Development of a new primary humidity measurement standard

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Abstract
A new primary humidity generator was developed at the National Standards Authority of Ireland (NSAI). The principles of operation and overall design of the generator are discussed. The flow regime within the saturators was analysed using computational fluid dynamics (CFD) to determine if sufficient mixing takes place within the final saturator. This was later investigated by experiment and it was found that sufficient mixing takes place for flow rates up to 3 l min\(^{-1}\) at standard flow conditions of 101 325 Pa and 293.15 K. The uncertainty due to non-ideal saturation at 90 °C dp is discussed. This uncertainty was analysed with respect to the effect that the flow speed and latent heating in the saturators has on the saturation efficiency, separately. The uncertainty associated with saturation efficiency at 90 °C dew point was estimated from the temperature gradients along the saturation path arising due to latent heating effects. The standard uncertainty due to non-ideal saturation was found to be ±3 mK dp.

Keywords: humidity, metrology, dew point generator, primary humidity standard, computational fluid dynamics, saturation efficiency, humidity generator

(Some figures may appear in colour only in the online journal)

1. Introduction

This paper describes the development of a new primary humidity generator at the National Metrology Laboratory (NML), a department of NSAI, in conjunction with the European Metrology Programme for Innovation and Research (EMPIR) 15RPT03 HUMEA project. The temperature and humidity section at NML provides traceability for temperature and humidity measurements in Ireland. The laboratory currently uses four chilled mirror reference hygrometers. One reference hygrometer is calibrated yearly against a primary standard in an external laboratory while the others are calibrated internally by comparison, resulting in an increased uncertainty. A primary humidity generator was designed at NML and Dublin City University (DCU) and fabricated by various specialist engineers around Ireland, resulting in the expansion of humidity measurement capabilities at NML. This allows all humidity reference standards at NML to be calibrated against a primary measurement standard as well as providing a new service to customers, several of whom operate chilled mirror hygrometers and send them overseas for calibration.

A primary measurement standard is based on a primary reference measurement procedure which is used to produce a measurement result without relation to a measurement standard of the quantity of interest (De Bièvre 2012). A primary humidity generator conditions a gas to be completely saturated with water vapour at a stable temperature and pressure. If the saturation temperature and pressure are measured...
with known uncertainty, then the output mole fraction and output dew/frost point can be calculated using empirical reference functions. This also requires that the uncertainty due to the non-ideal saturation conditions can be determined and that all other processes such as water vapour leakage and description of water vapour from surfaces in the sampling lines can be quantified in SI units, as a component of the combined uncertainty of the system.

There is no straightforward procedure for the development of a primary humidity generator and existing generators use different mechanisms to achieve complete saturation with known uncertainty. The system described in this paper was developed on the basis of existing designs and philosophies of operation, together with results obtained from computational fluid dynamics. The heat transfer and flow properties within several proposed saturator designs were compared using a conjugate heat transfer model. Of interest is the determination of the range of flow rates over which complete saturation occurs due to the level of mixing within the saturator, and the optimum temperature measurement location in the final saturator.

2. Background theory

The following equation, derived from the ideal gas law, relates the mole fraction \( x \) of water vapour in a gas with the partial pressure of water vapour \( e(T) \) and the total pressure \( P \) of the gas,

\[
x = \frac{e(T) \cdot f(T, P)}{P},
\]

where \( f(T, P) \) is an enhancement factor.

The parameter \( e(T) \) represents the saturated vapour pressure over water or ice as a function of temperature, and its value can be found from empirical reference functions, such as (Sonntag 1990) with stated uncertainty. The parameter \( f(T, P) \) is an enhancement factor used to correct for the difference in the water vapour pressure due to the addition of air (forming an air-vapour mixture) which increases the chemical potential of the condensed phase due to the increased total pressure, among other effects (Boyes and Bell 1999). Its value can be found from empirical reference functions such as (Greenspan 1976) with stated uncertainty. The value of the enhancement factor becomes significant at elevated pressures.

