PROCEEDINGS, Thirty-Sixth Workshop on Geothermal Reservoir Engineering
Stanford University, Stanford, California, January 31 - February 2, 2011
SGP-TR-191

THERMODYNAMIC ANALYSIS OF CYCLES OF THE CASCADE HEAT PUMP PLANT

Redko Andriy
Donbass National Academy of Construction and Architecture
Short address
Makiivka, Donetsk, 86123, Ukraine
e-mail: andrey.ua-mail@rambler.ru

ABSTRACT

The results of numerical studies of thermodynamic effectiveness of the cascade thermal scheme of the heat pump plant are presented. The cycles` optimum parameters are defined for working substances – R407C and R134A refrigerants in the temperature range 20°- 105°C (68° - 221°F). The possibility of using heat pump plants in heating systems of traditional buildings in the temperature regime of 90°/70 °C (194°/158°F) in the presence of geothermal heat supply is shown.

INTRODUCTION

Currently, widespread adoption of heat pump plants (HPP) in Ukraine, Russia and other Eastern Europe countries is constrained by the construction of new residential houses, administrative and public buildings (as well as in-use buildings) with a traditional system of water heating in the temperature regime of 90°/70°C (194° - 158°F). Single-stage HPPs type "air-to-water", "water-to-water" with ground heat exchanger, effectively starting to be used in low-temperature heating, mainly in cottage construction.

Rising the condensation temperature of the working substance in the HPP up to 90° - 95°C (194° - 203°F) is possible using, for example, sequential thermal scheme of HPP inclusion, two-stage and combined HPPs. [2-5, 7, 11-15]. Application of HPP requires the development of thermal schemes with a high-temperature coolant for the plant`s operation in the temperature regime of 85°/60 °C (185°/140°F) in existing heating system.

STATE OF THE PROBLEM

Existing needs and the demand for high-temperature HPPs pose problems for their development. Various coolants – organic liquids and their mixture, freon and mixtures thereof, ammonia, ammonia-water mixture, carbon dioxide and others uses in the development of HPPs as working substance. Freons of new generation - non-ozone depleting refrigerants are in common practice. When making use of geothermal energy, temperature voltage of waste geothermal fluid with temperature 15°-30°C (59° - 86°F) can be increased using HPP [3]. Choosing the most effective working substances for a specific temperature limits can be achieved by carrying out multivariate numerical studies.

In [5, 12, 14] the results of investigations of thermodynamic effectiveness of a single-stage HPP cycles with different working substances. In the papers mentioned above [3-5, 12-13, 15] were identified prospects for a two-stage compression system and binary heat pumps with two coolants. In cryogenic technology to achieve low temperatures are using HPP with double-and triple-cycles [7, 8].

In comparative evaluation of the efficiency of thermodynamic cycles with different working substances (R134A, R22, R21, R142B refrigerants) can be seen that the condensation temperature rises to 65°-70°C (149°-158°F) and even 80°-95°C (176° - 203°F) with the temperature in the supply of low-temperature heat source [5], that enables the regime of heat supply with the temperature regime 90/70°C (194°/158°F) . However, the evaporation temperature of the working substance must have a value of about 50°C (122°F) and above. As the results of the calculations, the working substance is effective in a range between vaporization temperature and condensation temperature of about 20-25°C (68°-77°F). Cycles with two-stage (multi-stage) compression and single working substance, or cascading (multi-stages) cycles and multiple working substances are effective in a wide temperature range.

OBJECTIVE

Studies are of optimal parameters of thermodynamic cycles` heat pump plants with multiple coolants.

MAIN RESULTS

Optimization results of the thermodynamic parameters cycles of the cascading HPP type "water-to-water" at a vaporization temperature of 20-25°C (68°-77°F) and at condensation temperature of 80-95°C are presented in the paper. Ozone-friendly refrigerants (R134A, R142B, R152A, R227EA, R245FA, R410, R407C, R404C and other) were studied as a working substance. We considered various combinations of the working substances in the lower and upper cycles of the cascade HPP. R134A, R142B, R404A, R407C, R410A were studied more fully from the working substances considered by us. The results are of a comparative evaluation of thermodynamic effectiveness of the two-stage thermal scheme and the cascade (multi-stage thermal scheme of the HPP (Fig.1). Refrigerants R407C, R410A, R134A, and R142B were studied as working substance.

