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(RESEARCH ARTICLE)

Finite time thermodynamic analysis of water cooled multi-spilit heat pipe system

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Abstract

Energy saving in data centers is increasingly important along with the rapidly developing IT industry. A integrated heat pipe air conditioning (AC) system which used chilled water unit and heat pipe heat exchanger (HPHE) was proposed.With the meteorological parameter of Changde city (31°53′N and 117°15′E) as the reference. The thermodynamic simulation software SIMULINK was employed in build the simulation model of the combined system. And it is compared with the traditional air-conditional system. The system include heat pipe evaporator cooling deliver unit（CDU）and the cold source. A time integral model for heat transfer of heat pipe evaporator is proposed. A heat pipe evaporator of a 12KW in the company was taking as an example. The results show that the time period of working fluid passing through the heat exchanger is about 6 seconds. When the dry degree is greater than 0.72, the heat transfer efficiency is low. When arranging the IT server, we should concentrate on the location below 1.3 meters. The CDU was treated as a condenser. Then, the cooling source is vapor compressor. The finite time thermodynamic and the exergy method were employed to analysis the heat exchanger and the compress processes.The results show that the former one has saved 26% of the energy. Then the using time and energy saving potential in major cities of China were investigated and studied.

Keywords: Thermodynamic simulation; Information data center (IDC) room; Heat pipe air-conditioning system; Exergy analysis; MATLAB/SIMULINK Software

1. Introduction

According to the great developing of the information technical, the IDC room is built and expanded. To maintain the normal operation of the run, the air-conditioning system is needed. The energy of the air - conditioning take about more than 40% of the total energy consumption of the room [1]. The traditional air-cooled method cannot fit the demand, so the water-cooled technique is employed in the design of the high efficiency cooling process. But for the safety of the room, the water pipe are prohibited to go into the IDC room, some are even not promised to the corridor of the room. On the other hand, the heat load of the room has characteristic: there is only heat load in room, the wet load is almost zero. And it need refrigerating all the year. The traditional air-conditioning system treat heat and wet load at the same time, then to keep the humidity of the room, the reheat component is added after the air-conditioning process. It waste a lot of energy. On the other hand, in the air distribution side, the traditional air conditioning system in the room, the cooled air is distributed by the wind pipe. The distance is long and there are cooling capacity loss in the process. To tackle these problems, the heat pipe air-conditioner is developed. The heat pipe has high heat transfer efficiency. The heat pipe system tackle the heat exhaust by the calculator sever locally, and combined the cold and heat channel closed technical. The power using efficiency (PUE) of the room can reach about 1.3[2].

They energy performance are studied by many scholars. In the heat pipe combined system aspect: A combined cooling solution is proposed to improve both thermal and energy performance for data centers with high heat density. Multistage heat pipe is introduced to make the internally cooled rack, which helps to illuminate the undesired mixing of hot and cold air, and makes a uniform distribution of indoor temperature [3]. An integrated cooling system for data

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centers which combines a heat pipe cooling cycle and a vapor compression cooling cycle. The operating mode of the system changes with the outdoor temperature. Key problems of the integrated system were solved such as the mix of the refrigerant and lubricant, the match of heat exchange areas and the durability of valves. A thermal equilibrium test was carried out to evaluate the system performance [4].In the measurement of the IDC room: Methodological approach to the energy efficiency optimization of high density data center, in a synergy with relevant performance analysis of corresponding case study. Related case study goal—to optimize energy efficiency to the level characterized by the power usage effectiveness nearly equal 1 (PUE ≈ 1), has been successfully realized, tracing an energy efficiency optimization road to a hyper energy efficient data center of the superb IT and energy performance [5]. In the influence of the environment aspect: A thermodynamic approach for evaluating energy performance (productivity) of information technology (IT) servers and data centers is presented. This approach is based on the first law efficiency to deliver energy performance metrics defined as the ratio of the useful work output (server utilization) to the total energy expanded to support the corresponding computational work. These energy performance metrics will facilitate proper energy evaluation and can be used as indicators to rank and classify IT systems and data centers regardless of their size, capacity or physical location [6]. Zou etal focused on the onsite test about self-adaptive capacity of a multi-spilit heat pipe system (MSHPS) in a real data center under 25%, 50%, 75% and 100% heating loads and various fan failures. The results show that the MSHPS abnormally operated under low heating loads, but it still met the cooling demands due to its superior self-adaptive capacity,but the refrigerant distribution characteristics of MSHPS is required to be further studied to optimize the structure of liquid pipes.

They simulation model is hard to be build, and the task of the experiment is large, to build a simple method to evaluate the heat-pipe combined system, a thermodynamic simulation software *SIMULINK* is employed then the energy and exergy efficiency will be calculated and compared with the traditional air-conditional system.