In a primary dew/frost point generator, the mole fraction at the point of realization (the final stage of saturation) can be found from equation (1) if the saturation temperature \( T_s \) and pressure \( P_s \) are known. It is assumed that the mole fraction at the point of measurement is unchanged (excluding the effects of leaks and adsorption which are included as uncertainties), so equating equation (1) for the point of realization and the point of measurement gives,

\[
e(T_d) \cdot f(T_d, P_{uat}) = \frac{e(T_s) \cdot f(T_s, P_s) P_{uat}}{P_s},
\]

where \( T_d \) is the dew/frost point at the point of measurement and \( P_{uat} \) is the pressure at the point of measurement.

The product, \( e(T_d) \cdot f(T_d, P_{uat}) \) can be solved for \( T_d \) iteratively because the vapour pressure equations used cannot be solved for \( T_d \) analytically. The uncertainty in the reference function, \( e(T) \), is given in Pa so the associated uncertainty of \( T_d \) is the pressure uncertainty propagated through the iterative solution.

In designing a humidity generator, it is useful to know the rate of water loss due to evaporation for various input and output humidities. This allows the saturators to be designed so that they can hold enough water to allow continuous operation for a sufficient time period. It is also desirable to calculate the rate of heat loss from the saturators due to the latent heat of vaporization. A high rate of heat loss from the saturators will increase temperature gradients across the saturators and increase the uncertainty of the output dew point. The estimated rate of heat loss from the saturators can be used in computer simulations to estimate the temperature gradients across the saturators. In order to calculate quantities such as the rate of water loss from the saturators and the rate of heat loss due to the latent heat of vaporization, it is necessary to find the rate of mass transport of \( H_2O \) based on the input and output mole fractions of water vapour. The rate of evaporation of \( H_2O \) in terms of mass \( \Delta m_w \) is found from (FitzGerald, Swift and Mac Lochlainn, 2019),

\[
\Delta m_w = M_{H_2O} \Delta n_T \left( x_{out} - x_{in} \frac{1 - x_{out}}{1 - x_{in}} \right),
\]

where \( M_{H_2O} \) is the molar mass of \( H_2O \), \( \Delta n_T \) is the total molar flow rate at the output, \( x_{in} \) is the mole fraction at the input and \( x_{out} \) is the mole fraction at the output.

Knowing the mass flow rate at the output and the molar mass of the output air the molar flow rate at the output can be found. If the volumetric flow rate at standard conditions is known at the output then the molar flow rate at the output can be calculated.

3. Design of the new NSAI primary humidity standard

3.1. Full schematic

The new humidity generator will operate in a single pass mode of operation; where dry, filtered air is passed through the full system only once before being exhausted. It is a single pressure generator, having only a small pressure drop from the point of realization in the final saturator to the point of measurement. A schematic diagram of the new NSAI humidity generator is given in figure 1.

A metered flow of dry, filtered air is passed through the system. The dry airstream can be passed fully or partially through a heated humidifier to condition the air to be near the output dew point, but not above it as the generator is designed to saturate air with water vapour using evaporation, not condensation.

The airstream is then sent to the pre-saturator which is submersed in the same thermostatic bath as the final saturator. The pre-saturator brings the dew point of the airstream
very close to the final saturation temperature but not above it. Net evaporation throughout the system is maintained by ensuring that the temperature along the saturation path is increasing towards the outlet. For low or midrange dew/frost points it cannot be easily determined whether the pre-saturator is over or under-saturating the process gas due to low latent heating, however, if it does oversaturate the air it will be condensed in the heat exchanger before reaching the final saturator.

After the final saturator, the airstream is sent to a reference chilled mirror hygrometer used to monitor the output, and the hygrometer under test, which is connected in parallel to the monitoring hygrometer. The system includes valves and tubing used to back purge the outlet tube while forward purging the hygrometers with dry air. This is necessary to reach low frost points and prevent condensation in the outlet tube while the temperature of the system is decreasing.

3.2. Humidifier

The heated humidifier is a polished stainless steel cylinder, as shown by the Computer Aided Design (CAD) model in figure 2.

