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Model Based Control of Intake Air Temperature and Humidity on the Test Bench

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Abstract

Engine test benches are crucial instruments to perform tests on internal combustion engines. Possible purposes of these tests are to detect the engine performance, check the reliability of the components or make a proper calibration of engine control systems managing the actuations. Since many factors affect tests results in terms of performance, emissions and components durability, an engine test bench is equipped with several conditioning systems (oil, water and air temperature, air humidity, etc.).
One of the most important systems is the HVAC (Heating, Ventilating and Air Conditioning), that is essential to control the conditions of the intake air. Intake air temperature, pressure and humidity should be controllable test parameters, because they play a key role on the combustion development. In fact, they can heavily affect the performance detected, such as power and specific consumption, and, in some cases, they may promote knock occurrence.
This work presents an HVAC model-based control methodology, where each component of the air treatment system (humidifier, pre-heating and post-heating resistors, chiller and fan) is managed coupling open-loop and closed-loop controls. Each branch of the control model is composed of two parts, the first one to evaluate the target for the given HVAC component, based on the system physical model, the second one is a PID controller based on the difference between the set-point and the feedback values.
The control methodology has been validated on an engine test bench where the automation system has been developed on an open software Real-Time compatible platform, allowing the integration of the HVAC control with all other functionalities concerning the test management. The paper shows the plant layout, details the control strategy and finally analyzes experimental results obtained on the test bench, highlighting the benefits of the proposed HVAC management approach.

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Keywords: HVAC; air conditioning

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1. Introduction

Nowadays in the automotive sector, complex solutions are studied and adopted to meet the need for high performance engines offering good reliability and, due to increasingly stringent anti-pollution standards, low emissions. A meticulous study is devoted to the development of mechanical components and innovative technical solutions. Moreover, engineers extensively use electronics to make 'intelligent' engines, capable of adapting to the requirements of the driver to offer the best performance in any condition.

A key part of development is the test activity [1] that is carried out to validate the adopted solutions, and a tool playing a key role during this process is the engine test bench. In fact, it allows performing different types of tests, producing large amounts of data in a very short time, thus contributing to the optimal setup, the approval or the rejection of the considered solutions [2].

The behavior of an engine can be influenced by many factors, and some of them can dramatically change the outcome of a test in terms of performance, emissions, and reliability. Therefore, it is necessary to monitor and accurately control, through appropriate systems, each of these parameters in order to carry out focused and repeatable tests. Fuel and air conditions obviously affect the combustion process: in particular, the conditions of the intake air can deeply affect the way the combustion develops, leading the engine to generate different performance according to air pressure, temperature and humidity [3].

Usually, engine test benches are equipped with an HVAC system to control the thermal conditions of the air introduced into the engine. It is a set of various components that need to be simultaneously controlled with the aim of conditioning air temperature and humidity (pressure conditioning is usually carried out with dedicated systems).

While in the literature there are many examples of works focusing on engine thermal management [4, 5, 6], model-based control of HVAC systems has not been previously addressed.

This paper introduces a model based control methodology of the HVAC system. To manage individual components of this plant, an algorithm putting together the outcomes of open loop and closed loop contributions has been designed. The open loop control is based on a physical heat exchange model between the intake air and the specific HVAC element to be controlled, while the closed loop part is based on the difference between target and measured air conditions.

The developed control model has been integrated into the test bench management software implementing the test bench automation system, running on a Real-Time platform. In this way, it has been possible to validate the HVAC system control methodology by performing air conditioning tests, feeding the model inputs with the test bench sensors outputs.

The paper is subdivided into sections. The first one describes the HVAC system component by component and briefly shows the hardware set-up of the test bench automation system where tests have been performed. In the second section, after a brief introduction to the test management software, the algorithm structure is presented, highlighting its components, leading from the inputs to the actuations commands driving the air conditioning system hardware. Finally, the last section reports the results of the HVAC system control assessment tests.

