THERMODYNAMIC STUDY OF AIRCRAFT AIR CONDITIONING AIR CYCLE MACHINES: 3-WHEEL x 4-WHEEL

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Abstract. *The main objective of this report is the development of aeronautic air conditioning system thermodynamic models of 3 and 4-wheel air cycle machines, aiming to understand the main thermodynamic advantages and disadvantages of each configuration. The EES – Equation Engineering Solver is used to develop the models, some parametric analysis and the graphic outputs. One's intention also is to evaluate the impacts of the re-heaters and condensers in the system efficiency. The thermodynamic models are tested under ground and flight operational conditions. Results showed that the second level of expansion from 4-wheel air cycle machine increases the thermodynamic efficiency of the cycle, and this effect was more evident for ground operation conditions. In addition, it was observed that the presence of a condenser decreases the coefficient of performance of both air cycle machines discussed, which is related to the fact that lines of constant pressure on a T-s diagram diverge with increasing entropy. One's concluded that the application of the software EES for thermodynamic analysis of air cycle machines can be really useful for advanced design analysis, mainly because it helps in understanding the thermodynamic process considered.*

Keywords*: Aircraft Air Conditioning, Air Cycle Machine (ACM), Coefficient of Performance (COP), 3-wheel, 4-wheel*

1. INTRODUCTION

According to SAE ARP 147, 2001, Environmental Control System (ECS) is defined as the group of systems responsible for the control of pressure, temperature, humidity and contaminants of an aircraft, aiming at the occupants' survival and comfort, as well as to guarantee the integrity of some electrical equipment carried onboard. In addition, ASHRAE 1997 defines this term also encompass functionalities such as rain and ice protection, oxygen distribution to the occupants and other pneumatic demands of the aircraft. The present work treats basically about thermodynamic models of the air conditioning system, which is the main system for the control of the aircraft thermal variables.

The main challenge

The air conditioning system has to comply with various requirements: aircraft thermal load, pressurization, ventilation and occupants' thermal comfort. The air conditioning system must assure an adequate minimum fresh airflow of not less than 0,047m^3/s (10 ft^3/min) per occupants, aiming at the compliance with the requirements imposed by airworthiness authorities. Moreover, it must supply enough air to pressurize the aircraft with an equivalent maximum altitude of 2438m (8000 feet) and respect all the pressure variations, complying with the physiological necessities of human beings. The air conditioning system acclimatizes the aircraft operating at extremely low temperatures bellow -60°C on high altitudes, and on the other extreme, on ground hot soaked conditions at sea level, with the presence of high levels of moisture and temperatures above 50°C. Considering all these requirements, the usual design point is the pull-down requirement, which is the transient condition where the aircraft must be cooled down within minutes to adequately receive the passengers and crew. The faster the aircraft gets acclimatized, the faster the aircraft is ready to receive the occupants and fly, minimizing the waste of time and money of its operators.

Aircraft air conditioning systems

There are two types of aircraft air conditioning systems, which can be applied individually or in a combined way:

- air cycle machines (ACM): based on the air refrigeration cycle (Brayton reverse), where the rejected heat is obtained by the transformation of thermal energy into expansion work;
- vapor cycle machine (VCM): based on the vapor refrigeration cycle, where the heat rejection is obtained by the phase transformation of a refrigerant fluid (usually applied in buildings and cars);

In general vapor cycle machines are more efficient than air cycle machines. In part this is explained by the vapor cycle operating principle (latent heat exchange with constant temperature), more efficient than the air cycle one (sensible heat exchange with constant pressure) according to Gigiel et al, 2001. To balance this effect, the air cycle machines need more refrigerant mass flow than the vapor cycle machines. However, when there is enough bleed air available (usually midsize jets and larger), the air cycle machine is more often employed mainly due to its simplicity and the fact its components are lighter, more compact and more reliable (SAE ARP 1168/3, 2004). Even with a reasonable coefficient of performance (COP), vapor cycle machines generally still require a considerable electrical power leading to electrical generator size increasing and impacting aircraft installation and weight. Recently, although the electrical generator technology improvement is benefiting the choose of vapor cycle technology, the air cycle is still more attractive. Table 1 presents some examples of the use of these refrigeration machines in aviation:

Table 1: Air cycle and vapor cycle applied in aviation

Air cycle machine	Largely applied on aviation, from midsize jets up to very large aircraft. Examples: B373, B747, A320, A340, A380, EMB120, EMB145, EMB170, GV, G200, etc.
Vapor cycle machine	Only used on light and very light jets. Examples: Embraer Phenom, CJ1, CJ2, CJ3, Lear45, etc.

Air cycle machines (ACM)

Among the air cycle machines available, four types are more commonly employed in aviation: simple cycle; bootstrap cycle; simple/bootstrap cycle (3-wheel) and the condensing cycle (4-wheel). In addition there exist other types that in fact are derived from the cited types.

The simple cycle was the first one to be developed and applied in aviation. Composed by a turbine and a fan in the same axe, and a heat exchanger, this is the lighter and most compact air cycle architecture. According to Rannenber, 1969, its low efficiencies for ground operation do not justify its application on modern commercial aircraft.

The bootstrap cycle is more efficient than the last one, which in turns brings a complexity and weight increasing. This is obtained with the addition of a compressor stage by replacing the exhausted fan by a compressor (see **Figure 1**). In spite of it, it needs an additional heat exchanger and an auxiliary electrical fan to forces the ram air through the heat exchangers when on ground.

