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First Principle Model of an Electric ECS Pack

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Abstract. Fully electric environmental control systems for aircraft are designed but currently rarely implemented. Simulation models of such systems in the early design phase have shown a lot of challenges: high electric power demand, complex architecture, difficult design of control strategies and difficult sizing.
In order to explore what performance figures are physically possible, a first principle model has been developed on the basis of purely thermodynamic considerations. Similar to a Carnot thermal engine, which tells what efficiency is reachable, the first principle model of an ECS can serve as a reference system which tells the minimum amount of electric power needed to fulfil the thermal requirements. Different to a specific technical design, this allows for more general conclusions.

1. Motivation

The project ENERGIZE deals with the Energy Management System for a More Electric Aircraft (MEA) with conventional propulsion but with an electric environmental control system (eECS). It is focused on the feedback cycle of electric power and thermal flows. Electric devices produce heat, which partially must be removed by the ECS, which needs more electric energy and so forth. In order to study these interdependencies, a comprehensive simulation model has been developed.

In such a system, the electric ECS, which does not receive bleed air from the engines, needs most of the generated electric power. Therefore, this component must be modelled in a precise and careful way. In particular answers to the following questions are important:

- How much power is required depending on the flight phase and the environmental conditions?
- How does the power demand change when the conditions change?
- What is an appropriate size for the ECS components (heat exchangers, compressors, …)?

However, a real system resp. real system data is not available to us. Setting up simulation for a potential technical design proved to be difficult [1] and revealed a number of subsequent challenges such as suitable control design, handling of sizing limitations and sufficient simulation performance. Even when all these challenges are met, there is clearly a great value in having a simpler more general model available.

Therefore, the idea came up to develop a First Principle Model, which is based on the essential physics and neglects the problems of a technical implementation. It would not show the same, but hopefully a comparable behavior.

2. Task and Requirements

The ECS receives the air from the environment of the aircraft and forwards it to the mixer unit, in which it is blended with recirculated air from the cabin. The inflow air with pressure p_{in}, temperature T_{in} and humidity mass ratio μ_{in} [g water / kg dry air] is transformed to outflow air with pressure p_{out}, temperature T_{out} and humidity mass ratio μ_{out}.

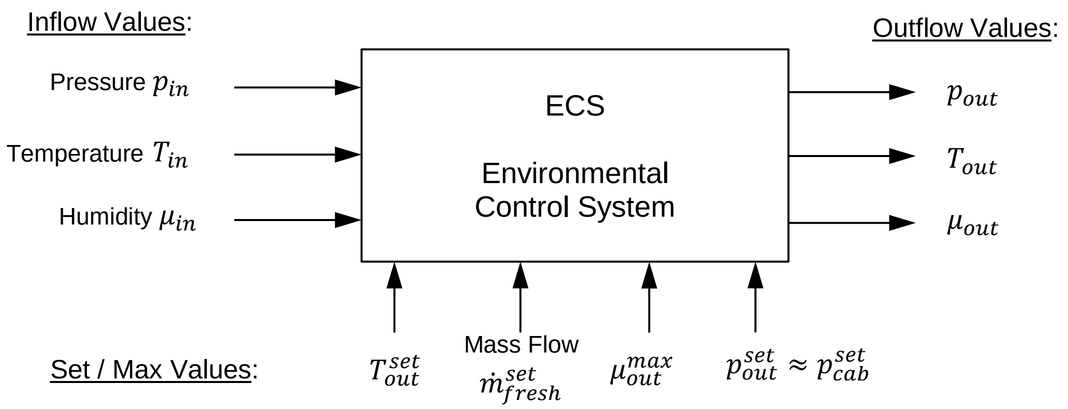

Figure 1. Input-Output-Scheme of an ECS

The ECS is controlled by the set values for the required fresh air flow $\dot{m}_{fresh}^{set}$, the discharge temperature T_{out}^{set} and by a maximum value for the humidity mass ratio μ_{out}^{max}. The outflow pressure must be slightly above the cabin pressure p_{cab}^{set}, which is controlled by the outflow valve. Hence the outflow pressure is quasi a given input variable. Therefore, we add p_{out}^{set} to the given set variables corresponding to the set cabin pressure.