2. The physical model of the water cooled multi-spilit heat pipe system

2.1. The system model of water cooled multi-spilit heat pipe system

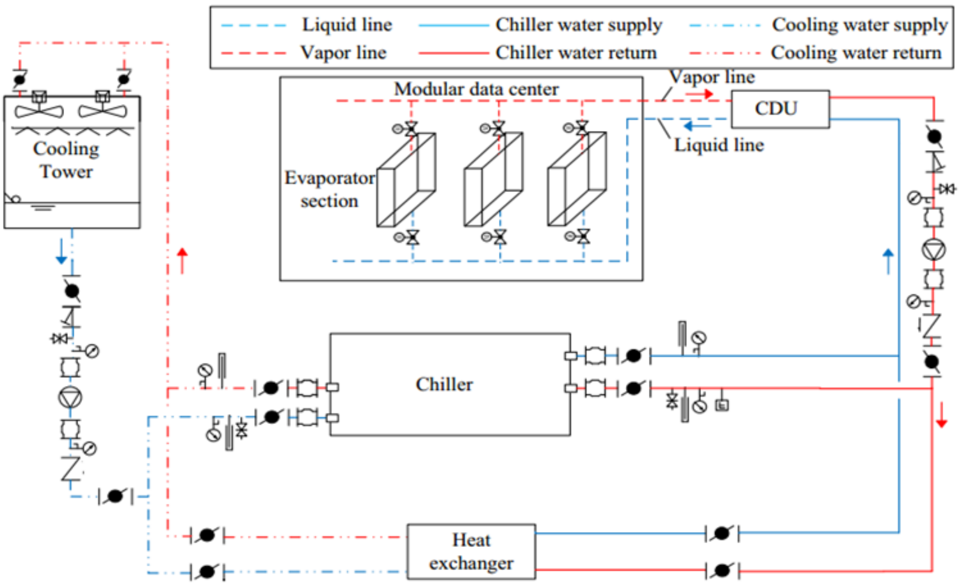

Figure 1 The backboard heat pipe air-conditioning system in the *IDC* room

Atypical heat pipe air-conditioning system can be seen in Fig. 1. The area of the IDC room is 384 m^2 , the indoor temperature demand is 24℃, and the humility is 50%. 72 backboard heat pipe air conditioners were designed in the system. The outdoor temperature is 35.4℃. The temperature of the chiller water can improved to 12/17℃. The exergy efficiency of the compressor is improved. And the chilling water machine can be shut down when the outdoor temperature is below 10℃ in winter, the free cooling mode can be used [7].

The average outdoor temperature of the room is showed in the Table. 1.

Table 1 The average outdoor temperature of the **IDC** room

Temperature gap	≥30	≥20, ＜30	≥10, ＜20	≥0, ＜10	＜0
Temperature distribute efficiency	11.5%	33.3%	27.1%	26.2%	1.9%

2.2. The Physical model of heat pipe evaporator

The 12 kW heat pipe heat exchanger is used as the object. The microchannel heat exchanger is used. The height of the heat exchanger is 2 m , and the width is 0.6 m . The wind rate of the fan is 2000 m^3/s , and the refrigerant is R134A.

2.3. The Physical model of CDU

The parameter of the CDU is show in table 2.

Table 2 The parameter of the CDU

Parameter	Side of refrigerant	Water side
Mass flow (kg/h)	187.6	1714
Pressure drop (kPa)	2100	-
Working pressure(bar)	9.08	0.1
Temperature(°C)	20	14/19
Tracheal branch(mm)	19	
Liquid pipe branch(mm)	16	
The heat transfer area(m^3)	0.9	
CDU outline size (high) ×(wide) ×(thick)	0.525 ×0.111×0.12	
Installation height(m)	1.8	

The Physical model of the cooling source

The cooling capacity of the compressor is 130KW;the condenser temperate is 35℃;the temperate of the cold water is 12℃。

3. The mathematical model of the heat-pipe air-conditioning system

3.1. The mathematical model of the evaporator of the heat-pipe

Referring to the micro channel heat exchanger model, based on the new Gungor-Winterton correlation formula[8], the hourly simulation model of the heat pipe evaporator is established.

$$\dot{h}_{tp} = 0.0455(S + \dot{F})R_{e1}P_r^{-1}We_{lo}^{-0.17}\frac{\lambda_1}{D_h} \qquad (1)$$

$$S = 1 + 4400BL^{0.86}Fr_{lo}^{-0.22} \qquad (2)$$

$$\dot{F} = 1.12\left(\frac{\dot{x}}{1-\dot{x}}\right)^{0.86}\left(\frac{\rho_l}{\rho_g}\right)^{0.41} \quad (3)$$

In the formula: $\dot{h}_{tp}$: Hourly two current heat transfer coefficient; R_{e1}: Liquid Reynolds number; P_r :Liquid Prandt number; BL :Boiling number; We_{lo}:Liquid Weber number; λ_l :Liquid thermal conductivity; D_h :Heat pipe diameter; $\dot{x}$:Dry degree; ρ_l :Liquid density; ρ_g :Gaseous density.

The total heat exchange:

$$Q = m \cdot \int_0^1 \dot{h}_{tp} d_x \quad (4)$$

The efficiency of the system:

$$\eta = \frac{Q}{T} \quad (5)$$

The T in the formula is a cycle, that is, the heat transfer time of the working fluid in the heat pipe evaporator:

$$T = \frac{L_{ev}}{v} \quad (6)$$

Medium: V is the velocity of the working fluid:

$$v = \frac{m}{\pi \cdot (\frac{D}{2})^2} \quad (7)$$

Medium: m is mass flow.

