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ANALYSIS OF WATER VAPOR CONDENSATION MODELS AT HIGH MASS FLUX DENSITIES INSIDE HELICAL TUBES

Summary

This paper presents an analysis of condensation models inside helical tubes. Due to the lack of reliable water vapor condensation models for high mass flux densities in helical tubes, in the paper is proposed usage of standard modification that correlate condensation heat transfer coefficient inside vertical tubes with condensation heat transfer coefficient inside helical tubes through enhancement factor. Although this modification is originally introduced for application without phase changes, in present work is shown that it can be also applied for phase changing application. Furthermore, modification was applied to two reliable condensation models for vertical tubes which are than compared by using both local and averaged approach for different mass flux densities.

Key words: *heat exchanger, helical tube, condensation, water vapor, mass flux density*

1. Introduction

Heat exchangers are used in a wide variety of applications including power plants, nuclear reactors, refrigeration and air-conditioning systems, automotive industries, heat recovery systems, chemical processing and food industries [1]. They can be also used in small-scale CCHP systems since such systems are very delicate regarding optimization of techno-economic parameters [2]. To meet the requirements of energy saving, size reduction and subsequent cost reduction of heat exchangers, many different heat transfer enhancements are available. One possible way of enhancing heat transfer rate, reducing the size and providing compactness in volume of the heat exchangers is by using helical tubes [3]. One of important advantages coming from usage of helical tubes is an excellent behavior in presence of thermal expansions. Helical tubes (Fig. 1) allow heat exchanger to behave as a spring, thus accommodating the stresses due to the thermal expansion [4]. In such analyses a dynamic model of helical springs can be used because pressure of working fluids is often time dependent [5]. In a last few decades heat exchangers with helical tubes have had a wide application in district heating systems (Fig. 2). They are used in district heating substations either as heaters in presence of hot water as a working medium, or as condensers in presence of water vapor as a working medium. The latter is the most common in the last years. In such applications, water vapor condenses inside helical tubes while around tubes is heated water. Although heat exchangers with helical tubes are widely used for more than few decades, tube-side filmwise condensation heat transfer coefficient in helical tubes is still not fully explored.

When fluid is flowing through a helical tube without phase change, the flow pattern is substantially different by comparison with the flow through straight tube, i.e. there is a velocity profile distortion due to the curvature of the helical tube. The reason is that in helical tubes a fluid is under influence of centrifugal mass forces which produce secondary flow of the fluid. Simultaneous influence of the friction on the tube wall and centrifugal mass forces results in formation of two vortices inside a helical tube which are carried by the primary fluid flow. Such combination of the primary and secondary flows leads to the enhancement of the heat transfer coefficient inside helical tubes, especially in laminar flow regimes where the fluid velocities and therefore heat transfer rates are both lower.

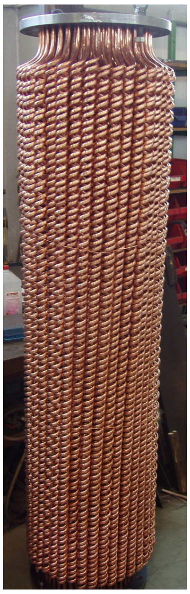

Fig. 1 Tube bundle from helical tubes

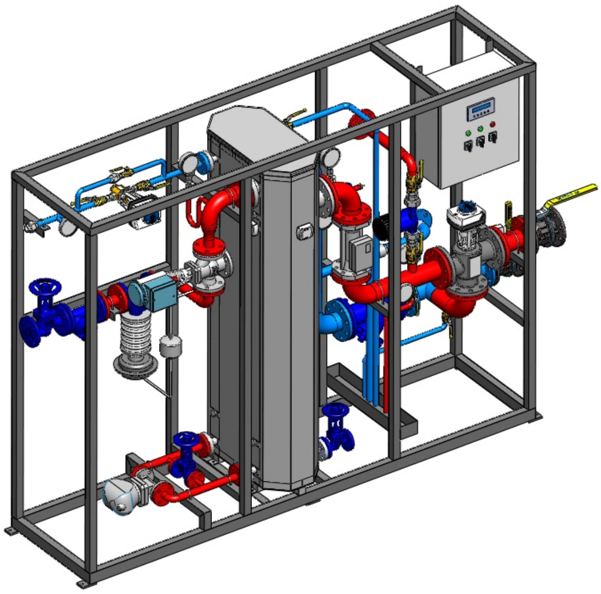

Fig. 2 District heating compact substation

Similar phenomenon experience a fluid in helical tube during a two phase flow. Significant difference comes from the fact that, in case of condensation, the secondary flow has increases the contact surface inside the tube between the vapor and the tube wall. This leads to a significant increase of heat transfer coefficient of condensation inside helical tubes.

