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RESEARCH ON CHARACTERISTICS OF DOUBLE-EVAPORATORS IN VRV AIR CONDITIONER

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ABSTRACT

In this paper, for studying the characteristics of double-evaporators in VRV air conditioner, a simulation on a distributed-parameter mathematical model is carried. The results present the static and dynamical characteristics of the system. Especially reveal the influencing between the double-evaporators. And a control method for reducing the influencing is proposed.
Key words: double-evaporators, VRV, simulation, suction pressure

INTRODUCTION

The evaporator is a main component for heat transfer in an air-conditioner system, its characteristics have effects on cooling capacity and COP directly. A double-systems air conditioner is composed of a compressor, a condenser, and two evaporators which are connected in parallel and controlled by two electric expansion valves separately. This configuration results in the same suction pressure of the two paralleled evaporators. Because the two evaporators are controlled separately, and their conditions for heat transfer are different, the two evaporators are independent to some extent. The valves not only have effects on controlling the superheat degrees, but also have effects on allocating the flow amount. So the double-evaporators have different characteristics compared with the single evaporator. Research is needed for the characteristics, which are the bases for control. In this paper, Research on the static and dynamical characteristics of the paralleled evaporators is carried, which is based on a distributed-parameter mathematical model.

NOMENCLATURES, SUBSCRIPTS AND SUPERSCRIPTS

Nomenclatures

A cross-section area (m^2)
A_{sur} area of pipe surface (m^2)
Cp specific heat ($kJ \cdot kg^{-1} \cdot K^{-1}$)
D diameter of tube (m)
EV evaporator
EEV electric expansion valve
F opening area of electric expansion valve
G refrigerant mass flow amount ($kg \cdot s^{-1}$)
h specific enthalpy ($kJ \cdot kg^{-1}$)
Mw pipe mass (kg)
P pressure ($N \cdot m^{-2}$)
V suction volume of per circle of compressor
N Rotational speed
α heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
Q Cooling capacity (kW)
X dryness fraction
t time step (s)
T temperature (K)
Z spatial step (m)
ρ density ($kg \cdot m^{-3}$)
ξ heat amplifying coefficient
v specific volume
1, 2 evaporators of 1 or 2

Subscripts

air air side
I spatial node number

c compressor
cd condenser
db dry bulb
e evaporator
f friction
i spatial nodal boundary number
R refrigerant
W pipe wall
wb wet bulb

Superscripts
in inside the tube, inlet
j node number of time
out outside the tube, outlet
suc suction

MODELING

In the air conditioner system, the components influence on each other. The evaporators are influenced by the EEVs and the compressor connected with them. If a whole model of the paralleled evaporators in double-systems air conditioner is established, the electric expansion valves and the compressor must be considered, described as fig.1.

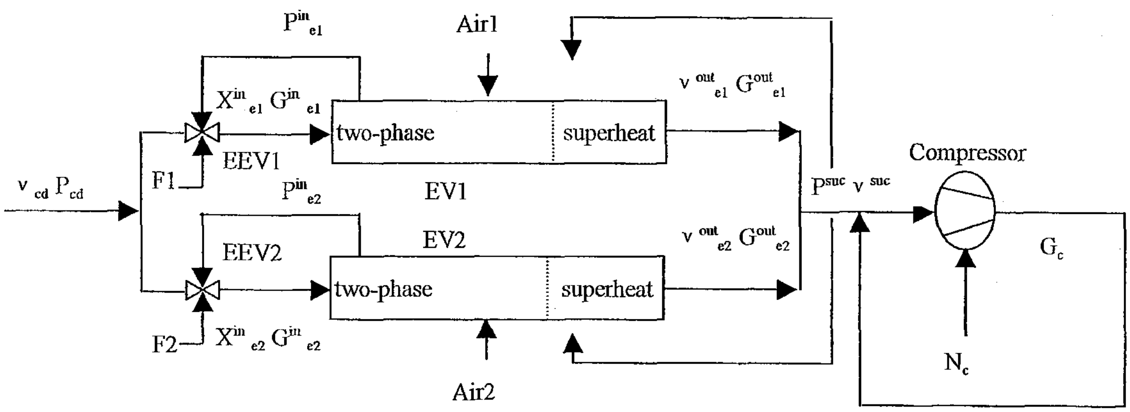


Fig.1 Model of the paralleled evaporators with the compressor and the electric valves

When model is established, the valves and the compressor are simplified as lumped parameter mathematical models.

