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Measurement of Tube Bundle Coolers Under Working Conditions

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ABSTRACT

This paper deals with the experiments on the two tube bundle coolers. The results are used for verification of theoretical relations for the heat transfer coefficient and pressure losses, and for the extending their limits of applicability. The coolers with the smooth and finned tubes, with clean and dirty surface are compared. The ability of the finned tubes for the coolers of the lubricated compressors were investigated.

INTRODUCTION

Out of the over-all production volume of coolers for water-cooled reciprocating compressors some 60 to 70% are taken up by tube bundle coolers. In spite of many years of development and in spite of their extensive application these coolers fall short of meeting the fundamental requirements imposed on modern apparatus: their specific thermal output relative to their heat transfer area, weight and volume of the assembly is relatively small. On the other hand, they exhibit a number of advantages in contrast to other well-established cooler designs: small floor area requirements, easily replaceable tube bundles, a fully mastered design and manufacturing technology. - Even though no essential increase of their heat output can be achieved by design modifications, it is necessary even in the future to pay attention to improving their design, operation, and heat transfer parameters, owing to the very high volume of production.

Experiments were carried out on coolers of two sizes /having inner transfer surface areas of 2.03 and 9.053 m², respectively/ using both smooth bundles and fin-and-tube radiators, at 600 kPa overpressure.

The experiments aimed at
- experimental verification of the suitability of the relations used for the heat transfer coefficient and the pressure loss, even under conditions exceeding their validity limits indicated in literature
- experimental verification of the suitability of using finned bundles for oil-lubricated air compressors
- experimental comparison of smooth and fin-end-tube assemblies, both clean and clogged
- comparison of bundles with segmented and annular diaphragms.

In these experiments, the Reynolds number at the water side varied within the limits of $Re_w = /2 - 8/.10^3$, whereas that at the air side was $Re_L = /2 - 6.5$ or $8/.10^4$.

Clogged bundles were measured following one month of three-shift operation in the compressor room of a mine.

1. VERIFICATION OF THE DESIGN FORMULAS

a/ Relationships used

Heat transfer coefficient

The heat transfer coefficient at the water side in the region of turbulent flow is calculated most frequently using Hausen's formula

$$Nu = 0.115 \left( Re^{2/3} - 125 \right) Pr^{1/3} \left[ 1 + \left( \frac{d_w}{L} \right)^{2/3} \right] \left( \frac{d_T}{d_s} \right)^{0.14}$$

The transfer coefficient at the air side in smooth bundles is calculated using Donohue's formula /valid up to $Re_L = 50,000/

$$Nu = C Re^{0.6} Pr^{0.33} \left( \frac{d_T}{d_s} \right)^{0.14}$$

or Kern's formula /valid for $Re_L = 210^3 - 10^6/

$$Nu = 0.36 Re^{0.55} Pr^{0.33}$$

For finned bundles the heat transfer coefficient is calculated by a corrected Donohue's formula
\[ \text{Nu} = 0.7 \, C \, \text{Re}^{0.6} \, p^{0.33} \left( \frac{d}{d_s} \right)^{0.14} \]

or Frenkel's formula
\[ \text{Nu} = C \, \text{Re}^{0.65} \left( \frac{d}{d_s} \right)^{-0.54} \left( \frac{h}{a} \right)^{-0.14} \]

where \( h \) and \( u \) are the height and spacing of the fins, resp.

**Pressure loss**

The pressure loss \( \Delta p \), at the gas side in smooth tubes is calculated most frequently using the simpler procedures by Grimison, Kern, or Frenkel. The more complicated Bell formula was not examined here. The calculation by Grimison's method can only be used up to \( \text{Re}_L = 40,000 \), owing to the limited extent of graphs for the \( \text{Nu} \) coefficient, and this is not enough.

