DEPARTMENT OF POWER SYSTEMS AND ENVIRONMENTAL PROTECTION FACILITIES

Thermodynamics

Laboratory classes

Heat pump cycle

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Aim of the exercise

This exercise aims to familiarize students with the different types of thermodynamic cycles, as well as define and determine the coefficients of performance for both power cycles (heat engines) and reverse cycles (refrigeration cycles and heat pumps). The aim of the exercise is achieved through the measurement and energy analysis of an air-to-water compression heat pump, the energy balance of the heat pump components, and the determination of the coefficient of performance ε_{tpc} for the Linde cycle.

1. Thermodynamic cycles

A thermodynamic cycle we call the process (or the cycle of processes) in which thermodynamic parameters of a thermodynamic agent at the initial state and the final state are the same. Thermodynamic cycles are used for the analysis of work of real devices working continuously or cyclically in which the thermodynamic agent every now and again comes back to the same state. For such devices one can define some reversible, comparative cycle, which enables to approximately valuate the quality of work of the device. This rough guess results from the fact, that each comparative cycle is always an ideal cycle which one cannot fully realize in reality and also from this, that in some devices (e.g. in internal combustion engines) take place chemical reactions which in a cycle (for accordance with its definition) one replaces with a suitable thermodynamic transformation.

A thermodynamic cycle can be realized in a closed system or in a suitable set of interconnected devices through which the mass can flow (the so called open system).

A thermodynamic cycle is reversible if all processes included are reversible and the heat exchange between a thermodynamic agent inside the system and its surroundings takes place at an infinitesimal temperature difference, $dT = 0$. If any of these conditions is not fulfilled, it makes the cycle irreversible (any irreversibility makes the whole cycle irreversible).

In every cycle there must appear a compression of the agent owing to the **compression work** l_{com} led to the system, and an expansion of the thermodynamic agent causing the return of the **expansion work** l_{exp} by the system. The net work (work of the cycle – l_{cycle}) determines the difference of the expansion work and the compression work:

$$l_{cycle} = l_{\exp} - |l_{com}| \tag{7.1}$$

Simultaneously, in order to perform the works of expansion and compression, to the circulating agent in the cycle the heat must be led q_{in} and given back from it q_{out}. From the definition of the thermodynamic cycle (the identity of the start state and the final state of the circulating agent) results, that the growth of the internal energy of the system in which this cycle is realized (as well as of all other parameters and functions of state), equals zero ($\Delta u = 0$). So – aside from the reversibility or irreversibility of the cycle – from the equation of the First Law of Thermodynamics results, that for the device realizing the cycle the energy balance has the following form (the net work output equals the net heat transfer to the cycle):

$$q_{in} - |q_{out}| = l_{cycle} \tag{7.2}$$

where:

q_{in} – represents heat transfer into the system (heat supplied),

q_{out} – represents heat transfer out of the system (heat released, also called; heat rejected or waste heat).

or the same for the closed system (with control mass):

$$Q_{in} - |Q_{out}| = L_{cycle} \tag{7.2a}$$

Notice – the modulus for the heats and the works results from the signs convention of heat and work (see subsection 4.2.2 – Fig. 4.1).

In consideration of the sign of the work l_{cycle}, thermodynamic cycles can be divided into:

- **Clockwise** (right-handed) – in which the change of state of the circulating thermodynamic agent progresses clockwise (independently from the coordinate system, wherein we demonstrate the cycle). For these cycles $q_{in} > |q_{out}|$ so, the net work $l_{cycle} > 0$ and hence the clockwise cycles are also called heat engines cycles or power cycles. One can say that the clockwise cycle (power cycle) converts heat into work.
- **Counter-clockwise** (left-handed) – in which the change of state of the circulating thermodynamic agent progresses counter-clockwise (also independently from the coordinate system). For these cycles $q_{in} < |q_{out}|$ so, the net work $l_{cycle} < 0$ and hence the counter-clockwise cycles are called also cycles of working machines (refrigerators and heat pumps). One can say that the counter-clockwise cycles (refrigerators or heat pumps cycles) convert work into heat.

The closed curve of the clockwise cycle is presented in figure 7.2a and of the counter-clockwise cycle in figure 7.2b, where the works executed during the cycle and the heat supplied and released in progress of the cycle are marked.

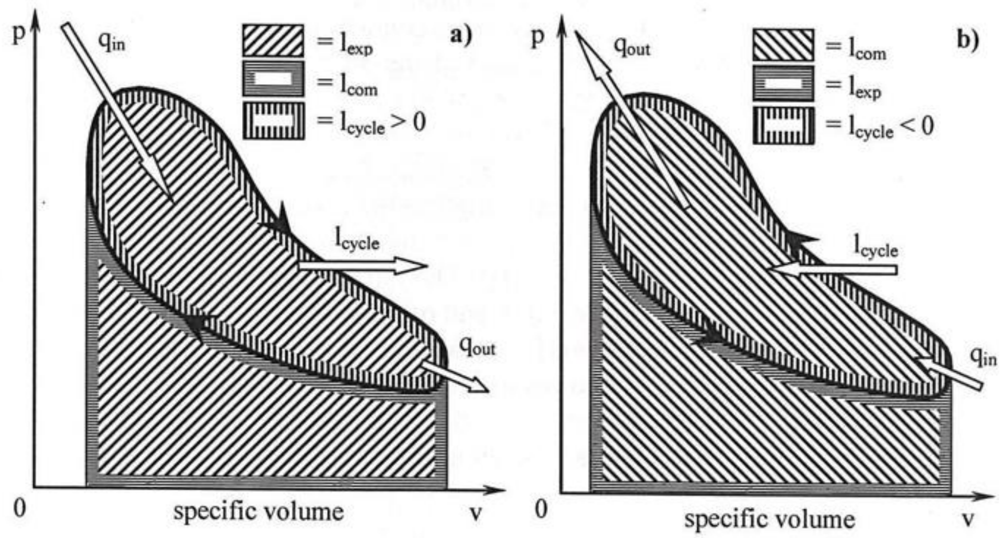

Fig. 7.2. Thermodynamic cycles in the *p–v* coordinate system: a) a clockwise cycle (heat engine); b) a counter-clockwise cycle (working machines – refrigerators and heat pumps)

As it can be seen, the expansion work of the clockwise cycle is greater than the compression work, hence the positive net work (obtained owing to the fact that the heat supplied is greater than the heat released) can be transferred to the surroundings and used, e.g. for propulsion of a working machine. Hence the name "heat engine cycle". Instead, the counter-clockwise cycle acts inversely and the expansion work is lesser than the compression work.

