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Heat transfer and pressure drop analysis of automotive HVAC condensers with two phase flows in minichannels

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ABSTRACT

The air cooled automotive condensers under study are of brazed aluminium tube and center, consisting of one row array of horizontal parallel multi-port flat tubes with louver fins on air side. Each tube is with a number of smooth parallel minichannels for internal flow of refrigerant. The analysis uses decomposition of the condenser along refrigerant flow path into specific different zones as follows: two single phase zones, namely: superheated and subcooled, and a few zones of two phase flow that can appear along some specific condensation paths to be: annular/intermittent/bubble or annular/annular-wavy/intermittent/bubble or annular/wavy/stratified. The approach presented is based on experimental correlations for heat transfer and pressure drop. The heat transfer prediction is performed using ε – NTU_o methodology. The results of the analysis refer to overall heat transfer rate, heat transfer in particular zones, pressure gradients heat transfer coefficients, vapour quality, condensation paths.

KEY WORDS

air cooled condenser, heat transfer and pressure drop analysis, minichannels

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Introduction

Vast fraction of passenger cars and trucks produced worldwide are vehicles equipped with air conditioning for which a refrigerant condenser is required to reject cycle energy gain to the environment. The condensers, like other automotive heat exchangers, subject to fulfil some specific automotive requirements such as compactness, low weight and low pressure drop. The compactness is of primary importance because of two reasons, small available chassis for the

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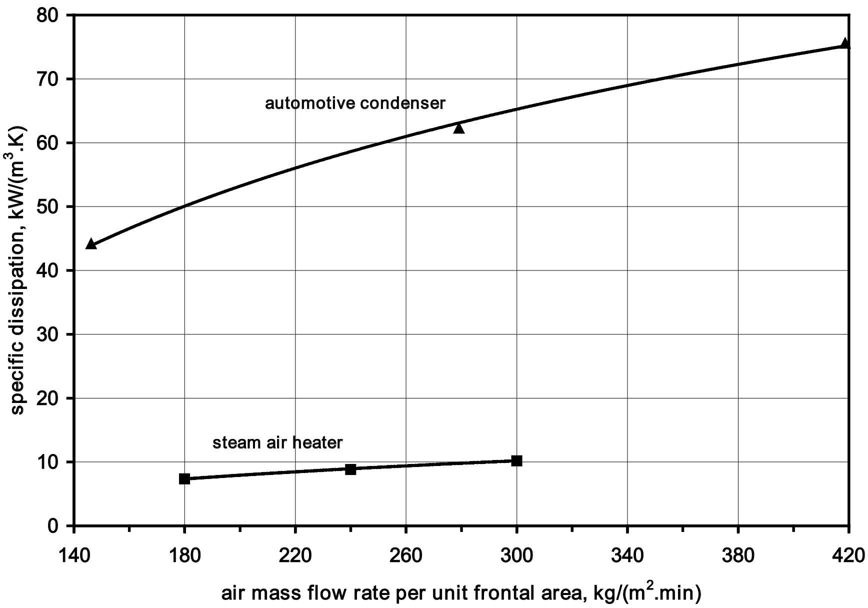


Fig. 1. Experimentally determined specific dissipation for a compact automotive condenser and steam air heaters

condensers as they are located in the vehicle engine compartment right behind the grill in front of the radiator and low weight significance because condenser weight contributes into total vehicle weight. To get a compact design the condenser high performance is necessary. Performance characteristics of automotive condensers, as being dependent on vehicle size and operation conditions, are the best presented if expressed by a specific figure of merit. Applicable to the case is the specific dissipation defined by heat rate dissipated per one degree of inlet temperature difference between the hot and cold fluid per volume of the exchanger core.

In Fig. 1 the specific dissipation as a function of air mass flow rate inflowing per unit frontal area of an aluminium condenser with minichannels for refrigerant flow is shown. For comparison purposes the corresponding data for an air heater by condensing steam assembled noncompact of one row finned steel round tubes arrangement are also presented. One can see in Fig. 1 that the specific dissipation for the condenser due to its compact design is on average six times higher than that for the heater.

The primary objective of this work is to present overall methodology to be purposed for thermal analysis of automotive condensers operating under mixed regimes in minichannels on the two-phase flow side of a vapour compression cycle with refrigerant R-134a as the working fluid.

1. Compact surfaces for automotive condensers

Because of the fact that compactness is the main feature of the condensers under study in this section we present an overview of compact heat transfer surfaces applied for automotive condensers. The condenser coil is composed of three main components: tubes, fins and side manifolds.

1.1. Air side

For automotive applications the air side thermal resistance contributes the most in overall resistance along the path of heat flow between the fluids. Hence particular attention is paid to improve performance of automotive exchangers by reduction the air side resistance. First of all extended surfaces of high compactness are employed. These surfaces by increasing the area for transfer can reduce the resistance by a factor of 3 ÷ 5 with respect to the resistance under air flow across bundles of small in diameter tubes. The other way leading for reduction the resistance is improvement of heat transfer coefficient on extended surfaces under air flow. Here shaped as well as interrupting surfaces are widely applied. These features of automotive exchangers are demonstrated in Figs. 2 and 3.

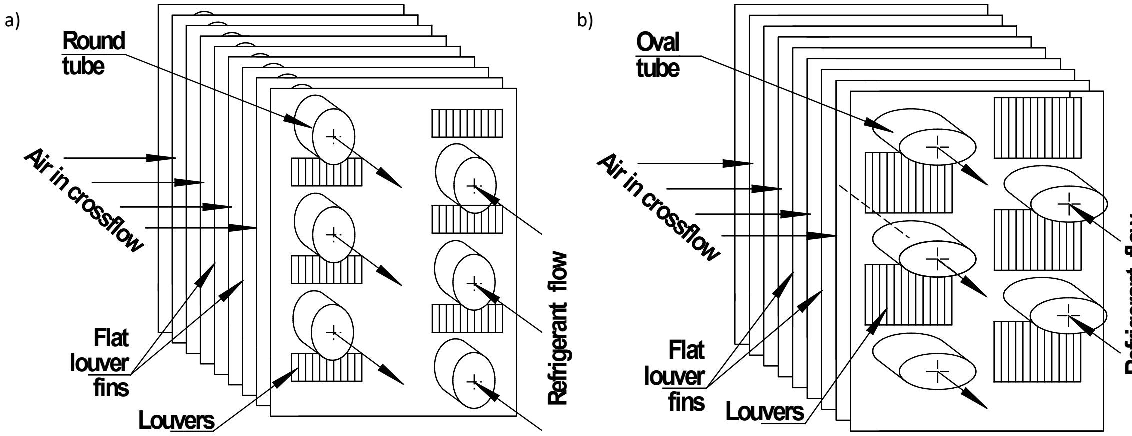


Fig. 2. Extended plain fin surfaces of interrupted geometry with multilouver fins: a) fins on round tubes; b) fins on oval (elliptic) tubes