Since the refrigeration field became very popular in the last few decades, helical tubes have been extensively studied and used in refrigeration applications. Therefore, there are a number of works in open literature dealing with condensation of refrigerants [3, 6, 7], but rare are those dealing with condensation of water vapor inside helical tubes. To the best of the author's knowledge, there has been one relevant work, carried out by Mohamed A. Abd Raboh et al. [4] dealing with condensation of water vapor inside helical tube. In that work an experimental study was done to investigate the influence of different operating parameters on the condensation heat transfer coefficient for water vapor flows inside helical tube. The obtained results showed that optimum operating parameters in the studied operating range are; tube inside diameter 4.95 mm, helical coil diameter 100 mm, helical coil pitch 40 mm and inclination angle for coil 45°. In the work authors also give an empirical correlation for Nusselt number as a function of Reynolds number and the examined operating parameters ($40 < Re < 230$, mass flux density < 20 kg/(m^2 s)).

In the absence of correlation for water vapor condensation heat transfer coefficient in helical tubes at mass flux densities higher than 20 kg/(m^2 s), the present paper proposes standard modification procedure for condensation heat transfer coefficients inside helical tubes. This modification procedure correlate condensation heat transfer coefficient inside vertical tubes with condensation heat transfer coefficient inside helical tubes through enhancement factor introduced in [8]. Although this correlation is originally used for helical tube heat exchangers without phase changes, it is shown that it can still be used in condensation application modeling. Even that a number of filmwise condensation heat transfer correlations for vertical tubes have been proposed, their predictions are often inconsistent [9], especially for larger mass flux densities. D. Papini and A. Cammi [10] also reported that well-known correlation by Shah [11] can be used for modelling of in-tube filmwise condensation, noting that its validity is questionable in certain operating conditions.

Therefore, among many correlations that have been checked, two relevant correlations which are compared in this paper are those proposed by D. Papini and A. Cammi in his work [10], and correlations from VDI Heat atlas [12] as the most relevant models dealing with this topic. The comparison of the results from those relevant correlations for the in-tube filmwise condensation models within vertical tubes is carried out through calculation of both local and averaged Nusselt numbers.

2. Modified heat transfer model for water vapor condensation in helical tubes

Many researchers have investigated in-tube condensation for vertical and horizontal tubes, but very few works have been performed to study the condensation within helical tubes in presence of water vapor as a working medium. In the absence of heat transfer coefficient models for filmwise water vapor condensation in helical tubes at mass flux densities higher than 20 kg/(m^{2}s), this work proposes modification of filmwise condensation model inside vertical tubes for heat transfer coefficient calculation inside helical tubes.

The proposed modification, introduced in [8], correlate heat transfer coefficients for water vapor condensation in vertical and helical tubes, according to the following expression,

$$\alpha_{\mathrm{hel}} = \left(1 + 3.5\,\frac{d_{\mathrm{i}}}{D_{\mathrm{H}}}\right) \cdot \alpha_{\mathrm{vert}} \tag{1}$$

where:

α_{hel}, W/(m^2 K)	- heat transfer coefficient for water vapor condensation in helical tubes
α_{vert}, W/(m^2 K)	- heat transfer coefficient for water vapor condensation in vertical tubes
d_{i}, (m)	- inside diameter of helical tube
D_{H}, (m)	- mean helical coil diameter

The term in the brackets is enhancement factor that accounts for heat transfer coefficient enhancement because of the influence of secondary flows appearing in helical tubes. This term is tested on applications dealing with heat transfer between working media without phase change. Therefore it is not obvious at the moment that the term is valid in heat transfer problems involving condensation. Still, thorough analysis of open literature dealing with refrigerants condensation shows very good agreement between enhancement factor calculated according to the expression (1) and measured heat transfer coefficient enhancement in the given literature, e.g. [6], [7]. In the [7] the agreement between heat transfer coefficient enhancement and enhancement factor according to (1) is within 4 %, while in [6] is within 1 %. It is important to emphasize that this analysis is based on the experiments involving refrigerants as the working medium. Identical analysis for condensing water vapor couldn't be made because there is a lack of suitable experimental data in the literature. Still, there is a strong indication to use eq. (1) as a general enhancement factor in helical tube condensation applications.

