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THE CALCULATION OF THE PIPE DIAMETER AND CHANGE WITH TEMPERATURE OF VAPOUR COMPRESSION REFRIGERATION SYSTEM USING VARIOUS ALTERNATIVE REFRIGERANTS

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Abstract

In this paper, the diameter of the pipe necessary for the lubrication oil to be entrained in the sucking and discharge line of the refrigerating systems using various alternative refrigerants calculated, its change range in accordance with temperature has been studied. The values of the minimum velocity of the refrigerant ($V_{r,min}$) and the minimum refrigeration load ($q_{r,min}$) necessary for the oil in the pipes in the suction and discharge lines to be entrained in accordance with the estimated pipe diameters determined at the first stage of this study have been calculated, and the diameters of the pipes to be used in the light of these values have been calculated. In a refrigeration system in which refrigerating load is 1 kw for copper pipes of K and L type, eco-friendly, alternative R134A, R410A and R423A refrigerants in place of refrigerants detrimental to ozone layer and leading to global warming have been used.

Key Words : Alternative refrigerants, Minimum velocity of refrigerant, Minimum refrigeration load, Pipe diameter

YENİ NESİL SOĞUTUCU AKIŞKANLARIN KULLANILDIĞI SOĞUTMA SİSTEMLERİNDE BORU ÇAPININ HESAPLANMASI VE SICAKLIKLA DEĞİŞİMİNİN İNCELENMESİ

Özet

Bu çalışmada, yeni nesil soğutucu akışkanların kullanıldığı soğutma sistemlerinin emme ve basma hatlarında yağlama yağının sürüklenebilmesi için gerekli boru çapı hesaplanarak sıcaklığa göre değişimi incelenmiştir. Çalışmanın ilk aşamasında belirlenen tahmini boru çaplarına göre emme ve basma hattında bulunan borulardaki yağın sürüklenebilmesi için gerekli olan minimum soğutucu akışkan hızı ($V_{r,min}$) ve minimum soğutma yükü ($q_{r,min}$) değerleri hesaplanmış, bu hesaplanan değerler ışığında kullanılması gereken gerçek boru çapları hesaplanmıştır. Analizler, soğutma yükünün 1 kW olduğu bir soğutma sisteminde K ve L tipi bakır borular için ozon tabakasına zarar veren ve küresel ısınmaya yol açan soğutucu akışkanların yerine çevre dostu alternatif akışkanlardan R134A, R410A ve R423A soğutucu akışkanları kullanılmıştır.

Anahtar Kelimler: Alternatif soğutucu akışkanlar, Minimum soğutucu akışkan hızı, Minimum soğutma yükü, Boru çapı

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Nomenclature

a	Oil thickness [m]
D_i	Pipe diameter [m]
g	Gravitational acceleration [m s^{-2}]
h_g	Vapour enthalpy [kJ kg^{-1}]
h_f	Liquid enthalpy [kJ kg^{-1}]
f_F	Darcy-Weisbach Friction factor
R	Pipe radius [m]
Re	Reynolds number
T	Temperature[^{0}C]
P	Pressure[Pa]
$U_{oil(y)}$	Velocity of oil distribution [m s^{-1}]
Q	Volumetric flow rate [m^3 s^{-1}]
ΔP	Pressure drop due to friction [Pa]
W	Flow
e	Relative roughness coefficicut
$V_{oil,min}$	Minimum oil velocity at the interface to stop oil flowing downwards [m s^{-1}]
$V_{r,min}$	Oil drifting to the minimum required rate of refrigerant
$q_{r,min}$	Minimum cooling load [kW]
μ	dynamic viscosity [kg m^{-1} s^{-1}]
ρ	density [kg m^{-3}]
ν	Kinematical viscosity [m^2 s^{-1}]
G	cooling flow mass rate in the system for unit time
KNS	Degree of volatility of the fluid
HSK	The cooling capacity per unit volume
ODP	Ozone depletion potential
GWP	Global warming potential

1. Introduction

After the 1940s, with the development of florocarbon refrigerants, the usage of the natural refrigerants as refrigerant has been decreased [1]. The usage of the CFCs have been increased, however it has been found out that they lead to breaching of the ozone layer and to global warming and that they are harmful to enviroment. Along with their high potential of causing global warming and breaching the ozone layer, CFCs are refrigerants that cause the highest destruction on the ozone layer. Therefore, certain precautions are taken and its worldwide usage is banned. One of the most crucial disadvantages of CFCs is that they exist for 75 – 120 years without the distortion of their chemical structures [2]. From 2011 on, Europe Union banned the usage of refrigerants GWP Value of which is more than150 [1, 3].

Refrigerants used in refrigerating systems have been replaced during two decades, owing to the breaching of the ozone layer, global warming and the harm that they do to the environment. With Montreal Protocol and Kyoto Protocol, Turkey also signed in 2009, refrigerants that do harm to the Ozone Layer, and that lead to global warming have been replaced by eco-friendly refrigerants with less effect on environment. In Europe the usage of the alternative refrigerants as refrigerants have become more and more common. Especially, vehicle air conditioners termed as mobile systems have become more and more common in mobile air conditioners and also in heat pump applications [4].

