Intelligent Computer Based Analysis of the Heat-Pipe Air Conditioning System for Internet Data Centers

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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. The water cooled multi-split heat pipe system ($MSHPS$) consists of three parts: the end of the indoor heat pipe evaporator, the cooling capacity distribution unit (CDU) and the outdoor cold source. The system is based on the principle of heat pipe phase transition. Because the water in the system does not enter the computer room to ensure the safety of the computer room. And according to the layout of the cabinet, flexible multi joint matching design. It is with the meteorological parameter of Changde city (31°53′N and 117°15′E) as the reference. It has 72 backboard heat pipes and three CDU . The thermodynamic simulation software $CYCLEPAD$ was employed to build the simulation model of the combined system. And it is compared with the traditional air-conditional system. The result shows that The heat transfer cycle of the heat exchanger takes about 6 seconds, and its total heat transfer Q is about 77KJ,the optimum flow rate of CDU is 0.82, and the corresponding heat transfer efficiency is 0.81 kg/s .And the former one had save 26% of the total energy. Then the using time and energy saving potential in major cities of China were investigated and studied. The software is a easy and rapid method to build the simulation model. The proposed method is fit for analysis of the real pipe-net systems.*

Keywords: *Thermodynamic simulation; Internet data center room; Heat pipe air-conditioning system; Exergy analysis; Software $CYCLEPAD$*

According to the great developing of the information technical, the Internet data center (IDC) room is built and expanded. To maintained the normal operation of the IDC room, the air-conditioning system is needed. The energy of the air - conditioning take about more than 40% of the total energy consumption of the IDC room[1]. The traditional air-cooled method can not fit the demand with the rapid increase of the high density heat load, so the water-cooled technique is employed in the design of the high efficiency cooling process. But for the safety reason of the IDC room, the water pipe are prohibited to go into the IDC room, or even not promised to the corridor of the IDC room. On the other hand, The characteristic of the heat load of the IDC room is: there is only heat load in IDC room, the wet load is very few and almost near to zero. And it need refrigerating all the year. The traditional air-conditioning system cheat 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. So it waste a lot of energy. On the other hand, the cooled air is distributed by the wind pipe of the traditional air conditioning system in the IDC room. 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 system tackle the heat exhaust by the calculator sever locally, and combined the cold and heat channel closed technical, the PUE of the IDC 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. Mufti-stage 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 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 to 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].The literatures are not focus on the water cooled multi-split heat pipe system ($MSHPS$).

In the simulation tool aspect, $CYCLEPAD$ is a thermal cycle mapping and analysis software. It is very standard to draw the schematic diagram of the thermodynamic cycle, and can automatically calculate the cycle under the given cycle parameters, which is very convenient. The equations used in the calculation can also be customized[7].The thermodynamic model of regenerative cycle is established by using the Engineering Thermodynamics Simulation Software $CYCLEPAD$. The thermodynamic parameters of each state point under different regenerative steam pressure can be quickly determined by setting the working state of each component and the initial parameters of working medium in the model. The variation rules of regenerative steam pressure and the percentage of regenerative steam, evaporator work, turbine work and regenerative cycle efficiency were obtained, and the fast analysis of optimal regenerative hot spot of single stage regenerative cycle was realized[8].A time series model is used to describe the progress of circulating direct condensation heat recovery of the compound condensing process which is made of two water cooling condensing processes in series for a centrifugal chiller[9].

They above simulation models are complex, and the task of the experiment is large. To build a simple method to evaluate the $MSHPS$, a thermodynamic simulation software $CYCLEPAD$ is employed. Then the energy and exergy efficiency will be calculated and compared with the traditional air-conditional system.

1. The physical model of the heat-pipe air-conditioning system

1.1 The physical model ofcombinedheat pipe system

Atypical heat pipe air-conditioning system can be seen in Fig 1**Error! Reference source not found.**. 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 in/out of the cooling deliver unit (CDU) is 12/17℃. And the chilling water machine can be shut down when the out door temperature is below 10℃ in winter, the free cooling mode can be used[10].

