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LONG NON-ADIABATIC CAPILLARY TUBE IN HOUSEHOLD REFRIGERATOR

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ABSTRACT

Non-adiabatic flows of R134a in a household refrigerator in a capillary tube were investigated in a frame of their mathematical models for one-phase and two-phase modes. Results of calculations allows to formulate and elaborate a technical decision concerning a capillary tube length. In this decision a minimal necessary pre-cooling of a refrigerant before the capillary tube has been provided and then the capillary tube's channel was placed in a channel of refrigerant flow in the evaporator and in a inlet tube of the compressor. Thus, the refrigerant flow in the capillary tube quickly cools down to the boiling temperature and one-phase flow along full length of tube with the refrigerant best quality in outlet of tube occurs. The elaborated technical decision has been realized in two-chamber refrigerators. They were tested by the standard method in a climatic thermochamber. Results of tests showed a more equal temperature field along height of the freezing chamber and this fact provides less power consumption than power consumption of similar freezers with classic short capillary tubes.

NOMENCLATURE

T	Temperature, K or ^{0}C
P	Pressure, Pa
ρ	Density, кg/m^3
$\dot{m}$	Mass flow rate, кg/s
D	Capillary tube diameter, m
C_v	Isochoric specific heat, J/(кg·K)
v	Velocity, m/s
K	Heat transfer coefficient, W/(m^2·K)
g	Gravity acceleration, m/s^2
Θ	Capillary tube inclination angle
h	Specific enthalpy, J/ kg
X	Refrigerant quality
ξ	Darsy friction factor
Z	Distance from capillary inlet, m

Subscripts

0	Inlet
V	vapor
L	liquid
W	wall

INTRODUCTION

This work was made in a frame of a general problem of conversion of an association to producing of household refrigerators working with new alternative refrigerants [1].

Parameters of the capillary tube like a restriction device influence essentially on specific power consumption of refrigerators. Also a correlation between length of cooled and non-cooled parts is important like a characteristic of a non-adiabatic flow of refrigerant along the capillary tube. However, it is known that influence of singles factors, capillary tube's parameters, conditions of heat exchange with environment on power consumption was not emphasized. Concerning this fact, selection of capillary tubes for household refrigerators often makes using an empirical base. On the other side, there is interest in elaboration of effective mathematical models of a refrigerant flowing in capillary tubes.

The best quality of the refrigerant outlet the capillary tube may be obtained if, for example, conditions for refrigerant boiling along the capillary tube will be not provided. It is possible if the capillary tube is cooled by immersing in the boiling refrigerant flow in the evaporator. On the other side, a necessary pressure fall only at one-phase refrigerant flow may be obtained in long capillary tube which would be longer than ones with two-phase flows. How long is common length of capillary tube and is it comparable with a necessary length of the evaporator channel and the inlet channel of the compressor in that case? Is it possible to make an applicable technical decision? How much and by what reasons this decision may be energetically effective? Deciding of this questions is made by as theoretical and as experimental methods in this work.

INVESTIGATION METHODS

A mathematical model of non-adiabatic refrigerant flow along a capillary tube (conventionally disposed along Z axis) is the Cauchy problem for a system of ordinary differential equations:

$$\frac{dp(z)}{dz} = -\xi_{WL} \cdot \frac{\rho_L v_L^2}{2D} - \rho_L g \sin(\Theta);$$

$$\frac{dT(z)}{dz} = \frac{1}{\rho_L c_{VL}} \cdot \frac{dp(z)}{dz} + \frac{k_W(z)}{\dot{m} c_{VL}} \cdot \left[T_0(z) - T(z)\right] + \frac{g \sin\Theta}{c_{vl}}. \quad (1)$$

An integrating this system at given values of p(0), T(0) in the inlet z=0 of the capillary tube provides to calculate values of temperature T(z) and pressure p(z) of the refrigerant along the tube, to compare values p(z) with values of vapor pressure $p_S(T)$ and to control a refrigerant boiling point coordinate and a beginning of two-phase flow evolution in the capillary tube.

The temperature $T_0(z)$ of medium at outside of the capillary tube and corresponding heat transfer coefficient $k_W(z)$ over surface of capillary tube may be assigned like a sectionally continuous function. In parts of the tube disposed in atmospheric air or the boiling refrigerant $T_0(z)$ may be assigned like a constants. However, for part of the tubes disposed in the inlet tube of the compressor it is necessary to make a coordinate calculation of $T_0(z)$ and T(z) in a frame of analysis of a cold heat exchangers. Calculation of $k_W(z)$ makes by using of available information from a literature on heat exchanger coefficients, particularly at refrigerant boiling inside tubes and in circular gaps.

