ANALYSIS OF THE DRIER SYSTEM OF IAE PILOT TRANSONIC WIND TUNNEL

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Abstract. *The Pilot Transonic Wind Tunnel (TTP) of the Aeronautics and Space Institute (IAE) is a conventional closed circuit tunnel, continuously driven by a main axial two-stage compressor of 830 kW of power. Its test section with 25 cm x 30 cm is provided with slotted walls and was designed for high quality flow tests. The tunnel control system guarantees stable conditions of pressure from 0.5 bar to 1.25 bar, Mach number from 0.2 to 1.3, temperature and humidity, related to test section. An injection system works intermittently in conjunction with the conventional tunnel operation to reach remote regions of the operational envelope, without demanding extra installed power. The humidity control is one important feature to achieve high flow quality at the test section. Beyond certain humidity level, moisture from the flow will condense out affecting the local properties, and also the condensation may clog pressure taps, disturbing pressure readings. So, it is necessary to establish a dew point criteria based on the local static temperature. Since it is impossible to perfectly isolate the tunnel circuit casing, there will always be leakage at various interfaces and seals introducing moist, or losing dry air from the circuit to the atmosphere, when the tunnel circuit is under or over pressurized, respectively. The worse case of leakage is through the gap between the main compressor shaft and the tunnel circuit. An useful mathematical model for the dryer system was performed, considering the physics characteristics of the dryer, the tunnel circuit and the ambient properties. Some experiments were performed with the real tunnel for comparison.*

Keywords: *transonic wind tunnel, dryer system, humidity control system*

1. INTRODUCTION

In high speed wind tunnel there is always a concern about the humidity levels in the local flow. The amount of moisture that can be held by a certain air volume increases with increasing temperature, but is independent of the pressure. When isentropic expansions occur the temperature drops and the air may become super cooled (cooled to a temperature below dew point temperature). Moisture will then condense out and, if the moisture content is sufficiently high, it will appear as a dense fog in the tunnel. Condensation can result in changes in local Mach number, and other flow characteristics such that data taken in a wind tunnel test may be erroneous (Pope and Goin, 1978).

Although this problem is more significant in supersonic wind tunnels (Burgess, 1951), its analysis in transonic wind tunnel is very important, because it can restrict the operational envelope of the tunnel. For example, in supercritical regions the local Mach number variation that is provoked by localized water vapor condensation alters on shock waves establishment conditions. Because of the sensitivity of shocks to minor disturbances in transonic flow, the overall aerodynamics of the vehicle can be significantly affected. Figure 1 shows a typical result of the influence of humidity on the drag coefficient for a fighter aircraft (Davis et al., 1986) – that the drag prediction was 0.001 greater at Mach number 0.9., and, unless the dew point temperature is maintained at a value of at least 5 K less than the free stream static temperature, water vapor condensation will have a measurable influence on the aerodynamic data.

Condensation will occur as a function of four parameters related to the air stream: the amount of moisture in it, its static temperature, its static pressure and the time during which the stream is at a low temperature. Pope and Goin (1978) discuss this issue with adequate details. Depending on the tunnel conceptual design and the test to be performed, some level of condensation may be tolerable during the tunnel operation. However, a practical and conservative rule, normally used in most transonic wind tunnel designs, is the one already presented: the dew point at the test section must be 5 K below the free stream static temperature.