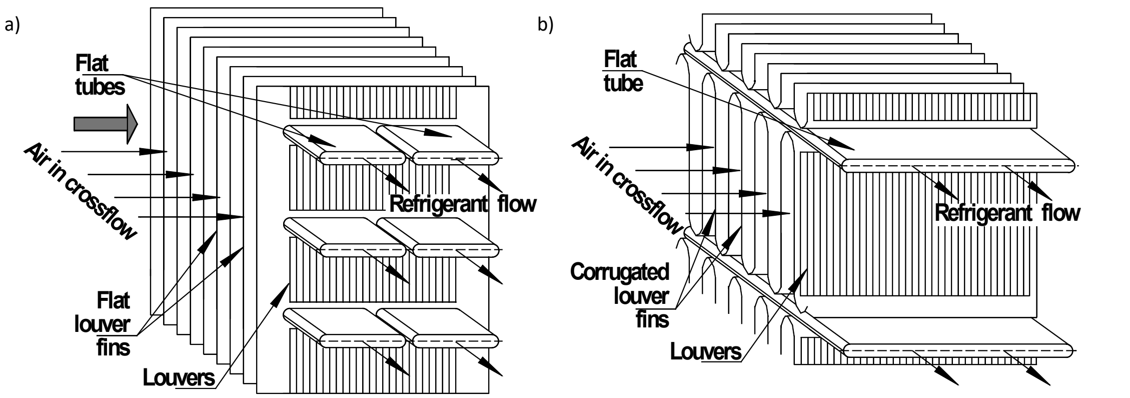


Fig. 3. Extended plain fin surfaces of interrupted geometry with multilouver fins: a) fins on two row flat tubes, b) fins on one row flat tubes

An ultimate design solutions are with the flat tubes with arrangement in one or a few rows, see in Fig. 3.

The designs with flat tubes, as shown in Fig. 3, can provide essential advantages such as:

- small wake zone that affects insignificantly the flow across louvers behind the tube,
- pretty high the average heat transfer coefficient for the overall finned surface because almost entire fin is louvered,
- much higher fin efficiency if fins are attached to the flat tube when compared to the round tube,
- lower aerodynamic drag because the exchanger can be designed with smaller frontal area,
- highest compactness of the exchanger core because of louvered fins are dense corrugated centers sandwiched in between the tubes and manifolds,
- easy way to make multi-port minichannels by extrusion.

Hence, the air cooled condensers with multilouver fins attached to the multiport flat tubes are very popular for automotive applications.

1.2. Refrigerant side

On the tube side there are three basic designs of plate fin-and-tube air cooled heat exchangers, namely: round tubes, oval tubes and flat tubes. The round tubes are in general of OD ~ 6 ÷ 16 mm by the tube wall thickness 0.3 ÷ 0.6mm. The inner tube surface can be smooth or covered with microfins in height of 0.1 ÷ 0.3mm at helical angle of $13^{o} \div 25^{o}$ to enhance the heat transfer coefficient during in tube condensation, see in Fig. 4.

The resultant enhance effect of the microfins is the heat transfer coefficient during in tube condensation of refrigerants increases by a factor 2 ÷ 3 over values of the coefficient for smooth tubes [1]. Availa-

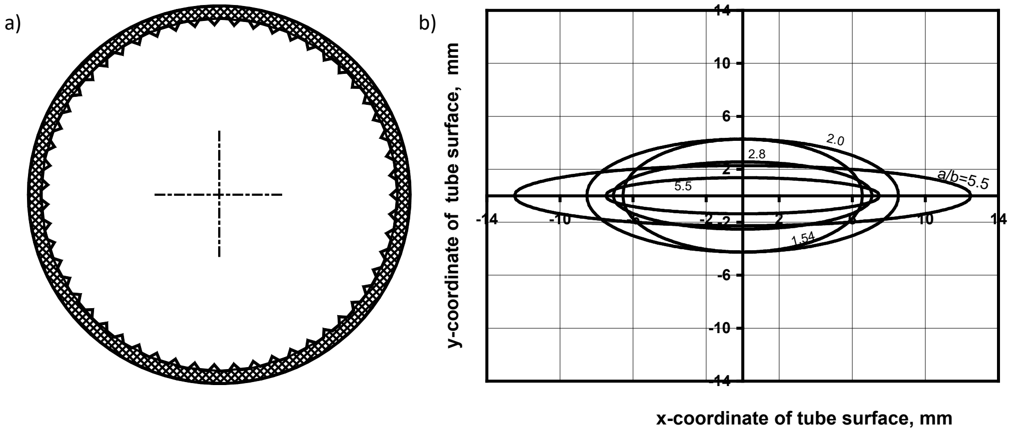


Fig. 4. Exemplary profiles of round and oval tubes applied with plain and louver fins: a) crossection of a round tube with internal microfins (magnification ~3x); b) oval tube profiles (a/b = longer semiaxis/shorter semiaxis)

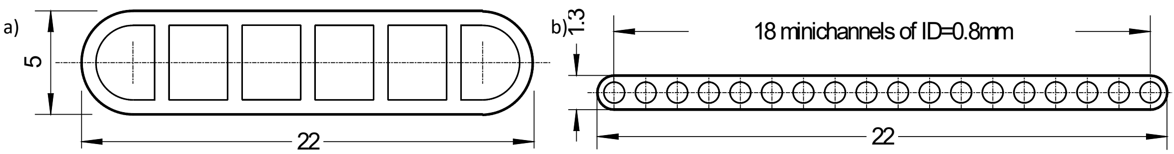


Fig. 5. The multiport extruded aluminium tubes: a) tube with square channels of medium size; b) tube with round minichannels

ble on the market are also multiport extruded aluminium tubes with the internal surface of mini channels smooth or enhanced by microfining. Such the tubes take advantage of low aluminium weight what results in profit from lower vehicle weight. Some exemplary profiles and mean dimensions are shown in Fig. 5.

Due to small inner diameter of the mini channels the inside flow occurs at low Reynolds numbers that values corespond to laminar flow regime. In turn, in tube heat transfer coefficient is higher for mini channels than for round tubes of medium size. This fact combined with dense packing of the core with aluminium flat tubes on air side permit to get much smaller units of equivalent condenser performance.

