kumari, 2010) Iyengar et al. (Iyengar, 2009) and Demetriou et al. (Demetriou, 2011), who presented holistic, analytic models taking into account the heat flow from servers to ambient air with a watercooled chiller and cooling tower as a cooling source.

The subject of presented paper is optimization model of the data center chilled water based precision cooling system with external, aircooled chiller and additional freecooling heat exchanger (water economizer) as a mechanical cooling source.

2 PROBLEM FORMULATION

Data center cooling system optimization problem includes formulation of the system model (including constant parameters and decision variables), definition of the limiting conditions and optimization criterion. Simplified technological scheme of the analyzed system is shown on Fig. 1.

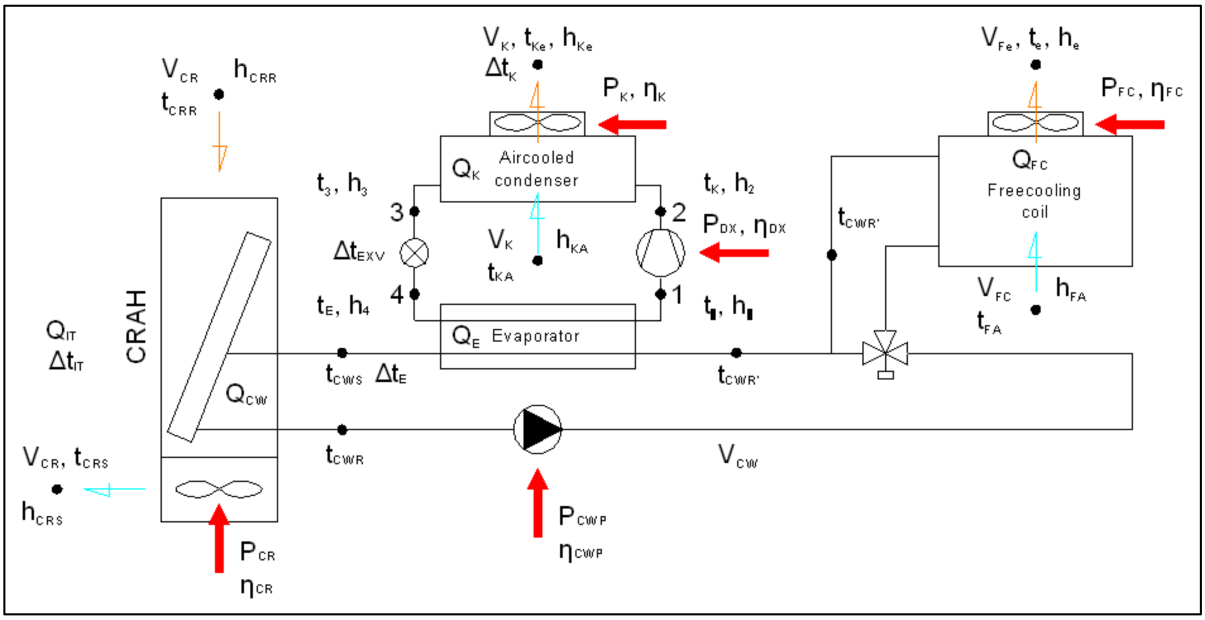

Figure 1. Technological scheme of the chilled water based data center cooling system

The data center cooling system model is described by the constant parameters and decision variables. Constant parameters, by definition, remain constant throughout optimization procedure (in general they can be a function of time). Decision variables, by definition, are changing throughout optimization procedure. Cooling system model described in this paper has following constant parameters: system structure, thermo-dynamical parameters of load carriers (air, water, refrigerant), ambient temperature distribution profile, server room heat load, mechanical efficiency of various components (pump, fans, compressor), difference between chilled water supply and evaporating temperature, minimum difference between refrigerant evaporating and condensing temperature, difference between ambient and refrigerant condensing temperature, length of the chilled water piping.

There are two types of decision variables used in the described model:

- incommensurable variables - server rack architecture. There are two variables of this kind included in the model: closed aisle architecture (separation of the hot and cold aisle, cooling system control according to supply air temperature, mass flow of the CRAH air as a function of server temperature rise) or open aisle architecture (no cold/hot aisle separation, cooling system control according to the return air temperature and CRAH unit airflow 30% higher compared to closed aisle architecture),
- measurable variables: server air temperature raise, supply and return chilled water temperatures (thus chilled water mass flow), freecooling coil operation starting ambient temperature, temperature difference to reach full required capacity of the freecooling coil.

