Control strategy development for optimized operational flexibility from humidified micro Gas Turbine: Saturation tower performance assessment

Ward De Paepe*
University of Mons (UMONS)
Thermal Engineering and Combustion Unit
Place du Parc 20, 7000 Mons, Belgium

Alessio Pappa
University of Mons (UMONS)
Thermal Engineering and Combustion Unit
Place du Parc 20, 7000 Mons, Belgium

Diederik Coppitters
University of Mons (UMONS)
Thermal Engineering and Combustion Unit
Place du Parc 20, 7000 Mons, Belgium

Marina Montero Carrero
Vrije Universiteit Brussel (VUB)
Thermo and Fluid Dynamics (FLOW)
Pleinlaan 2, 1050 Brussel, Belgium

Panagiotis Tsirikoglou,
Limmat Scientific AG
Industriestrasse 7, CH-6300 Zug, Switzerland

Francesco Contino
UCLouvain, Thermodynamics and Fluid Mechanics
Place du Levant, 2, 1348 Louvain-la-Neuve, Belgium

ABSTRACT

Waste heat recovery through cycle humidification is considered as an effective tool to increase the operational flexibility of micro Gas Turbines (mGTs) in cogeneration in a Decentralised Energy System (DES) context. Indeed, during periods with low heat demand, the excess thermal power can be reintroduced in the cycle under the form of heated water/steam, leading to improved electrical performance. The micro Humid Air Turbine (mHAT) has been proven to be the most effective route for cycle humidification; however, so far, all research efforts focused on optimizing the mHAT performance at nominal electrical load, in the absence of any thermal load. Nevertheless, in a DES context, the thermal and electrical load of the mGT need to be changed depending on the demand, requiring both optimal nominal and part load performances. To address this need, in this paper, we present the first step

*Address all correspondence to this author. Tel: +32 (0)65/37.44.71 Email: ward.depaepe@umons.ac.be

towards the development of a control strategy for a Turbec T100 mGT-mHAT test rig. First, using experimental data, the global performance, depending on the operating point as well as the humidity level, has been assessed. Second, the performance of the saturation tower, i.e. the degree of saturation (relative humidity) of the working fluid leaving this saturator, is analyzed to assess the optimal water injection system control parameter settings. Results show that optimal mHAT performance can only be obtained when the working fluid leaving the saturation tower is fully saturated, but does not contain a remaining liquid fraction. Under these conditions, a maximal amount of waste heat is transferred from the water to the mGT working fluid in the saturation tower. From these data, some general observations can be made to optimize the performance; being maximizing injection pressure and aiming for a water flow rate of $\approx 5\,\mathrm{m^3/h}$*. Besides these general recommendations, having a specific control matrix, that allows setting the saturation tower control parameters for any set of operational setpoint and the inlet conditions would lead to optimized performance. Therefore, future work involves the development of a control matrix, using advanced data post-processing for noise reduction and accuracy improvement, as well as an experimental validation of this methodology on the actual test rig.*

Nomenclature

CHP Combined Heat and Power

CIT Combustor Inlet Temperature

DES Decentralised Energy System

GT Gas Turbine

HAT Humid Air Turbine

LHV Lower Heating Value

mGT micro Gas Turbine

mHAT micro Humid Air Turbine

RICE Reciprocating Internal Combustion Engines

TIT Turbine Inlet Temperature

SVR Support Vector Regression

TOT Turbine Outlet Temperature

VUB Vrije Universiteit Brussel

INTRODUCTION

Cycle humidification is an interesting technique to enhance both fuel and operational flexibility of micro Gas Turbines (mGTs) in Decentralized Energy System (DES) applications. Indeed, in the current shift towards the use of alternative, renewable fuels, humidification can help stabilizing the combustion process, both for high-Lower Heatin Value (LHV) fuels, e.g. by reducing the risk of flashback under hydrogen combustion as a result of the reduced reactivity due the dilution [1], and low-LHV fuels, e.g. by reducing the temperature in the combustion chamber when using syngas or biogas and while doing so, reducing the formation of several pollutants [2]. From the operational point of view, cycle humidification allows to increase the specific power production [3]. Moreover, when humidification occurs before the recuperator, additional waste heat is recovered from the flue gases, leading to increased electric efficiency. This efficiency increase is especially interesting for mGTs operating in small-scale Combined Heat and Power (CHP) applications with periods of low or even no heat demand, typically residential CHP applications, allowing to make them competitive with Reciprocating Internal Combustion Engines (RICE) [4, 5].

From all possible options for humidification, cycles exploiting injection of liquid water in a saturation tower with a water recovery loop were identified, similar to large-scale Gas Turbine (GT) cycles [6], as the optimal route for mGT cycle humidification [7]. Especially the micro Humid Air Turbine (mHAT), proposed by Parente et al. [8] and based on the Humid Air Turbine (HAT) cycle from Rao [9], is considered the most promising solution: the cycle combines high performance with a still rather simple cycle layout, essential for the success of such an application [10]. Although the mHAT has been studied several times in literature, e.g. most recently by MosayebNezhad et al., who showed that a 500 kW_e mHAT could achieve 46.6 % efficiency when used in a waste water treatment plant and being fed with biogas [11][1], to the knowledge of the authors, only 2 full-scale demonstrator test rigs have been constructed and tested so far: a 100 kW_e mGT converted to mHAT test rig at the Institute of Engineering Thermal Physics, Chinese Academy of Science, Beijing, China [12] and again a 100 kW_e mGT converted to mHAT test rig at the Vrije Universiteit Brussel (VUB), Brussels, Belgium (test rig considered in this paper, details are presented in the *Methodology* section).

The main difference between both test rigs can be found in the cycle layout and in the used type of saturation tower. Indeed, on the one hand, the test rig of the Institute of Engineering Thermal physics has an aftercooler in the cycle, hence rather exploiting the mHAT+ layout from Parente et al. [8], making the cycle slightly more complex then the simpler mHAT cycle. Moreover, to humidify the air, a saturation tower with specific developed packing material was used [13], despite having first explored the potential of using a spray saturation tower [14]. On the other hand, the VUB test rig is using the classic mHAT layout without aftercooling, and is using a specific developed spray saturation tower [15]. Experimental results on both test rigs have demonstrated indeed the potential of the mHAT cycle: Xu et al. reported on a 22.2 % electric efficiency increase for the test rig of the Institute of Engineering Thermal physics [12], while Montero Carrero et al. reported an electric efficiency increase of up to 4.2 % absolute points in wet operation at constant rotation speed [16].