3. Difficulties

Modelling an ECS as described encounters a lot of difficulties. The three most important are:

- wide range of operating points
- complex hardware architecture
- complex control strategy

The wide range of operating points results from a wide span of environmental conditions as well as a wide span of discharge conditions. Ground conditions may vary between -20°C and dry air on cold days to 38°C and humid air on hot days. Conditions on cruise level with an average outside temperature of -55°C, a low pressure of 0.25 bar and very dry air represent a further extreme environment. On the discharge side a temperature between -20 and +20°C and a relatively low humidity is needed.

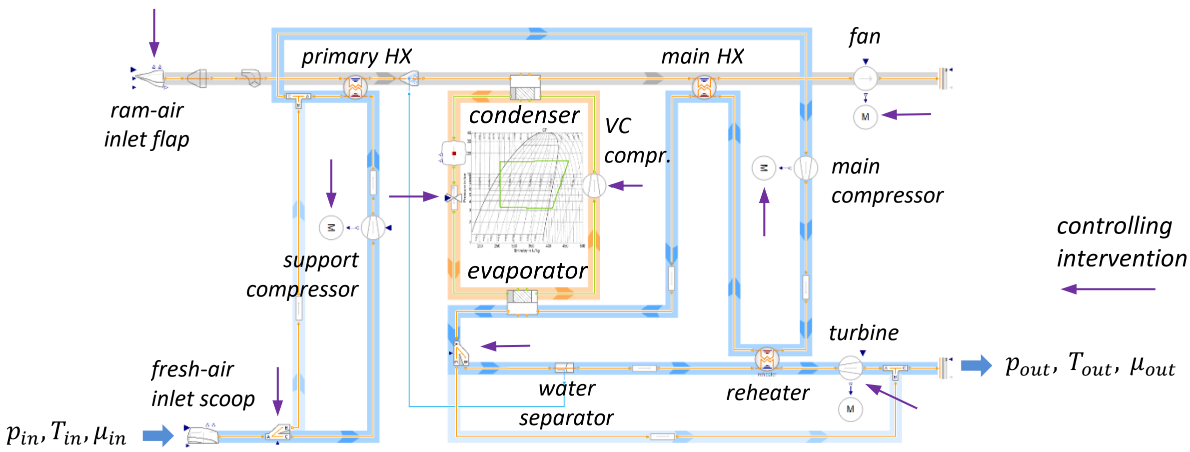

Figure 2. Example of a complex electric ECS architecture with embedded vapor cycle

As illustrated in figure 2, the hardware architecture is a labyrinth of pipes, valves, nozzles, flaps, heat exchangers and compressors and may even contain a vapor cycle for enhanced cooling.

Around 10 actuators are required and changing each of them has a significant effect on all outflow variables, and an embedded vapor cycle can make it even more difficult to control the system. Since the operating points are far from each other, a startup strategy and transfers between operating points are difficult tasks [1].

4. Features of the First Principle Model

The First Principle Model circumvents the described difficulties. It is restricted to the necessary physical processes and covers them in a minimalistic way. Control devices are not needed, since a backpropagation technique is applied. The modelled subprocesses do not have the technical limitations of real devices (e.g. compressors), which makes them work even under extreme conditions. Inertia effects caused by heat capacities are not considered as well.

This approach is supposed to be as simple as possible and promises to achieve a basic understanding of the system. As a simulation model it shall produce data in all operating points and help to study the required behaviour in all flight phases.

The following chapter describes how the thermodynamic process is modelled.