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

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Tab. 1 *The average outdoor temperature of the IDC room*

Outdoor Temperature (℃)	≥30	≥20, ＜30	≥10, ＜20	≥0, ＜10	＜0
Annual percent(%)	11.5	33.3	27.1	26.2	1.9

From the data shows in the Tab 1**Error! Reference source not found.**, there are 28.2% of annual time that the free cooling mode can be used.

The backboard heat pipe evaporators were divided to 3 groups, each group has 24 back board heat pipe evaporators. See in Fig. 1.

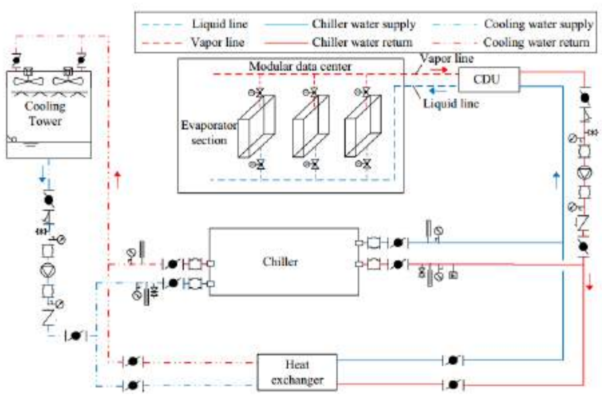

Fig. 2 *The backboard heat pipe air-conditioning system in the IDC room*

1.2 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/h, and the refrigerant is $R134A$.

1.3 The physical model of CDU

The parameter of the CDU is show in table 2.

Tab. 2 *The average outdoor temperature of the IDC room*

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(℃)	20	14/19
Tracheal branch(mm)	19	
Liquid pipe branch(mm)	16	
The heat transfer area(m^2)	0.9	
CDU outline size (high) ×(wide) ×(thick)	0.525 ×0.111×0.12	
Installation height(m)	1.8	

1.4 The physical model of the cooling source

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

2. The mathematical model of the $MSHPS$

2.1 The mathematical model of the evaporator of the heat-pipe

As shown in Fig. 3, the heat release process of the working fluid in CDU is carried out. The design of heat transfer temperature is about 19 ℃. 8-9 the process of absorbing the heat of the server in the evaporator for working fluids. The temperature is about 17 ℃. The condensation heat is equal to the enthalpy dropfrom 6 state to state point 7.

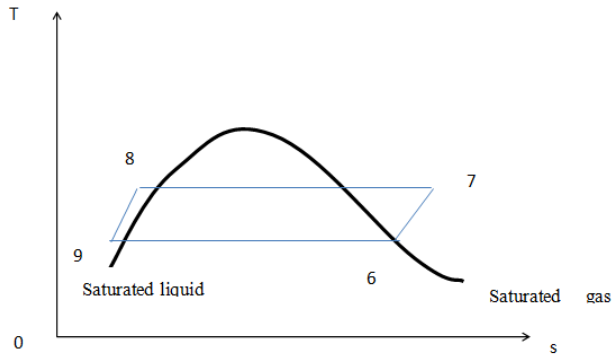

Fig. 4 *The T-S diagram of the evaporator of the heat pipe*

According to the thermo-entropy diagram of refrigerant, combined with the finite-time thermodynamics method and the balance analysis method, the mathematical model of the cooling capacity distribution unit (CDU) is established.

Referring to the micro channel heat exchanger model, based on the new Gungor-Winterton correlation formula[11], the hourly simulation model of the heat pipe evaporator is established.

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

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

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

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.

2.2 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 7to the state point 8[12].

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

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.

The exergy loss of the CDU :

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

And:

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

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 .

2.3 The mathematical model of the cooling source

2.3.1The mathematical model of the compressor

The mass flow of the refrigerant is m_{ref} , it comes to the compressor at the state 1. And the state of the outlet is state C. Because of the irreversible process of the compress. There are exergy loss and entropy generations. Seen in Fig. 5.

