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PREDICTION OF TWO-PHASE THERMOSYPHON THROUGHPUT BY SELF DEVELOPED CAE PROGRAM

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ABSTRACT

Computer Aided Engineering (CAE) program for prediction of two-phase closed thermosyphon throughput is developed. Thermosyphon working fluids under consideration are modern refrigerants (R404A, R407C, R134A etc.). Program is cross-platform - written in Java language. Solution process of temperatures and heat flux is done by iterative scheme. It is also a tool facilitating experimental process - automating documentation of measurements and providing fast validation. Throughput and mean temperatures computed are compared with experimental data. Empirical coefficients are matched expressing influence of different filling volumes (ratios) on heat transfer. Various modes of heat transfer are included for cooling and heating of condenser/evaporator section.

INTRODUCTION

Thermosyphons are passive heat transferring devices, which are basically solid containers filled with working fluid. Working fluid starts to operate in cycle when bottom section of container is heated. Liquid substance evaporates, causing increased pressure in evaporation region. Vapour flows from higher pressure region to lower pressure region which is also cooler. It results in condensation of working fluid in cooler region. Liquid film is flowing down to evaporator section what closes the cycle. Liquid transport in thermosyphon, unlike other heat pipe types, is induced only by gravity. Thermosyphon won't work in microgravity environment and against gravity (evaporator section positioned higher than condenser section).

Many of two-phase closed thermosyphon experimental research includes attempts to model heat transfer in this device or to compare experimental heat transfer coefficients for boiling and condensation with literature or self-developed correlations. Jouhara and Robins [1] have compared heat transfer coefficients obtained from measurements with number of correlations. They have researched small diameter closed two-phase thermosyphons charged with water and dielectric working fluids. The best fit in their study, was Labuntsov correlation for pool boiling, classical Nusselt correlation for thin film evaporation (overall evaporation heat transfer coefficient was based on this two, that were considered as thermal resistances in parallel), and own correlation, based on Nusselt film condensation. Authors report over prediction of mean condensation heat transfer coefficient by Nusselt theory. Work of Hashimoto and Kaminaga [2] confirms heat transfer deterioration in condensation process caused by shear stress between phases. Gross [3,4] in series of articles propose correlation for reflux (counter flow) condensation heat transfer coefficient. Some authors [5] report good agreement of experimental values with Gross correlation. Often Rohsenow correlation for pool boiling is compared with heat transfer coefficients at evaporator section [5, 6]. Present study propose simple and effective program, which uses mentioned correlations to predict thermosyphon throughput and in situ comparison with measurements.

C_{SF}	[-]	Working fluid-surface coefficient
c_p	[J/kg K]	Constant pressure specific heat
g	[m/s^2]	Gravitational acceleration on Earth (9.81 m/s^2)
Gr	[-]	Grashof number $(g\beta(T_w-T_{av})L_{ev})/\upsilon$
h	[W/m^2 K]	Heat transfer coefficient
$\dot{q}$	[W/m^2]	Heat transfer density (per unit area)
l_b	[m]	Bubble length scale $\sigma/(g(\rho-\rho_g))^{0.5}$
l_l	[m]	Film thickness scale $(\mu^2/g(\rho-\rho_g))^{1/3}$
P	[Pa]	Pressure
Pr	[-]	Prandtl number $(\mu c_p)/\lambda$
r	[J/kg]	Specific heat of condensation/evaporation
Ra	[-]	Rayleigh number $Gr\ Pr$
T	[K]	Temperature
Special characters		
α	[-]	Volume fraction ratio
β	[1/K]	Thermal expansion coefficient
υ	[m^2/s]	Kinematic viscosity
ρ	[kg/m^3]	Density
λ	[W/mK]	Thermal conductivity
μ	[Pa s]	Dynamic viscosity
σ	[N/m]	Surface tension

Subscripts

atm	Atmospheric
av	Average
g	Gaseous state
sat	Saturation state
i	Internal
NC	Natural convection
PNB	Pool nucleate boiling
w	Wall

EXPERIMENTAL SETUP

Experimental work was done at Institute of Turbomachinery Lodz University of Technology. Thermosyphon (550 mm length) was heated and cooled by two heat exchangers mounted respectively on evaporator and condenser section. Distilled water being the coolant and heating agent. There was also short adiabatic section present. Schematic drawing of thermosyphon assembly (with heat exchangers) is shown in Figure 1. Thermosyphon container wall thickness is 1 mm. Wall material is brass. Heat exchangers are made from stainless steel, and whole assembly is thermally insulated to avoid heat losses. Thermocouples was positioned on the outer surface of thermosyphon wall. Three thermocouples on evaporator section, two on adiabatic section, and three on condenser section. Precise longitudinal positions of thermocouples are shown also in Figure 1. Two-phase closed thermosyphon inside pressure was also measured by pressure transducer.

