Mechanical and Energy Engineering

Prediction of Heat Transfer Coefficient and Pressure Drop in Wire Heat Exchanger Working with R-134a and R-600a

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ABSTRACT

An experimental and theoretical works were carried out to model the wire condenser in the domestic refrigerator by calculating the heat transfer coefficient and pressure drop and finding the optimum performance. The two methods were used for calculation, zone method, and an integral method. The work was conducted by using two wire condensers with equal length but different in tube diameters, two refrigerants, R-134a and R-600a, and two different compressors matching the refrigerant type. In the experimental work, the optimum charge was found for the refrigerator according to ASHRAE recommendation. Then, the tests were done at 32°C ambient temperature in a closed room with dimension (2m*2m*3m). The results showed that the average heat transfer coefficient for the R-600a was higher than the R-134a, so the length of the wire tube was longer with R-134a than R-600a. The pressure drop for the smaller tube diameter was higher than the other tube. The second law thermodynamic efficiency was higher for R-600a, which reached 41%. The entropy generation minimization analysis showed that the R-600a refrigerant type and smaller tube diameter are approached the optimum point.

Keywords: wire condenser, heat transfer coefficient, pressure drop.

R-134a و R-600a

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1. INTRODUCTION

The static condenser is used due to its low cost and simple construction. Also, it’s performance is affected by the environment and the other components of the system. The study of wire condenser is very important, and clarifying its thermal behavior is the key to improve, especially for the saturated region. Theoretical and experimental work will be used to calculate the heat transfer coefficient and pressure drop in the condenser in three regimes (superheated, saturated, subcooled). Also, it shows the different performances of the used different refrigerants and different tube diameters of condenser, to optimize the condenser required to remove heat from refrigerant to air. This will be done by finding the flow pattern type and selecting the governing equations to calculate the heat transfer coefficient and the pressure drop in a detailed manner as well as to calculate the length of the heat exchangers, but few were directed to the static condenser as follows:

Ragazzi, and Pedersen, 1991 developed a computer simulation modeling to optimize the air-cooled condenser with the refrigerants R-12 and R-134a. The study used two methods to calculate the heat transfer coefficient and pressure drop in the condenser. The first method was fixed length, and the second method was fixed quality. This method works well for two-phase only. The result depicted that the refrigerant R-12 had more efficiency than refrigerant R-134a, but the refrigerant R-134a showed less damage to the ozone layer. El Hajal, et al., 2003 performed an experimental analysis work for condensation in horizontal tubes depending on the model of a two-phase flow pattern map. The experimental work was done with different refrigerants and for the following range of limitations: mass velocities from 24 to 1022 kg/m²·sec, tube internal diameters from 3.1 to 21.4 mm and vapor qualities from 0.03 to 0.97. The results displayed the maps of flow patterns region of all these refrigerants and determination of transition regions of flow patterns. Tanda, and Tagliafico, 1997 presented an experimental work to predict the natural convection and radiation heat despite from the external surface of the vertical wire-and-tube condenser and using water as a refrigerant. The study found the effects of the most important geometric and operating parameters, like the overall height of the exchanger, the spacing-to-diameter ratios of tubes and wires, and the mean tube to air temperature difference. Also, it was found that the radiation heat transfer represented 15% of the total heat transfer. Bansal, and Chin, 2003 carried out modeling and experimental study for the wire-and-tube condenser. A simulation model was developed using the finite element and variable conductance approach with a combination of thermodynamic correlations. The results showed that the outer heat transfer resistance contributed to about 80 and 83-95% of the total heat transfer for a single and two-phase flow, respectively and the heat transfer mode for wire-condenser was by convection, which contributed up to 65% of the total heat transfer. Dagilis, and Hofmanas, 2012 carried out experimental and numerical investigations to determine the influence of surrounding space around the condenser of a household refrigerator on the heat transfer efficiency. The study decision was a better performance done when more space for the condenser. In the experimental work, the condensing temperature was fixed at 40.3°C, but when the condenser was bent to avoid the heat influence from the compressor shell, the condensing temperature reduced to 36.7°C. The results indicated that the external heat transfer coefficient could rise by 14% in the case when the condenser was fully free and by 9% if the space between the condenser and wall-room was enlarged by 0.3 m. Heo, et al., 2012 presented an experimental work done to study the influence of vertical and horizontal pitches on the natural convection of two vertically staggered cylinders for both laminar and turbulent flows. The experimental work was conducted by varying the vertical and horizontal pitch-to-diameter ratios using a copper electroplating system, and the numerical simulations were performed using the FLUENT program. The results displayed that the heat transfer of the lower cylinder was similar to the single cylinder and demonstrated the effects of preheating, velocity, and side-flow from the lower cylinder on the upper cylinder. These effects weakened with increasing
horizontal pitch-to-diameter ratio. **Melo, and Hermes, 2009** conducted an analysis and experimental work to estimate the heat transfer coefficient between the external surfaces of natural draft wire-and-tube condensers and the surrounding air. The Buckingham-Pi theorem was used to derive a dimensionless multiplier in terms of the working temperatures and heat exchanger geometry, and this correlation predicted 79% of the measured data within an error band of +5%. **Lee, and Son, 2010** experimentally tested a horizontal double pipe heat exchanger. The work used the refrigerants R-290, R-600a, R-22 and R-134a and different inner diameters 10.07, 7.73, 6.54, and 5.8 mm, and mass flux varying from 35.5 to 210.4 kg/m²/sec. The results illustrated that the average condensation heat transfer coefficients of R-600a and R-290 were higher than those of R-134a and R-22, and the pressure drops of the four refrigerants were R-600a>R-290>R-134a > R-22. Also, the heat transfer coefficient of refrigerant was higher at 5.8 mm tube diameter and lower at 10.07 mm tube diameter for all the above refrigerants. Most of the researches don’t cover all the details of pressure drop or the equations that were used to determine the pressure drop. The current work will be done to find a thermodynamic analysis for the wire condenser for refrigerant side and airside depending on the entropy generation minimization theory for A. Bejan.

### 2. EXPERIMENTAL WORK

#### 2.1 Domestic Refrigerator

The domestic refrigerator used in the experimental work is shown in **Fig. 1**. It consists of a cabinet, a reciprocating compressor, a wire condenser, a roll bond evaporator, and a capillary tube. The refrigerator is made of pressed steel with paint and waterproof outside shell. The specifications of the main components are as follows; the two hermetic reciprocating compressors working with different refrigerant; the first compressor is working with R-134a refrigerant, with flow capacity is 8.1cm³, and cooling capacity is 210W, and the second is working with R-600a, with flow capacity of 11.2cm³, and cooling capacity of 198W. Both the compressors are working according to **ASHRAE, 1997**. The lengths of condensers were used in the experimental test is 10.25m with different tube diameters (4.76 mm and 6.35 mm). The evaporator is a roll-bond. The optimum inner diameter and length of the capillary tube are (0.66 mm) and (1.17 m), respectively. The cabinet size is 10 cubic feet. The back of the cabinet was covered with aluminum foil to prevent the heat transfer to the cabinet by radiation from the wire condenser. Also, the space between the frame and the door of cabinet was covered with a magnetic gasket to prevent heat loss and air infiltration.