To optimize the performance of the mHAT cycle, it is essential that the compressed air leaving the saturation tower is fully saturated [17]. This requires the accurate determination of the humidity level of the working fluid, i.e. relative humidity. Accurate online measurement of relative humidity of pressurized air is technically challenging and expensive (see later). Given the limited number of available mHAT test rigs, so far, no real-time monitoring method has been developed or presented. Nonetheless, aiming at determining online the relative humidity of the compressed air will only provide an indication on how optimal the saturation tower and in extend the cycle is operating, but no suggestions on how to improve

[1]For a complete overview on the available literature on humidified and more specific the mHAT cycles, we refer the reader to the review published by the authors of this work [3].

it. Hence, we believe it is more interesting to develop a control strategy based on global cycle performance, and thus not on the actual humidity level. Additionally, considering future commercialization, it must be avoided to include expensive measurement equipment, such as an accurate relative humidity sensor. So far, the most advance analysis presented of the HAT or mHAT cycle performance depending on the performance of the saturation tower is the work of Wang et al. [18]. In this work, the authors present an analysis of the impact of the air-to-water ratio on the performance of a HAT cycle at different operating points. Although presenting interesting trends towards performance optimization, these results are obtained during post-processing of experimental data, not allowing for online performance monitoring nor for dedicated control use. This expresses again clearly the need for the development of a control matrix, determining the control settings for the mHAT test rig and for the water injection network in particular, for the specific atmospheric conditions and requirements of the mGT, i.e. requested electrical power output or rotational speed. By focusing on the global cycle performance, and thus not on the humidity level of the compressed air, the actual humidity level will not be used or determined directly. However, since for a set of specific atmospheric conditions and operational setting, i.e. selected Turbine Outlet Temperature (TOT) and electrical power output, the mHAT performs optimal when the air is fully saturated (see later). So we can thus assume that, at maximal electrical performance for these specific atmospheric and operational conditions, the working fluid is indeed fully saturated. However, before such a control strategy can be developed, it is key to get an accurate assessment of the mHAT performance.

The aim of this paper is thus to assess the performance of the mHAT cycle depending on the saturation tower performance. This is an essential step towards the develop of a full control matrix that enables optimized performance of the mHAT test rig with a spray saturation tower. This control matrix will provide, as highlighted before, based on inlet air temperature and requested power output/rotational speed, the optimal settings in terms of mGT and more specific water injection system operating parameters. In the preparatory performance assessment study towards this control matrix development, presented in this paper, first the experimental data will be analyzed to assess the impact of the humidification level and saturation tower performance on the global performance. Followed by the identification of the critical parameters for optimal saturation tower performance. In a later stage, the control matrix can be obtained by means of advanced data post-processing for noise reduction and accuracy improvement using surrogate models (i.e. a Support Vector Regression (SVR) model).

In this paper, we first present the used methodology, including the mHAT cycle and test rig description. In the results section, first the experimentally obtained results for the global cycle performance are presented, followed by a presentation of the specific saturation tower performance, including the specific used operating conditions allowing to identify the main parameters of importance. Finally, the conclusion and future perspective are formulated in the last section.

METHODOLOGY

In this section, first the general mHAT cycle is presented, followed by the specific spray saturation tower, and finally by a brief discussion on the experimental setup.

mHAT cycle description

The classical recuperated mGT is converted in an mHAT cycle by introducing a saturation tower in between compressor outlet and recuperator inlet (Figure 1). In this saturation tower, a large amount of water is injected in the hot compressed air coming from the radial compressor. Due to the high temperature and low relative humidity of the incoming air, a fraction of the water is evaporated, leading to saturated air leaving the saturation tower at reduced temperature. After leaving the saturation tower, the humidified compressed air follows the classical pathway of the mGT operating according the recuperated Brayton cycle: being preheated in the recuperator by the heat from the hot flue gases; heated till maximum Turbine Inlet Temperature (TIT) in the combustion chamber by burning fuel; expanding over the turbine to deliver mechanical power to drive both compressor and a high speed generator for electricity production; and finally passing again through the recuperator. After having preheated the saturated compressed air in this recuperator, the flue gases still contain a lot of thermal power, which is used to heat the water in the injection circuit by means of an economizer. This water in turn will transfer the heat again to the cycle working fluid in the saturation tower.

In the saturation tower, it is thus the aim to fully humidify and saturate the compressed air with water to improve the mGT performance. By adding extra mass to the cycle behind the compressor, more power is available on the turbine shaft, leading to increased electric power output. In addition, the increased heat capacity of the working fluid (due to the additional water fraction) in combination with the lower recuperator inlet air temperature, results in a higher heat recovery in the recuperator, leading to increased electric efficiency as well. To optimize the performance of the mHAT, it is essential that, besides a mass transfer from the water phase to the gas phase due to water evaporation, a heat exchange is also effected. Indeed, the necessary heat to evaporate water can be taken entirely from the hot air (sensitive enthalpy), but ideally, the main fraction is taken from the water: sensible enthalpy is converted into latent heat, making that the water leaving the tower is typically at a lower temperature than when entering the tower. Due to this action, additional waste heat is recovered in the mGT cycle, leading to improved and optimized performance. As previously highlighted by the authors of this paper during an energy and exergy analysis [19], both actions — heat transfer from water to air in the saturation tower and humidified outlet at lower temperature — are essential to fully exploit the potential of the mHAT cycle.

Test rig description

The experiments used for the analysis presented in this paper have been conducted between 2012 and 2020 on the converted humidified mGT-VUB test rig [20]. The main parts of the test rig are a Turbec T100 series 2 mGT [21], a steam boiler for steam injection (not used for the experiments discussed in this paper), a custom developed spray saturation tower [15] and a data acquisition system to capture all cycle parameters (for a complete overview of installed sensors, their location and accuracy, we refer to [16]). The Turbec T100 mGT, operating according the recuperated Brayton cycle, is converted in an mHAT by routing the hot compressed air coming from the compressor (Figure 2(a)) through the spray saturation tower (Figure 2(b)), before routing the saturated air back to the recuperator inlet using a set of valves. In addition, a bleed and blow-off valve are installed behind the compressor to provide additional surge margin during water injection and prevent the compressor from going into surge when performing an emergency shut-down, respectively. This makes that the test rig can be operated in 5 different regimes [22]. For the experimental analysis, presented in this paper towards the development of a control matrix, only the results of operation in humidified mode, with water injection in the saturation tower (and air bleed for surge margin increase) have been used. The dry results, used as reference to demonstrate the impact of cycle humidification, were obtained by letting the working fluid, e.g. compressed air, passing through the saturation tower (without water injection, but with air bleed).