5. Modelling the Thermodynamic Process

5.1. Enthalpy Balance

For those who are not familiar with thermodynamic processes the hint is given that it is not sufficient to calculate the power demand from the enthalpy balance, which is the difference

$$\Delta\dot{H} = \dot{H}_{out} - \dot{H}_{in} \tag{1}$$

of the enthalpy outflow

$$\dot{H}_{out} = \left[c_p^{da}\, T_{out} + \mu_{out}\, c_p^{vap}\, T_{out}\right] \dot{m}_{fresh}^{da,set} \tag{2}$$

and the enthalpy inflow

$$\dot{H}_{in} = \left[c_p^{da}\, T_{in} + \mu_{in}\, c_p^{vap}\, T_{in}\right] \dot{m}_{fresh}^{da,set} \tag{3}$$

with the heat capacities of dry air c_p^{da} and water vapor c_p^{vap} and the the dry air mass flow $\dot{m}_{fresh}^{da,set}$.

The result is too far from a feasible physical implementation, since pressure and temperature cannot be changed independently. Particularly compression and expansion processes must be modelled explicitly.

In the following steps we will develop pressure-temperature diagrams (PT), which represent the required thermodynamic processes in an illustrative way. We will provide formulas, which enable to calculate the key points of the process explicitly.

5.2. Case Distinction

It turned out that four cases must be distinguished with the following conditions:

1) No Dehumidification required: $\mu_{in} \leq \mu_{out}^{max}$
   a) Cooling $T_{comp}(p_{out}^{set}) > T_{out}^{set}$
   b) Heating $T_{comp}(p_{out}^{set}) < T_{out}^{set}$
2) Dehumidification required: $\mu_{in} > \mu_{out}^{max}$
   a) Cooling without Reheating $T_{turb}^{in} \leq T_{dehum}$
   b) Cooling with Reheating $T_{turb}^{in} > T_{dehum}$

5.3. Case 1a: Cooling without Dehumidification

During cruise, on higher altitudes, this case is relevant. The pressure of the incoming air must be compressed to the required outlet pressure plus a small amount of pressure caused by losses in the ECS heat exchangers.

$$p_{comp} = p_{out}^{set} + \Delta p_{hex}\left(\dot{m}_{fresh}\right) \tag{4}$$

As the pressure rises by compression the temperature increases following an adiabatic resp. a polytropic characteristic. Since most often the compressor efficiency η_C is known, we calculate the temperature by a modified adiabatic characteristic with the adiabatic index of dry air $\kappa = 1.4$:

$$T_{comp} = T_{in} + T_{in} \cdot \frac{\left(\frac{p_{comp}}{p_{in}}\right)^{\frac{\kappa-1}{\kappa}} - 1}{\eta_C} \tag{5}$$

In this considered case the temperature of compressed air exceeds the required outlet temperature and therefore must be cooled. The whole thermodynamic process is shown in picture 3:

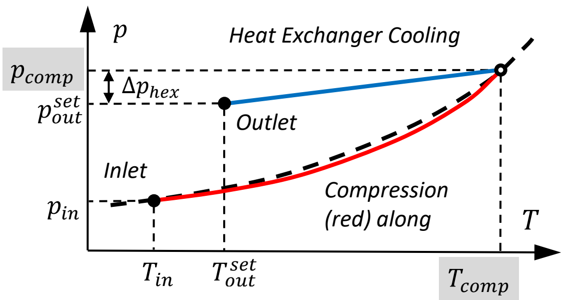

Figure 3. Compression and Cooling (Case 1a)

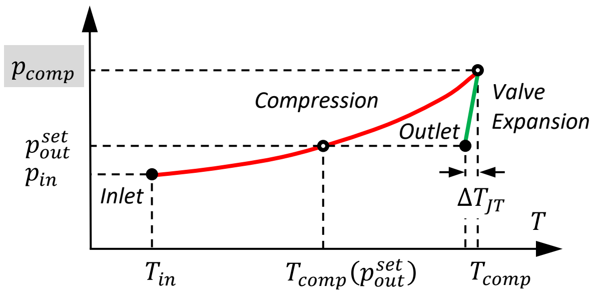

Figure 4. Compression and Expansion (Case 1b)

5.4. Case 1b: Heating without Dehumidification

This case may occur in winter on cold days, when the air is dry. As in the case before, the air is compressed, but the temperature does not reach the required outlet temperature. In order to reach this temperature a higher pressure is required:

$$p_{comp} = p_{in}\left[\eta_C\left(\frac{T_{comp}}{T_{in}} - 1\right) + 1\right]^{\frac{\kappa}{\kappa-1}} \tag{6}$$

