

## SHELL-AND-TUBE OR PLATE HEAT EXCHANGERS

## PROFILĒTU - RIBOTU CAURUĻU VAI PLĀKSNĪŠU SILTUMMAIŅI

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**The work is devoted to the memory of Mr. Peteris Savelyev, high-class expert and starry-eyed man**

*Key words: heat transfer, hydraulic friction, scales.*

### Introduction

Two kinds of heat exchangers are used in heat supply systems that are: shell-and-tube and plate-type. It is necessary to enhance heat transfer and to decrease pressure losses and scales. Modern plate heat exchangers are often compared with shell-and-tube apparatus, which has no intensifiers of heat transfer. Owing to heat transfer enhancement, it is possible to refine so much characteristics of shell-and-tube apparatus, that they will be comparable or better, then of plate heat exchangers.

Comparison of heat transfer, hydrodynamic and scales characteristics in shell-and-tube heat exchangers with different intensifiers of heat transfer was made in this work. It were chosen for intensification of heat transfer on the base of existing investigations [1-7]: a tube with circular knurl, a tube with spiral knurl, a twisted tube and a dimpled tube.

The calculation of heat and hydraulic characteristics and influence of scales on heat exchange of apparatus with intensifiers was made on the base of the known recommendations based on experimental data. Calculation data was obtained for the conditions of heat supply systems. The heat exchanger was considered in the capacity of an example, consisted of the beam of 37 tubes  $L = 1341$  mm,  $d = 10$  mm, material – steel, thickness of a wall – 1 mm. Parameters for hot heat-carrying agent are: flow rate  $G = 6.2 - 7.5$  t/h, inlet temperature -  $T \approx 70$  °C; parameters for cold heat-carrying agent are: flow rate  $G = 5.3 - 7.1$  t/h, inlet temperature -  $T \approx 11$  °C.

## Choosing of heat transfer intensifiers characteristics

The results of a number of researches were used for choosing heat transfer intensifiers characteristics for heat exchanger in heat supply systems.

*Mr. Peteris Savelyev* researched heat transfer and hydrodynamic in a tube with spiral knurl with different number of starts by  $t/d=3$ , where  $t$  – step of knurl,  $d$  – diameter of tube for a water in a range of Reynolds numbers  $Re = (5 - 50) \cdot 10^3$  in work [1] (Fig. 1). Dependence of ratio Nusselt numbers for spiral shaped and smooth tubes are shown on fig. 2. The data for the tubes with different curvature radius of knurl are identified 2.1 and 2.2 on fig. 2. As it established, the tubes with single-pass knurl are more effective, then double-pass and triple-pass. The ratio of Nusselt number increases with decreasing of Reynold's number. Especially, it is detected for a tube with double-pass knurl and curvature radius 3.5 mm.

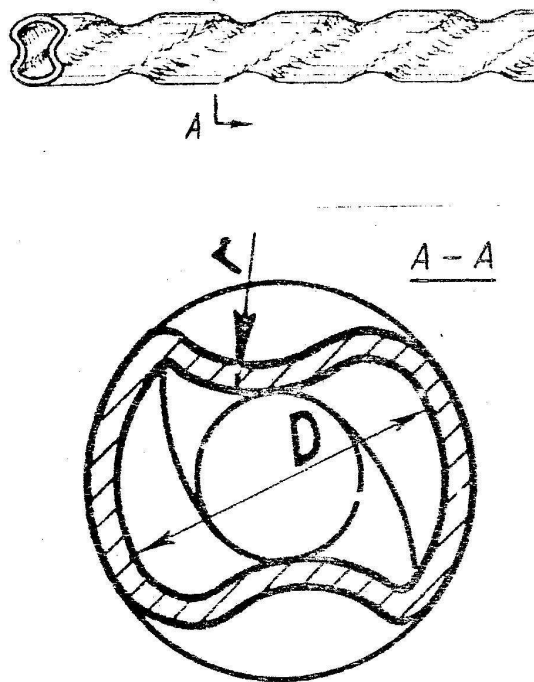


Fig. 1. Spiral shaped tube [1] (tube with spiral knurl)

