Numerical investigation on fin-tube three-fluid heat exchanger for hybrid source HVAC&R systems

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Abstract: The energy consumption of HVAC&R systems is increasing rapidly and application of hybrid source systems is an alternative solution. Utilizing three-fluid heat exchanger in hybrid source HVAC&R system can make the system more compact in construction, higher in thermal efficiency and lower in reliability risk. In this paper, a simulation model of a fin-tube three-fluid heat exchanger is built with distributed-parameter method and validated by experimental data. Based on the demand of a new type of hybrid source system, the optimal circuit arrangement type and tube diameter are achieved. The heat transfer rates of the three fluids under different working conditions are investigated. The results also show that to ensure that cold fluid 1 is superheated in the outlet for the stable running of the system, flow rate

and inlet pressure of the hot fluid and air inlet temperature should be higher than certain values, while flow rate and inlet pressure of cold fluid 1 should be lower than certain values. Air flow rate can be adjusted freely without range limit. This paper will help the design of fin-tube three-fluid heat exchanger and promote its application in HVAC&R systems.

Key words: HVAC&R, three-fluid heat exchanger, heat pump, air conditioning, simulation

Nomenclatures

A_1, A_2	areas as illustrated in Fig. 2, m^2
c_p	specific heat, J kg K^{-1}
G	mass flux, $\text{kg m}^{-2} \text{s}^{-1}$
g	acceleration of gravity, m s^{-2}
h_{fg}	liquid-vapor latent heat, J kg^{-1}
$\dot{m}$	mass flow rate, kg s^{-1}
p	pressure, Pa
Q	heat transfer rate, W
T	temperature, K
U_1, U_2	overall heat transfer coefficient, $\text{W K}^{-1} \text{m}^{-2}$
$\dot{v}$	volume flow rate, $\text{m}^3 \text{s}^{-1}$
x	vapor quality
z	height, m
Δp	pressure drop, Pa
Δv	change of specific volume, $\text{m}^3 \text{kg}^{-1}$
ρ	density, kg m^{-3}

Subscripts

a	acceleration
ave	average
cal	calculated
c1	cold fluid 1
c2	cold fluid 2
f	friction
g	gravity
h	hot fluid
in	inlet
out	outlet
sat	saturation
sup	supposed
TP	two-phase

Abbreviations

HVAC&R	heating, ventilation, air conditioning and refrigeration
ISMT	integrated system of mechanical refrigeration and thermosyphon

1 Introduction

Nowadays, Heating, Ventilation, Air Conditioning and Refrigeration (HVAC&R) systems are used in a large diversity of activities. The energy consumption of HVAC&R systems accounts for more than 35% of the total electrical energy consumed in the US [1]. Utilization of renewable energy is the most effective method to reduce this consumption. However, many kinds of renewable energy are not stable and adequate enough to be used as the only energy source [2-5]. Hybrid source

HVAC&R system is an ideal solution to achieve continuous working and shows great application potential, representative examples are hybrid source heat pumps [6-9] and hybrid source free cooling systems [10-12].

In order to use more than one energy sources, most hybrid source systems use two or more heat exchangers in series or parallel connection to transfer heat between three or more fluids. This kind of system has energy-saving effect while it still has the following shortcomings:

(1) To achieve the utilization of multi-source, series or parallel connection of heat exchangers exists, which makes the system complex in structure [13].

(2) Fluid flow inside the system is difficult to forecast due to series and parallel connection of heat exchangers, which affects the stable and efficient working [14].

(3) **Solenoid** valves are needed for switching between working modes [15], which brings reliability risk.

As we know, three-fluid heat exchanger is a **type** of heat exchanger**s** which can transfer heat synchronously between three fluids. It has been used in cryogenics, aerospace and chemical industries [16,17]. The application of three-fluid heat exchangers in these fields makes the equipments more compact in construction, higher in thermal efficiency [18] and lower in reliability risk [19]. **However**, few three-fluid heat exchangers have been used in HVAC&R systems until now, while they show great potential in this field to overcome the above shortcomings.

Analysis of three-fluid heat exchanger was first conducted by Morley in 1933 [20]. Ever since many researchers developed it further. Analytical solution was obtained for special cases [21]. Finite element models were developed for some **types** of three-fluid heat exchangers, including co-current parallel flow [22,23], counter-current parallel flow [17,22-25] and cross parallel flow [26-28]. Integral-mean

temperature difference model has also been developed to simplify the simulation of three-fluid heat exchangers [18]. Dimensionless design parameters that can be used to evaluate the performance of three-fluid heat exchangers have been studied [16,29,30]. Exergy theory and entropy generation number have been applied to describe and analyse the performance of three-fluid heat exchangers from irreversible thermodynamic theory [24,25]. However, all of the above studies only deal with **heat transfer without phase change** and do not include three-fluid heat exchangers that can be used for HVAC&R systems, in which two-phase fluid and air often exist.

