Design Principle and Application Case of HXSim Small Diameter Copper Tube Heat Exchanger Simulation Software

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Contents

1.Heat exchanger design criteria

1.1 Fin pattern and fin pitch of heat exchanger

1.1.1 Fin pattern and tube diameter of heat exchanger

The general principles for the selection of fin pattern and the tube diameter of heat exchanger include the following: (1) When the heat exchanger is used as condenser only, or as an evaporator under non-frosting conditions, or as an evaporator with extremely low moisture content and not easy to frosting, enhanced fins are recommended, such as slit fins, louver fins, supper slit fins (upper and lower bridges), etc. (Note: the heat transfer coefficient of the enhanced fin is 20% - 30% higher than that of the wavy fin; (2) Flat fin or wavy fin is recommended when the heat exchanger is used as evaporator under frosting condition (such as the outdoor unit of the heat pump), or when the applied condition is relatively bad, and it is easy to be blocked by particles; (3) For high-pressure refrigerant, smaller diameter tube is recommended. For example, 5 mm and 7 mm diameter copper tubes are widely used in the R32 system. For low-pressure refrigerant or ultra-low temperature heating system, a larger diameter copper tube is recommended; (4) for applications with low rated capacity, the smaller diameter copper tube should be used for heat exchanger, which can better reflect the capacity of the heat exchanger and reduce the cost.

Table 1-1 Application conditions of common fin and tube types

NO.	Tube type	Fin pattern	Recommended application conditions
1	5	Slit/Louver	● Heat exchange capacity of the heat exchanger less than 3500 W (including), and used for single cooling or both cooling and heating conditions, but not used for frosting conditions, for example, the indoor unit of the split air conditioner. ● Single cooling condenser with heat capacity less than 5000 W (including), for example, the outdoor

			unit of the single cooling split air conditioner. ● Other conditions that refrigerant filling capacity has clear restrictions or requirements, but the application environment is relatively clean.
2	5	Wavy	● Heat exchange capacity of the heat exchanger with less than 3500 W for both cooling and heating conditions, and the heat exchanger may be used in frosting conditions, for example, the outdoor unit of the heat pump split air conditioner. ● Other conditions that application environments are relatively harsh and easy to be blocked by particles, but the heat exchange is small, generally less than 3500 W.
2	7	Wavy	● The heat exchange capacity is 3500 W or above, which can be used as a heat exchanger for both cooling and heating, and the heat exchanger may be used for frosting conditions, such as the outdoor unit of the heat pump split air conditioner. ● Other conditions that application environments are relatively harsh and easy to be blocked by dust particles, and the heat exchange is generally more than 3500 W. ● It is used to replace the 9.52 mm wavy fin.
3	7	Slit/Louver	● The heat exchanger with heat exchange capacity of more than 3500 W (including) is used for cooling or both cooling and heating, but the heat exchanger is not used for frosting conditions, for example, the indoor unit of the split air conditioner. ● Single condenser with heat exchange capacity more

			than 5000 W (including), for example, the outdoor unit of the cooling only split air conditioner. ● It is used to replace the 9.52 mm slit fin.

1.1.2 Fin pitch of heat exchanger

The fin pitch of the fin and tube heat exchanger is generally between 1.3 mm and 2.0 mm. Generally speaking, (I) the smaller the fin pitch is, the more compact the heat exchanger is, the less copper tube is used, and the higher the relative economy is. (II) the smaller the fin pitch is, the worse the ability of the heat exchanger to resist dust, and the worse the long-term performance of the heat exchanger is, especially when the fin pitch of the slit fin is less than 1.2 mm. (III) for the frosting characteristics when the overall size of the heat exchanger is the same, generally, the smaller the fin pitch is, and the greater the output heat of the system in the whole frosting cycle is. This is because after the fin pitch is reduced, the heat exchange area is increased, which can improve the evaporation temperature of the heat exchanger.

Because of the requirements of energy efficiency standards, the selection of fin pitch should be as small as possible based on ensuring the processing quality to give full play to the performance of the heat exchanger at the minimum cost.

1.2 Flow path design of heat exchanger

1.2.1 General principles

Fin and tube heat exchanger is the core component of common air conditioner, refrigerated display cabinet, refrigerated cabinet, air-source heat pump, refrigerator, and other refrigeration systems. If the flow path design of the heat exchanger is unreasonable, the heat exchange performance of the heat exchanger will be reduced, and the cooling or heating efficiency of the system will be reduced by 10-30%. At present, the unreasonable design of heat exchanger flow path mainly occurs when it is used as an evaporator, such as the heat exchanger of the outdoor unit when heating and the heat exchanger of the indoor unit when cooling in the split air conditioner. The reason for the above phenomenon is that the inlet air volume in each area of the heat exchanger is not uniform due to the layout of the heat exchanger, and at the same time, the distributor itself will also cause a large gas-liquid maldistributed, especially when the liquid tube layout is unreasonable, which will lead to the partial dry out and overheating of the whole heat exchanger, and consequently greatly reducing the effective heat exchange area of the heat exchanger.

Therefore, the following principles should be followed when designing the flow path of the heat exchanger.

(1) When the heat exchanger is used as the condenser, the overall flow direction of the refrigerant should be counterflow; that is, the inlet gas tube should be designed on the leeward side, and the outlet liquid tube should be designed on the windward side.

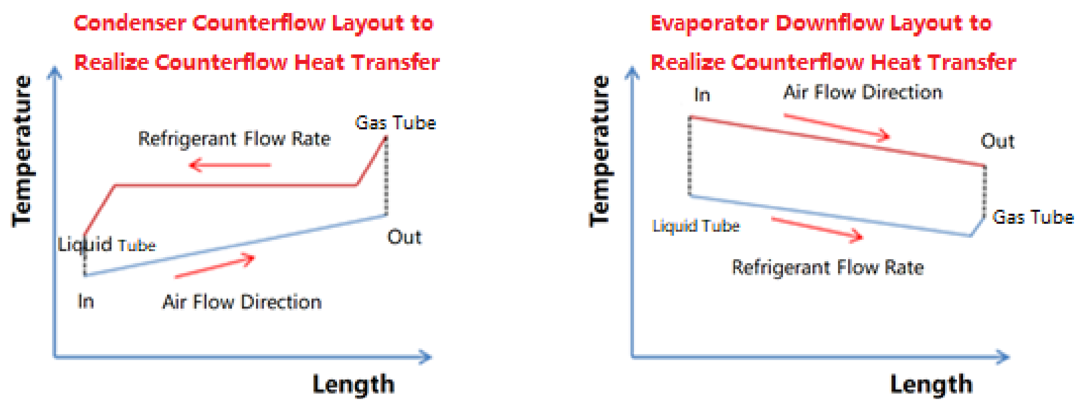
(2) When the heat exchanger is used as the evaporator, the overall direction of refrigerant should be parallel flow, that is the inlet liquid tube should be designed on the windward side, and the outlet gas tube should be designed on the leeward side; this design can make the heat exchanger have the maximum heat transfer temperature difference, as shown in Figure 3-1. It should be specially pointed out that the inlet temperature of the evaporator will be higher than the outlet temperature even if the refrigerant is superheated at the outlet due to the temperature slip of the refrigerant (especially when the small diameter copper tube is used, or the number of the path is less). Therefore, only the downstream flow path design can realize the counterflow heat transfer.

(3) Before the liquid tube module of the heat exchanger enters into the distributor, it is necessary to ensure that there is a straight tube section with a length of more than 25 times the outer diameter of the liquid tube, to ensure that the two-phase flow is fully mixed before entering the distributor; in the design, it is necessary to avoid that the refrigerant directly enters into the distributor through L-Bend or U-bend without any rectification. For the indoor unit with narrow space, if the length of the straight tube section of the liquid tube cannot be guaranteed, a special rectification structure should be adopted to mix the gas and liquid. In the case that rectification is not possible, and the length of the straight tube section cannot be guaranteed, the capillary should be adopted to adjust refrigerant distribution.

Further, the liquid tube L-shaped structure (U-shaped structure) should be used for positioning, and the hole position on the distributor should be one-to-one corresponding to the flow path number of the heat exchanger. During the manufacturing process, the sequence of each flow path should be fixed.

(4) The flow path length of each path of the heat exchanger should be consistent with the wind field of the heat exchanger. Under the condition of relatively uniform airflow distribution, each flow path should be symmetrical and of equal length; for non-uniform airflow distribution, the flow path length should be shorter where the airflow speed is high, and the flow path length should be longer where the airflow speed is low.

(5) When the heat exchanger is used as the evaporator, in principle, the combination of the diameter and length of the capillary tube after the outlet of the distributor should be used to adjust the refrigerant flow rate, to ensure that the area of the heat exchanger can be fully utilized and avoid the local dry out area.



(a) Counterflow heat transfer of condenser with counterflow layout

(b) Counterflow heat transfer of evaporator with downflow layout

Figure 1-1 Flow path layout of evaporation and condenser

1.2.2 Determination of path number

For a heat exchanger, the more number of paths, the lower mass flow rate in each path, and the lower heat transfer coefficient, but the lower the pressure drop and the smaller the temperature slip, and the more difficult to distribute the refrigerant, and the higher the cost of the distributor; The less the number of paths, the higher the flow rate and the higher the heat transfer coefficient, but the higher the pressure drop and the larger the temperature slip, the lower the difficulty in distributing the refrigerant and the lower the cost of the distributor. Therefore, whether the heat exchanger is used as an evaporator or condenser, there is an optimal number of paths, which makes the performance of the heat exchanger better and the cost lower. The following describes the design principle of the heat exchanger as a condenser only, an evaporator only, and both evaporator and condenser.

1.2.2.1 Heat exchanger as condenser only

When the heat exchanger is used as a condenser, due to the high working pressure of the refrigerant, the temperature slip caused by the pressure drop of the refrigerant is small, and the refrigerant velocity of the refrigerant during the condensation process is continuously reduced, and the accelerating pressure drop is negative. Therefore, when the heat exchanger is used as a condenser only, the number of flow paths is less, which can improve the heat exchange of the heat exchanger and the energy efficiency of the system. Generally, as condenser only, the pressure drop of the heat exchanger is about 5% of inlet pressure, which has the best performance, and the pressure drop of common refrigerant is shown in table 1-2. In actual design, HXSim software can be used for the calculation to find out the number of the path in line with the pressure drop range listed in table 3-1, or the quick calculation method of the path in table 1-2 can be used for a quick calculation.

In particular, when the pressure drop of one inlet and one outlet is too large for a single path, and the pressure drop of two inlets and two outlets is too small, the 2-in-1 flow path can be used. In the gas area and high-quality area, the flow path can be divided into two paths, and in the low quality or liquid area, the flow path length can be adjusted according to the pressure drop.

Table 1-2 Pressure drop range of condenser for common refrigerants

NO.	**Refrigerant**	**Working pressure of condenser**	**Condenser pressure drop range**	**Quick calculation of path number** *1

1	R410A	2500-3200 kPa	60-150 kPa	For 7 mm tube, the heat exchange capacity of a single path is about 3500-4000 W when the flow length is less than 15 m, and 3000-3500 W when the flow length is more than 15 m.
2	R32	2800-3500 kPa	60-150 kPa	For 7 mm tube, the heat exchange capacity of a single path is about 4000-4500 W when the flow length is less than 15 m, and 3500-4500 W when the flow length is more than 15 m.
3	R134a	1000-1500 kPa[*1]	40-100 kPa	For 7 mm tube, the heat exchange capacity of a single path is about 2500-3500 W when the flow length is less than 15 m, and 2000-3000 W when the flow length is more than 15 m.
4	R22/R404A	2000-2500 kPa	60-100 kPa	For 7 mm tube, the heat exchange capacity of a single path is about 3000-3500 W when the flow length is less than 15 m, and 2500-3000 W when the flow length is more than 15 m.

Note 1: For the single path heat transfer of any diameter tube, we can use the single path heat transfer of 7 mm diameter copper tube. The following formula was used for calculation.

$$Q_d = Q_7 \times \left(\frac{d}{6.88}\right)^2 \qquad (1\text{-}1)$$

Where d is the inner diameter of the copper tube and Q_d is the optimal single path heat transfer of the condenser with the inner diameter of d (unit: mm).

1.2.2.2 Heat exchanger as evaporator only

When the heat exchanger is used as an evaporator, the number of paths of the heat exchanger has a significant impact on the capacity and energy efficiency of the system. When the number of paths of the evaporator is small, it is conducive to the improvement of the heat transfer coefficient in the tube, but it will significantly increase the pressure drop and increase the temperature slip. Therefore, the temperature at the inlet of the heat exchanger will be significantly increased, making the heat transfer temperature difference smaller, as shown in Figure 1-2.

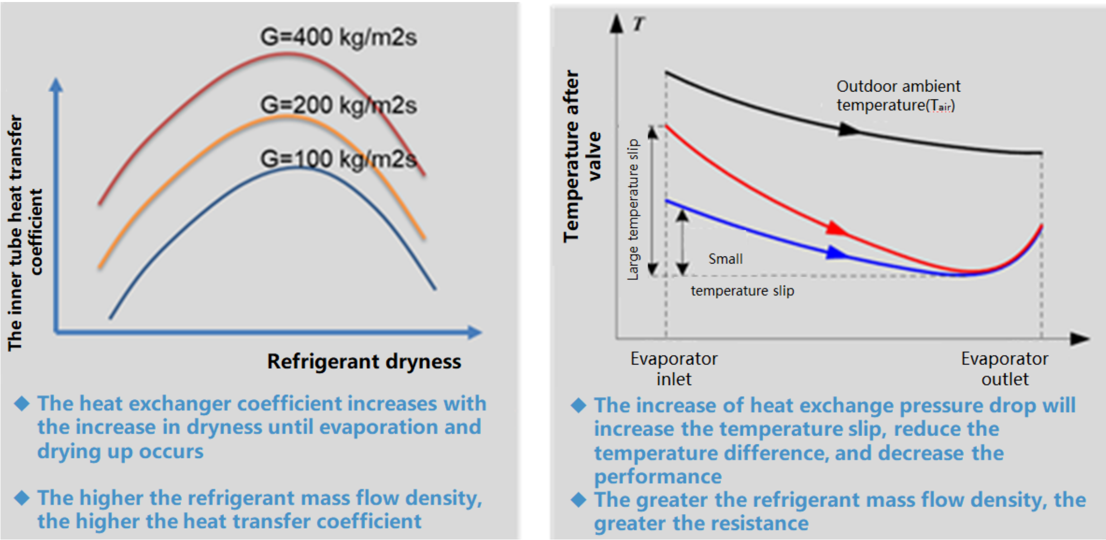


Figure 1-2 Heat transfer coefficient and heat transfer temperature difference on evaporation side

Because the heat transfer is directly proportional to the product of temperature difference and heat transfer coefficient, there is an optimal number of paths to maximize the heat transfer. Compared with the condenser, the working pressure of the evaporator is generally much lower, so the path of the evaporator needs more than that of the condenser. Especially, when the evaporator is used for low-temperature heating, or the system uses low-pressure refrigerant (such as R134a), the heat exchanger needs more paths.

Generally, the pressure drop of an evaporator is about 3% of the heat exchanger's inlet pressure. The pressure drop range of the evaporator and the quick calculation of the path number of the commonly used refrigerant is shown in Table 1-3. It should be pointed out that the pressure drop discussed here is the pressure drop at the inlet and outlet of the heat exchanger, excluding the pressure drop of the distributor and capillary tube of the heat exchanger. In actual design, HXSim can be used for the calculation to find out the number of the path in line with the pressure drop range listed in Table 1-3, or the quick calculation method of the path in Table 1-3 can be used for quick calculation.

In particular, when the resistance of one inlet and one outlet is too large for a single path, and the resistance of two inlets and two outlets is too small, the 2-in-1 flow path can be used. One path can be used in the low dryness area of the evaporator inlet, and it can be divided into two paths after reaching the medium and high dryness. The bifurcation position is generally set at 1 / 2 or later of the flow path, which can be adjusted according to the actual situation.

