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ABSTRACT

This study attempts to qualitatively quantify the thermal design aspects of the suction tube of a hermetic refrigeration compressor with high shell pressure configuration. The effects on heat transfer and fluid flow which affect the design dimensions and material selection of the suction tube will be shown and discussed. Results obtained suggested that the tube material plays a minimum part in restricting heat transfer. The reason for this being that a higher heat transfer resistance comes from the convective heat transfer inside the tube, which depends on the flow situation of the tube, rather than the tube materials. A good design should incorporate effects of reducing the flow velocity of the gas in the tube.

INTRODUCTION

In vapour compression refrigeration systems, the outlet of the evaporator is connected to the inlet of the compressor by a tube, which is generally called suction tube, see Fig. 1. The performance of the compressor depends greatly on the refrigerant conditions upstream of the suction port and the latter depends on, among other parameters, the design of the suction tube. The suction tube design is commonly known to have effects on the noise and volumetric efficiency of the compressor [1, 2].

The design criteria of the suction tube are always concentrated on reducing the following three criteria: noise, suction heating and pressure drop. To overcome the noise at the suction side, it is common to place a suction muffler at the end of the suction tube. The existence of the suction muffler also reduces pressure drop upstream of the suction valve during the suction process. Suction heating effects reduce the volumetric efficiency of the compressor. To reduce the suction heating effects, the thermal design aspect on the suction tube must be considered. This paper presents a simplified study on the thermal aspects of the suction tube design. It focused on the suction heating effects in relations to various design variables. The following are taken into consideration: convective heat transfer along the tube, conduction heat transfer outside the tube.

To simplify the analysis, the investigation begins by postulating the suction tube as a simple horizontal tube that is placed in a hot chamber. The latter representing the shell of the compressor. This arrangement is commonly found in hermetic compressors with high shell pressure configuration, where the refrigerant gas is first discharged into the shell before it is led to the condenser. The definition of the suction tube from compressor point of view is the section that extends from the shell of the compressor to the inlet port of the pump unit. Fig. 2 shows the schematic view of the simplified suction tube model applied in present analysis. The tube is not more than an ordinary tube with refrigerant gas flows inside. Outside the tube is the hot refrigerant gas at temperature $T_w$. The conditions of the refrigerant gas outside the tube is not known, but it is assumed that the flow velocity outside the tube may be simplified to an equivalent cross flow velocity $U$ across the tube. Results on the preliminary analysis show that under the postulating conditions, the tube materials, in general, has very little effects on the heat transfer to refrigerant flow inside the tube.

HEAT TRANSFER ANALYSIS

In this analysis, heat transfer correlations commonly found in literature are employed. The correlations appeared in the literature can be broadly divided into two categories depending on whether the problem considered is under constant wall temperature or constant heat flux. Under the situation described previously, it is assumed that the current situation may be treated as a constant wall temperature problem. It is however noticed that [3] for the Reynolds number ranging
from $10^4$ to $10^5$, and for a hydronamically and thermally fully developed flow, the error that may be involved in the calculation of Nusselt number is less than 5%. For the present qualitative analysis, it is believed that the assumption made is reasonable. And in fact, for heat transfer correlations available in the literature, higher error may come from the estimation of heat transfer coefficients.

For hydronamically and thermally fully developed turbulent gases flow with $0.5 \leq \text{Pr} \leq 1$, in the region where $\text{Nu}(\varphi) \neq \text{Nu}(T)$ the Colburn [3] correlations may be applicable:

When $L/D \geq 60$

\begin{align*}
\text{Nu} &= 0.022 \text{Re}^{0.8} \text{Pr}^{0.6} \\
\text{Nu} &= 0.021 \text{Re}^{0.8} \text{Pr}^{0.6}
\end{align*}

When $20 \leq L/D < 60$

\begin{align*}
\text{Nu}_L &= \text{Nu} \left( 1 + \frac{6D}{L} \right)
\end{align*}

When $2 \leq L/D < 20$

\begin{align*}
\text{Nu}_L &= \text{Nu} \left( 1 + \left( \frac{D}{L} \right)^{0.7} \right)
\end{align*}

In the above equations, $\text{Nu}$ and $\text{Nu}_L$ represent Nusselt Number, $D$ is the tube diameter, $L$ is the tube length. $\text{Nu}(\varphi)$ is the Nusselt number for constant heat flux whereas $\text{Nu}(T)$ is the Nusselt number for constant wall temperature.