A new **type** of three-fluid heat exchanger, fin-tube three-fluid heat exchanger, was proposed recently [31] as shown in Fig. 1. Its tube circuit consists of an inner tube, an outer tube and fins outside the outer tube. This **type** of heat exchanger can be used to achieve the heat transfer between air and two in-tube fluids. Therefore, it is suitable for application in hybrid source HVAC&R systems such as hybrid source heat pumps and free cooling systems. However, few studies have been conducted on this new type heat exchanger until now.

To overcome the above short comings of existing hybrid source systems, the authors proposed a new type of hybrid source free cooling system utilizing fin-tube three-fluid heat exchanger, called integrated system of mechanical refrigeration and thermosyphon (ISMT) [32]. The schematic diagram of the system is shown in Fig. 2. It includes two circulation loops: a refrigeration loop and a thermosyphon loop. A three-fluid heat exchanger is used to connect the two loops. The fluid in the thermosyphon loop can transfer heat with the other two fluids to achieve the simultaneous utilization of mechanical cooling source and free cooling source.

For better use of fin-tube three-fluid heat exchanger, performance investigation is essential. This paper focuses on performance investigation of fin-tube three-fluid

heat exchanger for application in ISMT, a new type hybrid source HVAC&R system. A simulation model of the fin-tube three-fluid heat exchanger is built with **distributed-parameter** method and validated by experimental data. With the validated simulation model, the performance and applicability of different structures and working conditions in the ISMT are investigated. This paper will help the design of fin-tube three-fluid heat exchanger and promote its application in HVAC&R systems.

2 Model of Three-fluid Heat Exchanger

The flow arrangement of the fin-tube three-fluid heat exchanger in the ISMT is shown in Fig. 3. Cold fluid 1 (working fluid of the refrigeration loop) flows and evaporates in the inner tube. Cold fluid 2 (air) flows through the fins outside the outer tube. The hot fluid (working fluid of the thermosyphon loop) flows and condenses in the annular space between the inner and outer tube, cooled by the other two fluids.

The numerical model is developed with **distributed-parameter** method, dividing the tube circuit into elements along the axial direction as shown in Fig. 3. **Each tube is divided into 30 elements.** The flow chart of three-fluid heat exchanger simulation is shown in Fig. 4. **The inlet conditions of the hot fluid and cold fluid 1 in one tube are the outlet conditions of the former tube in the flow circuit. The inlet air (cold fluid 2) conditions of each element in the second row along the airflow direction are the outlet conditions of the element in the same position in first row.** Compared with two-fluid heat exchangers, two heat transfer rates are needed to be supposed and two iterations are needed in **each** element. The heat transfer rate of the two sides in an element can be calculated as:

$$Q_{c1} = U_1 A_1 (T_{h,ave} - T_{c1,ave}) \quad (1)$$

$$Q_{c2} = U_2 A_2 (T_{h,ave} - T_{c2,ave}) \quad (2)$$

where, Q_{c1} and Q_{c2} are the heat transfer rate **in an element** of cold fluid 1 and cold

fluid 2 (air), respectively, W; $T_{h,ave}$, $T_{c1,ave}$, $T_{c2,ave}$ are the average temperature in the element of the hot fluid, cold fluid 1 and cold fluid 2, respectively, K; U_1A_1 and U_2A_2 are the overall heat conductance in an element as illustrated in Fig. 3, W K⁻¹. The heat transfer rate of the hot fluid is also the total heat transfer rate of the element:

$$Q_h = Q_{c1} + Q_{c2} \quad (3)$$

The two-phase pressure drop in an element is calculated as:

$$\Delta p = \Delta p_f + \Delta p_a + \Delta p_g = \Delta p_f + G^2 \Delta v + g \rho_{TP} z \quad (4)$$

where, Δp_f , Δp_a , Δp_g are the fiction, acceleration and gravity pressure drop, respectively, Pa; G is the mass flux, kg m⁻² s⁻¹; Δv is the change of specific volume, m³ kg⁻¹; ρ_{TP} is the density of two-phase fluid, kg m⁻³; z is the height difference of the inlet and outlet of the element, m; g is the acceleration of gravity, m s⁻².