![Diagram](image)

**Figure 1.** The diagram of the system and instrumentation.
2.2 Instrumentation
Several measuring instruments have been used with domestic refrigerator. Thermists of negative
temperature coefficient (NTC) type are used to measure the temperature. The pressure gauges were
used to indicate the pressure at high and low-pressure sides of the refrigerant circuit. Also, four
pressure transducers were used to measure the pressure across the compression cycle of the domestic
refrigerator. A turbine refrigerant volume flow meter made from stainless steel was used to measure
the flow rate of refrigerant. Digital power clamp meter was used to measure the current; the voltage
supplied to the refrigerator, and the power consumption per hour and day. The electronic refrigerant
scale was used to measure the refrigerant mass inventory of the domestic refrigerator.
All the measuring devices are connected with interface unit Data Acquisition DAQ (Arduino MEGA 2560).
The interface Arduino connected to the computer control system (Laptop) to view the data using
LABVIEW software, which enables the communication between the systems under study. The
refrigerants were charged using an electronic scale to measure the required amount of refrigerant
mass.

3. STATIC CONDENSER MODELING
The suitable and accurate designs of the wire condenser help the system to work in a good situation
and decrease the power consumption, but on another side, it means an acceptable cost. The following
assumptions are considered in the modeling of the wire-on-tube condenser: steady-state case and
one-dimensional analysis, and cross-flow heat exchanger, unmixed refrigerant side and mixed
airside. The bends affect heat transfer. The saturated (two-phase) region for the refrigerant is
considered homogenous mixture. Properties of the refrigerants are uniform thermodynamically.
There is no change in the temperature and pressure of the refrigerant with the radius of the tube.
Muller-Stenhagan, and Heck, 1986 correlations were used to calculate the fraction pressure drop in
the two-phase flow. Domanski, and Hermes, 2008 correlations were be used to calculate the bends
pressure drop in the two-phase flow. The thermodynamic model for a wire condenser is done
according to the two methods of calculation: Zones method and Integral method.

3.1 Zones Method
This method is done by dividing the condenser into the regions according to the process and phase
change, which is a single or two-phase flow. Also, the suitable equations are used to calculate the
heat transfer coefficient and pressure drop for each region. The wire tube condenser is divided into
the refrigerant side and the air side. The refrigerant side has three regions: superheat, two-phase flow,
and subcooled.
3.1.1 The superheat region
It’s the first part of the condenser and the refrigerant in the gas phase. The heat transfer coefficient
can be determined according to Dittus-Boelter correlation from reference Holman, 2010:
\[ Nu = 0.023 * R_g^{0.8} * P_r^{0.3} \]
Where: \( Nu = \frac{\alpha_{superheat} \cdot \Delta T_l}{k_g} \)
The properties are evaluated at the film temperature, \( T_{film} \).
\[ T_{film} = \frac{T_{wall} + T_{bulk}}{2} \]
Where: \( R_g = \frac{G \cdot \Delta T_l}{\mu_g} \)
The heat rejected is given by:
\[ Q_{superheat} = \dot{m}_r \cdot c_p \cdot (T_{in} - T_{cin-sat}) \]
\[ Q_{superheat} = \alpha_{superheat} \cdot A_{st} \cdot \left( T_{in} - T_{wi} \right) \]
Where: \( A_{st} = \pi \cdot D_t \cdot L_{superheat} \)
\( T_{in} = \) The inlet temperature to superheat region.
\( T_{cin-sat} \) = The inlet temperature to saturated region.
\( T_{wi} \) = The temperature of the inner tube wall.

The total pressure drop is the summation of the friction pressure drop, momentum pressure drop, and bending pressure drop as well as the gravity pressure drop for the vertical tube only. The friction pressure drop is calculated by the Darcy equation, **Bruce, and Donald, 2002**:

\[
\Delta P_{friction} = 2 \cdot f_g \cdot \frac{l_{superheat}}{D t_i} \cdot \frac{G^2}{\rho_{in}}
\]

(5)

The friction factors are given by **Dobson, 1994**:

\[
f_g = 0.046 \cdot Re_g^{-0.2}
\]

(6)

The momentum pressure drop is calculated from reference **Traviss, 1972**:

\[
\Delta P_{mom} = -G^2 \cdot \left( \frac{1}{\rho_{out}} - \frac{1}{\rho_{in}} \right)
\]

(7)

The bending pressure drop is calculated from reference **Bruce, and Donald, 2002**:

\[
\Delta P_{bends} = f_g \cdot \frac{G^2}{2 \cdot \rho_{in}} \cdot \frac{l_{bend}}{D t_i} \cdot N_{bends}
\]

(8)

Where: \( l_{bend} = \frac{\pi p_t}{2} \), \( N_{bends} \) is the number of bends, and \( p_t \) is the bend length.

The gravity pressure drop is the change due to the elevations of the discharge tube. This is calculated from hydrostatics:

\[
\Delta P_{gravity} = \rho_g \cdot g \cdot L_{discharge}
\]

(9)

3.1.2 The two-phase region

It’s the second part of the condenser, which covers the largest part of the total area of the condenser. The first step is to find the type of flow pattern to calculate the heat transfer coefficient, the two-phase flow may be stratified, stratified-wavy, annular, intermittent, and mist flow, shown in **Fig. 2**.

![Figure 2. The flow pattern types.](image)

The logarithmic mean void fraction will be used to calculate the flow pattern regions according to, **Hajal, 2003**:

The void fraction is:

\[
\varepsilon = \frac{\varepsilon_h - \varepsilon_{ra}}{\ln(\frac{\varepsilon_h}{\varepsilon_{ra}})}
\]

(10)

The homogeneous void fraction is:

\[
\varepsilon_h = \left[ 1 + \left( \frac{1-x}{x} \right) \left( \frac{\rho_V}{\rho_L} \right) \right]^{-1}
\]

(11)

The Rouhani-Axelsson void fraction is:

\[
\varepsilon_{ra} = \frac{x}{\rho_V} \left( \frac{1}{\rho_V} + \frac{1-x}{\rho_L} \right) + \left[ \frac{1}{\rho_V} + \frac{1-x}{\rho_L} \right] + \left[ \frac{1.18(1-x)\sigma(\rho_l-\rho_v)}{\rho_L^{0.5}} \right]^{-1}
\]

(12)

The liquid film Reynold number is:
\[ Re_L = \frac{4G(1-x)\delta}{(1-\varepsilon)\mu_L} \quad \text{or} \quad \frac{GDt(1-x)}{\mu_L} \]  

Fig. 3 shows the geometrical dimensions of a stratified flow where: \( p_L \) is the stratified perimeter around the bottom of the tube. \( p_V \) is the non-stratified perimeter around the top of the tube. \( h_L \) is the height of the stratified liquid. \( h_i \) is the length of the interface. \( A_L \) and \( A_V \) are the corresponding cross-sectional areas occupied by the liquid and vapor. Four of these dimensions would be found using the internal diameter of the tube.