For finned tubes, the same formulas are used as for smooth tubes. A more recent formula is that by Williams and Katz:

\[ \Delta p = 1548 \frac{Z+1}{6} \left( \frac{M_n}{m_{5d}} \right)^2 \left[ \left( \frac{n}{n_d} \right)^{0.14} + 0.542 \left( \frac{d}{f_p} \right)^2 \right] \text{[Pa]} \]

Here
- \( f_p \, [m^2] \) - free cross section between diaphragms
- \( f_d \) - \( \frac{(Z-1)c f_k + 2b f_k}{(Z-1)c + 2b} \)
- \( f_k \) - free cross section between diaphragms measured on shell diameter
- \( f_t \) - free cross section between the last diaphragm and the tube plate
- \( \zeta \) - drag coefficient
- \( Z \) - number of diaphragms
- \( n' \) - number of tube rows between the centers of gravity of neighbouring diaphragms

**Heat transfer coefficient**

A comparison of calculated heat transfer coefficient for a clean tube bundle is given in Table 1.

The results are compared graphically in Figs. 1 through 3 using following notation:
- \( \text{Re}_L \) - Reynolds number, air side
- \( \text{Re}_w \) - Reynolds number, water side
- \( G \) - smooth tubes, clean
- \( G' \) - smooth tubes, clogged
- \( B \) - finned tubes, clean
- \( B' \) - finned tubes, clogged
- \( k \) - experimental heat transfer coefficient, clean tubes
- \( k' \) - experimental heat transfer coefficient, clogged tubes

**Subscripts**
- \( t \) - theoretical
- \( s \) - calculated with dry air
- \( v \) - considering the effect of air humidity

For smooth tubes /Fig. 1, symbol G/ the experimental value of heat transfer coefficient coincides with the theoretical value calculated by the Donohue formula /k_tv/, which takes into consideration the effect of humidity condensation. The theoretical

<table>
<thead>
<tr>
<th>( \text{Re}_L )</th>
<th>( k , [W , \text{m}^{-2} , \text{K}^{-1}] )</th>
<th>( /k - k'/ : k[%] )</th>
</tr>
</thead>
<tbody>
<tr>
<td>experim.</td>
<td>Donohue</td>
<td>Kern</td>
</tr>
<tr>
<td>21370</td>
<td>108,5</td>
<td>102,4</td>
</tr>
<tr>
<td>41200</td>
<td>149,3</td>
<td>152,7</td>
</tr>
<tr>
<td>66530</td>
<td>266,5</td>
<td>286,0</td>
</tr>
<tr>
<td>88450</td>
<td>341,0</td>
<td>338,3</td>
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</tbody>
</table>

Mean deviation [%] = 0,28 - 14,4
course for a case free from the effect of humidty condensation is represented by
the curve k'ts.

It follows from the comparison of results for finned tubes /Fig. 1/, symbol B/, that the Donohue formula gives better approximations for the given type of cooler. Frenkel's formula gives values which are too high /this curve is not shown here/. In the same Figure a comparison is made of the theoretical curve of heat transfer coefficient k'tv /considering humidty condensation/ and k'ts /no kondensation/ with the experimental curve for a clean bundle with fins. With the finned bundle, the effect of oil deposits on the surfaces is manifested practically immediately, at the beginning of operation.

Pressure loss

Smooth tubes: The experimental curve of Δp_L /gas side, curve G/ is identical to that by Frenkel's formula /G Frenkel/, whereas the calculation by the Kern formula /G Kern/ gives lower values

Finned tubes: The calculation according to Williams and Katz /Fig. 2/, curve B ≡ B_w+K/ is an excellent representation of the experimental curve, whereas the calculation by Kern /curve B Kern ≡ G Kern/ is unsuitable.

The curves for experimental flow drag Δp_w at the wasser side are identical for both smooth and finned tubes /Fig. 3/, curve G ≡ G/; the theoretical drag curves are also the same /curve G_t ≡ G_t/ but the calculated drags are in fact one half of the experimental values.

<table>
<thead>
<tr>
<th>Table 2</th>
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<tbody>
<tr>
<td>Re_L</td>
</tr>
<tr>
<td>G</td>
</tr>
<tr>
<td>B</td>
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<tr>
<th>Table 3</th>
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<tbody>
<tr>
<td>Re_w</td>
</tr>
<tr>
<td>G</td>
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<tr>
<td>B</td>
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</tbody>
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2. EFFECT OF TUBE CLOGGING

a/ Effect on the heat transfer coefficient

This effect is also shown in Fig. 1. The values obtained with either a finned or a smooth bundle following a one-month three-shift operation /ca 500 operational hours/ are shown in dashed line /curves k'/.

