УДК 62-971.4

Ориентировочный расчет теоретического цикла парокомпрессионного фреонового контура воздушного теплового насоса

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В целях повышения эффективности эксплуатации воздушного теплового насоса актуальным является анализ термодинамических параметров цикла парокомпрессионного фреонового контура. Для того чтобы оценить все термодинамические процессы в теплохолодильной системе воздушного теплового насоса и произвести расчеты, как правило, используют тепловые диаграммы, разрабатываемые производителями. Авторы предлагают использовать для ориентировочного расчета парокомпрессионного фреонового контура таблицы на линии насыщения и линии перегретого пара рабочего хладагента. В результате расчета теоретического парокомпрессионного цикла были получены значения тепловой энергии, изымаемой рабочим телом в процессе его непрерывного фазового превращения: кипения, испарения, конденсации, которые определялись по точкам состояния энтальпии на соответствующих участках работы элементов ВТН (испаритель, компрессор, конденсатор). Расчеты показали, что при создании тепловой мощности на выходе теплового насоса определяющим параметром является фазовое превращение скрытой теплоты парообразования при кипении и конденсации рабочего тела в замкнутой системе компрессионного цикла.

Ключевые слова: парокомпрессионный цикл, фреоновый контур, энтальпия, воздушный тепловой насос

Для цитирования:
Федосов С.В., Федосеев В.Н., Логинова С.А., Зайцева И.А. Ориентировочный расчет теоретического цикла парокомпрессионного фреонового контура воздушного теплового насоса. *Умные композиты в строительстве.* 2021. Т. 2. №. 4. С. 24-34 URL: http://comincon.ru/index.php/tor/V2N4_2021

DOI: 10.52957/27821919_2021_4_24

Approximate calculation of a theoretical cycle of a vapor-compression freon loop in an air heat pump

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To improve the efficiency of using an air heat pump (AHP), it is important to analyze the thermodynamic parameters of the vapor-compression freon loop cycle. To evaluate all thermodynamic processes in the heat recovery system of the air heat pump and to make calculations, experts frequently use heat charts developed by the manufacturers. The authors propose to use tables on saturation line and superheated vapor line of working refrigerant for approximate calculation of vapor-compression freon loop. As a result of calculation of a theoretical vapor-compression cycle, the authors obtained heat energy values taken by the working body during its continuous phase change: boiling, evaporation, condensation, which were determined by the enthalpy state points at the corresponding sections of operation of AHP elements (evaporator, compressor, condenser). Calculations showed that at creating thermal power at the heat pump output, its determining parameter is the phase change of latent heat of vaporization at boiling and condensation of working body in the closed compression cycle system.

Key words: vapor-compression cycle, freon loop, enthalpy, air heat pump

For citation:
Fedosov, S.V., Fedoseev, V.N., Loginova, S.A., Zaitseva, I.A. Approximate calculation of a theoretical cycle of a vapor-compression freon loop in an air heat pump. *Smart Composite in Construction*. 2021. Vol. 2. No 4. P. 24-34 URL: http://comincon.ru/index.php/tor/V2N4_2021

DOI: 10.52957/27821919_2021_4_24

INTRODUCTION

The basis of modern research in increasing the energy efficiency of air heat pump (AHP) operation is the analysis and improvement of heat and mass transfers occurring in the thermodynamic cycle of the freon loop. Thermodynamic perfection of the vapor compression cycle to a large extent determines technical, economic and ecological efficiency of AHP. This is especially important when developing innovative environmentally friendly technologies for thermal energy generation [1].

The main advantage of compression-type heat pumps is the great efficiency among the latest heat pumps. Their proportions of received and given energy can reach 1:7 (at feeding 1 kilowatt of electric energy to the heat pump compressor, 7 kilowatts of heat will be extracted from cooled zone) [2, 3]. The operating mode of an AHP changes due to the environment, which, as a consequence, determines its thermal power output.

Heat and mass transfer within thermodynamic processes might intensify by converting different in quality energy flows in the freon loop. Science community actually has no materials on heat exchange that takes into account non-stationarity of processes in working zones of heat pump, and also on non-stationary transfer of thermal energy by mass transfer [3].

To use the heat pump efficiently, both internal and external process conditions should be considered [4]. For example, to calculate heat conversion under real non-equilibrium conditions, one should have knowledge of properties and parameters of working substances in different states.

Freons used for heat transformation in AHP are usually studied by manufacturers at different pressures and temperatures, and their parameters are presented in diagrams or tables [5].

A common idea is that the most common way to calculate the thermodynamic cycles of AHP is to use diagrams of states of working substances:

- diagrams s–T (entropy–temperature);
- diagrams p–V (pressure–specific volume);
- diagrams $i(h)$–P (enthalpy–pressure) or $i(h)$–lg$P$.

The vapor-compression cycle of the refrigeration machine can be calculated without using thermal diagrams [6]. Instead, the calculation may involve tables on the saturation line and superheated vapor line of the refrigeration agent (refrigerant).

