Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2014

Non-Adiabatic Capillary Tubes In Cycle Simulations

Martin Heimel
Graz University of Technology, Austria, martin.heimel@tugraz.at

Erwin Berger
Graz University of Technology, Austria, berger@ivt.tugraz.at

Stefan Posch
Graz University of Technology, Austria, posch@ivt.tugraz.at

Martin Eichinger
Graz University of Technology, Austria, Eichinger@ivt.tugraz.at

Raimund Almbauer
Graz University of Technology, Austria, almbauer@ivt.tugraz.at

Non Adiabatic Capillary Tubes in Cycle Simulations

Martin HEIMEL[1]*, Erwin BERGER[2], Stefan POSCH[3], Martin EICHINGER[4], Raimund ALMBAUER[5]

[1, 2, 3, 4, 5] Institute for Internal Combustion Engines and Thermodynamics, Graz University of Technology, Inffeldgasse 19, 8010 Graz, Austria

[1]heimel@ivt.tugraz.at	+43 316 873 30235
[2]berger@ivt.tugraz.at	+43 316 873 30234
[3]posch@ivt.tugraz.at	+43 316 873 30233
[4]eichinger@ivt.tugraz.at	+43 316 873 30234
[5]almbauer@ivt.tugraz.at	+43 316 873 30230

* Corresponding Author

ABSTRACT

Capillary tubes including a suction line heat exchanger are typical expansion devices used in nowadays domestic refrigeration appliances. To account for their functionality in transient cycle simulations, including the highly transient start-up and shut-down operations, different capillary tube models and their implementation in such a cycle are investigated. These non-adiabatic capillary tube models comprise dimensionless correlations, neural network methods and one-dimensional homogeneous models which stem partly from open literature and from previous work of the authors. The range of parameters is chosen according to the need of domestic applications using R600a - mass flow rates range between 0 and 3 kg/h and inlet pressures up to 6 bar. It is concluded that the direct implementation of the 1d-code bears the disadvantage of low speed, whereas common dimensionless correlations lose accuracy off its design point. Neural networks turn out to be a good trade-off between speed, reliability and accuracy.

1. INTRODUCTION

Energy efficient cooling in household refrigeration is a topic which attracts interest more and more over the past decades. In order to successfully understand the loss mechanisms and detect space for further improvement simulation seems to be a capable tool to support established experimental methods. Especially in refrigeration experiments are characterized by long stabilisation time and measuring expenditure - problems which could be circumvented in simulation. Many different cycle simulations have been described in literature yet, a review can be found in Ding (2007), Qiao *et al.* (2010) or Hermes and Melo (2008) where a clear tendency towards transient cycle simulations is discernible.

This work is related to the topic "cycle simulation" insofar as a sub-model of such a simulation is the expansion device including a heat exchanger to the suction line. The difficulty herein lies in the determination of the mass flow rate through the capillary tube and the transferred heat per time to the suction side for given geometric and thermodynamic boundary conditions. The kind of models therefore range from semi-empirical correlations mostly fitted to experimental data to more complicated one-dimensional finite volume methods. For implementation in a cycle simulation the requirements to sub-models are manifold. Not only accuracy matters, but also speed and reliability is of importance. A highly accurate but slow and unstable model may be second choice, thus also less accurate but fast and reliable models have their legitimation.

In the following, three classes of models are picked out and investigated under the scope of transient application. The necessary boundary conditions stem from an existing cycle simulation and should pose a frame for the comparison of the individual capillary tube models. Also, for the sake of comparability, only one special geometry, the coaxial design is used here (Figure 1). It is used since this case poses a nearly analytic case in terms of heat transfer compared to the big amount of lateral designs, where the heat transfer is influenced by the finish of the soldering, the insulation, if there is heat conduction paste used and so on. At the end, results for mass flow rate and transferred heat during a typical On-period of the refrigeration system are compared and discussed.

2. CAPILLARY TUBE MODELS

As mentioned above, the geometry of this study is of concentric design. This is due to the fact that some of the correlations from literature are specially designed to work for this kind of capillary tubes. Also the available experimental data from Melo *et al.* (2002) refers to the concentric design. The refrigerant in use is R600a, an important feature to mention, since not all correlations claim the ability to be designed for more than one refrigerant.