For fully developed laminar flow assuming constant wall temperature, the average heat transfer coefficient over the entire tube length may be expressed Hausen equation [4] by:

\begin{align*}
\text{Nu} &= 3.66 + \frac{0.0668(D/L) \text{Re Pr}}{1+0.4(D/L) \text{Re Pr}^{2/3}}
\end{align*}

Where $\text{Pr}$ is the Prandtl number of the fluid. In the above equation, if the tube is long enough, the Nusselt number approaches a constant value of 3.66.

Next consider the heat transfer outside the tube. The flow situation outside the tube is not known. In the compressor unit, the discharge process is first from the discharge port of the compressor into the shell, then, a tube connects the compressor shell to the condenser leads the refrigerant out of the shell. Thus, it is suspected that the situation outside the tube can be simulated by assuming firstly with an arbitrary cross velocity over the tube. In practice, the value of assumed cross flow velocity that gives good agreement between the predicted temperature of the tube wall with measured value may be taken as the equivalent cross flow velocity, representing the flow situation outside the tube.

For forced convection heat transfer across the tube, the Churchill and the Bernstein [4] correlations may be applicable:

\begin{align*}
\text{Nu} &= 0.3 + \frac{0.62 \text{Re}^{1/2} \text{Pr}^{1/3}}{1+\left( \frac{0.4}{\text{Pr}} \right)^{2/3} \left( 1 + \left( \frac{\text{Re}}{282000} \right)^{5/8} \right)^{4/5}}
\end{align*}

for $10^2 < \text{Re} < 10^7$ ; $\text{Pe} > 0.2$, where $\text{Pe} = \text{Re} \times \text{Pr}$.

Fig. 3 shows that the cross flow velocity of the gas outside the tube, $U$ has very little effects on temperature rise of the fluid, if its value is greater than 1 m/s.

The last item to consider is the conduction through the tube wall. This consideration allows the properties of the tube materials to be accounted for in the heat transfer consideration. This is given by the conduction equation through tube:

\begin{align*}
Q &= \frac{T_{\text{wo}} - T_{\text{wi}}}{\ln \left( \frac{r_0}{r_f} \right) / (2\pi K_m L)}
\end{align*}
Where $K_w$ is the thermal conductivity for tube wall, $L$ is the length of the tube, $T_w$ and $T_{wo}$ are inside and outside tube wall temperatures, whereas $r_i$ and $r_o$ are internal and outer tube radii respectively.

Considering the energy balance for heat transfer from the outer tube through the tube wall and into the inner tube, the rise in temperature $\Delta T$ as the refrigerant gas flows through the tube, may be expressed as:

$$\Delta T = \left( T_o - T_i \right) \left( m c_p \frac{U A}{2} \right) T_i - T_i$$

(4)

Where $T_o$ is the gas temperature in the shell away from the tube, $T_i$ and $T_o$ are fluid inlet and outlet bulk temperatures respectively. The term $U A$ may be expressed as:

$$U A = \frac{1}{R_1 + R_2 + R_3}$$

(5)

Where $R_1, R_2, R_3$ are thermal resistances defined by:

$$R_1 = \frac{1}{h_i A_i}, \quad R_2 = \frac{\ln(r_o/r_i)}{2 \pi K_m L}, \quad R_3 = \frac{1}{h_o A_o}.$$

Where $h$ and $A$ are convective heat transfer coefficient and heat transfer areas respectively, $i$ and $o$ represent inner and outer tube properties.