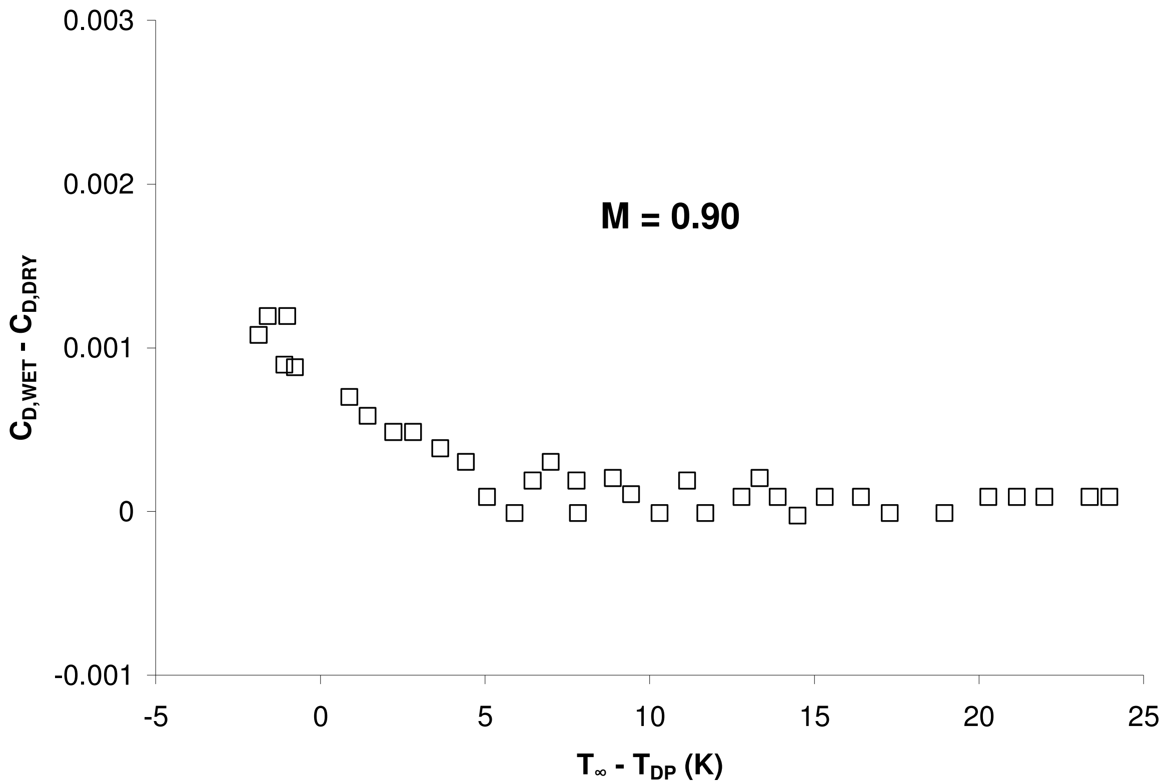

Figure 1. Influence of humidity on the drag coefficient of typical fighter aircraft (Davis et al., 1986).

Assuming a dew point temperature of 5 K below the static local temperature, Table 1 shows the possible Mach number for condensation free criterion at the test section, considering a stagnation temperature of 27 K. Observe that only in subsonic range it is possible to perform condensation free tests at typical ambient temperature condition. In the other hand, for low supersonic tests, it is very important an effective dryer system to guarantee the flow quality at the test section.

Table 1. Dew point temperature requirements for various test section Mach numbers (stagnation temperature of 27 C).

M	$T_{DP,des}$ (C)
0.2	19.5
0.3	16.5
0.4	12.5
0.5	7.6
0.6	1.7
0.7	-4.9

M	$T_{DP,des}$ (C)
0.8	-12.2
0.9	-20.0
1.0	-28.2
1.1	-36.6
1.2	-45.2
1.3	-53.9

The Pilot Transonic Wind Tunnel (TTP) of the Aeronautics and Space Institute (IAE) is a scaled down 1/8th from an industrial facility, intended to be build in the General-Command for Aerospace Technology (CTA), in Brazil. TTP is a modern conventional closed circuit transonic wind tunnel, continuously driven by a two-stage, 830 kW frequency controlled main compressor. The tunnel is now completely installed and under calibration campaigns. Its test section has 30 cm x 25 cm with semi-open slotted walls. A main control system integrates several subsystems actions in order to maintain desirable Mach number (from 0.2 to 1.3), pressure (from 0.5 bar to 1.2), temperature and humidity. The tunnel also incorporates an injection system that operates intermittently, providing high speed gas jet, in conjunction with the main compressor operation, in order to enlarge the tunnel operational envelope without demanding extra installed power.

A transonic wind tunnel is best characterized by its semi-open test section walls. They allow mass flow to leave the test section, preventing from flow chocking, reducing shock/expansion reflections and wall interference, mostly in transonic regime – special flaps at the end of test section help to control the amount of mass flow trough the walls by their opening position. In order to control the pressure outside of the test section walls a big volume (the plenum chamber) envelops the test section and an auxiliary compressor extracts mass from this chamber. Figure 2 shows details of the components into the plenum chamber of TTP. The flow, as indicated in the figure, comes from left to right, being accelerated in the first throat, passing through the test section and the re-entry flaps. Then the flow goes through the second throat (which has an important role in supersonic regime), through the injection mixing chamber, and finally reaches the high speed diffuser. More technical details may be found in Falcão Filho and Mello (2002).

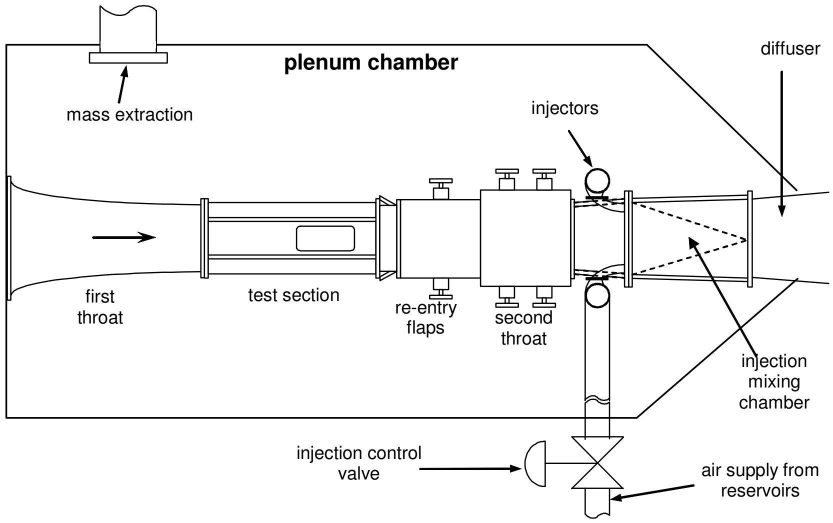

Figure 2. TTP plenum chamber, showing its internal components.