As mentioned in the introduction, rather than using a classical saturation tower with packing material, to enhance the contact surface between compressed air and water, we opted to use a spray saturation tower [15]. By not using packing material, the pressure loss over the saturation tower could be limited. The disadvantage of this approach; however, is that due to the presence of liquid droplets in the compressed air, an appropriate droplet separator must be added to the system to avoid liquid water from entering the recuperated. Indeed, this is highly unwanted, since the mHAT concept displays optimal performance when the air leaving the saturation tower is fully saturated (relative humidity of 100 %), but the presence of liquid water must be avoided (see *Results* section). To provide sufficient contact area, the water flow rate must be divided in a large amount of small droplets. Therefore, in the test rig, 7 full cone nozzles, each composed of 7 hollow cone nozzles from PNR [23], were installed (2 nozzles are highlighted in Figure 2(b)). The main challenge is to get a small average droplet size, while still injecting a large amount of liquid water and doing it with a minimal pressure difference to limit the power consumed by the pump (indicated on Figure 2(b)). These objectives are typically conflicting. Finally, although a pure counter-flow arrangement is preferred to get optimal saturation tower performance, as shown by Xu et al. in [14] and [24], in the VUB-test rig, a cross flow arrangement was selected, mainly for technical reasons. Firstly, due to space limitations, it was not possible to install a large tower for a pure counter-flow arrangement, nor was it possible to put the saturation tower close to the compressor outlet. The non-optimal placing leads to significant pressure losses forcing to operate in water injection mode always with the air bleed open (for surge protection). Secondly, to avoid interference of the different spray cones, leading to unwanted coalescence of small droplets to larger droplets, the water is injected perpendicular to the air flow. Indeed, by placing the saturation tower horizontally and putting the nozzles along the axis, more spacing between the nozzles could be respected. When using a counter-flow arrangement, the saturation tower needs to have a significantly larger diameter to place all nozzles while respecting a minimal distance. This would make the tower volume significantly larger, which is disastrous for the pressure loss as well as the dynamic performance of the cycle. Indeed, the additional volume will put a limit on the ramp rapes for rotational speed and power output, due to the significant surge margin reduction [25].

Concerning the operational control of the test rig, we opted to keep using the original mGT control system, to manage the mGT operation, in combination with a manual control approach for the water injection circuit. The Turbec T100 mGT control system consists in two parts: constant TIT control and constant power control. The TIT is kept constant by changing the fuel mass flow rate. Operation at maximal TIT of 950 °C leads to maximized electric efficiency. Since TIT cannot be measured, the control system links TIT with the measurement of TOT using fixed look-up tables. Under humidified conditions, these look-up tables are no longer valid, making that the mGT operates rather at constant TOT (of 645 °C). Due to the changing properties of the working fluid, in humidified conditions, at constant TOT, the actual TIT is lower. Although it might be technically possible to correct the look-up tables for the changing composition, at maximal TIT of 950 °C, still respecting the material constraints for the turbine, TOT would increase and lead to possible damaging of the recuperator hot side inlet. Hence it was decided to not adjust the temperature control. Operation at constant power output is achieved by adjusting the rotational speed of the engine. To achieve more flexibility in the humidified operation, the constant power control was slightly adjusted to also enable operation at constant rotational speed [16].

The manual control of the injection system includes 3 control parameters: pump rotational speed, number of used nozzles and ventilator speed on the air heaters. By changing the rotational speed of the pump and activating more or less nozzles in the system, the injection pressure could be varied, enabling the control of the total amount of injected water in the saturation tower. So far, no actual control strategy was used due to the limited available data. Pump speed and number of nozzles were varied based on qualitative parameters, i.e. the visual observation of the injection in the tower, as well as quantitative parameters, being injected water mass flow rate and injection pressure. Final aim was to obtain stable continuous injection leading to a fine fog of droplet (Figure 3). Unfortunately, it was technically impossible to measure the actual droplet side to validated whether the size distribution was as presented by the manufacturer [23] and conform the minimal required size, calculated when designing the saturation tower [15]. Therefore, a rather qualitative visual validation was performed. Next to the qualitative visual observation also two more quantitative parameters, being the amount of injected water (volumetric flow rate) and the injection pressure, were monitored as well to ensure an optimized injection. Both parameters were set close to the theoretical optimal setpoints, determined based on numerical simulations of the mHAT cycle (2.5 kg/s for the water injection flow rate [10]) and nozzle datasheets (pressure difference of 0.5 bar that should lead at the optimized flow rate to the desired droplet size [23]). Moreover, a third quantitative parameters was also used to control the pump rotational speed, being the Combustor Inlet Temperature (CIT). Indeed, as highlighted in previous work of the authors [22], a too large amount of liquid water entering the recuperator will lead to a reduced CIT and hence unwanted and disastrous flame out, since an unwanted and uncontrolled shutdown leads to compressor surge. A maximal threshold of 100 °C temperature reduction of the CIT (determined based on the operators experience) was used to decide to either reduce the amount of injected water or even to perform a controlled emergency shutdown, in an attempt to avoid this flame out.

The injection temperature could also be varied by changing the water flow rate by adjusting the pump speed. Indeed, given that the water circuit is a closed circuit (besides the small amount of feedwater added to compensate for the amount of evaporated water), changing the flow rate will affect the outgoing temperature of the saturation tower. This in combination with the changing convective heat transfer coefficient for the water in the economizer will change the actual water injection temperature. Additionally, two air heaters to dispose the thermal power produced by the mGT during operation in CHP mode are available to provide additional cooling of the water (especially during start-up, when the water flow rate is still limited). Changing the rotational speed of the ventilator of the air heaters will thus also affect the water temperature. However, during full water injection mode (full mHAT mode), these air heaters should be shut down. As will be shown later in the experimental data presentation, the size distribution of the droplets and injection water flow rate (and to a lesser extend the water injection temperature) are essential to obtain optimal performance, stressing again the need for a proper control matrix.

No relative humidity sensor was installed on the VUB test rig [16], since, as indicated in the introduction, measurement of the relative humidity of compressed air is challenging and expensive. Relative humidity can be determined either directly or indirectly. Direct determination includes the measurement of the water content of the stream (in most cases based on the O_2 content on wet basis). Typical O_2 sensors for online monitoring are expensive, especially when pressure compensation is needed. Moreover, they are rather insensitive in the operating range for typical mHAT application (O_2 fraction ranging from 18 to 21 %, depending on the relative humidity). Alternative approaches, focusing on the measurement of the amount of evaporated water by measuring the water streams entering and leaving the saturation tower, do not provide a solution for online monitoring: given that the amount of evaporated water is small compared to the total water mass flow rate (2 to 4 %), which is typically within the measurement accuracy of the sensors. An alternative approach, applied by Xu et al. [12] and Pedemonte et al. [26], includes the accurate measurement of the feedwater flow rate while keeping the level of liquid water in the saturation tower constant. This approach gives, however, only an accurate result when applied over a longer period and is thus not applicable from a control strategy point of view.

Finally, as mentioned before, next to the full saturation of the compressed air, it is essential that no liquid water remains present in the working fluid before entering the recuperator. This additional water will lead to lower recuperator outlet temperature, thus lower CIT. In that case, more fuel needs to be injected in the combustion chamber to compensate this lower CIT with an electric efficiency decrease as results [27]. As highlighted by Pedemonte et al., the relative humidity of the compressed air leaving the saturation tower typically exceed 100 % — experimental results indicated an average relative humidity of 110 % for all 126 conducted tests [26] —, which highlights that indeed liquid water is present. In an attempt to obtain the relative humidity of the performed wet experiments on the VUB test rig using SVR, the authors of this paper came to a similar conclusion, where even humidity levels of up to 150 % have been calculated [22]. As will be shown later when analyzing the electric efficiency, a too high water injection leads indeed to a negative impact and must be avoided. Hence, using a control strategy based on efficiency optimization by adjusting the water injection network parameters, as is the aim of the future work, offers a solution.

