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Report on Redesign of Cooling System at Zhejiang Aucma

Project No. MP/CPR/99/168
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The Redesign and Conversion of Circuits of Freezers and refrigerators produced by Aucma Zhejiang Electrical Appliance Co. Ltd, Jiaxing, Zhejiang, China

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0.1 Subtract

redesign of the cooling circuits of five current freezer/refrigerator models: two refrigerators (LCD-207, LCD-228), two freezers (BD/C193, BD/C216) and one Water dispenser (RW-60).
The redesign should replace R12 with R600a as refrigerant. The redesign should be performed in accordance with the standards ISO7371:1995, ISO8187:1991, ISO5155:1995 (GB/T8059.1-1995, GB/T8059.2-1995, GB/T8059.3-1995 are equal to these three ISO standards) based on theoretical calculations, laboratory tests, trial and optimization. And the energy consumption of each model should not exceed the former level.

0.2 Introduction to Zhejiang Aucma

0.2.1 General information of R600a background
Domestic freezer manufactures have for the past 50 years been using R12 as the refrigerant. In Aucma this refrigerant is now being phased out as it is environmentally harmful. R600a has been identified as the replacement. Its ODP and GWP are so little, its ODP is zero, yet its GWP is 1.
This study was to redesign five of Aucma freezers, to convert it from R12 to R600a. The brief was that its performance should be the same or better and few changes as possible were to be made. It was except that only the compressor, capillary tube and charge would need to be altered.

0.2.2 Introduction to the model characters
We have a large range of models. And we use a standard designation code for the model type. The typical code are BD/C-193, LCD-207 and RW-60. in the above case the B stands for Domestic, a commercial freezer would be designated S, the D refers to a freezer, the C refer to a chilled compartment, the CD refers to have both chilled and frozen compartments (refrigerator), D/C means the chilled frozen compartments can be converted into chilled compartments. The number refers to the capacity of the model in litres.
0.2.3 Background of Zhejiang Aucma

Qingdao AUCMA Zhejiang Electrical Appliance Co., ltd. is a 100% Chinese collectively owned enterprise. It operates under the auspices of the China household Appliance Association. The company employs 550 people, 542 of them are employed in the refrigeration appliance manufacturing departments. The company lives in Jiaxing. Jiaxing located between Shanghai and Hangzhou (the capital of Zhejiang).
Production of refrigerators started in 1982. The Yiyou refrigerators factory has gradually expanded its capacity. In 1989 several high productivity machines were imported from Perros, Italy and as a result the production capacity was increased to 300,000 units per year (one shift operation). The annual production in that year amounted to 175,000 units. Later on the production fluctuated around 100,000 units levels due to managerial and market difficulties. In 1996 The Yiyou refrigerators factory was taken over by the most successful Chinese freezer manufacturer (AUCMA Group). Base on the technical and managerial experience and excellent market reputation of the mother company, our company was revitalized and after the lowest production level in 1996 the enterprise returned to and in 1998 we have already surpassed our peak production.

No new investment was made after the taken over, except for the purchase of a new painting line. However, the existing machinery has been overhauled, the quality was improved and new products were developed. We are planed to maintain the production growth of the recent years and achieve the installed capacity. In the first half of 1999 the production has already reached 100,000 units.
We have a team that is highly specialized in the design and redesign of the cooling circuit. Due to the replacement of CFC-11 with 65% Cyclopentane along with 35% Isopentane and CFC-12 with Isobutane for the refrigerant, the products of each model will be changed. We could finish all the work concerned and the Energy consumption is at the same level with R12 as refrigerant.
We have developed my refrigerator models without assistance from my major international products. Today we produce 20 models of upright refrigerators, chest freezers and smaller showcases-all N design(38℃) with wall thickness about 38mm(refrigerator) and 65mm(freezer).

These products complement well the product mix of the mother company, which concentrates on chest freezer manufacturing. The plant produces also various components for the production of refrigerators or freezers, incl. Condensers, aluminum and plastic inner liners, etc. We do not export.
The technology and a part of the appliance are of Chinese origin, the foaming machines and a part of the dry lines are from Italy (Canon and Perros), and the refrigerant charging boards were purchased from IGF, Japan and Leybold/A'Gramkow (GWO), Germany.

We have two main production building. In Plant□there is one assembly line, three cabinet and two door foaming lines. In Plant 2 one assembly line and one cabinet line are installed. In the main season (October-April) the lines are operating each day (including the weekends). The major overhauls are carried out in the very hot summer period.
The materials and components used for freezer manufacturing are of various origins. The appliances are built from local steel in the metal sheet department of the company. We are coated with paint at the plant. The inner liner of the chest freezers is aluminum sheet produced I china, the refrigerators and all doors/lids have both glass and PU/ABS lids. Majority of the compressors are from Hangli, Hangzhou and from Dongbei, Huangshi. The appliances are equipped with coated bought-in copper coil or roll-bond evaporators and write/tube condensers made by Zhejiang from copper tubes.
The polyol, CFC-11 and CFC-12 are of Chinese origin, isocyanate imported from Japan.

0.2.4 The proportion testing room at the plant of Zhejiang Aucma

ISO9001 standard requires that a proportion of freezer cabinets are tested fully loaded with tylose packs and the temperature monitored in the four top corner with "M" pack. In the room only two air conditioners can keep the ambient temperature. And the temperature in it is not controlled well. One computer could read the temperatures and can draw a curve of each cabinet from the computer and the curve we can see the freezer or refrigerator the time of opening and halting.

0.2.5 The type test room at Zhejiang Aucma

The test room are used for type testing. In the room the temperature can be controlled from 16 centigrade degrees to 43 centigrade degrees. The humidity in it can be controlled too. There are no arrangement for the control of air velocity over the refrigerators and the position and spacing relative to each other and to the wall are not controlled and monitored.

0.2.6 Capillary Tubes

The capillary tubes are cut to length in the room near the type test room. Every day there are two times for testing the length of the tubes for whether to attain the flow rate. According to the length of each model the tubes are cut from the coil with pliers and with both ends capped. The cutting capillary will be transported to the assistant plant site to use. In the assistant plant there is a special man which cut the capillary tubes for about 4mm length of both ends with capillary pliers to ensure open ends. And then one end is welding to the end of the evaporator. After this use high press nitrogen to testing whether the whole system is good, then crap the open ends of capillary tubes.

0.2.7 Freezer construction

The inner liner is assembled and taped. It is then fitted with pre-coiled evaporators either steel coated inside and out with copper. The spacer are fitted, the evaporator coils are spaced, taped to the side the aluminum inner. The assembly then has the suction tube and the straight through accumulator added. The capillary tube is then attached. These are braised into position immediately and fresh and caps fitted over the open ends pipes. The outer skin of steel if fitted and the whole unit is foamed and cured.

0.2.8 Assembly

Condensers are assembled in the foamed cabinet with 4 screws. The compressor is next fitted and the overload and cut out attached to it. Wiring is completed. The suction line is then uncapped and braised a filter dryer fitted and capillary tube and compressor braised. The system is then evacuated for half an hour. The units are then charged at a charging station which first is connected to the unit using a special valve. It is then evacuated, the vacuum checked, the unit filled with a pre-weighed charge of refrigerant and disconnected. The tube is then crimped towards the compressor with this crimp remaining in place while the outer tube is snapped of at the first crimp and then end braised to seal the unit.

0.2.9 Testing

All units on the production line are tested electrically for insulation, earth leakage and earth continuity. The leak test is carried out in 2 stages, the low pressure braising joints are checked with the compressor off and the high pressure joints are checked with the compressor running. Checks are carried out using a electronic halogen leak detector. The units failed these leakage tests are shunted onto a side line for repair. It is repaired that the repair costs is very small, because the labor cost is cheap. The units are then put onto a 1.5 hour pull-down test during which time the temperature at one position (not contacting the inner) reduced below -18□. Temperature as low as

-32□ can be recorded at this test. 0.5% of production fails this test. 0.2% because of leakage, 0.1% because of vacuum problems, 0.1% because of other reasons.

Part 1 Training Course

1.1 Training Course at Zhejiang Aucma

1.1.1 the Timetable of the Training (table 1)

Day	Contents	Explanation	Remark
March 24	Familiarization with refrigerators, facilities and the production line.	Collect data from the workshop	
March 25	Lecture on measuring refrigerator performance.	Practical instrumentation of refrigerators	
March 26	Practical result of measurements.	Lecture on improving refrigeration circuits	Practical test
March 27	Practical results of test.	Lecture on improving refrigerator performance	Practical, modification of the freezers and retest
March 28	Practical lecture	Lecture	Practical
March 31	Practical lecture	Collect information on refrigerator components	Practical
April 1	Practical lecture	Workshop on product safety	Practical
April 2	Practical lecture	Lecture	Practical
April 3	Practical lecture	Lecture	Practical
April 4	Questions, summary and future program.		

Table 1 (by Mr. Wang Yizhao and Mr. Yuan Liqiang)

1.1.2 The material or equipment needed to be replaced, repaired and purchased for the redesign

The following materials and equipments will be needed for the practical workshops. And Mr Wang Yizhao and Mr. Yuan Liqiang will bring some equipment.

Δ the test room will be repaired

Δ 3 units of serval different freezer models

Δ 30 temperature sensors for attaching to freezer circuits

Δ data logger

Δ Refrigerant manifold and gauges

Δ Isobutane and MAX-95 (Charging Machine).

Δ Vacuum Pump

Δ Small charging cylinder and weighing balance. If not ready, we can use it somewhere near the factory.

1.2 Training report at Qingdao Aucma

1.2.1 Objective

A practical course to learn how to measure the performance of a domestic refrigerator or freezer covert it to operate on a different refrigerant and optimize its performance. The principal skills learnt will be conduction development tests, analysis data, calculations to predict performance changes and decision making for refrigeration system optimization.

1.2.2 Course Outline

The course will be lasted for 2.5 weeks. It will be practical course with participants performing test on a refrigerator (one per participant), learning to interpret the results and draw conclusions. They will then learn how to decide on what changes to make, carry out the modifications and measure the result.

Tuition and classroom sessions will be tailored suit the individual level of knowledge of every participant and will be based on the workshop notes supplied and taught in formal lectures in Chinese at Qingdao Aucma

1.2.3 Course content

April 7
Instrumentation
Measuring Temperature, pressure and energy consumption
Manufacture of thermocouples
Calibration of instruments
The pull down and continuous running test

April 8

Analysis data from pull-down and continuous running tests

April 9

How to choose new refrigerants

Charging a refrigerator or freezer

Measuring performance changes

April 10

Optimization of refrigeration systems – the refrigerant charge

Recovering refrigerant from a system

Workshop methods of charging

April 11

Optimization of refrigeration systems2 – the refrigerant charge

April 14

Optimization of refrigeration systems 3 – capillary tube

Modifying the capillary tube

April 15

Optimization of refrigeration systems 4 – the compressor calculation 1

April 16

Optimization of refrigeration systems 4 – the compressor calculation 2

April 17

Optimization of refrigeration system 5 – the evaporator size

April 18

Optimization of refrigeration system6 – the condenser size

April 21

Optimization of refrigeration system7 – heat loss calculation

April 22

Optimization of refrigeration system8 – the position of the thermostat

April 23

How to identify the faults from the results

Air in system

Blocked capillary tube

Damaged capillary tube
Much or little of the refrigerant charging
Long or short of the capillary tube

1.2.4 Program and fees

The course has been run from April 7 to April 23 in Qingdao Aucma Group. All the measuring equipment for the course will be provided by Qingdao Aucma. Testing will take place in the test room and lectures and tutorials in the conference room. And include some visiting to the scene spot.

Item	Man days	Cost (RMB Yuan)
Preparation and clearing up some documents	10	8,000
Travel to and from Jiaxing to Qingdao		12,000
Course teaching and supervision	20	12,000
Course materials, lunch and beverages		3,000
Provision of a cultural guide for 2 weekend	4	2,000
Traffic fees in Qingdao for 2.5 weeks		2,000
Cultural visits (entry fees etc total 6 days)		12,000
Total	34	51,000

1.3 Technical note of Isobutane (R600a)

1.3.1 Application

Isobutane is used as refrigerant for replacement of R12 and R134a. Isobutane is environmentally sound. The main difference between R12/R134a and R600a is that R600a is flammable.

1.3.2 Physical properties

Chemical formula C_4H_{10}
Molar mass: 58.1 kg/Kmol

Boiling point at 1013 mbar: -11.7℃

Liquid density:

Boiling point (-11.7℃) 594.1kg/m^3

At 20℃ 557.6 kg/m^3

At 25℃ 551.7 kg/m^3

Gas density:

At 0℃ 2.6970 kg/m^3

At 20℃ 2.4801 kg/m^3

At 25℃ 2.4355 kg/m^3

Vapor pressure (Absolute pressure)

At 20℃ 3.03 bar

At 30℃ 4.05 bar

At 40℃ 5.32 bar

At 50℃ 6.87 bar

1.3.3 Health and safety aspects:

Isobutane in high concentrations will displace oxygen at the risk of unconsciousness and in worst cases it might be fatal.

Isobutane has a low order of inhalational toxicity. Thus, there are no apparent ill-effects from breathing concentrations up to 5% isobutene for 2 hours.

Isobutane is very fire and explosion dangerous. Use of open fire and smoking when working with Isobutane are not allowed.

In case of skin and eye contact with liquid Isobutane, frostbite may occur. Always use gloves and goggles when working with Isobutane.

1.3.4 Explosion levels:

Autoignition temperature 462℃

Low explosion level (LEL) 1.8% vol. (45g/m^3, 20℃)

High explosion level (HEL) 8.4% vol. (210g/m^3, 20℃)

1.3.5 Environmental effect:

Environmental effect relative to R12 and R134a:

Item	R600a	R12	R134a
ODP (Ozone depletion potential)	0	1	0
GWP (Global warming potential (100 year)	3	8500	1300

1.3.6 Relative charges required

System Volume	Weight R600a	Weight R12	Weight R134a
50 ml	28g	67g	62g
100 ml	56g	135g	124g
200 ml	113g	269g	249g
500 ml	282g	673g	622g
1 liter	563g	1350g	1240g
2 liter	1130g	2690g	2490g
5 liter	2820g	6730g	6220g
10 litre	5630g	13460g	12430g

1.4 Compressor messages

1.4.1 Refrigerant

- The R12 is quite different from the R600a and R134a with the different refrigerant, the compressor's design will bi changed accordingly. So users should strictly respect the relative refrigerant-using regulation.
- The refrigerant should not be over-charged, the quantity the refrigerant charged is enough as long as the compressor can come into normal operation.
- Warning
  For the sake of safety, R600a refrigerant should not be charged over 150g. The density of the R600a in the atmospheres (calculate according to its volume) should not exceed 1.98% (LEL) to 8.5% (UEL), the kindling with more than 460℃ will ignite it.

1.4.2 Lubricant

- The compressor has been fully charged with the dewatered and clean lubricant which usually enough for all the compressor's service life. The lubricant should continuously flow back to the compressor's cylinder to avoid wandering in the refrigerating system and ensure the proper circulation of the gas.
- Based on the model BD/C191 with the refrigerant of R12, and according to some documents, we changed some configure

1.4.3 Transportation and Storage

- Keep the compressor upright and avoid heavy dashing during transportaion.
- Compressor should be kept in dry (the relative humidity should not exceed 80%) and well-ventilated place with no corrosive gas surrounding
- The compressor should not be piled more than 2 layers. Not downside permitted during loading and unloading.

1.4.4 Caution

- Ambient temperature of the compressor operation is 10-43℃
- The sealing rubber block is not permitted to be removed until compressor begins to use.
- The refrigerant should be charged as soon as possible within 15 minutes once the sealing rubber block is removed. (suggesting to remove the process tube's block first.)
- The compressor's grounding must be reliably connected the refrigerating device at all times.
- The requirements of the compressors' installation: the inkling angle with the level should not exceed 5 degree.
- The compressor, if in the vacuum circumstance, is forbidden to take the pressure test and starting test.
- Don't start the compressor if the starter is not fully prepared.
- The stocking period must be less than 6 months after the date of production. If longer, you have to check whether the filled nitrogen is sufficient. Replacement must be done if necessary.

Part 2 Theoretical Calculation

2.1 Theoretical calculation for the model BD/C-193

2.1.1 Design conditions of BD/C193

Freezer BD/C-193 Design Conditions (according to the standards GB/T8059.3-1995) :

1) Environmental conditions: ambient temperature t_a=32□, relative humidity φ=75±5%;
2) Inside temperature of the freezer: t≤-18□;
3) Storage volume: L=193L;
4) Refrigerant: R600a;
5) Blowing agent: isopentane

Freezer BD/C-193 Construction (shown in **Figure 1**):

1) Opening type: top-opening type
2) Overall dimensions:847mm*1073mm*548mm (height*width*depth);
3) Thickness of the adiabatic wall: right side=57mm, left side=57mm, front side=56mm, back side=56, up side=48.5mm, underside=58mm;
4) Condenser construction: steel wire-tube type, the diameter of the tube and wire are 6mm(d_b),1.6mm(d_w); and the thickness of the steel tube is 0.71mm; space between tubes is 48mm(s_b), and space between wires is 5.5mm(s_w);
5) Evaporator construction: steel tube-board type, the board is made of aluminium and the thickness is 0.4mm, the diameter of the steel tube is 8mm and the thickness is 0.71mm;
6) Expansion device: capillary, the diameter is 1.8mm;
7) Compressor type: hermetically sealed compressor refrigerating compressor (plunger type);

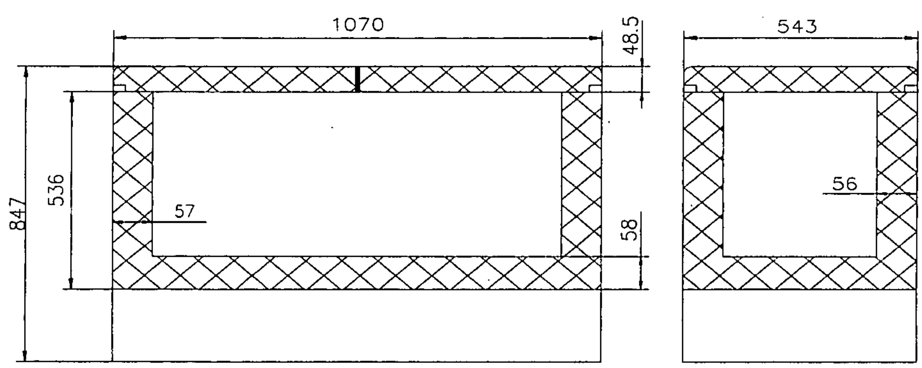

Figure 1

2.1.2 Theoretic Calculation of Cooling System

(1) Calculation of heat loss Q_F

The heat loss including three parts, one is the heat loss through the box of the freezer Q_{1F}, the second is for opening the freezer door Q_{2F}, the third is for cooling the goods Q_{3F}.

