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HUMIDIFICATION IMPACT ON THE PERFORMANCE IMPROVEMENT OF A NOVEL TWO-SHAFT MICRO GAS TURBINE: THERMODYNAMIC CYCLE PERFORMANCE ASSESSMENT

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ABSTRACT

Despite appearing as a promising technology for small-scale decentralized Combined Heat and Power (CHP), the relatively low electrical efficiency of micro Gas Turbines (mGTs) prevents them from being attractive to users with variable heat demand. Transforming the cycle into a micro Humid Air Turbine (mHAT) by adding a saturation tower in the cycle allows for an increase in the electrical efficiency of these units in moments of low heat demand. Although humidification is well studied and proven effective on the simple recuperated Brayton cycle mGTs, its potential for cycle performance improvement when applied on more advanced mGT cycles is currently unknown. Therefore, the numerical study presented in this paper aims to assess the impact of humidification on the performance of a novel two-shaft mGT from MITIS, exploiting the intercooled regenerative reheat gas turbine cycle concept. The benefits of water injection mostly rely on the increased heat capacity of the air-vapor mixture, and the more significant amount of heat recovered in the recuperator, both resulting a the lower fuel consumption. Simulation results show indeed that by introducing a saturation tower in this two-shaft turbogenerator system, waste heat is recovered, leading to increased electrical efficiency from 35.12% for the mGT cycle to 36.31% for the mHAT cycle while providing a flexible heat and power output. This rise in the efficiency is maybe small, but could be increased further by going towards more advanced cycle configuration (aftercooling) as well as by less limiting the TIT.

Keywords: CHP, electrical efficiency, Humid air turbine, micro gas turbine, simulation

NOMENCLATURE

Acronyms

AC	Air compressor
CC	Combustion chamber
CHP	Combined heat and power
ECO	Economizer
ELE	Electricity
GT	Gas turbine
HP	High pressure
IC	Intercooler
LHV	Lower heating value
LP	Low pressure
mGT	Micro gas turbine
mHAT	Micro humid air turbine
REH	Reheater
SAT	Saturator
TIP	Turbine inlet pressure
TIT	Turbine inlet temperature

Symbols

k	Heat capacity ratio

Greek symbols

η	Efficiency
π	Pressure ratio

Subscripts

Is	Isentropic
Turb	Turbine

1. INTRODUCTION

As a distributed energy technology, micro Gas Turbines (mGTs) are becoming increasingly important as an integral part of the heating, cooling, and electrical power industry. These small gas turbines can be used for small-scale power generation alone or operated as combined heating and power (CHP) systems with cogeneration efficiency of up to 80% (30% electrical plus 50% heat efficiency) for a 100 kW_e size engine [1]. However, these efficiencies are only reached when the heat in the exhaust gases is entirely used for external heating purposes. When there is no or low heat demand, the heat produced by the cogeneration unit has to be rejected. This means that the overall efficiency of the CHP system is reduced to the electrical efficiency of the unit. This negatively affects the economic performance of the units, even forcing the unit to shut down.

Micro Gas Turbines (mGT) humidification is a promising technique that can enhance electrical efficiency during periods of low or no external heat demand [2]. By introducing hot water or steam, which is auto-raised using the heat from the flue gases, additional mass can be added to the gas turbine cycle, increasing the produced electrical power and thus specific power output. Moreover, when humidification occurs in such a manner that waste heat is recovered from the flue gases and reintroduced in the cycle (via the humidification), the electrical efficiency of the cycle can be improved, offering a solution for the operation when there is no demand for heat [3].

Among all the various options available for the humidification of gas turbines, including water injection that fully evaporates, steam injection, and water injection in a saturation tower, the latter one, also known as the Humid Air Turbine (HAT) cycle (i.e., water injection in a saturation tower, as suggested by Rao and Day [4]) proposes the highest potential electrical efficiency increase according to the research carried out by Jonsson and Yan [5]. Many researchers have studied numerically and experimentally the positive impact of water injection by a saturation tower in gas turbines, known as the HAT cycle [6-12].

Nyberg et al., for instance, assessed different possible HAT cycle structures, focusing on the optimal introduction point and temperature of the make-up water, showing that the optimal position is subject to the design parameters of the compressor and the water circuit [6]. Furthermore, von Heiroth et al. performed static modelling to analyze the steady-state operation of a gas turbine with a recuperator and a humidifier at different loads of the HAT cycle in the Laboratory of Heat and Power Engineering at the Lund Institute of Technology discussed in [7]. Traverso et al. focused rather on the saturator, by studying experimentally a pressurized humidification tower, with structured packing over 162 working points, covering a relatively wide range of possible operating conditions [8]. They showed that the saturator behavior, in terms of air outlet humidity and temperature, is primarily driven by, in decreasing order of relevance, the inlet water temperature, the inlet water over inlet dry air mass flow ratio and the inlet air temperature. Furthermore, they successfully correlated the experimental results using a set of new non-dimensional groups: able to capture the air outlet temperature.

