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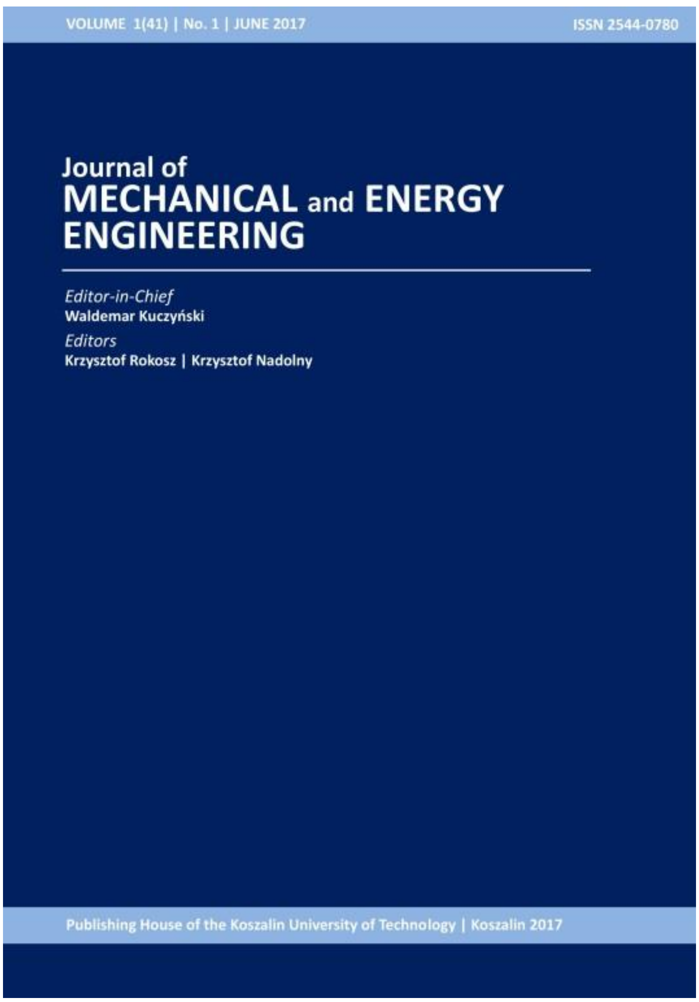

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ANALISYS OF HEAT TRANSFER COEFFICIENT DURING REFRIGERANT CONDENSATION IN VERTICAL PIPE MINICHANNEL

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Abstract: This article presents the results of experimental research of R404A, R407C and R410A high-pressure refrigerants condensation in vertical pipe minichannels with an internal diameter d below 2,5 mm. The study determined the local and average heat transfer coefficient in the full range of vapor quality, x = 1-0. On the basis of experimental investigations, the dependence of heat transfer coefficient on the vapor quality x, the mass flux density G and the channel internal diameter d was obtained.

Keywords: Heat Transfer Coefficient, heat flux density, minichannel, vapor quality

1. INTRODUCTION

The development of compact heat exchangers was initiated by the energy crisis of the 1970s. The surge in oil prices triggered by the embargo on the United States has prompted many countries to rationalize energy. In response to this situation, science started evolving towards miniaturization of energy systems simultaneously increasing their efficiency. Such behavior resulted in constant changes in the design of heat exchangers. According to the literature, the first mass-produced compact heat exchangers were created in SWE in Sweden in 1977. These were welded plate heat exchangers that are used today in refrigeration and air conditioning (up to 100 kW of heating power).

With the development of welding methods, including welding with laser beams, the internal diameter of the channels forming the key element of the exchangers has been successively reduced. Heat exchangers have been used in an increasingly wide range of temperatures and pressures, which has led to enhancement of application rates for the industry. At the end of the 20th century plate exchangers with channels with a hydraulic diameter of less than 1 mm were already in common use. This enabled the development of the production of equipment and power systems in mini and microscale. According to LI [1], miniature heat exchangers are excellent at acquiring and transmitting energy from low temperature sources. In the following years significant progress was made in the production of tubular heat exchangers, reducing the inner diameter of the channel to less than 1 mm and its wall thickness to 0.04 mm. Composite heat exchangers have been used in many engineering and scientific applications, including microelectromechanical systems, such as miniature pumps, sensors or actuators.