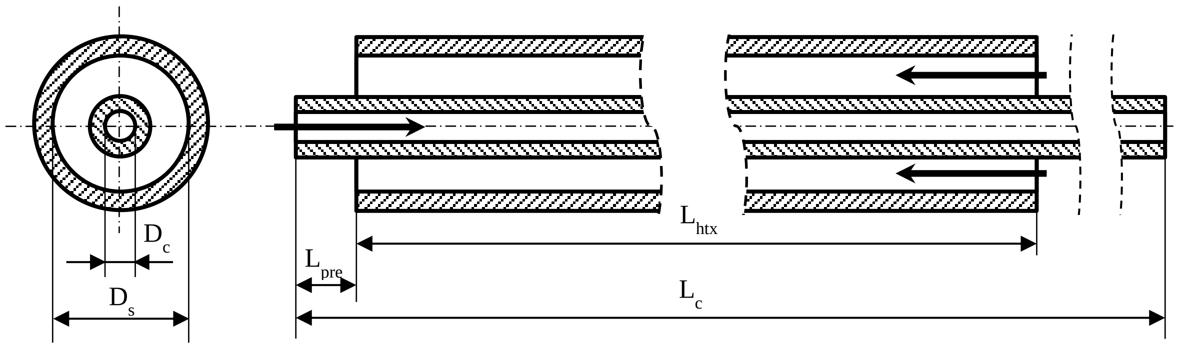

Figure 1: Sketch of the investigated concentric capillary tube.

The boundary conditions are manifold, the index *c* refers to capillary tube and *htx* to the suction line heat exchanger. Concerning geometry the inner diameters (D_c, D_s) as well as the length of the heat exchanger region (L_{htx}) and of the capillary (L_c) are of importance. Some models also use the adiabatic length (L_{pre}) as input quantity. In terms of thermodynamics, the inlet- and the outlet pressure ($p_{c,in}$, $p_{c,out}$) of the capillary and an average pressure of the suction line ($p_{s,in}$) must be given. Additionally a second property like enthalpy, quality or temperature must be given at each inlet (e.g. $h_{c,in}$, $h_{s,in}$) to determine the thermodynamic state.

2.1 1d-model

The most common treatment within this group of models is the homogeneous flow assumption where certain properties (e.g. viscosity) in the two phase region are calculated from saturation properties at the current pressure.

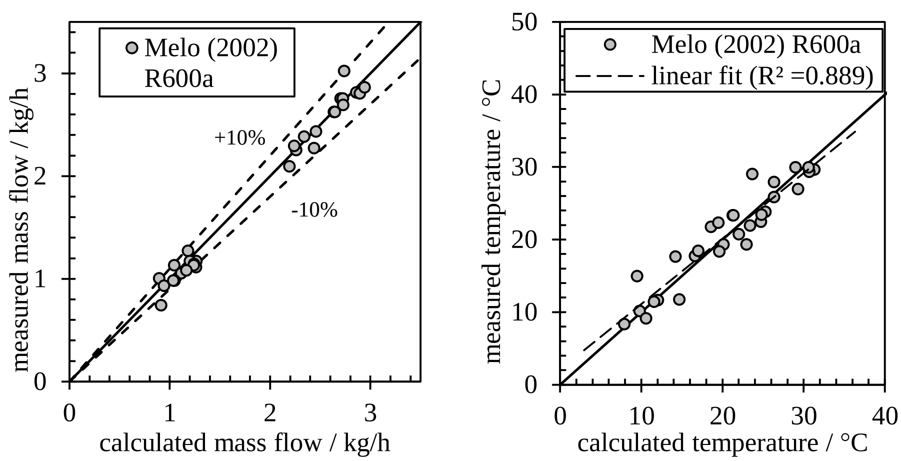

Figure 2: Validation by experiments from Melo *et al.* (2002) using a concentric capillary tube heat exchanger.

Applying this method and including heat transfer between suction line on basis of a finite volume formulation the flow within this expansion device can be described. Such a model is used in this study – for details the reader is referred to Xu and Bansal (2002) or Heimel *et al.* (2014). The model is able to calculate for steady state conditions the mass flow rate and suction line outlet temperature which are shown for measurement and experiment in Figure 2.

The adiabatic part of this model is presented in Heimel *et al.* (2012) and has been validated by now with over 500 points from literature and own experiments using R600a, R134a, R12, R152a and R22.

2.2 Artificial Neural Networks (ANN)

The method has been used to reproduce the characteristics of adiabatic and non-adiabatic capillary tubes based on experimental or simulation training data. The idea behind is that a set of matrices can be shaped in such a way that the mean squared error of given training data (input- and corresponding output values) reaches a minimum. This mathematical framework in combination with training data can be regarded as empirical model and is able to reproduce data within the range of the training data with low deviation. Application in the field of capillary tubes can be found in Islamoglu *et al.* (2009), Vins and Vacek (2005) or Heimel *et al.* (2014). Whereas in most applications ANNs are trained by experimentally gained datasets a different approach is used here. To overcome the shortcoming of having a too small parameter range like it is the case for experimental investigations where only a small number of geometrically different devices are tested, numeric training data is provided. Therefore, a randomly generated input file is created where the parameter range can be set for every single parameter. This input file is checked for plausibility and then evaluated by the 1d-code. The number of total points used for the ANN in this study is over 10^4 which prevents overfitting during training. The quality of the fit (1d-model / ANN) for mass flow rate is -0.0003 kg/h mean deviation and 0.042 kg/h standard deviation and for the transferred heat 0.01 W mean deviation and 2.25 W standard deviation. The network configuration is as follows: 10 inputs, 2 outputs, 30 neurons in the first hidden layer and 15 neurons in the second hidden layer are used. The accuracy of the model gets lower when one or more input parameters are close to the boundary of its range. Again, these specifications of accuracy refer to the comparison between 1d-model and ANN and not to measurements and ANN.