It is mounted horizontally and dry air is passed over the plane liquid surface. A thermal well allows the interior temperature and approximate output dew point to be measured. Heat tracing is wrapped around the unit and a proportional integral derivative (PID) controller is used to maintain the temperature. The output humidity can also be changed quickly by partially bypassing the humidifier with the use of two needle valves.

3.3. Pre-saturator

The pre-saturator is a polished stainless steel cylinder with a hole in its centre, as shown by the CAD model in figure 3.

The hole in its centre allows tubing and a standard platinum resistance thermometer (SPRT) to be fed through it to the final saturator (below). The unit contains a baffle plate at the input to prevent laminar streamlines. A thermal well is located near the output to estimate the output dew point from the pre-saturator.

3.4. Heat exchanger

The pre-saturator is connected to the final saturator via a heat exchanger, as shown in figure 4.

The heat exchanger was fabricated from approximately 4.5 m of ½ inch diameter stainless steel tubing using computer numerical control (CNC) tube bending. The interior walls are not polished in order to encourage condensation. It was found using the method described in (Hudoklin and Drnovšek, 2008) that the minimum length of this tubing required to produce a temperature change from 20 °C to −100 °C (within 0.1 mK) at conservatively chosen operating conditions is 1.83 m, so the length of tubing used ensures that the airstream is in thermal equilibrium with the surrounding bath liquid before reaching the final saturator.
the liquid bath near the outlet differed by 45 mK from the PRT inserted directly in the flow path, while an SPRT immersed in outlet differed in temperature by 4 mK compared to the PRT in the flow path. For this reason, thermal wells were included in the design of all saturators. At low frost points latent heating is negligible so the gradients across the final saturator and the surrounding bath liquid are significantly reduced. At low frost points the SPRT immersed in the liquid bath can be used to estimate the vapour/ice temperature, since the gradients are low and the capsule PRTs are less suitable to measure at temperatures as low as −40 °C due to increased drift.

Brass is chosen as the material for the base due to its high machinability and thermal conductivity. The base is plated with 40 µm of high phosphorous, electroless nickel plating (ENP) to provide corrosion resistance. High phosphorous ENP produces a pore free surface for plating thicknesses above 25 µm and is extremely unreactive and corrosion resistant. Electroless plating gives an even plating thickness on complex geometries unlike electroplating where variation of distance from the electrode produces a variation of plating thickness. This could be the reason for the corrosion of similar units over time, such as the saturator described in (Bosma and Peruzzi, 2017) which was electroplated with gold. The effect of impurities introduced by the ENP was estimated as the maximum drop in dew point over water soaked in glass, plastic, copper and stainless steel vessels for 13 months which was found to be less than 0.1 mK by (Vilbaste et al. 2013). This is conservative since elements like copper readily react with oxygen whereas high phosphorous ENP is extremely unreactive.

3.5. Final saturator

The final saturator, shown in figure 5, consists of a brass base and a stainless steel lid.

The lid and base are precision machined so the connecting surfaces are very flat, allowing a leak tight seal to be made. An indium wire gasket is used to create the seal. Connections to the flow path are made using stainless steel, metal gasket, face seal fittings. The unit contains a spiral path approximately 1.9 m in length, as shown in figure 6.

Thermal wells are located near the end of the saturation path and approximately half way along the path. The unit contains a hole in its centre, increasing thermal contact between the airstream and the thermostatic bath liquid and allowing an SPRT to be inserted near the outlet of the unit. The ability to measure at these locations allows temperature gradients and saturation efficiency uncertainties to be characterised. The position of temperature probes can be swapped during measurement, without significantly disturbing the stability of the system. This allows small temperature gradients to be measured with greater accuracy. Thermal wells allow temperature measurement of the interior of the saturator without compromising the leak tightness of the unit. This allows temperature measurement at these locations during every calibration, giving valuable information on the state of the system, such as the temperature homogeneity and latent heat loading within the final saturator. The unit also has connections for pressure measurements and an additional outlet approximately half way along the saturation path which can be used to validate the saturation efficiency.

