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MODELING OF THE FLOW DISTRIBUTION INSIDE THE COLLECTORS OF THE HIGH PERFORMANCE HEAT EXCHANGER

MODELOWANIE ROZPŁYWU CIECZY W KOLEKTORACH WYSOKOSPRAWNEGO WYMIENNIKA CIEPŁA

A b s t r a c t

In the paper the influence of the geometrical parameters of the collectors on flow distribution in a high performance heat exchanger is investigated. Using the commercial CFD code – ANSYS CFX software, it was possible to find the geometry that improves significantly flow distribution inside the device in comparison to the traditional design.

Keywords: *high performance heat exchanger*, *collector*, *flow distribution*

S t r e s z c z e n i e

W artykule zaproponowano modyfikacje geometrii stosowanych kolektorów wysokosprawnego wymiennika ciepła, która ma na celu poprawienie warunków pracy tego urządzenia. Przy użyciu programu ANSYS CFX przeanalizowano wpływ parametrów geometrycznych kolektorów wysokosprawnego wymiennika ciepła na rozpływ cieczy do poszczególnych rurek wymiennika ciepła.

Słowa kluczowe: *wysokosprawny wymiennik ciepła*, *kolektor*, *rozpływ cieczy*

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Nomenclature

$v_i^{\max}$ – maximum velocity in the tube of the i^{th} pass of the heat exchanger [m/s]
$v_i^{\min}$ – minimum velocity in the tube of the i^{th} pass of the heat exchanger [m/s]
T – bulk temperature of liquid in a tube [K]
$\dot{m}_{in}$ – mass flow rate of water [kg/s]
p_{op} – operating pressure of water [bar]
T_{in} – water temperature at the inlet [K]
q – heat flux density [W/m^2]
h_t – height of the tube, the collector is formed of [mm]
h_w – height of the wedge [mm]
x – distance between the side of the collector and the nozzle pipe [mm]
r_i – ratio of the maximum and the minimum area averaged velocity in the tubes of the i^{th} pass of the high performance heat exchanger

1. Introduction

High performance heat exchangers are widely used in many industries: the petrochemical, the automotive, the energetic and many others. This kind of the heat exchanger is called in consistency with its structure. The large heat transfer area, leads to high thermal efficiency of the device. Its working principle is to cool rapidly large amount of gaseous or liquid medium. Because of its compact size, it is possible to use it for easy installation in the heating, the drying, the air conditioning and the other systems.

The high thermal performance of this type of the heat exchanger is achievable thanks to the large packed rectangular fins on the tube surface as well as the shape of the tube. The oval tube ensures better distribution of velocity along the circumference during the flow, as well as the lower pressure drop in comparison to the circular tube. To use of the all-aforementioned advantages, it is necessary to ensure the uniformity of nearly uniform distribution of velocity in the all tubes. Not fulfilling this condition leads to large differences in mean temperature in the tubes. Consequently, the excessive thermal stress occurs that may cause the heat exchanger to break down. Small volume of the collectors of the kind of the heat exchangers implicates the possibility of improper flow condition inside the tubes, causing unsuitable distribution of thermal and mechanical loads inside. The images of the damaged tube, Fig. 1, as well as the image of the tube bundle prone to buckling, Fig. 2, prove, that such a failure is possible [1, 2].

This paper presents the analysis of the flow distribution in the high performance heat exchanger with the modified geometry of its collectors. The traditional geometry does not ensure proper velocity distribution in the tubes, especially when it is necessary to supply liquid with the velocity in the nozzle pipe exceeding 3 m/s.

Therefore, some modifications are carried out in order to improve the flow conditions inside the device. In general, the pipe from that collector is cut by half and the perforated bottom with the nozzle pipe is welded to it. However, for the modified construction of the collector, the pipe is rotated at an angle less than 5° and then cut. Next, the wedge is welded to it. This kind of construction is easy to made and cheep. Moreover, it ensures larger space for fluid to flow properly inside the tubes than the typical design. Both types of collectors are presented on Fig. 3.