Since this pressure is higher than the required outlet pressure, we add an expansion valve, which reduces the pressure by an isenthalpic process. Hereby the temperature changes slightly due to the Joule-Thomson effect with Joule-Thomson coefficient μ_{JT}.

$$\Delta T_{JT} = \mu_{JT}\left(p_{comp} - p_{out}^{set}\right) \tag{7}$$

Therefore, the air must be compressed to the temperature:

$$T_{comp} = T_{out}^{set} + \Delta T_{JT} \approx T_{out}^{set} \tag{8}$$

Since pressure and temperature cannot be calculated separately, at least one iteration cycle starting at $T_{comp} = T_{out}^{set}$ must be performed. The complete thermodynamic process is shown in figure 4.

5.5. Dehumidification

In order to avoid that water or ice condensates in ECS components, the relative humidity at the outlet (point of lowest temperature) must not exceed 100%.

$$\varphi_{out}^{max} < 100\% \tag{9}$$

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Since pressure and temperature at the outlet are given, we can determine what humidity mass ratio (vapor mass/dry air mass) must not be exceeded:

$$\mu_{out}^{max} = \frac{R_{da}}{R_{vap}} \cdot \frac{\varphi_{out}^{max}\, p_{vap}^{sat}(T_{out}^{set})}{p_{out}^{set} - \varphi_{out}^{max}\, p_{vap}^{sat}(T_{out}^{set})} \tag{10}$$

with the vapor saturation pressure p_{vap}^{sat} at the outlet temperature T_{out}^{set} and the specific gas constants for dry air R_{da} and water vapor R_{vap}.

We assume dehumidification is done by condensation. When humid air is cooled, the vapor saturation pressure and the ability to take water vapor decreases. However, the air must not be cooled below 0°C, since the components would ice up. Normally a dehumidification temperature of 4°C is used if the outlet temperature lies below.

$$T_{dehum} = \begin{cases} 4°C, & T_{out}^{set} \leq 4°C \\ T_{out}^{set}, & else \end{cases} \tag{11}$$

Under many conditions, dehumidification at the available pressure is not sufficient and a high-pressure dehumidification must be applied. With increasing pressure, the air can take less water vapor. The pressure required for dehumidification can be calculated by:

$$p_{dehum} = \left(\frac{R_{da}}{R_{vap}} \cdot \frac{1}{\mu_{out}^{max}} + 1\right) p_{vap}^{sat}(T_{dehum}) \tag{12}$$

Concerning the PT diagram, we must construct a path which starts at the inlet, ends with the outlet and passes the dehumidification point (p_{dehum}, T_{dehum}).

5.6. *Case 2a: Dehumidification and Cooling without Reheating*

High pressure dehumidification becomes necessary on ground or at low altitude, particularly when the air is humid.

In this case, compression to the dehumidification pressure leads to the much higher temperature

$$T_{comp} = T_{in} + T_{in} \cdot \frac{\left(\frac{p_{comp}}{p_{in}}\right)^{\frac{\kappa-1}{\kappa}} - 1}{\eta_C} \tag{13}$$

corresponding to a high power demand of the compressor. It is attempted to recuperate part of this power by using a turbine for the decompression. With the known turbine efficiency η_T, the outlet conditions and the dehumidification pressure we can calculate the turbine inflow temperature.

$$T_{turb}^{in} = \frac{T_{out}^{set}}{\eta_T\left[\left(\frac{p_{out}^{set}}{p_{dehum}}\right)^{\frac{\kappa-1}{\kappa}} + \frac{1}{\eta_T} - 1\right]} \tag{14}$$

The process, in which the turbine inflow temperature is below the dehumidification temperature, is shown in figure 5. In this case, the air is more dehumidified than required.