Nomenclature

T_{TGT}	[°C]	target air temperature at the engine intake
$HUrel_{TGT}$	[°C]	target air relative humidity at the engine intake
$HUass_{TGT}$	[g/kg]	target air absolute humidity at the engine intake
T_{ENV}	[°C]	temperature of the air introduced in the HVAC system
$HUrel_{ENV}$	[%]	relative humidity of the air introduced in the HVAC system
$HUass_{ENV}$	[g/kg]	absolute humidity of the air introduced in the HVAC system
T_{INT}	[°C]	temperature of the air at the engine intake
$HUrel_{INT}$	[%]	relative humidity of the air at the engine intake
$HUass_{INT}$	[g/kg]	absolute humidity of the air at the engine intake
P_{BARO}	[Pa]	barometric pressure
$\dot{m}$	[kg/s]	air mass flow rate
T_{DP}	[°C]	dew point temperature
H_{OL}	[W]	heat to transfer to the air calculated by the open loop control

H_{CL}	[W]	heat to transfer to the air calculated by the closed loop control
H_{deh}	[W]	heat to subtract to the air to dehumidify
Cp	[J/kgK]	air specific heat
ΔT	[°C]	temperature difference
C_c	[W]	chiller cooling capacity
SG_{OL}	[%]	steam generator open loop control output
SG_{CL}	[%]	steam generator closed loop control output
SG	[%]	duty cycle of the steam generator
$CH1_{OL}\%$	[%]	chiller open loop control output produced by the humidity control
$CH_{CL}\%$	[%]	chiller closed loop control output produced by the humidity control
$CH1\%$	[%]	duty cycle of the chiller produced by the humidity control
$CH2\%$	[%]	duty cycle of the chiller produced by the temperature control
$CH\%$	[%]	duty cycle of the chiller, the maximum between $CH1\%$ and $CH2\%$
$WRpostH$	[W]	post-heating resistance wattage
$WRpreH$	[W]	pre-heating resistance wattage
RpostH%	[%]	duty cycle post-heating resistance
RpreH%	[%]	duty cycle pre-heating resistance

2. HVAC system and hardware setup

The work presented in this paper has been developed and validated performing air conditioning tests on an engine test bench. In the following, the structures of the air conditioning system and the control hardware setup are presented.

2.1 HVAC plant layout

The HVAC system that has been tested is located in a technical room near the test bench room.