The pressure control system of TTP controls the test section pressure, the humidity level and the mass flow extraction percentage from the plenum chamber. Figure 3 shows a sketch of this system. There are other auxiliary systems in the facility, but for the purpose of clarity they are absent in the figure. The reader will find more details in Falcão Filho and Mello (2002). The pressure control system is based on a centrifugal compressor CENTAC (item 8 in Fig. 3) supplied by Ingersoll Rand, model 1ACV15M2LP, with intake capacity of 0.577 m^3/s, pressure compression ratio of 4, and driven by a 186 kW electric motor. The compressor (8) takes air from the plenum chamber (5), from circuit duct (6) and from the atmosphere, and delivers it back to the tunnel circuit passing through the dryer (9), or without passing through it, and also delivers the air direct to the atmosphere. All its functions are performed by a computer program developed in a LabView platform which automatically controls six flow control valves (FCV1, FCV2, ... FCV6).

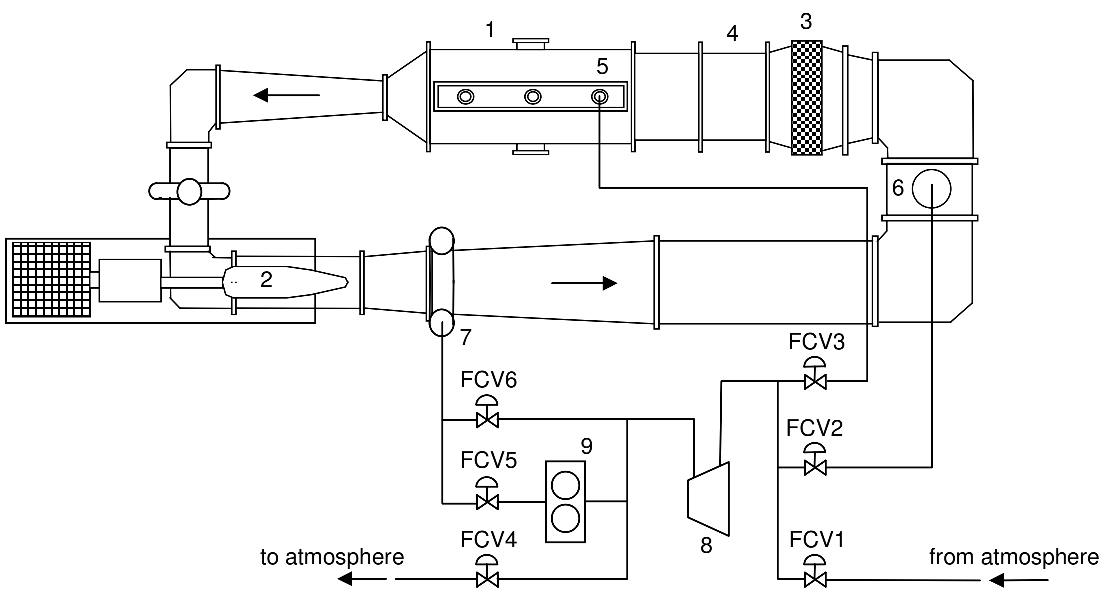

Figure 3. Diagram of the pressure control system of TTP.

To control the pressure level at the wind tunnel circuit, basically air is taken from port 6 and it is delivered to port 7, passing or not the dryer (9) – control valve VCF1 is required only when it is necessary to admit new air into the tunnel circuit to increase pressure while control valve VCF4 is required only when it is necessary to extract air from the circuit to atmosphere to decrease pressure. Depending on test conditions, part of the flow will pass through the dryer (9) and the control valves VCF5 and VCF6 will be used to select the amount of air to be dried.

Below Mach number 0.6, no plenum chamber mass flow extraction is required for typical tests and just a modest drainage is sufficient to comply with the humidity criterion (see Tab. 1). However, above Mach number 0.6 more and more mass flow extraction from the plenum chamber is required and also, lower and lower dew point temperature criterion is needed (see Tab. 1). In these situations, valves VCF2, VCF3, VCF5 and VCF6 will be required to operate in a logical fashion to obtain the best operation condition.

