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# Finite-Time Thermodynamic Simulation of Circulating Direct Condensation Heat Recovery on Chillers

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## Abstract

A time series model is used in this paper to describe the progress of circulating direct condensation heat recovery of the compound condensing process (CCP) which is made of two water cooling condensing processes in series for a centrifugal chiller in the paper. A finite-time thermodynamics method is used to set up the time series simulation model. As a result, an upper bound of recoverable condensation heat for the compound condensing process is obtained which is in good agreement with experimental result. And the result is valuable and useful to optimization design of condensing heat recovery.

## Keywords

Condensation Heat Recovery, Compound Condensing Process, Time Series, Finite-Time Thermodynamics

## 1. Introduction

One of the most widely used building heating and cooling equipment in China is the centrifugal chiller system. There are several problems associated with utilizing the system [1]. A great deal of useful waste energy, which can be used for other purposes, is directly dissipated to the environment. This dissipated heat not only wastes energy, but also causes severe pollution in the surrounding areas. On the other hand, in order to satisfy building sanitary water requirement, an external electrical or gas-fired boiler must operate all-day long [2]. Many methods have been attempted to tackle these problems. In recent years, there have been rapid developments in combined space cooling and water heating systems for residences [3] [4] [5]. This combined system utilizes the rejected condensation heat from air-conditioners to preheat domestic hot water, leading to claims that water heating in summer, primarily for bathing, can be made available virtually free whenever space cooling is required, and is considered one of the most cost effective energy conservation measures. This technology is particularly suitable for the applications in the subtropical regions because most residences are provided with air-conditioners for space cooling in the hot and humid summer. Condensation heat recovery on heat pumps [6] [7] [8] and other building-plant system [9] has also been studied. The results show it has little influence on the operating capability of an air-conditioning system using a condensation heat recovery system. With the development of computer technology, numerical simulation has been used to analyze the condensation heat recovery on experimental basis. Mason [10], K.C. Toh [11], Bong [12] and Baxter [13] have analyzed the capability and operating characteristics of a large scale system using numerical method. Their results show that the condensation heat recovery saves energy significantly.

As a new method to recovery condensation heat, compound condensation methods have been proposed and developed by Gong [1] [14] [15] [16] [17]. The compound system may recover the condensation heat wholly or partially to produce sanitary hot water, especially for hotel. A thermodynamic simulation study has been carried out for a single stage centrifugal chiller [18]. It meets a flexible requirement of sanitary hot water, and seldom affects the condensing effect of the air conditioning equipment during its operation, but there is also a question existed that is the upper limit of the condensation heat that can be recovered. In late 1990s, a high recovery ratio resulted in an accident and the equipment was damaged in system reconstruction for energy conservation.

In this paper, finite-time thermodynamics will be employed to describe the progress of condensation heat recovery. The upper limit of condensation heat which can be recovered in compound condensation mode of the centrifugal chiller is being worked out by time series method.

## 2. Physical Model of the Compound Condensing Process

A single stage centrifugal chiller in a hotel in south China has been selected as the example. The refrigeration capacity is about 1750 kW with the working fluid of R22. Although a single condensing unit plant may get good condensing effect, energy is not used rationally during the whole cooling process. A sanitary hot water system has been added to the chiller at July 2001 as a part of the whole condensation process in order to take full advantage of the condensation heat shown in **Figure 1**.

Comparing with the single condensation refrigeration cycle, the system has an additional condensation heat recovery device which locates between the compressor and the original condenser.

