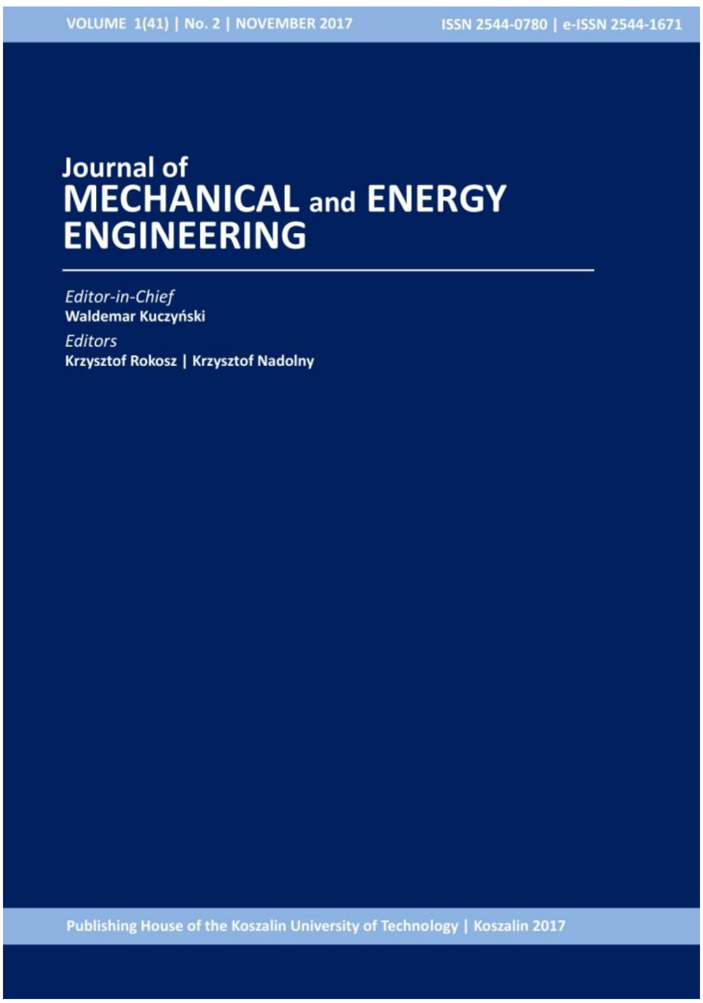
Modeling the instabilities of the condensation process of the R134a and 404A refrigerants in pipe minichannels in the conditions of periodic hydrodynamic disturbances

Waldemar KUCZYŃSKI, Aleksander DENIS

Online: http://www.jmee.tu.koszalin.pl/download_article/jmee_2017_02_179188.pdf

Cite this article as:
Kuczyński W., Denis A. Modeling the instabilities of the condensation process of the R134a and 404A refrigerants in pipe minichannels in the conditions of periodic hydrodynamic disturbances. Journal of Mechanical and Energy Engineering, Vol. 1(41), No. 2, 2017, pp. 179-188.



Journal of Mechanical and Energy Engineering

ISSN (**Print**): 2544-0780
ISSN (**Online**): 2544-1671
Volume: 1(41)
Number: 2
Year: 2017
Pages: 179-188

Article Info:
Received 5 October 2017
Accepted 25 November 2017

Journal of
MECHANICAL and ENERGY
ENGINEERING

ISSN: 2544-0780 | e-ISSN: 2544-1671
Vol. 1(41) | No. 2 | November 2017
pp. 179-188

MODELING THE INSTABILITIES OF THE CONDENSATION PROCESS OF THE R134A AND 404A REFRIGERANTS IN PIPE MINICHANNELS IN THE CONDITIONS OF PERIODIC HYDRODYNAMIC DISTURBANCES

Waldemar KUCZYŃSKI[1*], Aleksander DENIS[1]

[1*] Faculty of Mechanical Engineering, Department of Energetics, Koszalin University of Technology, Raclawicka 15-17, 75-620, Koszalin, Poland, e-mail: waldemar.kuczynski@tu.koszalin.pl

(Received 5 October 2017, Accepted 25 November 2017)

Abstract: In the present paper, an attempt was made to model the periodic hydrodynamic instabilities of the condensation process of a refrigerant in pipe minichannels. For this purpose, a homogenous thermodynamic model with an equilibrium condition on both sides was used, which was based on the equations of the balance of the mass, momentum and energy. These equations were used for numerical modeling of flow disturbances with a phase change. This model takes into account the complexity of multiphase flows. The accuracy of the model calculations was verified by means of the experimental results obtained. A satisfactory compliance was found for this comparison, which confirms the usefulness of the computational model proposed for the determination of the influence of periodically generated hydrodynamic disturbances on the condensation process of R134a and R404A refrigerants in pipe minichannels..

Keywords: condensation, minichannels, modeling of instabilities

1. INTRODUCTION

For many years now, investigations have been conducted concerning those phenomena that occur during multiphase flows of liquids and gases; attempts have been made to specify these processes with the aid of mathematical models. To a great extent, this research concerns the wave phenomena that accompany phase changes. Two modeling methods for these phenomena are described in the literature [1-5]. The first method involves an analysis of the multiphase flow structure in the form of a description of local extensive parameters of each of the phases occurring. This description pertains to areas outside of the interfacial area. The second approach makes use of a group of continuous models, in which averaging of the physical quantities is applied, i.e., all heterogeneities are averaged. As some of the authors have stated [1, 2], this group of models includes poly-liquid models, models of mixtures and homogenous models.