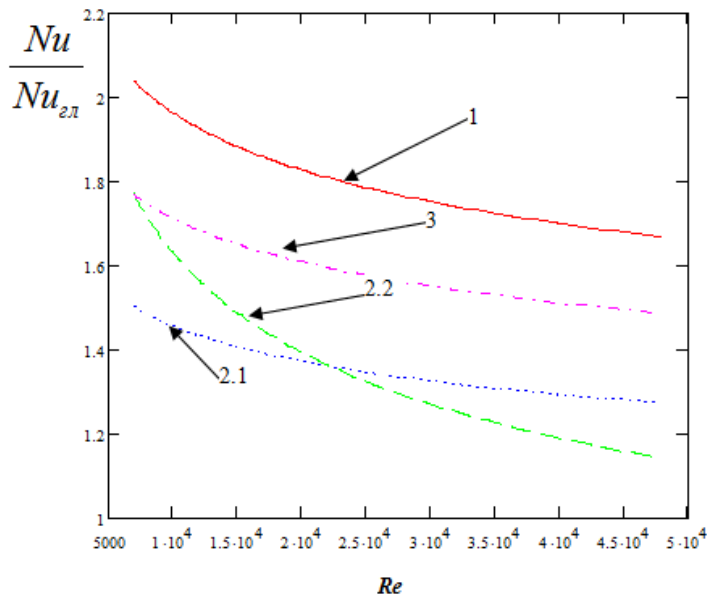


Fig. 2. Dependence of ratio  $\frac{Nu}{Nu_{27}}$  to

Reynold`s number  $Re$ .

- 1 - Tube with single-pass knurl
- 2.1 - Tube with double-pass knurl,  $r=3.2$
- 2.2 - Tube with double-pass knurl,  $r=3.5$
- 3 - Tube with triple-pass knurl

The tube with single-pass knurl with ratio step of knurl to diameter of tube  $t/d=3$  was chosen on the base of results presented in [1].

A choice circular knurl characteristic (Fig.3) was made on the base of Dreytser G.A. recommendations [2]. Owing to conducted calculations, it were chosen: step – 6 mm, depth of knurl – 0.5 mm. In this case sufficiently high heat transfer was obtained with acceptable pressure losses.

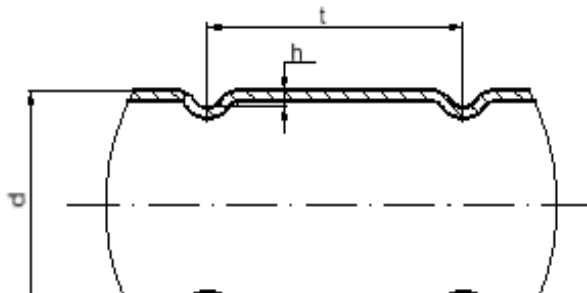


Fig. 4. The tube with circular knurl

A choice twisted tube characteristic (Fig. 4) was made on the base of Dzubenko B.V. research results [3]. It was shown in work [3] that twisted tube with  $S/d=8$  provides essential heat transfer increasing with relatively small hydraulic friction growth.



Fig. 4. Twisted tube

A choice dimpled tube characteristic (Fig. 5) was made on the base of Kiknadze G.I. research results [4]. Owing to recommendations of work [4], it was chosen: longitudinal step – 6 mm, dimple diameter – 3 mm, dimple depth  $h \approx 1$  mm.



Fig. 5. Dimpled tube

### Calculation results

Calculation results of heat transfer, pressure losses and scales in heat exchangers with heat transfer intensificators and without them are presented on Fig.6,7 and 8 for considered conditions. Dependence of ratio heat transfer coefficients to Reynolds number and flow rate of cold heat-carrier in tube space for heat exchangers with heat transfer intensificators and without them are presented on Fig.6 for: a tube with circular knurl, a tube with spiral knurl, a dimpled tube and a twisted tube.