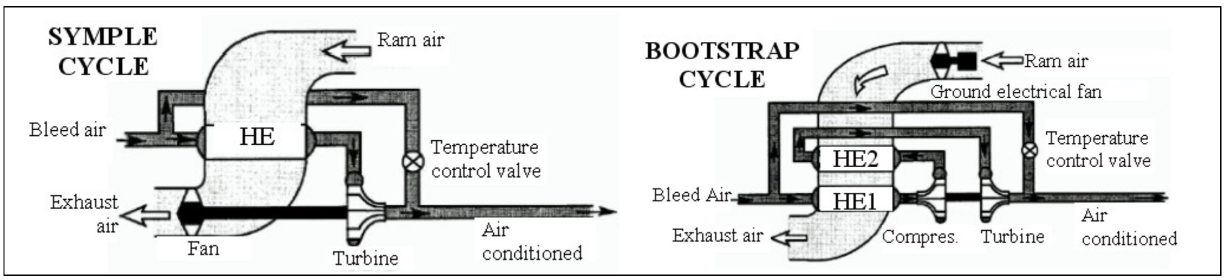

Figure 1: Simple cycle (turbine and fan) and Bootsrap cylcle (turbine and compressor), Andrade et al, 2004.

The simple/bootstrap cycle (3-wheel), introduced in the 80's in the majority of commercial aircraft (according to DeFrancesco, 1993, in the Boeing 757, Boeing 767, and Airbus 320), is a natural evolution of the last cycles, combining the fan from simple cycle with the compressor from bootstrap, mounted in the same turbine axe (see **Figure 2**). The exhausted fan avoids the necessity of an extra electrical fan for ground operation, and the compressor absorbs 85% of the turbine expansion power augmenting the cycle efficiency. The remaining 15% are absorbed by the fan, which supplies ram air to the heat exchangers.

The condensing cycle (4-wheel) is the last air cycle air conditioning system developed and was first installed in the Boeing 777, and afterwards in the Macdonald Douglas 12 (ASHRAE, 1999). It is similar to the 3-wheel machine, but with an additional expansion stage. It has two turbines; one compressor and a fan mounted in the axe (see **Figure 2**).

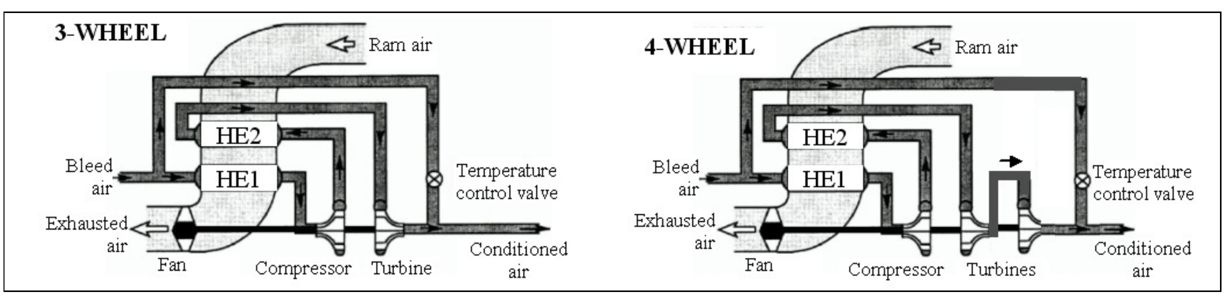

Figure 2: 3-wheel machine and 4-wheel machine, Andrade et al, 2004.

As presented in Table 2, the 3 and 4-wheel air cycle machines are each time more used in modern aviation.

Table 2: 3 and 4-wheel machines applied in aviation

MODEL	ACM
Concorde, CESSNA 550; EMB-120	2-wheels
SAAB 2000; EMB-145, G200, GV; A320; B757; B767; B777; MD12; EMB-170; EMB190	3-and 4-wheels

Motivation in the pre-design phase

The activity of design and selection of an air-conditioning system is usually faced in the pre-design phase where first aircraft characteristics are under definition. Changes during this phase are easier and cheaper to accomplish, and the need for taking the write decisions is a well-know verity and will preclude costs build up and unexpected delays later on. Therefore, the use of prediction tools such as digital mock-ups and mathematical models are each time more essential. With this in mind, the analyze methodology presented herein is intended to help engineers in the pre-design phase to select the air conditioning system, considering basic thermodynamic models for air cycle systems.

Main objective

The main objective of this report is the development of mathematical models of 3 and 4-wheel aeronautical air conditioning systems, aiming to understand the thermodynamic processes evolved, as well as to identify the main advantages from an energetic efficiency point of view. It was analyzed two operational conditions: flight at approximately 11.8km (39,000ft) and hot soaked ground condition. The Equation Engineering Solver (EES, 2004) is used to develop the models, some parametric analysis and the graphic outputs.

2. THERMODYNAMIC MODELS

Herein is described the thermodynamic models for the 3 and 4-wheel air cycle machines.

2.1. 3-WHEEL MODEL

The engine inlet diffuser promotes an initial compression (free stream pressure recovery) from “1-2”, as represented in the Figure 3.

