Journal of Power Technologies 101 (3) (2021) 81 -- 87

An improved Müller-Steinhagen and Heck model for two phase pressure drop modeling at high reduced pressures

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Abstract

Better understanding of two-phase fluid behavior is required to optimize the design models of the components containing a two-phase refrigerant. This is important since applications increasingly seek to operate in the region of high reduced pressure values, for instance the vapor generator, which is a key heat exchanger in the Organic Rankine Cycle system and the high temperature heat pump. Implementations are carried out at high evaporation saturation temperatures where the refrigerant transformation to vapor occurs at temperatures higher than 90°C. Analysis of the literature analysis shows there is a gap in knowledge regarding two-phase flow for synthetic refrigerants at high saturation temperatures. Reliable prediction of pressure drop in two-phase flows is an important prerequisite for accurate optimization of thermal systems. The total pressure drop of a fluid derives from the variation of potential and kinetic energy of the fluid and friction on the channel walls or between the phases (60-120°C) and moderate reduced pressures (0.2-0.5). This paper presents a modification to the established Müller-Steinhagen and Heck (1986) model for two phase pressure drop in relation to high values of reduced pressures. Model validation has been done in comparison to reliable experimental data obtained by Charnay et al. (2015) for R245fa at reduced pressures above 0.5. The modification constitutes a significant improvement on the calculations presented in the literature, including by the authors of experimental data.

Introduction

Better understanding of two-phase fluid behavior is required to optimize the design models of the components containing a two-phase fluid, especially in light of the fact that more and more applications are sought in the high reduced pressure ranges, for instance the vapor generator, a key heat exchanger in the Organic Rankine Cycle system and in the high temperature heat pump. Numerous implementations are sought at high evaporation saturation temperatures where the evaporation of working fluid occurs at temperatures higher than 90°C. Analysis of the literature shows there is a gap in knowledge regarding two-phase flow for synthetic refrigerants at high saturation temperatures. Reliable prediction of pressure drop in two-phase flows is an important prerequisite for accurate optimization of thermal systems.

A common approach to model pressure drop in two-phase flows is to express it in terms of the two-phase flow multiplier. Historically, the precursor model in that respect was postulated by Lockhart and Martinelli (1949), followed by models due to Chisholm (1967), Friedel (1979) and Müller-Steinhagen and Heck (1986). To date, many modifications of these correlations have been put forward, becoming new predictive methods. These methods were not very successful in capturing trends of pressure drop in the vicinity of the thermodynamic critical point. Based on analysis of two-phase pressure frictional pressure drop, a model first proposed by Cavallini et al. (2009) was updated and a new equation formulated taking into account the effects of surface roughness on the frictional pressure drop as a function of the liquid only Reynolds number. The updated model (Del Col et al. 2013) was validated against experimental data and compared with other available correlations, obtaining satisfying results, with a total absolute mean deviation of 7.4%. The new model predicted 93.3% of the measured pressure drop within a ±15% error band and 98.9% within ±20%, showing its superiority over well-known correlations. Charnay et al. (2015) concluded that, from amongst the models developed on the basis of the separated flow model, the best correlations for predicting the whole database for the experimental data of R245fa are those of Müller-Steinhagen and Heck (1986), Zhang and Webb (2001), and Friedel (1979), respectively. For other correlations, the deviation increases as the saturation temperature increases.

The present paper pursues the objective of further improving the predictions of Müller-Steinhagen and Heck model (1986) at high reduced pressures. In the model the effect of the small diameter of the channel has been taken into account through consideration of surface tension as well as the appropriate impact of that term through validation against experimental data obtained by Charnay et al. (2015).