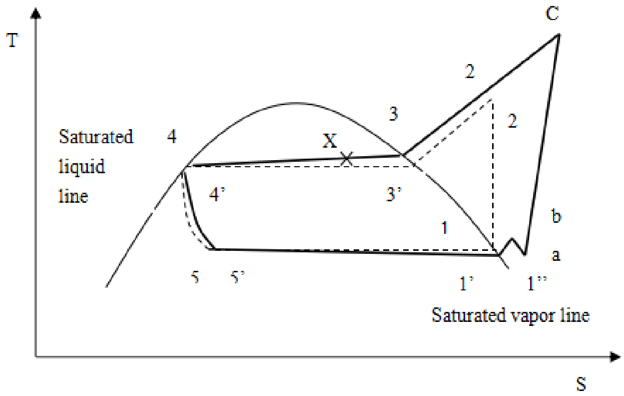

Fig. 6 *The T-S diagram of the cooling source*

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 :

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

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

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

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

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

2.3.2The mathematical model of the condensor

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

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

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:

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

2.3.3 The mathematical model of the valve

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

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

2.3.4 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].

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

(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.

3The simulation modelof the $MSHPS$ and the traditional air-conditional system

3.1The simulation modelof the $MSHPS$

3.1.1Building model

A The water cooling+ CDU + backboard mode:

The water cooling + CDU + backboard mode include 3 Sub-Cycle systems as follows:

(1) Sub-cycle(A):The evaporation process indoor (HX1).The liquid substance (S2) absorb heat in the IDC room (HX2), the temperature of the IDC room declines (S4) ,and the substance becomes to gas (S1).

(2) Sub-cycle(B): The heat exchange process in the CDU (HX2).The chilled water (S10) absorb heat in the CDU (HX2), the temperature increase (S11), the substance in the backboard exhaust heat in the CDU, and becomes to liquid (S2).

(3) Sub-cycle(C):The chilled water unit cycle.The compressor operating ($CMP1$), the chilled water offer at temperature 12℃(S10), it exchange heat in the CDU, and the temperature back to the chiller is 17℃(S11), see in Fig. 7:

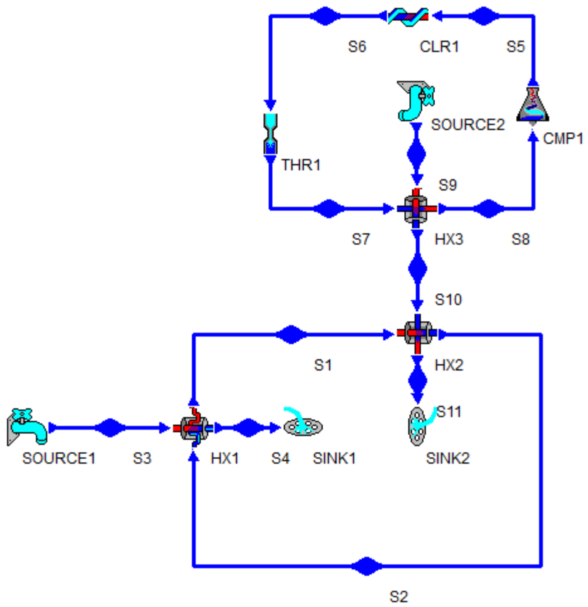

Fig. 8 The CYCLEPAD diagram of the chilled water unit +CDU + backboard mode

B The free cooling mode:

The free cooling mode include 2 Sub-cycle systems:

(1) The evaporation process in the IDC room(HX1): the liquid $R22$ (S2) absorb heat in the room (HX1), the temperature in the IDC room declines (S4), the $R22$ become to gas (S1).

The free cooling process: the cold wind (S5) absorb heat in the air-cooling heat exchanger (HX2), the temperature of the air increase (S6), the $R22$ exhaust heat, and becomes to liquid (S2); see in Fig. 9.

