Theoretical Analysis Of A Finned-Tube Condenser For A Residential Air-Conditioner Using R-161

Mohammmad Amin S, S G Taji, Dr. A S Padalkar
* Modern College of Engineering, Shivajinagar, Pune
# Assistant Professor, Modern College of Engineering, Shivajinagar, Pune
$ Principal, Flora Institute of Technology, Pune

Abstract

Current residential air-conditioners use hydrochlorofluorocarbon refrigerant, R-22, as a working fluid. In accordance with the Montreal Protocol, it will be banned in coming few year. Refrigerant R-161 is one of the primary candidates to replace refrigerant R-22 in residential air-conditioning applications. In the first part of this study, the design methodology for finned tube heat exchanger is presented. In later, a computational model that determines the COP of an air-conditioning system for various operating conditions and geometric configurations of the condenser is used. In addition, a methodology is detailed for optimizing the condenser design using the COP of the system.

1. Introduction

Due to impending ban on R-22, several alternatives, including binary and ternary blends of HFCs, as well as propane, are being considered as potential R-22 replacement fluids. One very promising replacement, from the viewpoint of zero ODP, is the R-161. Besides the basic characteristics such as thermal properties and flammability, very little heat transfer and pressure drop data for R-161 is available. Yet, knowledge of the performance characteristics of air-cooled refrigerant heat exchangers with alternative refrigerants is of practical importance in designing air-cooled heat exchangers required in air-conditioning equipment. Therefore, more knowledge of the two-phase flow heat transfer and pressure drops that occur in refrigerant R-161 heat exchangers is needed.

2. Literature Survey

Many researchers work on the design of air conditioners for residential application using alternative refrigerant. Yingwen Wu et al. investigate the feasibility of R-161 applied in residential air conditioner, the thermodynamic performance and comprehensive theoretical thermodynamic cycle of R-161, R-22 and R-290 under various air-conditioner operating condition. Furthermore, the cooling and heating performance of R-161 and R-22 under various operating condition was investigated experimentally in a 3.5kW residential heat pump air conditioner. Property and thermodynamic cycle comparison showed that R-161 has better thermodynamic performance than R-290, the rated cooling and heating capacity is lower than R-22 but higher than R-290, the rated cooling and heating COP is higher than both R-22 and R-290. The experimental rated cooling capacity reduced 7.6% and rated cooling EER increased 6.1%, rated heating capacity reduced 6.8% and rated heating COP increased 4.7%, refrigerant optimized charge reduced 43% compared to R-22 system, theoretical and experimental test revealed that R161 has lower discharge temperature than R-22 system

3. Air-Conditioning System and Component Modeling

Of the three basic refrigeration cycles (vapor compression, absorption, and thermo-electric), the cycle typically used in the HVAC industry is the vapor compression cycle. The working fluid for the system in this study is refrigerant R-161.

The vapor compression refrigeration cycle modeled for this study is shown in Figure 3.1. As the figure shows, low pressure, superheated refrigerant vapor from the evaporator enters the compressor (State 1) and leaves as high pressure, superheated vapor (State 2). This vapor enters the condenser where heat is rejected to outdoor air that is forced over the condenser coils. Next the refrigerant vapor is cooled to the saturation temperature (State 2b), and then cooled to below the saturation point until only sub-cooled liquid is present (State 3).

The high pressure liquid is then forced through the expansion valve into the evaporator (State 4). The refrigerant then absorbs heat from warm indoor air that is blown over the evaporator coils. The refrigerant is completely evaporated (State 4a) and heated above the saturation temperature before entering the compressor (State 1). The indoor air is cooled and dehumidified as it flows over the evaporator and returned to the living space.