2.3 Correlations (empirical, semi-empirical, analytic)

A popular method for describing the mass flow rate trough adiabatic capillary tubes are dimensionless correlations, also for non-adiabatic capillaries such formulations can be found. The following ones are applicable for R600a and predict mass flow rates. Some of them also predict the outlet temperature of the suction line heat exchanger.

- **Melo *et al.* (2002)**

$$T(°C) = 10.0861 + 2.3625\mathrm{p}_{in} + 2.4964T_{sub} + 5.3390D_c + 11.4987L_{htx} - 3.1265D_s + 0.1446p_{c,in}D_s - 4.4467T_{sub}D_c + 0.2263T_{c,in}L_{htx} - 0.0728L_cL_{pre} \tag{1}$$

$$\dot{m}(kg/h) = -7.1650 + 0.1755\mathrm{p}_{c,in} + 0.8454L_c + 12.7375D_c + 0.0276T_{sub} + 0.0960L_{htx} - 0.0005\mathrm{p}_{c,in}T_{c,in} - 0.0150T_{sub}L_{pre} - 1.6512DL + 0.0024L_{htx}D_s \tag{2}$$

The units are: pressure in bar, diameter in mm, length in m, temperature in °C. The mass flow rates as well as the suction pressure are assumed to be equal for the capillary tube and the heat exchanger. These equations are derived from steady state measurements which feature subcooled inlet and choked flow conditions.

- **Sarker and Jeong (2012)**

$$\frac{\dot{m}}{D_c^2\sqrt{p_{c,in}\rho_{c,in}}} = 4.7136\left(\frac{D_c}{L_c}\right)^{0.490041}\left(\frac{L_s}{L_c}\right)^{0.08562}\left(\frac{c_{p\,s,l}}{c_{p\,s,v}}\right)^{0.03136}\left(\frac{p_{c,in}-p_{c,sat}}{p_{crit}}\right)^{0.012101} \tag{3}$$

for subcooled inlet

$$\frac{\dot{m}}{D_c^2\sqrt{p_{c,in}\rho_{c,in}}} = 3.9976\left(\frac{D_c}{L_c}\right)^{0.4955}\left(1-\mathrm{x}_{c,in}\right)^{-0.6819}\left(\frac{c_{p\,s,l}}{c_{p\,s,v}}\right)^{-0.2895}\left(\frac{\lambda_{s,l}}{\lambda_{s,v}}\right)^{0.3118}\left(\frac{\rho_{s,l}}{\rho_{s,v}}\right)^{-0.3332} \tag{4}$$

for two-phase inlet

These correlations were developed by using training data which comes from a capillary tube model by Kim *et al.* (2011). Finally, the correlations were validated by experiments using R134a and R600a.

- **Hermes *et al.* (2010b)**

$$\dot{m}_{ad} = \Phi \sqrt{\frac{D_c^5}{L_c}\left[\frac{p_{c,in}-p_f}{\nu_f} + \frac{p_f - p_{c,out}}{a} + \frac{\nu_f p_f k}{a^2} ln\left(\frac{a p_{c,out}+b}{a p_f + b}\right)\right]} \tag{5}$$

$$\frac{\dot{m}}{\dot{m}_{ad}} = 1.11\left(\frac{L_s}{L_c}\right)^{1.11}\left(\frac{D_s}{D_c}\right)^{-0.335} \varepsilon^{-0.366}\left(\frac{\nu_f \mu_f}{\nu_v \mu_v}\right)^{-0.281} \tag{6}$$

ν is the specific volume and μ the dynamic viscosity at flash-point and at evaporating conditions. The heat exchanger effectiveness ε results by applying the ε-NTU method. The subscript f stands for conditions at the flash point – for further details the reader is referred to the original publication. Similar formulations can be found in literature for adiabatic conditions, for instance Zhang and Ding (2004). The experimental validation was performed using R600a and R134a as refrigerant.

$$\Phi = 6.0 \quad a = \nu_f(1-k) \quad b = \nu_f p_f k \quad k = 1.63 \cdot 10^5 {p_f}^{-0.72} \quad \varepsilon = \frac{NTU}{1+NTU} \tag{7}$$