The formula (1) - (7) constitutes a heat transfer model for heat pipe microchannel heat exchangers[9].

3.2. The mathematical model of the CDU

See in figure 2.The heat exchange of the CDU is equal to the enthalpy drop from the state point 7 to the state point 8[10].

$$\dot{Q}_{cond} = \dot{H}_7 - \dot{H}_8 \quad (8)$$

In the formula, $\dot{Q}_{cond}$ is the heat transfer for CDU by time. The $\dot{H}_7$ 、 $\dot{H}_8$ are enthalpy value for the state point 7 and 8.

According to the designed evaporation and condensation temperature, the state of T-S diagram 2, 7 and 8 points can be determined.

Exergy loss of the CDU:

$$\dot{I}_{cond} = \dot{m}_{ref} \cdot \dot{i}_{cond} \quad (9)$$

And:

$$\dot{i}_{cond} = \dot{m}_{ref} \cdot (\dot{h}_7 - \dot{h}_8) - \dot{T}_0 \cdot (\dot{s}_7 - \dot{s}_8) \qquad (10)$$

In the formula: $\dot{i}_{cond}$: the hourly loss of refrigerant for unit quality, $\dot{I}_{cond}$: the loss of working hours and $\dot{m}_{ref}$:the mass flow rate of working medium. $\dot{T}_0$:The entropy of hourly ambient temperature and $\dot{s}_7$ 、 $\dot{s}_8$:working hourly entropy. The formula (8) - (10) constitutes a finite time thermodynamic model of CDU.

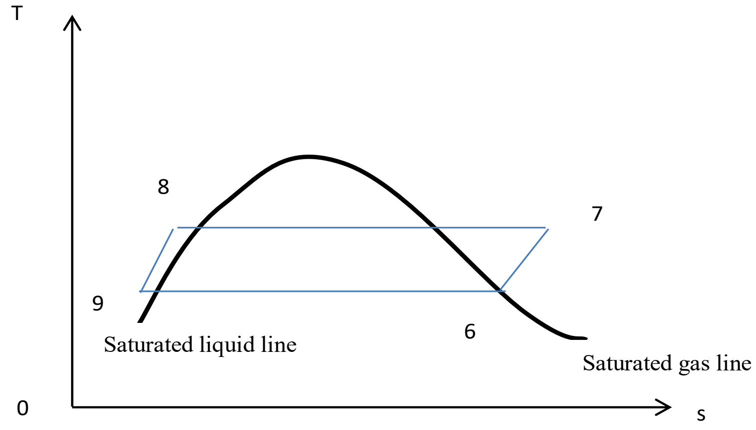

Figure 2 The T-S diagram of the evaporator of the heat pipe

3.3. The mathematical model of the Cooling source

3.3.1. The mathmatical model of the compressor

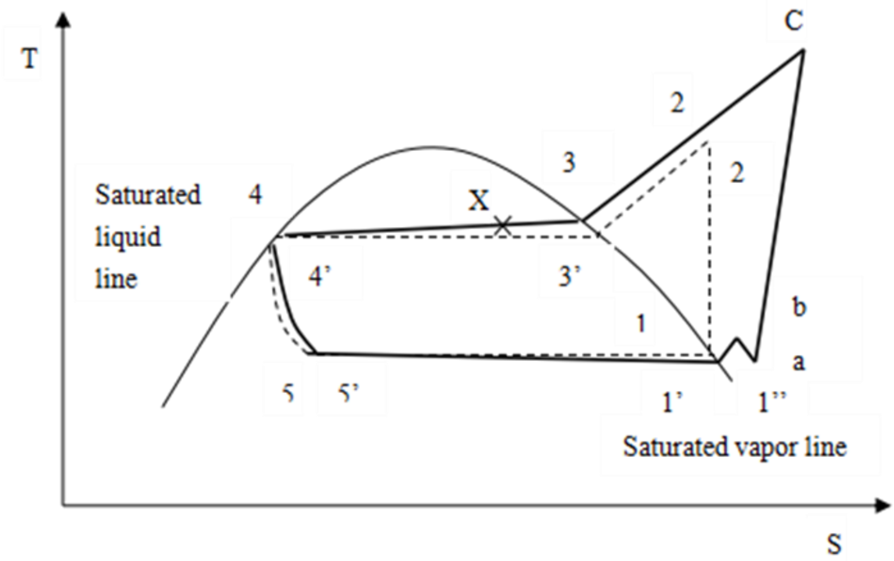

Figure 3 The T-S diagram of the cooling source

The mass flow of the refrigerant is m_{ref}, it comes to the compressor at the state 1. And the state of the outletis state 3. Because of the irreversible process of the compress. There are exergy loss and entropy generations. The exergy balance equation is as follows[11]:

$$\dot{e}_1 + \dot{w} = \dot{e}_2 + \dot{i}_{comp} \qquad (11)$$

The exergy input by the compressor is:

$$\dot{e}_{in,comp} = \dot{w} = \dot{m}_{ref}(\dot{h}_2 - \dot{h}_1) \qquad (12)$$