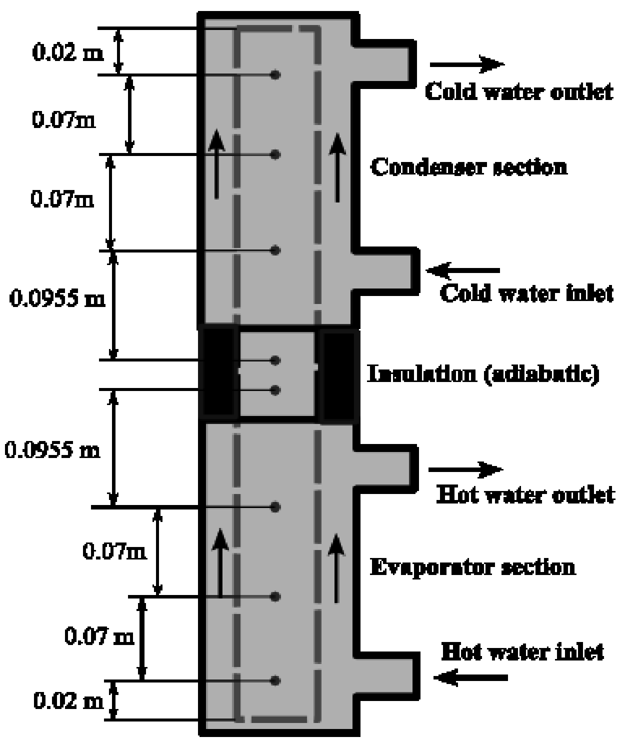

Figure 1 Schematic drawing of thermosyphon assembly with heat exchangers

From the measured pressure saturation temperature of boiling working fluid inside heat pipe container could be obtained. For further details of the experiment please refer to [7]. Results obtained from this study are used for CAE program validation.

WORKING FLUIDS

Working fluids considered in present study are modern refrigerants: R134a, R404a, R410a and R407c. Thermophysical properties of these substances are taken from ASHRAE Fundamentals [8]. Tabulated values of properties stored in text files are further linearly interpolated and used for CAE program computations. It has to be noted than refrigerant R134a is not a blend, but one component (pure) substance. Other mentioned refrigerants are blends (mixtures). Mixtures don't evaporate, or condense at constant temperature, but some temperature glide is observed (temperature varying with concentration) [8]. In Figure 2 standard T-x (temperature – mole fraction) diagram is shown. At temperature T_A mixture of two components starts to evaporate. This temperature for specified initial concentration is called bubble point temperature. Produced vapor is richer in more volatile component, and at the same time liquid concentration shifts towards less volatile component. As liquid is consequently boiled off, saturation temperature (at constant pressure) is rising. When vapor concentration attains initial liquid mixture concentration, it's the final concentration. There is no further temperature shift and when last drop of liquid vanish, vapor will be in the temperature corresponding to this final concentration. This temperature is called dew point temperature. Figure 2 refers to zeotropic blends. Azeotropic blend T-x curves touch at some point. This point will indicate final temperature and concentration.

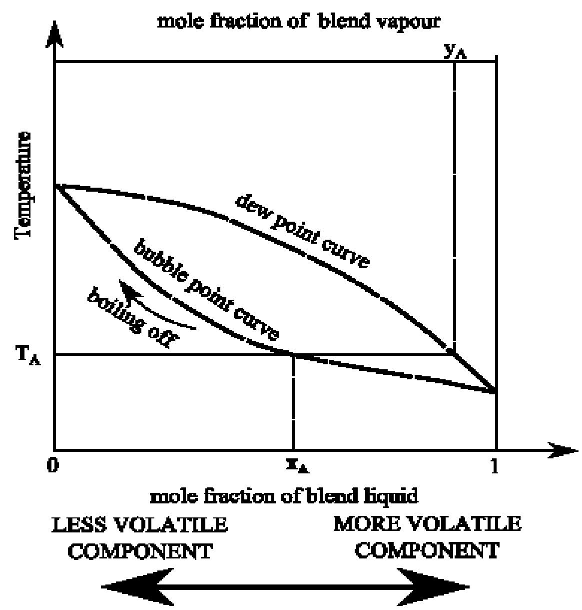

Figure 2 Temperature – mole fraction diagram for zeotropic blend at constant pressure

If mixture is not in this azeotropic proportions, it will change towards this fraction. Of course if phase change occurs and mixture is at azeotropic proportions, liquid has the same molar fraction as vapor. Refrigerant R404A is blend with near azeotropic proportions. “Near” means that small temperature glide is still present (for R404a the highest difference between bubble and dew point temperature is about 1K). R410A is even closer to azeotropic, being 50/50 blend of R32 and R125. R407C is most problematic because it is zeotropic. Saturation temperatures of R404A and R410A can be assumed as mean of boiling point temperature and dew point temperature. Temperature glide observed in case of R407C refrigerant can be as high as 7-8 K. For computations involving R407C refrigerant, bubble point temperature was assumed (as saturation temperature). Inside two-phase thermosyphon, in steady state conditions, liquid pool is constantly replenished with condensate. Liquid is not boiled off, so the assumption of nearly constant mole fraction (close to bubble point) seems to be valid. We know from experiments, that this is false assumption. Some effects are present – temperature of condenser is much lower than evaporator (relatively large temperature drop occurs along heat pipe). Because none of this phenomena were reported in the case of nearly azeotropic mixtures, this behaviour is most probably caused by zeotropic “nature” of refrigerant. Most important fact is that it makes this working fluid the worst (less heat throughput).