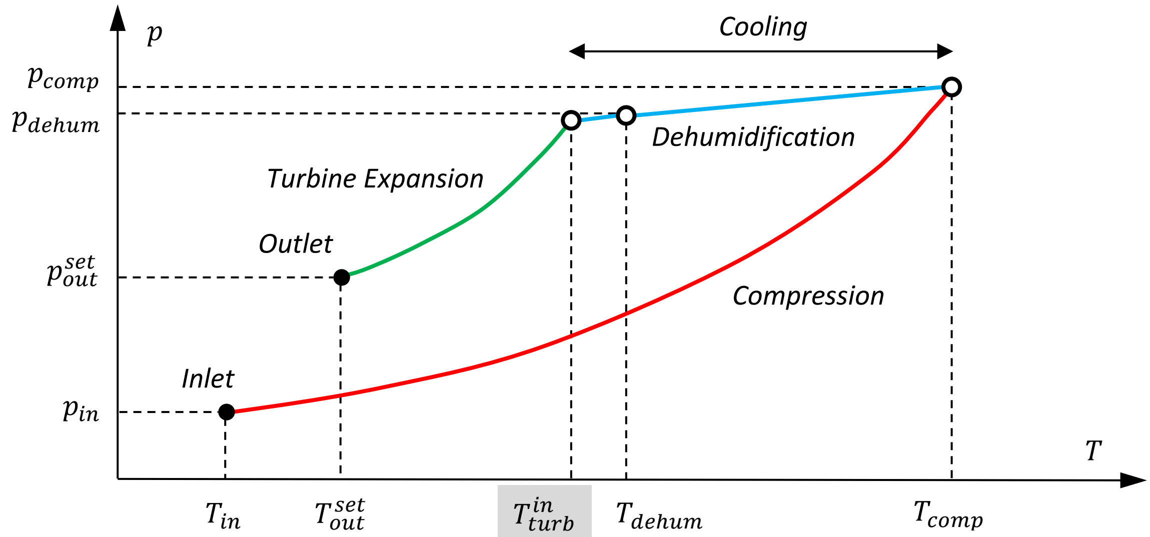

Figure 5. Turbine Expansion without Reheating (Case 2a)

5.7. Case 2b: Dehumidification and Cooling with Reheating

Under most conditions the turbine inflow temperature is higher than the dehumidification temperature. In this case the cooled air is reheated before it reaches the turbine. Thus, a great part of the power inherited in the step between dehumidification and outflow pressure can be recuperated.

The needed heat is taken from the first heat exchanger in the cooling chain. Since the enthalpy flow on the emitting side must equal the enthalpy flow on the absorbing side the heat exchanger called reheater cools the compressed air down to

$$T_{reheat}^{emit} = T_{comp} - \frac{c_p^{da} + \mu_{out}\, c_p^{vap}}{c_p^{da} + \mu_{in}\, c_p^{vap}} \left(T_{out}^{set} - T_{turb}^{in}\right) \tag{15}$$

