Discrete-rough Heat Exchange Surfaces

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Abstract. The article provides an overview of one of the methods of intensifying the convective heat transfer on discrete rough surfaces

1. Introduction
Heat exchangers have been widely used in many areas of the energy industry. In modern conditions, the improvement of the energy and technology installation efficiency requires continuous perfecting the heat exchange equipment with the help of introducing various methods of heat exchange intensification.

2. Urgency
Nowadays, the intensification of heat convection is one of the promising and complex challenges of the heat exchange theory. The choice of intensification method is individual and depends on purpose, design of the device, working fluid properties, etc. The achievement of intensification by the asperities, resulting by various types of rolling is common and fairly simple. Compared with other types of developed surfaces, the above mentioned method has a number of advantages:

- As a result of rolling on the outer surface, the intensification is also achieved in the inner surface.
- The technology of rolling is affordable and easy to date.
- The modern methods of the assembly of shell-and-tube heat exchangers are fully compatible with the rolled tubes.
- This method of intensification has relatively low price, since the cost of rolling is only a few percent of the cost of a smooth pipe.

3. Literature review
Based on [1] the concept of "discrete-rough" is applied to channels and pipes or heat exchange surfaces. Usually, discrete rough channels and pipes with various asperities on the inner surface are referred to. It is most often:
• Screw single- and multi-thread smoothly outlined asperities obtained by a well-known technology with the help of rolling rollers or discs (figure 1).

![Figure 1](image1.png)

**Figure 1.** Screw single- and multi-thread smoothly outlined asperities.

• Pipes with spiral-screwed wire inserts (figure 2).

![Figure 2](image2.png)

**Figure 2.** Pipes with spiral-screwed wire inserts.

• Cross-section, periodically arranged, smoothly contoured projections obtained with the aid of rollers or discs (figure 3).

![Figure 3](image3.png)

**Figure 3.** Cross-section, periodically arranged, smoothly contoured projections.

• Channels with an inner single- and multiple-thread screwed wall fins (figure 4).

![Figure 4](image4.png)

**Figure 4.** Channels with an inner single- and multiple-thread screwed wall fins.

• Spiral-shaped pipes (figure 5).
In this article, the thermal efficiency of pipes with internal asperities will be considered. The very effective ones are the low transverse asperities obtained by rolling [2], since, based on the experiments, at a relatively small Reynolds numbers and relatively large relative space of the asperities, the turbulence of the flow leads to an optimal ratio between the hydraulic resistance and a heat exchange increase. However, with a significant increase of Re number the positive effect of heat exchange against hydraulic resistance reduces. It could be explained by a significant increase of the pipe’s hydraulic resistance. A similar situation occurs when the projection height increases at a constant Reynolds. As a result of a more streamlined protrusion, the hydraulic resistance of the profile and the pipe as a whole is reduced, but this greatly complicates the production. The asperity shape has no practical effect on the heat exchange, but some contradictions are observed in the experiment [2]. The paper states that the successive transition from a triangular projection through a semicircular and rectangular – to a drop-like one the resistance coefficient is reduced by 24%. However, with a slight change in the rolling technology (with constant height and step of the asperities and small changes in their shape) $\xi$ changed by 25%. The authors make the conclusion that the pressure loss at triangular asperity the speed is 1.4 times more than with drop-like ones, and rectangular under – 1.33 times more.
than the semicircle is small. The experiment [3] states that the maximum losses are achieved at rectangular asperities.

The growth of the reduced roughness height $h/D$ at a constant step $t/h$ leads to a heat transfer increase only to a certain value. The reason is a significant distance of turbulent disturbances from the walls, which practically has no effect on the heat exchange near the wall. In addition, with an increase in the number of asperities per unit length, it leads to the appearance of stagnant zones with significant heat exchange worse. Based on [2, 4], the maximum heat transfer increase in a coasting pipe (3.8-4.3 times more than in a conventional pipe) was obtained at $d/D = 0.6$ and $t/h = 10$.

In the small heights area of asperities, the heat exchange rate and hydraulic growth are almost the same. This conclusion allows to reduce the size of the heat exchange apparatus, as with the same hydraulic resistance, the heat exchange of a pipe with a developed heat exchange surface is 25-40% higher compared to a smooth pipe. The optimum height of the asperities in the tube is in the range of $0.1 > 2h/D > 0.02$, and the optimal step is in the range of $25 > t/h > 10$, while increasing $h/D$ the optimum is moving into the area of large $t/h$, which is tested experimentally in the field of $Re = 104 - 105$. For annular channels the recommended values are $h/d_1 = 0.01 - 0.03$, for flat ones is $h/d_1 = 0.05 - 0.025$.

The paper [5] also shows the results of an experimental investigation of heat exchange, hydraulic resistance and the correlation between them for the air flow in pipes with the diameter $D = 15$ mm (pipe №1) and $D = 18$ mm (pipe №2), length $l = 573$ mm with transverse annular asperities made of wire by the diameter $h = 1.8$ mm. Parameters of asperities: step $t = 45$ mm, $t/h = 25$, $2h/D = 0.24$ and 0.2. The area of the $Re = 3 \cdot 10^2 - 6 \cdot 10^3$.

The generalization of experimental results on heat loss (line a) and their comparison with known experimental data (figure 8).

**Figure 8.** The generalization of experimental results on heat loss (line a) and their comparison with known experimental data.

Line a is approximated by the conformity equation

$$Nu = cRe^{0.8},$$

where in the range $Re = 1500-6000$ with taken $c = 0.048$ (error matching test points ±12%), and in the range $Re = 300-1500$ $c = 0.065$ (deviation of the experimental results from the line a not more than ±15%). These experimental results are consistent with the experiments [4] (line b) with 10% accuracy.

The results of the heat exchange experiment are consistent with the experimental Grass formula for Nusselt media at a constant number $Re = 2500$ (point b) with 18% error.

The results of the experiment (line a) are consistent with Uttawar’s experiments (line e) for tubes with helical wire inserts.

A satisfactory agreement is observed between the results of the experiment (line a) and the experimental data on the heat transfer of the oil for which the heat transfer of the gas in the knurled pipe is calculated ($d/D = 0.8$; $t/D = 1.94$) (line i).

The heat transfer of a pipe with a developed surface in the transfer mode significantly exceeds the heat transfer of a smooth pipe as for Hausen (line x) $Nu/Nu_0 = 3/44$. 

4. Conclusion
These facts do not have a complete, theoretical explanation yet; the influence of the asperity step on the current is not studied enough. However, as for the practical use, when deposits occur (for example, in dusty gas flows, where the thickness of the deposits can reach 1 mm), turbulators sink into them and the pipe profile passes to the initial (smooth) form, which does not greatly reduce the heat exchange. Often contaminated pipes with a developed surface work better than dirty smooth ones. The pipes ring knurled, as shown above, very “sensitive” at high Re numbers for an asperity form. This, in particular, explains the spread of data from different authors.

References
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