Electric expansion valve:

$$G^{in} = \frac{F}{\sqrt{\zeta}} \sqrt{(P_{cd} - P^{in}) / v_{cd}} \tag{1}$$

Compressor:

$$G_c = \lambda * N_c * V / v^{suc} \tag{2}$$

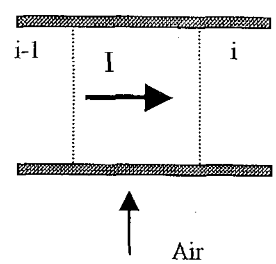


Fig.2 Unit of control volume

λ and $\frac{1}{\sqrt{\zeta}}$ are the characteristic coefficients of compressor and valve, which can be got by empirical equations[1].

The key of modeling is to set up the distributed-parameter model of the evaporator. In the paper, The model is developed on the interface equations and the conservation equations. The two-phase flow theory is used to analyze the flow of refrigerant R22. The flow is seen as one-dimensional flow along the axis of the tube. The thermal storage of the tube wall and the elbow of the tube are considered in the computation. The unit of control volume of No.I is described as fig.2. In fig.2, point I locates at the midpoint of the unit, and points i and i-1 locate at the boundary of the unit.

The finite difference equations:

Equations 3,4,5 are the mass, momentum, energy conservation equations, and the momentum equation can be assumed as time-independent [2].

$$\frac{\rho^j{}_I - \rho^{j-1}{}_I}{\Delta t} + \frac{G^j{}_i - G^j{}_{i-1}}{\Delta z} = 0 \tag{3}$$

$$\frac{P^j{}_{i-1} - P^j{}_i}{\Delta z} = \frac{\Delta P_f}{\Delta z} + \frac{1}{\Delta z}\left(\frac{G^j{}_i{}^2}{\rho^j{}_i} - \frac{G^j{}_{i-1}{}^2}{\rho^j{}_{i-1}}\right) \tag{4}$$

$$\frac{\rho^j{}_I h^j{}_I - \rho^{j-1}{}_I h^{j-1}{}_I}{\Delta t} + \frac{G^j{}_i h^j{}_i - G^j{}_{i-1} h^j{}_{i-1}}{\Delta z} = \left(\frac{\pi D^{in}}{A}\right)\alpha^{in}\left[T_W{}^{j-1}{}_I - T_R{}^{j-1}{}_I\right] \tag{5}$$

The finite difference equation outside the tube:

$$Cp_W M_W (T_W{}^j{}_I - T_W{}^{j-1}{}_I) = \left[A^{out}{}_{sur}\alpha^{out}\left(\frac{T^{in}{}_{air} + T^{out}{}_{air}}{2} - T_W{}^{j-1}{}_I\right) - A^{in}{}_{sur}\alpha^{in}(T_W{}^{j-1}{}_I - T_R{}^{j-1}{}_I)\right] \tag{6}$$

$$A^{out}{}_{sur}\alpha^{out}\left[\frac{T^{in}{}_{air} + T^{out}{}_{air}}{2} - T_W{}^{j-1}{}_I\right] = \xi G_{air} Cp_{air}(T^{in}{}_{air} - T^{out}{}_{air}) \tag{7}$$

For the computation, The upper hand finite difference algorithm is used. In the paper, the definite flow of the refrigerant is considered as annular flow. The Hughmark model is used for R22 [3][4]. In the equations above, ΔP_f, α^{in} are calculated out from the empirical equations associated with Hughmark model. In the two-phase section and the superheat section, these parameters are corresponding to different empirical equations [4]. Besides, α^{out} and ξ are calculated out from the empirical equations [2] too. The finite difference equations above are corresponding to dynamical characteristics. When the static characteristics are studied, the time items should be ignored. In the computation, there are two important boundary conditions, the condensing pressure in the outlet of the condenser and the rotational speed of the compressor.