The heatless regenerative dryer (9) was supplied by Ingersoll Rand, its model is HRD55, with capacity of 0.680 m^3/s. In order to achieve dew points lower than -40 C, a desiccant dryer was the preferred solution, working by the principle of adsorption – it has inherent advantages of high reliability and low maintenance. The dryer utilizes dual pressure vessels, one vessel is actively drying the compressed air, while the other vessel is being regenerated – the two vessels switch back and forth, with the regenerated vessel becoming the active vessel, as the other vessel regenerates. The vessels, also called here desiccant towers, have been designed to allow a minimum of 5 s contact time, which is essential for complete moisture adsorption and consistent dew points, and for air velocity less than 0.3 m/s. Figure 4 shows an operational scheme of this unit, when the left desiccant tower is in use – the thick arrows point the way the flow goes through the dryer. Air is admitted by inlet port and it is selected by a shuttle valve (3) to be admitted at the bottom of the left tower. Moisture is absorbed by the desiccant as the air flows upward through the vessel. Dry compressed air exits at the outlet port, being selected by another shuttle valve (7). While the left tower is in use, a portion of the dry air is diverted through restriction orifices (9) and (10), and metered through the purge adjusting valve, through the vessel being regenerated, drying the wetted desiccant and preparing it for reuse. This purge air is exhausted through the purge exhaust valve (6) to local atmosphere (see thin arrows). The shuttle valves (3) and (7) guarantee the isolation of the two flow circuits (thick and thin arrows). Each tower operates for 150 s in use and 150 s being regenerated in switching procedure that is repeated again and again, assuring a continuous flow of dry air from the unit.

It will be presented some analysis of the pressure control system of the TTP while the humidity control function is required. A mathematical model was elaborated to represent the system for comparison purpose.

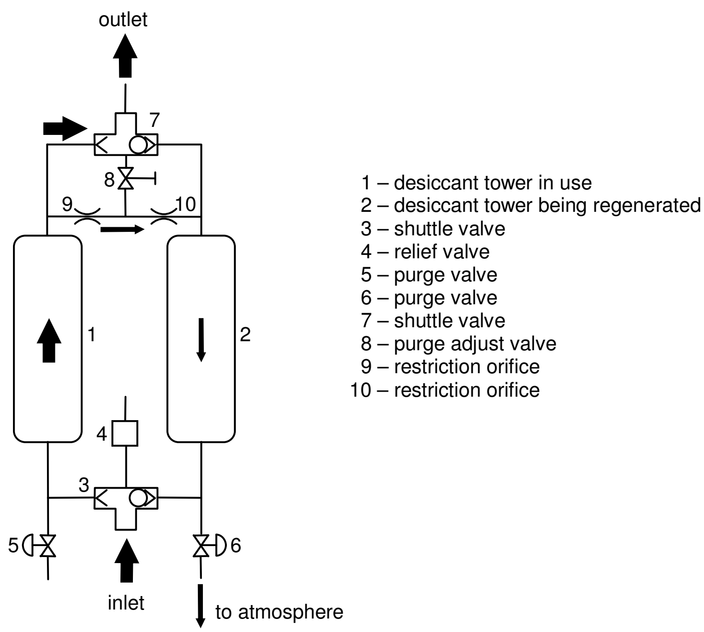

Figure 4. Operational scheme of the dryer.

2. METHODOLOGY

A mathematical model (Falcão Filho, 1990) was used to evaluate the pressure control system for the industrial facility. The same approach was used here for the TTP with some details incorporated. For example, at that time, there was not any information about the dryer system and a constant humidity was assumed for its output and, of course, there was not any tunnel at all. Now it is possible to compare the model result with the experimental test results. A brief explanation will be presented, followed by the methodology used in the experimental tests.

2.1 Mathematical Model

To analyze the dryer function of the pressure control system of TTP, a mathematical model was developed based on a simplified scheme for the wind tunnel circuit and for the pressure control system, according to the Fig. 5. This figure was obtained from Fig. 3, being now represented only the control lines that take role in the model, adding the external leakages in and out of the tunnel circuit, represented by white and black arrows, respectively. In the figure, there are two ways to operate the system. The first one, when the tunnel circuit is under pressurized. In this case, one can follow the white arrows and see that, there will be leakage from the outside atmosphere into the tunnel casing. So, in a simplified interpretation, the compressor will have to force the same amount of flow back to the atmosphere. The second situation will be when the tunnel circuit is over pressurized (black arrows) and the leakage flow is from the tunnel circuit out to the atmosphere – the compressor will have to take the same amount of new air from the atmosphere to maintain the pressure level into the circuit.

In Fig. 5, $\dot{m}_l$ is the mass flow leakage into the tunnel or out of it, $\dot{m}_e$ is the mass flow extracted from the tunnel circuit, $\dot{m}_1$ is the mass flow that is readmitted into the tunnel circuit by the control system, α is the fraction of mass that by passes the dryer back to the tunnel circuit. w_t is the absolute humidity in the tunnel circuit, w_a is the absolute humidity at the ambient conditions, w_d is the absolute humidity leaving the dryer and w_2 and w_3 are auxiliary absolute humidity values.