HEs Currently offered by the manufacturers are getting smaller and smaller, but still retaining their previous power. As the efficiency of the construction work increases, the wear and tear costs are rising [2]. The combination of these two factors requires efficient cooling. The heat flux emitted by the microchip is even 1000 W/cm^2 [3]. Increasing demands placed on refrigerators designers require the search for new design solutions. The authors of the article assume that the efficiency of miniature heat exchangers is significantly influenced by: the nature of the refrigerant flow, the type of refrigerant used and the internal diameter of the minichannel. Type of heat reception is also important for heat exchange as well as channel orientation. There is still a number of studies on heat transfer in air cooled minichannels. A similar situation occurs with vertical channels.

Bohdal [4] presents the results of thermo-flow studies for the zeotropic R404A, R407C, R410A and R134a homogeneous R134a. Tests were performed on minichannels with internal diameters d_w = 0.90-3.30 mm (0.90, 1.40, 1.60, 1.94, 2.30 and 3.30 mm). Daisuke [5] proposes a new heat exchange, pressure drop model with R134a, R32, R1234ze and R410A refrigerants in rectangular cross-section channels. The author points out that at low flow rates, the heat transfer coefficient does not change for the full range of x = (1-0). This model more closely takes into account the influence of surface tension on the condensation process in minichannels. López-Belchí [6] presents the results of tests for the heat transfer coefficient during condensation of refrigerant R32 and R410A in MCs. The work compares the value of the heat transfer coefficient and the heat flux depending on the vapor quality.

2. THE AIM AND SCOPE OF RESEARCH

The studies was conducted at three vertical minichannels with circular cross sections, made of stainless steel with internal diameters dw = 1,6 mm, 2 mm and 2.5 mm and length L = 1000 mm. The refrigerants R404A, R407C and R410A were used. The measuring section was cooled with air at a flow rate of 17 m/s and the refrigerant was driven with set and time-varying thermo-flow parameters. MC were placed in rectangular shaped air duct with 60x120 mm cross section. Experimental studies were performed in the following range of thermal-flow parameters:

Mass flux density: $G = 169 \div 1017$ kg/(m^2·s),
Heat flux density: $q = 0 \div 3$ (kW/m^2),
Saturation temperature: $t_s = 20 \div 40$ ^{o}C,
Vapor quality: $x = 0 \div 1$.

The design of the test stand allowed measurement of temperature and pressure of the refrigerant and the temperature of the cooling air along the length of the measuring section. The flow of refrigerant and coolant was also measured.

3. EXPERIMENTAL FACILITY

The heat and flow studies of the condensation process were carried out at a test stand that is shown in Figure 1. The superheated steam of the refrigerant was forced through the measuring section by the compressor after pre-cooling. A heat exchanger was installed in front of the measuring section, which was used to determine the vapor quality of the medium by the balance method. For this purpose, the volume flow of the cooling water and the refrigerant was measured, as well as the water and refrigerant temperature at the inlet and outlet of the exchanger. Adjustment of the water flow rate allowed to control the parameters of the medium (eg vapor quality) at the inlet to of the measuring section. Then the refrigerant condensed inside a vertical 950 mm stainless steel vertical tube minichannel (flow vertically down). MC was placed in a rectangular duct measuring 60×120×1100 mm. A counterflow currant of air at a speed of 17 m/s was forced into the duct. Refrigerant pressure was measured at the inlet and outlet of the measuring section with piezoresistive sensors fitted with the Endress + Hauser PMP 131-A1401A1W Transmitter, made in measuring class 0.5. In addition, a local pressure drop of 100 mm was measured with the Deltabar SPMP in class 0.075. Second heat exchanger was installed to obtain a homogenous liquid at the outlet of the measuring section. After cooling with water in the exchanger, the liquid medium was sent to the Coriolis 34XIP67 flowmeter in measuring class 0.52, where its mass flow was measured. The flow rate of the water through the exchanger was controlled by a RTU-06-160 type rotameter with accuracy class 2.5.