Two-phase frictional pressure drop

Flow resistance due to friction in two-phase flow is greater than in the corresponding case of single-phase flow at the same flow rate. The two-phase flow multiplier is defined as the ratio of friction pressure drop in two-phase flow, $(dP/dz)_{TP}$ to the friction pressure drop in single-phase flow with either liquid or vapor, $(dP/dz)_0$, as presented below:

$$\Phi^2 = \left(\frac{dP}{dz}\right)_{TP} \Big/ \left(\frac{dP}{dz}\right)_0 \tag{1}$$

Unfortunately, the correlations developed for conventional size tubes cannot be directly used in calculations of pressure drop in mini-channels, regardless of the fact that these correlations were developed based on a large volume of experimental data. These correlations, however, can serve as a starting point for a group of new models dedicated to predicting two-phase pressure drop in mini-channels. Selected correlations developed earlier for convectional channels can be tailored to predictions for mini-channels by incorporating surface tension effects.

The two-phase frictional pressure drop correlation, as scrutinized in this paper, was formulated by Müller-Steinhagen and Heck (MS) (1986). Originally, it assumed a form:

$$\Phi_{MS}^2 = \left[1 + 2\left(\frac{1}{f_1} - 1\right)x\right](1-x)^{1/3} + x^3\frac{1}{f_1} \tag{2}$$

In equation (2) the function f_1 represents the ratio of pressure drop in liquid to the pressure drop in gaseous phase and yields $f_1=(\rho_L/\rho_G)\ (\mu_L/\mu_G)^{0.25}$ for turbulent flow (following application of the Blasius equation for the determination of friction factor) and $f_1=(\rho_L/\rho_G)(\mu_L/\mu_G)$ for laminar flows, following application of the formula $f=64/Re$.

The modification made previously by the present authors was based on extending the applicability of the Müller-Steinhagen and Heck correlation to the case of mini-channels. D. Mikielewicz & J. Mikielewicz (2011) introduced the constraint number Con, which significantly improves predictions of the pressure drop in the mini-channels scenario. In the modified version, the Müller-Steinhagen and Heck (M-S-H) model is:

$$\Phi_{MS}^2 = \left[1 + 2\left(\frac{1}{f_1} - 1\right)x Con^m\right](1-x)^{1/3} + x^3\frac{1}{f_1} \tag{3}$$

where $Con=(\sigma/g/(\rho_L-\rho_G))^{0.5}/D$ and $m=0$ for conventional channels. The best consistency of results with experimental data, in the case of small diameter and mini-channels, is obtained for the value of exponent $m=-1$.

Jakubowska and Mikielewicz (2019) attempted to include the effect of reduced pressure in the Müller-Steinhagen and Heck correlation on the basis of a correction to the definition of the two-phase flow multiplier, given by equation (1). In the form examined it read:

$$\Phi^2 = \Phi_{MS}^2\left[1 - \left(\frac{P_{sat}}{P_{crit}}\right)^a\right] + 1 \tag{4}$$

Exponent a was found to perform best at $a=1$. The comparisons against experimental data proved that this approach was not very successful in predicting flows.

In a subsequent revisiting of the Müller-Steinhagen and Heck model applicable to conventional and small diameter channels, the exponent m, which shows the effect of the surface tension, was modified. The version of the Müller-Steinhagen and Heck (MSH1) model modified by authors yields:

$$\Phi^2_{MS} = \left[1 + 2\left(\frac{1}{f_1} - 1\right) xCon^{-0.875}\right](1-x)^{1/3} + x^3\frac{1}{f_1} \tag{5}$$

In the present study an additional two-phase flow multiplier was also studied, as formulated by Groneveld. The method is particularly recommended for refrigerants and in the original version it reads:

$$\Phi^2_{GR} = 1 + \left(\frac{dp}{dz}\right)_{fr}\left[\frac{\left(\frac{\rho_L}{\rho_G}\right)}{\left(\frac{\mu_L}{\mu_G}\right)^{0.25}} - 1\right] \tag{6}$$

In (6) the pressure drop $\left(\frac{dp}{dz}\right)_{fr}$ is determined from:

$$\left(\frac{dp}{dz}\right)_{fr} = f_{fr}\left[x + 4\left(x^{1.8} - x^{10} f_{fr}^{0.5}\right)\right] \tag{7}$$

If the Froude number for liquid, Fr_l is greater than 1.0 then $f_{fr} = 1$. Otherwise, it should be:

$$f_{fr} = Fr_l + 0.0055\left(ln\frac{1}{Fr_l}\right)^2 \tag{8}$$

The two-phase flow multiplier models selected for comparison are regarded as pertinent to model the flow near the thermodynamic critical point as they have a built-in feature whereby they approach, at least theoretically, unity when approaching the thermodynamic critical point. Actually, at that location the densities of liquid and vapor are equal.

