ANALYSIS OF AIRCRAFT COMPRESSOR SYSTEMS

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Abstract

Analysis of aircraft compressors for the pressurization of the potable water system reveals problems by applying the standard performance curve for the determination of compressor life cycles.
Measurements show, that the cabin pressure is a very important factor which is often not taken into account. Furthermore a simulation scheme for pressurized water tanks is presented and numerical solutions are compared to experimental results.

1 Introduction

Today's potable water systems in airplanes are pressure supplied by a combination of engine bleed air and additional compressed air. During flight pressure is supplied mainly by bleed air while the compressors are used on the ground and on demand.
To conduct bleed air from the engines to the water tanks, a complex system of pipelines and control valves is required. The valves have to reduce the pressure and if necessary a pressure switch has to activate the compressors. Airlines observed that compressor failures sometimes occurred before the expected life cycle. So far, exact reasons for the malfunction are not identified, but it is presumed that compressors have a longer operating time during flight because of lacks of bleed air or the influence of lower cabin pressure.
A dynamically adaptable compressor system as well as a pressurization system based on pumps could be an alternative to the complex bleed air based pressure systems in future. A simplified structure is more favorable and more reliable, however at present it is not yet state of the art in aircraft construction.
For the development of an innovative pressure system theoretical and experimental investigation of interaction effects between the subsystems pressure supply, water storage and potable water supply as well as investigations of the performance of different compressors are necessary.
For the simulation of the dynamic interaction processes the numerical model FLOWSIS [4] was developed. In this model the one-dimensional Navier-Stokes-Equations applied to pipes are coupled with the gas dynamics of the pressure system [8], [9]. Thus, switching cycles of the compressors as well as the variation of the pressure in the system can be simulated very fast.
FLOWSIS [4] can be used for dimensioning and optimizing pressure supply systems as well as potable water systems.
The standard performance curves of the compressors are very important boundary conditions of the numerical model. Since these curves depend on the ambient pressure, a test rig was developed to measure the performance. The resulting standard curves for different inlet pressures are compared to performance curves that were theoretically derived from the standard performance curve for an inlet pressure at sea level.

2 Numerical model for pressurized tanks

The presented mathematical model [8], [9] is able to simulate the pressurization process of tanks during flight and on ground. Fig. 1 shows the assumed system architecture.

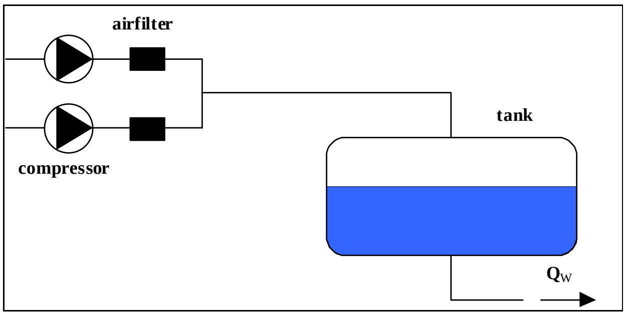

Fig. 1 Simplified form of the architecture of the pressurization subsystem

For the simulation, the performance of the compressors and the pressure loss at the air-filters are boundary conditions. It is assumed that the pressurization process is isothermal.
The General Law of Gases is applicable and the time derivative of this law is:

$$\frac{\partial p}{\partial t}\rho - \frac{\partial \rho}{\partial t} p = 0 \tag{1}$$

where p = air pressure and ρ = air density.
The total volume of the tank is constant and given by $V_T = V_W + V$. The time derivative of this equation leads to:

$$\frac{\partial V_W}{\partial t} + \frac{\partial V}{\partial t} = Q_W + \frac{\partial V}{\partial t} = 0 \tag{2}$$

where V = air volume; V_W = volume of the fluid; V_T = volume of the tank.
The air mass is given by $m = \rho V$, so that the time derivative is expressed by:

$$\frac{\partial m}{\partial t} = \rho \frac{\partial V}{\partial t} + V \frac{\partial \rho}{\partial t} \tag{3}$$

The mass flow depends on the performance of the compressors and on their steering (switch on/off) points.
These equations constitute a linear equation system that can be solved by numerical integration. The variables pressure p and flow Q_W depend on the consumption and therefore interact with the numerical model of the pipe network.
For a fast calculation, an adaptive time step control was integrated into the numerical procedure. The method showed high accuracy as well as a numerically stable behavior.