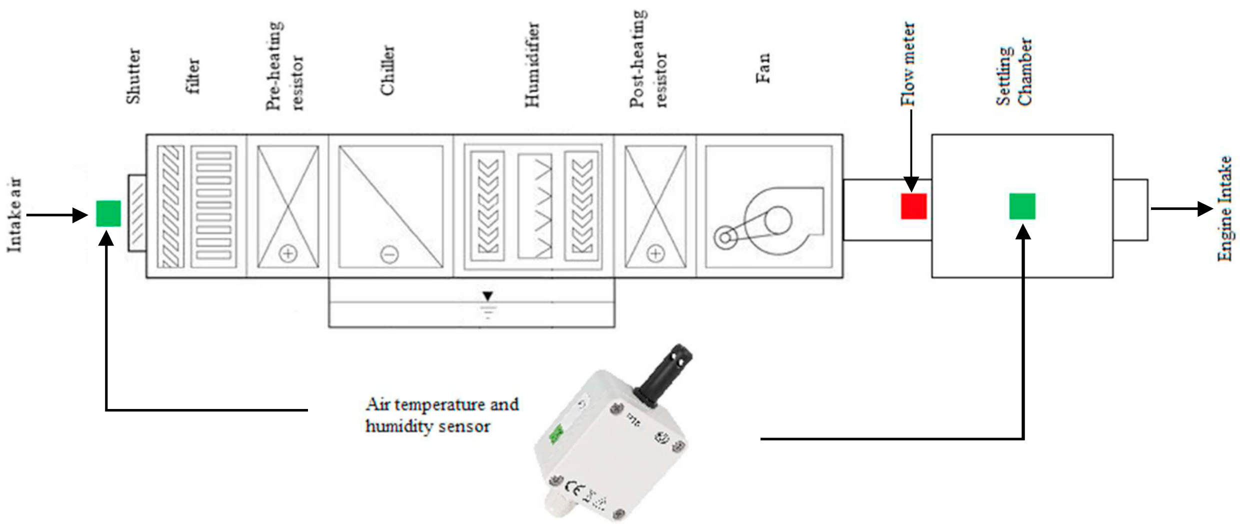

Fig. 1. Scheme of the HVAC system and sensors layout

As shown in Figure 1, starting from the air intake of the HVAC, a shutter permits to regulate the intake section establishing the air flow rate processed by the system; next, a filter avoids impurities from being introduced inside. Then a 12 kW pre-heating resistor allows increasing the air temperature. A maximum temperature thermostat switches the resistor off when a selectable threshold is exceeded.

The resistor is followed by a heat exchanger, where air is cooled by the liquid refrigerated by a dedicated chiller. If processed air exceeds the dew point temperature, airborne water vapor condenses, reducing the air absolute humidity. The producer of the chiller provides a table to estimate the cooling capacity, based on the treated air temperature and the coolant fluid temperature.
The air cooler is followed by a humidifier, a steam generator whose function is to increase air humidity introducing water vapour in the conditioned air through a nebulizer. This component can produce up to 26.4 kg/h of steam. Condensed water is collected in a tank and evacuated outside the technical room.
A 18 kW post-heating resistor performs the last heating process and finally a fan pushes the conditioned air towards the engine intake.

As regards the model outputs for the management of the HVAC components, the humidifier needs an analogue signal proportional to the steam flow and a digital enable command. Resistors and chiller are controlled by means of PWM signals, managing the percentage of the control period spent by the systems in ON or OFF states.

The HVAC control algorithm uses as input variables physical quantities available to the test bench automation system, specifically the HVAC and engine input air conditions, in terms of humidity and temperature, together with the atmospheric pressure and the air mass flow rate.

As shown in figure 1, two air humidity and temperature sensors are used, the first one outside of the HVAC system, the second one placed inside the settling chamber attached to the engine intake. The barometric pressure sensor is installed in the test bench room while the flow meter is installed downstream of the air conditioning system (before the settling chamber).

2.2 Hardware Setup

The engine test bench control system managing the HVAC and all other devices, is based on a Real-Time compatible hardware platform, whose scheme is shown in Figure 2. The hardware setup consists of a Real-Time controller, a high-performance desktop PC running Labview Real-Time OS, connected via MXI Express BUS interface with an NI MXI Chassis containing the I/O Modules. Finally, the Host PC, connected via Ethernet with the Real-Time controller, offers a user interface, allowing the operator to manage and monitor the tests.

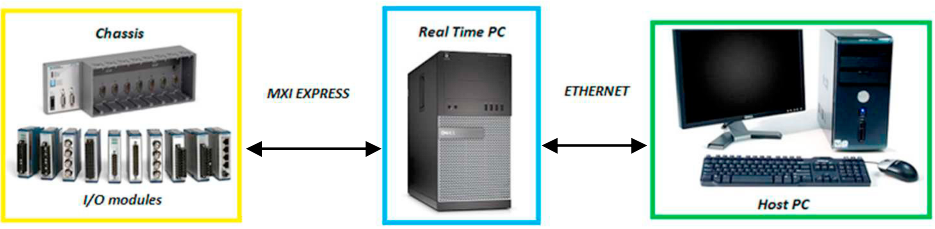

Fig. 2. Control system structure

3. The HVAC managing algorithm

The engine test bench control software has been developed in NI Veristand, a software environment for developing Real-Time testing applications that can run on Real-Time hardware. NI Veristand offers the possibility to include models, such as the HVAC control logic, developed in NI Labview and running on the Real-Time controller. The NI Veristand Project incorporates the hardware I/O functionalities, the control models, the user interface and automation sequences. HVAC systems are usually managed by means of PID controllers implemented on PLCs that manage single components, such as the steam generator, or the chiller: the proposed control methodology benefits of an open loop model-based contribution, managing the HVAC system components with a global approach.