![Diagram of two-phase flow in a circular tube](image)

**Figure 3.** The geometrical parameters for two-phase flow in a circular tube.

\[ h_{ld} = \frac{h_L}{dt} \quad P_{id} = \frac{p_L}{dt}, \quad A_{ld} = \frac{A_L}{dt}, \quad A_{vd} = \frac{A_V}{dt} \]  

The cross-sectional area occupied by liquid and vapor, \( Hjajl, 2003 \):

\[ A_L = A(1-\varepsilon) \]  \( A_V = A\varepsilon \)

The dimensionless liquid cross-sectional area, \( Hjajl, 2003 \):

\[ A_{ld} = \frac{1}{8} \left[ (2\pi - \theta_{strat}) - \sin(2\pi - \theta_{strat}) \right] \]  

The dimensionless liquid height:

\[ h_{ld} = 0.5 \left( 1 - \cos\left( \frac{2\pi - \theta_{strat}}{2} \right) \right) \]  

The dimensionless perimeter of interface, \( Hjajl, 2003 \):

\[ P_{id} = \sin\left( \frac{2\pi - \theta_{strat}}{2} \right) \]  

The stratified angle around the upper perimeter of the tube:

\[ \theta_{strat} = 2\pi - 2 \left\{ \pi(1-\varepsilon) + \left( \frac{3\pi}{2} \right)^{1/3} \left[ 1 - 2(1-\varepsilon) + (1-\varepsilon)^{1/3} - \varepsilon^{1/3} \right] - \frac{1}{200} (1-\varepsilon) \varepsilon [1 - (1-\varepsilon)] \left[ 1 + 4((1-\varepsilon)^2 + \varepsilon^2) \right] \right\} \]  

The mass velocity of the wavy flow, \( Hjajl, 2003 \):

\[ G_{wavy} = \left\{ \frac{16\Delta^2_{id}A_{id}^{2.5}D_{id}\rho_{LV}}{x^2\pi^2(1-2h_{ld})^2} \right\}^{0.5} \left[ \pi^2 \left( \frac{W_{e}}{Fr} \right)_L^{-1.023} + 1 \right]^{0.5} + 50 - 75e^{-\left( x^2 - 0.97 \right)^2 / x(1-x)} \]  

The mass velocity of the stratified flow, \( Hjajl, 2003 \):

\[ G_{strat} = \left\{ \frac{(226.3)^2A_{ld}^2\rho_{LV}(\rho_{L}-\rho_{V})\mu_{lv}^{0.81}}{x^2(1-x)\pi^3} \right\}^{1/3} + 20x \]  

The vapor quality at the transition from intermittent to annular flow, \( Hjajl, 2003 \):

\[ x_{IA} = \left\{ 0.2914 \left( \frac{\rho_{LV}}{\rho_{L}} \right)^{-1/1.75} \left( \frac{\mu_{LV}}{\mu_{lv}} \right)^{-1/7} + 1 \right\}^{1/2} \]  

The ratio of the liquid Weber number to the liquid Froude number, \( Hjajl, 2003 \):

\[ \left( \frac{W_{e}}{Fr} \right)_L = \frac{9.81 + Dt L^2 \rho_{L}}{\sigma} \]  

The factor \( \xi \), \( Hjajl, 2003 \):

\[ \xi = \left[ 1.138 + 2\log\left( \frac{\pi}{1.5A_{ld}} \right) \right]^{-2} \]  

The mass velocity of the mist flow, \( Hjajl, 2003 \):

\[ G_{mist} = \left\{ \frac{7680A_{id}^2 + 9.81 + Dt L^2 \rho_{LV}}{x^2\pi^2\xi} \left( \frac{Fr}{W_{e}} \right)_L \right\}^{0.5} \]
The mass velocity of the bubbly flow, **Hajal, 2003**:  
\[
G_{\text{bubbly}} = \left( \frac{256\Delta_{\text{p}}^2 \rho_{\text{L}} (\rho_{\text{L}} - \rho_{\text{g}}) + 9.81}{0.3164 (1-x)^{1.75} n_{\text{h}} d_{\text{h}}^{0.25}} \right)^{1/1.75} 
\]

(27)

To identify the flow pattern at a particular value of vapor quality x, the following limitations are to be applied, **Hajal, 2003**:

- Annular flow exists if \( G > G_{\text{wavy}} \), \( G < G_{\text{mist}} \) and \( x > x_{1A} \).
- Intermittent flow exists if \( G > G_{\text{wavy}} \) and \( G < G_{\text{mist}} \) or \( G < G_{\text{bubbly}} \) and \( x < x_{1A} \).
- Stratified-wavy flow exists if \( G_{\text{strat}} < G < G_{\text{wavy}} \).
- Fully stratified flows exist if \( G < G_{\text{strat}} \).
- Mist flow exists if \( G > G_{\text{mist}} \).

Three procedures can calculate the heat transfer coefficient according to the flow pattern type. If the flow pattern is stratified-wavy, then the heat transfer coefficients are calculated from **Hajal, 2003**:

\[
\alpha_{\text{tp}} = \frac{\alpha_{t,\text{tp}}^* \theta + [(2\pi - \theta) \alpha_{t,\text{tp}}^* \theta]}{2 + \pi \alpha_{t,\text{tp}}^* \theta} 
\]

(28)

The upper angle of the tube not wetted by stratified liquid:

\[
\theta = \theta_{\text{strat}} \left[ \left( \frac{G_{\text{wavy}} - G}{G_{\text{wavy}} - G_{\text{strat}}} \right) \right]^{0.5} 
\]

(29)

The convective condensation heat transfer coefficient:

\[
\alpha_c = c * \text{Re}_{t}^n * \text{Pr}_{t}^m * \frac{k_{L}}{\delta_{\text{w}}} * f_i 
\]

(30)

Where: \( c = 0.003 \), \( n = 0.74 \) and \( m = 0.5 \) from **Hajal, 2003**:

\[
f_i = 1 + \left[ u_{\text{w}} \right]^{0.5} \left[ \frac{(\rho_{\text{L}} - \rho_{\text{g}}) + 9.81 \cdot \delta_{\text{w}}^{2}}{\sigma} \right]^{0.25} 
\]

(31)