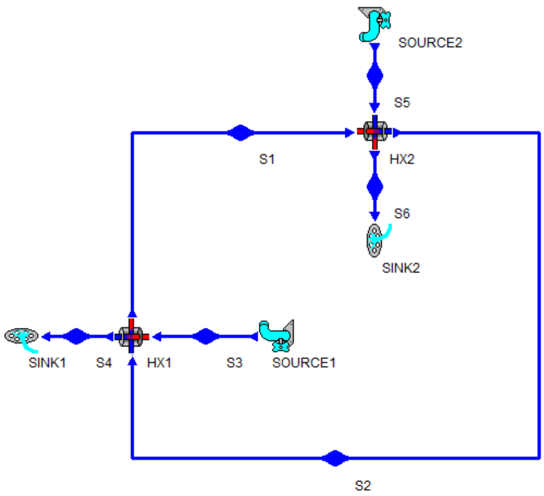

Fig. 10 *The CYCLEPAD diagram of the free cooling mode*

3.1.2Parameters input

A The water cooling + CDU +backboard mode:

The water cooling + CDU + backboard mode include three heat exchange process: The evaporation process、 the heat exchange process in the CDU and the chilled water producing process. The parameters input is as follows:

The chilled water producing process: The refrigerant (S8 Substance) is $R22$. The sanction pressure of the compressor is 500 KPa. The chilled water unit ($CMP1$) supply the chiller water to the CDU (HX2) at temperature of 12℃ (S10), the chilled water absorb heat in the CDU and the temperature improve to 17℃(S11). The mass flow of the chilled water unit is 0.018kg/s.

The heat exchange process in the : the refrigerant (S1) exhaust heat in the CDU, and the temperature reduce (S1).

The evaporation process: the refrigerant (S2

Substance) is $R22$, the power efficiency in the backboard is 3 kW, the temperature in the IDC room (S3) is 37℃, the temperature that the IDC room demand is 24℃. The temperature of the refrigerant (S2) in to the back board (HX1) is 17℃, and improved to 36℃(S1) after though the backboard. See in Fig. 11.

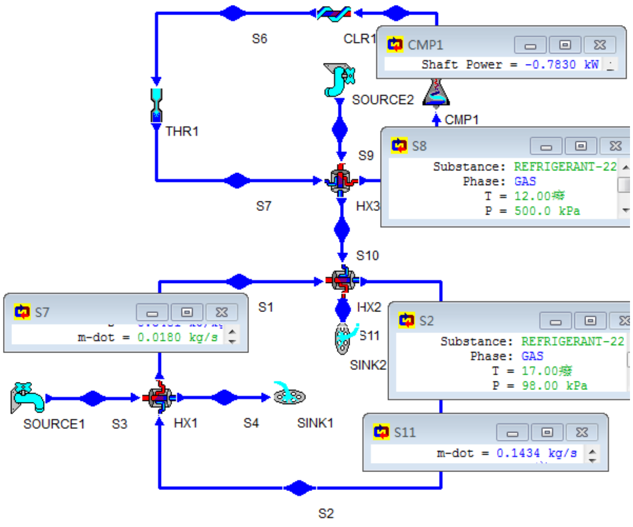

Fig. 6 *The CYCLEPAD diagram of thecompressor mode*

B The free cooling mode

The free cooling mode included two heat exchange process: evaporation process and the condensation process.

The evaporation process

The parameter input: the quantity of the heat absorbed by each of the back board heat pipe (HX1) is 3 kW (Q-dot); Suppose the temperature input at the IDC room is 37℃ (S3), it declined to 24℃(S4) through the backboard; The pressure (P) input and output is 98℃.

The substance in the backboard is $R22$ (Substance), the liquid temperature of the refrigerant is 17℃(S2), it absorbs heat in the room, and it becomes to gas (S1), the temperature increase to 36℃, the heat exchange process (HX1) is not ISOTRORIC and ISOBARIC.

3.1.3The calculation result

The power input to the chiller water unit is 0.783 kW, the mass flow of the chilled water is 0.14 kg/s,

When the heat lost is ignored, the heat exchange in the CDU is equal to the back board (HX1), 3 kW.