ALGORITHM OF THE PROGRAM

CAE program has been written in Java programming language. Java was chosen because of many advantages of this language (cross-platform, free – GNU license). For more detailed description of Java see [9]. Block diagram illustrating program algorithm is presented in Figure 3. Initial data can be imported from state files or filled manually by user. Properties of working fluids can be load from data files (linearly interpolated). Data and initial values are used by solve block that contains iterative algorithm. Algorithm simultaneously solves three balance equations: hot fluid stream (heating evaporator section), heat transfer through thermosyphon system and cold stream (cooling condenser section of heat pipe). Figure 4 shows screenshot of CAE program main window. Initially thermosyphon geometry, volumetric filling ratio and inside pressure can be entered. Inside pressure is experimental value that is used purely for validation – program itself calculates inside pressure. Also evaporator heating and condenser cooling heat transfer mode can be specified. There are two options present at evaporator section: convection and heat flux. Condenser is usually cooled by convection, so it is the only option available. Wall material can be selected (from popular metals) and also working fluid, which can be chosen from stated previously refrigerants. After hitting “Calculate” button solution is presented in the form of plots and diagrams: “Plots” tab. “Enter data” button allows to enter experimental data to validate results of computation with reality. In model menu item list of options is available for choosing respectively evaporation/condensation model. For convection “Real htc values” enables user to set convection heat transfer coefficients the same as obtained from experiment (if experimental data is available). Liquid - surface coefficient can be chosen if Rohsenow model for pool boiling was selected.

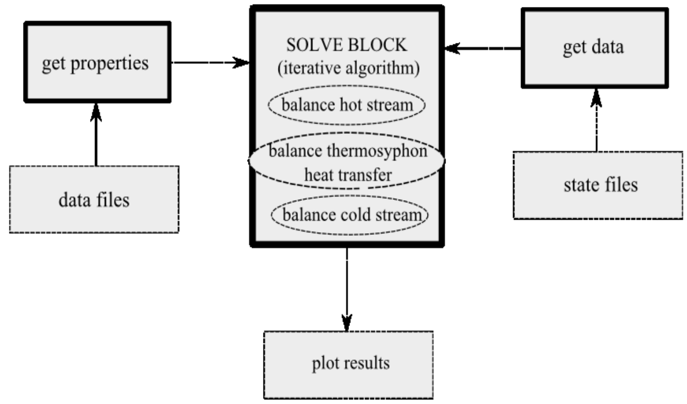

Figure 3 Block diagram illustrating CAE program algorithm

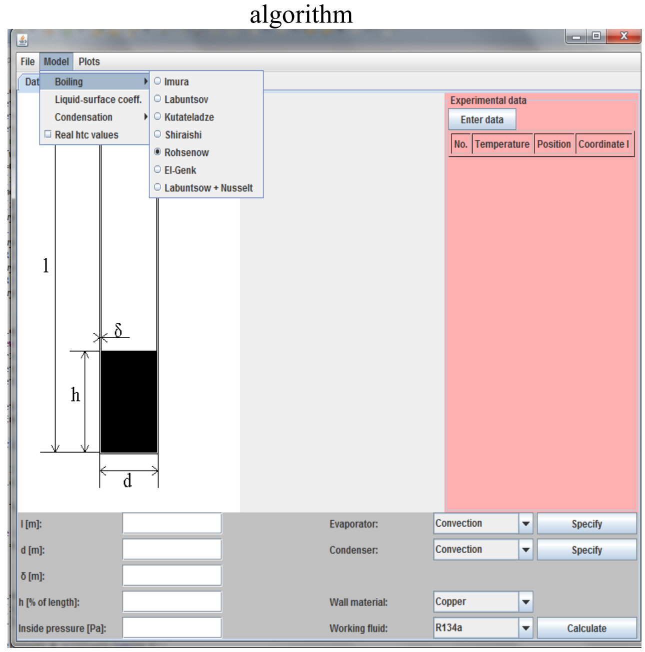

Figure 4 presents screenshot of CAE program with description.

HEAT TRANSFER COEFFICIENTS

Different boiling/evaporation models can be chosen. First of the options is Imura model, developed for thermosyphons:

$$h = 0.32 \cdot \left(\frac{\rho^{0.65} \lambda^{0.3} c_p^{0.7} g^{0.2}}{\rho_g^{0.25} r^{0.4} \mu^{0.1}} \right) \cdot \left(\frac{P}{P_{atm}} \right)^{0.3} \cdot \dot{q}^{0.4} \qquad (1)$$

Similar but slightly more sophisticated correlation was proposed by Labuntsov:

$$h = 0.075\left[1+10\left(\frac{\rho_g}{\rho-\rho_g}\right)^{0.67}\right]\left(\frac{\lambda^2}{\upsilon\sigma\left(T_{sat}+273.15\right)}\right)^{0.33}\cdot\dot{q}^{0.67} \quad (2)$$

Kutateladze modelled pool boiling as:

$$h = 0.44\left(\frac{\lambda}{l_b}\right)\left(\frac{1x10^{-4}\dot{q}\cdot P}{g\cdot r\cdot\rho_g\cdot\mu}\cdot\frac{\rho}{\rho-\rho_g}\right)^{0.7}\cdot\mathrm{Pr}^{0.35} \quad (3)$$