There is a distinction between the simulations of double-evaporators and single evaporator, it is how to deal with the mixing process of the refrigerant flowing out from the two evaporators. Because the conditions of inlet and heat transfer are different for the two evaporators, the outlet state of the two evaporators will be different. It is a process with enthalpy unchanged, while entropy becomes high. The state of mixture can be different from either outlet of the two evaporators. In the paper, the total enthalpy is divided by the total flow mass amount, then the specific enthalpy of the mixture is got. From the specific enthalpy, the state of the mixture can be known, whether superheat or saturation. If it is in saturation, the empirical equations associated with Hughmark model will be used, otherwise, the empirical equations associated with superheat state will be used.

RESULTS AND DISCUSSIONS

1.Static characteristics

The double-evaporators not only have twice areas for heat transfer, but also are controlled separately by the EEVs connected with them. Any parameters changed in one evaporator will influence on the other. The results of static simulation present these, described as below.

Fig.3,4 present the influence on EV2 when changing the opening area of EEV1(F1) with other conditions unchanged, conditions: $T_{db1}=T_{db2}=27°C$, $T_{wb1}=T_{wb2}=19.5°C$, $F2=4.8\times10^{-7}m^2$,the two evaporators have the same wind speed of $400m^3/h$, $N_c=1600rpm$.

Fig. 3 influncing on the suction pressure with F1 changed

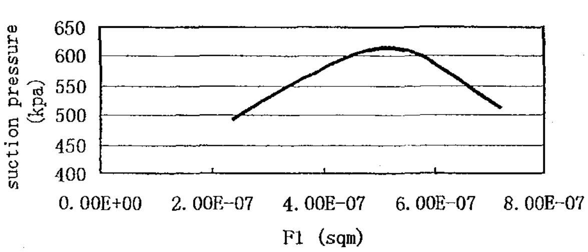


Fig. 4 influncing on cooling capacity of EV2 with F1 changed

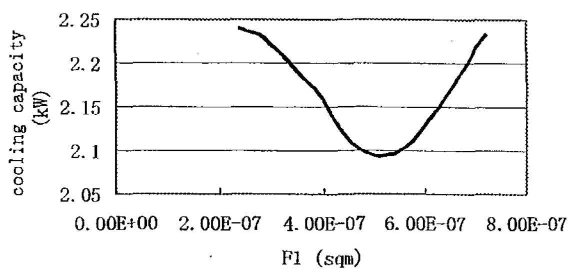


Fig.3 presents that the pressure drop doesn't change in direct ration with the opening area of the valves. When the area of EEV1 is enlarged, the loss of throttling is reduced, but pressure drops of friction and acceleration are increased. So there is a max point when the area of EEV1 being enlarged. In fig.4, when the suction pressure is improved, there is the same effects as lowing the speed of compressor to improve the suction pressure of EV2, so the average evaporating temperature of EV2 is improved and its flow amount becomes less, then the cooling capacity of EV2 becomes less. For the same reason, when the suction pressure is reduced, the cooling capacity of EV2 becomes big.