The heat transfer and pressure drop correlations are shown in Table 1. The refrigerant thermodynamic properties are fitted by the data provided by Refprop 8 [39], and the modelling code is developed based on VC6.0.

In order to validate the accuracy of the model, the performance of the three-fluid heat exchanger was also investigated by experiments. The experiments were carried out in a test room of air conditioners. The airside conditions were controlled by the air handling unit of the test room, and measured by a test chamber of enthalpy difference. The flow conditions inside the tube were measured by transducers of pressure, temperature and flow rate in the inlet and outlet of the heat exchanger. The front velocity of the airflow is around 1.1 m³ s⁻¹ and the temperature is 12~22 °C. The flow rates of the hot fluid and cold fluid 1 are 0.03~0.10 kg s⁻¹ and 0.02~0.04 kg s⁻¹, respectively. The calculated total heat transfer rate (heat transfer rate of the hot fluid) is compared to experimental results of the three-fluid heat exchanger as shown in Fig.

5. The calculated values agree well with the experimental ones with deviations of -10%

and +6%.

3 Results and Discussions

In this paper, the applicability of different structures of fin-tube three-fluid heat exchanger is studied firstly to choose an appropriate structure for the system. Then, the inlet fluid conditions are varied to investigate the performance of the heat exchanger in various working conditions of the system.

3.1 Performance of different structures

The outlet of cold fluid 1 connects the inlet of the compressor in the refrigeration loop (refer to Fig. 2). Therefore, cold fluid 1 in the outlet must be superheated to avoid liquid hammer, **which means liquid fluid entering the compressor and will damage its valve plate**. In this **section**, the required outlet condition of cold fluid 1 is given to choose between different structures. This typical working condition is shown in Table 2.

Three different circuit arrangement types of fin-tube three-fluid heat exchanger are shown in Fig. 6. All of the three types have two rows and 28 inner and outer tubes in each row while circuit numbers are different. In type A, there are 28 parallel circuits of cold fluid 1 and hot fluid, that is, one circuit of cold fluid 1 connects one circuit of the hot fluid. In type B and C, one circuit of cold fluid 1 connects four and seven circuits of the hot fluid, respectively, reducing the circuit number of cold fluid 1 compared to Type A. The geometric parameters of the three types are the same and shown in Table 3.

The total heat transfer rates (heat transfer rates of the hot fluid) of different arrangement types and inner tube diameters are shown in Fig. 7. The heat transfer rate of type A is relatively small because it has more circuits of cold fluid 1 therefore the flow rate in one single circuit is lower. However, it can be seen that less circuits do

not always mean higher heat transfer rate. The heat transfer rate of type C is lower than **that of** type B. This is mainly due to the higher pressure drop of cold fluid 1 in Type C so that when the outlet condition is the same, the average evaporation pressure of cold fluid 1 is higher. Therefore, the temperature difference between the hot fluid and cold fluid 1 in Type C is smaller than Type B so the heat transfer rate is smaller. The heat transfer rate increases with increasing inner tube diameter for smaller flow area and higher velocity of the hot fluid. While, the inner tube diameter should not be too close to the outer tube diameter, which will bring higher pressure drop of the hot fluid side and difficulty in manufacture. Considering these factors, type B is chosen for the system and the inner tube outside diameter is fixed at 6.0 mm.

3.2 Performance under different working conditions

With the optimal structure achieved above, the performance under different inlet conditions is investigated in this **section**. The inlet conditions are those often appear in the ISMT, which are shown in Table 4. When one inlet parameter is varied **in the variation range to investigate its effect on the performance**, other inlet parameters keep the base line values **in Table 4**.

The heat transfer rate and outlet temperature varied with the flow rate of the hot fluid are shown in Fig. 8 and Fig. 9, respectively. The heat transfer rates of the three fluids all increase with increasing flow rate of the hot fluid, because higher flow rate enhances the heat transfer between the hot fluid and other fluids. When cold fluid 1 is superheated in the outlet ($T_{c1,out} > T_{c1,sat}$, $T_{c1,sat}$ is the saturation temperature corresponding to the outlet pressure of cold fluid 1), this increase turns slower. As mentioned above, cold fluid 1 in the outlet must be superheated to avoid liquid hammer of the compressor in the ISMT. When the flow rate of the hot fluid is relatively low, cold fluid 1 in the outlet is in two-phase state, as shown in Fig. 9.

