УДК 621.564

O. Ostapenko ✉***, P. Stoyanov, O. Yakovleva, M. Khmelniuk***
Odessa National Academy of Food Technology, 112 Kanatnaja str., Odessa, 65039, Ukraine
✉ e-mail: ostapenkosc@gmail.com ORCID: 0000-0003-0990-7883

HEAT PUMP TECHNOLOGY – POTENTIAL IMPACT ON ENERGY EFFICIENCY PROBLEM AND CLIMATE ACTION GOALS WITHIN UKRAINIAN ENERGY SECTOR

The increasing demand of energy sources for urban, household and industrial facilities requires strategies development for seeking new energy sources. In recent years an important problem is to have energy storage, energy production and energy consumption which fulfill the environment friendly expectations. A lot of attention is devoted to renewable energy sources. One of the most attracting among them is energy production form geothermal sources. At a few meters below the earth's surface the underground maintains a constant temperature in an approximation through the year allowing to withdraw heat in winter for heating needs and to surrender heat during summer for air-conditioning purposes. Heat pump is a rapidly developing technology for heating and domestic hot water production. Using ground as a heat source, heat exchange is carried out with heat pumps compound to vertical ground heat exchanger tubes that allows the heating and cooling of the buildings utilizing a single unit installation. Heat pump unit provides a high degree of productivity with moderate electric power consumption. In this paper a theoretical performance study of a vapor compression heat pump system with various natural and synthetic refrigerants (HFCs) is presented. Operation mode of the heat pump unit was chosen according to European Standard EN14511-2:2007 and EN255-2. An influence of discharge temperature on system performance was evaluated at different boiling temperatures. The comparison of mass flow rate and coefficient of performance for considered refrigerants at constant cooling capacity and condensation temperature was performed.

Key words: *new energy sources, heat pump; natural refrigerant; discharge temperature; coefficient of performance; ground source heat pump.*

О. В. Остапенко, П. Ф. Стоянов, О. Ю. Яковлєва, М. Г. Хмельнюк
Одеська національна академія харчових технологій, вул. Канатна, 112, Одеса, 65039, Україна

ТЕПЛОНАСОСНІ ТЕХНОЛОГІЇ ТА ЇХ ПОТЕНЦІЙНИЙ ВПЛИВ НА ПРОБЛЕМИ ЕНЕРГОЕФЕКТИВНОСТІ ТА ЗАХИСТУ НАВКОЛИШНЬОГО СЕРЕДОВИЩА В ЕНЕРГЕТИЧНОМУ СЕКТОРІ УКРАЇНИ

Зростаючий попит на енергоносії для міських, побутових і промислових об'єктів потребує розробки стратегій для пошуку нових джерел енергії. В останні роки важливою проблемою є наявність методів зберігання, виробництва та споживання енергії, які не мають впливу на довкілля. Багато уваги приділяється поновлюваним джерелам енергії. Одне з найбільш привабливих поновлюваних джерел енергії це геотермальна енергія. У декількох метрах нижче поверхні землі підтримується постійна температура на протязі всього року, що дозволяє використовувати це тепло взимку для потреб опалення та відводити туди тепло протягом літа для потреб кондиціювання повітря. Тепловий насос являє собою технологію для опалення і виробництва гарячої води яка швидко розвивається. При використанні землі в якості джерела тепла, теплообмін здійснюється з тепловими насосами за допомогою з'єднання вертикальних грунтових теплообмінних труб, що дозволяє забезпечувати опалення та охолодження будівель, за допомогою однієї комбінованої системи. Тепловий насос забезпечує високий ступінь продуктивності з помірним споживанням електроенергії. У роботі проведено теоретичне дослідження продуктивності парокомпресійного теплового насосу з використанням різних природних і синтетичних холодоагентів. Режим роботи блоку теплового насоса прийнято згідно з Європейським стандартом EN14511-2:2007 і EN255-2. Було оцінено вплив температури нагнітання на продуктивність системи при різних температурах кипіння. Проведено порівняння масової витрати і коефіцієнту перетворення для розглянутих холодоагентів при постійній холодопродуктивності і температурі конденсації.