2. Previous works

Based on numerous references an overview of modelling for design of steam and refrigerant condensers is presented by Kakaç (1991) where considerations for air conditioning condensers are contributed by Pate (1991). Review of open literature shows that the first attempts on condensers design methodology were directed to steam turbine condensers (Šklover & Mil'man, 1985). More recently, the modelling for performance analysis of an automotive condenser is presented by Lee and Yoo (2000) where earlier attempts are also summarized. Their analysis is based on $\varepsilon - NTU_o$ methodology with each fin considered as a single transfer unit. Concerning the refrigerant flow (Lee & Yoo, 2000) assumed a simplified model of the condenser with the only annular regime on the two phase side. The condenser modelling which takes into account the annular and stratified flow regimes is presented by Bansal and Chin (2002). They determined condensation heat transfer coefficient for annular regime on correlation by Cavallini and Zecchin (1971) and by the Nusselt model (Kakaç, 1991) for stratified flow. In turn, by consideration only two flow regimes (Bansal & Chin, 2002) analysis is also limited because transition regime (wavy) that can occupy significant part (roughly about up to 20% of tube length) of condensation path is not included. For an automotive condenser with minichannels on the refrigerant side and louvered fins on the air side Jabardo and Mamani (2003) report the results of condenser modelling accompanied with experimental data related to. The model has been developed based on dividing the three pass condenser coil into three regions: the superheated vapor, the change of phase, and the subcooled liquid regions. Regarding the heat transfer coefficient on the refrigerant side that for single phase regions are determined in Jabardo and Mamani (2003) using well known Dittus and Boelter correlation and a correlation by Mamani (2001). His correlation is of the same form throughout all two phase region regardless two phase flow regimes occurring as the condensation progresses. In turn the analysis of two phase side proposed in Jabardo and Mamani (2003) is an approximation because one correlation cannot grasp different mechanisms of in tube condensation under different flow regimes. With respect to the air side Jabardo, Saiz and Mamani (Jabardo & Mamani, 2003) determine the heat transfer coefficient on correlation by Chang and Wang (1997).

With respect to thermal analysis of brazed aluminium condensers that use flat tubes with mini channels of small diameter for refrigerant flow under combined regimes annular/intermittent/bubble or annular/wavy/stratified or annular/wavy/intermittent/bubble no papers have been found to be published in open literature.

In the paper, at first an overview of heat transfer and pressure drop modelling applied for automotive condensers operating at mixed regimes on the two phase flow side is summarized. Then detailed solution procedure is outlined for the rating problem an automotive condenser using a specific set of input data and the results obtained are then presented.

3. Heat transfer modelling applied

We seek for heat transfer performance and pressure drop results for the condenser. To carry out the analysis a lumped approach is used with decomposition of the condenser coil along refrigerant flow into a number of zones established with respect to single phase and two phase flow regimes. In turn, the refrigerant flows (assuming at constant pressure) through either zone with its total mass flow rate, however, either zone is cooled in the crossflow with air stream of fractional mass flow rate that corresponds to total air mass flow rate via ratio the zone frontal area to total condenser frontal area. Since the refrigerant side heat transfer coefficients and pressure gradients can vary significantly along the flow the mean values within boundaries of each zone were calculated by integration.

3.1. The ε – NTU$_o$ relations

The case of crossflow with both fluids unmixed is applied for single pass zones of no phase change so that the formula for effectiveness for the zones in point is given by (Incropera & DeWitt, 1996)

$$\varepsilon = 1 - exp\{NTU_o^{0.22} \cdot [exp(-C^* \cdot NTU_o^{0.78}) - 1]/C^*\} \quad (1)$$

Although, full mixing is assumed at either cros-section of individual minichannel, the assumption that the tubeside fluid is unmixed can be justified because usually number of parallel minichanels (rows) is significant. The shell side fluid is also considered unmixed. This assumption is commonly recommended to use for the case as well as also applied in data reduction procedures for heat transfer coefficients (Wang et al., 2000).

Regarding zones of two phase flow, note that $C^* = 0$ holds. Then the effectiveness is given by

$$\varepsilon = 1 - exp(-NTU_o) \quad (2)$$

In order to develop relationship describing parameter NTU$_o$ of Eqs. (1) and (2) the total thermal resistance R_t in the path of heat flow between the fluids is required. The resistance Rt for the case is the sum of component resistances because the corresponding thermal circuit is along with resistances in series, hence

$$R_t = R_o + R_w + R_i \quad (3)$$

In turn, all the components of Rt envisioned in Eq. (3) need to be determined for either established zone of the condenser.

3.2. Component thermal resistances on the path of heat flow

The material of this section deals with the determination of individual resistances of Eq. (3) in which the main contributing considered in the paper are: convective resistances on the air and refrigerant sides and conductive resistance across the wall separating the fluids.

Thermal resistance on the air side. The resistance for a condenser zone of air side transfer area A_o extended by fins is determined on the following expression.

$$R_o = 1/(\eta_o \cdot h_o \cdot A_o) \quad (4)$$

Efficiency η_o of Eq. (4) is related to transfer areas A_o and A_f as well as fin efficiency η_f by

$$\eta_o = [1 - (A_f / A_o) \cdot (1 - \eta_f)] \quad (5)$$

where η_f is dependent on coefficient h_0 via the fin parameter. Hence, the most essential problem in determining R_0 is evaluation of heat transfer coefficient h_0 for extended surface with corrugated fins of louvered type brazed in between the flat tubes, as shown in Fig. 6. For the case with corrugated fins Chang and Wang (1997) proposed the following correlation describing the Colburn factor

$$j = Re_{L_p}^{-0.49} \cdot \left(\frac{\theta}{90}\right)^{0.27} \cdot \left(\frac{F_p}{L_p}\right)^{-0.14} \cdot \left(\frac{F_\ell}{L_p}\right)^{-0.29} \cdot \cdot \left(\frac{T_d}{L_p}\right)^{-0.23} \cdot \left(\frac{L_\ell}{L_p}\right)^{0.68} \cdot \left(\frac{T_p}{L_p}\right)^{-0.28} \cdot \left(\frac{\delta_f}{L_p}\right)^{-0.05} \tag{6}$$

An advantage of Eq. (6) is that it combines the Reynolds number and all the geometry parameters essential for fin design into an accurate formula developed based on fitting significant amount of experimental data for the louver fins.

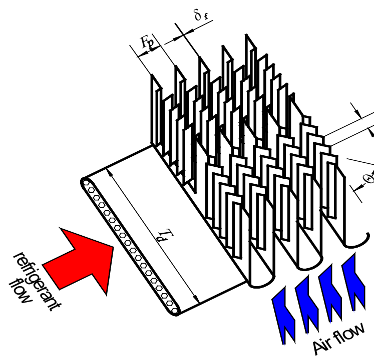


Fig. 6. Corrugated louver fins onto flat tube in a crossflow and associate dimensions

Correlation (6) determines the mean value of h_0 (note, η_o is not included into h_0) throughout all the extended surface (prime surface + fin surface) and is valid for air flow at Reynolds numbers bounded in between 50 ÷ 5000 for number of tube rows N = 1 and 2. It correlates experimental data on j factor at mean deviation ±7.55%.

Wall conduction resistance. The thermal resistance of the flat tube wall is determined by the conduction shape factor. To use available data defining the factors for transverse heat conduction a repeatable element of the tube crossection with a single circular minichannel is selected as shown in Fig. 7. The upper and bottom surfaces of the element as well as minichannel wall surface are diatermic but the left and right surfaces located in the middle of the wall between two adjacent minichannels can be considered as adiabatic (due to symmetricity).