Thermal scheme of two-stage HPP with a working substance (refrigerant R407C) is shown in Figure 1.

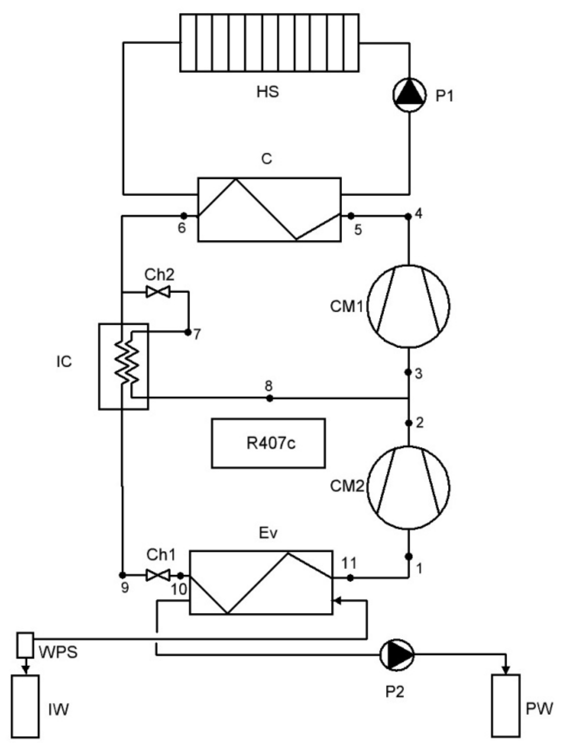

Figure 1. Thermal scheme of two-stage HPP with an intercooler
CM1, CM2 - compressor, IC. - intercooler; C. – Capacitor; Ev. – evaporator; Ch – choke, P1 – circulation pump, P2- force-pump, HS - heating system, PW, IW - production and injection wells, respectively, WPS - water purification system.

Thermal scheme of a cascade HPP is shown in Figure 2. The scheme sets out the calculated point of the cycle (Figure. 3)

Cascade HPP cycles are shown in Figure 3.

Geothermal fluid is cooled in the 9-10 the temperature from 35°C (95°F) to 26°C (78.8°F). The HPP transfers the heat in the condenser in the 11-12 to the heating-system water in the temperature range of 90/70°C (194°/158°F). Adiabatic compressor efficiency is assumed to be η_{ad}=0,80 and the product of mechanical efficiency to electrical efficiency of the wire is $\eta_{m.e.}$=0,95.

As a result of executed numerical variant computations of HPP's heat engineering parameters it was determined that the optimal working substance for lower cycle – refrigerant R407C and for upper – R143A. The vaporization temperature and condensation temperature are considerably affect the value of the conversion coefficient, because when t_{vap}=25°C (77°F), t_{con}=95°C (203°F) the conversion coefficient is φ=2,58, and for t_{con}=80°C - φ =3,44; value of the minimal temperature difference in the evaporator, condenser, intermediate evaporator, the pressure of coolant at the outlet of the compressor and other parameters.

The results of the calculation of heat engineering characteristics of HPP following:

- evaporation temperature in the lower cycle - 25°C (77°F)
- evaporation temperature in the upper cycle – 67.4°C (153.32°F)
- condensation temperature in the lower cycle – 72.4°C (162.32°F)
- condensation temperature in the upper cycle – 104.2°C (219.56°F)
- cooling capacity in the lower cycle – 133.5 kW
- heat productivity in the lower cycle – 159.2 kW
- cooling capacity in the upper cycle –159,2 kW
- heat productivity in the upper cycle – 188,3 kW
- compressor power in the lower cycle – 25,7 kW
- compressor power in the upper cycle – 29,1 kW

The differential pressure in the compressors:

- in the lower cycle – 1416.8 kPa
- in the upper cycle – 1359.1 kPa
- conversion coefficient – 3.44
- refrigeration coefficient – 1.41
- coefficient of thermal efficiency – 0.39

