Influence of surrounding space on heat transfer effectiveness of refrigerator's condenser

V. Dagilis*, I. Hofmanas**

*Kaunas University of Technology, A. Mickevičiaus 37, 44312 Kaunas, Lithuania, E-mail: vytautas.dagilis@ktu.lt
**Kaunas University of Technology, K. Donelaičio 20, 44239 Kaunas, Lithuania, E-mail: ignas.hofmanas@gmail.com

Nomenclature

c - specific heat, J/(kg K); d - diameter, m; g - gravitational acceleration, m/s^2; H - heat exchanger (condenser) height, m; Nu - Nusselt number; p - pitch, m; T - temperature, K; Ra - Rayleigh number.
Greek symbols: α - heat transfer coefficient, W/(m^2 K); β - coefficient of volume expansion, 1/K; λ - thermal conductivity, W/(m K); ρ - density, kg/m^3; μ - dynamic viscosity, kg/(m s).
Subscript: a - air; c - convective; t - tube; w - wire; cd - conden-sation/condenser.

1. Introduction

Household refrigerators are cyclic operating systems which effectiveness is being strongly affected by the heat exchange parts like evaporator and condenser. Condenser is one of the parts of a freezing system - it dissipates the heat into surroundings. The heat is being absorbed from a freezing zone. The heat transfer effectiveness of a condenser has an impact on power consumption.

The strict requirements for decreasing power consumption of household appliances, the refrigerators in particular, are forcing to look for the new ways to improve their effectiveness. One of the ways is to increase heat transfer effectiveness of the condenser that allows reduce pressure and power consumption.

The wire-tube condensers are ordinarily used in household refrigerators. They dissipate heat by natural convection and radiation. Wire-tube condenser usually consists of horizontal serpentine tube and vertical wires (Fig. 1) and it is fit in the back of the refrigerator. The back side is moved up to the wall with a distance of 0.02-0.05 m that defines a producer of refrigerators (Fig. 2). The producer have to choose what decision is better: more space for condenser or bigger volume of the refrigerating chambers.

In operating ant testing conditions the back wall of the refrigerator usually is kept with the distance of 0.05-0.08 m from the room wall. This limits the natural convection space for heat exchanger. This space has a certain influence on heat exchange effectiveness. Naturally flowing air is being disturbed by the walls of the both side. Thus, the effectiveness of heat transfer is being lowered. All the more the cooling air is coming from compressor shell with much higher temperature. Therefore arising air has a higher temperature than that of surroundings. The air temperature at the top of the condenser is higher by several degrees in comparison to underneath temperature. So heat transfer is lowering at the top of the condenser with decreasing temperature difference and at the same time is increasing
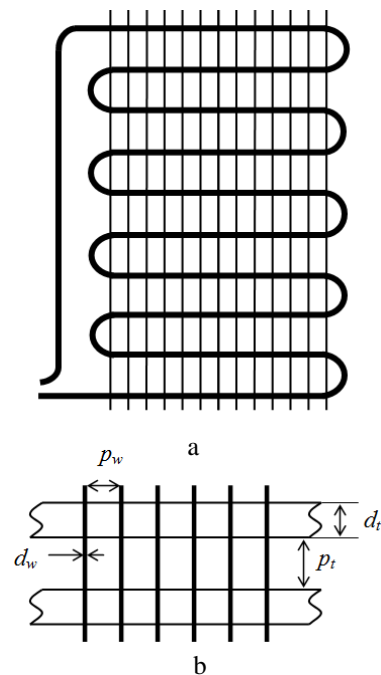

Fig. 1 a - the sketch of the wire-tube condenser; b - geometrical parameters of the condenser
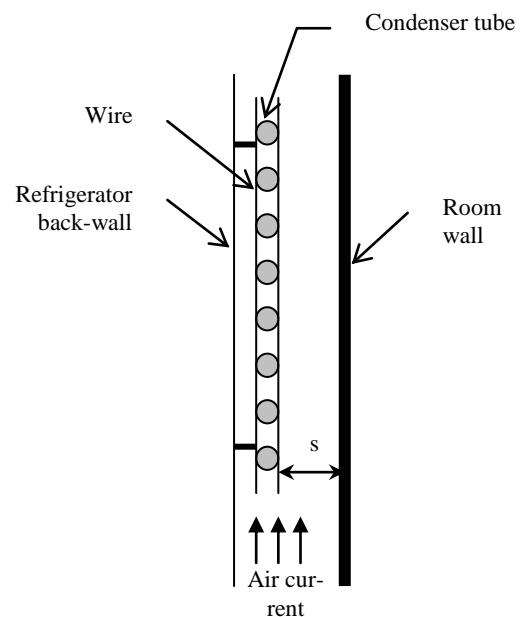

Fig. 2 Air convective space around condenser

because of rising air velocity.

