HEFAT2014
10th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
14 – 26 July 2014
Orlando, Florida

APPLICATION OF PLATE HEAT EXCHANGERS TO THE DEVELOPMENT OF PORTABLE HIGH-PRESSURE HUMID GAS GENERATORS

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ABSTRACT

The preliminary design and testing of a portable, high-pressure humidity generator based on a corrugated-plate heat exchanger (PHE) is described. It resulted in a very simple system, consisting of a bubbling pre-saturator followed by the PHE used as a condenser. The PHE does not need to be hosted in a liquid bath, resulting in a very compact design whose operation is largely independent from a specific bath - which may be available on site – used to condition the temperature of a circulating counter-flow fluid.
Experimental tests were performed on a prototype in order to demonstrate the principle and validate at an early stage the system operation. The efficiency of the PHE as condenser was assessed over a temperature range from 0 °C to 65 °C by varying input and influence parameters, such as the dew point temperature, the gas flow rate and its pressure up to 0.5 MPa. The saturation temperature of the humid gas flow compared favourably with the dew-point temperature as measured by a chilled-mirror hygrometer calibrated against a primary humidity standard. The temperature deviations were below 0.02 °C, with a maximum deviation of 0.04 °C, over the whole range of investigated temperatures and pressures.

INTRODUCTION

The INRIM humidity standard are based on two primary humidity generators to cover a dew/frost point temperature range from -75 °C to + 85 °C and a relative humidity range from 2 %rh to 95 %rh [1-3]. The generators were designed to operate at, or slightly above, the atmospheric pressure and are routinely used for laboratory calibrations of precision dew-point and relative humidity hygrometers, thus ensuring the measurement traceability to humidity and related quantities to the scientific and industrial users.
Because of their construction and size, they are not suitable whenever an industrial application would require instrument calibration at the point of use, either at atmospheric pressure or at higher pressures.
In order to achieve this goal, the design and development of a portable humidity generator targeted to have performance comparable with a laboratory system, at least in limited dew-point temperature range, was undertook. The preliminary results are discussed in this work.

NOMENCLATURE

A: total PHE heat transfer area [m^2]
$c_{p,h}$: hot fluid constant pressure specific heat [J/kg K]
$c_{p,c}$: cold fluid constant pressure specific heat [J/kg K]
$\dot{m}_c$: mass flow rate of cold fluid [kg/s]
$\dot{m}_h$: mass flow rate of hot fluid [kg/s]
L: PHE plate length [m]
N_p : number of plates
$\dot{Q}$: exchanged heat power [W]
$\dot{Q}_c$: heat power exchanged on the cold side [W]
$\dot{Q}_h$: heat power exchanged on the hot side [W]
T_{bath}: thermostatic bath temperature [°C]
$T_{dp,inlet}$: pre-sat dew-point temperature of PHE inlet gas [°C]
T_{CMH1}: dew-point temperature detected by CMH1 [°C]
T_{CHM2}: dew-point temperature detected by CMH2 [°C]
T_{sat}: saturator temperature [°C]
$u(T_{CMH}-T_{sat})$: combined standard uncertainty of the difference between CMH reading and saturator temperature [°C]
U_D: overall heat transfer coefficient [W/m^2 K]
W: PHE plate width [m]
ΔH_{vl}: latent heat of vaporization [J/kg]
Δp_1: pressure difference between PHE inlet and saturator outlet [Pa]
Δp_2: pressure difference between saturator outlet and CMH inlet [Pa]
Δq: specific humidity difference between PHE inlet and outlet[kg/kg]
ΔT_c: water temperature difference between PHE outlet and inlet [°C]
ΔT_h: hot (condensing) gas temperature difference between PHE outlet and inlet [°C]
ΔT_{ml}: logarithmic mean temperature difference [°C]
Φ: volumetric humid gas flow rate [l/min]

EXPERIMENTAL APPARATUS AND PROCEDURES

Characteristics of a Plate Heat Exchanger

Heat exchangers are, in general, system components used to promote the heat transfer between two non-mixing fluids at different initial temperatures. Depending on the flow pattern inside a heat exchanger, the final thermal states can be as follows:

1) if the fluids flow along the same direction (co-current), they tend to attain the same equilibrium temperature, which is somewhere between the initial temperatures of the two fluids;
2) if the fluids flow counter-current, then the "cold" fluid (i.e., that whose temperature decreases during the passage into the heat exchanger) can exit from heat exchanger with a temperature higher than that of the "hot" fluid (i.e., that whose temperature increases during the process).