3. Heat transfer models for water vapor condensation in vertical tubes

In-tube filmwise condensation in vertical and horizontal tubes has been widely studied and reviewed in the past. However, W. Xiaoyong et al. [9] showed that their predictions are often inconsistent. Therefore, in the present paper is performed comparison of the water vapor condensation in helical tubes based on two relevant models that provide realistic values of filmwise condensation heat transfer coefficients specified in literature [10]. The models are those proposed by D. Papini and A. Cammi in their work [10], and correlations from VDI Heat atlas [12]. Both models provide correlations for calculation of local and average filmwise condensation heat transfer coefficients in case of a downflow streaming.

Local Reynolds number in both models is defined as follows:

$$Re_{\mathrm{f,x}} = \frac{q_{\mathrm{m}}\left(1-x\right)}{\pi\, d_{\mathrm{i}}\, \mu_{\mathrm{f}}} \tag{2}$$

According to VDI Heat atlas [12], correlations for local Nusselt numbers (laminar, turbulent and overall) are defined as follows:

$$Nu_{\mathrm{x,l}} = \frac{\alpha_{\mathrm{x,l}}\, \mathit{\Gamma}}{\lambda_{\mathrm{f}}} = 0.693\left(\frac{1-\rho_{\mathrm{g}}/\rho_{\mathrm{f}}}{Re_{\mathrm{f,x}}}\right) \tag{3}$$

$$Nu_{\mathrm{x,t}} = \frac{\alpha_{\mathrm{x,t}}\, \mathit{\Gamma}}{\lambda_{\mathrm{f}}} = \frac{0.0283\, Re_{\mathrm{f,x}}^{7/24}\, Pr_{\mathrm{f}}^{1/3}}{1+9.66\, Re_{\mathrm{f,x}}^{-3/8}\, Pr_{\mathrm{f}}^{-1/6}} \tag{4}$$

$$Nu_{\mathrm{x}} = \sqrt{\left(f_{\mathrm{val}}\, Nu_{\mathrm{x,l}}\right)^{2} + Nu_{\mathrm{x,t}}^{\;2}} \tag{5}$$

Correlations for average Nusselt numbers (laminar, turbulent and overall) are defined as follows:

$$Nu_{\mathrm{m,l}} = \frac{\alpha_{\mathrm{m,l}}\, \mathit{\Gamma}}{\lambda_{\mathrm{f}}} = 0.925\left(\frac{1-\rho_{\mathrm{g}}/\rho_{\mathrm{f}}}{Re_{\mathrm{f}}}\right)^{1/3} \tag{6}$$

$$Nu_{\mathrm{m,t}} = \frac{\alpha_{\mathrm{m,t}}\, \mathit{\Gamma}}{\lambda_{\mathrm{f}}} = \frac{0.02\, Re_{\mathrm{f}}^{7/24}\, Pr_{\mathrm{f}}^{1/3}}{1+20.52\, Re_{\mathrm{f}}^{-3/8}\, Pr_{\mathrm{f}}^{-1/6}} \tag{7}$$

$$Nu_{\mathrm{m}} = \sqrt[1.2]{\left(f_{\mathrm{val}} Nu_{\mathrm{m,l}}\right)^{1.2} + Nu_{\mathrm{m,t}}^{\;1.2}} \tag{8}$$

D. Papini and A. Cammi, [10], for local condensation heat transfer coefficient calculation recommended the Kutateladze correlation:

$$\alpha_{\mathrm{x}} = \lambda_{\mathrm{f}}\left(\frac{\nu_{\mathrm{f}}^{2}}{g}\right)^{-1/3} \cdot 0.756\, Re_{\mathrm{f,x}}^{-0.22} \tag{9}$$