3 Compressor performance curves

The compressor performance curves are a major boundary condition of the numerical model for simulating a pressurization process. The model's accuracy mainly depends on the correct input of the compressor performance.
This information is not always obtainable since many compressor manufacturers only provide standard performance curves that are determined at sea level-conditions, that means with an ambient pressure of 1013 mbar-abs.
For a lower inlet pressure a rough estimation of the compressor performance by superposing different diagrams is recommended for example by Senior Aerospace.
As expected, the overall accuracy of these rough estimated standard performance curves is not good enough to be used for optimization tasks. Fig. 2 shows a comparison between estimated and measured values.

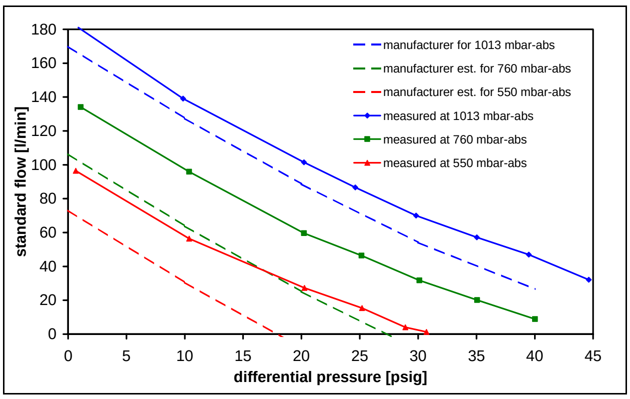

Fig. 2 Measured and calculated performance curves

For reciprocating compressors it is possible to determine standard performance curves by calculating the flow using equations (4) to (7). Reciprocating compressors are often used for the pressurization of aircraft potable water systems, because they have several advantages like good efficiency even in small size and at high pressure, good operating flexibility over a wide range of conditions, very simple design, field repair and lightweight [2].

The terms, variables and definitions used in the following derivations are in accordance with [1] and [10]. In reciprocating compressors the volume inflow $\dot{V}_H$ is usually determined by dividing the volume passed by the piston by unit of time. The theoretical volume flow is the product of piston displacement V_H multiplied by speed of rotation n.

Due to the expansion of the compressed gas remaining in the clearance volume, the leaks and the preheating of the gas drawn in, the effective volume flow $\dot{V}_{1,nu}$ referred to inlet conditions is much smaller than $\dot{V}_H$. The ratio of these volumetric flows is the effective volumetric efficiency:

$$\lambda_{nu} = \frac{\dot{V}_{1,nu}}{\dot{V}_H} \qquad (4)$$

The effective volumetric efficiency can be described by four independent efficiency factors:

$$\lambda_{nu} = \lambda_F \cdot \lambda_p \cdot \lambda_A \cdot \lambda_d \qquad (5)$$

where $\lambda_i \approx \lambda_F \lambda_p$ = indicated volumetric efficiency including throttling capacity losses, $\sigma = \lambda_A \lambda_d$ = warm-up factor for temperature increase including leakage losses.

For estimating λ_F equation (6) can be used. A derivation is presented in [5]. This estimation is also recommended by [1].

$$\lambda_F = \left[1 - \varepsilon_0 \left(\frac{Z_4}{Z_3} \cdot \psi^{\frac{1}{ne}} - 1 \right) \right] \qquad (6)$$

where ε_0 = cylinder clearance volume as decimal fraction of displaced volume, Z = compressibility factors, ψ = pressure ratio across cylinder (flange to flange) and n_e = isentropic volume exponent at operating conditions. For determining clearance and displaced volume, the compressor has to be dismantled.

Compressor valves and cylinder gas passages are in direct contact to the gas flowing into the cylinder displacement. There are pressure- and with it capacity-losses associated with this flow. These losses called throttling capacity losses λ_p range between 0,95 and 0,98 [1].