In their basic assumptions, each of these models is based on the equations of the conservation of mass, momentum and energy. These equations are adapted depending of the processes being modeled that occur during phase changes. In the case of the wave phenomena of two-phase media, it is important to define the sources of the occurrence of these phenomena [6-12].

2. MODELING OF THE INSTABILITIES OF THE CONDENSATION PROCESS

In the practice of modeling wave phenomena, it is assumed that we are dealing with a system of elementary phenomena that form a physical model. When solving elementary models, a model of the entire complex process is obtained.

Depending of the purpose of a simulation, physical phenomena are treated as either discrete systems in space (with concentrated constants) or as continuous processes in space (with distributed agglomerates) with continuous time.

2.1. Methodology of measurements

When following the procedure described above, the basic equations of conservation for the condensation process in a minichannel where periodic instabilities occur were considered.

For the purpose of describing the instabilities of the mass flow rate in a pipe minichannel, the following form of the univariate engineering equation [13, 14] for the conservation of mass in a two-phase flow was used:

$$\frac{\partial \rho}{\partial t} + w\frac{\partial \rho}{\partial z} + \rho\frac{\partial w}{\partial z} = -\rho w \frac{1}{A}\frac{\partial A}{\partial z}, \tag{1}$$

under the following assumption [2]:

$$\dot{m} = w\rho A = const. \tag{2}$$

The changes (oscillations) of the mass flow rate of the refrigerant that are produced by means of a periodic generation of dynamic disturbances may be treated as their displacement in the form of a monochromatic wave with a small amplitude and frequency along the axis of the horizontal channel [1, 2, 3, 9]. During condensation in a pipe minichannel under the conditions of periodic changes (an increase and decay) of the mass flux density (G) of the refrigerant, the average mass flux density ṁnst of the refrigerant in a transient state is determined with the following equation:

$$\dot{m}_{nst.} = (G \cdot A)_{nst} = A\cdot(G + G') \cdot \exp[i(\omega\tau - kz)], \tag{3}$$

where:
- G – mass flux density of the refrigerant in a stationary state,
- G' – mass flux density of the refrigerant in a transient state,
- i – imaginary value $i^2 = 1$,
- τ – time when disturbances occur,
- k – wave number, $k = \frac{2\pi}{\lambda} = \frac{\omega}{v_p}$,
- λ – wave length,
- v_p – phase velocity of disturbances, $v_p = \frac{l}{\Delta\tau}$,
- l – distance between pressure sensors,
- $\Delta\tau$ – displacement time of the pressure change signal,
- ω – pulsation (angular frequency) of the basic parameter, $\omega = \frac{2\pi}{T}$,
- T – wave period ($T = \tau = \tau_o + \tau_c$).

If the cross-sectional area of the channel is constant, $\left(\frac{\partial A}{\partial z}\right) = 0$, equation (3) takes also on the form [1,3]:

$$\dot{m}_{nst.} = (G \cdot A)_{nst} = A\cdot(G + G') \cdot \exp[i(\omega\tau - kz)]. \tag{4}$$

The assumption that the mass flow rate $\dot{m}_{nst}$. has an inertial nature as a result of the occurrence of periodical disturbances is accepted. This results in a pressure drop Δp on the flow path of the condensing refrigerant:

$$p = \Delta p_{res.} + \Delta p(t)\ , \tag{5}$$

where: Δp_{res} – linear resistances of flow in a channel in the form of $\Delta p_{res.} = c\cdot\frac{\rho w^2}{2}$, where $c = \lambda\frac{l}{d}$; and $\Delta p(t)$ – inertance of the refrigerant stream expressed with the dependence $\Delta p(t) = I\cdot\frac{dV(t)}{dt}$, where $I = \frac{\rho\cdot L}{A}$ is the inertance coefficient that characterizes the flow of liquid with density ρ in a channel with length L and cross section A.

Taking into consideration these dependencies, Equation (5) takes on the following form:

$$\Delta p = \frac{\lambda\cdot L\cdot\rho w^2}{2\cdot d} + \frac{\rho\cdot L}{A}\cdot\frac{dV(t)}{dt}, \tag{6}$$

whereas:

$$\frac{\rho\cdot L}{A}\cdot\frac{dV(t)}{dt} = \frac{L}{A}\cdot\frac{dm}{dt}\ . \tag{7}$$

Equation (6) can be written in the following form:

$$p_1 - p_2 = \frac{\lambda\cdot L\cdot\rho w^2}{2\cdot d} + \frac{L}{A}\cdot\frac{dm}{dt}, \tag{8}$$

where p_1 and p_2 are the pressure values in the inlet and outlet cross-sections of the minichannel, respectively. In compliance with the procedure of the discretization process [13, 14], the pressure on the boundaries of the ith segment is considered as an arithmetic average pressure value in the next segment. Using the notation from Fig. 1, the following is obtained:

$$p' = \frac{p_{i-1} + p_i}{2} \text{ and } p'' = \frac{p_i + p_{i+1}}{2}. \tag{9}$$

After substituting Equation (8) in (9), the following was obtained:

$$\frac{dm_i}{dt} = \frac{A}{l_i}\cdot\left(\frac{p_{i-1} - p_{i+1}}{2} - \frac{\lambda\cdot l_i\cdot\rho w_i^{\ 2}}{2\cdot d}\cdot sign(w_i)\right), \tag{10}$$

where *sign* (signum) is the function that combines the flow direction with the pressure drop.