Ключові слова*: нові джерела енергії, тепловий насос; природний холодоагент; температура нагнітання; коефіцієнт перетворення; геотермальний тепловий насос.*

DOI: http://dx.doi.org/10.15673/0453-8307.6/2015.44781

I. INTRODUCTION

In recent years heat pumps market is growing rapidly. It is directly linked to increasing energy needs of humanity and the simultaneous depletion of traditional non-renewable energy resources in the world as a whole and in individual regions. Heat pumps are widely used in the chemical and food industries, housing and communal services. Most of the power equipment has high emission rates of energy flows to the atmosphere (refrigeration plants condensation heat, waste water enterprises, the flue gases from internal combustion engines and boilers). The use of heat pumps allows reducing greenhouse gas and carcinogenic substances emissions and, thus reducing human impact on the environment.

The heat pump performance is based with following factors: the temperature of heat source, schematic diagram of the heat pump, climate conditions of the region, working fluids of heat pump (refrigerants and intermediate coolants), heat pump elements (type of compressors, heat exchangers, control systems).

There are various types of heat pumps, such as: vapor compression, adsorption, absorption and ejector heat pump systems which are able to use a variety of low-grade heat sources. Due to a simple circuit design vapor compression heat pumps are widely used, especially for domestic applications. For the analysis of refrigerant choice global trends in the field of legislation on environmental protection, international agreements governing the use of refrigerants were taken into account [7]. In particular, according to the decision of the European Parliament in 2011 the use of air conditioning systems with fluorine-containing CFCs (HFC) with a global warming potential (GWP) above 150 was banned [1-6].

In 2008 more than 25% of European heat pump market share was represented by ground-source heat pumps (Forsén 2008). According to EUObserver (2007) and EHPO (2008) more than 690,000 heat pump units with 7.300 MW of capacity were installed in Europe in 2006. The total installed capacity have being increasing in recent years and in 2013 reached 24 GW with estimate useful energy production of 13 TWh. Integration of renewable energy sources in heating and cooling applications was approximately 8.26 TWh avoiding 2.12 Mt of CO_2-equivalent emissions. The global installed capacity has reached about 15.400 MW, and annual energy use is estimated to be 87.500 TJ (Lund 2005). The worldwide GSHP capacity has seen high growth recently. According to Le Feuvre (2008) annual growth rates have exceeded 10% over the last 10 years, mostly in North America and European countries. Geothermal energy is available everywhere and Ukraine has a good geothermal energy potential. Nowadays ground-source heat pumps are a small but growing fraction of the global installed base of space-conditioning equipment [11-13].

This study aims at ground-source heat pump system with 20 kW of heating capacity at constant condensation temperature $t_к$=40 ℃ (according to test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating standards EN 14511-2:2007, EN 255-2) and variable boiling point t_0 = - 5 ÷ + 6 ^{0}C [14-15]. Schematic diagram is the single stage heat pump system with suction line heat exchanger (SLHX).

Table 1. *Refrigerants properties*

Parameter	Refrigerant					
	R410a	R404a	R134a	R600a	R290	R1270
Chemical formula	R-22/125 (50/50)	R125/R143a/ R134a (44/52/4)	CH_2FCF_3	C_4H_{10}	C_3H_8	$(CH_3)CH=CH_2$
M, kg / kmol	72.59	97.60	102.03	58.12	44.10	42.09
t_s, ^0C	-51.3	-46.7	-26.5	-12	-42.1	-47.7
t_{ct}, ^0C	72.3	72.7	101.5	135	96.7	103.6
P_{ct}, *bar*	49.3	37.35	40.6	36.5	42.5	46.7
R, kJ / kg	270.1	201	231.3	366.2	425.6	475.2
ODP	0	0	0	0	0	0
GWP	1890	3750	1300	0.001	0.01	1.8

M – molecular weight, t_s – boiling point at atmospheric pressure, t_{ct} – critical temperature, P_{ct} – critical pressure, ODP – ozone depletion potential, GWP – global warming potential; R – latent heat of vaporization (condensation)

As the working fluids for the heat pump the following natural refrigerants was considered (R600a, R290, R1270) such as modern alternative refrigerants (R404a, R134a, R410a).