The warm-up factor σ depends on many different conditions e.g. pressure ratio, cooling, speed of rotation and compressor type. As explained in [3] and [10], it is not possible to calculate the warm-up factor so far. In [5] an equation for estimating σ is presented, but the factor C can only be determined by experimental investigations:

$$\sigma = 1 - \frac{C}{\left(D \cdot \rho \cdot c_{K,m} \right)^{0,2}} \cdot \frac{D + 2s}{D} \left(\psi^{\frac{\kappa-1}{\kappa}} - 1 \right) \qquad (7)$$

where D = diameter of the cylinder, s = piston stroke, $c_{K,m}$ = averaged speed of the piston, ρ = density of the gas, C = experimental determined factor.

Experimental results of [7] show the dependency of the warm-up factor on the inlet pressure and on the density of the gas. This dependency is very important for calculating the runtime and the expected life cycle of aircraft compressors. For example, the cabin pressure during flight averages about 760 mbar-abs.

In Fig. 3 σ-values for different inlet pressures of a water-cooled compressor at 1300 rpm [7] and of a small air-cooled aircraft compressor at 2200 rpm are compared.

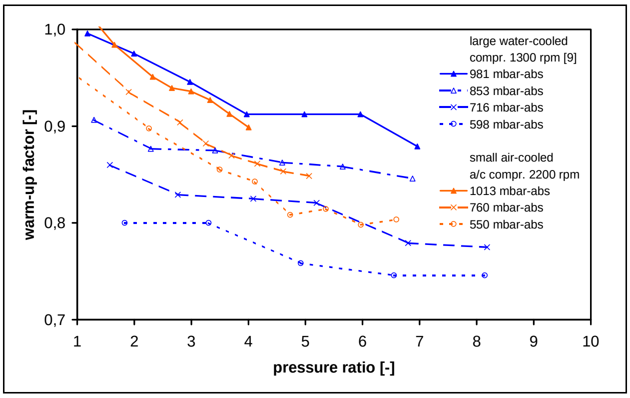

Fig. 3 Warm-up factor for compressors at different inlet pressures

In fact there is no exact calculation of the flow at different inlet pressures possible since σ–values are not known for different conditions and for any aircraft compressor design.
Aircraft compressors are optimized in weight and size and not in performance. For example, larger reciprocating aircraft compressors often have a small defined leakage in the piston head or cylinder head to reduce the necessary power for restarting at higher outlet pressures and with it to minimize the weight of the electrical motor (e.g. Fig. 4). Due to these leakages it is not possible to estimate the cylinder clearance volume and thus the flow.
Therefore, standard performance curves of aircraft compressors can only be accurately determined by a special test rig.

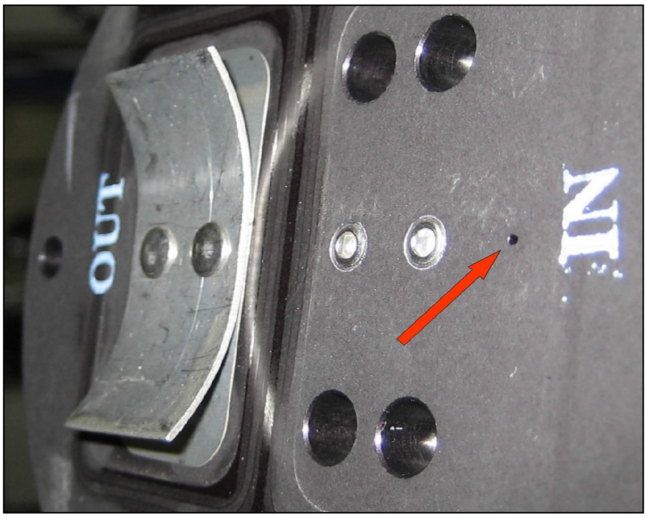

Fig. 4 Defined leakage of an aircraft compressor

4 Compressor test rig

At Hamburg University of Technology a test rig for measuring compressor performance curves for different ambient conditions was developed together with the industrial partner Luftfahrtgeräte Gauting GmbH (LGG).
The test rig consists of two parts: the inlet-section (1) and the outlet-section (2) (Fig 5).
where V = proportional pressure regulator with proportional magnet valve, s = input signal (pressure presetting), Q = rotary gas meters, T = temperature sensor, p = absolute pressure sensor, C = tested compressor, W = cooling.