A key parameter for selecting a heat exchanger is the so-called "transfer area" which is related to the heat transfer and others process variables by means of the following equation:

$$A = \frac{\dot{Q}}{U_D \times \Delta T_{ml}} \tag{1}$$

where ΔT_{ml} is a temperature that represents the thermal behaviour of the fluids during their passage into the heat exchanger and whose estimation depends on the fluids flow pattern (co-current or counter-current).

Among several different types of heat exchangers, for our application we focused on the corrugated-plate heat exchanger (PHE) type. It consists of a number of parallel plates, suitably patterned in order to increase the transfer area and the flow turbulence and welded each other in order to make leak-free between micro-channels and therefore preventing any physical contact and/or mixing between the two fluids. The two fluids alternately flow within each pair of plates, so that a single plate is in contact with a "hot" fluid on one side and with a "cold" fluid on the other side; each plate pair has four passageways at the corners, to provide suitable flow channels for the fluids into the PHE.

Figure 1 shows the schematic design of the commercial PHE model used in our experiments [4]. It is a 10-plate heat exchanger, where five gas flow channels are intermixed with five liquid flow channels. The manufacturer specified a temperature range of operation from about -180 °C to 220 °C at a maximum pressure of 2.3 MPa, well exceeding the design operation range of the humid gas generator. The PHE actual heat transfer area, as calculated from [5], resulted:

$$A = N_p \times L \times W = 0{,}134\ m^2 \tag{2}$$

The maximum heat power exchanged on the PHE "cold side", was experimental evaluated from the difference between the PHE outlet and inlet temperatures (ΔT_c) as follows [6]:

$$\dot{Q}_c = \dot{m}_c \times c_{p,c} \times \Delta T_c \tag{3}$$

where $\dot{m}_c$ is the cold fluid mass flowrate and $c_{p,c}$ is its specific heat at constant pressure. Water was used as the cooling fluid at a constant volumetric flow rate of 10 l/min.

Assuming an adiabatic system (i.e., a negligible heat loss toward the surrounding environment) the heat power on the "cold side" would be equal to that on the "hot side", which is estimated as follows:

$$\dot{Q}_h = \dot{m}_h \times c_{p,h} \times \Delta T_h + \dot{m}_h \times \Delta H_{vl} \times \Delta q \tag{4}$$

where the second term (latent heat) takes into account the hot fluid condensation. In the current experiment, the maximum heat power exchanged between the fluids, estimated according to eq. (4), was about 60 W.

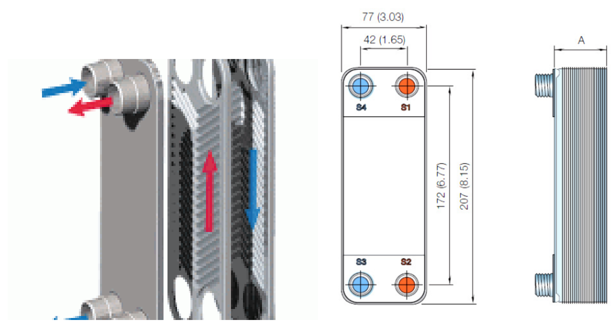

Fig.1 Internal design of the PHE with counter-current fluid flow between the plates (left). Size and port location of the commercial 10-plate PHE used in this work (right).