The high temperature refrigerant from the compressors passes the sanitary

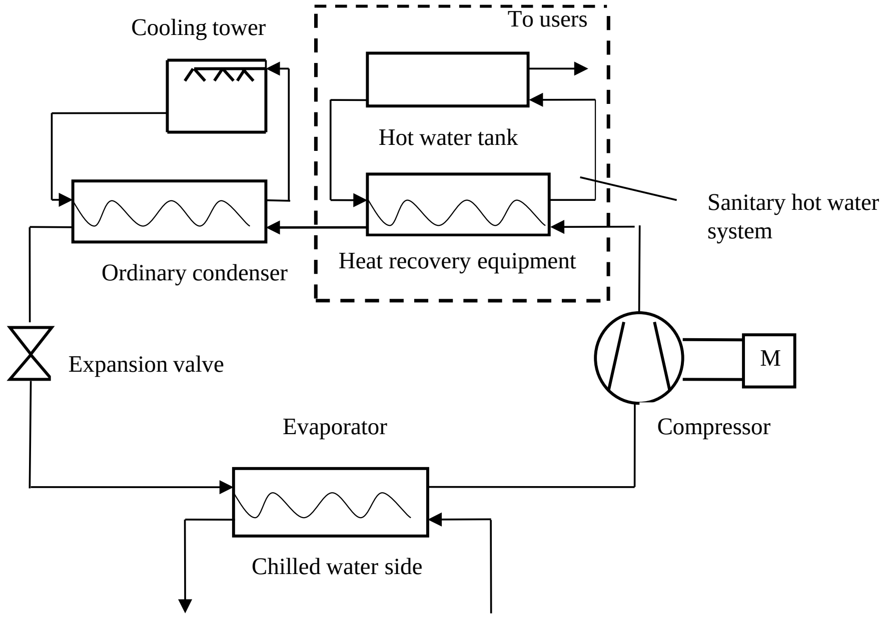



**Figure 1.** Schematics of a vapor-compression refrigeration cycle.

hot water supply system and the ordinary condenser in turn. The sanitary hot water system produces sanitary hot water about 6t/h with the water temperature of 50˚C. A conventional chiller system on a T-S diagram is illustrated in **Figure 2**. Cycle 1-2-3'-4'-5-1 shows a theoretical chiller cycle. Process 1-2 is isentropic compression. Process 2-3'-4-4' is isobaric cooling. Process 4'-5 is an isenthalpic expansion. And process 5-1 is isobaric heating. There are several modifications in a practical chiller cycle. The modifications include: 1) there are refrigerant pressure drops in the heat exchangers (cooling tower and evaporator); 2) the thermostatic expansion valve is an isenthalpic process; 3) there are heat losses to the ambient from the system components; 4) the cycle is working with super-heating and sub-cooling regions. Therefore, the practical cycle is depicted by cycle 1'-1''-a-b-c-2'-3-x-4-4'-5'-1' as shown in **Figure 2**, where a-b-c is an irreversible compression process; c-x is the process of heat absorbed by sanitary hot water in hot water supply; x-3-4-4' is the cooling process which rejects heat to a high-temperature sink by condenser; 4'-5' is an isenthalpic expansion process; and 5'-1' is an isobaric evaporation process where heat is absorbed from a low-temperature source and cooling is produced. The area surrounded by the cycle 1'-1''-a-b-c-2'-3-x-4-4'-5'-1' on the T-S diagram is the power input to the compressor. The area surrounded by 9-c-2'-3-x-4-4'-6-9 is the condensation heat exhausted to the two condensers.

It has been estimated that the condensation heat is 1.3 times of refrigerating effect in the compressing system, and is 2.5 of that in absorbing system. Unfortunately, the plenteous condensation heat is not utilized efficiently. Rather, the heat is directly exhausted to the surrounding environment in most cases. Therefore, natural air and water energy can be utilized with the compound-mode, *i.e.*, the system can employ two water systems cooling. In the summer and transitional

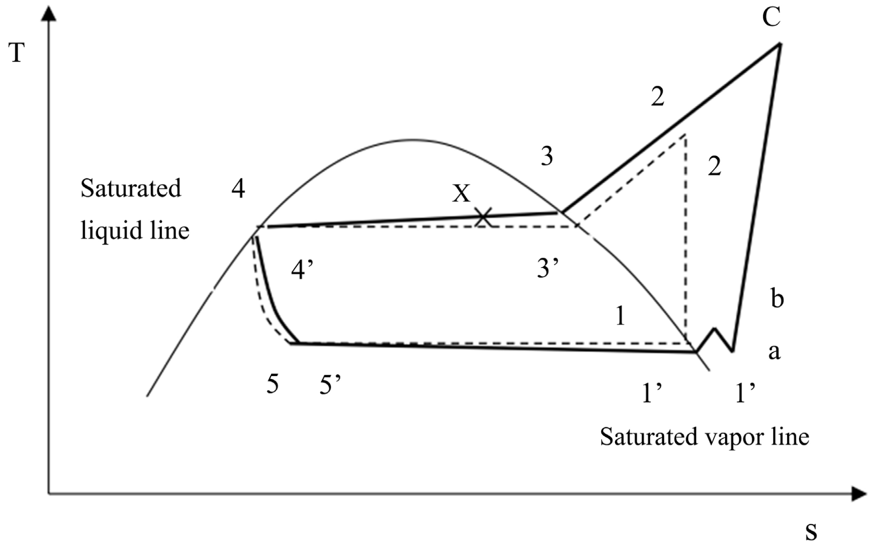



**Figure 2.** Theory of refrigerating cycle.

periods, the compound-model takes the condensation heat exhausted to the surrounding environment into heating sanitary hot water. It has been testified the new heat recovery technique decreases the heat contamination and waste heat dramatically in the summer or transition period. The chiller system works more efficiently and economically.