Results

The original experimental dataset of pressure drop per Charnay et al. (2015) was obtained from a horizontal tube of 3mm inner diameter during adiabatic flow with R245fa as a working fluid. The considered mass velocity ranged from 100 to 1500 kg/(m^{2}s), whereas the saturation temperature varied from 60 to 120 °C and the inlet vapor quality from 0 to 1. The correlation of Müller-Steinhagen and Heck (1986) was found overall to be the best at predicting that data. Nevertheless, it performed slightly worse at smaller values of saturation temperature and consistency increased at higher saturation temperatures. For example, the consistency of predictions versus experimental data at the saturation temperature of 60°C was 75% with an error margin of ±30%; at 120°C consistency was a little over 80% with the same error margin. The general observed trend was that when increasing the saturation temperature, vapor density increased, hence the slip ratio and vapor velocity decrease. In consequence, the two-phase frictional pressure drop decreases. The maximum value of frictional pressure drop is achieved at lower vapor quality when the saturation temperature decreases.

Below presented are calculations of frictional pressure drop in a channel obtained by means of two models of two-phase flow multipliers, namely those per Müller-Steinhagen and Heck described by equations (3) and (5) as well as the Groneveld model. The difference between the versions of the Müller-Steinhagen and Heck models is the value of exponent m in equations (3) and (5). The reasons why the Groenveld model was selected are that (i) in the course of analyses it produces very satisfactory results, and (ii) it was not considered by Charnay et al. (2015). Distributions of frictional pressure drop for two different saturation temperatures, namely t_{sat}=80°C and 100°C, corresponding to reduced pressures p*=0.52 and 0.65 are presented in Figs 1-8. The mass velocity varied from 100 to 1000 kg/(m^{2}s).

At first glance the predictions obtained by the standard Müller-Steinhagen and Heck model, i.e., m=0, differ significantly from the predictions using the modified versions of the model with m=-1 and m=-0.875. The somewhat aberrant behavior of the pressure drop is observed in the case of G=100 kg/(m^{2}s), which indicates that the standard M-S model fails to predict that dataset. In the case of p*=0.52 the original M-S model significantly underpredicts the distributions of experimental pressure drop, especially with large mass velocities. The situation improves when the M-S-H and MSH1 versions with different values of m are used. The models still underpredict the pressure drop but the differences are much smaller. Additionally, the predictions are much closer to experimental evidence for m=-0.875. For comparison, the predictions as per Groenveld are included, see Fig. 4. Practically the same results are obtained as in the case of the MSH1 model with m=-0.875 .