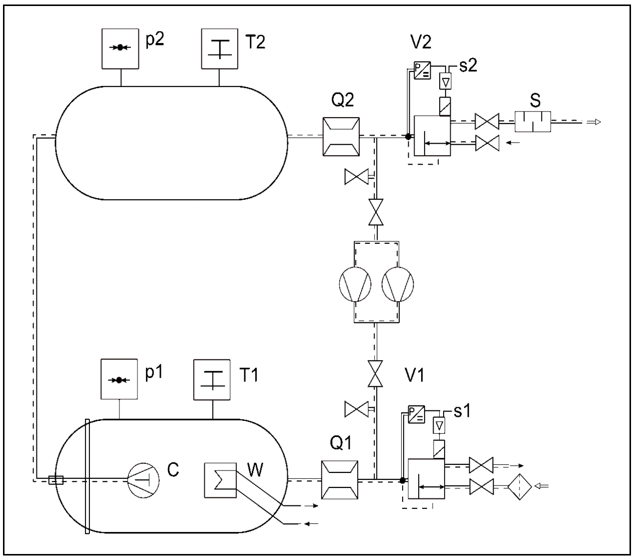

Fig. 5 Compressor test rig

The proportional regulated valves of the compressor test rig keep the pressure in the inlet- or outlet-section constant. An integrated PI-controller compares the measured pressure with the pre-selected value and controls a proportional magnet-valve.
Rotary gas meters are volume-measuring devices for gaseous media and operate according to the positive displacement principle. By using rotary gas meters it is possible to register the suction and flow capacity of the tested compressor under operating conditions and with highest accuracy over a wide pressure and temperature range.

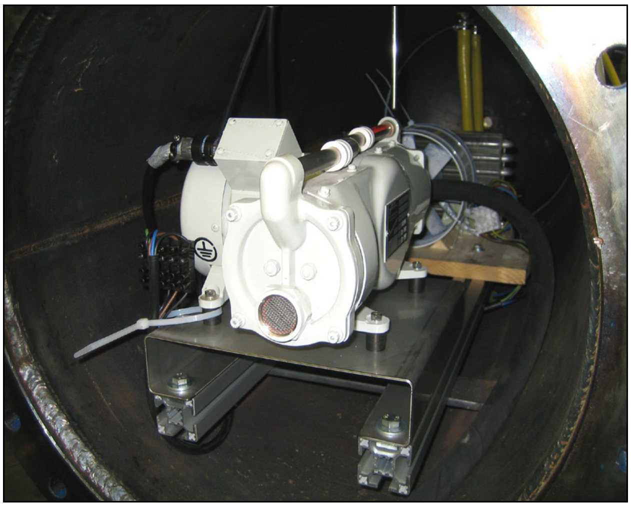

Fig. 6 Aircraft compressor inside the inlet-tank (cooling unit in the background.)

The tested compressor is operating inside the inlet-tank. Thus, compressor leakages have no effect on the accuracy of the measured flow rates at different inlet-pressure conditions. For keeping the temperature constant a cooling unit is installed inside the inlet-tank (Fig. 6).
With two additional compressors connected to the inlet- and the outlet-section of the test rig it is possible to control any pressure in a range of 400 to 1100 mbar-abs without depending on atmospheric pressure conditions.
The values of flow, pressure and temperature are measured continuously at the inlet and the outlet of the compressor. By using the General Law of Gases (Eq. 8) it is possible to determine the standard performance curve of the compressor for a defined inlet pressure and to check the accuracy of the measured values by comparing the standard flow of the inlet and the outlet (Eq. 9). This equation is obtained for steady flow conditions.

$$\frac{p_1 \cdot V_1}{T_1} = \frac{p_2 \cdot V_2}{T_2} \qquad (8)$$

$$\frac{p_S \cdot \dot{V}_S}{T_S} = \frac{p_1 \cdot \dot{V}_1}{T_1} = \frac{p_2 \cdot \dot{V}_2}{T_2} \qquad (9)$$

where p_s = standard pressure (1013,25 mbar-abs), $\dot{V}_S$ = standard flow, T_s = standard temperature (15°C).
The standard performance curve of a small reciprocating aircraft compressor at different inlet pressures is shown in Fig. 7.