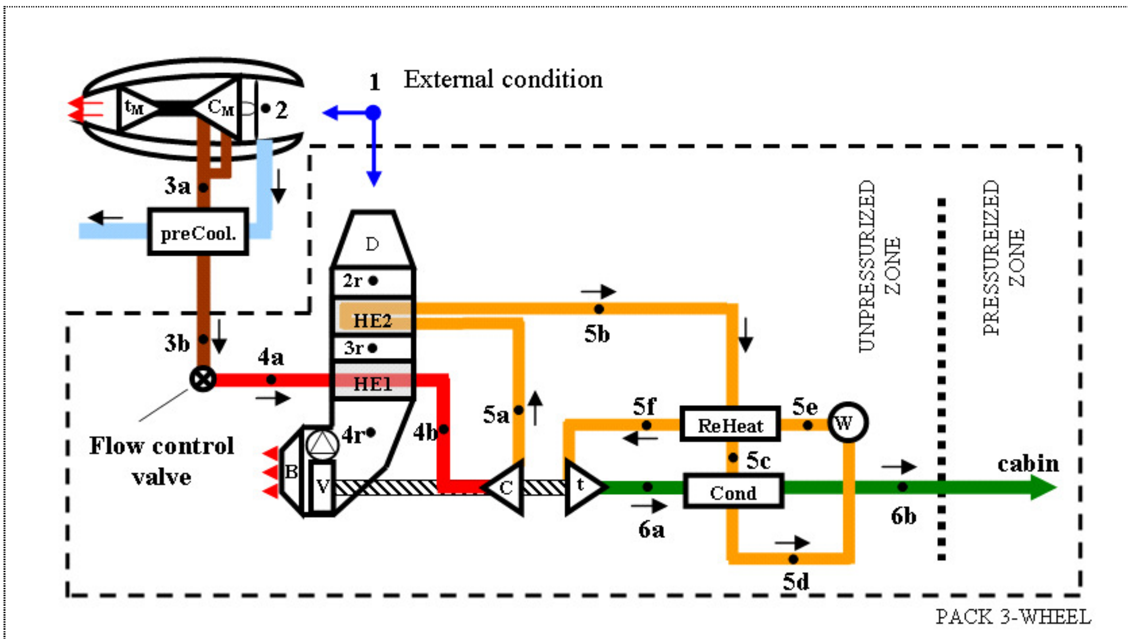

Figure 3: Three-wheel air cycle machine with series heat exchangers, condenser and re-heater

Considering its isentropic efficiency “η_d”, the stagnation pressure ”P_{2i}” and the outlet real pressure “P_2” are obtained with:

$$\frac{P_{2i}}{P_1} = \left(\frac{T_{2i}}{T_1}\right)^{\frac{k}{k-1}}, \text{ and } \eta_d = \frac{P_2 - P_1}{P_{2i} - P_1}. \tag{1}$$

The engine compressor, also called primary compressor, pressurizes the air from “2-3a", leading it to its ideal temperature “T_{3i}” and real temperature “T_3”according to the isentropic equations:

$$\frac{T_{3ai}}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{k-1}{k}}, \text{ and } \eta_{c1} = \frac{T_{3ai} - T_2}{T_{3a} - T_2}, \tag{2}$$

Where “η_{c1}” is the primary compressor isentropic efficiency and “P_3” is the bleed port pressure. The primary compressor real work per mass of dry air “w_{c1}”, in [J/kg], is obtained by:

$$w_{c1} = c_p\left(T_{3a} - T_2\right). \tag{3}$$

The pressure and temperature regulation upstream the pack (state “4a") is done by the aircraft pneumatic system, herein represented by the bleed air ports, the pre-cooler and the flow control valve. It is considered the pneumatic system is capable of keep the following conditions:

$$P_3 = 250\text{kPa}\ (36{,}3\text{psi});\ T3b = 200°C(392°F);\ P4 = P3 - 50\text{kPa}\ ;\ T4a = T3b. \tag{4}$$

The heat rejected in the pre-cooler is calculated with:

$$q_{pr} = C_p\left(T_{3a} - T_{3b}\right). \tag{5}$$

Considering the thermal capacity in the hot side is smaller or equal to the cold side thermal capacity (typical design condition), the pre-cooler effectiveness is obtained with the following relation (Stoecker, 1989):

$$\varepsilon_{pr} = \frac{C_q\left(T_{q,e} - T_{q,s}\right)}{C_{min}\left(T_{q,e} - T_{f,e}\right)} = \frac{\left(T_{3a} - T_{3b}\right)}{\left(T_{3a} - T_{2r}\right)}, \tag{6}$$

Where: “q” = hot; “f” = cold; “e” = inlet; “s” = outlet; “C_q” = Thermal capacity of hot side, obtained by “$m.c_p$”, being “m” = hot mass flow [kg/s]; “C_{min}” = Minimal thermal capacity between both sides (here consider equal to “Cq”); “T_{2r}” = Heat sink inlet temperature, equal to “T_2”.

In the process “4a-4b”, the air is cooled by the primary heat exchanger. The pressures losses are neglected, and the heat rejected by dry air mass flow “q_{tc1}” is:

$$q_{tc1} = \frac{Q_{tc1}}{\dot{m}} = C_p\left(T_{4a} - T_{4b}\right). \tag{7}$$

The temperature “T_{4b}” and the heat rejected in the primary heat exchanger can be calculated in the same way as shown for pre-cooler.

After an isentropic compression in the secondary compressor (ACM compressor), the temperatures “T_{5ai}” and “T_{5a}”, and its real work per dry air mass flow are obtained as presented for the primary compressor (similar to Eq. (1) and (2)).

Considering isentropic expansion, the air temperature downstream turbine is “T_{6i}”. Neglecting the pressure losses in the distribution system, one’s can consider $P_6 = P_{cabin}$.

The turbine outlet temperatures “T_{6i}” and “T_6” and the turbine real work is obtained with the following relations:

$$\frac{T_{6i}}{T_{5b}} = \left(\frac{P_6}{P_5}\right)^{\frac{k-1}{k}},\ \eta_{c1} = \frac{T_{5b} - T_6}{T_{5b} - T_{6i}},\ \text{and}\ \ w_t = c_p\left(T_{5b} - T_6\right). \tag{8}$$

The pressure “P_5” is obtained solving the implicit work balance in the air cycle machine: the work from turbine is equal to the sum of the compressor work plus the exhaust fan work.

$$w_t = w_c + w_v. \tag{9}$$

To distribute the turbine generated work among the consumers it is used the non-dimensional factor first proposed by Andrade et al, as follows

$$w_c = \alpha\ (w_t). \tag{10}$$

Therefore, the work available for the exhaust fan is $(1\text{-}\alpha)w_t$. When α=0, the exhaust fan burns the whole turbine power (simple cycle) and when α=1, the compressor itself is the only consumer (bootstrap cycle). For intermediate values of “α”, the turbine power is divided between the compressor and the exhaust fan (simple/bootstrap cycle, or three-wheel cycle).