□Heat loss through the box of the freezer Q_{1F}

Q_{1F} including two parts, one is the heat loss through the adiabatic wall Q_a, the second is through the door Q_b.

$$Q_a = kA(t_1 - t_2)$$

$$K = \frac{1}{\frac{1}{a_1} + \frac{\delta}{\lambda} + \frac{1}{a_2}}$$

K— Coefficient of transfer heat, unit $(W/(m^2 \cdot k))$;
t_1— Outside exterior temperature of the freezer, unit (□);
t_2— Inside exterior temperature of the freezer, unit (□);
δ— thickness of the adiabatic wall, unit (m);
λ— heat conductivity of the adiabatic wall, unit $(W/(m \cdot k))$; λ=0.04 $W/(m \cdot k)$
a_1— exterior temperature of the air outside the freezer, unit $(W/(m^2 \cdot k))$; a_1=11.3 $W/(m^2 \cdot k)$
a_2— exterior temperature of the air inside the freezer, unit $(W/(m^2 \cdot k))$; a_2=1.16 $W/(m^2 \cdot k)$
A— area of the freezer exterior, unit(m^2)

Heat loss of every sides of the Freezer (shown in **Table 1**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	58	56	56	57	57
$K\,(W/(m^2 \cdot k))$	0.462	0.418	0.425	0.425	0.421	0.421
$A(m^2)$	0.567	0.567	0.549	0.549	0.272	0.272

t_1- t_2(□)	50	62	50	50	50	50
Q(w)	13.09	14.69	11.66	11.66	5.72	5.72

Table 1

Q_a=13.09+14.69+11.66+11.66+5.72+5.72=62.54w

Q_b=15%Q_a=0.15*62.54=9.38w

Q_{1F}= Q_a+ Q_b=62.54+9.38=71.92w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.193m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to -18□,106.75KJ/kg;

V_a— bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.193 \times 1 \times 106.75}{3.6 \times 0.9} = 6.36w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ/(kg·k);

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L=0.965kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.965 \times 4.19 \times 25 + 0.965 \times 333 - 0.965 \times 2 \times (-2)]}{2 \times 3.6} = 59.20w$$

$$Q_F = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F = 71.92 + 6.36 + 59.20 = 137.48w$$

(2) Circular thermodynamic calculation

□Parameter for design (Shown in **Table 2** and **3**)

Rated states of cooling system

State parameter	Condensation temperature (t_k)	Evaporator temperature (t_0)	Inspiratory temperature (t_G)	Super-cooling temperature (t_s)

Value of design	54.4	-23.3	32(80)	17
Parameter reference	t_k=32+22.4	t_0=-18-5.3	Ambient temperature	t_s=32-15

Table 2

Physical parameter

Parameter name	Sign	Unit	Design value
Condensation pressure	P_k	MPa	0.7814
Evaporator pressure	P_o	MPa	0.0623
Saturated steam enthalpy after the evaporator	h_1	KJkg	523.63
Steam-gas enthalpy before entering the compressor	h_1'	KJ/kg	612.41
Steam-gas bulking value before entering the compressor	v_1'	M^3/kg	0.68371
Steam-gas enthalpy before entering the cylinder	h_1''	KJ/kg	700.36
Steam-gas bulking value before entering the cylinder	v_1''	M^3/kg	0.80290
Temperature of the steam-gas	t_{2s}	℃	95.9
Enthalpy of the saturated steam at the condensation temperature	h_2	KJ/kg	629.76
Enthalpy of the steam-gas	h_2''	KJ/kg	1105.8
Enthalpy of the super-cooling liquid (32℃)	h_3	KJ/kg	275.44
Enthalpy of the super-cooling liquid (17℃□	h_3'	KJ/kg	241.23
Enthalpy of the refrigerant at the entrance to the evaporator	h_4	KJ/kg	241.23
Enthalpy of the compression gas □32℃□	h_{2s}	KJ/kg	714.04

Enthalpy of the compression gas □80℃□	h_{2s}'	KJ/kg	836.31

Table 3

□Calculation

- Refrigerating capacity per mass

$$q_0 = h_1 - h_4 = 523.63 - 241.23 = 284.4kJ/kg$$

- Refrigerating capacity per volume

$$q_v = \frac{q_o}{v_1^1} = \frac{h_1 - h_4}{v_1^1} = \frac{284.4}{0.68371}kg/m^3 = 415.96kg/m^3$$

- Compress work per mass

$$w_i = h_{2s} - h_1^1 = 714.04 - 612.41.04 = 101.63kJ/kg$$

- Refrigerating coefficient

$$\varepsilon = \frac{q_o}{w_i} = \frac{284.4}{101.63} = 2.79$$

- Condensing heat capacity per mass

$$q_k = h_2^1 - h_3 = 832.654 - 275.44w = 557.21w$$

- Cycle mass of refrigerant

$$G_a = \frac{Q}{q_o} = \frac{137.48 \times 3.6}{284.4}kg/h = 1.73kg/h$$

- Condensing heat capacity

$$Q_k = G_a \bullet q_k = G_a \bullet (h_2 - h_3) = \frac{1.81 \times 557.21 \times 10^3}{3600}w = 267.76w$$

- Overheat vapor suctioning by compressor

$$v_s = G_a \bullet v_1^1 = 1.73 \times 0.68371m^3/h = 1.182m^3/h$$

(3) Calculation for choosing the compressor type

□Calculate the coefficient of gas transmission

- Volume coefficient

$$\lambda_v = 1 - c\left[\left(\frac{p_k + \Delta p_k}{p_o}\right)^{\frac{1}{m}} - 1\right]$$

$$\Delta p_k = 0.1p_k$$

c — relative clearance volume, c=0.025;

m— expand coefficient, m=1.

$$\lambda_v = 1 - 0.025 \times \left[\left(\frac{0.7814 + 0.1 \times 0.7814}{0.0623} \right)^{\frac{1}{1}} - 1 \right] = 0.68$$

- Pressure coefficient

$$\lambda_p = 1 - \frac{1+c}{\lambda_v} \bullet \frac{\Delta p_o}{p_o}$$

$$\Delta p = 0.05 p_o$$

$$\lambda_p = 1 - \frac{1+0.025}{0.68} \times 0.05 = 0.924$$

- Temperature coefficient

$$\lambda_t = \frac{T_1''}{aT_k + b\theta}$$

a,b — Coefficient, a=1.15,b=0.25;

T_1'' —Inspiratory temperature, T_1''=273+80=353k;

T_k— Condensation temperature, T_k=273+54.4=327.4k;

θ— Superheat degree, θ= T_1''-(t_o+273)=103.3k.

$$\lambda_t = \frac{353}{1.15 \times 327.4 + 0.25 \times 103.3} = 0.87$$

- Leak coefficient

$\lambda_l = 0.99$ (According to the experience)

$$\lambda = \lambda_v \bullet \lambda_p \bullet \lambda_t \bullet \lambda_l$$

$$\lambda = 0.68 \times 0.924 \times 0.87 \times 0.99 = 0.542 m^3/kg$$

□Theoretic carry capacity

$$V_h = \frac{V_s}{\lambda} = \frac{1.182}{0.542} = 2.18 m^3/h$$

□Cooling capacity of compressor

$$Q_o = q_v \bullet V_h \bullet \lambda$$

$$Q_o = 415.96 \times 2.18 \times 0.542 = 491 kJ/h = 0.136 kw$$

□Power of the compressor

a) Theoretical adiabatic power

$$p_o = \frac{G_a \bullet w_i}{3600}$$

$$p_o = \frac{1.73 \times 101.63}{3600} = 48.8w$$

b) Indicated power

$$p_i = \frac{p_o}{\eta_i}$$

$$\eta_i = \frac{T_o}{T_k} + bt_o$$

b=0.0005, T_o=273-23.3=249.7k, T_k=273+54.4=327.4k;

$$\eta_i = \frac{249.7}{327.4} + 0.0005 \times (-23.3) = 0.704$$

$$P_i = \frac{P_o}{\eta_i} = \frac{48.8}{0.704} = 69.32w$$

c) Friction power

$$P_m = \frac{p_m \bullet v_h}{36.72}$$

P_m— average friction, p_m=0.45Mpa

$$P_m = \frac{0.45 \times 2.18}{36.72} = 26.71w$$

d) Motor shaft horse power

$$P_e = P_i + P_m = 69.32 + 26.71 = 96.03w$$

e) Electric power and motor efficiency

$$P_{el} = \frac{P_e}{\eta_{mo}}$$

$$\eta_{mo} = 0.82$$

$$P_{el} = \frac{96.03}{0.82} = 117w$$

According to the result of the theoretic calculation including the theoretic cooling capacity, power of the compressor, electric power and motor efficiency etc, and the design parameter of R600a, we choose the compressor which is produced by JIAXIPERA company, model BM1112CY, external voltage 220V, input power 126w, external cooling capacity 148w.

(4) Calculation of condenser

□Head load in overheat Zone Q_1 and in saturated zone Q_2

$\beta = \frac{Q_1}{Q_2} = \frac{h_{2k} - h_2}{h_{2k} - h_3}$ (The percent of the heat load in overheat zone and heat load in saturated zone)

h_{2k}— enthalpy, states t=80℃,p=0.7814Mpa, h_{2k}=681.02kJ/kg;

$$\beta = \frac{681.02 - 629.76}{681.02 - 275.44} = 0.126$$

$$Q_1 = 0.126 Q_k = 0.126 \times 267.76 = 33.74 w$$

$$Q_2 = (1 - 0.126) Q_k = 0.874 \times 267.76 = 234.02 w$$

□Deference in temperature of overheat □t_1 and saturated zone □t_2

a) $\Delta t_1 = \dfrac{t_h - t_k}{\ln \dfrac{t_h - t_k}{t_k - t_a}}$

t_h— overheat gas temperature, t_k=80℃;

$$\Delta t_1 = \frac{80 - 54.4}{\ln \frac{80 - 32}{54.4 - 32}} = 33.59\ ℃$$

b) $\Delta t_2 = t_k - t_a$

$$\Delta t_2 = 54.4 - 32 = 22.4\ ℃$$

□natural convection heat transfer coefficient

c) overheat zone

$$a_{of} = 0.94 \frac{\lambda_f}{d_e} \left[\frac{(s_b - d_b)(s_w - d_w)}{(s_b - d_b)^2 + (s_w - d_w)2} \right]^{0.155} (P_{rf} G_{rf})^{0.16}$$

de —Equivalent diameter;

$$d_e = s_b \left[\frac{1 + 2 \frac{s_b}{s_w} \bullet \frac{d_w}{d_b}}{\left(\frac{s_b}{2.76 d_b} \right)^{0.25} + 2 \frac{s_b}{s_w} \bullet \frac{d_w}{d_b} \eta_f} \right]^4$$

η_f— rib efficiency, η_f=0.85;

$$d_e = 0.048 \times \left[\frac{1 + 2 \times \dfrac{0.048}{0.0055} \times \dfrac{0.0016}{0.006}}{\left(\dfrac{0.048}{2.76 \times 0.006} \right)^{0.25} + 2 \times \dfrac{0.048}{0.0055} \times \dfrac{0.0016}{0.006} \times 0.85} \right]^4 = 0.0637$$

g— acceleration of gravity, $g=9.8m/s^2$;

β— expend coefficient, $\beta=0.0031$;

r— Sport glutinosity of the air, $r=17.84*10^{-6}m^2/s$

$$G_{rf} = g\beta\Delta t_1 d^3 / \gamma^2$$

$$G_{rf} = 9.8 \times 0.0031 \times 33.59 \times \frac{(0.0637)^3}{(17.84 \times 10^{-6})^2} = 826997$$

$$P_{rf} = 0.698$$

$$a_{of}' = 0.94 \times \frac{2.82 \times 10^{-2}}{0.0637} \times \left[\frac{(48-6)(5.5-1.6)}{(48-6)^2 - (5.5-1.6)^2} \right]^{0.155} \times (0.698 \times 826997)^{0.26} = 9.07 w/m^2 \bullet k$$

d) Saturated zone

λ_f— when the temperature is 43.2℃, thermal conductivity of the air, 0.278W/(m·k)

r— Sport glutinosity of the air, $r=17.28*10^{-6}m^2/s$

$$P_{rf} = 0.699$$

β— expend coefficient, $\beta=0.00316$;

$$G_{rf} = 9.81 \times \frac{1}{273+43.2} \times 22.4 \times \frac{(0.0637)^3}{(17.28 \times 10^{-6})^2} = 599872$$

$$a_{of}' = 0.94 \times \frac{2.78 \times 10^{-2}}{0.0637} \times \left[\frac{(48-6)(5.5-1.6)}{(48-6)^2 - (5.5-1.6)^2} \right]^{0.155} \times (0.699 \times 599872)^{0.26} = 8.35 w/m^2 \bullet k$$

□Radiation heat transfer coefficient

e) Overheat zone

$$a_{or}' = 6.67\varepsilon \frac{\left(\dfrac{T_1 + 273}{100} \right)^4 - \left(\dfrac{T_2 + 273}{100} \right)^4}{\Delta t_1}$$

ε— Dark coefficient, $\varepsilon=0.97$;

T_1— average temperature of the condenser, $T_1=65.59$℃

T_2— ambient temperature, T_2=32 ℃

$$a_{or}' = 5.67 \times 0.97 \times \frac{\left(\frac{65.59+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{33.59} = 7.35 w/m^2 \bullet k$$

f) Saturated zone

T_1— average temperature of the condenser, T_1=54.4℃

$$a_{or}'' = 5.67 \times 0.97 \times \frac{\left(\frac{54.4+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{22.4} = 6.96$$

□ Heat transfer area

$$\eta_o = \frac{a_b + a_p \eta_f}{a_b + a_p} \text{ (Exterior efficiency)}$$

a_b— Outside exterior area per m, $a_b = 0.006\pi$

a_p— Inside exterior area per m,

$$a_p = 2 \times \left(\frac{1}{s_w} + 1\right) \pi d_w s_b$$

$$\eta_o = \frac{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \times 0.0016\pi \times 0.048 \times 0.85}{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \pi \times 0.048 \times 0.0016} = 0.8763$$

g) Overheat zone

$$A' = \frac{Q_1}{\left(a_{of}' + a_{or}'\right) \eta_o \Delta t_1}$$

$$A' = \frac{33.74}{(9.07 + 7.35) \times 0.8763 \times 33.59} = 0.0700 m^2$$

h) Saturated zone

$$A'' = \frac{234.02}{(6.96 + 8.35) \times 0.8763 \times 22.4} = 0.7786 m^2$$

Total heat transfer area

$$A = A' + A'' = 0.0700 + 0.7786 = 0.8486 m^2$$

□The size of the condenser

i) The length of the steel tube

$$L = \frac{A}{a_b + a_p}$$

$$L = \frac{0.8486}{0.088 + 0.0188} = 7.94m$$

Because when we use the freezer, the condenser would be dirty, in order to make no influence on it, we should increase the length, so the actual length is 1.05L, 8.30m, and we also do not want to change the mold of the condenser, so the width of the condenser is the same as the mold, its width is o.45m, the depth is 0.768m. Generally the length of the steel wires is longer than the condenser depth, so the actual length is 0.788m.

(5) Calculation of the evaporator

The area of the evaporator is not changed, its is 1.23m^2(A)

The rib efficient is 3.5(β)

So the exterior of the steel tube is

$$A_e = \frac{A}{\beta} = \frac{1.23}{3.5} = 0.353m^2$$

$$L = \frac{A_e}{0.008\pi} = 14m$$

(6) Calculation of Capillary(according to the calculation of R12)

In the conditions of the same states, the capillary length of R12.

a) $$L = \frac{\Delta P \bullet R_e^{\ 0.25} \bullet d}{0.1582 w^2 \bullet \rho}$$

b) $$W = \frac{G_a v}{\frac{\pi}{4} \bullet d^2}$$

d— inside diameter of the capillary, d=0.00066m;

v— bulking value of the liquid, t=15℃,v=4.21*10^{-3}m^3/kg;

G_a— Cycle mass of refrigerant R12, G_a=2.96kg/h;

$$W = \frac{2.96 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00066)^2 \times 3600} m/s = 10.12 m/s$$

R_e— Reynolds number

- $R_e = \dfrac{Wd\rho}{\mu}$

ρ— average density, ρ=237.5kg/m3;

μ— average glutinosity, $\mu=0.217*10^{-3}$Pa·s

$$R_e = \frac{10.12 \times 0.00066 \times 237.53}{0.217 \times 10^{-3}} = 7455.70$$

$$L = \frac{1.216 \times 10^6 \times (7455.70)^{0.25} \times 0.00066}{0.1582 \times (10.12)^2 \times 237.55} m = 1.94m$$

According to the experiment, we find the capillary length which using the R600a is shorter than the capillary which using the R12, so the actual length is 0.75L, 1.44m.

(7) The mass of the perfusion(According to the R12)

$V_n = 0.2892m^3$—The inside volume of the evaporator

$V_k = 0.1193m^3$—The inside volume of the condenser

$$m = 0.441 V_n + 0.228 V_k$$

$$m = 0.441 \times 0.2892 + 0.228 \times 0.1193 = 154g$$

Because the perfusion of R600a is less than R12, so the perfusion mass of R600a is 0.45m, is 68g.