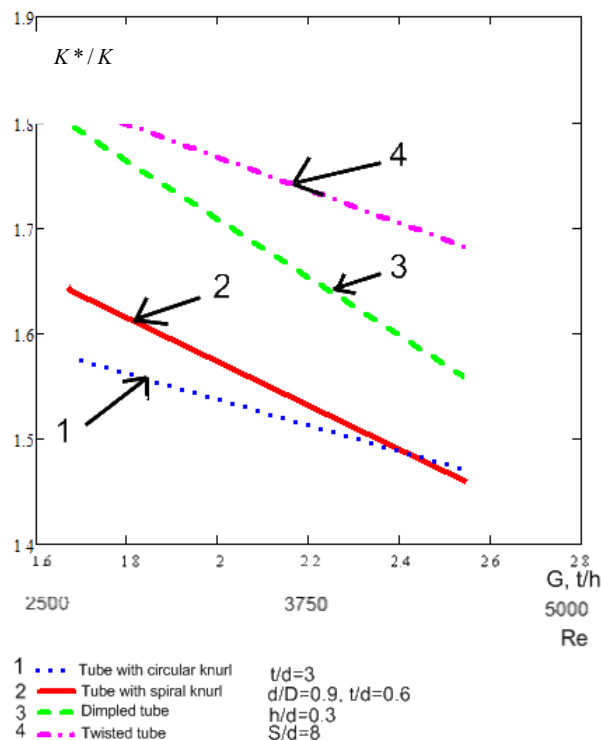


Fig.6. Dependence of heat transfer coefficients ratio to Reynold`s number and flow rate of cold heat-carrier in tube space

As it was acquired, the most growth of heat transfer coefficient (up to 80% against heat exchanger without heat transfer intensificators) was obtained in case of twisted tube. In case of dimpled tube increasing of heat transfer coefficient up to 80% is also observed by low Reynold's numbers, but this effect loses with growth of Reynold's numbers. The tube with spiral knurl and the tube with circular knurl provide a growth of heat transfer coefficient in heat exchanger up to 60%. Essential growth of heat transfer in heat exchangers with investigated heat transfer intensificators is explained, that the data is obtained in the Reynold's number range, agreed to transition zone from laminar to turbulent flow.

Dependence of ratio pressure losses Reynold's number and flow rate of cold heat-carrier in tube space for heat exchangers with heat transfer intensificators and without them are presented on Fig.7 for: a tube with circular knurl, a tube with spiral knurl, a dimpled tube and a twisted tube.

It was estimated, that the most growth of ratio  $\Delta p^*/\Delta p$  (up to 45%) was obtained in the tubes with circular knurl. Spiral knurl reduced to growth of ratio  $\Delta p^*/\Delta p$  up to 35%. The growth of ratio  $\Delta p^*/\Delta p$  up to 25% was obtained for twisted tube, and in case of dimpled tube – 20%.

Time dependence of heat transfer coefficient is presented on Fig. 8 for: smooth tube, twisted tube and tube with circular knurl with carbonate hardness of water  $C = 10$  mg-equiv/l and Reynold's number  $Re=3000$ . Formulas, offered by Dzubenko B.V. in work [5], were used for calculation of changing heat transfer coefficient in time.

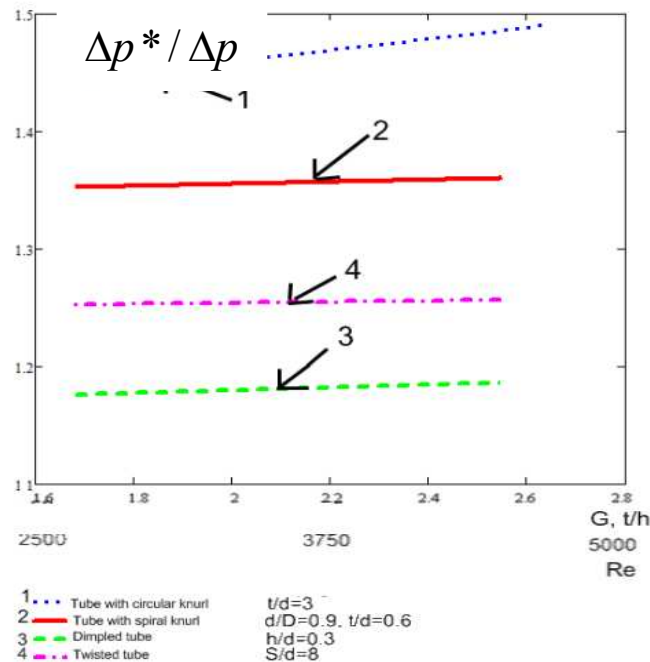


Fig.7. Dependence of ratio pressure losses to Reynolds number and flow rate of cold heat-carrier in tube space