$$T_{s,out} = T_{s,out} + \varepsilon\left(T_{c,in} - T_{s,in}\right) \tag{8}$$

- **Wolf and Pate (2002)**

$$\frac{\dot{m}}{D\mu} = 0.07602\left(\frac{L_c}{D_c}\right)^{-0.4583}\left(\frac{L_s}{D_c}\right)^{0.07751}\left(\frac{p_{c,in}{D_c}^2}{\mu_{c,l}^2 \nu_{c,l}}\right)^{0.7342}\left(\frac{p_s {D_c}^2}{\mu_{c,l}^2 \nu_{c,l}}\right)^{-0.1204}\left(\frac{T_{c,sub} c_p {D_c}^2}{\mu_{c,l}^2 \nu_{c,l}^2}\right)^{0.03774}$$
$$\left(\frac{T_{s,sup} c_p {D_c}^2}{\mu_{c,l}^2 \nu_{c,l}^2}\right)^{-0.04085}\left(\frac{\mu_{c,l}-\mu_{c,v}}{\mu_{c,l}}\right)^{0.1768} \tag{9}$$

for 1K < $T_{c,sub}$ < 17K

$$\frac{\dot{m}}{D_c \mu_{c,l}} = 0.01960\left(\frac{L_c}{D_c}\right)^{-0.3127}\left(\frac{p_{c,in}{D_c}^2}{\mu_{c,l}^2 \nu_{c,l}}\right)^{1.059}\left(\frac{p_s {D_c}^2}{\mu_{c,l}^2 \nu_{c,l}}\right)^{-0.3662}\left(1 - \mathrm{x}_{c,in}\right)^{4.759}\left(\frac{T_{s,sup} c_p {D_c}^2}{\mu_{c,l}^2 \nu_{c,l}^2}\right)^{-0.04965} \tag{10}$$

for 0.02 < $x_{c,in}$ < 0.1

These correlations base on experiments applying R600a, R410a, R22 and R134a and are formulated for subcooled and two-phase inlet conditions seperately.

The common correlation by Bittle *et al.* (1995) for mass flow rate and suction line outlet temperature is designed specifically for R152a and therefore not used in this study. Comparing these correlations with R12-measurements, the deviation in mass flow appears to be 20 % and in suction line outlet temperature 2 to 3 °F, respectively.

3. TEST-ENVIRONMENT: CYCLE SIMULATION

As mentioned earlier, the boundary conditions for the comparison are taken from a simulation model which is adapted for reproducing the transient behaviour of an upright freezer (A+ / 160 l / R600a). The heat exchangers are modelled as one-dimensional finite volumes, the capillary tube model with suction line heat exchanger is based on before-mentioned Artificial Neural Network. The compressor model is based on a semi-empirical model for discharge temperature, mass flow rate and electric power. The compartment consists of six walls where one-dimensional transient heat transfer is treated. Also a door heating and an accumulator are considered. Figure 3 shows

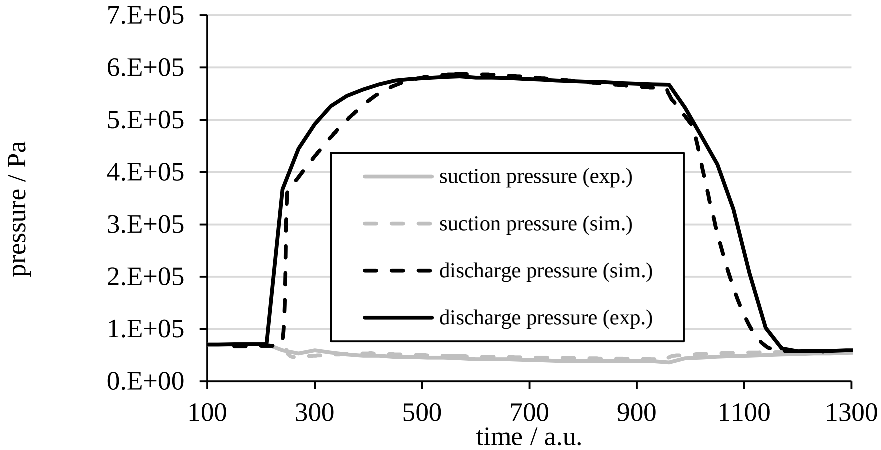

Figure 3: Comparison of pressure. Discharge- and suction pressure from measurements and simulation.

the pressure over time for cycling transients at -18 °C compartment temperature and 25 °C ambient temperature. Since not only the pressure but also other parameters like electric power, duty cycle ration and temperatures fit well to the experiments, there is thus reason to presume that also other thermodynamic parameters like enthalpy or quality are within a reasonable range if measurements of these quantities could be available. The observed case from Figure 3 leads to a set of boundary conditions in terms of pressure, temperature and quality where the capillary tube faces two phase inlet conditions also during the On-periods. This can be observed experimentally too, that is where the temperature at capillary inlet equals the saturation temperature at discharge pressure. A second set of boundary conditions has also been derived from this configuration where the inlet condition for the capillary tube is subcooled most of the time.

4. DERIVED TEST CASE AND SETUP

The obtained boundary conditions are shown in Figure 4 in terms of pressure, quality and temperature. For the first case Figure 4 (a, b) the inlet conditions are always in the two phase regime. At the suction-line inlet superheated refrigerant is always present. In Figure 4 (c, d) the capillary tube operates under subcooled inlet-conditions but still, at the beginning and end of each On-phase the refrigerant at the inlet becomes superheated. This is because most of the refrigerant is collected in the evaporator, hence the condenser runs dry. This dry-out clearly can be seen in Figure 4 (a). The slope of the discharge pressure changes the first time when the compressor switches off, the second change in the slope is reached when the condenser runs out of liquid refrigerant.