Considering there is no latent heat exchanger in the re-heater (there’s no water condensation), the sensible heat rejected by dry air mass airflow is:

$$q_{reaq} = C_p\left(T_{5b} - T_{5c}\right) = C_p\left(T_{5e} - T_{5d}\right). \tag{11}$$

Whenever the external moisture is significant, the work fluid is considered a mixture of dry air $\dot{m}_{AS}$, water vapor $\dot{m}_{VA}$, and free moisture $\dot{m}_L$:

$$\dot{m} = \dot{m}_{AS} + \dot{m}_{VA} + \dot{m}_L. \tag{12}$$

Where “ $\dot{m}$ “ is the total flow [kg/s];

The absolute moisture of the saturated humid air and the partial pressure of the saturated water vapor are calculated respectively with (ASHRAE, 1997):

$$W_{sat} = 0.622\frac{P_{VA\_SAT}}{P - P_{VA\_SAT}}, \tag{13}$$

and

$$P_{VA\_SAT} = \exp\left(\frac{C_1}{T} + C_2 + C_3T + C_4T^2 + C_5T^3 + C_6T^4 + C_7\ln(T)\right). \tag{14}$$

According to SAE AIR 1168/6, 2004, ground operation air conditioning system design point must consider external air absolute moisture of 22g/kg. Although condensation occurs in other components, only the water separator is capable of removing free moisture. Therefore, its inlet total moisture is the same as the external air. Furthermore, it is considered that the condenser downstream total moisture is 7g/kg (typical value, according to DeFrancesco, 1993).

It is also considered the air leaves the condenser within its dew point temperature, calculated as follows:

$$T_{cond,s}=T_O\,; \quad T_O = \text{TEMP}(AirH2O;\ W=W_{cond,s}\ ;\ P=P_{cond,s};\ UR=1)\ , \tag{15}$$

Where: "T_O" = humid air dew point temperature [K]; "TEMP" = thermodynamic function to calculate the dew humid air temperature function of total moisture "W" [kg/kg], total pressure "P" [Pa], and relative humidity "UR"[---].

It is important to add that whenever the air is considered a mixture of humid air and free moisture (not only dry air), the thermodynamic equations discussed herein is not only function of its temperatures and pressures, but of its enthalpies.

The cabin cooling effect per mass flow can obtained with the following equation:

$$q_c = c_p(T_{cabin} - T_{in}), \tag{16}$$

Where "T_{cabin}" is the cabin air temperature and "T_{in}" is the temperature of the air supplied to the cabin.

Aiming to comply with human physiological necessities, a percentage of the primary compressor work is used for the cabin air pressurization during altitude flights. This amount is calculated as follows:

$$w_p = \frac{C_pT_2}{\eta_{c1}}\left[\left(\frac{P_{cabine}}{P_2}\right)^{\frac{k-1}{k}} - 1\right] + w_d\ . \tag{17}$$

The air cycle machine coefficient of performance (COP) which consider the work of pressurization, and the one which does not consider it (COP_P) are obtained with the following equations:

$$COP_P = \frac{q_c}{w_d + w_{cp}} \text{ and } COP = \frac{q_c}{w_d + w_{cp} - w_p}\ . \tag{18}$$

Considering the air cycle machine as a control volume, and assuming the following hypothesis:

- steady state;
- negligible kinetic and potential energy variations between inlet and outlet;
- modeling the air as a perfect gas;
- assuming the mass inflow is identical to the mass outflow,

the energy balance is obtained with (Moran et al, 1996):

$$\frac{dE_{cv}}{dt} = \dot{Q}_{CV} - \dot{W}_{CV} - \dot{m}C_p\left(T_{cabin} - T_1\right) = 0\ , \tag{19}$$

Where: "$\dot{Q}_{CV}$" = resultant heat exchanged through the control volume [W]; "$\dot{W}_{CV}$" = resultant work done by the control volume [W].

2.2. 4-WHEEL MODEL

The equations for 4-wheel and 3-wheel are identical up to the condenser outlet (Figure 4, point "6b").

After an isentropic expansion, the second turbine outlet temperature is "T_{7i}". The real temperature "T_7" and the work generated "w_{t2}" can be calculate knowing the turbine isentropic efficiency "η_{t2}" – similarly to the first turbine, as presented with the Eq. (8).

The pressure "P_5" is determined solving the air cycle machine balance equation: the sum of the work from turbines 1 and 2 is equal to the sum of compressor work and exhaust fan work: $w_{t1} + w_{t2} = w_{c2} + w_{v.}$

The work distribution between the users can be done using the same factor employed for the 3-wheel machine:

$$w_{c2} = \alpha\ (\ w_{t1} + w_{t2}\ )\ , \tag{20}$$

Where "α" is the percentage of the total work generated by both turbines, used by the ACM compressor. Thus, the work available for the fan is $(1-\alpha)(w_{t1}+w_{t2})$. When α=0, the exhausted fan absorb the whole work, and when α=1, the ACM compressor does. For intermediate values of "α", the work from turbines is divided between compressor and fan.

The responsibility division between both turbines regarding work generation is defined by:

$$w_{t1} = \beta \, (w_{c2} + w_v) \, , \tag{21}$$

Where "β" is the percentage of total work generated by turbine 1. Thus, the turbine 2 work is $(1- \beta)(w_{c1} + w_v)$.

When β=0, w_{t1}=0, and the work is totally produced by turbine 2. When β=1, turbine 1 does produce the whole work. For intermediate values of "β", the work is produced by both turbines.