RESULTS

In this result section, first the analysis of the experimental results of the global cycle performance are presented, highlighting the impact of an incomplete saturated working fluid (*Under-saturation*) or fully saturated working fluid still composing liquid droplets leaving the saturation tower (*Over-saturation*). Second, the specific performance of the saturation tower for the different operating points was analyzed, together with the used control settings in an attempt to provide some preliminary indications on the optimal control strategy.

Global cycle performance

When looking at the obtained corrected electric efficiencies ($\eta_{el,corr}$) as function of the corresponding relative humidity (ϕ), no trend at all can be observed (Figure 4), which is not in agreement with what would be expected. As explained in previous work of the authors [22], the measured electric efficiency for the different test ($\eta_{el,meas}$) was corrected for the variations of the inlet air temperature of the compressor as well as for the produced electrical power, using following equation:

$$\eta_{\text{el,corr}} = \frac{\eta_{\text{el,meas}}}{\eta_{\text{el,ref}}}. \tag{1}$$

For this correction, the technical data provided by the manufacturer was used to determine the corresponding reference electric efficiency ($\eta_{\text{el,ref}}$) for the specific inlet air temperature and selected power output [21]. The relative humidity (ϕ) is defined as follows:

$$\phi = \frac{p_v}{p_{\text{sat}}(t)}, \tag{2}$$

where $p_{\text{sat}}(t)$ is the water saturation pressure at the (measured) temperature of the working fluid, while p_v is the vapor pressure of the water present in the working fluid. This vapor pressure is determined based on the equation linking absolute humidity x with relative humidity ϕ:

$$x = \frac{p_v}{p_{\text{tot}} - p_v} \frac{M_{water}}{M_{\text{air}}} = 0.622 \frac{\phi p_{\text{sat}}(t)}{p_{\text{tot}} - p_{\text{sat}}(t)}, \tag{3}$$

where p_{tot} is the total pressure (measured) of the working fluid, M_{water} and M_{air} the molar mass of water and air respectively. Finally, the absolute humidity x is calculated using:

$$x = \frac{\dot{m}_{\text{water}}}{\dot{m}_{\text{air}}}, \tag{4}$$

where $\dot{m}_{\text{water}}$ and $\dot{m}_{\text{air}}$ are the respectively water and air mass flow rates. These mass flow rates were obtained using measurements of the compressor operating point (inlet and outlet temperature and pressure), compressor operating map, fuel mass flow rate and finally the composition of the flue gases, according to the calculation method previously developed by the authors [22].

As highlighted in this previous work [22], no exact assessment of the uncertainty on this approach could be performed, since the accuracy of the operating maps is unknown; however, an early adoption of this method to predict the air mass flow rate under steam injection indicated a relative high uncertainty on this mass flow rate of 4 %. This can explain the rather large scatter on the data (and was the main reason for the use of a SVR method for the direct determination of the relative humidity [22]). However, the large uncertainty cannot explain the absence of a clear trend between relative humidity and efficiency, since the amount of water introduced in the mGT cycle will determine the produced power output as well as the electric efficiency [28]. Hence, the explanation for the missing trend can be found in the performance of the mGT system itself. As highlighted in [29], the Turbec T100 mGT test rig displays a different part load behavior compared to what is reported by the manufacturer. Indeed, at reduced power output, first a slight increase (with a maximum at 90 kW_e) in efficiency is observed, followed by a mild decrease (until 65 kW_e where 80 kW_e should achieve similar performance as operating at the nominal load of 100 kW_e) and finally a steep decay starting from 65 kW_e [21]. For the mGT at VUB, a more or less linear decrease in efficiency with decreasing requested power output was observed and validated in the past [29]. Hence, this discrepancy will not allow to only compare $\eta_{el,cor}$ for different ϕ to study any possible trend. A more precise discussing comparing data of similar operating point is thus required to observe and explain these potential trends.

To allow for a more accurate assessment of the impact of the relative humidity on the mHAT performance, the experimental results were analyzed for each test performed at constant rotational speed (Figure 5(a) and (b)) and constant power output (Figure 5(c) and (d)) individually. The data obtained under humidified conditions was sorted in three categories: $\phi < 0.8$ or *under-saturated*, corresponding to tests operated with insufficient mass transfer leading to incomplete saturation of the working fluid; $0.8 < \phi < 1.2$ or *fully saturated*, corresponding to tests with a (nearly) complete saturation of the working fluid, but a minimal (or no) liquid water present in the working fluid; and $\phi > 1.2$ or *over-saturated*, corresponding to fully saturated working fluid, with a clear liquid water fraction present. The majority of the tests performed at constant rotational speed could be categorized as *fully saturated* (Figure 5(a)), with also a significant amount *over-saturated* (e.g. some tests even displayed a relative humidity exceeding 200 %). On the other hand, most tests at constant power output can be categorized as *under-saturated* (Figure 5(c)), which can be explained by the fact that the initial tests with limited water injection on the test rig have been conducted at constant power output, and afterward a switch towards constant rotational speed was made, once the behavior of the test rig was better known and understood.

Results at constant rotational speed highlight indeed that running in humidified mode, the electrical power output (P_{gen} will increase (as expected, Figure 5(a)), as well as the electric efficiency ($\eta_{el,corr}$, Figure 5(b), corrected values have been used to remove the effect of the altering inlet air temperature). Operation at *over-saturated* conditions does not directly affect the power output, but leads to a reduced electric efficiency (explanation is discussed later). When selecting the option of constant power output, although having fixed the requested power output, only under humidified conditions this requested power output could be achieved (Figure 5(c)), due to the limitations of the control system (see [16] for the full explanation). Finally, despite only a minor amount of runs could be categorized as *fully saturated*, one could observe again that full saturation leads to superior electrical performance compared to *under* or *over-saturation* (Figure 5(d)).