It should be pointed out that considering the actual cost and the difficulty of distribution debugging, the path number of the evaporator can not be set too much just for the sake of low-pressure drop. With the increase of the heat exchanger's path number, the number of holes in the distributor is correspondingly increased, and the cost of the heat exchanger will also be increased.

Table 1-3 pressure drop range of evaporator for common refrigerants

NO.	**Refrigerant**	**Working pressure of evaporator**	**Evaporator pressure drop range**	**Quick calculation of shunt number** [*1]
1	R410A	936-1218kPa	20-40 kPa	For 7 mm tube: ● flow length<10 m, the heat exchange capacity of a single path is about 1400-1700 W. ● 10 m < flow length < 15 m, the heat exchange capacity of a single path is about 1200-1500 W. ● 15 m < flow length < 20 m, the heat exchange capacity of a single path is about 1000-1300 W. ● flow length > 20 m, the heat

				exchange capacity of a single path is about 800-1100 W. If the evaporator is used in the working condition (such as heat pump) where the evaporation temperature Te is lower than 0 ℃, the correction factor SF should be multiplied based on the above heat exchange to correct. The correction factor SF is as follows: (i) If -10<Te<0°C, SF = 0.85; (II) if -20 °C<Te<-10 °C, SF = 0.7; (III) if Te<-20 °C, SF = 0.5; Note: if the evaporation temperature Te of the evaporator covers two or more temperature zones at the same time, the correction factor SF shall be determined according to the main assessment conditions. If the main assessment condition is rated heating condition with 7/6°C environmental temperature (Te=-1 °C), the evaporation temperature belongs to category (I), the SF factor shall be taken as 0.85.
2	R32	951-1244 kPa	20-65kPa	For 7 mm tube: flow length < 10 m, the heat exchange capacity of a single path is about 2000 – 2500 W; 10 m < flow length < 15 m, the heat exchange capacity of a single path is

				about 1500-2000 W; 15 m < flow length < 20 m, the heat exchange capacity of a single path is about 1300-1600 W; flow length > 20 m, the heat exchange capacity of a single path is about 1000-1300 W. If the evaporator is used in the working condition (such as heat pump) where the evaporation temperature Te is lower than 0 ℃, the correction factor SF should be multiplied based on the above heat exchange to correct. The correction factor SF is the same as that of R410A.
3	R134a	349-443kPa	20-40kPa	For 7 mm tube: flow length < 10 m, the heat exchange capacity of a single path is about 800-1000 W; 10 m < flow length < 15 m, the heat exchange capacity of a single path is about 700-900 W; 15 m < flow length < 20 m, the heat exchange capacity of a single path is about 600-800 W; flow length > 20 m, the heat exchange capacity of a single path is about 500-700 W. If the evaporator is used in the

				working condition (such as heat pump) where the evaporation temperature Te is lower than 0 ℃, the correction factor SF should be multiplied based on the above heat exchange to correct. The correction factor SF is the same as that of R410A.

Note 1: For the single path heat transfer of any diameter tube, we can use the single path heat transfer of 7 mm diameter copper tube. The following formula was used for calculation.

$$Q_d = Q_7 \times \left(\frac{d}{6.88}\right)^2 \quad (1\text{-}2)$$

Where d is the inner diameter of the copper tube and Q_d is the optimal single path heat transfer of the condenser with the inner diameter of d (unit: mm).

1.2.2.3 The heat exchanger is used as evaporator and condenser at the same time

The heat exchanger is used as the evaporator and condenser at the same time. The general principle should be based on the path design principle of the heat exchanger as the evaporator (see 1.2.2.2). Specifically, due to the different application environments of the indoor unit and outdoor unit, there are some differences, which are explained below.

(1) Heat exchanger as indoor unit

As an indoor unit, if the heat exchanger has the demand for cooling and heating at the same time, it should be designed according to the cooling demand, that is, according to the design requirements of the evaporator. See section 1.2.2.2 for details.

For the system with high energy efficiency, the number of flow paths can be reduced by 10% - 20% based on the number of flow paths calculated in section 1.2.2.2. For air conditioners, if the energy efficiency evaluation indexes adopted have partial load requirements, such as APF, IEER, SCOP, etc., the reduction of the internal unit flow path number is conducive to improving the energy efficiency of the system under half load condition, and energy efficiency of the system under the rate and half load heating condition, and low-temperature heating condition. It can significantly improve the comprehensive energy efficiency of the system (such as APF, IEER).

For the case of non-uniform wind speed, the flow length of each path of the heat exchanger should be inversely proportional to the wind speed.

(2) Heat exchanger as outdoor unit

As an outdoor unit, if the heat exchanger has the demand for cooling and heating at the same time, it should be designed according to the heating demand, that is, according to the design requirements of the evaporator. See section 1.2.2.2 for details.

Due to objective reasons such as heat exchange capacity, tube length, and cost of the distributor, the number of paths calculated directly according to section 1.2.2.2 may not be able to be operated in practice. The following is an explanation of these special cases (Note: the explanation can not cover all the actual situations. If it is not listed here, it can be designed according to similar principles)

(i) Too many paths

With the increase of the outdoor unit's capacity, the path number of heat exchangers increases sharply. At this time, increasing the number of paths can reduce the pressure drop of the heat exchanger and improve the heat transfer performance in theory, but in practice, it faces the following difficulties: (I) too many paths leads to difficulty in refrigerant distribution debugging, especially for the model with serious uneven wind speed; (II) the significant increase of the number of holes in the distributor leads to the cost increase.

The solution is to divide one flow path into two, which can reduce the pressure drop of the heat exchanger and reduce the holes of the distributor by half. After the fluid is divided into two paths, the number of upwind tubes and leeward tubes of the two paths should be as consistent as possible (except for those not allowed in the structure), to ensure the uniformity of the outlet temperature of the two paths.

(II) the heat exchange capacity of the single heat exchanger is less than 5000 W, and there are more U-tubes

In this case, the number of flow paths is usually 2-4 by pressure drop calculation. If the number of U-tubes is large (the number of U-tubes in each path is more than 4U), and the flow path is simply divided into several paths, the performance of heat exchanger and system is usually not optimal. Figure 1-3 shows the heat transfer characteristics of outdoor heat exchanger at different tube side positions. The physical properties of refrigerant in different positions of outdoor heat exchanger are different, and the number of flow paths required is also different. At the inlet stage of the flow path, the refrigerant dryness is small, the mass of the liquid phase refrigerant is much greater than that of the gas phase refrigerant so that the flow rate of the refrigerant in the tube is very low. At this time, the number of flow paths of the outdoor heat exchanger should be reduced to increase the flow rate of the refrigerant and the heat transfer coefficient; When the refrigerant flows to the middle of the tube side of the outdoor heat exchanger, the dryness of the refrigerant increases gradually, and more liquid-phase refrigerant is converted into gas-phase refrigerant with very low density so that the flow rate of the refrigerant in the tube rises gradually, and the number of flow paths of the outdoor heat exchanger is a reasonable value; When the refrigerant is in the outlet stage of the outdoor heat exchanger, the dryness of the refrigerant is large, and the mass of the gas-phase refrigerant is far greater than that of the liquid phase refrigerant, which makes the flow rate of the refrigerant in the tube larger. At this time, the flow path number of the outdoor heat exchanger should be increased to reduce the flow rate of the refrigerant and avoid excessive pressure drop of the refrigerant.

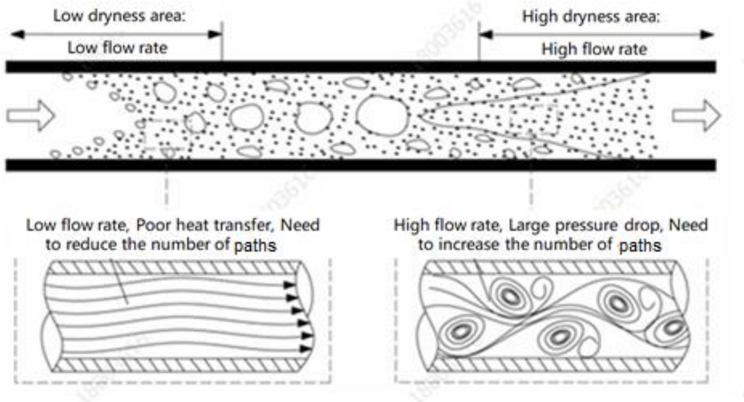


Figure 1-3 heat transfer characteristics of outdoor heat exchanger at different tube side positions

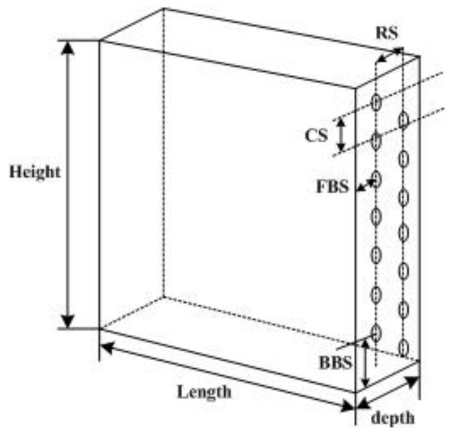
2 HX Sample for Air Conditioner ODU

2.1 Structure and performance of original HX

(1) Structure of the original HX

The heat exchanger is a single cooling air conditioner ODU. The original heat exchanger is a 7 mm heat exchanger, which has two rows, 12 U tubes. The detailed geometry is listed in table 2-1. The flow circuitry of the original heat exchanger is listed in Fig. 2-1.

Table 2-1 structure parameters of newly designed condenser

No.	Parameters	Value	Note
1	Tube diameter, mm	7	
2	Tube length(Length), mm	770	
3	HX depth (Depth), mm	36.4	
4	HX height(Height), mm	504	
5	Row	2	
6	Column	24	
7	Row space(RS), mm	18.2	
8	Column space(CS), mm	21	
9	Fin pitch, mm	1.4	
10	Fin type	Wavy	

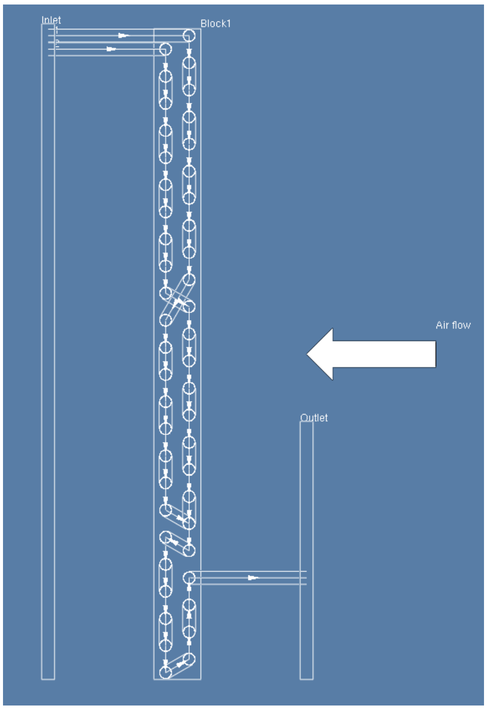


Figure 2-1 Flow circuitry of prototype heat exchanger

(2) Performance of the original HX

The EER of air conditioner is 3.6, and the rated cooling capacity is 3500 W. The refrigerant is R32. The discharge pressure is 2861 kPa (condensing temperature is 46 ℃), and discharge temperature is 68 ℃. The Airflow rate of heat exchanger is 1800 m3/h. The detailed working condition is shown in Table 2-2.

To match the requirements of the air conditioner, the heat exchange capacity should be over 4470 W, with subcooling over 5℃ under the air inlet temperature 35/24℃. The mass flow rate of the heat exchanger is equal to 59 kg/h, which can be calculated as the following equations.

$$m = \frac{Q}{h_{in} - h_{out}} \tag{2-1}$$

Where m is the mass flow rate, kg/s; Q is desired heat exchange capacity, W; h_{in} is the specific enthalpy of refrigerant flowing into the heat exchanger, kJ/kg; h_{out} is the specific enthalpy of refrigerant flowing out from the heat exchanger, kJ/kg;

Table 2-2 Working conditions of 5 mm ODU heat exchanger of air conditioner

Items		Value
Airside	Dry bulb temperature, ℃	35
	Wet bulb temperature, ℃	24

	Airflow rate, m3/h	1800
Refrigerant side	Refrigerant type	R32
	Discharge temperature, ℃	68
	Discharge pressure, kPa (condensing temp. ℃)	2851 (45.5)
	Subcooling, ℃	5
Capacity requirement, W		Over 4470

Table 2-3 list the simulation results of the prototype heat exchanger. The heat exchange capacity is 4500 W, which is over the requirements of 4470, and the subcooling is 5.1 ºC.

Table 2-3 Summary of simulation result of original heat exchanger

COIL SIDE			
Fin Type	Corrugated	Utilized Tubes	48
Fin Material	Aluminum	Non Utilized Tubes	0
Fin Spacing [mm]	1.40	Circuits	3
Fin Thinkness [mm]	0.105	Tubes Per Circuit	16.00
Tube Type	Grooved	Coil Length [mm]	770.00
Tube Material	Copper	Coil Depth [mm]	36.40
Tube Dimension [mm]	7.00*0.23*0.10	Coil Height [mm]	504.00
Holes	24	Outer Area [m2]	19.380
Rows	2	Inner Area [m2]	0.759
Tube Vertical Space [mm]	21.00	Coil Face Area [m2]	0.39
Tube Horizontal Space [mm]	18.20	Inner Volume [L]	1.241
Header In [mm]	9.5	Header Out [mm]	9.5
AIR SIDE		REFRIGERANT SIDE	
Air Inlet DB. Temp. [°C]	35.0	Refrigerant	R32
Relative Humidity %	40.3	Discharge Superheat [°C]	22.50
Air Outlet DB. Temp. [°C]	42.7	Condenser Temp.[°C]	45.50
Relative Humidity %	26.6	Subcooling [°C]	5.16
Air Flow [m3/h]	1831.2	Mass Flow [kg/h]	59.0
Air Mass Flow [kg/h]	2352.0	Pressure Drop [kPa]	28.238
Frontal Velocity [m/s]	1.3	Outlet Pressure [kPa]	2799.510
Air Pressure Drop [Pa]	22.0	Ref. Charge [kg]	0.45
Atmospheric Pressure [kPa]	101.3	Ref. Side H.T.C. [W/m2*K]	3119.427
Air Side H.T.C. [W/m2*K]	71.916		
CAPACITY			
Total Capacity [kW]	4.500		

The modeling process is listed as below:

(1) Open HXSim, and press the new case button, as Fig. 2-2.

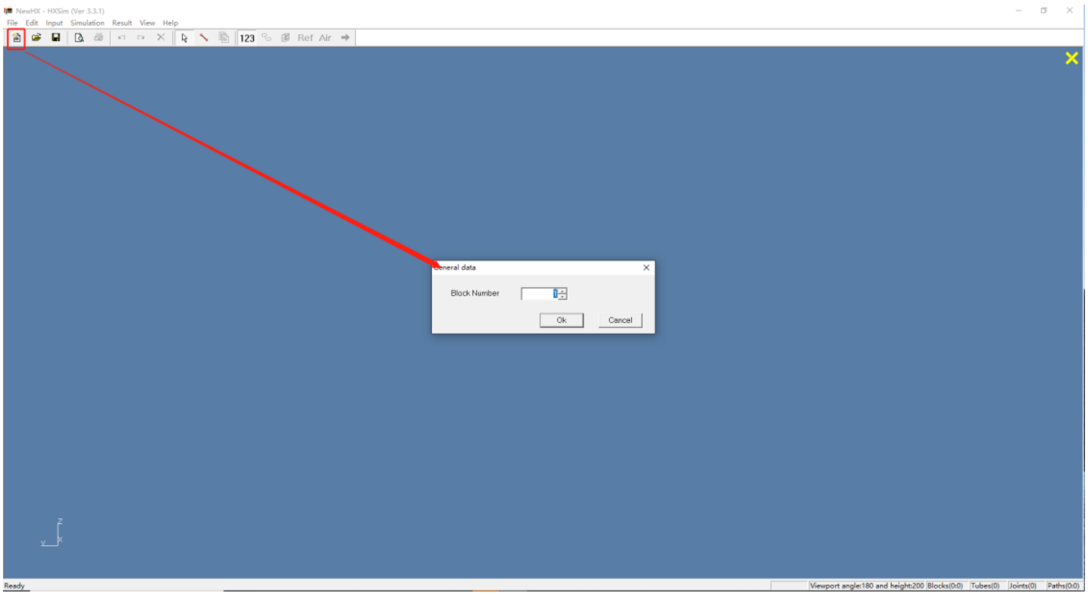


Figure 2-2 New a case

(2) Press button “block” to open block dialog, and input the parameters listed in Table 2-1 like the following Fig. 2-3. Then, press the button “tube type”, select all the tubes in the table (Caution, the grids will be highlighted as blue when they are selected), and then press the button “specify tube type” to choose the specified 7 mm grooved tube, as shown in Fig. 2-4.