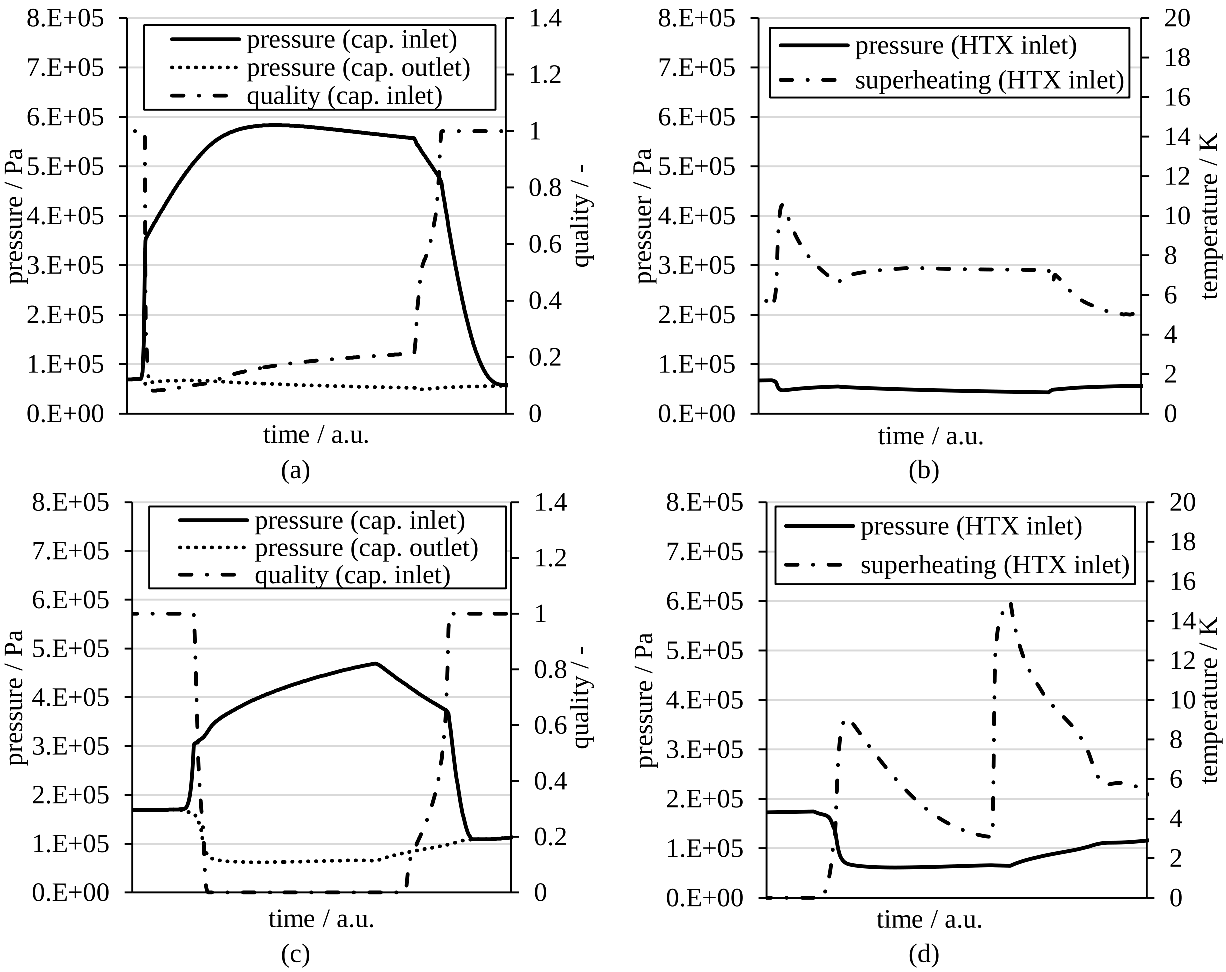

Figure 4: Pressure, temperature and quality boundary conditions from the test case 1 (subcooled inlet during On-phase) and test case 2 (two-phase inlet during On-phase).

This phase continues until equalisation of suction- and discharge pressure. It has to be noted, that the mass flow rate through the capillary tube and the suction line heat exchanger are not necessarily the same, particularly during changes of the compressor speed. The same applies for the suction pressure - given high pressure drops in a long evaporator the difference can be as much as 0.3 bar under certain transient conditions - therefore the pressure at capillary tube outlet and suction-line heat exchanger differ in reality. The geometry is similar to the real device and shown in Table 1, where also the maximum and minimum values of the main thermodynamic properties are listed.

Table 1: Main thermodynamic min-, max values and geometry of the assumed non-adiabatic capillary tube used for the test-cases.