In the light of all these, the calculations of the pipe equipment and diameters of alternative refrigerants to be used in the refrigerating systems rise a new requirement. In literature, there are too many studies on the calculations of pipe diameters in the systems in which alternative refrigerants are used. The patterns obtained are related to refrigerants of old generation. Owing to the different qualities between refrigerants of old generation and that of new generation, during application, being revised, the existing methods should be applied on new refrigerants.

In refrigeration and heat pump systems, in order to increase the pressure of the refrigerant, compressors with piston(s) are used, and by the effect of the compressor, refrigerating oil is entrained to the discharge line. While the oil together with the refrigerant forms the balance mixture, it is sometimes entrained, due to the high velocity of refrigerant in the outlet of the compressor. Otherwise, proper lubrication does not occur, in this case, the break down of the compressor is inevitable, because of the overload in the compressor. Another problem is that the oil in the pipes causes plugging, and that, in a short while, it turns to covering on all the surface. And this is an undesired case, as it requires an added resistance in terms of heat transfer.

The oil leaking from the compressor is entrained by the refrigerant through the cycle, it reaches the suction line. Hence, the pipe equipment is important as much as any element of the refrigeration system. Especially, in the vertical pipes' segment, the design of the high pipes as well as proper fixing methods should be cared for, as the elevation of the oil is too difficult against the gravitation. Therefore, it is necessary that the refrigerant has determined velocities for determined altitudes, and also, as this leads to the pressure loss, the pressure loss must be kept on determined levels [5].

The issue of the oil cycle in refrigeration systems occurs in the suction line in which the temperature is the extreme low. Here the viscosity of the oil is considerably high. At this point, because of the high viscocity, the oil entrainment becomes difficult. In this paper, the estimation of the minimum liquid velocity and the minimum quantity of liquid, depending on the fluid behaviours that investigate the relation between the gravity and the entrainment force has been shown.

Kesim et al. [5] studied the velocity of refrigerant required in the minimum refrigeration load for the oil to be dragged in the vertical pipes in refrigeration systems. Taking the saturation field determined between the condenser and evaporator into consideration, they related the refrigeration capacity to the velocity of the refrigerant. Through the equations they obtained, for R134A refrigerant, for a situation where copper pipes are used in the suction and discharge line, they charted the minimum refrigeration capacity. Garland and Hadfield [6] studied the environmetal effects of refrigeration systems with hermetic compressors in which hydrocarbon refrigerants were used in terms of tribology. They studied the corrossion mechanism and friction factors, carrying out tests on the critical components of the sytem for different refrigerants. Cremaschi et al. [7] carried out experimented the oil grip characteristics of the condenser and the evaporator in the suction and discharge lines. With the mixing and unmixing lubricants in R22, R410A and R134A refrigerants, they studied the physics of the oil grip in its largest transmission characteristics. Through a parametric analiysis in the suction line, they showed the oil grip is dependent on oil mass rate and mass flow of the refrigerant vapor and the viscocity rate of the mixture and the pipe distribution.

Hwang et al. [8] investigated the oil distribution in the refrigeration system with transcritic CO_2. In the experimental results, they showed that the oil concentration of the refrigerant discharged from the compressor increased the oil grip volume in the temperature convertor and in the suction line. And, in the experimental results, they also observed that the evaporator pressure dropped, as the oil concentration increased in the fixed refrigerant flow [9]. Zoellick and Hrnjak [10] studied the oil grip mass and pressure drop in the horizontal and vertical piping in a polyster (POE) oiled system in which R410A was used as refrigerant.

They determined the vapor velocity, measuring the heat and the mass. They calculated the velocities of the refrigerant vapour, considering the characteristics such as the oil grip, the pressure drop suggested by ASHRAE. Sethi [11] investigated quantitatively studied the alternative refrigerant R123yf and the oil-grip and the pressure drop in the horizantal and vertical pipes in the suction line of the system run with POE32 oil.

2. Numerical Analysis

The aim of this study is to show the calculation of the diameters of the pipes, with the estimation of the minimum velocity and minimum refrigerant quantity depended on the refrigerant principles studying the force between the gravity and entrainment.

The minimum refrigeration load of the oil in the vertical segments of the refrigeration systems and the desired refrigeration velocity have been calculated. Then, the pipe diameter to be used in the system has been obtained, depending on the minimum velocity of the refrigerant required for the oil obtained to be entrained. In these calculations, it is assumed that the downward flow of the thick oil layer covering the internal face of the pipe depended on the gravity and the cutting force of upward flow balanced the upward flow of the oil layer.

There are two different mechanisms required to transmit the oil to the compressor in the refrigeration systems in which the oil is mixable with the refrigerant. These are the mixture of the gravitation and vapor cycle. Gravitation causes the mass accumulation of the oil in the bottom of the pipe that is the only outlet through which the mixture can be carried. The minimum velocity of the refrigerant vapor is taken into consideration, the flow of the oil layer on the wall of the pipe primarily being regarded, while a formula for mixing the oil is being derived. The velocity of the oil on the surface of the refrigerant vapor, as shown in figüre 1, is calculated, the profile velocity of the oil being determined. Here ‘a’ stands for the thickness of the oil layer, $V_{oil,min}$ stands for the minimum velocity of the oil stopping the downward flow of the oil [12].

