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NUMERICAL ANALYSIS OF THE CONDENSATION CHARACTERISTICS OF DIFFERENT HEAT-TRANSFER MEDIA IN AN AIR-COOLED CONDENSER[1]

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A mathematical model of an air-cooled condenser (ACC) section is developed. It allows comprehensive study of its operation for various geometries, conditions, and heat-transfer media. A software program for numerical analysis of an ACC section developed using Excel spreadsheet software is used to study the dependence of rates, thermal and aerodynamic characteristics of the condensation of a number of heat-transfer media in a standard ACC section on the velocity of the cooling air. The model can be used for design of thermal power plants and refrigerating machines with air-cooled condensers.

Keywords: air-cooled condenser (ACC); mathematical model; freon; consumption; heat-transfer coefficient; thermal transmittance.

The advantages of air-cooled condensers (ACCs) and their importance for power engineering are well-known and widely discussed in the Russian and foreign literature. If equipped with air-cooled condensers, TPPs become independent of a water source, and there is no need to construct and maintain a water supply system. Ecologically, ACCs prevent environmental contamination with salts and saturation of the atmosphere with steam. Reducing the number of systems of TPPs improves their operating conditions and reliability [1, 2].

Since the 1970s, ACCs have been installed at steam power plants and combined-cycle power units (CCUs). ACCs were designed and constructed for 150 to 850 MW CCUs and 600 and 660 MW pulverized-coal power units. In Russia, they have been installed at the Verkhne-Mutnovsky GeoPP, Sochi CHPP, and Adler TPP.

ACCs are characterized by:

— large dimensions because of low thermal transmittance;

— use of finned tubes, often with fins made of nonferrous metals;

— outdoor installation, outside the machine hall of the TPP;

— long large-diameter pipelines and vapor pressure losses between the turbine and the condenser;

— risk of freezing of the condensate and failure of tubes when water vapor is used.

Because of the large size and low thermal transmittance, ACCs are more preferable at TPPs with relatively low vapor flow rates to the condenser, such as CHPPs with CCU.

A promising application of ACCs may be power units with organic or binary Rankine cycle low-boiling fluids (LBFs) with high vapor density [3]. In this case, ACCs and vapor pipelines are much smaller. The vapor of most LBFs can be condensed at temperatures below 0°C, which raises the cycle efficiency by decreasing the temperature of the removed heat. The condensation of the vapor of many LBFs even at negative temperatures occurs at pressure above atmospheric, which prevents air inleakage in the condenser. Most LBFs freeze below –40°C and do not cause corrosion of the equipment.

Problem Statement. ACCs are fabricated from industrially produced standard sections [4]. Several sections are assembled into a module through which air is pumped over by a fan. The tubing system of a section consists of two headers and several rows of finned tubes welded to them (Fig. 1). To ensure strength and simplify transportation and installation, the tubing system is fixed in a frame. The basic geometrical parameters of a section: tube length L, tube bundle width B, total number N of tubes, number Z of rows of tubes along the path of air, finned surface area F.

The configuration and geometry of tubes are shown in Fig. 2, where S_1 is the spacing between tubes in a row; S_2 is the spacing between rows of tubes along the path of air; D is

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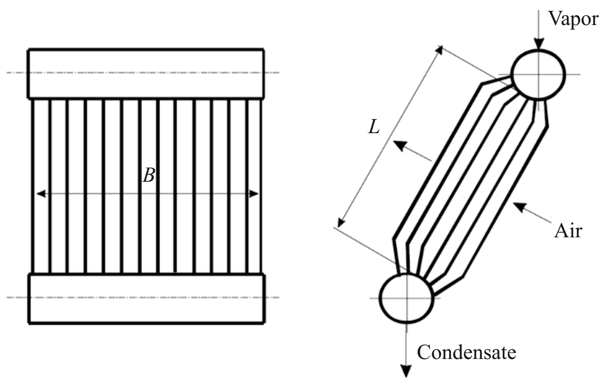

Fig. 1. Tubing of the ACC section.