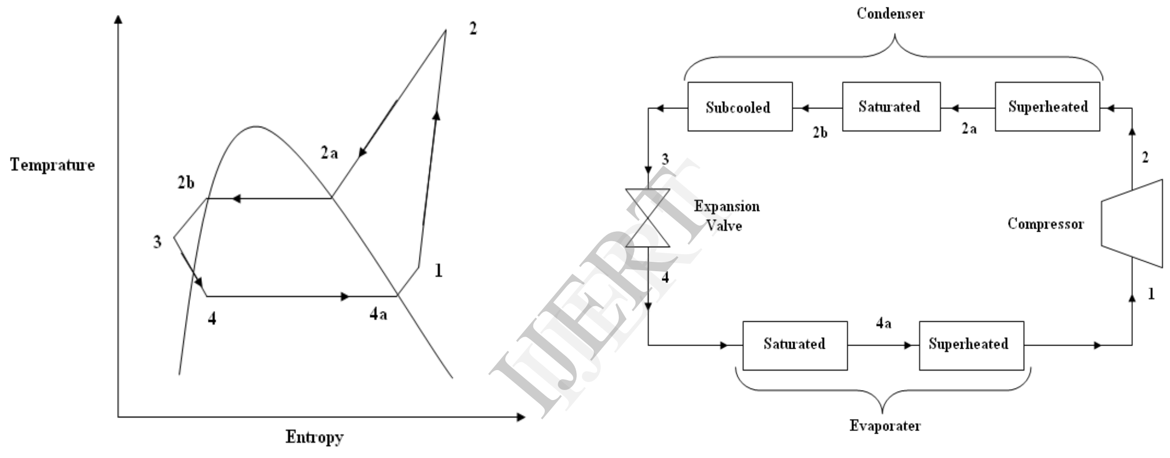

Figure 3.1: Actual Vapor-Compression Refrigeration Cycle

3.1 Compressor

The purpose of the compressor is to increase the working pressure of the refrigerant. The compressor is the major energy-consuming component of the refrigeration system. For this study scroll type positive displacement compressors, which dominate the residential air-conditioning industry, is used.

The amount of specific work (work per unit mass of refrigerant) done by an ideal compressor can be expressed with the following

$$w_{s,com} = (h_{2s} - h_1) \tag{3.1}$$

where h is the refrigerant enthalpy. For a non-ideal compressor, the actual amount of work done depends on the efficiency,

$$w_{a,com} = \frac{w_{s,com}}{\eta_c} = (h_2 - h_1) \tag{3.2}$$

3.2 Condenser

The condenser is a heat exchanger that rejects heat from the refrigerant to the outside air. Although there are many configurations of heat exchangers, finned-tube heat exchangers are the type most commonly used for residential air conditioning applications

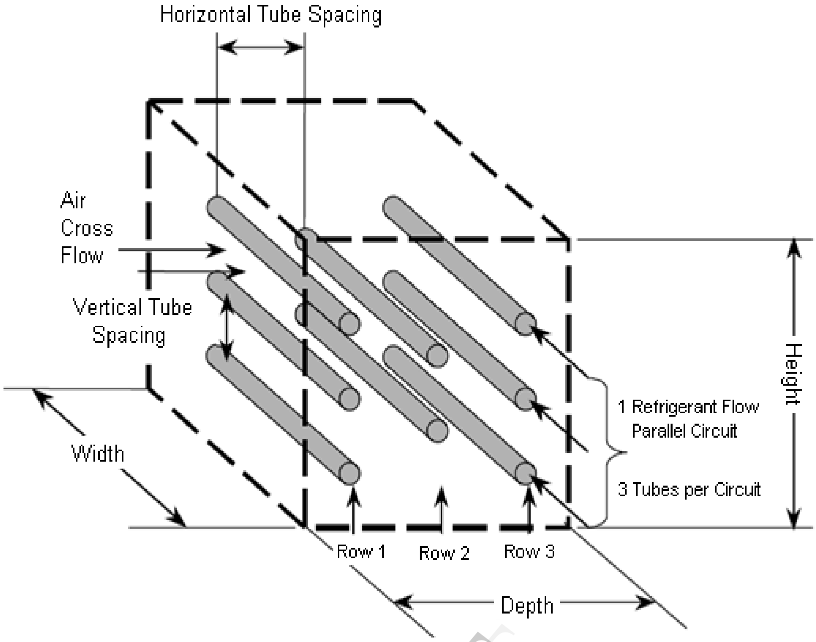

Figure 3.2: Typical Cross Flow Heat Exchanger (fins not displayed)

The amount of heat per unit mass of refrigerant rejected from each section can be expressed as the difference between the refrigerant enthalpy at the inlet and at the outlet of each section.

$$q_{con,sh} = h_2 - h_{2a} \tag{3.3}$$

$$q_{con,sat} = h_{2a} - h_{2b} \tag{3.4}$$

$$q_{con,sc} = h_{2b} - h_3 \tag{3.5}$$

The total heat rejected from the hot fluid, which in this case is the refrigerant, to the cold fluid, which is the air, is dependent on the heat exchanger effectiveness and the heat capacity of each fluid

$$\dot{Q} = \varepsilon\, C_{min}\left(T_{h,i} - T_{c,i}\right) \tag{3.6}$$

The heat capacity C is expressed as

$$C = \dot{m}c_p \tag{3.7}$$

The heat exchanger effectiveness discussed earlier in this chapter is the ratio of the actual amount of heat transferred to the maximum possible amount of heat transferred.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{3.8}$$

