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Optimal operating conditions for an electric ECS in ground parking status

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Abstract

The Electric Environmental Control System (EECS), which uses an electric compressor to draw in outside air instead of the engine bleed, is under consideration. In this study, the case of EECS operating in a grounded parking status is discussed. The pressure ratios for the electric compressor and the air cycle machine compressor are set as variable parameters. A thermodynamic analysis is performed to clarify the operating conditions of the system, taking dehumidification into account. The coefficient of performance is obtained, and it is found that the maximum value is obtained under the condition where the pressure ratio of the electric compressor is the smallest.

Keywords: Environmental control system; Aircraft; Electrification; Air cycle machine; Electric compressor

1. Introduction

In recent years, the electrification of aircraft has been rapidly advancing as a countermeasure against environ- mental problems and to achieve higher performance due to decarbonization and rising fuel prices. Electrification of aircraft eliminates the need for complex hydraulic, pneumatic, and mechanical systems for aircraft operation. As a result, it is expected to increase design freedom, improve maintainability and increase fuel efficiency.

The Environmental Control System (ECS) is one of the essential equipment systems for commercial aircraft used primarily for passenger transport, providing life-sustaining, temperature-controlled air to passengers in extreme environments at high altitudes. In a conventional ECS, compressed ambient air is extracted by the engine and passed through the Air Cycle Machine (ACM) to deliver temperature-regulated air to the passengers. This conventional system uses a portion of the compressed air produced by the engines of aircraft flying at cruise altitude, which has the disadvantage of significantly impairing engine output. Therefore, a new method has been devised in which compressed air is generated by an electric compressor instead of using the engine bleed air [1, 2, 3]. The main purpose of this electric ECS (EECS) is to operate at high altitude cruise. At an altitude of about 35,000 feet, the outside air is dry and there is no need to consider the effects of humidity.

On the other hand, the percentage of humidity in the ambient air increases with decreasing altitude. In particular, when an aircraft is parked on the ground, the situation is different from that at high altitude, and the effect of humidity on the EECS cannot be ignored. In this study, a thermodynamic analysis of EECS operating under grounded aircraft is performed to clarify the conditions for optimum system operation, taking dehumidification into account.

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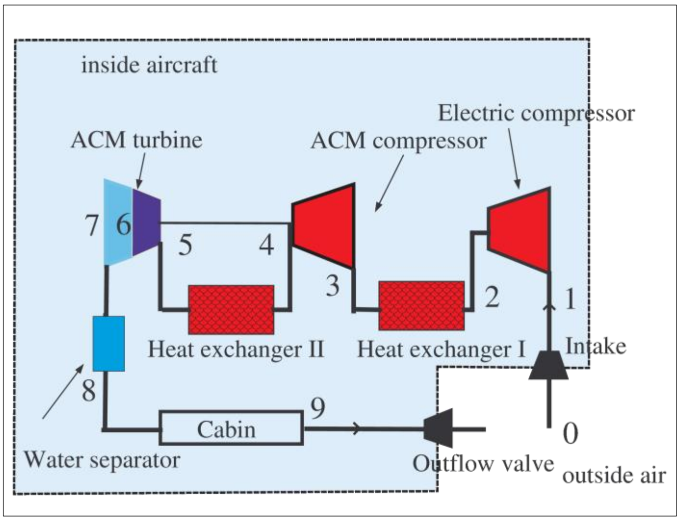
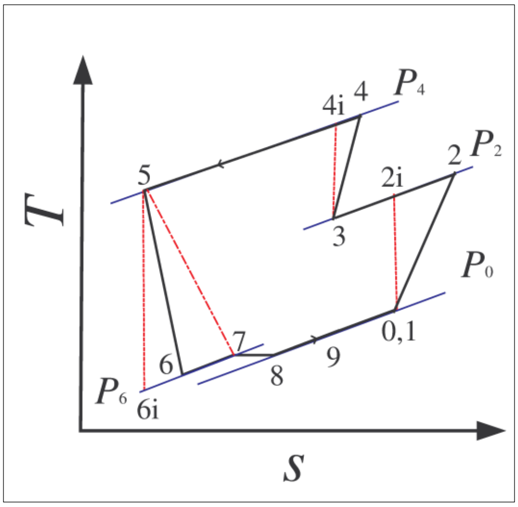
2. ECS model

