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DESIGN AND APPLICATION RESEARCH ON DIGITAL SCROLL COMPRESSOR IN AIR CONDITIONING SYSTEM

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ABSTRACT

New methods of numerical design on air conditioning system in buildings are presented in this paper. In order to realize numerical design, the design principle of the digital scroll compressor is introduced and analyzed, the capacity-duty ratio calculation models are built up, and the thermodynamic calculations of every link in the digital scroll compressor air-conditioning system are made, what's more, the room transient heat load is considered together with capacity adjustable air conditioner. Numerical design is analyzed from both design principle and experiment. The performance experimental results demonstrate the well adjustable capacity of the digital scroll compressor air-conditioning system.

1. INTRODUCTION

The description about the rotary machinery could be found first in U.S. Patent 1905. A French named Leon Creux invented it. But due to the problem of the rotary plate's machining precision, it hadn't entered into commercial application field. In 1970s, the appearance of numerical controlled machining devices with high precision made the research of rotary compressors possible. We can review the development of rotary machinery.

In the patented Creux's Rotary Engine, the vortex ring adopted circular involutes, and the moving vortex plate adopted revolution structure (Creux, 1905). Later, people made many improvements on rotary engines. Nordi (1925) obtained a patent for the Rotary Liquid Pump. The moving rotary plate was droved by two crankshafts that were synchronous through two gears. The conception of "Scroll Compressor" was first presented in Jones's patented Rotary Pump. Since 1970, the researches on rotary machinery have been increased and a series of theories were presented.

At the beginning of 1980, rotary machinery was applied for the first time in air conditioning compressor. The rotary machinery was used in motorcars and in floor-type air conditioner. Since then, the manufacturing technique of scroll compressors has been developed quickly, and the two types of scroll compressors have formed: one is represented by the low-pressure-cavity-flexible scroll-compressor manufactured by Corpland Company, and the other is represented by the high-pressure-cavity compressor produced by Hitachi Company. Because it had many advantages, such as continuous discharge, zero clearance volume, high efficiency, low vibration, low noise and so on, the scroll compressor is widely used in commercial products replacing most of piston compressors.

For the people are paying more attention to protect the environment, saving energy became the main trend of the design of Air Conditioner. Because the indoor heat load is changing constantly, the air conditioner is required to automatically adjust the capacity according to the change of the environmental temperature. Thus the compressor must be better. At the beginning of 1990, in Hitachi and Dakin Company, the inverter technique was applied to develop a compressor whose rotate speed could be controlled through adjusting its supply frequency and voltage. The experiment showed, that after applying the changeable rotational speed technique, the air conditioner could keep the room at a certain temperature and save much energy. Except the research on adjustable-rotational-speed, the U.S. Corpland Company began to research the capacity-adjusting-technique on scroll compressors in 1990. At the beginning of 2000, they developed successfully a volume-adjustable compressor with high efficiency and energy-saving features. Through adjusting the axial clearance between the moving and static plate of the scroll compressor, 0% and 100% discharge is realized. Then through combining the time of 0% and 100% discharge, continuous-integral-control of the compressor capacities realized. This method could be described as the combination of 0(unload) and 1(load). Thus this type of compressor was named Digital Scroll Compressor. It avoided the AC/DC converting consumption in inverter compressors, and greatly improved the energy efficiency. Numerical control is a high efficient method. This paper will focus on the refrigerating principle of the digital scroll compressor and its application in air conditioners.

2. DESIGN PRINCIPLE

In order to realize numerical design on air conditioning system, two problems must be solved. Firstly, the transient heat load of the construction must be sensed, that is, the heat load must be numerically simulated through the measurement of the sensor; secondly, the cooling capacity of the air conditioning system must be controlled continuously according to the requirement. The combination of these two aspects can realize the numerical design. The following Figure 1 shows Air Conditioner Interconnected System.

The numerical simulation on the heat load of constructions is a complex task. Even though there is much research (Gu *et al*., 1998), many problems still aren't clear now. There was no better way to control the capacity of air conditioning system before. If an air conditioning system's capacity can't be controlled continuously, the system can't be named as the nmerical air conditioning system. The popular inverter system or simple capacity-controlled-system can't meet numerical requirements.

2.1 Numerical simulation on air conditioning system's capacity

A numerical air conditioning system must have a numerical output compressor. After many years of research, U.S. Corpland Company developed the digital scroll compressor which output discharge rate is combined with 0 and 1, thus it realizes continuous control. The principle is as shown in Figure 1.