For a cross flow heat exchanger with both fluids unmixed, the effectiveness can be related to the number of transfer units (NTU) with the following expression

$$\varepsilon = 1 - \left\{\left(\frac{1}{C_r}\right)(NTU)^{0.22}\left[exp(-C_r(NTU)^{0.78}) - 1\right]\right\} \tag{3.9}$$

The NTU is a function of the overall heat transfer coefficient, U, and is defined as

$$NTU = \frac{UA}{C_{min}} \tag{3.10}$$

3.3 Condenser Fan

Natural convection is not sufficient to attain the heat transfer rate required on the airside of the condenser used in a reasonably sized residential air-conditioning system. Therefore a fan must be employed to maintain the airflow at a sufficient velocity. The power required by the fan is directly related to the air-side pressure drop across the condenser and to the velocity of air across the condenser

$$\dot{W}_{f,con} = \frac{V_{a,con}\,\Delta P_{a,con}\,A_{fr,con}}{\eta_{fan,con}} \tag{3.11}$$

3.4 Expansion Valve

The expansion valve is used to control the refrigerant flow through the system under normal operating conditions, the expansion valve opens and closes in order to maintain a fixed amount of superheat in the exit of the evaporator. In this study, the superheat will be maintained at the typical 6^0C. The energy equation shows that the enthalpy is constant across the expansion valve.

$$h_3 = h_4 \tag{3.12}$$

3.5 Evaporator

The purpose of the evaporator is to transfer heat from the room air in order to lower its temperature and humidity. Because the refrigerant enters the evaporator as a liquid-vapor mixture, it is only divided into saturated and superheated sections. No sub-cooled section is necessary. The total enthalpy change of the air is thus the sum of the enthalpy change due to the decrease in temperature (sensible heat), and the enthalpy change due to condensation (latent heat). The effective specific heat can thus be expressed in terms of the specific heat for dry air only,

$$c_{p,eff} = c_p + \left(\frac{0.25\ \Delta h_{sens}}{0.75\ \Delta T}\right) = 1.33c_p \tag{3.13}$$

3.6 Evaporator Fan

The power required by the evaporator fan depends on the losses in these ducts and can vary from configuration to configuration. Here, the default power requirement used by the Air-conditioning and Refrigeration Institute (ARI) of 365 Watts per 27 m^3/minute of air will be used.

3.7 Refrigerant Mass Inventory

Since the compressor contains only vapor, the mass of refrigerant in the compressor is also neglected. Therefore the total mass of the system includes the mass of refrigerant in the sub-cooled, saturated, and superheated portions of the condenser, and in the saturated and superheated portions of the evaporator.

$$m_{sat,evap} = \frac{A_{ci} L_{sat,evap}}{(1-x_i)(v_v - v_l)} ln\left[\left(\frac{v_v}{x_i(v_v - v_l) + v_l}\right)\right] \tag{3.14}$$

The mass of refrigerant in the superheated portions of the condenser and evaporator are expressed simply as

$$m_{con.sh} = \rho_v\, A_{ci}\, L_{con,sh} \tag{3.15}$$

$$m_{evap,sh} = \rho_v\, A_{ci}\, L_{evap,sh} \tag{3.16}$$

Finally, the mass of refrigerant in the sub-cooled section of the condenser is expressed as

$$m_{con.sc} = \rho_v\, A_{ci}\, L_{con,sc} \tag{3.17}$$

4. Design and Optimization of Operating Parameters

The performance of air conditioning systems is highly dependent on operating conditions and parameters. To determine the effects of the various operating parameters on the COP, a typical evaporator and condenser coil pair is arbitrarily selected for the "base configuration". All of the characteristics of the condenser are specified, and all but the frontal area of the evaporator are specified

Table 4.1: Base Case Condenser and Evaporator Characteristics

Dimension	Condenser	Evaporator
Tube Spacing (mm x mm)	32 x 28	26 x 16
Tube inner diameter (in)	0.349	0.349
Tube outer diameter (in)	0.375 (3/8")	0.375 (3/8")
Height (mm)	76	46
Finned width (mm)	91	N/A
Fin pitch (fin/in)	12	12
Number of rows	3	4
Number of circuits	12	9
Number of tubes per circuit	2	2

It is determined that systems with between 6^0C and 9^0C degrees sub-cool in the condenser and air flowing over the condenser with velocities ranging from 1.8 m/s and 3.6 m/s will yield the optimum COP for the base configuration investigated in this study.