We consider an Electric Environmental Control System (EECS) which consists of two compressors, two turbines and two heat exchangers as shown in Fig. 1(a). The moist air as working fluid is inspired from the outside through the intake (process 0-1) not bled from engine, compressed in an electric compressor (process 1-2), where the electric compressor executes a driving power of the EECS. The air is sent to an Air Cycle Machine (ACM). The air goes through the first heat exchanger for cooling by outside ram air (process 2-3, isobaric process), is compressed again in an ACM compressor (process 3-4) and goes through the second heat exchanger (process 4-5, isobaric process) for cooling. After that, it expands through an ACM turbine (process 5-6-7), arrives at a water separator device (7-8), and the cabin with appropriate pressure and temperature. It should be noted that the ACM turbine work is equal to the ACM compressor work. Then, the air leaves the cabin, and is discharged through an outflow valve to the outside (process 9-0). A schematic $T-s$ diagram corresponding to the successive air cycle behaviors is shown in Fig. 1(b). In our thermodynamic analysis, we define two pressure ratios in front of and behind the two compressors as a control parameter, where $\gamma_{12} = P_2/P_1$ is the ratio for the electric compressor and $\gamma_{23} = P_3/P_2$ for the ACM compressor. When calculating the state quantities for each state point in Fig. 1, it is necessary to consider the effect of humidity due to water vapor on mass flow rate, specific heat, and other parameters. In this study, the moist air, the working fluid, is treated as an ideal gas.



(a)Electric air-cycle system



(b)$T-s$ diagram

Figure 1 Schematic figure of the system

Since this study assumes aircraft ground parking conditions, the outside air contains water vapor and is humid. In such a case, a portion of the moist air, which is the working fluid, condenses during turbine expansion, and the condensed moisture must be dehumidified. The adiabatic expansion and condensation process in a turbine is shown in Fig. 2(a) as a $P - T$ diagram, where the subscripts are the state point values in Fig. 1. Figure 2(b) shows the relationship between absolute humidity X and temperature T to show the corresponding change in absolute humidity. In the actual turbine expansion and condensation process, a portion of the water vapor in the moist air condenses and releases latent heat as the temperature and pressure decrease. The temperature of the dry air rises as the dry air receives the released latent heat of condensation (5-7). In Fig. 2(a), the temperature and pressure of the working fluid (moist air) decrease from the turbine inlet (state point 5) and intersect the saturated water vapor line at point DP. The working fluid then moves along the saturated water vapor line to state point 7 as the pressure and temperature decrease and condensation occurs.

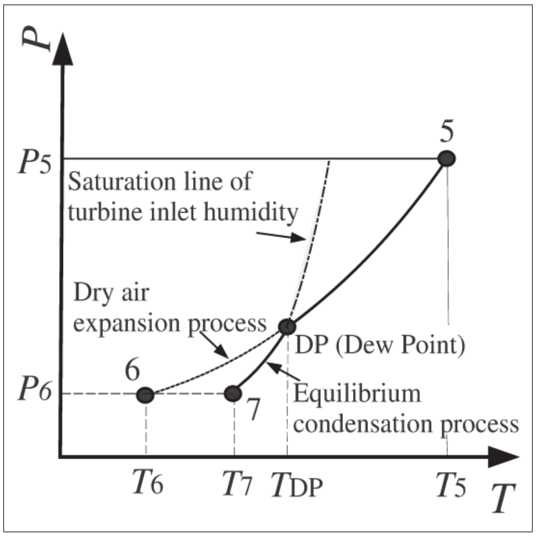
In this study, however, the expansion and condensation process, which involves a complex series of phase changes, is simplified by dividing it into two processes: the turbine expansion process (without phase change) and the condensation process (with phase change). In this case, in Fig. 2(a), after the temperature and pressure of the moist air decrease from the turbine inlet (state point 5) and intersect the saturated water vapor line at point DP, the moist air becomes supercooled from point DP to point 6. State point 6 is the state at the turbine outlet where the moist air is considered to have expanded adiabatically without condensation. This is the end of the turbine expansion process. After that, some of the water vapor in the moist air condenses and releases latent heat of condensation, and the dry air receives the condensation heat and is isostatically heated to the final equilibrium state at point 7. This is the condensation process. Figure 2(b) shows the expansion and condensation process in the corresponding turbine on a moist air diagram ($X - T$ diagram). The figure shows that the temperature of the working fluid decreases from the turbine inlet state (point 5) to the turbine outlet state (point 6) without changing the absolute humidity value X_0. In Fig 2(b), this is the turbine expansion process (undercooling process) without phase change. After that, the condensation process occurs and the absolute humidity drops to X_8 due to the decrease in water vapor in the moist air. At the same time, the dry air in the working fluid receives the latent heat of condensation and its temperature rises to reach the state 7. This is the condensation process. The change from state 6 to 7 in Fig. 2(b) is an adiabatic change in an open flow system, and since there is no external work, it is an isenthalpy change. The change from state 6 to 7 occurs along the isenthalpy line, and the temperature at the turbine outlet can be determined using the conditions of this isenthalpy process.

