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PERSPECTIVE WAYS OF HEAT TRANSFER ENHANCEMENT FOR HEAT EXCHANGERS IN FOULING CONDITIONS

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Abstract: *The paper discusses the possibilities of intensifying the heat transfer of liquid working fluids in heat exchangers under fouling conditions. It presents a perspective on heat exchangers according to the geometry of the heat exchanger surface. The shell and tube heat exchanger with the helical flow and plate heat exchanger with the spiral flow are introduced in greater detail. These two heat exchangers are compared concerning fouling and thermal-hydraulic behavior in industrial cases. The paper shows clear advantages of these flow structures for improving these aspects. The article places a strong recommendation on the use of spiral plate heat exchangers wherever the working conditions allow.*

Keywords: Heat transfer, fouling, thermal-hydraulic analysis, heat exchangers, spiral, helical.

1. Introduction

The research on heat exchangers currently focuses on optimizing known heat exchanger (HE) geometries. In addition, the thermal–hydraulic aspects of the use of nanoparticles in the working fluid are being investigated. The HE geometry can also be used as a criterion for dividing heat exchangers. The basic division of heat recovery HEs is according to the heat exchange surface based on tubular and plate. State-of-the-art heat exchanger designs are microchannel HEs. Opposite to these research trends are working fluids that exhibit significant heat transfer surface fouling. Fouling is a time-dependent phenomenon that negatively affects the thermal-hydraulic behavior of heat exchangers. In this work, the focus lies on liquid working fluid with fouling properties.

2. Tubular heat exchangers in fouling conditions of working liquids

For fouling conditions of liquid working fluids, shell and tube heat exchangers (STHEs) are commonly used. The deployment of SHTEs requires a more detailed structural design of the aiming. They are used for the most demanding applications in terms of pressure and temperature. A negative phenomenon is the formation of so-called dead zones, on the shell side close to segmental baffles (see Fig. 1). In dead zones, there is no required flow velocity, and therefore sufficient turbulence does not occur. Consequently, the heat exchange surface becomes fouled, thus impairing heat transfer, and leading to uneven structure loading.

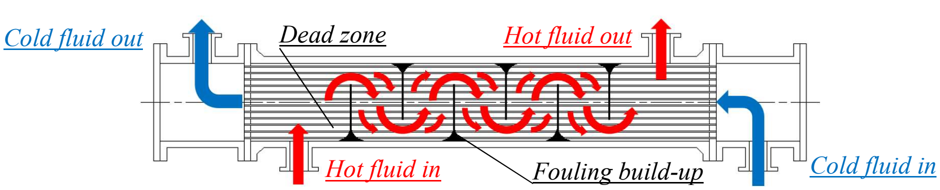

Fig. 1: Principle of shell-side fluid flow and dead zones in STHE.

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The following Tab. 1 shows the process data for the operation of the proposed STHE for cooling the vacuum side distillate pump around (VSDPA) stream by atmospheric residue (ATRES) stream to be heated.

Working fluid	VSDPA	ATRES
Side	Tube side	Shell side
Mass flow	31.2 kg/s	55.6 kg/s
Inlet temperature	274.3 °C	223.8 °C
Fouling factor	0.00053 $m^2 \cdot °C/W$	0.00088 $m^2 \cdot °C/W$

Tab. 1: Process data on STHE.

For the process data in Tab. 1, a thermo-hydraulic design of the STHE has been made considering the prescribed pressure losses, the working fluids, heat duty, and the position constraint, to get to basic design dimensions. Tab. 2 shows the main design parameters of the STHE for the shell side on which the fouling working fluid (ATRES) flows.

Shell internal diameter	**Tube length**	**Baffle spacing**	**Pressure drop**
1 200 mm	6 327 mm	516 mm	87 020 Pa

Tab. 2: Main results of shell side design data for STHE design.

There are several possibilities for heat transfer intensification in STHE (Akpomiemie and Smith, 2016). Intensification is implemented both on the shell side and on the tube side. On the tube side, these are mainly twisted tapes and coiled wires. Changing a baffle system can intensify heat transfer on the shell side. By tilting the baffles under a certain angle (5 ° to 45 °), a helical bypass of tubes is achieved (Stehlík et al., 1991), see Fig. 2.