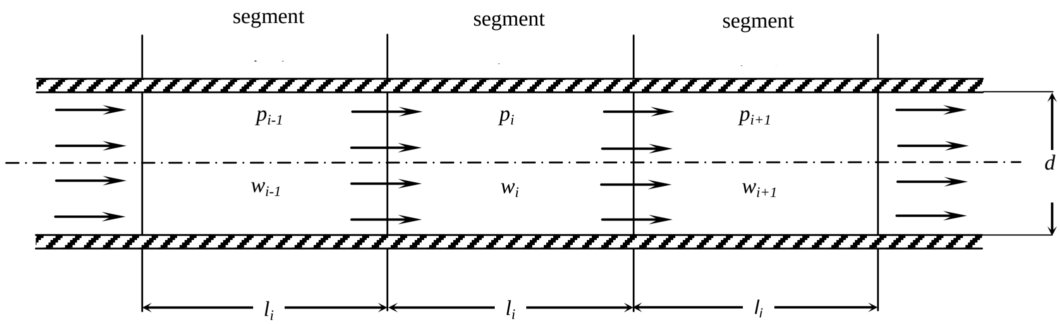


Fig. 1. Discretization model of the pipe minichannel

Equation (10) constitutes a computational algorithm that determines the oscillations of the mass flow rate produced by a hydrodynamic instability [9, 13, 14].

2.2. Oscillation model of the mass flow rate

The equation of the balance of the momentum transport takes on the following form [2]:

$$\rho \frac{\partial w}{\partial t} + \frac{\partial w_i w_j}{\partial z} = \kappa_i \rho + \frac{\partial T_{ij}}{\partial z} , \tag{11}$$

where:

$\frac{\partial T_{ij}}{\partial z}$ – divergence of the tensor of shearing stresses,

$\frac{\partial w}{\partial t}$ – acceleration of element with mass ρdV,

κ – unitary mass forces.

It was accepted for the wave and dispersion medium that $\frac{\partial w}{\partial t} = 0$ and $\frac{\partial w}{\partial z} = \frac{1}{V}\frac{dV}{dt}$; hence, Equation (11) can be written as follows:

$$\frac{\partial \rho}{dt} \int_{V9t)} dV = \frac{\partial m}{dt} , \tag{12}$$

while assuming that $\rho = const$:

$$\frac{dm}{dt} = \frac{dV}{dp} \rho \frac{dp}{dt} , \tag{13}$$

and from there:

$$\dot{m}_{nst.wej.} - \dot{m}_{nst.wyj.} = \frac{dV}{dp} \rho \frac{dp}{dt} . \tag{14}$$

From Equation (14), dependence was obtained for the calculation of the pressure oscillation under the conditions of periodic hydrodynamic instabilities:

$$\frac{dp}{dt} = \frac{p_{i-1} - p_{i+1}}{\left(V_{i-1} + V_{i+1}\right) \cdot \rho} \cdot \left(\dot{m}_{i-1} - \dot{m}_{m+1}\right) . \tag{15}$$

Equation (15) is compliant with the one provided by [13,14].

2.3. Oscillations of the channel wall temperature

Based on the energy balance equations that comply with the Newton equation for the stationary state, the following is obtained:

$$\dot{Q} = A \cdot \alpha \cdot \left(T_{sat} - T_{wall}\right) , \tag{16}$$

$$\dot{Q} = \dot{m} \cdot c_p \cdot \left(T_{in.} - T_{out}\right) , \tag{17}$$

$$A \cdot \alpha \cdot \left(T_{sat} - T_{wall}\right) = \dot{m} \cdot c_p \cdot \left(T_{in.} - T_{out}\right) = 0 . \tag{18}$$

Using the equation of the first law of thermodynamics with the following form:

$$\dot{Q} = \rho \cdot v \cdot c_p \cdot \frac{dT}{dt} , \tag{19}$$

the following is obtained:

$$\begin{aligned} & A \cdot \alpha \cdot \left(T_{sat} - T_{wall}\right) - \dot{m} \cdot c_p \cdot \left(T_{in.} - T_{out}\right) = \\ & = \rho \cdot v \cdot c_p \cdot \frac{dT}{dt} . \end{aligned} \tag{20}$$

After the conversion of Equation (20) and taking into consideration the notation from Fig. 1, dependence was obtained that describes oscillations of the channel wall temperature T_{wall}:

$$T_{wall} = T_{sat.} - \frac{\dot{Q} + \left(\dot{m}_{i-1} h_{i-1} - \dot{m}_{i+1} h_{i+1}\right)}{A \cdot \alpha} . \tag{21}$$

Dependencies (10), (15) and (21) describe the oscillations of the mass flow rate $\dot{m}$, pressure p_s and channel wall temperature $T_{ści}$ that are the result of the interaction of periodic disturbances that produce hydrodynamic instabilities. In the modeling process, the fact that the feeding valve is periodically opened and closed was taken into account. The rate of mass flow $\dot{m}$ through the valve is calculated from the following dependence:

$$\dot{m} = C \cdot A \cdot \sqrt{2 \rho_i \cdot \Delta p} , \tag{22}$$

where:
C – relative flow ratio of the valve,
A – flow cross section,
ρ_i – density of the refrigerant before the valve,
Δp – drop of the pressure at the valve.