Shiraishi model is the same as Imura with slight correction. Rohsenow well-known pool boiling model:

$$h = \frac{\dot{q}^{2/3}}{\frac{C_{SF}\cdot r}{c_p}\cdot\left[\frac{l_b}{r\cdot\mu}\right]^{0.33}\cdot\mathrm{Pr}^{1.7}} \quad (4)$$

Where C_{SF} coefficient takes into account surface-working fluid pair. Mentioned models do not include liquid film evaporation. El-Genk [10] correlation takes falling, thin film evaporation/boiling into account, but was used with omission of film boiling part (assumption that film boiling is not present). El-Genk correlation consist of natural convection heat transfer coefficient:

$$h_{NC} = 0.475\cdot\left(\frac{\lambda}{d_i}\right)Ra^{0.35}\left(\frac{l_b}{d_i}\right)^{0.58} \quad (5)$$

Pool boiling heat transfer coefficient:

$$h_{PNB} = \left(1.0+4.95\psi\right)h_{Ku} \quad (6)$$

Where additional terms:

$$h_{Ku} = 6.95\times10^{-4}\cdot\left(\frac{\lambda}{l_b}\right)\cdot\mathrm{Pr}^{0.35}\left(\frac{\dot{q}\cdot l_b}{\rho_g\cdot r\cdot\nu}\right)^{0.7}\left(\frac{P\cdot l_b}{\sigma}\right)^{0.7} \quad (7)$$

and,

$$\psi = \left(\frac{\rho_g}{\rho}\right)^{0.4}\left\{\left(\frac{P\cdot\nu}{\sigma}\right)\left[\frac{\rho^2}{\sigma\cdot g\cdot\left(\rho-\rho_g\right)}\right]^{0.25}\right\}^{0.25} \quad (8)$$

Heat transfer coefficient of evaporating liquid film is given by equation:

$$h_x = \left(4/3\right)^{1/3}\left(\frac{\lambda}{l_l}\right)\cdot\mathrm{Re}^{-1/3} \quad (9)$$

All introduced terms are combined in one heat transfer coefficient. For nucleate boiling film term reader can refer to [10]. Last boiling/evaporation model uses Labuntsov correlation for pool boiling and Nusselt model for film evaporation, as suggested in [3]. As condensation model Nusselt correlation is used:

$$h = 0.943\left[\frac{g\cdot\rho\left(\rho-\rho_g\right)\cdot r^*\cdot\lambda^3}{\mu\cdot\left(T_{sat}-T_{wall}\right)\cdot L}\right]^{1/4} \quad (10)$$

Also Gross model for reflux condensation is used but because of its complexity it will not be presented, for more information see [3]. Above correlations were chosen due frequent use in literature on thermosyphons.

GRAPHICAL PRESENTATION OF RESULTS

Results obtained from computer program are bar charts and plots. Every form of graphical representation is adapted to validate computational with experimental data. As can be seen in Figure 5 modelled (computed) heat transfer rate can be compared with heat transfer rate obtained from experiment. It gives instantaneous estimate of accuracy of result obtained from CAE program. User can also check temperature distribution along thermosyphon (Figure 6) and validate it against experimental one. Real variation of measured temperature can be seen or averaged wall temperature on evaporator and condenser section (from numerical integration).

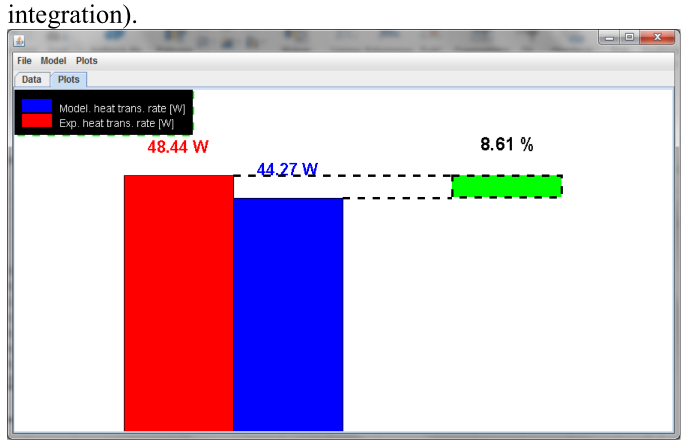

Figure 5 Model heat transfer rate (throughput) vs experimental heat transfer rate and their relative difference

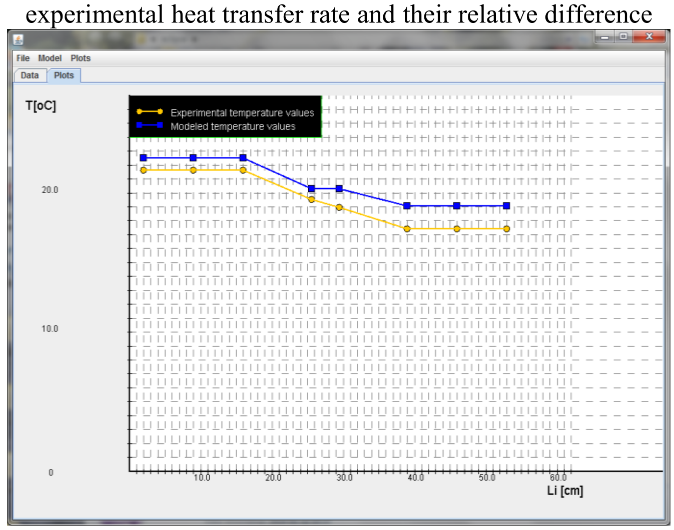

Figure 6 Experimental temperature distribution along thermosyphon vs modelled temperature distribution