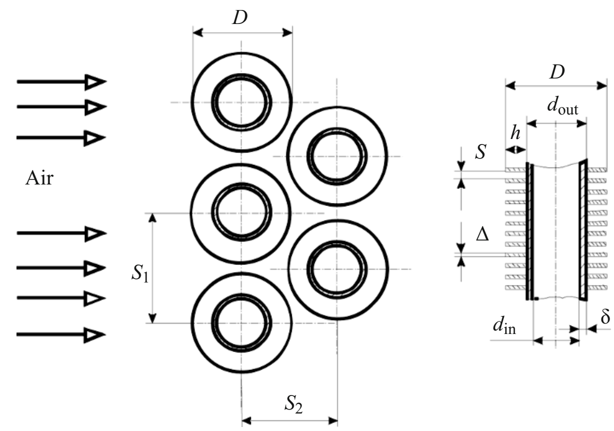

Fig. 2. Configuration and geometry of tubes in the ACC section.

the fin diameter; h is the fin height; d_{out} and d_{in} are the outer and inner diameter of tube; δ is the tube wall thickness; Δ is the fin thickness; S is the spacing between fins.

Our goal here is to develop a mathematical model of an ACC section for numerical study of the condensation characteristics of various substances based on the specified geometrical parameters and operating conditions of the section.

The mathematical model has been developed using techniques described in [2, 4]. The model disregards the effects of nonuniform distribution of vapor among tubes and of air in the intertubular space, solar radiation, wind, and rain. Air draught is provided with fans.

The input data for the model: geometrical parameters of the section (Fig. 1), materials and geometrical parameters of tubes and fins (Fig. 2).

The operating conditions are represented by:

air: temperature toa at the inlet of the section, °C; barometric pressure p_b, kPa; air velocity ω_a in the throat of the section, m/sec;

condensation: temperature t_c, °C; vapor dryness fraction at the inlet, x_{in}, and outlet, x_{out}, of the section.

Model-Based Algorithm.

1. Inner flow area of tubes for vapor and condensate F_f, m^2

$$F_f = \frac{N\pi d_{in}^2}{4}.$$

2. Extension ratio φ

$$\varphi = 1 + \frac{2h(d_{out} + h + \Delta)}{S d_{out}}.$$

3. Finned surface area F, m^2

$$F = N\varphi\pi d_{out}.$$

4. Contraction coefficient χ

$$\chi = 1 - \left[1 + \frac{2h\Delta}{d_{out} S}\right]\frac{d_0}{S_1}.$$

5. Hydraulic diameter d_h for air, m

$$d_h = \frac{2[S(S_1 - \Delta) - 2\Delta h]}{2h + S}.$$

6. Number n_f of fins on one tube

$$n_f = \frac{L}{\Delta + S} + 1.$$

7. Equivalent diameter d_{eq} of tubes, m

$$d_{eq} = \frac{d_{out}}{\varphi} + \frac{\varphi - 1}{\varphi}\sqrt{\frac{F}{2Nn_f}}.$$

8. Determine the condensation pressure p_c (MPa) from the condensation temperature t_c.

9. Average air temperature $t_{a.av}$ in the section

$$t_{a.av} = (t_{oa} + t_c - 10)/2.$$

10. Specify the vapor flow rate G_v (kg/sec) to the section.

11. Vapor velocity ω_v at the section inlet, m/sec

$$\omega_f = \frac{G_f}{F_f \rho_f},$$

where ρ_v is the vapor density determined from p_c and x_{in}.

12. Heat flow Q from vapor to air, kW

$$Q = G_v(h_{in} - h_{out}),$$

where h_{in} and h_{out} are the enthalpies determined from p_c, x_{in}, and x_{out}.

13. Heat flux q from vapor to air through unit area of finned surface, W/m^2

$$q = Q/F.$$

14. Volume airflow rate V, m^3/sec

$$V = \chi \omega_a L B.$$

15. Mass airflow rate G_a, kg/sec

$$G_a = V\rho_a,$$

where ρ_a is the air density determined from $t_{a.av}$ and p_b.

16. Air temperature increment Δt_a in the section, °C

$$\Delta t_a = \frac{Q}{G_a c_a},$$

where c_a is the heat capacity of air determined from $t_{a.av}$ and p_b.