(1)Energy consumption of the chiller mode:

1. The power input of the compressor ($CMP1$) of each end is 0.78 kW, the power input of the 72 backboard is 56kW.
2. The power input of the chiller water pump is about 9.2 kW.
3. The power input of the fan in the backboard is 0.16 kW each, the total electric power input is 11.5 kW.

The power input of the chilled water unit + CDU +back board mode is:

（56+9.2+11.5） kW x24h x365x(100-49.3)%

=310 892 $kW \cdot h$

See in Fig.6.

(2)The calculation results of free cooling mode

The calculation results is showed in Fig 7.The mass flow of the $R22$ (m-dot) is 0.23 kg/s.

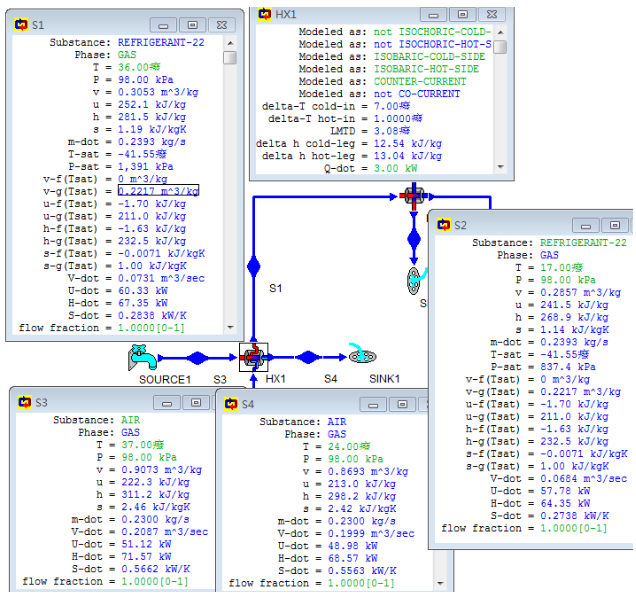

Fig. 7 *The calculation result of the evaporation process*

Energy consumption of the free cooling mode:

The energy consumption is the fan of the air – cooling system.

There are 6 fans, the total electric input is 10.8 kW;

$$10.8\,kW \text{x} 24\,h \text{x} 365 \text{x} 49.3\%$$

$$=46\,641\,kW \cdot h$$

3.1.4 The sensitivity analysis

The power input of the compressor vs. the mass flow of the refrigerant can see in Fig.8.The air flow of the outdoor vs. the outdoor temperature can seen in Fig.9.The mass flow of the air in the IDC room vs. the indoor temperature can seen in Fig.10.