Kutateladze correlation for average condensation heat transfer coefficient is defined as follows:

$$\alpha_{\mathrm{m}} = \lambda_{\mathrm{f}}\left(\frac{\nu_{\mathrm{f}}^{2}}{g}\right)^{-1/3} \frac{Re_{\mathrm{f}}}{1.08\, Re_{\mathrm{f}}^{1.22} - 5.2} \tag{10}$$

In equations (6), (7) and (10) Reynolds number, Re_f, is calculating according to (2) setting the vapor quality to zero.

Comparison of the presented calculation models was performed in order to select appropriate one to use for helical tubes according the above proposed modification. Reference data for thermal calculations are summarized in Table 1. Constructional and technical data needed for the calculations are taken from a heat exchanger with helical tubes installed and operating in a real application.

Table 1 Heat exchanger data, taken as reference for the calculations

Description	*Value*	*Dimension*
Tube outer diameter	10	mm
Tube inner diameter	8	mm
Tube thickness	1	mm
Mean helical coil diameter	50	mm
Tube number	38	
Thermal conductivity (W.Nr.1.4307)	16,5	W/(m·K)
Shell side cross section area	0,02323	m^2
Thermal power	450	kW
Water vapor saturation pressure	7	bar,a
Water vapor saturation temperature	165	°C
Mass flowrate per tube	0,0057	kg/s
Mass flux density per tube	114	$kg/(m^2s)$
Shell side inlet temperature	65	°C
Shell side outlet temperature	80	°C

Thermal calculations of both models [10, 12] were performed using local and average approach assuming condensation occurs immediately. Fig. 3 shows temperature distributions and segmental subdivision within referent heat exchanger used in local approach calculations. The calculations are performed according to proposed correlations and methodology in references [10], [12] and [13].

In local calculation model representing a local approach, a heat exchanger tube is divided into five sections according to vapor quality x, with step Δx = 0.2 kg/kg. Then, heat transfer coefficient is calculated for the middle of each section i.e. for the average values of vapor quality in the section according to aforementioned correlations [13]. Thus calculated local heat transfer coefficients are then averaged and compared with an average heat transfer coefficient calculated according to average calculation models. Note that average calculation models, which represent average approach, do not require subdivision of heat exchanger tube into segments.

Important result for the discussion is also averaged local heat transfer coefficient (denoted with index m,x) which is calculated according to the following expression:

$$\alpha_{\mathrm{m,x}} = \frac{\sum_{i=1}^{5} \alpha_{\mathrm{x},i} \, L_{\mathrm{x},i}}{L} \tag{11}$$

To perform above averaging, (11), it is necessary to calculate the tube length of each section in heat exchanger tube using the following procedure. First, the shell-side heat transfer coefficients, α_s, are calculated according to correlations from [14], and then heat transfer coefficients inside helical tube according to correlations from [10, 12]. Finally, with known heat transfer coefficients and heat flux the heat transfer area and the tube length can be easily calculated.

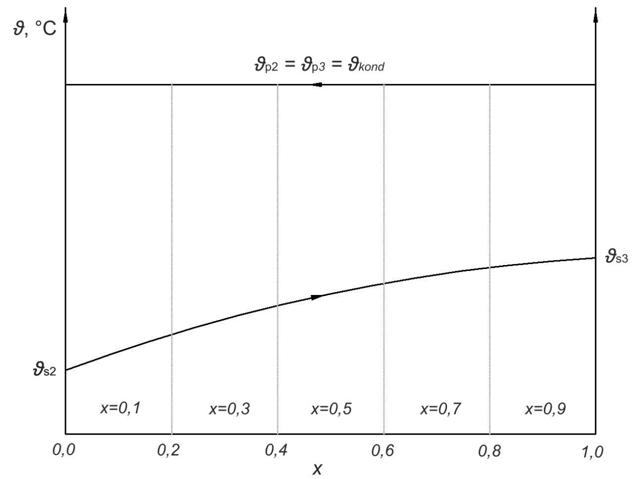

Fig. 3 Temperature distributions in referent heat exchanger

4. Results and discussion

Table 2 presents the thermal calculation results of referent heat exchanger for identical operating and construction parameters. Thermal calculations are performed according to local and average calculation models for both proposed heat transfer models [10, 12]. Furthermore, the table also gives averaged local results of heat transfer coefficients.