Natural working substance such as isobutene (R600a) and propane (R290) are widely available and possesses good thermodynamic properties, "zero" greenhouse effect, don't affect the ozone layer, and are cheap. However use of these refrigerants entails with higher safety requirements according to high flammability and explosiveness. Mixed refrigerants (R404a, R134a, R410a) don't fall under the restrictions of the Kyoto and Montreal protocols, are promising in the long term forecasting of use [16]. The properties of refrigerants are submitted in table 1.

II. HEAT PUMP MATHEMATICAL MODEL AND DESIGN

A schematic diagram of the ground source heat pump system is given on fig. 2. The thermodynamic cycle on diagrams temperature-entropy and pressure-enthalpy is shown on fig. 1. Many techniques have been recently proposed in order to improve the cycle performance, more details are given by Wang, 2000, Chap.9 (Wang, 2000). In the current work, a heat exchanger has been added between the suction line and liquid line [17].

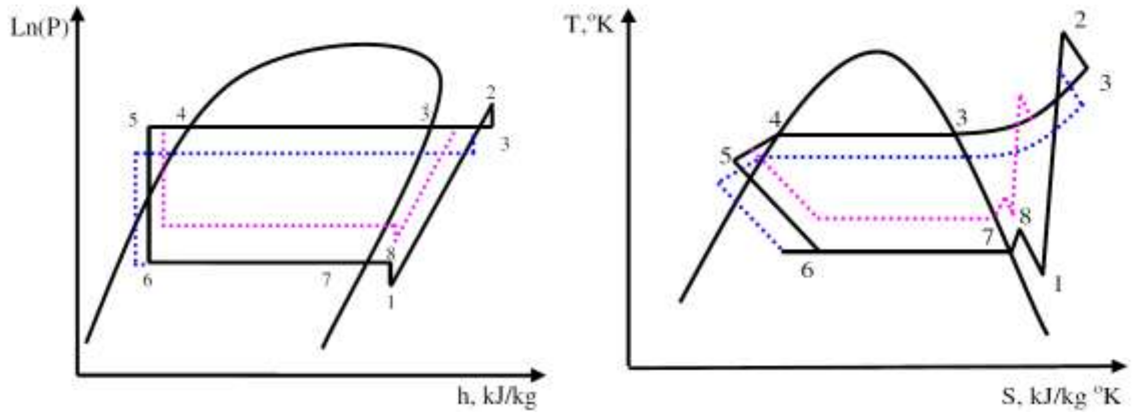

Figure 1 *– Thermodynamic cycle on diagrams temperature-entropy and pressure-enthalpy*

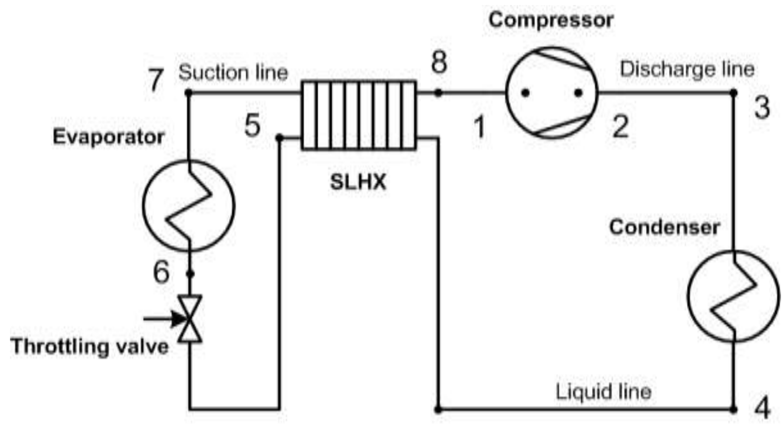

Figure 2 *– Schematic diagram of the heat pump system components*

The mathematical model for the cooling cycle is provided by Ayyildiz [17]. The equations (1-15) from this model for the compressor, condenser and throttling valve are first obtained.