Herewith effectiveness of the condenser is very important to low consumption of metal to produce it. Price

of wire-tube condensers mainly depends on the price of metal because of high developed technology of manufacturing. Furthermore the construction the wire-tube condenser is based on thorough and narrow optimization. It would be irrational to refuse applying such a good product. It would have much more sense to manage more effective heat transfer without changing the main construction of the condenser. It is very important to find out the effect in case when the condenser operates in real conditions, i.e. in a household refrigerator system. Thus experimental methods and numerical simulation related to air convective space around the condenser are represented in the present article.

2. Previous studies

Several studies related to natural convection problem with wire-tube condensers are done in the literature. Tagliafico and Tanda [1] show a detailed experimental research with a wide range of various geometrical forms of the wire-tube condenser. However, the experiments were made without any space limitations by special test apparatus. Hot water was used inside the tubes whereas air was a cooling fluid that took aside heat. The experiments did not hold an isothermal condition, while the condensation process takes place at constant pressure and temperature. It is worth to point out that the temperature difference between the condenser surface and surrounding air does not hold for a real operating conditions of a refrigerator. The research was done with a too large difference of a temperature exceeding 46°C, while in a real condition this difference usually below 15°C. Nevertheless, the wire-tube calculation method of a wire-tube heat exchange was presented in the article. Numerical simulation results were with a good agreement to experimental ones.

Authors of the study [2] made an overview of heat transfer by natural convection as well as made optimization of a wire-tube condenser. The method presented by authors [1] was used in the study. It was demonstrated that the ratio of the best transfer effectiveness to minimum metal consumption is in the ranger of smaller wire diameter and higher number of them. In comparison with conventional construction of a condenser it leads to a lower mass of metal of 6%, while the heat transfer effectiveness increases by 3%. The dominating heat transfer manner of the wire-tube condenser is natural convection, which may exceed up to 65% of the total heat transfer capacity.

Natural convection heat transfer was examined in the articles [3, 4] as well as in the mentioned above. Ameen et al [3] presented the analysis of refrigerant effect on condensation heat transfer as well. R12 refrigerant was replaced by ecological R134a agent. Comparative analysis showed that the change of the refrigerant does not affect the effectiveness of heat transfer from the side of condensation. Therefore the old condensers might be further used. In the Ref. [5, 6] the analysis on wire-tube condenser is presented. However forced convection was used in this case. Natural convection in this case was negligible so the effect of space limiting factor is quit different.

Authors Afonso and Matos [7] presented the investigation related to improvement of refrigerator by decreasing heat inflow from the side of condenser. The authors suggested to cover the back wall of a refrigerator and upper part of the compressor shell by aluminum foil. It was demonstrated that the coverage decreases the temperature

in the inside of the refrigerator by approximately 2K. Such a simple improvement could sufficiently decrease the power consumption.

The surrounding air temperature is an important parameter, since it strongly effects the heat exchange of condenser. Saidur et al [8] proposed the study related to surrounding temperature impact on power consumption of the refrigerator. Room temperature was changed from 14°C to 32°C. According authors about 60-70% of power consumed by the refrigerator is energy for covering heat inflow due to temperature difference as well as energy for increased power requirement due to higher pressure of condensation. Energy losses because of the system cycling and hydraulic losses are not high. About 30% of energy losses are related to friction and electrical losses of the compressor. Hence the most important factor is quality and quantity of insulation of the refrigerator.

Other studies [9, 10] demonstrated a numerical simulation of the space influence on effectiveness of natural and forced convection. Authors [9] have used the method of calculations reported in the previous papers [1, 2] and have compared the theoretical result with experimental one made by authors [9] as well. Authors also draw a conclusion that the heat transfer is sufficiently higher when condenser space is large enough ($s > 200$ mm).