For conducting system shown in Fig. 7 the conduction shape factor SL expressed per unit length of the element is given by (Zweidimensionale Wärmeleitung, 1984)

$$S_L = \frac{2 \cdot \pi}{\pi \cdot (D_m/2)/s + ln[(s/2)/(\pi \cdot D_i/2)]} \tag{7}$$

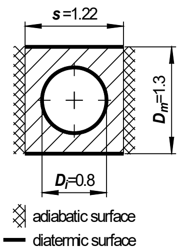


Fig. 7. Selected element of the flat tube with a single circular minichannel

Hence, for a zone of length L having N parallel flat tubes either with n mini channels the conduction resitance R_w can be expressed as

$$1/R_w = \lambda_w \cdot S_L \cdot L \cdot N \cdot n \tag{8}$$

Usually R_w values are the only a small fraction of R_t, order of ~1% or less. Consequently, idealization as shown in Fig. 7 affects the effectiveness insignificantly.

Thermal resistance on the refrigerant side. The scope of this paper is limited to mini channels of smooth inner surface. Because no fins are present onto this surface the thermal resistance either under single or two phase flow is given by

$$R_i = 1/(h_i \cdot A_i) \tag{9}$$

The heat transfer coefficient h_i in the foregoing expression is a complex function of flow and heat transfer conditions. For duct flows, as it is the case, there is a variety of available correlations that define coefficient h_i for single phase as well as two phase flows. The correlations proposed are all of recognized accuracy and have been selected as the most suitable for thermal analysis of automotive condensers.

Zones of single-phase flow. For zones of single-phase flow the correlation by (Gnielinski, 1976) is used to determine h_i so that the corresponding Nusselt number is given by

$$Nu_i = \frac{\left(Re_{D_i} - 1000\right) \cdot Pr_i \cdot (f_i/2)}{1 + 12.7 \cdot \sqrt{f_i/2} \cdot \left(Pr_i^{2/3} - 1\right)} \tag{10}$$

Correlation (10) is valid for fully developed internal flows in circular and noncircular (in this case use instead of) ducts under Reynolds numbers 3000 ÷ $5 \cdot 10^6$ and correlates experimental data at uncertainty of ±10% if Prandtl numbers are within $0.5 < Pr_i < 2000$, however if $2 < Pr_i < 140$ inaccuracy is lower to be ±6%.

Zones of two phase flow. Correlations for prediction coefficient hi for zones with the two phase flow refer to particular flow structure regimes that develop along the tube when the condensation progresses.

Hence, before setting up the correlations let us remind descriptively basic flow regimes for in horizontal tube condensation.

For condensers considered in the paper the refrigerant flows in minichannels of ID ~ 1 mm or a bit less so the surface tension forces can essentially affect the flow and in turn the flow regimes development. As influence of gravity decreases with decreasing tube diameter and at the same time the surface tension effect is progressively increasing one can expect in minichannels such flow regimes as annular, slug, plug and bubble (or transient regimes existing in between specified). These regimes are of mutual action of shear and surface tension forces. There are some experimental evidences confirming this conclusion. For example (Niño et al., 2003) recently reported that for two phase adiabatic flow in multiport minichannel test facility of 1.58 mm in hydraulic diameter of each port the flow regimes they observed are the following: annular, intermittent (slug and plug) and bubble. Hence, involving surface tension influence into recognition of flow regimes on the refrigerant side for the condenser under study is of primary importance. However, prevailing maps can be of used for predict flow regimes in tubes of larger diameters. Consequently, thermal analysis of condensers with minichannels should be based on new maps developed as specific for two phase flow regime evaluations in minichannels.

Steam traps are usually installed at the condenser exit or inlet to the subcooling pass permits only the liquid phase to flow through. In turn, the bubbly flow (or stratified regime at some flow conditions) ends the condensation with all the tube section filled with liquid phase. Concluding, variety of complex flow regimes precludes establishing a single correlation that can describe the heat transfer coeffcient hi along the complete condensation path.

Selecting appropriate correlations for heat transfer on the two phase flow side for purposes of the condenser design is influenced also by significance of thermal resistance on the two phase flow side R_i for total thermal resistance R_t of Eq. (3) when compared to significance of thermal resistance R_o on the coolant side, (Srinivasan & Shah, 1997). Preliminary evaluations done based on the condenser under study showed that for the zone with annular flow regime the resistances are: $R_i \approx 0.0004$ K/W, $R_o \approx 0.002$ K/W and $R_w \approx 0.000005$ K/W. Hence, one can see that the resistance on the air side does control accuracy of the condenser modelling.

Annular flow. This regime is generally regarded to be the dominant pattern existing over the most part of the condensing length. In turn, for the annular flow regime the correlation proposed by (Shah, 1979) of well recognized accuracy of ±15.4% has been used in a form as follows

$$h_{i,an}(x) = h_{\ell o} \cdot \left[(1-x)^{0.8} + \frac{3.8 \cdot x^{0.76} \cdot (1-x)^{0.04}}{p_r^{0.38}} \right] \quad (11)$$

where coefficient $h_{i,an}(x)$ refers to the inner tube surface at a crossection where the vapor mass fraction is x. Heat transfer coefficient $h_{\ell o}$ in expression (11) attributes to entire two phase flow (of liquid + vapor) as the liquid flow at saturated properties. Accordingly development of (Shah, 1979) coefficient $h_{\ell o}$ is calculated using a modified Dittus and Boelter equation (Winterton et al., 1998), hence

$$h_{\ell o} = 0.023 \cdot Re_{\ell o}^{0.8} \cdot Pr_{\ell}^{0.4} \cdot \lambda_\ell / D_i \quad (12)$$

Validity of correlation (11) refers to annular two phase flow at mass velocity of entire flow $G_i = m_i/A_i > 200$ kg/(m^{2}s). Accordingly Eq. (11) a local value of coefficient $h_{i,an}(x)$ depends essentially on the vapor mass fraction. Hence, mean value of coefficient $h_{i,an}(x)$ along a channel of length L is determined by the integration

$$h_{i,an} = \frac{1}{L} \cdot \int_0^L h_{i,an}(x) \cdot dz \quad (13)$$

where x = x(z). Relation x(z) can be found only by analysis of a distributed condenser model in which the condenser is divided into a number of small subexchangers along the refrigerant flow. However, if component thermal resistance on air side is much bigger than the other resistances of Eq. (3) then one can obtain reasonably accurate results of the condenser analysis assuming that x = x(z) is linear. Under this assumption, mean value of coefficient $h_{i,an}(x)$ is given by

$$h_{i,an} = \frac{1}{x_{o,an} - x_{i,an}} \cdot \int_{x_{i,an}}^{x_{o,an}} h_{i,an}(x) \cdot dx \quad (14)$$

where $x_{o,an} - x_{i,an}$ is the drop in vapor mass fraction under annular flow regime.