In consequence the net work is negative, that is to say, the work must be led to the device realizing this cycle. So, at the expense of work led into the system, the device gives back to the surroundings more heat than it receives from it. Hence, the name "working machine cycle", which refers to refrigerators and heat pumps.

In every cycle, both clockwise and counter-clockwise, there can be distinguished four characteristic points, as shown in figure 7.3.

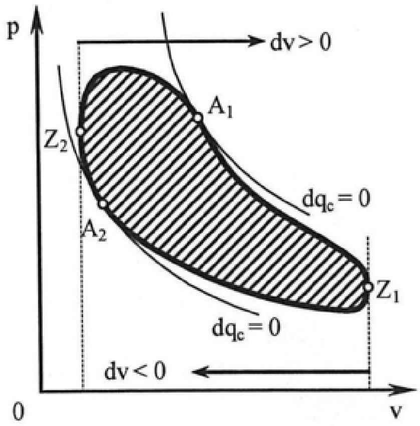

Fig. 7.3. Characteristic points of cycle in the *p–v* coordinate system. Z_1 and Z_2 – dead centers, A_1 and A_2 – adiabatic points

Independently from the kind of cycle which the device is realizing, from the above considerations results that within the surroundings contacting with the system, it is necessary to distinguish two heat sources. From one of the sources the heat is supplied to the system and to the second one it is rejected. The dead centers (shown in Fig. 7.3) divide the cycle into two parts; of the compression (between Z_1 and Z_2 the volume decreases, $dv < 0$) and of the expansion (between Z_2 and Z_1 the volume increases, $dv > 0$). The adiabatic points also separate the cycle into two parts; the one in which the heat is supplied to the system from the first source (between A_2 and A_1) and the one in which the heat is given back to the second source (between A_1 and A_2). The adiabatic points we can determine by finding the points of contact of the cycle curve and of the two isentropes tangent to it.

It is worth to stress, that the net work l_{cycle} we can appoint both as the difference of mechanical works (of the expansion and compression works) and as the difference of flow works. It results from the fact, that for the reversible cycle this difference is always equal to the surface area inside the cycle curve, as shown in figure 7.4.

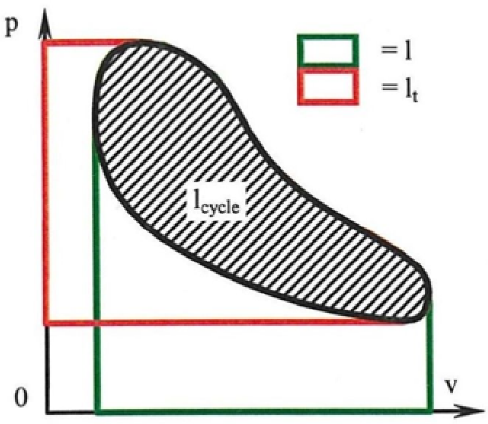

Fig. 7.4. Net work l_{cycle} as the difference of the mechanical and flow works of transformations constituting the cycle curve

7.3. THERMAL EFFICIENCY OF THERMODYNAMIC CYCLES

In general, the "efficiency" or "effectiveness" we can determine using the definition which is true for all conditions and for any process. It means, that by the efficiency of a process we understand the ratio calculated by dividing the useful result which we want to attain by the costs which we have to pay to get it done. Hence, a larger value is better. Of course in our considerations, by the results and the costs we understand the "energy effects" and "energy inputs". In order to determine the efficiency of the thermodynamic cycle in accordance with its above understanding, we will consider it separately for the heat engine cycle and separately for the working machines cycles.

7.3.1. Thermal Efficiency of the Heat Engine (the Clockwise Cycle)

The cycle of the heat engine in the p–v coordinate system is shown in figure 7.2a. The engine receives the heat q_{in} from the top source (usually named a "hot reservoir", so $q_{in} = q_{hot}$) at the temperature T_h, executes the net work $l_{cycle} > 0$ and gives back the remaining part of heat q_{out} to the bottom source (usually named a "cold reservoir", so $q_{out} = q_{cold}$) at the temperature T_c. Of course, in order to realize this direction of the heat flow, the following relation of the temperatures of heat sources; $T_h > T_c$ must be retained. The schematic diagram of this system is shown in figure 7.5.

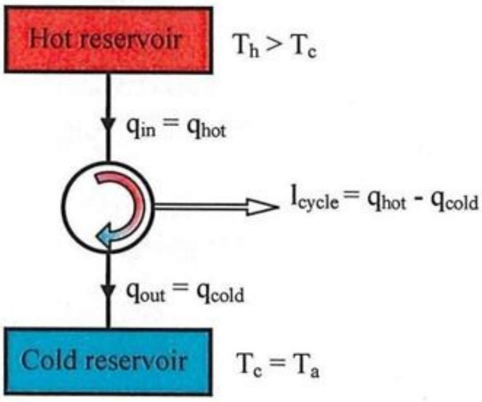

Fig. 7.5. Representation of a system undergoing a heat engine cycle (clockwise cycle)

Most often, the temperature of the cold reservoir T_c for heat engines is equal to the ambient temperature T_a (temperature of the surroundings, e.g. atmospheric air temperature). In compliance with the above mentioned definition, the efficiency is the ratio of performed useful work l_{cycle} ("energy effect") to the heat energy consumed from the hot reservoir ("energy input"). So, taking into account the energy balance of the system – equation (7.2) – we can write:

$$\eta_{he} = \frac{l_{cycle}}{q_{in}} = \frac{q_{in} - |q_{out}|}{q_{in}} = 1 - \frac{|q_{out}|}{q_{in}} = 1 - \frac{|q_{cold}|}{q_{hot}} \tag{7.3}$$

As we will see in the further discussion with reference to the Second Law of Thermodynamics, because both heat q_{in} and q_{out} are not equal to zero, the efficiency of a heat engine must always be less than unity ($\eta_{he} < 1$).

Of course, if the analyzed heat engine cycle were irreversible, its efficiency would be lesser than for the reversible cycle because some part of the work of the cycle would be used to overcome the energy dissipation of the system.

7.3.2. Thermal Efficiency of the Working Machine (the Counter-Clockwise Cycle)

The cycle of the working machine in the p–v coordinate system is shown in figure 7.2b. The machine receives the heat q_{in} from the bottom source ("cold reservoir", so $q_{in} = q_{cold}$) at the temperature T_c, consumes the work of the cycle $l_{cycle} < 0$ and gives back the sum of energy (the heat supplied and the net work input) equal to the heat q_{out} to the top source ("hot reservoir", so $q_{out} = q_{hot}$) at the temperature T_h. Because the heat flow from the bottom source to the top source (at the expense of the led work $- l_{cycle}$) can be realized between various temperature levels, we can differentiate two kinds of the working machines. The schematic diagrams of these systems are shown in figure 7.6.