There are following limiting conditions used in the optimization procedure: CRAH unit return air temperature (for open aisle option) or server inlet air temperature, thus supply air temperature range according to ASHRAE 2011 A1 class recommended envelope – for closed aisle architecture.

The optimization criterion used in the described model was minimum annual electrical power consumption of the cooling system according to the below account:

$$E_{TOT} = \min\left(E_{TOT1}, \ldots E_{TOTj}, \ldots E_{TOTj\,max}\right) \tag{1}$$

3 MODEL ASSUMPTIONS AND SOLVING PROCEDURE

For each of the components of the cooling system model (CRAH, freecooling coil, water chiller) balance equations have been formulated including power input of the components, i.e. cooling capacity delivered by the chilled water system includes the additional heat dissipated by the CRAH unit fans according to the equation:

$$Q_{CW} = Q_{IT} + P_{CR}\left(1 - \eta_{CR}\right) \tag{2}$$

CRAH unit fan power input depends on the volume flow of the server room air, which further depends on the mass flow and supply air temperature. It has been calculated as a function of the reference values of the fan power input and volume airflow according to the following equations:

$$P_{CR} = P_{CR\,ref}\left(\frac{\dot{V}_{CR}}{\dot{V}_{CR\,ref}}\right)^3 \tag{3}$$

$$\dot{m}_{CR} = \frac{Q_{IT}}{C_P\left(t_{CRR} - t_{CRS}\right)} \tag{4}$$

$$\dot{V}_{CR} = \frac{\dot{m}_{CR}}{\rho_{CRS}} \tag{5}$$

$$\rho_{CRS} = \frac{p}{R_p T_{CRS}} \tag{6}$$

Cooling capacity of the external freecooling coil has been calculated in a simplified way, including decision variables $\Delta t_{100\%FC}$ and $t_{A,0\%FC}$ according to the equations:

$$t_{FC,100\%} = \frac{t_{CWS} + t_{CWR}}{2} - \Delta t_{FC,100\%} \tag{7}$$

$$Q_{FC} = \begin{cases} 0, \;\; t_A \geq t_{A,0\%FC} \\ Q_{CW} - \left(t_A - t_{FC,100\%}\right)\Delta Q_{CF}, \;\; t_{FC,0\%} > t_A > t_{FC,100\%} \\ Q_{FC} = Q_{CW}, \;\; t_A \leq t_{FC,100\%} \end{cases} \tag{8}$$

$$\Delta Q_{FC} = \frac{Q_{CW}}{t_{FC,0\%} - t_{FC,\,100\%}} \tag{9}$$

Power input of the freecooling coil fan has been calculated using equations (3,4,5,6) described for the CRAH unit, with the assumption of constant ambient temperature raise across the coil.

Cooling load of the evaporator has been calculated according to the equation:

$$Q_E=Q_{CW}-Q_{FC} \tag{10}$$

Heatload of the aircooled condenser includes power input of the compressor and has been calculated according to the equation:

$$Q_K=Q_E+P_{DX} \tag{11}$$

Power input of the compressor has been calculated according to following equations:

$$P_{DX}=\frac{Q_E}{COP_{DX}} \tag{12}$$

$$COP_{DX}=\left(\frac{T_E}{T_K-T_E}\right)\eta_{DX} \tag{13}$$

$$t_E=t_{CWS}-\Delta t_E \tag{14}$$

$$t_K=\begin{cases} t_A+\Delta t_K, & (t_K-t_E)\geq\Delta t_{EXV} \\ t_E+\Delta t_{EXV}, & (t_K-t_E)<\Delta t_{EXV} \end{cases} \tag{15}$$

where η_{DX}, Δt_{EXV}, Δt_K are decision variables based upon chiller manufacturer data.