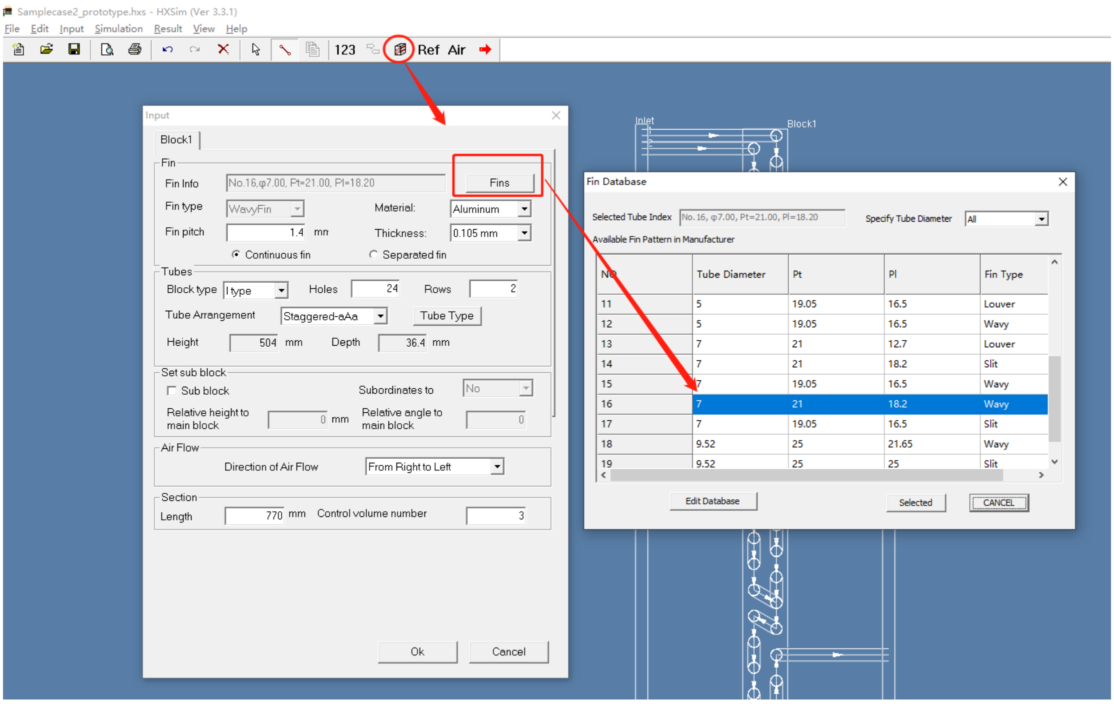


Figure 2-3 Setting the parameter of HX

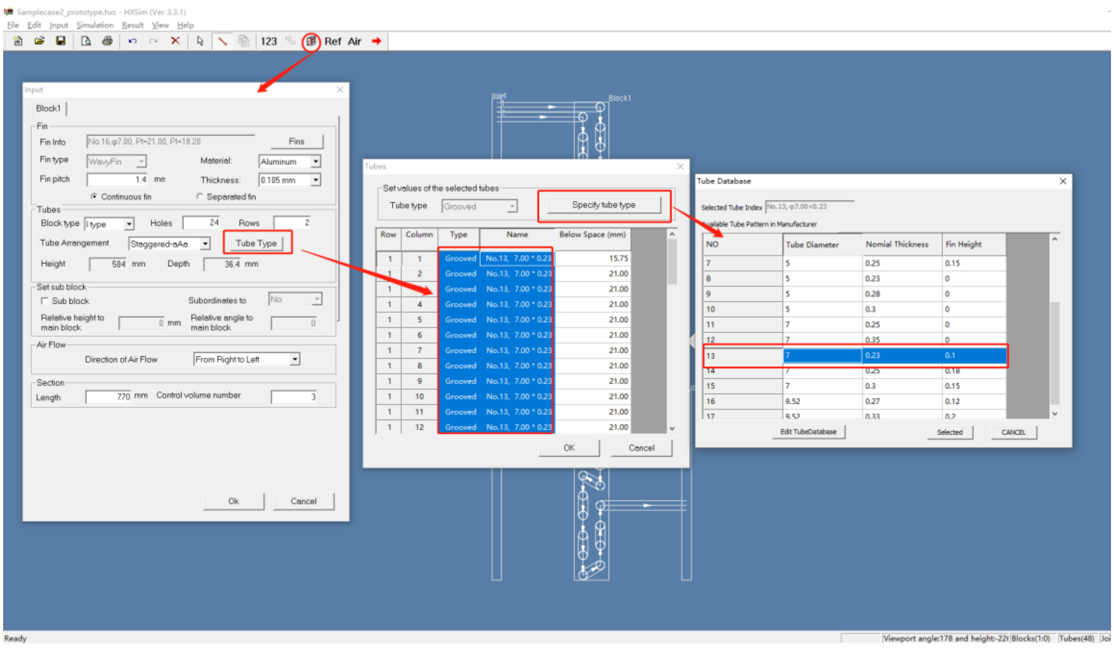


Figure 2-4 Setting the tube parameters

(3) Connect the flow circuits like the follows as shown in Fig. 2-1.

(4) Setting the refrigerant side input and airside input

Input the refrigerant side working conditions listed in Table 2-2 into the refrigerant setting dialog, as shown in Fig. 2-5. Caution, the refrigerant mass flow rate is unknown at first, but its value can be obtained by either iterative calculation or theoretical calculation using the rated capacity dividing the specific enthalpy difference of inlet and outlet of heat exchangers as listed in Eq. (2-1). Due to the discharge temperature and subcooling are known, the specific enthalpy of inlet and outlet can be calculated by the pressure, discharge temperatures, and pressure, subcooling temperatures, respectively.

Then, input the airside working conditions listed in Table 2-2, including air volume flow rate, dry bulb temperature, and wet bulb temperature, as shown in Fig. 2-6.

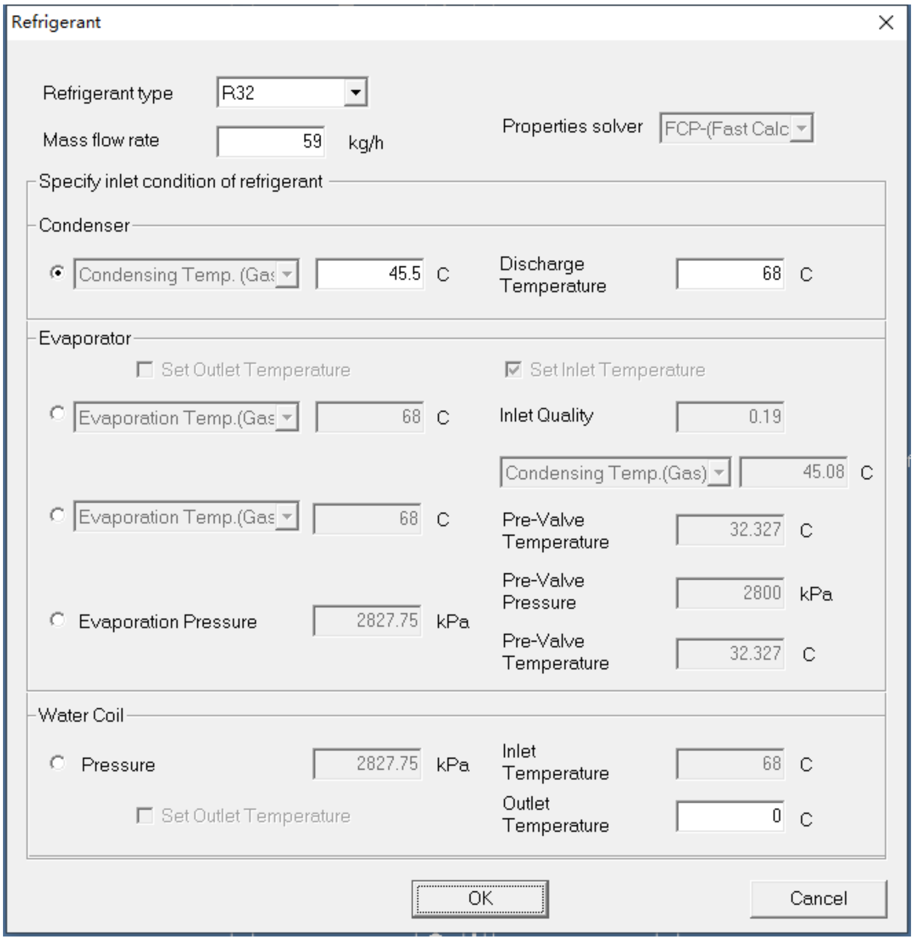


Figure 2-5 Setting the working condition of refrigerant side

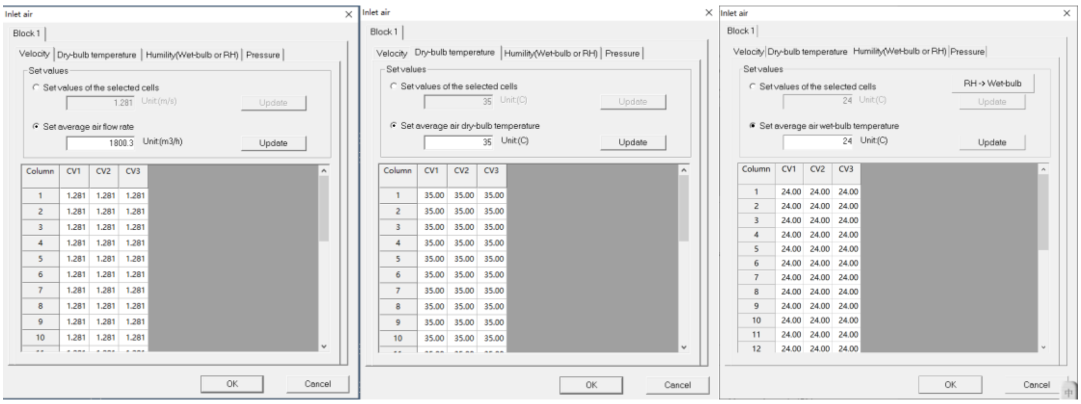


Figure 2-6 Setting the working conditions of airside

(5) Click the simulation/run button to run the simulation, and the user can obtain the simulation results listed in table 2-3.

2.2 Design and optimization of small diameter copper tube HX

Due to the restriction of ODU box, the shapes of new 5 mm heat exchanger are the same as those of original heat exchanger, including the length, height, and et al.

(1) Fin type selection

According to the 1.1.1 (Table 1-1, No.1), the ODU heat exchanger is used for single cooling, the slit/louver is selected. The column space and row space of fin usually depend on the fin die which manufacturer build. In this case, the manufacturer already has fin die (19.5*11.6 slit fin), and thus this fin type is used in the design.

(2) Fin pitch selection

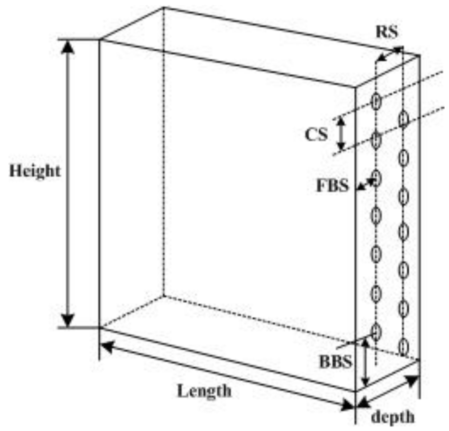
According to the 1.1.2, the fin pitch is selected as 1.3 mm.

(3) U tubes and row number selection

Due to the column space 19.5, and the total height around 504 (original heat exchanger height), the total U tubes number can be calculated out, which is equal to 13. The row number is 2, the same as original heat exchanger.

Based on the above selection and calculation, the detailed structure is shown in Table 2-4.

Table 2-4 structure parameters of new designed condenser

No.	Parameters	Value	Note
1	Tube diameter, mm	5	
2	Tube length(Length), mm	770	
3	HX depth (Depth), mm	23.2	
4	HX height(Height), mm	507	
5	Row	2	
6	Column	26	
7	Row space(RS), mm	11.6	
8	Column space(CS), mm	19.5	
9	Fin pitch, mm	1.3	
10	Fin type	Slit	

(4) Flow circuits design

According to 1.2.2.1(Table 1-2) and equation (1-1), the heat exchange capacity of a single path of heat exchanger can be figured out as eq. (2-2), which is equal to

2433 W. The total heat exchange requirements (listed in Table 2-2) is over 4470. As a result, the best refrigerant flow circuits are between 1 and 2, but approach to 2.

$$Q_d = Q_7 \times \left(\frac{d}{6.88}\right)^2 = 5000 \times \left(\frac{5.2-0.2*2}{6.88}\right)^2 = 2433 \text{ W} \tag{2-2}$$

According to 1.2.2.1:

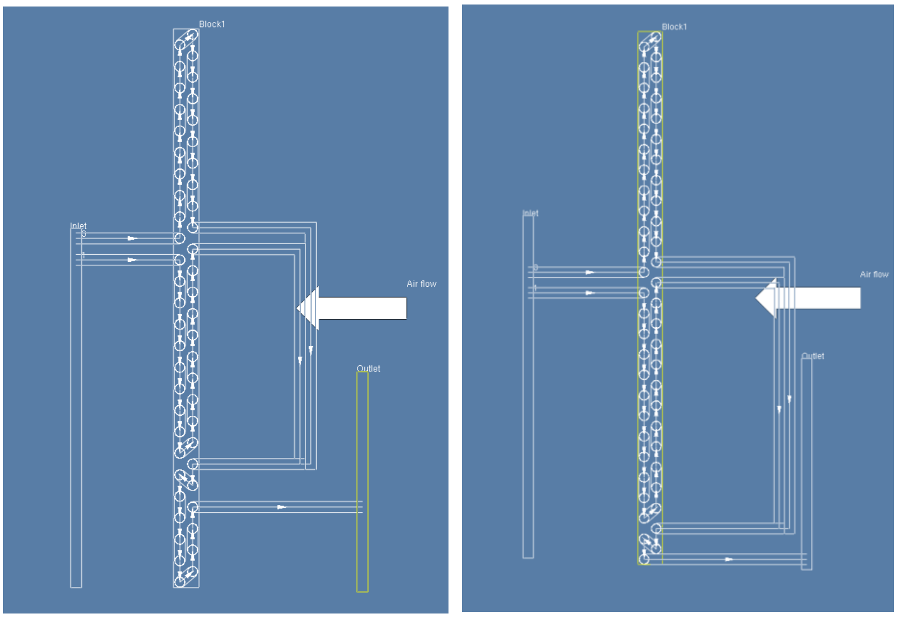
In particular, when the flow resistance of 1 path is too large, while the resistance of 2 paths is too small, a two-in-one flow path can be used. The gas and high dryness zones can be divided into two paths, while the low dryness and liquid zones can be synthesized into one path. The length can be adjusted according to the pressure drop.

As a result, in this case, 2 paths converging into one path is preferred. Because we have 13 U tubes in each row, and the flow circuits should be designed symmetry, there are two options are available:

i) 5 U tubes in each path in each row, and then converge into 1 path (3 U tubes in each row)

ii) 6 U tubes in each path in each row, and then converges into 1 path (1U tubes in each row)

These two options are listed in Figure. 2-7.