For such calculation of heat conversion and evaluation of AHP efficiency, it is required to know the properties and parameters of refrigerants in different states [5]. The thermodynamic and thermophysical characteristics of refrigerants are considered in three aspects: in terms of influence on thermodynamic cycle efficiency, on AHP performance indicators, and on compressor design characteristics.

EXPERIMENT

Calculation of the vapor-compression cycle according to the specified external conditions – parameters of the low-temperature source (ambient air) and the level of the coolant (water) heating – is selected taking into account environmentally safe refrigerants used in AHPs.

The method to calculate the vapor-compression cycle of the freon loop of AHP is based on the transformation (transfer) of heat due to phase transitions of the working substance (freon): liquid – vapor-liquid mixture – gas [7]. These processes of continuous phase change of the working body – boiling, evaporation, condensation – can be determined by the so-called reference points of enthalpy state at the corresponding sections of operation of the AHP elements (evaporator, compressor, condenser).

For example: on the circuit diagram (Fig. 1), let us represent as points all thermodynamic processes of vapor-compression cycle of evaporation and condensation unit for heat pump A20/W30 with average heat power 7 kW and electric power 1.84 kW [8], where A – ambient air at 20 °C, W – heat carrier in heating circuit, water temperature at 30 ˚C.

Next, we present the calculation of enthalpies by control points corresponding to certain sections of the cycle.

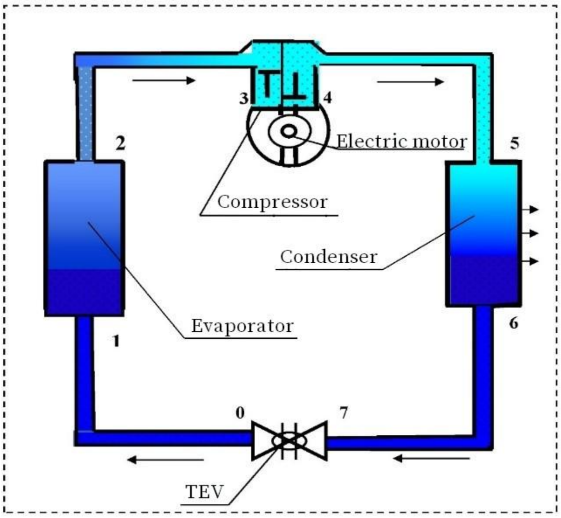

Fig. 1. Circuit diagram of the vapor compression cycle and working sections of the evaporation and condensation unit of the heat pump system

In accordance with ISO/DIS 17584, Table 1 presents thermodynamic index values, which is enthalpy taken on liquid line and saturated vapor line on the working sections of the freon loop vapor-compression cycle [8].

Table 1. Enthalpy values on the line of saturation and superheated vapor line, kJ/kg [8]

Freon	h_1	h_2	h_3
R22	249.81	416.69	452.3
R32	275.9	514.3	560
R134A	265	413	445
R404A	262.7	387.2	420
R407C	272	425	465
R410A	270.4	436.2	470
R507A	260.4	383.1	420

Table 1 shows that the highest heat transfer (h_2) will be observed at section 3-4. This happens because of the processes of irreversibility inside the compressor [9]. The enthalpy increase in it is greater than in the ideal cycle, and this, in turn, leads to an increase in temperature and pressure.

In reference tables of thermodynamic properties [5, 10], enthalpy values are given because enthalpy as an auxiliary function in thermodynamic calculations is how heat is supplied in heat exchangers (evaporator, condenser) at constant pressure.

RESULTS

To study the cycle, it is convenient to start at the point where the refrigerant is in a slightly supercooled liquid state. This is because the location of this point tends to vary little regardless of the various modifications to the main cycle.

Control point 1 of section 0-1 (liquid phase) – the liquid refrigerant at the evaporator entrance displaced through the throttling device (TEV – thermostatic expansion valve).

The amount of useful energy withdrawn from the environment (air) from the initial cold circuit and required to transform into vapor the liquid obtained at the inlet to the evaporator and the freon loop line (q_1), is

$$q_1 = h_2 - h_1. \tag{1}$$

Section 1-2. The liquid remaining in the evaporator boils at a pressure close to atmospheric pressure with temperature t_2 = +5 °C. It then evaporates in a «cold» heat exchanger, absorbing heat and therefore cooling the medium, which has a higher temperature than the boiling point for this pressure, taking away the heat of vaporization. The design of the evaporator has a decisive influence on the freon boiling. With most of the gas being evacuated by the compressor, the pressure does not rise.

Section 2-3. The gas refrigerant from the evaporator with temperature t_2 = +5 °C is sucked in by the compressor.

Reference point at section 3-4 (gas). Then the compressor, pumping out the gas refrigerant, compresses it adiabatically with temperature increase up t_3 = +40 °C and higher with much higher pressure P_2 = 15–17 bars [12].

The compressor exerts force over freon when compressing, which according to the first law of thermodynamics increases thermal energy of freon flow circulating in an AHP loop.