The entropy generation of the irreversible process is:

$$\dot{s}_{gen,12} = \dot{s}_2 - \dot{s}_1 \qquad (13)$$

Then the exergy loss of the compressor is:

$$\dot{I}_{comp} = \dot{e}_1 - \dot{e}_2 + \dot{w} = \dot{m}_{ref} T_0 (\dot{s}_2 - \dot{s}_1) \qquad (14)$$

In the ideal reversible process of the compress, the power consumption is :

$$\dot{w}_c = \dot{m}_{ref} \cdot (\dot{h}_{2'} - \dot{h}_1) \qquad (15)$$

The parameter of the the real outlet state 3 of the compressor can be defined by the formula:

$$\dot{\eta}_{is,c} = \frac{\dot{h}_{2'} - \dot{h}_1}{\dot{h}_2 - \dot{h}_1} \qquad (16)$$

In the formula, is the compress efficiency, it is decentralized compressor, suppose it is 0.85.

$$\dot{h}_2 = \frac{\dot{h}_{2'} - \dot{h}_1}{\dot{\eta}} + \dot{h}_1 \qquad (17)$$

3.3.2. The mathematical model of the condensor

The heat of condensation is equal to the enthalpy drop from the 2 state to the state point 4.

$$\dot{Q}_{cond} = \dot{H}_2 - \dot{H}_4 \qquad (18)$$

According to the T-S diagram, the state of the 4 point can be determined.

$$\dot{I}_{cond} = \dot{m}_{ref} \cdot \dot{i}_{cond} \qquad (19)$$

And:

$$\dot{i}_{cond} = \dot{m}_{ref}\cdot[(\dot{h}_2-\dot{h}_4)-\dot{T}_0\cdot(\dot{s}_2-\dot{s}_4)] \quad (20)$$

3.3.3. *The mathematical model of the valve*

As shown in Fig. 2, the throttling process line is 4~5. The exergy loss of throttle valve is from exergy balance equation and adiabatic throttling equation.

$$\dot{I}_{\exp} = \dot{T}_0\cdot \dot{m}_{ref}(\dot{s}_5-\dot{s}_4) \quad (21)$$

3.3.4. *The mathematical model of evaporator*

In the evaporator, the circulating refrigerant absorbs heat from the low temperature heat source to transform into gaseous state and enter the compressor. For example, the process line 5~1 in Figure 1. When the cooling capacity is utilized, the exergy loss is estimated from the exergy balance equation [12].

$$\dot{I}_{ev} = \dot{m}_{ref}\cdot[(\dot{h}_5-\dot{h}_1)-\dot{T}_0\cdot(\dot{s}_5-\dot{s}_1)]+\dot{m}_{ev,w}\cdot[(\dot{h}_{evwo}-\dot{h}_{evwi})-\dot{T}_0\cdot(\dot{s}_{evwo}-\dot{s}_{evwi})] \quad (22)$$

The formula m_{evw} is the flow of cold water, h_{evwi} 、 h_{evwo} are the enthalpy of cold water for inlet and outlet, and h_{evwi} 、 h_{evwo} are the entropy of cold water import and export.

4. Simulation model of unit type water cooled multi continuous heat pipe system

4.1. The simulation model of the evaporator of the heat pipe

According to formula 3, the formula of F is calculated. A simulation model for solving F is established. The constant module 1.12 (constant) is multiplied by X divided by 1 (Constant1) minus the 0.86 th power of X and multiplied by the 0.41 th power (Math Function1)[13].

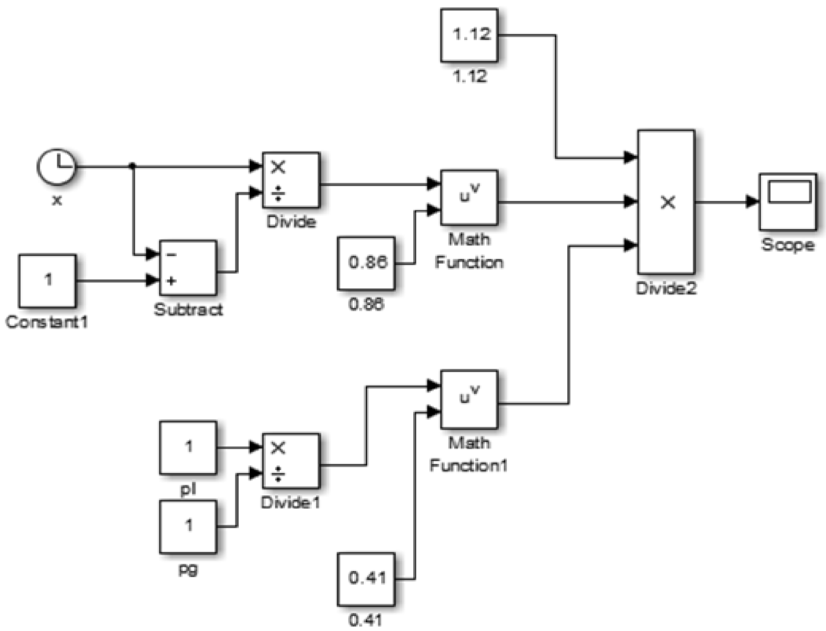

Figure 4 The simulation model of F value in heat transfer coefficient of two phase flow

Similarly, a simulation model of heat transfer coefficient and heat transfer can be established.