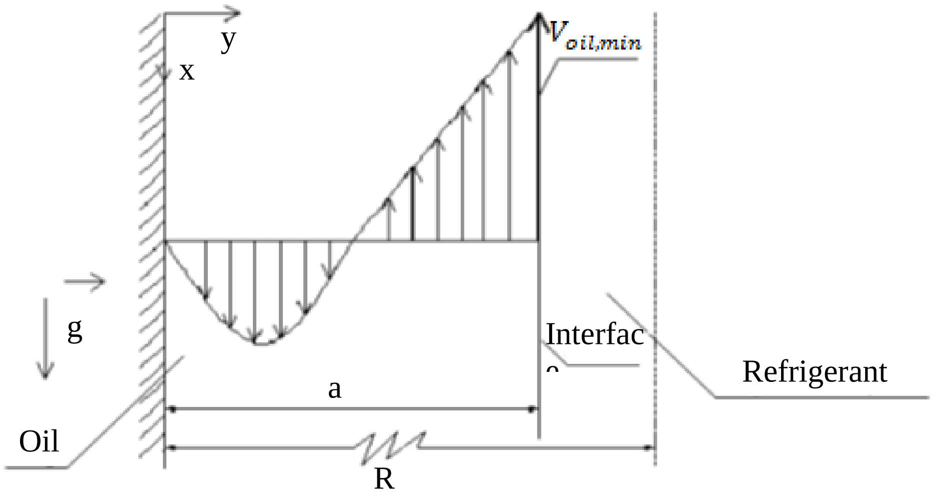

Fig 1. In a refrigeration system, the profile of oil flow in the internal surface of the pipe [5].

The vapor velocity in the interface causes the oil to be entrained downward in more than average velocity. The relation between the average velocity and the surface velocity is dependent on the pipe diameter and on the vapor density [13] In this case, the density and viscosity of the oil are also crucial. In order to carry the oil up in the vertical line, the net volumetric flow rate of the oil should be greater than or equal to zero. The limit of the volumetric flow rate of the oil also helps to calculate the velocity of the oil on the interface of the refrigerant vapor. The profile of the oil velocity can be calculated by the help of Navier-Stokes in the oil side and the known equations [14, 15, 16].

$$u_{oil}(y) = \left(\frac{1}{\rho_{oil} v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{y^2}{2} + A_y + B \tag{1}$$

Here if $y = 0$ take $u_{oil} = 0$ and $B = 0$ if $y = a$ take $u_{oil} = V_{oil,min}$ becomes.

$$V_{oil,min} = \left(\frac{1}{\rho_{oil} v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{a^2}{2} + A_a \tag{2}$$

$$u_{oil} = \left(\frac{1}{\rho_{oil} v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{y^2}{2} + \frac{V_{oil,min}}{a} - \left(\frac{1}{\rho_{oil} v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{a^2}{2} \tag{3}$$

If this obtained equation of velocity distribution is derived again.

$$u_{oil}(y) = \left(\frac{1}{\rho_{oil} v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\left(\frac{y^2}{2} - \frac{a}{2}y\right) + \frac{V_{oil,min}(y)}{a} \tag{4}$$

The equation is derived and the volumetric flow velocity of the oil can be found.

$$Q_{oil} = 2\pi R \int_0^a u_{oil}(y)dy \tag{5}$$

If the equation obtained from the Eq. (4) is subtituted,

$$Q_{oil} = 2\pi R\left[\left(\frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\right]\int_0^a \left(\frac{y^2}{2} - \frac{a}{2}y\right)dy + \int_0^a \left(\frac{V_{oil,min}(y)}{a}\right)dy \tag{6}$$

equation is found, and the following equation is obtained, the integrals being calculated, and the equation being reorganized.

$$Q_{oil} = \left(\frac{g}{v_{oil}} - \frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx}\right)\frac{a^3\pi R}{6} + V_{oil,min}\,a\pi R \tag{7}$$

In case of boundary condition, volumetric velocitiy of the oil is equal to zero, for this case of boundary condition, the minimum oil velocity on the vapor surface is found as follows [17].

$$Q_{oil} = 0 \ \rightarrow 0 = \frac{a^3\pi R}{6}\left(\frac{g}{v_{oil}} - \frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx}\right) + V_{oil,min}\,a\pi R \tag{8}$$

$$V_{oil,min} = \frac{a^2}{6}\left(\frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right) \tag{9}$$

The gradian of pressure diversity in the Eq. (1) can be determined, the side of the refrigerant being considered. To determine the pressure drop owing to friction, Darcy Weisbach Formula (an emprical Formula) being used, it can be written as follows [18].

$$\Delta P = f\rho_s \frac{V_r^2 L}{2D_i} \tag{10}$$

Here, V_r is average refrigerant velocity and it has been given as counterpart relation in the straight pipe for *f* Blasius factor.