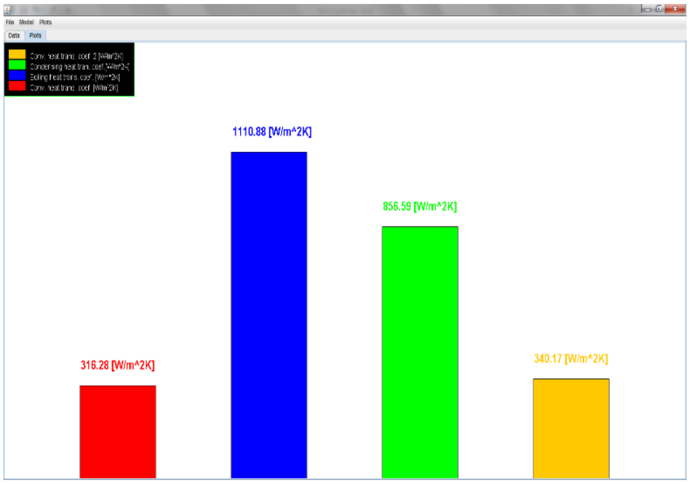

Figure 7 Experimental heat transfer coefficients, from left convection (of water heating evaporator section), boiling/evaporation, condensation and convection 2 (cooling water)

In Figure 7 option of display heat transfer coefficients on bar chart is presented. User can obtain experimental and respectively computed coefficients, along with thermal resistances.

VALIDATION OF COMPUTED RESULTS

Computationally obtained values of heat transfer coefficients and thermosyphon throughputs were validated against experimental ones. Some of correlations available in CAE program have been omitted. For example Imura correlation is represented by Shiraishi equation because they are very similar. Refrigerant chosen for validation procedure is R410A. R407C was rejected because of mentioned zeotropic behavior that is not fully understood and poor thermal performance. Two remaining refrigerants were not considered due to limited article space. Neither full analysis of R410A was given. Interested reader can refer to [7] for results for every refrigerant (R134a, R407C, R404A, R410A). In Figure 8 boiling/evaporation heat transfer coefficients are shown versus hot water inlet temperature (heating evaporator section). Condenser section is cooled by cold water of approximately constant temperature (about 10°C). Shiraishi correlation gives clearly over predicted results for 10% thermosyphon filling ratio. The same "growing" trend be seen for Rohsenow correlation, heat transfer coefficients are not as high but also considerably over predicted. Surface-working fluid pair coefficient has value 0.004-0.005 as often seen in literature. El-Genk equation, taking liquid film and pool heat transfer into consideration, exhibits downward trend with increasing temperature. Even obtained values are close to experimental, trend seems to be not realistic. Similar trend and values can be obtained from Labuntsov and Labuntsov plus Nusselt as recommended in [1]. Labuntsov exhibits sharper slope and Labuntsov plus Nusselt more "plane". Experimental coefficient decreases, starting from 35°C temperature. It is caused by dry out limit [10], which characterizes by lower throughputs, elevated evaporator section temperature and lower inside pressure. Proper operation in whole considered range (for 20% filling ratio) can be seen in Figure 9, where Shiraishi and Rohsenow correlations were not presented. They were omitted because of large over prediction. If dry out limit is not present experimental data exhibits increasing trend as expected. El-Genk correlation still shows downward slope. The most accurate are Labuntsov based correlations. Labuntsov steeper, and Labuntsov plus Nusselt slightly increasing. The unexpected result is that pure Labuntsov equation fits better than Labuntsov plus Nusselt, even the latter seems to be more physical due to explicit filling ratio consideration.

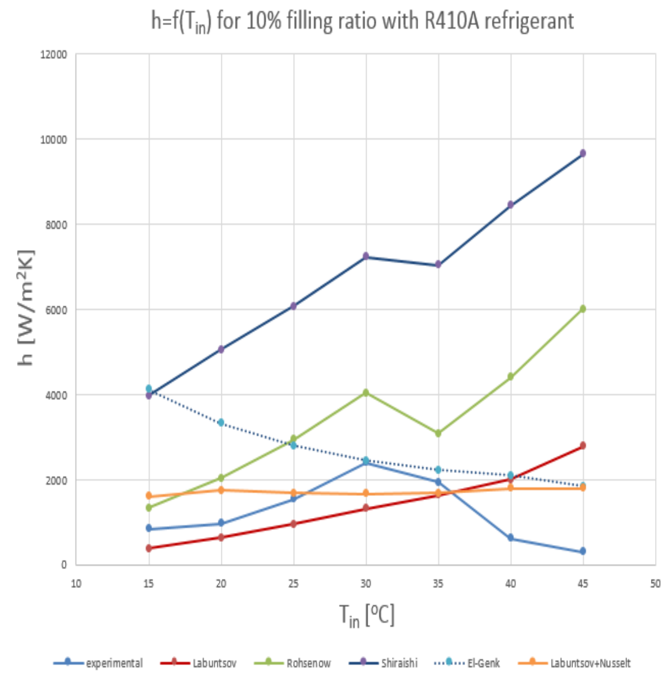

Figure 8 Boiling/evaporation heat transfer coefficients vs hot water inlet temperature (heating evaporator section) for 10% thermosyphon filling ratio with R410A refrigerant