Prior to fabrication, tests were performed using CFD with a conjugate heat transfer model to compare the temperature homogeneity and flow patterns in several proposed designs. All of the proposed designs performed well with the deciding factor being ease of fabrication. However, the main outcome in the comparison was from the investigation of the effect of measuring temperature at different locations in and around the saturator. In the simulation the input air to the final saturator was at 94 °C dp and the output air was 95 °C dp. This was chosen as a conservative estimate for the purpose of comparison rather than to obtain absolute results. In reality the input air to the final saturator should be closer to the output dew point. It was assumed that the best representation of the vapour/condensate temperature is from a PRT inserted directly in the flow path near the outlet.

It was found that a PRT inserted in a thermal well near the outlet differed in temperature by 4 mK compared to the PRT inserted directly in the flow path, while an SPRT immersed in the liquid bath near the outlet differed by 45 mK from the PRT in the flow path. For this reason, thermal wells were included in the design of all saturators. At low frost points latent heating is negligible so the gradients across the final saturator and the surrounding bath liquid are significantly reduced. At low frost points the SPRT immersed in the liquid bath can be used to estimate the vapour/ice temperature, since the gradients are low and the capsule PRTs are less suitable to measure at temperatures as low as −40 °C due to increased drift.

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3.6. Summary

The final saturator holds 400 ml of water with sufficient space remaining for air flow. Using equation (3), it was found that this is sufficient to saturate air to 90 °C dp at 1 l min⁻¹ (at standard conditions of 101325 Pa and 273.15 K) for 163 days, assuming that the pre-saturator conditions the air to be within 0.025 °C dp of the final output dew point (in-line with measurements performed at 90 °C dp). The pre-saturator holds 600 ml of water and it was found that this is sufficient to saturate air from 89 °C dp to 90 °C dp for 7 days in continuous operation. Based on tests during characterization it is reasonable to assume that the humidifier saturates air to within 1 °C of the output dew point of 90 °C dp. The humidifier can run for 30h in continuous operation at 90 °C output dew point (at 1 l min⁻¹ at standard conditions) but it is not immersed in the liquid bath and can be easily refilled.

The resistance measurements of the PRTs in the pre-saturator and final saturator and the SPRT in the liquid bath, are made using a precision resistance bridge. Pressure measurements are made using a precision barometer and the output from the generator is monitored using a chilled mirror hygrometer. The pressure at the point of measurement is measured by an integrated pressure transducer which is corrected against the reference barometer at the beginning and end of a calibration and is characterized with respect to drift, non-linearity, repeatability, hysteresis, temperature dependence and stability. The full system excluding valves is controlled by one computer. All units are sampled by one software package and calculations are performed automatically.
4. Modelling flow

Mixing of air and water vapour within the final saturator is crucial to maintaining efficient saturation. The dimensions of the flow path are large and the flow rate is low so the flow speed and Reynolds number are low, meaning that turbulent flow is not expected. The flow patterns at 90 °C generated dew point were analysed in Ansys Discovery AIM using a CFD conjugate heat transfer model to investigate the range of mass flow rates over which sufficient mixing is expected to occur. The geometric model used for simulations was a simplified version of the detailed CAD model of the final saturator. The generated mesh consisted of 3.13 × 10^6 polyhedral elements in a total volume of 2.07 × 10^{-2} m^3. The cell size varied depending on the local geometry. The mesh had an average orthogonal quality of 0.78.