The liquid film thickness flow for the stratified-wavy **Hajal, 2003**:

\[
\delta_{\text{w}} = \frac{D_{\text{t}} - \left[ D_{\text{t}}^2 - A_{\text{t}} \frac{8}{2\pi - \theta} \right]^{0.5}}{2} 
\]

(32)

The Nusselt film condensing coefficient on the top perimeter of the tube according to:

\[
\alpha_f = 0.728 \left[ \frac{\rho_{\text{L}} (\rho_{\text{L}} - \rho_{\text{g}}) + 9.81 \cdot h_{fg} T_{\text{sat}}}{\mu^{*} D_{\text{c},t} \Delta T} \right]^{0.25} 
\]

(33)

\[
h_{fg} = \frac{J}{kg} \cdot \Delta T = T_{\text{sat}} - T_{\text{w}} 
\]

If the flow is fully stratified, then the heat transfer coefficients calculate from:

\[
\alpha_{\text{tp}} = \frac{\alpha_{t,\text{tp}}^* \theta_{\text{strat}} + [(2\pi - \theta_{\text{strat}}) \alpha_{t,\text{tp}}^* \theta]}{2 + \pi \alpha_{t,\text{tp}}^* \theta} 
\]

(34)

The convective condensation heat transfer coefficient:

\[
\alpha_c = c * \text{Re}_{t}^n * \text{Pr}_{t}^m * \frac{k_{L}}{\delta_{\text{s}}} * f_i 
\]

(35)

Where: \( c = 0.003 \), \( n = 0.74 \) and \( m = 0.5 \) from **Hajal, 2003**:

\[
f_i = 1 + \left[ u_{\text{w}} \right]^{0.5} \left[ \frac{(\rho_{\text{L}} - \rho_{\text{g}}) + 9.81 \cdot \delta_{\text{w}}^{2}}{\sigma} \right]^{0.25} \left( \frac{G}{G_{\text{strat}}} \right) 
\]

(36)

The liquid film thickness flow for the fully-stratified:

\[
\delta_{\text{s}} = \frac{D_{\text{t}} - \left[ D_{\text{t}}^2 - A_{\text{t}} \frac{8}{2\pi - \theta_{\text{strat}}} \right]^{0.5}}{2} 
\]

(37)

The important point is the length of the two-phase area; with an inner diameter of 4.42mm and 3.25mm, so the area is the target. The approximate length can be found from the relation of the energy balance:

\[
Q_{\text{two-phase}} = \dot{m}_{\text{r}} * h_{fg} 
\]

(38)

\[
\alpha_{\text{tp}} * A_{\text{s-tp}} * \Delta T = \dot{m}_{\text{r}} * h_{fg} * \Delta x 
\]

(39)
\[ G = \frac{m_v}{A_c}, \text{ and } A_c = \pi * D t_i^2 / 4 \]

\[ \alpha_{tp} * \pi * D t_i * l * \Delta T = G * \frac{\pi + D t_i^2}{4} * h_{fg} * \Delta x \]

Let \( l = dz \) and \( \Delta x = dx \)

Re-arrangement yield:

\[ \frac{dz}{dx} = \frac{G + h_{fg} * d_i}{4 * \alpha_{tp} + \Delta T} \int dz = \frac{G + h_{fg} * d_i}{4 * \alpha_{tp} + \Delta T} \int dx \]

It is the length of the tube required to cover the change in the quality, and the average heat transfer coefficient can be found from the following:

\[ \frac{1}{\alpha_{tp}} = \frac{1}{x_{i} - x_{e}} \int_{x_{i}}^{x_{e}} \frac{dx}{dz} \] This expression is a function for quality only.

\[ Q_{two-phase} = \alpha_{tp} * A_{i-tp} * (T_R - T_{win}) \]

Where: \( A_{i-tp} = \pi * D t_i * L_{tp} \)

The total pressure drop in the saturated region is the summation of pressure drop due to, friction, momentum. The gravity pressure drop is the horizontal layout of the tubes will be neglected, according to ASHRAE, 1997.

\[ \Delta P_{tp-total} = \Delta P_{tp-friction} + \Delta P_{tp-mom} + \Delta P_{tp-bends} \] (41)

The friction pressure drop is calculated using Müller-Steinhagen and Heck correlation according to ASHRAE, 1997:

\[ \left( \frac{dp}{dz} \right)_{tp-friction} = \Lambda \cdot (1 - x)^{\frac{1}{3}} + \left( \frac{dp}{dz} \right)_{lo} \cdot x^3 \] (42)

\[ \Lambda = \frac{\left( \frac{dp}{dz} \right)_{lo}}{2} + 2 \cdot \left( \frac{dp}{dz} \right)_{g_{lo}} - \left( \frac{dp}{dz} \right)_{lo} \cdot x \] (43)

Where: \( \left( \frac{dp}{dz} \right)_{lo} = f_i \cdot \frac{2 G_{tot}}{D_i \rho_i} \), \( \left( \frac{dp}{dz} \right)_{g_{lo}} = f_g \cdot \frac{2 G_{tot}}{D_i \rho_g} \), \( f = 0.079 * Re^{-0.25} \) and \( Re = \frac{G_{tot} D_i}{\mu} \)

The momentum pressure drop calculated from ASHRAE, 1997:

\[ \frac{dp}{dz_{mom}} = -G^2 \cdot \frac{d}{dz} \left( \frac{x^2}{\rho_g \varepsilon_{zivic}} + \frac{1 - x}{\rho_l (1 - \varepsilon_{zivic})} \right) \] (44)

Where \( \varepsilon \) is the void fraction for Zivic:

\[ \varepsilon_{zivic} = \left[ 1 + \frac{(1-x)}{x} \left( \frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \] (45)

The final form used to calculate the momentum pressure drop is:

\[ \Delta P_{tp-mom} = G^2 - \left\{ \left( \frac{x^2}{\rho_g \varepsilon_{zivic}} + \frac{(1-x)^2}{\rho_l (1 - \varepsilon_{zivic})} \right) - \left( \frac{x^2}{\rho_g \varepsilon_{zivic}} + \frac{(1-x)^2}{\rho_l (1 - \varepsilon_{zivic})} \right) \right\} \] (46)

Bending pressure drop: Domanski, 2008 proposed a new correlation based on the two-phase friction pressure drop correlation for straight tubes by Müller-Steinhagen and Heck Correlation and a multiplier to accounts for the bend curvature.

\[ \Delta P_{tp-bend} = 0.0065 \cdot \left( \frac{G \cdot D \cdot x}{\mu_g} \right)^{0.54} \cdot \left( \frac{1}{x - 1} \right)^{0.21} \cdot \left( \frac{\rho_l}{\rho_g} \right)^{0.34} \cdot \left( \frac{2 R}{d_i} \right)^{-0.67} \cdot \Delta P_{tp-st} \] (47)

The total pressure drop for bends is:

\[ \Delta P_{tp-bends} = \Sigma \Delta P_{tp-bend} \] (48)

3.1.3 The subcooled region:

It's the last part of the condenser and the refrigerant in the liquid phase. The heat transfer coefficient and pressure drop are the same as the superheated region.