The refrigeration system with condensation heat recovery has been operated steadily for more than 14 years. With such a compound system, the sanitary hot water supply can be satisfied by the condensation heat during the summer and the transition period. It has served its purpose of environment protection and energy-efficiency enhancement successfully. Several new compound systems have been built at other locations. Based on these experimental data, a method of time series would be proposed in the following part to express the maximum ratio of the condensation heat ban be recovered theoretically.

## 3. Time Series Model of Compound Condensing Process (CCP)

As shown in **Figure 2**, a water tank can be seen as the source and the sink of the sanitary water which would be heated by high temperature refrigerant fluid while cycling between the water tank and the condensation heat recovery device (condenser 2).

Ignoring the heat loss of the water tank, then the heat progress in the water tank could be described by the following Equations (1) and (2).

$$t_{wt,\tau+\Delta\tau} = t_{wt} + \frac{m_w}{M} \cdot \Delta t_{w,\tau} \tag{1}$$

$$t_{wt,\tau} = t_0 \quad \tau = 0 \tag{2}$$

in which, $t_R$ and $t_W$ are the temperature of refrigerant and sanitary water. The symbol $\tau$ means time. The parameter $t_0$ is the initial temperature of sanitary water. It could be the temperature of the surroundings. The parameter $m$ and $M$

are the quality flow of sanitary water and total amount of sanitary water respectively.

Sanitary water flow through the condenser 2, the subscripts "in" and "out" are used in this work to express the inlet and outlet of the sanitary water. The temperature of sanitary water changes with the different utilized condition and different working condition of the chiller. Ignoring the heat loss of the all tubes, the relationship of the temperature of the inlet and outlet of condenser 2 could be described by the following Equations (3)-(5).

$$t_{w,out,\tau} = t_{w,in,\tau} + \Delta t_{w,\tau} \tag{3}$$

$$t_{w,in,\tau} = t_0 \quad \tau = 0 \tag{4}$$

$$t_{w,in,\tau+\Delta\tau} = t_{wt,\tau} \tag{5}$$

An integrated coefficient of performance $EER$ is employed to estimate the energy utilization effectiveness of a compound condensing system. It is the ratio of utilized energy effectively and compressor power input. The energy utilization includes both the effective cooling capacity $Q_{ck}$ and the heating sanitary water capacity $Q_w$. See in Equation (6).

$$EER = \frac{Q_{CK} + Q_W}{W} \tag{6}$$

In which, $W$ is the input power. $Q_w$ can be expressed as

$$Q_{W,\tau} = m_W \cdot c_p \cdot \left(t_{w,out,\tau} - t_{w,in,\tau}\right) \tag{7}$$

Ignoring the heat loss in the heat transfer process, Equation (7) can be rewritten as Equation (8):

$$q_R \left(h_C - h_{4'}\right) \cdot x = m_W \cdot c_p \cdot \left(t_{w,out,\tau} - t_{w,in,\tau}\right) \tag{8}$$

where $q_R$ is the flow of refrigerant; $h_c$ and $h_{4'}$ are the enthalpy of the refrigerant at state point C and 4′ in **Figure 1**; parameter $x$ is the ratio of condensation heat recovered.

So:

$$x = \frac{m_W \cdot c_p \cdot \left(t_{w,out,\tau} - t_{w,in,\tau}\right)}{q_R \cdot \left(h_C - h_{4'}\right)} \tag{9}$$

In Equation (9), the flow of refrigerant $q_R$ and the flow of sanitary water $m_W$ are both fixed value in particular system; the enthalpy of the refrigerant at state point C and 4′ $h_c$ and $h_{4'}$ are both fixed in particular operating condition; the specific heat at constant pressure of sanitary water $c_p$ is can be regarded as a constant; the temperature of sanitary water at the outlet and inlet of the condensation heat recovery device $t_{W,out,\tau}$ and $t_{W,in,\tau}$ are function of time $\tau$. Although time $\tau$ is a variable, and the maximum of $\tau$ is variable with the condition of the utilization, the scale of $t_{W,out,\tau}$ and $t_{W,in,\tau}$ are fixed in a particular system(the maximum is set and the minimum is the temperature of the environment). The heating progress of the sanitary is nearly the same.