The work input rate to the compressor is:

$$\dot{W_{net}} = \dot{m} \cdot (h_2 - h_1) \quad (1)$$

where

$$h_2 = h_1 + \frac{(h_{2s} - h_1)}{\eta_{isn}} \quad (2)$$

Power consumed by compressor is:

$$\dot{W}_{comp} = \frac{\dot{W_{net}}}{\eta_m \eta_e} \quad (3)$$

The heat rejection rate in condenser is calculated by:

$$\dot{Q}_r = \dot{m}_r \cdot (h_3 - h_4) \quad (4)$$

$$\dot{Q}_c = \dot{m}_w \cdot c_{p,w}(T_{w,o} - T_{w,i}) \quad (5)$$

$$\dot{Q}_r = \dot{Q}_c \quad (6)$$

$$\dot{Q}_c = K_c A_c F_c \Delta T_{lm,c} \quad (7)$$

where $\Delta T_{lm,c}$ – logarithmic mean temperature difference

$$\Delta T_{lm,c} = \frac{(T_c - T_{w,o}) - (T_4 - T_{w,i})}{ln\frac{(T_c - T_{w,o})}{(T_4 - T_{w,i})}} \quad (8)$$

The heat transfer rate in the evaporator is:

$$\dot{Q}_e = \dot{m}_r \cdot (h_7 - h_6) \quad (9)$$

$$\dot{Q}_e = \dot{m}_b \cdot c_{p,b}(T_{b,i} - T_{b,o}) \quad (10)$$

$$\dot{Q}_e = K_e A_e F_e \Delta T_{lm,e} \quad (11)$$

$$\Delta T_{lm,e} = \frac{(T_{b,i}-T_7)-(T_{b,o}-T_6)}{ln\frac{(T_{b,i}-T_7)}{(T_{b,o}-T_6)}} \quad (12)$$

The liquid refrigerant is expanded by throttling process and the expansion valve and enthalpy is:

$$h_6 = h_5 \quad (13)$$

Heat balance of suction line heat exchanger:

$$h_4 - h_5 = h_8 - h_7 \quad (14)$$

From the first law of thermodynamics, the heat energy balance of a heat pump system is:

$$\dot{W}_{comp} + \dot{Q}_e = \dot{Q}_c \quad (15)$$

The conventional parameter that has been used to describe the heat pump performance is COP (Coefficient of Performance), which is the ratio of the quality of the useful heat output to the quantity of work driving the compressor:

$$COP = \frac{\dot{Q}_c}{\dot{W}_{comp}} \quad (16)$$

For analysis purposes, the following assumptions were made:

- P2-P3=25 kPa respectively, see Fig.1.
- The pressure drop through the pipe is negligible.
- The isentropic efficiency of the compressor is 75%.
- The mechanical efficiency of the compressor is 75%.
- Sub-cooling in the condenser is 5K useless superheat in suction line is 5K.
- Thermal efficiency of the suction line heat exchanger, is 70%
- Heat loss factor of the compressor, i.e. ratio between heat loss of the compressor to the surroundings and the energy consumption of the compressor, is 15%.
- Superheat in compressor’s motor 3K (for hermetic scroll compressor)
- Operation modes according to EN 14511-2:2007 are B0/W35 and B0/W55.
- Cooling capacity assumed to be constant 20 kW, thus a change in temperature will affect the flow rate of refrigerant through the cycle.
- Temperature difference in condenser/evaporator is 5K.