3.1.4 Final calculation:

To find the total length for condenser:
\[ L_c = L_{\text{superheat}} + L_{\text{tp}} + L_{\text{subcooled}} \]  

The average heat transfer coefficient is:  
\[ \alpha_{ic} = \frac{(L_{\text{superheat}} \cdot \alpha_{\text{super}}) + (L_{\text{tp}} \cdot \alpha_{\text{avtp}}) + (L_{\text{subcooled}} \cdot \alpha_{\text{subcooled}})}{L_c} \]  

3.1.5 Airside:  
The condenser shape and configuration allow a good understanding of the behaviors of the heat transfer and the pressure drop. Before clarifying the analysis of the condenser airside, the fin efficiency, the finned tube surface effectiveness, the overall heat transfer coefficient, and the $\varepsilon$-NTU method should be determined as follows:  
The actual fin efficiency is calculated using the approximation for fin geometry relation Holman, 2010:  
\[ \eta_f = \frac{\tanh(m_{\text{fin}} \cdot L_{\text{fin}})}{m_{\text{fin}} \cdot L_{\text{fin}}} \]  
Where:  
\[ m_{\text{fin}} = \frac{L_{\text{fin}}}{K_{\text{fin}} D_f} \]  
\( K_{\text{fin}} \) is the thermal conductivity of the fin. \( D_f \) is the diameter of the wire fin.  
The finned tube surface is described using its areas, where:  
\[ A_{\text{tot}} = A_f + A_{\text{tube}} \]  
\[ A_f = \pi \cdot D_f \cdot L_f \cdot \text{No. fins} - A_{\text{contact}} \]  
\[ A_{\text{tube}} = (\pi \cdot D_{\text{to}} \cdot L_c - A_{\text{contact}}) \]  
\[ A_{\text{contact}} = (D_f \cdot D_{\text{to}} \cdot 0.75) \cdot \text{No. legs} \cdot \text{No. fins} \]  
The heat transfer coefficient for horizontal surfaces (tubes) \( \alpha_{c\text{ah}} \) equations, Holman, 2010:  
\[ Ra_h = \frac{9.81 \cdot \beta \cdot (T_{\text{cs}} - T_{\text{amb}}) \cdot D_c^2 \cdot Pr_a}{\nu^2} \]  
If \( [Ra_h < 10^9] \)  
\[ \alpha_{c\text{ah}} = 1.32 \cdot \left[ \frac{T_{\text{cs}} - T_{\text{amb}}}{D_c} \right]^{0.25} \]  
If \( [Ra_h > 10^9] \)  
\[ \alpha_{c\text{ah}} = 1.52 \cdot \left[ \frac{T_{\text{cs}} - T_{\text{amb}}}{D_c} \right]^{0.333} \]  
The heat transfer coefficient for vertical surfaces (fins) is \( \alpha_{c\text{av}} \) equations, Holman, 2010:  
\[ Ra_h = \frac{9.81 \cdot \beta \cdot (T_{\text{cs}} - T_{\text{amb}}) \cdot H_c^2 \cdot Pr_a}{\nu^2} \]  
If \( [Ra_h < 10^9] \)  
\[ \alpha_{c\text{av}} = 1.42 \cdot \left[ \frac{T_{\text{cs}} - T_{\text{amb}}}{H_c} \right]^{0.25} \]  
If \( [Ra_h > 10^9] \)  
\[ \alpha_{c\text{av}} = 1.31 \cdot \left[ \frac{T_{\text{cs}} - T_{\text{amb}}}{H_c} \right]^{0.333} \]  
The radiation heat transfer coefficient\( \alpha_{c\text{rad}} \), Holman, 2010:  
\[ \alpha_{c\text{rad}} = \varepsilon \cdot 5.67 \cdot 10^{-8} \cdot \left[ \frac{(T_{\text{cs}})^4 - (T_{\text{amb}})^4}{T_{\text{cs}} - T_{\text{amb}}} \right] \]  
To calculate the surface efficiency:  
\[ \eta_s = 1 - \frac{A_{\text{fin}}}{A_{\text{tot}}} (1 - \eta_{f\text{in}}) \]  
The heat transfer from the wire condenser, Holman, 2010:  
\[ Q_{\text{fin}} = [\alpha_{c\text{av}} + \alpha_{c\text{rad}}] \cdot A_f \cdot [T_{\text{cs}} - T_{\text{amb}}] \cdot \eta_{f\text{in}} \]  
\[ Q_{\text{tube}} = [\alpha_{c\text{ah}} + \alpha_{c\text{rad}}] \cdot A_{\text{tube}} \cdot [T_{\text{cs}} - T_{\text{amb}}] \]  
\[ Q_{c\text{c\text{a\text{oth}}} = \text{a\text{ta\text{al}}} = Q_{\text{fin}} + Q_{\text{tube}} \]  
\[ R_{oc} = \frac{[\alpha_{c\text{ah}} + \alpha_{c\text{rad}}] \cdot A_{\text{tot}} \cdot \eta_s}{1} \]  
\[ R_{ic} = \frac{1}{\alpha_{ic} \cdot A_i} \]
The overall heat transfer coefficient is determined by the following equation, Holman, 2010:

\[
\frac{1}{UA} = \frac{1}{\eta_s \alpha_{ac, A}} + \frac{\ln \frac{\alpha}{l}}{2\pi \kappa_p L_e} + \frac{1}{\alpha_{ic, A_t}}
\]  

(69)

The effect of fouling inside and outside is neglected because the refrigerator is brand new.

The effectiveness of the wire condenser is calculated by assumption the heat exchanger unmixed-mixed and cross-flow configuration, Holman, 2010:

\[
\varepsilon_{H,E} = 1 - \exp\{- (1/C^*) \cdot [1 - \exp(-NTU \cdot C^*)]\}
\]  

(70)

Where: \( C^* = \frac{c_{min}}{c_{max}} \) and \( C = \dot{m} \cdot c_p \)

Also \( \varepsilon_{H,E} \) can be defined as:

\[
\varepsilon_{H,E} = \frac{Q_{act}}{Q_{max}} \quad \text{with} \quad Q_{max} = \dot{m} \cdot c_{min} \cdot (T_{cl} - T_{co})
\]  

(71)

This leads to:

\[
\varepsilon_{H,E} = \frac{(T_{cl} - T_{co})}{(T_{hi} - T_{cl})} \quad \text{or} \quad NTU = \frac{c_{min}}{U_A}
\]  

(72)

3.2 Integral Method:

The integral method was done according to Ragazzi, and Pedersen, 1991, for the saturated region only by dividing this region to elements and calculate the heat transfer coefficient and pressure drop by integrating. This method was done by integrating the equations that depend on dryness friction of refrigerant from 0.95 to 0.05. In this method, the parameter can be analyzed and find when the flow pattern changes from stratified-wavy flow to fully-stratified flow. Also, the heat transfer coefficient and pressure drop can be found locally according to dryness friction percent. In this method, an integral solution was used to find the heat transfer coefficient. The equations and the procedure are the same as the two-phase region in zones method but with integral according to the dryness friction.