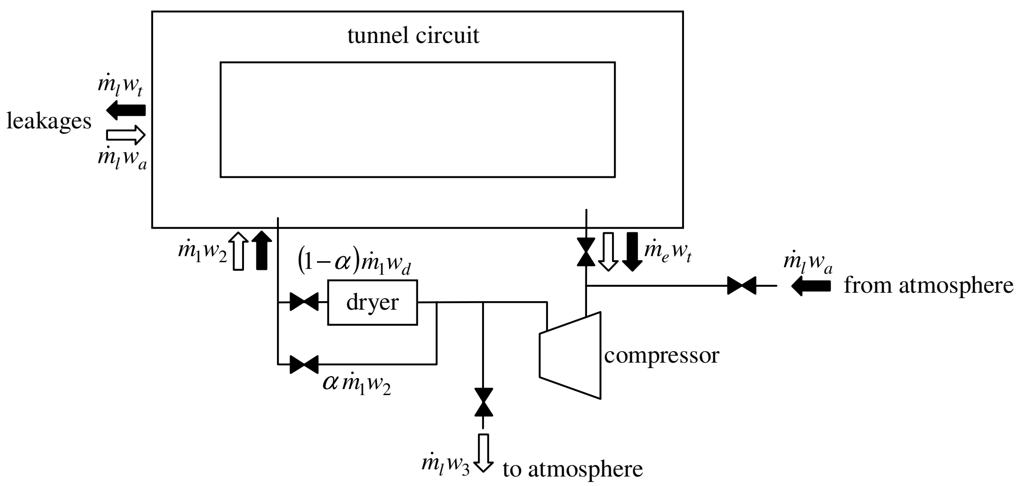

Figure 5. Simplified scheme mathematically modeled. White and black arrows represent the flow directions when the tunnel circuit is under pressurized and over pressurized, respectively.

Applying the general relations for two gas-vapor mixture (Potter and Somerton, 1995),

$$w_{mix} = \frac{w_{gas1}\dot{m}_{gas1} + w_{gas2}\dot{m}_{gas2}}{\dot{m}_{gas1} + \dot{m}_{gas2}}, \tag{1}$$

and the principle of mass conservation, to the mathematical model described in Fig. 4, which the general equation in a differential form is given by

$$d\ w_t = \frac{\dot{m}_{H_2O,in} + \dot{m}_{H_2O,out}}{M_{air}}\, dt\,, \tag{2}$$

where $\dot{m}_{H_2O,in}$ and $\dot{m}_{H_2O,out}$ are the mass flow into the tunnel circuit and out of it, respectively, M_{air} is the total amount of air into the tunnel circuit and dt is time differential, then it is possible to obtain finite difference equations, given by

$$w_t^{(n+1)} = w_t^{(n)} + \frac{\Delta t(\dot m_l + \dot m_e)}{M_{air}\left[1+w_t^{(n)}\right]} \frac{A_1 \dot m_e + A_2 \dot m_l}{A_3 \dot m_e + A_4 \dot m_l} \tag{3}$$

and

$$w_t^{(n+1)} = w_t^{(n)} + \frac{\Delta t\ \dot m_e}{M_{air}\left[1+w_t^{(n)}\right]} \frac{A_1 \dot m_e - A_5 \dot m_l}{A_3 \dot m_e - A_6 \dot m_l}\ , \tag{4}$$

corresponding to the solutions for under pressurized and over pressurized tunnel circuit, respectively. The auxiliary terms in these equations are $A_1 = (1-\alpha)\left(w_d - w_t^{(n)}\right)$, $A_2 = \alpha\ w_a + (1-\alpha)\ w_d - w_t^{(n)}$, $A_3 = 1 + (1-\alpha)\ w_d + \alpha\ w_t^{(n)}$, $A_4 = 1 + (1-\alpha)\ w_d + \alpha\ w_a$, $A_5 = (1-\alpha)\ w_d - w_a - \alpha\ w_t^{(n)}$ and $A_6 = -(1-\alpha)\ w_d - w_a + \alpha\ w_t^{(n)}$.

As initial conditions for the model, it is necessary to inform the model the extracted mass flow from the circuit ($\dot m_e$) and the leakage mass flow ($\dot m_l$). This last one is more difficult to estimate. Since the shape and size of the holes of the circuit casing are not predictable, normally an equivalent hole for the circuit casing is estimated. From the Saint Venant e Wantzel equation (Chapman and Walker, 1971), one can calculate the mass flow through a hole as a function of the pressure relation between both sides of a wall, that is given by

$$\dot m_l = \frac{p_1 A_l}{\sqrt{\gamma\, R\, T_0}} \sqrt{9.8 \left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}\ , \tag{5}$$

where p_1 is the higher pressure, p_2 the lower pressure, A_l is the equivalent hole area, γ is the relation between the specific heats at constant pressure and at constant volume, R is the gas constant and T_0 is the stagnation temperature. This equation is valid for $p_2/p_1 > 0.5283$ – for lower values, the mass flow is constant and equal to the critical value, which corresponds to the critical solution, with sonic speed at the hole.