Oscillations on the supply to the pipe minichannel were modeled with the aid of a programmed timer, the so-called *regulator of feeding valve setpoint* [7, 8]:

$$Pu(s) = 0 \leq \left| \left(P + I\frac{1}{s} + Ds \right) \cdot \left(i_n - i_p \right) \right| \leq \max , \quad (23)$$

where:
Pu – signal of the output from the controller,
P – proportional term,
I – integral term,
s – Laplace operator,
$(i_n - i_p)$ – control error.

The model simulations were conducted in MATLAB 2008 numerical software with the use of the SYMULINK package. This package offers the ability to conduct simulations both with discrete and continuous time. It is used, among other applications, in the simulation and analysis of control theory [13, 14].

3. EXPERIMENTAL EXAMINATIONS AND MEASURING PROCEDURES

The experimental tests were conducted using a specially designed testing facility, which is presented in the photo in Fig. 2, and its schema is found in Fig. 3.

Stainless steel pipe minichannels with a circular cross section were used for the investigations. The following internal diameters were used: d = 0.64, 0.90, 1.40, 1.44, 1.92, 2.30 and 3.30 mm. The measuring section was cooled with water. The R134a and R404A refrigerants were used in the examinations. With the aid of evenly arranged Cerabar TPMP131–type piezoelectric pressure sensors, the saturation pressure p_s of the refrigerant was measured during its condensation phase in flow through a straight, axial segment of the minichannel. These sensors were combined with an electronic signal processing system and a computer-based monitoring module. In addition, the pressure pin of the refrigerant at the inlet to the measuring section, the pressure pout at the outlet from this section and the pressure drop Δp over the minichannel length were all measured.

The measurements of the temperature T_{wall} of the external surface of the minichannel wall were conducted with the aid of *K*-type thermocouples that were evenly arranged over its length (in the cross sections of the pressure measurement). The temperature T_{in} of the refrigerant in the inlet cross section to the measuring section, the temperature T_{out} in the outlet cross section and the temperature T_{water} of the cooling water in characteristic cross sections of the water cooling system were also measured. The measurement of the mass flow rate was carried out with a *Coriolisa Promass 80A* mass flowmeter.

All of the pressure sensors, thermocouples and flow rates were connected to a *USB PersonalDaq/3000*-type measuring set with a *PDQ30* and *USB 1616HS* extensions with an *AI-EXP48* extension. The measuring system was connected to a computer, making data archiving possible for the purpose of later analysis of the data.