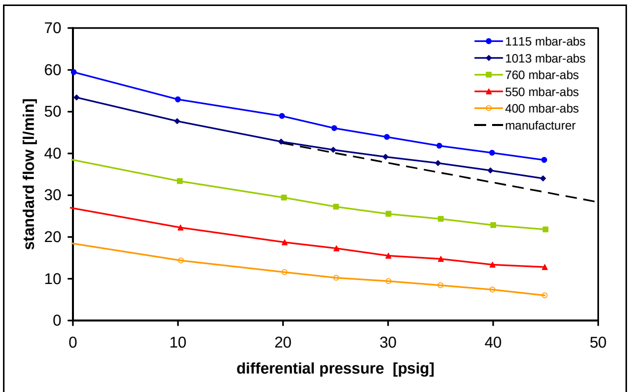

Fig. 7 Standard performance curve for different inlet pressures

The experimental results show that the measured compressor performance curve compares very well with the curve given by the manufacturer for ground pressure conditions.

5 Comparison of numerical and experimental results

In order to validate the numerical model, several experiments were performed on a full size aircraft potable water system test rig (Fig. 8). The experimental apparatus consists of two aircraft water tanks, each with a capacity of 370 l that are half filled.

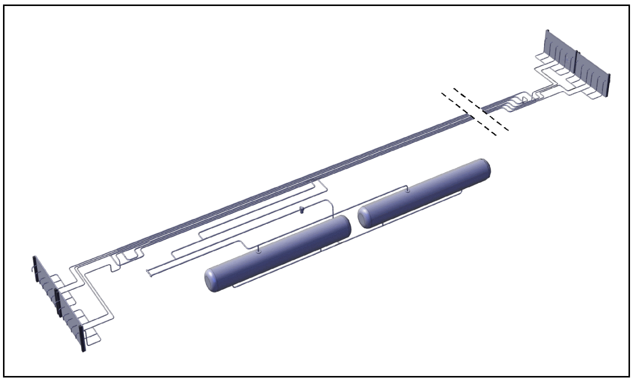

Fig. 8 System test rig – a full size aircraft potable water system

Eight times water is withdrawn during the experiment time of one hour. Boundary conditions are the working range of the compressor of 35 to 40 psig and its performance curve.

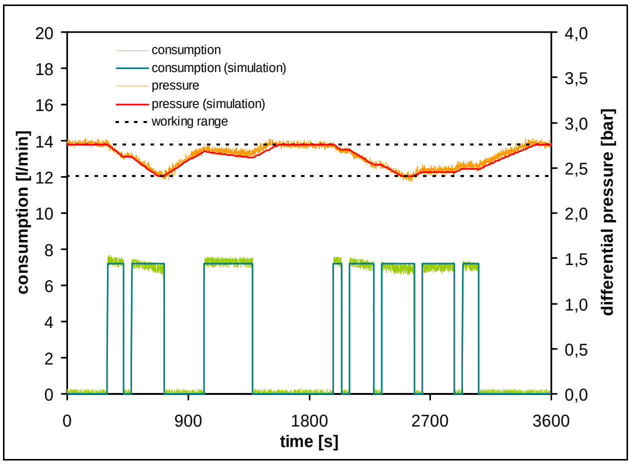

Fig. 9 Tank pressure and consumption

Pressure, consumption as well as the switch on and off process of the compressor were recorded.
Computational results agree very well to the measured values (Fig. 9). It became evident, that the model's accuracy mostly depends on the correct performance curve of the compressor.
The compressor runtime to build up the necessary tank pressure is much longer having lower cabin pressure than at sea level. With the numerical tool this runtime can be calculated and applied to the estimation of the expected life cycle of aircraft compressors.

6 Conclusion

With the presented numerical methods it is possible to simulate the interaction of the pressurization of aircraft water tanks in dependency of the consumption and the cabin pressure with high resolution and very satisfying accuracy in short calculation time.
With the knowledge of the exact performance curves of a compressor for different inlet pressures based on experimental research, very close to reality results are obtained by numerical simulation.
There are some formulas for estimating the compressor performance for different inflow conditions available, but for exact modeling and optimization only experimental performance curves should be used.
The presented tools and ideas will lead to more precise estimations of compressor life cycles which will reduce costs.

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