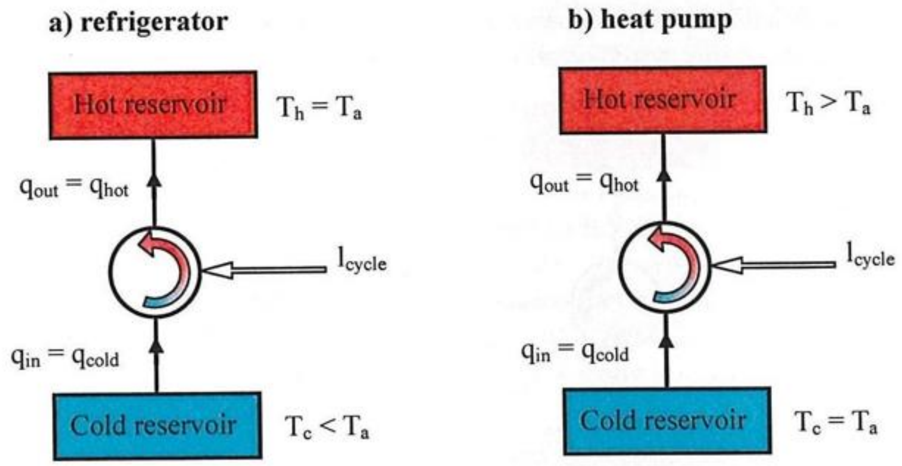

Fig. 7.6. Representation of a system undergoing a counter-clockwise cycle: a) a refrigerator cycle; b) a heat pump cycle; T_a – ambient temperature

Both the shown above devices realize the counter-clockwise cycle at the expense of the work led to the system, however they have various destinations and consequently we must differently determine their efficiency.

→ **Refrigerator**

The refrigerator (Fig. 7.6a) receives the heat from the cold reservoir at the temperature lower than the ambient temperature (e.g. from a cooling chamber) and transfers the heat to the hot reservoir (mostly to the surroundings). Thus, from the definition, the hot reservoir temperature $T_h = T_a$ and the cold reservoir temperature must be $T_c < T_a$. Hence, in compliance with the definition of the efficiency, the useful result for the refrigerator is the heat received from the bottom source (cooling chamber), instead the cost is the work input to the cycle, l_{cycle}. So, taking into account the energy balance of the system, we can write:

$$\varepsilon_r = \frac{q_{in}}{|l_{cycle}|} = \frac{q_{in}}{|q_{out}| - q_{in}} = \frac{q_{cold}}{|q_{hot}| - q_{cold}} \tag{7.4}$$

In accordance with the dependency (7.4), **the refrigerator efficiency may be both greater than unity and less than unity ($\varepsilon_r < 1$ or $\varepsilon_r > 1$)**. For that reason we denote the efficiency (effectiveness) of this cycle with ε (most often termed "coefficient of performance" – COP), instead of η for which we reserve rather lesser than unity values and the name "efficiency".

→ **Heat pump**

As far as a destination of a refrigerator is cooling of the bottom source, insomuch the heat pump has an inverse target – it has to heat the top source. That is why this device (Fig. 7.6b) receives the heat from the cold reservoir (mostly from the surroundings at the ambient temperature T_a) and gives it back to the hot reservoir, e.g. to the heated room. So, by definition, the cold reservoir temperature $T_c = T_a$ and the hot reservoir temperature must be, $T_h > T_a$. For that reason, according to the efficiency definition, the useful result of the heat pump is heat transferred to the top source (hot reservoir) whereas the cost is the work input to the cycle, l_{cycle} (in other words, the propulsion work). So, taking into account the energy balance for this system, we can write:

$$\varepsilon_{hp} = \frac{|q_{out}|}{|l_{cycle}|} = \frac{|q_{out}|}{|q_{out}| - q_{in}} = \frac{|q_{hot}|}{|q_{hot}| - q_{cold}} \tag{7.5}$$

In accordance with the formula (7.5), **the heat pump efficiency is always greater than unity ($\varepsilon_{hp} > 1$)**. For the same reasons as for the refrigerators, also for the heat pumps the efficiency (effectiveness) of this cycle we denote with ε and call the "coefficient of performance" – COP, instead of the "efficiency".

The modulus in formulas (7.3), (7.4) and (7.5) we used for the accordance with the signs convention of heat and work. We can write down these dependencies also without using the modulus. Then however one ought to rigorously comply with the notation of the balance of energy of a given device, in accordance with the turn of arrows of the flow of the energy, applied in diagrams 7.5, 7.6a and 7.6b.

2. The procedure of the laboratory exercise. Development of the measurement results.

The object of the study is an air-to-water compression heat pump. The laboratory stand represents a model of a compression heat pump [3]. The system contains all the components found in actual heat pumps. Additionally, the stand is equipped with elements that allow for adjusting the heat pump's load, enabling the determination and measurement of the pump's operating parameters under varying conditions. The inclusion of sensors and control elements allows for the reading of operating parameters, which enable the calculation of indicators characterizing the heat pump's performance. The cold reservoir is air, while the hot reservoir is the water used to cool the condenser. The working fluid in the heat pump cycle is refrigerant R134a. A detailed description of the setup can be found in the technical manual [3].

2.1. Refrigerant – the working fluid in the heat pump cycle

The working fluid in the heat pump cycle is the refrigerant R134a. **Tetrafluoroethane R-134a** ($C_2H_2F_4$) – is an organic chemical compound from the group of alkyl halides (Freons), specifically a fluorinated derivative of ethane, widely used in refrigeration systems.

R134a is designed for use in small domestic and commercial refrigeration devices, as well as air conditioning systems, particularly in automotive air conditioners. The compound requires a high degree of system tightness due to its ability to absorb moisture from the air during routine maintenance, such as charging the system. The presence of water in the system leads to various negative effects, such as oil breakdown When different oils are used, copper plating may occur. In this case, as moisture content in the mixture increases, the oil undergoes hydrolysis, producing acid that "transports" copper. R134a reacts with components made of zinc, magnesium, lead, and aluminum alloys containing 2% magnesium. Materials containing sodium, potassium, or calcium should be avoided. It does not have harmful effects on ferritic steel, copper and its alloys, or aluminum components. R134a is non-flammable and non-explosive under normal conditions, but under high pressure and with 60% air, it can form an explosive mixture. Therefore, air or oxygen must not be used for tightness tests. The refrigerant has high permeability, and while it is not toxic, some of its breakdown products are. Inhaling larger quantities of R134a vapor can cause narcotic effects, irritation of mucous membranes, and heart arrhythmias, potentially leading to a heart attack in extreme cases. Being heavier than air, it can cause asphyxiation by displacing air in confined spaces.