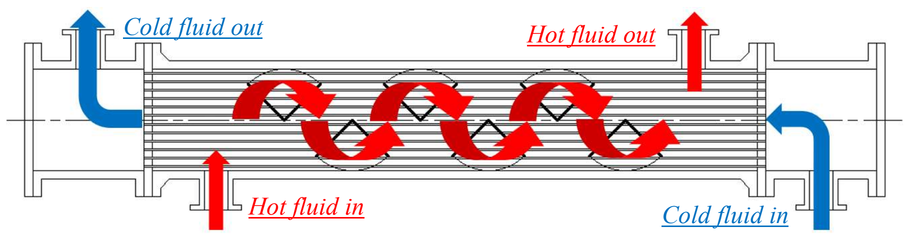

Fig. 2: Principle of inclined baffles and shell-side helical fluid flow in STHE.

According to the process data in Tab. 1, the optimal geometry of STHE with helical baffles was designed to minimize fouling on the shell side. Tab. 3 shows its main design data and the pressure drop on the shell side.

Shell inner diameter	**Tube length**	**Baffle inclination**	**Pressure drop**
1 200 mm	5 800 mm	15 °	81 894 Pa

Tab. 2: Main results of shell side design data for STHE design.

3. Plate heat exchanger

Tubular heat exchangers are predominant in process applications. However, they can be replaced by plate heat exchangers (PHE) with profiled plates, especially in processes where lower temperature and pressure are involved. The reason for choosing PHEs in new designs is their modular construction. Their deployment in processes with fouling agents has faced a lot of scrutiny.

In addition to the above-mentioned profiled plate design, PHEs also include spiral plate heat exchangers (SPHE). Spiral plate heat exchangers have, according to the manufacturer, a self-cleaning effect. The curvature of the channels of working fluids creates great turbulence along their entire length. If sediments appear, despite high turbulences, they are swept away because of the increased local velocity, i.e. the self-cleaning effect. A schematic of the SPHE is shown in Fig. 3.

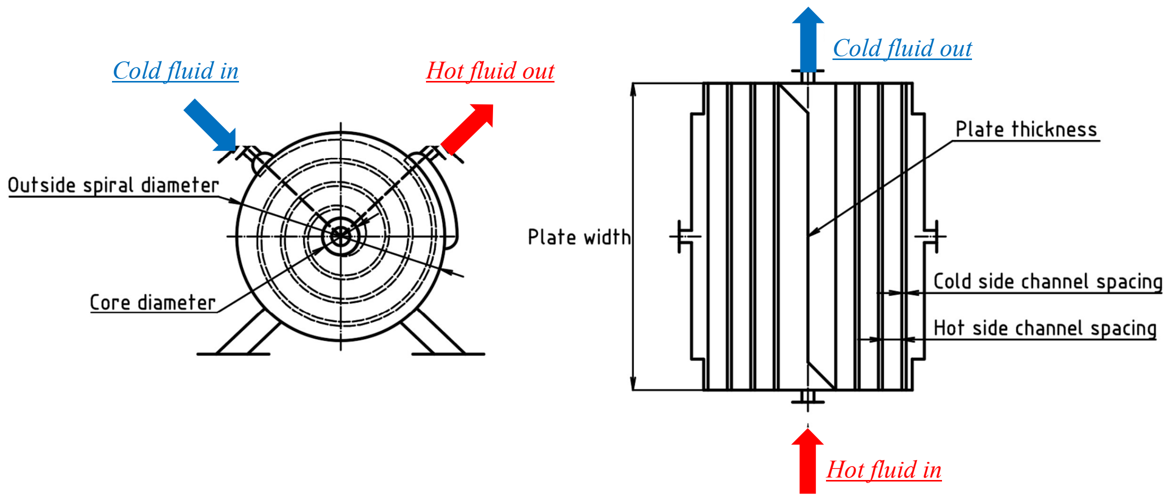

Fig. 3: Schematic drawing of a spiral plate heat exchanger.

Fig. 3 shows the principle of the SPHE function with a description of the basic geometry. The hot working fluid enters through the center of the unit and flows through a channel from the inlet toward the shell. The cold working fluid enters radially and flows towards the center. The arrangement of the working fluids in the SPHE is countercurrent, hence the correction factor to determine the mean temperature difference, $f = 1$ (Moretta, 2010). The heat exchange surface of the curved plate can be fitted with pins, which contribute to the intensification of heat transfer. To illustrate the characteristics of the SPHE, process data from an industrial case of heat recovery from hot wastewater for a 210 kW SPHE are presented in Tab. 4.