2.2 Theoretical calculation for the model BD/C-216

2.2.1 Design conditions of BD/C-216

Freezer BD/C-216 Design Conditions (according to the standards GB/T8059.3-1995) :

1) Environmental conditions: ambient temperature t_a=32□, relative humidity φ=75±5%;
2) Inside temperature of the freezer: t≤-18□;
3) Storage volume: L=216L;
4) Refrigerant: R600a;
5) Blowing agent: isopentane

Freezer BD/C-216 Construction (shown in **Figure 2**):

1) Opening type: top-opening type
2) Overall dimensions:895mm*1073mm*548mm (height*width*depth);
3) Thickness of the adiabatic wall: right side=57mm, left side=57mm, front side=56mm, back side=56, up side=48.5mm, underside=58mm;
4) Condenser construction: steel wire-tube type, the diameter of the tube and wire are 6mm(d_b),1.6mm(d_w); and the thickness of the steel tube is 0.71mm; space between tubes is 48mm(s_b), and space between wires is 5.5mm(s_w);
5) Evaporator construction: steel tube-board type, the board is made of aluminium and the thickness is 0.4mm, the diameter of the steel tube is 8mm and the thickness is 0.71mm;

6) Expansion device: capillary, the diameter is 1.8mm, the thickness is 0.71mm;
7) Compressor type: hermetically sealed compressor refrigerating compressor (plunger type);

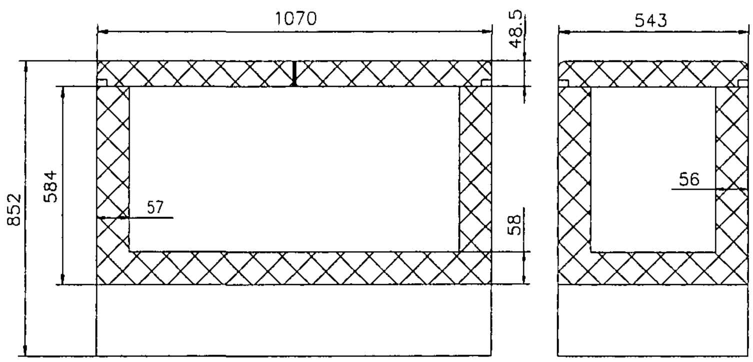

Figure 2

2.2.2 Theoretic Calculation For Cooling System

(1) Calculation of heat loss Q_F

The heat loss including three parts, one is the heat loss through the box of the freezer Q_{1F}, the second is for opening the freezer door Q_{2F}, the third is for cooling the goods Q_{3F}.

□Heat loss through the box of the freezer Q_{1F}

Q_{1F} including two parts, one is the heat loss through the adiabatic wall Q_a, the second is through the door Q_b.

$$Q_a = kA(t_1 - t_2)$$

$$K = \frac{1}{\frac{1}{a_1} + \frac{\delta}{\lambda} + \frac{1}{a_2}}$$

K— Coefficient of transfer heat, unit ($W/(m^2 \cdot k)$);
t_1— Outside exterior temperature of the freezer, unit (□);
t_2— Inside exterior temperature of the freezer, unit (□);
δ— thickness of the adiabatic wall, unit (m);
λ— heat conductivity of the adiabatic wall, unit ($W/(m \cdot k)$); λ=0.04 $W/(m \cdot k)$
a_1— exterior temperature of the air outside the freezer, unit ($W/(m^2 \cdot k)$); a_1=11.3 $W/(m^2 \cdot k)$
a_2— exterior temperature of the air inside the freezer, unit ($W/(m^2 \cdot k)$); a_2=1.16 $W/(m^2 \cdot k)$
A— area of the freezer exterior, unit(m^2)

Heat loss of every sides of the Freezer (shown in **Table 4**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	58	56	56	57	57
$K\ (W/(m^2 \cdot k))$	0.462	0.418	0.425	0.425	0.421	0.421
$A(m^2)$	0.567	0.567	0.593	0.593	0.285	0.285
t_1- t_2(□)	50	62	50	50	50	50

Q(w)	13.09	14.69	12.60	12.60	5.99	5.99

Table 4

Q_a=13.09+14.69+12.60+12.60+5.99+5.99=64.96w

Q_b=15%Q_a=0.15*64.96=9.74w

Q_{1F}= Q_a+ Q_b=64.96+9.74=74.70w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.216m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to -18□,106.75KJ/kg;

V_a— bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.216 \times 1 \times 106.75}{3.6 \times 0.9} = 7.12w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ/(kg·k);

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L=1.08kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[1.08 \times 4.19 \times 25 + 1.08 \times 333 - 1.08 \times 2 \times (-2)]}{2 \times 3.6} = 65.95w$$

$$Q_F = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F = 74.70 + 7.12 + 65.95 = 147.77w$$

(2) Circular thermodynamic calculation

□Parameter for design (Shown in **Table 5** and **6**)

Rated states of cooling system

State parameter	Condensation temperature (t_k)	Evaporator temperature (t_0)	Inspiratory temperature (t_G)	Super-cooling temperature (t_s)
Value of design	54.4	-23.3	32(80)	17

Parameter reference	t_k=32+22.4	t_0=-18-5.3	Ambient temperature	t_s=32-15

Table 5

Physical parameter

Parameter name	Sign	Unit	Design value
Condensation pressure	P_k	MPa	0.7814
Evaporator pressure	P_o	MPa	0.0623
Saturated steam enthalpy after the evaporator	h_1	KJkg	523.63
Steam-gas enthalpy before entering the compressor	h_1'	KJ/kg	612.41
Steam-gas bulking value before entering the compressor	v_1'	M^3/kg	0.68371
Steam-gas enthalpy before entering the cylinder	h_1''	KJ/kg	700.36
Steam-gas bulking value before entering the cylinder	v_1''	M^3/kg	0.80290
Temperature of the steam-gas	t_{2s}	℃	95.9
Enthalpy of the saturated steam at the condensation temperature	h_2	KJ/kg	629.76
Enthalpy of the steam-gas	h_2''	KJ/kg	1105.8
Enthalpy of the super-cooling liquid (32℃)	h_3	KJ/kg	275.44
Enthalpy of the super-cooling liquid (17℃□	h_3'	KJ/kg	241.23
Enthalpy of the refrigerant at the entrance to the evaporator	h_4	KJ/kg	241.23
Enthalpy of the compression gas □32℃□	h_{2s}	KJ/kg	714.04
Enthalpy of the compression gas □80℃□	h_{2s}'	KJ/kg	836.31

Table 6

□Calculation

- Refrigerating capacity per mass

$$q_0 = h_1 - h_4 = 523.63 - 241.23 = 284.4kJ/kg$$

- Refrigerating capacity per volume

$$q_v = \frac{q_o}{v_1^1} = \frac{h_1 - h_4}{v_1^1} = \frac{284.4}{0.68371} kg/m^3 = 415.96kg/m^3$$

- Compress work per mass

$$w_i = h_{2s} - h_1^1 = 714.04 - 612.41.04 = 101.63kJ/kg$$

- Refrigerating coefficient

$$\varepsilon = \frac{q_o}{w_i} = \frac{284.4}{101.63} = 2.79$$

- Condensing heat capacity per mass

$$q_k = h_2^1 - h_3 = 832.654 - 275.44w = 557.21w$$

- Cycle mass of refrigerant

$$G_a = \frac{Q}{q_o} = \frac{147.77 \times 3.6}{284.4} kg/h = 1.87kg/h$$

- Condensing heat capacity

$$Q_k = G_a \bullet q_k = G_a \bullet (h_2 - h_3) = \frac{1.87 \times 557.21 \times 10^3}{3600} w = 289.43w$$

- Overheat vapor suctioning by compressor

$$v_s = G_a \bullet v_1^1 = 1.87 \times 0.68371 m^3/h = 1.278m^3/h$$

(3) Calculation for choosing the compressor type

□Calculate the coefficient of gas transmission

- Volume coefficient

$$\lambda_v = 1 - c\left[\left(\frac{p_k + \Delta p_k}{p_o}\right)^{\frac{1}{m}} - 1\right]$$

$$\Delta p_k = 0.1 p_k$$

c — relative clearance volume, c=0.025;

m— expand coefficient, m=1.

$$\lambda_v = 1 - 0.025 \times \left[\left(\frac{0.7814 + 0.1 \times 0.7814}{0.0623} \right)^{\frac{1}{1}} - 1 \right] = 0.68$$

- Pressure coefficient

$$\lambda_p = 1 - \frac{1+c}{\lambda_v} \bullet \frac{\Delta p_o}{p_o}$$

$$\Delta p = 0.05 p_o$$

$$\lambda_p = 1 - \frac{1+0.025}{0.68} \times 0.05 = 0.924$$

- Temperature coefficient

$$\lambda_t = \frac{T_1''}{aT_k + b\theta}$$

a,b — Coefficient, a=1.15,b=0.25;

T_1'' —Inspiratory temperature, T_1''=273+80=353k;

T_k— Condensation temperature, T_k=273+54.4=327.4k;

θ— Superheat degree, θ= T_1''-(t_o+273)=103.3k.

$$\lambda_t = \frac{353}{1.15 \times 327.4 + 0.25 \times 103.3} = 0.87$$

- Leak coefficient

$\lambda_l = 0.99$ (According to the experience)

$$\lambda = \lambda_v \bullet \lambda_p \bullet \lambda_t \bullet \lambda_l$$

$$\lambda = 0.68 \times 0.924 \times 0.87 \times 0.99 = 0.542 m^3 / kg$$

□Theoretic carry capacity

$$V_h = \frac{V_s}{\lambda} = \frac{1.278}{0.542} = 2.357 m^3 / h$$

□Cooling capacity of compressor

$$Q_o = q_v \bullet V_h \bullet \lambda$$

$$Q_o = 415.96 \times 2.357 \times 0.532 = 490 kJ / kg = 0.147 kw$$

□Power of the compressor

- Theoretical adiabatic power

$$p_o = \frac{G_a \bullet w_i}{3600}$$

$$p_o = \frac{1.87 \times 101.63}{3600} = 52w$$

- Indicated power

$$p_i = \frac{p_o}{\eta_i}$$

$$\eta_i = \frac{T_o}{T_k} + bt_o$$

b=0.0005, T_o=273-23.3=249.7k, T_k=273+54.4=327.4k;

$$\eta_i = \frac{249.7}{327.4} + 0.0005 \times (-32) = 0.704$$

$$P_i = \frac{P_o}{\eta_i} = \frac{52}{0.704} = 73.86w$$

- Friction power

$$P_m = \frac{p_m \bullet v_h}{36.72}$$

P_m— average friction, p_m=0.45Mpa

$$P_m = \frac{0.45 \times 2.357}{36.72} = 28.8w$$

- Motor shaft horse power

$$P_e = P_i + P_m = 73.86 + 28.8 = 102.66w$$

- Electric power and motor efficiency

$$P_{el} = \frac{P_e}{\eta_{mo}}$$

$$\eta_{mo} = 0.82$$

$$P_{el} = \frac{102.66}{0.82} = 125.19w$$

According to the result of the theoretic calculation including the theoretic coolingg capacity, power of the compressor, electric power and motor efficiency etc, and the design parameter of R600a, we choose the compressor which is produced by JIAXIPERA company, model BM1114CY, external voltage 220V, input power 130w, external cooling capacity 160w.

(4) Calculation of condenser

□Head load in overheat Zone Q_1 and in saturated zone Q_2

$$\beta=\frac{Q_1}{Q_2}=\frac{h_{2k}-h_2}{h_{2k}-h_3}$$ (The percent of the heat load in overheat zone and heat load in saturated zone)

h_{2k}— enthalpy, states t=80℃,p=0.7814Mpa, h_{2k}=681.02kJ/kg;

$$\beta=\frac{681.02-629.76}{681.02-275.44}=0.126$$

$$Q_1=0.126Q_k=0.126\times 289.43=36.46w$$

$$Q_2=(1-0.126)Q_k=0.874\times 289.43=252.96w$$

□Deference in temperature of overheat □t_1 and saturated zone □t_2

- $$\Delta t_1=\frac{t_h-t_k}{\ln\frac{t_h-t_k}{t_k-t_a}}$$

t_h— overheat gas temperature, t_k=80℃;

$$\Delta t_1=\frac{80-54.4}{\ln\frac{80-32}{54.4-32}}=33.59℃$$

- $$\Delta t_2=t_k-t_a$$

$$\Delta t_2=54.4-32=22.4℃$$

□natural convection heat transfer coefficient

- overheat zone

$$a_{of}=0.94\frac{\lambda_f}{d_e}\left[\frac{(s_b-d_b)(s_w-d_w)}{(s_b-d_b)^2+(s_w-d_w)2}\right]^{0.155}(P_{rf}G_{rf})0.16$$

de —Equivalent diameter;

$$d_e=s_b\left[\frac{1+2\frac{s_b}{s_w}\bullet\frac{d_w}{d_b}}{\left(\frac{s_b}{2.76d_b}\right)^{0.25}+2\frac{s_b}{s_w}\bullet\frac{d_w}{d_b}\eta_f}\right]^4$$

η_f— rib efficiency, η_f=0.85;

$$d_e = 0.048\times\left[\frac{1+2\times\dfrac{0.048}{0.0055}\times\dfrac{0.0016}{0.006}}{\left(\dfrac{0.048}{2.76\times0.006}\right)^{0.25}+2\times\dfrac{0.048}{0.0055}\times\dfrac{0.0016}{0.006}\times0.85}\right]^4 = 0.0637$$

g— acceleration of gravity, g=9.8m/s^2;

β— expend coefficient, β=0.0031;

r— Sport glutinosity of the air, r=17.84*10^{-6}m^2/s

$$G_{rf} = g\beta\Delta t_1 d^3/\gamma^2$$

$$G_{rf} = 9.8\times0.0031\times33.59\times\frac{(0.0640)^3}{(17.84\times10^{-6})^2} = 826997$$

$$P_{rf} = 0.698$$

$$a_{of}' = 0.94\times\frac{2.82\times10^{-2}}{0.0637}\times\left[\frac{(48-8)(5.5-1.6)}{(48-8)^2-(5.5-1.6)^2}\right]^{0.155}\times(0.698\times826997)^{0.26} = 9.07w/m^2\bullet k$$

- Saturated zone

λ_f— when the temperature is 43.2℃, thermal conductivity of the air, 0.278W/(m·k)

r— Sport glutinosity of the air, r=17.28*10^{-6}m^2/s

$$P_{rf} = 0.699$$

β— expend coefficient, β=0.00316;

$$G_{rf} = 9.81\times\frac{1}{273+43.2}\times22.4\times\frac{(0.0637)^3}{(17.28\times10^{-6})^2} = 599872$$

$$a_{of}' = 0.94\times\frac{2.78\times10^{-2}}{0.0637}\times\left[\frac{(48-6)(5.5-1.6)}{(48-6)^2-(5.5-1.6)^2}\right]^{0.155}\times(0.699\times599872)^{0.26} = 8.35w/m^2\bullet k$$

□Radiation heat transfer coefficient

- Overheat zone

$$a_{or}' = 6.67\varepsilon\frac{\left(\dfrac{T_1+273}{100}\right)^4-\left(\dfrac{T_2+273}{100}\right)^4}{\Delta t_1}$$

ε— Dark coefficient, ε=0.97;

T_1— average temperature of the condenser, T_1=65.59℃

T_2— ambient temperature, T_2=32 ℃

$$a_{or}' = 5.67 \times 0.97 \times \frac{\left(\frac{65.59+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{33.59} = 7.35 w/m^2 \bullet k$$

- Saturated zone

T_1— average temperature of the condenser, T_1=54.4℃

$$a_{or}'' = 5.67 \times 0.97 \times \frac{\left(\frac{54.4+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{22.4} = 6.96$$

□ Heat transfer area

$$\eta_o = \frac{a_b + a_p \eta_f}{a_b + a_p}$$ (Exterior efficiency)

a_b— Outside exterior area per m, $a_b = 0.006\pi$

a_p— Inside exterior area per m,

$$a_p = 2 \times \left(\frac{1}{s_w} + 1\right) \pi d_w s_b$$

$$\eta_o = \frac{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \times 0.0016\pi \times 0.048 \times 0.85}{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \pi \times 0.048 \times 0.0016} = 0.8763$$

- Overheat zone

$$A' = \frac{Q_1}{\left(a_{of}' + a_{or}'\right) \eta_o \Delta t_1}$$

$$A' = \frac{36.46}{(9.07 + 7.35) \times 0.8763 \times 33.59} = 0.0745 m^2$$

- Saturated zone

$$A'' = \frac{252.96}{(6.96 + 8.35) \times 0.8763 \times 22.4} = 0.8376 m^2$$

Total heat transfer area

$$A = A' + A'' = 0.0745 + 0.8376 = 0.9121 m^2$$

□The size of the condenser

- The length of the steel tube

$$L = \frac{A}{a_b + a_p}$$

$$L = \frac{0.9121}{0.088 + 0.0188} = 8.53m$$

Because when we use the freezer, the condenser would be dirty, in order to make no influence on it, we should increase the length, so the actual length is 1.05L, 8.95m, we do not want to change the mold of the condenser, so the width of the condenser is the same as the mold, its width is o.45m, the depth is 0.798m. Generally the length of the steel wires is longer than the condenser depth, so the actual length is 0.800m.

(5) Calculation of the evaporator

The area of the evaporator is not changed, its is $1.36m^2$(A)

The rib efficient is 3.5(β)

So the exterior of the steel tube is

$$A_e = \frac{A}{\beta} = \frac{1.36}{3.5} = 0.389m^2$$

$$L = \frac{A_e}{0.008\pi} = 15.50m$$

(6) Calculation of Capillary(according to the calculation of R12)

In the conditions of the same states, the capillary length of R12.