Mathematical modeling of a two-phase refrigerant flow was made in a frame of well-known two fluid approximation, using values temperatures $T_L(z)$ and $T_V(z)$ of phases, their velocities

$V_L(z)$ and $V_V(z)$, pressure p(z) and volume vapor void y(z). Five modeling differential equations considering a non-adiabatic flow were made on base of the axiom of mass, impulse and energy conservation. With accuracy up to terms considering the non-adiabatic flow this equations principally coincide with equations [2]. The sixth equation was made for vapor void or refrigerant quality X(z) of the flow:

$$X(z)=\left\{h_v-h_L+\frac{v_V^2}{2}-\frac{v_L^2}{2}\right\}^{-1}\cdot\left\{h_0+\int_0^z\frac{k_W(y)[T_0(y)-T(y)]}{\dot{m}}dy-h_L-\frac{v_L^2}{2}\right\}, \qquad (2)$$

where h_0 - enthalpy of refrigerant at p(0) and T(0) inlet the capillary tube. A relation equation between y(z) and x(z) with (2) allows to close the system of differential equations. General problems during integrating this system connect with achievements for represent an over-phases interaction as in [2].

Integrating corresponding systems of differential equations at different beginning conditions and conditions of heat exchange was made for flows of R134a. Thermophysical properties of R134a as for one-phase as on saturation curve were calculated by using equation [3]. Probation of mathematical models and their calculation algorithms was made by comparison calculated values p(z), T(z) with experimental literature information of different authors. Base calculations were made for capillary tubes with inner diameters D=0.80~0.85 mm, inner rough $(0.4\sim0.5)\cdot10^{-6}$ m and external diameter 2.1 mm.

A predesined evaporator for freezing chamber of a refrigerator - freezer had a channel required length of capillary tube from 12 to 14 m. Length of inlet tube of the compressor with inner diameter 5 mm was approximately 1.8 m. Thus, common length of the capillary tube was approximately up to 17 - 18 m.

An object for experimental researches was the two-chamber refrigerator - freezer with common volume of chambers 0.23 m^3 and 0.1 m^3 freezing chamber volume. Cooling chamber had a roll-welding evaporator and freezing chamber had a tube-sheet evaporator.

For testing in the climatic thermochamber two groups of sample refrigerator - freezers were prepared:

- with a short capillary tube having volume consumption from $4.4\cdot10^{-3}$ m^3/min. to $6.4\cdot10^{-3}$ m^3/min.;
- with a long capillary tube having volume consumption from $1.8\cdot10^{-3}$ m^3/min. to $3.8\cdot10^{-3}$ m^3/min.

In the second case the capillary tube were disposed in the tube-sheet evaporator along full its length up to plugged end and refrigerant in the capillary tube flow with against direct to the refrigerant flow in the evaporator.

In the refrigerator - freezer have been used a modify compressor XKB 6.65-1M described in [1] with lubricant viscosity 22. R12 and R134a has been used for experimental samples. After ending of using one refrigerant the apparatus carefully cleaned, pumped out and charged by another refrigerant.

During tests of refrigerator - freezers in the climatic thermochamber daily power consumption was determined. Also distribution of temperature along the height of freezing chamber charged by model packets was determined using five thermocouples. Experiments were made using 18 samples of refrigerator - freezers.

RESULTS

Probation of the mathematical problem and realizing calculations by comparison calculated values p(z), T(z) and experimental information of different authors demonstrates that elaborated procedures allow satisfactorily to represent experimental data. In the further model calculations an influence of different factors on characteristics of the flow in the capillary tube researched. Dependencies of minimal pre-cooling of refrigerant before the capillary tube on the length of inlet non-cooled part of tube were obtained.

Fig. 1 demonstrates a typical character of pressure and temperature changes at one-phase non-adiabatic flow in the capillary tube with diameter D=0.83 mm, rough $0.4 \cdot 10^{-6}$ m, common length 15.7 m, inlet refrigerant pressure 1072 kPa and temperature 29.5 ^{0}C (pre-cooling 4 ^{0}C). Mass consumption is $1.030 \cdot 10^{-3}$ kg/s, length of inlet tube of the compressor 1.8 m. Analysis of uncertainties of calculations demonstrated that uncertainties of determination of capillary tube common length are about 3% and uncertainties of determination of refrigerant consumption at given capillary tube length are 1~5 %.

Exactly capillary tubes having near above listed characteristics were applied in samples of refrigerator - freezers of the second group.

The first experimental results obtained at determining daily power consumption was finding out a power consumption minimum dependence on the volume consumption capability of the capillary tube. This minimum was more significant for the first group of refrigerator - freezers and less significant for the second group. Besides, this minimum occurred as working with R12 as working with R134a.

On the other side, a comparison of values of daily power consumption at volume consumption providing a minimal power consumption in each group of refrigerator - freezers demonstrated that objects with the long capillary tube have 12 % power consumption less at working with R12 and 18 % less with R134a.

