SOFTWARE FOR THE RAPID DESIGN AND ANALYSIS OF CONDENSER HEAT EXCHANGERS

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1. INTRODUCTION

Heat transfer theory concerning both sensible and latent heat transfer is historically well-researched, and analytical solutions for the analysis of heat exchangers of various types (e.g. Shell-and-Tube and Double-Pipe) are presented in topical literature. The intention for this software, "Heat Exchange Designer" (HXD), is to incorporate selected, computationally inexpensive approximations into a package for the rapid design and analysis of heat exchangers. Previous iterations of this software have incorporated such updates as the expansion of heat exchanger types, cost-estimation, and multi-objective optimisation using a Teaching Learning Based Optimisation algorithm [1-3].

Many applications of a Shell-and-Tube Heat Exchanger (STHE) include phase-change processes, such as condensation or evaporation. This iteration of the software expands capability to that of latent (condensation) heat transfer within STHE. The cold fluid flows inside the horizontal tubes while the hot vapour flows perpendicular to the tube bundle. Condensation occurs on a surface when the surface temperature is lower than the saturation temperature of a vapour. Modelling the thermal behaviour of the outside tube condensation is one of challenging points for STHE (condenser) design and analysis.

In 1916, based on the number of assumptions, Nusselt [4] developed theoretical solutions to determine the heat transfer coefficient for film-wise condensations of a pure vapour on a single tube. This Nusselt correlation for film condensation heat transfer coefficient has still been used in many cases [5-7]. Further developments from the Nusselt model have been made over the years to consider various effects which were neglected in the original approach. Several available theoretical models for condensation on banks of tubes were well summarized and their validity and predictability have been compared [8]. It was found that the McNaught [9] model gave better agreement with the steam data, whereas the model of Fujii and Oda [10] was the best for non-steam cases.

This paper describes a new user-friendly computer program developed for the design and anlaysis of shell-and-tube condensers. The developed program has been tested through the application cases of feedwater condensers in powerplants as well as emerging sustainable-energy technologies.

2. MATHEMATICAL MODELS FOR HEAT TRANSFERS

Sensible heat transfer, where the energy transfer is associated with the exchange of heat energy from a hot medium to a cold medium, can be modelled according to well-known formulae such as for the Heat Transfer Rate, $\dot{Q}_s$, given in Eq. (1).

$$\dot{Q}_s = U\,A\,\Delta T_{LM} \tag{1}$$

Where U is the overall heat transfer coefficient, A is the surface area for heat transfer, and ΔT_{LM} is the Log Mean Temperature Difference (LMTD) (for a counter-current flow). The energy transfer rate offered by latent heat transfer, $\dot{Q}_l$, where the transferred energy is sourced from the condensation of a vapour to a liquid, is calculated via Eq. (2).

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$$\dot{Q}_l = \dot{m}_{vap}\, h_{l,vap} \tag{2}$$

Where $\dot{m}_{vap}$ is the mass flow rate of the condensing vapour and $h_{l,vap}$ is the specific latent heat of vaporisation, a property of the condensing fluid. Condensation is one of many types of phase change and the focus of this study for its use in power generation systems. Importantly for heat exchange, the bulk temperature of the condensing fluid is constant whilst the fluid undergoes the phase change therefore the temperature of the condensate remains high. As can be seen in Eq. (1), maintaining a large temperature difference across two fluids, increases the rate of energy transfer. As the temperature of the hot fluid remains high under this process, and the process itself heats the coolant further, the net effect is a greater transfer of energy to the coolant than possible for a similar net temperature difference without phase change. The temperature at which condensation occurs is referred to as the saturation temperature.