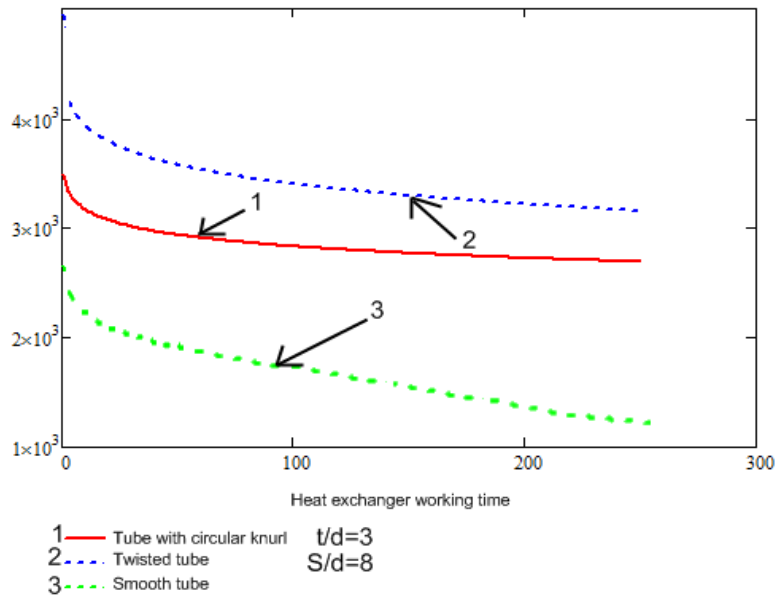


Fig.8. Scales influence on heat transfer coefficient in heat exchanger

As it was acquired, heat transfer coefficient for a smooth tube continuously decreasing in result of scales and for 250 hours decreased in 2,5 times. Heat transfer coefficient for heat exchangers with twisted tubes and tubes with circular knurl in the beginning is decreasing and then it's magnitude is stabilizing. Thereby, efficiency of using turbulence stimulator in heat exchangers with the course of time increases.

## Conclusions

Comparison heat transfer, hydrodynamic and scale characteristics in shell-and-tube heat exchangers with different heat transfer intensifiers was made in this work. For heat transfer intensification it was chosen: a tube with circular knurl, a tube with spiral knurl, a dimpled tube and a twisted tube.

The calculation of heat and hydraulic characteristics and influence of scales on heat exchange of apparatus with intensifiers was made on the base of the known recommendations based on experimental data. Calculation data was obtained for the conditions of heat supply systems in a range of Reynold's numbers  $Re = 2500 - 5000$ .

The most growth of heat transfer coefficient (up to 80% against heat exchanger without heat transfer intensifiers) was obtained in case of using in heat exchanger the twisted tubes. In case of using in heat exchanger the dimpled tubes increasing of heat transfer coefficient up to 80% are also observed by low Reynold's numbers, but this effect looses with growth of Reynold's numbers. The tube with spiral knurl and the tube with circular knurl provide a growth of heat transfer up to 60%.

The most growth of ratio  $\Delta p^*/\Delta p$  (up to 45%) was obtained in the tubes with circular knurl. Spiral knurl reduced to growth of ratio  $\Delta p^*/\Delta p$  up to 35%. The growth of ratio  $\Delta p^*/\Delta p$  up to 25% was obtained for heat exchanger with twisted tubes, and in case of dimpled tubes – 20%.

Heat transfer coefficient for heat exchanger with a smooth tubes with carbonate hardness of water  $C = 10$  mg-equiv/l and Reynolds number  $Re=3000$  continuously decreasing in results of scales and for 250 hours decreased in 2,5 times. Heat transfer coefficient for a twisted tube and tube with circular knurl in the beginning is decreasing and then its magnitude stabilized. Thereby, efficiency of using turbulence stimulator with the course of time increases.