Description of the Experiments

The high-pressure humidity generator was designed as a condensing-type system where a humid gas stream flows through the PHE "cold side" and is brought to saturation at a given temperature T_{sat} by condensing the excess water vapour of the humid mixture. To this end, a dry gas is firstly pre-saturated at a temperature slightly higher than T_{sat} by a suitable pre-saturator, then the humid gas stream – flowing inside the PHE micro-channels - exchanges sensible and latent heat with a separate counter-current cooling flow, at a constant flow rate, whose temperature T_{bath} ($\approx T_{sat}$) is set by a thermostatic circulation bath.

At the PHE "cold side" outlet, the condensed phase is collected in a small cylindrical reservoir, where the humid gas flows before leaving the system. The small reservoir operates as an equaliser/final saturator to compensate for any small dew-point temperature drop caused by the flow pressure drop into the PHE. Such a small equaliser/final saturator together with the PHE were kept in a small external reservoir. The cooling water is made to flow in a series circuit to feed the PHE first and the water reservoir thereafter. The length of the equaliser/final saturator path was chosen to allow the humid gas to reach a full saturation at any T_{sat} in the dew-point temperature range of operation[7].

Figure 2 shows a sketch of the experimental apparatus set up to validate the design principle and to carry out initial testing of the humid gas generator.

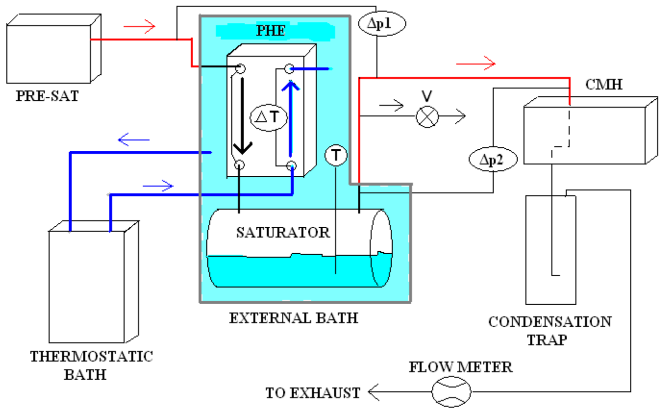

Fig. 2 Set-up of the experimental apparatus used to validate the principle and to test the humid gas generator. (CMH = chilled-mirror hygrometer; PHE = plate heat exchanger; ΔT = differential thermocouple; T = Pt-100 thermometer; Δp1, Δp2 = differential pressure gauges).

The initial tests involved the use of a commercial humidity generator system (termed PRE-SAT in Fig. 2) to supply a pre-saturated gas stream to the PHE system over a wide range of conditions, i.e., by varying the inlet dew-point temperature $T_{dp,inlet}$ between 1 °C and 65 °C and the gas flow rate from 2 l/min to 20 l/min. Of course, in the final design the humidity pre-saturator is expected to be replaced by a compact bubbling pre-saturator. The PHE outlet gas stream was delivered to a chilled-mirror hygrometer CMH (Fig. 1) by means of a heated hose. A by-pass needle valve (V) was used to insure a constant volumetric gas flow rate of 0.5 l/min to the CMH, irrespective of the actual gas flow rate flowing through the PHE system. During all experiments the final saturator temperature was measured by means of a calibrated metal-sheathed platinum resistance thermometer (PRT) located near the humid gas outlet in contact with the shallow water layer in the saturator. A compression fitting feed-through insured the pressure integrity to the system. A differential thermocouple was used to detect the temperature difference ΔT_c between the PHE outlet and inlet cooling fluid flow. A differential capacitance pressure gauge (1333 Pa f.s.) detected the overall flow pressure drop across the system.

Experiments

One of the main concerns in using a PHE as a condenser in a humidity generator comes from the micro-channel structure of the plates, with small passageways for the gas flow, which may cause a comparatively large pressure drop and consequently unacceptable a dew-point temperature error at the saturator outlet.
An initial test was carried out to detect such a pressure drop (Δp_1) as a function of the volumetric gas flow rate - at a constant saturator temperature T_{sat} set to 5 °C – with increasing condensation heat load, simulated by progressively increasing the pre-saturator temperature from 5 °C to 60 °C (Figure 3).