Properties of the refrigerant: tetrafluoroethane R-134a [2]

Molar mass: μ = 102.03 kg/kmol
Specific gas constant: R = 81.49 J/kgK
Liquid density at 25°C: ρ = 1206.0 kg/m^3
Specific heat of the liquid at 25°C: c = 1.44 kJ/kgK
Specific heat of the at vapor 25°C (1.013 bar): c_p = 0.85 kJ/kgK

2.2. Scheme of the measurement stand for an air-to-water compression heat pump

The scheme of the air-to-water compression heat pump is shown in Figure 2.1. The scheme highlights points corresponding to the state parameters in the comparative Linde cycle, as well as points and physical quantities whose values are measured. The basic components of the heat pump are the condenser, expansion valve, evaporator, and compressor. The operating parameters of the condenser are determined by measuring the temperature of the working fluid before (T2) and after the condenser (T3) along with the pressure (P2/3) (Fig. 10). . It is assumed that the refrigerant pressure in the condenser remains constant ($p_2 = p_3$). The operation of the evaporator is characterized by measuring the temperature before (T4) and after the evaporator (T1) as well as the pressure (P1/4) (Fig. 10). It is assumed that the refrigerant pressure in the evaporator is constant ($p_4 = p_1$). To increase the intensity of heat flux delivered to the working fluid in the evaporator, the air flow is forced by fans. he operation of the compressor is evaluated by measuring the electrical power (P) supplied to the compressor motor. the state parameters of the working fluid in the compression process are the temperature (T1) and pressure (P1/4) before the compressor, and the temperature (T2) and pressure (P2/3) after the compressor (Fig. 2.1). In the condenser, the R134a refrigerant releases heat to the cooling water. The amount of heat transferred to the water is determined by measuring the temperature (T5) of the cooling water before the condenser and the temperature (T6) after the condenser, as well as the water flow rate (F1) (Fig. 2.1).

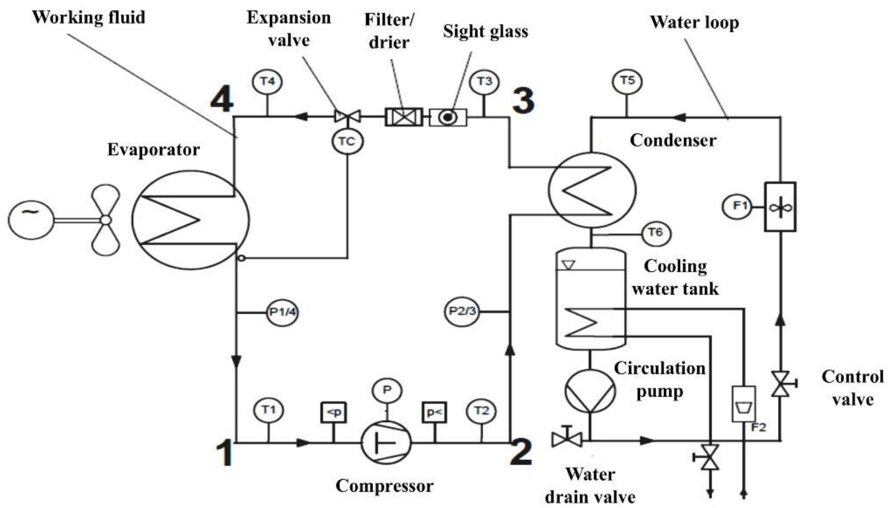

Fig. 2.1. Scheme of the laboratory stand for the heat pump with marked measurement points [3]
T – temperaturę senors, *p* – pressure sensors, *F* – measurement of the working fluid flow rate,
P – measurement of electrical power

2.3. Table of measured quantities

Measurements of the operating parameters of the heat pump should be conducted under steady-state conditions. For the given load parameters of the heat pump (operation of the evaporator fans and the cooling water flow through the condenser), several readings should be taken. The calculations are carried out based on the averaged values of the measured quantities. The measured physical quantities and the measurement results are presented in Table 1.

Table 1. Measured quantities

No.	*Measured quantity*	*Unit*	*Value*	*SI Unit*	*Value*
1	Temperature after evaporator (before compressor) t_1	ºC			
2	Temperature before condenser (after compressor) t_2	ºC			
3	Temperature after condenser t_3	ºC			
4	Temperature before evaporator t_4	ºC			
5	Pressure in evaporator p_{4-1} (absolute)	bar			
6	Pressure in condenser p_{2-3} (absolutne)	bar			
7	Electrical power supply to compressor P_{el}	W			
8	Temperature of water before condenser t_5	ºC			
9	Temperature of water after condenser t_6	ºC			
10	Water flow rate $\dot{V}_w$	l/h			

To determine the energy fluxes in the system and calculate the coefficient of performance (COP) of the heat pump, it is necessary to define the enthalpy of the working fluid at specific points in the cycle (measurement points). he values of enthalpy at various points in the cycle should be obtained from the ln$p$ – i diagram (Fig. 11). Additionally, the specific volume of the superheated vapor of the working fluid at points 1 and 2 must be calculated. The values obtained from the diagram are presented in Table 2.

Table 2. Parameters of the working fluid R134a at characteristic points in the heat pump cycle

No.	*Parameter*	*Unit*	***1***	***2***	***3***	***4***
1	Temperature ***t***					
2	Pressure ***p***					
3	Specific volume ***v***				------	-------
4	Specific enthalpy ***i***					

Note: The values of the physical quantities in the table should be read from the lnp – i diagram (Fig. 11).

3.1. Energy Analysis of an Air-to-Water Compression Heat Pump

The basis for the energy/thermodynamic analysis of the compression heat pump is the data obtained from the measurement of parameters at the laboratory setup. By plotting the points obtained during the measurements on the ln$p$ – i for the R134a refrigerant, he Linde cycle for the actual values of the working fluid's state parameters is obtained. The Linde cycle plotted for the actual parameters of the working fluid is shown in Figure 3.1. Comparing the cycle in Figure 3.1. with the comparative Linde cycle, differences between the real and comparative cycles can be observed.