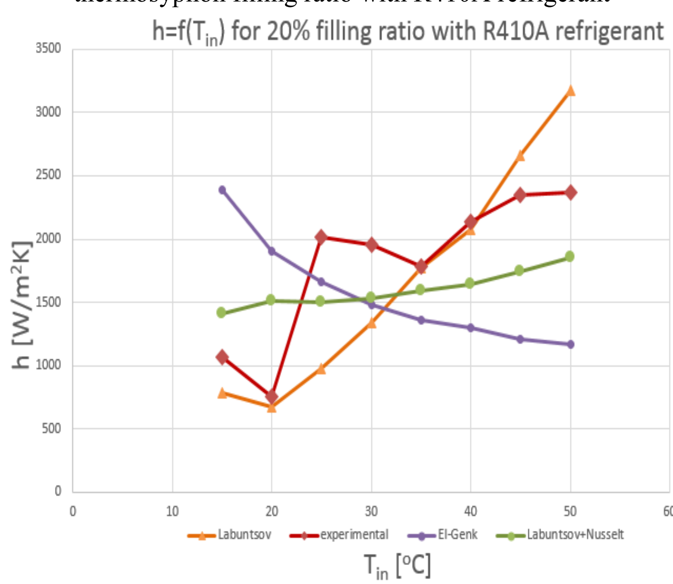

Figure 9 Boiling/evaporation heat transfer coefficients vs hot water inlet temperature (heating evaporator section) for 20% thermosyphon filling ratio with R410A refrigerant

These correlations can be further validated against filling ratio dependence. As can be seen in Figure 10, Labuntsov equation loses its advantage. Experimental heat transfer coefficients decrease almost linearly while these computed from El-Genk correlation decrease more sharply and Labuntsov exhibits nearly constant values. Best fit in this context is Labuntsov plus Nusselt, not so sharp, under predicted but almost linear and closest to experimental. Labuntsov plus Nusselt correlation seems to be the best choice, and will be chosen in throughput computation. Condensation heat transfer coefficients are presented in Figure 11. Gross correlation gives considerably higher heat transfer coefficients than Nusselt, and both are decreasing with increasing hot water inlet temperature. This fact can be explained by thinner liquid film at lower throughputs. However experimental heat transfer coefficients steadily increase (not decrease) and for lowest considered temperatures they differ up to 700% with Gross correlation. To show this results are not coincidental similar plot is presented in Figure 12 for 20% filling ratio. Second set of experimental results show similar trend to this seen in Figure 11. Clearly, at low throughputs both correlations fail. New correlation is needed, probably shearing stress here is deteriorating heat transfer coefficient, but more experimental data is necessary to develop it. Figure 13 present condensation heat transfer coefficient vs volumetric filling ratio of thermosyphon. Gross correlation produces higher heat transfer coefficients that are observed in reality and this behaviour prevails for every considered filling ratio. All coefficients are slightly increasing, except for 40% filling ratio sharp decrease is reported for experimental value. This has to be treated as experimenter error. Nusselt keeps very close agreement with experimental results. In this case filling ratio dependence seems to be very weak and of secondary importance. Heat throughput

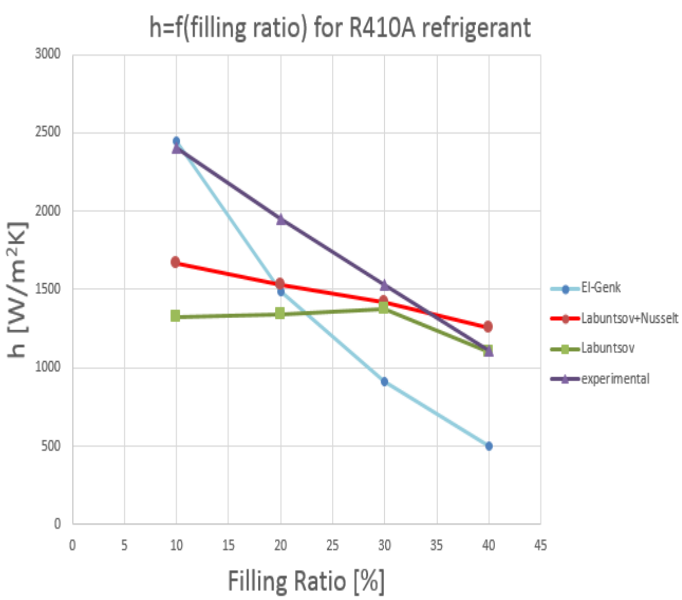

Figure 10 Boiling/evaporation heat transfer coefficients vs various filling ratios of thermosyphon with R410A refrigerant (for 30ºC isotherm)