Therefore, flow rate of the hot fluid in the system should be higher than a certain value, which is 0.044 kg s^{-1} in this study.

The heat transfer rate and outlet temperature varied with the flow rate of cold fluid 1 are shown in Fig. 10 and Fig. 11, respectively. The heat transfer rates of the hot fluid and cold fluid 1 increase with increasing flow rate of cold fluid 1, while the heat transfer rate of cold fluid 2 slightly decreases. That is, increasing flow rate of cold fluid 1 will reduce the heat transfer rate of the other side. When the flow rate of cold fluid 1 is relatively high, cold fluid 1 in the outlet is in two-phase state, as shown in Fig.11. Therefore to ensure that cold fluid 1 is superheated in the outlet, the flow rate of cold fluid 1 should be lower than a certain value, which is 0.033 kg s^{-1} in this study.

The heat transfer rate and outlet temperature varied with the flow rate of cold fluid 2 (air) are shown in Fig. 12 and Fig. 13, respectively. The heat transfer rates of the hot fluid and air increase with increasing air flow rate, while the heat transfer rate of cold fluid 1 slightly decreases. That is, increasing air flow rate will reduce the heat transfer rate of the fluid inside the inner tube. Cold fluid 1 is always superheated when the air flow rate is varied from 0 to $2.24 \text{ m}^3 \text{ s}^{-1}$, which is quite a wide range, thus air flow rate can be adjusted freely according to demand of this system.

The heat transfer rate and outlet temperature varied with the inlet pressure of the hot fluid are shown in Fig. 14 and Fig. 15, respectively. With increasing inlet pressure of the hot fluid, the temperature differences between the hot fluid and the other two fluids increase therefore the heat transfer rates of all the three fluids increase, while the increase of cold fluid 1 is not as obvious as cold fluid 2 (air), due to superheated occurs in its outlet. When the inlet pressure of the hot fluid is relatively low, cold fluid 1 in the outlet is in two-phase state. To ensure that cold fluid 1 is superheated in the

outlet, the inlet pressure of the hot fluid should be higher than a certain value, which is 0.801 MPa in this study.

The heat transfer rate and outlet temperature varied with the inlet pressure of cold fluid 1 are shown in Fig. 16 and Fig. 17, respectively. With increasing inlet pressure of cold fluid 1, the temperature difference between the hot fluid and cold fluid 1 decreases therefore the heat transfer rates of the hot fluid and cold fluid 1 decrease, while the heat transfer rate of cold fluid 2 (air) slightly increases. When the inlet pressure of cold fluid 1 is relatively high, cold fluid 1 in the outlet is in two-phase state. To ensure that cold fluid 1 is superheated in the outlet, the inlet pressure of cold fluid 1 should be lower than a certain value, which is 0.638 MPa in this study.

The heat transfer rate and outlet temperature varied with the inlet temperature of cold fluid 2 (air) are shown in Fig. 18 and Fig. 19, respectively. With increasing air inlet temperature, the temperature difference between the hot fluid and air decreases therefore the heat transfer rate of the airside decreases. Meanwhile, the heat transfer rate of the other side increases. When the air inlet temperature is relatively low, cold fluid 1 in the outlet is in two-phase state, as shown in Fig.19. To ensure that cold fluid 1 is superheated in the outlet, air inlet temperature should be higher than 10 °C in this study. While, it should not be higher than the temperature of the hot fluid to maintain free cooling effect.

4 Conclusions

This study mainly focuses on three-fluid heat exchangers for application in hybrid source HVAC&R systems. A distributed-parameter model of fin-tube three-fluid heat exchanger is built and validated by experimental data. The performance of different geometric structures and working conditions is investigated based on the demand of ISMT, a new type of hybrid source HVAC&R system. The main results

are summarized as follows:

- 1) Circuit arrangement type has significant effect on the performance of fin-tube three-fluid heat exchanger. The optimal type with appropriate circuit number is achieved. The heat transfer rate increases with increasing inner tube diameter and it is fixed at 6.0 mm considering manufacture.
- 2) The heat transfer rates of the three fluids all increase with increasing flow rate of the hot fluid. To ensure that cold fluid 1 is superheated, flow rate of the hot fluid should be higher than a certain value, which is 0.044 kg s^{-1} in this study. Increasing flow rate of cold fluid 1 or cold fluid 2 (air) will improve the heat transfer rate of the hot fluid (total heat transfer rate) and its own side, while reduce the heat transfer rate of the other side. To ensure that cold fluid 1 is superheated, the flow rate of cold fluid 1 should be lower than a certain value, which is 0.033 kg s^{-1} in this study. However, air flow rate can be adjusted freely according to demand of application.
- 3) Increasing inlet pressure of the hot fluid will improve the heat transfer rates of all the three fluids. While, increasing inlet pressure of cold fluid 1 will reduce the heat transfer rate of the hot fluid and its own side, while improve that of the other side. To ensure that cold fluid 1 is superheated in the outlet, the inlet pressure of the hot fluid should be higher than a certain value, which is 0.801 MPa in this study, and the inlet pressure of cold fluid 1 should be lower than a certain value, which is 0.638 MPa in this study.
- 4) With increasing inlet temperature of cold fluid 2 (air), the heat transfer rate of the hot fluid and airside decreases while the heat transfer rate of the other side increases. To ensure that cold fluid 1 is superheated, air inlet temperature should be higher than 10°C in this study.

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Figure Captions

Fig. 1. Structure of fin-tube three-fluid heat exchanger.

Fig. 2. Schematic diagram of the ISMT [32].

Fig. 3. Flow arrangement of one tube circuit.

Fig. 4. Flow chart of simulation.

Fig. 5. Comparison of calculated and experimental values of heat transfer rate.

Fig. 6. Three different circuit arrangement types.

Fig. 7. Heat transfer rate of different structures.

Fig. 8. Heat transfer rate varied with flow rate of hot fluid.

Fig. 9. Outlet temperature varied with flow rate of hot fluid.

Fig. 10. Heat transfer rate varied with flow rate of cold fluid 1.

Fig. 11. Outlet temperature varied with flow rate of cold fluid 1.

Fig. 12. Heat transfer rate varied with flow rate of cold fluid 2 (air).

Fig. 13. Outlet temperature varied with flow rate of cold fluid 2 (air).

Fig. 14. Heat transfer rate varied with inlet pressure of hot fluid.

Fig. 15. Outlet temperature varied with inlet pressure of hot fluid.

Fig. 16. Heat transfer rate varied with inlet pressure of cold fluid 1.

Fig. 17. Outlet temperature varied with inlet pressure of cold fluid 1.

Fig. 18. Heat transfer rate varied with inlet temperature of cold fluid 2 (air).

Fig. 19. Outlet temperature varied with inlet temperature of cold fluid 2 (air).

Table Captions

Table 1. Heat transfer and pressure drop correlations used for simulation.

Table 2. Working condition for structure investigation.

Table 3. Geometric parameters of three-fluid heat exchanger.

Table 4. Working conditions for performance investigation.

Table 1 Heat transfer and pressure drop correlations used for simulation

Item	Correlation
Evaporation heat transfer	Kandlikar [33]
Condensation heat transfer	Shah [34]
Single-phase heat transfer	Dittus and Boelter [35]
Airside heat transfer	Yu [36]
Two-phase friction pressure drop	Lockhart and Martinelli [37]
Single-phase friction pressure drop	Blasius [38]

Table 2 Working condition for structure investigation

Parameter	Value
Inlet enthalpy of the hot fluid (kJ kg^{-1})	392.3
Inlet pressure of the hot fluid (MPa)	0.8207
Mass flow rate of the hot fluid (kg s^{-1})	0.0462
Outlet enthalpy of cold fluid 1 (kJ kg^{-1})	409.4
Outlet pressure of cold fluid 1 (MPa)	0.6089
Mass flow rate of cold fluid 1 (kg s^{-1})	0.031
Temperature of cold fluid 2 (air) ($^{\circ}\text{C}$)	12.0
Relative humidity of cold fluid 2 (air)	50%
Volume flow rate of cold fluid 2 (air) ($\text{m}^3 \text{s}^{-1}$)	1.114
Type of cold fluid 1 and the hot fluid	R22

Table 3 Geometric parameters of three-fluid heat exchanger

Parameter	Value	Parameter	Value
Tube length (mm)	700	Number of rows	2
Inner tube outside diameter (mm)	5.2-6.4	Tube pitch (mm)	25.0
Inner tube wall thickness (mm)	0.25	Row pitch (mm)	21.65
Outer tube outside diameter (mm)	9.52	Fin pitch (mm)	1.5
Outer tube wall thickness (mm)	0.26	Fin thickness (mm)	0.12