3. Experimental study

The experimental study was carried out in the united laboratory of Kaunas University of Technology and JSC "Snaigė". Household refrigerator RF-31 was used with conventional construction condenser. The main geometrical parameters of condenser are given below:
- outer diameter of the tube (with the coating layer) $d_t = 4.8$ mm;
- inner diameter of the tubes $d_{cd} = 3.35$ mm;
- tube coil pitch $p_t = 50.0$ mm;
- number of the tube coils − 21;
- total length of the tube (including nonfinned part) $l = 16.5$ mm;
- diameter of wire with a coating $d_w = 1.5$ mm;
- wire pitch $p_w = 8.8$ mm;
- number of wires pair $n = 50$ mm;
- total high of condenser $H = 1.20$ m.

The condenser is fitted to the back side of the refrigerator within a distance of 0.03 m from it. The experiments were carried out in a special thermal camera, where the standard temperature of 25°C was kept during all tests.

The refrigerator was moved close to the camera wall with the distance of 0.05 m to the back wall of the refrigerator. Temperature of the surface of the condenser was measured at three different places – upper part, middle and lower part, t_{cd1}, t_{cd2}, t_{cd3} respectively. The t_{cd2} temperature corresponds the condensing temperature the best, because temperature t_{cd1} may be of somewhat higher in the upper part of the condenser and t_{cd3} − of 1 to 3°C lower the lower part. At the upper part the condensing process is not started yet and is finished at the lower part.

The experiment allows investigating the effect of a limiting space for a heat exchange. In Fig. 3 the dynamics of a temperature during the couple of working cycles is shown. It can be seen that mean condensing temperature lowered by almost 2°C when distance between room wall

and condenser was enlarged up to 1m. The Fig. 4 demonstrates the condensing temperature change in case when the condenser is bent from compressor shell. The position enables to avoid the influence of rising heat from compressor. The temperature t_{cd3} decreased much because of subcooling effect. Heat transfer at the lower part of the condenser is of somewhat different in this case. Rising cool air gets out heat from refrigerant liquid but not from gas. The heat transfer of such kind is not isothermal one as it is while refrigerant condensates. As it can be seen the mean temperature t_{cd2} (as well as t_{cd1}) is lower by approximately 2°C compare to previous and by almost 4°C compared to conventional position of the condenser. It means that the heat from compressor has sufficient influence on condensation temperature t_{cd2}.
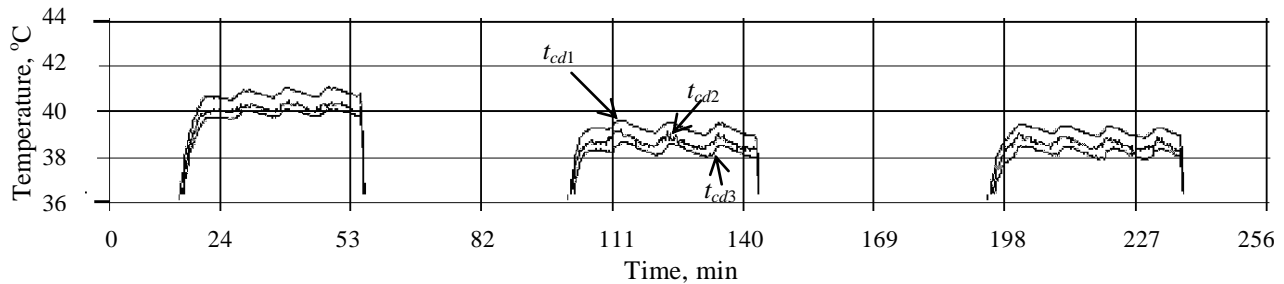
Fig. 3 Condensing temperature with conventional air space (first working cycle) and when it was enlarged (second two cycles)
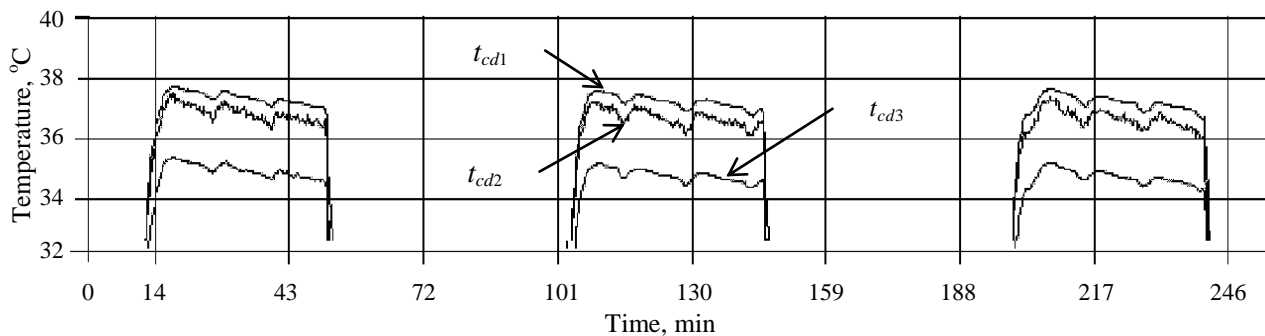
Fig. 4 The change of condensing temperatures when the condenser is bent to avoid heat from compressor shell (as it shown in Fig. 5)

The view of experimental refrigerator with the bent condenser is demonstrated in Fig. 5.