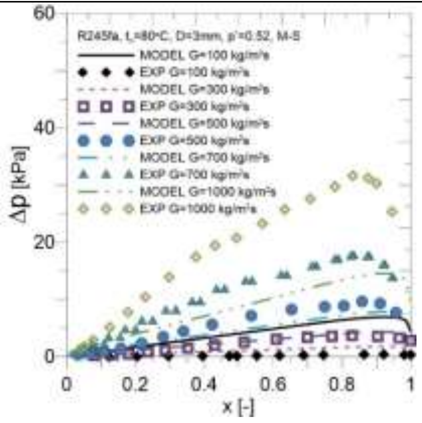

Figure 1: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=0, p=0.52.*

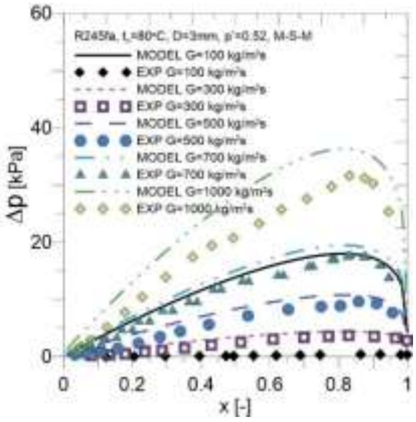

Figure 2: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=-1, p=0.52.*

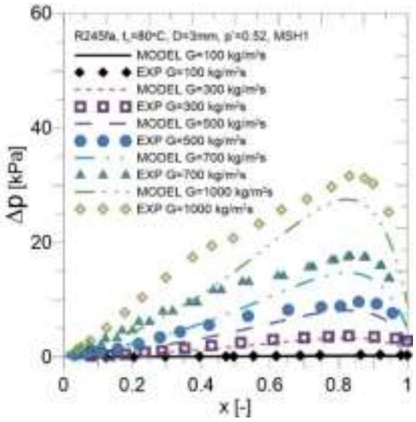

Figure 3: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=-0.875, p=0.52.*

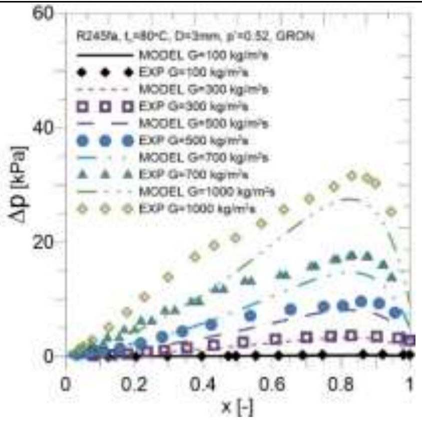

Figure 4: Frictional pressure drop as a function of quality in the Groneveld model; p=0.52.*

Attention now turns to the results obtained using the considered models in the case of increased values of reduced pressure, Figures 5-8. Here, the values of pressure drop are lower than with the smaller reduced pressures. As in the former case the M-S-H model underpredicts the experimental results and a significant improvement is observed when the exponent m in eq. 5 is set to m=-0.875. The predictions using the Groneveld model are in best agreement with the experimental data.

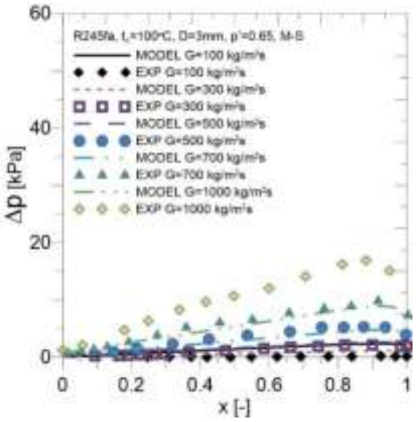

Figure 5: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=0, p=0.65.*

The target here is performance of the M-S model at different saturation conditions at a constant value of mass velocity, G=500 kg/(m^2s). As can be seen, best consistency of prediction is achieved at m=-0.875.

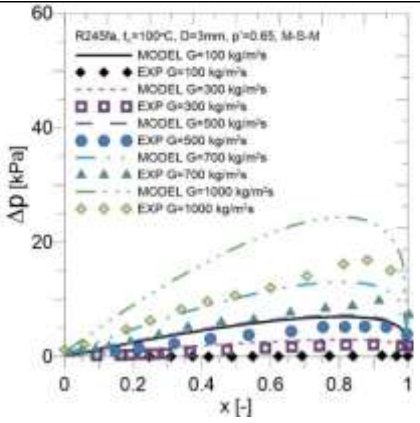

Figure 6: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=-1, p=0.65.*

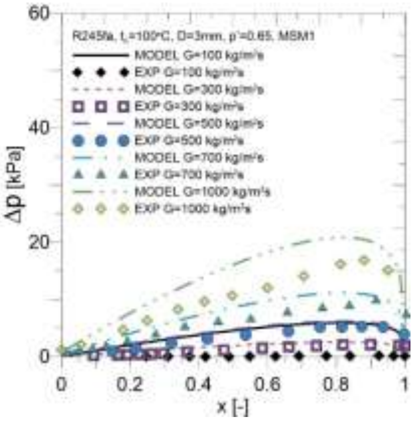

Figure 7: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model; m=-0.875, p=0.65.*