As expected, the maximum pressure drop was found at the highest volumetric flow rate (10 l/min) corresponding to the largest heat load (~60 W) which, for a condenser, also corresponds to the highest flow friction in the micro-channels due to the increased water condensation layer on the plates. The worst-case experimental pressure drop was 440 Pa (at 10 l/min gas flow rate with a pre-saturator temperature 55 °C higher than the T_{sat}). The above pressure drop would correspond to a dew-point temperature error of -0.06 °C.
It should be pointed out that in a more realistic operating condition, the expected pressure drop is below 160 Pa corresponding to a dew-point temperature error of only -0.02 °C.

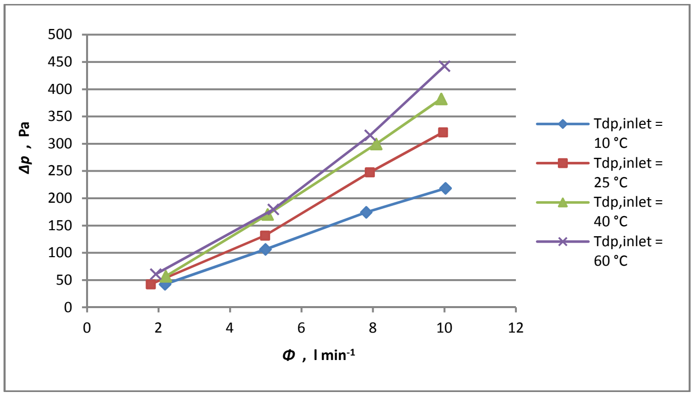

Fig. 3 Flow pressure drop through the PHE system, operating at atmospheric pressure, as a function of the gas flow rate and the condensation heat load (at increasing inlet pre-sat temperature $T_{dp,inlet}$) at constant saturator temperature T_{sat} = 5 °C.

To fully characterise the PHE-based humid gas generator, the design of the experiment involved three different sets of testing. Typical and extreme operating conditions were explored as follows:

Test 1) PHE in vertical position with the following conditions:

a. Saturator temperatures ($T_{bath} \approx T_{sat}$) set to 1 °C, 15 °C, 25 °C, 40 °C, and 60 °C;
b. Pre-saturator (inlet dew-point) temperature, $T_{dp,inlet} = T_{sat} + 5$ °C;
c. Gas flow rates at the PHE inlet set to Φ = 2 l/min, 4 l/min, 8 l/min, and 12 l/min;

Test 2) PHE in vertical position with:

a. Constant saturator temperatures T_{sat} = 5 °C;
b. Pre-saturator (inlet dew-point) temperature, $T_{dp,inlet}$ = 0 °C, 10 °C, 30 °C, and 60 °C;
c. Gas flow rates Φ = 2 l/min, 4 l/min, 8 l/min, and 12 l/min;

Test 3) PHE in horizontal position with:

a. Saturator temperatures T_{sat} = 1 °C, 25 °C, 40 °C, and 60°C;
b. Pre-saturator (inlet dew-point) temperature, $T_{dp,inlet} = T_{sat}$ + 5 °C (see test 1b. above);
c. Gas flow rates Φ = 2 l/min, 4 l/min.

Test 1 was designed to assess the overall system efficiency , i.e. the capability to fully saturate the carrier gas in a range of conditions similar to those expected for an optimized use of the humid gas generator (i.e., $T_{dp,inlet} = T_{sat}$ + 5 °C) at increasing flow rates.
Test 2 was useful to evaluate the full efficiency of the PHE as a condenser, i.e. its ability to work with increasing heat loads and with a progressively ticker condensing layer on the plates. To assess such performance index the dew-point temperatures of the inlet gas were increased up to T_{sat} + 55 °C, which correspond to a maximum condensing mass flow rate of water of about 90 g/h.
Test 3 was similar to Test 1, except it was carried out with the PHE in a horizontal position. The main goal of the test was to confirm the system efficiency in view of the engineered design of the generator, as the PHE horizontal operation would allow a more compact layout and a better portability.