Table 2 Thermal calculations results

section	x	α_x	$\alpha_{s,x}$	k_x	$\Delta\vartheta_{m,x}$	A_x	L_x
	-	W/(m^2 K)	W/(m^2 K)	W/(m^2 K)	K	m^2	m
calculation acc. VDI Heat Atlas [12] – local calculation model							
1	0.9	11969	4282	2647	86.5	0.3930	0.412
2	0.7	10182	4222	2527	89.5	0.3979	0.417
3	0.5	10348	4162	2516	92.5	0.3868	0.405
4	0.3	10859	4101	2522	95.5	0.3738	0.392
5	0.1	11454	4038	2528	98.5	0.3615	0.379
				$\alpha_{m,x}$ = 10955	W/(m^2 K)		L=2,004 m
calculation acc. Kutateladze [10] – local calculation model							
1	0.9	17988	4282	3372	86.5	0.3086	0.323
2	0.7	14126	4222	3181	89.5	0.3161	0.331
3	0.5	12625	4162	3072	92.5	0.3168	0.332
4	0.3	11724	4101	2990	95.5	0.3153	0.330
5	0.1	11093	4038	2921	98.5	0.3129	0.328
				$\alpha_{m,x}$ = 13495	W/(m^{2}K)		L=1,644 m

	α_m W/(m^2 K)	α_s W/(m^2 K)	k_m W/(m^2 K)	$\Delta\vartheta_{m,}$ K	A_m m^2	L_m m
calculation acc. VDI Heat Atlas [12] – average calculation model						
	11469	4162	2577	92.3	1.8921	1.982
calculation acc. Kutateladze [10] – average calculation model						
	13285	4162	3110	92.3	1.5679	1.643

In Fig. 4 and Fig. 5 the condensation heat transfer coefficient values as a function of vapor quality x from both models are shown. The model according to Kutateladze [10] generally gives higher values of local heat transfer coefficients then VDI Heat Atlas model, [12]. At lower vapor qualities (up to 0.3 kg/kg) both models give approximately identical results but at higher vapor qualities, as its value approaches one, the differences increase significantly resulting with 67 % higher heat transfer coefficient at 90 % vapor quality.