3.3 Exergy Analysis of the Condenser:

The condenser represents the second component in the vapor compression refrigeration system. The condenser is made of iron tube and fins are painted with black color, according to Bejan, 1996:

The energy:

\[
\dot{m}_r \cdot h_4 + \dot{m}_a \cdot h_{a\text{in}} = \dot{m}_r \cdot h_5 + \dot{m}_a \cdot h_{a\text{out}} + Q_{\text{cond losses}}
\]  

(74)

\[
Q_{\text{cond losses}} = \dot{m}_r \cdot (h_4 - h_5) + \dot{m}_a \cdot (h_{a\text{in}} - h_{a\text{out}})
\]  

(75)

The exergy analysis:

\[
ED_{cond} = \left(1 - \frac{T_{amb}}{T_{wall con}}\right) Q_{\text{cond losses}} + \Sigma_{in} ex - \Sigma_{out} ex
\]  

(76)

Where:

\[
\Sigma_{in} ex = \dot{m}_r \cdot \left( ex_4 + \frac{P_{\text{scond}}}{\rho_4} \right) + \dot{m}_a \cdot \left( ex_{a\text{in}} + \frac{P_{\text{ain}}}{\rho_{\text{ain}}} \right)
\]

\[
\Sigma_{out} ex = \dot{m}_r \cdot \left( ex_5 + \frac{P_{\text{scond}}}{\rho_5} \right) + \dot{m}_a \cdot \left( ex_{a\text{out}} + \frac{P_{\text{aout}}}{\rho_{aout}} \right)
\]

\[
ex_4 = (h_4 - h_o) - T_o \cdot (s_4 - s_o)
\]

\[
ex_5 = (h_5 - h_o) - T_o \cdot (s_5 - s_o)
\]

\[
ex_{a\text{in}} = (h_{a\text{in}} - h_o) - T_o \cdot (s_{a\text{in}} - s_o)
\]

\[
ex_{a\text{out}} = (h_{a\text{out}} - h_o) - T_o \cdot (s_{a\text{out}} - s_o)
\]

The final form for the exergy analysis is:

\[
ED_{cond} = \left(1 - \frac{T_o}{T_{wall con}}\right) Q_{\text{losses cond}} + \dot{m}_r \cdot \left[ (ex_4 - ex_5) + \frac{\Delta P_{\text{scond}}}{\rho_4} \right] + \dot{m}_a \cdot (ex_{a\text{in}} - ex_{a\text{out}})
\]  

(77)

\[
\zeta_{\text{cond}} = \frac{ED_{\text{cond}}}{\text{Power total}}
\]  

(78)

The exergy efficiency is given by:

\[
\eta_{\text{cond}} = 1 - \zeta_{\text{cond}}
\]  

(79)
3.4 Optimum Tube Diameter of the Condenser:
The optimum diameter of tube condenser is divided into two cases; the first is internal flow optimization, and the second is external flow optimization.

3.4.1 Internal flow optimization:
The optimum size of the internal tube diameter of the condenser is found according to Entropy-Generation Minimization from Bejan, 2006. This theory is based on the relation between the entropy generation number and the Reynolds number.
The entropy generation number is calculated from:
\[
N_s = \frac{S_{gen}}{S_{gen,min}} = 0.856 \ast \left( \frac{Re}{Re_{opt}} \right)^{-0.8} + 0.144 \ast \left( \frac{Re}{Re_{opt}} \right)^{4.8}
\]  
(80)
The optimum Reynolds number is calculated from:
\[
Re_{opt} = 2.023 \ast Pr^{-0.071} \ast B_o^{0.358}
\]  
(81)
The Parameter B_o calculated from:
\[
B_o = \frac{\dot{m} \ast q}{\mu^{5/2} (k_f \ast T_{sat})^{0.5}}
\]  
(82)
The Reynolds number: \( Re = \frac{4 \ast \dot{m}}{\pi \ast D_t \ast \mu} \)  
(83)

3.4.2 External flow optimization:
The optimum size of the external tube diameter of the condenser is found according to Entropy-Generation Minimization theoretical analysis for external flow around the horizontal cylinder at low Reynolds number from Mahdi, 2018. This theory depends on the relation between the entropy generation number and Reynolds number ratio, which is calculated from:
\[
N_s = \frac{S_{gen}}{S_{gen,min}} = \left( \frac{Re}{Re_{opt}} \right)^{-0.33} + \left( \frac{Re}{Re_{opt}} \right)^{0.216}
\]  
(84)
\[
N_s2 = \frac{S_{gen}}{S_{gen,min}} = \left( \frac{Re}{Re_{opt}} \right)^{-0.385} + \left( \frac{Re}{Re_{opt}} \right)^{0.698}
\]  
(85)
\[
N_s3 = \frac{S_{gen}}{S_{gen,min}} = \left( \frac{Re}{Re_{opt}} \right)^{-0.466} + \left( \frac{Re}{Re_{opt}} \right)^{0.8}
\]  
(86)
\[
N_s4 = \frac{S_{gen}}{S_{gen,min}} = \left( \frac{Re}{Re_{opt}} \right)^{-0.618} + \left( \frac{Re}{Re_{opt}} \right)^{0.8}
\]  
(87)
Equations are used according to the range of Reynolds number (Re):
If \( 0.4 < Re < 4 \) then use equation (84) and the optimum Reynolds number is calculated from:
\[
Re_{opt} = 0.139 \ast \beta_o^{1/0.546}
\]  
(88)
If \( 4 < Re < 40 \) then use equation (85) and the optimum Reynolds number is calculated from:
\[
Re_{opt} = 0.722 \ast \beta_o^{1/1.083}
\]  
(89)
If \( 40 < Re < 4000 \) then use equation (86) and the optimum Reynolds number is calculated from:
\[
Re_{opt} = 1.824 \ast \beta_o^{1/1.266}
\]  
(90)
If \( 4000 < Re < 40000 \) then use equation (87) and the optimum Reynolds number is calculated from:
\[
Re_{opt} = 14.64 \ast \beta_o^{1/1.618}
\]  
(91)
Where \( Re = \frac{\rho_o U_o D_t \mu}{\mu_o} \)
Where \( \beta_o \) is the duty parameter calculated from:
\[
\beta_o = \frac{q}{U_o \ast k_s \ast \mu \ast T_o \ast Pr^{1/3}}
\]  
(92)
The analysis and the optimization of the condenser were carried out using EES software.