The efficiency tests were all based on a comparison between the local saturator temperature (T_{sat}), as measured by the Pt-100 PRT, and the dew-point temperature as measured by a chilled-mirror hygrometer (T_{CMH}) calibrated against the INRIM primary humidity standard. The dew-point temperature readings were corrected for the pressure drop (Δp_2) from the generator outlet to the point of application. The agreement between T_{sat} and T_{CMH} - to within the measurement uncertainty - demonstrates the ability of the system to fully saturate the gas at the given temperature.

Preliminary Testing at Pressure

A preliminary testing of the generator at pressure was carried out with pressurized nitrogen up to 0.5 MPa with the saturator temperature ranging from 5 °C to 60 °C. With respect to the set-up depicted in Fig. 2, the following changes were made to the experimental apparatus:

- the commercial humidity generator PRE-SAT was replaced by a pressure- and temperature-controlled bubbling pre-saturator equipped with a cartridge heater and a PID controller;
- the CMH1 was replaced by a chilled-mirror dew-point hygrometer (CMH2) able to operate at pressures up to 2 MPa;
- a double-diaphragm forward pressure regulator was inserted between the inlet gas from a high-pressure cylinder and the pre-saturator in order to insure a stable pressure operation above atmospheric pressure;
- an absolute pressure transducer (MKS Baratron, 1.33 MPa f.s.) connected to the pre-saturator inlet was used to monitor the system pressure.

The tests at pressure mirrored the Test 1 above, except for the gas flow rate which was kept constant at 2 l/min.

Uncertainty Estimate

As discussed, in the experiments the measurand was the quantity $T_{CMH} - T_{sat}$ which summarizes the comparison results between the saturation state of the generated humid gas flow and the actual dew-point temperature, detected by a hygrometer traceable to humidity standards.
In order to evaluate the combined standard uncertainty (k=1) associated to the measurand, the input components that were accounted for and estimated are:

- the calibration uncertainty of the PRT employed for measuring T_{sat};
- the worst-case temperature measurement repeatability of T_{sat};
- the calibration uncertainty of both CMHs;
- the worst-case dew-point temperature measurement repeatability of T_{CMH} for both hygrometers.

The combined standard uncertainty was estimated by assuming statistical independence of the temperatures and, consequently, adding their variances:

$$u(T_{CMH} - T_{sat}) = \sqrt{u(T_{CMH})^2 + u(T_{sat})^2} \tag{5}$$

where each variance does include the calibration uncertainty and the measurement repeatability for the relevant CMH.

RESULTS AND DISCUSSION

Table 1 summarizes the set of results as obtained from Test 1. The saturator temperature and the difference with respect to chilled-mirror hygrometer reading is reported, together with the measurand standard uncertainty (k=1), as a function of the temperature and the volumetric flow rate.

The saturator temperature compared favourably with the dew-point temperature as detected by chilled-mirror hygrometer in the whole range of operation. It is worth noting the temperature difference never exceeded its expanded uncertainty (coverage factor k=2) which for the present experiments was estimated to be between 0.05 °C and 0.06 °C.
These results demonstrated the ability of the system to bring the gas to a complete saturation in the whole range of dew-point temperatures and humid gas flow rates.