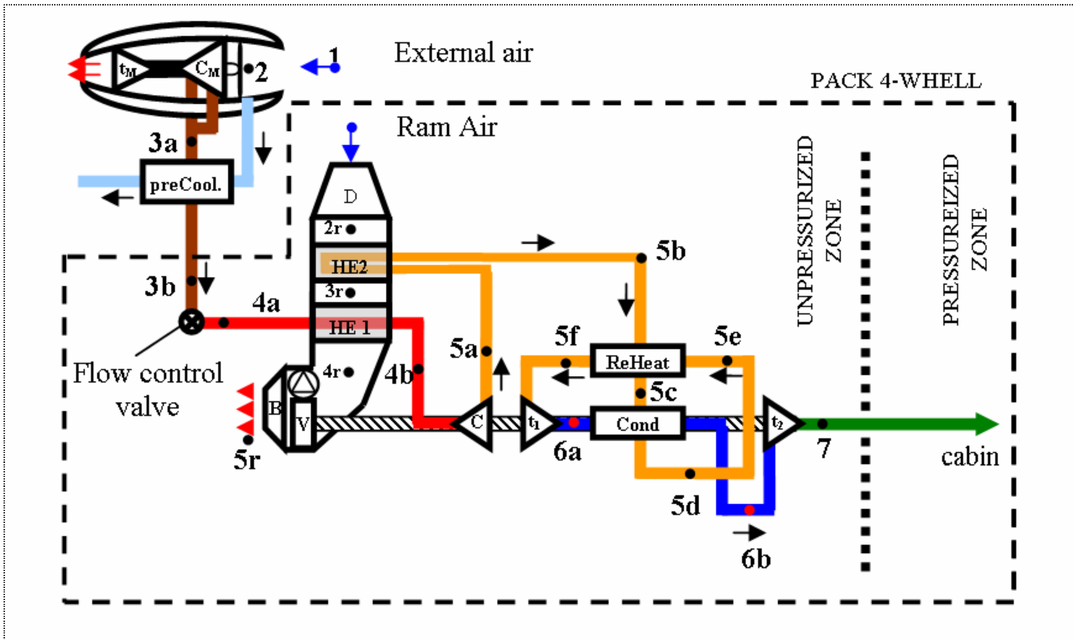

Figure 4: Four-wheel air cycle machine with series heat exchangers, condenser and re-heater

2.3. VALIDATION

The thermodynamic model was validated based on the simulation results obtained by Andrade et al, 2004. The authors have modeled a 3-wheel air cycle machine with heat exchangers installed in parallel, and simulated a cruise condition at 11.8km (38,700ft), at mach 0.47. The compressor, turbine and diffuser efficiencies as well as the heat exchangers effectiveness was kept constant. The flow control valve was considered adiabatic with a constant pressure loss of 50kPa. Other component and distribution duct losses were neglected.

Table 3 presents the COP obtained, showing a good proximity between the present work results and the reference.

Table 3: Output values comparison

Parameter	Description	Present work	Andrade et al, 2004
COP	Coefficient of performance	0.7565	0.7564
COP_P	COP including pressurization work	0.4279	0.4278

In addition, Figure 5 presents a parametric study where the influence of the Mach number on the coefficient of performance is analyzed. It is observed the results are very similar, and that the Mach number increasing promotes the reduction of the coefficient of performance, since the heat sink temperature is augmented.

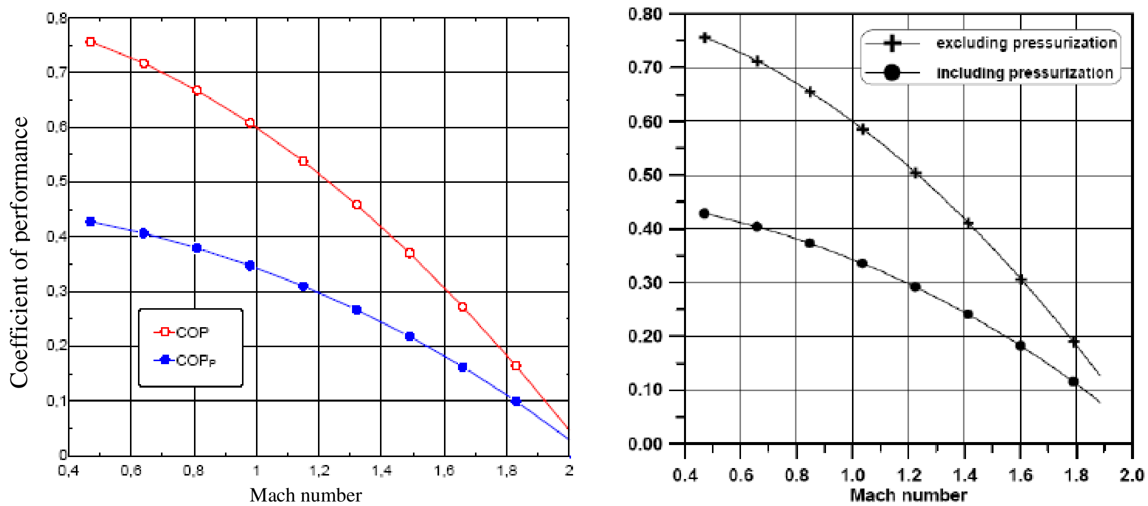

Figure 5: Coefficient of performance function of Mach number. Left curves obtained with the present work, and right curves obtained by Andrade et al, 2004.

3. SIMULATION INPUTS

The main parameters used for the simulations are listed in Table 3.

Table 4: Simulation input parameters

	ON GROUND		*ON CRUISE*	
	3-WHEEL	*4-WHEEL*	*3-WHEEL*	*4-WHEEL*
State "1" (external air):	T_1=40°C; P_1=101kPa;		T_1 = -57°C; P_1 = 20 kPa (ISA);	
State "3b" (pre-cooler outlet):	T_{3b} = 180°C; P_3 = 350 kPa;		T_{3b} = 200°C; P_3 = 250 kPa;	
State "4a" (1HE inlet):	$T_{4a} = T_{3b}$; $P_4 = P_3$ - 50 kPa;			
Turbine, compressor, diffuser efficiency:	$\eta_{t1} = \eta_{t2} = 0.77$; $\eta_{c1} = \eta_{c2} = 0.82$; η_d =0.84;			
HE and re-heater effectiveness:	$\varepsilon_{tc1} = \varepsilon_{tc2} = 0.80$; $\varepsilon_{reaq} = 0.2$			
Cabin internal air:	P_{cabin}= 101kPa; T_{cabin}= 24°C;		P_{cabin} = 2.4km; T_{cabin} = 24°C;	
Aircraft operational conditions:	Altitude = 0 m; Mach = 0;		Altitude = 11.7km; Mach =0.47;	
Condenser total moisture [g/kg]:	$W_{T,cond,e}$ = 22; $W_{cond,s}$ = 7;		N/A	
Condenser boundary condition:	$T_{cond,s}$ =dew point temp.		ε_{cond}=0.2	
Factors: α and β	α = 0.8	α = 0.8; β = 0.5	α = 1	α = 1; β = 0.5