4.2. The simulation model of the CDU

According to the mathematical model of heat exchanger, a simulation model is established. According to the equivalent heat transfer between refrigerant and water side, the refrigerant side is a phase change process, and the heat transfer $\dot{Q}_t$ is latent heat of phase change (multiplied by refrigerant mass flow rate $\dot{m}_{ref}$. The heat transfer at the water side is the difference between the inlet and outlet temperature ($\dot{T}_{wo}-\dot{T}_{wi}$) times the specific heat (C_w) and the flow rate . Seen in Fig.5.

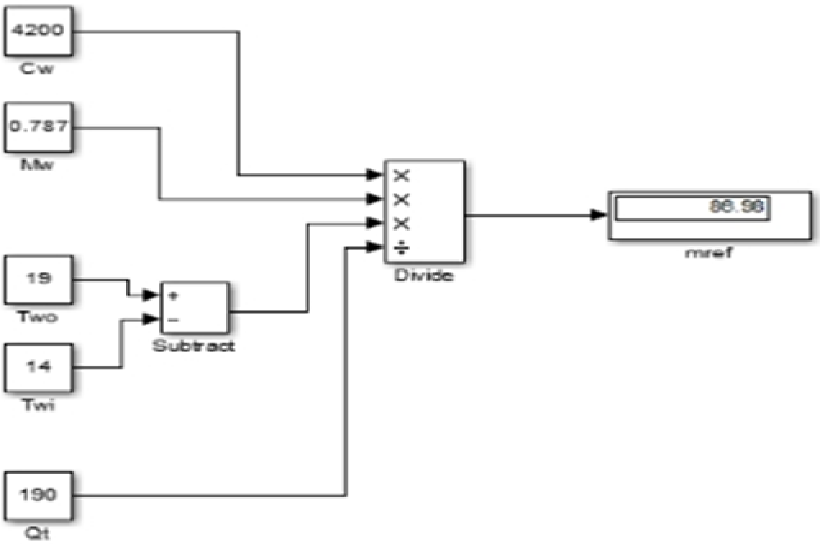

Figure 5 The simulation model for solving the flow of refrigerant in CDU

4.3. The simulation model of the compressor

According to the evaporating temperature, the entropy of the refrigerant at state 1 can be gotten. Then the entropy of the refrigerant at state 2 can be gotten; then according to the Te\Tc and compressor efficiency η, the differ of the ($\dot{h}_2-\dot{h}_1$) can be gotten. Then put the ($\dot{h}_2-\dot{h}_1$) sublet the accumulate of the initial temperature $\dot{T}_0$ multi ($\dot{s}_2-\dot{s}_1$), at last, let the calculation result multi the mass flow of the refrigerant m_{ref} . The exergy loss of the compressor can be gotten. See in Fig.6[14].

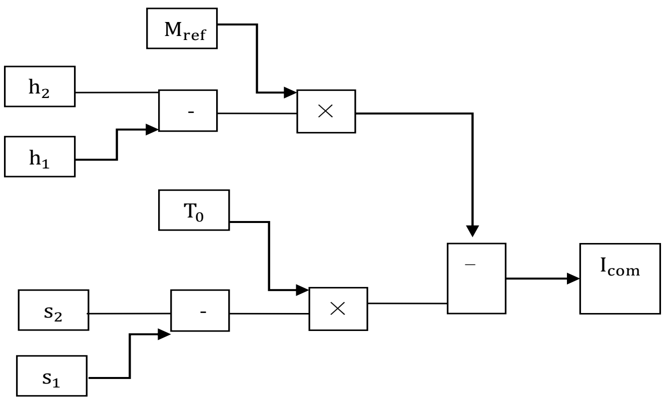

Figure 6 The simulation model of the condenser

4.4. The Simulation model of condenser

The condensing heat is equal to the enthalpy drop from state 3 to state 4.

According to the T-S diagram, the state at outlet of the condenser can be defined.

According to the condensing temperature Tc, the exergy loss of the condenser can be gotten. Then value of the simulation model $M_{ref} \cdot [(h_2 - h_4) - T_0 \cdot (s_2 - s_4)]$ can be calculated, see in Fig.7.