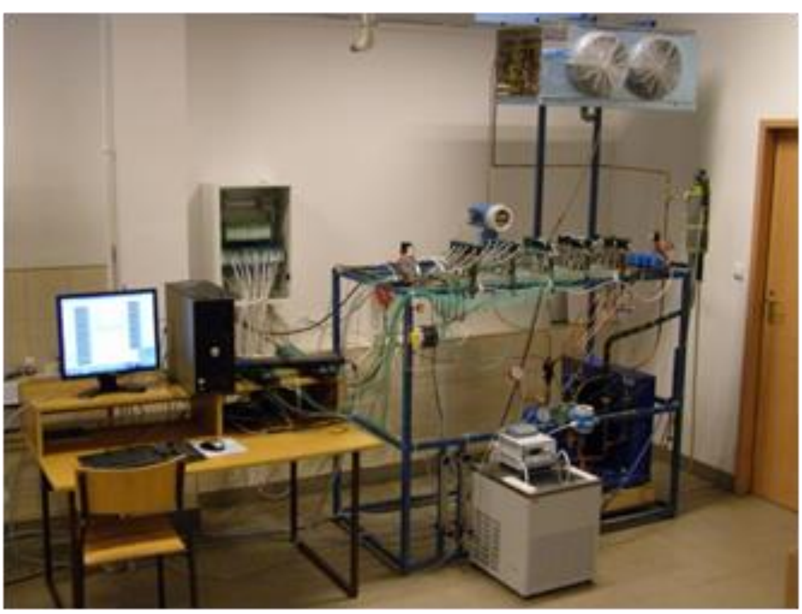


Fig. 2. View of the measuring facility

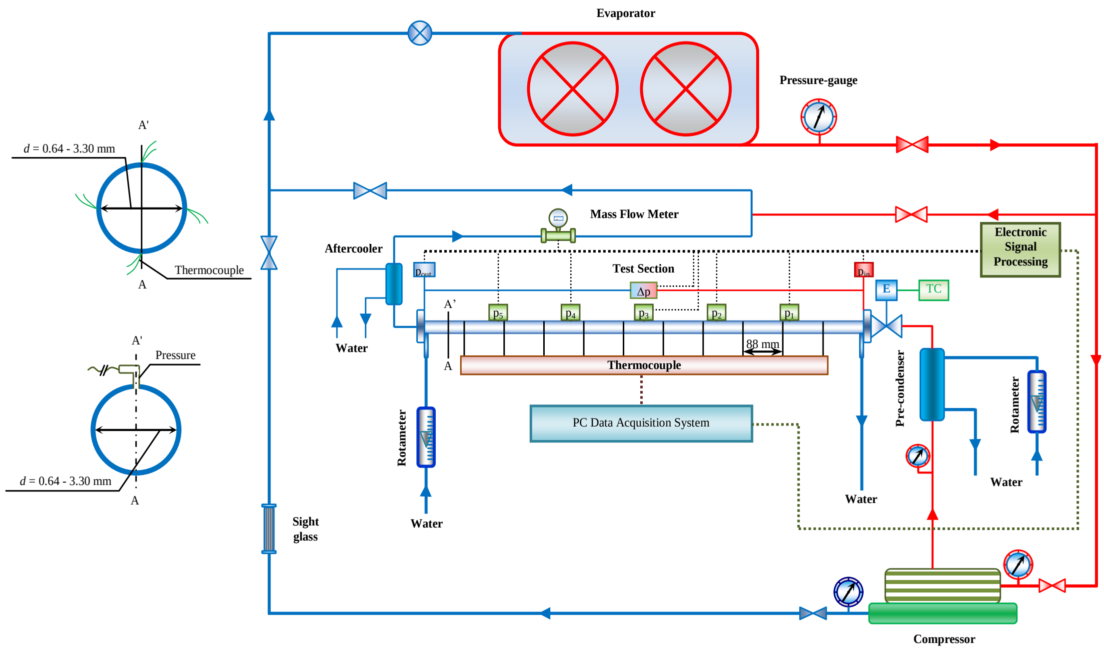


Fig. 3. Schema of measuring facility

Prior to opening valve *E* (Fig. 1), which produced periodical disturbances, the measuring system was brought to a stationary state, and an initial mass flow rate value $\dot{m}_{initial}$ ≈1.85 kg/h was obtained for the refrigerant. In this state, the researchers began to generate external disturbances of a periodic nature by closing and opening the electromagnetic valve *E*. This valve was installed on the inlet to the measuring segment, and it was controlled by means of a *LOGO 230R TC* timer manufactured by Siemens.

The concept of "*periodically generated disturbances*" is to be understood as periodic changes in the feeding of a pipe minichannel with a refrigerant. These changes are the result of a change in the opening time τ_o and closing time τ_c of valve *E*, which was installed on the inlet to the measuring section. In a given measuring series, the condition was maintained that $\tau_o = \tau_c$. An increment in the opening and closing time of the valve was 0.05 s for each measuring series. It was accepted in the schedule of experiments that the initial closing and opening time of the valve *E* was $\tau_o = \tau_c = 0.1$ s. This was the result of an empirical determination of the lower range of those frequencies at which the impact of the disturbances generated during the condensation process in the pipe minichannel was already evident [15,16].

The sum of the opening and closing time of the valve constituted the basis for the determination of the frequencies of the disturbances generated *f* [Hz], in compliance with the dependence provided by [17]:

$$f = \frac{1}{\tau_o + \tau_c}, \tag{24}$$

where: τ_o and τ_c denote the opening and closing time, respectively, of the valve, expressed in seconds.

The examinations were carried out until the moment when the periodic influences resulted in an instantaneous stopping of the flow of the refrigerant. The tests were conducted for the following parameters [15,16, 18-21]:

– mass flux density of the refrigerant G = 100-4100 kg/(m^2·s),
– saturation temperature of the refrigerant T_s = 20-55°C,
– frequency of disturbances f = 0.20-5 Hz.

4. ANALYSIS OF THE RESULTS OF THE INVESTIGATIONS

The results of the numerical calculations (performed with MATLAB 2008 software) of the condensation process for R134a and R404A refrigerants under the conditions of hydrodynamic instabilities were compared with the experimental data. The collation is presented in Figs. 4 to 6. A satisfactory compliance was found for the description of the phase and the distribution of the amplitude of the disturbances occurring.

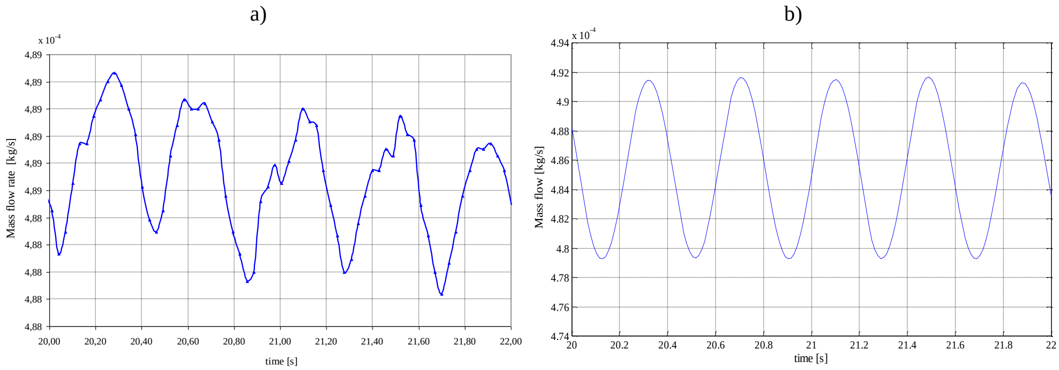


Fig. 4. Comparison of the mass flow rate results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrodynamic disturbances on the condensation process of the R134a refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency of disturbances f = 3.33 Hz; a) Experiment, b) Symulation Matlab 2008 [15, 16]