4. RESULTS AND DISCUSSIONS

The thermodynamic processes obtained for 3 and 4-wheel air cycle machines for cruise operation are presented on Figure 6. The dashed lines are the isobaric process, and the continuous are the compression and expansion processes on turbines and compressors. Each state temperature is presented on the legends.

The 4-wheel machine obtains a lower turbine outlet temperature (188.8K against 193.8K) due to the fact that its turbine pressure ratio is lower than that of the 3-wheel unique turbine. Conceição, 2006, has shown the lower is the pressure ratio in the turbine the lower is the entropy increasing at its expansion, which is related to the fact for perfect gases the lines of constant pressure diverge on a T-s diagram with increasing entropy. Since the entropy increasing in the 4-wheel turbines expansion is minimized when compared to 3-wheel, its power generation is higher leading to a higher pressure at the compressor outlet (P_5). Both machines condensers sensible heat exchange processes is represented by: $T_{5b} \rightarrow T_{5c}$ - heat rejecter at the condenser hot side; $T_{6a} \rightarrow T_{6b}$ - heat recovered at the condenser cold side. Since the 4-wheel condenser cold side pressure is higher than the 3-wheel's, its cold side inlet temperature is higher (216.6K versus 178.3K) and thus the heat rejected is minimized. All these factors contribute to make the 4-wheel COP 5% higher than the 3-wheel`s for cruise condition as shown on top part of Table 5. As expected, the COP, which does not consider the work of pressurization, is higher than COP_P.

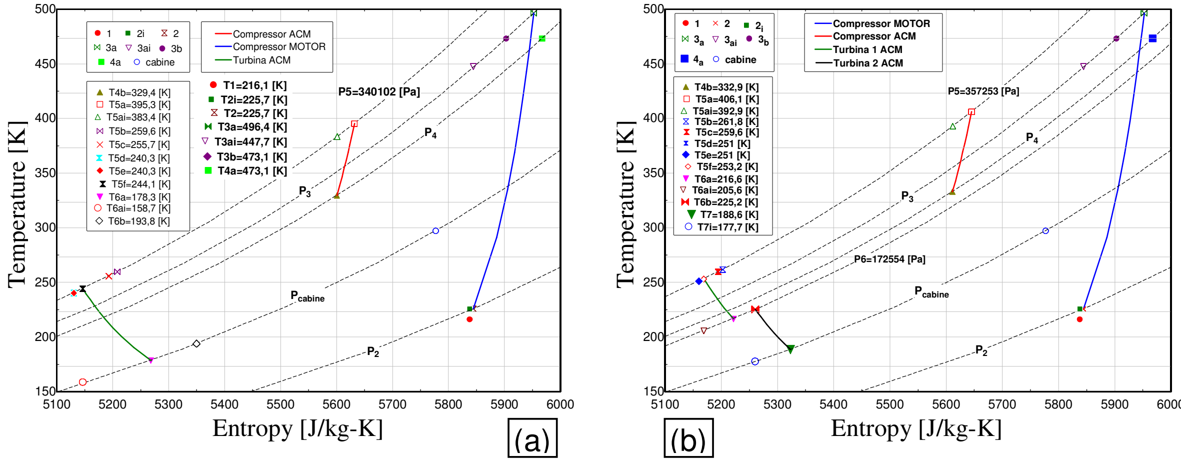

Figure 6: Thermodynamic process obtained for cruise operation. (a) 3-wheel; (b) 4-wheel.

Figure 7 presents the thermodynamic processes obtained for 3 and 4-wheel air cycle machines on ground operation. If compared to diagrams from last figure, it is noticeable the isobaric lines are closer because the external pressures on ground are higher. Since the external air water content is considered for ground operation, the latent heat exchanger at the condensers produce a superior entropy variation on its cold side ($T_{6a} \rightarrow T_{6b}$). As the cabin pressure is equal to the external pressure, COP is equal to COP_P for this case. All the advantages of the 4-wheel machine discussed for cruise operation are maximized for ground operation, and its COP is 91% higher than that obtained for the 4-wheel configuration, according to Table 5.

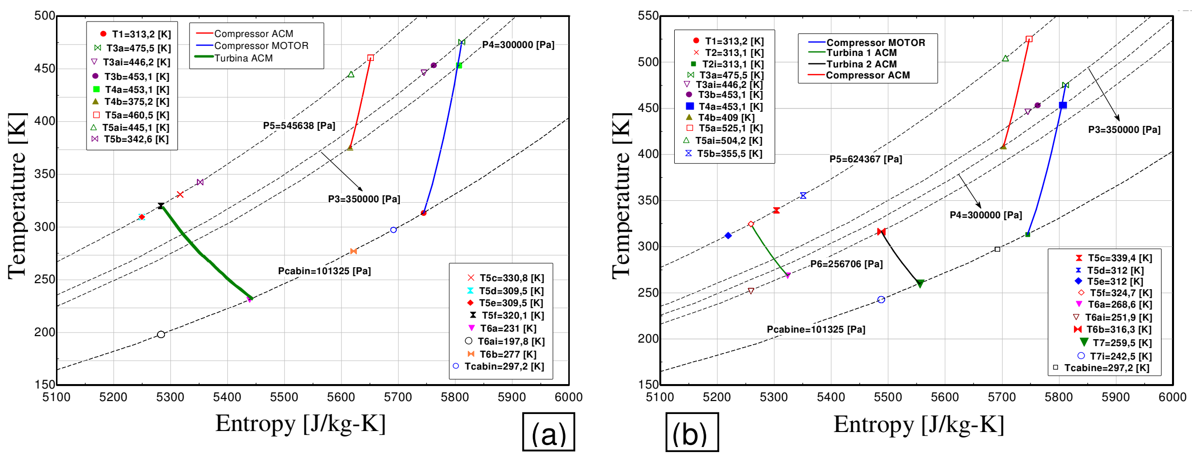

Figure 7: Thermodynamic process obtained for ground operation: (a) 3-wheel; (b) 4-wheel.