To explain the worse performance of the mHAT when there is *under* or *over-saturation* of the working fluid, an analysis of the temperatures in the cycle must be made (Figure 6)[2]. A first parameter to look at is the temperature of the working fluid that is entering the recuperator ($T_{rec,in}$, Figure 6(a)). When the working fluid is not entirely saturated, this temperature remains significantly higher than the temperature in *fully* and *over-saturated* cases. Due to the higher temperature, less heat is recovered in the recuperator, explaining the lower electric efficiency. For the *fully* and *over-saturated* cases, no clear

[2]For simplicity and clarity, both data taken in constant rotational speed or constant requested power output mode are plotted on the same figure, since no distinct difference could be observed between both operation modes in the analysis. Experimental results are plotted as function of the rotational speed, which represents the operating point of the compressor, and thus of the mGT, best for both operating modes.

difference in terms of recuperator inlet temperature can be observed. However, the main difference can be found here in the CIT (Figure 6 (b)). Due to the presence of liquid water in the working fluid, more heat needs to be exchanged to first evaporate this water and then increase the temperature. Since the surface of the recuperator is fixed, this results in a significantly lower CIT than in the *fully saturated* case, which in turn is below the dry case. As highlighted when analyzing the recuperator performance, more heat is exchanged in the humidified cases (increased heat flux) [22]. However, given the fixed surface, the effectiveness reduces, displaying lower CIT. With a larger recuperator, the CIT could be increased further, leading to even more improved performance [22]. This increased heat flux in the recuperator is also displayed in the lower temperature of the flue gases leaving the recuperator ($T_{\mathrm{rec,out}}$, Figure 6(c)). Especially in *over-saturated* cases, where the temperature drops the most, leaving less heat available for the economizer.

Saturation tower performance

Next to the outlet humidity level of the working fluid from the saturation tower, the evolution of the water temperature over the saturation tower is a second parameter to assess the performance of the saturation tower. As highlighted before, in ideal operating conditions, part of the evaporation enthalpy is taken from the water flow, leading to an energy transfer and thus a temperature reduction, as was observed in counter-flow saturation towers [14, 13]. Looking at this temperature evolution of the water between injection and extraction from the saturation tower ($\Delta T_{\mathrm{w,sat}}$, Figure 7), one can observe that there is indeed a minor temperature decrease. This temperature decrease is maximal, as expected, for the *fully saturated* cases. For *under-saturation* and in some case of *over-saturation*, there is indeed some transfer; however, the temperature decrease is limited. Finally, the remainder of the *over-saturation* cases even lead to a water temperature increase over the saturation tower (negative difference). This indicates that in that case, no sensible heat is transferred from the water to the compressed air, thus not fully exploiting the potential of the mHAT (see later).

For optimal performance of the mHAT, all thermal power recovered from the flue gases in the economizer needs to be transferred to the working fluid in the saturation tower [7]. Based on the temperature difference measured in the water circuit, the recovered thermal power from the flue gases, Q_{T100}, and exchanged between the water and the working fluid in the saturation tower, Q_{sat}, could be calculated (Figure 8). As mentioned before, in the water circuit, behind the saturation tower, two air heaters were installed to dump the thermal power of the mGT when operating in CHP mode. By varying the speed of the ventilator of the air heater, the exchanged thermal power could be controlled. In case the saturation tower performs optimal, all thermal power of the water is exchanged in the saturation tower and hence, there is no power exchanged in these air heaters. However, during wet operation, it was noticed that the temperature in the water circuit was increasing, meaning not all power could be exchanged in the saturation tower. Therefore, the air heaters were used to dump the excess power, ensuring steady-state operation.

Results of the calculated thermal powers indicate that under *fully saturated* conditions, indeed the maximal thermal power exchange takes place (Figure 8(a)). However, even in these cases, not all thermal power produced by the mGT (Figure 8(b)) is exchanged, leading to a non-zero heat dump to the atmosphere in the air heaters for constant temperature control (Figure 8(c)), indicating that the saturation tower is not recovering the entirely waste heat. Two explanations can be given: first, due to the cross-current injection, the evolution of the water temperature in the saturation tower is already different. The water is injected along the axis of the tower perpendicular to the air stream at constant temperature, while at the same time, the temperature of the air is dropping. Hence, the potential for actually reducing the water temperature along the tower is already limited: there is no real operation along the saturation line making that the water temperature could not be reduced until the wet bulb temperature of the incoming compressed air. Second, the bottom of the tower served as water reservoir (as is the case for counter-flow saturator, with or without packing material). In a counter flow arrangement, this water gets only minimally in contact with the hot compressed air, while in this case, air is moving parallel with the liquid water, leading to some convective heat transfer between air and water and thus a possible reheat of the water.

The importance of the correct functioning of the saturation tower for optimal performance is highlighted again when looking what happens when the working fluid is not fully saturated (*under-saturation*) or liquid droplets are still present in the working fluid (*over-saturation*). Indeed, in *fully saturated* conditions, there is some exchange of sensible heat between the water and the working fluid in the saturation tower. However, looking at the total available thermal power in the flue gases (Q_{T100}), only a small fraction is recovered and most is dumped by the air heaters. The *under-saturation* is also clearly reflected in the total available amount of thermal power, Q_{T100}, which is only slightly lower than in dry conditions due to the very limited extra heat exchange in the recuperator (see before). Finally, when the working fluid is leaving the saturation tower in *over-saturated* conditions, there is, as mentioned before, no sensible heat transferred from the water to the compressed air ($Q_{\mathrm{sat}} \approx 0$ or even negative). The reason for this can be found in the fact that the heat that would normally be exchanged in the saturation tower is already recovered before in the recuperator (evaporation enthalpy for the liquid fraction). As explained in previous subsection, this leads to a lower temperature of the flue gases when they enter the economizer and thus penalizes the energy recovery to the water (lower Q_{T100}). Since there is indeed some additional heat recovery in the *over-saturated* case, an electric efficiency increase is observed (Figure 4). However, since this recovery does not occur via the most optimal pathway from an exergetic point of view, i.e. evaporation at variable temperature in a saturation tower, the efficiency increase is lower than in the *fully saturated* conditions.

As last part of the experimental data analysis, the water circuit control parameters used during the different humidified test, including water flow rate (Q), injection pressure (Δp), and water inlet temperature ($T_{\mathrm{w,in}}$), have been analyzed (Figure 9). Although the system was designed for a nominal water flow rate of $10\,\mathrm{m^3/h}$, the highest performance is obtained at a reduced flow rate (around $5\,\mathrm{m^3/h}$). A higher injection rate leads to more droplets leaving the saturation tower still in liquid phase, resulting in *over-saturated* conditions. Concerning the injection pressure, clearly a higher pressure is requested, since this leads to a better pulverization of the liquid droplets (Figure 3(b)), hence more contact area. Very small injection pressures lead to thicker droplets (Figure 3(a)) with less contact area and an *under-saturation*. Finally, the impact of *over-saturation* can be seen on the water injection temperature. As mentioned before, due to the lower temperature of the flue gases when leaving the recuperator, less heat is available in the economizer to heat the water, leading to lower water injection temperatures. A lower injection temperature is nefast for the evaporation of the water in the saturation tower.

From these data, some general observation can be made to optimize the performance; however, it highlights again the real need for a specific control matrix, that helps setting the saturator control parameters for any set of operating setpoint (constant requested power output or rotational speed) and inlet conditions (inlet air temperature). Therefore, for future work, it is foreseen to construct such a control matrix.