Fig. 5 influncing on the suction pressure with rooml's temperature changed

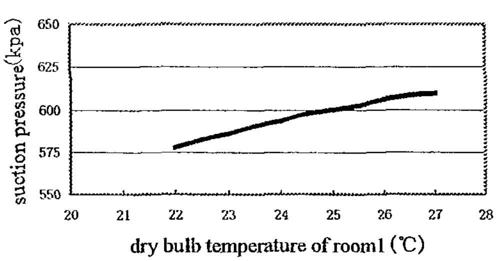


Fig. 6 influncing on cooling capacity of EV2 with rooml's temperature changed

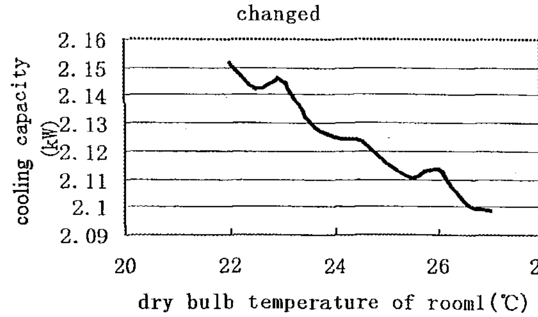


Fig.5, 6 present the influence on EV2 when changing room1's indoor temperature with other conditions unchanged, conditions: the two evaporators have the same wind speed of $400m^3/h$, N_c=1600rpm. $F1=F2=4.8\times10^{-7}m^2$, T_{db1}=22~27℃, T_{db2}=27℃, $T_{wb1}=T_{wb2}$=19.5℃.

In fig.5, The suction pressure is increased when the room1's indoor temperature ascends. This results in reducing the EV2's cooling capacity. From the results, we can draw a conclusion that when conditions of one evaporator are changed, there are influences on the other evaporator via the same suction pressure.

2.Dynamical characteristics

It is important to know the dynamical characteristics for deciding the control parameter. Especially, the air conditioner system is usually controlled in closed loop. So the dynamical characteristics of step-changing should be studied. In the paper, research on responses to step-changing is carried. The results are described as below.

Response to step-changing of EEV1's opening area

Conditions: $F2=4.8\times10^{-7}m^2$, $T_{db1}=T_{db2}$=27℃, $T_{wb1}=T_{wb2}$=19.5℃, N_c=1400rpm. The opening area of EEV1 is changed from $4.8\times10^{-7}m^2$ to $2.4\times10^{-7}m^2$, the two evaporators have the same wind speed of $400m^3/h$. The effects are described as below.

Fig7 influncing on pressure with F1 step-changed

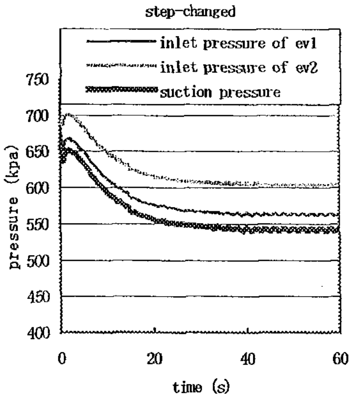


Fig8 influncing on superheat degree with F1 step-changed

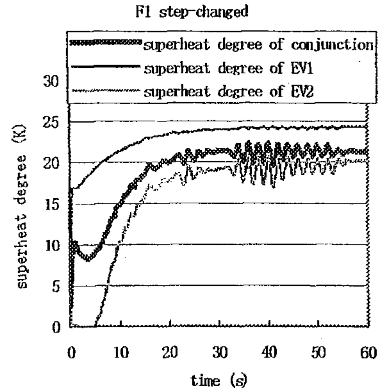
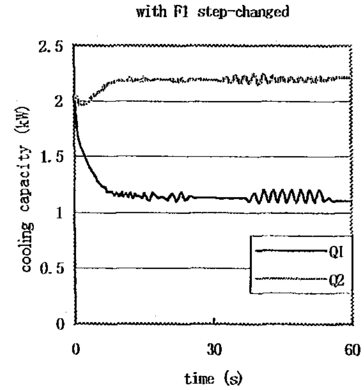