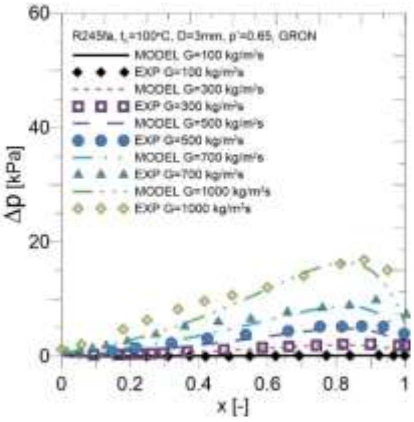

Figure 8: Frictional pressure drop as a function of quality in the Groneveld model; p=0.65.*

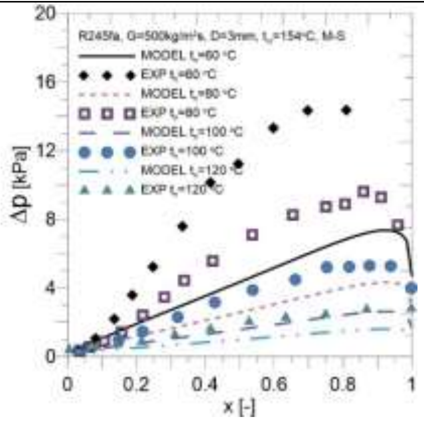

Figure 9: Frictional pressure drop as a function of quality in the M-S model at different saturation temperatures; m=0.

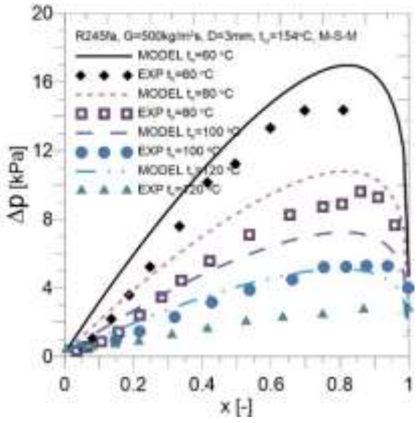

Figure 10: Frictional pressure drop as a function of quality in the M-S-H model at different saturation temperatures; m=-1.

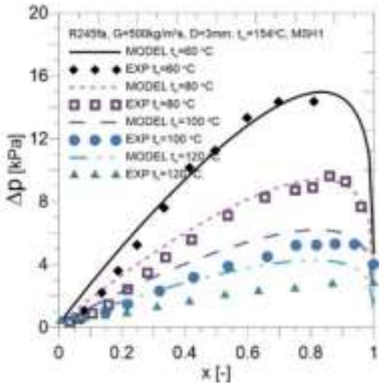

Figure 11: Frictional pressure drop as a function of quality in the Müller-Steinhagen and Heck model at different saturation temperatures; m=-0.875.

Conclusions

The paper considers two-phase flow modeling using a modified Müller-Steinhagen and Heck model. The postulated model is applicable to mini-channels and additionally is suitable for dealing with the reduced pressure effect. Incorporated into the model is a term responsible for modeling the influence of surface tension, wherein lies the true originality of the model. The Constraint number, Con, was introduced for that purpose. The influence of the Con number was examined through selection of the appropriate exponent m, which in the authors' earlier papers was set at a value of m=-1. In the present study the exponent m was further adjusted to a value of m=-0.875.

Another finding of the paper is that the data obtained by Charnay et al. (2015) is also well captured by the two-phase multiplier model created by Groneveld.

Acknowledgments

The work presented in this paper was carried out within project 2017/25/B/ST8/00755 funded by the National Science Center, Poland in the period 2018-2021.

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