CONCLUSION AND FUTURE WORK

In this paper, we presented the first but essential step towards the development of a control strategy for a Turbec T100 mGT-mHAT test rig. Previous work had indicated that online measurement of the relative humidity of the working fluid leaving the saturation tower could not be done with great accuracy. Moreover, when it would be determined by means of an indirect method, e.g. SVR method, accurate data on how to improve the performance is still missing. The aim of the control strategy is then to optimize the mHAT performance without having actual knowledge on the conditions of the working fluid leaving the saturation tower, but only focusing on the global cycle performance. In this work, we present the analysis of the electric efficiency improvement, and more generally the global cycle performance, as function of the obtained degree of saturation of the working fluid.

Using experimental data, the global performance, depending on the operating point i.e. constant rotational speed or constant power output, as well as the specific performance of the saturation tower were assessed. Experimental results indicated that complete saturation is necessary to obtain maximal electrical performance. Under these conditions, a maximal amount of waste heat is transferred from the water to the working fluid in the saturation tower. When the working fluid is not fully saturated, but also when it is over-saturated, i.e. the compressed air is fully saturated and liquid droplets are still present, the electrical performance of the mHAT is increased less due to a too limited waste heat recovery in the first case and a limited contact area of the recuperator withholding essential additional heat recovery for water evaporation for the latter case. In both cases, the full potential of the mHAT cycle is not exploited. When analyzing the specific performance of the saturation tower, it was confirmed again that only under fully saturated conditions, a maximal amount of waste heat is transferred from the hot water from the economizer back to the cycle working fluid in the saturation tower. When the working fluid is not entirely saturated, a fraction of the waste heat is recovered, but due to the incomplete humidification, less heat is recovered in the recuperator, leading to increased loss of thermal power through the stack. When the working fluid is over-saturated, there is no net heat exchange between the water and the compressed air in the saturation tower as a results of the reduced available heat in the flue gases when entering the economizer. Finally, analysis of the used operational control settings for the water injection system shows that optimal mHAT performance can be obtained when the injection pressure is maximized (to ensure a good pulverization of the air) and aiming at a water flow rate of around $5\,\mathrm{m}^3/\mathrm{h}$. However, due to the significant scatter on the results, no clear control settings could be achieved.

Future works involves the complete work on data production for monitoring of the mHAT performance, including data collection, pre-processing (clustering and noise estimation) and data production/forecasting with an experimental validation. The specific goal is the development of an online control strategy for optimal mHAT performance with potentially the development of a virtual *humidity sensor*. Once successfully implemented, this work will be extended with as ultimate target the development of a digital twin for the humidified mGT-VUB test rig.

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List of Figures

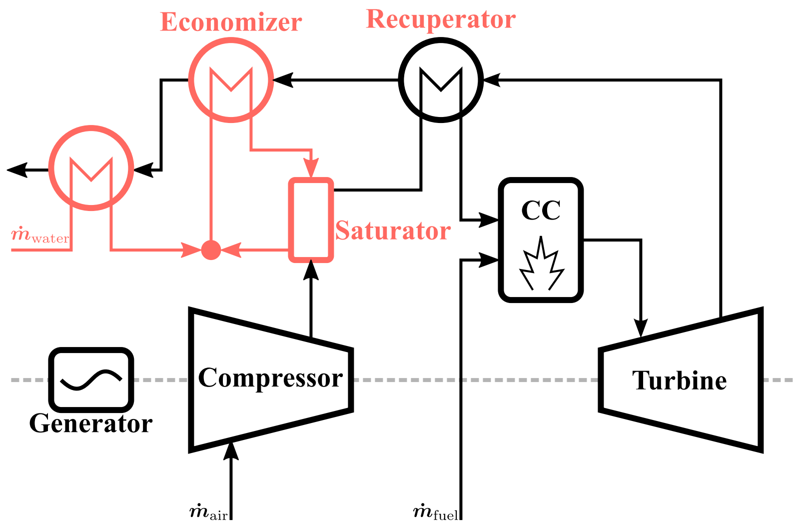

Fig. 1. The classical mGT (black parts) is converted in an mHAT by introducing a saturation tower in the cycle between compressor and recuperator (red parts).

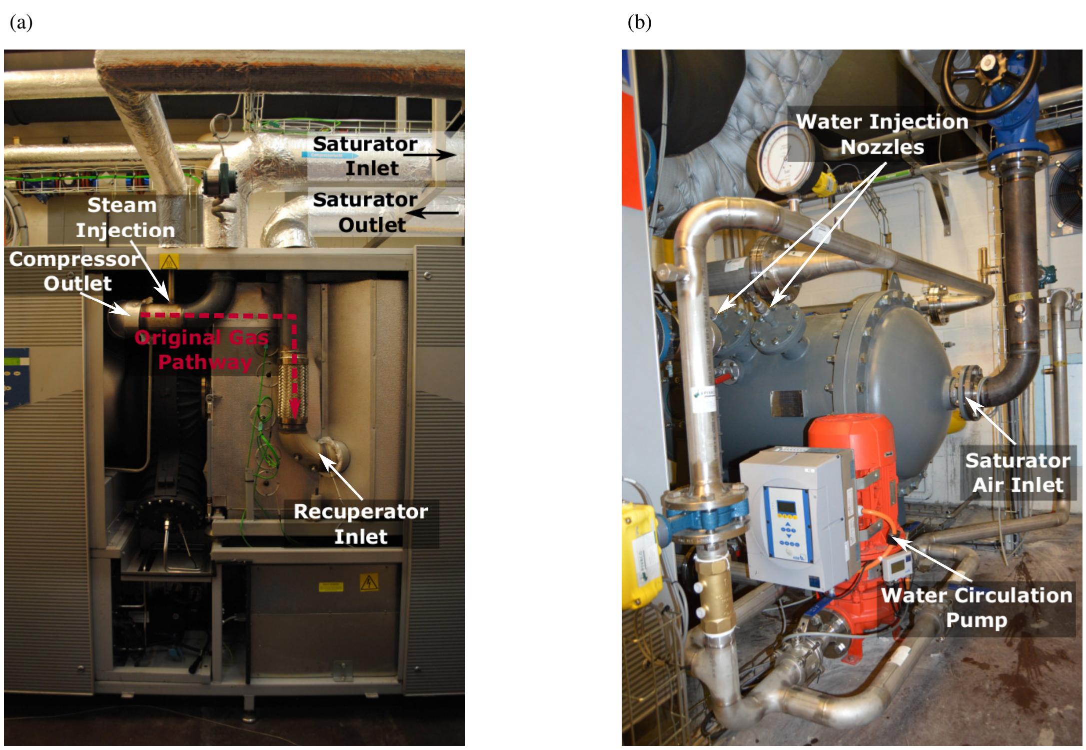

Fig. 2. Pictures of the integration of the saturation tower in the mGT cycle in between the compressor outlet and recuperator inlet of the Turbec T100 mGT series 2 (a) and of the saturation tower with circulation pump, valves and sensors (b).

