Influence of the capillary pipe geometry on the energy efficiency of household refrigerator

Hristo Hristov[1], *Apostol* Simitchiev[1*], and *Donka* Stoeva[1]

[1]University of Food Technologies, Plovdiv, Bulgaria, Department of MAFFI, Maritza 26, blvd.

Abstract. The energy efficiency of household refrigerating appliances was measured according to EN ISO15502. A Liebherr IKP 1650 built-in Domestic refrigerator is used to determine the influence of the capillary pipe geometry. The temperatures at the injection site of the refrigerant, the inlet and outlet of the evaporator receiver were measured. The obtained results show that there is no universal optimal geometry of the capillary pipe. There is no element of the refrigeration cycle whose change has no effect on the refrigerant flow rate passing through the capillary pipe. A change in the geometry of capillary pipe affects all other components of the refrigeration cycle and can greatly improve or impair the efficiency of the refrigerator. The potential for switching from one to two different geometries of the capillary pipe is approximately 8% improvement in energy consumption. Possible implementation should take into account the potential savings ~ 8% and the life cycle of the particular appliance (~ 15 years). For the particular measured device, the consumption per year is about 75kWh, which should take into account that the device itself has a low volume and low energy consumption.

1 Introduction

Anyone who nowadays would like to be engaged in scientific research on throttling capillary pipes in refrigeration technique should ask themselves if there is anything new to be discovered in this field. Capillary pipes have been used as an expansion body in refrigeration since the 1920s to the present day [1, 2]. The first scientific publications were published in 1946 and the first practical optimizations were made in 1957 [3]. Studies in this area are of great interest to many scientists involved in refrigeration. The reason for this is the huge use of the capillary pipe – till 1953 only 300,000 refrigerators per year were produced. Today only in Plovdiv, Bulgaria the factories produce 3,000 daily or 1 million per year. Each household has at least one refrigerator or air conditioner in which the refrigerant is throttled by a capillary pipe.

Capillary pipes are currently the cheapest mass-produced expansion part in refrigeration [4]. This is mainly due to the simple construction – usually 2 to 6 meters long copper pipes with an inside diameter of 0.5 to 2 mm. They have no moving parts or are sensitive to external influences and, unlike conventional expansion valves (thermostatic or electric)

*Corresponding author: asimitchiev@gmail.com

have much higher reliability [5]. However, despite its simple design, as a result of the regenerative heat exchange with suction pipe used in domestic refrigerators, the capillary pipe is one of the most complex elements [6, 7]. The complexity stems from the ever changing ambient temperature and heat load of the system. This leads to a number of problems in their design, which in many cases prove ineffective.

1.1 Thermodynamic processes during throttling in a capillary pipe

The thermodynamic processes in a capillary pipe are most clearly illustrated by a diagram of temperature and pressure Fig.1 [8, 9]. The depiction assumes that the refrigerant leaves the condenser in a state of cooled liquid and that there is no heat exchange in the regenerative heat exchanger.

The capillary pipe connects the condenser to the evaporator. The refrigerant enters it with condensation pressure and at the outlet it exits with approximate evaporation pressure. At the inlet, it may be a cooled liquid, a saturated liquid or a vapor mixture. Due to the sharp shrinkage of the cross section at the inlet of the capillary pipe, the refrigerant undergoes pressure loss. In the diagram shown in Fig.1, this phenomenon is highly overexposed and practically the losses are within a few millibars, which is difficult to prove with measurements.

However, with the slight cooling in the domestic refrigerators, it plays a significant role as steam bubbles can form, which can lead to noise and flow instability. Therefore, refraction of the capillary pipe should be avoided. The picture presented at Fig.2 shows a refraction of a capillary pipe.

By using coils around the filter dehydrator, heat exchange is obtained between the refrigerant in the filter dehydrator and the capillary pipe. This leads to the presence of steam bubbles and thus loss of refrigeration capacity. The coils around the filter dehydrator are classified as an energy-innefficient design solution.