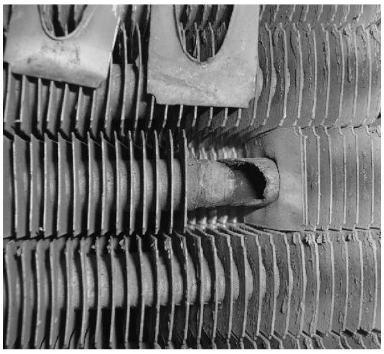

Fig. 1. The damaged tube
Rys. 1. Pęknięta rurka

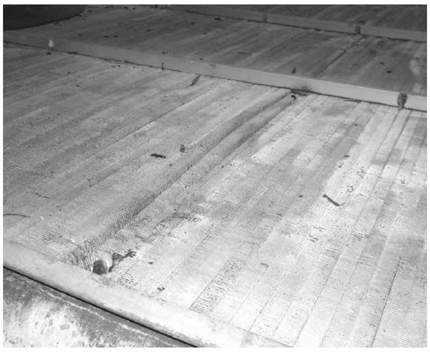

Fig. 2. The buckled tubes of the high performance heat exchanger
Rys. 2. Wyboczone rurki w wysokosprawnym wymienniku ciepła

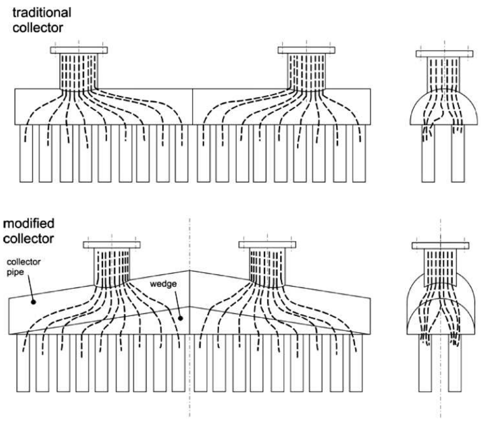

Fig. 3. The traditional and the modified collectors
Rys. 3. Tradycyjny oraz zmodyfikowany kształt kolektora

The investigations carried out using the commercial CFD [3–5] code ANSYS CFX allowed to find the proper geometrical parameters of the modified shape of the collector. The construction ensures significantly better flow distribution inside the tubes compared to the traditional one.

2. High performance heat exchanger working conditions

The two-pass heat exchanger, presented schematically on Fig. 4, is a subject of investigation.

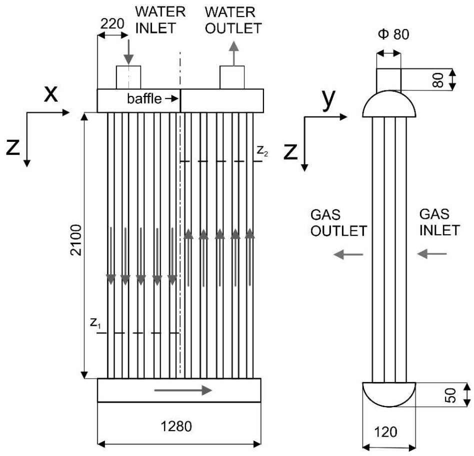

Fig. 4. The scheme of the two-pass high performance heat exchanger
Rys. 4. Schemat dwubiegowego wysokosprawnego wymiennika ciepła

The hot gaseous medium flows perpendicularly to the bundle of the fined elliptical tubes (36 mm × 14 mm × 2 mm), and exchanges its thermal energy with cold water flowing inside the tubes. There are two rows of the tubes. Each of them consists of 42 tubes.

Because the baffle is situated in the middle, water flows through the upper collector into the half of the tubes. Next, it turns round in the lower collector to the tubes in the second pass and finally flows out of the heat exchanger through the upper collector and the outlet nozzle pipe.

The parameters at which the device operates are presented in Tab 1. These parameters imitate the working conditions of the high performance heat exchanger that broke down in one of the Polish companies.

Table 1

The working parameters of the heat exchanger

Flow parameter	Value
$\dot{m}_{in}$ [kg/s]	19
p_{op} [bar]	6
T_{in} [K]	373
q [W/m^2]	60 000

3. The computational procedure

As aforementioned, the typical collector is replaced with the modified one, with the parameterized shape. Three design variables, see Fig. 5, are the height of the tube sector h_t that the collector is formed of, the height of wedge h_w that connects the perforated bottom and the collector, and the distance between the side of the collector and the axis of the nozzle pipe x.