The numerical indication of the compressor's capacity output: the working conditions of the compressor are 100% and 0% capacity outputs, that are named condition 1(100%) and condition 0(0%). Through adjusting the axial clearance between two vortex plates, it realized the inversion between 0 and 1 is realized. When the compressor is at condition 1, the two vortex plates are at normal design position (Figure 2 (a)), and the compressor works at full load. When at condition 0, the two vortexes are axially separated (Figure 2 (b)), because there is no compressed gas, the discharge rate is zero, and the net watt and cooling capacity are very small. Thus through adjusting the time length (duty ratio) of condition 0 and condition 1, continuous integral control of the compressor's discharge rate is realized (Figure 3, 4).

The duty ratio is defined as

$$\alpha = \frac{\sum_{i=1}^{n}(\Delta t_1)_i}{\sum_{i=1}^{n}(\Delta t_1 + \Delta t_2)_i} \tag{1}$$

In one period change, the discharge rate is the function of α , that is

$$q_p = \psi(\alpha) \tag{2}$$

Through adjusting the value α , we can get the result of approximately continuous change.

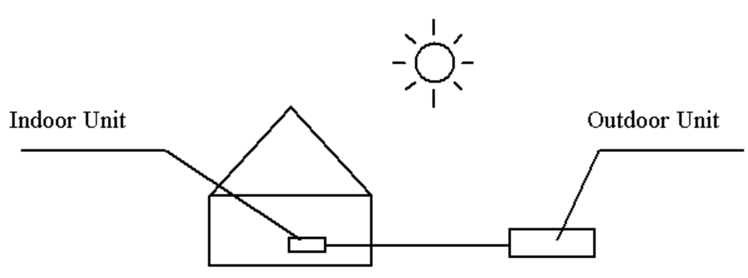


Figure 1: Air conditioner interconnected system

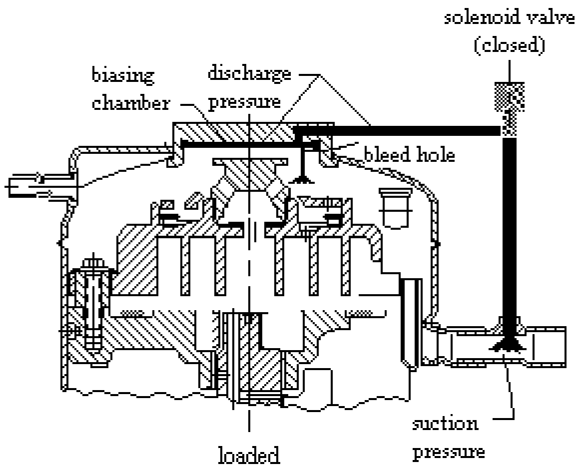


Figure 2(a): 100% Compressed condition (condition 1)

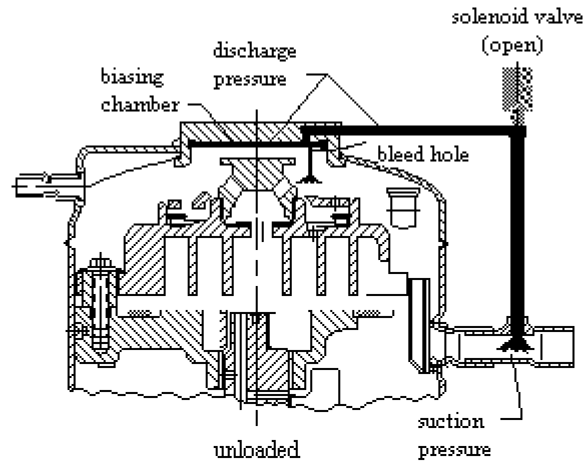


Figure 2(b): 0% Compressed condition (condition 0)