As it can be noticed, the condensation temperature is lower by almost 4°C at the same conditions of surroundings. It means that the escape of heat influence from compressor and larger space for natural air convection gives a possibility to decrease refrigerator's power consumption by approximately 9%.

In order to experimentally determine the average heat transfer coefficient $\bar{\alpha}$ of the condenser, it is necessary to find heat capacity Q. The coefficient $\bar{\alpha}$ is determined as a ratio of the Q to temperature difference multiplied by area f_{cd}, i.e. $\bar{\alpha} = Q / (f_{cd} \Delta t)$. The total heat exchange area is known: $f_{cd} = 0.814$ m^2.

Temperature difference between the condenser and the ambient air was measured at several locations, and the average difference Δt was determined.

In the case of a natural convection conventional space, the condensing temperature was fixed 40.3°C. When the condenser was bent from compressor (see Fig. 5) the condensing temperature was 36.7°C.

The first case is not strictly defined due to the impact of the compressor: operation temperature of the compressor is high and slightly increases the temperature in surrounding of the condenser. Therefore, experiments were carried out with a non limiting space, when the condenser was placed like is shown in Fig. 5
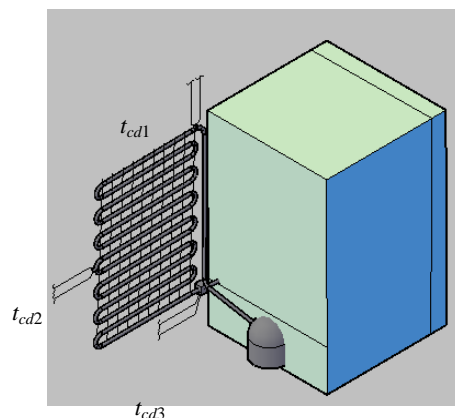
Fig. 5 View of experimental refrigerator with the bent condenser avoiding heat influence from compressor shell

Measurements of ambient air temperature were done at five different heights from the room floor starting 0.3 m for the first height. The temperature sensors were placed in five equal distances at each 0.3 m. The condenser surface temperatures were placed like Fig. 5 demonstrates. The three termocouple sensors were carefully tested. The second sensor t_{cd2} corresponds the condensing temperature

the best. The temperatures around the condenser were slightly different from an average temperature of thermo camera, which appeared due to the air stratification. The average air temperature during the test was equal to 24.1°C and temperature difference in respect of condensing temperature t_{cd2} - Δt = 12.6°C. The heat capacity of condenser was calculated as follow

$$Q = g_{cd}\left(h_{cd2} - h_{cd1}\right) \qquad (1)$$

As we can see, the heat capacity depends on mass flow capacity of refrigerant R600a (isobutene) g_{cd}. The enthalpies h_{cd1} and h_{cd2} are ingoing and outgoing refrigerant enthalpies. The enthalpies were estimated when temperatures t_{cd1}, t_{cd2}, t_{cd3} were measured during the test. The temperature was measured in each 15 seconds. Then the temperature of evaporating dropped to -25°C, the measured temperatures correspond values as follows: t_{cd1} = 37.3°C, t_{cd2} = 36.7°C, t_{cd3} = 34.8°C. Accordance to „CoolPack" software, the gas and liquid enthalpies are h_{cd1} = 606.5 kJ/kg, h_{cd2} = 282.2 kJ/kg respectively.