III. RESULTS AND DISCUSSION

The compressor's discharge temperature has significant influence on heat pump performance for heating an intermediate medium (air, water or coolant). The temperature of the refrigerant vapor at the outlet of the compressor determines the highest possible temperature limit of heating the intermediate medium during refrigerants condensation process in condenser or removing refrigerants heat overheating (desuperheater). It is possible to withdraw approximately 10-15% of the condensation heat in a heat pump desuperheater. The heat flow in desuperheater allows to heat intermediate coolant or water up to 60-70 °C.

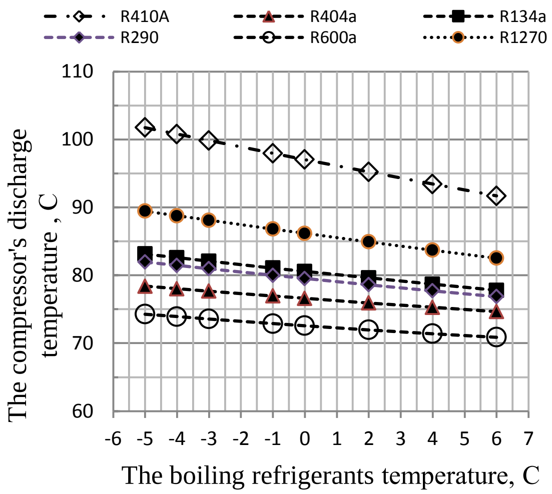

Figure 3 – *Dependence of the* compressor's *discharge* temperature $t_2=f(t_0)$

Figure 3 shows the dependence of the compressor's discharge temperature for different refrigerants of the boiling temperature. Consider the option of maintaining a minimum boiling point t_0 = = –5 °C and condensation temperature t_c = 40 °C. The graph shows that the maximum compressor's discharge temperature is equal to 102 °C which corresponds to the R410a, the minimum compressor's discharge temperature corresponds to R600a 74.5 °C. From the standpoint of the compressor's discharge temperature and the optimal compressor performance the use of R600a is more preferably than R410A. However, high compressor's discharge temperature allows heating the coolant in heat exchangers of heat pump to a higher temperature. The compressor's discharge temperatures of R290 and R134a are almost equal at the same operating conditions (t_c and t_0). The refrigerant R1270 (propylene) allows to implement a single-stage compression, the compressor's discharge temperature is optimal for all types of compressors (piston, scroll, screw etc.) and is quite high compared to other heat pump working fluids.

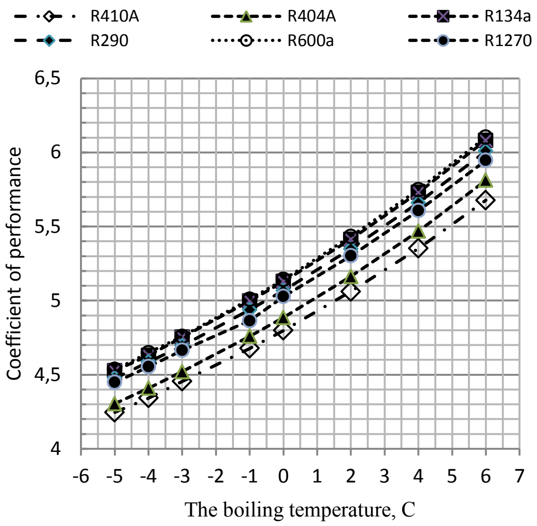

***Figure 4** – Coefficient of performance from refrigerants boiling point dependence COP=f(t$_0$)*

Coefficient of performance from refrigerants boiling point dependence is shown on figure 4. Results show that the maximum thermodynamic efficiency of the heat pump corresponds to R600a and R134a as a working fluid, the minimum – R410a. In the case of R600a and R134a heat pump COP is 7.6% higher in comparison with refrigerant R410a. These results are also connected with molecular weight of different refrigerants (see table 1). Large values of evaporation enthalpy are found for substances with light molecule weight [9] and the energy losses across a compressor's valves are high when the molecule weight is high [10]. The thermal transformation coefficient for R290 and R1270 are practically the same range. Analysis of the of R134a and R600a performance shows that in terms of ease of use and maintenance R600a is more preferred. Refrigerant R600a is a natural working substance, in the case of depressurization of the system can be easily refueled in contrast to the blend refrigerant R134a.