From the analysis above, the condensation heat transferred to the sanitary is nearly a constant when the system is operating steadily. Then the temperature increase of the sanitary $\Delta t_{W,i}$ is nearly fixed in a cycle, see in Equation (10).

$$\Delta t_{W,i} = \left(t_{w,out,\tau} - t_{w,in,\tau}\right) = \frac{q_R\left(h_C - h_{4'}\right)\cdot x}{m_W \cdot c_p} \tag{10}$$

The parameter $\Delta t_{W,i}$ = 3˚C - 5˚C (this is an experiment value). Then the Equation (10) can be changed as follow, see in Equation (11).

$$x = \frac{m_W \cdot c_p \cdot \sum_{1}^{n} \Delta t_{w,i}}{q_R\left(h_C - h_{4'}\right)} \tag{11}$$

In which, $n$ is the cycle times of sanitary in the condensation heat recovery device, see in Equation (12).

$$n = CINT\left(\frac{\Delta t_{\max}}{\Delta t_{W,i}}\right) \tag{12}$$

In which, *CINT* is the function of rounded to integer. $\Delta t_{\max}$ is the maximum of the temperature difference of the sanitary water, see in Equation (13).

$$\Delta t_{\max} = t_{out,\max} - t_{in,\tau=0} \tag{13}$$

Here all the parameters of equation are known. The maximum of $x$ can be found by the simulation of compound condensing model with the help of CYCLEPAD.

To the Equation (11), some change must be done also. The temperature of the refrigerant and sanitary water $t_{R,\tau}$ and $t_{W,\tau}$ changed with not only time $\tau$, but also the position of the heat exchanger. It was difficult to describe them with a sample function.

The whole circulate is separated into many small segments along the direction of the refrigerant to find an average temperature difference for calculating the heat transfer between the two fluids. Then a logarithmic mean temperature difference (*LMTD*) can be used as the equivalent with the actual temperature difference in each segment, see in Equation (14):

$$LMTD = \frac{\left(T_{Hout} - T_{Lin}\right) - \left(T_{Hin} - T_{Lout}\right)}{Ln\left[\left(T_{Hout} - T_{Lin}\right)/\left(T_{Hin} - T_{Lout}\right)\right]} \tag{14}$$

where $T_{H,out}$ and $T_{L,out}$ are the outlet temperature of the refrigerant and that of cooling water in each segment, $T_{H,in}$ and $T_{L,in}$ are the input temperature of the refrigerant and that of cooling water in a segment, respectively.

Then the condensation heat can be recovered can be found by Equation (15):

$$Q_{W,\tau} = \int K\left(LMTD\right)A \tag{15}$$

From the analysis above, the parameter of *LMTD* is the only unknown value.

As the sanitary hot water circulated in the condensation heat recovery device, its temperature changes with time and these changes lead to the changes of the *LMTD*. So *LMTD* is a function of time as well.

As the *LMTD* has a maximum value in a given system and it is positive, so Equation (15) can be rewritten as Equation (16):

$$Q_{w,\tau} = \int_{\Delta t_{\max}}^{0} K \cdot A \cdot LMTD(\tau) \tag{16}$$

where $\Delta t_{\max}$ is the maximum of the logarithmic mean temperature difference between sanitary water and refrigerant, and it is given by Equation (17):

$$\Delta t_{\max} = t_{(R,0)} - t_{(w,0)} \tag{17}$$

where $t_{(R,0)}$ and $t_{(W,0)}$ are the temperature of refrigerant and sanitary hot water in the initial time of the cycle.

The sanitary water circulated in the condensation heat recovery device. This progress of heat exchange would be stopped when the temperature different is 0. In the real operating progress, when the temperature different is below 2˚C, the efficiency of condensation heat recovery is low, and the condensation heat can be recovered is almost possible to ignore. Then the equation can be rewritten as Equation (18):

$$Q_{W,\tau} = \int_{\Delta t_{\max}}^{2} K \cdot A \cdot LMTD(\tau) \tag{18}$$

Since the sanitary water is cycling through the system, its temperature increases by more and more cycling. Therefore the temperature difference $LMTD(\tau)$ is a function of time. And it is a continual curve in the bond of 2˚C and $\Delta t_{\max}$.