Power input of the aircooled condenser fan has been calculated using equations (3, 4, 5, 6) with the assumption of constant temperature rise of the ambient air. Power consumption of the circulating pump has been calculated using following equations:

$$P_{CWP}=\frac{\dot{V}_{CW}\Delta p_{CWP}}{\eta_{CWP}} \tag{16}$$

$$\Delta p_{CWP}=\Delta p_P+\Delta p_{CR}+\Delta p_{FC}+\Delta p_E \tag{17}$$

$$\Delta p_P=1{,}3Rl \tag{18}$$

with the simplifying assumption, that local pressure drop is 30% of the linear. Pressure drop of the CRAH unit, freecooling coil and evaporator has been calculated using below equations:

$$\Delta p_{CR}=\Delta p_{CR\ ref}\left(\frac{\dot{V}_{CW}}{\dot{V}_{CW\ ref}}\right)^2 \tag{19}$$

$$\Delta p_{FC}=\Delta p_{FC\ ref}\left(\frac{\dot{V}_{CW}}{\dot{V}_{CW\ ref}}\right)^2 \tag{20}$$

$$\Delta p_E=\Delta p_{E\ ref}\left(\frac{\dot{V}_{CW}}{\dot{V}_{CW\ ref}}\right)^2 \tag{21}$$

where values of reference pressure drops have been taken from manufacturers datasheets.

To find optimum mix of the design parameters according to the given criterion a complete review optimization method was used. Starting point of the method was creation of the mathematical models of the particular components of the cooling system, which influence the total power consumption (CRAH, chiller, freecooling coil, hydraulic system and pump). For each of the components system of the equations for the load carriers and energy balance were created. After applying the limiting conditions, a set of permissible variants was formulated, including the mix of design parameters (aisle architecture, chilled water supply and return temperatures, supply and return CRAH unit air temperatures).

For each of the permissible variants, according to the ambient air temperature profile in the $\langle t_{A,min}, t_{A,max} \rangle$ range with a step of 1°C and assigned duration (time) of each temperature, power consumption of the particular components has been calculated according to the following accounts:

$$E_{TOT}(t_A)=E_{CR}(t_A)+E_{FC}(t_A)+ +E_{DX}(t_A)+E_K(t_A)+E_{CWP}(t_A) \quad (22)$$

$$E_{CR}(t_A)=P_{CR}\Delta\tau \quad (23)$$

$$E_{FC}(t_A)=P_{FC}(t_A)\Delta\tau \quad (24)$$

$$E_{DX}(t_A)=P_{DX}(t_A)\Delta\tau \quad (25)$$

$$E_K(t_A)=P_K(t_A)\Delta\tau \quad (26)$$

$$E_{CWP}(t_A)=P_{CWP}\Delta\tau \quad (27)$$

$$E_{TOT}=\sum_{t_{A\,min}}^{t_{A\,max}} E_{TOT}(t_A) \quad (28)$$

For each of the permissible variants power usage effectiveness ratio of the mechanical systems has been calculated according to following equations:

$$PUE(m)=\frac{E_{TOT}+E_{IT}}{E_{IT}} \quad (29)$$

$$E_{IT}=8760P_{IT} \quad (30)$$

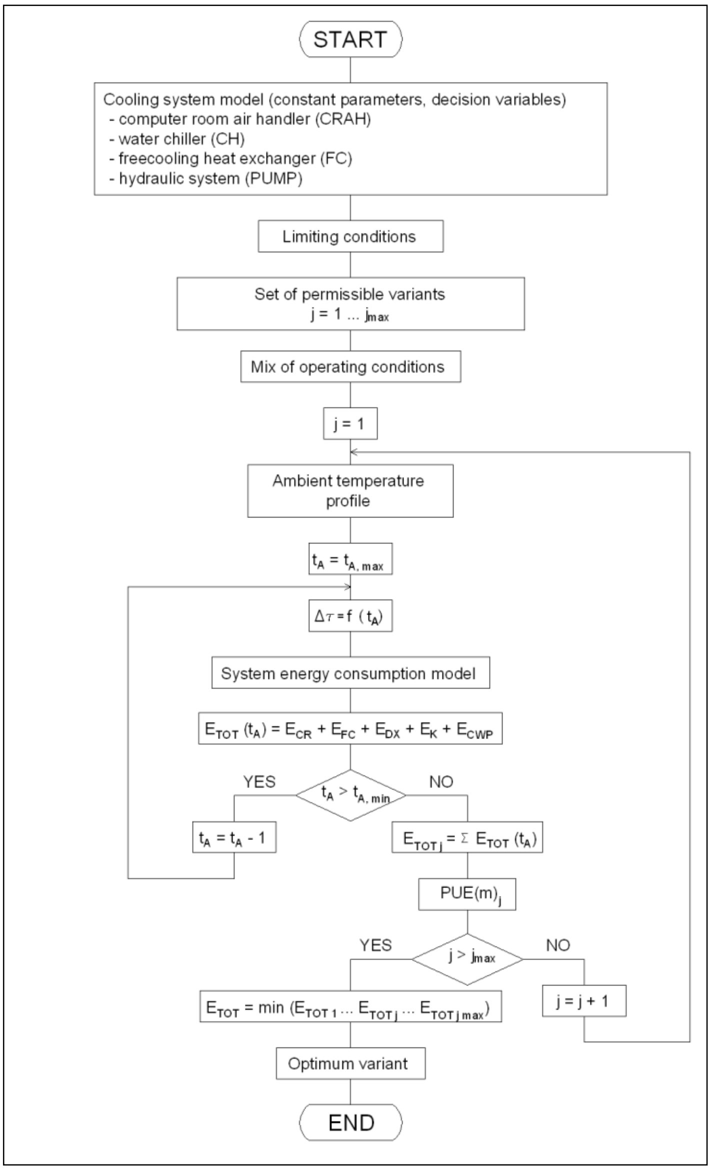

Figure 2. General algorithm of the optimization procedure

Optimum variant according to minimum annual power consumption criterion (1) has been determined. General algorithm of the optimization procedure is shown on figure 2.

4 CASE STUDY INPUT DATA AND RESULTS

The permissible variants (mix of decision variables and limiting conditions) calculated for the case study purposes include:

- open aisle room architecture: chilled water regime range t_{CWS}/t_{CWR}= (5-10/13-15)°C, CRAH unit return temperature t_{CRR}=24°C, CRAH unit temperature difference Δt_{CR}= 11,5°C,
- closed aisle room architecture: chilled water regime range t_{CWS}/t_{CWR}= (12-21/20-26)°C, temperature of the cold aisle (server inler) according to ASHRAE 2011 A1 Data Center class recommended range (A. T. C. TC 9.9., 2011), CRAH unit temperature difference Δt_{CR}= 15°C.

There have been 24 permissible variants calculated. List of decision variables of the selected five calculations for the case study example is shown in Table 1.

Table 1. Case study calculation selected input data

Variant No.	Architecture	Q_{IT}	t_{CWS}	t_{CWR}	Δt_{CR}	$\Delta t_{FC0\%}$	$\Delta t_{FC100\%}$
[-]	[-]	[kW]	[°C]	[°C]	[°C]	[°C]	[°C]
1	open	200	8	13	11,5	1	12
2	open	200	8	15	11,5	1	12
3	closed	200	15	22	15	1	12
4	closed	200	16	24	15	1	12
5	closed	200	18	26	15	1	12

The case study calculation results of the selected permissible variants are shown in Table 2., which includes annual power consumption of the respective components of the modelled system, total annual power consumption of the system and PUE(m) ratio. Energy consumption of the optimum variant (No. 5) based on the minimum annual energy consumption criterion (variant No. 5) is 60% lower compared to most energy consuming system (variant No. 5). Figure 3. shows energy consumption split of the optimum calculated system, while Figure 4. shows energy consumption split of the maximum calculated variant (No. 1).

Table 2. Case study calculation selected results

t_{CWS}/t_{CWR}	E_{DX}	E_K	E_{FC}	E_{CWP}	E_{CR}	E_{TOT}	PUE(m)
[°C]	[kWh/year]	[kWh/year]	[kWh/year]	[kWh/year]	[kWh/year]	[kWh/year]	[-]
8/13	176020	26541	43332	50189	83134	379215	1,22
8/15	165007	23416	47376	22547	83861	342208	1,20
15/22	77413	10382	46793	22547	40514	197649	1,11
16/24	63051	7687	44699	16887	41108	173432	1,10
18/26	46337	5265	40973	16887	41958	151420	1,09