Comparison of heat pump refrigerants by mass flow in the system with the same cooling capacity is shown on fig.4. Based on the mass flow chart for the systems refilled with natural refrigerants (R290, R600a and R1270) have highest performance characteristics. This is due to the fact that these refrigerants have higher values of the latent heat of vaporization unlike R410a, R404a and R134a. These results are consistent with investigation other researchers [5, 8, 3, 16].

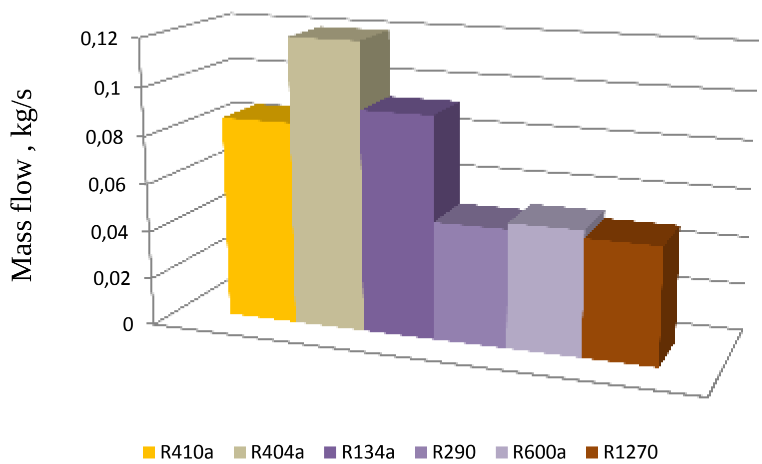

***Figure 5** – Mass flow rate chart for considered refrigerants*

VI. CONCLUSIONS

«Energy Strategy of Ukraine for the period until 2030» provides radical changes in the structure of heat sources. The main factor that causes these changes is a sharp increase in world prices for natural gas, oil and petroleum products. Therefore, it is forecasted the gradual replacement of gas boilers and CHP plants majority that currently provide the vast proportion of the thermal energy production with new technologies, including the use of environmental heat, and especially heat pumps. The conducted research of the heat pump characteristics allowed us to analyze efficiency of the ground-source heat pump on environmentally friendly refrigerants. The heat pump characteristics on natural refrigerants indicate high performance with low environmental impact (ODP=0, GWP < 2). The discharge temperature analysis shows advantages of R410 and R1270 for high temperature applications (domestic hot water production). Refrigerant choice for analysis was based on its availability on the market and long term use prospective, so natural refrigerants R290, R600a, R1270 are more preferable. Selection of the working fluid of the heat pump for a particular purpose (space heating, domestic hot water production, etc.) can achieve a significant increase in the performance of the equipment and energy systems as a whole.

NOMENCLATURE

W_{net}	Work input rate to the compressor (kW)	$c_{p,b}$	Specific heat of brine (kJ/kg °C)
m_r	Mass flow rate of the refrigerant (kg/s)	$c_{p,w}$	Specific heat of water (kJ/kg °C)
m_w	Mass flow rate of the water (kg/s)	$T_{w,o}$	Outlet temperature of water from condenser (°C)
m_b	Mass flow rate of the intermediate coolant (kg/s)	$T_{w,i}$	Inlet temperature of water to the condenser (°C)
η_{isn}	Isentropic efficiency	$T_{b,o}$	Outlet temperature of brine from evaporator (°C)
η_e	Electric engine efficiency	$T_{b,i}$	Inlet temperature of brine to the evaporator (°C)
η_m	Mechanical efficiency of compressor	h	Enthalpy (kJ/kg)
W_{net}	Power consumed by compressor (kW)	h_{2s}	Isentropic enthalpy of the refrigerant (kJ/kg)
K_c	Total heat transfer coefficient of condenser (kW/m^2 °C)	ΔT_{lm}	Logarithmic temperature difference
A_c	Total heat transfer area of condenser (m^2)	Q_r	Rejected heat in condenser (kW)
F_c	Correction coefficient of condenser ($F_c=1$)	Q_c	Heat of condensation (kW)
K_e	Total heat transfer coefficient of evaporator (kW/m^2 °C)	Q_e	Heat transfer rate in evaporator (kW)
A_e	Total heat transfer area of evaporator (m^2)	COP	Coefficient of performance
F_e	Correction coefficient of evaporator ($F_b=1$)		

