Design and analysis of heat exchanger (evaporator) type of staggered tube cross-flow as modified evaporator AC-Split

F H Napitupulu¹, Fellix¹, and H V Sihombing¹

¹Mechanical Engineering Department, Faculty of Engineering, Universitas Sumatera Utara, Jl. Dr. Mansur Kampus USU Medan 20155, Indonesia

Email: farelnapatupulu@yahoo.com

Abstract. This study aims to make and analysis a tube group type evaporator design alternately with crossing flow as a result of modification of split PK 1 evaporator AC and to evaporate the working fluid, namely R-22, to environmental so that it also simultaneously cools the room. The evaporator designed has a length of 26 cm, a width of 16 cm and a height of 84.2 cm, with a tube diameter of 6.35 mm, a tube length of 58 cm and as many as 25 passes, consisting of 5 rows and 5 columns with alternating positions, distance between tubes 15 mm vertically and horizontally. With this design, the evaporator reduce the temperature air inlet of 0.08.

1. Introduction
The increase of global environmental problems, efficient energy utilization becomes an ever more urgent target in science and technology. Among diverse elementary techniques to be improved, the heat exchanger is one of the major components common in a wide variety of thermal energy handling processes, such as conversion, transport, consumption and storage [1]. Improvement of heat exchanger performance affects both directly and indirectly the performance of various devices and systems, and it would lead to better utility and industrial energy plants, air-conditioning and water heater, manufacturing processes, transportation systems, and even information devices, all of which should contribute to reduction of emission of greenhouse effect gases [2-4]. One of the methods for improving performance refrigeration is modification evaporative cooling.

Several studies on the applied heat exchanger (evaporator) on air conditioning systems have been found in the literature. Seong-Yeon Yoo et al [5] investigated the heat transfer characteristics for staggered tube banks. This paper analyzes the performance for various tube spacing, tube locations and Reynolds numbers. Haobo Jiang et al [6] simulated and design tools for air-to-refrigerant heat exchanger for improving effectiveness and efficiency. Simulated used a variety of working fluids and correlation of heat transfer and pressure drop. Jader R Barbosa, Claudia melo and Christian hermes [7] studied the heat transfer and pressure drop characteristics of tube-fin evaporators. Liang Pu et al [8] analysis flow and heat transfer characteristic of evaporator using tube shape. It showed the average heat transfer coefficient is approximately 2.2 %, 4.2%, and 11.2 % higher than the circular tube. Enlu wang et al [9] experiment the gas condensation in a multi-row staggered tube bundle heat exchanger. There is a suitable cooling water flow rate with an adequately low wall temperature and an adequately high transfer coefficient and condensate capture rate. It’s showed that the increase in the cooling water temperature, the condensate capture rate decrease, however, the heat transfer coefficient increases under a similar condition.
The objective of the current work is to design and analysis a tube bank of the evaporator with staggered tube cross-flow. The staggered heat exchanger was design based on the cooling capacity needed. It was expected that the staggered heat exchanger would be applied in the air conditioning system.

2. Method

In designing this evaporator, the capability and purpose of the evaporator will be determined first. Evaporator designed to cool the room with a cooling capacity of 1 PK. The cooling fluid or refraction used is R-22 refraction. The designed evaporator is a type of tube banks with a crossing flow and without fins. The reason for choosing the tube banks type where the tube banks type can produce lower air temperatures than the types of open pipes which are generally used in Split AC, can reduce the length of the tubes used, as well as easier maintenance. On the other hand, the consequence of using this type is that it will use more reforestation and more complicated manufacturing. The position of the staggered tube is chosen in this design because it has better heat transfer than the parallel tube position.

![Figure 1](image)

Figure 1 Schematic vapor compression cycle with modification the evaporator

The design is viewed from the ideal cycle of vapor compression, the amount of cooling capacity can be obtained, then the assumptions needed to obtain the dimensions of the evaporator are designed such as ambient air temperature, etc. done to obtain the number of tubes to be used and also the dimensions of the evaporator designed will be obtained.