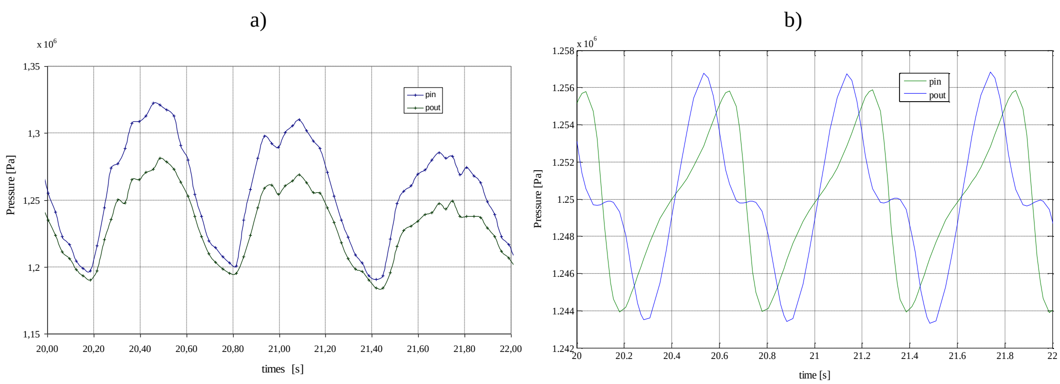


Fig. 5. Comparison of the condensation pressure results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrody-namic disturbances on the condensation process of the R134a refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency f = 1.67 Hz; a) Experiment, b) Symulation Matlab 2008 [15, 16]

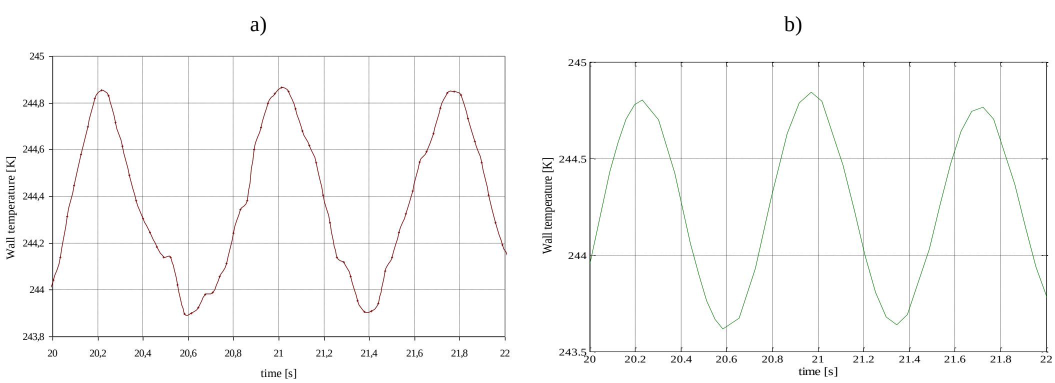


Fig. 6. Comparison of the channel wall temperature results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrodynamic disturbances on the condensation process of the R134a refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency f = 1.25 Hz; a) Experiment, b) Symulation Matlab 2008 [15, 16]

a) b)

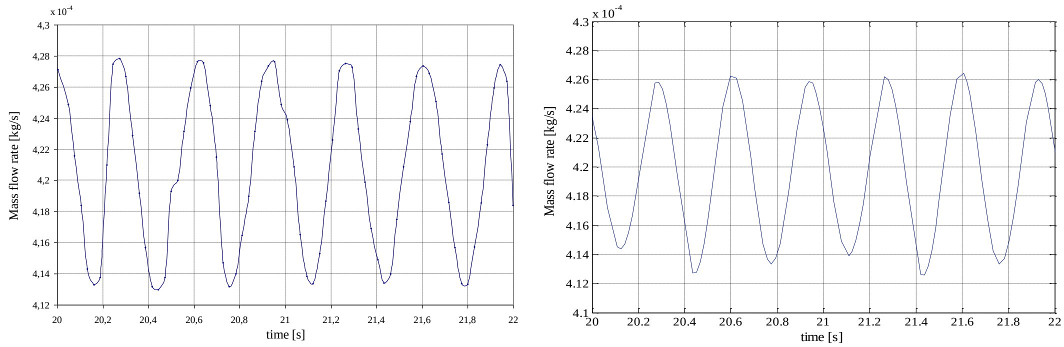


Fig. 7. Comparison of the mass flow rate results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrodynamic disturbances on the condensation process of the R404A refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency of disturbances f = 3.33 Hz; a) Experiment, b) Symulation Matlab 2008 [15, 16]

a) b)