3. Results

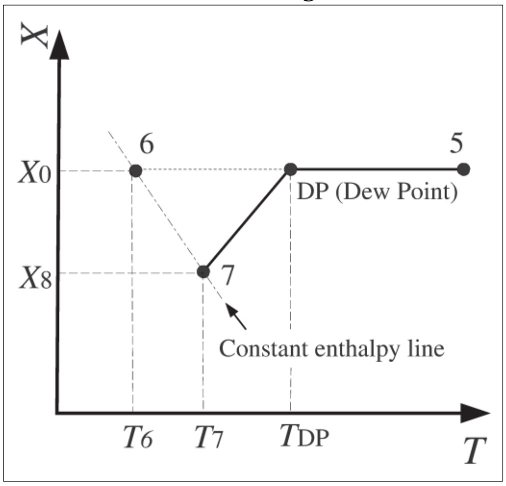
3.1. $T - s$ diagram

The aircraft is assumed to be parked on the ground in this study, with an open door at the entrance and exit, and the cabin pressure is assumed to be 1 atm. The outside air environment is assumed to be a high- temperature region such as Southeast Asia, where the outside temperature is 40 °C and the relative humidity is 60%. The mass flow rate of dry air, which is required for moist air as the working fluid, is 0.2495 kg/min per person. Assuming 200 passengers and crew, the total mass flow rate is 0.8317 kg/s. The temperature of the working fluid delivered to the cabin increases due to heat from the passengers in the cabin and heat absorption as it passes through the avionics. Q_C is the amount of heat per unit time emitted from the person in the cabin, and Q_A is the amount of heat generated per unit time from the avionics; Q_C is 15kW based on physiological data, and Q_A is 18 kW based on past experimental data. The cabin inlet temperature is 276.3 K.

Figure 3(a) shows the $T - s$ diagram for an electric compressor pressure ratio γ_{12} = 5.0 and ACM compressor pressure ratios γ_{23} = 1.5, 3.0, and 4.5 for an ambient air relative humidity of 60%. In the case of γ_{23}= 3.0, the pressure of the working fluid compressed by the electric compressor increases and the working fluid reaches the heat exchanger I. In the cooling process (2-3) in the heat exchanger I, the temperature and entropy decrease as the working fluid is cooled. The pressure and temperature then increase due to the compression of the ACM compressor and reach the heat exchanger II. In the cooling process (4-5) in heat exchanger II, the temperature and entropy also decrease. In both the heat exchanger I and II, the working fluid is cooled by the ambient air. Therefore, the outlet temperature of both the heat exchangers should be larger than the ambient air temperature. In the case of γ_{23} = 3.0, the temperatures at the outlet of the two heat exchangers are both larger than the ambient air temperature (the same temperature at state 0 indicated by T_0 in the figure).



