DYNAMIC MODELING OF AN AERONAUTICAL AIR CONDITIONING SIMPLE AIR CYCLE MACHINE

Eng. Dayvis Dias da Silva, dayvis.silva@embraer.com.br
Prof. Dr. Edson Luiz Zaparoli, zaparoli@ita.br
ITA - Instituto Tecnológico de Aeronáutica, Praça Marechal Eduardo Gomes, n° 50 - CEP 12228-900, São José dos Campos, SP.

MsC. Wallace H. L. Turcio, wturcio@embraer.com.br
EMBRAER - Empresa Brasileira de Aeronáutica, Av. Brigadeiro Faria Lima, n° 2170 – CEP 12227-901, São José dos Campos, SP.

Abstract. *This work presents a methodology applied to modeling and simulation of an aeronautical air conditioning simple Air Cycle Machine (ACM). Using Matlab/Simulink tools, an ACM was modeled and the model was evaluated comparing the results with available experimental data.The model is able to represent the ACM operating in open and closed control loop (with ACM outlet air temperature control).For the model construction, the ACM was separated in its several components and specific models were built for each one (turbine, fan, heat exchanger, temperature control valve and controller). The components models were integrated, in order to represent the ACM and simulations were run exposing the model to several inlet pressure and temperature conditions.During simulations the air was taken as dry, therefore it was not modeled the ACM's water separator component.*

Keywords*: Air Conditioning, System Modeling, Dynamic Simulation.*

1. INTRODUCTION

During system design an engineer shall consider the several aspects and variables that can affect the system performance and behavior. Such aspects can be of thermal, electric, pneumatic, magnetic and others natures. Looking for better results and more efficient products, it is clear the growing necessity of modeling and analysis of systems making use of computers programs – Karnopp *et al.* (2000).

For an aeronautical environment control system design, this system can be represented and analyzed with the aid of dynamic models able to represent the pneumatic system, the air conditioning machine and the aircraft cabin, all them with considerable complexity and with equal importance and contribution to the design process. An improvement in the Air Cycle Machine (ACM) model representation is important for aircraft manufacturer, since more accurate simulation results allow system engineers to act in advance, adjusting important system parameters and control laws, leading to less flight tests and reducing the system development cost.

There are few publications about ACM modeling and generally the available literature is dedicated to steady state modeling and analysis. Conceição (2006) is an example of a good contribution in this domain, who developed a steady state model for an aeronautical air conditioning system aiming the understanding of thermodynamic advantages and disadvantages for different air cycle machines configurations by permitting the thermodynamic processes visualization through T-s diagrams. In the dynamic domain, a simplified linear dynamic model obtained from an ACM manufacturer was used by Turcio (2003) when studying and comparing strategies for controlling passenger aircraft air conditioning systems.

Helping to reduce this lack of publications, this work proposes a dynamic modeling methodology and presents the simulation results for an aeronautical air conditioning air cycle machine dynamic model. This work is a synthesis of the study performed by Silva (2010).

2. AIR CONDITIONING SYSTEM

An aeronautical air conditioning system is composed by a set of equipments (heat exchanger, fan, compressor, valves, etc) that supply and distribute fresh air into cabin and compartments for ventilation, pressurization and temperature control. An introduction to the fundamentals and functions of an aircraft air conditioning system as well as a brief description of its main components is presented by Ebeling (1968).

Two types of air conditioning machine are used for aircraft applications: vapor cycle machine and air cycle machine. Usually air cycle machines are chosen due to their better reliability and lower weight - SAE (1990). This type of machine operates in an open-loop Brayton refrigeration cycle similar to the Brayton power cycle of gas turbine engines. High-pressure air extracted from the gas turbine engine compressor is cooled in a heat exchanger using ambient air as the heat sink, and then this air is refrigerated by expansion in a turbine. The power exerted on the turbine shaft is absorbed by an ambient air fan and/or a compressor. The air cycle machine possible configurations include simple cycle (turbine – fan), bootstrap (turbine – compressor), three – wheel (turbine – compressor – fan) and four – wheel (turbine – turbine – compressor – fan). There are publications describing briefly how an aeronautical air conditioning system operates covering different air cycle machine configurations, for example Hunt and Space (1995) discuss some engineering and operational aspects of a Boeing 767 aircraft three – wheel air cycle machine and DeFrancesco (1993) describes the advantages of a four – wheel air cycle machine used in the Boeing 777 over a three – wheel unit. In this work the simple cycle, shown in Fig. 1, was taken for study.