Wavy flow. Foregoing correlations refer to either pure annular or pure stratified flows. However there is a transition specified as the wavy flow in between those pure regimes. To find the heat transfer coefficient for such the transition under R134a flow Skiepko (2004) proposed to follow Jaster and Kosky (1976) proration developed for condensing steam flow. However, comparisons of experimental data and model evaluations showed that such the proration can result in increased inaccuracy of condenser performance data obtained from the modelling particularly at lower mass velocity of the entire two phase flow. Hence for this work we use a correlation for wavy regime proposed by Chato and Dobson (1998). The heat transfer coefficient (averaged over entire tube inner perimeter) for film condensation in the upper part of the horizontal tube is expressed (Chato & Dobson, 1998).

$$h_U\left(x\right)=\frac{0.23\cdot Re_{\upsilon o}^{0.12}}{1+1.11\cdot X_{tt}^{0.58}}\cdot\left(\frac{Ga\cdot Pr_\ell}{Ja_\ell}\right)^{0.25}\cdot\frac{\lambda_\ell}{D_i} \qquad (15)$$

where the turbulent-turbulent Martinelli parameter X_{tt} is given by

$$X_{tt}=\left[(1-x)/x\right]^{0.9}\cdot\left(\rho_\upsilon/\rho_\ell\right)^{0.5}\cdot\left(\mu_\ell/\mu_\upsilon\right)^{0.1} \qquad (16)$$

Intermittent and bubbly flow. Modeling of in tube condensation heat transfer lacks of relevant experimental data to be used for analysis if flow regime develops to be intermittent or bubbly. Hence, at present the corresponding contributions can be evaluated in approximate manners only. If such regimes occur at lower vapor qualities they contribute little in condenser overall heat transfer performance and due to this fact the approximations cannot significantly deteriorate in the final outcomes. If the intermittent regime develops at higher qualities the resistance on the refrigerant side is essentially lower than on the air side and hence approximations in determining heat transfer coefficient affect little the transfer. In turn, Cavallini et al. (2003) proposes to prorate linearly the heat transfer coefficient at the boundary down with respect to vapor quality to that for entire liquid flow, if after annular regime the slug flow develops, hence one gets

$$h_{i,int}=h_{i,\ell o}+\frac{x}{x_{an,o}}\cdot\left(h_{i,an}\Big|_{x=x_{an,o}}-h_{i,\ell o}\right) \qquad (17)$$

Flow regime prediction. As aforementioned review shows, different types of correlations are necessary for prediction heat transfer coefficients in different two phase flow regimes. Hence, flow regime prediction is especially important for condenser heat-flow analysis. A simple way for regime predictions under condensation of steam during in tube flow is proposed by Jaster and Kosky (1976). Butterworth (1977) demonstrated consistency of their approach for condensation of refrigerant R-12. The main fault of method by Jaster and Kosky (1976) is its limited applicability only for a condensation path along annular – transition (wavy) – stratified regimes. More general method of flow regime prediction is by Taitel and Dukler (1976) map. Although this map refers to adiabatic two phase flows, Breber et al. (1980) showed that the parameters of the map Taitel and Dukler (1976) are indeed significant for condensation under horizontal flow. However, neither parameter of the original map Taitel and Dukler (1976) does take into account surface tension forces. Hence, application of the maps (Jaster & Kosky, 1976; Taitel & Dukler, 1976; Breber et al., 1980) for prediction two phase flow regimes in minichannles of small diameter is greatly limited unless some modifications are introduced. In this respect Dobson and Chato (1998) suggest to use of Taitel and Dukler (1976) map but with surface tension involved into prediction the flow regimes in a manner by Galbiatti and Andreini (1992). Their final result for the transition refers to annular – stratified regimes. Hence, involving of the surface tension as proposed by Galbiatti and Andreini (1992) is irrelevant for flow regime predictions in minichannels because the stratified in minichannel flows may marginal to occur, if any, what is a fact confirmed experimentally (Niño et al., 2003).

In turn, for the present work we use a new of two-phase flow map developed by Tabatabai and Faghri (2001). Their map accounts for the surface tension effects under two-phase flow in horizontal minichannelss. Basically, map Tabatabai and Faghri (2001) is split into two regions, namely: shear dominated and surface tension dominated. The transition boundary between shear dominated regimes (annular, mist, stratified, wavy) and surface tension dominated regimes (slug, plug, bubbly) has been determined on force balance resulted from mutual action of shear, bouyancy (due to gravity) and surface tension forces. Specifically, the transition from annular to slug/plug or bubble regime is developed based on instability of liquid collars in annular-wavy regime leading to formation of originating liquid slug. Hence, following Tabatabai and Faghri (2001) stabilizing influence of surface tension that augments stabilizing influence of gravity can be taken into evaluations of the flow regimes in ducts of small inner diameter.

3.3. Pressure drop evaluation

The pressure drop on the air side. The pressure drop in point is due to the following contributions: the friction effect, change in the momentum effect, and entrance and exit losses. To calculate the friction contribution a correlation for the Fanning friction factor developed by Chang et al. (2000) is proposed to employ. The main motivation for correlation is it involves not only various flow friction effects by terms dependent on the Reynolds number but also influence of the finned surface form drag (independent on Reynolds number) is accounted in by factors dependent on exhaustive design set of dimensionless geometry parameters of the louvered surface. Concluding, the friction factor for the case is given by

$$f = f_1 \cdot f_2 \cdot f_3 \tag{18}$$

where factor f_1 is described as

$$\text{if } Re_{L_p} < 150, \tag{19}$$

$$f_1 = 14.39 \cdot Re_{L_p}^{-0.805 \cdot F_p / F_\ell} \cdot \left\{ ln\left[1 + \left(F_p / L_p\right)\right]\right\}^{3.04}$$

$$\text{and for } 150 < Re_{L_p} < 5000, \tag{20}$$

$$f_1 = 4.97 \cdot Re_{L_p}^{0.6049 - 1.064/\theta^{0.2}} \cdot \left\{ ln\left[\left(\delta_f / F_p\right)^{0.5} + 0.9\right]\right\}^{-0.527}$$

Regarding factor f_2, the following expression is developed

$$\text{if } Re_{L_p} < 150, \tag{21}$$

$$f_2 = \left\{ ln\left[\left(\delta_f / F_p\right)^{0.48} + 0.9\right]\right\}^{-1.435} \left(\frac{D_h}{L_p}\right)^{-3.01} \cdot \left[ln\left(0.5 \cdot Re_{L_p}\right)\right]^{-3.01}$$

$$\text{if } 150 < Re_{L_p} < 5000, \tag{22}$$

$$f_2 = \left[\frac{D_h}{L_p} \cdot ln\left(0.3 \cdot Re_{L_p}\right)\right]^{-2.966} \cdot \left(\frac{F_p}{L_\ell}\right)^{-0.7931 \cdot T_p / \left(T_p - D_m\right)}$$

Factor f_3, as below, completes necessary empirical information for air side pressure drop modelling, and hence

$$\text{if } Re_{L_p} < 150, \tag{23}$$

$$f_3 = \left(\frac{F_p}{L_\ell}\right)^{-0.308} \cdot \left(\frac{L}{L_\ell}\right)^{-0.308} \cdot \left(e^{-0.1167 \cdot T_p / D_m}\right) \cdot \theta^{0.35}$$

$$\text{if } 150 < Re_{L_p} < 5000, \tag{24}$$

$$f_3 = \left(\frac{T_p}{D_m}\right)^{-0.0446} \cdot \left\{ ln\left[1.2 + \left(L_p / F_p\right)^{1.4}\right]\right\}^{-3.553} \cdot \theta^{-0.477}$$

Note also that correlation (18) does not include coefficients for the entrance and exit pressure losses. Therefore, these should be determined based on other sources, for the case using data charts presented by Kays and London (1984). In turn, based on the above information, the conventional pressure drop relation for compact heat exchangers described by Kays and London (1984) is applied for determining the pressure drop on the air side and due to this fact is not quoted here.