5. Design and Optimization of Geometric Parameters of Condenser

Figure 5.1 shows the effect of the air velocity on the COP for varying numbers of rows with optimum sub-cool at 35^0C ambient temperature. According to the figure, for much of the range of air velocities shown, the optimum COP occurs for configurations utilizing 3 rows of tubes. The figure also shows that as the number of rows decreases, the optimum air velocity increases. This trend is summarized in Table 5.1, which shows the optimum operating conditions for each row configuration.

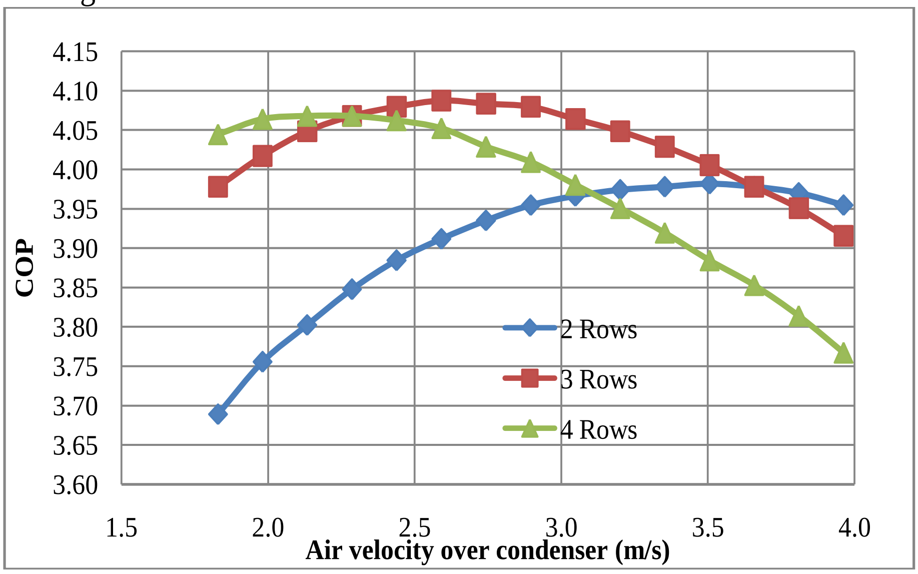

Figure 5.1: Effect of Air Velocity over Condenser on COP for Various Numbers of Rows at Optimum Sub-Cool with Fixed Condenser Frontal Area

Table 5.1: Optimum Operating Conditions for Varying Number of Rows with Fixed Condenser Frontal Area

Number of Rows	COP	Cost Factor	Air Velocity (m/s)	Degrees Sub-cool at 35°C (°C)
2	3.98	0.75	3.35	7
3	4.09	1.00	2.60	8
4	4.07	1.32	2.13	7

Figure 5.2 shows the effect of air velocity on the COP for varying fin pitch with optimum sub-cool at 35^{0}C ambient temperature. As the figure shows, varying the fin pitch has a small affect on the optimum COP when keeping the frontal area of the condenser fixed (optimums range from 4.00 to 4.10). According to the figure, the recommended range of operation is between air velocities of 2.4 m/s and 3.4 m/s. The optimum air velocity increases from 2.4 m/s to 3.4 m/s as the fin pitch decreases from 14 fins per inch to 8 fins per inch.