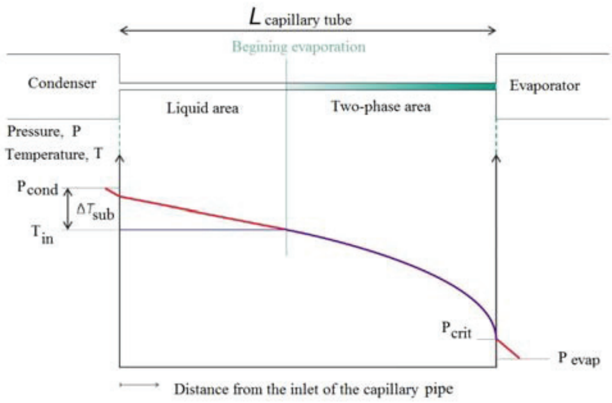

Fig. 1. Diagram of temperature and pressure in a capillary pipe in the absence of regenerative heat exchange.

After entering the capillary pipe, the refrigerant in the liquid condition is throttled approximately isothermic Fig.1. The pressure decrease in proportion to the length. Fluid heating around 0.6 K/m due to dissipation could be expected here. In real conditions, the pressure at which evaporation begins is reached well below the saturation pressure. An overheated liquid is formed, as shown in Fig.1 with the intersection of pressure and temperature lines. The moment the evaporation begins leads to a sharp decrease of the overheating. In the next two-phase flow, steam and liquid are in thermos-dynamic equilibrium due to the intense mixing. Lowering the pressure causes the liquid to evaporate, which is why the average density decreases and the speed of the two phases increases (Table 1). This leads to two mutually reinforcing effects: due to the acceleration, the pressure decreases and the flow at a higher speed has greater pressure losses (friction). These two effects lead to an increasingly upward pressure line. The mass flow rate and density of the refrigerant reaches critical value Pcrit, which depends only on the input parameters. Subsequent lowering of the pressure after the capillary pipe cannot lead to an increase of the mass flow rate. Therefore, when considering the processes in the capillary pipe, the evaporation pressure does not play a role.

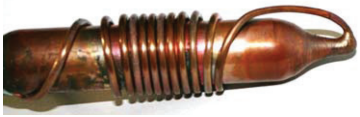

Fig. 2. Refraction and winding of the capillary pipe on the filter dehydrator.

When cooled liquid enters the capillary pipe, the fluid area increases. Unlike the two-phase region, the relative pressure losses in the liquid region per unit length of the capillary pipe are much smaller. In other words: there is an inverse correlation between the mass flow rate in the capillary pipe and the relative fraction of the two-phase region.

The heat exchange between the suction and capillary pipes in the regenerative heat exchanger leads to an increase in the specific cooling capacity, but also to an increase in the energy consumption of the compressor. In the case of domestic refrigerators, the specific operation of the compressor at R600a is little less dependent on the suction temperature, and the suction gases from the compressor are warmed almost to ambient temperature due to the lack of insulation on the suction pipe. This is to avoid hydraulic shock. Refrigeration capacity that cannot be supplied to the capillary pipe by the suction pipe is lost in the area after the heat exchanger and the compressor inlet into the environment.

Table 1. Refrigerant parameters at inlet and outlet of adiabatic capillary pipe (homogeneous flow).

Parameter	Inlet ofcapillary pipe	At the beginning of two-phase region	Outlet of capillary pipe
Vapor content%,	0	5	23
Density ,kg/m³	551	119	12
Speed ,m/s	2	8	80

2 Materials and methods

2.1 Energy efficiency and standards

The European certificate for energy efficiency of large domestic appliances was introduced in 1996. This resulted to an increase in production of highly efficient refrigerators.