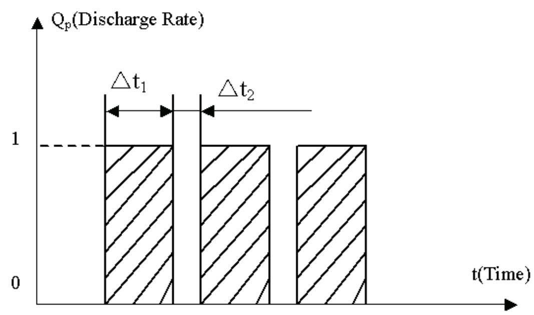


Figure 3: Interconnected drawing of time and discharge rate

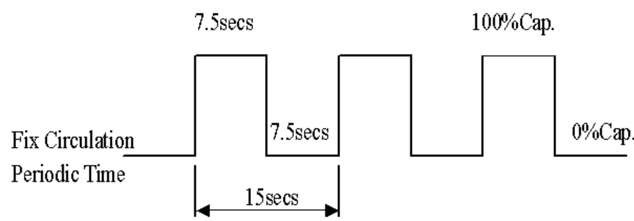


Figure 4: Principle of capacity control

When at condition 1, full load compression ratio $\varepsilon = P_d / P_i$, where P_d means discharge pressure and P_i means suction pressure. When at condition 0, the moving vortex plate and the static vortex plate are separated, $P_i \approx P_d$, $\varepsilon \approx 1$. The compressor is completely unloaded. At this time, the discharge rate is zero and consumption of watt is very small. Through the combination between 0 and 1, the real cooling capacity $Q = f(\alpha)$, which means the cooling capacity, is the function of duty ratio.

In order to establish the mathematic models of the digital scroll compressor, the following supposes are made. Firstly, the leakage loss during the compression process is ignored. Secondly, the lubricant's influence on refrigeration system is ignored. Thirdly, the compression is a polytropic process.

Concerning the digital scroll compressor showed in Figure 2 (a) and Figure 2 (b), the energy relation and controlled-body drawing are made in Figure 5 and Figure 6, respectively. In Figure 6, Q_1 is the heat rejection value from discharge to suction, Q_2 is the heat transmission value from discharge to middle compression, Q_3 is the heat transmission value from discharge to middle discharge, P_i, h_i denote the suction pressure and enthalpy, P_d, h_d denote the discharge pressure and enthalpy, Q_A, Q_B, Q_C denote the heat transmission value from the next link to its former.

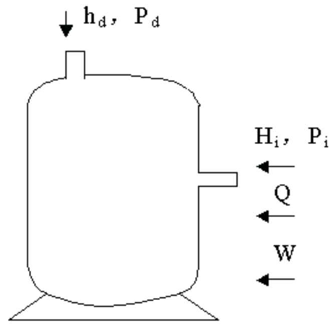


Figure 5: The total energy relation of the scroll compressor

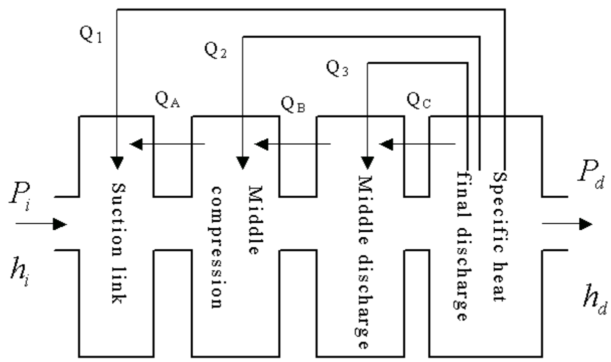


Figure 6: Energy relation between every link and their controlled-bodies

When at condition 1(100% compressed), the controlled-body equations in the above four link are the following.

2.1.1 Suction link: When suction inside pressure $p < p_i$, from the conservation of energy law, we can get

$$h_{iv}\frac{dm}{dt} - 2p\frac{dV}{dt} + \frac{dQ_1}{dt} + \frac{dQ_A}{dt} = h\frac{dm}{dt} + m\frac{dh}{dt} - V\frac{dp}{dt} \tag{3}$$

Where m is the mass, t is the time.

2.1.2 Middle compression link: Similarly, when $p_i < p < p_0$, it has

$$2p\frac{dV}{dt} + \frac{dQ_2}{dt} + \frac{dQ_B - dQ_A}{dt} = m\frac{dh}{dt} - V\frac{dp}{dt} \tag{4}$$

2.1.3 Middle discharge link: When the inside pressure $p < p_0$, it has

$$2p\frac{dV}{dt} + \frac{dQ_3 - dQ_B}{dt} - 2h\frac{dm}{dt} = m\frac{dh}{dt} - mv\frac{dp}{dt} \tag{5}$$

2.1.4 Specific heat link:

$$h_d^{'}\frac{dm}{dt} - \frac{dQ^*}{dt} - h_0\frac{dm}{dt} = h\frac{dm}{dt} + m\frac{dh}{dt} - V\frac{dp}{dt} \tag{6}$$

Where Q^* is the heat rejection value of discharge and other parts of the compressor.