	D_c	L_c	D_s	L_s	L_{pre}	$p_{c,in}$	$x_{c,in}$	T_{sub}	$p_{c,out}$	$p_{s,in}$	T_{sup}
	mm	m	mm	m	m	bar	-	°C	bar	bar	°C
nominal	0.75	3.26	4.6	1.6	0.2						
TEST 1: max						5.83	1	0	0.69	0.67	10.5
TEST 1: min						0.56	0.08	0	0.49	0.43	5.0
TEST 2: max						4.69	1	7.4	1.70	1.74	14.9
TEST 2: min						1.09	0	0	0.62	0.61	> 0

Concerning time scales, the duration of the On-cycle lies within the range of minutes, the duty cycle ratio is around 30 %.

5. RESULTS AND COMPARISON

The before-mentioned boundary conditions are applied to the models shown in section 2.3. as well as to the 1d-code and a Neural Network derived from the 1d-code. Due to its universality the 1d-code is believed to be the best estimate of the mass flow rate compared to other models and serves as reference in this investigation. The results in terms of capillary tube mass flow rate and suction line heat exchanger outlet temperature are plotted, irrespective of the range of applicability since a tool is sought after which should be able to cover the whole cycle. The only criteria is that the model has to be designed to work with R600a as refrigerant for a coaxial heat exchanger design. All of the following models apart the 1d-model are comparably fast in calculational terms – how they behave under real world operating conditions is investigated next.

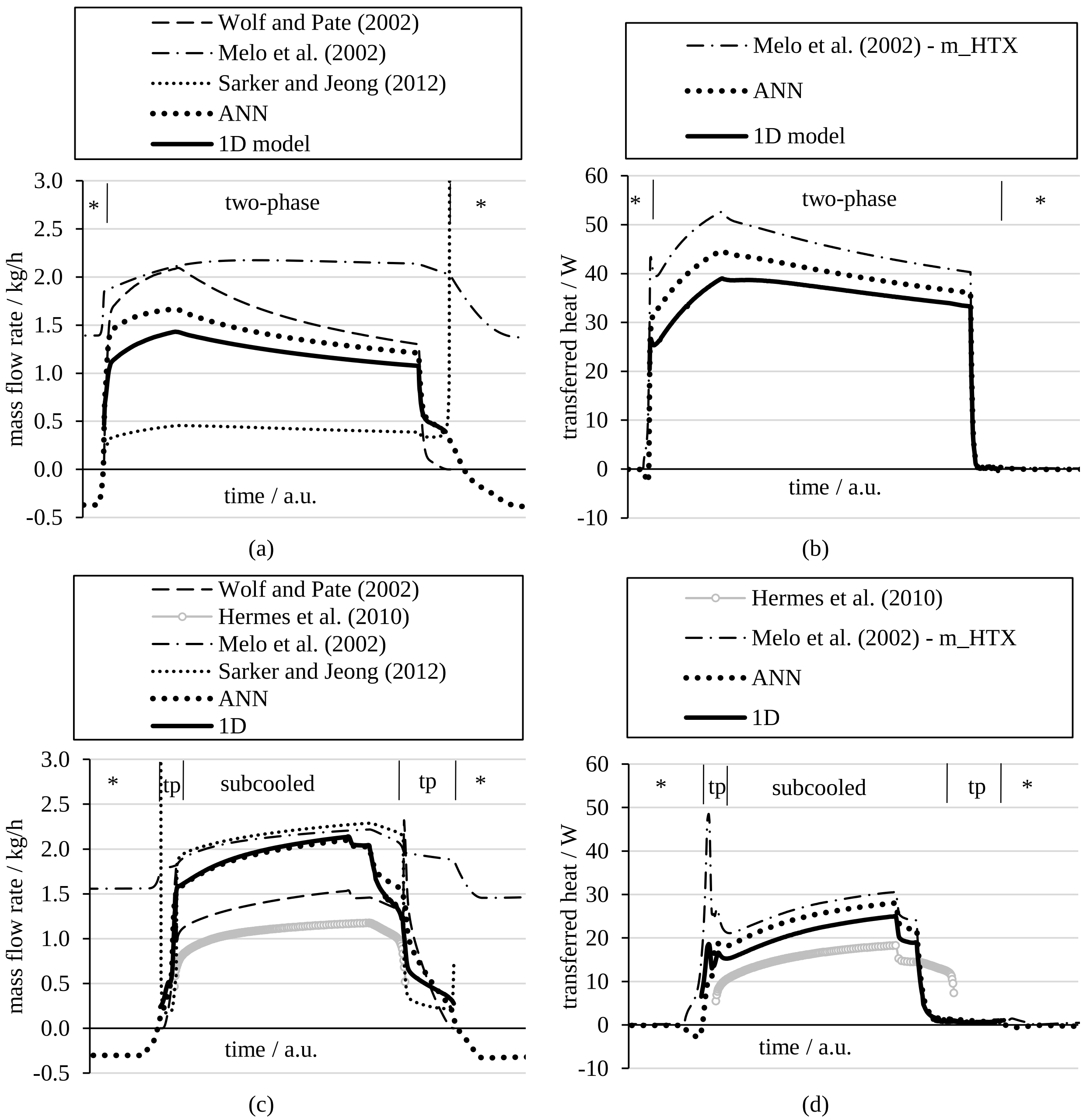

Figure 5: Calculated mass flow rates and transferred energy for two phase inlet situation (a, b) and subcooled situation (c, d). * stands for superheated vapour.