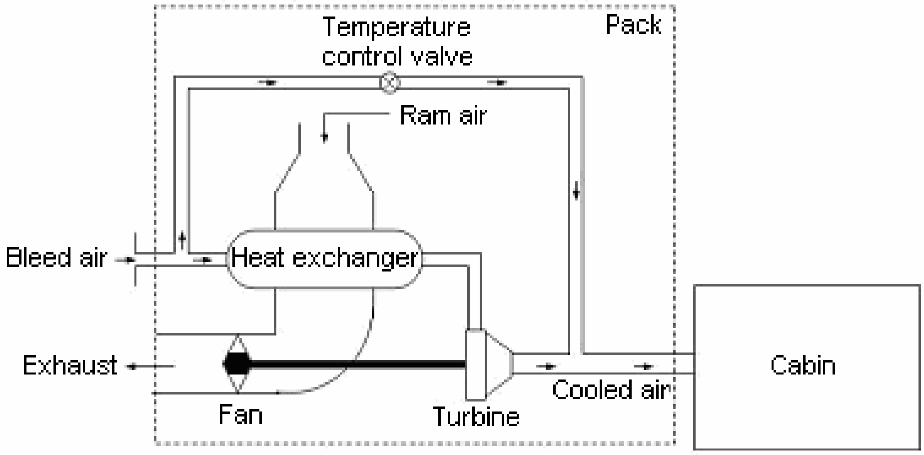

Figure 1. Air conditioning simple air cycle machine

In a simple cycle ACM, high pressure and temperature air extracted from the engine flows throughout a heat exchanger and part of its thermal energy is rejected to the cold air stream obtained from the ambient. This cold air flows through the heat exchanger due to pressure energy increased by the fan either when the aircraft is on ground or flying in low speed. For high speed flights the cold air is forced in the system due to its impact pressure. The fan is installed in the end of the line to avoid air temperature increase before flowing through the heat exchanger. After leaving the heat exchanger the pressurized air expands in the turbine. During the expansion, thermal and pressure energy is converted in turbine shaft power. Part of this power is absorbed by the fan and part is dissipated as friction energy in the bearings. The ACM outlet air temperature control is guaranteed with a by pass valve, which deviate part of the pressurized air before it reaches the heat exchanger. This hot air is then mixed with the air from the turbine outlet. After that, the air flows through a low pressure water separator, where 60% to 80% of entrained moisture is removed.

3. MATHEMATICAL MODEL DESCRIPTION

A mathematical model is a representation of real system using equations that describes its physical process – Zaparoli and Andrade (2006). In the air conditioning simple air cycle machine model proposed in this work, these equations represent the relationship between air flow rate, pressure differences and temperature. Besides physical equations, it was also used performance maps for some ACM components. For the model construction, the ACM was separated in its several components and specific models were built for each one (turbine, fan, heat exchanger, temperature control valve and controller). The air was taken as dry and the water separator component was not modeled. The components models were integrated in order to represent the ACM behavior. All equations are presented in SI (metric) units.

3.1. Ducts

The ducts were modeled as a resistor – capacitor arrangement. The resistor represents the duct pressure loss effect and the capacitor represents the mass and energy accumulation in the duct.

The resistance effect was modeled considering isentropic orifice equations – Van Wylen *et al* (1998). The Mach number M in the orifice throat is obtained by the Eq. (1), where P is the pressure and k is the ratio between specific heat at constant pressure and specific heat for constant volume. The subscripts “1” and “2” refer to orifice inlet and outlet respectively.

$$M = \left\{ \frac{2}{k-1} \left[\left(\frac{P_1}{P_2} \right)^{\frac{(k-1)}{k}} - 1 \right] \right\}^{\frac{1}{2}} \tag{1}$$

The temperature T_G of the flow in the orifice throat is calculated according to Eq. (2), where T_1 is the flow temperature in the orifice inlet.

$$T_G = \frac{T_1}{1 + \frac{k-1}{2} M^2} \tag{2}$$

The pressure P_G in the orifice throat is given by Eq. (3).

$$P_G = \frac{P_1}{\left(1 + \frac{k-1}{2} M^2\right)^{\frac{k}{(k-1)}}} \tag{3}$$

The mass flow rate $\dot{m}$ in the orifice throat can be obtained by Eq. (4), where A is the orifice throat area and R is the gas (air) constant.

$$\dot{m} = P_G A M \sqrt{\frac{k}{R T_G}} \tag{4}$$

From Eq. (4) and using Eq. (5) it is also possible to obtain the "enthalpic flow" $\dot{J}$ through the orifice (duct):

$$\dot{J} = \dot{m} \frac{kR}{(k-1)} T_1 \tag{5}$$

The capacitive effect in the duct is characterized by its mass and energy balance – Van Wylen *et al* (1998). The duct (volume control) mass variation rate $\frac{dm_{v.c.}}{dt}$ can be different from zero if the duct inlet mass flow rate is different from the duct outlet mass flow rate. This rate is calculated according to Eq. (6), where the subscripts "e" and "s" refer to the duct inlet and outlet respectively.