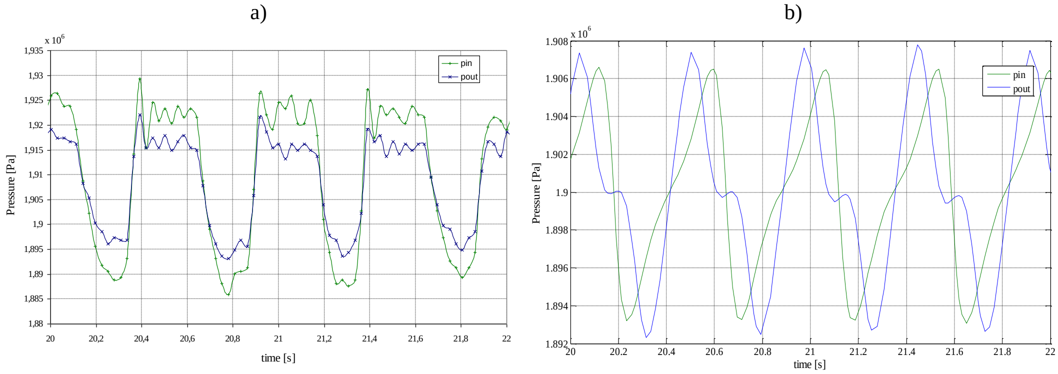


Fig. 8. Comparison of the condensation pressure results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrodynamic disturbances on the condensation process of the R404A refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency f = 2 Hz; a) Experiment, b) Symulation Matlab 2008 [15, 16]

a) b)

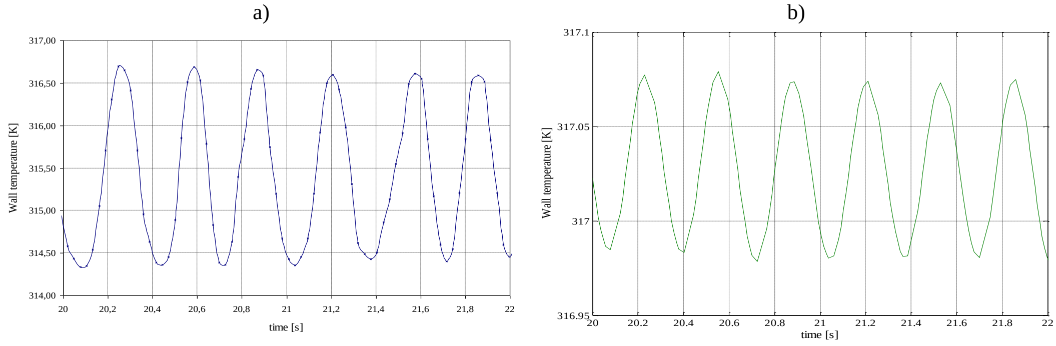


Fig. 9. Comparison of the channel wall temperature results from the numerical calculations with the results of the experimental investigations that describe the influence of hydrodynamic disturbances on the condensation process of the R404A refrigerant in a pipe minichannel with an internal diameter d = 0.64 mm and a frequency f = 3.33 Hz [15, 16]

5. CONCLUSIONS

The aim of this paper was to try to model the hydrodynamic instabilities during condensation of refrigerants, homogeneous R134a and zeotropic mixture R404A, in tubular mini-channels. Those instabilities were periodic, with wave-length and frequency set in advance. Thermal-mechanical equations based on mass, momentum and energy balance of the system were employed to do so. Results calculated according to the model were compared with the experimental data. Satisfying agreement between theoretical calculations and experimental results was obtained for the phase and the amplitude distribution of instabilities:

- oscillations of refrigerant's mass flux: 90% overlap for R134a and 54% for R404A,
- oscillations of refrigerant's pressure: 62% for R134a and 52% for R404A,
- oscillations of wall-temperature: 8% for R134a and 78% for R404A.

Thus, the theoretical results should be considered as preliminary. It is necessary to adopt other computational methods or software in order to obtain the required accuracy.

Nomenclature

Symbols

d – internal diameter of the mini-channel, mm
f – frequency, Hz
G – mass flux density, kg/(m^2·s)
L – mini-channel length, m
m – mass flow rate, kg/h
p – pressure, MPa
T – temperature, ^{o}C

Greek letters

λ – wave length, m
τ – time, s
$\Delta\tau$ – time interval, s

Subscripts

c – closing of cut-off valve
in. – inlet cross-section, moment measurement begins
o – opening of cut-off valve
out. – outlet cross-section
s – condensation (liquefaction)

