SPECIFIC FEATURES OF HIGH PRESSURE WATER SEPARATION IN AIRCRAFT ENVIRONMENTAL CONTROL SYSTEMS

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Abstract

Theoretical analysis of high pressure water separation process for aircraft environmental control systems is presented. It allows better understanding of the required system elements properties, including a two-stage cooling turbine.

1 Introduction

Most of modern airliner environmental control systems are equipped with a high-pressure water separation loop. The simplified diagram of such a system fragment is given in Fig. 1.

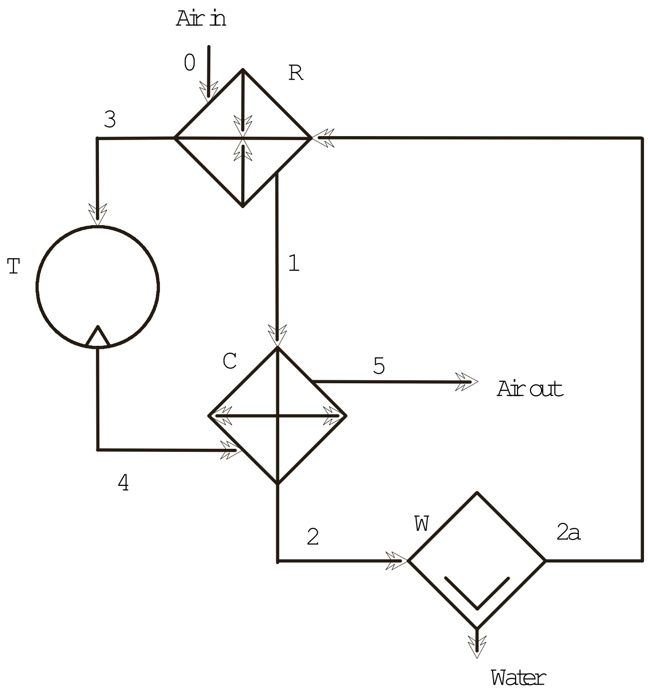

Fig. 1. Simplified diagram of high pressure water separation loop

The loop consists of the following main components: cooling turbine (**T**), condenser (**C**), reheater (**R**) and condensed water extractor (**W**). Lines connecting the elements are numbered from 0 to 5.

Humid air coming from the previous elements of the environmental control system (ECS) goes into a hot side of re-heater **R** through line **0** and, being preliminary cooled in this re-heater, enters, through line **1,** into hot side of condenser **C**. This air is cooled by the low-pressure and the lowest temperature air coming through line **4** from the turbine **T**. The temperature in the high pressure side of condenser **C** decreases down to the dew point, followed by a condensation process on heat transfer surfaces, creating films, droplets and streams of water.

The water and air mixture goes, through line **2,** to water extractor **W** which can be of the simplest layout. After water extractor the saturated air moves through line **2a** to the cold side of the re-heater where it is heated up by the hot air with temperature t_0, the index showing the line number. The heated-up air with a temperature of t_3 goes to turbine **T** where it expands from pressure p_3 ($\approx p_0$) down to pressure p_4 ($\approx p_5$). In accordance with the gas dynamics laws the air temperature decreases in the turbine and residual water vapor partially transforms into water (or – below freezing temperature – into mixture of overcooled water and ice particles).

This cold mixture goes into low pressure side of condenser **C** and is warmed there. But there exists a danger of ice formation on cold front section of condenser. Therefore some antiicing arrangements should be incorporated in its structure. Condensed or frozen water can be completely evaporated (even below freezing temperature) making air in line **5** unsaturated. Obtaining low temperature unsaturated air provides additional advantage to high pressure water separation systems.

2 Theoretical Analysis of High Pressure Water Separation Loop for Dry Air

For such a system, a set of equations can be derived with an assumption that air is dry:

$$\begin{aligned}
&T_1 = T_0 - \eta_1(T_0 - T_2);\\
&T_2 = T_1 - \eta(T_1 - T_4);\\
&T_3 = T_2 - \eta_1(T_0 - T_2);\\
&T_4 = T_3(1 - K_\mathrm{T});\\
&T_5 = T_4 - \eta(T_1 - T_4);\\
&K_\mathrm{T} = \eta_\mathrm{T}\left(1 - \pi_\mathrm{T}^{-\frac{k-1}{k}}\right);\\
&\pi_\mathrm{T} = p_3/p_4 \approx p_0/p_5,
\end{aligned} \qquad (1)$$

where π_T – turbine pressure ratio; k – adiabatic exponent; η_T – turbine thermodynamic effectiveness; η – condenser thermal effectiveness[1]; η_1 – reheater thermal effectiveness; T – temperatures expressed in absolute (Kelvin degrees) scale, the subscripts indicating to the line numbers).