- $L = \dfrac{\Delta P \bullet R_e^{0.25} \bullet d}{0.1582w^2 \bullet \rho}$

- $W = \dfrac{G_a v}{\frac{\pi}{4} \bullet d^2}$

d— inside diameter of the capillary, d=0.00066m;

v— bulking value of the liquid, t=15℃,$v=4.21*10^{-3}m^3/kg$;

G_a— Cycle mass of refrigerant R12, G_a=3.55kg/h;

$$W = \frac{3.55 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00071)^2 \times 3600} m/s = 10.48m/s$$

R_e— Reynolds number

- $R_e = \frac{Wd\rho}{\mu}$

ρ— average density, ρ=237.5kg/m3;
μ— average glutinosity, μ=0.217*10^{-3}Pa·s

$$R_e = \frac{11.37 \times 0.00071 \times 237.53}{0.217 \times 10^{-3}} = 8144$$

$$L = \frac{1.216 \times 10^6 \times (8144)^{0.25} \times 0.00071}{0.1582 \times (10.48)^2 \times 237.55} m = 1.98m$$

According to the experiment, we find the capillary length which using the R600a is shorter than the capillary which using the R12, so the actual length is 0.75L, 1.50m.

(7) The mass of the perfusion(According to the R12)

$V_n = 0.3170m^3$—The inside volume of the evaporator

$V_k = 0.1305m^3$—The inside volume of the condenser

$$m = 0.441\ V_n + 0.228\ V_k$$

$$m = 0.441 \times 0.3170 + 0.228 \times 0.1305 = 169g$$

Because the perfusion of R600a is less than R12, so the perfusion mass of R600a is 0.45m, is 76g.

2.3 Theoretical calculation for the model LCD-207

2.3.1 Design conditions of LCD-207

Freezer LCD207 Design Conditions (according to the standards GB/T8059.1-1995) :

1) Environmental conditions: ambient temperature ta=32□, relative humidity φ=75±5%;
2) Inside temperature of the freezer: td≤-18□, tc≤5□;
3) Storage volume: L_d=94L, L_c=113L;
4) Refrigerant: R600a;
5) Blowing agent: isopentane

Freezer LCD207 Construction (shown in Figure 3):

1) Opening type: top-opening type
2) Overall dimensions:847mm*1073mm*547mm (height*width*depth);
3) Thickness of the adiabatic wall: fresh food storage compartment, right side=43mm, left side=39mm, front side=39mm, back side=39, up side=48.5mm, underside=51mm; frozen food storage compartment, right side=57mm, left side=43mm, front side=56mm, back side=56, up

side=48.5mm, underside=61mm.

4) Condenser construction: steel wire-tube type, the diameter of the tube and wire are 6mm(d_b),1.6mm(d_w); and the thickness of the steel tube is 0.71mm; space between tubes is 48mm(s_b), and space between wires is 5.5mm(s_w);
5) Evaporator construction: steel tube-board type, the board is made of aluminium and the thickness is 0.4mm, the diameter of the steel tube is 8mm and the thickness is 0.71mm;
6) Expansion device: capillary, the diameter is 1.8mm, the thickness is 0.71mm;
7) Compressor type: hermetically sealed compressor refrigerating compressor (plunger type);

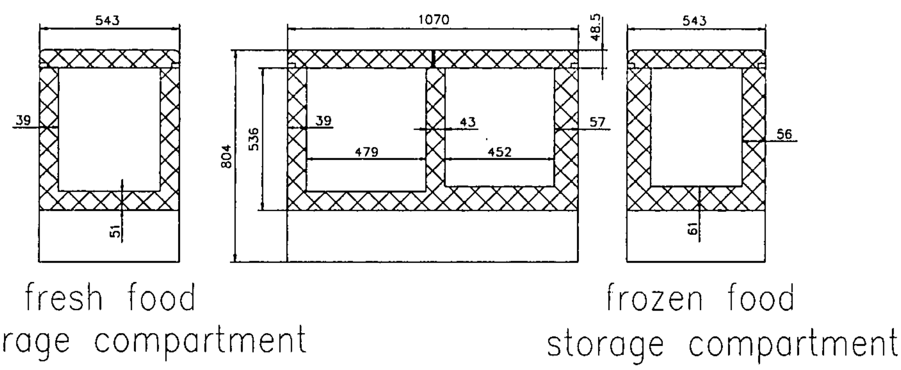

Figure 3

2.3.2 Theoretic Calculation For Cooling System

1) Calculation of heat loss Q_F

The heat loss including three parts, one is the heat loss through the box of the freezer Q_{1F}, the second is for opening the freezer door Q_{2F}, the third is for cooling the goods Q_{3F}.

□Heat loss through the box of the freezer Q_{1F}

Q_{1F} including two parts, one is the heat loss through the adiabatic wall Q_a, the second is through the door Q_b.

$Q_a=KA(t_1- t_2)$

$$K=\frac{1}{\frac{1}{a_1}+\frac{\delta}{\lambda}+\frac{1}{a_2}}$$

K— Coefficient of transfer heat, unit ($W/(m^2\cdot k)$);

t_1— Outside exterior temperature of the freezer, unit (□);

t_2— Inside exterior temperature of the freezer, unit (□);

δ— thickness of the adiabatic wall, unit (m);

λ— heat conductivity of the adiabatic wall, unit ($W/(m\cdot k)$); λ=0.04 $W/(m\cdot k)$

a_1— exterior temperature of the air outside the freezer, unit ($W/(m^2\cdot k)$); a_1=11.3 $W/(m^2\cdot k)$

a_2— exterior temperature of the air inside the freezer, unit ($W/(m^2\cdot k)$); a_2=1.16 $W/(m^2\cdot k)$

A— area of the freezer exterior, unit(m^2)

Heat loss of every sides of the frozen food storage compartment (shown in **Table 7**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	61	56	56	43	57
$K\ (W/(m^2 \cdot k))$	0.462	0.404	0.425	0.425	0.494	0.421
$A(m^2)$	0.214	0.214	0.213	0.213	0.256	0.256
t_1- t_2(□)	50	58	50	50	23	50
Q(w)	4.94	5.01	4.53	4.53	2.90	5.38

Table 7

Q_a=4.94+5.01+4.53+4.53+2.90+5.38=27.32w

Q_b=15%Q_a=0.15*27.32=4.09w

Q_{1F}= Q_a+ Q_b=27.32+4.09=31.42w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.094m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to -18□,106.75KJ/kg;

V_a—bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.094 \times 1 \times 106.75}{3.6 \times 0.9} = 3.10w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ·k/kg

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_d=0.47kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.47 \times 4.19 \times 25 + 0.47 \times 333 - 0.47 \times 2 \times (-2)]}{2 \times 3.6} = 28.84w$$

$$Q_F' = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F' = 31.42 + 3.10 + 28.84 = 63.36w$$

Heat loss of every sides of the fresh food storage compartment room (shown in **Table 8**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	51	39	39	39	43
K $(W/(m^2 \cdot k))$	0.462	0.449	0.519	0.519	0.519	0.494
$A(m^2)$	0.228	0.228	0.225	0.225	0.256	0.256
t_1- t_2(□)	27	52	27	27	27	-23
Q(w)	2.84	5.32	3.15	3.15	3.58	-2.90

Table 8

Q_a=2.84+5.32+3.15+3.15+3.58-2.90=15.14w

Q_b=15%Q_a=0.15*15.14=2.27w

Q_{1F}= Q_a+ Q_b=15.14+2.27=17.41w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.111m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to 5□,71.6KJ/kg;

V_a— bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.113 \times 1 \times 71.6}{3.6 \times 0.9} = 2.49w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ/(kg·k);

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_c=0.565kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.565 \times 4.19 \times 25 + 0.565 \times 333 - 0.565 \times 2 \times (-2)]}{2 \times 3.6} = 34.66w$$

$$Q_F'' = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F'' = 17.14 + 2.49 + 34.66 = 54.29w$$

□ Total heat loss

$$Q_F = Q_F' + Q_F'' = 63.36 + 54.29 = 117w$$

2) Circular thermodynamic calculation

□Parameter for design (Shown in **Table 9 and 10**)

Rated states of cooling system

State parameter	Condensation temperature (t_k)	Evaporator temperature (t_0)	Inspiratory temperature (t_G)	Super-cooling temperature (t_s)
Value of design	54.4	-23.3	32(80)	17
Parameter reference	t_k=32+22.4	t_0=-18-5.3	Ambient temperature	t_s=32-15

Table 9

Physical parameter

Parameter name	Sign	Unit	Design value
Condensation pressure	P_k	MPa	0.7814
Evaporator pressure	P_o	MPa	0.0623
Saturated steam enthalpy after the evaporator	h_1	KJkg	523.63
Steam-gas enthalpy before entering the compressor	h_1'	KJ/kg	612.41
Steam-gas bulking value before entering the compressor	v_1'	M^3/kg	0.68371
Steam-gas enthalpy before entering the cylinder	h_1''	KJ/kg	700.36
Steam-gas bulking value before entering the cylinder	v_1''	M^3/kg	0.80290
Temperature of the steam-gas	t_{2s}	℃	95.9
Enthalpy of the saturated steam at the condensation temperature	h_2	KJ/kg	629.76
Enthalpy of the steam-gas	h_2''	KJ/kg	1105.8
Enthalpy of the super-cooling liquid (32℃)	h_3	KJ/kg	275.44
Enthalpy of the super-cooling liquid	h_3'	KJ/kg	241.23

(17℃□			
Enthalpy of the refrigerant at the entrance to the evaporator	h_4	KJ/kg	241.23
Enthalpy of the compression gas □32℃□	h_{2s}	KJ/kg	714.04
Enthalpy of the compression gas □80℃□	h_{2s}'	KJ/kg	836.31

Table 10

□Calculation

- Refrigerating capacity per mass

$$q_0 = h_1 - h_4 = 523.63 - 241.23 = 284.4kJ/kg$$

- Refrigerating capacity per volume

$$q_v = \frac{q_o}{v_1^1} = \frac{h_1 - h_4}{v_1^1} = \frac{284.4}{0.68371} kg/m^3 = 415.96kg/m^3$$

- Compress work per mass

$$w_i = h_{2s} - h_1^1 = 714.04 - 612.41.04 = 101.63kJ/kg$$

- Refrigerating coefficient

$$\varepsilon = \frac{q_o}{w_i} = \frac{284.4}{101.63} = 2.79$$

- Condensing heat capacity per mass

$$q_k = h_2^1 - h_3 = 832.654 - 275.44w = 557.21w$$

- Cycle mass of refrigerant

$$G_a = \frac{Q}{q_o} = \frac{117 \times 3.6}{284.4} kg/h = 1.48kg/h$$

- Condensing heat capacity

$$Q_k = G_a \bullet q_k = G_a \bullet (h_2 - h_3) = \frac{1.48 \times 557.21 \times 10^3}{3600} w = 229.07w$$

- Overheat vapor suctioning by compressor

$$v_s = G_a \bullet v_1^1 = 1.48 \times 0.68371 m^3/h = 1.01 m^3/h$$

3)Calculation for choosing the compressor type

□Calculate the coefficient of gas transmission

- Volume coefficient

$$\lambda_v = 1 - c\left[\left(\frac{p_k + \Delta p_k}{p_o}\right)^{\frac{1}{m}} - 1\right]$$

$$\Delta p_k = 0.1 p_k$$

c — relative clearance volume, c=0.025;
m— expand coefficient, m=1.

$$\lambda_v = 1 - 0.025 \times \left[\left(\frac{0.7814 + 0.1 \times 0.7814}{0.0623}\right)^{\frac{1}{1}} - 1\right] = 0.68$$

- Pressure coefficient

$$\lambda_p = 1 - \frac{1+c}{\lambda_v} \bullet \frac{\Delta p_o}{p_o}$$

$$\Delta p = 0.05 p_o$$

$$\lambda_p = 1 - \frac{1+0.025}{0.68} \times 0.05 = 0.924$$

- Temperature coefficient

$$\lambda_t = \frac{T_1''}{aT_k + b\theta}$$

a,b — Coefficient, a=1.15,b=0.25;
T_1'' —Inspiratory temperature, T_1''=273+80=353k;
T_k— Condensation temperature, T_k=273+54.4=327.4k;
θ— Superheat degree, θ= T_1''-(t_o+273)=103.3k.

$$\lambda_t = \frac{353}{1.15 \times 327.4 + 0.25 \times 103.3} = 0.87$$

- Leak coefficient

$\lambda_l = 0.99$ (According to the experience)

$$\lambda = \lambda_v \bullet \lambda_p \bullet \lambda_t \bullet \lambda_l$$

$$\lambda = 0.68 \times 0.924 \times 0.87 \times 0.99 = 0.542 m^3/kg$$

□Theoretic carry capacity

$$V_h = \frac{V_s}{\lambda} = \frac{1.01}{0.542} = 1.86 m^3/h$$

□Cooling capacity of compressor

$$Q_o = q_v \bullet V_h \bullet \lambda$$

$$Q_o = 415.96 \times 1.86 \times 0.542 = 421 kJ/h = 0.117 kw$$

□Power of the compressor

- Theoretical adiabatic power

$$p_o = \frac{G_a \bullet w_i}{3600}$$

$$p_o = \frac{1.48 \times 101.63}{3600} = 42w$$

- Indicated power

$$p_i = \frac{p_o}{\eta_i}$$

$$\eta_i = \frac{T_o}{T_k} + bt_o$$

b=0.0005, T_o=273-23.3=249.7k, T_k=273+54.4=327.4k;

$$\eta_i = \frac{249.7}{327.4} + 0.0005 \times (-32) = 0.704$$

$$P_i = \frac{P_o}{\eta_i} = \frac{42}{0.704} = 59.65w$$

- Friction power

$$P_m = \frac{p_m \bullet v_h}{36.72}$$

P_m— average friction, p_m=0.45Mpa

$$P_m = \frac{0.45 \times 1.86}{36.72} = 23w$$

- Motor shaft horse power

$$P_e = P_i + P_m = 59.65 + 23 = 82.65w$$

- Electric power and motor efficiency

$$P_{el} = \frac{P_e}{\eta_{mo}}$$

$$\eta_{mo} = 0.82$$

$$P_{el} = \frac{82.65}{0.82} = 100.79w$$

According to the result of the theoretic calculation including the theoretic cooling capacity, power of the compressor, electric power and motor efficiency etc, and the design parameter of R600a, we choose the compressor which is produced by BaiXue company, normal model QDHC90 external voltage 220V, input power 113w, external cooling capacity 128w.

4) Calculation of condenser

□Head load in overheat Zone Q_1 and in saturated zone Q_2

$\beta = \frac{Q_1}{Q_2} = \frac{h_{2k} - h_2}{h_{2k} - h_3}$ (The percent of the heat load in overheat zone and heat load in saturated zone)

h_{2k}— enthalpy, states t=80℃,p=0.7814Mpa, h_{2k}=681.02kJ/kg;

$$\beta = \frac{681.02 - 629.76}{681.02 - 275.44} = 0.126$$

$$Q_1 = 0.126Q_k = 0.126 \times 229.07 = 28.85w$$

$$Q_2 = (1 - 0.126)Q_k = 0.874 \times 229.07 = 200.20w$$

□Deference in temperature of overheat □t_1 and saturated zone □t_2

- $$\Delta t_1 = \frac{t_h - t_k}{\ln \frac{t_h - t_k}{t_k - t_a}}$$

t_h— overheat gas temperature, t_k=80℃;

$$\Delta t_1 = \frac{80 - 54.4}{\ln \frac{80 - 32}{54.4 - 32}} = 33.59℃$$

- $$\Delta t_2 = t_k - t_a$$

$$\Delta t_2 = 54.4 - 32 = 22.4℃$$

□natural convection heat transfer coefficient

- overheat zone

$$a_{of} = 0.94\frac{\lambda_f}{d_e}\left[\frac{(s_b - d_b)(s_w - d_w)}{(s_b - d_b)^2 + (s_w - d_w)2}\right]^{0.155}(P_{rf}G_{rf})0.16$$

de —Equivalent diameter;

$$d_e = s_b \left[\frac{1 + 2\frac{s_b}{s_w} \bullet \frac{d_w}{d_b}}{\left(\frac{s_b}{2.76 d_b}\right)^{0.25} + 2\frac{s_b}{s_w} \bullet \frac{d_w}{d_b}\eta_f} \right]^4$$

η_f— rib efficiency, η_f=0.85;

$$d_e = 0.048 \times \left[\frac{1 + 2 \times \frac{0.048}{0.0055} \times \frac{0.0016}{0.006}}{\left(\frac{0.048}{2.76 \times 0.006}\right)^{0.25} + 2 \times \frac{0.048}{0.0055} \times \frac{0.0016}{0.006} \times 0.85} \right]^4 = 0.0637$$

g— acceleration of gravity, g=9.8m/s^2;

β— expend coefficient, β=0.0031;

r— Sport glutinosity of the air, r=17.84*10^{-6}m^2/s

$$G_{rf} = g\beta\Delta t_1 d^{\ 3} / \gamma^2$$

$$G_{rf} = 9.8 \times 0.0031 \times 33.59 \times \frac{(0.0637)^3}{(17.84 \times 10^{-6})^2} = 826997$$

$$P_{rf} = 0.698$$

$$a_{of}{}' = 0.94 \times \frac{2.82 \times 10^{-2}}{0.0637} \times \left[\frac{(48-6)(5.5-1.6)}{(48-6)^2 - (5.5-1.6)^2} \right]^{0.155} \times (0.698 \times 826997)^{0.26} = 9.07 w / m^2 \bullet k$$

- Saturated zone

λ_f— when the temperature is 43.2℃, thermal conductivity of the air, 0.278W/(m·k)

r— Sport glutinosity of the air, r=17.28*10^{-6}m^2/s

$$P_{rf} = 0.699$$

β— expend coefficient, β=0.00316;

$$G_{rf} = 9.81 \times \frac{1}{273 + 43.2} \times 22.4 \times \frac{(0.0637)^3}{(17.28 \times 10^{-6})^2} = 599872$$

$$a_{of}{}' = 0.94 \times \frac{2.78 \times 10^{-2}}{0.0637} \times \left[\frac{(48-6)(5.5-1.6)}{(48-6)^2 - (5.5-1.6)^2} \right]^{0.155} \times (0.699 \times 599872)^{0.26} = 8.35 w / m^2 \bullet k$$

□Radiation heat transfer coefficient

- Overheat zone

$$a_{or}{}' = 6.67\varepsilon \frac{\left(\frac{T_1+273}{100}\right)^4 - \left(\frac{T_2+273}{100}\right)^4}{\Delta t_1}$$

ε— Dark coefficient, ε=0.97;

T_1— average temperature of the condenser, T_1=65.59℃

T_2— ambient temperature, T2=32 ℃

$$a_{or}{}' = 5.67 \times 0.97 \times \frac{\left(\frac{65.59+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{33.59} = 7.35 w/m^2 \bullet k$$

- Saturated zone

T_1— average temperature of the condenser, T_1=54.4℃

$$a_{or}{}'' = 5.67 \times 0.97 \times \frac{\left(\frac{54.4+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{22.4} = 6.96$$

□ Heat transfer area

$$\eta_o = \frac{a_b + a_w \eta_f}{a_b + a_w}$$ (Exterior efficiency)

a_b— Outside exterior area per m, $a_b = 0.006\pi$

a_p— Inside exterior area per m,

$$a_p = 2 \times \left(\frac{1}{s_w} + 1\right) \pi d_w s_b$$

$$\eta_o = \frac{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \times 0.0016\pi \times 0.048 \times 0.85}{0.006\pi + \left(\frac{2}{0.0055} + 2\right) \pi \times 0.048 \times 0.0016} = 0.8763$$

- Overheat zone

$$A' = \frac{Q_1}{\left(a_{of}{}' + a_{or}{}'\right) \eta_o \Delta t_1}$$

$$A' = \frac{28.85}{(9.07 + 7.35) \times 0.8763 \times 33.59} = 0.0584 m^2$$

● Saturated zone

$$A'' = \frac{209.67}{(6.96 + 8.35) \times 0.8763 \times 22.4} = 0.6174m^2$$

Total heat transfer area

$$A = A' + A'' = 0.0584 + 0.6174 = 0.6758m^2$$

□The size of the condenser

● The length of the steel tube

$$L = \frac{A}{a_b + a_p}$$

$$L = \frac{0.7091}{0.088 + 0.0188} = 6.32m$$

Because when we use the freezer, the condenser would be dirty, in order to make no influence on it, we should increase the length, so the actual length is 1.05L, 6.64m. we do not want to change the mold of the condenser, so the width of the condenser is the same as the mold, its width is 0.45m, so the horizontal amount N is 6.64/0.45=14, the depth is 0.672m. Generally the length of the steel wires is longer than the condenser depth, so the actual length is 0.674m.