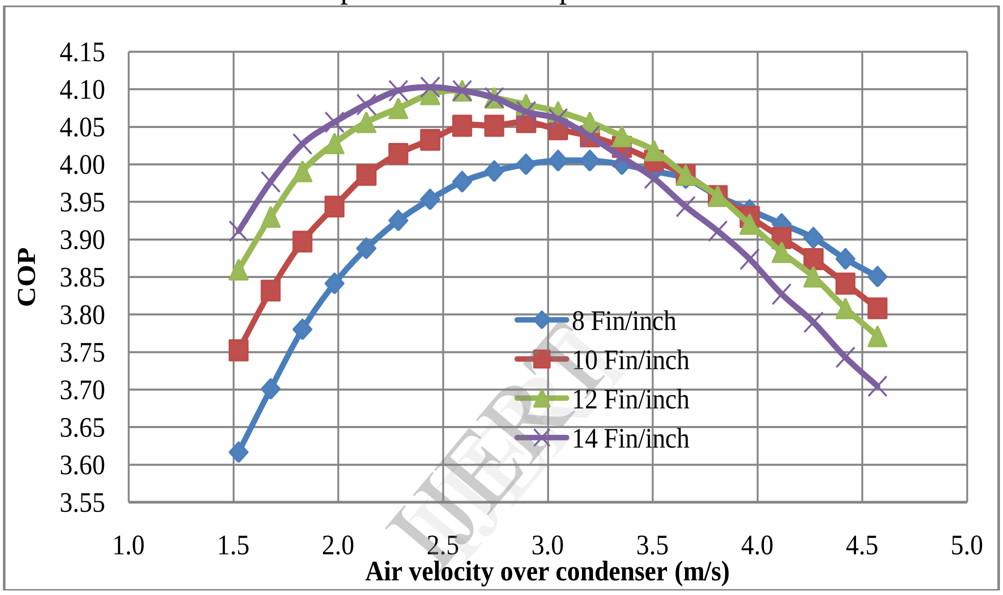

Figure 5.2: Effect of Air Velocity on COP for Varying Fin Pitch with Optimum Sub-Cool for Fixed Condenser Frontal Area

Table 5.2: Optimum Operating Conditions and Cost Factor for Varying Fin Pitch with Fixed Frontal Area

Fin Pitch	Optimum Sub-cool at 35^{0}C ambient Temperature (°C)	Optimum Air Velocity (m/s)	Optimum COP	Cost Factor
8	8	3.2	4.00	0.78
10	8	2.9	4.05	0.89
12	8	2.6	4.09	1.00
14	8	2.4	4.10	1.10

Figure 5.3 shows the effect of the air velocity on the COP for various tube diameters at optimum sub-cool. According to the figure, the absolute maximum COP is 4.11 and occurs at a tube diameter of 1/2”. The COP increases by approximately 5.4 % from 3.88 to 4.09 as the tube diameter is increased from 5/16” to 3/8”. The COP then increases by only 0.5% from 4.09 to 4.11 when the tube diameter increases from 3/8” to 1/2”. When the diameter is further increased from 1/2” to 5/8”, the optimum COP decreases by only 2.8 % from 4.11 to 4.00. These results, along with the optimum operating conditions and cost factors for varying tube diameters, are shown in Table 5.3.

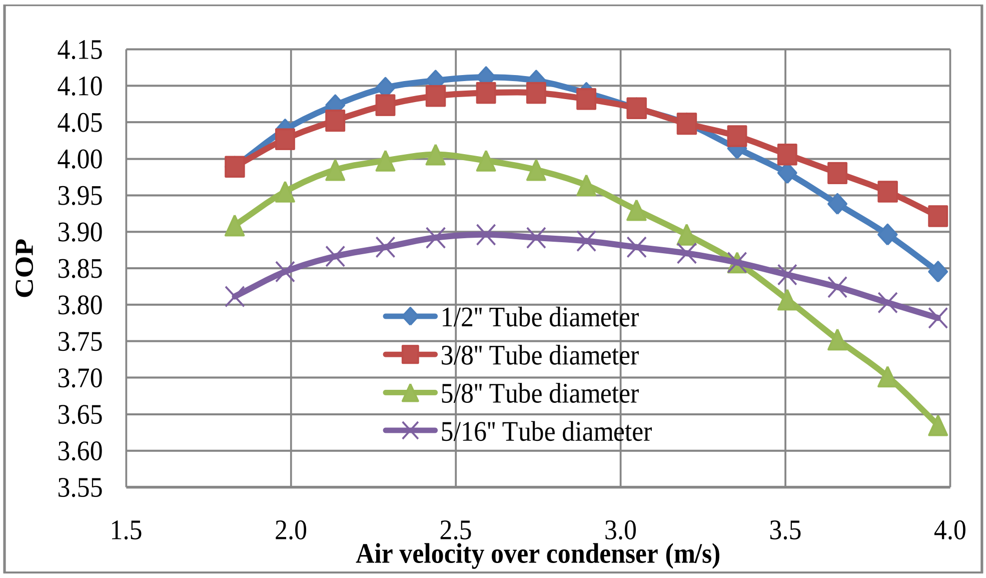

Figure 5.3: Effect of Air Velocity on COP for Varying Tube Diameter at Optimum Sub-Cool for Fixed Condenser Frontal Area