Not very complicated manipulations give the following results:

$$T_5 = T_0\left[1 + \frac{\left(\dfrac{\eta + \eta_1 - 2\eta\eta_1 - 1}{1 - \eta_1 + \eta\eta_1}\right)K_\mathrm{T}}{1 - \left(\dfrac{\eta - \eta\eta_1}{1 - \eta_1 + \eta\eta_1}\right)(1 - K_\mathrm{T})}\right];$$

$$T_4 = T_0(1 - K_\mathrm{T}) *$$

$$* \left\{1 + (1 - \eta_1)\left[\frac{(1 - \eta(2 - K_\mathrm{T}))(1 - \eta_1) + \eta(1 - K_\mathrm{T})}{1 - (1 - \eta(2 - K_\mathrm{T}))\eta_1 - \eta(1 - K_\mathrm{T})} - 1\right]\right\}$$

where T_0 – predetermined air temperature at the inlet to the loop. Similar expressions can be obtained for all the other temperatures.

These solutions become more obvious if plotted as graphs. An example of such a case, when the reheater is excluded from the system (its effectiveness being taken $\eta_1 = 0$) is given in Fig. 2. Figures near the graphs correspond to the scheme line numbers.

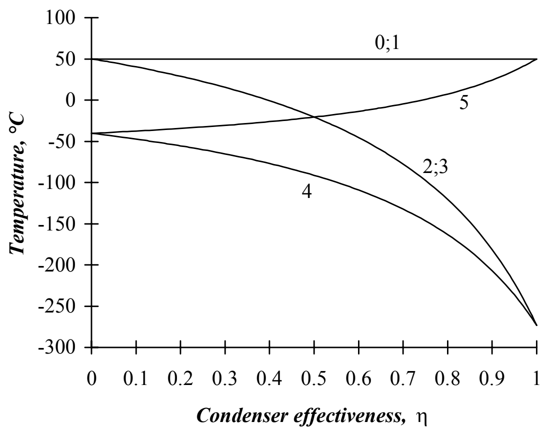

Fig. 2. Temperatures distribution versus condenser effectiveness for a loop without reheater

The calculations were made with $\pi_\mathrm{T} = 4.5$ and $\eta_\mathrm{T} = 0.8$. It can be seen that increasing condenser thermal effectiveness leads to decreasing turbine inlet temperature $\boldsymbol{T_3}$ and consequently temperature differential $(T_3 - T_4)$ that is followed by decreasing the overall system effectiveness: outlet temperature "5" gets increased. It can be also seen that at full condenser effectiveness, when η equals a unit, outlet temperature "5" achieves the level of inlet temperature "0", and, theoretically, the temperatures in lines "2", "3" and "4" may reach absolute zero.

Actually there is no necessity in very high condenser effectiveness because the equality of its outlet temperatures "5" - in low-pressure line - and "2" - in high-pressure line - is achieved at moderate effectiveness value of approximately

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[1] Heat exchanger thermal effectiveness is expressed by the relation

$$\eta = \left[c_p \dot{m}_h\left(t_{h.in} - t_{h.out}\right)\right] / \left[c_p \dot{m}_{\min}\left(t_{h.in} - t_{c.in}\right)\right],$$

where c_p – specific heat capacity; $\dot{m}$ - flow rate, with subscript h for hot side and min for minimal of two interacting flows; t – temperature, with subscript in for the temperature at the inlet section, out – for the temperature at the outlet section, c - for cold side of heat exchanger.

0.5. This means that if at temperature T_2 humid air is saturated and condensed water is removed from the system through water extractor **W** then at temperature $T_5 = T_2$, but at low pressure the humid air is unsaturated even at temperatures below freezing point.