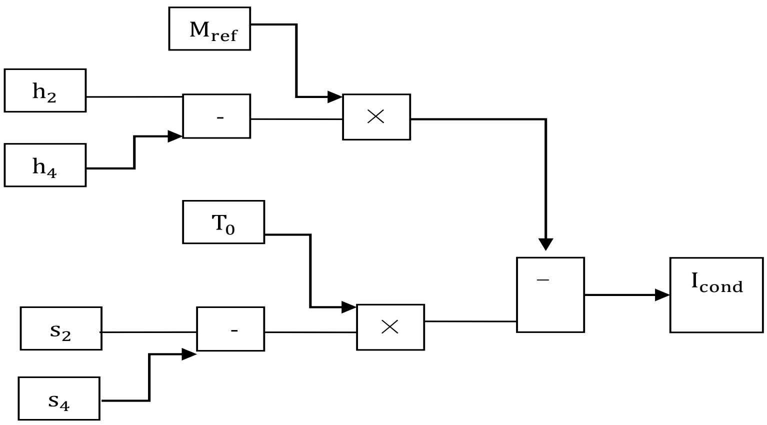

Figure 7 The simulation model of the condensor

4.5. The simulation model of expansion valve

Seen in figure 1, the throttle process is 4-5, due to the exergy balance equation and the adiabatic throttle process, the exergy loss of the valve $T_0 \cdot m_{ref} \cdot (s_4 - s_5)$ is got.

According to the condensing temperature the entropy at state 4 ,s4 can be gotten. Then the entropy at state 5 (s5) can be calculated by the evaporation temperature. Let s4 sublet s5, the results multi the base point temperature T0 and the mass flow of the refrigerant. The exergy loss of the expansion valve can be gotten. Seen in Fig.8.

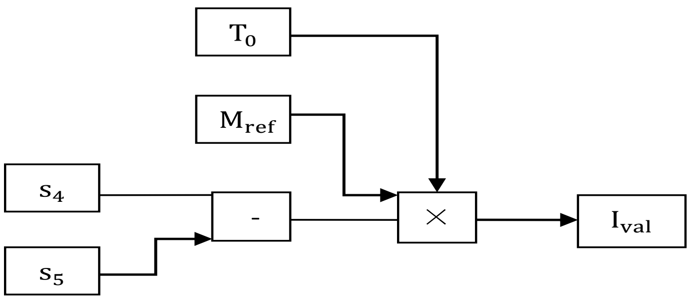

Figure 8 The simulation model of the valve

4.6. The simulation model of the evaporator

According to the inlet temperature and outlet ,the enthalpy and entropy of the inlet and outlet chilled water can be calculated. Then put the environment temperature T0 multi the , then let the differ of the enthalpy of inlet and outlet sublet, at last, put the result multi the mass flow of the chilled water, the exergy loss of the chilled water side can be gotten. Similarly, the exergy loss at refrigerant side can be gotten, then put the two add together, the exergy loss of the evaporator $m_{ev,w}[(h_{evo} - h_{evi}) - T_0(s_{evo} - s_{evi}) + m_{ref}[(h_5 - h_1) - T_0(s_5 - s_1)]$ is gotten. Seen in Fig .9.

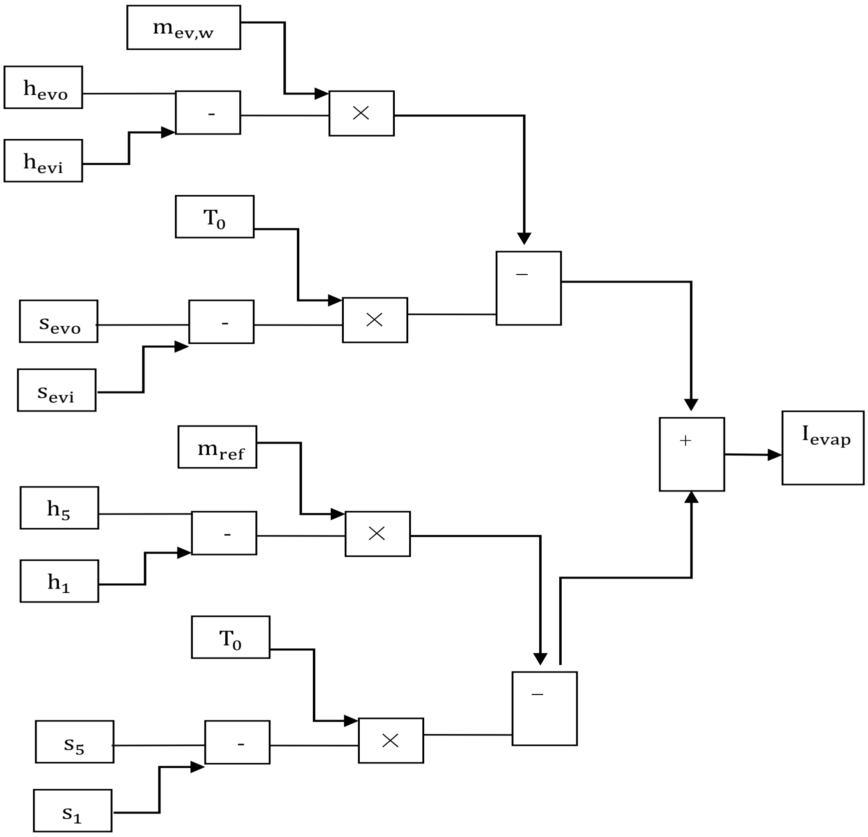

Figure 9 The simulation model of the condenser

5. Results and discussion

It can be seen from the relationship between the dry degree of the heat exchanger and the height and time of the heat exchanger that the heat transfer efficiency of the refrigerant is higher when the liquid refrigerant is many, and the heat transfer efficiency is lower when the dry degree is greater than 0.72. When arranging the IT server, it is advisable to concentrate on the location below 1.3 meters, seen in Fig.10.