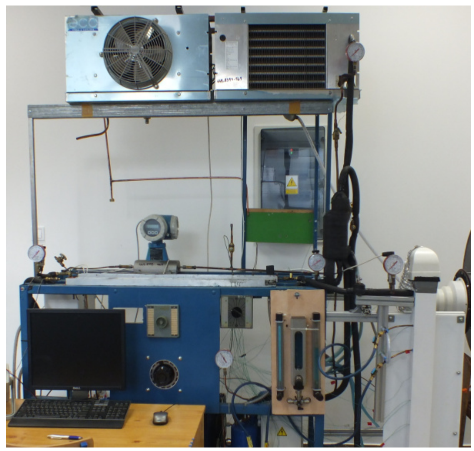


Fig. 1. The overall view of the experimental facility

The measuring equipment installed on the test bench allowed direct measurement of following values:

- the temperature of the refrigerant T_{F1} at the inlet and outlet T_{F2} from the measuring section,
- surface temperature of the outer wall of the tube minichannel using thermocouples in nine sections T_{W1} to T_{W9},
- cooling air temperature in three measuring sections in the air duct over the length of the measuring section $T_{P1} \div T_{P3}$,
- refrigerant mass flow rate through the measuring section $\dot{m}_r$,
- air mass flow rate through the measuring section $\dot{m}_p$,
- refrigerant pressure at the inlet and outlet of the measuring section p_{n1}, p_{n2},
- refrigerant pressure drop during the flow through the measuring section (Δp/ L),
- refrigerant temperature T_{F01}, T_{F02} at the inlet and outlet of the pre-cooling heat exchanger,

- T_{H01} temperature of cooling water at the inlet and T_{H02} at outlet of the exchanger,
- the mass flow rate of the water through the exchanger $\dot{m}_{H2O}$.

Indirectly the following values were calculated:

- heat flux density q,
- density of mass G,
- vapor quality x.

Measuring section is presented in Fig. 2.

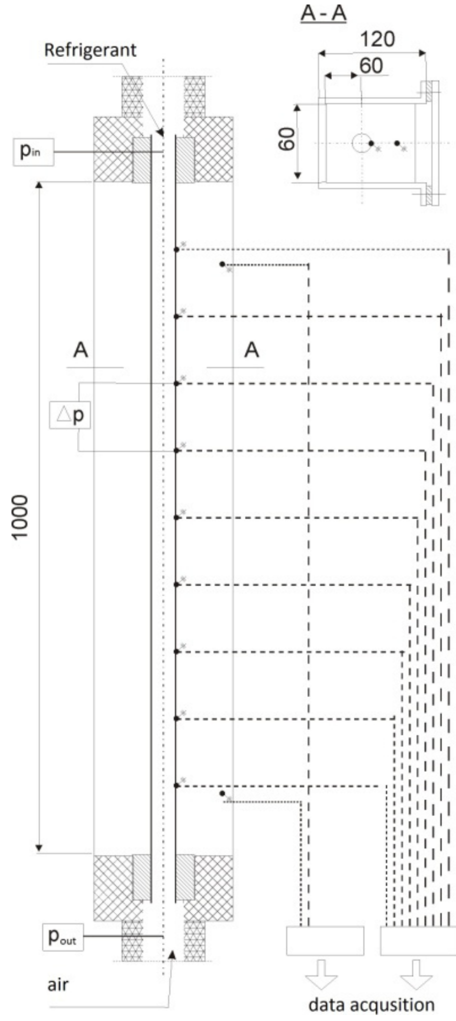


Fig. 2. Framework of the measuring section

The schematic diagram of experimental facility is shown in Figure 3.