Table 4 Working conditions for performance investigation

Parameter	Base line	Variation range
Inlet enthalpy of the hot fluid (kJ kg^{-1})	392.3	--
Inlet pressure of the hot fluid (MPa)	0.8207	0.79-0.84
Mass flow rate of the hot fluid (kg s^{-1})	0.0462	0.037-0.066
Inlet enthalpy of cold fluid 1 (kJ kg^{-1})	256.2	--
Inlet pressure of cold fluid 1 (MPa)	0.6282	0.60-0.65
Mass flow rate of cold fluid 1 (kg s^{-1})	0.031	0.026-0.035
Inlet temperature of cold fluid 2 (air) ($^{\circ}\text{C}$)	12.0	8.0-16.0
Relative humidity of cold fluid 2 (air)	50%	--
Volume flow rate of cold fluid 2 (air) ($\text{m}^3 \text{s}^{-1}$)	1.114	0-2.24

Fig. 1 Structure of fin-tube three-fluid heat exchanger

Fig. 2 Schematic diagram of the ISMT [32]

Fig. 3 Flow arrangement of one tube circuit

Fig. 4 Flow chart of simulation

Fig. 5 Comparison of calculated and experimental values of heat transfer rate

Fig. 6 Three different circuit arrangement types

Fig. 7 Heat transfer rate of different structures

Fig. 8 Heat transfer rate varied with flow rate of hot fluid for $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{v}_{c2} = 1.114 \text{ m}^{-3} \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^{\circ}\text{C}$

Fig. 9 Outlet temperature varied with flow rate of hot fluid for $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$,

$$\dot{v}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}, p_{h,in} = 0.8207 \text{ MPa}, p_{c1,in} = 0.6282 \text{ MPa}, T_{c2,in} = 12.0 \text{ }^{\circ}\text{C}$$

Fig. 10 Heat transfer rate varied with flow rate of cold fluid 1 for $\dot{m}_h = 0.0462 \text{ kg}$

s^{-1} , $\dot{V}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}$, $p_{h,\text{in}} = 0.8207 \text{ MPa}$, $p_{c1,\text{in}} = 0.6282 \text{ MPa}$, $T_{c2,\text{in}} = 12.0 \text{ }^\circ\text{C}$

Fig. 11 Outlet temperature varied with flow rate of cold fluid 1 for $\dot{m}_h = 0.0462 \text{ kg s}^{-1}$

$\dot{v}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ °C}$

Fig. 12 Heat transfer rate varied with flow rate of cold fluid 2 (air) for $\dot{m}_h = 0.0462$

kg s^{-1} , $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^\circ\text{C}$

Fig. 13 Outlet temperature varied with flow rate of cold fluid 2 (air) for $\dot{m}_h = 0.0462$

kg s^{-1} , $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^{\circ}\text{C}$

Fig. 14 Heat transfer rate varied with inlet pressure of hot fluid for $\dot{m}_h = 0.0462 \text{ kg s}^{-1}$,

$$\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}, \dot{v}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}, p_{c1,\text{in}} = 0.6282 \text{ MPa}, T_{c2,\text{in}} = 12.0 \text{ }^\circ\text{C}$$

Fig. 15 Outlet temperature varied with inlet pressure of hot fluid for $\dot{m}_h = 0.0462 \text{ kg}$

s^{-1} , $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{V}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}$, $p_{c1,in} = 0.6282 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^\circ\text{C}$

Fig. 16 Heat transfer rate varied with inlet pressure of cold fluid 1 for $\dot{m}_h = 0.0462 \text{ kg}$

s^{-1} , $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{v}_{c2} = 1.114 \text{ m}^3 \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^\circ\text{C}$

Fig. 17 Outlet temperature varied with inlet pressure of cold fluid 1 for $\dot{m}_h = 0.0462$

kg s^{-1} , $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{v}_{c2} = 1.114 \text{ m}^{-3} \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $T_{c2,in} = 12.0 \text{ }^{\circ}\text{C}$

Fig. 18 Heat transfer rate varied with inlet temperature of cold fluid 2 (air) for $\dot{m}_h = 0.0462 \text{ kg s}^{-1}$, $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{v}_{c2} = 1.114 \text{ m}^{-3} \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$

Fig. 19 Outlet temperature varied with inlet temperature of cold fluid 2 (air) for $\dot{m}_h = 0.0462 \text{ kg s}^{-1}$, $\dot{m}_{c1} = 0.031 \text{ kg s}^{-1}$, $\dot{v}_{c2} = 1.114 \text{ m}^{-3} \text{ s}^{-1}$, $p_{h,in} = 0.8207 \text{ MPa}$, $p_{c1,in} = 0.6282 \text{ MPa}$