It is observed the efficiencies for cruise operation are greater than those for ground operation, mainly due to the lower heat sink temperature (225.7K versus 313.2K). The following table summarizes the results just discussed.

Table 5: Results obtained in the simulation

	Model	*COP*	COP_P	q_c [kW/kg/s]	T_{insufl} [K]	w_t [kW/kg/s]	P_5 [kPa]	q_{cond} [kW/kg/s]
CRUISE	3-wheel	0.6519	0.3687	103.4	193.8	65.8	340.1	15.5
	4-wheel	0.6848	0.3874	108.6	188.6	73.2	357.2	8.6
	%	+5	+5	+5	-3	+11	+5	-44
GROUND	3-wheel	0.1735		29.4	277.0	111.3	545.6	58.5
	4-wheel	0.3321		56.3	259.5	151.7	624.4	67.2
	%	+91		+91	-6	+36	+14	+15

5. PARAMETRIC ANALYSIS

The increase of the air cycle pressure ratio produces a reduction in its coefficient of performance (COP), which can be observed in the Figure 8 and Figure 9. These curves were developed using the 4-wheel model operating on cruise. At first, the independent variable "P_3" was increased, resulting in the increase of "P_5", what finally produces the cycle pressure ratio rise. Then the cabin pressure is reduced by means of cabin altitude variation, producing the cycle pressure ratio augmentation as well. At the same time the COP is reduced, both figures show that for higher cycle pressure ratio the cooling effect "q_c" is increased. It means that the maximization of the coefficient of performance is obtained with the minimization of the cooling effect. In real cases, the COP optimization by means of pressure ratio minimization is not possible because the resultant cooling effect is a function of the aircraft pull-down requirements, and can not be reduced. A second point is that lower pressure ratios could mean lower aircraft sizes, but on that sort of aircraft, the air cycle machine is not viable due to the engine bleed air limitations.

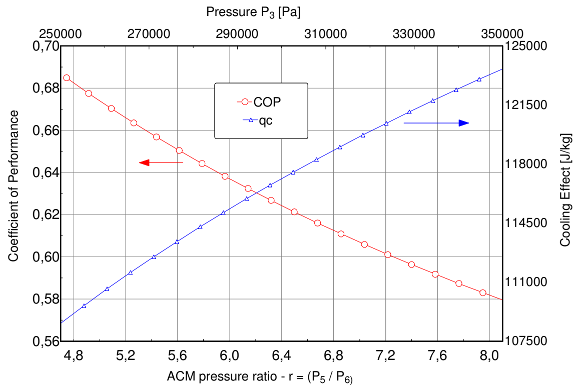

Figure 8: Coefficient of performance and cooling effect function of the ACM pressure ratio (increasing of P_3)

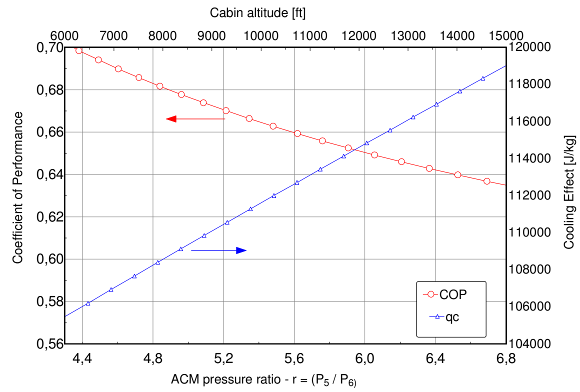

Figure 9: Coefficient of performance and cooling effect function of the ACM pressure ratio (reduction of P_{cabin})

A second parametric analysis is presented in Figure 10 and Figure 11, where the impacts of the condenser effectiveness and moisture outlet on the cycle COP is analyzed for cruise and ground condition, respectively. It is observed the greater the condenser heat rejected the lower is the cycle COP for both conditions. It happens because the higher the condenser heat, the higher is the temperature variation on both condenser sides and therefore, the higher will be the entropy increasing in the condenser low pressure side. As a result, the inflow temperature is increased, impacting on the cycle COP.

Both figures show the 4-wheel is less impacted because, as already presented in Figure 6(a) and (b), its condenser cold side has a higher pressure than 3-wheel's, resulting in higher cold side temperatures. Finally, these higher cold side temperatures minimize the condenser heat rejected and also improve the second turbine expansion power.

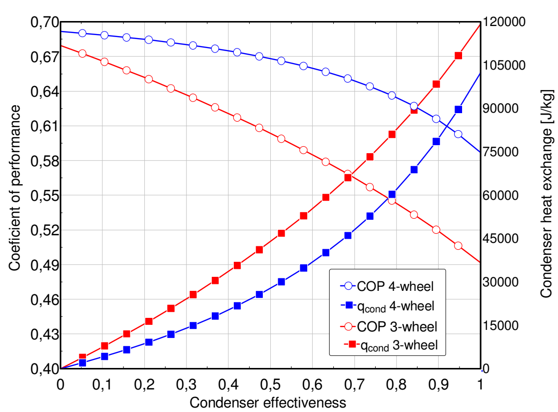

Figure 10: Coefficient of performance and condenser heat rejected function of condenser effectiveness (cruise operation)