(a) $P-T$ diagram



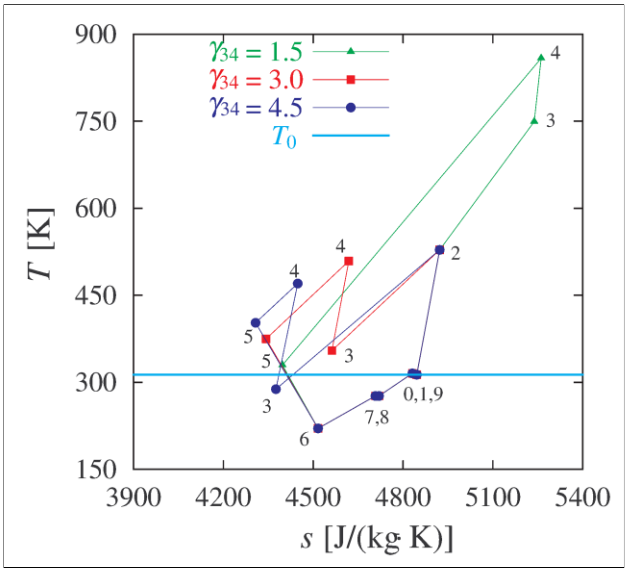
(b) $X-T$ diagram

Figure 2 Process for phase change

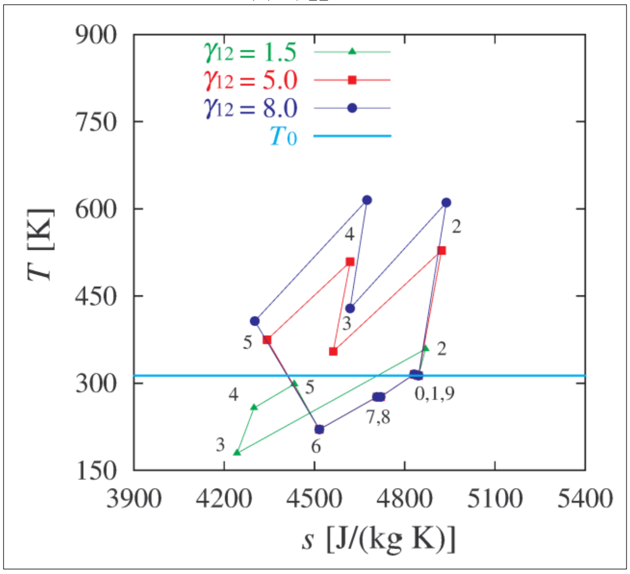
When the pressure ratio of the ACM compressor becomes as small as γ_{23} = 1.5, the compression process in states 3-4 moves to the high entropy side to ensure the temperature difference in compressor work in order for the ACM compressor and ACM turbine work to be equal in the ACM. This is because the larger the entropy region, the larger the temperature difference for the same pressure ratio due to the coarser density of the isobaric pressure line. In this case, the entropy increases in the process of heat exchanger I, and the working fluid that should be cooled is heated, which is inappropriate for EECS. At the same time, we can see the outlet temperature of heat exchanger II is lower than in the case where γ_{23} = 3.0, and if γ_{23} is further reduced, the outlet temperature of heat exchanger II is expected to fall below the ambient temperature. If the outlet temperature of the heat exchanger becomes smaller than the ambient air temperature T_0 of the cooling heat source, the system is not viable. On the other hand, when γ_{23} is large as 4.5, the outlet temperature of heat exchanger I is expected to decrease and become smaller than the ambient temperature. For larger values of γ_{23}, the entropy change or temperature difference becomes negative in the process of heat exchanger II, and the working fluid is expected to be heated. Furthermore, Fig. 3(b) shows the $T-s$ diagram when γ_{23} = 3 is constant and γ_{12} is varied. As γ_{12} becomes smaller, the outlet temperature of heat exchanger I becomes smaller and may become smaller than the ambient temperature, which is the cold heat source. Furthermore, when γ_{12} = 1.5, the entropy difference and temperature difference when passing through heat exchanger II become negative, creating a situation where the working fluid is heated in heat exchanger II. Thus, it can be seen that there is an appropriate operating range for the pressure ratios of the electric compressor and ACM compressor in order for the electric ECS to operate properly. Figure 4 shows the operating range of the pressure ratio for normal operation of the ECS, which is indicated by the red line and the relative humidity of the ambient air is 60%.

Table 1 Calculation condition

c_{dry}	[kJ/kgK]	1.004	$\dot{m}_{dry}$	[kg/s]	0.832
c_{vapor}	[kJ/kgK]	1.805	$P_0 = P_8$	[kPa]	101.3
h_0	[kJ/kg]	2501	R	[kJ/kgK]	0.008341
T_0	[K]	313.2	X_0	[kg/kgDA]	0.02841
T_8	[K]	273.6	Q_C	[kW]	15
Q_A	[kW]	18			



(a) $\gamma_{12} = 5$



(b) $\gamma_{34} = 3$

Figure 3 T-s diagram

3.2. Cooling process in the heat exchanger

In the EECS we have been dealing with in this study, the humidity in the working fluid is condensed by the expansion process of the ACM turbine, where the target humidity is unknown, but the system proceeds to change to the target temperature and then to saturation state. The system is designed so that all condensed water droplets are removed by the water separator. On the other hand, depending on the operating conditions of the electric compressor pressure ratio and ACM compressor pressure ratio, and the humidity conditions of the ambient air taken into the system, the condensation may occur during the cooling process in the two heat exchangers in the system. In this study, we avoid dealing with the details of the condensation process in the heat exchangers in detail, and dehumidification operations including condensation are performed in the ACM turbine and water separator. Therefore, in order to ensure stable operation of the system, the conditions under which the condensation process does not occur in the heat exchanger cooling process are necessaly to be further clarified, even in the pressure ratio operating range shown earlier. In order for condensation not to occur in the two heat exchangers I and II, the outlet temperature of each heat exchanger must be above the saturation temperature of the water vapor in the moist air. That is, if the outlet temperatures of heat exchangers I and II are T_3 and T_5, respectively, and the saturation temperature of water vapor in moist air at each heat exchanger's pressure is T_{dew1} and T_{dew2}, the following conditions must be satisfied to prevent condensation from occurring in the heat exchangers.