$$f = \frac{0.3164}{Re^{0.25}}, \qquad Re = \frac{V_r D_i}{v_r} \tag{11}$$

The ones given in the Eq. (11) being subtituded for the ones in the Eq. (10), the pressure gradian in the *X* direction is found as follows.

$$\frac{dP}{dx} = f\,\frac{0.1582\,\rho_r v_r^{0.25} V_r^{1.75}}{D_i^{1.25}} \tag{12}$$

From the equation, miminun oil velocity on the interface of the refrigerant vapor is calculated as follows.

$$V_{oil,min} = \left(\frac{1}{\rho_{oil}V_{oil}}\,\frac{0.1582\,\rho_r v_r^{0.25} V_r^{1.75}}{D_i^{1.25}} - \frac{g}{V_{oil}}\right)\frac{a^2}{6} \tag{13}$$

In order to calculate the average velocity of the refrigerant vapor that ensures the returning criteria, another equation is required between the refrigerant vapor and the minimum oil velocity on the interface. This equation can be calculated, the refrigerant on the interface and the shear stress being equated to each other [5]. The shear stress of the oil shown in fig.1 can be formed as follows.

$$\tau_{oil,a} = \mu_{oil}\left(\frac{du_{oil}(y)}{dy}\right)_{y=a} \tag{14}$$

The Formula derived in the Eq. (3) being formed in place of U_{oil} in the Eq. (14), the following equation is found.

$$\tau_{oil,a} = \mu_{oil}\left(\frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{a}{2} + \frac{v_{oil,min}}{a} \tag{15}$$

For the turbulent flow, the following emprical equation can be used in order to determine the shear stress on the side of the refrigerant.

$$\tau_{r,a} = 0.0225\rho_r V_r^2\left(\frac{2v_r}{V_r D_i}\right)^{0.25} \tag{16}$$

The following equation is derived, the shear stresses given in the Eq. (15) and the Eq. (16) being equated.

$$\tau_{oil,a} = \tau_{r,a} = \mu_{oil}\left(\frac{1}{\rho_{oil}v_{oil}}\frac{dP}{dx} - \frac{g}{v_{oil}}\right)\frac{a}{2} + \frac{v_{oil,min}}{a} = 0.0225\rho_r V_r^2\left(\frac{2v_r}{V_r D_i}\right)^{0.25} \tag{17}$$

The term of the pressure gradient given in the Eq. (12) being substituted in the Eq. (17).

$$V_{oil,min} = 0.0268\frac{a\rho_r v_r^{0.25}V_r^{1.75}}{\rho_{oil}V_{oil}D_i^{1.25}} + \left(\frac{g}{V_{oil}} - \frac{1}{\rho_{oil}V_{oil}}\frac{0.1582\rho_r v_r^{0.25}V_r^{1.75}}{D_i^{1.25}}\right)\frac{a^2}{2} \tag{18}$$

The equation above is formed, and $V_{oil,min}$ determined in the Eq. (18) and in the Eq. (13) being equated,

$$\left(\frac{1}{\rho_{oil}V_{oil}}\frac{0.1582\rho_r v_r^{0.25}V_r^{1.75}}{D_i^{1.25}} - \frac{g}{V_{oil}}\right)\frac{a^2}{6} = 0.0268\frac{a\rho_r v_r^{0.25}V_r^{1.75}}{\rho_{oil}V_{oil}D_i^{1.25}} + \left(\frac{g}{V_{oil}} - \frac{1}{\rho_{oil}V_{oil}}\frac{0.1582\rho_r v_r^{0.25}V_r^{1.75}}{D_i^{1.25}}\right)\frac{a^2}{2} \tag{19}$$

V_s is derived as follows [5].

$$V_s = \left(\frac{ag\rho_{oil}D_i^{1.25}}{0.1582a\rho_r v_r^{0.25} - 0.0402\rho_r v_r^{0.25}D_i}\right)^{1/1.75} \tag{20}$$

The item derived in the equation in the Eq. (18) being used, the minimum oil velocity on the interface is found as follows so that the downward flow of the oil can be prevented,

$$V_{oil,min} = \frac{0.0067a^2 g}{v_{oil}}\left(\frac{D_i}{0.1582a - 0.0402D_i}\right) \tag{21}$$

Fully developed refrigeration flow slips on the interface in $V_{oil,min}$ velocity. So, minimum average flow velocity of the refrigerant are found from the addition of the Eq. (20) and the Eq. (21) to each other [5].

$$V_{r,min} = \left(\frac{ag\rho_{oil}D_i^{1.25}}{0.1582a\rho_r v_r^{0.25} - 0.0402\rho_r v_r^{0.25}D_i}\right)^{1/1.75} + \frac{0.0067a^2 g}{v_{oil}}\left(\frac{D_i}{0.1582a - 0.0402D_i}\right) \tag{22}$$

The minimum refrigerating load required to entrain the oil back in the cycle can be derived as follows.

$$q_{r,min} = \frac{\pi \rho_{oil} (0.96 D_i)^2 V_{r,min}}{4} \left(h_g - h_f\right) \tag{23}$$