(a) (b)

Figure 2-7 Flow circuit of 5 mm ODU heat exchanger

2.3 Modeling process of small diameter copper tube HX

(1) Open HXSim, and press new case button, as Fig. 2-8.

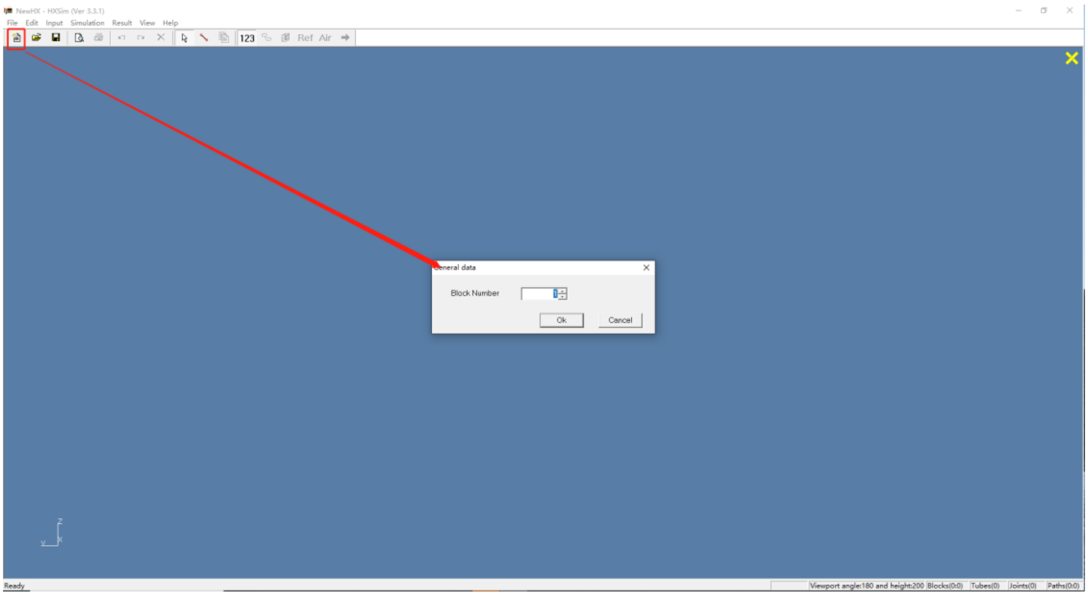


Figure 2-8 New a case

(2) Press button “block” to open block dialog, and input the parameters listed in Table 2-4 like the following Fig. 2-9. Then, press the button “tube type”, select all the tubes in the table (Caution, the grids will be highlighted as blue when they are selected), and then press the button “specify tube type” to select the specified 5 mm grooved tube, as shown in Fig. 2-10.

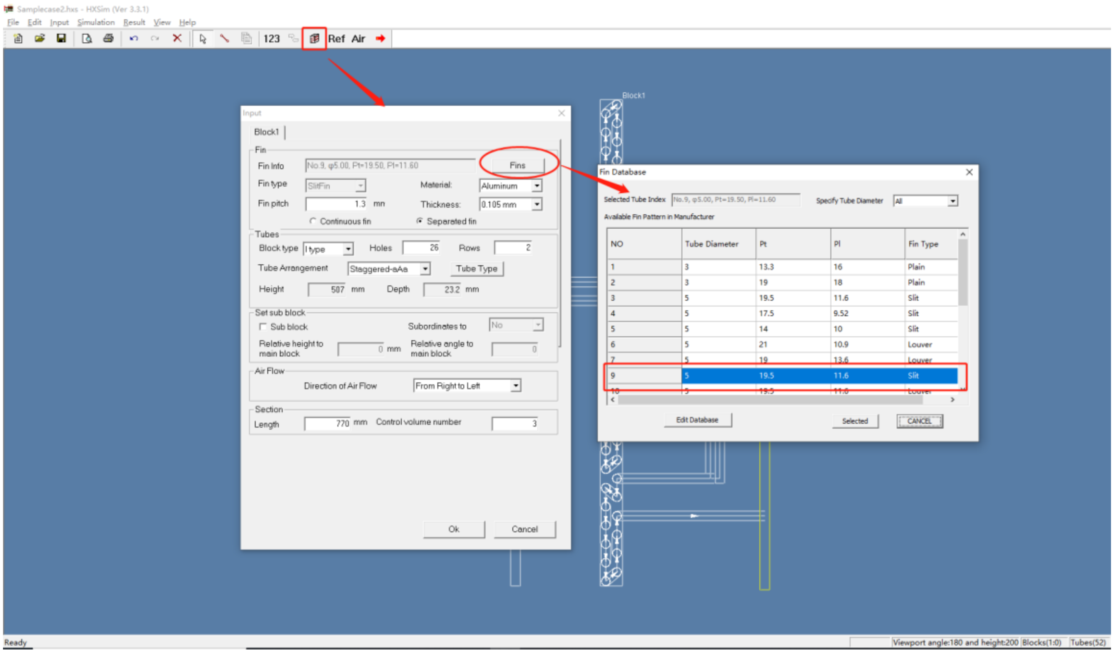


Figure 2-9 Setting the parameter of HX

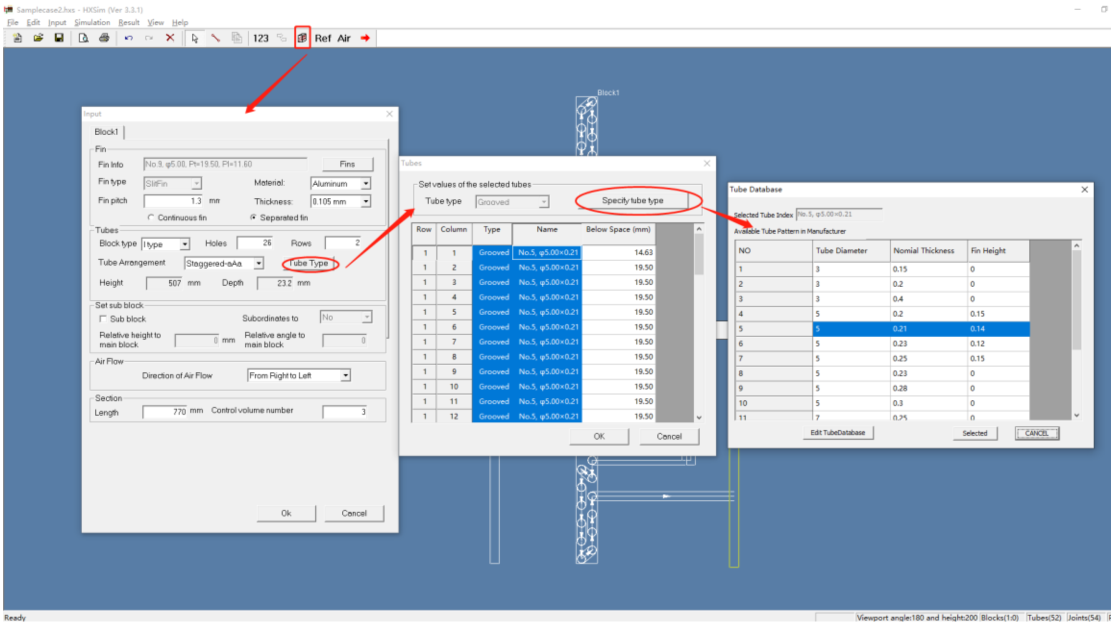


Figure 2-10 Setting the tube parameters

(3) Connect the flow circuits like the follows, as shown in Fig. 2-11.

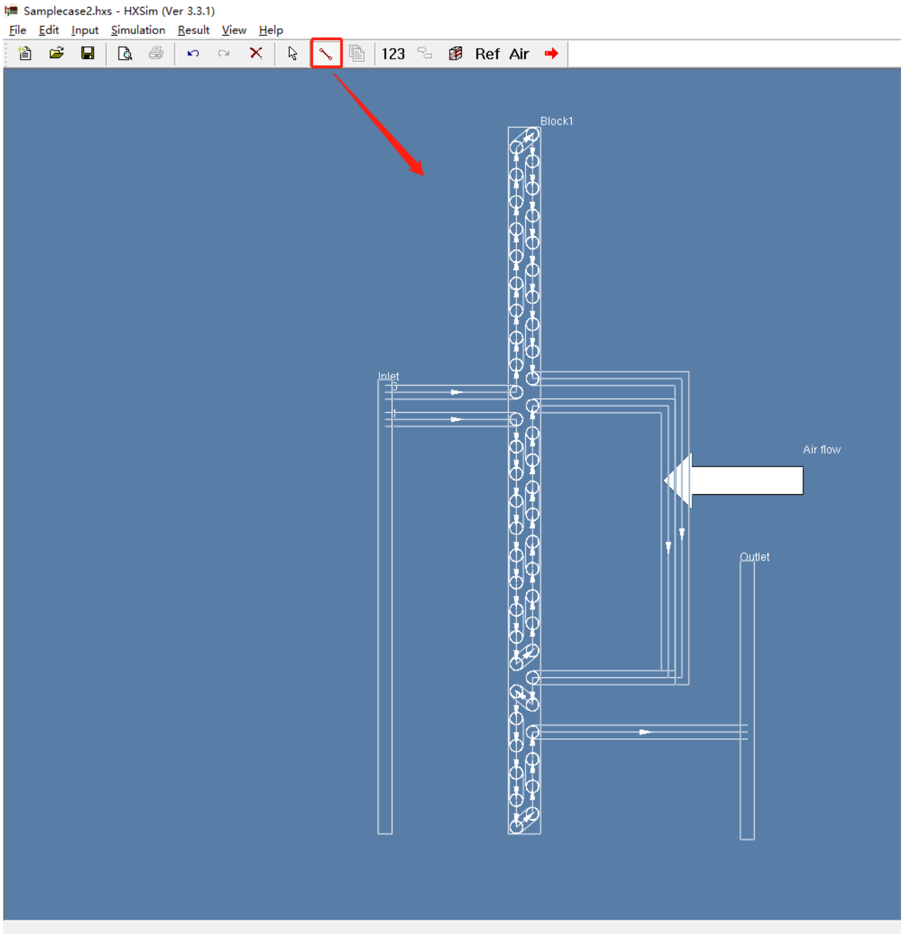
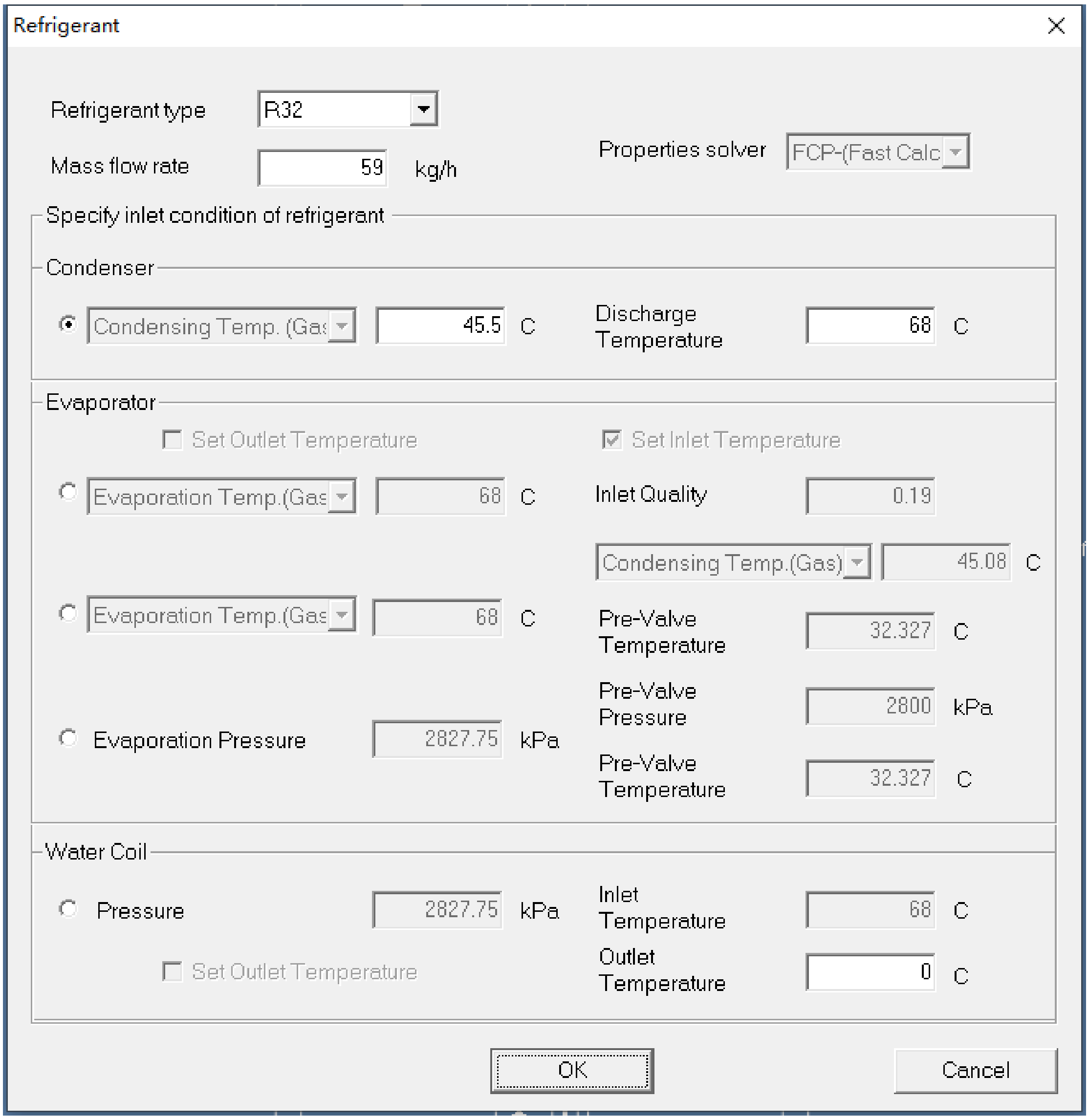


Figure 2-11 Connecting flow circuits

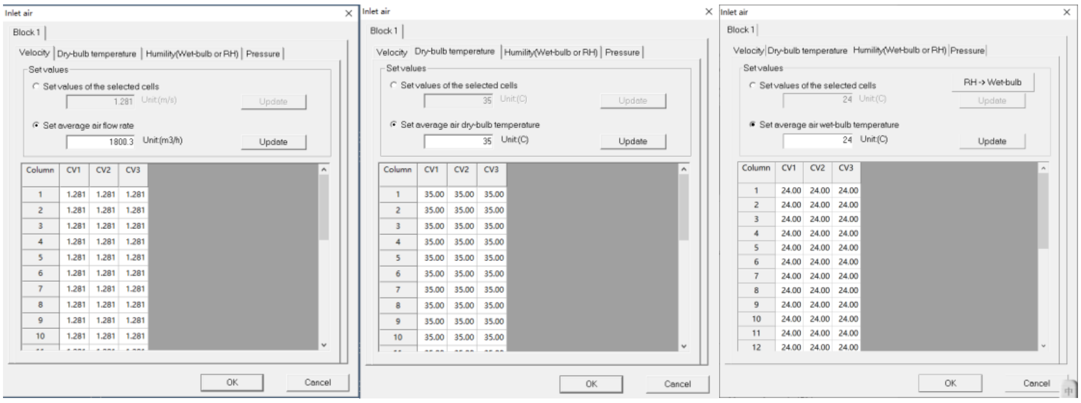
(4) Setting the refrigerant side input and airside input

Input the refrigerant side working conditions listed in Table 2-2 into the refrigerant setting dialog, as shown in Fig. 2-12 (a). The mass flow rate is set as the original heat exchanger.

Then, input the airside working conditions listed in Table 2-2, including air volume flow rate, dry bulb temperature, and wet bulb temperature, as shown in Fig. 2-12 (b).