The pressure drop on the refrigerant side. Because the superheated vapor enters and subcooled liquid exits the condenser the overall pressure drop consists of the drop under single phase flow and drop under two phase flow. Note that the overall flow is horizontal throughout all the condenser tubing. In turn, either pressure drop, regardless single or two phase, is the sum of friction loss contribution, change in the momentum contribution and local pressure losses, if any. The pressure drop for the single phase duct flows includes friction and change in the momentum contributions and can be estimated using a variety of formulas presented elsewhere, e.g. in Gersten and Ducts (1998), hence not quoted here.

Under two phase flow for in tube condensation the pressure gradient along the flow varies significantly due to decreasing vapor quality. Hence, if one assumes that x depends linearly on coordinate z along duct length, then pressure change between outlet and inlet $\Delta p = p_o - p_i$ in a condenser duct of length L under two phase flow is given by

$$\Delta p = L \cdot \underbrace{\frac{1}{x_o - x_i} \int_{x_i}^{x_o} \frac{dp}{dz} \cdot dx}_{mean\ value\ of\ dp/dz} \tag{25}$$

Pressure gradient dp/dz of Eq. (25) is due to friction and change in momentum effects. Thus the gradient in point is

$$-\left(dp/dz\right) = \frac{2 \cdot f_{\ell o} \cdot G^2}{\rho_\ell \cdot D_i} \cdot \Phi_{\ell o}^2 + G^2 \cdot \frac{x_o - x_i}{L} \cdot \frac{d}{dx}\left[\frac{x^2}{\alpha \cdot \rho_\upsilon} + \frac{(1-x)^2}{(1-\alpha) \cdot \rho_\ell}\right] \tag{26}$$

The two phase flow pressure drop multiplier Φ_{lo}^2 is determined on an empirical correlation by Friedel (1979), due to space limitation not quoted here. The void fractions required in formula (26) have been calculated with constant C_o given by Thome (2003) as

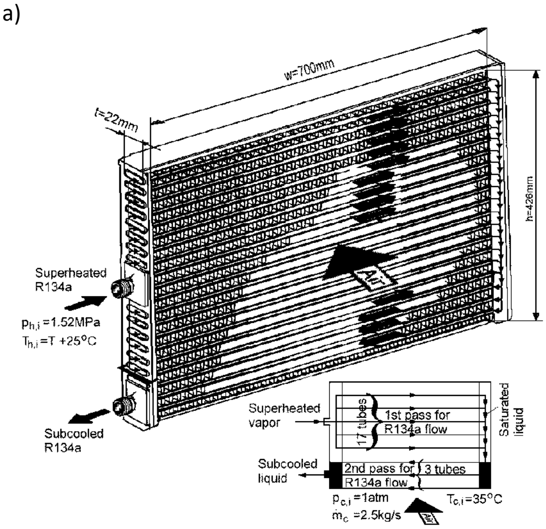
$$C_o = 1 + 0.12 \cdot \left(1 - x\right) \tag{27}$$

An advantage of Friedel's correlation is that it involves the influence of the ratio of inertia to surface tension forces on pressure drop multiplier Φ_{lo}^2. Hence, for in tube two phase flow, evaluation of the frictional contribution into the presuure gradient by Friedel correlation is regarded Cavallini et al. (2002) as one of the best prediction of available experimental data. Eq. (27) completes necessary empirical information applied in evaluation of pressure drop on the refrigerant side of the condenser.

4. Condenser under study

A condenser of R134a considered for this work is cooled in crossflow with air as shown in Fig. 8. The air inlet temperature, pressure, humidity and mass flow rate are: 35°C, 1.01325bar, 60% and 2.5kg/s, respectively. The refrigerant (R134a) inlet temperature is 25°C above its saturation temperature at pressure of 1.52MPa. On R134a side the condenser is of two pass horizontal flow arrangement where the second pass is predicted for subcooling. The air flows across corrugated louver fins brazed onto one row array of flat aluminium tubes having internal circular minichannels where R134a flows. As shown in Fig. 8 the refrigerant enters the header, flows through minichannels along the first pass and condenses. Then the opposite header turns the liquid flow into the second pass in back direction along the subcooling zone towards the exit.

a)



b)

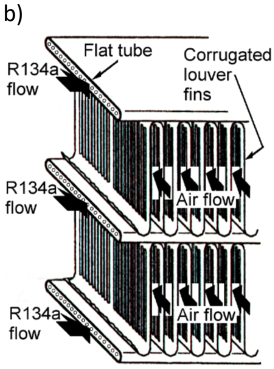


Fig. 8. Basic dimensions and flow arrangement of the condenser under study:
a) view of the condenser assembling, and flow arrangement;
b) view of brazed fins onto flat tubes of the condenser

On input, condenser core geometry configuration data are given in Table 1. For needs of the present study thermodynamic properties of R134a are quoted from Lemmon et al. (2005) and air properties from Kakaç (1991).

Tab. 1. Condenser core geometry configuration data

Configuration quantity	Data value
Height, h, [mm]	426
Width, w, [mm]	700
Depth, t, [mm]	22
Number of passes	2
Number of flat tubes in condensation pass, N_{cd}	17
Number of flat tubes in subcooling pass, N_{sc}	3
Number of flat tubes in air flow direction, N	1
Tube material thermal conductivity, λ_w, [W/m.K]	204
Flat tube	
Major flat tube dimension, T_d, [mm]	22
Minor flat tube dimension, D_m, [mm]	1.3
Flat tube pitch, T_p, [mm]	21.3
Minichanel diameter, d_o, [mm]	0.8
Number of minichannels in one flat tube, n_r,	18
Fins	
Fin pitch, F_p, [mm]	2.0
Fin thickness, δ_f, [mm]	0.075
Fin length between tubes, F_ℓ , [mm]	20
Louver length, L_ℓ , [mm]	18.0
Louver angle, ϑ, [°]	28.0
Louver pitch, L_p, [mm]	1.5
Fin length in air flow direction, L, [mm]	22.0
Free flow area to frontal area ratio	0.902
Heat transfer area/total volume, Y, [m²/m³]	1089.6
Hydraulic diameter on air side, D_h, [mm]	3.31
Fin transfer to total transfer area ratio	0.911