(5)Calculation of the evaporator

The area of the evaporator is not changed, its is 1.23m^2(A)

The rib efficient is 6.05(β)

So the exterior of the steel tube is

$$A_e = \frac{A}{\beta} = \frac{1.47}{6.05} = 0.242m^2$$

$$L = \frac{A_e}{0.008\pi} = 9.63m$$

Because the temperature of the fresh food storage compartment and frozen food storage compartment is deferent, so the area of transfer heat is deferent in them, and the evaporator tube is longer in the fresh food storage compartment than in the frozen food storage compartment. According to the experience, the length of the evaporator tube in the frozen food storage compartment is 8.13m, the length of the evaporator in the fresh food storage compartment is 1.5m.

(6) Calculation of Capillary(according to the calculation of R12)

In the conditions of the same states, the capillary length of R12.

● $$L = \frac{\Delta P \bullet R_e^{0.25} \bullet d}{0.1582 w^2 \bullet \rho}$$

- $W = \dfrac{G_a v}{\dfrac{\pi}{4} \bullet d^2}$

d— inside diameter of the capillary, d=0.00066m;

v— bulking value of the liquid, t=15℃,v=4.21*$10^{-3}m^3/kg$;

G_a— Cycle mass of refrigerant R12, G_a=2.69kg/h;

$$W = \frac{2.69 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00066)^2 \times 3600} m/s = 9.20 m/s$$

R_e— Reynolds number

- $R_e = \dfrac{Wd\rho}{\mu}$

ρ— average density, ρ=237.5kg/m3;
μ— average glutinosity, μ=0.217*10^{-3}Pa·s

$$R_e = \frac{9.20 \times 0.00066 \times 237.53}{0.217 \times 10^{-3}} = 6646$$

$$L = \frac{1.216 \times 10^6 \times (6646)^{0.25} \times 0.00066}{0.1582 \times (9.20)^2 \times 237.55} m = 2.27m$$

According to the experiment, we find the capillary length which using the R600a is shorter than the capillary which using the R12, so the actual length is 0.75L, 1.70m.

(7) The mass of the perfusion(According to the R12)

$V_n = 0.1989m^3$—The inside volume of the evaporator

$V_k = 0.0954m^3$—The inside volume of the condenser

$$m = 0.441V_n + 0.228V_k$$

$$m = 0.441 \times 0.1989 + 0.228 \times 0.0954 = 109g$$

Because the perfusion of R600a is less than R12, so the perfusion mass of R600a is 0.45m, is 49g.

2.4 Theoretical calculation for the model LCD-228

2.4.1 Design conditions of LCD-228

Freezer LCD228 Design Conditions (according to the standards GB/T8059.1-1995) :

1) Environmental conditions: ambient temperature ta=32□, relative humidity φ=75±5%;
2) Inside temperature of the freezer: td≤-18□, tc≤5□;
3) Storage volume: L_d=104L, L_c=124L;
4) Refrigerant: R600a;
5) Blowing agent: isopentane

Freezer LCD228 Construction (shown in **Figure 4**):

1) Opening type: top-opening type
2) Overall dimensions:894mm*1073mm*543mm (height*width*depth);
3) Thickness of the adiabatic wall: fresh food storage compartment, right side=43mm, left side=39mm, front side=39mm, back side=39, up side=48.5mm, underside=51mm; frozen food storage compartment, right side=57mm, left side=43mm, front side=56mm, back side=56, up side=48.5mm, underside=61mm.
4) Condenser construction: steel wire-tube type, the diameter of the tube and wire are 6mm(d_b),1.6mm(d_w); and the thickness of the steel tube is 0.71mm; space between tubes is 48mm(s_b), and space between wires is 5.5mm(s_w);
5) Evaporator construction: steel tube-board type, the board is made of aluminium and the thickness is 0.4mm, the diameter of the steel tube is 8mm and the thickness is 0.71mm;
6) Expansion device: capillary, the diameter is 1.8mm, the thickness is 0.71mm;
7) Compressor type: hermetically sealed compressor refrigerating compressor (plunger type);

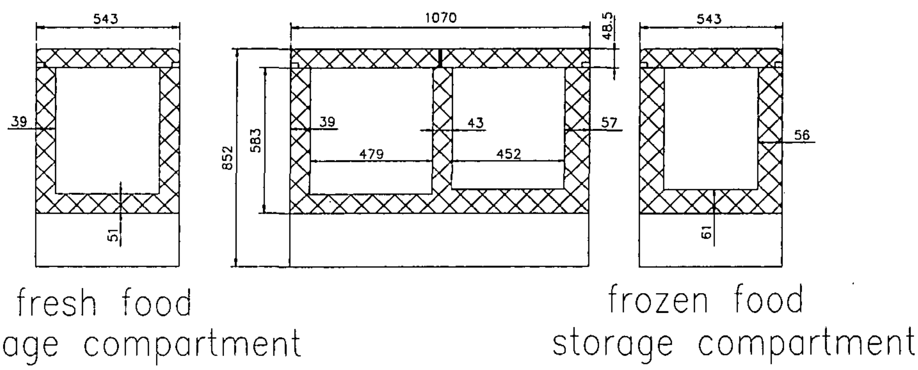

Figure 4

2.4.2 Theoretic Calculation For Cooling System

1) Calculation of heat loss Q_F

The heat loss including three parts, one is the heat loss through the box of the freezer Q_{1F}, the second is for opening the freezer door Q_{2F}, the third is for cooling the goods Q_{3F}.

□Heat loss through the box of the freezer Q_{1F}

Q_{1F} including two parts, one is the heat loss through the adiabatic wall Q_a, the second is through the door Q_b.

$Q_a=KA(t_1-t_2)$

$$K=\frac{1}{\frac{1}{a_1}+\frac{\delta}{\lambda}+\frac{1}{a_2}}$$

K— Coefficient of transfer heat, unit ($W/(m^2\cdot k)$);

t_1— Outside exterior temperature of the freezer, unit (□);

t_2— Inside exterior temperature of the freezer, unit (□);

δ— thickness of the adiabatic wall, unit (m);

λ— heat conductivity of the adiabatic wall, unit ($W/(m\cdot k)$); λ=0.04 $W/(m\cdot k)$

a_1— exterior temperature of the air outside the freezer, unit ($W/(m^2\cdot k)$); a_1=11.3 $W/(m^2\cdot k)$

a_2— exterior temperature of the air inside the freezer, unit ($W/(m^2\cdot k)$); a_2=1.16 $W/(m^2\cdot k)$

A— area of the freezer exterior, unit(m^2)

Heat loss of every sides of the frozen food storage compartment (shown in **Table 11**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	61	56	56	43	57
$K(W/(m^2\cdot k))$	0.462	0.404	0.425	0.425	0.494	0.421
$A(m^2)$	0.220	0.220	0.233	0.233	0.296	0.296
t_1-t_2(□)	50	58	50	50	23	50
Q(w)	5.08	5.15	4.95	4.95	3.36	6.23

Table 1

Q_a=5.08+5.15+4.95+4.95+3.36+6.23=29.72w

Q_b=15%Q_a=0.15*29.72=4.45w

Q_{1F}= Q_a+ Q_b=29.72+4.45=34.17w

□Heat loss for opening the door Q_{2F}

$$Q_{2F}=\frac{v_B n\Delta h}{3.6v_a}$$

V_B— storage volume, 0.104m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to -18□,106.75KJ/kg;

V_a—bulking value of the air, 0.9m^3/kg.

$$Q=\frac{0.104\times1\times106.75}{3.6\times0.9}=3.43w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F}=\frac{m_c ct_1+m_c r-m_c c_b t_2}{2\times3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ·k/kg

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_d=0.52kg;
t_1 — temperature of the water, t_1=25□;
t_2— temperature of the ice, t_2=-2□.

$$Q_{3F}=\frac{[0.52\times 4.19\times 25+0.52\times 333-0.52\times 2\times(-2)]}{2\times 3.6}=31.90w$$

$$Q_F^{'}=Q_{1F}+Q_{2F}+Q_{3F}$$

$$Q_F^{'}=34.17+3.43+31.90=69.5w$$

Heat loss of every sides of the fresh food storage compartment (shown in **Table 12**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	48.5	51	39	39	39	43
K $(W/(m^2\cdot k))$	0.462	0.449	0.519	0.519	0.519	0.494
$A(m^2)$	0.225	0.225	0.242	0.242	0.296	0.296
t_1- t_2(□)	27	52	27	27	27	-23
Q(w)	2.81	5.25	3.39	3.39	4.14	-3.36

Table 2

Q_a=2.81+5.25+3.39+3.39+4.14-3.36=15.62w
Q_b=15%Q_a=0.15*15.62=2.34w
Q_{1F}= Q_a+ Q_b=15.62+2.34=17.96w

□Heat loss for opening the door Q_{2F}

$$Q_{2F}=\frac{v_B n\Delta h}{3.6v_a}$$

V_B— storage volume, 0.124m^3;
n — times of opening the door per hour, n=1;
□h —enthalpy difference of the air from 32□ to 5□,71.6KJ/kg;
V_a— bulking value of the air, 0.9m^3/kg.

$$Q=\frac{0.124\times 1\times 71.6}{3.6\times 0.9}=2.74w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F}=\frac{m_c c t_1+m_c r-m_c c_b t_2}{2\times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ/(kg·k);
r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;
c_b— heat of fusion of the ice, the temperature is □,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_c=0.62kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.62\times4.19\times25+0.62\times333-0.62\times2\times(-2)]}{2\times3.6} = 38.04w$$

$$Q_F'' = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F'' = 17.96 + 2.74 + 38.04 = 58.74w$$

□ Total heat loss

$$Q_F = Q_F' + Q_F'' = 69.5 + 58.74 = 128.24w$$

2) Circular thermodynamic calculation

□Parameter for design (Shown in **Table 13 and 14**)

Rated states of cooling system

State parameter	Condensation temperature (t_k)	Evaporator temperature (t_0)	Inspiratory temperature (t_G)	Super-cooling temperature (t_s)
Value of design	54.4	-23.3	32(80)	17
Parameter reference	t_k=32+22.4	t_0=-18-5.3	Ambient temperature	t_s=32-15

Table 13

Physical parameter

Parameter name	Sign	Unit	Design value
Condensation pressure	P_k	MPa	0.7814
Evaporator pressure	P_o	MPa	0.0623
Saturated steam enthalpy after the evaporator	h_1	KJkg	523.63
Steam-gas enthalpy before entering the compressor	h_1'	KJ/kg	612.41
Steam-gas bulking value before entering the compressor	v_1'	M^3/kg	0.68371
Steam-gas enthalpy before entering the cylinder	h_1''	KJ/kg	700.36

Steam-gas bulking value before entering the cylinder	v_1''	M^3/kg	0.80290
Temperature of the steam-gas	t_{2s}	℃	95.9
Enthalpy of the saturated steam at the condensation temperature	h_2	KJ/kg	629.76
Enthalpy of the steam-gas	h_2''	KJ/kg	1105.8
Enthalpy of the super-cooling liquid (32℃)	h_3	KJ/kg	275.44
Enthalpy of the super-cooling liquid (17℃□	h_3'	KJ/kg	241.23
Enthalpy of the refrigerant at the entrance to the evaporator	h_4	KJ/kg	241.23
Enthalpy of the compression gas □32℃□	h_{2s}	KJ/kg	714.04
Enthalpy of the compression gas □80℃□	h_{2s}'	KJ/kg	836.31

Table 14

□Calculation

- Refrigerating capacity per mass

$$q_0 = h_1 - h_4 = 523.63 - 241.23 = 284.4 kJ/kg$$

- Refrigerating capacity per volume

$$q_v = \frac{q_o}{v_1^1} = \frac{h_1 - h_4}{v_1^1} = \frac{284.4}{0.68371} kg/m^3 = 415.96 kg/m^3$$

- Compress work per mass

$$w_i = h_{2s} - h_1^1 = 714.04 - 612.41.04 = 101.63 kJ/kg$$

- Refrigerating coefficient

$$\varepsilon = \frac{q_o}{w_i} = \frac{284.4}{101.63} = 2.79$$

- Condensing heat capacity per mass

$$q_k = h_2^1 - h_3 = 832.654 - 275.44w = 557.21w$$

- Cycle mass of refrigerant

$$G_a = \frac{Q}{q_o} = \frac{128.24 \times 3.6}{284.4} kg/h = 1.61 kg/h$$

- Condensing heat capacity

$$Q_k = G_a \bullet q_k = G_a \bullet (h_2 - h_3) = \frac{1.61 \times 557.21 \times 10^3}{3600} w = 249.19w$$

- Overheat vapor suctioning by compressor

$$v_s = G_a \bullet v_1^1 = 1.61 \times 0.68371 m^3/h = 1.1 m^3/h$$

3) Calculation for choosing the compressor type

□Calculate the coefficient of gas transmission

- Volume coefficient

$$\lambda_v = 1 - c\left[\left(\frac{p_k + \Delta p_k}{p_o}\right)^{\frac{1}{m}} - 1\right]$$

$$\Delta p_k = 0.1 p_k$$

c — relative clearance volume, c=0.025;
m— expand coefficient, m=1.

$$\lambda_v = 1 - 0.025 \times \left[\left(\frac{0.7814 + 0.1 \times 0.7814}{0.0623}\right)^{\frac{1}{1}} - 1\right] = 0.68$$

- Pressure coefficient

$$\lambda_p = 1 - \frac{1+c}{\lambda_v} \bullet \frac{\Delta p_o}{p_o}$$

$$\Delta p = 0.05 p_o$$

$$\lambda_p = 1 - \frac{1 + 0.025}{0.68} \times 0.05 = 0.924$$

- Temperature coefficient

$$\lambda_t = \frac{T_1''}{aT_k + b\theta}$$

a,b — Coefficient, a=1.15,b=0.25;
T_1'' —Inspiratory temperature, T_1''=273+80=353k;

T_k— Condensation temperature, T_k=273+54.4=327.4k;

θ— Superheat degree, θ= T_1''-(t_o+273)=103.3k.

$$\lambda_t = \frac{353}{1.15\times 327.4+0.25\times 103.3} = 0.87$$

- Leak coefficient

$\lambda_l = 0.99$ (According to the experience)

$$\lambda = \lambda_v \bullet \lambda_p \bullet \lambda_t \bullet \lambda_l$$

$$\lambda = 0.68\times 0.924\times 0.87\times 0.99 = 0.542m^3/kg$$

□Theoretic carry capacity

$$V_h = \frac{V_s}{\lambda} = \frac{1.1}{0.542} = 2.03m^3/h$$

□Cooling capacity of compressor

$$Q_o = q_v \bullet V_h \bullet \lambda$$

$$Q_o = 415.96\times 2.03\times 0.542 = 457kJ/h = 0.127kw$$

□Power of the compressor

- Theoretical adiabatic power

$$p_o = \frac{G_a \bullet w_i}{3600}$$

$$p_o = \frac{1.61\times 101.63}{3600} = 45w$$

- Indicated power

$$p_i = \frac{p_o}{\eta_i}$$

$$\eta_i = \frac{T_o}{T_k} + bt_o$$

b=0.0005, T_o=273-23.3=249.7k, T_k=273+54.4=327.4k;

$$\eta_i = \frac{249.7}{327.4} + 0.0005\times(-32) = 0.704$$

$$P_i = \frac{P_o}{\eta_i} = \frac{45}{0.704} = 63.92w$$

- Friction power

$$P_m = \frac{p_m \bullet v_h}{36.72}$$

P_m— average friction, p_m=0.45Mpa

$$P_m = \frac{0.45 \times 2.03}{36.72} = 25w$$

- Motor shaft horse power

$$P_e = P_i + P_m = 63.92 + 25 = 88.92w$$

- Electric power and motor efficiency

$$P_{el} = \frac{P_e}{\eta_{mo}}$$

$$\eta_{mo} = 0.82$$

$$P_{el} = \frac{88.92}{0.82} = 108.43w$$

According to the result of the theoretic calculation including the theoretic cooling capacity, power of the compressor, electric power and motor efficiency etc, and the design parameter of R600a, we choose the compressor which is produced by BaiXue company, normal model QDHC90 external voltage 220V, input power 128w, external cooling capacity 113w.