Working fluid	Hot wastewater	Cold feedwater
Mass flow	120 kg/s	20 kg/s
Inlet temperature	~25.4 °C	10 °C
Outlet temperature	25 °C	~12.6 °C
Fouling factor	0.000088 m^2.K/W	0 m^2.K/W

Tab. 4: Process data on SPHE.

The procedure in (Moretta, 2010) can be followed for SPHE design. In this procedure, the unknown is the heat transfer surface. The size of the heat exchange area is designed with an overdesign of 20 to 30 percent (Moretta, 2010). From the known width of the plate, the minimum length of the plate is determined and rounded to a value within the given overdesign range. In the present case, the overdesign is 28 %. The length of the plate is calculated using the Eq. (1) the outer diameter of SPHE, which for given process data together with other basic dimensions is given in Tab. 5 and recorded in Fig. 3.

$$D_S = \sqrt{15.36 \cdot L \cdot (S_h + S_c + 2 \cdot t) + C^2} \tag{1}$$

	Plate		**Channel spacing**		**Diameter**	
Thickness	**Width**	**Length**	**Hot side**	**Cold side**	**Core**	**Outside**
t	w	L	S_h	S_c	C	D_S
3 mm	1 000 mm	4 500 mm	60 mm	10 mm	125 mm	688 mm

Tab. 5: Main results for SPHE design.

4. Discussion of results

The two optimally designed STHEs presented in Chapt. 2 are identical in design except for the baffle system on the shell side. By changing the baffle system, fouling is reduced, and heat transfer is intensified. Both designs were implemented to allow the STHE to transfer around 2.7 MW of heat. To achieve the same heat duty, a smaller heat transfer area is required in the case of the STHE with helical baffles, resulting in a reduced heat exchanger length (approx. 10 %). Comparing Tabs. 2 and 3, the pressure drop is reduced due to all the factors mentioned above in favor of the STHE with helical baffles.

The pressure drop of working fluids in the presented SPHE is calculated according to Eq. (2). Eq. (2) accounts for the fitting of 60 x 60 mm intensification pins on the heat exchange surface (Moretta, 2010). The pressure loss of both waste and feed water is around 15.3 kPa. The identity of achieved losses results from achieving the same flow velocity ($v = 2\ \mathrm{m/s}$) in differently sized channels (see Tab. 5), and at approximately the same densities (ρ) of both liquids.

$$\Delta P = \frac{1.45 \cdot (L \cdot v^2 \cdot \rho)}{1.705} \tag{2}$$

Lastly, a comparison of SPHE and STHE with helical baffles is made with the support of the in-house calculation program based on (Stehlík et al., 1991). The design of the new STHE with helical baffles is based on the case solved in Chap. 2 so that it meets the process data for SPHE in Tab. 4. The new inner diameter of the shell and length of the tubes have been set to maintain approximately the original ratio of these two dimensions and to achieve the stated heat duty of 210 kW.

$$p = 2 \cdot \sqrt{2} \cdot D \cdot \tan(\varphi) \tag{3}$$

Subsequently, according to Eq. (3), taken from (Stehlík et al., 1991), the baffle spacing (p) was calculated for the angle of baffle inclination (φ), which should be in the range of 40 to 60 percent of this value (Stehlík et al., 1991). The resulting dimensions of the STHE with helical baffles are given in Tab. 6 along with the pressure loss data.

Shell inner diameter	**Tube length**	**Baffle inclination**	**Baffle spacing**	**Pressure drop**	
				Shell	**Tubes**
400 mm	2 000 mm	15 °	150 mm	88 254 Pa	7 889 Pa

Tab. 6: Main results of shell side design data for STHE design.

The cold feedwater is placed on the tube side and the hot fouling wastewater on the shell side. The hydraulic loss of the fouling working fluid is lower in the SPHE than in the case of STHE with helical baffles.

5. Conclusions

The paper presents perspective ways to increase heat transfer in heat exchangers operating in fouling conditions of liquid working fluids. Industrial examples demonstrate these intensification methods. From the examples presented, it is possible to establish a clear superiority of spiral plate heat exchangers in terms of fouling rate, built-up area, and above all, pressure losses. The positive influence of helical and spiral flow on fouling and thermal-hydraulics behavior has also been demonstrated. Therefore, it seems promising to deal with such types of flows in the future and to investigate their influence on the presented aspects also for non-liquid working fluids and their operating conditions.

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