According to the second law of thermodynamics, energy transfer in the heat pump cycle continues under the influence of mechanical and electrical energy of the compressor drive, which uses electrical energy to compress the gas refrigerant, resulting in increased pressure and temperature of the refrigerant.

The heat dissipation from the compressor motor can be calculated using the formula

$$Q\text{motor} = N(1 - \eta)/\eta, \text{kW}, \tag{2}$$

where N – motor power, kW; η – efficiency.

During heat exchange between the working body and the compressor, the amount of energy released by our compressor with an electric motor power of 1.84 kW is

$$Q\text{motor} = 1.84 \cdot (1 - 0.8)/0.8 = 0.46 \text{ kW}.$$

Heat dissipation from the compressor can be calculated according to the Bitzer recommendations [13].

The amount of energy used for compression, received at the compressor outlet from the freon loop (q_2), is

$$q_2 = h_3 - h_2. \tag{3}$$

Section 4-5. The compressor forces the gas into the condenser where it condenses at a high constant pressure and a high operating temperature from t_3 = +40 °C to t_3 = +50 °C, giving the heat energy to the heat transfer medium at the desired consumer heating temperature.

Section 5-6. The TEV regulates the flow of liquid, thereby reducing the release of vapor. The fluid expands in the valve but does not exchange energy with the environment. The expansion takes place at a constant enthalpy. Nothing happens during pressure reduction. The temperature is virtually unchanged. When the fluid pressure reaches saturation, a further decrease in pressure also implies a decrease in temperature, since otherwise the fluid overheats, which implies the formation of a thermodynamically unstable state [14].

As a result of cooling the liquid, energy is released, which is spent on the evaporation of part of the liquid. Consequently, liquid evaporation in this case depends on pressure: the smaller it is, the more liquid evaporates.

Section 6-7. The fluid pressure reaches its final value. The volume of liquid evaporated is determined by means of equal concentration lines. In this example, the refrigerant expands to a 5-bar pressure (–5 °C) with a 50% vapor concentration.

At section 5-6-7 from condenser, thermostatic expansion valve (TEV) to evaporator inlet, we assume that TEV in this case only creates conditions to change the state of vapor-liquid mixture at evaporator inlet under different temperatures of working medium, environment, and pressure. Since TEV regulation in this process is the reverse process of the compressor operation mode, the TEV does not have any additional effect on freon, everything is done by the compressor. The TEV only separates the pressure zones with the required freon capacity. Therefore, the connecting tubes of the TEV must be matched to the compressor capacity [15].

When generating the heat output from a heat pump, two parameters matter the most:

- phase transformation of latent heat of vaporization at boiling and condensation of working medium in a closed system of compression cycle,

- dynamics of influence of air flows on thermophysical and thermodynamic properties of freon, processes occurring in it.

The heat and cooling efficiency derived from the freon loop (COP) is

$$COP = q_1/q_2. \tag{4}$$

Table 2. Values of the energy received from the freon loop, at different sections of the vapor-compression cycle of AHP and COP

Freon	q_1, kJ/kg	q_2, kJ/kg	COP
R22	166.88	35.61	4.69
R32	238.4	45.7	5.22
R134A	148	32	4.63
R404A	124.5	32.8	3.80
R407C	153	40	3.83
R410A	165.8	33.8	4.91
R507A	122.7	36.9	3.33

To estimate all thermodynamic processes in the refrigeration system and to make calculations, we use thermal diagrams or approximate calculation of the vapor-compression freon loop according to tables on the saturation line and the superheated vapor line of the working refrigerant [16].

Due to the wide spread of vapor-compression heat pumps, all options to increase their efficiency are being considered. The first option is to increase the efficiency of a theoretically equilibrium ideal cycle. Another option is to reduce losses from non-equilibrium (irreversibility) of processes.

It is important to properly select a working medium (refrigerant) and optimal parameters of vapor-compression cycle. An important property of freons is the heat of phase transition. The higher the value, the lower the freon consumption and therefore the lower the cost of the compressor. On the other hand, AHP performance can also be assessed by two indicators: mechanical efficiency (compressor drive) and volumetric efficiency (compressor chamber size) [15-17].

In a real cycle, unlike ideal cycle, the processes of expansion and compression of a working body are considered irreversible and they do not follow an adiabatic curve, but constant enthalpy line – polytropically, which leads to some increase in energy losses not only in compressor, but also in evaporator and condenser. Therefore, it is important to consider the design, piping (trunking) configuration, and location of the main elements of the air heat pump.

CONCLUSIONS

As an AHP is a complex device with advanced automation, the heat pump system requires proper calculation of the operating mode and the selection of basic structural elements by qualified designers and competent installers. It should be noted that the result of the calculation of an AHP will always depend on the conditions in which it is operated.

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Поступила в редакцию 29.11.2021
Принята к опубликованию 06.12.2021

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Received 29.11.2021
Accepted 06.12.2021