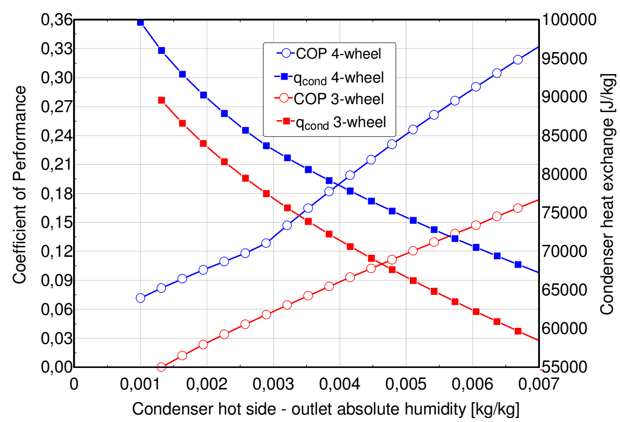

Figure 11: Coefficient of performance and condenser heat rejected function of condenser absolute humidity outlet (ground operation)

Even if both air cycle machines rejected the same amount of heat in the condensers (60kJ/kg, for instance), Figure 10 shows that the 4-wheel machine has a higher COP than 3-wheel's (approximately 0.64 against less than 0.61). For ground operation, Figure 11 indicates that for the same water removal in the condenser, 4-wheel air cycle machine have a significantly higher COP than 3-wheel's, even rejecting more heat on its condenser. When the condenser outlet absolute moisture is 0.007kg/kg, the 3 and 4-wheel COP are respectively 0.18 and 0.33, according to Table 5.

6. CONCLUSIONS

The insertion of a second expansion stage in an air cycle machine brings theoretical thermodynamic gains basically due to an increasing in the turbines expansion efficiency, resultant from a lower pressure ratio felt by each turbine. Conceição, 2006 has shown this improving effect is related to the isobaric lines divergence in the T-s diagram, and is independent of the existence of intermediate heat exchange.

It was observed the 4-wheel machine has a lower impact on its coefficient of performance than 3-wheel's when operating on cruise and on ground. According to DeFrancesco 1993, it occurs basically due to the recovery of the condensation heat as useful power in the second turbine. In reality, analyzing the results from cruise operation, one's concluded this occurs even if there is not water condensation inside the condenser. Probably more important is the condenser low working pressure, which dictates the temperature levels on its cold side.

The presence of a condenser is justified only when the external air moisture levels are significant. For cruise operation, where the moisture can be neglected, the condenser plays against the system efficiency. According to Warner, 1992, the 4-wheel machine can have a valve to bypass the first turbine and condenser, linking the secondary heat exchanger directly to the second turbine, which minimizes this negative effect.

Analyzing the ground operational cases it is observed the thermodynamic advantages of the 4-wheel in relation to the 3-wheel machine are maximized when considering critical cooling conditions for hot days.

The methodology of analysis of thermodynamic cycles developed and presented herein helps in the design of aeronautical air cycle machines, mainly due to the following reasons: makes the thermodynamic processes visualization easier through T-s diagrams; assist the understanding of the thermodynamic cycles under analysis; improve the trade-off study of each configuration and helps in understanding the contribution of each component in the system and leads to a more robust / adequate configuration definition process.

7. REFERENCES

[1] Andrade, R. A.; Zaparoli, E. L., 2004, "A Thermodynamic Study Of An Aircraft Air-Conditioning Air Cycle Machine", CONEM 2004, Congresso Brasileiro de Engenharia Mecância, Belem PA.

[2] ASHRAE Handbook, 1997, "Fundamentals Volume", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

[3] ASHRAE Handbook, 1999, "Applications Handbook", chapter 9 "Aircraf", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

[4] Conceição, S. T., 2006, "Estudo Termodinâmico de Máquinas de Ar Condicionado Ciclo a Ar de Aplicação Aeronáutica". Master Degree Thesis – Instituto Tecnológico de Aeronáutica, São José dos Campos.

[5] DeFrancesco, G. L., 1993, "Condensing Cycle Air Conditioning System", 23rd International Conference on Environmental Systems, Colorado Springs, Colorado, July 12th-15th .

[6] EES, 2004, "Engineering Equation Solver Manual for Microsoft Windows Operating Systems", F-Chart Software, http://www.fchart.com/

[7] Gigiel, A. J., Granryd, E., et al, 2001, "Guideline for the Application and Design of Air Cycle Systems for Heating, Ventilationg, and Air Conditoining in Buildings", Editado por Marcel J. E. Vershoor, Department of Refrigeration and Heat Pump Technology.

[8] Gigiel, A. J., Granryd, E., et al, 2001, "Guideline for the Application and Design of Air Cycle Systems for Heating, Ventilationg, and Air Conditoining in Buildings", edited by Marcel J. E. Vershoor, Department of Refrigeration and Heat Pump Technology.

[9] Moran, M. J., Shapiro, H. N., 1996, "Fundamentals of Engineering Thermodynamics", John Wiley & Sons, Inc.

[10] Rannenberg, G. C., 1969, "Simple/bootstrap cooling system for a new transport airplane". United Aircraft Corp., Hamilton Standard Div., Windsor Locks.

[11] Society of Automotive Engineers. ARP147 Rev. E, 2001, "Environmental Control Systems Terminology", Aerospace Recommended Practice, SAE InternationalPublicações SAE, Warrrendale PA.

[12] Society of Automotive Engineers. SAE AIR 1168/3, 2004, "Aerothermodynamic Systems Engineering and Design", SAE International.

[13] Society of Automotive Engineers. SAE AIR1168/6, 2004, "Characteristics of Equipment Components,Equipment Cooling System Design, and Temperature Control System Design", SAE International.

[14] Stoecker, W. F., 1989, "Design of Thermal Systems", McGraw-Hill Inc.

[15] Warner, J. L., 1992, "Environmental Control System Condensing Cycle", Patente USA N° 5,086,622, United Technologies Corporation, Hartford, Conn.