The amount of circulating refrigerant, i.e. mass flow g_{cd} was determined during special calorimetric test of the compressor. The developed testing rig in the meant united KTU and "Snaigė" laboratory allows carry out calorimetric tests at various conditions. Moreover, it was possible to fix measurements during the test but not only at the end of the calorimetric process. So in this case the capacity and efficiency is being determined at refrigerator's working conditions but not at standard conditions (CECOMAF or ASHRAE). These conditions are sufficient different. For example, the compressors must be tested at standard conditions when temperature of condensation is 55°C, while refrigerator's condensing temperature does not exceed 40°C. Therefore, when the condensation temperature is 36.7°C and evaporating temperature -27.6°C (it corresponds evaporating temperature at refrigerator cycle conditions minus 25°C) the determined HTK80AA compressor cooling capacity Q_o = 109 W and effectiveness 1.92. Evaporating temperature -27.6°C was estimated during calorimetric test of the refrigerator when hydraulic losses of the suction line were evaluated. During refrigerator test conditions when evaporating temperature exceeds -25°C precisely, the pressure inside the compressor shell is estimated by the special pressure-gauge. According the „CoolPack" the evaporating pressure at -25°C is 0.587 bar. The difference between 0.587 bar and measured pressure is due hydraulic losses in the trunk-line between evaporator and compressor. So estimated pressure value of 0.522 bar corresponds the boiling temperature equal to minus 27.6°C.

The amount of circulating refrigerant g_{cd} is determined as

$$g_{cd} = Q_o \,/\left(h_1 - h_4\right) \qquad (2)$$

where h_1, h_4 are enthalpies of vapor and liquid at the compressor testing condition, which corresponds refrigerator working conditions, i.e. gas suction temperature t_1 = 22°C and liquid temperature $t_4 = t_{cd3}$ = 34.8°C.

Gas suction temperature (22°C) was measured during the standard test of the refrigerator. Within these conditions we have enthalpies: h_1 = 592.5 kJ/kg,

h_4 = 282.2 kJ/kg and amount of circulating refrigerant g_{cd} = 0.000351 kg/s. Then heat capacity of the condenser (according Eq. (1)) is Q = 113.9 W.

According to the equation $\bar{\alpha} = Q/\left(f_{cd}\Delta t\right)$, the coefficient of heat transfer is $\bar{\alpha} = 11.1$ W/(m² K). This quantity is slightly smaller than that theoretically estimated (see previous author's article, Ref. [1]), which was $\bar{\alpha} = 12.6$ W/(m² K).

4. Numerical calculations

Numerical simulation was carried out in order to take in more thoroughly natural convective phenomenon when wire-tube heat exchanger is examined. The simulation was accomplished by the „Ansys CFX" software. First of all temperature distribution on the surfaces of wire-tube condenser that allow analyze fin (wire) effectiveness visually was determined.

Air heat transfer coefficient $\bar{\alpha}$ makes an impact on the fin effectiveness [1, 3] and at the same time the difference of the temperatures between the wire and air, i.e.

$$E = \frac{\left[tanh\left(mp_t/2\right)\right]}{\left[mp_t/2\right]} \text{ and } m = \sqrt{\frac{4\alpha}{\lambda_w d_w}} \qquad (3)$$

The difference of temperatures between the outer surface of condenser and air around it may decrease or increase local heat transfer coefficient α as well as average one $\bar{\alpha}$. Modeling the above mentioned software allowed us to get the thermovision view of the surface of condenser (Fig. 6).

The temperature of wires is lower compare to tube temperature because of fin efficiency is lower than unity. Temperature decreases across the edge of the wire from 38.8°C to 35.9°C. Meanwhile the tube temperature varies in much narrow interval: from 38.8°C at a contact with tube to 39.2°C where the wires are not attached. In the middle of a bend the temperatures increases up to 39.7°C (Fig. 6, b), while condensing temperature inside is 40.0°C.

The average temperature of the condenser is calculated as well. At the done conditions the average surface temperature is 37.4°C.

The calculation of natural convection heat transfer requires thermal and physical fluid properties that form Gr but not Re numbers. However, for the local heat transfer calculation the estimation of local velocity is necessary. The used program allows calculate the local air velocity, although fluid moves at a very low velocity. Thermovisional image of numerical simulated air velocities is presented in Fig. 7 (see the right side of the picture). As can be seen the horizontal tubes make slower velocity of air. Velocity reaches 1 m/s in the distance of 0.02-0.03 m from vertical surface and about 0.6 m/s closer to it. Analogous results to this present Berianche at al [10]. Authors give the thermovisional image of simulated velocities of about 0.5 m/s at small cross-flow areas of a heat exchan-ger.

The left side of Fig. 7 presents air temperature spread around vertical surface of the condenser. The figure demonstrates the thermovisional image of numerical simulation results. In this figure it is seen that the temperature towards the middle of the condenser varies only slightly.