After knowing enthalpy (i) and entropy (s) at each point on the cycle, then looking for the mass flow rate in the tube is sought by connecting the power formula on the compressor equation (1) with enthalpy changes, cos θ which has been determined by the potential difference, and the current strength has also been determined the calculation is as follows:

$$m_h = \frac{V \times l \times \cos\theta}{(i_2-i_1)k_f/k_g}$$  \hspace{1cm} (1)

To find the value of the heat transfer rate or maximum capacity of the evaporator that will be designed calculated by the equation (2)

$$Q_{e,max} = \dot{m}(i_1 - i_4)$$  \hspace{1cm} (2)

The efficiency, $\eta_f$ determined by the equation (3)

$$\eta_f = \frac{\tanh(mL_c)}{mL_c}$$  \hspace{1cm} (3)
Where, \( m = \frac{h}{kA} \) and convection coefficient \( (h) \), and \( A \) is determined the following equation (4)

\[
A = \frac{n\pi d t}{2}
\]

To calculate the effectiveness of the evaporator, determined by the equation (5)

\[
\varepsilon = \frac{t_{\text{air,in}} - t_{\text{air,out}}}{t_{\text{air,in}} - t_{\text{air}}(T_{\text{ref,in}})}
\]

The capacity of evaporator tube banks can be obtained as follows (6)

\[
Q_e = \frac{\varepsilon}{\eta_f} \times Q_{e,m}\text{ax}
\]

Use the design of the type and dimensions of the model tube heat exchanger tube banks with the settlement of zigzag (staggered) because it is more effective than arranged in parallel. The number of tube rows can be determined by the following equation (7)

\[
Dp > x(n - 0.5) + x + d
\]

Fluid properties (air) are obtained from the air properties table, and the average temperature difference of logarithms is obtained by equation (9)

\[
\Delta T_{lm} = \frac{(T_s - T_{\infty}) - (T_s - T_{\infty})}{\ln ((T_s - T_{\infty})/(T_s - T_{\infty}))}
\]

The air velocity through a heat exchanger is obtained by the fluid flow discharge equation (10)

\[
A_{in}V_{in} = A_{out}V_{out}
\]

The maximum speed of the transverse determined from equation (11)

\[
S_p = \sqrt{15^2 + \left(\frac{15}{2}\right)^2}
\]

The heat transfer value per unit tube length \( (q') \), the heat transfer capacity that is designed and the tube length is adjusted to the length of the blower used, so that it is calculated by the equation (12)

\[
q' = \frac{Q}{h}
\]

3. Results and Discussions

3.1. Design of heat exchanger (evaporator) type of staggered tube cross-flow

Before designing a heat exchanger, the capability of a heat exchanger can be determined in advance. Heat transfer or the maximum capacity of evaporator is 813 W. In designing this evaporator using the ideal cycle of vapor compression. Analysis of the ideal vapor compression cycle, the pressure on the evaporator 60 psi and the pressure on the condenser is 290 psi. So that the enthalpy and entropy can be determined using table thermodynamic refrigerant R22.

The model of heat exchanger designed is a tube banks model with an arrangement of tubes in a staggered because they are more effective than arranged in parallel. The dimension of the tube heat exchanger is \( \frac{1}{4} \) " . The kind of propeller used is horizontal blower with height 58 cm, diameter 9 cm and tube spacing is 1.5 cm.

- A number of tube heat exchanger calculated with equation 8:

\[
Dp > x(n - 0.5) + x + d
9 > 1.5(n - 0.5) + 1.5 + 0.635
n < 5.076
\]

- Properties of air

\[
T_e = 30 ^oC = 303 K
C_p = 1007.12 J/kgK
\nu = 16.1918 \times 10^{-6} m^2/s
k = 26.522 \times 10^{-3} W/mK
Pr = 0.70658
\rho = 1.1514 kg/m^3
T_s = 4 ^oC = 277 K
\]
\( \rho = 1.1514 \text{ kg/m}^3 \)
\( Pr = 0.71298 \)
\( T_f = \frac{4+30}{2} + 273 = 290 \text{ K} \)
\( v = 15 \times 10^{-6} \text{ m}^2/\text{s} \)
\( k = 25.5 \times 10^{-3} \text{ W/mK} \)
\( Pr = 0.7096 \)

- The Different mean of temperature with equation 9:
  \[ \Delta T_{im} = \frac{(4 - 30) - (4 - 24)}{\ln \left( \frac{4 - 30}{4 - 24} \right)} \]
  \[ \Delta T_{im} = 22.868 \]

- The velocity of air in a heat exchanger with equation 10:
  \[ 58 \times 8.5 \times V_{in} = 58 \times 5.5 \times 3 \]
  \[ V_{in} = 1.941 \text{ m/s} \]

- Velocity transversal with equation 11:
  \[ S_D = \sqrt{15^2 + \left( \frac{15}{2} \right)^2} \]
  \[ S_T + D = \frac{15 + 6.35}{2} = 10.675 \]

  Because, \( S_D > \frac{S_T + D}{2} \) the maximum of velocity,
  \[ V_{max} = \frac{15}{15 - 6.35} \times 1.941 \]
  \[ V_{max} = 3.36 \text{ m/s} \]

- Reynold number \( (Re) \) can be calculated:
  \[ Re_{D,max} = \frac{3.36 \times 0.00635}{16.1918 \times 10^{-6}} \]
  \[ Re_{D,max} = 1317.704 \]