2.2 Experimental Tests Procedures

Some tests were performed to obtain an operational envelope of the dryer, by measuring the humidity at its outlet port, and varying the pressure ratio of the compressor. The compressor ratio was changed controlling the valves FCV1 and FCV5, being kept completely closed all other valves: the air was admitted always from the local atmosphere (through valve FCV1) and the compressor ratio was adjusted by positioning the valve FCV5.

The instrument used to measure the humidity was furnished by Delta OHM, model DO 9861T. The instrument readings were in dew point temperature units and a conversion to absolute humidity was necessary, to be used in the mathematical model described in item 2.1. Many empirical equations from psychrometric chart may be obtained, by the one from Sverdrup (1989) was used,

$$w = \phi\ \ 10^{7.582 - \frac{2676}{T_{DP}}}\ , \tag{6}$$

valid for dew point temperatures below 273.16 K and

$$w = \phi\ \ 0.01744 \left(\frac{288.72}{T_{DP}}\right)^{5.1724} e^{23.6785\left(1 - \frac{288.72}{T_{DP}}\right)}\ , \tag{7}$$

valid for dew point temperatures equal or above 273.16 K. In these equations, ϕ is the ratio between molecular weights of water vapor and dry air (equal to 0.622), T_{DP} is the dew point temperature in K and w is the absolute humidity is kg of water per kg of dry air.

Some other tests were performed with the dew point instrument installed into the tunnel circuit to measure the humidity for various operation conditions of the pressure control system.

3. RESULTS

Figure 6 shows a complete cycle of the dryer system during a 300 s run. One can see the dew point temperature and the compression ratio variations during three shifts of the dryer towers: each tower performed one single cycle “in duty”. The shifts are best observed in the compression ratio curve – during tower shifts a pressure disturbance appears. Initially the pressure increases due to the two purge valves closing (5 and 6 in Fig. 4), followed by a quick drop in the pressure due to the opening of the purge valve corresponding to the new tower being regenerated and, finally, the pressure is stabilized. This transient period lasts about 15 s and does not affect significantly the available time for tests.

From the tests performed it was observed that the humidity level at the outlet of the dryer is greatly dependent on the compression ratio, as will be shown later.

In Figure 6 the averaged compression ratio was 2.30 and the corresponding averaged dew point temperature was -8.55 C. Considering a stagnation temperature of 27 C, with this dew point temperature it would be possible to run the tunnel, with condensation free condition, up to Mach number 0.75 (see Tab. 1). However, only with measurements into the tunnel circuit this issue will be clarified.

It is evident from the figure the behavior of the dryer during a cycle. Just after a tower shift, the dew point temperature has a strong drop. It denotes the great capacity of dryness of the tower just after being regenerated: the dew point temperature reaches the lowest value in about 20 s. After that, the capacity of dryness is slowly decreased, being observed by the dew point temperature rising, almost asymptotically, until a new shift of towers occurs. It is interesting to note that the behavior is not the same for both towers. Of course the maintenance procedures, the quantity of dryer element and the instant conditions may explain this.

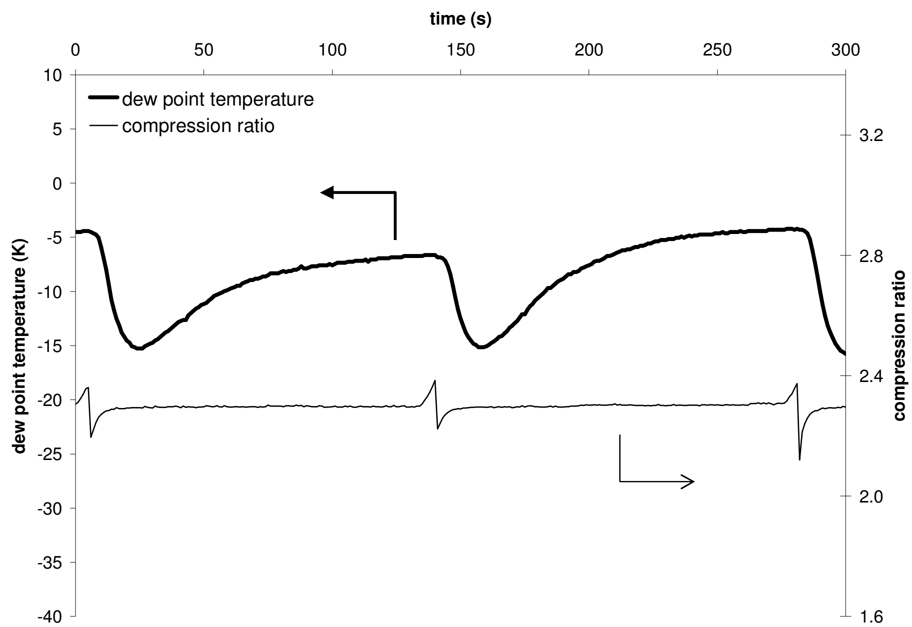

Figure 6. Results from a complete cycle of the dryer.