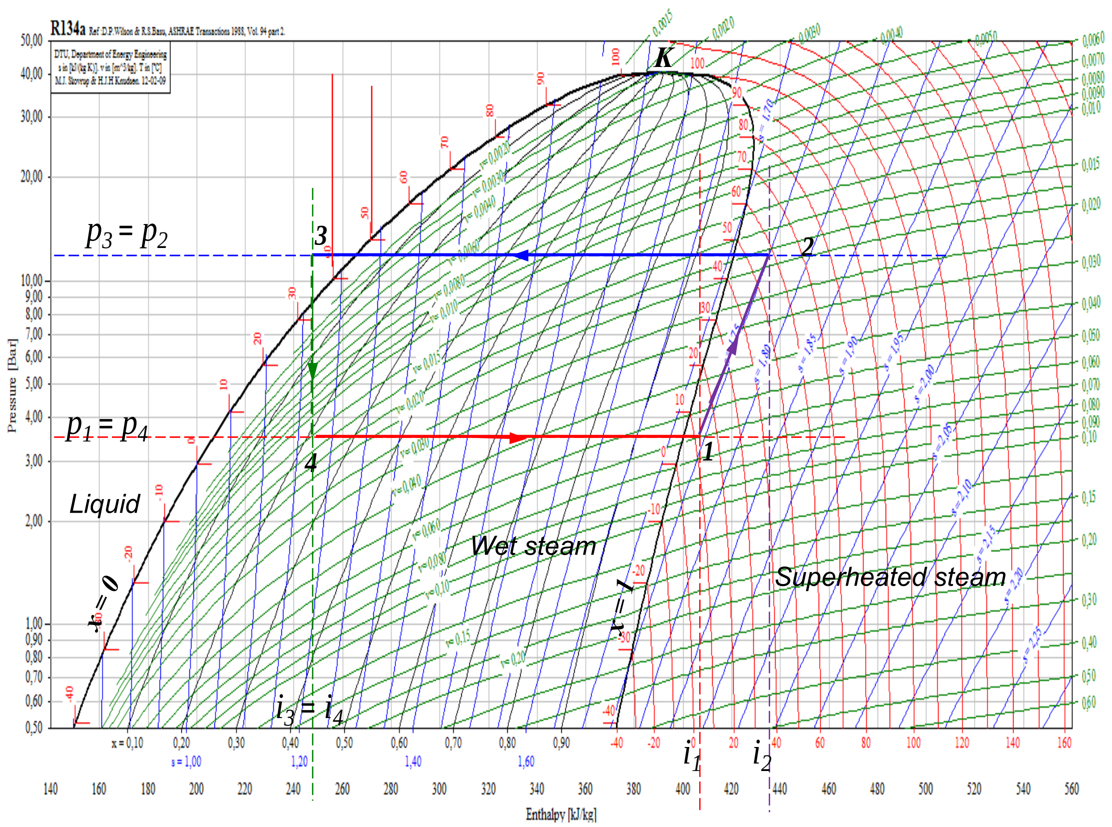

Fig. 3.1. Heat pump cycle on a pressure-enthalpy (ln$p$ – i) diagram for the refrigerant R134a, based on the measured operating parameter values.

In the actual cycle, the vapor of the working fluid leaving the evaporator is slightly superheated, point 1 (Fig. 3.1.) is located in the superheated region. The actual compression process 1 – 2 is an irreversible adiabatic proces. Isobaric heat rejection 2 – 3 in the condenser leads to cooling the working fluid below the saturation temperature at pressure p_3. Point 3 on the figure (Fig. 3.1.) is located in the liquid region. The isenthalpic throttling process 3 – 4 moves the state of the working fluid into the wet steam region. The pressure $p_4 = p_1$ and enthalpy $i_4 = i_3$ (quality about $x_4 = 0.20$ in Fig. 3.1.) define the position of point 4.. Izobaric heat addition 4 – 1 transitions the working fluid state back to point 1, where the vapor is already superheated. After determining the positions of the Linde cycle points, the enthalpy values (for all points) and the specific volume of the superheated vapor at points 1 and 2 are read from the ln$p$ – i diagram (Fig. 3.1.). The values obtained are summarized in Table 2.

3.1.1. Energy balance of the compressor

Process 1 – 2 compression of the working fluid vapor in the compressor

It is assumed that the compression process of the working fluid in the compressor is an irreversible adiabatic process, meaning there is no heat exchange with the surroundings. The First Law of Thermodynamics for adiabatic compression defines the energy balance in this process.

$$\dot{Q}_{1-2} = \Delta \dot{I}_{1-2} + \dot{L}_{t1-2} \qquad (44)$$

$\dot{Q}_{1-2}$ – heat flux between the working fluid/system and the surroundings during the compression process1 – 2,
$\Delta \dot{I}_{1-2}$ – Change in enthalpy flux of the working fluid during compression1 – 2,
$\dot{L}_{t1-2}$ – Flow work flux supplied to the working fluid in the compression process 1 – 2.

In the adiabatic compression process, there is no heat exchange between the system and the surroundings $\dot{Q}_{1-2} = 0$. From Eq. (44) following relationship is obtained:

$$\dot{L}_{t1-2} = -\Delta \dot{I}_{1-2} = -(\dot{I}_2 - \dot{I}_1) \qquad (45) \qquad\qquad \dot{L}_{t1-2} = \dot{M}_o (i_1 - i_2) \qquad (45a)$$

$\dot{M}_o$ – mass flow rate of the working fluid (R134a) in the heat pump cycle,
i_1, i_2 – specific enthalpy of the working fluid at states 1 and 2.

The efficiency η_{spo} of the compressor is defined as the ratio of the energy flux (flow work flux) supplied to the working fluid to the electrical power supplied to the compressor drive.

$$\eta_{spo} = \frac{\dot{L}_{t1-2}}{P_{el}} \qquad (46)$$

η_{spo} – compressor efficiency,
P_{el} – compressor drive power.

Applying Eq. (45a) in Eq. (46), the flow work flux $\dot{L}_{t1-2}$ supplied to the working fluid is determined by the mass flow rate M_o of the thermodynamic working fluid in the heat pump cycle

$$\eta_{spo} = \frac{\dot{M}_o (i_1 - i_2)}{P_{el}} \qquad (46a) \qquad\qquad \dot{M}_o = \frac{\eta_{spo} P_{el}}{i_1 - i_2} \qquad (47)$$

In the analysis of the measurement results, the compressor efficiency should be assumed: $\boldsymbol{\eta_{spo} = 0.52}$

According to the sign convention, the power supplied to the compressor is less than zero. (In equation (47), the power should be substituted with a negative sign or the absolute value of the difference in enthalpy Δi_{1-2} should be used in the denominator)
The ratio of the absolute pressure p_2 after the compressor to the absolute pressure p_1 before the compressor is called the compression ratio π_{sp}

$$\pi_{sp} = \frac{p_2}{p_1} \qquad (48)$$

3.1.2. Energy balance of the condenser

Process 2 – 3 isobaric heat rejection in the condenser to the cooling water

The equation for the First Law of Thermodynamics applied to the isobaric process in the condenser is expressed as

$$\dot{Q}_{c2-3} = \Delta\dot{I}_{2-3} + \dot{L}_{t2-3} \tag{49}$$

$\dot{Q}_{c2-3}$ – total heat flux in the process 2 – 3 in the condenser,
$\Delta\dot{I}_{2-3}$ – change in enthalpy flux of the working fluid in the condenser during the process 2 – 3,
$\dot{L}_{t2-3}$ – flow work flux in the process 2 – 3.