As one can see from Figure 5, some models are not capable of dealing with two phase inlet conditions (a, b). Despite the fact that this situation is not desirable for normal operation in a refrigeration device, it occurs during the startup and shutdown of the compressor. It also happens in the case of off-design conditions or in devices with unfortunate layout of components. Also visible on a first glance is the big spreading of mass flow rates and transferred heat, even during the nearly stable conditions. The mass flow rates vary by the order of 80 % (a) and 50 % (c) given the highest value as base. Better agreement is achieved when it comes to transferred heat where the spreading is 25 % (b) and 40 % (d), respectively. The ANN represents the 1d-model best in every case although in (a) and (b) the difference is a little higher than expected from training. The transferred heat is computed by taking the mass flow rate through the suction line which is given and the temperature difference between suction line in- and outlet.

Two phase inlet conditions: Only three of the five models from literature were able to predict mass flow rates. The model of Wolf and Pate (2002) stays closest to the 1d-model and follows its trend without any discontinuities. The correlation of Sarker and Jeong (2012) predicts a quite low mass flow rate which rises steeply when getting close to superheated conditions because of the term (1-x) in combination with a negative exponent. The correlation of Melo *et al.* (2002) is originally not designed for two phase inlet conditions but still predicts mass flow rates within a reasonable range although the trend is not the same as the reference shows. Concerning transferred heat, this model is the only one applicable to this situation although, again, it was not designed to be used for two phase inlet.

Subcooled inlet conditions: Among these tests, the algebraic solution of Hermes *et al.* (2010b) can be applied. This unique model predicts rather low mass flow rates and transferred heat compared to others for this test case. Its application ends where two phase flow starts to be present at the inlet. The correlations of Wolf and Pate (2002) stick close to the reference 1d-model but show significant peaks during transition from subcooled to two phase region. That is when Equation (10) is suddenly used instead of Equation (9). Sarker and Jeong's correlations for mass flow rate shows very good agreement with results of Melo *et al.* (2002) and, as mentioned above, turns to high values close to superheated conditions. The correlations of Melo *et al.* (2002) meet the 1d-model in the subcooled region and remain at a nearly constant offset. This is due to its nature of summing up the single contributions of influencing factors.

Table 2: Relative calculating time of each capillary tube model referred to ANN.

	Hermes *et al.* (2010)	Melo *et al.* (2002)	Sarker and Jeong (2012)	Wolf and Pate (2002)	ANN	1d
relative calculating time (referred to ANN)	56.2	3.3	5.3	7.8	1.0	> 10^6
property calls per calculation	28	2	4	6	0	n.a.

The calculation times of the models compared to ANN are given in Table 2. This test was carried out using REFPROP v9.0 (Lemmon et al., 2010) for all thermodynamic property calculations. The property functions are embedded in Visual Basic programming environment where the test was carried out. The 1d code was evaluated using a different programming language but due to its high calculating time this fact is of minor importance. For 10^3 datapoints the time was measured, neglecting the duration of input and output. First of all, it can be seen, that the 1d code is slowest of all and that the rest of the models differs only by a factor of 50. Noteworthy, the thermodynamic property conversions take its time and are the main reason for the differences in speed. The model of Hermes et al (2010) depends due to its iterative nature also on the quality of the first guess – 4 iterations were necessary in this case.

5. CONCLUSION

- Applying available correlations from literature to a coaxial capillary tube suction line heat exchanger working with R600a, a rather large spread is observed for both a cycle under subcooled and two-phase inlet conditions. Irrespective of the actual mass flow rate or transferred heat this indicates that the presented correlations should be used with great care. Especially at transition regions from subcooled to two-phase inlet.

- In literature many publications rely on experimental methods for data generation. The charm of this approach is that also physical effects which may be omitted in simulation are captured and are therefore, although unconsciously, considered. A big disadvantage, though, is the restriction in time and parameter range. Experiments in the field of refrigeration are time consuming, especially when steady state conditions are required. Also, some quantities which characterize the thermodynamic state of the fluid (e.g. vapour quality or density) are not measurable satisfactorily at the inlet of the capillary tube and the suction line without changing the system by means of measurement equipment. Therefore, the suggested approach for training data generation for capillary tube models which should cover the range from beginning of the On-phase until its end is, to use a validated model to avoid a too small span of parameters.