Figure 3 shows a schematic representation of the HVAC control model, highlighting the components I/O.

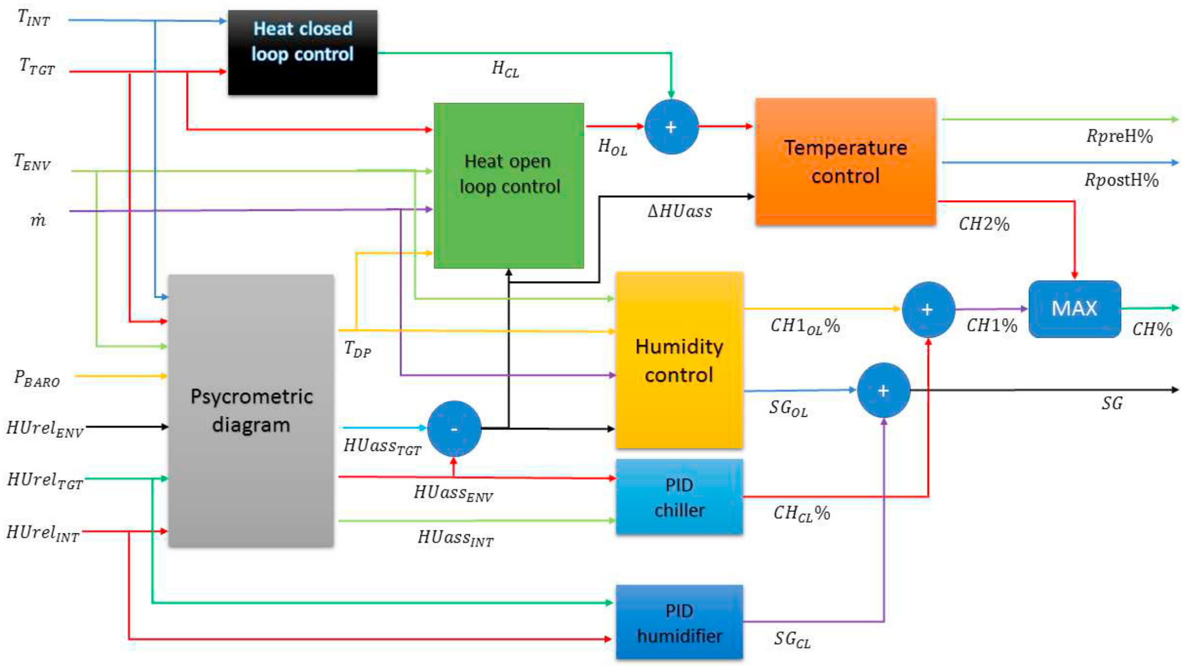

Fig. 3. Inputs and outputs of the HVAC control model

As previously remarked, each output of the HVAC control model presented in this paper is composed of 2 terms, an open loop and a closed loop component. The parameters of the PID controllers have been set according to the slowness of the controlled phenomena. A step variation of the demands could quickly lead PIDs to saturation. For this reason, the derivative of the targets is saturated with an appropriate rate limiter.

The open loop calculus is based on the estimation of the heat and water vapor that the air conditioning system has to transfer to (or from) the air processed by the plant. To do this, the model uses the formulation of the heat transfer, shown in Eq.(1), where ΔT depends on the specific case.

$$H_{OL} = \dot{m} * Cp * \Delta T \quad (1)$$

In order to assess the quantity of water vapor to introduce or remove from the inlet air, the absolute humidity of the air introduced in the plant is compared with the target, evaluated from target air temperature and relative humidity required by the operator. The absolute humidity, expressed in terms of grams of water per kg of dry air, is derived from relative humidity, temperature and barometric pressure through the psychrometric diagram.