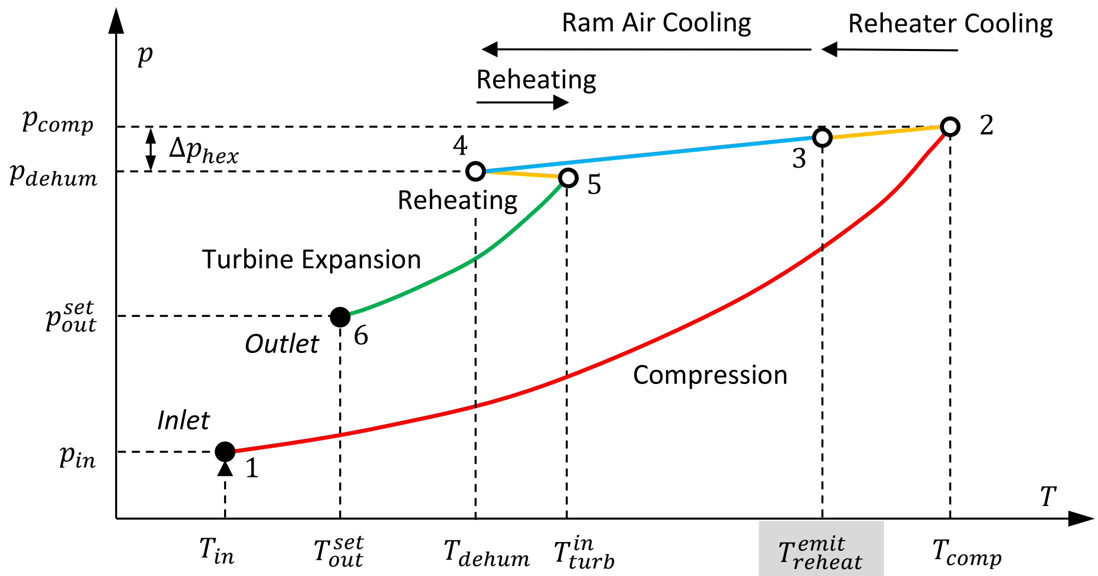

Figure 6. Turbine Expansion and Reheating (Case 2b)

As shown we were able to calculate all intermediate points 2-5 of the PT diagram explicitly in a straight forward way from the given points 1 and 6.

5.8. Model Structure

The path through the PT diagram represents a process which is shown in figure 7 below. The numbers point out in which order the process variables are to be calculated.

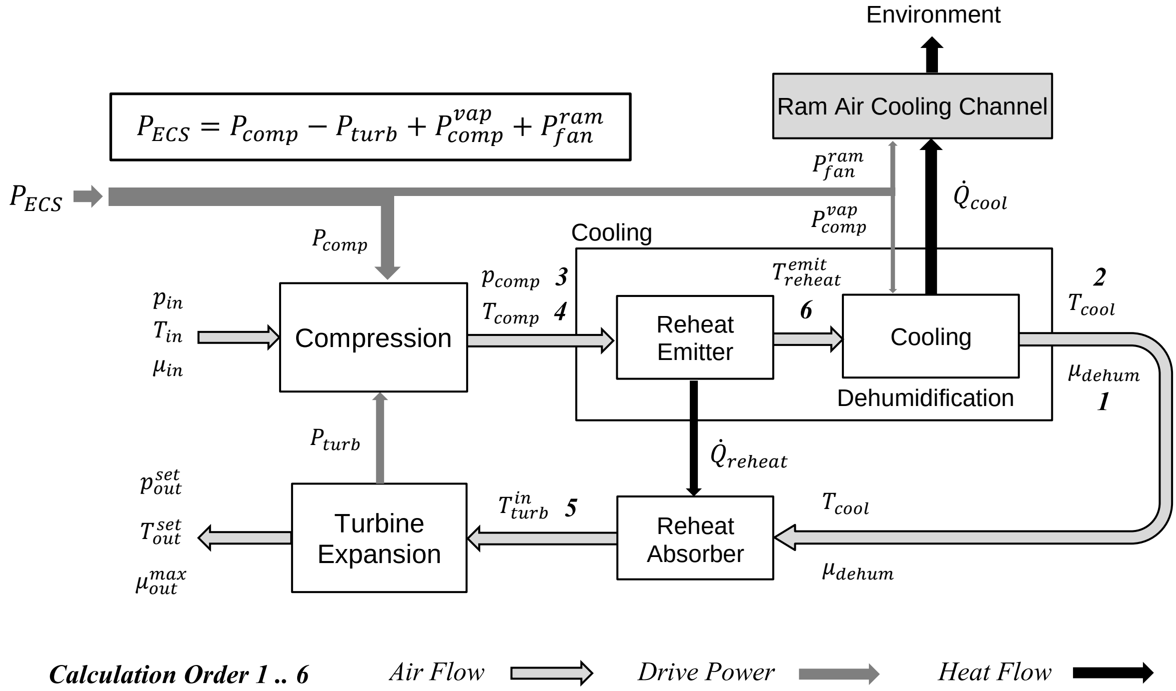

Figure 7. Process Diagram of the First Principle Model

6. Model Results

The first principle model yields the enthalpy flows at all given and calculated key points, e.g. at the inlet, after the compressor and after the reheat emitter:

$$\dot{H}_{in} = \left(c_p^{da} + \mu_{in}\, c_p^{vap}\right) T_{in}\, \dot{m}^{set}_{fresh} \tag{16}$$

$$\dot{H}_{comp} = \left(c_p^{da} + \mu_{in}\, c_p^{vap}\right) T_{comp}\, \dot{m}^{set}_{fresh} \tag{17}$$

$$\dot{H}^{emit}_{reheat} = \left(c_p^{da} + \mu_{in}\, c_p^{vap}\right) T^{emit}_{reheat}\, \dot{m}^{set}_{fresh} \tag{18}$$