2.1 MODELLING FILM-WISE CONDENSATION

Film-wise condensation is assumed, where condensing vapour forms a continuous film around the tubes as the probability of occurrence is greater and produces conservative designs. The modelling of film-wise condensation has been considered by numerous groups, but the most widely implemented model was derived by Nusselt [4]. Nusselt considered a smooth, laminar film of condensate developing on the outside of a tube. It was proposed that the vapour-side mean Heat Transfer Coefficient (HTC), $\bar{\alpha}$, can be modelled as,

$$\bar{\alpha} = 0.728\left[\frac{{k_L}^3 \rho_{cond}\left(\rho_{cond} - \rho_{vap}\right) g h_{l,vap}}{\mu_{cond}(T_{sat} - T_W)D}\right]^{0.25} \tag{3}$$

Where ρ refers to the density, k refers to the thermal conductivity, μ refers to dynamic viscosity, T_{sat} and T_W are the saturation and wall temperatures, respectively, and D is the tube outside diameter. The Nusselt method is considered conservative. It often underpredicts the HTC in comparison with experimental data as it does not consider the shear effects of the vapour flow.

McNaught [9] proposed a method to compensate for shear effects by relating the shear component of the HTC to that of an idealised liquid-only flow and corrected for two-phase flow via the Lockhart-Martinelli Parameter, equated as,

$$\alpha_{shear} = \alpha_{liq} \times 1.26\left[\frac{1}{X_{tt}}\right]^{0.78} \tag{4}$$

Here, α_{shear} models the shear component of the HTC of high-Reynolds number flows, α_{liq} is the HTC for the hypothetical liquid-only flow, which per this software is calculated via the Petukhov equations [3]. X_{tt} is the Lockhart-Martinelli Parameter for the flow at a given vapour quality. A root mean squared operation combines both the gravity effect of Nusselt model and the vapour shear effect of McNaught model for the overall HTC [8, 11].

2.2 ACCOUNTING FOR MULTIPLE, HORIZONTALLY-ALIGNED TUBES

Modelling condensation on banks of tubes adds complexity. The condensation from higher rows of tubes falling onto lower rows contributes to the liquid film that forms and insulates the heat transfer process. As such the heat transfer for lower tube rows is reduced versus that of the higher tube rows. Methods for accounting for this effect of condensate inundation have been summarised and evaluated by Honda [12]. Several equations were proposed; however, based on empirical data, literature [13, 14] suggests that a method put forward by Kern [15] is most applicable to the condensation of steam. The equation is given as,

$$\bar{\alpha}_n = \bar{\alpha}_1 n^{-1/6} \tag{5}$$

Where $\bar{\alpha}_n$ is the average HTC across the tube in n-th row, $\bar{\alpha}_1$ is the HTC for the top row calculated per section 2.1, and n is a given tube's row number from top to bottom. Upon completion of the program, the results with Eq. (5) agreed well with literature.

3. SUPERHEATED VAPOURS AND CONDENSATE COOLING

A key goal with this tool is to allow for the modelling of oncoming vapours with temperatures above the saturation temperature. In this case, the flow exists in a superheated state and must first be cooled to the saturation temperature before condensation can begin. Additionally, an exit temperature of the condensate lower than the saturation temperature would be required in a case, which requires further condensate cooling following the total phase change of the vapours.

To model these pre- and post-condensation temperature drops, a multi-zone model [11] is proposed which splits the heat exchanger into three zones: the desuperheating zone (gas phase), the condensation zone (two-phase, phase-change), and the condensate-cooling zone (liquid phase). This presents a challenge to the existing program which uses convergence loops to calculate the individual zone properties. With the expansion of zones there is a proportional increase in computational expense, but also the interaction between the zones must be considered. The implicit nature of the functions that connect these zones adds complexity.