Many graphs like the one in Fig. 6 were obtained by varying the compression ratio from 2.0 to 3.3. These results were all put together in an operational envelope for the dryer. For each compression ratio it was calculated a maximum, an average and a minimum value of the dew point temperature of a complete cycle. Figure 7 shows these results.

All the tests were repeated in a second day when the ambient conditions were slightly different – in Fig. 7 they are distinguished by day 1 and day 2. During the experiences, it was observed that the dew point instrument was unable to measure values below –40 C. So, in some cases the minimum and, consequently, the averaged values were affected by the instrument limitation. In these cases they are marked differently in the figure. A curve fitting was determined using a first order polynomial, ignoring those values affected by the instrument limitation.

It is evident from the Fig. 7 that the capacity of dryness is greatly improved with increasing of the compression ratio in the dryer.

The same battery of tests was repeated, now the dew point temperature being measured into the tunnel circuit. The instrument probe was installed at the lowest flow speed region of the tunnel circuit – the stilling chamber. The results are shown in Fig. 8. This figure is very important, as it represents a real humidity system operation. During these tests, the pressure control system was required to operate to maintain the circuit pressure at 1.013 bar, while the ambient pressure was 0.944 bar. Of course, flow leakage through the tunnel walls to the atmosphere is expected in this case.

The results are reasonable and low dew point temperature is reached with compression ratios not too high. However, in order to run the tunnel at high speed tests (according to Tab. 1) it will be necessary lower dew point temperatures. From Fig. 8, for a compression ratio of 2.8 the dew point temperatures are: maximum -18.2 C, average -29.1 C, minimum -38.7 C. In a conservative analysis, the maximum value of dew point temperature (-18.2 C) corresponds to tests at Mach number 0.9 without condensation and all the test period the dew point temperature guarantees a condensation free condition. Another case, considering the average value of -29.1 C, it is possible perform tests up to Mach number 1 at the test section. For this case, for a complete run of 200 s, it is observed two periods of about 50 s when the dew point temperature is at condensation free condition.

To reach higher Mach number condensation free conditions, the tests will have shorter time duration or the dryer will operate at a higher pressure ratio.

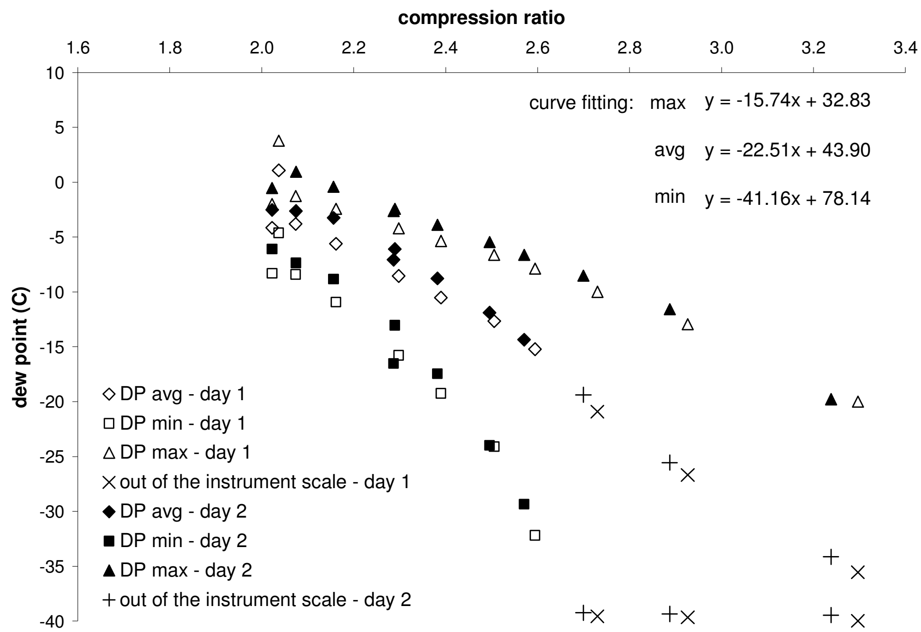

Figure 7. Operational envelope for the dryer. Tests were taken in two distinct days.