Here a definition of $\Delta t_{w,\max}$ that is the maximum temperature difference between the inlet temperature of the first circulation and outlet temperature of the last circulation in sanitary hot water is presented. Actually, in general, $\Delta t_{w,\max}$ always is less than or equal to $\Delta t_{\max}$. So we suppose that the $\Delta t_{\max} = \Delta t_{w,\max}$ in the paper. Thus, Equation (15) can be dispersed as a time series. See in Equation (19):

$$Q_W = \sum_{i=1}^{n} K \cdot A \cdot LMTD_i \tag{19}$$

where $n$ is the number of circular times that the sanitary water cycling through the system, $\Delta t_{w,i}$ = 3˚C - 5˚C (this is an experiment value)it is given by Equation (20):

$$n = CINT\left(\frac{\Delta t_{w,\max}}{\Delta t_{w,i}}\right) \tag{20}$$

In which $\Delta t_{w,\max}$ is the maximum of the temperature difference between sanitary hot water inlet and its outlet. CINT is the function of rounded to an integer number.

In the Equation (21):

$$LMTD_i = \frac{\left(T_{Hout} - T_{Lin,i}\right) - \left(T_{Hin} - T_{Lout,i}\right)}{Ln\left[\left(T_{Hout} - T_{Lin,i}\right) / \left(T_{Hin} - T_{Lout,i}\right)\right]} \tag{21}$$

in which, $T_{Lin,i}$ and $T_{Lout,i}$ is separately the temperature of sanitary hot water at the inlet and outlet of the condensation heat recovery device at the $i^{th}$ cycle, see in Equation (22) - Equation (23).

$$T_{Lin,i}=\begin{cases}T_{Lin,i-1}+\Delta t_{w,i}\cdot(n-1) & i=1\sim(n-1)\\ T_{Lin,n-1}+MOD\left(\dfrac{\Delta t_{w,\max}}{\Delta t_{w,i}}\right) & i=n\end{cases} \tag{22}$$

$$T_{Lout,i}=\begin{cases}T_{Lout,i-1}+\Delta t_{w,i}\cdot(n-1) & i=1\sim(n-1)\\ T_{Lout,n-1}+MOD\left(\dfrac{\Delta t_{w,\max}}{\Delta t_{w,i}}\right) & i=n\end{cases} \tag{23}$$

in which MOD means to seek the remainder of $\Delta t_{w,\max}$ divided by $\Delta t_{w,j}$.

Then the Equation (9) can be written at last as follow, see in Equation (24):

$$x=\frac{\sum_{i=1}^{n}K\cdot A\cdot LMTD_i}{q_R\cdot\left(h_c-h_{x,i}\right)} \tag{24}$$

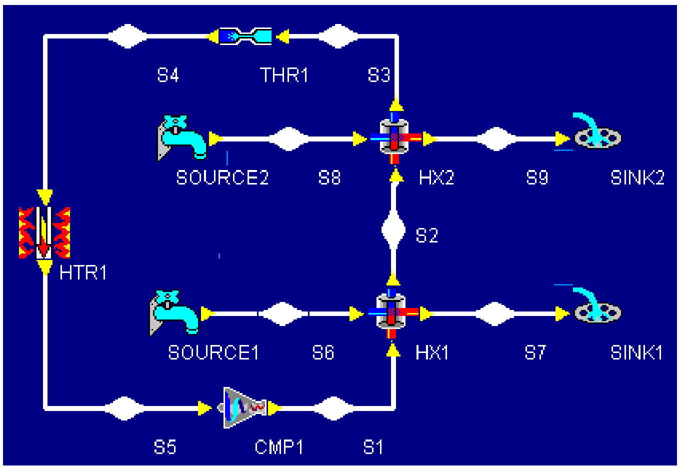
Since $LMTD_i$ is supposed based on the practical operating situation, the statement of sanitary hot water is definite at each cycle.

## 4. Simulation of Compound Condensing Process

Compound condensation heat recovery is simulated with CYCLEPAD in this paper, as shown in Figure 3. The symbols in Figure 3 and the components represented by them are shown in Table 1, as also defines of all components of the cycle. The parameters in this model were all based on the experiment result of the hotel in south china.