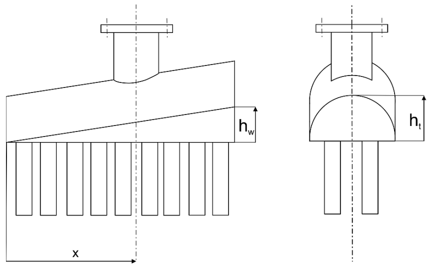

Fig. 5. The geometrical parameters of the modified construction of the collector
Rys. 5. Parametry geometrii zmodyfikowanego kolektora wymiennika ciepła

The constraints of the design variables are presented in Tab. 2.

Table 2

The constraints of the design space

Design variable	Constraint
h_t [mm]	$35 < h_t \leq 60$
h_w [mm]	$40 \leq h_w \leq 70$
x [mm]	$200 \leq x \leq 500$

The hight of the tube sector may be less or equal to its radius, therefore the values presented in Tab. 2 are taken into consideration. Moreover, the total volume of the device should be as small as possible, that ensures the low manufacturing costs, and therefore the h_w value is less than 8% of the collector width. It is also important to find the proper distance x between the inlet nozzle pipe and the side of collector. This parameter influences the flow distribution inside the collector, too.

It is the last constraint that the difference between the temperature of water in the inlet nozzle pipe and the outlet one should not exceed 20 K. This condition is fulfilled for the value of average heat flux density $q =$ 60 000 W/m^2.

The computation is carried out, using the element based finite volume method. The example discredited domain, for h_t = 60 mm, h_w = 70 mm and x = 400 mm is shown on Fig. 6.

The ANSYS CFX software solves simultaneously the mass, the momentum and the energy conservation equations for all the finite elements inside the computational domain. The standard $k - \varepsilon$ model is used to model the turbulence effect.

The objective function is to minimize the ratio of the maximum and the minimum area averaged velocities in the tubes for the first and the second pass:

$$r_1(h_t, h_w, x) = \frac{v_1^{\max}}{v_1^{\min}} \to \min \tag{1}$$

$$r_2(h_t, h_w, x) = \frac{v_2^{\max}}{v_2^{\min}} \to \min \tag{2}$$

The area-averaged velocities v are computed for the tubes in the first and the second pass for the subsequent values of the presented "z" coordinate (the distance between the upper perforated bottom and the cross section of the tube) – z_1 = 1800 mm and z_2 = 300 mm. These values are chosen, because it is necessary to find the stationary flow conditions inside the tubes, not disturbed by inlet effect, when liquid flows in to the tube.

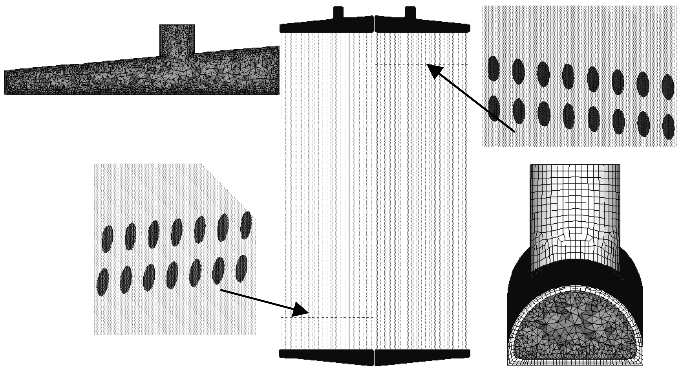

Fig. 6. The high performance heat exchanger divided into finite volumes
Rys. 6. Wysokosprawny wymiennik ciepła podzielony na objętości kontrolne

Because the heat transfer coefficient of liquid depends strongly on velocity, the smaller r_1 and r_2 values are, the better the thermal effectiveness of the device is being obtained. Moreover, it also ensures lower thermal loading of the construction, and therefore it is less prone to the damage.

4. Results and discussion

The streamlines, that illustrates flow distribution in the high performance heat exchanger with traditional collectors, are presented on Fig. 7. It is possible to observe that only part of the tubes is fed properly. The blank areas indicate the zones where the flow doesn't exist at all, or liquid flows with very low velocity, that may lead to liquid overheating and its phase change.