In contrast to the condenser properties, the re-heater influence is quite opposite: the higher its effectiveness, the closer the outlet temperature "5" to its lowest value (in our case to

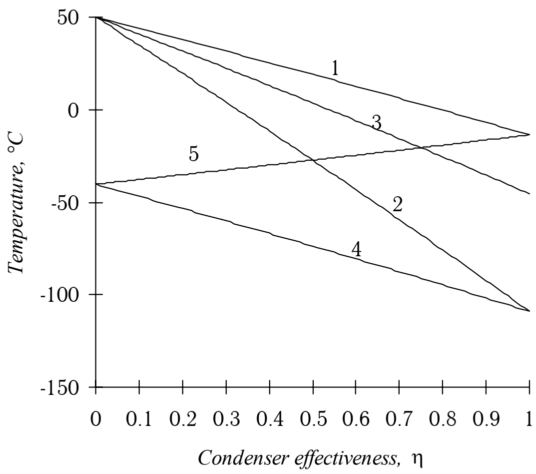

Fig. 3. Temperatures distribution vs. condenser effectiveness for a loop with the reheater effectiveness $\eta_1 = 0.4$

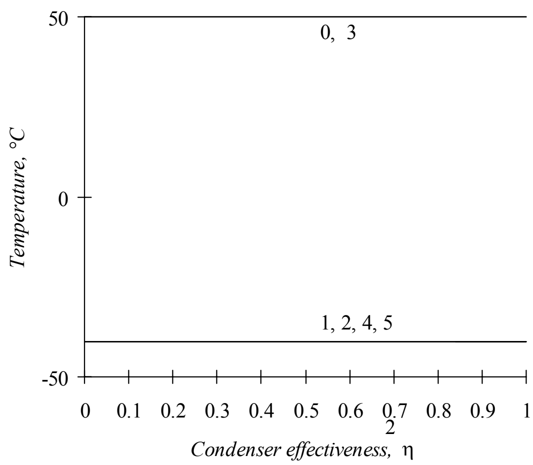

Fig. 4. Temperatures distribution vs. condenser effectiveness for a loop with the reheater effectiveness $\eta_1 = 1.0$

approximately –40 °C) for any condenser effectiveness. It means that increasing reheater effectiveness provides increasing overall system effectiveness. This is illustrated in Fig. 3 and Fig.4. Calculations for both figures were made with the previous values of $\pi_T = 4.5$ and $\eta_T = 0.8$.

Most expressive is Fig. 4. It shows that with a re-heater effectiveness equal to a unit, possible only theoretically, temperatures "1", "2", "4" and "5" remain unchanged at the minimum value for any condenser effectiveness. This result shows also that too high reheater effectiveness can cause less than freezing temperatures at high pressure side of the heat exchangers (condenser and reheater) which is inadmissible, because of possible ice clogging at heat exchangers channels. Therefore the re-heater thermal effectiveness should be limited as well as that of the condenser.

3 Theoretical Analysis of High Pressure Water Separation Loop for Humid Air

Analytical solution of the problem is impossible for humid air due to transcendental character of enthalpy equations. Therefore numerical methods were used. The results of calculations are presented in Fig. 5 and 6.

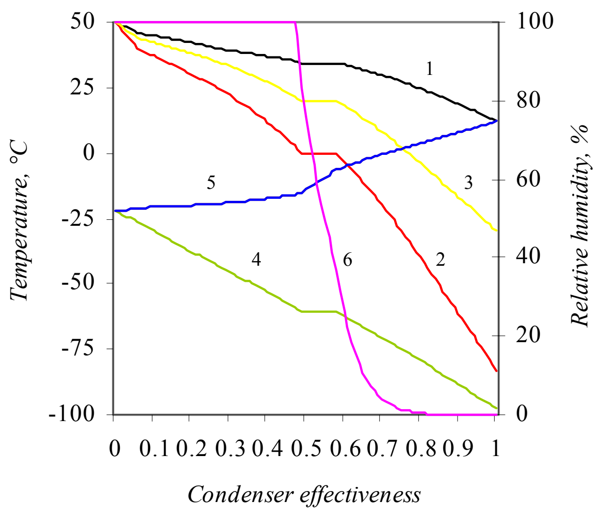

Fig. 5. Temperatures and relative humidity distribution vs. condenser effectiveness for humid air inlet moisture content 10 g/kg dry air