2.1.5 Mass flow ratio and volume change ratio: The mass flow ratio is defined as

$$\frac{dm}{dt} = \alpha_1 A\sqrt{2\rho_i(p_i - p_i')} \tag{7}$$

Where α_1 is the inlet flow coefficient, A is the active flow area, ρ_i is the refrigerant density before entering the chamber (kg/m^3), $p_i - p_i'$ the pressure drop after passing the inlet, p_i' is the actual pressure in the chamber. According to the geometry theory of scroll compressor (Cao, 1998), the volume change ratio is given by

$$\frac{dV}{dt} = \begin{cases} \left|\pi A(A-2t)H\right| - \dfrac{\omega}{\pi}\right| & p < p_d' \quad (8a) \\ \left[2\pi(\varphi+\theta) - 4\pi^2\right]\omega R^2 H + 4\varphi R^2 \omega \left|\dfrac{3}{2}\pi - \theta\right| & p > p_d' \quad and \quad \theta < \theta^* \quad (8b) \\ \left[8\pi^2 - 2\pi(\varphi+\theta)\right]\omega R^2 H + 4\varphi R^2 \omega \left|\dfrac{7}{2}\pi - \theta\right| & p > p_d' \quad and \quad \theta > \theta^* \quad (8c) \end{cases}$$

Where A is the scroll pitch (m), t is tooth thickness (m), H is the tooth depth (m), ω is the angular velocity of motor (rad/s), R is the base radius (m), θ is the corner of crank shaft (rad), θ^* is the discharge angle (rad).
2.1.6. The thermodynamic model of the compressor is defined as

$$W = \eta_v h_v P_1 \frac{\kappa}{\kappa - 1}\left[\left(\frac{P_1}{P_2}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \tag{9}$$

$$h_2 = h_1 + P_1 \frac{\kappa}{\kappa - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \Big/ \rho \tag{10}$$

Where ρ is the refrigerant density at the compressor's inlet.

2.2 Computation of room heat load

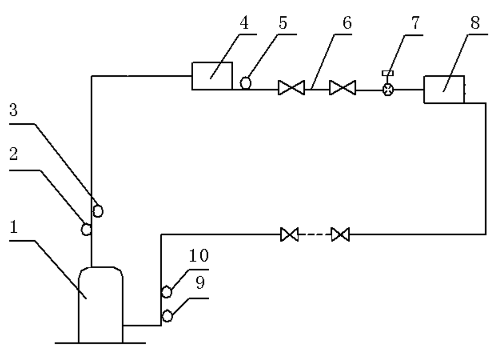
In this paper, we just consider the heat load formed by building enclosure and room heat producer. So the target function about the air-conditioner cooling capacity can be expressed as

$$Q = f(P_1, P_2, \alpha) \tag{11}$$

Where α denotes the duty ratio.

In the air conditioning interconnected system showed in Figure 1, the refrigerating principle is as Figure 7. If we set the indoor temperature sensor t_1 and outdoor ambient air temperature t_2, set the discharge temperature sensor T_d and discharge pressure sensor P_d near compressor, set the suction pressure sensor P_s and suction temperature sensor T_s at suction side, then the compressed temperature and the pressure change can be sensed. At the same time, we set mass flow detector in the system. Thus the cooling capacity Q can be defined as:

$$Q = g \times EER \cdot (1 - \delta_s) P_s \lambda_v \cdot V_h \frac{n}{n-1}\{[\varepsilon(1+\delta_0)]^{\frac{n-1}{n}} - 1\}\frac{Z_1 Z_2}{2Z_1} \cdot \alpha \tag{12}$$



1-compressor 2-discharge temperature sensor T_d 3-discharge pressure sensor P_d 4-condenser 5-mass flow detector g 6-connecting pipe 7-capillary 8-indoor heat exchanger 9-suction temperature sensor T_s 10-suction pressure sensor P_s
Figure 7: Refrigerating principle about air conditioning system

Where g is the circulation mass flow (kg/s), δ_0 is the total relative pressure consumption in the suction and discharge process, δ_s is the relative pressure consumption in the suction process, n is the polytropic exponent, λ_v is the suction coefficient, λ_h is the suction volume, Z_1, Z_2 denote the gas compressibility coefficient, EER is the energy efficiency(EER may be viewed as a constant value).