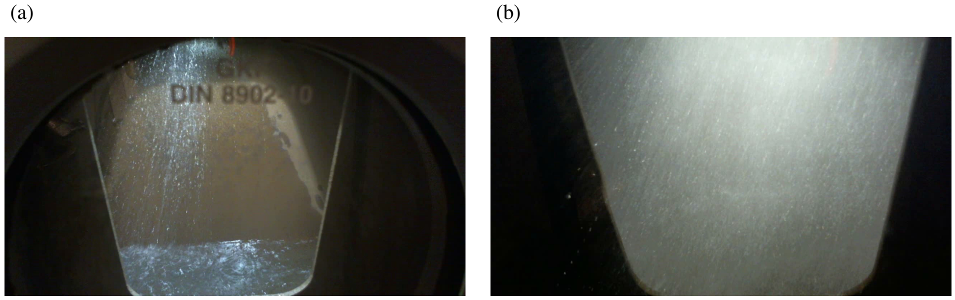

Fig. 3. The droplet size and distribution was monitored visually through the saturation tower loop holes. The injected water flow rate and droplet distribution from (a) corresponds to the low flow rates and low injection pressure, while (b) shows the water flow rate and distribution after increasing the injection pressure.

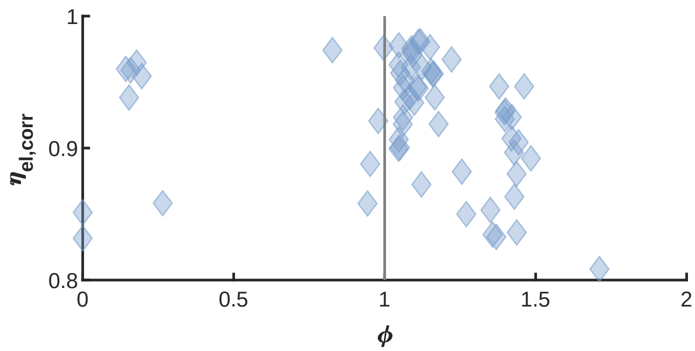

Fig. 4. When comparing the obtained corrected electric efficiency ($\eta_{el,corr}$) as function of the corresponding relative humidity of the working fluid (ϕ) exiting the saturation tower, unlike what would be expected, no trend can be observed.

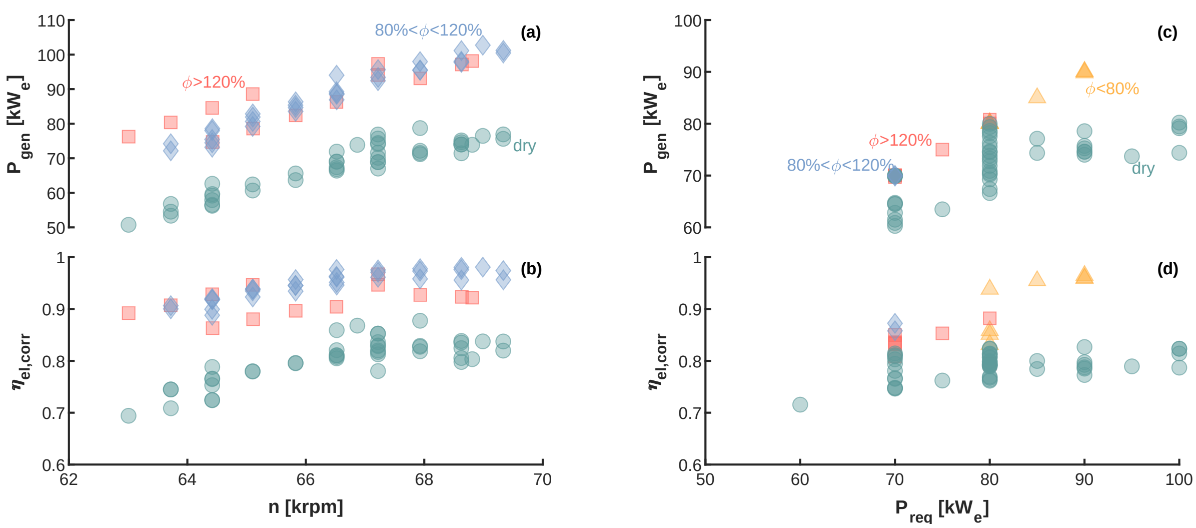

Fig. 5. Operation at incomplete saturation of the working fluid leaving the saturation tower ($\phi < 0.8$), as well as over-saturated conditions with liquid water present in the working fluid ($\phi > 1.2$), result in reduced electrical performance of the mHAT at constant rotational speed, n (reduced generated electric power, P_{gen} (a), and reduced electric efficiency, $\eta_{el,corr}$(b)) and at constant requested power output, P_{req}: although the requested power output could be generated in all humidified cases, even at under- or oversaturation, which is not the case under dry conditions, reduced electric efficiency, $\eta_{el,corr}$ observed (d).

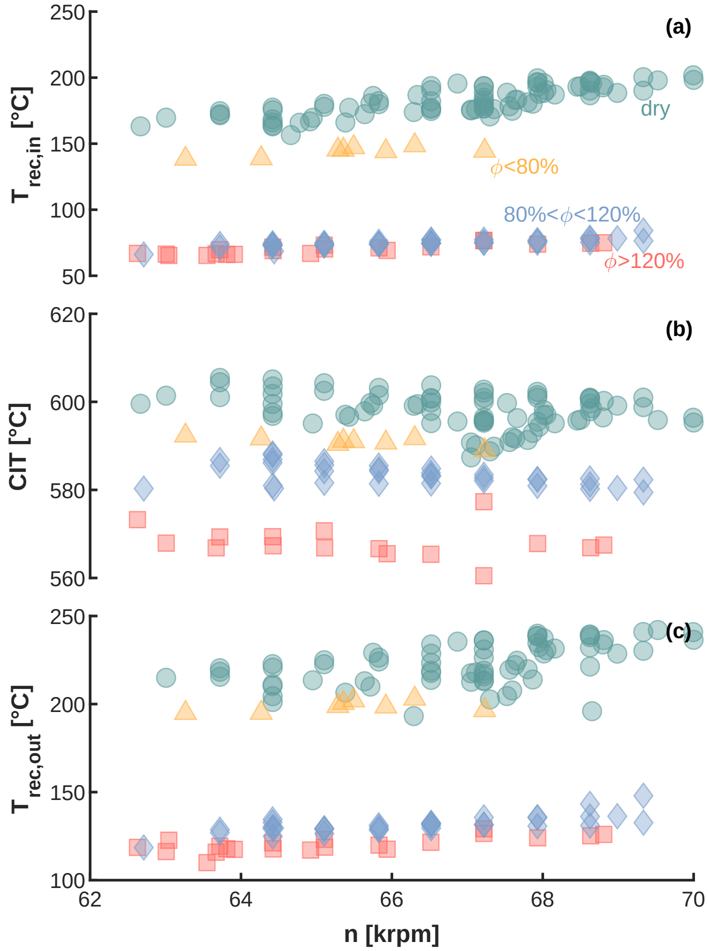

Fig. 6. Analysis of the temperatures in the mHAT cycle allows to explain the performance under the different conditions: *over-saturation* leads to highly decreased recuperator inlet temperatures $T_{\mathrm{rec,in}}$ followed by decreasing CIT (b) and flue gas recuperator outlet temperatures (c) due to the limited surface of the recuperator. *Under-saturation* leads on the other hand to significant higher recuperator inlet temperatures (a), having a reduced heat recovery in the recuperator as result (increased recuperator outlet temperature (c)).