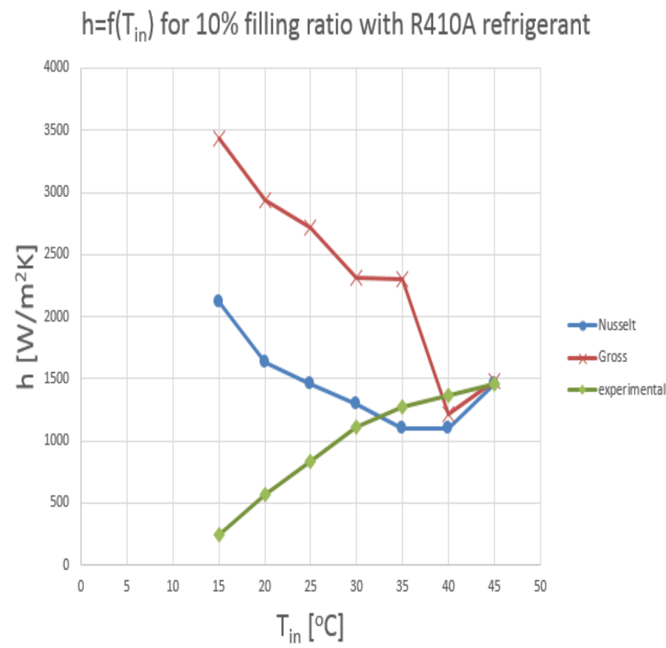

Figure 11 Condensation heat transfer coefficients vs hot water inlet temperature (cooling condenser section) for 10% thermosyphon filling ratio with R410A refrigerant

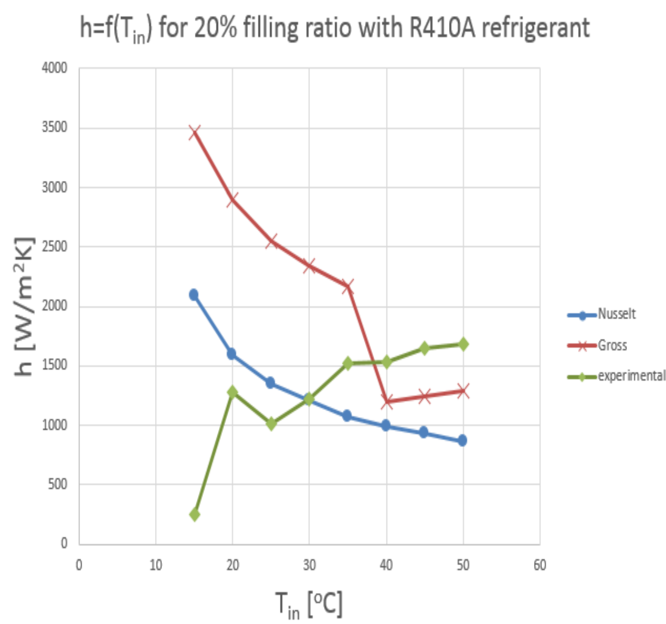

Figure 12 Condensation heat transfer coefficients vs hot water inlet temperature (cooling condenser section) for 20% thermosyphon filling ratio with R410A refrigerant

of thermosyphon was calculated with CAE program and presented in Figure 14. Gross correlation was chosen for modelling condensation heat transfer and Labuntsov plus Nusselt for boiling heat transfer. Generally relative difference between modelled throughputs and experimental does not exceed 25%. Due to over prediction of condensation heat transfer coefficients at lower temperatures (and lower throughputs) modelled heat transfer rate is higher than value obtained from experiment. For higher temperatures this tendency reverses and modelled throughputs are under predicted. Filling ratio dependence of throughputs is similar for modelled and experimental results. Experimental throughputs for 30^{o}C isotherm are higher than modelled. Considerable drop is reported for 40% filling ratio.

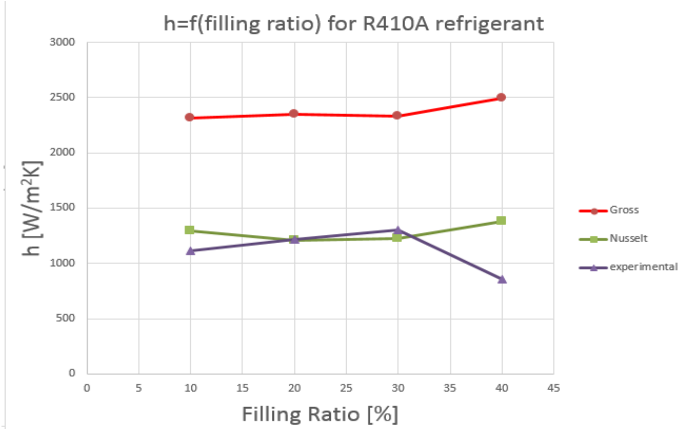

Figure 13 Condensation heat transfer coefficients vs filling ratio of thermosyphon with R410A refrigerant (for 30^{o}C isotherm)