$$\frac{dm_{v.c.}}{dt} = \sum \dot{m}_e - \sum \dot{m}_S \tag{6}$$

At each instant of time the mass m accumulated in the duct is equal to the integral of Eq. (6), as in Eq. (7):

$$m = \int \frac{dm_{v.c.}}{dt} dt \tag{7}$$

Considering a similar approach, the energy E accumulated in the duct is obtained in Eq. (8). The thermodynamics first law is applied to the duct volume, where $\dot{Q}_{v.c.}$ is the heat flux supplied to the flow, if any.

$$\frac{dE_{v.c.}}{dt} = \dot{Q}_{v.c.} + \sum \dot{J}_e - \sum \dot{J}_S \tag{8}$$

The accumulated energy for each instant of time is given by Eq. (9):

$$E = \int \frac{dE_{v.c.}}{dt} dt \tag{9}$$

As the process occurs at constant volume, the temperature T inside the duct can be obtained by Eq. (10), where c_v is the air specific heat for constant volume.

$$T = \frac{E}{mc_v} \tag{10}$$

After obtaining the air mass accumulation in the duct and its temperature, it is possible to calculate the duct internal pressure with the equation for a perfect gas. For a duct with internal volume Vol, Eq. (11) is used to provide the pressure.

$$P = \frac{m}{Vol} RT \tag{11}$$

3.2. Turbine

Turbine is one of the categories in which turbomachinery are divided. According to Shepherd (1965), turbomachinery is a term that includes equipments that produce pressure energy, such as pumps and compressors, and equipments that produce power, as the turbines, in which the primary elements are rotating.

In the developed model the turbine is of radial type. Figure 2 and Fig. 3 are part of the required data for turbine modeling. In the model the turbine behaves as a mass flow rate restriction in which this mass flow rate depends on the turbine inlet and outlet pressure, turbine inlet air temperature and the turbine rotational speed. The numeric values of the turbine performance maps have been removed due to intellectual property rights of the manufacturer.

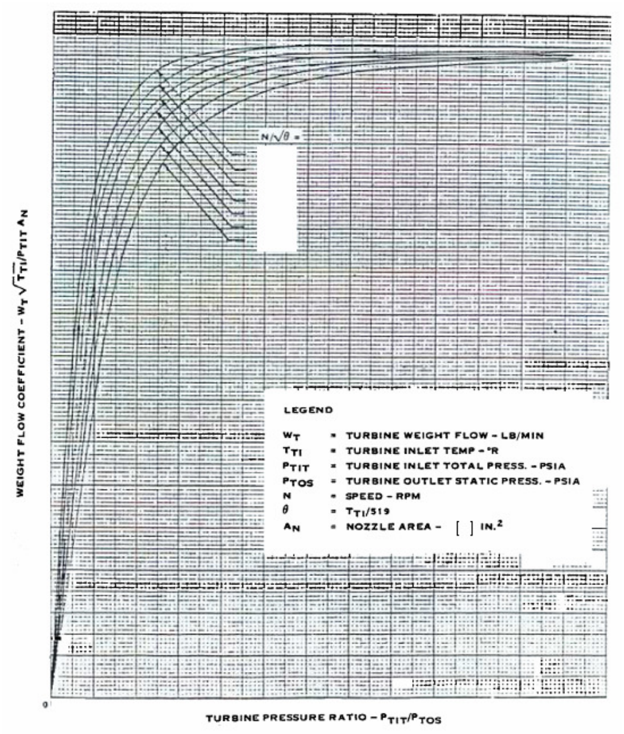

Figure 2. Turbine pressure ratio versus turbine flow coefficient – Turcio (2010)

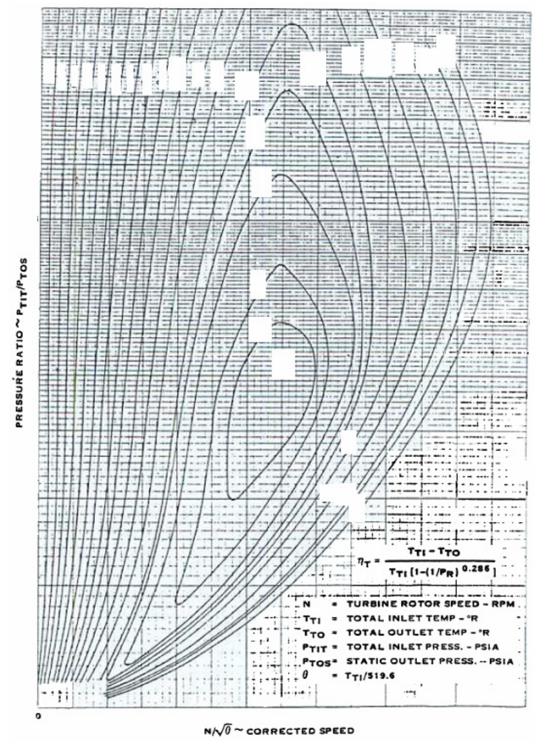

Figure 3. Turbine corrected speed versus turbine pressure ratio – Turcio (2010)