The solver uses the finite volume method of discretization and the governing equations are in the conservational form. The model included computational domains representing the thermostatic liquid bath, the brass saturator, the standing water inside the saturator, and the air flow within the saturator. Domains were also included to represent the stainless steel PRTs but these are not relevant to the analysis of flow. The bath liquid flowed from the bottom to the top of the computation domain to simulate an upwelling bath. The boundary conditions at the air input were defined for various mass flow rates from 1 l min$^{-1}$ to 10 l min$^{-1}$ at standard conditions, with the input air at a temperature of 90 °C, since the air temperature is expected to be very near to the liquid bath temperature at the output of the heat exchanger. The air output boundary was defined to be at atmospheric pressure as it will be near atmospheric pressure in operation. The bath liquid input boundary was defined to have a mass flow rate of 0.05 kg s$^{-1}$ and a temperature of 90 °C. At this flow rate the gradient from bath input to output was 1 mK. The bath output was defined to be at atmospheric pressure. All other exterior surfaces were defined to be insulated walls and all region interfaces were defined to be thermally coupled, no-slip walls.

The domain representing the standing water inside the saturator was defined to contain a heat source of −0.06 W. This is the magnitude of latent heating estimated using equation (3), and assuming an increase in dew point of 25 mK across the final saturator based on a temperature difference of 25 mK from the outlet of the pre-saturator to the outlet of the final saturator at the midpoint of the defined operating conditions. This value of latent heating will remain constant for simulations of different mass flow rates. The results will show the effect of increased flow speed on mixing for pre-conditioned air where latent heating is constant. The latent heating is included in the simulation as the cooling of air near the water surface is expected to retard convective mixing. Mixing is expected to occur for low flow speeds due to small temperature gradients within the saturator. This is accounted for by using a buoyancy model which includes a term in the momentum equation for the buoyancy force due to density gradients arising from fluctuations in the temperature field.

The evaporation of water is not simulated as it would require the use of a multiphase model and would significantly complicate the simulation. Instead the latent heating due to evaporation is estimated using equation (3) and included as a boundary condition. It is assumed that excluding the simulation of evaporation will produce a conservative result regarding the observation of mixing, since air nearest to the water surface will be less dense if vapour pressure gradients exist, because humid air is less dense than dry air, creating a density gradient that will encourage mixing. It would be beneficial to simulate evaporation using a multiphase model so that water vapour pressure gradients could be studied, however, in this analysis streamlines will be plotted to determine if mixing of the process gas will occur or if the flow will be laminar. If ample mixing of the process gas occurs it is assumed that vapour pressure gradients will be negligible. Mixing of water vapour by diffusion may occur for laminar flows and is not accounted for in the simulation, which may produce a conservative result if diffusion of water vapour for laminar flows is significant. However, since turbulence is expected to be minimal, the amount of diffusion is expected to be small.

As mentioned previously, turbulence is not expected due to the low flow rate, large flow path dimensions and therefore low Reynold’s number. Turbulence could be modelled as well as evaporation of water using a multiphase model, and a design could be optimised to produce turbulence, possibly with the use of partial recirculation to increase the flow speed, and validated by observing vapour pressure gradients in a CFD model for various flow rates. This however would significantly increase the complexity of both the CFD model and the overall design and philosophy of operation. The model described in this paper is a simplified approach intended to produce a conservative result. Figure 7 shows the streamlines plotted in the flow path of the final saturator for an output flow rate of 1 l min$^{-1}$ at standard conditions.

In figure 7 the simulated flow rate through the final saturator is stated in units of 1 min$^{-1}$ at standard conditions of 293.15 K and 101 325 Pa. From this point on, if a flow rate is stated in units of 1 min$^{-1}$, it can be assumed that this refers to airflow at standard conditions. The colour of the streamlines represents temperature for the purpose of illustration, however for this simulation the temperature is unimportant and the direction of the streamlines is the quality of interest. In figure 7 streamlines can be seen to travel in random directions,
regularly traveling from bottom to top of the saturator, indicating that sufficient mixing will occur.

Figure 8 shows side views of the wireframe model of the final saturator with streamlines plotted for various flow rates through the unit. In figure 8 the scale is removed from the temperature contours as it changes for each simulation and its purpose is mainly for illustration and to show relative temperature change across the saturator.

