Analysis of increase in efficiency of air-cooled heat exchangers due to intensification of heat exchange

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Abstract. Currently, the issues of improving the heat transfer processes in air-cooled heat exchangers play a huge role in terms of increasing the cooling efficiency and thus reducing the size of heat exchangers. This significantly saves resources. There are several basic methods for increasing the heat transfer of fluids in air coolers, for example, such as: finning of tubes and heat exchange intensification with the help of tube inserts. The purpose of the study is to analyze the efficiency of different fin types and the use of tube inserts, as well as changes in the heat transfer coefficients of air and natural gas, depending on the changes in the types of fins and tube inserts, and their geometry. The application software is Xchanger Suite provided by the company HTTRI to ITMO University. The result is the optimally selected geometry of the heat exchange intensifiers based on the analysis, what allows reducing the dimensions of the air-cooled heat exchangers, while providing the necessary heat load, the heat exchange surface reserve coefficient and the heat transfer coefficients.

1. Introduction. To date, such issues as obtaining the cheap energy with minimum efforts are widely actual among the engineers and researchers. One of the most popular solutions is use of an air-cooled heat exchanger that implies air-cooling implementation at environment temperature or at temperatures below zero degrees Celsius. Obviously, such type of heat exchangers has significant advantages in comparison with other heat exchangers where the second working fluid is needed to calm a heat exchange fluid down. For air-cooled heat exchangers it is not necessary to have a cooling medium due to using air energy. It allows one to save energy costs and to make a heat exchanger less in dimensions. Another advantage is no fouling and corrosion of tube outside surface. The construction of air-cooled heat exchanger (Figure 1) consists of a tube bundle, fan, louver, diffuser and supporting construction. The purpose of fan application is to pump the airflow through the heat exchanger. The fan is installed in the diffuser that is intended for efficiency increase and uniform distribution of airflow. The diffuser is elongated uniform shell along the whole tubes where the fan is located inside. The heat exchange section consists of the tube bundle where the fluid enters the tubes [1].

In this study, a force draught air-cooled heat exchanger is involved in the winter mode for the purpose of natural gas cooling in the pre-cooling cycle [2]. The input data is presented in Figure 2 where natural gas enters at environment temperature in a high compressed condition and is calmed down to ~26 °C. The purpose of this study is to investigate the ways to decrease the dimensions of the device increasing the effective heat transfer area that can be improved due to the tube- and airside heat transfer enhancement.
2. Methods & Materials.

2.1 Tube heat exchange intensification.

In the tubeside there are several basic ways to increase heat transfer. One of them is using of tube inserts. The most applicable technique is Twisted Tape which can twist a stream along the tube surface what triggers turbulization. Due to turbulization, medium flows are mixed with each other what enhances heat convection. Twisted Tape method is particularly effective in the turbulent flow mode of the medium when Reynolds number is more than 2300. However, due to turbulization, the mean flow velocity can be increased too much and leads to tube vibration what causes destruction of the tube bundle structure. In practical terms, Twisted Tape is able to increase heat transfer coefficient 2–3 times.

One another technique is improving the inner-tube surface by creating micro finning to twist a flow in the curved direction. This MicroFin technology is showed in Figure 4 and allows dividing the flow into its component parts.

![Figure 1. Air-cooled heat exchanger construction.](image1)

![Figure 2. Input data and scheme of streams movement.](image2)

![Figure 3. Twisted Tape tube insert.](image3)

![Figure 4. MicroFin tube surface enhancement.](image4)

2.2 Airside heat exchange intensification.

The choice for airside heat transfer enhancement is widespread and suggests many types that differ in geometry forms. The finning can be low and high and has many different forms. It is usually used to force the airflow to flush the outside surface of the tube more tightly, which increases heat transfer as well. The basic types of fins considered in the study are shown in Figure 5.
2.3 Evaluation of tube intensification efficiency

To estimate how tube inserts and tube surface enhancement may be high it is necessary to calculate thermal performance factor $\eta$ and associated values such as Nusselt number $\text{Nu}$ and friction factor $f$:

$$\eta = \frac{\text{Nu}}{\text{Nu}_0} \left(\frac{f}{f_0}\right)^{1/3}; \quad \text{Nu} = \frac{\alpha l}{\lambda}; \quad f = \frac{\Delta \rho}{(\rho u^2 / 2) \left(\frac{l}{D}\right)}.$$

Thermal Performance Factor (TPF) shows interrelation of heat exchange with such physical quantities as pressure and velocity and thermo physical properties as heat conductivity and density as well as with geometry of tube [3].