Mutual comparisons of smooth and finned bundles, both clean and clogged, are given in Fig. 4 /as a function of Re_L /air side/, and in Fig. 5 /as a function of Re_w /water side/.

The average relative drop of the heat transfer coefficient due to clogging, \( \frac{k - k'}{k} \), calculated from three or four values obtained at different Re_L /air side/ or Re_w /water side/, are shown in Tables 2 and 3.

It can be observed that

1. with smooth tubes, the relative drop of the heat transfer coefficient increases with increasing quantity of water but remains nearly constant with increasing quantity of air
2. with finned tubes, the relative drop of the transfer coefficient increases with increasing quantity of water but decreases with increasing quantity of air.

These experimental results confirm the well-known fact that one and the same thermal resistance will always have a greater effect upon the higher heat transfer coefficient /i.e., that at the water side/, and also confirm that the oil film is entrained and pulled away from the tube surface at higher air flow velocities; in finned tubes this is so to such an extent that even the relative drop of the heat
transfer coefficient decreased.

It follows from the diagrams that the finned tubes, even when clogged, permit a better heat transfer than the smooth tubes. The transfer coefficient of clogged finned bundle is higher by 10 to 30% than that of a clean smooth bundle. The value of the coefficient of thermal conductivity of a wet clayey deposit was calculated on the basis of the above experiments and was confirmed later by direct measurement. The average value is about 0.7 W/m²K. Most of literature sources give a value which is about one half of this.

b/ Effect of tube clogging on the pressure loss

The pressure loss \( \Delta P_L \) at the air side of smooth tubes is not increased when the tubes become clogged /Fig. 2, curve \( G = G' \). For finned tubes /curve \( B' \) the value \( \Delta P_L \) is somewhat higher when the surface is clogged, and decreases with decreasing air flow velocity. This is probably due to entrainment of the oil film stuck in-between the fins at a higher velocity, thus enlarging the cross-sectional area for the passage of air.

The pressure loss \( \Delta P_w \) at the water side, which is roughly the same for clean bundles, whether smooth or finned /Fig. 3, curve \( G = B' \), is higher for the clogged smooth bundle /curve \( G' \) than for the clogged finned bundle /curve \( B' \). This fact can be explained so that the finned bundle becomes somewhat less clogged, owing to a slightly higher velocity and more turbulence provoked by the regular projections at the interior surface originating from the fin rolling operation.

3. COMPARISON OF BUNDLES WITH SEGMENTED AND ANNULAR DIAPHRAGMS

According to literature, the annular diaphragms are more advantageous for heat transfer than the segmented diaphragms at equivalent diameters above 0.0237 m. Experiments were conducted with clean bundles of smooth tubes. The experimental results of the heat transfer coefficient are given in Fig. 5. Whereas the theoretical increase of heat transfer coefficient for a bundle with annular diaphragms /symbol \( k_{RING} \) is 13 to 17% against that with segmented diaphragms /\( k_{SBHE} \), the experimental increase as per Fig. 6 was 19 to 27%.

The difference in drag resistance at the gas side /Fig. 6/ increases with increasing quantity of gas. An increased pressure loss \( \Delta P_L \) is encountered with the annular diaphragms; as against the segmented diaphragms it constitutes about 14.5% at \( Re_L = 60,000 \) and about 32.0% at \( Re_w = 80,000 \).

CONCLUSION

Measurement has confirmed the suitability of the design relationships used to calculate the heat transfer and the pressure loss, and their correctness was verified up to Reynolds numbers of \( Re_L = 8 \times 10^4 \) and \( Re_w = 9.10^3 \). The effect of clogging of the two transfer surfaces was examined. It has been shown that tubes provided with low, rolled-on fins provide a better heat transfer even when covered with an oil deposit than clean smooth tubes.

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