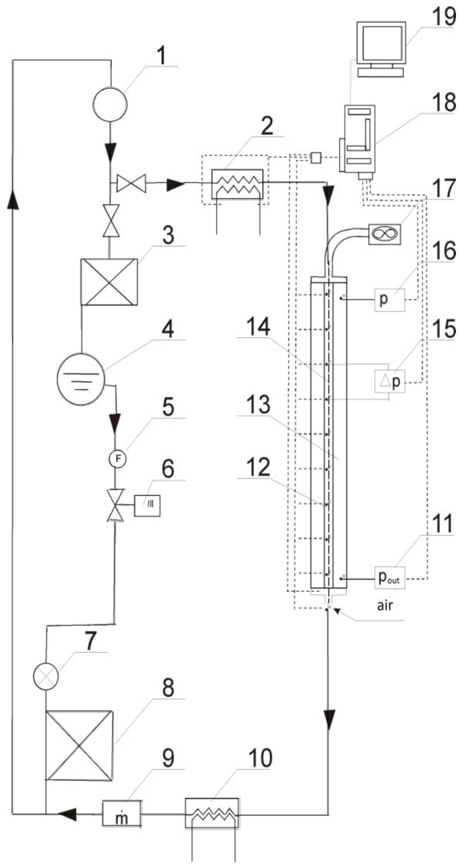


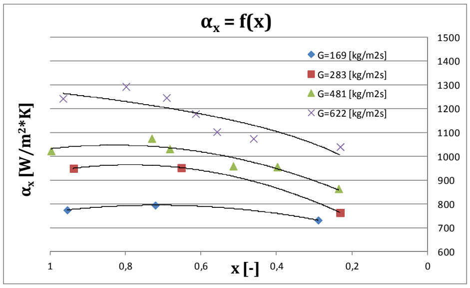
Fig. 3. Schematic diagram of the test stand: 1) compressor, 2) heat exchanger, 3) condenser, 4) refrigerant liquid tank, 5) filter, 6) electromagnetic valve, 7) expansion valve, 8) fan cooler, 9) refrigerant mass flow meter, 10) heat exchanger, 11) inlet pressure sensor, 12) K-type thermocouples, 13) air channel, 14) pipe minichannel, 15) differential pressure measurement, 16) outlet pressure sensor, 17) fan, 18) data acquisition system, 19) computer of the measuring section

4. EXPERIMENTAL RESULTS

Test results present local and mean values of heat transfer coefficient in vertical tube minichannels. Figure 4. shows the dependence of HTC on vapor quality in the range of x = 1÷ 0 in minichannel of diameter d_w = 2.5 mm during refrigerants condensation a) R404A with G = 169 ÷ 509 kg/(m^2·s), b) R407C with G = 169 ÷ 736 kg/(m^2·s), and c) R410A with G = 169 ÷ 622 kg/(m^2·s).

The heat transfer coefficient takes highest values in the specific condensing area (vapor quality x in the range of 0,6 to 0,8). The heat transfer coefficient decreases along with the decrease of vapor quality x.

a)



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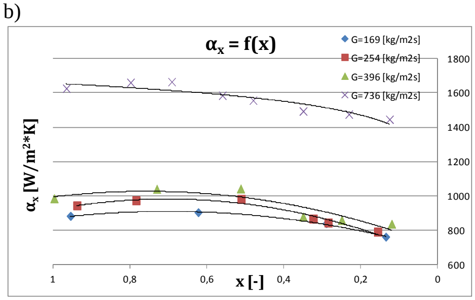

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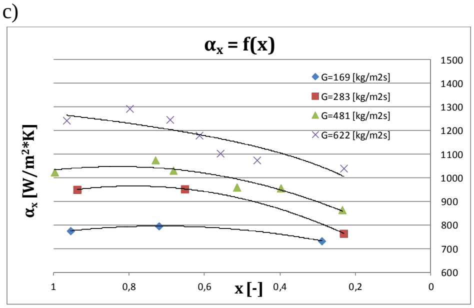
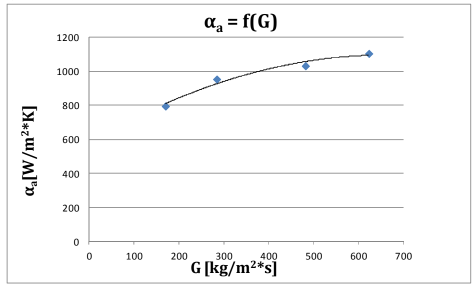


Fig. 4. Experimental dependence of local heat transfer coefficient on vapor quality during condensation of refrigerants a) R404A, b) R407C, c) R410A in vertical pipe minichannel with internal diameter d_w = 2,5 mm.

a)