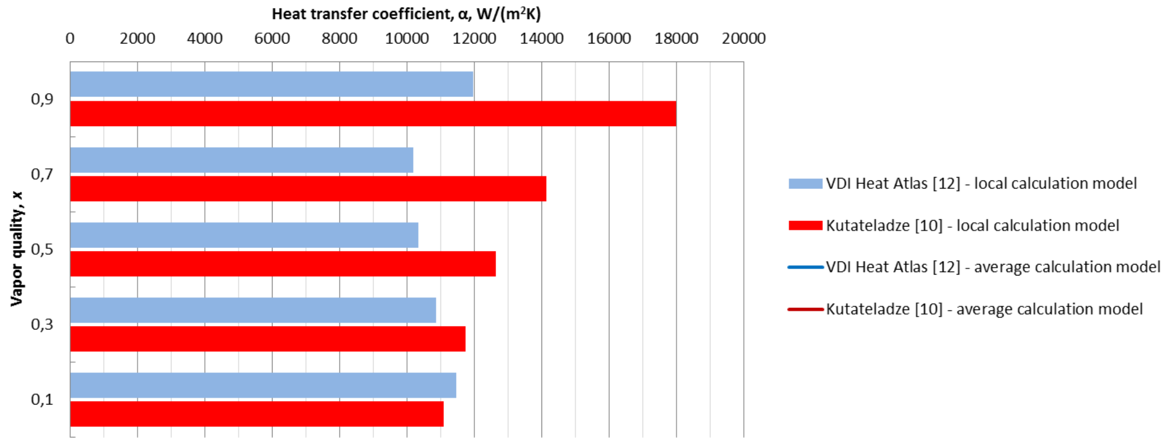

Fig. 4 Heat transfer coefficients

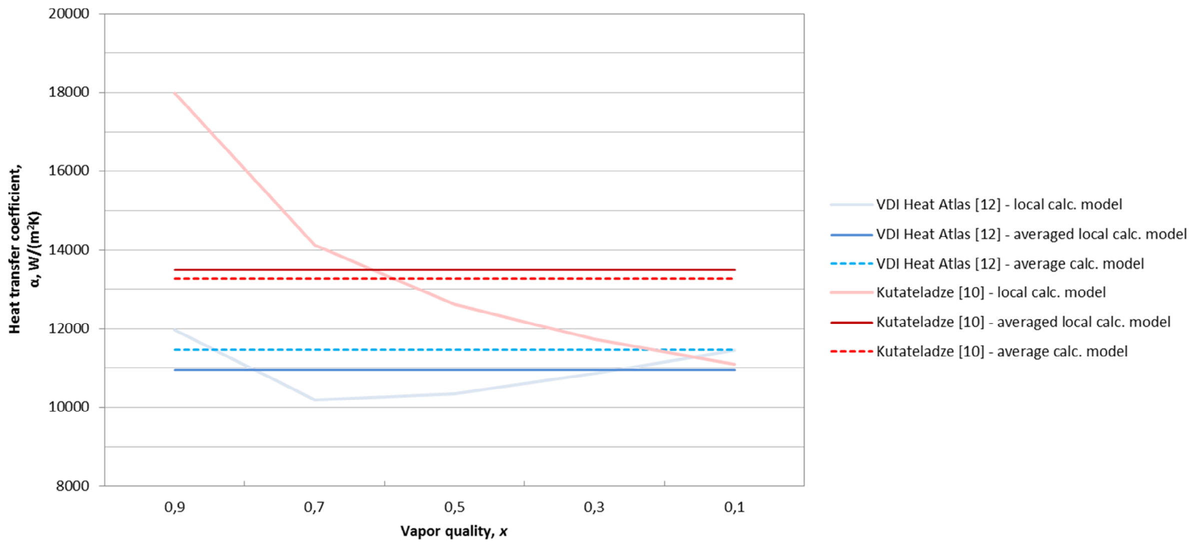

Fig. 5 Heat transfer coefficients

When comparing averaged local model and average model from the same literature, it can be seen that Kutateladze model is more consistent giving only 1.3 % difference in values of average heat transfer coefficients. This difference in VDI Heat Atlas models is slightly higher, that is 4.7 % Furthermore, when comparing average models, Kutateladze average approach calculation gives 15% higher values then the same model in VDI Heat Atlas. This difference is even higher when comparing averaged values obtained by local approach. Kutateladze model gives in 23% higher values than those obtained by VDI Heat Atlas model. The reason is that Kutateladze models, [10], takes that kinematic viscosity dominantly influence heat transfer what makes him more sensitive to the vapor quality change. This means that this model should be used carefully when dealing with condensation inside helical tubes because the quality change rate can be significantly different compared to the condensation inside straight tubes due to the appearance of secondary flows.

To obtain complete comparison results, the same calculation procedure is performed for mass flow densities above 40 kg/(m^2 s). The results are shown in Fig. 6. It is observed that, at lower mass flow densities differences between values obtained from average and averaged local calculation models [10, 12] are significant, while at higher mass flow densities has tendency of decreasing.