The humidity management logic provides two possible paths:

- If absolute air humidity has to be increased, the steam generator is activated. Using Eq.(2), the model calculates the amount of steam to produce in terms of percentage of the humidifier's nominal capacity, that is 26.4 kg/h of steam flow, corresponding to an output signal of 10 V.

$$SG_{OL} = \frac{(\dot{m}*3600)*(HUass_{TGT} - HUass_{ENV})}{1000*26.4} * 0.1 \quad (2)$$

This open loop contribution is added to the output of the humidifier PID controller that compares the absolute humidity target with the absolute humidity of the engine intake (based on the relative humidity and temperature inputs).

The sum result, conveniently saturated between specific limits is the Voltage output managing the humidifier.

$$SG = SG_{OL} + SG_{CL} \quad (3)$$

It has to be noticed that the desaturation of the integral term in the PID controller has to take into account the effect of the open loop contribution [7, 8]. Therefore, the output of the PID controller will be as follows, being SAT the PID authority (i.e., the maximum allowed contribution of the closed loop controller). The parameter SAT has an important impact on the wind-up behavior of the PID controller.

$$\begin{cases} [-SAT;\ 10 - SG_{OL}]\ if\ SG_{OL} \geq 10 - SAT \\ [-SAT;\ SAT]\ if\ 10 - SAT \geq\ SG_{OL} \geq SAT \\ [-SG_{OL};\ SAT]\ if\ SG_{OL} < SAT \end{cases} \tag{4}$$

- If absolute air humidity has to be decreased, the chiller is activated to decrease air temperature until it reaches the dew point and the airborne water vapor condenses decreasing air humidity. First of all it is necessary to determine the dew point through the psychrometric diagram, considering the absolute humidity target and a relative humidity of 100 %. Then, Eq.(1) is applied, where in this case ΔT is the difference between T_{ENV} and T_{DP}: this allows determining the heat H_{deh} to subtract from the airflow in order to reach the target absolute humidity. Finally, once determined the chiller cooling capacity for the measured T_{ENV}, the model output managing the chiller is calculated through Eq.(5).

$$CH1_{OL}\% = \frac{H_{deh}}{Cc} * 100 \tag{5}$$

This term is added to the output of the chiller PID controller that compares the relative humidity target with the relative humidity of the engine intake. The result, saturated between proper limits is the duty cycle of the chiller PWM for the humidity control. As stated previously, saturation limits must take into account the presence of the open loop contribution.

$$CH1\% = \ CH1_{OL}\% + CH_{CL}\% \tag{6}$$

As regards the temperature management logic, the closed loop control consists of a PID controller that compares T_{TGT} with T_{INT} and produce a contribution, in terms of heat, named H_{CL}. This term is added to the heat H_{OL} calculated in open loop. Three are the possible cases that may occur:

- Air temperature must be reduced. This case involves the activation of the chiller, while both the resistors remain deactivated. $CH2\%$, that is the PWM duty cycle to manage the Chiller in the temperature control, can be estimated through Eq.(7), where H_{OL} is calculated through Eq.(1) with a ΔT that is the difference between T_{TGT} and T_{ENV}.
  Both humidity and temperature control produce a duty cycle for the chiller PWM, the higher among these is used to manage the device.

$$CH2\% = \frac{H_{OL}+H_{CL}}{Cc} * 100 \tag{7}$$

- Air temperature must be increased, while humidity must be reduced. In this case, the humidity control activates the chiller and the temperature control switches on the post-heating resistance. In fact, as previously explained, to decrease the amount of water in the air, it is essential to bring the air at a temperature below the dew point. So, the best way to increase the air temperature at the same time the chiller is working is to turn on the post-heating resistance which is placed downstream of the chiller [9]. For this reason, in this specific case, the amount of heat to transfer to the air is calculated in open loop through Eq.(1) where ΔT is the difference between T_{TGT} and T_{DP}. Eq.(8) shows how to calculate the duty cycle of the resistor PWM for this case.