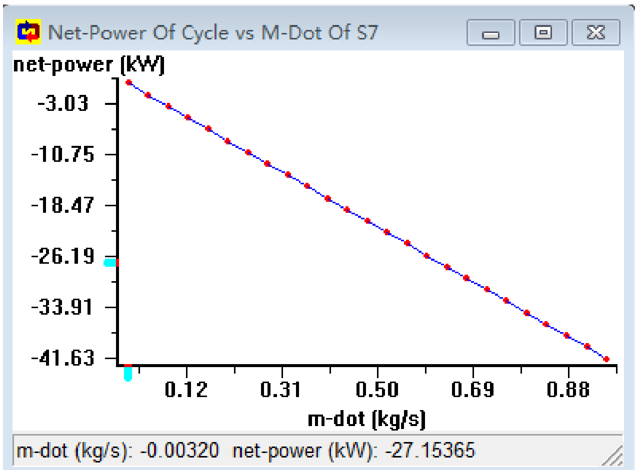

Fig. 8. The power input of the compressor vs. the mass flow of the refrigerant

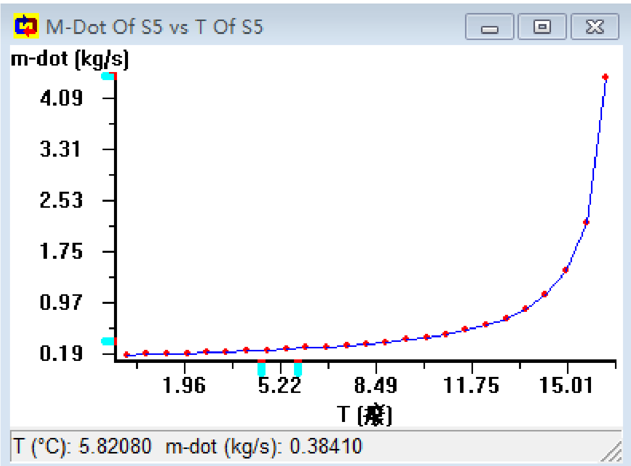

Fig.9 The air flow of the outdoor vs. the outdoor temperature

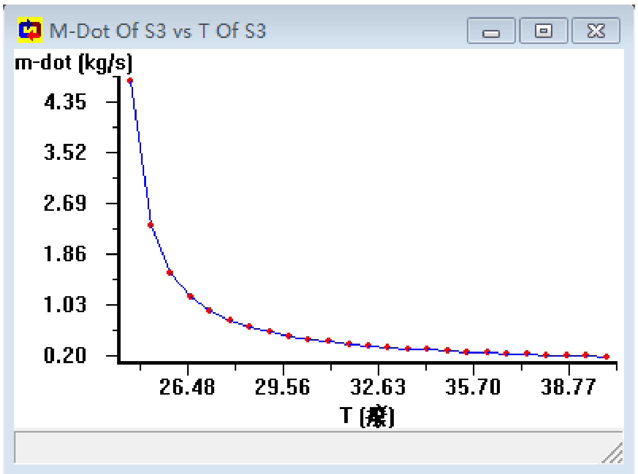

Fig.10. *The mass flow of the air in the IDC room vs. the indoor temperature*

3.1.5AEER:

$$310892+46641$$

$$=357\,533\,kW \cdot h$$

3.2The simulation model of traditional air conditioning system

3.2.1The simulation model of traditional air conditioning system with CYCLEPAD

The traditional air-conditional mode: the compressor ($CMP1$) absorb gas (S1), and compressed (S2); then exhaust heat to the environment though the condenser($CLR1$); and then through the expansion devices (THR1); at last, absorb the heat in the IDC room in the evaporator (HTR1), comes to the compressor. The refrigeration cycle is finished. See in Fig. 11.

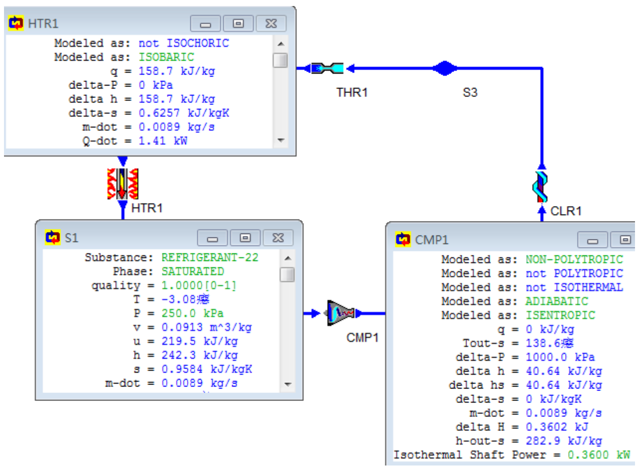

Fig.11 *The calculation results of the air-conditioning system*

3.2.2The parameter input of the traditional air-conditioning system

Parameter input: the compress process is NON-POLYTROPIC、not ISOTHEMAL; the absorb gas pressure is 250 kPa (S1), the phase of the absorb gas is SATURATED, the quality is 1.000. the Isothermal Shaft Power is 0.36 kW; the heat exchange process in the evaporator is ISOBARIC.