Note that in the above equations, the gas property values are evaluated at the mean bulk temperature, $T_b$. Where $T_b$ is given by:

$$T_b = \frac{T_i + T_o}{2}.$$

The above equations (1,2,4,5) can be solved with an assumed $T_o$ to begin with. Thus the fluid properties values required may be evaluated and hence the heat transfer calculation may be carried out. Subsequently, the value of $T_o$ may be updated with the most recent value obtained from equation (5). This calculating procedure is repeated until the converged solution is obtained. The convergence is assumed when the most recent $T_o$ value changes within the pre-set tolerance, say 0.001°C, when compared with the previous $T_o$.

In the above analysis, the refrigerant used is R134a. The thermophysical properties of the refrigerant gas are obtained from the equations given in Ref. [5]. The computer program has been written to compute the temperature rise, $\Delta T$ of the refrigerant gas flowing through the tube. The pressure drop $\Delta P$ caused by tube wall friction based on the smooth tube has also been calculated.

RESULTS AND DISCUSSIONS

The standard input parameter used are: mass flow rate, $\dot{m} = 1.6$ g/s, inside tube diameter, $D = 7$ mm and wall thickness of 1 mm, tube length, $L = 50$ mm, nominal suction pressure = 1.03 bar, nominal discharge pressure = 13 bar, shell bulk temperature, $T_{\infty} = 120^\circ C$, fluid inlet temperature, $T_i = 40^\circ C$, copper tube with conductivity, $K_m = 385 W/mK$.

The results obtained are shown in Fig. 4 to 7. In general the results show that the tube flow is always turbulent, for a typical of tube diameter of 7 mm, the Re is about 22000. The results also show that the value of $Re$ governs the heat transfer. This also implies that the highest thermal resistance comes from the thermal resistance in tube. The tube wall material has insignificant thermal resistance as compared to convective terms. Apart from these, the results also show that:

Fig. 4(a) to (c) show variation of results with varying tube diameter. The results show that as the tube diameter increases heat transfer and pressure drop reduce, which is caused by reduction in flow velocity. Pressure begins to drop significantly only when the tube diameter is less than 5 mm.

The results show that the tube material with thermal conductivity of less than 1 W/mK and below will affect the heat transfer in tube, see Fig. 5(b). Results also show that the gas exit temperature increases linearly with tube length. The increase in heat transfer in this case is found to cause by bigger heat transfer area, provided by longer tubes, see Fig. 6.

The investigation also carried to see the effects of compressor speed on heat transfer. The results show that heat transfer increases as speed increases but the temperature gain of the fluid reduces. This situation is due to the fact that the rate of heat transfer is lower then the rate at which the mass flow rate increases. The results on heat transfer with compressor speed are shown as the variations with mass flow rate instead, in Fig. 7.
CONCLUSIONS

The investigation shows that the flow inside the tube is always turbulent. It also shows that the tube material has an insignificant thermal resistance when compared with the convective thermal resistance in the tube. The results suggested that to reduce suction heating and pressure drop and thus to improve the volumetric efficiency of a compressor, for a known refrigerant flow rate, efforts should be concentrated on reducing the gas flow velocity and hence $Re$. This situation can be achieved by, such as, increasing the suction tube diameter and reducing the tube length.

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REFERENCES


Fig. 1 A sectional view of a compressor showing the suction tube

Fig. 2 Schematic of suction tube

Fig. 3 Variations of $\Delta T$ and $Q$ with $U$
Fig. 4(a) Variations of Temperature rise and Pressure drop

Fig. 4(b) Variations of Re and heat transfer rate

Fig. 4(c) Comparing the thermal resistances

Fig. 5(a) Variations of Temperature rise and Pressure drop

Fig. 5(b) Variation of Re and heat rate

Fig. 5(c) Comparing thermal resistances
Fig. 6(a) Variations of Temperature rise and pressure drop

Fig. 6(b) Variations of Re and heat rate

Fig. 6(c) Comparing thermal resistances

Fig. 7(a) Variations of temperature rise and pressure drop

Fig. 7(b) Variations of Re and heat rate

Fig. 7(c) Comparing thermal resistances