Domestic refrigerators intended for the European market were measured in accordance with EN62552; EN153; EN ISO15502. After 2020 to assess energy efficiency and better compare different products worldwide (incl. Australia, China, Europe) we have moved towards a unified so-called Global standard IEC 62552 (Global Standard). For ISO 15502:2005 measurements, only one ambient temperature (25 ^{o}C) is required to determine energy consumption. Systems marketed in Europe are now optimized for these conditions only. The main feature that matters in the choice of the geometry of the capillary tube is the fact that in the new standard, energy efficiency will be measure at 2 ambient temperatures (16 ºC and 32 ºC) as a opposed to the current 25 ºC. Summary of the difference between the two standards is shown on Table. 2.

Table 2. Standards.

Standard	EN62552; EN153; EN ISO15502	IEC 62552 (Global Standard)
Measuring elements	Refrigerating part: PT100	Refrigerating part: PT100
	Chiller part: packages	Chiller part: PT100
	Freezer part: packages	Freezer part: PT100
	Wineries: packages	Wineries: PT100
Ambient temperature	25°C	16°C and 32°C
Target temperature	Refrigerating part: $T_M = 5{,}0°C$ (Average of three PT100)	Refrigerating part: $T_M = 4{,}0°C$ (Average of three PT100)
	Chiller part: $T_{MAX} = 3{,}0°C$ (maximum for all packages)	Chiller part: $T_M = 2{,}0°C$ (maximum of three PT100)
	Freezer part: $T_{MAX} = -18°C$ (maximum temperature of all packages)	Freezer part: $T_M = -18°C$ (Mean average of all PT100)
	Winery: $T_M = 12°C$ (Average of all packages)	WT: $T_M = 12°C$ (Average of three PT100)

2.2 Measuring energy efficiency

The energy efficiency class shows the energy consumption of the refrigeration unit when compared to another with the same volume and purpose. It is determined in accordance with Regulation (EC) № 643/2009 of the European Parliament. The so-called energy efficiency index is used to determine the energy efficiency class. It represents the ratio of actual consumed electricity per year to standard consumption per year [10, 11]

$$EEI = \frac{AE}{SAE}.100 = \frac{E_{24h}.365}{V_{eq} + M + N + CH}.100 \quad ,\% \tag{1}$$

Where AE – actual consumed electricity per year, kWh; SAE – standard consumption per year, kWh; Veq – equivalent volume of domestic refrigerating appliance; M, N – values selected from tables, depending on the category; CH -50 kWh/per year in the compartment for perishable products with a volume of at least 15 liters.

2.3 Nitrogen flow rate

Determining the inside diameter of a capillary pipe at a given pressure drop is a difficult task. Even the slightest deviation of the inner diameter of the tolerances has a noticeable effect. For this reason, in practice, instead of the inside diameter, the most commonly used is the nitrogen (N_2) flow rate, under certain initial conditions [12]. The methodology for measuring (N_2) flow rate is defined in DIN8905 Part 3. In the measurements, despite the different physical processes of refrigerant throttling and a practically ideal gas (such as N_2), conclusions can be drawn regarding the linear relationship between the flow rates of N_2 and the actual flow rate of the refrigerant in the system, subject to the initial conditions:

1. Increasing the N_2 flow rate of a capillary pipe by about 10 %, by reducing the length or choosing a larger internal diameter, results in an increase in the refrigerant flow rate by about 10 %. Unless the length of the regenerative heat exchanger changes.

2. If the capillary pipe is replaced by a shorter one, which, however has the same nitrogen flow rate, it can be expected that if the operating parameters are maintained the refrigerant flow rate remains unchanged. Unless the regenerative heat exchanger changes (especially the length) in all other cases the flow rates must be calculated on a case by case basis. Different flow rates will also lead to a different optimal refrigerant volumes.

2.4 Determining the optimum amount of refrigerant

Determining the optimum amount of refrigerant per refrigerator for the time being is an empirical process on the principle of sample error. Refrigerant is added until overheating of the evaporator outlet is within the desired range, and the suction pipe temperature must be monitored [13]. When the compressor is not running, the refrigerant is collected in the evaporator. After the start of the compressor there is a lack of refrigerant in the condenser. The refrigerant must first be moved to the condenser.