Using the Fig. 2 it is possible to obtain the turbine mass flow rate from the Eq. (12) for a given turbine inlet pressure P_{TIT}, turbine outlet pressure P_{TOS}, turbine air inlet temperature T_{TI} and turbine rotational speed N. In the Eq. (12) W_{coef} and A_N are the turbine flow coefficient and turbine nozzle area respectively.

$$W_T = \frac{11.72 W_{coef} P_{TIT} A_N}{6894.75\sqrt{1.8 T_{TI}}} \tag{12}$$

On the map of Fig. 3, given the turbine inlet and outlet pressure, turbine rotational speed and turbine air inlet temperature, the turbine efficiency η_T can be obtained. The turbine outlet temperature T_{TO} is calculated by Eq. (13), where P_R is the ratio between turbine inlet and outlet pressure.

$$T_{TO} = T_{TI} - \eta_T T_{TI}\left[1 - \left(\frac{1}{P_R}\right)^{0.286}\right] \tag{13}$$

The available power on the turbine shaft is given by Eq. (14), where c_p is the air specific heat for constant pressure.

$$Pot_T = W_T c_p \left(T_{TI} - T_{TO}\right)\eta_T \tag{14}$$

The turbine rotational speed is obtained from Newton's second law. For a given resultant torque τ_R, being I the moment of inertia of the turbine – fan assembly, b the viscous bearings friction coefficient and w the turbine angular speed, the rotational speed in RPM is given by Eq. (15):

$$N = \frac{\int \frac{\tau_R - bw}{I}}{2\pi} 60 \tag{15}$$

3.3. Heat exchanger

The heat transfer process between two fluids that are at different temperatures and are separated by a solid wall occurs in many engineering applications – Incropera and Dewitt (1998). In the ACM a compact heat exchanger is used, since air is the working fluid and the refrigerant, it has a low coefficient of heat transfer and greater exchange area is needed. The heat exchanger is modeled as a component that restricts air mass flow rate due to its pressure loss and also promotes heat transfer between hot and cold side flows. Figure 4 and Fig. 5 show the maps that are being used to estimate mass flow rate and heat exchange in this component.

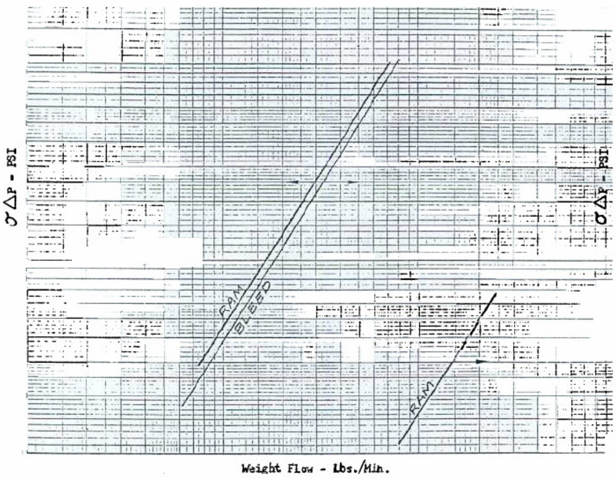

Figure 4. Heat exchanger pressure loss map – Turcio (2010)

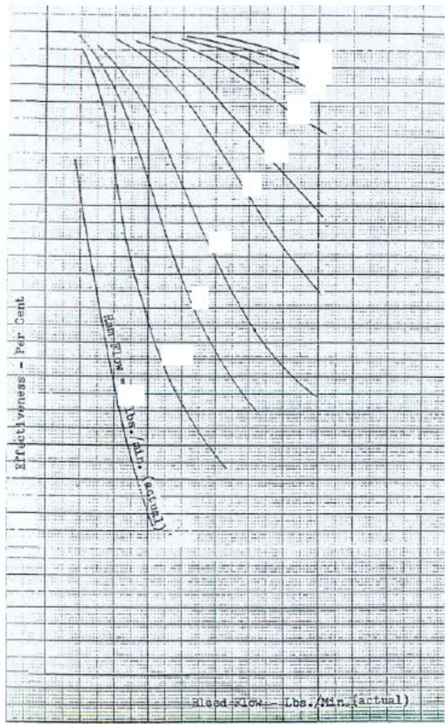

Figure 5. Heat exchanger effectiveness map – Turcio (2010)