2.4 Evaluation of efficiency intensification in the airside

Fin efficiency shows how effective can be use of finning and depends on fin geometry and the material of finning. Fin efficiency is calculated using theoretical $\Omega_T$ and actual $\Omega_A$ fin efficiency [4]:

$$\Omega_T = \tanh \left(\frac{m_f l_e}{m_f l_e}\right); \quad \Omega_A = \Omega_T \cdot Fe,$$

where $l_e$ is effective fin height and depends on the form of finning:

- for circular fins $l_e = l \left(1 + L_f / 2\right) \left[1 + 0.35 \ln \left(D_f / D_f\right)\right]$;
- for serrated fins $l_e = l + L_f / 2$;
- for rectangular fins $l_e = (D_f / 2) \left[1 + 0.35 \ln C_j \left(C_j - 1\right)\right]$.

Rectangular fin height function can be calculated using the follow formula:

$$C_j = 1.28 \left(\frac{l_{sw}}{D_f} \right) \left(\frac{l_{sw} - 0.2}{l_{sw}}\right)^{0.5}.$$

The material of finning is defined by thermal conductivity of fin type and is different for copper and carbon steel. This also defines fin efficiency correction factor. $Fe = 1.2 - 0.2 / \Omega_T$ for cases $k_f \geq 155.7 \text{ W/(m·°C)}$, and in other cases $Fe = 1$ if fin height $l \leq 1$.

2.5 Overall evaluation of use heat exchange enhancement

It is important to estimate how given above ways of heat exchange intensification can increase the heat exchange surface needed to achieve the specified temperatures. Calculation of actual and required overall heat transfer coefficient (6) demonstrates how much heat quantity is conveyed from natural gas that is more heated to air through 1 m² of heat exchange surface when the temperature difference between fluids is 1 °C.

The difference between both coefficients is that required overall heat transfer coefficient is calculated as obligatory rate for the specified geometry while actual overall heat transfer coefficient demonstrates heat transfer in the real conditions:
The overdesign coefficient \( c \) (%) determines the quantity of amount of heat transfer surface required. By calculating this rate, it is possible to assess whether there is enough heat transfer surface to achieve required temperatures. If there is not enough surface it is recommended to increase the effective area of heat exchanger. For successful and reliable heat exchange, the overdesign coefficient must be approximately 5–15 %:

\[
c = 100 \left( \frac{k^c}{k^r} - 1 \right).
\]

3. Results & Discussion.

The geometry for Twisted Tape and MicroFin changes as basic geometry dimensions increase and is shown in Tables 1 and 2.

| Table 1. Twisted Tape geometry |
|-----------------------------|
| **No. of experiment** | 1 | 2 | 3 | 4 | 5 | 6 |
| **Thickness, mm** | 0.1 | 0.5 | 2 | 3 | 4 | 5 |
| **L/w for 360° twist** | 1/2/3/4 | 1/2/3/4 | 1/2/3/4 | 1/2/3/4 | 1/2/3/4 | 1/2/3/4 |

| Table 2. MicroFin geometry |
|-----------------------------|
| **No.** | 1 | 2 | 3 | 4 |
| **Number of Fins** | 5 | 10 | 15 | 20 |
| **Fin height, mm** | 2 | 4 | 6 | 8 |
| **Fin thickness, mm** | 0.2 | 0.4 | 0.6 | 0.7 |

The calculated thermal performance factor for every number of experiments is represented in Figure 6 for Twisted Tape and in Figure 7. Looking at figure 6 the maximum rate of thermal performance factor is achieved at about 2.85 and gets maximum values by L/w for 360° twist among all calculations. For all numbers of experiments TPF takes maximum with the thickness of 4 mm and then decreases. In figure 7 TPF rises as the geometry increases and the maximum achieved value is 2.57.