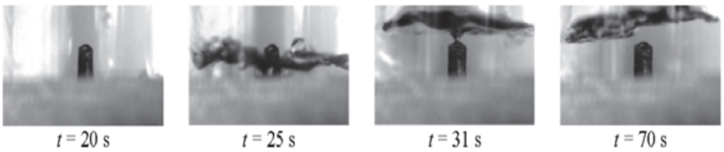

Fig. 3. Increasing the liquid piller after starting the compressor. Influence of gas amount on liquid level at condenser outlet/ capillary inlet.

2.5 Experiment planning

The measurements were performed using ANOVA of the type 3^2 with two independent variables (nitrogen flow rate and amount of refrigerant R600a) and one response surface (energy consumption). For convenience and easier reproduction of the results, the nitrogen flow rate was presented instead of the capillary pipe's length. The measuring range was selected on the basis of the serial characteristics of the refrigerator and the practical limitations. All experiments were performed in triplets.

Multi Channel Process System (MCPS) software product was used for measurements and registration. The temperature reading is accurate to ±0.15 °C, and the energy consumption is ± 0.01%. Energy efficiency research was conducted in Liebherr's laboratory. A Standard IKP 1650 refrigerator was used. It was optimized for operation at

25°C. The capillary pipe is 1 to 4 m long and 0,6 ± 0,025 mm in diameter. Length of 2,52 m corresponds to 4,5 l/min nitrogen flow rate according to the method shown in DIN8905 part 3.

2.6 Experimental installation

A Liebherr IKP 1650 built-in domestic refrigerator is used to determine the influence of the geometry of the capillary pipe.

The temperatures at the injection site of the refrigerant, the inlet and outlet of the evaporator receiver were measured.

Table 3. Liebherr IKP 1650 built-in domestic refrigerator specification.

Appliance	**IKP1650**	
Serial number	31.485.678.2	
Measurement number	LHG150138	
Evaporator	roll bond	Nitrogen flow rate 4,5 l/min
Condenser	Wire	
Vaccum panel in the door	7429116-00	640x395x11 mm
Vaccum panels in the housing	7429116-00	640x395x11 mm
Compressor	EmbracoVES D5	Inverter control
Amount of refrigerant		32 g [R600a]
Volume of the refrigerant part [l]		151
Climatic class		SN-T
Energy index		21,9 %
Energy efficiency class		A+++
Energy consumption according to ISO15502		0,178 Wh/24h

2.7 Temperature measurements

Cold storage temperature t_m represents the mean of the three PT100 elements during the tests - t_{1m}, t_{2m}, t_{3m} .

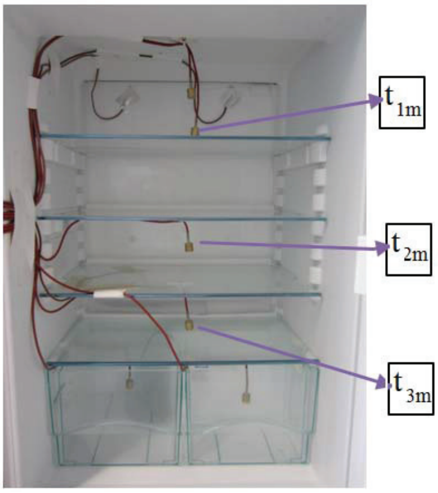

Fig.4. Refrigeration compartment.

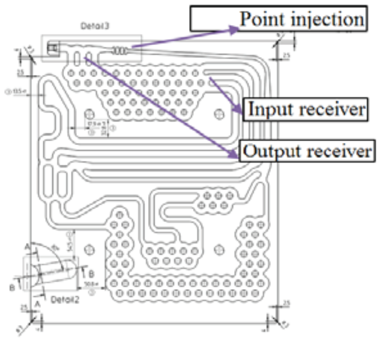

Fig.5. Evaporator.