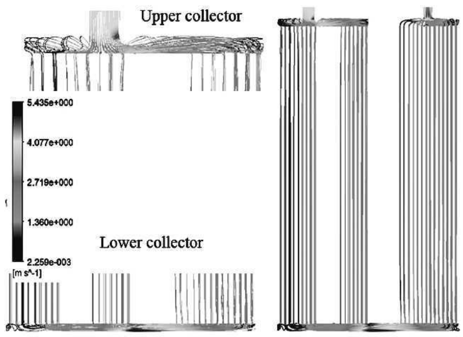

Fig. 7. The flow distribution in high performance heat exchanger with traditional collector
Fig. 7. Rozpływ cieczy w wysokosprawnym wymienniku ciepła z tradycyjnym kolektorem

Aiming to overcome this problem, the modified construction is being tested. As mentioned before, the computation is carried out to check which configuration of the geometrical parameters h_t, h_w and x ensures the best flow distribution in all the tubes in the first and the second pass.

The first results of computation are shown on Fig. 8 and Fig. 9. The values of r_1 and r_2 are presented as a function of x and h_t. The value of h_w is 40 mm.

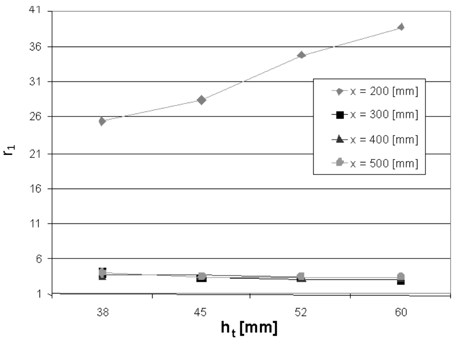

Fig. 8. The values of r_1 as a function of x and h_t for h_w = 40 mm
Rys. 8. Zmienna r_1 w funkcji x oraz h_t dla h_w = 40 mm

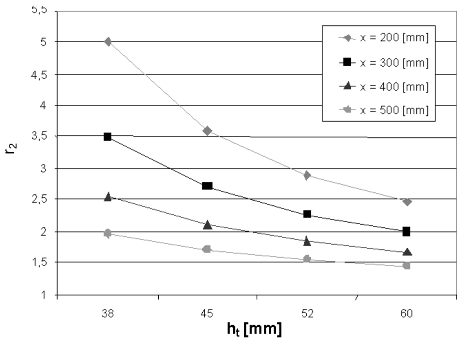

Fig. 9. The values of r_2 as a function of x and h_t for h_w = 40 mm
Rys. 9. Zmienna r_2 w funkcji x oraz h_t dla h_w = 40 mm

It is possible to observe on Fig. 8, that the flow distribution in the first pass is improper especially for the smallest value of x parameter: 200 mm. The r_1 value exceeds 25, that means, that liquid flows very fast in some tubes and the mass flow rate is very low in the others. As mentioned before, this may result in the phase change of liquid due to overheating.

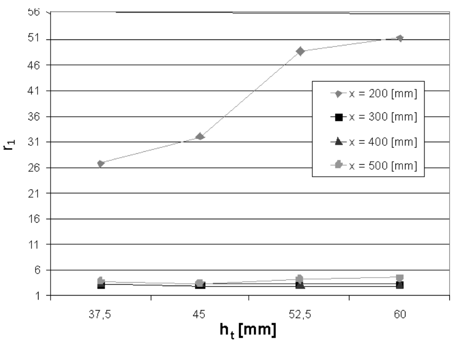

Fig. 10. The values of r_1 as a function of x and h_t for h_w = 40 mm
Rys. 10. Zmienna r_1 w funkcji x oraz h_t dla h_w = 50 mm

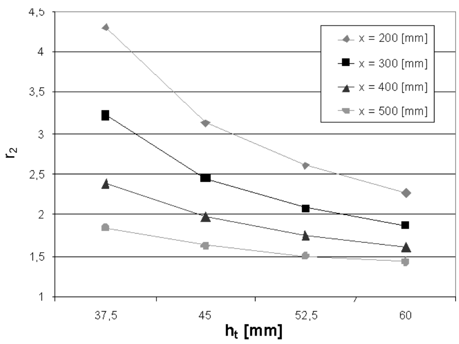

Fig. 11. The values of r_2 as a function of x and h_t for h_w = 40 mm
Rys. 11. Zmienna r_2 w funkcji x oraz h_t dla h_w = 50 mm

It is also worth to mention, that despite the difference between the outlet and the inlet mass flow averaged temperatures does not exceed 20 K, and the automatic device, controlling this difference indicates that the heat exchanger works properly, the tubes may still break down due to excessive thermal loadings.