The calculations were fulfilled for the following conditions: inlet air temperature of

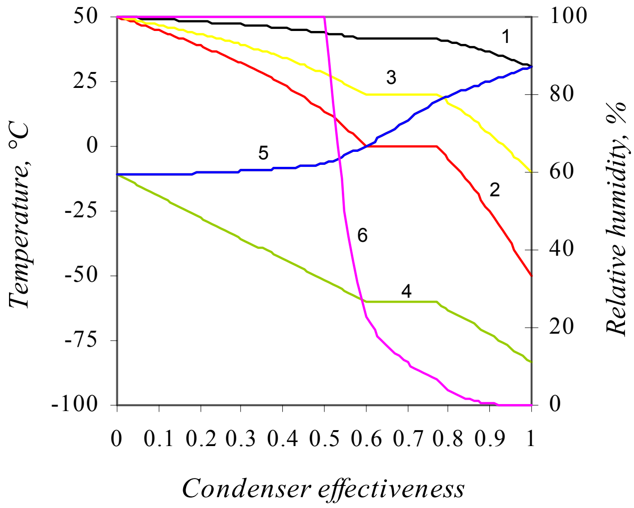

Fig. 6. Temperatures and relative humidity distribution vs. condenser effectiveness for humid air inlet moisture content 20 g/kg dry air

+50 °C; inlet air pressure of 0.45 MPa; outlet air pressure of 0.1 MPa (the pressures are considered to be unchanged along the lines before the cooling turbine and after it, the hydraulic resistances being neglected); cooling turbine efficiency of 0.8; reheater thermal efficiency of 0.4; initial air moisture contents of 10 g/kg (Fig. 5) and 20 g/kg (Fig. 6); all the condensed water completely removed through the water extractor **W**.

Curves 6 of Figs 5 and 6 represent the relative humidity of the outlet air in line 5, that is measured by the right vertical axis. All the other curve numbers reflect the temperatures in corresponding lines of Fig. 1.

It can be seen from Figs 5 and 6 that the pattern of the curves mainly corresponds to those of Figs 2 and 3. Detectable difference first manifested in temperature increase of the line 2, and consequently in all other lines, is caused by condensation process in condenser **C** which can be seen better in the upper-left corner of Fig. 5.

More essential deviations start only after the beginning of freezing process in condenser **C**. Such a mode of operation shown in Fig. 5, with a condenser effectiveness of more than ~0.5, and in Fig. 6, with a condenser effectiveness of more than ~0.6, is considered in practice as a system failure. At the same time the whole range is worth of being considered from a theoretical viewpoint. While the freezing process continues, the temperature "2" remains constant and horizontal portion of the curve "2" reflects it. Horizontal steps in curves "3", "4" and "1" are consequently conditioned by the curve "2" step. The more the initial air moisture contents, the wider are the steps.

It also can be seen from Figs 5 and 6 that outlet air relative humidity decreases abruptly after condenser effectiveness surpasses the value of approximately 0.5 (slightly depending upon initial air moisture contents).

In spite of general similarity of the curves in Figs 3 and 5 or 6 the temperatures quantitative difference is quite significant and may approximate up to 50 degrees centigrade. At the same time it means that for lower air humidity the freezing temperature in the condenser high-pressure side can occur at lower effectiveness (down to 0.325). So ice formation in condenser high-pressure side appears quite probable while aircraft flies at altitudes of 3…5 km. At the same time it becomes impossible to obtain unsaturated air at the loop outlet. Such regimes are supposed to be undesirable.

To avoid ice formation in heat exchangers two-stage turbine is used in Boeing-777 environmental control system (Fig. 7 [1]).