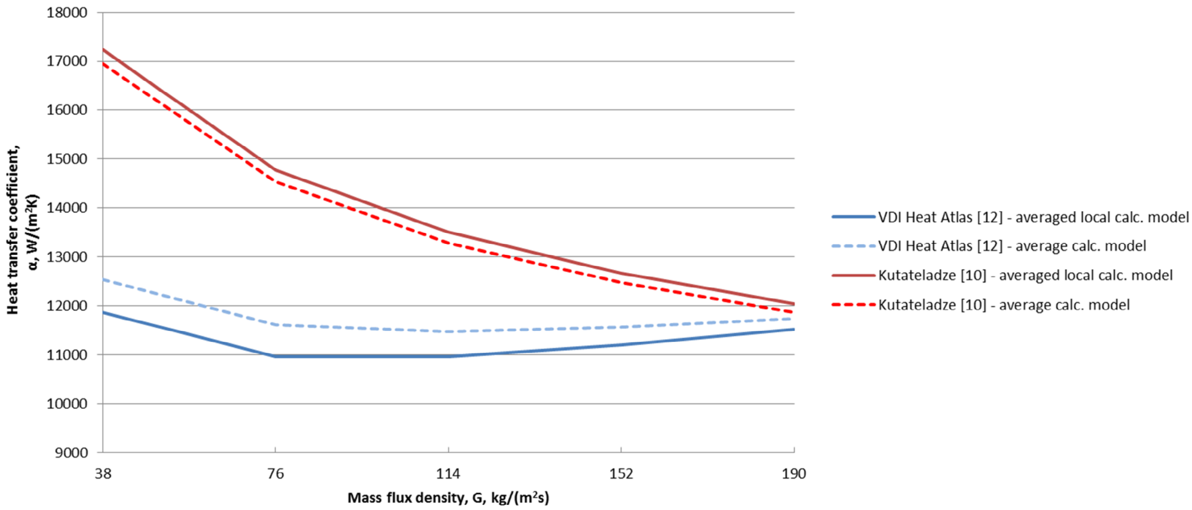

Fig. 6 Heat transfer coefficients as a function of different mass flux densities

According to the calculated data, it is obvious that these models show very good agreement for high mass flux densities which indicates that the models are more or less reliable in this region. However, for low mass flux densities the models show significant discrepancy in calculated results. There are two main reasons for such behavior. First, reason is the imperfection of the condensation models for condensation inside vertical tubes and the second, and the most important is the expression for enhancement factor.

Imperfection of the models comes from the fact that the quality is changing through the tubes. In both models the properties of the saturated water vapor are taken as constant averaged properties both across the tube cross section and along a section of the tube length. Due to the unsteady vapor quality change applied approach produce certain discrepancy in low mass flux densities regime.

The existing discrepancy between the models for low mux flux density is even more increased by the use of constant enhancement factor according to (1). Since the physical background of enhancement factor lies in the fact that secondary flows alter the flow pattern, it is expected that enhancement factor should, among others, depend on mass flux density because secondary flows depend on fluid velocity inside the tube. This is confirmed by the presented results.

5. Conclusion

Although many condensation models inside helical tubes have been introduced for media in the field of refrigeration, there is a serious lack of suitable models involving condensation of water vapor, especially when dealing with high mass flux densities. The most common approach is correlating heat transfer coefficient in helical tubes with well investigated water vapor condensation inside straight vertical tubes. Such correlations are necessary because of the secondary flows that appear in helical tube heat exchangers.

This work proposes the use one of well-known correlation that has been originally introduced for application without phase change of participating media. For that reason the correlation was checked whether it applies for phase changing application through available experimental works in the literature. A very good agreement was found between the proposed correlation and the available experimental data.

This correlation was then applied to two reliable models for vertical condensation inside straight tubes, Kutateladze and VDI Heat Atlas models. The comparison is carried out using both local and averaged approach for the 120 kg/m^{2}s water vapor mass flux density. It has been shown that VDI Heat atlas gives significantly lower, more conservative results, that is, 15 % or 23 % lower values of average heat transfer coefficient, depending whether local or average approach is used, respectively. The reason is that Kutateladze models accounts dominantly for viscosity of condensate as the property that influences heat transfer, which makes him more sensible to vapor quality change at relatively high mass flux densities. Furthermore, the work has shown that when using VDI Heat Atlas models, even more conservative results can be obtained by the local approach which gives about 5 % lower values of heat transfer coefficient.

Finally, the proposed calculation was then carried out for different mass flux densities above 40 kg/m^{2}s. The results showed significant difference between the two models up to mass flux density of 180 kg/m^{2}s. Such difference is increased as the mass flux densities decreases which imply that the enhancement factor should have different expressions and depending variables for different flowing regimes. That is because secondary flows in helical tubes depend on primary fluid velocity resulting in different heat transfer enhancement. Logical approach in order to model the enhancement factor could be dimensionless analysis, similar to the heat transfer model approach. It is necessary to form particular dimensionless numbers from the available physical properties, geometrical and other data and to use them in modeling the enhancement factor for each flow regime, which has to be previously defined.