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#### **Kuzma-Kichta Yu.A., [Saveljevs P.A.], Lodvikovs K.M. Profilētu - ribotu cauruļu vai plāksnīšu siltummaiņi**

*Darbā veikta termisko un hidraulisko raksturojumu aprēķināšana, kā arī izmēru ietekmes analīze aparātiem ar siltumapmaiņu intensificējošiem elementiem, pamatojoties uz agrāk zināmu ekeperimentālo pētījumu rezultātiem un rekomendācijām. Izejas dati aprēķiniem tika izvēlēti parametru diapazonā, kas ir raksturīgi siltuma un ūdens apgādes sistēmām. Aprēķinu analīze rāda, ka aplūkotajā Reinoldsa skaitļa ( $Re$ ) diapazonā visaugstākās siltumapmaiņas koeficienta vērtības ir caurulēm ar grumbuļaini profilētu virsmu pie  $Re < 3 \cdot 10^3$  un vītnes profila caurulēm pie vērtībām  $Re > 3 \cdot 10^3$ . Izvēlētajā izmēru diapazonā siltumapmaiņas koeficienta vērtības vītām caurulēm un caurulēm ar ievalcētiem cilindriskiem iedobumiem sākotnēji samazinās, bet pēc tam stabilizējas un paliek aptuveni konstantas, kamēr gludām caurulēm šīs vērtības monotoni samazinās līdz ar ekspluatācijas laiku. Spiediena zudumi vītās caurulēs ir praktiski tādi paši kā gludām caurulēm. Spiediena zudumi caurulēm ar grumbuļaini profilētu virsmu ir mazāki salīdzinājumā ar gludām caurulēm.*

#### **Kuzma-Kichta Yu.A., [Savelyev P.A.], Lodvikov K.M. Shell-and-tube or plate heat exchangers**

*The calculations of heat and hydraulic characteristics and influence of apparatus with intensifiers size analysis to heat exchanger potential of equipment were made onto the basis of known experimental data recommendations. Calculations data were received in a range of parameters, that are typical for a heat and water supply systems.*

*It was obtain, that in studied range of Reynolds's number, the most better heat transfer coefficient value is for the tubes with dimpled interface with  $Re < 3 \cdot 10^3$  and in case a swirl tubes by  $Re > 3 \cdot 10^3$ . Under the stipulation of scales, the heat transfer coefficient for a swirl tube and tube with circular knurl in a beginning is decreasing, and then comes out on a constant, whereas for a smooth tube – monotonically decrease in time. Pressure losses in heat exchanger with twisted tubes are almost the same as in case of smooth tubes. Pressure losses in heat exchanger with dimpled tubes are less than in case of smooth tubes.*

**Кузьма-Кичта Ю.А., Савельев П.А., Лодвиков К.М. Анализ работы теплообменников с профилированными трубами**

Расчёты тепловых и гидравлических характеристик, а также влияния размеров на теплообмен в аппаратах с интенсификаторами теплообмена был сделан на базе известных ранее рекомендаций и основанных на экспериментальные данные. Вычисления были проведены в пределах параметров, которые являются типичными для систем теплоснабжения. В изученном диапазоне чисел Рейнольдса было установлено, что наибольшее значение коэффициента теплопередачи достигается в тех случаях, когда используется труба с покрытыми ямочками поверхностью при  $Re < 3 \cdot 10^3$ . При  $Re > 3 \cdot 10^3$  эффективнее работает спирально закрученная труба. и когда труба водоворота  $Re > 3 \cdot 10^3$ . В рассмотренных режимах коэффициент теплопередачи для спирально закрученной трубы и трубы с круговой насечкой первоначально уменьшается, а затем стабилизируется, тогда как для гладкой трубы – монотонно уменьшается со временем. Потери давления в теплообменнике с кручеными трубами почти такие же, как и в случае с гладкими трубами. Потери давления в теплообменнике с покрытыми радиальными углублениями трубами меньше, чем в случае использования гладких труб.