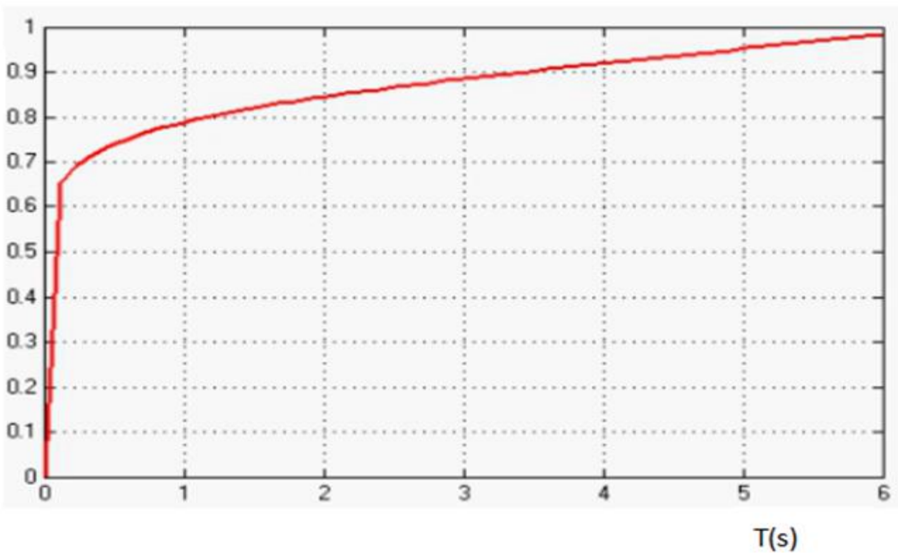

Figure 10 The x vs. the operating time in the evaporator

The calculation results show that heat transfer gradually slows down with time. CDU has a heat transfer of about 74KW after 6 seconds of heat transfer.Seen in Fig.11.

Figure 11 The total heat transfer vs.the heat transfer time of the heat pipe exchanger

The heat transfer efficiency of CDU is calculated by changing the flow rate of cold water. As shown in the diagram, the optimum flow rate of CDU is 0.82, and the corresponding heat transfer efficiency is 0.81.Seen in Fig.12