4.1. Flow regimes evaluation

Because correlations for prediction coefficient h_i as well as for multiplier $\Phi_{\ell 0}^2$ refer to particular flow structure regimes the primary step in condenser heat-flow analysis is evaluation of the regimes that can develop along the tube when condensation progresses. For needs of this presentation the flow regimes have been identified with the use of the map (Tabatabai & Faghri, 2001) developed for two phase flows in minichannels because this map involves the surface tension into evaluation of the flow regimes. In this map the transition boundary between the annular and intermittent (slug) regimes occurs if the liquid volume flow rate fraction determined based on superficial velocities j_ℓ and j_υ of the liquid and gas phase, respectively, is

$$\dot{\varepsilon}_\ell = \frac{j_\ell}{j_\ell + j_\upsilon} = 0.06 \qquad (28)$$

For comparison purposes let us provide some data on terminal vapor quality for the annular regime at the transition evaluated with the use of criterion Jaster and Kosky (1976), maps Taitel and Dukler (1976) and Tabatabai and Faghri (2001). The results of this evaluation are shown in Table 2. One can see in Table 2 that the terminal point for the annular regime is essentially affected by inclusion the surface tension into considerations. Hence, accordingly Tabatabai and Faghri (2001) the condensation flow under annular regime terminates at x = 0.531 when the surface tension is included when compared to x = 0.151 if one neglects the surface tension. This result does mean that the annular regime terminates at about four times higher vapor quality when the condensation is affected by the surface tension forces.

As the condensation continues down under the slug and then plug regimes, the vapor quality x decreases what results in decreasing the ratio of vapor superficial velocity j_υ to liquid superficial velocity j_ℓ. At the same time the void fraction α is also decreasing what, in accordance with Tabatabai and Faghri (2001), makes the increase in pressure gradient due to surface tension. However, due to decrease in vapor quality the shear pressure gradient also decreases. In turn, the condensation path on map Tabatabai and Faghri (2001) continues down right into the region of slug and plug regimes because ratio j_υ/j_ℓ decreases below 15.7 (value resulted from m = = 0.06) and ratio $(dp/dz)_{surface\ tension}/(dp/dz)_{shear}$ increases. This conclusion indicates that nor wavy neither stratified regimes can occur for the case considered at entire flow mass velocity G_h = 421.3 kg/(m²s). Hence, the only possible post intermittent regime is the bubble flow. To determine the transition between the intermittent and bubble regimes use on their map the same boundary as proposed by Taitel and Dukler (1976). In turn, intersection of $T_{TD}(X)$ values determined for horizontal flows from

$$T_{TD} = \left\{ \left(\frac{dp}{dz} F \right)_\ell \Big/ \left[(\rho_\ell - \rho_\upsilon) \cdot g \right] \right\}^{0.5} \qquad (29)$$

with boundary D(X) of map Taitel and Dukler (1976) provides value of the Martinelli parameter X at the transition. Then solving formula for X

$$X = \sqrt{\left[\frac{(dp/dx)_\ell}{(dp/dx)_\upsilon} \right]} \qquad (30)$$

with respect to vapor quality provides x_D value at the transition.

Experimental evidences (see e.g. Collier & Thome, 1994) show that the slug regime always preceeds plug flow as x decreases (corresponding X values increase). Hence, at first slug flow is formed for the case as condensation progresses after the annular regime. The liquid slugs coalescence locking large vapor bubbles (plugs) within liquid what forms the plug flow regime. As the plugs of vapor continue along the flow, condensation continues also what reduces the size of plugs. Then the bubble flow is formed when turbulent fluctuations can break the plugs into small bubbles what is the transition of intermittent to bubble regime as described above. For the case it occurs at X_{tt} = 70.9 what corresponds to x = 0.00262 and α = 0.0261. The bubble flow condensation continues with disappearing smaller and smaller bubbles by the end of condensation.

Development of this condensation scenario is depicted in Fig. 9 that illustrates how flow regimes can vary along the flow length (see also data in Table 2) accompanied by decreasing vapor quality. Note in Fig. 9 that as condensation starts at x = 1 the first flow regime is annular, as expected. After that the intermittent flow develops, initially the slug and then plug. The flow regime becomes bubbly as condensation continues down the intermittent regime.

The following observations can be made based on Fig. 9. Hence, when the surface tension is disregarded one can see in Fig. 9 a) the length of the annular flow regime is the biggest in the zone of two phase flow. The intermittent flow spans on the length by four 4 times shorter that the that for the annular

Tab. 2. Vapor quality and liquid volume flow rate fraction at termination point of the annular regime

Steam, surface tension neglected (Jaster & Kosky, 1976)	R134a, surface tension neglected (Taitel & Dukler, 1976)	R134a, surface tension included (Tabatabai & Faghri, 2001)
x = 0.127	x = 0.151	x = 0.531
$\dot{\varepsilon}_\ell$ = 0.332	$\dot{\varepsilon}_\ell$ = 0.289	$\dot{\varepsilon}_\ell$ = 0.06

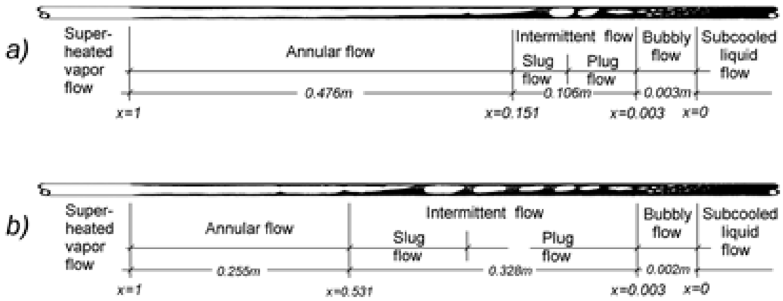


Fig. 9. Two phase flow regimes for the condenser under study determined based on (note: length and x scales are not preserved):
a) map (Taitel & Dukler, 1976) at G = 424.7 kg/(m^2s) with surface tension neglected;
b) map (Tabatabai & Faghri, 2001) at G = 421.3 kg/(m^2s) with surface tension included

and the bubbly flow occurs at negligible length almost directly at the minichannel exit port. On the contrary, it is shown in Fig. 9 b) that the surface tension essentially affects development of the flow regimes under condensation in minichannels. In turn, the biggest length is under the intermittent flow and the annular flow preceding the intermittent takes the length shorter by about 20% when compared to the former one. Insignificant length is of the bubbly flow of the same order if one disregards the surface tension. One can also see in Fig. 9 that regardless the surface tension is included or not the stratified regime is not present. This result is in accordance with aforementioned experimental observations done by Niño et al. (2003).

4.2. Overall calculation scheme

The input data for the calculations are: core geometry data and design conditions such as the air inlet temperature, pressure, humidity, air mass flow rate and the refrigerant inlet temperature and pressure. To manage the task a computational strategy has been developed which performs the analysis taking into considerations zone after zone along the refrigerant in tube flow. Calculations are iterative because the refrigerant mass flow rate should be initially guessed. Because of complexity of the expressions involved into the modelling presented an extensive spreadsheet Excel has been prepared to be used in calculations after passing thorough checking for its convergence and accuracy.

