Exergy Analysis of Refrigeration Evaporators

Sanat Gapurovih Zakirov  
*Tashkent State Technical University*

Kudrat Fuadovich Karimov  
*Tashkent State Technical University*

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ABSTRACT
The experiments have been performed and the evaporators with rolled-up transversal annular grooves were analyzed. Exergies, exergic losses and efficiency were determined depending on refrigerant boiling points, temperature of brine. The objective of this work is evaluation of efficiency of evaporators with rolled-up pipes by the exergic analysis method.

1. INTRODUCTION
One of the actual and important problems standing before the engineers, who design and develop refrigerating plants, is reducing of metal intensity of heat exchangers and energy consumption for their operation. Intensification of heat transfer is regarded as an effective way to resolve this problem. To the present time there are researched and proposed the various methods of convective heat transfer intensification.

The number of optimal surfaces includes the tubes with rolled-up transversal annular grooves explored in heat exchangers of aeronautic engineering, power engineering, petroleum and metallurgy industries. Efficiency of these heat exchangers with rolled-up tubes was generally assessed, as we know, by energy method only. That method does not allow to appreciate the thermohydrodynamic perfection of a heat exchanger or heat transfer surface but can only be used for their comparison.

2. BASIC SECTION
An evaporator, which specification is given below, was developed for performing of experiments:
- "tube in tube" type, with boiling in the inter-tube space;
  - outer tube – copper, $\varnothing$ 24 x 2 mm;
  - inner tube – copper, $\varnothing$ 12 x 1 mm;
- heat transfer surface, internal, m$^2$ – 0.1475;
- overall length, m – 4.7.

Four options of tubes were examined: smooth one and rolled-up ones with relative height of diaphragm: d/D=0.876; 0.91; 0.946. Relative spacing of turbulators is the same for all tubes and is equal to d/D = 0.4 (Dreitser et al., 1989). It must be noted that the presence of transversal annular grooves did not affect the heat transfer during boiling of ammonia, as the operation of evaporators corresponded to bubble boiling mode. Tubes with rolled-up transversal annular grooves only give positive heat transfer effect in film boiling mode. Heat transfer intensification was achieved through raising of heat transfer coefficient from the part of refrigerating medium. As the refrigerating
medium there was used 23% water solution of sodium chloride. Refrigerant pump grade is X14-22M. Ammonia boiling temperature range is between –10º and –27ºC. Water brine temperatures at inlet and outlet are 0ºC to –17ºC and –5ºC to –22ºC respectively. Specific cold-productivity is 7,000 W/m².

Evaluation of efficiency of the evaporator with the studied tubes has been made by exergic method.

Exergy delivered by the refrigerant was calculated by the Equation (1).

\[ E_a = Q_o \left( l - \frac{T_{o,c}}{T_o} \right) \]  

where: \( Q_o \) is evaporator cold productivity.

The value of exergy delivered by the pump transferring sodium chloride water solution was determined by the Equation (2).

\[ E_p = N_p \cdot \eta_{p,m} \cdot \eta_p^e \]  

where: \( N_p \) is pump output determined by the Equation (3).

\[ N_p = \frac{G_s \cdot \Delta P_s}{\rho_s \cdot \eta_p} \]  

\( \eta_p^e = 0.75 \), \( \eta_{p,m} = 0.9 \), \( \eta_p^e = 0.8 \) – are respectively internal, electromechanical and exergic efficiency of the pump.

Exergy gained by the refrigerating medium was determined by the Equation (4).

\[ E_s = Q_o \left( l - \frac{T_{o,c}}{T_o} \right) \]  

Exergy losses due to the final temperature difference were calculated by the Equation (5).

\[ D_T = (E_a + E_p) - E_s \]  

Exergy losses due to hydraulic resistances were determined by the Equation (6).

\[ D_p = G_s \cdot \Delta E_s \cdot \frac{l}{\eta_p^e} \]  

where \( \Delta E_s \) is the change of specific exergy of the water solution during change of its pressure, was calculated by the Equation (7).

\[ \Delta E_s = \Delta P_s \left[ \frac{\theta_s}{T_{s,m} - T_{o,c}} \left( \frac{\Delta V_s}{\Delta T_s} \right) \right] \]  

Overall loss of exergy was calculated by the Equation (8).

\[ \sum D = D_{o,c} + D_t + D_p \]  

Exergic efficiency of the condenser was determined by the Equation (9).

\[ \eta_E^e = \frac{E_s}{E_a + E_p} \]
The results of exergetic analysis are given in the Table 1 (only for $\theta = 6.8; 7.6^\circ C$). As seen from the Table, for all values of $\theta$ or $|t_o - t_{s2}|$ in the smooth surface tubes, the dominating part of the overall exergy losses is represented by the losses due to the final temperature difference, therefore the exergy losses from hydraulic resistances and heat exchange with environment may be neglected. For all rolled-up tubes the share of $D_p$ losses is significant and they may