The temperatures of the inlet, outlet and the midpoint of the condenser were measured.

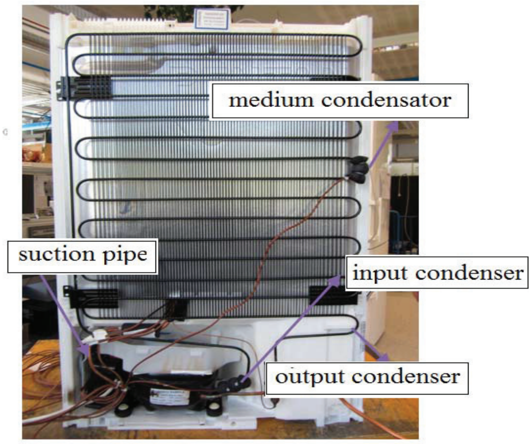

Fig. 6. Condenser.

3 Results and discussion

Table 4 presents two optimal variants (LM21 and LM32) of the measurements and table 5 summarizes the results. The first measurement is on the serial device. The measurements were made by changing the geometry of the capillary pipe and the amount of refrigerant without changing the length of the regenerative heat exchanger.

Table 4. Optimal variants of the measurements.

		LM01*		LM21		LM32	
Ambient temperature		16°C	32°C	16°C	32°C	16°C	32°C
Nitrogen flow rate	l/min	4,5	4,5	6,0	6,0	3,0	3,0
Refrigerant R600a	g	32,0	32,0	38,0	38,0	38,0	38,0
Setting	°C	4,5	4,5	4,5	4,5	5,0	5,5
Working time	min	12,3	22,0	9,1	25,8	15,9	17,3
Rest time	min	51,5	22,8	49,8	24,3	55,8	20,6
Working time	%	19,3	48,9	15,4	51,2	22,3	45,5
T_m	°C	4,8	5,1	4,8	4,9	5,1	5,2
Energy consumption	kWh/24	0,107	0,314	0,92	0,328	0,121	0,295
PT100 (T1)	°C	4,6	4,5	4,6	4,5	5,0	6,2
PT100 (T2)	°C	3,9	3,3	3,8	3,2	4,4	5,3
PT100 (T3)	°C	6,0	7,5	6,1	7,2	6,0	4,1
TKE01	min	-17,6	-13,2	-13,8	-12,2	-19,9	-10,5
Injection	°C	-15,6	-12,8	-13,2	-11,9	-18,1	-10,1
Inlet in the receiver	°C	-14,7	-12,0	-10,5	-10,5	-16,5	-10,9
Outlet in the reciver	°C	-6,4	-4,8	-6,0	-6,8	-7,4	-5,8
Inlet in the condenser	°C	29,3	48,9	28,7	50,1	29,0	47,4
Midpoint condenser	°C	24,0	44,7	26,4	45,6	28,2	46,3
Outlet in the condenser	°C	16,1	31,8	16,1	45,8	20,5	44,5
Suction pipe	°C	12,5	27,0	15,3	29,9	16,2	24,9

*Serial refrigerator data

Table 5. Summary of measurement results.

Measurement №	Nitrogen flow rate, l/min	Amount R600a , g	Energy consumption Wh/24h	
			16°C	32°C
LHG150138_M01	4,5	32	107	314
LHG150138-M38	4,5	33	103	322
LHG150138_M21	6	38	92	328
LHG150138_M25	3	33	130	310
LHG150138-M32	3	38	121	295
LHG150138_M30	4,5	33	101	322

LHG150138-M34	6	33	101	330
LHG150138_M24	3	38	110	301
LHG150138_M29	4,5	28	122	326
LHG150138_M12	3	28	122	307
LHG150138-M37	6	28	107	340

The ANOVA table (Table 6) partitions the variability in EC 16 into separate pieces for each of the effects. It then tests the statistical significance of each effect by comparing the mean square against an estimate of the experimental error. In this case, 2 effects have P-values less than 0.05, indicating that they are significantly different from zero at the 95.0% confidence level.