Although an accurate value is not available to determine the thickness 'a' on the internal surface of the pipe, similar applications being considered, the oil thickness is used as $D_i/50$ [19].

$$q_{r,min} = \frac{\pi \rho_r (0.96 D_i)^2 V_{r,min}}{4} \left(h_g - h_f\right) \left[\left(\frac{g \rho_{oil} D_i^{1.25}}{1.8518 \rho_r v_r^{0.25}} \right)^{1/1.75} \left(\frac{D_i^2 g}{13819.40\, v_{oil}} \right) \right] \tag{24}$$

All of these equations being considered, minimum refrigerant velocity and minimum refrigeration load in the suction and discharge lines being considered in this study, for the alternative refrigerants R134, R410A and R423A, the change range of the pipe diameters of the lines forming the refrigeration system in accordance with the temperature has been determined. In the analysis, copper pipes of K type and L type being considered, calculations have been carried out. In a system chosen as 1 kW, minimum refrigerant velocity required for the oil to be entrained in accordance with the estimated pipe diameters and the field being multiplied, the flow can be found as follows.

$$W = V_{r,min} A \tag{25}$$

Then, to ensure the desired refrigeratin capacity, the quantity of the refrigerant required to cycle in the unit of time,

$$G = \frac{860}{(h_g - h_f) 0.24} \tag{26}$$

is calculated with the formula above. The friction factor must be calculated to determine the pipe diameter. To calculate the friction factor, relative roughness factor (ε) ve Reynolds sayısı (Re) is calculated as follows.

$$\varepsilon = \frac{0.0015}{D_i\,1000} \tag{27}$$

$$R_\varepsilon = \frac{1}{30\mu_r}\left[\frac{\rho_r G V_{r,min}}{\pi}\right]^{1/2} \tag{28}$$

These values obtained being used in the Formula of friction factor,

$$f_F = 0.001375\left[1 + \left(20000\varepsilon + \frac{10^6}{R_\varepsilon}\right)^{1/3}\right] \tag{29}$$

Friction factor is found. All of these equations being used, the Formula that gives the minimum pipe diameter required for the refrigeration system is derived as follows.

$$D_i = \left[\frac{8\left(\frac{G_\varepsilon}{3600}\right)^2 30}{\rho_r 9.81 P_{AF} 3.14^2} f_F\right]^{1/5} \tag{30}$$

3. Results and discussion

In the calculations, refrigeration capacity has been taken as 1 kW, evaporator temperature as between -40 oC and 10 oC, codenser temperature as 40 oC [20]. All of the equations derived being considered, and being calculated, minimum refrigerant loads and velocities in the suction and discharge lines, and the pipe diameters (Type K, TypeL) for alternative refrigerants of the R134A, R410A and R423A have been charted as follows (Table 1 – Table 12)

Table 1. For the refrigerant R134A, for a system of 1 kW in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.01368	0.01299	0.0124	0.01192	0.0115	0.01116
0.00785	0.0153	0.01452	0.01387	0.01332	0.01286	0.01247
0.0102	0.01627	0.01544	0.01475	0.01416	0.01367	0.01326
0.0135	0.01738	0.01649	0.01575	0.01512	0.0146	0.01416
0.0165	0.01822	0.01729	0.01651	0.01586	0.0153	0.01484
0.019	0.01884	0.01788	0.01707	0.01639	0.01582	0.01534
0.0254	0.02019	0.01915	0.01829	0.01756	0.01694	0.01643
0.0317	0.02128	0.02019	0.01928	0.01851	0.01786	0.01732
0.0377	0.02218	0.02104	0.02009	0.01929	0.01861	0.01805
0.05	0.02372	0.02251	0.02149	0.02063	0.01991	0.01931
0.062	0.02498	0.0237	0.02263	0.02172	0.02096	0.02032
0.074	0.02606	0.02472	0.02361	0.02266	0.02187	0.0212
0.0861	0.02702	0.02564	0.02448	0.0235	0.02268	0.02199
0.0981	0.02788	0.02646	0.02526	0.02425	0.0234	0.02269

Table 2. For the refrigerant R134A, for a system of 1 kW in the suction line, the pipe of TYPE L being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.01375	0.01305	0.01246	0.01197	0.01156	0.01121
0.00805	0.01539	0.01461	0.01395	0.0134	0.01293	0.01255
0.0109	0.01653	0.01568	0.01498	0.01438	0.01388	0.01346
0.014	0.01753	0.01663	0.01588	0.01525	0.01472	0.01428
0.0169	0.01833	0.01739	0.0166	0.01595	0.01539	0.01492
0.02	0.01907	0.0181	0.01728	0.01659	0.01601	0.01553
0.0262	0.02034	0.01929	0.01842	0.01769	0.01707	0.01655
0.0322	0.02136	0.02026	0.01935	0.01858	0.01793	0.01738
0.0383	0.02226	0.02112	0.02016	0.01936	0.01868	0.01812
0.0505	0.02378	0.02256	0.02154	0.02068	0.01996	0.01935
0.0627	0.02504	0.02376	0.02269	0.02178	0.02102	0.02038
0.0749	0.02614	0.0248	0.02367	0.02273	0.02193	0.02127
0.0871	0.0271	0.02571	0.02455	0.02357	0.02274	0.02205
0.0993	0.02797	0.02653	0.02533	0.02432	0.02347	0.02275