Φ (l/min)	$T_{dp,inlet}$ (°C)	T_{sat} (°C)	$T_{CMH} - T_{sat}$ (°C)	$u(T_{CMH} - T_{sat})$ (°C)
1.95	4.990	1.091	0.024	0.024
3.92	4.981	1.089	0.015	0.024
7.91	4.970	1.087	0.008	0.024
11.63	5.021	1.100	-0.003	0.024
1.92	10.010	5.111	-0.010	0.024
4.12	10.010	5.110	-0.007	0.024
8.09	10.000	5.117	-0.014	0.024
11.89	9.990	5.119	-0.011	0.024
1.93	20.008	15.040	0.020	0.025
4.11	20.016	15.038	0.016	0.025
8.18	19.999	15.037	0.008	0.025
11.84	20.011	15.038	-0.002	0.025
1.95	30.008	25.041	0.025	0.025
3.92	30.006	25.036	0.042	0.025
8.18	30.035	25.043	0.034	0.025
11.80	30.023	25.041	0.000	0.025
1.99	44.990	40.007	-0.028	0.027
4.12	45.003	40.013	0.001	0.027
8.01	45.010	40.013	0.001	0.027
12.02	44.988	40.013	-0.020	0.027
1.71	65.000	59.922	0.008	0.029
3.99	64.990	59.913	0.007	0.029
8.10	64.980	59.919	0.012	0.029
12.24	64.980	59.931	-0.003	0.029

Table 1 Summary of the saturator temperature measurement T_{sat} and its comparison with the dew-point temperature reading of a calibrated chilled-mirror hygrometer T_{CMH} as a function of the gas flow rate, with the PHE in vertical position; $T_{dp,inlet} = T_{sat} + 5$ °C.

Figure 4 summarizes the results of Test 2. The difference between the chilled-mirror hygrometer reading and the saturator temperature is plotted in the graph, together with its standard uncertainty (k=1), as a function of the pre-sat temperature $T_{dp,inlet}$ and volumetric flow rate, at constant saturator temperature ($T_{sat} = 5$ °C).

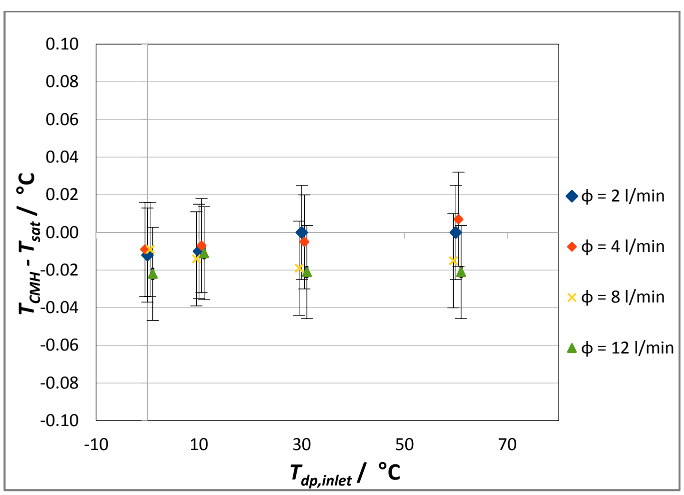

Fig. 4 Difference between CMH reading and saturator temperature as a function of the inlet dew-point and flow rate. Error bars correspond to the measurement standard uncertainty (k=1).

The saturator temperature compared favourably with the dew-point temperature as read by the CMH1, showing the ability of the system to saturate the carrier gas even at increasing inlet thermal load. This result together with the Test 3 results, which repeated Test 1 with a horizontally-oriented PHE [8-9] is believed to fully validate the principle for a PHE-based humidity generator operated at or near the atmospheric pressure.

Figure 5 shows the results of preliminary testing at higher pressure. A pressure-controlled N_2 gas flow was pre-saturated at $T_{dp,inlet} = T_{sat} + 5$ °C by flowing it through a bubbling pre-saturator in the temperature range from 5 °C to 60 °C. It should be noted that for $T_{sat} = 5$ °C (points at the left-hand side of the graph), the pre-saturated inlet gas corresponded to the condition $T_{dp,inlet} = T_{sat} + 15$ °C. The tests were repeated at three pressures, i.e. 0.1 MPa, 0.3 MPa and 0.5 MPa.