Table 1. Components represented by the symbols in Figure 3 and the definitions of all components.

| symbol | component | definition |
|---|---|---|
| CMP1 | compressor | adiabatic, isentropic |
| HX1 | condensation heat recovery device | counter-current, isobaric |
| HX2 | condenser | counter-current, isobaric |
| THR1 | expansion valve | |
| HTR1 | evaporator | isobaric |



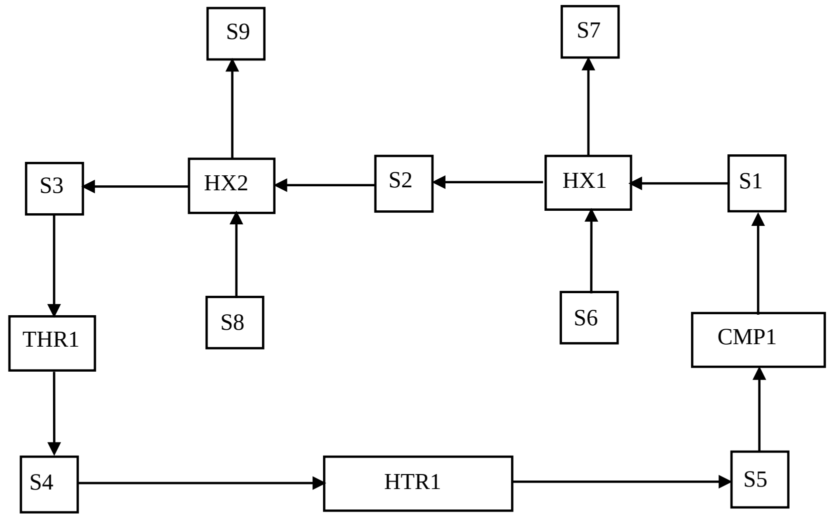



**Figure 3.** The compound condensation heat recovery cycle.

There are three sub-cycles in the compound condensing cycle. The first sub-cycle is the refrigerant cycle: S1-HX1-S2-HX2-S3-THR1-S4-HTR1-S5-CMP1-S1. In this cycle the refrigerant absorbs heat from the air-conditioning room (HTR1) and releases it to the environments. The second sub-cycle is the sanitary hot water cycle: S6-HX1-S7. Sanitary water is heated in the heat-exchanger (HX1) by the condensation heat. In which S6 is the temperature of water comes from the sanitary water tank and S7 shows the temperature of sanitary water goes back to the water tank. The third sub-cycle is the cooling water cycle: S8-HX1-S9. The residual condensation heat is brought to the outdoor environment. In which S8 means the temperature of cooling water back from the cooling tower and S9 shows the temperature of cooling water goes to the cooling tower. If the sanitary water cycle is not in need, the second sub-cycle does not operate. The third sub-cycle works full time.

## 5. Result and Discussion

In this paper two equations were both discussed, the result of them is nearly the same. At state S2, with Equation (14)-(15), the percent which recoverable condensation heat takes of the whole can be taken out if the ratio $x$ is known. When the value of S2 is changed, the position of $x$ in Figure 1 moved on the curve 3-4′. The temperature difference is 5˚C in the second sub-cycle.

COP of the refrigeration cycle with compound condensation heat recovery versus with the temperature of the outlet of compressor (S1) and the condenser (S3) are shown in Figure 4 and Figure 5.

As the temperature of the outlet of compressor (S1) increases, the COP value decrease. Figure 6 shows the refrigeration effect (the heat is transferred to the cold water in evaporator: HTR1) versus with the temperature of the inlet of compressor (S5).

The ratios of condenser heat can be recovered are shown in Figure 7 and the experimental result are shown in Figure 8.