3.2.3The calculation results ofthe traditional air-conditioning system

The calculation result: the compress process is NON-POLYTROPIC、ADIABATIC、ISSENTROPIC; the heat exchange load of the compressor with the environment is 0; the mass flow of the refrigerant (m-dot) is 0.0089kg/s; the heat exchange (HTR1) of the evaporator is 1.41 kW. see in Fig. 12.

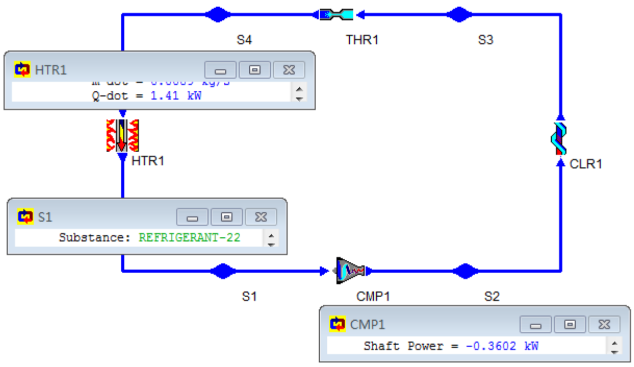

Fig.12 *The calculation results of the air-conditioning system*

So, the COP of the traditional air conditional system is 3.92. When the heat load in theroom is 216 kW, the electric power input of the compressor is 55 kW.

$AEER$:

55.24 kW ×24h×365d

=483 928 $kW \cdot h$

4. The results and discussions

Assuming that the heat source of the server is one-dimensional and steady, it is uniform along the height direction. According to the calculation results of the simulation model, at the beginning, the heat transfer rate decreases gradually with the heat absorbed by the working medium because of the more components of the liquid working medium in the evaporator. The heat transfer cycle of the heat exchanger takes about 6 seconds, and its total heat transfer Q is about 77KJ, as shown in Figure 13.

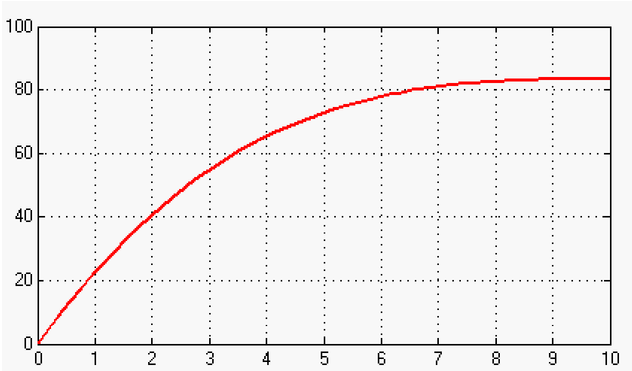

Fig. 13 *The relationship between heat transfer and heat transfer time of heat pipe exchanger.*

The height of the heat pipe evaporator is two meters. Along the height direction, the dry degree of refrigerant gradually changed from 0 to 1. The relationship between the dryness of heat exchanger tubes and the height and time of heat exchanger is as follows:

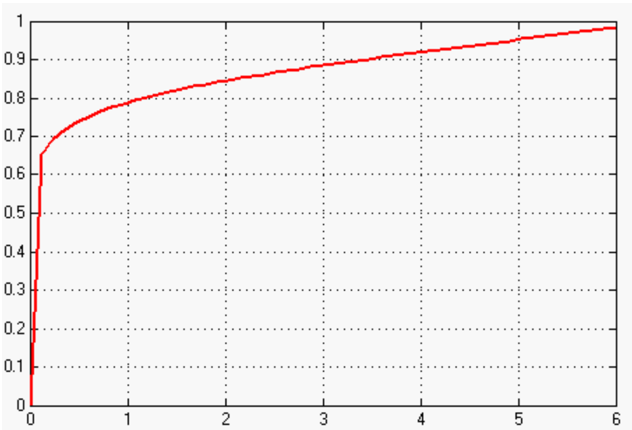

Fig. 14 *The dryness and height of the heat exchanger VS. the operation time*

Energy efficiency calculation of heat pipe evaporator:

The refrigeration efficiency of the system is:

$$COP = \frac{Q}{W} = \frac{\int_0^{\tau} \dot{Q} dt}{w \cdot \tau}$$

$$=\frac{77}{0.4\times 6}$$

=32.5

Theinput electric power for fan, is 0.4KW.