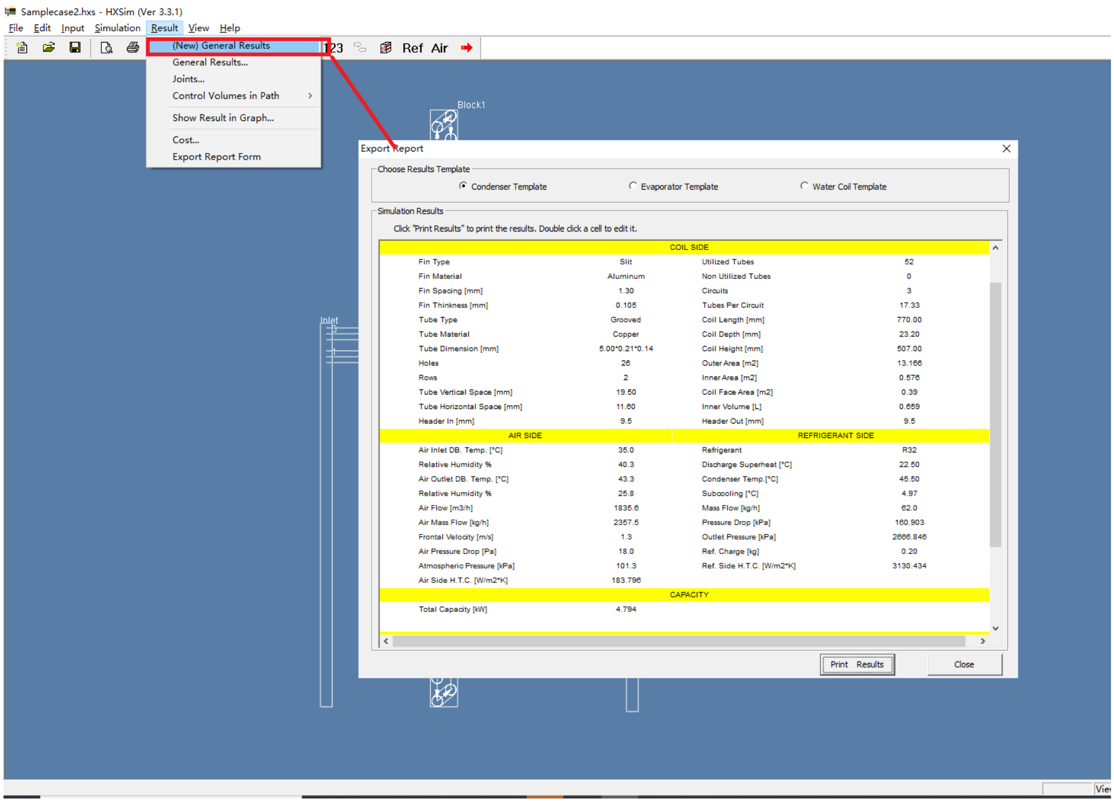
(a) Refrigerant side



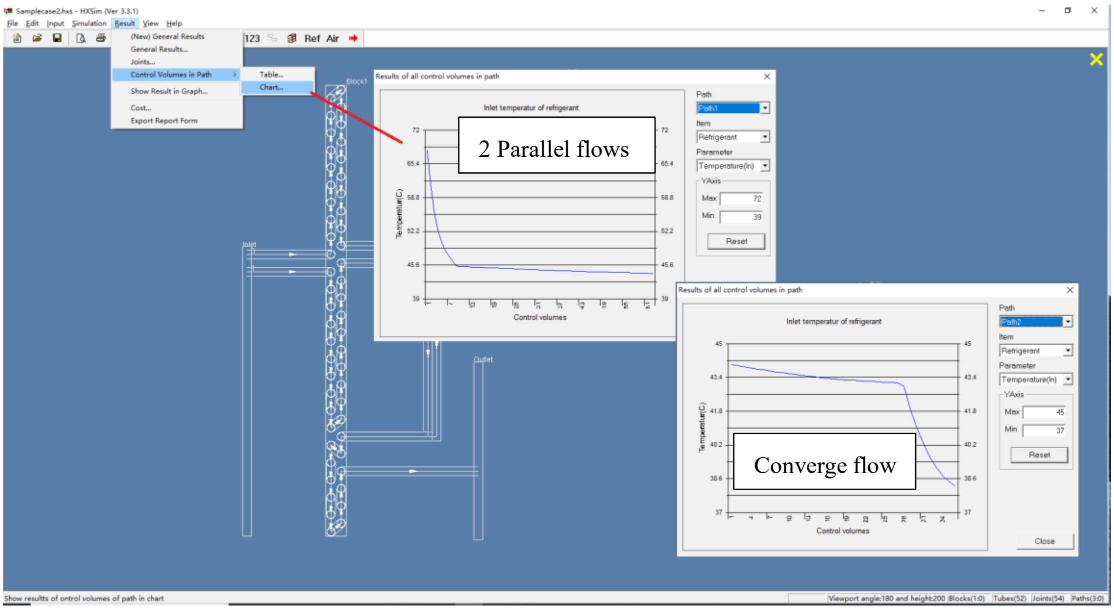
(b) Airside

Figure 2-12 Setting the working conditions

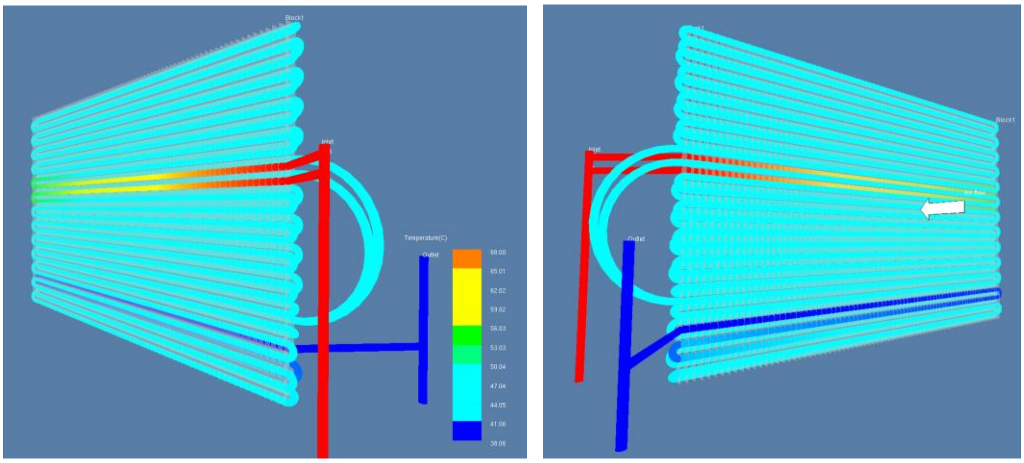
(5) Click the simulation/run button to run the simulation. After simulation, the simulation results can be obtained by clicking the menu result as shown in Fig. 2-13.



(a) Dialog to show summary of simulation result



(b) Dialog to show temperature trend of flow path



(c) dialog to show temperature gradation

Figure 2-13 Dialogs to show simulation results

(6) repeat the process (1)-(5), the simulation results of flow circuits Fig.2-7(b) can be obtained.

2.4 Simulation result summaries and analysis

The simulation results of flow circuits in Fig. 2-7(a) and Fig. 2-7(b) are listed in Table 2-5. It shows that the flow circuits in Fig. 2-7(a) have better performance. As a result, the flow circuits 2-7(a)

From these results, we can conclude that: 5 mm heat exchanger works well as ODU heat exchanger of single cooling heat exchanger. The pressure drop is over 120 kPa due to 5 mm diameter, however, the slide of condensing temperature is within 3 ℃, and further, the slides of condensing temperature in the 2 parallel flow paths are within 1.5 ℃ (see Fig. 2-13). The pressure drop in cooling condition has rather smaller impact on heat exchanger performance, compared to evaporation conditions.

5 mm tube heat exchanger has lower cost and relatively good performance. As a result, 5 mm tube heat exchanger is widely used in ODU of single cooling air conditioners.

Table 2-5 Summary of simulation result

Items	**Flow circuitry in Fig. 2-7(a)**	**Flow circuitry in Fig. 2-7(b)**
Capacity, W	4612	4564
Pressure drop, kPa	128.0	103.6
Subcooling, oC	6.917	5.89

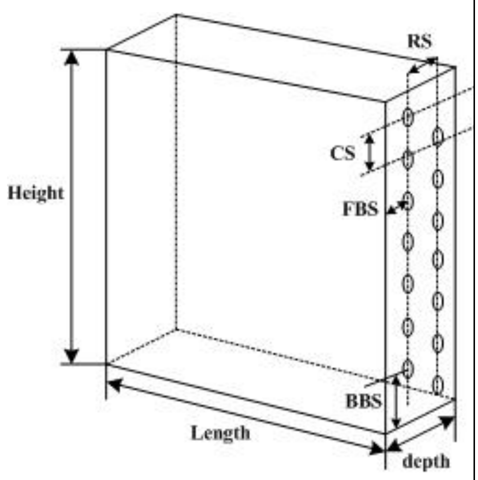
3 HX Sample for Cooling Cabinet

3.1 Structure and performance of original HX

(1) Structure of the original HX

The heat exchanger is a condenser of cooling cabinet. The original heat exchanger is a 9.52 mm heat exchanger, which has four rows, and each row has 8 tubes. The detailed geometry is listed in table 3-1. The flow circuitry of the original heat exchanger is listed in Fig. 3-1.

Table 3-1 Structure of condenser of prototype

No.	Parameters	Value	Note
1	Tube diameter, mm	9.52	
2	Tube length(Length), mm	220	
3	HX depth (Depth), mm	65	
4	HX height(Height), mm	200	
5	Row	3	
6	Column	8	
7	Row space(RS), mm	21.65	
8	Column space(CS), mm	25	
9	Fin pitch, mm	1.8	

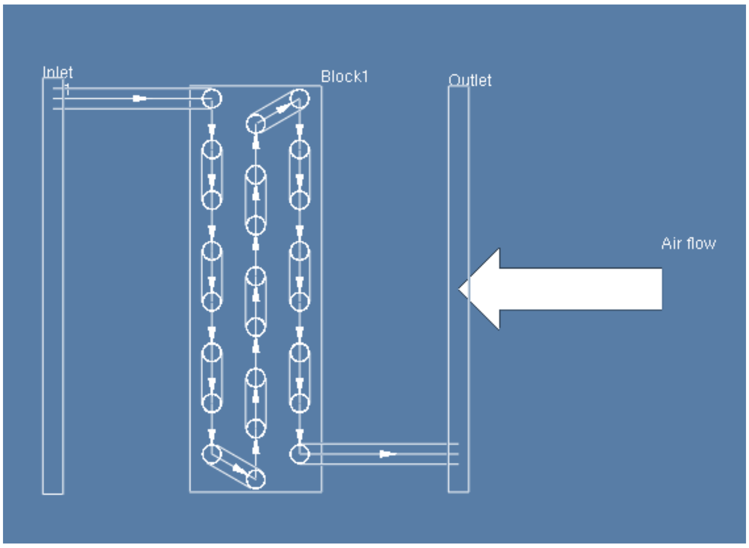


Figure 3-1 Flow circuit of original condenser of cooling cabinet

(2) Performance of the original HX

The refrigerant is R404A. The discharge pressure is 2193 kPa, and discharge temperature is 62 ℃. The Air flow rate of heat exchanger is 250 m3/h. The detailed working condition is shown in Table 3-2.

To match the requirements of the cooling cabinet, The required heat exchange capacity is over 690 W, with subcooling over 5 ℃. The mass flow rate of the heat exchanger is equal to 18 kg/h, which can be calculated as the following equations.

$$m = \frac{Q}{h_{in} - h_{out}} \tag{3-1}$$

Where m is the mass flow rate, kg/s; Q is desired heat exchange capacity, W; h_{in} is the specific enthalpy of refrigerant flowing into the heat exchanger, kJ/kg; h_{out} is the specific enthalpy of refrigerant flowing out from the heat exchanger, kJ/kg;

Table 3-2 Working conditions of 5 mm condenser of cooling cabinet

<table>
<tr><th colspan="2">Items</th><th>Value</th></tr>
<tr><td rowspan="3">Airside</td><td>Dry bulb temperature, ℃</td><td>35</td></tr>
<tr><td>Wet bulb temperature, ℃</td><td>24</td></tr>
<tr><td>Airflow rate, m3/h</td><td>250</td></tr>
<tr><td rowspan="4">Refrigerant side</td><td>Refrigerant type</td><td>R404A</td></tr>
<tr><td>Discharge temperature, ℃</td><td>62</td></tr>
<tr><td>Discharge pressure, kPa
(condensing temp. ℃)</td><td>2193
(48)</td></tr>
<tr><td>Subcooling, ℃</td><td>5</td></tr>
<tr><td colspan="2">Capacity requirement, W</td><td>Over 690</td></tr>
</table>

Table 2-3 list the simulation results of the prototype heat exchanger. The heat exchange capacity is 696 W, which is over the requirements of 690, and the subcooling is 5.27 ºC.

Table 3-3 Summary of simulation result of original heat exchanger

COIL SIDE			
Fin Type	Corrugated	Utilized Tubes	24
Fin Material	Aluminum	Non Utilized Tubes	0
Fin Spacing [mm]	1.80	Circuits	1
Fin Thinkness [mm]	0.105	Tubes Per Circuit	24.00
Tube Type	Grooved	Coil Length [mm]	220.00
Tube Material	Copper	Coil Depth [mm]	64.95
Tube Dimension [mm]	9.52*0.33*0.20	Coil Height [mm]	200.00
Holes	8	Outer Area [m2]	3.196
Rows	3	Inner Area [m2]	0.147
Tube Vertical Space [mm]	25.00	Coil Face Area [m2]	0.04
Tube Horizontal Space [mm]	21.65	Inner Volume [L]	0.325
Header In [mm]	9.5	Header Out [mm]	9.5
AIR SIDE		REFRIGERANT SIDE	
Air Inlet DB. Temp. [°C]	35.0	Refrigerant	R404A
Relative Humidity %	40.3	Discharge Superheat [°C]	14.00
Air Outlet DB. Temp. [°C]	43.7	Condenser Temp.[°C]	48.00
Relative Humidity %	25.2	Subcooling [°C]	5.27
Air Flow [m3/h]	255.9	Mass Flow [kg/h]	18.0
Air Mass Flow [kg/h]	328.6	Pressure Drop [kPa]	0.335
Frontal Velocity [m/s]	1.6	Outlet Pressure [kPa]	2193.028
Air Pressure Drop [Pa]	56.6	Ref. Charge [kg]	0.16
Atmospheric Pressure [kPa]	101.3	Ref. Side H.T.C. [W/m2*K]	1137.794
Air Side H.T.C. [W/m2*K]	86.675		
CAPACITY			
Total Capacity [kW]	0.696		

The modeling process is listed as below:

(1) Open HXSim, and press the new case button, as Fig. 3-2.

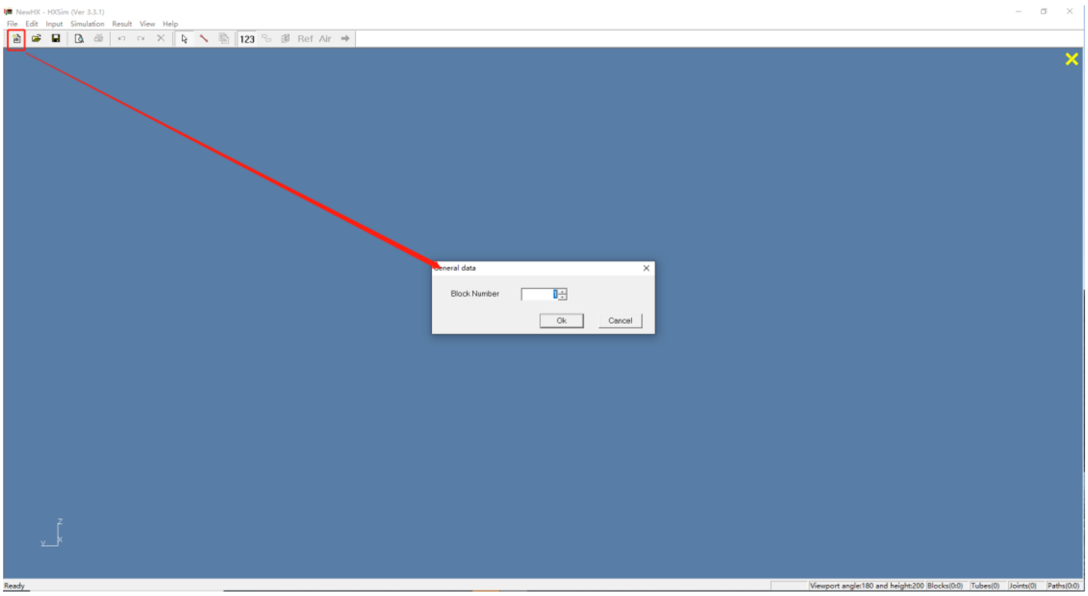


Figure 3-2 New a case

(2) Press button “block” to open block dialog, and input the parameters listed in Table

3-1 like the following Fig. 3-3. Then, press the button “tube type”, select all the tubes in the table (Caution, the grids will be highlighted as blue when they are selected), and then press the button “specify tube type” to choose the specified 9.52 mm grooved tube, as shown in Fig. 3-4.