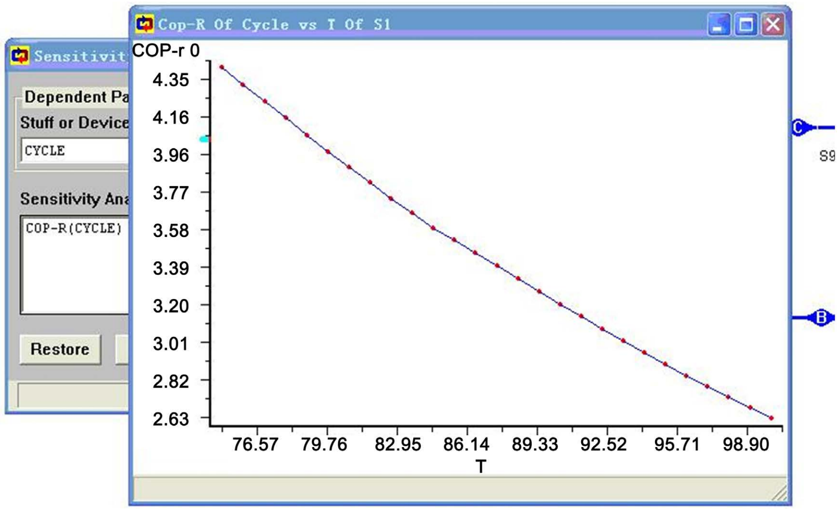



**Figure 4.** The relation of COP with the temperature of the outlet of compressor (S1).

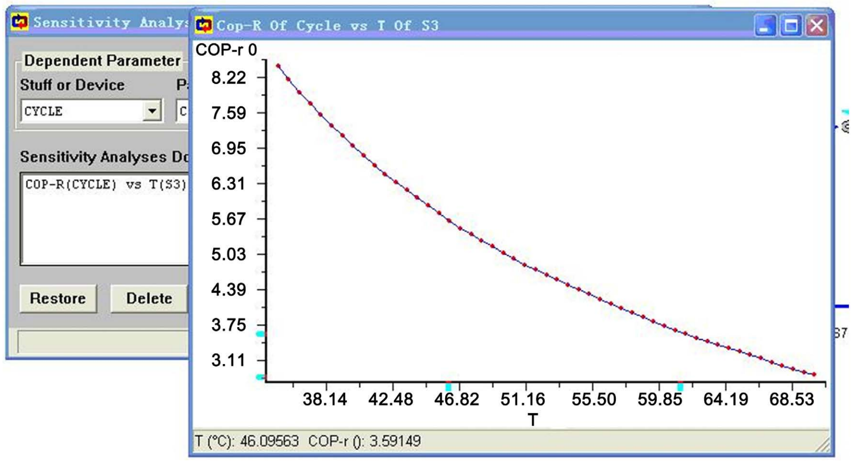



**Figure 5.** The relation of COP with the temperature of the outlet of the condenser (S3).

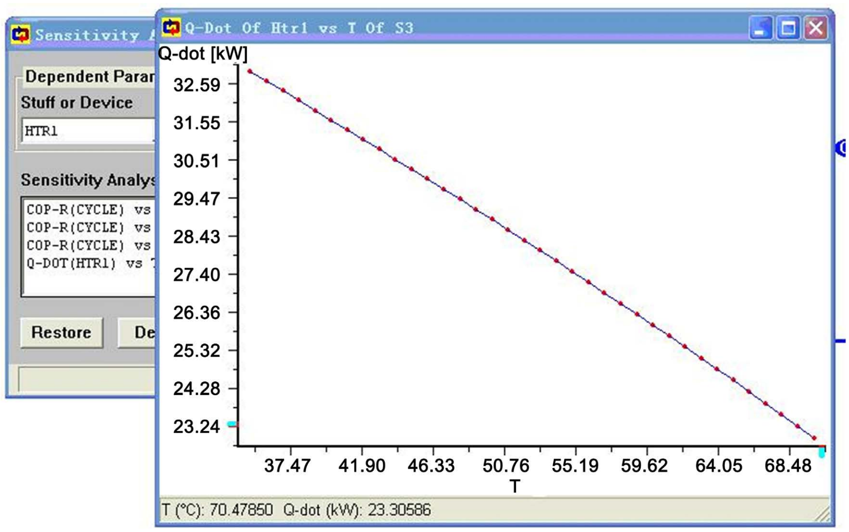



**Figure 6.** The relation of refrigeration effect with the temperature of outlet of the condenser (S3).

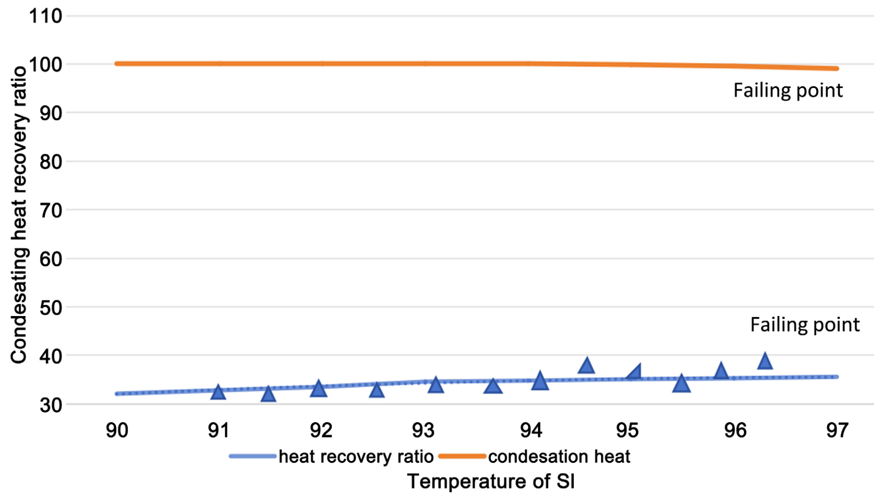



**Figure 7.** Some points of the relation of temperature of outlet of compressor with the ratio of the condensation heat can be recovered.