17. Air temperature $t_{a2'}$ at the section outlet, °C

$$t_{a2'} = t_{oa} + \Delta t_a.$$

18. Average air temperature $t_{a.av}$ in the section, °C

$$t_{a.av} = \frac{t_{oa} + t_{a2'}}{2}.$$

19. Terminal temperature difference θ, °C

$$\theta = t_c - t_{a2'}.$$

20. Temperature drop Δt in the section, °C

$$\Delta t = \frac{t_{a2'} - t_{oa}}{\ln \dfrac{t_c - t_{oa}}{t_c - t_{a2'}}}.$$

21. Aerodynamic air drag Δp_a, kPa

$$\Delta p_a = 10^{-3} \frac{Z \,\mathrm{Eu}\, \rho_a \omega_a^2}{2},$$

where $\mathrm{Eu} = 5.4(d_{eq}/d_h)^{0.3} \mathrm{Re}_a^{-0.25} C_Z$ is the Euler number; $\mathrm{Re}_a = \omega_a d_h \rho_a$ is the Reynolds number; $C_Z = 0.0006Z^4 - 0.0166Z^3 + 0.1661Z^2 - 0.758Z + 2.5989$ is a coefficient depending on the number of tubes along the air flow.

22. Heat-transfer coefficient α_v in tubes, W/($m^2 \cdot$ K)

$$\alpha_f = \mathrm{Nu}_{av} \frac{\lambda_c}{d_{in}},$$

where the average heat-transfer coefficient from the condensed vapor to the inner wall of tubes is calculated from the following equation [5]:

$$\mathrm{Nu}_{av} = C\,\mathrm{Re}_c^{0.8}\,\mathrm{Pr}_c^{0.43}\, 0.5 \times \left[\sqrt{1 + x_{in}\left(\frac{\rho_c}{\rho_v} - 1\right)} + \sqrt{1 + x_{out}\left(\frac{\rho_c}{\rho_v} - 1\right)}\right],$$

where C is a coefficient depending on the material of the tubes (0.024 for steel, 0.026 for brass, and 0.032 for copper); Nu is the Nusselt number; Re_c is the Reynolds number; Pr_c is the Prandtl number; ρ_c and ρ_v are the densities of the condensate and vapor.

An advantage of this equation is that for $\mathrm{Re}_c > 5 \times 10^3$ the slope of the tubes does not affect the intensity of heat transfer [6], the Reynolds number being defined as

$$\mathrm{Re}_c = 4G_v/\rho d_{in}\mu_c,$$

where μ_c is the coefficient of dynamic viscosity of the condensate.

All physical parameters appearing in the expressions for the numbers Nu, Re, Pr and the densities of condensate and vapor are determined from the saturation temperature.

23. The heat-transfer coefficient α_2 from the finned surface to air, W/($m^2 \cdot$ K)

$$\alpha_2 = \mathrm{Nu}_a \lambda_a / d_h,$$

where λ_a is the thermal conductivity of air; $\mathrm{Nu}_a = \mathrm{StPr}_a^{2/3} \mathrm{Re}_a \mathrm{Pr}_a^{1/3}$ is the Nusselt number, where $\mathrm{StPr}_a^{2/3} = -7.5 \times 10^{-7} \times \mathrm{Re}_a + 0.0135$; λ_a, ρ_a, and Pr_a are determined from $t_{a.av}$ and p_b.

It is recommended in [4] that the heat-transfer coefficient to air $\alpha_a = 0.8\alpha_2$; fin efficiency $E = \tanh(mh_{f.eq})/(mh)$, where tanh is hyperbolic tangent; $h_{f.eq} = h(1 + 0.35)\ln(d_{out}/d_{in})$ is the equivalent height of a circular fin of rectangular cross-section in meters; $m = \sqrt{2\alpha_a/(\lambda_f \Delta)}$, where λ_f is the thermal conductivity of fins, W/(m $\cdot$ K).

The reduced heat-transfer coefficient α_r from fins to air, W/($m^2 \cdot$ K)

$$\alpha_r = \alpha_a \frac{[(\varphi - 1)E + 1]}{\varphi}.$$

24. Thermal transmittance K, W/($m^2 \cdot$ K)

$$K = \left[\frac{1}{\alpha_r} + \left(R_{tc} + R_{dep}\frac{d_{in}}{d_{out}} + \frac{d_{in}}{2\lambda_{tf}} \ln \frac{d_{in}}{d_{out}} + \frac{1}{\alpha_v}\frac{d_{in}}{d_{out}}\right)\varphi\right]^{-1},$$

where R_{tc} is the thermal insulance of the tube-fin interface, $m^2 \cdot$ K/W; R_{dep} is the thermal insulance of deposits in tubes, $m^2 \cdot$ K/W.