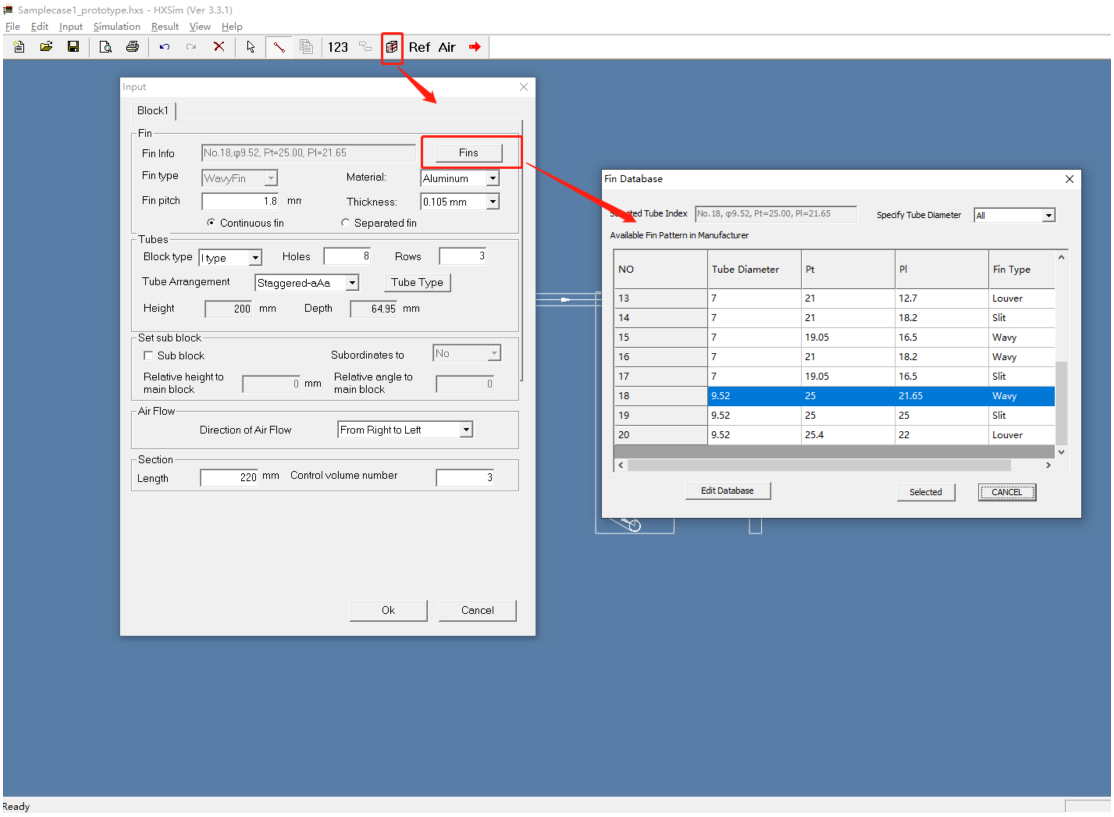


Figure 3-3 Setting the parameter of HX

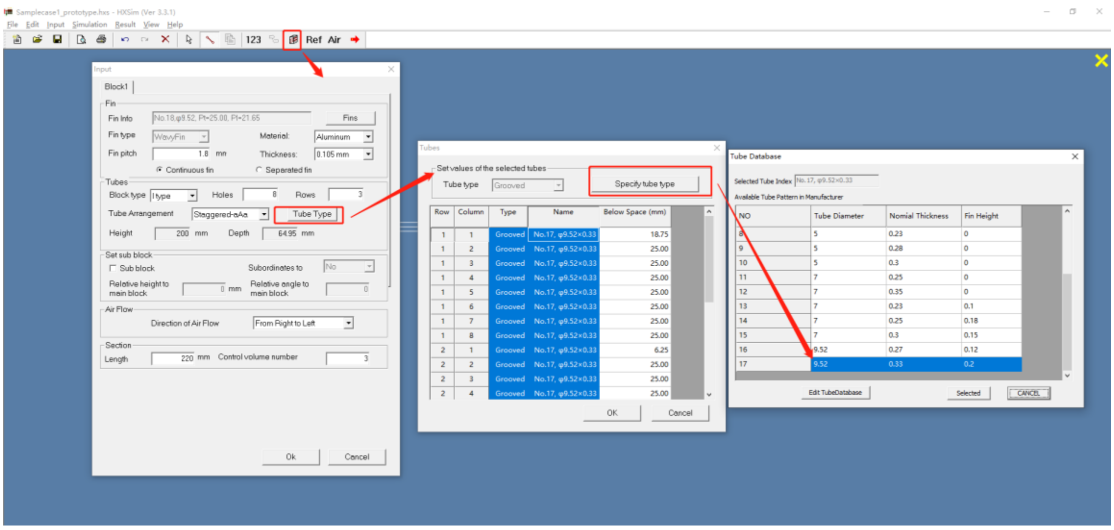


Figure 3-4 Setting the tube parameters

(3) Connect the flow circuits like the follows as shown in Fig. 3-1.

(4) Setting the refrigerant side input and airside input

Input the refrigerant side working conditions listed in Table 3-2 into the refrigerant setting dialog, as shown in Fig. 3-6 (a). Caution, the refrigerant mass flow rate is unknown at first, but its value can be obtained by either iterative calculation or theoretical calculation using the rated capacity dividing the specific enthalpy difference of inlet and outlet of heat exchangers as listed in Eq. (3-1). Due to the discharge temperature and subcooling is known, the specific enthalpy of inlet and outlet can be calculated by the pressure, discharge temperatures, and pressure, subcooling temperatures, respectively.

Then, input the airside working conditions listed in Table 3-2, including air volume flow rate, dry bulb temperature, and wet bulb temperature, as shown in Fig. 3-6 (b).

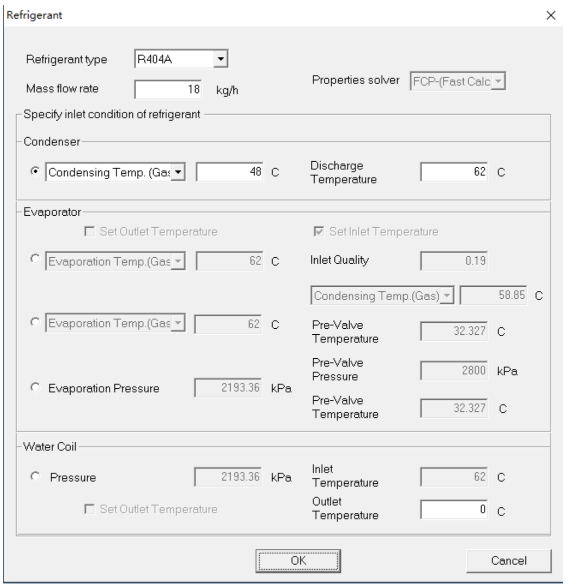


Figure 3-5 Setting the inputs of refrigerant

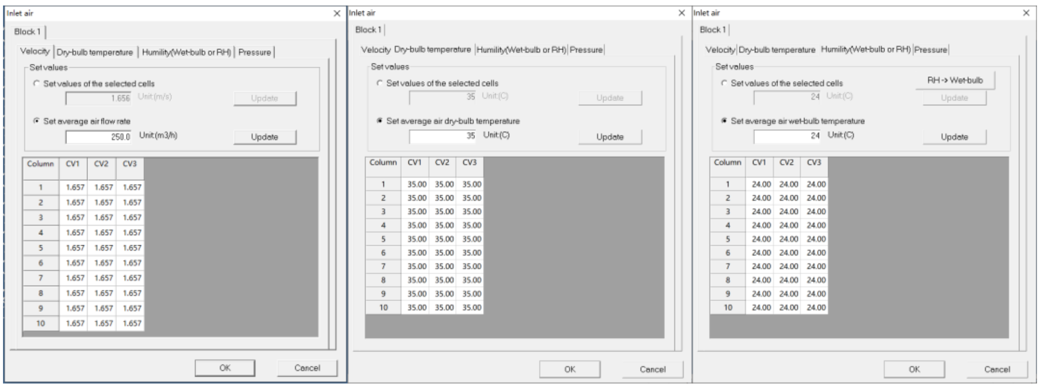


Figure 3-6 Setting the inputs of air

(5) Click the simulation/run button to run the simulation, and user can obtain the simulation results listed in table 3-3.

3.2 Design and optimization of small diameter copper tube HX

Due to the restriction of available volume for the condenser of cooling cabinet, the shapes of new 5 mm heat exchanger are almost the same as those of original heat exchanger, including the length, height, and et al.

(1) Fin type selection

According to the 1.1.1 (Table 1-1, No.1), the heat exchanger is used for condenser under not very clean condition without any maintenance, and thus wavy fin is selected. The column space and row space of fin usually depend on the fin die which is available for manufacturers. In this case, the fin die (19.05*16.5 wavy fin) is available for the manufacturer, and thus this fin type is used in the design.

(2) Fin pitch selection

According to the 1.1.2, the fin pitch is selected as 1.8 mm (the same as original heat exchanger) due to long term use and no maintenance.

(3) U tubes and row number selection

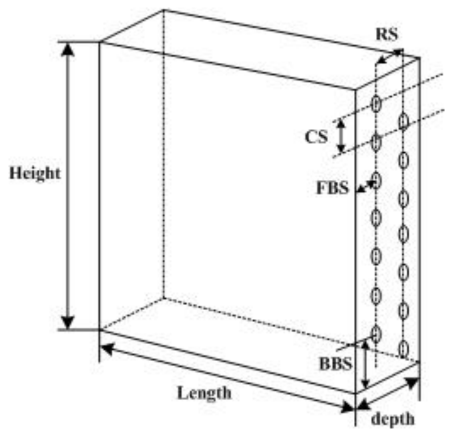
Due to the column space 19.05, and the total height around 200 (original heat exchanger height), the total tube number can be calculated out, which is equal to 10.

Due to the row space 16.5, and the total height around 65 (original heat exchanger height), the total row number can be calculated out, which is equal to 4.

Based on the above selection and calculation, the detailed structure is shown in Table 3-4.

Table 3-4 structure parameters of new designed condenser

No.	Parameters	Value	Note

1	Tube diameter, mm	5	
2	Tube length(Length), mm	220	
3	HX depth (Depth), mm	66	
4	HX height(Height), mm	190.5	
5	Row	4	
6	Column	10	
7	Row space(RS), mm	16.5	
8	Column space(CS), mm	19.05	
9	Fin pitch, mm	1.8	

(4) Flow circuits design

According to 1.2.2.1(Table 1-2) and equation (1-1), the heat exchange capacity of a single path of heat exchanger can be figured out as eq. (3-2), which is equal to 2433 W. The total heat exchange requirements (listed in Table 2-2) is just 690 W. As a result the best refrigerant flow circuits is 1.

$$Q_d = Q_7 \times \left(\frac{d}{6.88}\right)^2 = 5000 \times \left(\frac{5.2-0.2*2}{6.88}\right)^2 = 2433 \text{ W} \qquad (3\text{-}2)$$

According to 1.2.1, item no.(1):

When the heat exchanger is used as the condenser, the refrigerant flow direction should be countercurrent. Namely, the inlet pipe should be designed on the leeward side, and the outlet liquid pipe should be designed on the upwind side.

As a result, the flow circuit is one, and the flow path is counterflow, as shown in Fig. 3-7.

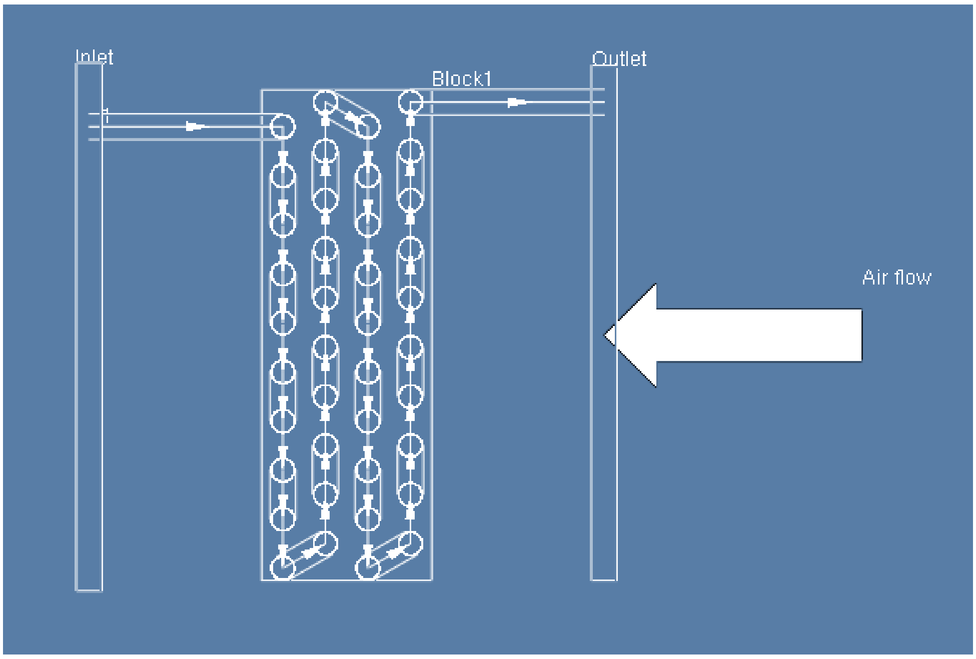


Figure 3-7 Flow circuit of 5 mm condenser of cooling cabinet

3.3 Modeling process of small diameter copper tube HX

(1) Open HXSim, and press new case button, as Fig. 3-8.

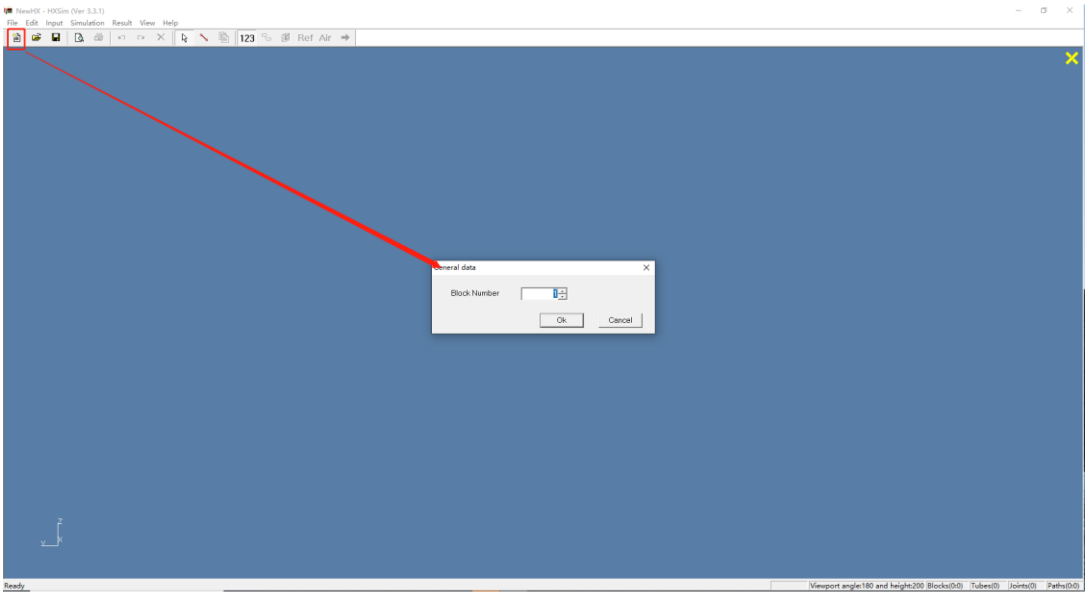


Figure 3-8 New a case

(2) Press button “block” to open block dialog, and input the parameters listed in Table 3-1 like the following Fig. 3-9. Then, press the button “tube type”, select all the tubes in the table (Caution, the grids will be highlighted as blue when they are selected), and then press

the button “specify tube type” to choose the specified 5 mm grooved tube, as shown in Fig. 3-10.