5. RESULTS AND DISCUSSION:
The experimental work done for four cases is given in Table (1):
Figs. 4 and 5 show the T-s and P-h diagrams of the vapor compression refrigeration cycle with R-134a as a working fluid for two condensers with different diameters 6.35 mm and 4.76 mm at 32˚C ambient temperature. The compression ratio remained constant, which means that the compression ratio was not affected by the change of condenser tube diameter. The condenser and evaporator pressures and temperatures decrease with the increase in the condenser tube diameter because the subcooled region was increased. The dryness fraction at the evaporator inlet decreases with the increase in the condenser tube diameter, which gives a better cooling effect in the evaporator. The maximum and minimum temperatures of each cycle are shifted up with the decrease in the condenser tube diameter due to the pressure drop, which increases at small diameter.

Table 1. Table of all the tests.

| No. | Diameter (mm) | Refrigerant type | Test No. | Charge (g) | Tevp °C | Time (min) |
|-----|---------------|-----------------|----------|------------|---------|------------|
| 1   | 4.76 (3/16") | R-134a          | 1        | 60g        | -18     | 150 min    |
|     |               |                 | 2        | 70g        | -22     | 135 min (optimum) |
|     |               |                 | 3        | 80g        | -19     | 160 min    |
| 2   | 6.35 (1/4")  | R-134a          | 1        | 80g        | -15     | 120 min    |
|     |               |                 | 2        | 100g       | -22     | 90 min (optimum) |
|     |               |                 | 3        | 120g       | -16     | 130 min    |
| 3   | 4.76 (3/16") | R-600a          | 1        | 20g        | -17     | 120 min    |
|     |               |                 | 2        | 30g        | -22     | 90 min (optimum) |
|     |               |                 | 3        | 40g        | -15     | 130 min    |
| 4   | 6.35 (1/4")  | R-600a          | 1        | 30g        | -15     | 120 min    |
|     |               |                 | 2        | 50g        | -23     | 90 min (optimum) |
|     |               |                 | 3        | 70g        | -18     | 120 min    |

Figure 4. The P-h diagram of vapor compression refrigeration cycle with R-134a and different tube diameter of condensers.

Figure 5. The T-s diagram of vapor compression refrigeration cycle with R-134a and different tube diameter of condensers.
Figs. 6 and 7 depict the T-s and P-h diagrams of the vapor compression refrigeration cycle with R-600a as a working fluid for two condensers with different diameters 6.35 mm and 4.76 mm at 32°C ambient temperature. The compression ratio is not constant, because the change in the tube diameter of condenser affected the flow stream in the tube because the density of R-600a is greater than R-134a and mass inventory for the large tube is higher than the small tube. The condenser and evaporator pressures and temperatures decrease with the increase in the condenser tube diameter because of the same reason given above. The dryness fraction decreases with the increase in the condenser tube diameter, which gives a better cooling effect in the evaporator. The maximum and minimum temperatures of each cycle are also shifted up with the increase in the condenser tube diameter.

Fig. 8 depicts the heat transfer coefficient for the refrigerant side in the superheat region, the two-phase flow region, and subcooled region, in the condenser for four tests. This figure shows that the average heat transfer of the refrigerant R-600a is higher than the refrigerant R-134a for all tests. This happens due to the high turbulence in the refrigerant flow. The heat transfer coefficient in the subcooled is the lowest for all tests, and the heat transfer coefficient of the two-phase region is the highest for all tests because the tube for the two-phase flow is longer than the other regions.

Fig. 9 demonstrates the total pressure drop in the condenser for the superheat region, two-phase flow region, subcooled region, and the total condenser. This figure shows that the pressure drop in the condenser of the refrigerant R-600a is higher than the refrigerant R-134a for all tests because the density of R-600a is higher and the properties for R-600a is different for R134a. Also, the pressure drop in the condenser with 4.76 mm tube diameter is higher than the condenser with 6.35 mm tube diameter for all tests, because the velocity is higher in the small diameter.

Figs. 10, 11, and 12 clarify the pressure drop in the condenser due to the friction, momentum, and bends for all zones. These figures show that the pressure drop in the condenser working with the refrigerant R-600a is higher than the condenser working with refrigerant R-134a for all tests because the density is higher and the properties for R-600a is different for R134a. Also, the pressure drop in the condenser with a 4.76 mm tube diameter is higher than the condenser with 6.35 mm tube diameter for all tests due to the high turbulence of refrigerant flow.

Fig. 13 indicates the exergy efficiency of the condenser for tests 1, 2, 3, and 4 is 93.96%, 92.32%, 88.46%, and 87.14%, respectively, it is concluded that for the same refrigerant, the exergy efficiency of condenser with tube diameter 4.76 mm is smaller than the condenser with tube diameter 6.35 mm because the pressure drop is higher. And, for the same tube diameter of condenser, the exergy efficiency of condenser working with R-600a is smaller than the condenser working with R-134a because the heat losses are higher.

Fig. 14 reveals the C.O.P for all tests compared with C.O.P for cycle Carnot. This figure shows the performance of the system, which usually looks where the Carnot is higher than the actual test three times.

Fig. 15 elucidates the second law thermodynamic efficiency for all tests. This figure shows that the test four (R-600a, 4.76 mm) has higher thermodynamical efficiency than the other.

Fig. 16 reveals the calculated total length of condenser required to remove the heat from the refrigerant to ambient, and show the length of the necessary condenser for all zones. This figure explains that the total length of condenser working with R-134a is greater than the actual length, but the total length of condenser working with R-600a is lower than the actual length, because the heat transfer coefficient of the refrigerant R-600a is greater than the refrigerant R-134a.

Fig. 17 depicts the optimum diameter for the outer conditions of the condenser. All tests are located on the heat dominated region because in the natural convection, the losses due to the friction are very low, and most of these losses are due to the heat. The nearest test on the optimum point is the test (R-134a, 6.35 mm) because the outer area is larger than the other.

Fig. 18 reveals the optimum diameter for the inside tube of the condenser. The condenser of test one
(R-134a, 6.35 mm) is located on the friction dominated region nearly the optimum point, and the condenser of test two (R-134a, 4.76 mm) is located on the friction dominated region, also far away from the optimum point, because the friction losses in the condensers are dominated. Also, this figure shows that test three (R-600a, 6.35 mm) and four (R-600a, 4.76 mm) are located on the heat dominated region apart from the optimum point because the heat losses are dominated. Test four (R-600a, 4.76 mm) is the closed test to the optimum point as referred before in the exergy efficiency section and the best.

Fig. 19 manifests the distribution of heat transfer coefficient with the dryness fraction by the integral method. The heat transfer coefficient of the test (R-600a, 4.76 mm) is greater than other tests, but the heat transfer coefficient of the test (R-134a, 6.35 mm) is lower than other tests, which are identical to the zone method.