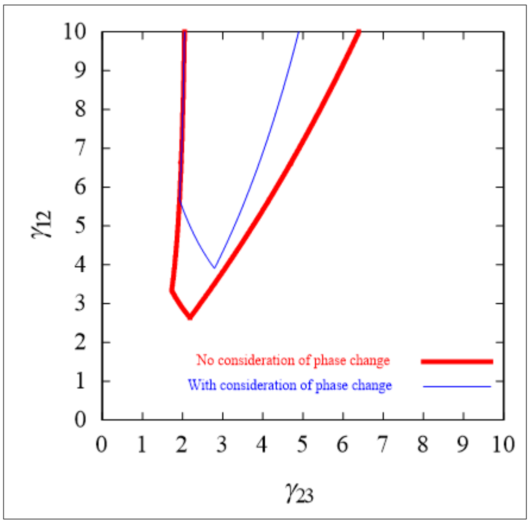


Figure 4 Operating range of the pressure ratios

$$T_3 > T_{dew1},\ T_5 > T_{dew2} \text{......................} (1)$$

A partial pressure of water vapor P_W is obtained in the heat exchangers from

$$P_W = \frac{PX}{X+0.622} \text{.....................} (2)$$

Substituting the total pressure in the heat exchangers and absolute humidity of moist air, which is the working fluid passing through heat exchangers I and II, into Eq. (2), the water vapor partial pressure in each heat exchanger can be obtained. The conditions of Eq. (1) can be confirmed by calculating the water vapor saturation temperature corresponding to the water vapor partial pressure calculated in Eq. (2) from PROPATH [4] comparing it with the outlet temperature of each heat exchanger. The results for the pressure ratio operating range when condensation in the heat exchanger is shown in Fig. 4 with blue line. When condensation is considered, the operation range becomes smaller.

3.3. Coefficient of performance

The EECS in this study can be regarded as a refrigeration system. Since the ACM compressor work is offset by the ACM turbine work, the coefficient of performance, *COP*, is defined as the ratio of the input power of the electric compressor to the cooling capacity of the cabin, as defined in the following equation.

$$COP = \frac{Q_C + Q_A}{\dot{m}_{dry}(1+X_0)\cdot(c_{dry}+c_{vapor}\cdot X_0)\cdot T_0 \frac{\gamma_{12}^{\frac{k-1}{k}}}{\eta}} \quad \text{............(3)}$$

The values of the performance coefficients are shown in Fig. 5 for several ambient relative humidities. It can be seen that the performance coefficients decrease monotonically with respect to the pressure ratio γ_{12} and with increasing humidity.

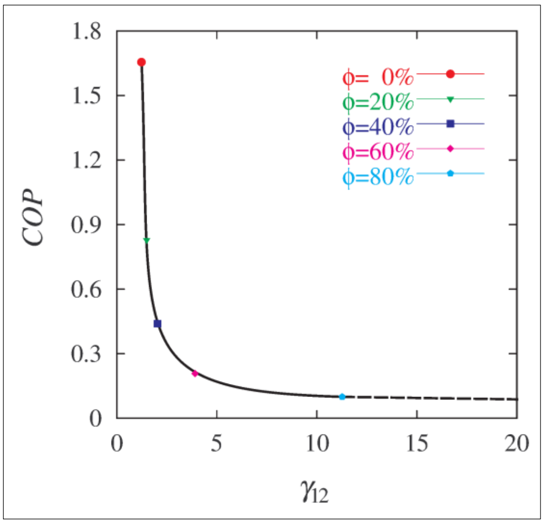


Figure 5 Coefficient of performance vs. γ_{12}

4. Conclusion

A thermodynamic analysis has been performed taking dehumidification into account. We have clarified the operating conditions for pressure ratios, where maximum value of *COP* is obtained for the smallest the pressure ratio of the electric compressor.

Compliance with ethical standards

Disclosure of conflict of interest

No conflict of interest to be disclosed.

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