Table 3. For the refrigerant R410A, for a system of 1 kW in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.01142	0.01094	0.01054	0.01021	0.009945	0.009734
0.00785	0.01277	0.01223	0.01179	0.01142	0.01112	0.01088
0.0102	0.01358	0.01301	0.01253	0.01213	0.01181	0.01156
0.0135	0.01451	0.01389	0.01338	0.01296	0.01262	0.01235
0.0165	0.01521	0.01456	0.01403	0.01359	0.01323	0.01294
0.019	0.01572	0.01506	0.0145	0.01405	0.01367	0.01338
0.0254	0.01684	0.01613	0.01554	0.01505	0.01465	0.01433
0.0317	0.01776	0.017	0.01638	0.01586	0.01544	0.0151
0.0377	0.01851	0.01772	0.01707	0.01653	0.01609	0.01574
0.05	0.0198	0.01895	0.01826	0.01768	0.01721	0.01684
0.062	0.02084	0.01996	0.01922	0.01861	0.01812	0.01773
0.074	0.02174	0.02082	0.02005	0.01942	0.0189	0.01849
0.0861	0.02255	0.02159	0.0208	0.02014	0.0196	0.01918
0.0981	0.02327	0.02228	0.02146	0.02078	0.02023	0.01979

Table 4. For the refrigerant R410A, for a system of 1 kW in the suction line, the pipe of TYPE L being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.01148	0.01099	0.01059	0.01026	0.009991	0.00978
0.00805	0.01285	0.01231	0.01185	0.01148	0.01118	0.01094
0.0109	0.01379	0.01321	0.01272	0.01232	0.012	0.01174
0.014	0.01463	0.01401	0.01349	0.01307	0.01272	0.01245
0.0169	0.01529	0.01465	0.01411	0.01366	0.0133	0.01302
0.02	0.01592	0.01524	0.01468	0.01422	0.01384	0.01354
0.0262	0.01697	0.01625	0.01565	0.01516	0.01475	0.01444
0.0322	0.01782	0.01707	0.01644	0.01592	0.01549	0.01516
0.0383	0.01857	0.01779	0.01713	0.01659	0.01615	0.0158
0.0505	0.01984	0.019	0.0183	0.01772	0.01725	0.01688
0.0627	0.0209	0.02001	0.01927	0.01866	0.01817	0.01777
0.0749	0.02181	0.02088	0.02011	0.01948	0.01896	0.01855
0.0871	0.02261	0.02165	0.02085	0.02019	0.01966	0.01923
0.0993	0.02334	0.02234	0.02152	0.02084	0.02029	0.01985

Table 5. For the refrigerant R423A, for a system of 1 kW in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.01461	0.01385	0.01322	0.0127	0.01225	0.01188
0.00785	0.01634	0.01549	0.01478	0.01419	0.0137	0.01328
0.0102	0.01737	0.01647	0.01572	0.01509	0.01456	0.01412
0.0135	0.01856	0.01759	0.01679	0.01611	0.01555	0.01507
0.0165	0.01946	0.01844	0.0176	0.01689	0.0163	0.0158
0.019	0.02012	0.01907	0.0182	0.01746	0.01685	0.01634
0.0254	0.02155	0.02043	0.01949	0.01871	0.01805	0.0175
0.0317	0.02272	0.02153	0.02054	0.01972	0.01902	0.01844
0.0377	0.02368	0.02244	0.02141	0.02055	0.01982	0.01922
0.05	0.02533	0.02401	0.0229	0.02198	0.02121	0.02056
0.062	0.02667	0.02527	0.02411	0.02314	0.02233	0.02164
0.074	0.02782	0.02637	0.02516	0.02414	0.02329	0.02258
0.0861	0.02885	0.02735	0.02609	0.02504	0.02416	0.02342
0.0981	0.02977	0.02822	0.02692	0.02584	0.02493	0.02416

Table 6. For the refrigerant R423A, for a system of 1 kW in the suction line, the pipe of TYPE L being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.01468	0.01392	0.01329	0.01276	0.01231	0.01194
0.00805	0.01644	0.01558	0.01487	0.01428	0.01378	0.01336
0.0109	0.01765	0.01673	0.01596	0.01532	0.01479	0.01434
0.014	0.01872	0.01774	0.01693	0.01625	0.01568	0.0152
0.0169	0.01957	0.01855	0.0177	0.01699	0.01639	0.01589
0.02	0.02036	0.0193	0.01842	0.01768	0.01705	0.01653
0.0262	0.02171	0.02058	0.01963	0.01884	0.01818	0.01763
0.0322	0.0228	0.02161	0.02062	0.01979	0.01909	0.01851
0.0383	0.02377	0.02253	0.02149	0.02063	0.0199	0.01929
0.0505	0.02539	0.02406	0.02296	0.02203	0.02126	0.02061
0.0627	0.02674	0.02534	0.02418	0.0232	0.02239	0.0217
0.0749	0.0279	0.02645	0.02523	0.02421	0.02336	0.02265
0.0871	0.02893	0.02742	0.02616	0.02511	0.02422	0.02348
0.0993	0.02986	0.0283	0.027	0.02591	0.025	0.02423