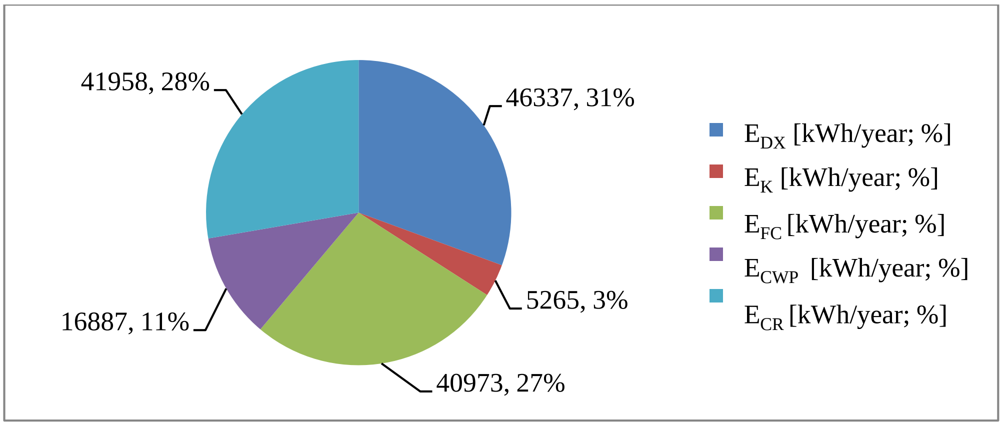

Figure 3. Cooling system energy consumption split (optimum variant – No. 5)

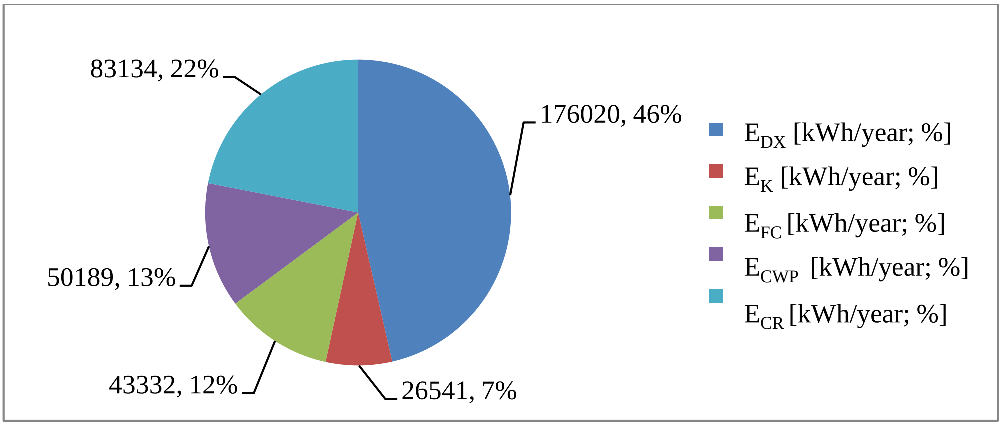

Figure 4. Cooling system energy consumption split (maximum variant – No. 1)

Figure 5. shows power input of the respective components of the cooling system, total power input and annual energy consumption of the optimum calculated system versus ambient temperature range.