Figure 8 shows that at 1 l min$^{-1}$ the direction of streamlines is random and mixing occurs due to the low flow speed of the gas. At 2 l min$^{-1}$ the flow becomes slightly more directional in that the horizontal component of velocity increases, and mixing decreases slightly. At 3 l min$^{-1}$ mixing has decreased but streamlines can still be seen to travel from top to bottom of the flow path and complete saturation may still occur over the distance of the flow path. At 4 l min$^{-1}$ the flow continues to become more laminar. Some streamlines can still be seen to travel from top to bottom of the flow path so complete saturation may still occur over the full path length. Above 4 l min$^{-1}$ the flow continues to become more laminar with the plotted streamlines approximately resembling the ones in figure 8 at 4 l min$^{-1}$, so the results for flow rates above 4 l min$^{-1}$ are not shown. Based on the CFD results it is expected that sufficient mixing of the gas will occur at flow rates up to 2 l min$^{-1}$ due to the random direction of the streamlines plotted. Above 2 l min$^{-1}$ it is expected that mixing will be reduced. The reduced level of mixing may still be enough to provide efficient saturation over the full 1.9 m path length in the saturator. The diffusion of water vapour may also provide efficient saturation over the full path length of the saturator. The change in saturation efficiency with respect to changes in flow speed was investigated experimentally and the results are given in the following section.

5. Uncertainty due to non-ideal saturation

The uncertainty due to non-ideal saturation is the only uncertainty component which will be discussed in this paper. In a primary humidity generator air and water vapour are leaving the system in order to allow measurement of the gas by the hygrometer under test. This means that the conditions in the final saturator, where the dew/frost point temperature is realized, differs from the ideal conditions that the vapour pressure reference functions are based on. This can cause differences in the output water vapour mole fraction compared to what is expected based on temperature and pressure measurements in the final saturator.

Various methods exist to characterize the uncertainty associated with saturation efficiency including, variation of flow rate in order to observe the point where the saturation efficiency begins to decrease, dew point measurement of the process gas at intermittent points along the saturation path and measurement of the difference in temperature between the vapour and condensate. In this paper, saturation efficiency will be investigated by first determining the flow rate dependence of the generator by varying the output flow rate under normal operating conditions and observing the change in the output dew point as measured by a chilled mirror hygrometer. The saturation efficiency will then be investigated by varying the flow rate, while keeping latent heating constant and by varying latent heating by varying the input dew point, while output mass flow rate remains constant. Variation of flow was compared against the output dew/frost point and the effects of latent heating were compared against the output dew point and temperature gradients within the final saturator.

Figure 9 shows a graph of the hygrometer error (monitoring hygrometer reading minus the expected output dew point) plotted against mass flow rate through the generator.

The results in figure 9 were taken under normal operating conditions without maintaining constant latent heating in the system. For elevated mass flow rates both the flow speed and the latent heating in the saturators increases due to the
increased mass transport of water vapour from the saturators. The data in figure 9 shows that for flow rates below 4 l min$^{-1}$ the output dew point varies by less than 3 mK. When the flow rate is increased to 5 l min$^{-1}$ the peak to peak spread of values increases to 5.3 mK and above 5 l min$^{-1}$ the variation becomes more significant. The data in figure 9 was taken by varying the flow rate throughout the system while keeping only the flow through the monitoring hygrometer constant (it can be assumed that the flow through the chilled mirror hygrometers always remains constant unless otherwise stated). The variation of the output dew point for elevated flow rate is assumed to be a combination of increased latent heating within the final saturator and reduced mixing at elevated flow rates, as discussed in section 4. It can be seen in figure 9 that at 7 l min$^{-1}$ the hygrometer reading increases relative to the expected output dew point based on temperature measurements, before continuing to decrease at 10 l min$^{-1}$. This could be due to the fact that for increased flow rate, latent heating increases, therefore cooling the PRTs in the flow path and increasing temperature and vapour pressure gradients across the flow path, meaning that the output air can be supersaturated relative to the temperature measured by the PRTs in the flow path. The increased flow rate can also cause a decrease in the output dew point relative to the temperature measured in the flow path due to the decreased mixing and increased vapour pressure gradients cause by laminarity of the flow due to the increased flow speed. The variation in the output dew point for various output flow rates is therefore a combination of different effects and it is for this reason that the output dew point will now be analysed with respect to variation of latent heating and flow speed separately.