4) Calculation of condenser

□Head load in overheat Zone Q_1 and in saturated zone Q_2

$$\beta = \frac{Q_1}{Q_2} = \frac{h_{2k} - h_2}{h_{2k} - h_3}$$ (The percent of the heat load in overheat zone and heat load in saturated zone)

h_{2k}— enthalpy, states t=80℃,p=0.7814Mpa, h_{2k}=681.02kJ/kg;

$$\beta = \frac{681.02 - 629.76}{681.02 - 275.44} = 0.126$$

$$Q_1 = 0.126 Q_k = 0.126 \times 249.19 = 31.40w$$

$$Q_2 = (1 - 0.126) Q_k = 0.874 \times 249.19 = 217.79w$$

□Deference in temperature of overheat □t_1 and saturated zone □t_2

- $$\Delta t_1 = \frac{t_h - t_k}{\ln \frac{t_h - t_k}{t_k - t_a}}$$

t_h— overheat gas temperature, t_k=80℃;

$$\Delta t_1 = \frac{80-54.4}{\ln\frac{80-32}{54.4-32}} = 33.59℃$$

- $\Delta t_2 = t_k - t_a$

$$\Delta t_2 = 54.4 - 32 = 22.4℃$$

□natural convection heat transfer coefficient

- overheat zone

$$a_{of} = 0.94\frac{\lambda_f}{d_e}\left[\frac{(s_b-d_b)(s_w-d_w)}{(s_b-d_b)^2+(s_w-d_w)2}\right]^{0.155}(P_{rf}G_{rf})^{0.16}$$

de —Equivalent diameter;

$$d_e = s_b\left[\frac{1+2\frac{s_b}{s_w}\bullet\frac{d_w}{d_b}}{\left(\frac{s_b}{2.76d_b}\right)^{0.25}+2\frac{s_b}{s_w}\bullet\frac{d_w}{d_b}\eta_f}\right]^4$$

η_f— rib efficiency, η_f=0.85;

$$d_e = 0.048\times\left[\frac{1+2\times\frac{0.048}{0.0055}\times\frac{0.0016}{0.006}}{\left(\frac{0.048}{2.76\times0.006}\right)^{0.25}+2\times\frac{0.048}{0.0055}\times\frac{0.0016}{0.006}\times0.85}\right]^4 = 0.0637$$

g— acceleration of gravity, g=9.8m/s^2;

β— expend coefficient, β=0.0031;

r— Sport glutinosity of the air, r=17.84*10^{-6}m^2/s

$$G_{rf} = g\beta\Delta t_1 d^{\,3}/\gamma^2$$

$$G_{rf} = 9.8\times0.0031\times33.59\times\frac{(0.0637)^3}{(17.84\times10^{-6})^2} = 826997$$

$$P_{rf} = 0.698$$

$$a_{of}' = 0.94\times\frac{2.82\times10^{-2}}{0.0637}\times\left[\frac{(48-6)(5.5-1.6)}{(48-6)^2-(5.5-1.6)^2}\right]^{0.155}\times(0.698\times826997)^{0.26} = 9.07w/m^2\bullet k$$

- Saturated zone

λ_f— when the temperature is 43.2℃, thermal conductivity of the air, 0.278W/(m·k)

r— Sport glutinosity of the air, $r=17.28*10^{-6}m^2/s$

$$P_{rf} = 0.699$$

β— expend coefficient, β=0.00316;

$$G_{rf} = 9.81 \times \frac{1}{273+43.2} \times 22.4 \times \frac{(0.0640)^3}{(17.28 \times 10^{-6})^2} = 599872$$

$$a_{of}'' = 0.94 \times \frac{2.78 \times 10^{-2}}{0.0637} \times \left[\frac{(48-6)(5.5-1.6)}{(48-6)^2 - (5.5-1.6)^2} \right]^{0.155} \times (0.699 \times 599872)^{0.26} = 8.35 w/m^2 \bullet k$$

□Radiation heat transfer coefficient

- Overheat zone

$$a_{or}' = 6.67\varepsilon \frac{\left(\frac{T_1+273}{100}\right)^4 - \left(\frac{T_2+273}{100}\right)^4}{\Delta t_1}$$

ε— Dark coefficient, ε=0.97;

T_1— average temperature of the condenser, T_1=65.59℃

T_2— ambient temperature, T2=32 ℃

$$a_{or}' = 5.67 \times 0.97 \times \frac{\left(\frac{65.59+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{33.59} = 7.35 w/m^2 \bullet k$$

- Saturated zone

T_1— average temperature of the condenser, T_1=54.4℃

$$a_{or}'' = 5.67 \times 0.97 \times \frac{\left(\frac{54.4+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{22.4} = 6.96$$

□ Heat transfer area

$$\eta_o = \frac{a_b + a_p \eta_f}{a_b + a_p}$$ (Exterior efficiency)

a_b— Outside exterior area per m, $a_b = 0.006\pi$

a_p— Inside exterior area per m,

$$a_p = 2\times\left(\frac{1}{s_w}+1\right)\pi d_w s_b$$

$$\eta_o = \frac{0.006\pi+\left(\dfrac{2}{0.0055}+2\right)\times 0.0016\pi\times 0.048\times 0.85}{0.006\pi+\left(\dfrac{2}{0.0055}+2\right)\pi\times 0.048\times 0.0016} = 0.8763$$

- Overheat zone

$$A' = \frac{Q_1}{\left(a_{of}'+a_{or}'\right)\eta_o\Delta t_1}$$

$$A' = \frac{31.40}{(9.07+7.35)\times 0.8763\times 33.59} = 0.0650m^2$$

- Saturated zone

$$A'' = \frac{217.79}{(6.96+8.35)\times 0.8763\times 22.4} = 0.6722m^2$$

Total heat transfer area

$$A = A' + A'' = 0.0650 + 0.6722 = 0.7372m^2$$

□The size of the condenser

- The length of the steel tube

$$L = \frac{A}{a_b + a_p}$$

$$L = \frac{0.7372}{0.088+0.0188} = 6.90m$$

Because when we use the freezer, the condenser would be dirty, in order to make no influence on it, we should increase the length, so the actual length is 1.05L,7.30m, and we do not want to change the mold of the condenser too, so the width of the condenser is the same as the mold, its width is o.45m, the depth is 0.768m. Generally the length of the steel wires is longer than the condenser depth, so the actual length is 0.788m.

5) Calculation of the evaporator

The area of the evaporator is not changed, its is 1.23m^2(A)

The rib efficient is 6.05(β)

So the exterior of the steel tube is

$$A_e = \frac{A}{\beta} = \frac{1.67}{6.05} = 0.276m^2$$

$$L = \frac{A_e}{0.008\pi} = 10.98m$$

Because the temperature of the fresh food compartment and frozen food compartment is deferent, so the area of transfer heat is deferent in them, and the evaporator tube is longer in the frozen food compartment than in the frozen food compartment. According to the experience, the length of the evaporator tube in the frozen food compartment room is 9.48m, the length of the evaporator in the fresh food compartment is 1.5m.

6) Calculation of Capillary(according to the calculation of R12)

In the conditions of the same states, the capillary length of R12.

- $$L = \frac{\Delta P \bullet R_e^{\ 0.25} \bullet d}{0.1582 w^2 \bullet \rho}$$

- $$W = \frac{G_a v}{\frac{\pi}{4} \bullet d^2}$$

d— inside diameter of the capillary, d=0.00071m;

v— bulking value of the liquid, t=15℃,$v=4.21*10^{-3}m^3/kg$;

G_a— Cycle mass of refrigerant R12, G_a=2.96kg/h;

$$W = \frac{2.96 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00071)^2 \times 3600} m/s = 8.74 m/s$$

R_e— Reynolds number

- $$R_e = \frac{Wd\rho}{\mu}$$

ρ— average density, ρ=237.5kg/m3;

μ— average glutinosity, $\mu=0.217*10^{-3}$Pa·s

$$R_e = \frac{8.74 \times 0.00071 \times 237.53}{0.217 \times 10^{-3}} = 6792$$

$$L = \frac{1.216 \times 10^6 \times (6792)^{0.25} \times 0.00071}{0.1582 \times (8.74)^2 \times 237.55} m = 2.71m$$

According to the experiment, we find the capillary length which using the R600a is shorter than the capillary which using the R12, so the actual length is 0.75L, 2.03m.

7) The mass of the perfusion(According to the R12)

$V_n = 0.2268m^3$ —The inside volume of the evaporator

$V_k = 0.1107m^3$ —The inside volume of the condenser

$$m = 0.441 V_n + 0.228 V_k$$

$$m = 0.441 \times 0.2268 + 0.228 \times 0.1107 = 125g$$

Because the perfusion of R600a is less than R12, so the perfusion mass of R600a is 0.45m, is 56g.

2.5 Theoretical calculation for the model RW-60

2.5.1 **Design conditions of RW-60**

Freezer RW60 Design Conditions (according to the standards GB/T8059.1-1995) :

1) Environmental conditions: ambient temperature ta=32□, relative humidity φ=75±5%;
2) Inside temperature of the freezer: td≤-18□, tc≤5□;
3) Storage volume: L_d=25L, L_c=35L;
4) Refrigerant: R600a;
5) Blowing agent: isopentane

Freezer RW60 Construction (shown in Figure 5):

1) Opening type:upright type
2) Overall dimensions:847mm*1073mm*547mm (height*width*depth);
3) Thickness of the adiabatic wall: fresh food storage compartment, right side=34mm, left side=34mm, front side=41mm, back side=30, up side=41.5mm, underside=42.5mm; frozen food storage compartment, right side=42.5mm, left side=42.5mm, front side=41mm, back side=33, up side=42.5mm, underside=40mm.
4) Condenser construction: steel wire-tube type, the diameter of the tube and wire are 4.76mm(d_b),1.6mm(d_w); and the thickness of the steel tube is 0.75mm; space between tubes is 48mm(s_b), and space between wires is 5.5mm(s_w); placed vertically.
5) Evaporator construction: steel tube-board type, the board is made of aluminium and the thickness is 0.4mm, the diameter of the steel tube is 8mm and the thickness is 0.71mm;
6) Expansion device: capillary, the diameter is 1.8mm, the thickness is 0.71mm;
7) Compressor type: hermetically sealed compressor refrigerating compressor (plunger type);

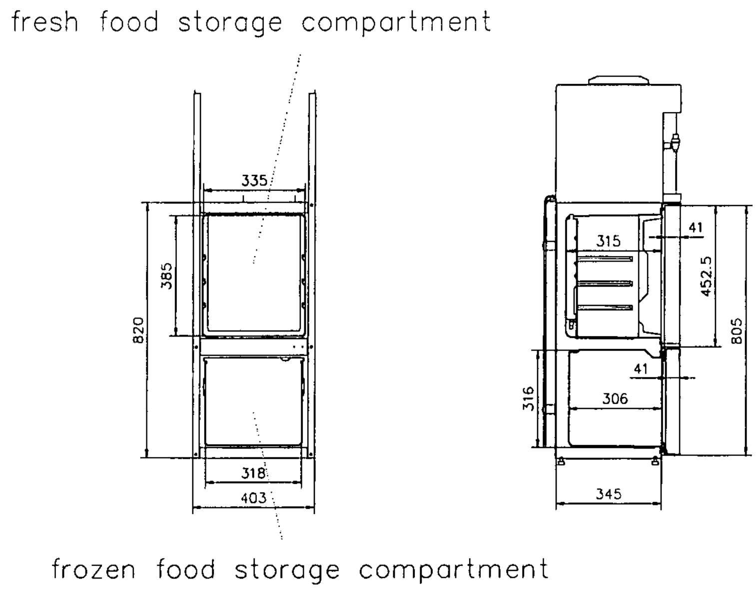

Figure 5

2.5.2 **Theoretic Calculation For Cooling System**

(1) Calculation of heat loss Q_F

The heat loss including three parts, one is the heat loss through the box of the freezer Q_{1F}, the second is for opening the freezer door Q_{2F}, the third is for cooling the goods Q_{3F}.

□Heat loss through the box of the freezer Q_{1F}

Q_{1F} including two parts, one is the heat loss through the adiabatic wall Q_a, the second is through the door Q_b.

$Q_a=KA(t_1- t_2)$

$$K = \frac{1}{\frac{1}{a_1}+\frac{\delta}{\lambda}+\frac{1}{a_2}}$$

K— Coefficient of transfer heat, unit ($W/(m^2\cdot k)$);

t_1— Outside exterior temperature of the freezer, unit (□);

t_2— Inside exterior temperature of the freezer, unit (□);

δ— thickness of the adiabatic wall, unit (m);

λ— heat conductivity of the adiabatic wall, unit ($W/(m\cdot k)$); λ=0.04 $W/(m\cdot k)$

a_1— exterior temperature of the air outside the freezer, unit ($W/(m^2\cdot k)$); a_1=11.3 $W/(m^2\cdot k)$

a_2— exterior temperature of the air inside the freezer, unit ($W/(m^2\cdot k)$); a_2=1.16 $W/(m^2\cdot k)$

A— area of the freezer exterior, unit(m^2)

Heat loss of every sides of the frozen food storage compartment (shown in **Table 15**)

	Up side	Under side	Front side	Back side	Left side	Right side

$\Delta(m)$	42.5	40	41	39	42.5	42.5
$K(W/(m^2{\cdot}k))$	0.513	0.513	0.506	0.519	0.497	0.497
$A(m^2)$	0.126	0.126	0.168	0.168	0.126	0.126
t_1- t_2(□)	23	50	50	52	50	50
Q(w)	1.48	3.24	3.29	3.51	3.13	3.13

Table 15

Q_a=1.48+3.24+3.29+3.51+3.13+3.13=17.78w

Q_b=15%Q_a=0.15*17.78=2.67w

Q_{1F}= Q_a+ Q_b=17.78+2.67=20.44w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.025m^3;

n — times of opening the door per hour, n=2;

□h —enthalpy difference of the air from 32□ to -18□,106.75KJ/kg;

V_a—bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.025 \times 2 \times 106.75}{3.6 \times 0.9} = 1.646w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ·k/kg

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_d=0.125kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.125 \times 4.19 \times 25 + 0.125 \times 333 - 0.125 \times 2 \times (-2)]}{2 \times 3.6} = 7.67w$$

$$Q_F' = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F' = 20.44 + 0.823 + 7.67 = 28.93w$$

Heat loss of every sides of the fresh food storage compartment room (shown in **Table 16**)

	Up side	Under side	Front side	Back side	Left side	Right side
$\Delta(m)$	41.5	42.5	41	30	34	34

K $(W/(m^2 \cdot k))$	0.503	0.497	0.506	0.588	0.556	0.556
$A(m^2)$	0.127	0.127	0.27	0.27	0.145	0.145
t_1- t_2(□)	33	-23	27	27	27	27
Q(w)	2.11	-1.45	3.68	4.28	2.18	2.18

Table 16

Q_a=2.11-1.45+3.68+4.28+2.18+2.18=12.99w

Q_b=15%Q_a=0.15*12.99=1.95w

Q_{1F}= Q_a+ Q_b=12.99+1.95=14.94w

□Heat loss for opening the door Q_{2F}

$$Q_{2F} = \frac{v_B n \Delta h}{3.6 v_a}$$

V_B— storage volume, 0.035m^3;

n — times of opening the door per hour, n=1;

□h —enthalpy difference of the air from 32□ to 5□,71.6KJ/kg;

V_a— bulking value of the air, 0.9m^3/kg.

$$Q = \frac{0.035 \times 1 \times 71.6}{3.6 \times 0.9} = 0.77w$$

□Heat loss for cooling the foods Q_{3F}

$$Q_{3F} = \frac{m_c c t_1 + m_c r - m_c c_b t_2}{2 \times 3.6}$$

c— specific heat capacity of the water, the temperature is 25□,c=4.19kJ/(kg·k);

r— heat of fusion of the water, the temperature is 25□,r= 333kJ/kg;

c_b— heat of fusion of the ice, the temperature is -2□,c_b=2 kJ/(kg·k);

m_c— the mass of the water, m_c=0.005L_c=0.175kg;

t_1 — temperature of the water, t_1=25□;

t_2— temperature of the ice, t_2=-2□.

$$Q_{3F} = \frac{[0.175 \times 4.19 \times 25 + 0.175 \times 333 - 0.175 \times 2 \times (-2)]}{2 \times 3.6} = 10.73w$$

$$Q_F'' = Q_{1F} + Q_{2F} + Q_{3F}$$

$$Q_F'' = 14.94 + 0.77 + 10.73 = 26.44w$$

□ Total heat loss

$$Q_F = Q_F' + Q_F'' = 28.93 + 26.44 = 55.37w$$

The actual heat loss is greater than the value of calculation, so he actual heat loss is 1.2Q_f, 66w.