Figs. 20, 21, and 22 displays the distribution of the friction, momentum, and total pressure drop in the condenser with dryness fraction. These figures show the pressure drop of the tests with a 4.76 mm tube diameter is higher than the other tests, which also agrees with the total zones method.

Fig. 23 demonstrates the total length of condenser distribution with a dryness fraction. This figure shows that the tests working with R-134a are higher than the other tests due to the difference in the heat transfer coefficient.

Fig. 24 shows the comparison between the two methods of calculating the length: zone method and integral method. There is a different length calculated by two methods. The difference is 10% between the two methods, and the integral method gives the longer condenser.

Fig. 25 shows a comparison between the two methods of calculating the heat transfer coefficient. The zone method produces lower average heat transfer coefficient by 11% than the integral method integral.

Fig. 26 exhibits the total pressure drop difference; also, the integral method is higher than the zone method by 20%.

Fig. 27 indicates that the pressure drop in the experimental tests is higher than the pressure drop calculated in the two methods because the result of the two methods is calculated for the pure refrigerant, but in fact, the refrigerant is not pure. Also, the friction pressure drop in the bends is higher than the pressure drop calculated from the two methods caused by the errors of instrumentation and variable input data.

**Figure 6.** The P-h diagram of vapor compression refrigeration cycle with R-600a and different tube diameter of condensers.

**Figure 7.** The T-s diagram of vapor compression refrigeration cycle with R-600a and different tube diameter of condensers.
Figure 8. Heat transfer coefficient for all zones in condenser with the tests.

Figure 9. The total pressure drop for all zones.

Figure 10. The friction pressure drop for all zones.

Figure 11. The momentum pressure drop for all zones.

Figure 12. The bends pressure drop for all zones.

Figure 13. The exergy efficiency for the condenser.
Figure 14. The C.O.P and C.O.P Carnot with tests.

Figure 15. The second law thermodynamic efficiency.

Figure 16. The length of condenser for all zones.

Figure 17. The optimum air flow of outer tube diameter condenser.

Figure 18. The optimum refrigerant flow of inner tube diameter condenser.

Figure 19. The relation between the heat transfer coefficient and dryness fraction.
Figure 20. The relation between the friction pressure drop and dryness fraction.

Figure 21. The relation between the momentum pressure drop and dryness fraction.

Figure 22. The relation between the total pressure drop and dryness fraction.

Figure 23. The relation between the Length of condenser and dryness fraction.

Figure 24. The comparison between the length calculated by zone method and the length calculated by integral method.

Figure 25. The comparison between the heat transfer coefficients calculated by zone method and integral method.
6. CONCLUSIONS

The following concluding remarks are drawn from the tests:
1. The heat transfer coefficient calculated in the condenser for the test working with the refrigerant R-600a and tube diameter of condenser 4.76 mm is higher than the other tests.
2. The pressure drop of the condenser for the tests two (R-134a, 4.76 mm) and four (R-600a, 4.76 mm) is higher than the other tests.
3. The length of condenser required to remove the heat to the environment for tests working with fluid R-600a is smaller than the tests working with R-134a.
4. The optimum inner diameter of the condenser for the system working with R-134a is 6.35 mm, and the optimum inner diameter of the condenser for the system working with R-600a is 4.76 mm.
5. The optimum outer diameter of the condenser for all tests is the large exterior area.
6. The zone method was found slightly deviated from the integral method which approved that the current model for the zone method had a good agreement with integral method; that means the zone method is easy and reliable.
7. The integral method is more accurate than the zone method by 14%.

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### NOMENCLATURE

| Symbol | Definition | Unit | G          |
|--------|------------|------|------------|
| A      | Area       | m²   | Total mass velocity of liquid and vapor kg/m².s |
| B      | Constant   |      | Gravitational acceleration m/s² |
| B₀     | Parameter  |      | Liquid height m |
| a,b    | Kays & London power coefficient |      | Dimensionless liquid height |
| C      | Thermal capacitance (ṁ *C_p) kW/K |      | Latent heat J/kg |
| C*     | Thermal capacitance ratio Cmin/Cmax |      | Enthalpy at state n (n=1,2,3 …) kJ/kg |
| C_p    | Specific heat at constant pressure kJ/kg.K |      | Thermal conductivity W/m.°C |
| D      | Diameter   | m    | Length of tube m |
| ED     | Exergy destruction | W | Mass flow rate kg/s |
| ex     | Exergy     | J/kg | Nu | Nusselt number --- |
| Fr     | Froude number |      | NTU | Number of transfer unit --- |
| f      | Friction factor |      | p | Pressure kPa |
| f_i    | Interfacial roughness factor |      | P | Power W |
Pr  Prandtl number  ---
p_
 Perimeter of liquid-vapor interface m
p_
 Dimensionless perimeter of interface m
p_L  Wetted perimeter m
Q  Heat transfer rate W
Ra  Rayleigh number ---
Re  Reynolds number ---
r  Radius m
St  Stanton number ---
S  Entropy kJ/kg.K
S_gen  Entropy generation rate W/K
T  Temperature °C
t_fin  Fin thickness m
u  Mean velocity m/s
UA  Overall heat transfer coefficient W/K
W  Work W
We  Weber Number ---
x  Quality or dryness fraction ---

GREEK CHARACTERS
Symbol  Definition  Unit
α  heat transfer coefficient W/m². °C
α
 Two phase local heat transfer coefficient W/m². °C
α_a  Air-side average convective heat transfer coefficient W/m². °C
α_r  Refrigerant side average convective heat transfer coefficient W/m². °C
β  Extend coefficient for air in natural convection l/K
β_o  duty parameter number ---
ε  Emissivity ---
ε  void fraction ---
δ  Liquid film thickness m
η  Efficiency ---
η_fin  Fin efficiency ---
η_s  Air-side surface efficiency ---
ρ  Density kg/m³
σ  Surface tension N/m
σ_B  Stefan-Boltzmann constant ---
θ  The upper angle of the tube not wetted by stratified liquid rad
θ_strat  Stratified angle rad
ζ  Exergy dissipation ---
ξ  Factor ---
Λ  Factor in Müller-Steinhagen and Heck correlation.

SUBSCRIPTS
Symbol  Definition  Definition
a  Air
amb  Ambient
dis  Discharge
g  Gas
go  Gas only
H  Homogeneous
h  Hydraulic
i  In
l  Liquid state
Ld  Dimensionless liquid
Lo  Liquid only
max  Maximum
min  Minimum
mist  Mist flow
o  out
r  Refrigerant
s  Surface
sat  Saturation
sub  Sub-cool
sh  Super-cool
strat  Stratified flow
suc  Suction
tot  Total
tp  Two phase
t  Tube
Π  Second law of thermodynamics
v  Vapor