Fig9 influncing on cooling capacity with F1 step-changed



Figures as above indicate, because of change of mass accumulation behind EEV1, the pressures are sharply changed and restored in the beginning. Because F1 is reduced immediately, EV1's inlet pressure drops from 690kpa to 650kpa sharply, which influences on EV2's and suction pressure too, all the pressures are reduced. When the pressure descending, more refrigerant flows into the evaporators, so the pressures are restored a little. Since the throttling effect of EEV1 is dominant finally, EV1's inlet pressure becomes lower then EV2's, and the suction pressure becomes lower. Because of the lower suction pressure, EV2 gets more mass flow and cooling capacity. Because more mass flow can't compensate the effect of heat transfer strengthened and suction pressure reduced, EV2's superheat degree becomes bigger. According to EV1, mass flow is so little that EV1's superheat degree goes up quickly and capacity is cut down. From fig.8, the mixing process can be found. In the figures, all the parameters have transitional periods because of the transitional periods of the pressures. In fig.8, the conjunction superheat degree is the superheat degree after mixing, other superheat degrees are the ones before mixing in the two evaporators.

Response to step-changing of the wind speed of EV1

Conditions: the wind speed of EV1 is step-changed from 400m^3/h to 500m^3/h, $F1=F2=4.8\times10^{-7}m^2$, N_c=1400rpm, $T_{db1}=T_{db2}=27$℃, $T_{wb1}=T_{wb2}=19.5$℃, the wind speed of EV2 is 400m^3/h.

Fig.10 influncing on pressure with EV1's wind speed step-changed

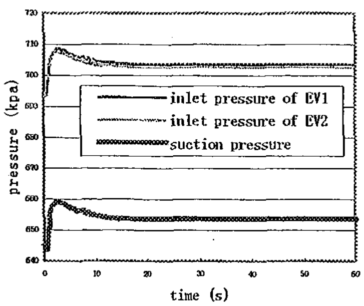


Fig.11 influncing on superheat degree with EV1's wind speed step-changed

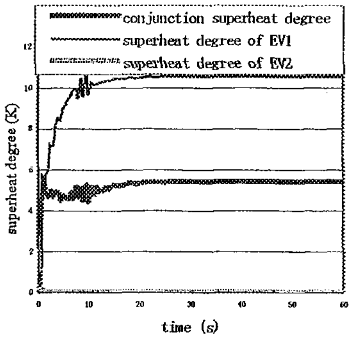


Fig. 12 influncing on cooling capacity with EV1's wind speed step-changed

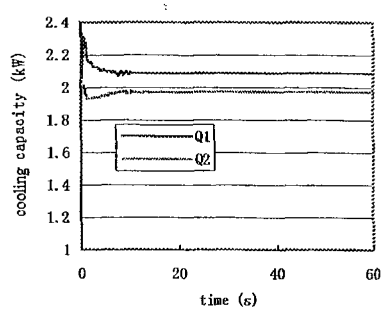


Figure 10 also shows there is an increasing and restoring process of pressure, because mass accumulation is changed in evaporators. In the beginning, increase of wind speed of EV1 causes its heat transfer coefficient to increase sharply, which results in EV1's superheat degree (as fig.11) and capacity to rise drastically (as fig.12). Then the refrigerant mass flow sucked by the compressor is less than what is discharged from EEVs, which results in the increase of mass accumulation and pressure in the evaporators. The higher pressure leads less mass flow discharged from EEVs, so the pressure and mass accumulation are restored a little. In the end, the heat transfer strengthened has such a dominant effect that all the pressures, EV1's cooling capacity and its superheat degree are raised. According to EV2, because the suction pressure is raised, its cooling capacity is reduced and it is still kept in saturation.