25. Design terminal temperature difference θ_d

$$\theta_f = (t_c - t_{oa}) \exp[-KF/(10^3 c_a G_a)].$$

26. Air temperature $t_{a2''}$ at the section outlet, °C

$$t_{a2''} = t_c - \theta_f.$$

27. If $|t_{a2'} - t_{a2''}| < 0.01$°C, stop the process; otherwise, change the vapor flow rate G_v and go to step 11.

The model has been implemented as a software package in an Excel spreadsheet environment using a REFPROP dynamic library to define the thermophysical parameters of the substances of interest[3]. The bisection method is used for automatic offset of heat balance between vapor and air to satisfy the condition of step 27.

Selection of Substances. An extensive review of the relevant foreign and Russian literature allows us to conclude that the following properties should be allowed for in selecting substances as working fluids for Rankine cycle:

— a substance should have high density and molecular weight to reduce the size of the section;

— the saturation vapor pressure of a substance in the temperature range of the cycle should be not very high, yet not lower than atmospheric to avoid problems with generating vacuum and providing strength and tightness of heat exchangers, pipelines, and valves;

— the heat capacity of a substance in liquid state should be as low as possible, while the heat of evaporation should be as high as possible;

— the specific work in the turbine in the cycle temperature range should be high;

— the viscosity of the liquid and vapor phases should be low to ensure low friction losses and high heat-transfer coefficients;

— the heat conductivity of the liquid and vapor phases should be high to ensure effective heat exchange;

— the derivative $dT/dS = \xi$ for the saturated vapor line should be close to 0 to ensure the operation of the turbine without losses due to moisture and to avoid vapor superheating;

— thermal stability at high cycle temperatures is needed;

— the triple point should be below the minimum cycle temperature;

— a substance should not be toxic, flammable, and explosive;

— a substance should be inexpensive, readily available, and environmentally friendly;

— a substance should have low ozone-depletion and global-warming potentials (ODP and GWP).

Currently, there are no substances that meet all the above requirements.

The main substance used in power engineering as a working fluid for Rankine cycle is water, which has the majority of the properties listed above. The main disadvantages of water are as follows:

— a great value of ξ (–17.78), which requires superheating of steam and leads to considerable losses due to moisture in the turbine;

— high triple-point temperature (–0°C), which creates problems in winter and does not allow the heat rejection temperature in the cycle to be below 10°C;

— to provide high efficiency, the pressure and temperature in the upper cycle must by high and the vacuum in the lower cycle must be high too, which creates many problems with the selection of structural materials and the maintenance of vacuum.

Since ACCs with water as a working fluid are widely covered in the literature, we select some freons (R123, R236fa, R245fa, R365mfc) and hydrocarbons (butane, pentane, isobutane, isohexane) to do some studies and to test the algorithm. The freons chosen are not ozone-depleting, are not inflammable, and have relatively low GWR. The hydrocarbons chosen have often been recently recommended in the domestic and foreign literature as a working fluid for heat pumps, refrigerators, and organic Rankine cycle. The main disadvantage of hydrocarbons is flammability.

The most important characteristics of the chosen substances are summarized in Table 1, where M is the molecular weight; T_{cr} and p_{cr} are the temperature and pressure at the critical point; T_{ul} is the upper limit temperature; T_{tp} is the triple-point temperature; the values of ξ are borrowed from [7].

Calculated Results. The following parameters of a standard section are used as input data [4]: tube material is steel 20, fin material is duralumin, fins are circular, $D = 0.057$ m, $h = 0.015$ m, $d_{out} = 0.027$ m; $\delta = 0.002$ m, $\Delta = 0.000735$ m, $S = 0.0025$ m, $S_1 = 0.084$ m, $L = 12$ m, $B = 1.85$ m, $Z = 6$, the number of tubes in the section $N = 132$, $F = 2690$ m^2. It is also assumed that $R_{tc} = R_{dep} = 0$.