The indoor heat load is defined as

$$Q_y = Fk(t_\tau - t_1) + CQ_r \tag{13}$$

Where Q_y is the indoor heat load, F is the computation area of outside walls and roofs, k is the conductivity coefficient, t_1 is the indoor temperature, t_τ is the cool load temperature transient value (related with the indoor/outdoor ambient air temperature and building enclosure), Q_r is the body heat radiation, C is the coefficient.

It's not easy to simulate the dynamic indoor heat load. If the computational value is transferred to air conditioning system as an initial value, it is defined as

Q_y (Indoor heat load)= Q (cooling capacity of air conditioning system)

In another word, it is

$$Q_y = f(\alpha) = F\kappa(t_\tau - t_1) + CQ_r \tag{14}$$

We can see $t_1 = P(\alpha)$, that is, the indoor temperature is related to duty ratio. The target function is defined as

$$t_1 = P(\alpha) \tag{15}$$

The indoor heat load value can be transferred to the air conditioner as a reference value. A feedback system must be added in the air conditioner. Then continuously compare the difference between t_1 and set temperature t_N, correct the duty ratio, that is, target function $t_1 = P(\alpha)$, correct α according to the difference between t_1 and t_N. Through adjusting the duty ratio α , numerical design on air conditioning system can be realized.

3. PERFORMANCE EXPERIMENT

Figure 8 shows a simulating experiment that was done in an ambient simulation lab. Maintain Room2 temperature of D.B.35°C /W.B.24°C. Room2 was heat insulated, the wall heat-transfer coefficient of Room1 could be controlled. Outside Room1, maintains a heat producer with constant heat flow density, that is, supply heat for Room1 with a constant heat producer Q_0 .

3.1 Experiment 1: Indoor Temperature Curve

For Room1, the initial temp, t_0 is 29°C, and set t_N=17°C, the indoor temperature curve is the following, as shown in Figure 9. In the curve, asterisks means theoretic design value. Because the value EER was set constant, there existed a difference between theory and experiment. In fact, when the indoor temperature is too high or too low, the value EER also will change, so the curve in theory is different from the fact, and the duty ratio in fact must be corrected through $t_1 - t_N$.

3.2 Experiment 2:Capacity-Duty ratio Curve

In Figure 10, the real line means experimental value, the broken line means theoretic value, and some differences exit between of them. The main reason is: when at condition 0 (that is the two vortex plates are separated), the digital compressor will consume some work. Though no compression, the motor still rotates and consumes some work. Thus the energy efficiency decreases when the duty ratio is small.

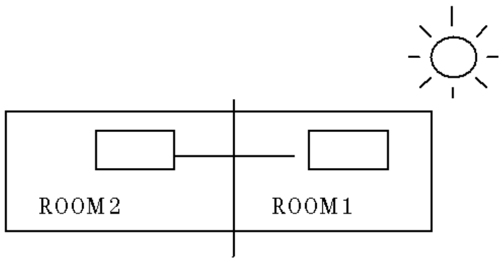


Figure 8: Sketch drawing of simulation experiment

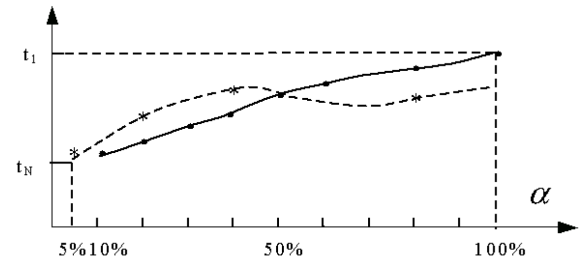


Figure 9: Indoor temperature curve

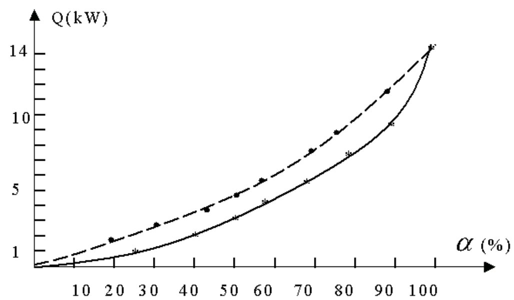


Figure 10: Capacity-Duty ratio curve

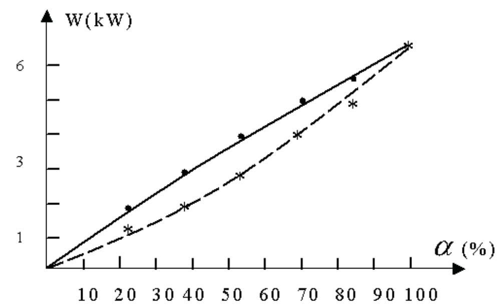


Figure 11: Work Consumption-Duty ratio curve

3.3 Experiment 3: Work Consumption-Duty Ratio Curve

Under the rated indoor/outdoor ambient temperature, the related curve of work consumption and duty ratio is approximately a beeline in theory. But similarly, the motor consumes some work when unloaded, so the experimental value is a little bigger than the theoretic value, shown as Figure 11.