The values of r_2 for different x and h_t are presented on Fig. 9. It is possible to observe, that the values of r_2 are significantly lower that r_1. It means, that it is easier to ensure better flow distribution into the tubes in the second pass of the heat exchanger. The r_2 value decreases with the increase of x and h_t.

Figures 10 and 11 present the values of r_1 and r_2 as a function of x and h_t. The value of h_w is 50 mm.

Figures 12–15 present the values of r_1 and r_2 as a function of x and h_t. The value of h_w is 60–70 mm.

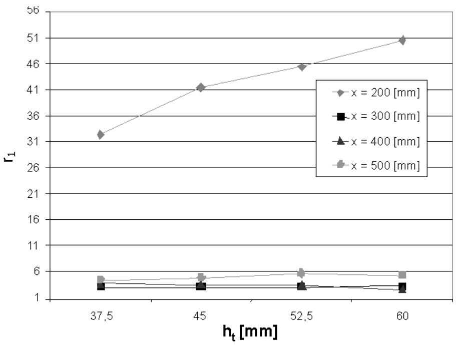

Fig. 12. The values of r_1 as a function of x and h_t for h_w = 60 mm
Rys. 12. Zmienna r_1 w funkcji x oraz h_t dla h_w = 60 mm

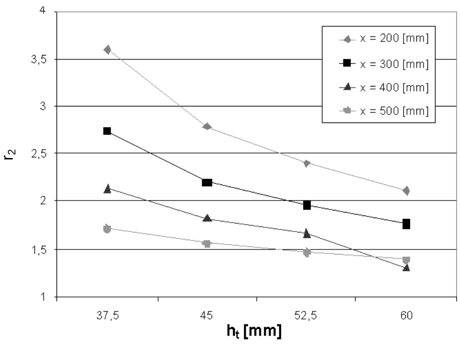

Fig. 13. The values of r_2 as a function of x and h_t for h_w = 60 mm
Rys. 13. Zmienna r_2 w funkcji x oraz h_t dla h_w = 60 mm

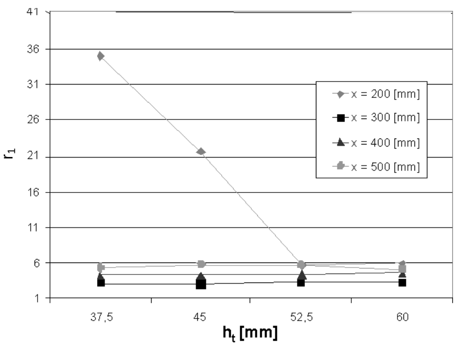

Fig. 14. The values of r_1 as a function of x and h_t for h_w = 70 mm
Rys. 14. Zmienna r_1 w funkcji x oraz h_t dla h_w = 70 mm

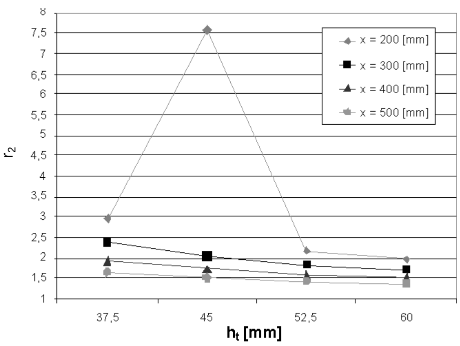

Fig. 15. The values of r_2 as a function of x and h_t for h_w = 70 mm
Rys. 15. Zmienna r_1 w funkcji x oraz h_t dla h_w = 70 mm

For h_w = 60 mm (see Fig. 12–13) it is possible to observe, that the increase of h_t slightly improves the r_1 and r_2 values for x = 300 mm, x = 400 mm and x = 500 mm. However in the case of x = 200, the r_1 value worsens significantly.