[3] www.nist.gov/srd/nist23.cfm

TABLE 1

Substance	M, g/mol	T_{cr}, K	p_{cr}, MPa	T_{ul}, K	T_{tp}, K	ξ, J/(kg · K^2)	G_v, kg/sec
Butane	58.12	425.1	3.80	575	134.9	1.03	3
Isobutane	58.12	407.8	3.63	575	113.7	1.03	3
Pentane	72.15	469.7	3.37	600	143.5	1.51	11
Isohexane	86.18	497.7	3.04	550	119.6	—	—
R236fa	152.04	398.1	3.20	500	179.5	0.76	6300
R245fa	134.05	427.2	3.64	440	171.0	0.19	930
R123	152.93	456.8	3.66	600	166.0	0.26	90
R365mfc	148.07	460.0	3.27	500	239.0	—	1500

Also, the outdoor temperature t_{oa} = 15°C; the condensation temperature of freon is 20°C higher than t_{oa}, i.e.; t_c = 35°C; barometric pressure is 98 kPa; the dryness fraction of vapor is equal to 1 at the inlet and to 0 at the outlet.

The air velocity ω_a was varied from 1 to 9 m/sec. At all points, we have $Re_c > 5 \times 10^3$, which means that the tubes can slope at any angle. The calculated results are presented in Figs. 3 – 5.

Analysis of the Results. The results obtained allow us to analyze the capabilities of the ACC section depending on which of the substances is used as a working fluid.

The air velocity for the operation of the fans can be found from Fig. 3. For commercial axial-flow fans, ω_a = 5 – 6 m/sec is the best choice.

Figure 4*a* indicates that the substances can be arranged in the order of increasing condensed-vapor flow rate as follows: R236fa, R123, R365mfc, R245fa, isohexane, pentane, butane and isobutane. The condensed vapor flow rate is strongly dependent on the air velocity: as ω_a changes from 1 to 9 m/sec, the vapor flow rate increases by a factor of approximately 6.

Figure 4*b* characterize the efficiency of the ACC section. It can be seen that the substances are arranged in the order of increasing efficiency as follows: isohexane, pentane,

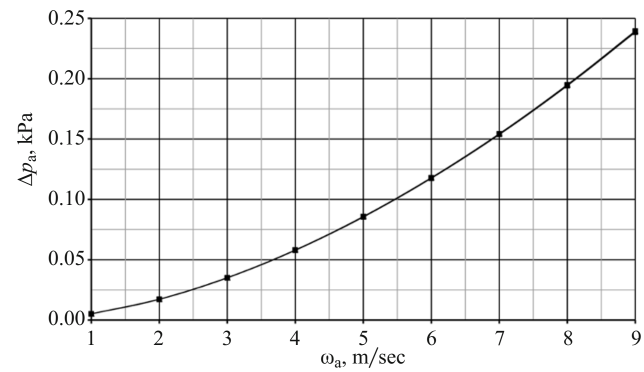

Fig. 3. Dependence of aerodynamic air drag Δp_a of the section on the air velocity ω_a.

R365mfc, R123, R245fa, butane, R236fa, isobutane. Figure 4*c* allows us to determine the necessary number of ACC sections from the known heat flux. It can also be seen that the heat flux *q* of butane is approximately 50 W/(m² · K) higher than that of isobutane.

Figure 4*d* shows the variation in θ with ω_a. It can be seen that θ is with_{in} the limits (~10°C) recommended in [4] for air-cooled heat exchangers.