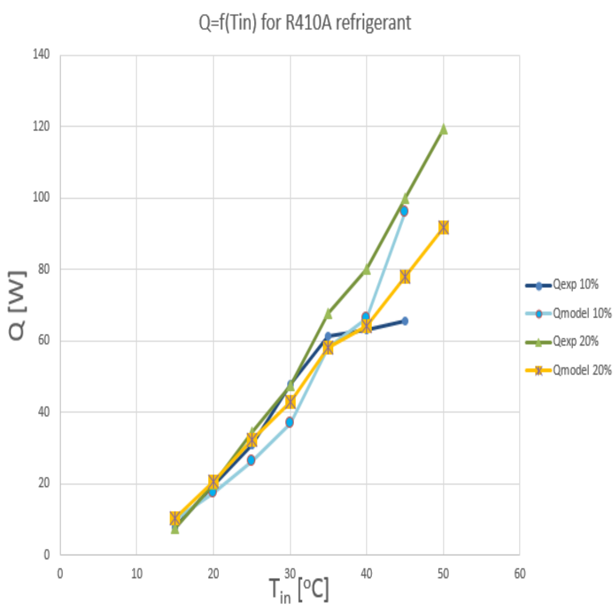

Figure 14 Thermosyphon throughput vs hot water inlet temperature for 10% and 20% filling ratios

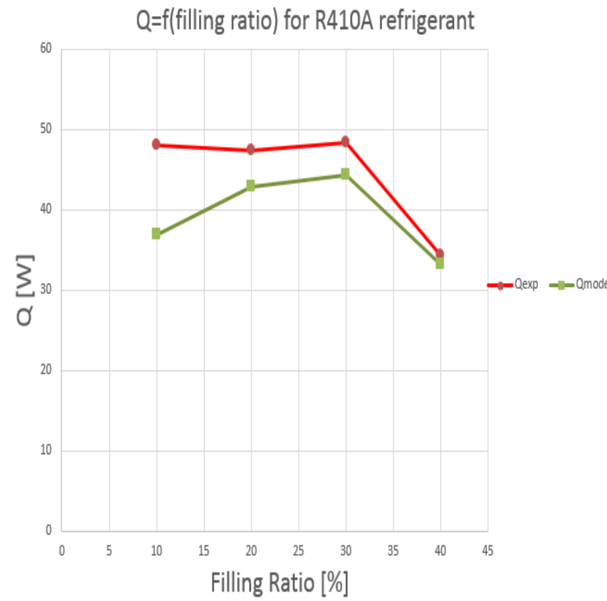

Figure 15 Thermosyphon throughput vs filling ratio of thermosyphon with R410A refrigerant (for 30^{o}C isotherm)

CONCLUSIONS

Generally good agreement of experimental throughputs with CAE program computations is observed. The best heat transfer coefficients correlations are chosen. Labuntsov, Labuntsov plus Nusselt and El-Genk gives the closest values to experiments for boiling/evaporation heat transfer coefficients. For condensation heat transfer coefficients Gross correlation was chosen. This was dictated by throughputs agreement, even Nusselt correlation was in some cases better (i. e. Figure 13). For low temperature ranges dry out limit can occur and it is reported for 10% filling ratio. Proposed CAE program cannot predict onset of dry out limit. Proposed correlations for falling thin film breakdown tend to produce over predicted results and were not presented (in whole considered temperature range breakdown of the film should occur). New model should be made or some corrections for old models should be introduced.

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REFERENCES

[1] Hussam Jouhara, Anthony J. Robinson, Experimental investigation of small diameter two-phase closed thermosyphons charged with water, FC-84, FC-77 and FC-3283, *Applied Thermal Engineering,* 2002, vol. 30, 2010 , pp. 201-211

[2] H. Hashimoto, F. Kaminaga, Heat transfer characteristics in a condenser of closed two-phase thermosyphon: effect of entrainment on heat transfer deterioration, *Heat Transfer-Asian Research*, vol. 31, 2002, pp. 212–225

[3] U. Gross, Reflux condensation heat transfer inside a closed thermosyphon, *International Journal of Heat and Mass Transfer*, vol. 35, 1992, pp. 279–294.

[4] U. Gross, Ch. Philipp, Conjugated shear stress and Prandtl number effects on reflux condensation heat transfer inside a vertical tube, *International Journal of Heat and Mass Transfer*, vol. 49, 2006, pp. 144–153.

[5] Yong Joo Park, Hwan Kook Kang, Chul Ju Kim., Heat transfer characteristics of a two-phase closed thermosyphon to the fill charge ratio, *International Journal of Heat and Mass Transfer*, Vol. 45, 2002, pp. 4655–4661

[6] S.H. Noie, Heat transfer characteristics of a two-phase closed thermosyphon, *Applied Thermal Engineering,* vol. 25, 2005, pp.495–506

[7]G. Gorecki, The Efficiency of the Heat Pipe of Known Geometry Using a Number of Refrigerants., submitted to HEFAT 2014.

[8] American Society of Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE Handbook: Fundamentals, 2005

[9] Schildt, H., Java The Complete Reference 9/E, McGraw-Hill Education, 2014.

[10] El-Genk M. S., Saber H. H., Determination of operation envelopes for closed two-phase thermosyphons, *International Journal of Heat and Mass Transfer*, vol. 42, 1999, pp. 778-892