The effect that varying flow rate has on the saturation efficiency without the effects of increased latent heating will be studied by varying the flow rate, while varying the input dew point in order to keep the latent heating in the pre-saturator and final saturator constant. The latent heating in the system can be found from equation (3) for various input and output mole fractions and output flow rates. The input dew point was varied using the by-pass valves of the humidifier and measuring the input dew point with a chilled mirror hygrometer. Figure 10 shows the hygrometer error plotted against the mass flow rate at the output for an approximately constant latent heating of 10.4 W. This value was chosen as it is the rate of latent heating produced for a flow rate of 0.5 l min$^{-1}$ at 90 °C output dew point if the input gas is dry air at −50 °C fp. This is higher than the rate of latent heating during use which should allow the drop off in output dew point for increased flow speed to be more easily observed.

Figure 10 shows a peak to peak variation of 8 mK for flow rates up to 3 l min$^{-1}$. Above 3 l min$^{-1}$ the output as measured by the chilled mirror hygrometer decreases significantly. It was found after this experiment was conducted that the water level in the final saturator was higher than expected due to condensation occurring in previous measurements. The reduced volume of the flow path caused the flow velocity to be higher than usual and exaggerated the drop off in output dew point seen in figure 10. These measurements will be repeated in the future when the equipment needed is available. Despite the fact that the drop off in output dew point is exaggerated, figure 10 agrees with what was expected from the discussion of the results in figure 8 in that sufficient mixing to provide complete saturation occurs for flow rates up to 2 l min$^{-1}$ and based on figure 10 flow rates as high as 3 l min$^{-1}$ produce enough mixing to achieve complete saturation over the path length of the final saturator. Since the generator will rarely be used above 1 l min$^{-1}$ for dew points this is a satisfactory result.

The effect that varying flow rate has on the saturation efficiency without the effects of increased latent heating was also studied by varying the flow rate at −40 °C generated frost point. Since latent heating effects are negligible at low frost points, this will show the effect that flow speed has on mixing although the drop off in output frost point may not be as significant as in figure 10 since saturation is readily achieved for frost points. A graph of hygrometer error against flow rate through the saturators is shown in figure 11, with error bars representing the standard deviation of 40 measurements from the monitoring hygrometer.

Figure 11 shows that below 4 l min$^{-1}$ there is less than 3 mK variation in the output frost point measured by the

![Figure 10](image-url)  
**Figure 10.** Graph of hygrometer error against mass flow rate at 90 °C generated dew point for constant latent heating of approximately 10.4 W.

![Figure 11](image-url)  
**Figure 11.** Graph of hygrometer error against mass flow rate at −40 °C generated frost point with error bars representing the standard deviation of 40 measurements taken from the monitoring hygrometer.
monitoring hygrometer. Above 4 l min\(^{-1}\) the output frost point as measured by the monitoring hygrometer begins to decrease relative to what is expected from temperature measurements at the outlet of the final saturator. This is assumed to be due to the decreased mixing in the final saturator due to laminar flow as discussed in section 4.

The effect of increased latent heating on the saturation efficiency was investigated by varying the latent heating in the pre-saturator and final saturator from approximately 0 W to 20 W (by varying the input dew point from the humidifier) and observing the change in the dew point measured by the monitoring hygrometer compared to the expected output dew point from temperature and pressure measurements in the final saturator. The rate of latent heating was calculated using equation (3). Figure 12 is a graph of the hygrometer error plotted against the rate of heat loss from the pre-saturator and final saturator due to the latent heat of vaporization, with error bars representing the standard deviation of 40 measurements from the monitoring hygrometer.

Below 9 W the hygrometer error varies by less than 3 mK dp. Above 9 W the hygrometer reading increases relative to the expected dew point based on temperature measurements in the saturation path. This is assumed to be due to the increased latent heating in the saturation path and the increased temperature gradient along the path.