The working fluid does not perform work in the condenser (dp = 0). Equation (49) thus simplifies to:

$$\dot{Q}_{c2-3} = \Delta\dot{I}_{2-3} = \dot{I}_3 - \dot{I}_2 \quad (49a) \qquad\qquad \dot{Q}_{c2-3} = \dot{M}_o(i_3 - i_2) \quad (50)$$

i_2, i_3 – specific enthalpy of the working fluid at states 2 and 3.

The heat flux $\dot{Q}_{c2-3}$ released by the working fluid in the condenser is transferred to the cooling water $\dot{Q}_{wch}$ and to the surroundings $\dot{Q}_{str}$. The heat flow transferred to the surroundings represents a heat loss

$$\dot{Q}_{c2-3} = \dot{Q}_s = \dot{Q}_{wch} + \dot{Q}_{str} \tag{51}$$

$\dot{Q}_s$ – heat flux released by the working fluid in the condenser;
$\dot{Q}_{wch}$ – heat flux absorbed by the cooling water in the condenser;
$\dot{Q}_{str}$ – heat flux lost to the surroundings.

The heat transferred by the working fluid in the condenser is the result of cooling the superheated refrigerant vapor to the state of dry saturated vapor, followed by the condensation of the vapor. The specific amount of heat transferred $q_{c2\text{-}3}$ is determined by the change in enthalpy during the isobaric process 2 – 3:

$$q_{c2-3} = q_w = q_s = \Delta i_{2-3} = i_3 - i_2 \tag{50a}$$

q_w – specific heat removed (released) from the system,
q_s – specific heat released by the working fluid in the condenser.

Heat flux $\dot{Q}_{c2-3}$ released by the working fluid in the condenser is less than zero $\dot{Q}_{c2-3} < 0$ ($i_3 < i_2$).

In the condenser, the heat from the working fluid is transferred to the cooling water. The heat flux absorbed by the water $\dot{Q}_{wch}$ can be defined as

$$\dot{Q}_{wch} = \dot{M}_w c_w \Delta T_w \tag{52}$$

$$\dot{Q}_{wch} = \dot{M}_w c_w (T_6 - T_5) \quad (52a) \qquad\qquad \dot{Q}_{wch} = \dot{V}_w \rho_w c_w (T_6 - T_5) \quad (52b)$$

$\dot{Q}_{wch}$ – heat flux absorbed by the cooling water in the condenser,;
$\dot{M}_w$ – mass flow rate of the cooling water in the condenser;
$\dot{V}_w$ – volumetric flow rate of the cooling water in the condenser;
c_w – average specific heat of water in the temperature range $t_5 - t_6$;

The average specific heat of water: $c_w\Big|_{t_5}^{t_6} = \dfrac{c_w\big|_0^{t_6} t_6 - c_w\big|_0^{t_5} t_5}{t_6 - t_5}$ (Table Z2.2);

ρ_w – water density; (calculated as the arithmetic mean of the densities at t_5 and t_6 (Table Z2.1));
T_5, T_6 – temperature of the cooling water before and after the condenser, respectively.

From equation (51), the heat loss to the surroundings can be determined.

$$\dot{Q}_{str} = \dot{Q}_s - \dot{Q}_{wch} \tag{53}$$

The efficiency coefficient η_s of heat transfer in the condenser can be defined as the ratio of the heat flux absorbed by the cooling water to the heat flux released by the working fluid (the absolute value is due to the energy flow sign convention – the heat flux released in the condenser is less than zero):

$$\eta_s = \frac{\dot{Q}_{wch}}{\left|\dot{Q}_s\right|} \tag{54}$$

$$\eta_s = \frac{\dot{V}_w \rho_w c_w (T_6 - T_5)}{\left|\dot{M}_o (i_3 - i_2)\right|} \tag{54a}$$

3.1.3. Process 3 – 4 isenthalpic throttling of the working fluid in the expansion valve

In the isenthalpic throttling process 3 – 4 there is no heat exchange with the surroundings $Q_{3\text{-}4} = 0$. The working fluid does not perform any work as it passes through the expansion valve $L_{t3\text{-}4} = 0$. Consequently, the enthalpy of the working fluid remains unchanged $i_3 = i_4$.

3.1.4, Energy Balance of the Evaporator

Process 4 – 1 Isobaric Heat Addition to the Working Fluid in the Evaporator

Process 4 – 1 is isobaric heat addition to the working fluid in the evaporator. The fluid entering the evaporator is wet steam, with parameters defined by state 4. In this state, the fluid has a pressure $p_4 = p_1$ and quality x_4, which is the result of isenthalpic throttling in the expansion valve from pressure p_3 to p_4 with constant enthalpy $i_3 = i_4$ (Fig. 11). The First Law of Thermodynamics for process 4 – 1 in the evaporator:

$$\dot{Q}_{c4-1} = \Delta\dot{I}_{4-1} + \dot{L}_{t4-1} \tag{55}$$

$\dot{Q}_{c4-1}$ – total heat flux added to the working fluid in the evaporator during process 4 – 1,
$\Delta\dot{I}_{4-1}$ – change in enthalpy flux of the working fluid in the evaporator during process 4 – 1,
$\dot{L}_{t4-1}$ – flow work flux during process 4 – 1.

The working fluid does not perform work in the evaporator (dp = 0). Eq. (55) simplifies to:

$$\dot{Q}_{c4-1} = \Delta\dot{I}_{4-1} \tag{55a}$$

$$\dot{Q}_{c4-1} = \dot{M}_o (i_1 - i_4) \tag{56}$$

i_1, i_4 – specific enthalpy of the working fluid at states 1 and 4.

Using equation (56), the heat flow added to the working fluid from the low-temperature heat source is determined.

3.2. Coefficient of Performance ε_{tpc} of the heat pump

Two indicators are defined to characterize the performance efficiency of the heat pump: the thermodynamic coefficient of performance ε_{tpc} of the heat pump, and the actual coefficient of performance ε_{pcrz} (COP) of the heat pump.