3.Control method proposed

The static and dynamical simulation results above show, the influences on each other can not be avoided between the paralleled evaporators. But in a double-systems air conditioner, it is unavoidable to adjust one evaporator's action to match its own load while the other evaporator doesn't need any change. How to maintain one evaporator to run constantly while the other is changed? A control method must be put forward. The simulation results tell us that the suction pressure is the key for the two evaporators influencing on each other. In the paper, a control method controlling the suction pressure is proposed to harmonize the actions to match the loads of the two evaporators. In an air conditioner system, the superheat degree is a control object too, and the optimum superheat degree for an air conditioner is rather small. Since it is a double-systems air conditioner, the method should put emphasis on the conjunction superheat (after mixing), instead of the one of independent evaporator (before mixing). So the control method is to control the suction pressure and conjunction superheat degree as the set values according to the loads. Figure 13,14,15,16 describe relations between the parameters when the suction pressure is set at 400kpa and 450kpa, and the conjunction superheat degree is set at 6K.

Fig.13, 14 describe the corresponding relation between the opening areas of EEVs and the rotational speed of the compressor in static state, when the control objects of suction pressure and superheat degree are accomplished. The three control actions should respond to one another as the figures. Fig.15, 16 are the results of control. The results show, when the mixing superheat degree is rather small, the evaporator with bigger cooling capacity is always in saturation with the other in superheat. When the paralleled evaporators get the same superheat degree, the loads are equal to each other, and the refrigerant flow amount is the biggest. The evaporator in saturation get its EEV's opening area and cooling capacity nearly unchanged, which can be decided by the suction pressure with the conjunction superheat degree set. The evaporator in superheat can change its cooling capacity and superheat degree with its EEV's opening area varied. The cooling capacity is corresponding to its superheat degree. When EEV's opening area in superheat is enlarged, its superheat degree becomes small, while cooling capacity and flow amount in it become big. Then the amount of saturating refrigerant, which is needed to compensate the superheat for maintaining the conjunction superheat degree, is changed little, which is the reason that EEV's opening area in saturation is nearly unchanged. With EEVs' opening area enlarged, the rotational speed of the compressor should be raised. After the rotational speed of the compressor gets the max point, if more refrigerant flow amount and cooling capacity are needed, the lower suction pressure object must be set for control.

Fig.13 relation of two EEVs

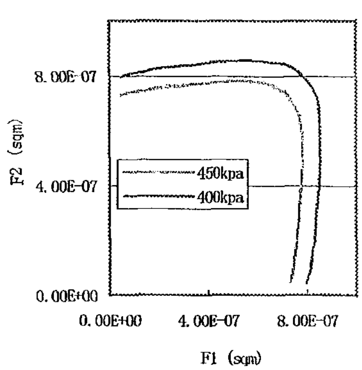


Fig.14 relation of the rotationl speed of the compressor and F1

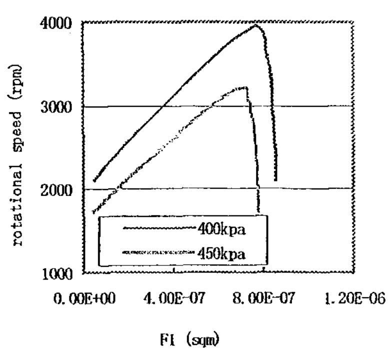
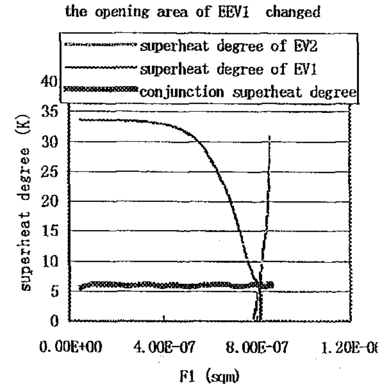
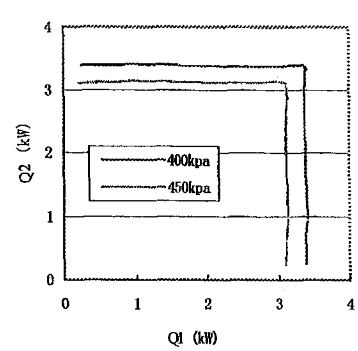