The temperature of air varies sufficiently by vertical direction. Temperature increases by near 6°C (experimental measures give increase of nearly 5°C in case when heat influence of the compressor is not eliminated). Obviously, the increase of air temperature is due to heat exchange. Higher air temperature reduces heat transfer coefficient from external side of the condenser. The heat transfer coefficient α depends on temperature difference between the surfaces and fluid because $Ra = GrPr$ number determines the α in natural convection case.
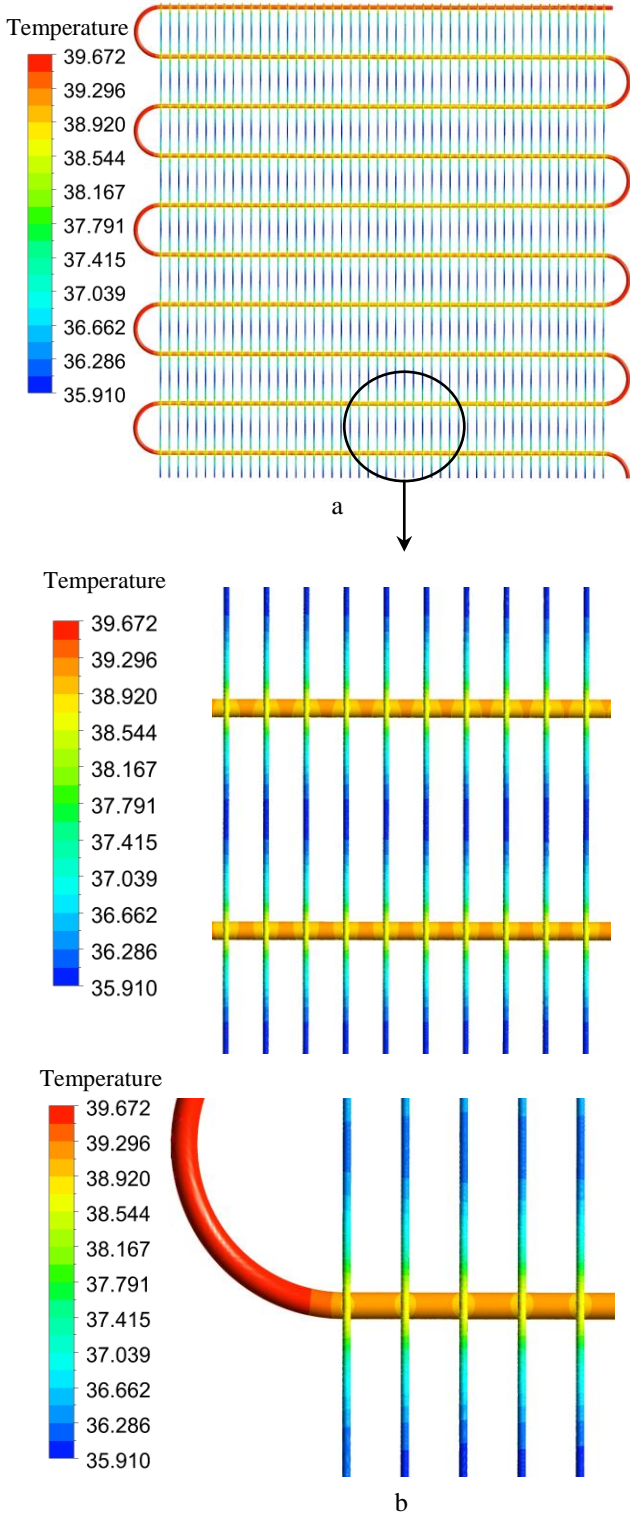