- Heat transfer as long the tube \( (q') \) calculate with equation 12:
  \[ q' = -813.34 \]
  \[ q' = -1402.31 \text{ W/m} \]

  Because the number of tubes not specified, so calculate the coefficient of heat transfer:
  \[ q' = N(\bar{h} \times \pi \times D \times \Delta T_{im}) \]
  \[ -1402.31 = N \times \bar{h} \times \pi \times 0.00635 \times 22.868 \]
  \[ \bar{h} = \frac{2416.306}{N} \]

- Nusselt number with Zukauckas equation:
  \[ \bar{N}u_D = C_2 \times C_1 \times Re_{D,max}^m \times Pr_r^{0.36} \times \left( \frac{Pr}{Pr_s} \right)^{0.25} \]
  \[ \bar{N}u_D = C_2 \times 0.35 \times 1317.704^{0.6} \times 0.70658^{0.36} \times \left( \frac{0.70658}{0.71298} \right)^{0.25} \]
  \[ \bar{N}u_D = C_2 \times 22.944 \]
• Number of column ($N_L$)

$$\bar{h} = \frac{Nu_D}{D} \frac{k}{D}$$

$$\frac{2416.306}{N} = C_2 \times 22.944 \times 0.026522$$

$$\frac{2416.306}{N_L \times 5} = C_2 \times 22.944 \times 0.00635$$

$$N_L \times C_2 = 5.04$$

• Value of $N_L \times C_2$ with variation longitudinal of the tube ($S_L$):

| No | $S_L$(mm) | $N_L \times C_2$ |
|----|-----------|------------------|
| 1  | 14        | 4.97             |
| 2  | 15        | 5.04             |
| 3  | 16        | 5.11             |
| 4  | 17        | 5.17             |

From an above, the value of $N_L \times C_2$ number of columns is 5.

3.2. Design testing of heat exchanger (evaporator) type of staggered tube cross-flow

Figure 2 show the design vapor compression with the modification evaporator of the system. The analysis of the system includes one compressor, condenser, expansion valve, and evaporator. After one hour or the temperature in room constant, the testing is done.

![Figure 2. Design vapor compression system](image)
Figure 3. Impact of the mass flow variation to temperature air inlet and temperature air outlet

Figure 3 shows the correlation of air temperature inlet and outlet of the evaporator with a mass flow of air, based on the ambient temperature. In the figure, the blue line is a temperature air inlet to the evaporator and the green line is the temperature air outlet from the evaporator. The average discrepancy of the temperature air inlet and outlet evaporator is 0.08. In addition, from the curve trend of temperature change parallel with mass flow rate, when the mass flow rate increases, it made temperature air outlet evaporator rise up.

4. Conclusions
Tube type heat exchanger type staggered tube cross-flow has been designed and built; the heat exchanger (evaporator) can reduce the temperature of 0.08% from the temperature air inlet. The designed evaporator has a length of 26 cm, a width of 16 cm and a height of 84.2 cm, with a tube bank passageway diameter of 6.35 mm, a tube length of 58 cm and as many as 25 passes, consisting of 5 lines and 5 staggered positions. The distance between the tubes is 15 mm vertically and horizontally.

5. Reference
[1] Shah K.N, Mahant K.V, Patel P.D, Yadav C.O. 2013. International Journal of Engineering Research & Technology ISSN: 2278-0181
[2] Pintoro A, Ambarita H, Nur T.B, Napitupulu F.H. 2018. International Conference Numerical Analysis in Engineering 308 012026
[3] Napitupulu F.H, Sabri M, Sihombing F, Sihombing H.V. 2018. IOP Conf. Series: Material Science and Engineering 420 012014.
[4] Rao J.B.B, Raju V.R. 2016. International Journal of Mechanical and Material engineering 11:6
[5] Yoo S.Y, Kwon H.K, Kim J.H. 2007. Journal of Mechanical Science and Technology 21 505-512.
[6] Jiang H, Aute V, Radermacher R. 2006. International Journal of Refrigeration 29 601-610
[7] Barbosa J.R.Jr, Melo C, Hermes C.J.L. 2008. International Refrigeration and Air Conditioning Conference 874 2310
[8] Pu L, Li Q, Shao X, Ding L, Li Y. 2018. Applied Thermal Engineering. S1359-4311(18)32233-6
[9] Wang E, Li K, Husnain N, Li D, Mao J, Wu W, Yang T. 2018. Applied Thermal Engineering 141 819-827.
[10] Incropera Frank P 2011 Fundamentals of Heat and Mass Transfer, Seventh Edition. United States of America