The results shows that the saturator temperature compared favourably with the dew-point temperature measured by the high-pressure chilled-mirror hygrometer (CMH2) in the above temperature and pressure ranges, except for $T_{sat} = 5$ °C.

The temperature deviations are to within the measurement standard uncertainty (k=1), which was estimated between 0.04 °C and 0.06 °C.

Figure 5 suggests that when $T_{sat} = 5$ °C, the temperature deviation between the hygrometer reading and the saturator temperature increases with the gas pressure. At 0.5 MPa, the deviation was beyond the statistical threshold set by a 95 % confidence interval ($U = 0.12$ °C).

The higher gas dew-point temperature detected by the CMH2 was tentatively attributed to a lower condenser efficiency with pressure; in fact, with a higher heat load (i.e., $T_{dp,inlet} = T_{sat} + 15$ °C) the system was unable to fully condensate the excess water vapor phase. This result would suggest the need of a careful control of the inlet dew-point to insure an optimal operation of the PHE condenser. Further investigations are needed to confirm the above tendency.

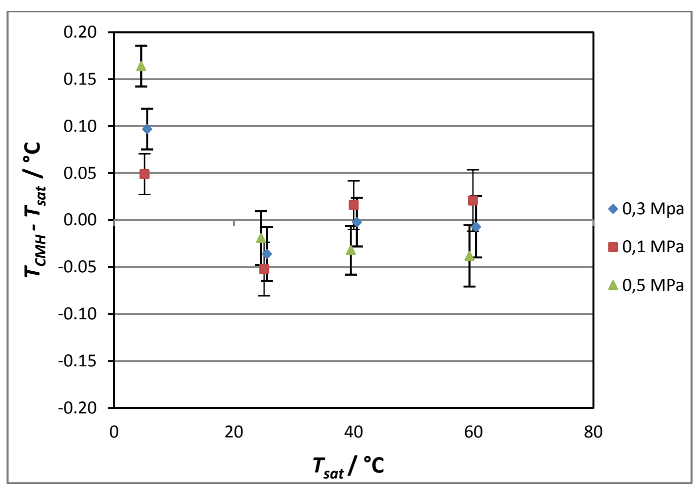

Fig. 5 Difference between CMH reading and saturator temperature at pressures of 0.1 MPa, 0.3 MPa and 0.5 MPa . PHE in horizontal position and flow rate of 2 l/min. Error bars correspond to the measurement standard uncertainty (k=1).

CONCLUSIONS

The experimental tests carried out at atmospheric pressure have proved the principle of operation of a PHE-based humidity generator. The system was found suitable to generate a humid gas flow at any dew-point temperature between 0 °C and 60 °C, with high accuracy and without any dependence from the orientation the heat exchanger. The system performance showed a high efficiency, to be able to operate in a large interval of volumetric flow rates and with a large range of inlet thermal loads (up to 60 W) with basically no effect on the saturator temperature. On the average the temperature deviation between the saturator temperature and the dew-point temperature measured by a calibrated CMH was lower than 0.02 °C, with a maximum deviation of 0.04 °C.

At pressure higher than the atmospheric pressure, preliminary testing showed a satisfactory behaviour for moderate heat load, but a temperature deviation larger than the expanded uncertainty (k=2) was found when the inlet dew-point temperature was 15 °C higher than the saturation temperature. This result needs further investigations in view of the expected generator operation at 1 MPa or higher.

The present assessment validated the working principles and showed that a portable humidity generator can be designed and easily operated at the following conditions:

- dew-point temperature range from 0 °C to 60 °C
- inlet pre-saturated gas with a dew-point 5 °C higher than the saturation temperature;
- gas pressure between 0.1 MPa and 0.5 MPa;
- volumetric flow rates up to 10 l/min;
- PHE either in horizontal or vertical orientation.

ACKNOWLEDGEMENT

The work was performed within a joint research project (JRP SIB64 METefnet) performed under the European Metrology Research Programme (EMRP). The project is jointly funded by the EMRP participating Countries within EURAMET and the European Union.

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