Fig. 6 a - view of temperature distribution of the condenser; b - zoomed fragments of condenser
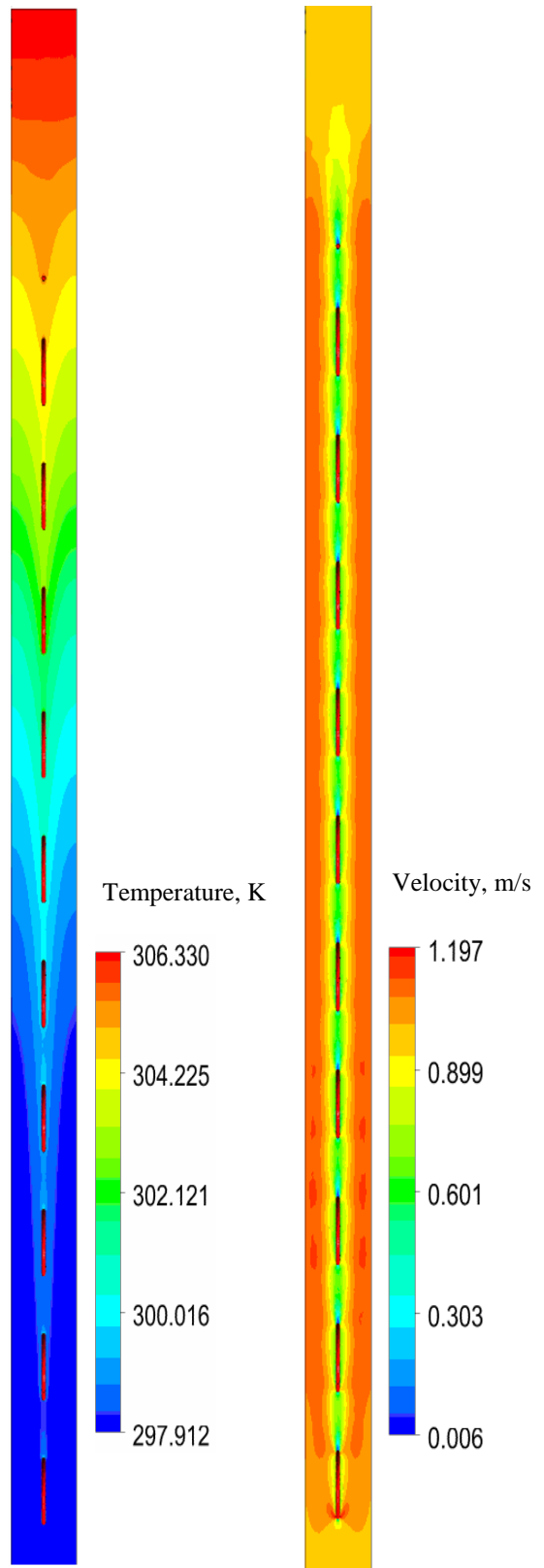

Fig. 7 Thermovisional image of numerical simulation of temperatures (left) and velocities (right) around condenser. Condensing temperature 40°C, condenser height 1.2 m; condenser space 0.06 m

The expression of the Rayleigh number Ra is

$$Ra = \left(\frac{\beta_a \, \rho_a^2 c_a}{\mu_a \, \lambda_a} \right)_c g \left(T_{cd} - T_a \right) H^3 \qquad (4)$$

The complete equation for the calculation of the heat transfer coefficient is as follow [1, 2]

$$Nu = 0.66 \left(\frac{Ra\,H}{d_t} \right)^{0.25} \times$$

$$\times \left\{ 1 - \left[1 - 0.45 \left(\frac{d_t}{H} \right)^{0.25} \right] exp\left(-\frac{s_w}{\varphi} \right) \right\} \quad (5)$$

where

$$\varphi = \left(28.2 / H \right)^{0.4} s_w^{0.9} s_t^{-1.0} +$$

$$+ \left(28.2 / H \right)^{0.8} \left[264 / \left(T_{cd} - T_a \right) \right]^{0.5} s_w^{-1.5} s_t^{-0.5} ;$$

$$s_w = \left(p_w - d_w \right) / d_w ; \quad s_t = \left(p_t - d_t \right) / d_t .$$

Then heat transfer coefficient is

$$\alpha_c = \frac{Nu \lambda_{ac}}{H} \quad (6)$$

Eq. (5) is developed for wire-tube type heat exchangers with the vertical wires. It assesses all geometric factors of exchanger such as condenser height H, distance between tubes p_t and wires p_w, tube and wire diameter – d_t and d_w respectively. Analysis of the equation proposes an interesting distinction concerning the dependence $\alpha = f(H)$. Heat transfer theory proposes that the higher H, the lower heat transfer coefficient α.

However Eq. (5) does not approve this relation. Nevertheless the accuracy of the Eq. (5) is satisfactory compared to other equations which could be used for vertical plane surfaces or vertical tubes [11]. Heat transfer coefficient calculated by (5) gives $\bar\alpha = 12.6$ W/(m^{2}K). Two different experimental investigation results are $\bar\alpha = 11.1$ W/(m^{2}K) and $\bar\alpha = 10.7$ W/(m^{2}K) [11]. Meanwhile the other theoretical equations (for vertical wall or vertical tube) give much higher value of the $\bar\alpha$ [12, 13] including Zhukauskas theory [14].