Table 7. For the refrigerant R134A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.008684	0.008915	0.009128	0.00933	0.009531	0.009733
0.00785	0.009703	0.009961	0.0102	0.01042	0.01065	0.01087
0.0102	0.01031	0.01059	0.01084	0.01108	0.01132	0.01156
0.0135	0.01101	0.0113	0.01157	0.01183	0.01208	0.01234
0.0165	0.01154	0.01185	0.01213	0.0124	0.01267	0.01293
0.019	0.01193	0.01225	0.01254	0.01282	0.01309	0.01337
0.0254	0.01278	0.01312	0.01343	0.01373	0.01402	0.01432
0.0317	0.01347	0.01383	0.01416	0.01447	0.01478	0.01509
0.0377	0.01403	0.01441	0.01475	0.01508	0.0154	0.01573
0.05	0.01501	0.01541	0.01578	0.01613	0.01647	0.01682
0.062	0.0158	0.01622	0.01661	0.01698	0.01734	0.01771
0.074	0.01649	0.01693	0.01733	0.01771	0.01809	0.01847
0.0861	0.0171	0.01755	0.01797	0.01837	0.01876	0.01916
0.0981	0.01764	0.01811	0.01854	0.01895	0.01936	0.01977

Table 8. For the refrigerant R134A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE L being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.008724	0.008957	0.009171	0.009374	0.009575	0.009778
0.00805	0.00976	0.01002	0.01026	0.01049	0.01071	0.01094
0.0109	0.01047	0.01075	0.01101	0.01125	0.01149	0.01174
0.014	0.0111	0.0114	0.01167	0.01193	0.01219	0.01244
0.0169	0.01161	0.01192	0.0122	0.01247	0.01274	0.01301
0.02	0.01208	0.0124	0.01269	0.01297	0.01325	0.01353
0.0262	0.01287	0.01322	0.01353	0.01383	0.01413	0.01442
0.0322	0.01352	0.01388	0.01421	0.01452	0.01483	0.01515
0.0383	0.01409	0.01446	0.01481	0.01513	0.01546	0.01579
0.0505	0.01505	0.01545	0.01582	0.01617	0.01651	0.01686
0.0627	0.01585	0.01627	0.01666	0.01702	0.01739	0.01776
0.0749	0.01654	0.01698	0.01738	0.01776	0.01814	0.01853
0.0871	0.01714	0.0176	0.01802	0.01842	0.01881	0.01921
0.0993	0.01769	0.01816	0.0186	0.01901	0.01941	0.01982

Table 9. For the refrigerant R410A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.007411	0.007516	0.007635	0.007771	0.007929	0.008117
0.00785	0.008284	0.0084	0.008533	0.008684	0.008861	0.00907
0.0102	0.008805	0.008929	0.009069	0.00923	0.009418	0.00964
0.0135	0.009402	0.009534	0.009684	0.009856	0.01006	0.01029
0.0165	0.009857	0.009995	0.01015	0.01033	0.01054	0.01079
0.019	0.01019	0.01033	0.0105	0.01068	0.0109	0.01115
0.0254	0.01091	0.01107	0.01124	0.01144	0.01167	0.01195
0.0317	0.0115	0.01167	0.01185	0.01206	0.0123	0.01259
0.0377	0.01199	0.01216	0.01235	0.01257	0.01282	0.01312
0.05	0.01282	0.013	0.01321	0.01344	0.01371	0.01403
0.062	0.0135	0.01369	0.0139	0.01415	0.01444	0.01477
0.074	0.01408	0.01428	0.01451	0.01476	0.01506	0.01541
0.0861	0.01461	0.01481	0.01504	0.01531	0.01562	0.01598
0.0981	0.01507	0.01528	0.01552	0.01579	0.01611	0.01649

Table10. For the refrigerant R410A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.007446	0.007551	0.00767	0.007807	0.007966	0.008155
0.00805	0.008332	0.00845	0.008583	0.008735	0.008913	0.009123
0.0109	0.008943	0.009068	0.009211	0.009374	0.009565	0.00979
0.014	0.009483	0.009616	0.009767	0.00994	0.01014	0.01038
0.0169	0.009913	0.01005	0.01021	0.01039	0.0106	0.01085
0.02	0.01031	0.01046	0.01062	0.01081	0.01103	0.01129
0.0262	0.011	0.01115	0.01132	0.01152	0.01176	0.01203
0.0322	0.01155	0.01171	0.01189	0.0121	0.01235	0.01264
0.0383	0.01203	0.0122	0.01239	0.01261	0.01287	0.01317
0.0505	0.01285	0.01303	0.01324	0.01347	0.01375	0.01407
0.0627	0.01354	0.01373	0.01394	0.01419	0.01447	0.01481
0.0749	0.01413	0.01432	0.01455	0.0148	0.0151	0.01546
0.0871	0.01465	0.01485	0.01508	0.01535	0.01566	0.01603
0.0993	0.01511	0.01533	0.01557	0.01584	0.01616	0.01654