$$R\text{postH}\% = \frac{H_{OL}+H_{CL}}{WRpostH} * 100 \tag{8}$$

- Air temperature must be increased, and humidity must be increased. In this case, since there is a need to increase the presence of water vapor in the air, the model activates the steam generator and the pre-heating resistance. The amount of heat to transfer to the air is calculated in open loop through Eq.(1) where ΔT is the difference between T_{TGT} and T_{ENV}. Eq.(9) shows how to calculate the duty cycle for the resistance PWM in this case.

$$RpreH\% = \frac{H_{OL}+H_{CL}}{WRpreH} * 100 \quad (9)$$

If the heat quantity is so high that it exceeds the pre-heating resistor wattage, also the post-heating resistor is switched on, with a duty cycle established with Eq.(10)

$$RpostH\% = \frac{H_{OL}+H_{CL}-WRpreH}{WRpostH} * 100 \quad (10)$$

4. Tests Results

Two different types of test have been carried out. First of all, the HVAC has been tested without engine on the bench and at fixed air flow, generating sudden variations of target temperature or humidity.
The HVAC system reaction to a sudden change of the target temperature (from 25°C to 30°C), at a fixed demand of humidity of 50%, is shown in Figure 4 a): although the reaction is slow, the temperature variation is obtained without critical oscillations in relative humidity, which is in the range 50±2%. The reason of the slow response is to be ascribed to several factors: on one hand, the target derivative is saturated, and the actual target is reached after more than 330 seconds. On the other hand, the resistors switching period is very long (1 minute). Finally, temperature changes are damped out by the presence of the settling chamber, which means that the plant reactions are very slow. This is a disadvantage when air conditions must be changed abruptly, but it is useful to maintain air conditions during engine speed and load transients. It has to be remarked that typically, once target air conditions are reached, they are maintained through the test duration.

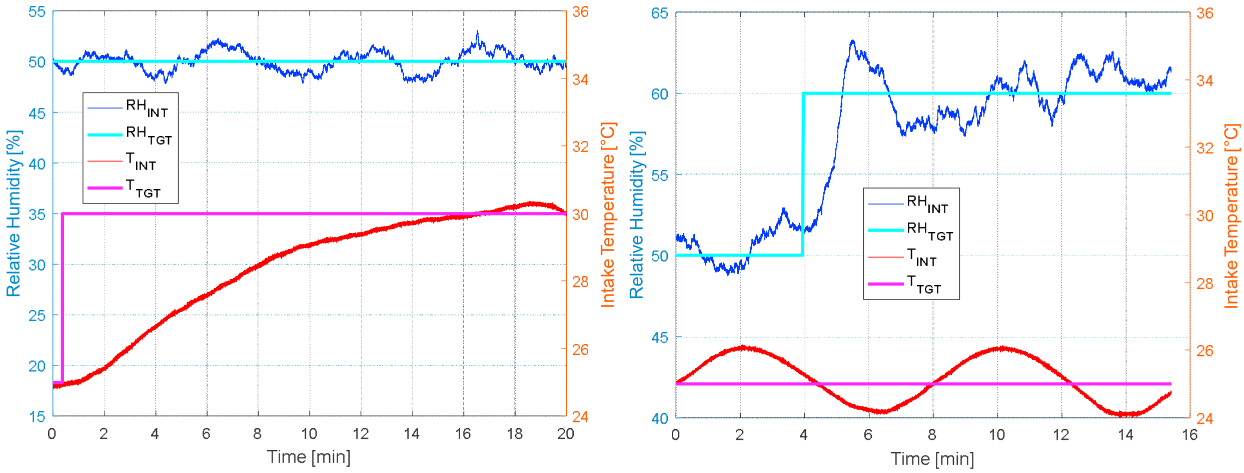

Fig. 4. a) change of target temperature at fixed humidity and air flow rate; b) change of target humidity at fixed temperature and air flow rate