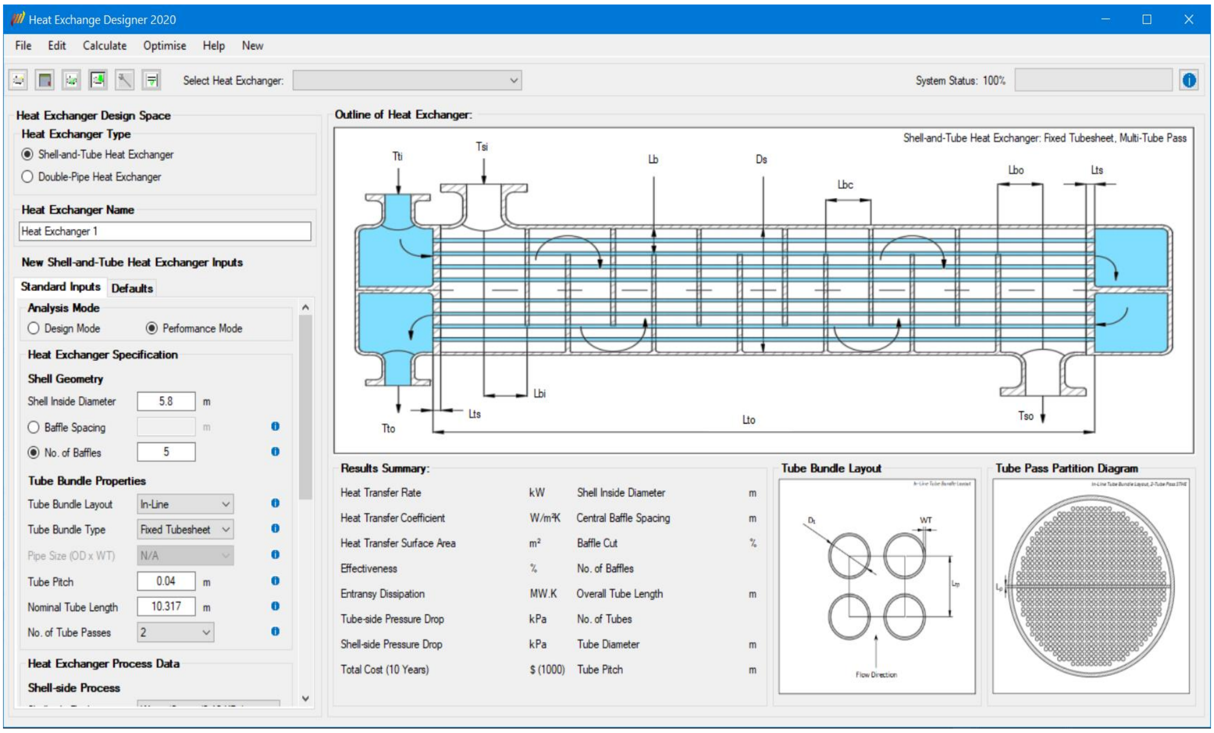

Figure 1: Snapshot of the Graphical User Interface for the Heat Exchange Designer (HXD) program

4. PRESSURE DROP

Pressure drops in the single-phase zones (the desuperheating zone and the condensate-cooling zone) can be calculated according to the existing Bell-Delaware model [3]. For the two-phase zone, condensation zone, the pressure drop multiplier for the condensation region is calculated as,

$$\emptyset_{cond}^{2} = 1 + (Y^{2} - 1)(0.25x^{0.77}(1 - x^{0.77})x^{1.54}) \tag{6}$$

Where Y is the Chisholm factor, which relates to the Lockhart-Martinelli parameter, but is concerned with the holistic flow rather than the separated phases. Finally, the two-phase pressure drop is calculated by applying this multiplier on the liquid phase pressure drop [16].

5. SOFTWARE DEVELOPMENT

The Heat Exchange Designer (HXD) program has been developed in the VB.NET programming language via the Visual Studio platform. An appropriate Graphical User Interface was developed to permit simple operation by a user as shown in Figure 1. The inputs window was expanded to permit the specification of required parameters concerning the condensation model.

6. VALIDATION AND RESULTS

Sourcing validation data concerning Shell-and-Tube condenser experiments from open literature has been a significant challenge, perhaps in part due to companies safeguarding condenser data as valuable intellectual property. Despite this, several cases were found that serve as adequate experiments for assessing the accuracy of the implemented condensation model.

Validation Case 1 concerns a surface condenser onboard a United States Navy ship with data [17] that concerns both sensible and latent heat exchange, permitting a holistic assessment of the model. Validation Case 2 regards a surface condenser in use at a 210MW thermal power plant facility [18] where the condenser sits as part of a larger system of turbines and compressors. This data focuses purely on the condensation process and will therefore be more useful in evaluating the Nusselt and McNaught combined models. The results are presented in Table 1.