3.2.1. Thermodynamic Coefficient of Performance ε_{tpc} of the heat pump

The thermodynamic Coefficient of Performance ε_{tpc} of the heat pump is defined by Eq. (28)

$$\varepsilon_{tpc} = \frac{|q_w|}{|l_{ob}|} \tag{28}$$

By incorporating into Eq. (28) the heat flux to the surroundings $\dot{Q}_{c2-3}$ from Eq. (50) and flow work $\dot{L}_{t1-2}$ supplied to the working fluid from Eq. (45a), following relationship can be expressed:

$$\varepsilon_{tpc} = \frac{|\dot{M}_o(i_3 - i_2)|}{|\dot{M}_o(i_2 - i_1)|} \tag{28a}$$

$$\varepsilon_{tpc} = \frac{|(i_3 - i_2)|}{|(i_2 - i_1)|} \tag{57}$$

The thermodynamic coefficient of performance ε_{tpc} evaluates the quality of the thermodynamic processes occurring in the cycle. It relates to the processes of heat rejection and work performed on the working fluid. It does not take into account the imperfections of the thermodynamic processes or the components of the heat pump (compressor, condenser).

3.2.2. Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump

Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump takes into account the impact of the compressor's work on the amount of energy that must be supplied to the system, as well as the actual amount of heat absorbed by the upper heat source (the water cooling the condenser).

Actual Coefficient of Performance ε_{pcrz} (COP – Coefficient of Performance) is a coefficient of efficiency, defined as the ratio of the energy flux absorbed by the upper heat source to the energy flux input supplied to drive the compressor. In the considered compression heat pump, the Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump is the ratio of the energy flux absorbed by the water cooling the condenser to the electrical power supplied to the compressor:

$$\varepsilon_{pcrz} = COP = \frac{\dot{Q}_{wch}}{P_{el}} \tag{58}$$

By expressing the heat flux absorbed by the water using equation (52b), the following relationship for the Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump is obtained:

$$\varepsilon_{pcrz} = COP = \frac{\dot{V}_w \rho_w c_w (T_6 - T_5)}{P_{el}} \tag{58a}$$

Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump is an indicator that reflects the real effects of the heat pump's operation. It is important to note that the temperatures of the sources fluctuate, meaning that the value of the coefficient changes during the operation of the heat pump.

4. Report

The report from the laboratory exercises should include:
1. The aim of the laboratory exercise.
2. The scheme of the laboratory setup.
3. Table of measurement results with conversion to the applicable unit system (Table 1).
4. Plotting of the heat pump cycle on the ln$p$ – i diagram (diagram – Attachment 1),
5. Reading from the diagram the enthalpy values at specific points and the specific volumes at points 1 and 2 (Table 2).
6. Energy analysis of the compression heat pump.
    6.1. Energy balance of the compressor: determination of work in the cycle, mass flow rate of the working fluid, and compression ratio (section 3.1.1).
    6.2. Energy balance of the condenser: determination of the total heat flux released by the working fluid, heat flux absorbed by the water, heat flux lost to the surroundings, and condenser efficiency (section 3.1.2).
    6.3. Energy balance of the evaporator: determination of the total heat flux supplied to the working fluid (section 3.1.3.).
7. Heat pump efficiency coefficients.
    7.1. Thermodynamic coefficient of performance of the heat pump (section 3.2.1).
    7.2. Actual coefficient of performance of the heat pump (section 3.2.2).
8. Summary of calculation results in a table (section 3).
9. Conclusions.

Literature:

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4. Moran M.J., Shapiro H. N.: Fundamentals of Engineering Thermodynamics, 5th Edition John Wiley & Sons, Inc., 5th Edition, 2006.
5. Cengel Y.A., Boles M.A., Kanoglu M.: Thermodynamics, An Engineering Approach, 9th Edition. McGraw-Hill Education, 2019.
6. Szewczyk W.: Lectures in Engineering Thermodynamics. Selected Problems. AGH University of Science and Technology Press, Kraków 2009.

Attachments:

Attachment 1. ln$p$ – i diagram for refrigerant R134a
Attachment 2. Properties of water

Table 3. Results of the Energy Analysis of the Compression Heat Pump Operation

No.	*Parameter*	*Unit*	*Result*
1	Flow work supplied to the working fluid $\dot{L}_{t1-2}$		
2	Mass flow rate of the working fluid $\dot{M}_o$		
3	Compression ratio of the compressor π_{sp}		
4	Total heat flux released by the working fluid in the condenser $\dot{Q}_{c2-3}$		
5	Heat flux absorbed by the cooling water $\dot{Q}_{wch}$		
6	Heat flux lost to the surroundings $\dot{Q}_{str}$		
7	Efficiency coefficient of the condenser η_s		
8	Total heat flux supplied to the working fluid in the evaporator $\dot{Q}_{c4-1}$		
9	Thermodynamic Coefficient of Performance ε_{tpc}		
10	Actual Coefficient of Performance ε_{pcrz} (COP) of the heat pump		