Table11. For the refrigerant R423A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE K being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00485	0.00978	0.009892	0.01001	0.01015	0.0103	0.01046
0.00785	0.01093	0.01105	0.01119	0.01134	0.01151	0.01169
0.0102	0.01162	0.01175	0.01189	0.01205	0.01223	0.01243
0.0135	0.0124	0.01254	0.0127	0.01287	0.01306	0.01327
0.0165	0.013	0.01315	0.01331	0.01349	0.01369	0.01391
0.019	0.01344	0.01359	0.01376	0.01394	0.01415	0.01438
0.0254	0.0144	0.01456	0.01474	0.01493	0.01515	0.0154
0.0317	0.01517	0.01535	0.01553	0.01574	0.01597	0.01623
0.0377	0.01581	0.01599	0.01619	0.0164	0.01664	0.01691
0.05	0.01691	0.01711	0.01732	0.01755	0.0178	0.01809
0.062	0.01781	0.01801	0.01823	0.01847	0.01874	0.01904
0.074	0.01858	0.01879	0.01902	0.01927	0.01955	0.01987
0.0861	0.01927	0.01948	0.01972	0.01999	0.02028	0.0206
0.0981	0.01988	0.0201	0.02035	0.02062	0.02092	0.02126

Table12. For the refrigerant R423A, for a system of 1 kW in T_{cond}=40ºC in the suction line, the pipe of TYPE L being used, the change of the calculated value of D_i to the estimated value of D_i in accordance with the evaporator temperature.

Temperature	-40	-30	-20	-10	0	10
Estimated D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]	Cal. D_i [m]
0.00495	0.009826	0.009938	0.01006	0.0102	0.01035	0.01051
0.00805	0.01099	0.01112	0.01126	0.01141	0.01157	0.01176
0.0109	0.0118	0.01193	0.01208	0.01224	0.01242	0.01262
0.014	0.01251	0.01265	0.01281	0.01298	0.01317	0.01338
0.0169	0.01308	0.01322	0.01339	0.01356	0.01376	0.01399
0.02	0.01361	0.01376	0.01393	0.01411	0.01432	0.01455
0.0262	0.0145	0.01467	0.01485	0.01505	0.01526	0.01551
0.0322	0.01523	0.0154	0.01559	0.0158	0.01603	0.01629
0.0383	0.01587	0.01605	0.01625	0.01647	0.01671	0.01697
0.0505	0.01696	0.01715	0.01736	0.01759	0.01784	0.01813
0.0627	0.01786	0.01806	0.01828	0.01852	0.01879	0.01909
0.0749	0.01863	0.01884	0.01907	0.01933	0.01961	0.01993
0.0871	0.01932	0.01954	0.01978	0.02004	0.02033	0.02066
0.0993	0.01994	0.02016	0.02041	0.02068	0.02098	0.02132

4. Conclusions

During the performance of a refrigeration system, for the oil to be ensured to return to the compressor, and for the pressure-loss and the overnoise to be prevented, the velocity of refrigerant and the diameter of the pipe should be determined properly. In this study, the formulas derived theoretically have been applied on different refrigerants such as R134, R410, R423A, and the estimated pipe diameters being regarded, the real values of the pipe diameters have been calculated.

It is observed that, the temperature of the evaporator rising, the pipe-diameters calculated become narrow, when the R134A is used as refrigerant and copper pipe of typ K is used, the cause of this is that the refrigerating force drops as a result of the rise of the evaporator temperature. Depending on this, the pipe diameter becomes narrow. Under the same conditions, the copper pipe of L type being used in place of the copper pipe of type K, it has been observed that the pipe diameter similarly becomes narrow in accordance with the rise of the evaporator temperature. The temperature of the evaporator being raised respectively from -40 ^{0}C by 10 ºC (-40 ºC, -30 ºC, -20 ºC, -10 ºC, 0 ºC, 10 ºC), it is observed from the calculated values that the pipe diameters become narrow approximately % 5

The same conditions being regarded, analysis have been carried out in the other refrigerants for the copper pipes of K type and type. Similarly, ıt has been observed that the pipe diameters become narrow, as the temperature of the evaporator rises. The refrigerants being compared, R410A has been obtained from the narrowest pipe diameter and the largest one from the R423A. The pipe diameters obtained being regarded, the pipe diameters, in accordance with the refrigerant changes, from the narrowest one to the largest one are R410A<R134A<R423A. In the same system, in the discharge line, in the R134A and R423A refrigerants, the temperature of the condenser being fixed as 40 ºC, and the temperature in the evaporator being raised by 10 ºC (-40 ºC, -30 ºC, -20 ºC, -10 ºC, 0 ºC, 10 ºC), average 2.5% rise in the diameter has been obsevered.

The pipe diameters calculated for the suction and discharge lines being studied, the pipe diameters in the discharge line are, as expected, narrower than the pipe diameters in the suction line.

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