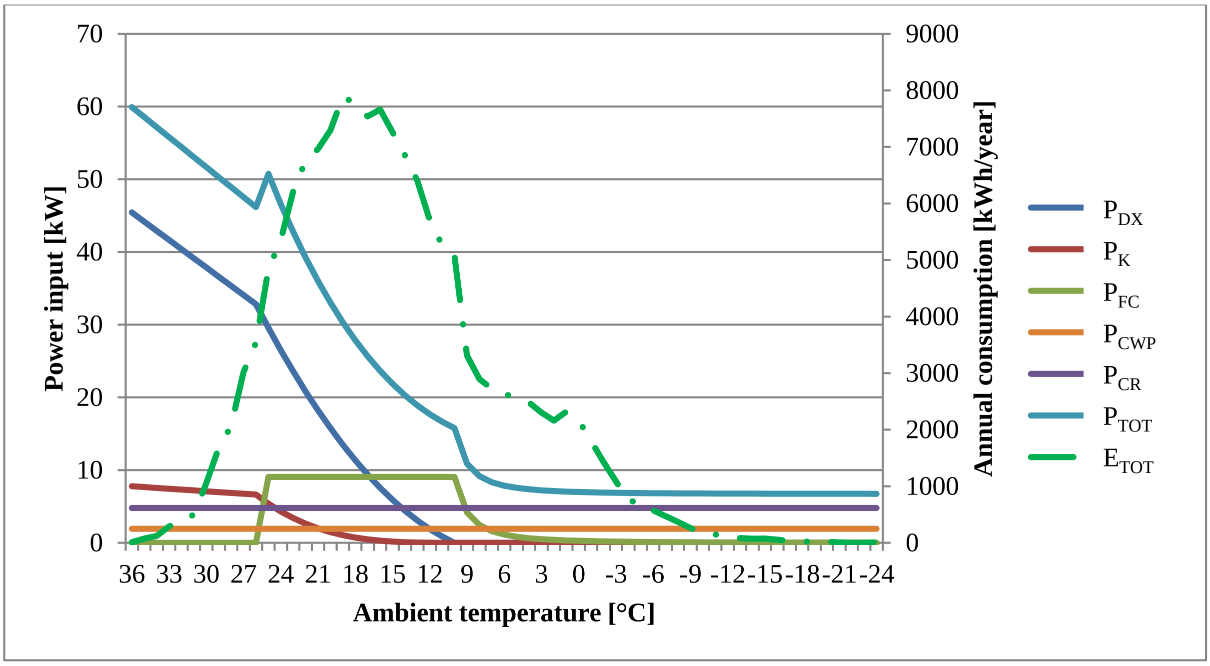

Figure 5. System components power input and total energy consumption versus ambient temperature –
– optimum variant – No. 5

5 CONCLUSIONS

The presented paper shows universal model of the optimization procedure and partial results of the calculations of an example case study. Complete review of the permissible variants is used as an optimization method. Based on a case study calculations is has been shown, that presented algorithm can provide an energy-optimized mix of design parameters of the data center cooling system, including server rack architecture (open/closed), chilled water temperature regime, chilled water mass flow and server inlet air temperature. Example calculation results presented in this paper are showing big potential of possible energy conservation depending on design parameters mix. For the case study calculation the difference between maximum and minimum calculated annual power consumption of the cooling system is 60%. Authors are conducting further research on the discussed topic, purpose of the research is further refinement of the cooling system particular components and introduction of more complex optimization procedures.

6 REFERENCES

A. T. C. TC 9.9. (2011). Thermal Guidelines for Data Processing Environments – Expanded Data Center Classes and Usage Guidance. *American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc*.

Breen, T. J.; Walsh, E. J.; Punch, J.; Shah, A. J.; Bash, C. E. (2010). From chip to cooling tower data center modeling: Part II Influence of chip temperature control philosophy. *IEEE.*

Breen, T. J.; Walsh, E. J.; Punch, J.; Shah, A. J.; Kumari, N.; Bash, C. E.; Lyon, G. (2010). From Chip to Cooling Tower Data Center Modeling: Validation of a Multi-Scale Energy Management Model. *13th IEEE ITHERM Conference*.

Breen, T. J.; Walsh, E. J. P. J.; Shah, A. J.; Bash. C. E. (2010). From chip to cooling tower data center modeling: Part I Influence of server inlet temperature and temperature rise across cabinet. *IEEE.*

Demetriou, D. W.; Khalifa, H. E.; Iyengar, M.; Schmidt, R. R. (2011). Development and experimental validation of a thermo-hydraulic model for data centers. *HVAC&R Research,* vol. 17:4, pp. 540-555.

Iyengar, M.; Schmidt, R. R. (2009). Analytical Modeling for Thermodynamic Characterization of Data Center Cooling Systems. *Journal of Electronic Packaging*, vol. 131, no. June, pp. 021009-1 - 021009-9.

Lu, L.; Cai, W.; Chai, Y. S.; Xie, L. (2005). Global optimization for overall HVAC systems - Part I problem formulation and analysis. *Energy Conversion & Management*, vol. 46, pp. 999-1014.

Lu, L.; Cai, W.; Xie, L.; Li, S.; Soh. C. (2005). HVAC system optimization - in-building section. *Energy and Buildings,* vol. 37, pp. 11-22.

Porowski, M. (2011). Strategia wyboru energooptymalnego systemu wentylacyjnego lub klimatyzacyjnego. Wydawnictwo Politechniki Poznańskiej, Poznań.