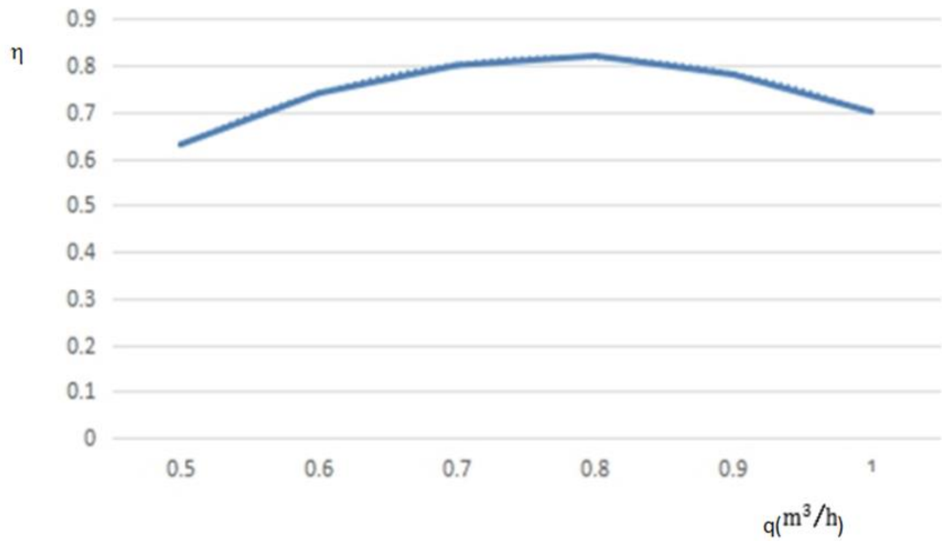

Figure 12 The η of the evaporator vs. the mass flow of the cooling water

The energy saving potential in major city of China is seen in Fig.13

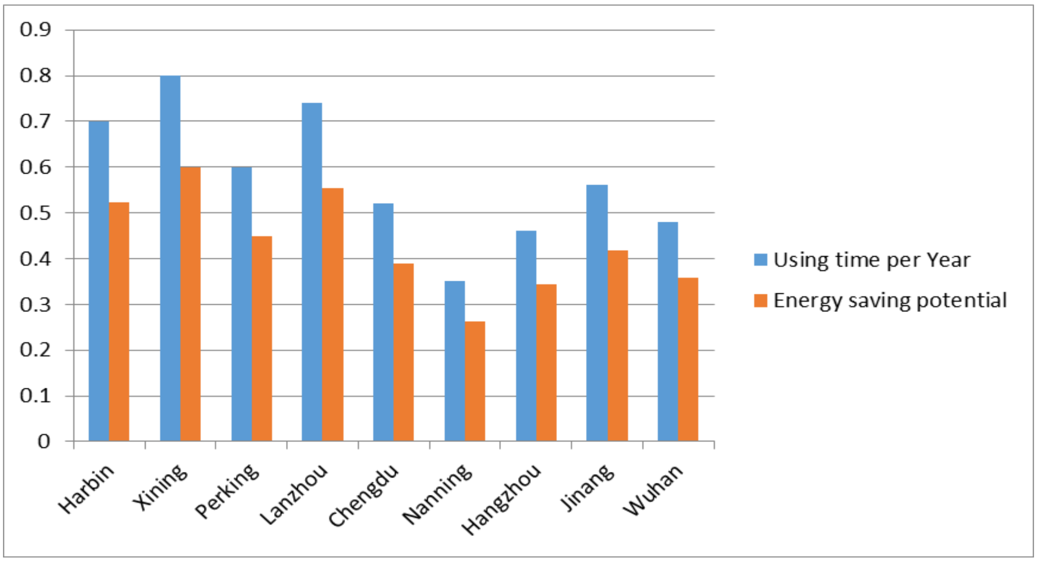

Figure 13 The energy saving potential in major city of China

6. Conclusion

In this paper, a time integration model for heat transfer in heat exchangers is established based on finite time thermodynamic analysis. The model can reflect the actual heat transfer process of heat exchanger. The model can be used to optimize the charging capacity of heat exchangers. And the model can reflect the relationship between heat transfer and heat exchanger height. Provide judgment and reference for the layout of IT equipment.

The hourly simulation model of CDU is simple and the simulation error is small. Compared with classical thermodynamic analysis, it can reflect the actual operation of the system. The simulation precision is high. Moreover, the model can provide reference for the design and optimization of CDU parameters. The model provides a new method for the simulation of thermal system.

Compliance with ethical standards

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Disclosure of conflict of interest

There are no conflicts of interest.

References

[1] Lingen CH, Fengrui S, and Chih W. (2004). Optimal allocation of heat-exchanger area for refrigeration and air-conditioning plants.Applied Energy,77(3) , 339-354.

[2] Hassan MAM. (2013). Investigation of performance of heat pipe as heat exchanger using alternative refrigerants. Journal of Energy Engineering, 139(1), 18-24.

[3] Jeon Sung-Taek and Cho Jin-Pyo. (2015). An experimental study on the performance of the separate type heat pipe in accordance with the refrigerant charge Journal of the Korea Academia-Industrial cooperation Society, 16(3), 1600-1604 .

[4] Naphon P, Thongkum D and Assadamongkol P. (2009). Heat pipe efficiency enhancement with refrigerant–nanoparticles mixtures. Energy Conversion and Management, 50(3), 772-776.

[5] L Y Zh, Y Y L, and X G. (2017). Experimental investigation and economic analysis of gravity heat pipe exchanger applied in communication base station Applied Energy, 194(15), 499-507.

[6] Chengchu Y, Wenxing S, Xianting L, etal. (2016). A seasonal cold storage system based on separate type heat pipe for sustainable building cooling. Renewable and Sustainable Energy Reviews, 58, 666-673.

[7] lMustafa AliErsöz, and AbdullahYıldız. (2016). Thermoeconomic analysis of thermosyphon heat pipes.Renewable Energy,85, 880-889.

[8] Chao D, LiJia Q and Yi L. (2017). Investigation on thermal design of a rack with the pulsating heat pipe for cooling CPUs. Applied Thermal Engineering, 110(5) , 390-398.

[9] Xingxing ZH, Jingchun SH, Wei H, etal. (2015). Comparative study of a novel liquid–vapour separator incorporated gravitational loop heat pipe against the conventional gravitational straight and loop heat pipes – Part II: Experimental testing and simulation model validation. Energy Conversion and Management, 93(15), 228-238.

[10] Hainan Zh, Shuangquan Sh, HongboX, etal. (2015). Numerical investigation on integrated system of mechanical refrigeration and thermosyphon for free cooling of data centers. International Journal of Refrigeration, 60, 9-18.

[11] Li L, Quan ZH, and YuebinY. (2017). Simulation of a micro channel separate heat pipe (MCSHP) under low heat flux and low mass flux Applied Thermal Engineering, 119(5) , 25-33.

[12] Tao D, Zhi guang H, Tian H, etal. (2016). Application of separated heat pipe system in data center cooling.Applied Thermal Engineering.109, Part A, 207-216.

[13] Tao D, Han wen C, Zhi guang H, etal. (2017). Experiment research on influence factors of the separated heat pipe system, especially the filling ratio and Freon types. Applied Thermal Engineering, 118, 357-364.

[14] Penglei ZH, Baolong W and Wei W. (2015). Heat recovery from Internet data centers for space heating based on an integrated air conditioner with thermosyphon Renewable Energy, 80, 396–406.

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