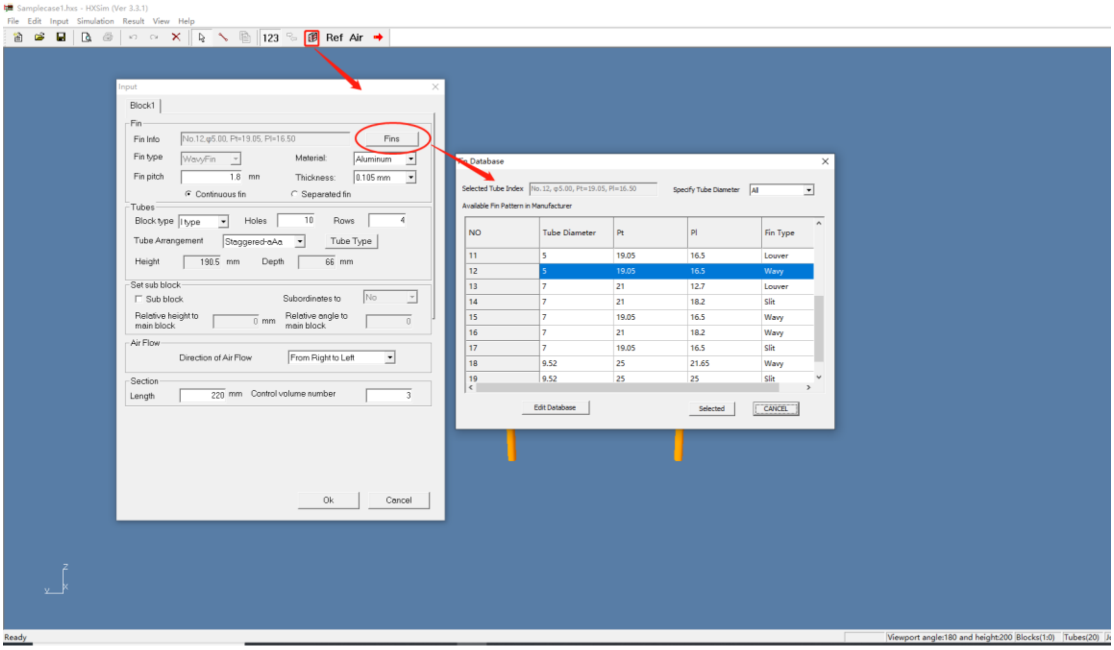


Figure 3-9 Setting the parameter of HX

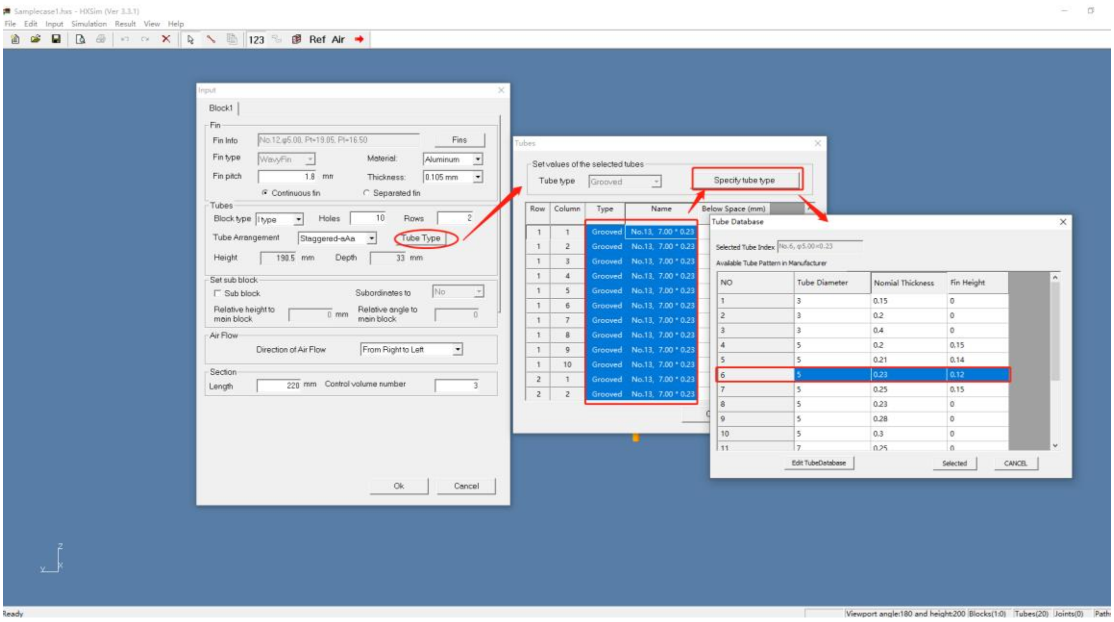


Figure 3-10 Setting the tube parameters

(3) Connect the flow circuits like the follows

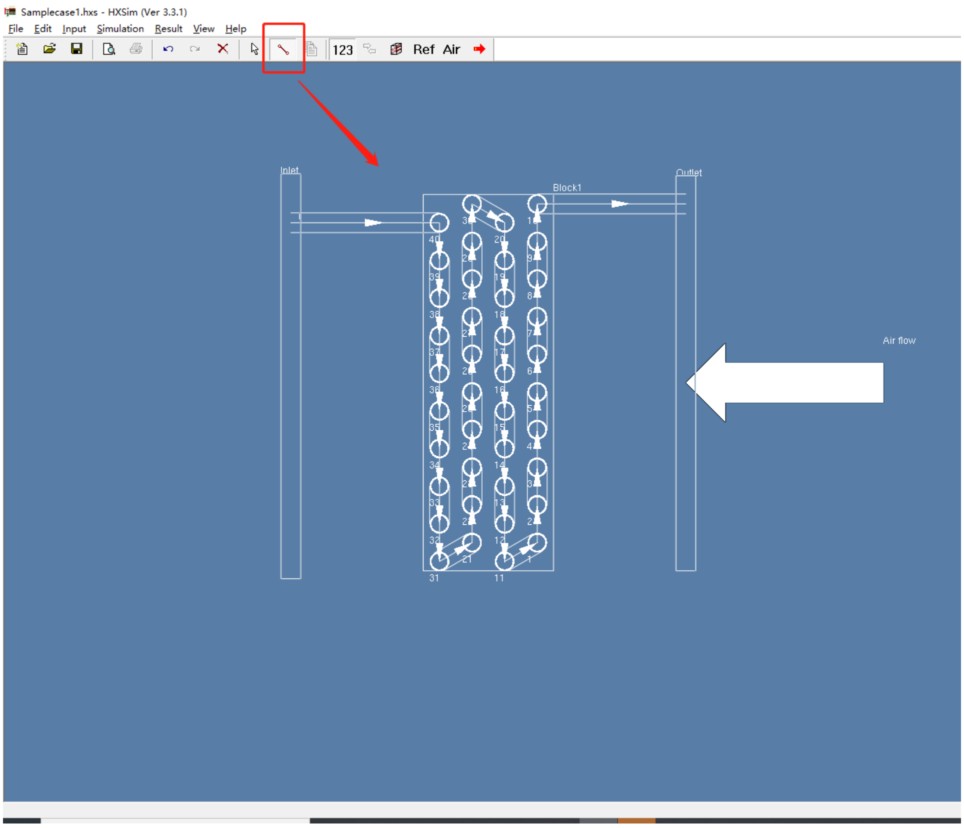
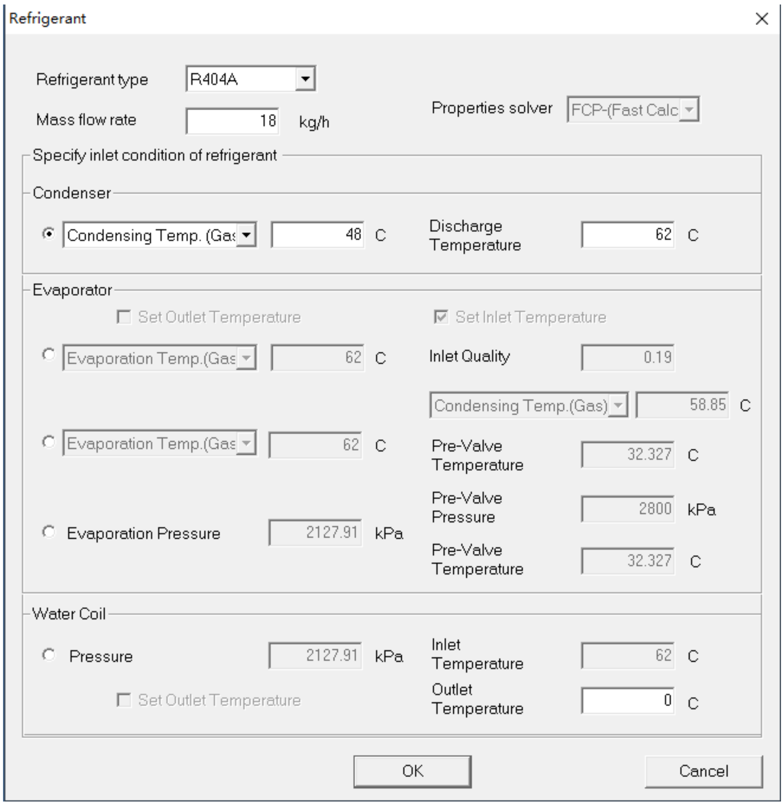
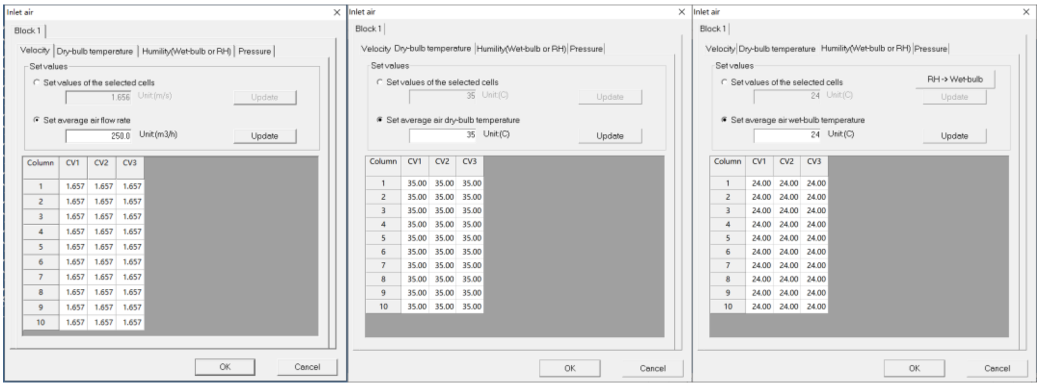


Figure 3-11 Connecting flow circuits

(4) Setting the refrigerant side input and airside input



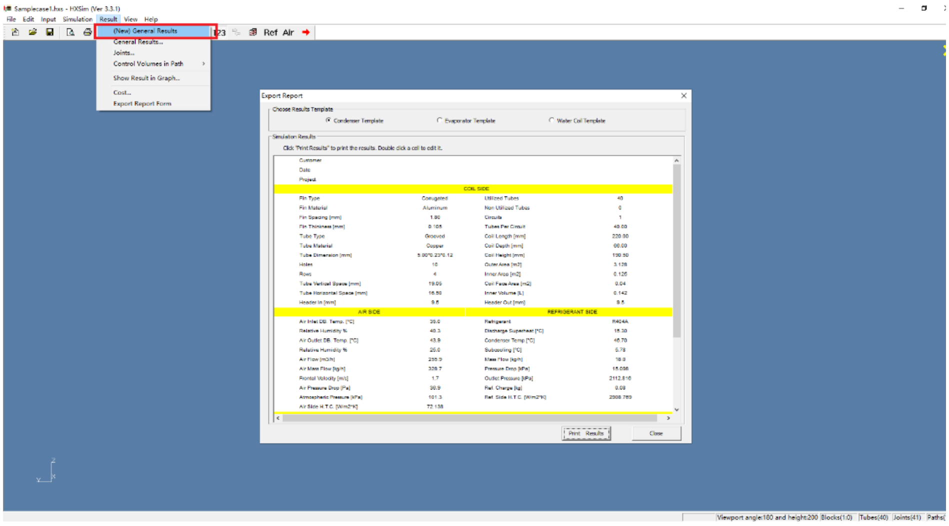
(a)



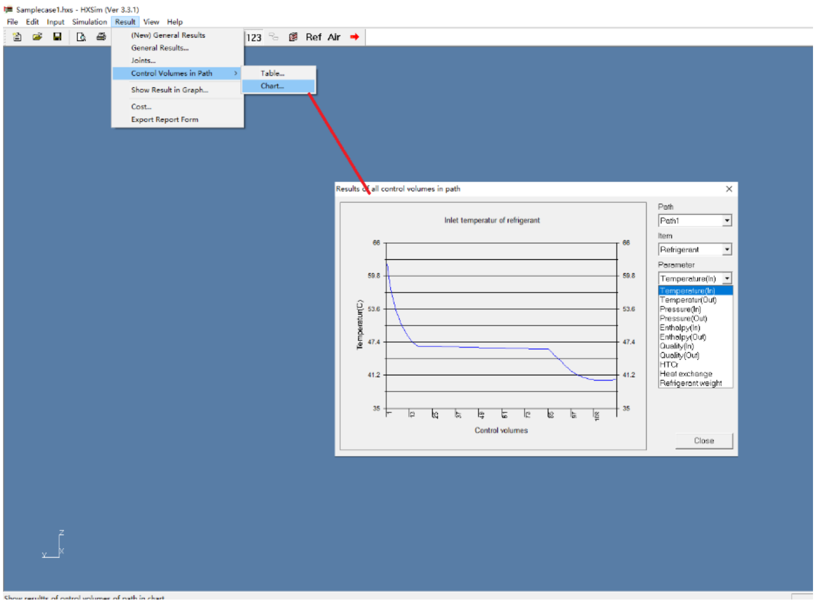
(b)

Figure 3-12 Setting the inputs of refrigerant and air

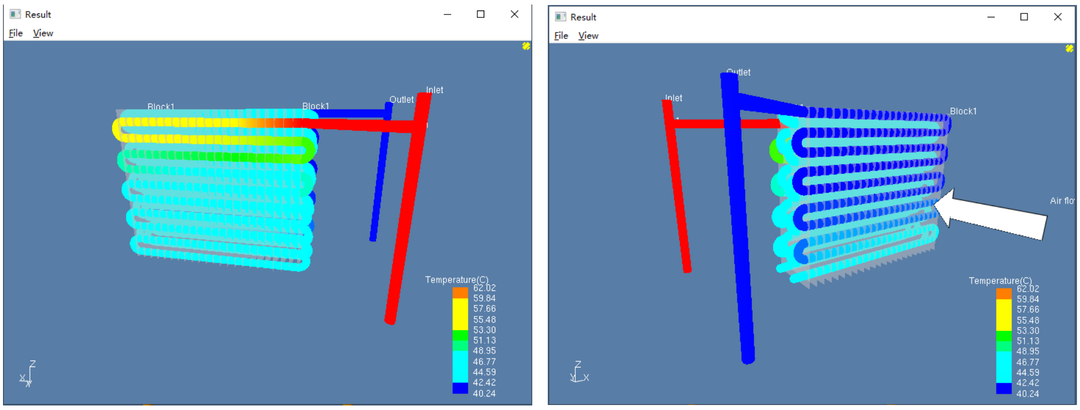
(5) Click the simulation/run button to run the simulation. After simulation, the simulation results can be obtained by clicking the menu result as shown in Fig. 3-13.