Thus, the non-adiabatic restriction of refrigerant in the outside cooled long capillary tube provides improvement of energy characteristics of a refrigerator - freezer. From thermodynamical point of view, this fact is impossible. Explanation of this fact possibly connected with cycle working of domestic refrigerators and a high quality of the refrigerant inlet the evaporator. Fig. 2 possibly demonstrates this fact, where temperature fields along height of freezing chamber for both group of refrigerator - freezers are presented. In a case of the long capillary tube the temperature field was more equal. This fact provides less heat transfer from outside, and longer time of stopping of the refrigerator - freezer machine.

CONCLUSION

Elaboration of a new modify compressor [1] and an evaporation system including a long capillary tube provided to decrease daily power consumption from 2.01 kW at hour down to 1.37 kW at hour. Such decrease is equal to 32 %.

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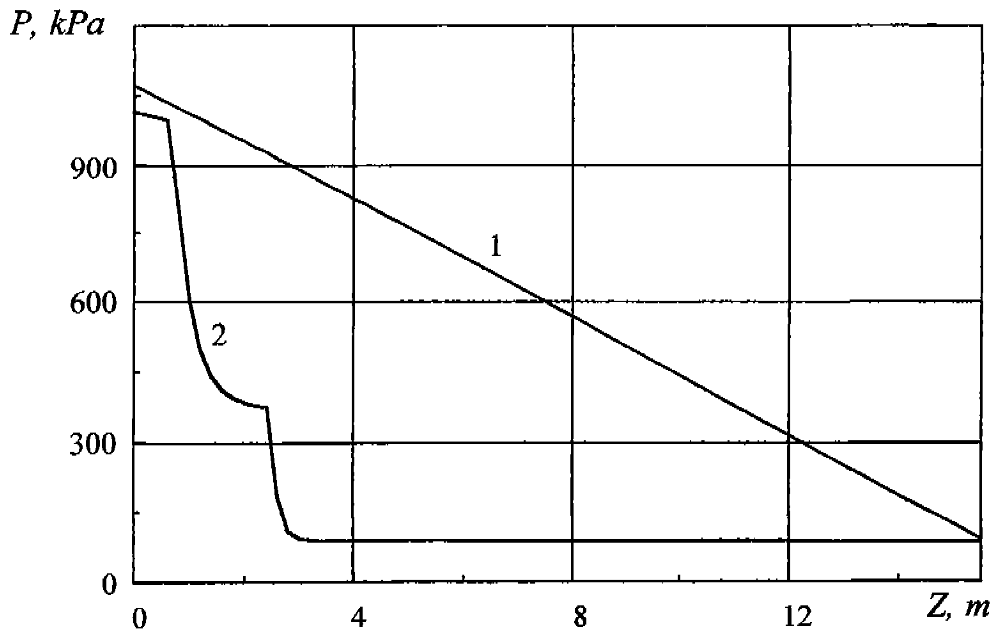

Fig. 1: Typical change of pressure at R134a flow in a long cooled capillary tube: 1- refrigerant pressure; 2 - vapor pressure corresponding to refrigerant temperature.

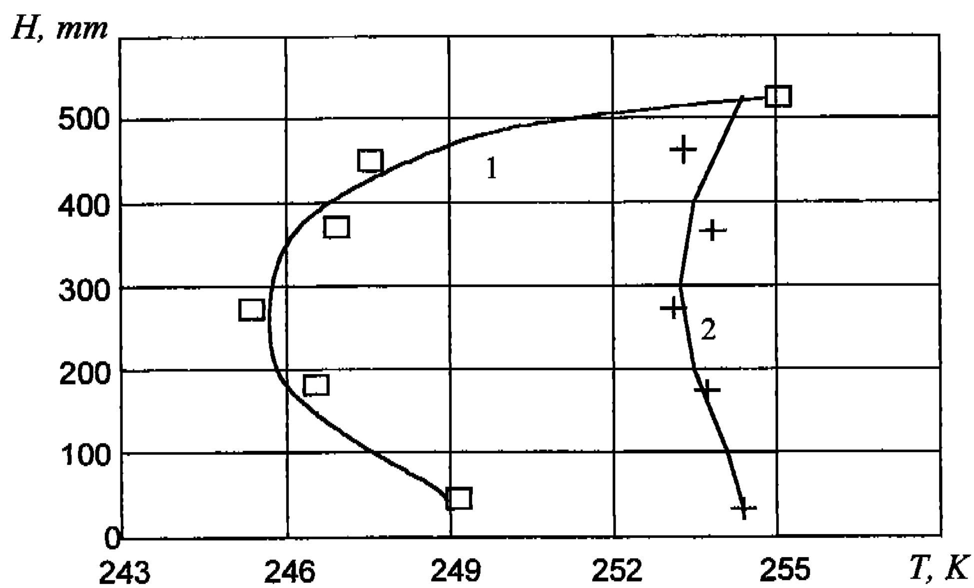

Fig. 2 : Temperature distribution along the height of freezing chamber: 1 - short capillary tube; 2 - long capillary tube (□, + - experimental data).