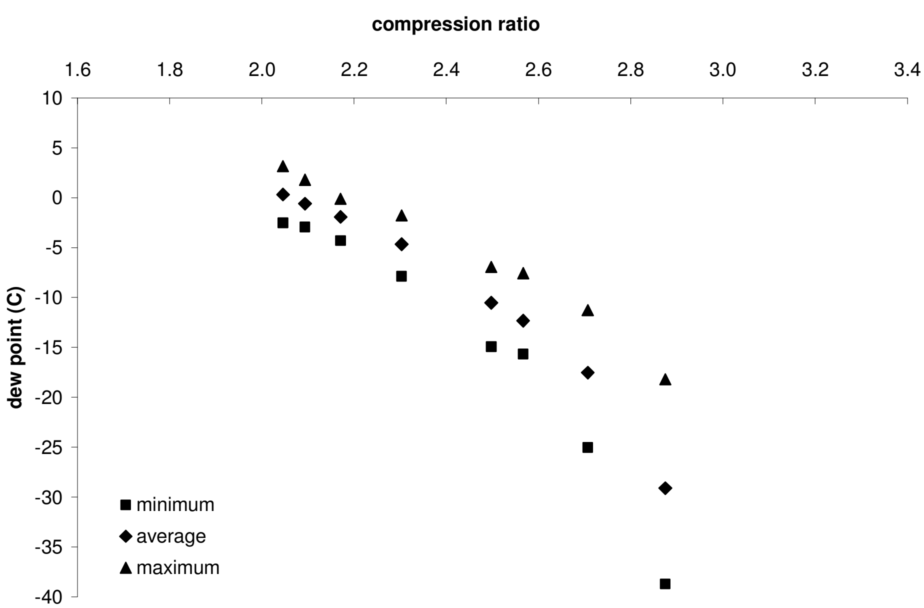

Figure 8. Results of dew point temperature measurements into the tunnel circuit.

It is very instrumental to compare Figs. 7 and 8. The range of dew point measurements (between maximum and minimum value) observed into the tunnel circuit (Fig. 8), for a determined compression ratio, was smaller than the corresponding dew point temperature at the exit of the dryer (Fig. 7). For example, for a compression ratio of 2.5 these temperatures difference observed into the tunnel was 8 C while at the dryer exit it was 18 C.

To clarify this point, another graph with both experiences with this same compression ratio was obtained. Figure 9 shows these results and compares them with the numerical calculation as presented in item 2.1. All time scales were shifted to precisely coincide the moments when the towers are shifted. The numerical code had as input data the absolute humidity variation at the dryer outlet.

As observed before and from the Fig. 9, it is clear that the great drop in the dew point temperature observed at the dryer outlet (see also Fig. 6) is not present in the tunnel circuit measurements. In some way, the tunnel circuit flow flattens this effect, although the same maximum value can be noted.

Another interesting fact is that the numerical result reproduces reasonably well the dew point temperature into the tunnel circuit, although the curve seems to be flatter than the experimental data. This can be explained by the fact that the mathematical modeling assumes the tunnel circuit as a large control volume with a global effect more noticeable.

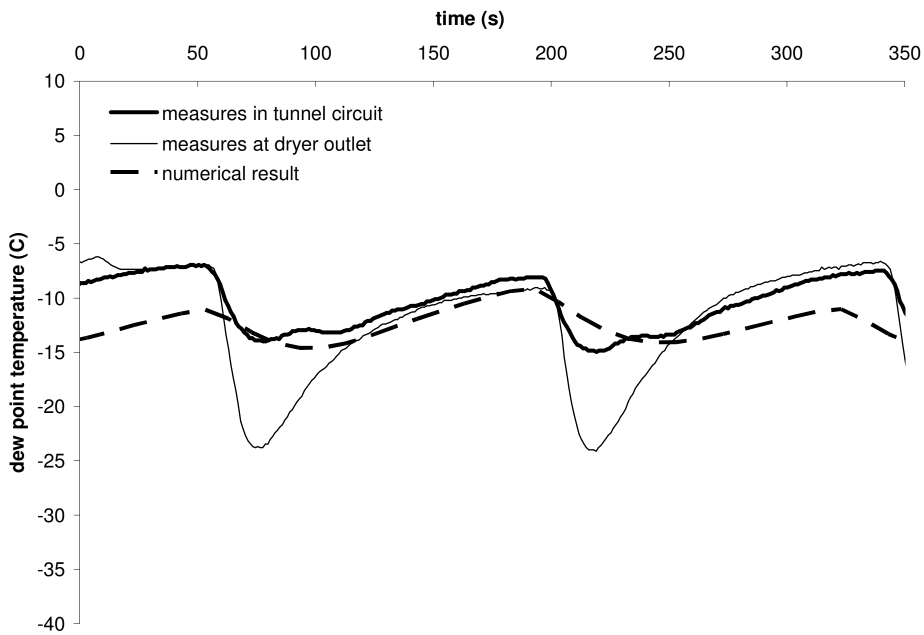

Figure 9. Comparison, for a determined test condition, of dew point temperature measurements done into the tunnel circuit, at the dryer outlet and the numerical calculation results.

4. CONCLUSION

A mathematical model was implemented to obtain the transient results of the pressure control system of the TTP, executing the humidity control function. These mathematical calculation results were compared with the experimental results performed in the TTP with the pressure control system operating.

An operational envelope of the dryer was obtained and implemented in the mathematical code. The results of the global numerical calculation were used to compare with the experience.

Some important aspects about the dryer functions were tested. Among them it was possible to verify the dryness capacity dependence with the compression ratio applied to the dryer. Based on these results it is possible to determine the right compression ratio to be selected for determine Mach number at the test section.

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