Table 6. Anova for EC16 and EC 32

EC 16				
Source	Sum of squares	Mean square	F-ratio	P-value
A:Flow	534,998	534,998	22,55	0,0177
B:Charge	240,349	240,349	10,13	0,0500
AA	101,038	101,038	4,26	0,1310
AB	16,8348	16,8348	0,71	0,4614
BB	20,9426	20,9426	0,88	0,4167
Lack-of-fit	199,97	39,9941	1,69	0,3540
EC 32				
A:Flow	1086,79	1086,79	113,73	0,0018
B:Charge	165,686	165,686	17,34	0,0252
AA	3,25952	3,25952	0,34	0,6002
AB	1,15	1,15	0,12	0,7516
BB	4,17811	4,17811	0,44	0,5557
Lack-of-fit	61,6704	12,3341	1,29	0,4439

The lack of fit test is designed to determine whether the selected model is adequate to describe the observed data, or whether a more complicated model should be used. The test is performed by comparing the variability of the current model residuals to the variability between observations at replicate settings of the factors. Since the P-value for lack-of-fit in the ANOVA table is greater or equal to 0,05, the model appears to be adequate for the observed data at the 95,0% confidence level.

The obtained regression equations which has been fitted to the data are:

$$EC\ 16 = 177{,}39 - 7{,}55 * \text{Nitrogen flow rate} - 1{,}04 * \text{Amount R600a} \quad (2)$$

$$EC\ 32 = 315{,}55 + 9{,}39 * \text{Nitrogen flow rate} - 1{,}16 * \text{Amount R600a} \quad (3)$$

Judging from the presented results and the literature analysis, it is clear that there is no universal optimal geometry of the capillary pipe. There is no element of the refrigeration cycle, whose change has no effect on the refrigerant flow rate passing through the capillary pipe. A change in the geometry of the capillary pipe affects all other components of the refrigeration cycle and can greatly improve or impair the efficiency of the refrigerator.

The potential for optimizing a single device when switching from EN ISO15502 to IEC 62552 (Global Standard) is presented in Table 7.

https://doi.org/10.1051/e3sconf/202020701013

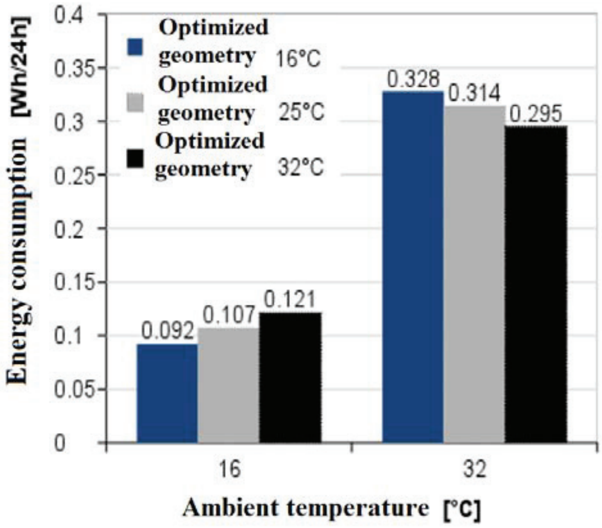

Fig. 7. Energy consumption at different geometries of the capillary pipe.

Table 7. Potential for optimizing a single device when switching from EN ISO15502 to IEC 62552 (Global Standard).