By first assessing temperature errors, it is apparent that in Case 1, the shell-side temperature change exhibits a bit larger percentage error of 17.42% (0.92 °C), compared to 3.57% (0.21 °C) for the tube-side flow. This may be attributed to the differences in mass flow rates between each flow. From a practical perspective, however, the discrepancy of actual temperature changes (°C) could be considered small since it is less than 1 °C. Finally, the model can predict reasonably well.

Reviewing the heat transfer rate, it is evident that the program is getting very close to the experimentally derived values with a peak error of just 5.84%. The energy required to undergo phase-change is somewhat fundamental per Eq. (2). Any deviation in the heat transfer rate can be isolated to two contributors, either the condensate mass flow rate or the latent heat of vaporisation ($h_{l,vap}$). Given that $h_{l,vap}$ is calculated according to the standard IF97 correlations (widely accepted relationships for the properties of water [19]), it is more probable that the condensate mass is the primary source of error. Whilst in both validation cases, total (full) phase change was the target, neither author acknowledges the exact exit quality and it is therefore likely that in both experiments, less vapour condenses than claimed. In contrast, the HXD program was run with the assumption of total (full) phase change. This may be the source of discrepancy.

With the HTC, the results are somewhat consistent with deviations circa to 10% in both cases. In addition, the multi-row averaging technique may be another source of error. The Kern method [15] was embedded based on empirical results, and whilst in comparison with the other models, it performs the best, it may still not be perfectly applicable to the test cases. There is insufficient information provided on both cases. Due to the scarcity of available data, to properly isolate the causes of error it may therefore be necessary to setup a controlled experiment to specifically evaluate this tool.

Of concern with the HTC overestimations, is that the tool is evidently calculating designs that are more efficient than reality. In the case of the HTC, it would preferable to underpredict rather than overpredict, as this would yield a conservative design. Similar considerations can be made for the pressure drop calculation, whilst the error is relatively low in Case 1 (no data available for Case 2), it is an underestimation which may cause engineers to undersize compressors/pumps as part of a wider system.

In summary, the discrepancies are generally small in all the cases, within acceptable bounds for an initial design tool. The design outputs of this tool would naturally be advanced and improved through the application of high-fidelity simulations in practice.

Table 1: Comparison between Experimental Data and Program Results

Parameters	Case 1			Case 2		
	Reference [17]	Program	Error	Reference [18]	Program	Error
Shell-Side Outlet Temperature, $T_{s, out}$ *(°C)*	38.34	37.4	(-0.94°C)	*45.9*	45.9	(0°C)
Shell-Side Temperature Change, $T_{s, out} - T_{s, in}$ *(°C)*	-5.28	-6.2	17.42% (0.92°C)	0	0	0.00% (0°C)
Tube-Side Outlet Temperature, $T_{t, out}$ *(°C)*	30.71	30.9	(0.19°C)	42	42.9	(0.9°C)
Tube-Side Temperature Change, $T_{t, out} - T_{t, in}$ *(°C)*	5.89	6.1	3.57% (0.21°C)	9	9.9	10.00% (0.9°C)
Heat Transfer Rate, Q *(MW)*	46.74	48	2.70%	292.6	309.7	5.84%
Overall Heat Transfer Coefficient, U *($W\ m^{-2}\ k^{-1}$)*	3604.4	3962.1	9.92%	3005	3283.6	9.27%
Pressure Drop *(Pa)*	5178	4910	-5.18%	-	4661	-

7. CONCLUSIONS

Implementation of a condensation model within an existing software has been successful with validation exercises indicating sufficiently close agreement with the experimental data for condensing steam. Findings are concluded as follows:

- Governing models for film-wise condensation based on the works of Nusselt and McNaught were implemented to model condensation.

- An innovative multi-zone model has been developed which permits the holistic design of a heat exchanger system that facilitates both latent and sensible heat transfer.

- Adequate validation indicated reasonable accuracy of the newly embedded models, suitable for the initial design phase of a condensing heat exchanger system.

As discussed, errors may be associated with the row averaging methods and a future optimisation exercise is recommended to fine tune the embedded method. It is recommended generally that further validation is necessary through assembling a purpose-built experiment where all parameters pertinent to the functionality of this tool are known or controlled.

8. REFERENCES

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