(a) Dialog to show summary of simulation result



(b) Dialog to show temperature trend of flow path



(c) dialog to show temperature gradation

Figure 3-13 Dialogs to show simulation results

3.4 Simulation result summary

The summary of simulation results is shown in Table 3-5. The total heat exchange is 726 W, and the subcooling is 5.7 ℃. The total pressure drop is 12 kPa, which is relatively small compared to the total pressure (over 2000 kPa). The heat transfer coefficient along the flow path is shown in Fig. 3-8. From these results, we can conclude that: 5 mm heat exchanger has very good performance in cooling cabinet, especially the capacity is less than 1000 W. Normally, the 5 mm tube will increase the pressure drop, and the heat transfer performance due to larger mass flux. However, when the total capacity is small, the impact of pressure drop is negligible.

As a result, 5 mm tube heat exchanger is very suitable for cooling cabinet condenser.

Table 3-3 Summary of simulation result

		COIL SIDE	
Fin Type	Corrugated	Utilized Tubes	40
Fin Material	Aluminum	Non Utilized Tubes	0
Fin Spacing [mm]	1.80	Circuits	1
Fin Thinkness [mm]	0.105	Tubes Per Circuit	40.00
Tube Type	Grooved	Coil Length [mm]	220.00
Tube Material	Copper	Coil Depth [mm]	66.00
Tube Dimension [mm]	5.00*0.23*0.12	Coil Height [mm]	190.50
Holes	10	Outer Area [m2]	3.128
Rows	4	Inner Area [m2]	0.125
Tube Vertical Space [mm]	19.05	Coil Face Area [m2]	0.04
Tube Horizontal Space [mm]	16.50	Inner Volume [L]	0.142
Header In [mm]	9.5	Header Out [mm]	9.5
AIR SIDE		**REFRIGERANT SIDE**	
Air Inlet DB. Temp. [°C]	35.0	Refrigerant	R404A
Relative Humidity %	40.3	Discharge Superheat [°C]	14.00
Air Outlet DB. Temp. [°C]	44.0	Condenser Temp.[°C]	48.00
Relative Humidity %	24.9	Subcooling [°C]	8.25
Air Flow [m3/h]	255.5	Mass Flow [kg/h]	18.0
Air Mass Flow [kg/h]	328.2	Pressure Drop [kPa]	12.221
Frontal Velocity [m/s]	1.7	Outlet Pressure [kPa]	2181.143
Air Pressure Drop [Pa]	36.8	Ref. Charge [kg]	0.10
Atmospheric Pressure [kPa]	101.3	Ref. Side H.T.C. [W/m2*K]	2462.536
Air Side H.T.C. [W/m2*K]	72.077		
		CAPACITY	
Total Capacity [kW]	0.726		

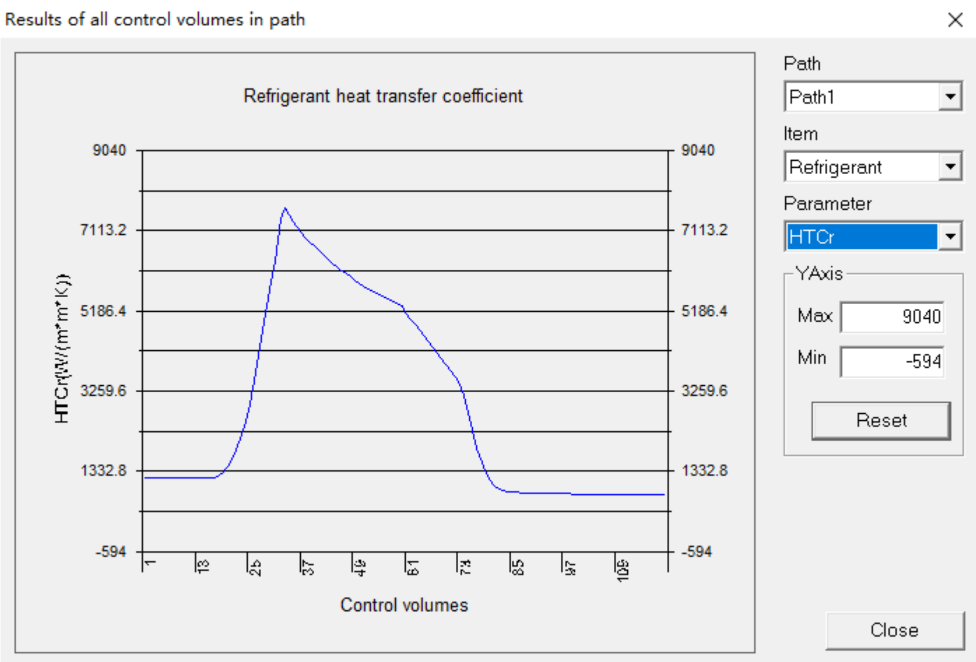


Figure 3-14 results of heat transfer coefficient along the flow path

Appendix A: Heat Transfer Correlation

A.1 Evaporation heat transfer of R32

Jige et al. [1] investigated the boiling heat transfer and flow characteristic of R32 in a small-diameter microfin tube, and he found that Diani et al.'s correlation[2] has a relative good prediction with the error less than 30% for the most of data. The detailed correlation is explained as below. Further, this correlation also works well for R1234ze(E) because this correlation is developed for R1234ze(E) at first.

$$HTC = HTCnb + HTCcv \tag{A-1}$$

$$HTCnb = 0.473HTCcooper * S \tag{A-2}$$

$$HTCcooper = 55p_{red}^{0.12}\left[-log_{10}^{p_{red}}\right]^{-0.55}M^{-0.5}HF^{0.67} \tag{A-3}$$

$$HF = \frac{q}{\pi DL} \tag{A-4}$$

$$S = 1.36X_{tt}^{0.36} \tag{A-5}$$

$$Xtt = \begin{cases} \left[\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}\right] \\ 1 \;\; if\; Xtt > 1 \end{cases} \tag{A-6}$$

$$HTCcv = 1.465HTC_{LO}\left[1 + 1.128x^{0.8170}\left(\frac{\rho_l}{\rho_v}\right)^{0.3685}\left(\frac{\mu_l}{\mu_v}\right)^{0.2363}\left(1-\frac{\mu_v}{\mu_l}\right)^{2.144}Pr_L^{-0.1}\right]Rx^{2.14}(Bo \cdot Fr)^{0.15}\left(\frac{G_0}{G}\right)^{0.36} \tag{A-7}$$

Where G0=100 $kgm^{-2}s^{-1}$.

$$HTC_{lo} = 0.023\frac{\lambda_L}{D}Re_{lo}^{0.8}Pr_L^{0.333} \tag{A-9}$$

$$Pr_L = \frac{\mu_L C_{p,L}}{\lambda_L} \tag{A-10}$$

$$Re_{Lo} = \frac{GD}{\mu_L} \tag{A-11}$$

$$Fr = \frac{G^2}{\rho_v^2 gD} \tag{A-12}$$

$$Bo = \frac{g\rho_L h\pi D}{8\sigma n} \tag{A-13}$$

A.2 Condensation heat transfer of R32

Sunil S. Mehendale[3] had built a general condensation heat transfer correlation for pure refrigerants and refrigerant mixtures. The detailed correlation is introduced below.

$$Nu_{\mathrm{cond}} = \frac{h_{\mathrm{cond}} d_r}{k_1} = C_0 \cdot \Pi_1^{C_1} \cdot \Pi_2^{C_2} \cdot \Pi_3^{C_3} \cdot \Pi_4^{C_4} \cdot \Pi_5^{C_5} \cdot \Pi_6^{C_6} \cdot \Pi_7^{C_7} \cdot \Pi_8^{C_8} \cdot \Pi_9^{C_9} \tag{A-14}$$

$$\Pi_1 = Nu_{Gn,\ \mathrm{homo}} \tag{A-15}$$

$$\Pi_2 = Ja_l \tag{A-16}$$

$$\Pi_3 = \mathrm{Pr}_l \tag{A-17}$$

$$\Pi_4 = \frac{g \Delta \rho e d_r}{\sigma n_f} \tag{A-18}$$

$$\Pi_5 = \frac{\mu_l^4 \cdot g}{\rho_l \sigma^3} \tag{A-19}$$

$$\Pi_6 = \frac{\tau_{i,Smith} \rho_v}{G^2} = \frac{f_{i,\ \mathrm{Smith}}}{2} \left(\frac{\chi}{\varepsilon_{\mathrm{Smith}}} \right)^2 \tag{A-20}$$

$$\Pi_7 = \frac{1-x}{x} \tag{A-21}$$

$$\Pi_8 = \frac{\Delta \rho}{\rho_l} \tag{A-22}$$

$$\Pi_9 = E_a = \left(\frac{2e \cdot n_f}{\pi \cdot d_r} \right) \cdot \left(\sqrt{\left(\frac{1}{\cos^2 \gamma} + \tan^2 \left(\frac{\beta}{2} \right) \right)} - \tan \left(\frac{\beta}{2} \right) \right) + 1 \tag{A-23}$$

Where the C1 to C9 is listed below:

$C_0 = 4.5992 \times 10^{-7}, C_1 = 0.4724, C_2 = -0.3458, C_3 = 2.9947, C_4 = 0.4280, C_5 = -0.6949, C_6 = 0.2553, C_7 = -0.0452, C_8 = -2.8747, C_9 = -0.5295$

The Nusselt number of homo is calculated as below:

For the turbulent flow:

$$\mathrm{Nu}_{\mathrm{turb}} = \frac{\left(f_{\mathrm{turb}} / 8 \right) \cdot \left(\mathrm{Re}_{\mathrm{homo}} - 1000 \right) \cdot \mathrm{Pr}_{\mathrm{homo}}}{1 + 12.7 \cdot \sqrt{f_{\mathrm{turb}} / 8} \cdot \left(\mathrm{Pr}_{\mathrm{homo}}^{2/3} - 1 \right)} \tag{A-24}$$

$$f_{\mathrm{turb}} = \left(1.8 \log_{10} \left(\mathrm{Re}_{\mathrm{homo}} \right) - 1.5 \right)^{-2} \tag{A-25}$$

For the laminar flow:

$$Nu_{lam} = 3.66 \tag{A-26}$$

The void fraction is calculated as below.

$$\varepsilon_{\text{Smith}}=\left(1+\left(\frac{\rho_v}{\rho_l}\frac{1-x}{x}\right)\cdot 0.4+0.6\cdot\sqrt{\frac{\frac{\rho_v}{\rho_l}+0.4\cdot\left(\frac{1-x}{x}\right)}{1+0.4\cdot\frac{1-x}{x}}}\right)^{-1} \tag{A-27}$$

$$f_{i,\text{Smith}}=0.005\cdot\left(1+300\cdot\frac{\left(1-\sqrt{\varepsilon_{\text{Smith}}}\right)}{2}\right) \tag{A-28}$$

The prediction result listed in the paper is listed as below.

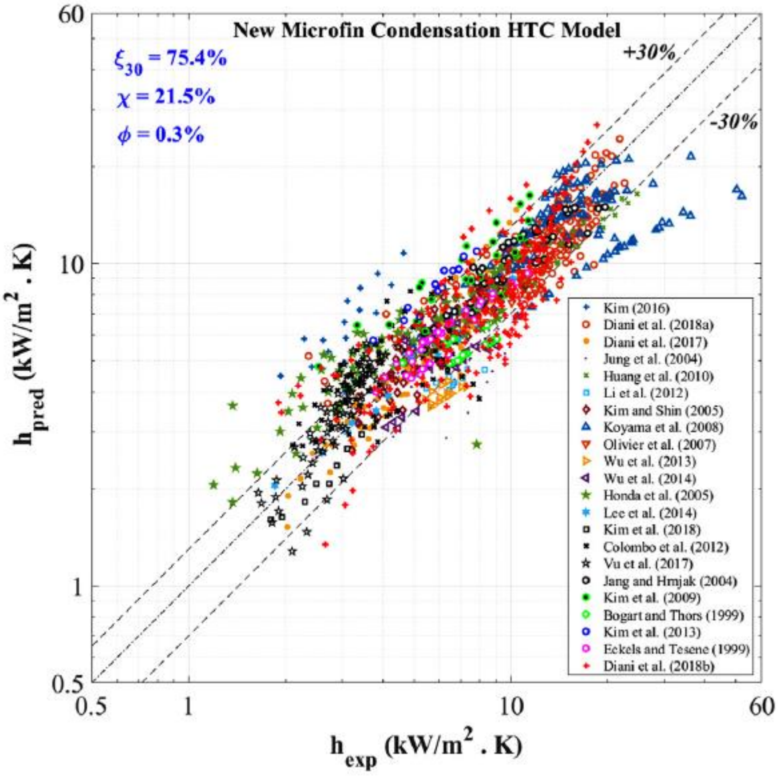


Fig. A-1 Prediction result of correlation[3] (Origination: Fig. 6 in literature [3])

A.3 Evaporation heat transfer of R404A

Klaus spindler.Hans Muller-Steinhagen[4] investigated the flow boiling heat transfer of R134a and R404A in a microfin tube, and he found that Kandlikar's correlation[5] has a relative good prediction with an error less than 15% for the most of data. The detailed correlation is explained below.

$$HTC/h_l=C_1Co^{C_2}(25Fr_{lo})^{C_5}+C_3Bo^{C_4}F_{fl} \tag{A-29}$$

$$h_l=0.023\frac{\lambda_L}{D}Re_{lo}^{0.8}Pr_L^{0.4} \tag{A-30}$$

The value of F_{fl} is a correction factor, and For R404A, F_{fl} is equal to 1.24.

$$Bo = {}^{q}/_{Gh_{fg}} \tag{A-31}$$

$$Fr_{lo} = \frac{G^2}{\rho_l^2 gD} \tag{A-31}$$

Where G is mass flux, g is the acceleration due to gravity, q is heat flux.

The constants C_1 to C_5 are listed in Table 1-1. The two sets of values given in Table 1-1 corresponding to convective boiling and nucleate boiling regions, respectively. The heat transfer coefficient at any given conditions is evaluated using the two sets of constants for the two regions, and since the transition from one region to another occurs at the intersection of the respective correlations, the higher of the two heat transfer coefficient values represents the predicted value from the proposed correlation. This method provides a continuity between the convective and nucleate boiling regions.

Table A-1 Constants in the proposed correlation in Eq. (29)

Constant	Convective region	Nucleate boiling region
C_1	1.1360	0.6683
C_2	-0.9	-0.2
C_3	667.2	1058
C_4	0.7	0.7
C_5	0.3	0.3

Note: C_5=0 for vertical tubes, and for horizontal tubes with Fr_l>0.4

A.4 Condensation heat transfer of R404A

The correlation is the same as R32. Sunil S. Mehendale[3] had built a general condensation heat transfer correlation for pure refrigerants and refrigerant mixtures.

Appendix B: References

[1]. Dasuke Jige, Kentaro Sagawa, Shota lizuka, Norihiro Inoue. Boiling heat transfer and flow characteristic of R32 inside a horizontal small-diameter microfin tube. International Journal of Refrigeration. DOI: https://doi.org/10.1016/j.ijrefrig.20189.08.019.

[2]. Andrea Diani, Simone Mancin, Luisa Rossetto. R1234ze(E) flow boiling inside a 3.4 mm ID microfin tube. International Journal of Refrigeration 47 (2014): 105-109.

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[4]. Klaus spindler.Hans Muller-Steinhagen. Flow boiling heat transfer of R134a and R404A in a microfin tube at low mass fluxes and low heat fluxes. Heat Mass Transfer (2009) 45: 967-977.

[5]. S.G. Kandlkar. A general Correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes. Journal of heat transfer (1990) 112: 219-228.

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