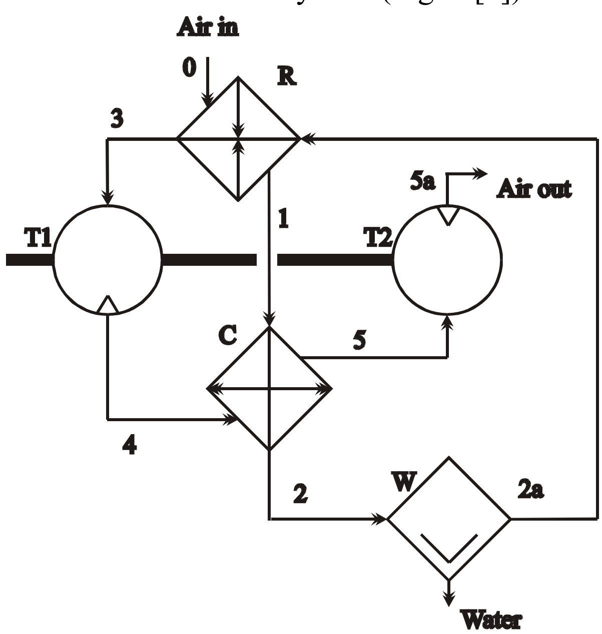

Fig. 7. High-pressure water separation loop of Boeing-777 environmental control system

The water separation loop in this system is arranged between the turbine two stages **T1** and **T2**.

Calculations were made for such a variant with an assumption that initial pressure of 0.45 MPa is divided evenly between the stages, the intermittent pressure being equal to geometric average of 0.212 MPa; the turbine second stage efficiency is the same as of the first – 0.8. The results for initial moisture content 17.5 g/(kg dry air) are presented in Fig. 8.

It can be seen that both heat exchangers (condenser **C** and reheater **R**) actually can be supported at the temperatures t_1, t_2, t_3, t_4 - higher than the freezing point, which is provided for all possible initial moisture contents. A lower temperature t_{5a} is obtained after the turbine second stage. It is interesting to note that this temperature is almost independent of condenser effectiveness, its changes being less 10 degrees centigrade on the whole argument range. Some quantity of the ice particles appearing in the outlet air can be evaporated with the help of recirculating cabin air.

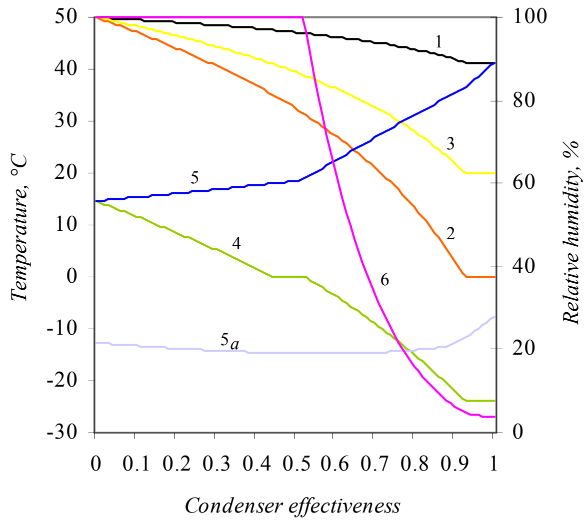

Fig. 8. Temperatures and relative humidity distribution vs. condenser effectiveness for humid air inlet moisture content 17.5 g/kg dry ai1

4 Conclusion

High-pressure water separation was introduced into aircraft environmental control systems mainly as a measure to overcome the disadvantages of low-pressure water separators. The disadvantages are caused by very small sizes (of the order of microns) of water droplets appearing in the air stream moving through the cooling turbine. It is very difficult to separate such droplets from air in any kind of centrifugal devices. Therefore special coalescent filters were used at the inlet portion of separator to make the droplets bigger. These filters are subjected to dust clogging and should be periodically replaced in the course of maintenance.

In addition, sometimes the effectiveness of low-pressure water separators cannot be taken as sufficient because the rate of separation is not high enough. Besides, the temperature in separators (at the ECS outlet) should always be higher than the water freezing point. Consequently there were reasons to find new solution of the problem and it was found in the use of high-pressure water separation loops.

But it is a common practice that new solutions entail new problems. One of them is ice formation on condenser high-pressure side surfaces when the inlet moisture content decreases during the flight. A two-stage turbine settles this problem, but the amount of ice or overcooled water at the outlet of the loop increases up to ~8 g/(kg dry air) instead of less than 1 g/(kg dry air) for one-stage turbine schemes. Theoretical analysis can help making more accurate decisions.

References

[1] Warwick G. Boeing-777, the inside story. *Flight International*, p. 33, 25 December 1991 – 7 January 1992.