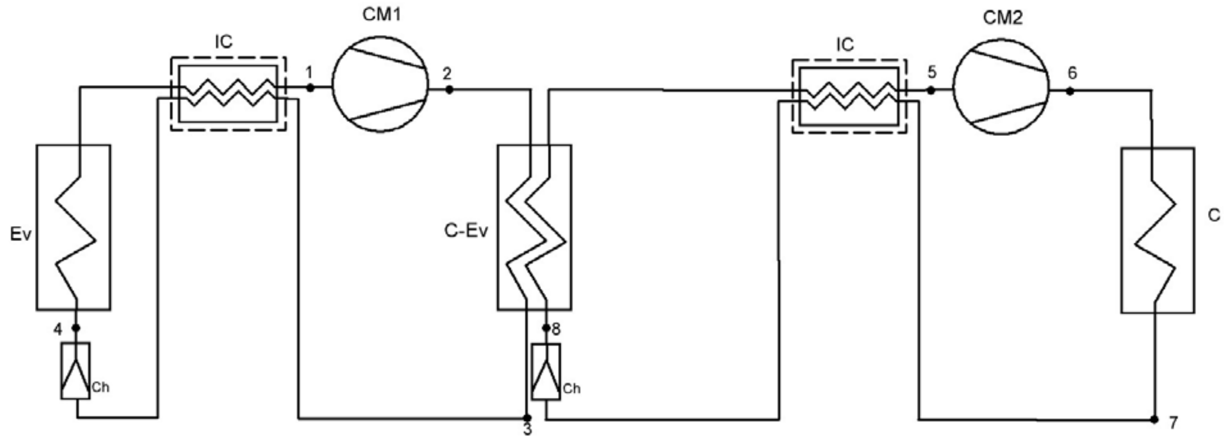

Figure 2. Thermal scheme of a cascade HPP
CM1, CM2 - compressor, IC - intercooler; C. – Capacitor; Ev. – evaporator; Ch – choke

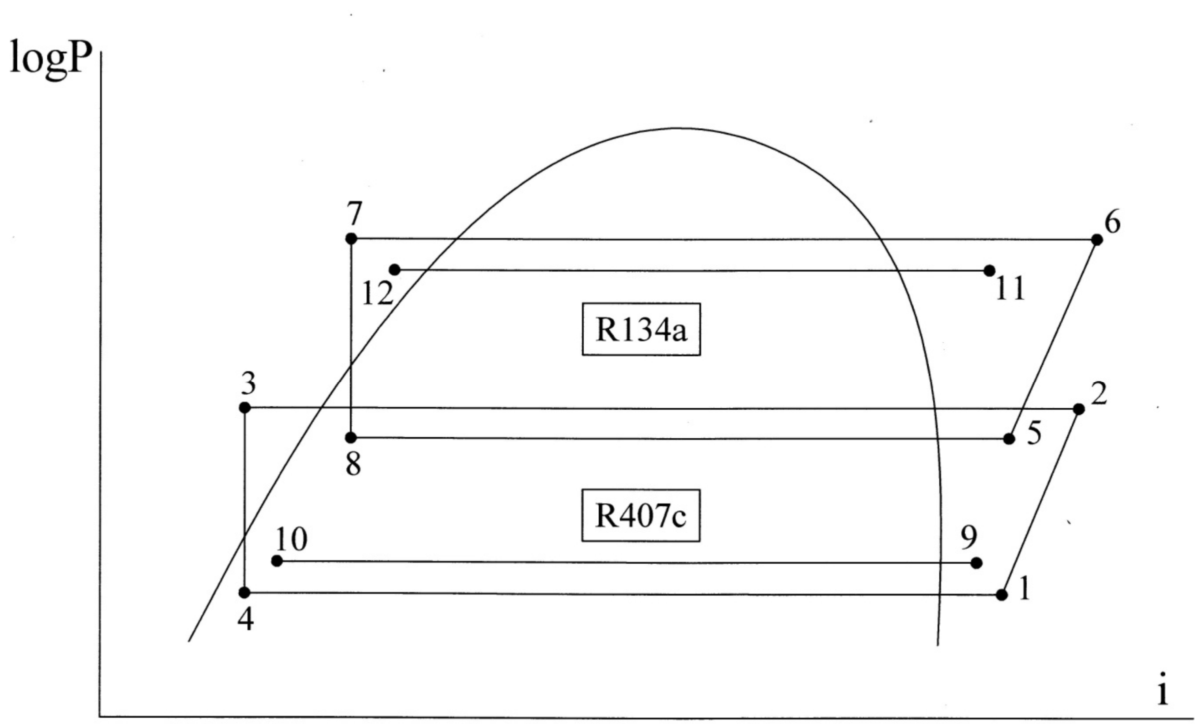

Figure 3. Cycles of the cascade HPP

Calculations were performed with the restriction of the differential pressure in the compressor.