Measurement	Nitrogen flow rate	Amount R600a	Energy consumption, Wh/24h		Annual energy consumption	Potential
№	l/min	g	16°C	32°C	kWh	%
LM01	4,5	32	107	314	76,8	-
LM21	6	38	92			
LM32	3	38		295	70,6	8,1

4 Conclusion

The potential for switching from one to two different geometries of the capillary pipe is approximately 8% improvement in energy consumption. Possible implementation should take into account the potential savings ~ 8% and the life cycle of the particular appliance (~ 15 years). For the particular measured device, the consumption per year is about 75 kWh, which should take into account that the device itself has a low volume and low energy consumption. In the specific case, after the measured improvements, it could save over 6,2 kWh annualy, or about 93 kWh annualy for entire life cycle of the appliance. At an average price of 0,16 BGN (0,08 Euros) (which is likely to increase each year, taking into account the trend in recent years), such optimization would be cost-effective at a cost less than 14,9 BGN (7,62 Euros). If we consider the initial investment, in the variant with 16 ^{o}C ambient temperature, the length of the capillary pipe decreases (nitrogen flow rate N_2 = 6 l/min; R600a = 38 g), respectively the prime cost of the refrigerator decreases. At a high ambient temperature of 32 ^{o}C, the length of the capillary pipe increases twice (nitrogen flow rate N_2 = 3 l/min; R600a = 38 g), which increases the cost of the refrigerator.

The authors would like to thank Liebherr Hausgerate Maritsa for the assistant provided.

References

1. R. H. Swart, Capillary Tube Heat Exchangers, Refrigerating Engineering, Nr. **9**, S. 221–224, 248–249 (1946)
2. L. Cooper, C.K. Chu, W.R. Brisken, Simple selection method for capillaries derived from physical flow conditions, Refrigerating Engineering 65, Nr. **7**. p.88 (1957)
3. R. Plank, *Kapitel Entwicklung/Wirtschaftliche Bedeutung, Werkstoffe (Band I)*, Handbuch der Kältetechnik. Springer Verlag (1954)
4. N. Ludwig, Füllen von Kälteanlagen mit Kapillareinspritzung, Die Kälte+ Klimatechnik, Nr. **2**, S. 28–34 (2009)
5. L. A. Staebler, Theory and Use of a Capillary Tube for Liquid Refrigerant Control, Refrigerant Engineering 55, Nr. **1**, S. 55–59, 102–105 (1948)
6. L.E. Dietsch, Das Kapillarrohr und seine Anwendung an kleinen Kältesystemen, Nr. **10 /** Teil 4.1, S. 19–22 (1974)
7. H. Hristov, K. Angelov, Improving the energy efficiency of a household refrigerator – I, Scientific works of the union of scientists in Plovdiv, Series B, Engineering and Technology, ISNN 1311-9419, Volume **XIII**, p.187-90 (2016)
8. H. Hristov, K. Angelov, Improving the energy efficiency of a household refrigerator – II, Scientific works of the union of scientists in Plovdiv, Series B, Engineering and Technology, ISNN 1311-9419, Volume **XIII**, p.191-194 (2016)
9. C. Melo, C.B. Neto, R.T.S. Ferreira, R.H. Pereira, Constitutive Equations for Capillary Tube Modeling, *Proceedings of the International Refrigeration Conference in Purdue*. Paper 449, 1998, S. 431–436 (1999)
10. C. Melo, C.B. Neto, R.T.S. Ferreira, Empirical Correlations for the Modeling of R134a Flow Through Adiabatic Capillary Tubes, ASHRAE Transactions 105 (1999), Nr. **2**, S. 51–59 (1999)
11. W. Bohl, W. Elmendorf, *Technische Strömungslehre,* **15**. Auflage. Vogel Buchverlag, 2014 (Kamprath-Reihe) (2014)
12. M. Schenk, L. R. Oellrich, *Experimentelle Untersuchung des Kältemitteldurchflusses von Isobutan (R600a) durch adiabate Kapillaren,* Tagungsband zur DKV-Tagung 2012, Würzburg DKV, Deutscher Kältetechnischer Verein, **AA II.2**, S. 1–11 (2012)
13. D. Hartmann, C. Melo, Popping Noise in Household Refrigerators: Fundamentals and Practical Solution, Applied Thermal Engineering **51** (s 1-2);40-47 (2013)