Figure 4 b) shows the reaction to the variation of target humidity (from 50% to 60%), at a requested temperature of 25 °C: similar considerations can be applied. It can be noticed that air temperature oscillates in the range ±1°C, due to the PID constant calibration. This setting, obtained with a fine tuning following the application of Ziegler-Nichols approach, could be considered improvable in steady flow conditions, but it has been considered the best-compromise solution taking into consideration also unsteady flow operation, typical of engine testing. The overall controller stability has been proved against the most extreme working conditions (maximum difference of target Temperature and Humidity with respect to feedback values).
Figure 5 shows that even in tests with highly variable flow rate and high target temperature humidity remains near the target, with a maximum variation of 3.5% and a standard deviation of 1.45%. The test has been performed running an engine on the test bench, alternating full throttle steps at different engine speeds to low throttle conditions at low engine speed.

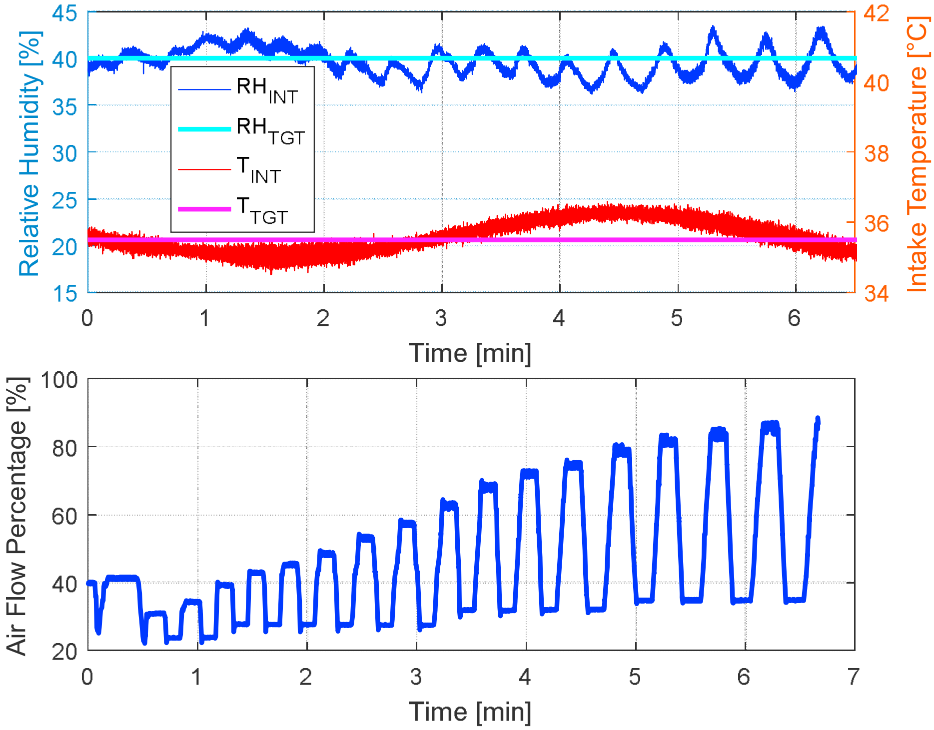

Fig. 5. Intake air humidity and temperature during a test with highly variable flow rate

5. Conclusions

The paper describes the implementation of an HVAC system control: each component control setting is determined according to the combination of an open loop model-based contribution, and a closed loop PID contribution. The control system has been implemented in the real-time test automation system controller, and tested with and without the engine. Even though the control of the HVAC system could take advantage of some hardware modifications, such as the possibility of controlling the chiller fluid temperature, or the increase in the resistors switching frequency, the approach demonstrated a good control capability, allowing engine testing in different steady temperature and humidity intake air conditions, even with very unsteady air flow.

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