The calculations were performed for the cascade HPP with refrigerant R404A in the lower cycle, shows that under the similar conditions, the conversion coefficient is somewhat lower and is equal to φ=3,17.

The conversion coefficient of two-stage HPP is below 3.0 at a vaporization temperature of 20°C (68°F) and condensation temperature of 100-105ºC (212°-221°F).

The thermodynamic cycle’s perfection of cascade HPP is determined by energy losses in individual elements of the thermal scheme of HPP. We calculated the irreversible energy losses, by the entropy method [4]. The energy balance of HPP are formulated, energy loss are defined for the separate units. Analysis of energy losses revealed units with high losses and the need to improve them.

Also, defined the exergy efficiency of the HPP (1):

$$\eta_{ex} = \frac{Q_{C_2}\tau_C}{W_1 + W_2 + Q_{ev}\tau_{ev}}, \qquad (1)$$

where, τ_c, τ_{ev} - Carnot temperature factor for the evaporator and condenser respectively, Q_{c_2}, Q_{ev} - heat productivity in the upper and lower cycles, W_1, W_2 - power in the upper and lower cycles, respectively. The exergy efficiency of HPP is equal to $\eta_{ex} = 0.626$

CONCLUSIONS

Performed numerical studies show the effectiveness of the thermal scheme of the cascade HPP. The cycles` optimal parameters are defined for working substances –R407C and R134A refrigerants in the temperature range of 20°C (68°F) to 105°C (221°F), which shows the possibility of using HPPs in traditional high-temperature heating systems of buildings in temperature regime 90/70°C (194°/158°F) in the presence of geothermal heat supply. Comparative analysis of thermodynamic effectiveness of the two-stage and cascade HPP`s thermal schemes shows the advantage of the latter.

REFERENCES

1. Denisov A.E (2003). *Integrated system of alternative heating for energy saving technologies:* Abstract. Dis. for the scientific. stage Dr. Tekhn. Science: special. 05.14.06 "Engineering Thermophysics and industrial heat-and-power engineering", 36.
2. Dilevskaya E.V.(1978) *Cryogenic micro heat exchangers*, 165.
3. *Geothermal heat and cold-supply of residential and public buildings and constructions.*(1989) Design Rules: VSN 56-87. - [Effective from 01.07.1988], (Departmental building codes), 70
4. Goshteyn D.P (1951). *Entropy method for calculating the energy loss*, 111.
5. Martynovsky V.S. (1959) *Thermodynamic characteristics of cycles of heating and refrigeration*, 162
6. Morozyuk T.V (2006). *Theory of refrigeration machines and heat pumps*, 721
7. Ogurechnicov L.A. (2005) “Geothermal resources in the energy”, *Alternative Energy and Ecology*, **11,** 58-66
8. Redko A.A, Krasnopolsky V.A., Redko I.A., Kharlampidi D.H (2008), “Stepped heating system with heat pumps”, *Utilities cities*. **84**,155-158
9. Redko A.A., Kharlampidi D.H (2008) “Analysis of modes of geothermal heat pump plants”, *Nova theme*. **4**, 21-25.
10. Redko A.A., Kharlampidi D.H (2009) Studies of the thermodynamic regime of geothermal heat pump plants:, *Bulletin of the Donbas National Academy of Engineering and Architecture*. - **2 (76)**, 86-98.
11. Redko A.O, Kharlampidi D.H (2008) “Methods of calculation of geothermal heat pump plants”, *Ventilation, lighting and heat gas supply*, **12,** 80-91.
12. Rosenfeld, L.M. (1952.) “Thermodynamic theory of dynamical heating by refrigeration machine”, *Journal of Technical Physics*. **8, 22,** 18-27.
13. Sokolov E.Y., Brodyanskiy V.M. (1981) *Energetic basis of transformation of heat and cooling processes*, 320 .
14. Stein L.L., Bychkov A.A. (1973) Construction and test results of the MKS at 4.2 K on the basis of the heat pum, *Abstracts I All-Union Conference on Cryogenic Engineering*, 143-144.
15. Yantovsky E.I., Pustovalov Y. V. (1982) *Vapor compression heat pumps*, 144.