Acronyms

HE – Heat Exchanger
HTC – Heat Transfer Coefficient
MC – Microchannel

References

1. Ardron K.H., Duffey R.B. (1977). Acoustic wave propagation in a flowing liquid-vapor mixture. *Int. J. Multiphase Flow*, Vol. 4, pp. 303-322.
2. Bilicki Z., Downar-Zapolski P. (1992). Wave phenomena in two-liquid and homogenous two-phase flow models. *Trans. IFFM*, Vol. 94, pp. 19-42.
3. Bohdal T., Kuczyński W. (2005). Investigation of boiling of refrigeration medium under periodic disturbance conditions. *Int. J. of Experimental Heat Transfer*, Vol. 18 , No. 3, pp. 135-151.
4. Bohdal T., Kuczyński W. (2011). Boiling of R404A Refrigeration Medium Under the Conditions of Periodically Generated Disturbances. *Heat Transfer Engineering*, Vol. 32 , No. 5, pp. 359-368.
5. Celmerowski A. (2008). *Modeling and simulation of physical systems Matlab/Simulink*. Publishing House of the Bialistok University of Technology. Bialistok. (in Polish)
6. Comakli O., Karsli S., Yilmaz M. (2002). Experimental investigation of two phase flow instabilities in a horizonatl in-tube boiling system. *E. Conv. and Management*, No. 43, pp. 249- 268.
7. Kacprowski J. (1960). *Outline of Electroacoustics*. Communications Publishing Houses. Issue II, Warsaw. (in Polish)
8. Kakac S., Bon B. (2008). A review of two-phase flow dynamic instabilities in tube boiling systems, *Int. J. of Heat and Mass Transfer* , Vol. 51, No. 3-4, pp. 399-433.
9. Kuczyński W., Charun H., Bohdal T. (2012). Influence of hydrodynamic instability on the heat transfer coefficient during condensation of R134a and R404A refrigerants in pipe mini-channels. *International Journal of Heat and Mass Transfer*, Vol. 55, pp. 1083-1094.
10. Nakoryakov V.E., Pokusaev B.G., Pribaturin N.A., Shreiber I.R. (1988). The wave dynamics in vapor liquid medium. *Int. J. of Multiphase Flow*, Vol. 14, No. 6, pp. 655-671.
11. Ramshaw J.D., Trapp J.A. (1978). Characterisyics, stability and short-wavelenght phenomena in two-phase flow equation systems. *Nuc. Sci. and Engineering*, Vol. 66, pp. 378-396.
12. Tian Ch., Dou Ch., Yang X., Li X. (2005). Instability of automotive air conditioning system with a variable displacement compressor. Part 1. Experimental investigation. *Int. J. Refrigeration*, Vol. 28, pp. 1102-1110.
13. Tarnowski W. (2004). *Modeling of systems*. Publishing House of the Koszalin University of Technology, Koszalin. (in Polish)
14. Tarnowski W. (2001). *Simulation and Optimalization in MATLAB*. Publishing of the WSM Gdynia, Gdynia. (in Polish)
15. Kuczyński W. (2013) Modeling of the propagation of a pressure wave during the condensation process of R134a refrigerant in a pipe minichannel under the periodic conditions of hydrodynamic disturbances, *International Journal of Heat and Mass Transfer*, Vol. 56, pp. 715-723.
16. Kuczyński W. (2013). Characterization of pressure-wave propagation during the condensation of R404A and R134a refrigerants in pipe mini-channels that undergo periodic hydrodynamic disturbances. *International Journal of Heat and Fluid Flow*, Vol. 40, pp. 135-150.
17. Kartaschoff P. (1985). *Frequency and Time*. Publishing Houses of Communications, Warsaw. (in Polish)
18. Tian Ch., Dou Ch., Yang X., Li X. (2005). Instability of automotive air conditioning system with a variable displacement compressor. Part 2. Experimental investigation. *Int. J. Refrigeration*, Vol. 28, pp. 1111-1123.
19. Xu J., Chen T. (2000). Acoustic wave prediction in flowing steam-water two-phase mixture. *Int. J. of Heat and Mass Transfer*, Vol. 43, pp. 1079-1088.
20. Zauderer E. (1983). *Partial differential equation of applied mathematics*. Wiley, New York.

21. Zhang T., Wen J., Peles Y., Catano J., Zhou R., Jensen M. (2011). Two-phase refrigeration flow instability analysis and active control in transient electronics cooling systems. *Int. J. of Multiphase Flow*, Vol. 37, pp. 84-97.

Biographical notes

Waldemar Kuczyński is a graduate of the Koszalin University of Technology, Poland. Since 1999, he has been working as the Chair of Thermal Engineering and Refrigerating Engineer-ing. In the year 2002, he was awarded the title of master of science in the field of machine construction and operation in the specialty of thermal engineering. In the years 2002 – 2007, he was a Ph.D. student at the Faculty of Mechanical Engineering of the Koszalin University of Technology. In the year 2008, under a project of the State Committee for Scientific Research, he defended his doctoral thesis entitled “Boiling Testing in Refrigerant Flow under the Conditions of Periodically Generated Disturbances" and in the year 2014 D.Sc. degree in Machinery Construction and Operation from Koszalin University of Technology. Since September 2008 he has been the head of the laboratory of the Chair of Thermal Engineering and Refrigerating Engineering and from 2016 he is also Deputy Dean of the Faculty of Mechanical Engineering of Koszalin University of Technology. He has authored and co-authored more than 100 articles and co-authored nine research projects (and the director of three). His chief interests focus on the issues of wave phenomena and instability during the condensation and boiling of refrigerants in conventional channels and in mini-channels.

Aleksander Denis received BSc (2013) at Wroclaw University of Technology and double MSc (2015) at Warsaw University of Technology and Ecole des Mines de Nantes; currently PhD student at Koszalin University of Technology (Department of Power Engineering). His scientific interests focus on instabilities of condensation of refrigerants in mini-channels. So far he published 3 scientific papers in national journals.