4.3. Results of the analysis

The results of the analysis are presented in tabulated and graphical forms. Table 3 shows the most essential outcomes. All the computed values presented in Table 3 have been obtained with inclusion of the surface tension into the analysis.

One can see in Table 3 the significance of transfer processes in particular zones for the overall condenser performance and pressure drop on the refrigerant side. In this respect the intermittent flow regime zone contributes the most providing ~39% of the overall condenser heat transfer. The corresponding figure for annular regime zone is lower by ~4% to be about ~35%. The zone of the bubble regime contributes in overall heat rate transfer by the only a small fraction ~0.2%. Hence, the bubble flow may not be distinguished for the case and the entire two phase region can be regarded as composed of the annular and intermittent zones only. A interesting comment arises if one compares the above figures evaluated at inclusion the surface tension with those presented in Breber et al. (1980) where the influence of surface tension is disregarded. For this case the corresponding figures are: ~63% (zone of the annular regime), ~11% (zone of the intermittent regime), ~0.2% (under the bubble regime). In turn, the surface tension promotes the transfer under intermittent regime and reduces contribution the annular flow zone in the transfer. The bubble regime contribution is insensitive on the surface tension.

Regarding pressure drop, the biggest contribution of ~73% is under single phase flow of subcooled liquid. For two phase flow region the biggest contribution of ~13% in overall pressure drop is under annular flow. All pressure drop in the two phase flow zone is ~0.2bar what means ~21% of that by overall condenser. This drop can decrease the saturation temperature by ~0.6oC what changes the mean temperature difference air – R134a in zone of two phase flow by ~1.6%, hence negligibly. By comparing thermal resistances on refrigerant and air sides given in Table 3, one can see that the accuracy of heat transfer prediction for two phase flow region of the condenser is controlled by accuracy of the air side modelling.

In Fig. 10, referenced to the annular flow regime, the variations of heat transfer coefficient $h_{i,an}(x)$ and

Tab. 3. Elected results of condenser analysis with annular, intermittent and bubble regimes (surface tension included)

Property	Super-heated zone	Annular regime zone	Intermittent regime zone	Bubble regime zone	Sub-cooling zone
$T_{h,i}$, °C	80.8	55.8	55.8	55.8	55.8
$T_{h,o}$, °C	55.8	55.8	55.8	55.8	42.6
$x_{h,i}$	-	1.0	0.531	0.00298	-
$x_{h,o}$	-	0.531	0.00262	0	-
G, kg/(m²s)	421.3	421.3	421.3	421.3	2387.4
$T_{c,i}$, °C	35.0	35.0	35.0	35.0	35.0
$T_{c,o}$, °C	40.5	40.6	39.9	38.1	38.5
C_h, W/K	77.85	∞	∞	∞	100.83
C_c, W/K	356.0	790.7	1017.9	8.0	383.3
C^*	0.2187	0	0	0	0.2631
R_i, K/W	7.28_{10}^{-3}	6.97_{10}^{-4}	1.06_{10}^{-3}	4.51_{10}^{-1}	1.59_{10}^{-3}
R_w, K/W	2.11_{10}^{-5}	9.50_{10}^{-6}	7.38_{10}^{-6}	9.38_{10}^{-4}	1.96_{10}^{-5}
R_o, W/K	7.46_{10}^{-3}	3.36_{10}^{-3}	2.61_{10}^{-3}	3.32_{10}^{-1}	6.98_{10}^{-3}
NTU_o	0.8705	0.3112	0.2671	0.1594	1.1546
ε	0.5463	0.2675	0.2344	0.1473	0.6322
L, m	0,1147	0,2548	0.3280	0.00240	0.70
Δp_h, Pa	4074	12136	7411	9	66194
	Refrigerant side pressure drop : 91182 Pa				
$\dot{Q}$, kW	1.9479	4.3942	4.9571	0.0245	1.3245
	Overall condenser heat transfer rate: 12.65 kW				

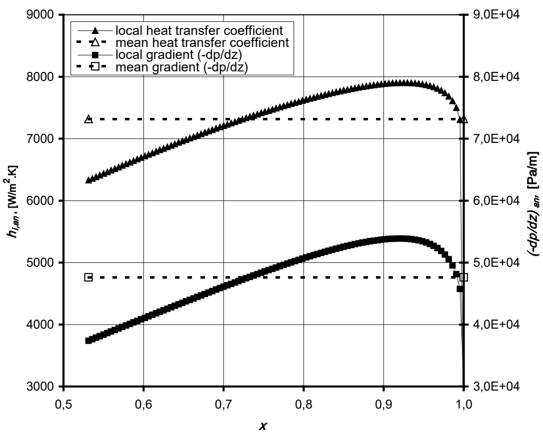


Fig. 10. Variation of heat transfer coefficient and pressure gradient with vapor quality for annular flow at G = 421.3 kg/(m²s)

pressure gradient $(dp/dz)_{an}$ with vapor quality are shown. Here coefficient $h_{i,an}(x)$ has been determined on correlation by Shah (1979) and gradient $(dp/dz)_{an}$ was evaluated using Eq. (26).

As forced convection driven by vapor shears is the only mechanism under this regime, note strong nonlinear variability in $h_{i,an}(x)$ as well in $(-dp/dz)_{an}$ with vapor quality so that both the curves are like parallel. There are also remarkably higher values of $h_{i,an}$ and (-dp/dz)an for the case considered of D_i = 0.8 mm in tube inner diameter when compared to cases with larger D_i values order of a few mm. There is a maximum also in coefficient $h_{i,an}$ as well as in gradient $(-dp/dz)_{an}$, either at about x ~ 0.9.

Conclusions

In the paper at first modelling of horizontal flow in tube condensers is summarized with particular attention on evaluation of refrigerant flow regimes in minichannnels with the use of two phase flow regime map accounting for the surface tension. Based on the modelling a computerized spreadsheet has been developed for design analysis of the condensers. Then, results of numerical experiments generated with the use of the spreadsheet are presented for a crossflow automotive condenser with R134a flow in minichannels of 0.8mm in inner diameter.

Presentation of the results begins with flow regimes predictions where effects of surface tension are also included. The condensation path for assumed input data of the condenser under study goes through the annular regime, then passes intermittent (slug and plug) regime and ends with bubble regime. Evaluations of thermal resistances show that the accuracy of heat transfer prediction for two phase flow region of the condenser is controlled by accuracy of the air side modelling. Regarding pressure drop, on R134a side the biggest contribution of ~73% in overall pressure drop is under single phase flow of subcooled liquid and for two phase flow region the biggest contribution of ~13% is under annular flow. The overall pressure drop under two phase flow results in change the mean air – R134a temperature difference in zone of two phase flow by ~1.6%, hence negligibly. Two additional remarks arise also from the results. The first is that involving surface tension into flow regime predictions for minichannels is necessary. In this respect map (Tabatabai & Faghri, 1979) directed on two phase flow regime predictions in minichannels has been employed in the paper. Worthy also to mention is lack of experimental data that can be used for modelling of condensation under intermittent flow regime.

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