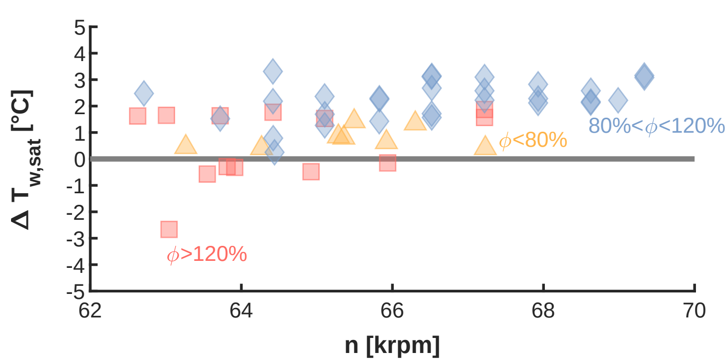

Fig. 7. Analysis of the temperature evolution of the water injected in the saturation tower highlights the importance of a proper operation of the tower: Only in the *fully saturated* case, $80\% < \phi < 120\%$, actual heat was recovered from the water flow (positive temperature difference).

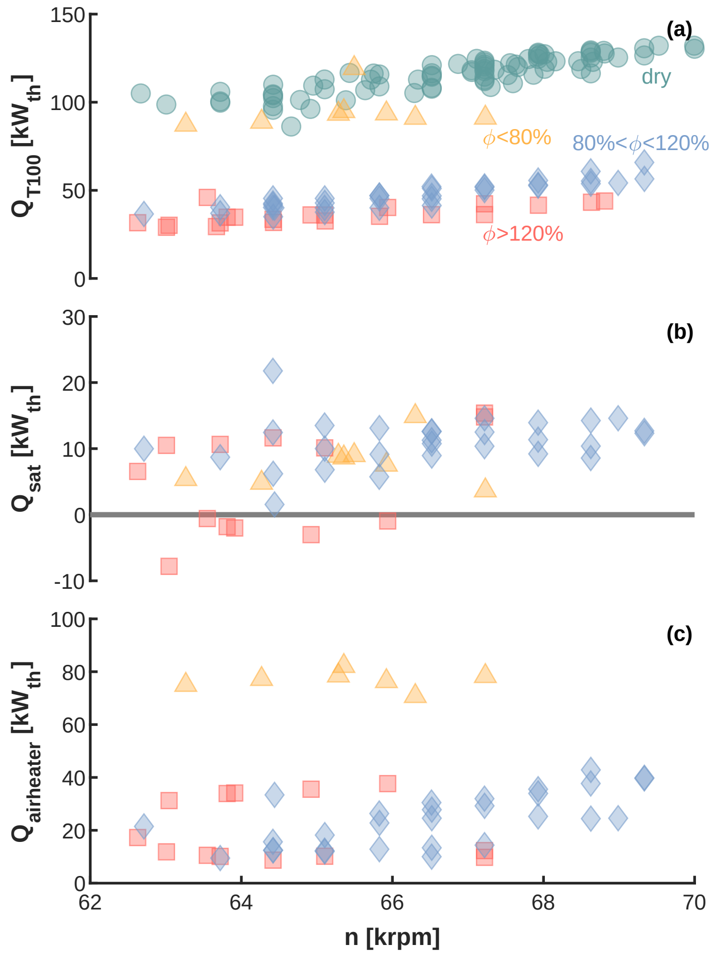

Fig. 8. Analysis of the thermal power recovered in the economizer, Q_{T100} (a), exchanged in the saturation tower, Q_{sat} (b), and dumped in the atmosphere using air heaters, Q_{airheat} (c), highlight that under *fully saturated* conditions, the largest transfer between the hot water and the mHAT working fluid is effected, explaining the superior performance of the mHAT, while in *over-saturated* conditions, a zero net-exchange occurs in the saturation tower.

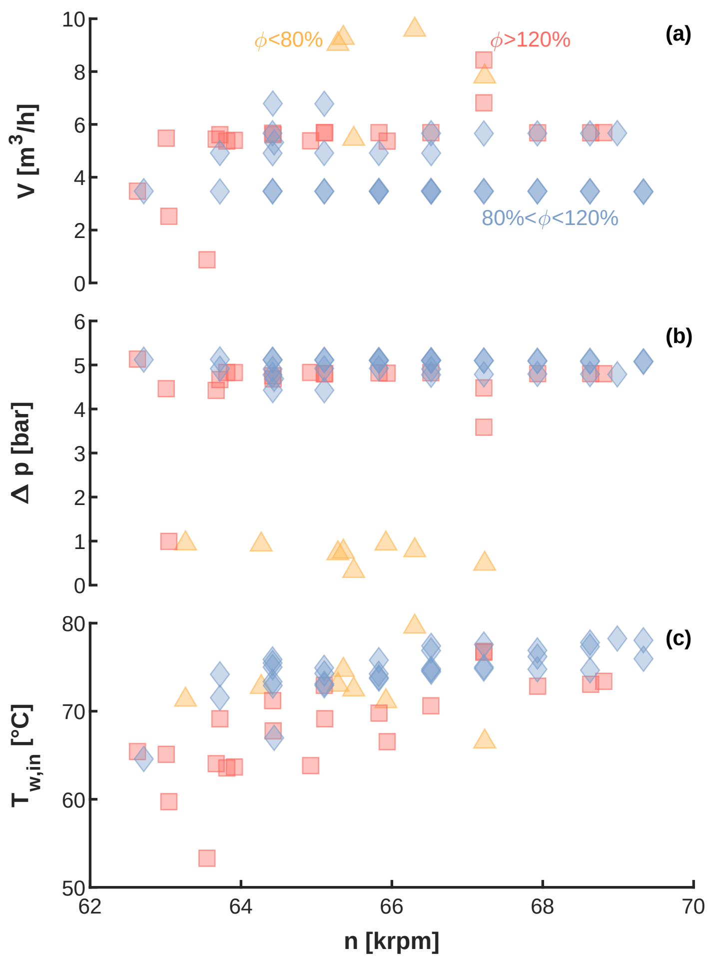

Fig. 9. Experimental results indicate that a maximal injection pressure, Δp (b), is needed to obtain optimized performance, while the water flow rate (V) should not exceed $5\,\mathrm{m^3/h}$ (a). For the water temperature ($T_{\mathrm{w,in}}$), in general, a higher injection temperature is favored (c).