Additionally it is possible to observe that the increase in h_w significantly improves the r_1 and r_2 values for x = 300 mm and x = 400 mm. It is also worth to note that for the parameters: x = 400 mm and h_t = 60 mm the obtained values of r_1 = 2.7 and r_2 = 1.3 are smaller than for all solutions presented before.

The values of r_1 and r_2 for h_w = 70 mm are presented on Fig. 14 and 15. It is possible to observe, that the further increase in the h_w doesn't improve the overall solution.

The bulk temperature distribution for the investigated control areas of the tubes for the first and the second pass of the high performance heat exchanger is presented on Fig. 16 and Fig 17.

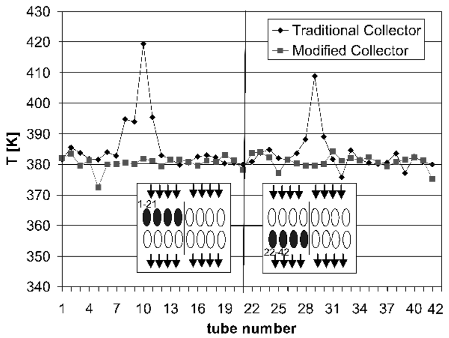

Fig. 16. The bulk temperature distribution in tubes in the first pass of the heat exchanger

Rys. 16. Rozkład temperatury czynnika w rurkach pierwszego biegu wymiennika

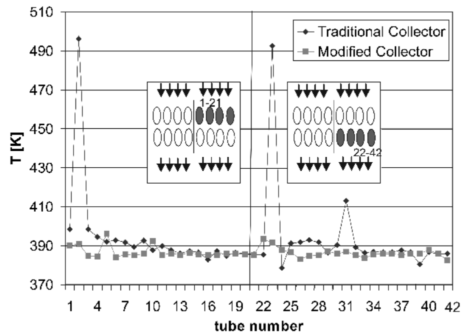

Fig. 17. The bulk temperature distribution in tubes in the second pass of the heat exchanger

Rys. 17. Rozkład temperatury czynnika w rurkach drugiego biegu wymiennika

The comparison between the typical design of the collector and the best design of the modified one (x = 400 mm, h_t = 60 mm, h_w = 60) mm shows, that the temperature difference inside the tubes is significantly lower for the proposed solution. It means, that there is no danger in liquid to overheat, as for the typical design, where liquid temperature inside the tube exceeds the saturation temperature for water at the 6 bar operation pressure.

5. Conclusions

In the presented work, the modified design of the collector of the high performance heat exchanger is presented. The study on its geometrical parameters was carried out. The best design of the modified collector improves significantly the flow distribution inside the tubes and consequently the heat transfer condition for the device in comparison to the typical design.

In the future, to prove that the construction is save to use, it is necessary to investigate the stress distribution inside the high performance heat exchanger with the modified collectors. In the investigation, the emphasis should be put on the connection between the perforated bottom and the tubes, where the stress level might be higher than in the other parts of the construction.

References

[1] Ł o p a t a S., O c ł o ń P., *Investigation of the flow conditions in high performance heat exchanger*, Archives of Thermodynamics, Vol. 31(2010), No. 3, Wydawnictwo IMP, Gdańsk 2010, 37-53.

[2] O c ł o ń P., Ł o p a t a S., *Wpływ rozpływu cieczy na w wysokosprawnym wymienniku ciepła na jego warunki pracy*, Praca zbiorowa: *Systemy, technologie I urządzenia energetyczne*, pod red. J. Taler, t. 2, Wydawnictwo Politechniki Krakowskiej, Kraków 2010.

[3] C h u n g T.J., *Computational Fluid Dynamics*, Cambridge Univ. Pres, Cambridge 2002.

[4] B l a z e k J., *Computational Fluid Dynamics Principles and Applications*, ELSEVIER, Oxford 2001.

[5] L e w i s R.W., N i t h i a r a s u P., S e e t h a r a m u K.N., *Fundamentals of the finite element method for heat and fluid flow*, Wiley, West Sussex, England 2004.