(2) Circular thermodynamic calculation

□Parameter for design (Shown in **Table 17 and 18**)

Rated states of cooling system

State parameter	Condensation temperature (t_k)	Evaporator temperature (t_0)	Inspiratory temperature (t_G)	Super-cooling temperature (t_s)
Value of design	54.4	-23.3	32(80)	17
Parameter reference	t_k=32+22.4	t_0=-18-5.3	Ambient temperature	t_s=32-15

Table 17

Physical parameter

Parameter name	Sign	Unit	Design value
Condensation pressure	P_k	MPa	0.7814
Evaporator pressure	P_o	MPa	0.0623
Saturated steam enthalpy after the evaporator	h_1	KJkg	523.63
Steam-gas enthalpy before entering the compressor	h_1'	KJ/kg	612.41
Steam-gas bulking value before entering the compressor	v_1'	M^3/kg	0.68371
Steam-gas enthalpy before entering the cylinder	h_1''	KJ/kg	700.36
Steam-gas bulking value before entering the cylinder	v_1''	M^3/kg	0.80290
Temperature of the steam-gas	t_{2s}	℃	95.9
Enthalpy of the saturated steam at the condensation temperature	h_2	KJ/kg	629.76
Enthalpy of the steam-gas	h_2''	KJ/kg	1105.8
Enthalpy of the super-cooling liquid (32℃)	h_3	KJ/kg	275.44
Enthalpy of the super-cooling liquid (17℃□	h_3'	KJ/kg	241.23

Enthalpy of the refrigerant at the entrance to the evaporator	h_4	KJ/kg	241.23
Enthalpy of the compression gas □32℃□	h_{2s}	KJ/kg	714.04
Enthalpy of the compression gas □80℃□	h_{2s}'	KJ/kg	836.31

Table 18

□Calculation

- Refrigerating capacity per mass

$$q_0 = h_1 - h_4 = 523.63 - 241.23 = 284.4kJ/kg$$

- Refrigerating capacity per volume

$$q_v = \frac{q_o}{v_1^1} = \frac{h_1 - h_4}{v_1^1} = \frac{284.4}{0.68371} kg/m^3 = 415.96kg/m^3$$

- Compress work per mass

$$w_i = h_{2s} - h_1^1 = 714.04 - 612.41.04 = 101.63kJ/kg$$

- Refrigerating coefficient

$$\varepsilon = \frac{q_o}{w_i} = \frac{284.4}{101.63} = 2.79$$

- Condensing heat capacity per mass

$$q_k = h_2^1 - h_3 = 832.654 - 275.44w = 557.21w$$

- Cycle mass of refrigerant

$$G_a = \frac{Q}{q_o} = \frac{55.37 \times 3.6}{284.4} kg/h = 0.835kg/h$$

- Condensing heat capacity

$$Q_k = G_a \bullet q_k = G_a \bullet (h_2 - h_3) = \frac{0.835 \times 557.21 \times 10^3}{3600} w = 129.31w$$

- Overheat vapor suctioning by compressor

$$v_s = G_a \bullet v_1^1 = 0.835 \times 0.68371m^3/h = 0.57m^3/h$$

(3) Calculation for choosing the compressor type

□Calculate the coefficient of gas transmission

- Volume coefficient

$$\lambda_v = 1 - c\left[\left(\frac{p_k + \Delta p_k}{p_o}\right)^{\frac{1}{m}} - 1\right]$$

$$\Delta p_k = 0.1 p_k$$

c — relative clearance volume, c=0.025;

m— expand coefficient, m=1.

$$\lambda_v = 1 - 0.025 \times \left[\left(\frac{0.7814 + 0.1 \times 0.7814}{0.0623}\right)^{\frac{1}{1}} - 1\right] = 0.68$$

- Pressure coefficient

$$\lambda_p = 1 - \frac{1+c}{\lambda_v} \bullet \frac{\Delta p_o}{p_o}$$

$$\Delta p = 0.05 p_o$$

$$\lambda_p = 1 - \frac{1+0.025}{0.68} \times 0.05 = 0.924$$

- Temperature coefficient

$$\lambda_t = \frac{T_1''}{aT_k + b\theta}$$

a,b — Coefficient, a=1.15,b=0.25;

T_1'' —Inspiratory temperature, T_1''=273+80=353k;

T_k— Condensation temperature, T_k=273+54.4=327.4k;

θ— Superheat degree, θ= T_1''-(t_o+273)=103.3k.

$$\lambda_t = \frac{353}{1.15 \times 327.4 + 0.25 \times 103.3} = 0.87$$

- Leak coefficient

$\lambda_l = 0.99$ (According to the experience)

$$\lambda = \lambda_v \bullet \lambda_p \bullet \lambda_t \bullet \lambda_l$$

$$\lambda = 0.68 \times 0.924 \times 0.87 \times 0.99 = 0.542 m^3 / kg$$

□Theoretic carry capacity

$$V_h = \frac{V_s}{\lambda} = \frac{0.57}{0.542} = 1.05 m^3 / h$$

□Cooling capacity of compressor

$Q_o = q_v \bullet V_h \bullet \lambda$

$Q_o = 415.96 \times 1.05 \times 0.542 = 237.09 kJ/h = 65.86 kw$

□Power of the compressor

- Theoretical adiabatic power

$$p_o = \frac{G_a \bullet w_i}{3600}$$

$$p_o = \frac{0.835 \times 101.63}{3600} = 23.6w$$

- Indicated power

$$p_i = \frac{p_o}{\eta_i}$$

$$\eta_i = \frac{T_o}{T_k} + bt_o$$

b=0.0005, T_o=273-23.3=249.7k, T_k=273+54.4=327.4k;

$$\eta_i = \frac{249.7}{327.4} + 0.0005 \times (-23.3) = 0.704$$

$$P_i = \frac{P_o}{\eta_i} = \frac{23.6}{0.704} = 33.52w$$

- Friction power

$$P_m = \frac{p_m \bullet v_h}{36.72}$$

P_m— average friction, p_m=0.65Mpa

$$P_m = \frac{0.65 \times 1.07}{36.72} = 18.9w$$

- Motor shaft horse power

$$P_e = P_i + P_m = 33.52 + 18.9 = 52.42w$$

- Electric power and motor efficiency

$$P_{el} = \frac{P_e}{\eta_{mo}}$$

$$\eta_{mo} = 0.70$$

$$P_{el} = \frac{52.42}{0.70} = 74.89w$$

According to the result of the theoretic calculation including the theoretic cooling capacity, power of the compressor, electric power and motor efficiency etc, and the design parameter of R600a, we choose the compressor which is produced by BaiXue company, normal model QDHC51 external voltage 220V, input power 80w, external cooling capacity 68w.

(4) Calculation of condenser

□Head load in overheat Zone Q_1 and in saturated zone Q_2

$$\beta = \frac{Q_1}{Q_2} = \frac{h_{2k} - h_2}{h_{2k} - h_3}$$ (The percent of the heat load in overheat zone and heat load in saturated zone)

h_{2k}— enthalpy, states t=80℃,p=0.7814Mpa, h_{2k}=681.02kJ/kg;

$$\beta = \frac{681.02 - 629.76}{681.02 - 275.44} = 0.126$$

$$Q_1 = 0.126 Q_k = 0.126 \times 129.31 = 16.28w$$

$$Q_2 = (1 - 0.126) Q_k = 0.874 \times 129.31 = 113.02w$$

□Deference in temperature of overheat □t_1 and saturated zone □t_2

- $$\Delta t_1 = \frac{t_h - t_k}{\ln \frac{t_h - t_k}{t_k - t_a}}$$

t_h— overheat gas temperature, t_k=80℃;

$$\Delta t_1 = \frac{80 - 54.4}{\ln \frac{80 - 32}{54.4 - 32}} = 33.59\,℃$$

- $$\Delta t_2 = t_k - t_a$$

$$\Delta t_2 = 54.4 - 32 = 22.4\,℃$$

□natural convection heat transfer coefficient

- overheat zone

$$a_{of} = 0.94 \frac{\lambda_f}{d_e} \left[\frac{(s_b - d_b)(s_w - d_w)}{(s_b - d_b)^2 + (s_w - d_w)2} \right]^{0.155} (P_{rf} G_{rf})^{0.16}$$

de —Equivalent diameter;

$$d_e = s_b \left[\frac{1 + 2\dfrac{s_b}{s_w} \bullet \dfrac{d_w}{d_b}}{\left(\dfrac{s_b}{2.76 d_b}\right)^{0.25} + 2\dfrac{s_b}{s_w} \bullet \dfrac{d_w}{d_b} \eta_f} \right]^4$$

η_f— rib efficiency, η_f=0.85;

$$d_e = 0.048 \times \left[\frac{1 + 2 \times \dfrac{0.048}{0.0055} \times \dfrac{0.0016}{0.00476}}{\left(\dfrac{0.048}{2.76 \times 0.00476}\right)^{0.25} + 2 \times \dfrac{0.048}{0.0055} \times \dfrac{0.0016}{0.00476} \times 0.85} \right]^4 = 0.0648$$

g— acceleration of gravity, g=9.8m/s^2;

β— expend coefficient, β=0.0031;

r— Sport glutinosity of the air, r=17.84*10^{-6}m^2/s

$$G_{rf} = g\beta\Delta t_1 d^3 / \gamma^2$$

$$G_{rf} = 9.8 \times 0.0031 \times 33.59 \times \frac{(0.0648)^3}{(17.84 \times 10^{-6})^2} = 870583$$

$$P_{rf} = 0.698$$

$$a_{of}' = 0.94 \times \frac{2.82 \times 10^{-2}}{0.0648} \times \left[\frac{(48 - 4.76)(5.5 - 1.6)}{(48 - 4.76)^2 - (5.5 - 1.6)^2} \right]^{0.155} \times (0.698 \times 870583)^{0.26} = 8.99 w/m^2 \bullet k$$

- Saturated zone

λ_f— when the temperature is 43.2℃, thermal conductivity of the air, 0.278W/(m·k)

r— Sport glutinosity of the air, r=17.28*10^{-6}m^2/s

$$P_{rf} = 0.699$$

β— expend coefficient, β=0.00316;

$$G_{rf} = 9.81 \times \frac{1}{273 + 43.2} \times 22.4 \times \frac{(0.0648)^3}{(17.28 \times 10^{-6})^2} = 631488$$

$$a_{of}'' = 0.94 \times \frac{2.78 \times 10^{-2}}{0.0648} \times \left[\frac{(48 - 4.76)(5.5 - 1.6)}{(48 - 4.76)^2 - (5.5 - 1.6)^2} \right]^{0.155} \times (0.699 \times 631488)^{0.26} = 8.16 w/m^2 \bullet k$$

□Radiation heat transfer coefficient

- Overheat zone

$$a_{or}{}' = 6.67\varepsilon \frac{\left(\frac{T_1+273}{100}\right)^4 - \left(\frac{T_2+273}{100}\right)^4}{\Delta t_1}$$

ε— Dark coefficient, ε=0.97;

T_1— average temperature of the condenser, T_1=65.59℃

T_2— ambient temperature, T2=32 ℃

$$a_{or}{}' = 5.67 \times 0.97 \times \frac{\left(\frac{65.59+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{33.59} = 7.35 w/m^2 \bullet k$$

- Saturated zone

T_1— average temperature of the condenser, T_1=54.4℃

$$a_{or}{}'' = 5.67 \times 0.97 \times \frac{\left(\frac{54.4+273}{100}\right)^4 - \left(\frac{32+273}{100}\right)^4}{22.4} = 6.96$$

□ Heat transfer area

$$\eta_o = \frac{a_b + a_w \eta_f}{a_b + a_w}$$ (Exterior efficiency)

a_b— Outside exterior area per m, $a_b = 0.00476\pi$

a_p— Inside exterior area per m,

$$a_p = 2 \times \left(\frac{1}{s_w} + 1\right) \pi d_w s_b$$

$$\eta_o = \frac{0.00476\pi + \left(\frac{2}{0.0055} + 2\right) \times 0.0016\pi \times 0.048 \times 0.85}{0.00476\pi + \left(\frac{2}{0.0055} + 2\right) \pi \times 0.048 \times 0.0016} = 0.8726$$

- Overheat zone

$$A' = \frac{Q_1}{\left(a_{of}{}' + a_{or}{}'\right) \eta_o \Delta t_1}$$

$$A' = \frac{13.75}{(8.99 + 7.35) \times 0.8726 \times 33.59} = 0.0339 m^2$$

● Saturated zone

$$A'' = \frac{95.44}{(6.96 + 8.16) \times 0.8726 \times 22.4} = 0.3824m^2$$

Total heat transfer area

$$A = A' + A'' = 0.0339 + 0.3824 = 0.4163m^2$$

Because the condenser is placed vertically, the coefficient of heat transfer is less than the condenser which is placed flatly, so the actual area of the condenser is greater than the result of theoretic calculation, the actual area is 1.4A, $0.5828m^2$.

□The size of the condenser

● The length of the steel tube

$$L = \frac{A}{a_b + a_p}$$

$$L = \frac{0.5828}{0.088 + 0.0149} = 5.66m$$

Because when we use the freezer, the condenser would be dirty, in order to make no influence on it, we should increase the length, so the actual length is 1.05L, 5.94m. we do not want to change the mold of the condenser, so the width of the condenser is the same as the mold, its width is 0.45m, so the horizontal amount N is 5.94/0.45=12, the depth is 0.576m. Generally the length of the steel wires is longer than the condenser depth, so the actual length is 0.58m.

(5) Calculation of the evaporator

The evaporator is not manufactured by our company, we buy it from other factory, so we do not want to change the size of it, because changing it will increase the cost. We do a lot of experiments for the matching, the result of the experiments is the size of the evaporator and the area of the heat transfer satisfied with the cooling system. So the evaporator is as the same as before.

(6) Calculation of Capillary(according to the calculation of R12)

In the conditions of the same states, the capillary length of R12.

● $$L = \frac{\Delta P \bullet R_e^{0.25} \bullet d}{0.1582w^2 \bullet \rho}$$

● $$W = \frac{G_a v}{\frac{\pi}{4} \bullet d^2}$$

d— inside diameter of the capillary, d=0.00066m;

v— bulking value of the liquid, t=15℃, $v=4.21*10^{-3}m^3/kg$;

G_a— Cycle mass of refrigerant R12, G_a=1.95kg/h;

$$W = \frac{1.95 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00061)^2 \times 3600} m/s = 7.81 m/s$$

R$_e$— Reynolds number

- $R_e = \frac{Wd\rho}{\mu}$

ρ— average density, ρ=237.5kg/m3;
μ— average glutinosity, μ=0.217*10^{-3}Pa·s

$$R_e = \frac{7.81 \times 0.00061 \times 237.53}{0.217 \times 10^{-3}} = 5215$$

$$L = \frac{1.216 \times 10^6 \times (5215)^{0.25} \times 0.00061}{0.1582 \times (7.81)^2 \times 237.55} m = 2.7m$$

According to the experiment, we find the capillary length which using the R600a is shorter than the capillary which using the R12, so the actual length is 0.75L, 2.0m.

(7) The mass of the perfusion(According to the R12)

$V_n = 0.1117m^3$—The inside volume of the evaporator

$V_k = 0.0520m^3$—The inside volume of the condenser

$$m = 0.441V_n + 0.228V_k$$

$$m = 0.441 \times 0.1117 + 0.228 \times 0.0520 = 71g$$

Because the perfusion of R600a is less than R12, so the perfusion mass of R600a is 0.45m, is 32g.

Part 3 Optical Design, Conclusions and Testing Data

3.1 Introduction to Energy Consumption Test

Pull down have optimized the capillary tube and charge in the R600a version on of the freezer. The storage test had shown that either the R12 nor the R600a freezer could maintain the storage temperature of the center, top M pack to -18℃ so the load line was reduced to 75mm below the lid.

Both freezers then could maintain the temperature of the warmest M pack to -18℃. The aim of the test described in this report was to measure the energy consumption of the converted R600a and the R12 freezers according to the method described in ISO7371:1995, ISO8187:1991 and ISO5055:1995(the standards). These standards are basically equal to the Chinese Standards GB/T8059.1-1995, GB/T8059.2-1995 and GB/T8059.3-1995.

3.2 Optimized Design

3.2.1 The model of BD/C193

The length of the capillary and perfusion of the refrigerant is very important for a cooling system, through theoretical calculation to them is not exact, it must be confirmed through experiment which shows the relations of the length of the capillary and perfusion of the refrigerant, so we do a lot of experiments through change the length and perfusion of the refrigerant, then we can confirm the most measurable value of the length of the capillary and perfusion of the refrigerant.

Because of the capillary tube length of R600a is too short to operate, and the area of the heat transfer is so small, we choose the scale which the inner diameter is 0.71mm.
Calculation the length of the capillary tube
The other parameters is not changed

$$W = \frac{2.96 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00071)^2 \times 3600} m/s = 8.74 m/s$$

$$R_e = \frac{8.74 \times 0.00071 \times 237.53}{0.217 \times 10^{-3}} = 6792$$

$$L = \frac{1.216 \times 10^6 \times (6792)^{0.25} \times 0.00071}{0.1582 \times (8.74)^2 \times 237.55} m = 2.71 m$$

The actual length is 0.75L, 2.03m, the changed length answered for the demand of operation and heat transfer.
The results of the experiment shows the relations of the length of the capillary (shown in the **Table 19**), through the experiment we find when the length is 1.93m, the energy consumption is the least.

Length (m)	1.73	1.83	1.93	2.03	2.13	2.23	2.33	2.43
Energy consumption (kw/24h)	1.09	1.07	1.05	1.06	1.07	1.09	1.10	1.11

Table 19

After doing the experiment of the length of capillary, we do the experiment for the perfusion of the refrigerant R600a, aiming at every length of the capillary ,we do the experiment through changing the perfusion of refrigerant, the results of the experiment shows in **Table 20** and **Figure 6**.