<table>
<thead>
<tr>
<th>Tube with d/D=0.945, $\theta=6.8^\circ C$</th>
<th>$E_a$, J/s</th>
<th>$E_s$, J/s</th>
<th>$E_{ps}$, J/s</th>
<th>$D_{oc}$, J/s</th>
<th>$D_{ot}$, J/s</th>
<th>$D_{op}$, J/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube with d/D=0.91, $\theta=6.8^\circ C$</td>
<td>88.992</td>
<td>71.436</td>
<td>1.644</td>
<td>0.702</td>
<td>12.555</td>
<td>2.082</td>
</tr>
<tr>
<td>Tube with d/D=0.876, $\theta=6.8^\circ C$</td>
<td>103.428</td>
<td>85.506</td>
<td>1.246</td>
<td>0.717</td>
<td>17.922</td>
<td>1.547</td>
</tr>
<tr>
<td>Tube with d/D=0.945, $\theta=7.6^\circ C$</td>
<td>122.536</td>
<td>103.934</td>
<td>0.918</td>
<td>0.744</td>
<td>18.602</td>
<td>1.113</td>
</tr>
<tr>
<td>Tube with d/D=0.91, $\theta=7.6^\circ C$</td>
<td>132.319</td>
<td>133.354</td>
<td>0.774</td>
<td>0.759</td>
<td>18.965</td>
<td>0.926</td>
</tr>
<tr>
<td>Smooth tube, $\theta=6.8^\circ C$</td>
<td>142.612</td>
<td>123.235</td>
<td>0.483</td>
<td>0.775</td>
<td>19.377</td>
<td>0.571</td>
</tr>
</tbody>
</table>

| Tube with d/D=0.910, $\theta=7.6^\circ C$ | 88.381 | 71.436 | 2.030 | 0.678 | 16.944 | 2.570 |
| Tube with d/D=0.876, $\theta=7.6^\circ C$ | 102.822 | 85.506 | 1.512 | 0.693 | 17.317 | 1.878 |
| Tube with d/D=0.945, $\theta=7.6^\circ C$ | 121.917 | 103.934 | 1.092 | 0.719 | 17.983 | 1.323 |
| Tube with d/D=0.91, $\theta=7.6^\circ C$ | 131.693 | 113.355 | 0.909 | 0.734 | 18.338 | 1.087 |
| Smooth tube, $\theta=7.6^\circ C$ | 141.974 | 123.235 | 0.546 | 0.750 | 18.739 | 0.645 |

| Tube with d/D=0.876, $\theta=7.6^\circ C$ | 87.85 | 71.436 | 2.526 | 0.657 | 16.413 | 3.199 |
| Smooth tube, $\theta=7.6^\circ C$ | 102.296 | 85.506 | 1.855 | 0.672 | 16.790 | 2.302 |
| Tube with d/D=0.945, $\theta=7.6^\circ C$ | 121.379 | 103.934 | 1.314 | 0.698 | 17.445 | 1.592 |
| Tube with d/D=0.91, $\theta=7.6^\circ C$ | 131.148 | 113.355 | 1.081 | 0.712 | 17.793 | 1.294 |

The results of exergetic analysis are given in the Table 1 (only for $\theta = 6.8; 7.6^\circ C$). As seen from the Table, for all values of $\theta$ or $|t_o - t_{s2}|$ in the smooth surface tubes, the dominating part of the overall exergy losses is represented by the losses due to the final temperature difference, therefore the exergy losses from hydraulic resistances and heat exchange with environment may be neglected. For all rolled-up tubes the share of $D_p$ losses is significant and they may
only be neglected at the temperatures below –20°C. With reducing of the boiling temperature \( \Delta T \) are not changing, and some reduction of \( D_p \) was noticed. Thus, for example, at \( \theta = 6.8^\circ C, t_0 = -10^\circ C \) the share of \( D_p \) made up ~4.3
% of $\Delta D = 25.22$ J/s for smooth surface tube, and $D_P$ made up ~13 % of $\Delta D = 20.19$ J/s for tube with $d/D=0.91$. 
And at $\theta = 6.8^\circ$C, $t_0= -21^\circ$C the $D_P$ share was ~2.3 % of $\Delta D = 25.881$ J/s for smooth surface tube, and $D_P$ was 
~6.6 % of $\Delta D = 20.025$ J/s for rolled-up tube.
Exergetic efficiency of evaporator is reducing with increasing of $\theta$ (or $\theta - t_{s2}$ ) and boiling temperature. As seem 
from the figures 1, 2, 3 in all the values of $\theta$ and $t_0$, the exergetic efficiency of rolled-up tubes are higher than that for smooth surface tube. Lines 1 – 5, in figures 1 - 3, concerns to smooth tube, lines 6 - 10 to rolled-up tubes at $t_0$=var.

![Figure 3: $\eta = f(\theta, t_0)$ for $d/D=0.876$](image)

3. CONCLUSIONS

- Application of evaporators with rolled-up tubes with studied relative diaphragm heights is thermodynamically more beneficial than that of usage of smooth surface tube.
- Optimal relative diaphragm height is 0.876. It is advisable to study rolled-up tubes with other relative spacing.

REFERENCES


ACKNOWLEDGEMENT

Gulandom Teshakulovna Nazarova and her private concern "PARVINA-S.D." financed a part of research.