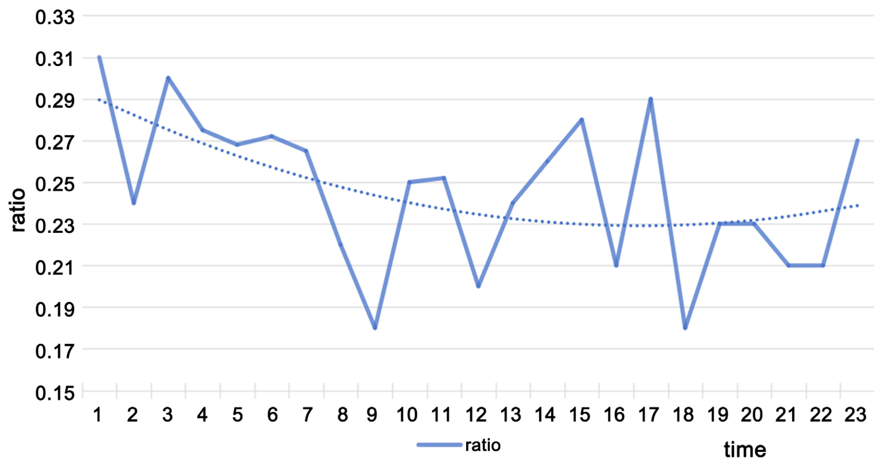



**Figure 8.** The relation of operation hours with the ratio of the condensation heat can be recovered of the experimental in the hotel in south china.

By comparing **Figure 7** and **Figure 8**, the ratio is about 30% when the temperature of the outlet of the compressor is changed between 90˚C and 100˚C (nearly 100˚C, but not 100˚C). The simulation result is very agreeable to the experimental results of the hotel in south china. When $x$ is 0, the condensation heat recovery device stops working, so the system works as a traditional chiller. The highest ratio of the circulating direct condensation heat recovery is about 36.5% in simulation, which is verified by the experimental results. Of course the ratio in real heating and cooling systems cannot reach this ratio.

## 6. Conclusion

In this paper, compound condensing technique, as a new method of condensation heat recovery on centrifugal chiller, is expressed with time series method based on finite-time thermodynamics. Process of each components of the system is based on our experimental data. The upper bound ratio of recoverable condensation heat is found and shown at the end of this paper. The ratio fits well with the results from our successful experiment.

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## Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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## Nomenclature

| | |
|---|---|
| $Q_{ck}$ | the cooling capacity (kW) |
| $Q_w$ | the sanitary water heating capacity (kW) |
| $COP$ | coefficient of performance |
| $EER$ | energy effect ratio |
| $T_{Hout}$ $T_{Hin}$ | the temperature of Fluid H at outlet or inlet (K) |
| $T_{Lout}$ $T_{Lin}$ | the temperature of Fluid L at outlet or inlet (K) |
| $W$ | the power of input (kW) |
| $LMTD$ | the logarithmic mean temperature difference (K) |
| $T_L$ $T_H$ | the temperature of Fluid H/L (K) |
| $\tau$ | the time (s) |
| $q_R$ | the flow of refrigerant (kg/s) |
| $K$ | the coefficient of heat transfer (w/(m$^2$·K)) |
| $A$ | the area of heat transfer (m$^2$) |
| $H$ | the enthalpy of the working fluid (kJ/kg) |

## Subscript

| | |
|---|---|
| *Fluid H* | high temperature fluid |
| *Fluid L* | Low temperature fluid |
| 1,2,4,$x$ | statement of the refrigerant on **Figure 1** |
| $R$ | refrigerant |
| $W$ | sanitary water |
| *Max* | maximum |
| *CINT* | a round number, integer number |
| *MOD* | means to seek the remainder of $\Delta t_{w,\max}$ |