Perfusion (g)	1.73	1.83	1.93	2.03	2.13	2.23	2.33	2.43
48	1.07	1.06	1.05	1.06	1.08	1.09	1.11	1.13
53	1.06	1.04	1.03	1.04	1.06	1.08	1.09	1.12

58	1.05	1.03	1.01	1.04	1.06	1.08	1.09	1.11
63	1.09	1.07	1.05	1.06	1.07	1.09	1.10	1.11
68	1.08	1.09	1.06	1.08	1.08	1.10	1.11	1.13
73	1.11	1.10	1.08	1.09	1.08	1.10	1.12	1.14
78	1.13	1.12	1.11	1.10	1.09	1.09	1.12	1.13
83	1.15	1.13	1.13	1.12	1.11	1.12	1.12	1.13

Table 20

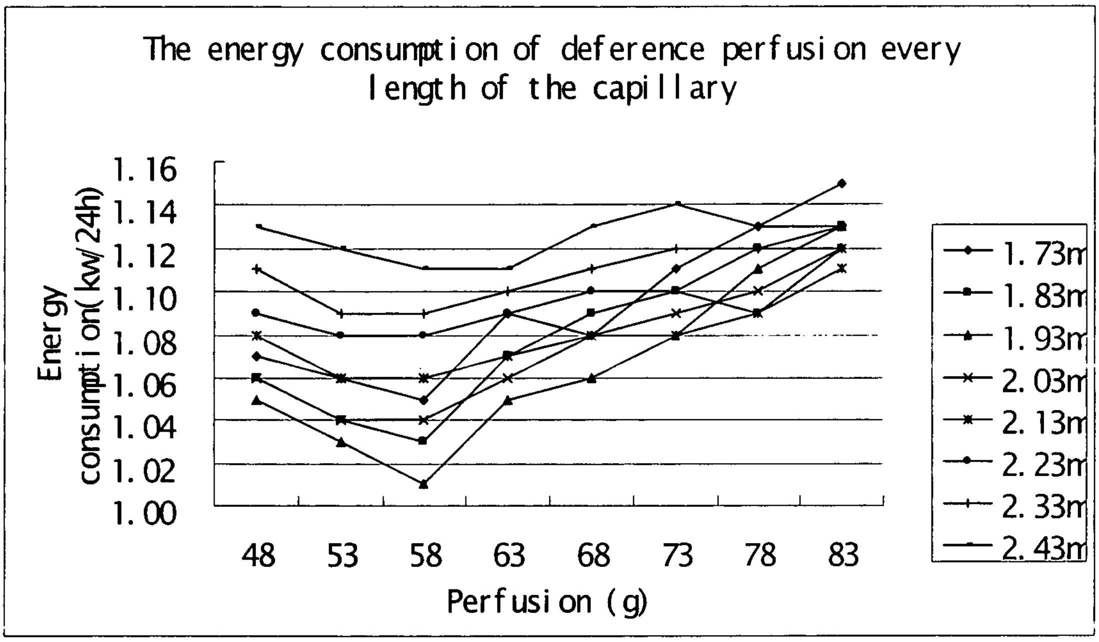

Figure 6

Conclusions

Through the experiment for changing the length of the capillary and perfusion of the refrigerant, we can find when the length is 1.93m and the perfusion is 58g, the energy consumption is the least.

3.2.2 The model of BD/C-216

The length of the capillary and perfusion of the refrigerant is very important for a cooling system, through theoretical calculation to them is not exact, it must be confirmed through experiment which shows the relations of the length of the capillary and perfusion of the refrigerant, so we do a lot of experiments through change the length and perfusion of the refrigerant, then we can confirm the most measurable value of the length of the capillary and perfusion of the refrigerant.

Because of the capillary tube length of R600a is too short to operate, and the area of the heat transfer is so small, we choose the scale which the inner diameter is 0.75mm.
Calculation the length of the capillary tube
The other parameters is not changed

$$W = \frac{3.55 \times 4.21 \times 10^{-3}}{\frac{3.14}{4} \times (0.00075)^2 \times 3600} m/s = 9.40 m/s$$

$$R_e = \frac{10.19 \times 0.00075 \times 237.53}{0.217 \times 10^{-3}} = 7716$$

$$L = \frac{1.216 \times 10^6 \times (7716)^{0.25} \times 0.00075}{0.1582 \times (9.40)^2 \times 237.55} m = 2.60m$$

The actual length is 0.75L, 1.95m, the changed length answered for the demand of operation and heat transfer.

The results of the experiment shows the relations of the length of the capillary (shown in the **Table 21**), through the experiment we find when the length is 1.90m, the energy consumption is the least.

Length (m)	1.80	1.85	1.90	1.95	2.00	2.05	2.10	2.15
Energy consumption (kw/24h)	1.19	1.17	1.16	1.18	1.19	1.20	1.21	1.21

Table 21

After doing the experiment of the length of capillary, we do the experiment for the perfusion of the refrigerant R600a, aiming at every length of the capillary, we do the experiment through changing the perfusion of refrigerant, the results of the experiment shows in **Table22** and **Figure 7**

Perfusion (g)	1.80	1.85	1.90	1.95	2.00	2.05	2.10	2.15
51	1.20	1.19	1.18	1.18	1.19	1.20	1.21	1.22
56	1.18	1.17	1.16	1.17	1.18	1.19	1.20	1.21
61	1.17	1.16	1.15	1.16	1.17	1.18	1.19	1.20
66	1.18	1.17	1.16	1.17	1.18	1.19	1.20	1.21
71	1.19	1.17	1.16	1.18	1.19	1.20	1.21	1.21
76	1.20	1.19	1.17	1.18	1.19	1.20	1.21	1.22
81	1.21	1.20	1.18	1.19	1.21	1.21	1.22	1.23
86	1.22	1.20	1.19	1.20	1.21	1.22	1.23	1.24

Table 22

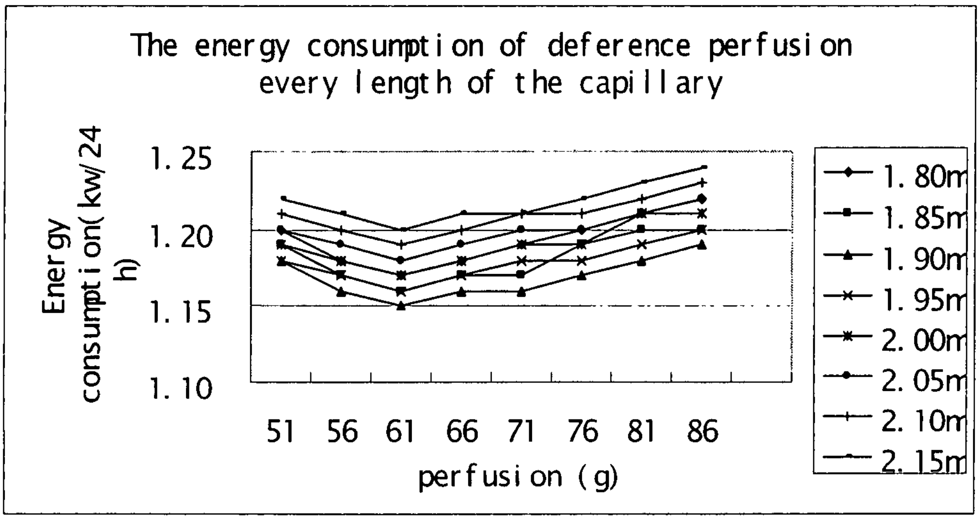

Figure 7

Conclusions

Through the experiment for changing the length of the capillary and perfusion of the refrigerant, we can find when the length is 1.90m and the perfusion is 61g, the energy consumption is the least.

3.2.3 The model of Lcd207

The length of the capillary and perfusion of the refrigerant is very important for a cooling system, through theoretical calculation to them is not exact, it must be confirmed through experiment which shows the relations of the length of the capillary and perfusion of the refrigerant, so we do a lot of experiments through change the length and perfusion of the refrigerant, then we can confirm the most measurable value of the length of the capillary and perfusion of the refrigerant.

The actual length is 0.75L, 2.03m, the changed length answered for the demand of operation and heat transfer.
The results of the experiment shows the relations of the length of the capillary (shown in the **Table 23**), through the experiment we find when the length is 1.98m, the energy consumption is the least.

Length (m)	1.60	1.70	1.80	1.90	2.00	2.10	2.20	2.30
Energy consumption (kw/24h)	1.03	1.02	1.01	1.00	1.01	1.02	1.03	1.04

Table 23

After doing the experiment of the length of capillary, we do the experiment for the perfusion of the refrigerant R600a, aiming at every length of the capillary ,we do the experiment through changing the perfusion of refrigerant, the results of the experiment shows in **Table 24** and **Figure 8**

Perfusion (g)	1.60	1.70	1.80	1.90	2.00	2.10	2.20	2.30
37	1.02	1.01	1.00	0.99	1.00	1.01	1.02	1.03
41	1.01	1.00	0.99	0.98	0.99	1.00	1.01	1.02
45	1.00	0.99	0.98	0.97	0.98	0.99	1.00	1.01
49	0.99	0.98	0.97	0.96	0.97	0.97	0.99	1.00
53	0.97	0.96	0.95	0.94	0.95	0.96	1.00	1.00
57	0.98	0.97	0.96	0.95	0.96	0.97	1.01	1.02
61	0.99	0.98	0.97	0.96	0.97	0.98	1.02	1.03
65	1.00	0.99	0.98	0.97	0.98	0.99	1.03	1.04

Table 24

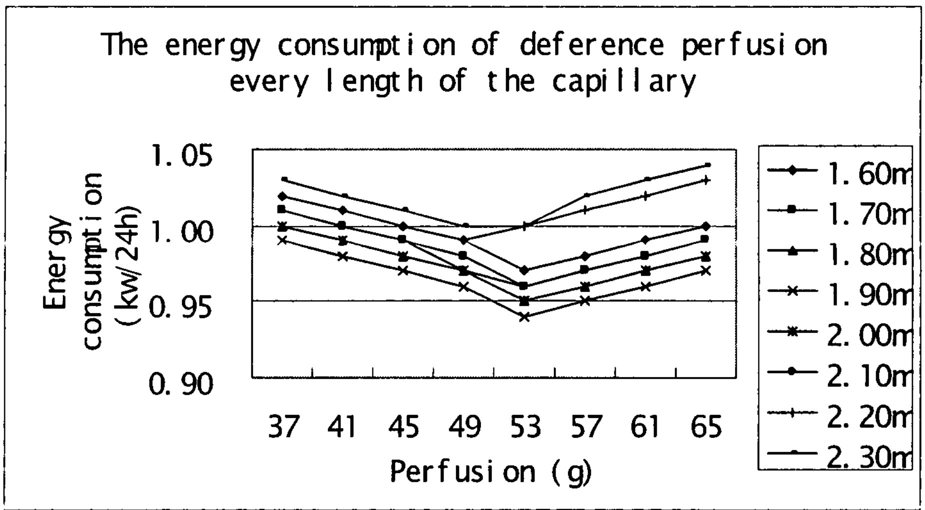

Figure 8

Conclusions

Through the experiment for changing the length of the capillary and perfusion of the refrigerant, we can find when the length is 2.00m and the perfusion is 53g, the energy consumption is the least.

3.2.4 The model of LCD228

The length of the capillary and perfusion of the refrigerant is very important for a cooling system, through theoretical calculation to them is not exact, it must be confirmed through experiment which shows the relations of the length of the capillary and perfusion of the refrigerant, so we do a lot of experiments through change the length and perfusion of the refrigerant, then we can confirm the most measurable value of the length of the capillary and perfusion of the refrigerant.

The actual length is 0.75L, 2.03m, the changed length answered for the demand of operation and heat transfer.

The results of the experiment shows the relations of the length of the capillary (shown in the **Table 25**), through the experiment we find when the length is 1.98m, the energy consumption is the least.

Length (m)	1.88	1.93	1.98	2.03	2.08	2.13	2.18	2.23
Energy consumption (kw/24h)	1.12	1.11	1.10	1.11	1.12	1.13	1.14	1.15

Table 25

After doing the experiment of the length of capillary, we do the experiment for the perfusion of the refrigerant R600a, aiming at every length of the capillary, we do the experiment through changing the perfusion of refrigerant, the results of the experiment shows in **Table 26** and **Figure 9.**

Perfusion (g)	1.88	1.93	1.98	2.03	2.08	2.13	2.18	2.23
48	1.12	1.11	1.10	1.11	1.12	1.13	1.14	1.15
50	1.11	1.10	1.09	1.10	1.11	1.12	1.13	1.14
52	1.10	1.09	1.08	1.09	1.10	1.11	1.12	1.13
54	1.11	1.10	1.09	1.10	1.11	1.12	1.13	1.14
56	1.12	1.11	1.10	1.11	1.12	1.13	1.14	1.15
58	1.13	1.12	1.11	1.12	1.13	1.14	1.15	1.16
60	1.14	1.13	1.12	1.13	1.14	1.15	1.16	1.17
62	1.15	1.14	1.13	1.14	1.15	1.16	1.17	1.18

Table 26

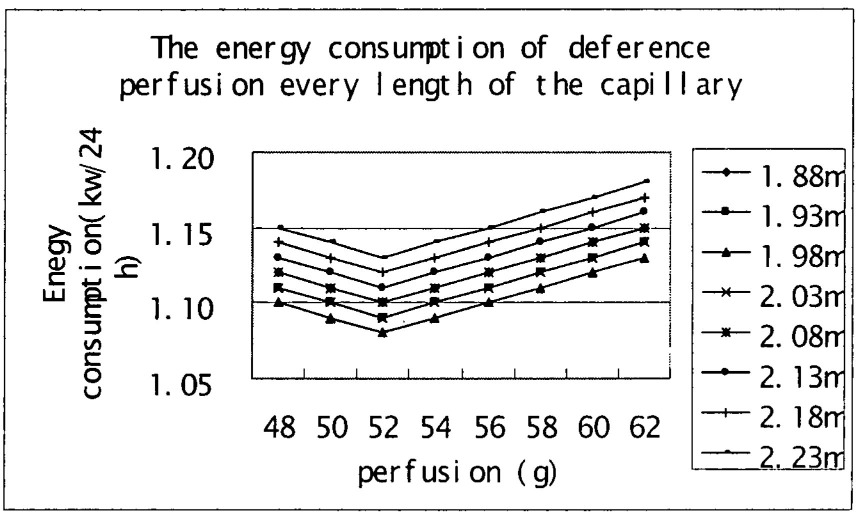

Figure 9

Conclusions

Through the experiment for changing the length of the capillary and perfusion of the refrigerant, we can find when the length is 1.98m and the perfusion is 54g, the energy consumption is the least.

3.2.5 The model of Rw60

The length of the capillary and perfusion of the refrigerant is very important for a cooling system, through theoretical calculation to them is not exact, it must be confirmed through experiment which shows the relations of the length of the capillary and perfusion of the refrigerant, so we do a lot of experiments through change the length and perfusion of the refrigerant, then we can confirm the most measurable value of the length of the capillary and perfusion of the refrigerant.

The actual length is 0.75L, 2.00m, the changed length answered for the demand of operation and heat transfer.
The results of the experiment shows the relations of the length of the capillary (shown in the **Table 27**), through the experiment we find when the length is 1.70m, the energy consumption is the least.

Length (m)	1.40	1.55	1.70	1.85	2.00	2.15	2.30	2.45
Energy consumption (kw/24h)	0.80	0.79	0.78	0.79	0.80	0.81	0.82	0.83

Table 27

After doing the experiment of the length of capillary, we do the experiment for the perfusion of the refrigerant R600a, aiming at every length of the capillary ,we do the experiment through changing the perfusion of refrigerant, the results of the experiment shows in **Table 28** and **Figure 10**

Perfusion (g)	1.40	1.55	1.70	1.85	2.00	2.15	2.30	2.45
20	0.79	0.78	0.77	0.78	0.79	0.80	0.81	0.82
23	0.78	0.77	0.76	0.77	0.78	0.79	0.80	0.81
26	0.77	0.76	0.75	0.76	0.78	0.79	0.80	0.82
29	0.78	0.77	0.76	0.77	0.79	0.80	0.81	0.83
32	0.80	0.79	0.78	0.79	0.80	0.81	0.82	0.83
35	0.81	0.80	0.79	0.80	0.81	0.82	0.82	0.84
38	0.82	0.81	0.80	0.81	0.82	0.83	0.83	0.84
41	0.83	0.82	0.81	0.82	0.83	0.84	0.84	0.85

Table 28

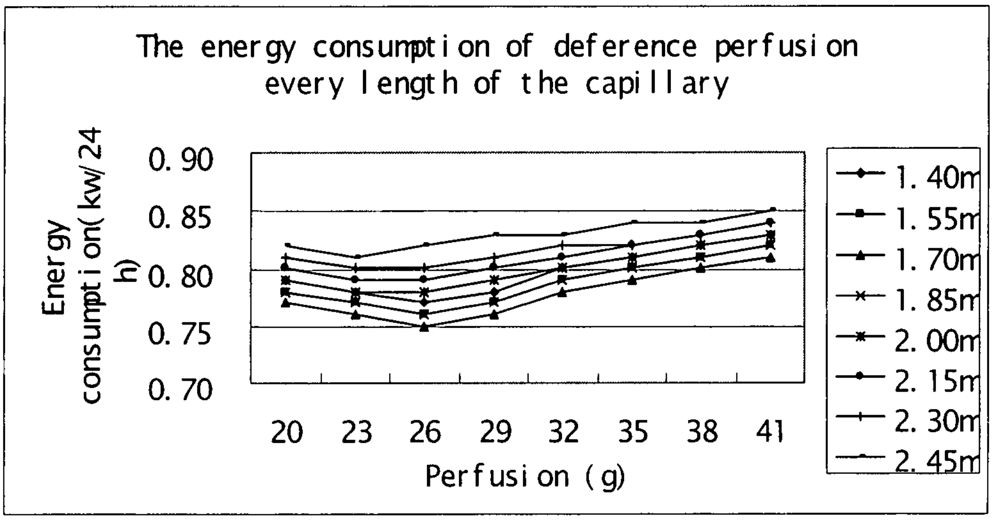

Figure 10

Conclusions

Through the experiment for changing the length of the capillary and perfusion of the refrigerant, we can find when the length is 1.70m and the perfusion is 26g, the energy consumption is the least.

3.3 Testing Data (See the accessory files)

3.4 Conclusions

From the testing data for each models, we can see that all the sample machine's storage temperature can attain the standards for climate type of "N" and "ST". And energy consumption of each model does not exceed the former level with R12 as the refrigerant.