The numeric values were also removed from Fig. 4 and Fig. 5 respecting intellectual property rights of the manufacturer. By the map of Fig. 4, given the inlet and outlet heat exchanger values of pressure and temperature it is possible to obtain the mass flow rate through it. The effectiveness ε is obtained from the map of Fig. 5. At steady state condition the heat exchanger hot and cold side outlet air temperature are calculated according to Eq. (16) and Eq. (17) respectively. The subscript "q" refers to hot side and the subscript "f" refers to cold side. The subscript "min" refers to the lower value of mass flow rate between hot and cold side.

$$T_{q,s} = T_{q,e} - \frac{\varepsilon \dot{m}_{\min}\left(T_{q,e} - T_{f,e}\right)}{\dot{m}_q} \tag{16}$$

$$T_{f,s} = T_{f,e} + \frac{\varepsilon \dot{m}_{\min}\left(T_{q,e} - T_{f,e}\right)}{\dot{m}_f} \tag{17}$$

The thermal inertia φ of the heat exchanger is represented by a first-order differential equation as per Eq. (18). The reference temperature, T_{s_ref}, is the temperature at steady state condition obtained from Eq. (16) or Eq. (17). T_s is the air temperature at the heat exchanger outlet at each instant of time, including transient conditions. At steady state condition T_{s_ref} and T_s are equal.

$$\varphi \dot{T}_s + T_s - T_{s\_ref} = 0 \tag{18}$$

3.4. Fan

A fan is a device that converts mechanical power of rotation applied to its shaft in pressure energy into the gas, usually air. For the fan model development, it was used a map which relates the mass flow rate and the pressure energy increase caused by the fan. In order to calculate the fan shaft power consumption it is necessary to obtain its efficiency. Figure 6 shows the fan performance map with the information discussed above.

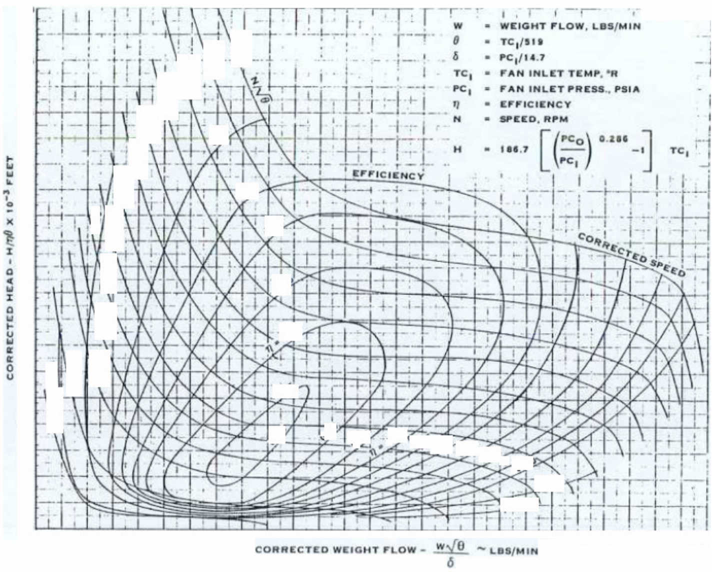

Figure 6. Fan performance map – Turcio (2010)

With the map of Fig. 6 and Eq. (19) the pressure P_{vo} in the fan outlet can be obtained. P_{vi}, T_{vi} and H_v are the fan inlet air pressure, fan inlet air temperature and the head (in meters of air) caused by the fan in the air respectively.

$$P_{vo} = \left[\left(\frac{3.28 H_v}{336.06 T_{vi}}\right) + 1\right]^{\frac{1}{0.286}} P_{vi} \tag{19}$$

The air temperature T_{vo} in the fan outlet is obtained from the Eq. (20).

$$T_{vo} = \left(\frac{P_{vo}}{P_{vi}} \right)^{\frac{k-1}{k}} T_{vi} \tag{20}$$

The fan shaft power Pot_v is calculated according to Eq. (21), where ρ is the air density, η_v is the fan efficiency obtained from Fig. 6 and ΔP_v is the air pressure variation considering fan inlet and outlet.

$$Pot_v = \frac{\dot{m} \Delta P_v}{\rho \eta_v} \tag{21}$$

The torque τ_v in the fan shaft is given by Eq. (22), where the fan angular speed w is identical to turbine speed.

$$\tau_v = \frac{Pot_v}{w} \tag{22}$$

3.5. Temperature control valve

A temperature control valve is used to regulate ACM outlet air temperature. This valve was modeled as a variable area orifice. The area value depends on the valve butterfly angle position in each instant of time. It was adopted a normally closed valve for modeling, with butterfly angle position proportional to control current. For the considered valve, zero mA of current corresponds to zero degree of butterfly opening angle and fifty (50) mA of current corresponds to ninety (90) degrees. The relationship between the valve area and its butterfly opening angle is illustrated in Fig. 7.