Table 5.3: Optimum Operating Conditions and Cost Factor for Varying Tube Diameters with Fixed Frontal Area

Tube Diameter (in)	Optimum Sub-cool at 35^0C ambient Temperature (^{0}C)	Optimum Air Velocity (m/s)	Optimum COP	Cost Factor
5/16”	8	2.6	3.88	0.92
3/8”	8	2.6	4.09	1.00
1/2”	8	2.6	4.11	1.20
5/8”	8	2.4	4.00	1.59

6. Conclusions and Result Discussion

The specific conclusions drawn from this study are as follows

1. Condenser tubes of smaller diameter enhance performance.
2. When packaging and space constraints are not present; the condenser configuration with the largest frontal area possible yields the best system performance.
3. When typical volume and space constraints are imposed; condensers employing 3 rows of tubes yield the best performance. However, increasing the number of rows to 4 actually increases the material cost of the coil and decreases the system performance when space constraints are imposed.
4. For all geometric configurations investigated, a refrigerant charge producing between 6 and 8 degrees sub-cool at 35^0C ambient temperature produces the optimum performance.
5. For all geometric configurations investigated the optimum velocity of air flow over the condenser coil ranges from roughly 1.8 m /s and 3.6 m /s.
6. If the material cost of the condenser is to be reduced; decreasing the fin pitch from the base configuration value of 12 fins per inch to 10 fins per inch produces a smaller increase in operating cost than decreasing either the number of rows or the frontal area.
7. If the cost of materials is allowed to increase by a specified amount, increasing the frontal area produces the largest reduction in the operating cost. However, increasing the number of rows or the fin pitch actually increases the operating cost for the base configuration.

Acknowledgement

I take immense pleasure in thanking my friend A R Kadam for his generous help in successful completion of my work.

Nomenclature

c_p - Specific heat at constant pressure
$c_{p,eff}$ - Effective specific heat at constant pressure
$c_{p,l}$ - Specific heat of fluid in the liquid phase
C_{min} - Minimum heat capacity between that of the air and the refrigerant
C_{max} - Maximum heat capacity between that of the air and the refrigerant
Cr - Ratio of the minimum heat capacity to the maximum heat capacity
h_1 - Specific enthalpy of refrigerant entering the compressor
h_2 - Actual specific enthalpy of refrigerant exiting the compressor
h_{2s} - Ideal specific enthalpy of refrigerant exiting the compressor
h_{2a} - Specific enthalpy of refrigerant exiting the superheated portion of the condenser
h_{2b} - Specific enthalpy of refrigerant entering the sub-cooled portion of the condenser
h_3 - Specific enthalpy of refrigerant entering the expansion valve
h_4 - Specific enthalpy of refrigerant exiting the expansion valve
$\dot{m}_{a,sat}$ - Mass of flow rate of air f owing over the saturated portion of the condenser
$\dot{m}_{a,tot}$ - Total mass flow rate of air flowing over the condenser
$\dot{m}_{air}$ - Mass flow rate of air flowing over heat exchanger
$m_{con,sat}$ - Mass of refrigerant in the saturated portion of the condenser
$m_{con,sc}$ - Mass of refrigerant in the sub-cooled portion of the condenser
$m_{con,sh}$ - Mass of refrigerant in the superheated portion of the condenser
$m_{evap,sat}$ - Mass of refrigerant in the saturated portion of the evaporator
$m_{evap,sh}$ - mass of refrigerant in the superheated portion of the evaporator

References

[1] “Handbook for the Montreal Protocol on Substances that Deplete the Ozone Layer”, Seventh edition (2006), pp 30, 33
[2] Yingwen Wu, Xiangfei Liang, Xiaoping Tu, Rong Zhuang, “Study of R-161 Refrigerant for Residential Air-conditioning Applications”, International Refrigeration and Air Conditioning Conference at Purdue, July 16-19, 2012
[3] S. F. Pearson, “A More Efficient Halocarbon Substitute for R-22”, Star Refrigeration Ltd., Thornliebank Industrial Estate, Glasgow, 6th April 2000
[4] Incropera, Dewitt, Bergaman, Lavine, “Fundamentals of Heat and Mass Tranfer”, John Wiley, 6e, pp 669-705
[5] T Kuppan, “Heat Exchanger Design Handbook”, Marcel Dekker, Inc. pp 192-199