Attachment 1. ln*p* – *i diagram for* refrigerant R134a

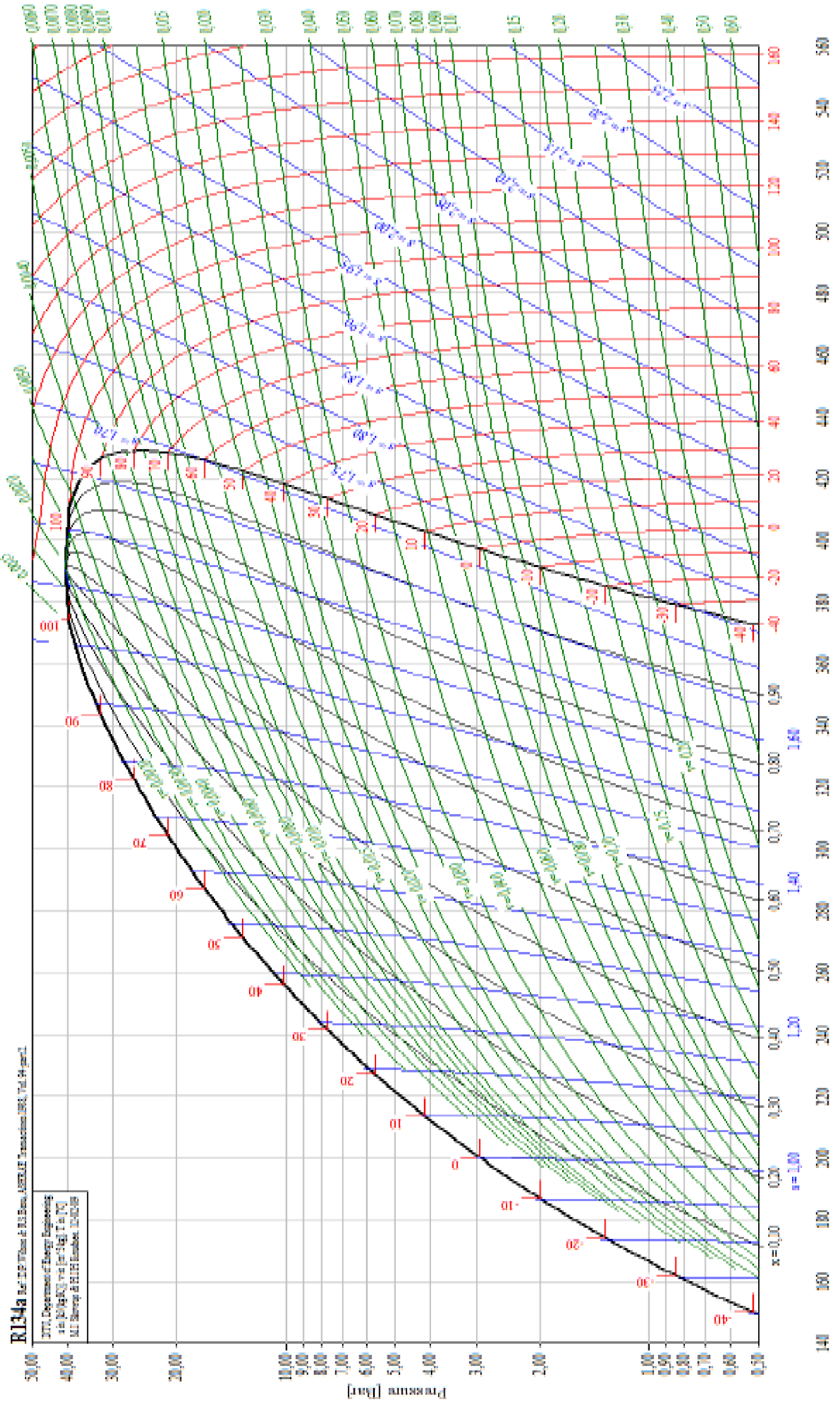

Fig. 12. ln*p* – *i* diagram for refrigerant R134a

Attachment 2. Water properties

Table Z2.1. Water density as a function of temperature

Tempera-ture *t*	Density *ρ*	Tempera-ture *t*	Density *ρ*	Tempera-ture *t*	Density *ρ*	Tempera-ture *t*	Density *ρ*	Tempera-ture *t*	Density *ρ*
°C	**kg/m^3**	**°C**	**kg/m^3**	**°C**	**kg/m^3**	**°C**	**kg/m^3**	**°C**	**kg/m^3**
-10	998.15	6	999.97	22	997.8	38	992.99	54	986.21
-9	998.43	7	999.93	23	997.56	39	992.63	55	985.73
-8	998.69	8	999.88	24	997.32	40	992.24	60	983.24
-7	998.92	9	999.81	25	997.07	41	991.86	65	980.59
-6	999.12	10	999.73	26	996.81	42	991.47	70	977.81
-5	999.3	11	999.63	27	996.54	43	991.07	75	974.89
-4	999.45	12	999.52	28	996.26	44	990.66	80	971.83
-3	999.58	13	999.4	29	995.97	45	990.25	85	968.65
-2	999.7	14	999.27	30	995.67	46	989.82	90	965.34
-1	999.79	15	999.13	31	995.37	47	989.4	95	961.92
0	999.87	16	998.97	32	995.05	48	988.96	100	958.38
1	999.93	17	998.8	33	994.73	49	988.52	110	951.0
2	999.97	18	998.62	34	994.4	50	988.07	120	943.4
3	999.99	19	998.43	35	994.06	51	987.61	130	935.2
4	1000.0	20	998.23	36	993.71	52	987.15	140	926.4
5	999.99	21	998.02	37	993.36	53	986.69	150	917.3

Table Z2.2. Average specific heat of water in the temperature range 0 – *t*

Temperature *t*	Specific heat *c*	Temperature *t*	Specific heat *c*	Temperature *t*	Specific heat *c*
°C	***kJ/kgK***	**°C**	***kJ/kgK***	**°C**	***kJ/kgK***
1	4.2118	21	4.1801	41	4.1776
2	4.2098	22	4.1793	42	4.1780
3	4.2078	23	4.1788	43	4.1784
4	4.2059	24	4.1784	44	4.1788
5	4.2040	25	4.1780	45	4.1797
6	4.2023	26	4.1776	46	4.1801
7	4.2006	27	4.1772	47	4.1809
8	4.1989	28	4.1768	48	4.1818
9	4.1974	29	4.1763	49	4.1826
10	4.1959	30	4.1763	50	4.1835
11	4.1893	31	4.1763	60	4.1910
12	4.1881	32	4.1759	70	4.1950
13	4.1872	33	4.1759	80	4.1990
14	4.1860	34	4.1759	90	4.2075
15	4.1851	35	4.1759	100	4.2160
16	4.1839	36	4.1763		
17	4.1830	37	4.1763		
18	4.1822	38	4.1768		
19	4.1814	39	4.1768		
20	4.1809	40	4.1772		

Development of the heat pump measurement results

1. Measurement results

Table 1. Measured quantities

No.	*Measured quantity*	*Unit*	*Value*			*Averaged value*	*SI Unit*	*Value*
1	Temperature after evaporator (before compressor) $\boldsymbol{t_1}$	ºC						
2	Temperature before condenser (after compressor) $\boldsymbol{t_2}$	ºC						
3	Temperature after condenser $\boldsymbol{t_3}$	ºC						
4	Temperature before evaporator $\boldsymbol{t_4}$	ºC						
5	Pressure in evaporator $\boldsymbol{p_{4-1}}$ (absolute)	bar						
6	Pressure in condenser $\boldsymbol{p_{2-3}}$ (absolute)	bar						
7	Electrical power supply to the compressor $\boldsymbol{P_{el}}$	W						
8	Temperature of water before condenser $\boldsymbol{t_5}$	ºC						
9	Temperature of water after condenser $\boldsymbol{t_6}$	ºC						
10	Water flow rate $\dot{V}_w$	l/h						

2. Heat pump cycle on the pressure-enthalpy (ln$p$ – i) diagram

3. State parameters at characteristic points of the cycle

Table 2. Thermodynamic parameters of R134a at characteristic points in the heat pump cycle

No.	*Parameter*	*Unit*	***1***	***2***	***3***	***4***
1	Temperature $\boldsymbol{t}$					
2	Pressure $\boldsymbol{p}$					
3	Specific volume $\boldsymbol{v}$				------	-------
4	Specific enthalpy $\boldsymbol{i}$					

In the isenthalpic throttling process 3 – 4 in the expansion valve, the enthalpy remains unchanged.: $\boldsymbol{i_4} = \boldsymbol{i_3}$