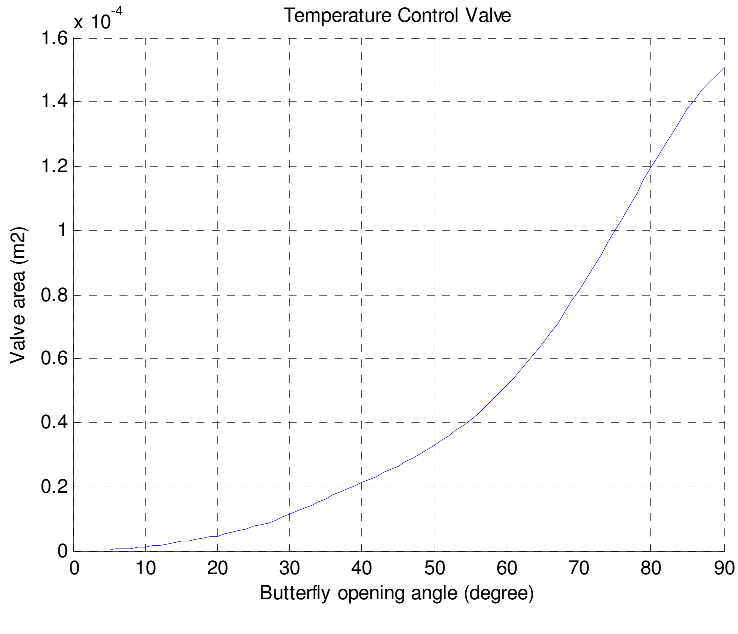

Figure 7. Valve butterfly angle opening versus valve area

The valve dynamic response is represented by a first-order differential equation as per Eq. (23), where A_θ is the butterfly opening angle at each instant of time, A_{θ_ref} is the reference angle and λ is the valve time constant.

$$\lambda \dot{A}_\theta + A_\theta - A_{\theta\_ref} = 0 \tag{23}$$

3.6. Controller

A temperature control law was designed to guarantee that the ACM outlet air temperature follows the reference temperature (set point temperature). Based on the difference between outlet air temperature and reference temperature, the closed control loop acts reducing this error – Ogata (2003). In this work it was considered a PID controller. Figure 8 shows the configuration of the PID controller adopted.

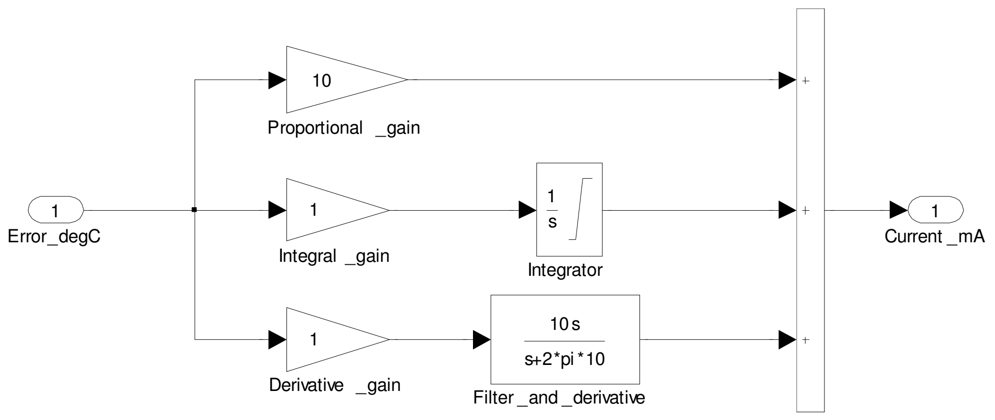

Figure 8. PID controller for ACM air outlet temperature control

4. METHODOLOGY AND SIMULATIONS

Making use of the commercial software Matlab/Simulink, the developed equations for each component were implemented and solved. The integration of the models of the components represents the whole air conditioning machine. The model is able to represent the ACM operating in open and closed control loop of its outlet air temperature.

Herein is shown the results for closed control loop simulation. Available experimental data were used to simulate and check the model representativeness. The experimental tests were not designed for model validation and it was not possible to obtain enough information for complete and formal validation. Tests were run with the aircraft on ground, parked. The main variables are engine bleed air pressure and temperatures. Figure 9 and Fig. 10 show these values, which were used as model input. The results are shown from Fig. 11 to Fig. 14.