REFERENCES

1. **Jyoti Soni, R.C. Gupta** Exergy Analysis of Vapour Compression Refrigeration System with Using R-407C and R-410A//*International Journal of Engineering Research & Technology (IJERT)*, Vol. 1 Issue 7, September - 2012.

2. **Rotchana N. Prapainop, K. Suen O.** Effects of refrigerant properties on refrigerant performance comparison: A review / *International Journal of Engineering Research and Applications (IJERA)*// Vol. 2, Issue 4, July-August 2012, pp.486-493

3. **Devotta, S**. Alternative heat pump working fluids to CFCs. *Heat Recovery Systems and CHP*, 1995. 15(3): p. 273-279.

4. **Lee, H., J. Yoon, Kim, P.K. and Bansal I**, Characteristics of condensing and evaporating heat transfer using hydrocarbon refrigerants. *Applied Thermal Engineering*, 2006. 26: p. 1054-1062.

5. **Rajapaksha, L.**, Performance evaluation of reversible heat pumps when using refrigerant mixtures. 2003, University College London: London

6. **Granryd, E.**, Hydrocarbons as refrigerants, an overview. *International Journal of Refrigeration*, 2001. 24(1): p.15-24.

7. UNEP. Handbook for the Montreal Protocol on Substances that Deplete the Ozone Layer. 2006 [cited;: [Available from: http://ozone.unep.org/ Publications /MP_Handbook/index.shtml.]

8. **Wen, M. and Ho C.** Evaporation heat transfer and pressure drop characteristics of R290 (propane), R600 (butane) and a mixture of R290/R600 in the three-lines serpentine small-tube bank. *Applied Thermal Engineering*, 2005. 25: p. 2921-2936.

9. **Gosney, W.B.**, Principles of refrigeration. 1982, Cambridge: Cambridge University Press.

10. **Woollatt, D.** Estimating valve losses when dynamic effects are important. *Proceeding of Purdue compressor technology conference 1982*: Purdue university

11. **Goetzler W., Zogg R., Lisle H., Burgos J.** Ground-Source Heat Pumps: Overview of Market Status, Barriers to Adoption, and Options for Overcoming Barriers. 2009, - Navigant Consulting, Inc. – 14-18 p.

12. **Angelino L., Dumas P., Latham A.** EGEC Market Report Update December 2014. *European Geothermal Energy Council*. – 2014. 24p.

13. GROUND-REACH. Inventory of EU Legislation on Ground Coupled Heat Pumps (GCHPs). 2008. – 164p.

14. EUROPEAN STANDARD NBN EN 14511-2 (2008), Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – *Part 2: Test conditions*, approved by CEN on 12 October 2007

15. Final Report IEA HPP Annex 28. Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating. 2005. – 114p.

16. **Winandy, Eric and Hundy, Guy**, "Refrigerant and Scroll Compressor Options for Best Performance of Various European Heat Pump Configurations" (2008). *International Refrigeration and Air Conditioning Conference*. Paper 895. http://docs.lib.purdue.edu/iracc/895

17. **Kharseh M**. Ground-Source Heat Pumps and Energy Saving, Heat Exchangers - Basics Design Applications, 2012. – Dr. Jovan Mitrovic (Ed.), ISBN: 978-953-51-0278-6, InTech, 459-476p. Available from:http://www.intechopen.com/books/heat-exchangers-basics-design-applications/ground-source-heat-pumpsand-energy-saving

Received 03 October 2015
Approved 03 November 2015
Available in Internet 25 December 2015