The temperature gradients between the SPRT immersed in the liquid bath near the output of the final saturator, the platinum resistance thermometer (PRT) inserted in the thermal well near the midpoint of the saturation path (PRT1), and the PRT inserted in the thermal well near the end of the saturation path (PRT2), were measured for various rates of latent heating in the system and plotted in figure 13.

This shows that the gradients within the final saturator remain reasonably consistent below 9 W latent heating, after which the gradients increase significantly. The increase in the gradient from the midpoint of the saturation path and the surrounding thermostatic bath liquid is greater than the increase in the gradient near the end of the saturation path and the surrounding thermostatic bath liquid. This indicates that saturation continues after the midpoint of the path and the air at the end of the saturation path is closer to complete saturation than air at the midpoint. Therefore, it is assumed that the output air will have a dew point greater than the vapour/condensate temperature at the midpoint of the saturation path. It is assumed that; if net evaporation is occurring in the final saturator, then the output air from the final saturator will not be supersaturated with respect to the temperature of the surrounding thermostatic bath liquid. Therefore, it will be assumed that the dew point of the output air from the generator will be best approximated by measurement of the vapour/condensate temperature by the PRT inserted in the thermal well near the end of the saturation path (for high dew points where latent heating is significant). The uncertainty associated with non-ideal saturation will be taken as the temperature gradient from the PRT near the end of the saturation path and the SPRT immersed in the liquid bath near the outlet of the final saturator, since the dew point will continue to increase after the PRT near the end of the saturation path but will not become supersaturated with respect to the surrounding thermostatic bath liquid, provided that net evaporation is occurring. Figure 14 is a graph showing the gradients between different points in the final saturator, normalized to remove temperature gradients due to the inhomogeneity of the thermostatic bath.

Temperature gradients due to the thermostatic bath are removed because they are included elsewhere in the uncertainty budget and the temperature gradients which are being examined with respect to the saturation efficiency are those that arise due to increased latent heating in the saturators. For latent heating below 9W, the maximum temperature difference measured between the SPRT and the PRT near the end of the saturation path is below 5 mK. This will be taken as the halfwidth of the rectangular distribution of values of output dew point attributed to the uncertainty due to non-ideal saturation.
By dividing this value by \( \sqrt{3} \) as described in (JCGM, 2008) it is found that this corresponds to a standard uncertainty of \( \pm 3 \) mK dp. This agrees with the data in figures 9, 11 and 12 where the output dew/frost point varies by less than 3 mK in the range of interest.

Based on the data in figures 9–11, the generator will produce a maximum output flow rate of 3 l min\(^{-1}\) when used as a primary humidity generator. This is sufficient to supply 6 chilled mirror hygrometers with 0.5 l min\(^{-1}\) of output air or 3 chilled mirror hygrometers at 1 l min\(^{-1}\) for frost points.

Based on the data given in figures 12–14 the generator will be operated with latent heating at a rate below 5 W across the pre-saturator and the final saturator. This will be controlled by varying the temperature of the humidifier and varying the position of the humidifier bypass valves. Figure 15 is a graph of the temperature difference from the pre-saturator to the final saturator plotted against the latent heating in the system.

Based on figure 15, maintaining a temperature difference of no more than 0.05 °C between the pre-saturator and final saturator (with the pre-saturator colder than the final saturator), ensures that the latent heating in the system remains approximately below 5 W (for 90 °C generated dew point). The uncertainty associated with saturation efficiency at −40 °C fp is discussed in (FitzGerald et al 2019) and the uncertainty due to non-ideal saturation at intermediate set points will be investigated in future analyses.

6. Conclusion

A dew point generator was developed at NSAI with the intention to be operated as a primary humidity standard. The design was tested using CFD prior to fabrication and the flow regime was analysed to determine the range over which sufficient mixing is expected to occur. The uncertainty associated with non-ideal saturation was analysed with respect to both the flow speed and latent heating in the saturators. The results were sufficient based on the operating requirements of the generator and were consistent with what was expected based on the CFD analysis of mixing.

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