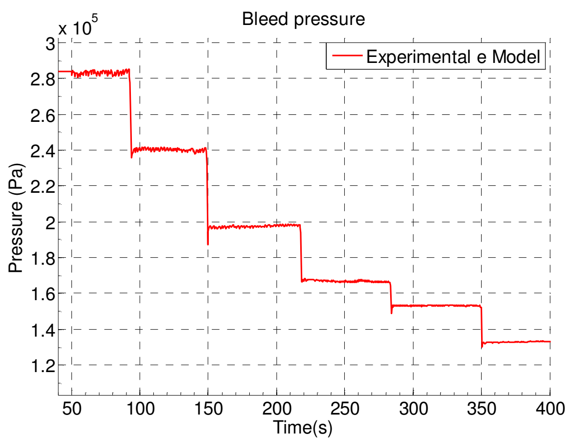

Figure 9. Bleed air pressure

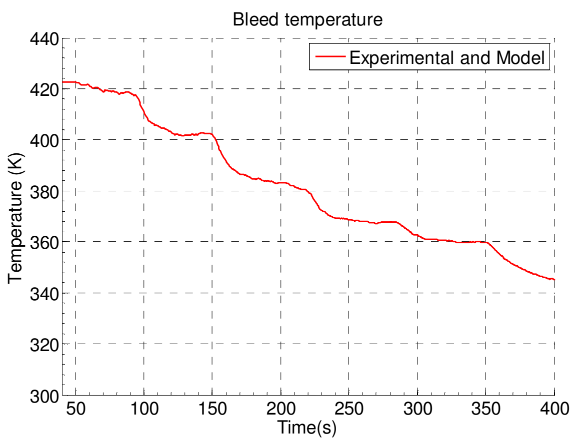

Figure 10. Bleed air temperature

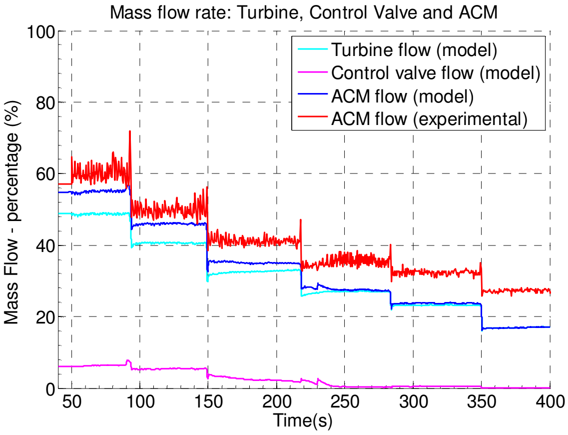

Figure 11. ACM air mass flow rate

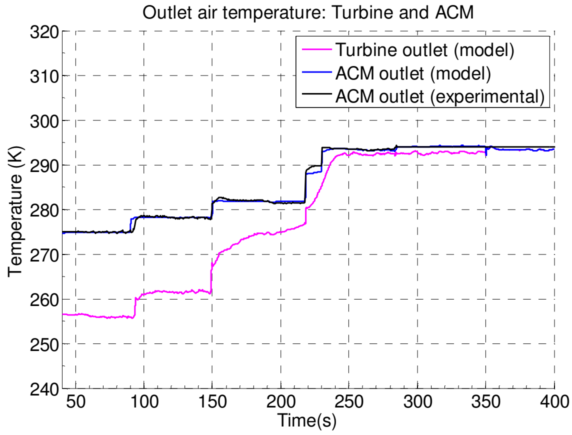

Figure 12. ACM outlet controlled temperature

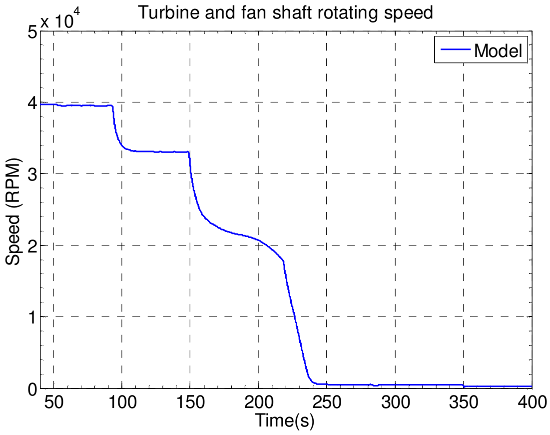

Figure 13. Turbine and fan rotating speed

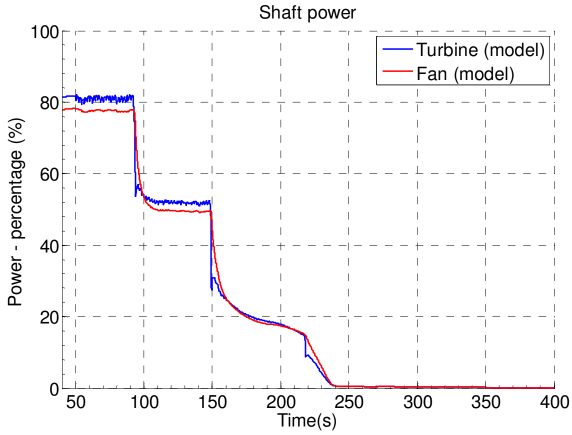

Figure 14. Turbine and fan shaft power

In order to protect the manufacturer intellectual property rights, in Fig. 11 and Fig. 14 the SI units [kg/s] and [W] were substituted by percentage representations based on a reference and arbitrary number.

5. RESULTS AND CONCLUSIONS

The simulation results show that the model has a good matching with experimental data when the ACM air mass flow rate and ACM outlet temperature are considered, as per Fig. 11 and Fig. 12. Besides matching quality, an important aspect is the fact that the model is representing well the behavior of the ACM and its components, for example the model simulation results presented in Fig. 11, Fig. 12 and Fig. 13 show that reducing the inlet pressure and temperature leads to a reduction in the ACM air mass flow rate and in the turbine-fan speed and leads also to an increase of the turbine outlet air temperature, showing that the pressure effect overcomes the temperature effect. In addition, as showed in Fig. 14, it is also verified that the ACM stabilizes (that is, shaft speed assumes a constant value) when turbine and fan shaft power and bearing power dissipation are in balance. When unbalance occurs the system will either accelerate if power generated by turbine is higher than power consumption (fan and bearings) or decelerate if turbine power is lower than power consumption, until a new balance state is reached. The results for speed variations due to operating conditions changes are within the expected values.

The proposed methodology for ACM modeling using components performance maps and energy and mass conservation laws is appropriate to represent an aeronautical air conditioning simple air cycle machine. The proposed ACM model can improve the representation of the whole air management system, contributing to a better understanding of ACM steady and unsteady state operations. This allows the anticipation of problems by system design engineering what implies in reduced system development costs.