3.4 Experiment 4: Energy Efficiency-Duty Ratio Curve

The energy efficiency is the most important parameter to measure the air-conditioner, so the experiment to test the energy efficiency is very important. Theoretically, under constant indoor/outdoor temperature, that is, when the condensing pressure and evaporator pressure is constant, the EER value is approximately a beeline. But in fact, for the digital scroll air-conditioning system, it's a curve. When the duty ratio is small, the time is longer at condition 0, and the unloaded power integral value of the motor is big, thus EER value decreases. With the increase of the duty ratio, the EER value increases gradually, shown as Figure 12.

3.5 Experiment 5: Pressure and current change during discharge and suction

For the digital scroll system, the experimental pressure and current curves during the discharge and suction are described in Figure 13. As we can see, in one duty ratio period, the discharge pressure in the high pressure region changes a little as wave, the suction pressure in the low pressure region and the compressor current change remarkably as approximate rectangle.

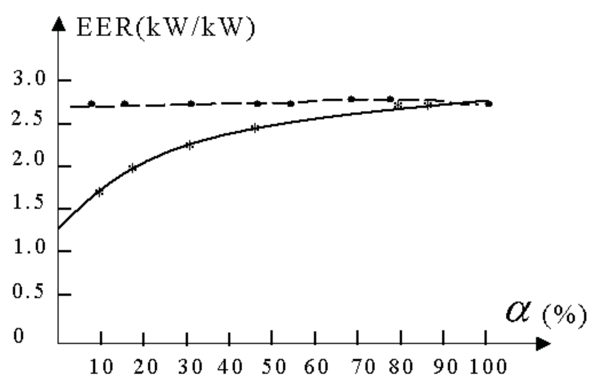


Figure 12: Energy Efficiency-Duty ratio curve

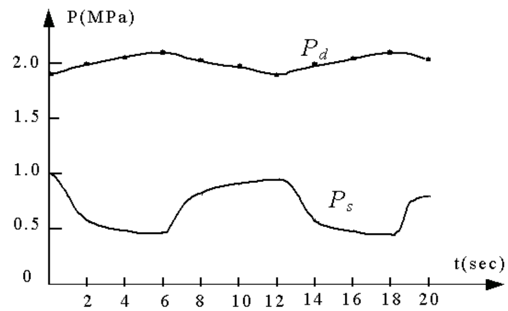


Figure 13 (a): Discharge pressure and suction pressure curves

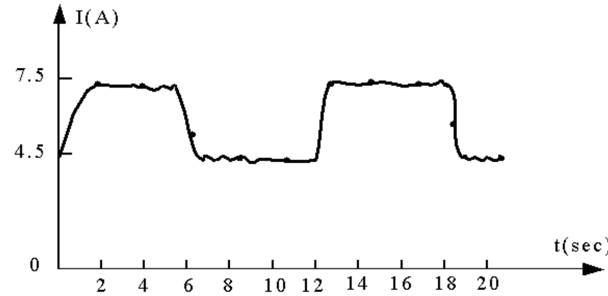


Figure 13 (b): Current curve

4. Conclusion

In order to realize numerical design onair conditioning s ystem, two problems must be solved: one is how to compute the room heat-load and the other is how to control the system capacity numerically. This paper made exploration into these two problems and the new methods on numerical design, which provided useful reference for further theoretic and experimental research.

REFERENCES

Cao X., 1998, Mathematic Model and Dimension Parameters Optimization of Scroll Compressor with Random Real Number Circles, Ph.D. thesis, Gansu Industry University, p. 12-16.

Creux, L., 1905, Rotary Engine, *U.S. Patent 801*, 182.

Gu Z.L., Yu Y.Z., Feng S.Y., 1998, *Scroll Compressor and Other Scroll Machine*. Shanxi Science and Technology Press, Shanxi, p. 131-155

Nordi, L., 1925, Improvements in or Relating to Fluid Pumps and the Like, *U.K. Patent Specification 220*, 296 January 8.