- Neural Networks, are able to represent any 1d-capillary tube model well far beyond standard boundaries and experimental limits. The output is continuous which might be an interesting feature for application. The drawbacks of this method are, firstly, a capillary tube model is necessary to generate enough training data which is time consuming and requires experimental validation. Secondly, ANNs are non-physical and may therefore predict unnatural values at and beyond the boundaries - for instance negative mass flow rates. Therefore, special care has to be taken at these positions and suitable mechanism have to be applied to avoid misinterpretation.

- Concerning calculational speed, the usage of any correlation mentioned above as well as the usage of ANN lies in the same order of magnitude. The exact timespan depends on the programming language, the implemented thermodynamic property tables and the programming skills. For methods other than the 1d-code the duration of a single calculation is in the order of milliseconds whereas a 1d-code takes seconds up to several minutes due to iterative solution algorithms and countless calls of thermodynamic property-functions.

NOMENCLATURE

T	temperature	(°C)
D	inner diameter	(m)
L	length	(m)
p	pressure	(Pa)
x	quality	(-)
$\dot{m}$	mass flow rate	(kg/h)
c_p	specific heat	(J/kg K)

Greek symbols

v	specific volume	(m^3/kg)
μ	dynamic viscosity	(Pa s)
ε	heat exchanger eff.	(-)
ρ	density	(kg/m^3)
λ	thermal conductivity	(W/m K)

Subscripts

sat	saturation	(-)
sub	subcooled	(-)
sup	superheated	(-)
v	vapour	
in	inlet	(-)
out	outlet	(-)
crit	critical	(-)
f	flash point	(-)
s	suction line	(-)
l	liquid	(-)

REFERENCES

Bittle, R.R., Stephensson, W.R., Pate, M.B., 1995, An experimental evaluation of capillary tube-suction line heat exchanger performance with R-152a, *ASHRAE Trans. Res.*, 101: p. 124-35.

Ding, G. L., 2007, Recent developments in simulation techniques for vapour-compression refrigeration systems, *Int. J. Refrigeration*, vol. 30: p. 1119-1133.

Heimel, M., Lang, W., Berger, E., R., A., 2012, A Homogenous Capillary Tube Model - Comprehensive Parameter Studies Using R600a, *International Refrigeration and Air Conditioning Conference at Purdue*, West Lafayette, USA.

Heimel, M., Lang, W., Almbauer, R., 2014, Performance predictions using Artificial Neural Network for isobutane flow in non-adiabatic capillary tubes, *Int. J. Refrigeration*, vol. 38: p. 281-289.

Hermes, C. J., Melo, C., 2008, A first-principles simulation model for the start-up and cycling transients of household refrigerators, *Int. J. Refrigeration*, vol. 31: p. 1341-1357.

Hermes, C. J., Melo, C., Knabben, F. T., 2010b, Algebraic solution of capillary tube flows. Part II: Capillary tube suction line heat exchangers, *Appl. Therm. Eng.*, vol. 30: p. 770-775.

Islamoglu, Y., Kurt, A., Parmaksizoglu, C., 2005, Performance prediction for non-adiabatic capillary tube suction line heat exchanger: an artificial neural network approach, *Energy Convers. Manage.*, vol. 46: p. 223-232.

Lemmon, E. W., Huber, M. L., McLinden, M. O., 2010, NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties - REFPROP, Version 9.0, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, MD, USA.

Melo, C., Vieira, L. A. T. and Pereira, R. H., 2002, Non-adiabatic capillary tube flow with isobutane, *Appl. Therm. Eng.*, vol. 22: p. 1661-1672.

Wolf, D. A., Pate, M. B., 2002, Capillary Tube-Suction Line Heat Exchanger Performance with Alternative Refrigerants, ASHRAE Research Project, Report 948.

Qiao, H., Radermacher, R., Aute, V., 2010, A Review for Numerical Simulation of Vapor Compression Systems, International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, USA.

Sarker, D., Jeong, J., 2012, Development of empirical correlations for non-adiabatic capillary tube based on mechanistic model, *Int. J. Refrigeration*, vol. 35: p. 974-983.

Vins, V., Vacek, V., 2009, Mass flow rate correlation for two-phase flow of R218 through a capillary tube, *Appl. Therm. Eng.*, vol. 29: p. 2816-2823.

Xu, B. and Bansal, P. K., 2002, Non-adiabatic capillary tube flow: a homogeneous model and process description, *Appl. Therm. Eng.*, vol. 22: p. 1801-1819.

Zhang, C., Ding, G., 2004, Approximate analytic solutions of adiabatic capillary tube, *Int. J. Refrigeration*, vol. 27: p. 17-24.

ACKNOWLEDGEMENT

This work has been carried out within the framework of ECO-COOL, a research project initiated and funded by the FFG (Austrian Research Promotion Agency). Furthermore the authors particularly acknowledge the technical support by Secop Austria GmbH, formerly ACC Austria GmbH and Liebherr-Hausgeräte Lienz GmbH.