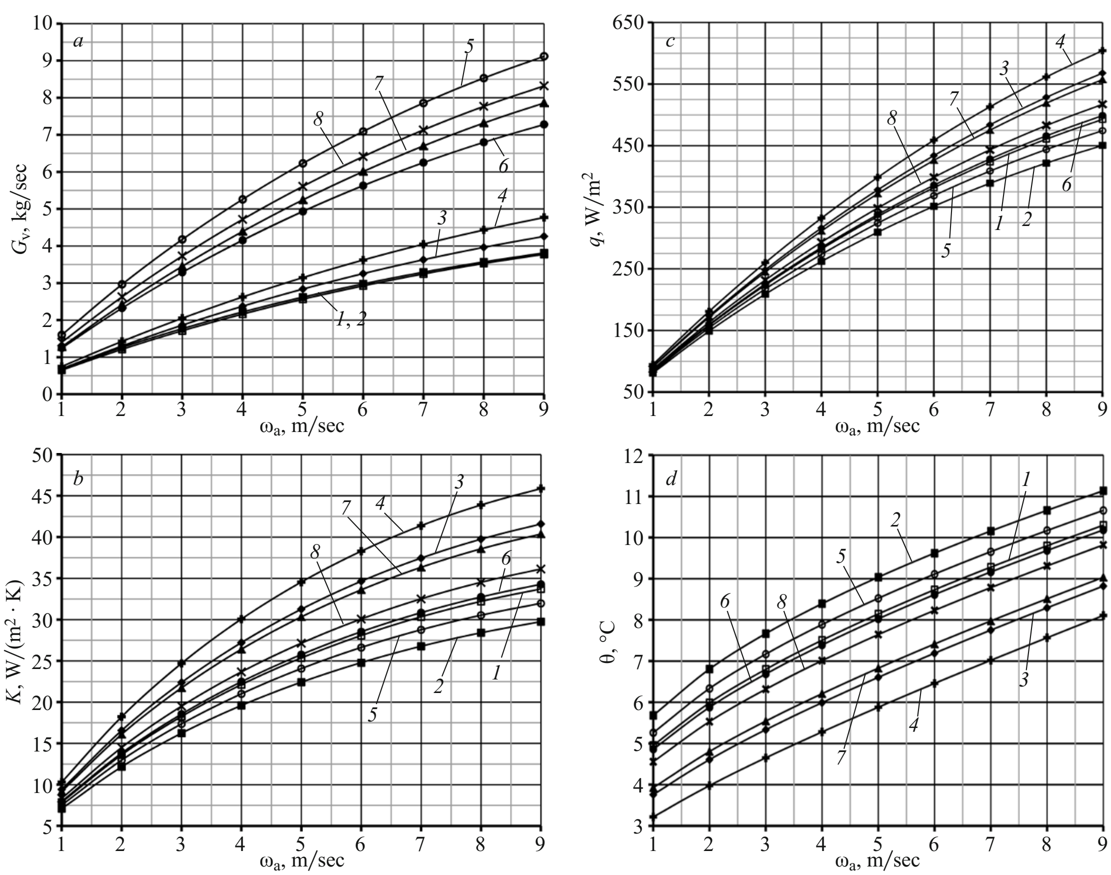

Fig. 4. Calculated results (dependence on ω_a) for some substances: *a*, vapor flow rate G_v; *b*, thermal transmittance K; *c*, heat flux from vapor to air *q*; *d*, terminal temperature difference θ; *hydrocarbons*: *1*, butane; *2*, isobutane; *3*, pentane; *4*, isohexane; *freons*: *5*, R236fa; *6*, R245fa; *7*, R365mfc; *8*, R123.

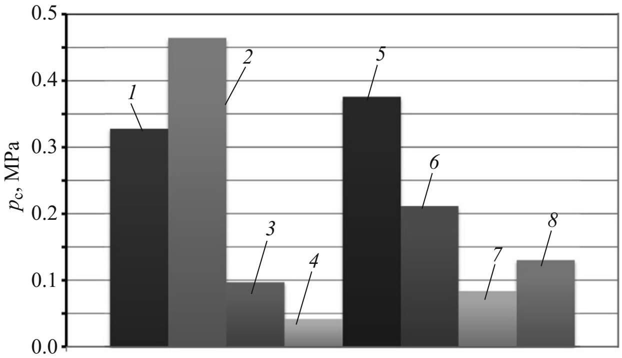

Fig. 5. Condensation pressure p_c in ACC tubes (for the notation, see Fig. 4).

Figure 5 shows that if $t_c = 35°C$, the substances can be arranged in the order of decreasing condensation pressure (which is higher than atmospheric) as follows: isobutane, R236fa, butane, R245fa, R123. The condensation pressure of R365mfc and isohexane is lower than atmospheric, which will cause problems with vacuum maintenance.

CONCLUSIONS

1. The proposed mathematical model of ACC section and associated software allow conducting comprehensive studies on the condensation of a wide range of substances included in the REFPROP database.

2. The model allows computing the air velocity in the section and heat flux, which can then be used to select fans and to determine the necessary number of sections and, thus, to assess the electricity costs for fans and the capital costs for ACC.

3. The substances examined can be arranged in the order of increasing efficiency of the ACC section: isohexane, pentane, R365mfc, R123, R245fa, butane, R236fa, isobutane.

4. Isobutane, R236fa, butane, R245fa, and R123 are condensed at pressures higher than atmospheric, whereas pentane, R365mfc, and isohexane are condensed at pressures lower than atmospheric even at a temperature of 35°C, which requires a vacuum generation and maintenance system.

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