From the differences we can determine all required powers resp. required heat flows, e.g. the compressor power or the heat flow rate of the reheater.

$$P_{comp} = \dot{H}_{comp} - \dot{H}_{in} \tag{19}$$

$$\dot{Q}_{reheat} = \dot{H}^{emit}_{reheat} - \dot{H}_{comp} \tag{20}$$

These results are minimum requirements and can be used for a first sizing of compressors, turbines, heat exchangers or the cooling ram air channel.

7. Simulation Studies

So far, the first principle model has mainly been used for a sensitivity analysis. It was of interest how an eECS behaves in various flight phases and under different environmental conditions. As an example, the following plot in figure 8 shows comparing simulation runs between the power demand of the first principle model and the model of a realistic ECS system in the early design phase. The dashed line shows the (simplistic) flight altitude to give an idea of the flight phase.

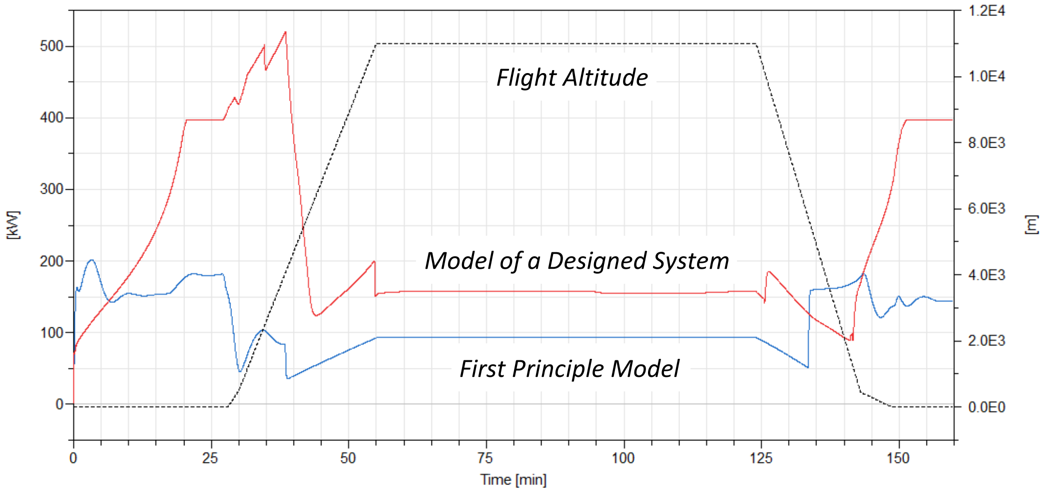

Figure 8. Example of Simulation Results

Obviously, the designed system has a lot of saving potential, particularly during the phases on ground. The reasons could be found in an undersized main heat exchanger. It may surprise that during landing there is a phase where the designed system needs less power than the first principle model. The reason is that the controller of the designed system was too slow to meet the required outlet conditions in time. It can be seen that the first principle model helps to identify many sorts of weakness in the architecture model.

8. Conclusions

We presented a first principle model of an electric ECS. In contrast to the model of a real eECS it

- has not a complex architecture
- does not need control facilities due to backward calculation from the set points
- works under all possible conditions
- can be computed easily (even with an Excel sheet) and runs fast as a simulation model

As a reference system it can serve

- to find a first sizing of the components
- to study the sensitivity in various flight phases and environmental conditions
- to find bad working conditions in real system by comparison
- as a starting system for the construction of a real system

and it can be used to improve resp. simplify the control strategy by a pre-set of the operating points.

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