The results of simulation enable to analyse the problem of increasing effectiveness of the condenser. It is clear that the air from the flank of the condenser is not involved in heat transfer process. Air temperature in the flank has the same temperature as the room temperature. For making transfer more effective it would be rational to involve cool air from the flanks. This can be done by using forced of mixed convection, for example. Somewhat faster increasing air would create higher dynamical pressure and this would involve the side air in to the heat exchange zone. A very small, cheep and low power using fan can arouse mixed convection when it is directed upwards. The mixed convection is more rational in sense of fin efficiency. Compared to fully forced convection the mixed convection increases $\bar\alpha$ only by 20-30% and the fin effectiveness of the wire decreases because of this by 10-15% (see Eq. 3). It means that material of the wires is employed quit effectively [11].

The solution of raising effectiveness by adapting a small fan that arouses mixed convection is novel. The first experiments were done with employing fan of 0.5 W of mechanical power. The condenser temperature was decreased by 2°C. Power consumption of the refrigerator lowed by 4% with the estimation of the fan power con-sumption. The more detailed studies of wire-tube condenser with employed mixed convection will be presented in the dr. thesis.

5. Conclusions

Operating ant testing conditions of the refrigerators require keeping the distance between condenser and room wall of 0.05-0.08 m. This limits the natural convection space for heat transfer. This space has a certain influence on heat exchange effectiveness. Naturally rising air is being disturbed by the walls of the both side. The producers of the refrigerators must choose what a decision is better: more space for condenser or bigger volume of the refrigerating chambers.

Experimental investigations of the influence of the condenser space for natural convection show that the external heat transfer coefficient could arise by 14% in case when the convection is fully free and by 7-9% if the space is enlarged by 0.3 m. The first case is not allowed because of refrigerator running instruction. The second case is complicated due to defined value of the refrigerator deepness.

Heat transfer and aerodynamic numerical simulation demonstrates natural convection weakness in sense of increase heat transfer effectiveness. Cooler air from flank is not involved into the region of heat transfer. It could be seen from the thermovisional image of numerical simulation of temperatures and air velocities around the condenser. The investigation presented in the article leads to novel solutions – applying of a mixed convection, which allows involving the air from the flaks of refrigerator.

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BUITINIO ŠALDYTUVO VIRBALINĮ KONDENSATORIŲ SUPANČIOS ERDVĖS POVEIKIO ŠILUMOS MAINAMS TYRIMAS

R e z i u m ė

Buitinio šaldytuvo kondesatoriaus šilumos mainų efektyvumui įtakos turi jį ribojanti erdvė. Darbe pateikti virbalinį kondesatorių supančios erdvės įtakos jo šilumos mainams eksperimentiniai ir skaitiniai tyrimai. Eksperimentai buvo atlikti realiomis sąlygomis, t.y. kondesatoriui dirbant šaldytuvo sudėtyje. Nustatytas vidutinis šilumos atidavimo koeficientas kondesatoriui esant neapribotoje erdvėje. Skaitinio modeliavimo būdu gauti vaizdai leido vizualiai atlikti kondensariaus šilumos mainų analizę. Tyrimų rezultatai parodė, kad erdvės įtaka šilumos mainams gali siekti iki 14%. Be to, gauti rezultatai sukūrė prielaidas naujam sprendimui – efektyvesniam kondensacijos šilumos nuvedimui reikia sukurti nedidelį dirbtinį oro judėjimą pavymui natūraliai kylančiam. Taip gauname mišrios konvekcijos atvejį, efektyviai derantį esamai kondensatoriaus konstrukcijai.

V. Dagilis, I. Hofmanas

INFLUENCE OF SURROUNDING SPACE ON HEAT TRANSFER EFFECTIVENESS OF REFRIGERATOR'S CONDENSER

S u m m a r y

Surrounding space around condenser of household refrigerator has sufficient influence on heat transfer efficiency. The experimental and numerical investigations are presented in the article in order determine the influence. Experimental investigation is carried out with condenser at real conditions when it operates in refrigerating system. The average heat transfer coefficient is determined when condenser space is not limited. The results of numerical simulation give a possibility to do analysis of heat transfer efficiency. The results demonstrate that condenser space impacts the heat transfer by up to 14%. Moreover, the results allowed doing a novel solution for more effective heat transfer, i.e. the creation of small artificial air flow upwards by the natural direction. Thus the combined convection case is getting, which is in good accordance with the conventional construction of the condenser.