Geometry data for low circular finning is in Table 3, for high serrated finning in Table 4 and for high rectangular finning in Table 5. For all types, the finning frequency grows along with the increase in the geometry. The calculations were conducted for two materials: copper and carbon steel to compare materials with different thermal conductivity to show how it influences the fin efficiency.
Table 3. Circular low fins geometry

| No. | Fins per unit length, fin/meter | Fin root diameter, mm | Fin height, mm | Fin thickness, mm | Outside area/length, m²/m |
|-----|---------------------------------|-----------------------|----------------|------------------|--------------------------|
| 1   | 100                             | 25                    | 5              | 0.2              | 0.5                      |
| 2   | 200                             | 25                    | 10             | 0.4              | 0.5                      |
| 3   | 300                             | 25                    | 15             | 0.6              | 0.5                      |
| 4   | 400                             | 25                    | 20             | 0.8              | 0.5                      |
| 5   | 500                             | 25                    | 25             | 1                | 0.5                      |

As a result, for circular low finning actual, the fin efficiency increases smoothly along the curve as heat transfer coefficient and fin geometry decrease. In case of two types of high finning actual, the fin efficiency takes the maximum with high numbers of heat transfer coefficients and changes unevenly along the parabola.

Table 4. Serrated and rectangular high fins geometry

| No. | Fins per unit length, fin/meter | Fin root diameter, mm | Fin thickness, mm | Fin height, mm | Fin width (only for rectangular finning), mm |
|-----|---------------------------------|-----------------------|------------------|----------------|--------------------------------------------|
| 1   | 100                             | 25                    | 0.2              | 5              | 30                                         |
| 2   | 200                             | 25                    | 0.4              | 10             | 40                                         |
| 3   | 300                             | 25                    | 0.6              | 15             | 50                                         |
| 4   | 400                             | 25                    | 0.8              | 20             | 60                                         |
| 5   | 500                             | 25                    | 1                | 25             | 70                                         |

Figure 8. Actual fin efficiency dependence on heat transfer coefficient.

Figure 9. Actual fin efficiency dependence on heat transfer coefficient for serrated finning.

Figure 10. Actual fin efficiency dependence on heat transfer coefficient for rectangular finning.
The results of heat transfer application assessment are in table 6 where the considered air-cooled heat exchanger is calculated with and without intensification. It is estimated how much we can reduce the dimensions of air cooler not changing the specified input and output temperatures.

### Table 5. Results of effectiveness increase

| Parameter                        | Without intensification | With intensification |
|----------------------------------|-------------------------|----------------------|
| Tube longitude, mm               | 3 000                   | 1 000                |
| Outside tube diameter, mm        | 25                      | 25                   |
| Tube thickness, mm               | 2                       | 2                    |
| Overdesign coefficient c, %      | 5.1                     | 11.05                |

### 4. Conclusions

With all the obtained results, it is possible to state the following.

1. The most effective type of tube intensifier is Twisted Tape technology with the maximum achieved thermal performance factor of 2.85. It is ascertained that TPF rises along with increase in the thickness and $L/w$ for 360° twist right up to the defined rate and then it declines. In contrast to this TPF for MicroFin technology growth until it runs out of space in the tubes for the fins.

2. The most effective type of airside intensifier is copper circular low finning with the maximum achieved finning efficiency of 98%. Actually other types have shown almost the identical maximum values. Therefore, it can be stated that all three types are effective.

3. The use of both types of intensifiers allows reducing the length of the tube bundle by 66% from 3000 mm to 1000 mm, while maintaining the overdesign coefficient at the level of 5–15%.

### Table 6. Nomenclature

| Symbol | Definition                                                                 |
|--------|---------------------------------------------------------------------------|
| Nu     | Nusselt number with enhancement technique                                |
| Nu0    | Nusselt number without enhancement technique                            |
| f      | Friction factor with enhancement technique                              |
| f0     | Friction factor without enhancement technique                           |
| $u$    | mean velocity of the fluid, m/s                                          |
| $D$    | hydraulic diameter of the test section, m                               |
| $l$    | fin height, m                                                            |
| $t_f$  | fin thickness, m                                                         |
| $D_f$  | outside fin diameter, m                                                 |
| $D_i$  | inside fin diameter, m                                                  |
| $m_f$  | fin efficiency parameter, m⁻¹                                           |
| $k_f$  | thermal conductivity of fin, W/(m·°C)                                    |
| $k_a$  | actual overall heat transfer coefficient, W/m²·°C                        |
| $k_r$  | required overall heat transfer coefficient, W/m²·°C                      |
| $\delta$ | wall thickness, m                                                      |
| $\lambda$ | thermal conductivity of wall, W/m²·°C                                    |
| $\alpha_{air}$ | heat transfer coefficient for air and natural gas, W/(m²·°C)        |
| $\alpha_NG$ | heat transfer coefficient for natural gas, W/(m²·°C)       |
| $\nu$  | Prandtl number                                                           |
| $Re$   | Reynolds number                                                          |

### References

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