6. REFERENCES

Conceição, S.T., 2006, "Estudo Termodinâmico de Máquinas de Ar Condicionado Ciclo a Ar de Aplicação Aeronáutica", Dissertação de Mestrado – Instituto Tecnológico de Aeronáutica, São José dos Campos, Brasil.

DeFrancesco, G. L., 1993, "Condensing cycle air conditioning system", In:SAE 23rd International Conference on Environmental Systems, Colorado, USA.

Ebeling, A., 1968, "Fundamentals of aircraft environmental control", Hyden Book Company.

Hunt, E.H. and Space, D.R., 1995, "Commercial Airliner Environmental Control System", In: Aerospace Medical Association Annual Meeting, California, USA.

Incropera, F.P. and Dewitt D.P., 1998, "Fundamentos de Transferência de Calor e de Massa", LTC Editora S.A., Rio de Janeiro, Brasil.

Karnopp, D.C., Margolis, D.L. and Rosenberg, R.C., 2000, "System Dynamics: Modeling and Simulation of Mechatronic Systems", Wiley-Interscience, USA, 503 p.

Ogata, K., 2003, "Engenharia de Controle Moderno", Prentice Hall, São Paulo, Brasil.

SAE AIR 1168/3: Aerospace Information Report - Aerothermodynamic Systems Engineering and Design. Warrandale: SAE, 1990.

Shepherd, D.G., 1965, "Principles of Turbomachinery", The Macmillan Company, New York, USA.

Silva, D.D., 2010, "Modelagem Dinâmica de uma Máquina de Ar Condicionado Aeronáutica Ciclo Simples", Dissertação de Mestrado – Instituto Tecnológico de Aeronáutica, São José dos Campos, Brasil.

Turcio, W.H.L., 2003, "Estratégia de Controle para Sistema de Condicionamento de Ar em Aeronaves de Passageiros", Dissertação de Mestrado – Escola Politécnica da Universidade de São Paulo, São Paulo, Brasil.

Turcio, W.H.L. Curvas de desempenho do Pack. [Mensagem Pessoal]. Mensagem recebida por: <dayvis.silva@embraer.com.br> em 13 de Maio de 2010.

Van Wylen, G.J., Sonntag, R.E., Borgnakke, C., 1998, "Fundamentos da Termodinâmica", Editora Edgar Blucher Ltda., São Paulo, Brasil.

Zaparoli, E.L. and Andrade, C.R., 2006, "ME-701 Air Conditioning", Apostila do curso de Sistemas Ambientais do Programa de Especialização em Engenharia, ITA-Embraer, São José dos Campos, Brasil.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.