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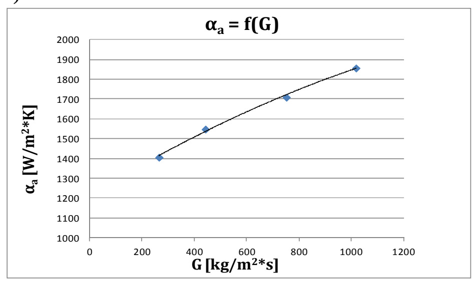

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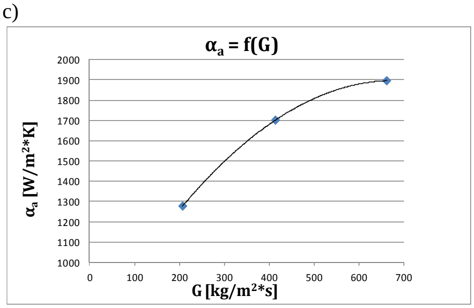


Fig. 5. The dependence of average heat transfer coefficient α_a on the mass flux density G for minichannel internal diameter a) dw = 2,5 mm, b) dw = 2 mm and c) 1,6 mm during condensation of refrigerant R410A.

Figure 5 shows the dependence of the avarage values of heat transfer coefficients on the mass flux density G during refrigerant condensation R410A in tube minichannel with diameter a) d_w = 2.5 mm, b) d_w = 2 mm and c) 1.6 mm. As you can see with the increase in density of the mass flux, the average heat transfer coefficient increases.

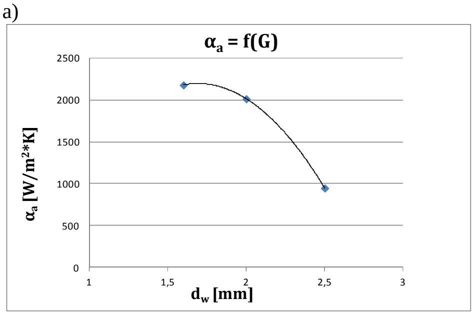
Value of HTC during refrigerant condensation R410A in minichannels with an internal diameter of d_w = 2.5 mm varies in the range of 800 W/(m^2·K) at mass flux density G = 169 kg/(m^2·s) to 1150 W/(m^2·K) for G = 622 kg/(m^2·s). For a channel with a diameter of d_w = 2 mm, the mean values of heat transfer coefficient oscillate in range of 1400 ÷ 1850 W/(m^2·K). As can be seen along with channels internal diameter decrease, HTC values are relatively higher at similar mass flux density G level. Another examined characteristic is the influence of MC's internal diameter on the heat transfer coefficient within the similar range of mass flux density level.

Figure 6 presents the dependence of internal diameter on HTC during refrigerants condensation of a) refrigerant R404A, b) refrigerant R407C and c) refrigerant R410A.

Along with the decrease of internal diameter d_w average heat transfer coefficient decreases. The dependence of MC's internal diameter is characteristic for all three types of tested refrigerants.

Fundamental heat characteristics for high pressure refrigerants condensation in single vertical pipe minichannels is congruent to the horizontal minichannels. The results of experimental studies were compared with the results of calculations by other authors, such as Shah, Akers or Tang.

a)



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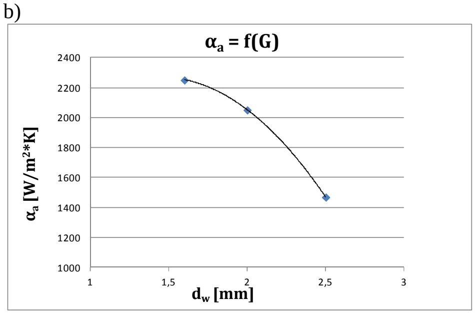

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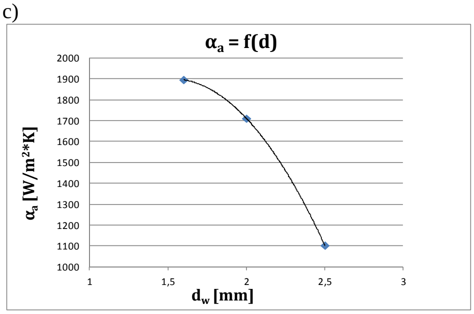

Fig. 6. The dependence of minichannels internal diameter on average heat transfer coefficient α_a during refrigerant condensation a) R404A, b) R407C and c) R410A