Fig.15 curve of superheat degree when the opening area of EEV1 changed



A new control method for double-systems air conditioner can be drawn from the presentation above. It is to set the suction pressure for matching the bigger one of the rooms' loads. According to the evaporator with smaller load of the room, control its superheat degree via its EEV, the superheat degree object is decided by its load and the setting suction pressure. At the same time, control the conjunction superheat degree via the other EEV, which is connected with the EV with bigger load. This is a method to separate the control objects to the different components. If the control objects are got, the different loads can be matched separately. When the loads are changed, the control objects should be reset. Then the

Fig.16 relation of cooling capacities



influences between the paralleled evaporators are reduced. The control block diagram is described as figure 17. In figure 17, the load of EV1 is supposed to be smaller than EV2's.

CONCLUSION

In the paper, the simulation results on a distributed-parameter mathematical model are listed. Then a control method is proposed. Several viewpoints can be drawn from these as below.

(1) In a double-systems air conditioner, when the boundary conditions of one evaporator are changed, there will be influences on the other evaporator definitely.

(2) The paralleled evaporators have influences on each other via the suction pressure.

(3) Since the suction pressure is the key for influencing between the paralleled evaporators, we can control the suction pressure to confine the influencing. The bigger load of the two evaporators can be matched via setting the suction pressure, basing on the setting suction pressure, the other evaporator's load can be matched via setting its superheat degree. Then the control objects are separated, and influencing between the paralleled evaporators is reduced.

(4) The control method of controlling the suction pressure is feasible for another multi-system air conditioners. It is to control the suction pressure to match the biggest load via the compressor, while controlling other evaporators' superheat degrees to match their own loads via their EEVs separately.

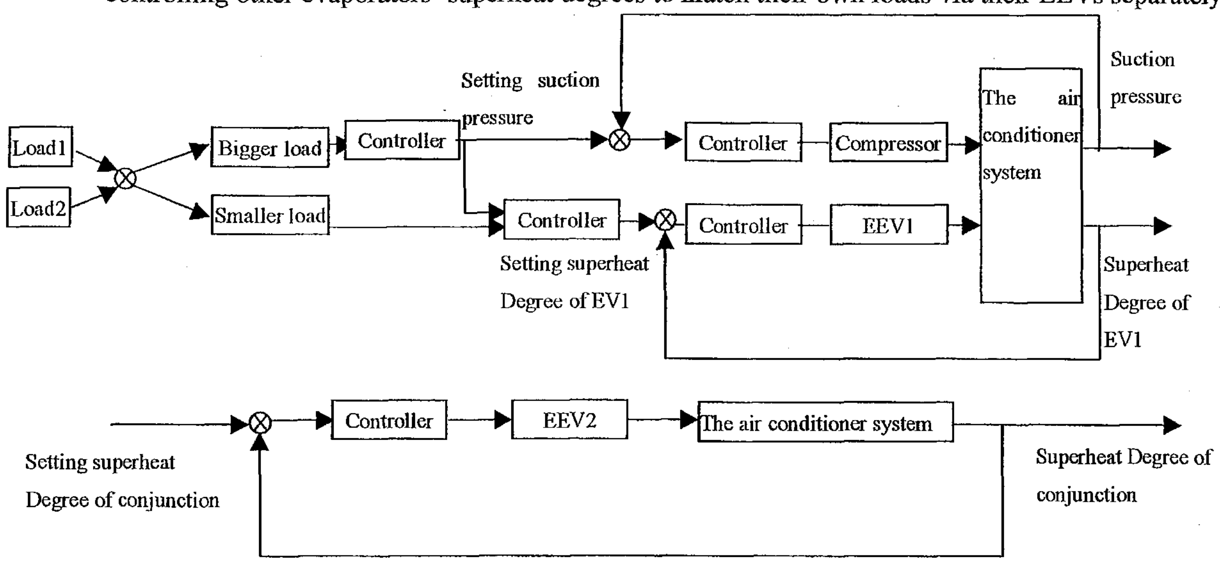


Fig.17 control block diagram

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