According to the total heat transfer of the evaporator in one cycle, the fillingrefrigerant is 3.85Kg.

The heat transfer gradually slows down with time. CDU has a heat transfer of about 9.8J after 4.2 seconds of heat transfer.Seen in Fig.15.

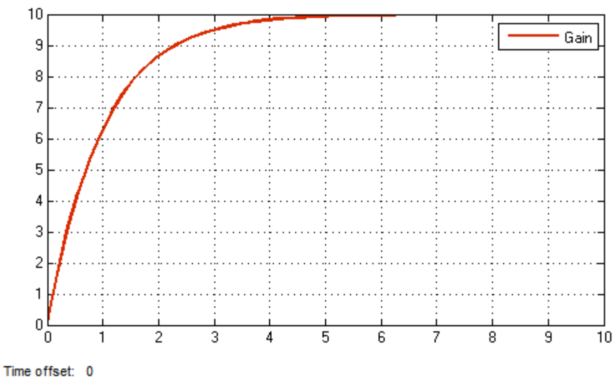

Fig.15 *The relationship between total heat transfer and running time of CDU*

The heat transfer efficiency ηof 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 kg/s. Seen in Fig.16.

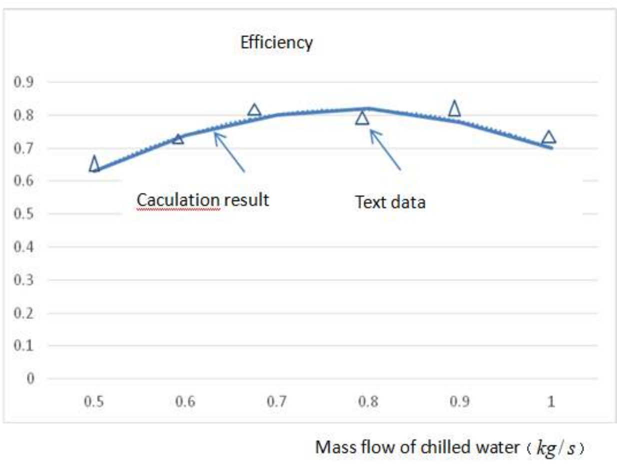

Fig. 16 *The relationship between CDU heat transfer efficiency and cold water flow rate*

The heat-pipe using time and energy saving potential in major city of China is issued in the Fig. 17.

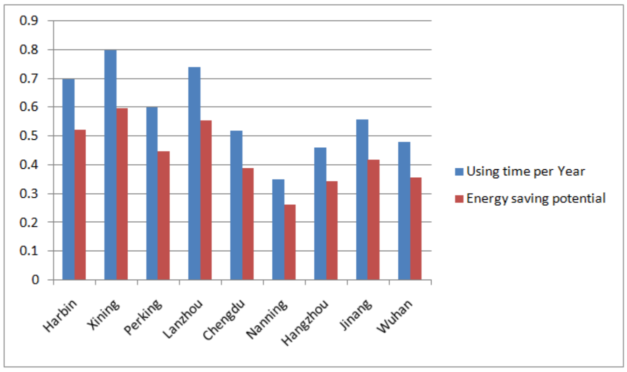

Fig. 17 *The energy saving potential in major city of China*

5. Conclusions

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

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.

The chiller + heat pipe mode and the free cooling mode has been simulated and compared with the traditional air-conditioning system. The energy saving capacity has been carried out. The *CYCLEPAD* software is a simulation tool for theory cycle, and it can be applied in the real pipe-network systems.This paper is based on server heat source for steady state. In fact, the heat load is fluctuating. The next step is to establish a stochastic model of heat load, which can further reflect the actual operation.

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