These and such other questions set the quality basis and motivation for future investigations in this field which is the aim of author's future work.

NOMENCLATURE AND SYMBOLS

Nomenclature:

A	Area (m^2)	Re	Reynolds number (-)
d_i	Inside tube diameter (m)	x	Vapor quality (-)
D_H	Mean helical coil diameter	α	Heat transfer coefficient (W/(m^{2}K))
f_val	Correction factor (-)	Γ	Characteristic length (m)
g	Gravitational acceleration (m/s^2)	ϑ	Temperature (°C)
k	Overall heat transfer coefficient (W/(m^{2}K))	λ	Thermal conductivity (W/(mK))
L	Tube length	μ	Dynamic viscosity (Pa·s)
Nu	Nusselt number (-)	ν	Kinematic viscosity (m^2/s)
Pr	Prandtl number (-)	ρ	Density (kg/m^3)
q_m	Mass flowrate (kg/s)		

Subscripts:

hel	Helical	m,x	Averaged local
f	Saturated liquid	s	Shell-side
g	Saturated vapor	vert	Vertical
m	Mean	x	Local
m,l	Mean laminar	x,l	Local laminar
m,t	Mean turbulent	x,t	Local turbulent

REFERENCES

[1] N. D. Shirgire, P. Vishwanath Kumar: *Review on Comparative Study between Helical Coil and Straight Tube Heat Exchanger*, Journal of Mechanical and Civil Engineering, volume 8, Issue 2, p. 55-59, IOSR, 2013

[2] I. Glavan, Z. Prelec, B. Pavković: *Modelling, simulation and optimization of small-scale CCHP energy systems*, International Journal of Simulation Modelling, volume 14, no. 4, p. 571-583, 2015

[3] M. H. Al-Hajeri, A. M. Koluib, M. Mosaad, S. Al-Kulaib: *Heat transfer performance during condensation of R-134a inside helicoidal tubes*, Energy Conversion and Management, volume 48, p. 2309-2315, Elsevier, 2007

[4] M. A. Abd Raboh, H. M. Mostafa, M. A. M. Ali, A. M. Hassaan: *Experimental Study of Condensation Heat Transfer Inside Helical Coil*, Evaporation, Condensation and Heat transfer, InTech, 2011

[5] S. Ayadi, E. Hadj-Taïeb: *Finite element solution of dynamic response of helical springs*, International Journal of Simulation Modelling, volume 7, no. 1, p. 17-28, 2007

[6] C. X. Lin, M. A. Ebadian: *Condensation heat transfer and pressure drop of R134a in annular helicoidal pipe at different orientations*, International Journal of Heat and Mass Transfer, volume 50, p. 4256-4264, Elsevier, 2007

[7] J. T. Han, G. P. Su, J. H. Pan: *Condensation heat transfer of R-134a in horizontal straight and helically coiled tube-in-tube heat exchangers*, Journal of Hydrodynamics, volume 19(6), p. 677-682, 2007

[8] D. Q. Kern: *Process heat transfer*, McGraw-Hill, 1965.

[9] W. Xiaoyong, F. Xiande, S. Rongrong: *A Comparative Study of Heat Transfer Coefficients for film Condensation, Energy Science and Technology*, Energy Science and Technology, volume 3, p. 1-9, CSCanada, 2012

[10] D. Papini, A. Cammi: *Modelling of Heat Transfer Phenomena for Vertical and Horizontal Configurations of In-Pool Condensers and Comparison with Experimental Findings*, Science and Technology of Nuclear Installations, volume 2010, ID 815754, Hindawi, 2010

[11] M. M. Shah: *A general correlation for heat transfer during film condensation inside pipes*, International Journal of Heat and Mass Transfer, Pergamon Press, 1979

[12] Verein Deutscher Ingenieure: *VDI Heat Atlas*, Springer, 2010

[13] H. D. Baehr, K. Stephan: *Heat and Mass Transfer*, second edition, Springer, 2006

[14] D. A. Donohue, *Heat transfer and pressure drop in heat exchangers*, Industrial & Engineering Chemistry, volume 41(11), p. 2499-2511, 1949

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