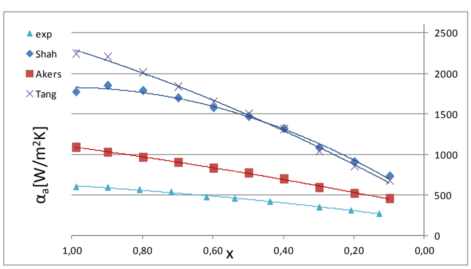

Fig. 7. Comparison of experimental investigation results with calculation results from correlations by other authors[8]. Dependence of heat transfer coefficient on vapor quality, for minichannel with internal diameter d = 2,5 mm and G = 62 kg/m^2·s during R407C refrigerant.

Correlations presented above are mainly used for the condensation process in horizontal channels, as shown in figure 7, there is a large discrepancy between results of experiment and other authors calculations results. Nearby experiment calculations results are Aker's correlation.

5. CONCLUSIONS

1. The heat and flow measurements of R407C, R404A and R410A refrigerants condensation was conducted in pipe minichannels with an internal diameter d = 1.6 mm, 2 mm and 2.5 mm. The study included the determination of average and local pressure drop and heat transfer coefficient in the model conditions.
2. It was found that the heat transfer coefficient depends not only on the size of the inner diameter d of the pipe minichannels but also on the mass flow density G and local vapor quality.
3. Based on the above it is recognized that there is a need for further research and elaboration of own empirical corelation describing heat exchange during condensation of refrigerants in vertical pipe minichannels.

Nomenclature

Symbols

α_a – average heat transfer coefficient, W/m^2·K
α_x – local transfer coefficient, W/m^2·K
d_w – internal diameter, mm
G – mass flux density, kg/m^2·s
q – heat flux, kW/m^2

Acronyms

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

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Biographical notes

Tadeusz Bohdal is the author of more than 300 scientific and technical publications (domestic and foreign), 10 books and over 100 documented studies for business entities. He directed 6 MNiSWW and NCN grants, promoted 8 doctors, is the author of numerous qualification testimonials, research grants and academic papers. He is a member of the Committee of Thermodynamics and Combustion of the Polish Academy of Sciences, Scientific Committee of the monthly “Chłodnictwo” and a scientific consultant of the monthly "Refrigeration and Air Conditioning Technique". He is also an expert of SIMP and an expert at Koszalin District Court in the field of heat and heat measurement and refrigeration. He has promoted more than 200 engineers and engineering masters in the field of Machine Building and Engineering and Agricultural and Forestry Engineering (specializations: Thermal Power Engineering and Refrigeration, Food Processing and Chemical Engineering, Computer Engineering, Engineering and Management, Food Engineering, Food Processing Technology).

Marcin Kruzel received his M.Sc. degree in Economics at the Institute of Economics and Management of Koszalin University of Technology (2008). Until 2010 academic and didactic staff of the Institute of Economics and Management. Currently a PhD student at the Faculty of Mechanics of Koszalin University of Technology. In his work he deals with refrigeration and the economic and technical aspects of using renewable sources of energy. He is an author of 18 papers printed in national and international magazines. Since 2016 works as a scientific specialist at Laboratory of Energetics in Koszalin University of Technology.

Małgorzata Sikora received her M.Sc. degree in Environmental Engineering (specialization: Heating and air conditioning) and next Ph.D (with honors) as well as D.Sc. degree in Machinery Construction and Operation from Koszalin University of Technology, in 2008 and 2011 respectively. Since 2011 she has been an assistant in the Department of Heating and Refrigeration Engineering at the Koszalin University of Technology. Currently she works as an assistant professor in Department of Power Engineering. Her scientific interests concern a heat and flow phenomenon during refrigerants condensation, refrigeration, heat pumps, etc. She has participated a in 4 national research projects, 1 international education project (Tempus Energy). She presenting results of her work at 4 international and numerous national conferences, she published 4 articles in journals from the Philadelphia list (LIST A) and 15 articles in national magazines (LIST B) and 46 papers printed in national and international conferences materials. Dr. Eng. Małgorzata Sikora is also co-author of 1 monograph published in English, and 1 didactic textbook.