However, despite the large amount of numerical works on the HAT cycle (for a full overview, we refer to the review paper of Jonsson and Yan [5]), only one small-scale humid air gas turbine pilot plant, including a water circuit and flue gas condensation system with a power output of 600 kW, has been constructed and tested at Lund University in Sweden since the development of the HAT cycle layout, achieving an electrical efficiency of around 35% [9]. After that, different variants of the HAT cycle have been proposed, such as the Cascaded Humidified Advanced Turbine (CHAT) [10] and the Advanced Humid Air Turbine Cycle (AHAT) [11]. Nakhamkin et al. [10] showed that the CHAT power plant could be offered with specific capital costs up to 20 percent lower than the combined cycle plant, and with competing efficiency. Compared to a combined cycle plant, the CHAT plant offers lower emissions (due to air humidification) and other significant operating advantages with regard to start-up time and costs, better efficiency at part load, lower power degradation at higher ambient temperatures, and simpler operations and maintenance due to elimination of the complexities and costs associated with steam production. The AHAT system, differentiating from the HAT cycle by the implementation of Water Atomized inlet Cooling (WAC) replacing the inlet cooling, has been studied by Araki et al. [11] to improve operational flexibility and thermal efficiency of the gas turbine power generation system. However, neither the HAT cycle concept nor any of its variants have been commercialized to date.

The impact of humidification on the electrical performance of single-shaft mGTs has also been studied in the literature. Lee et al. [12] compared the performance of a mGT with an injection of hot water and steam generated through the same heat recovery unit at two locations in the mGT cycle: in the recuperator inlet line and the combustor. Their results showed that the injection at the recuperator inlet exhibits a higher efficiency than injection at the combustor for both water and steam injection cases. Also, they indicated that steam injection provides a higher power generation efficiency than water injection. The research group of the last author of this paper [2] presented an exhaustive review of existing methods for humidifying the recuperated mGT cycle with a focus on the advantages and disadvantages of each option, technology challenges, and economic potential. In their other work [13], they studied the impact of different advanced humidified mGT cycles on the mGT performance. The results pointed out that depending on the cycle layout, more or less waste heat could be recovered and, in all cases, led to higher electrical output and reduced fuel consumption, resulting in increased electrical efficiency. They also indicated that the micro Humid Air Turbine (or mHAT, as proposed by Parente et al. [14]) was the most promising cycle, combining electrical efficiency increase with relatively limited cycle alterations. Furthermore, Zhang et al. [15] studied the coupling effect of key variables of this mHAT cycle, which uses a water-air mixture as working fluid, on the specific power output, heat recovery, and electrical efficiency. The interactive variables considered in their research are water-air ratio (W/A), regenerator effectiveness (RE), and after-cooler effectiveness (AE). Their results are presented for the identical and different TIT and pressure ratios (PR). They conclude that as the pressure ratio increases, the W/A and AE have a major impact on the HAT cycle efficiency, and the influence of RE on the HAT cycle efficiency considerably weakens. The HAT cycle efficiency, specific work output, and heat recovery achieved values of around 44%, 556 kJ/kg, 5505 kW, where its corresponding TIT, PR, W/A, AE, RE values are 1280°C, 8, 1.5, 0.75, 0.9, respectively. Finally, Montero Carrero et al. [16] showed by using Sankey and Grassman diagrams the impact of humidification on the electrical efficiency of a single-shaft micro gas turbine, the Turbec T100 mGT. Their study showed that the electrical efficiency of the T100 increases by around 1.5% by water injection, while the total exergy efficiency decreases by about 5%. However, while there is an enthalpy gain in the saturation tower; exergy decreases in this component due to the increase in entropy related to the evaporation of water. Hence, the beneficial effect of water injection in the mHAT is primarily due to the rise in the heat capacity of the air-vapor mixture, leading to a more significant amount of heat recovered in the recuperator and, thus, lower fuel consumption in the combustion chamber, finally leading to increased efficiency (when operating at constant power). Enhancing the waste heat recovery in the recuperator is thus crucial for maximizing mHAT performance [17].

On the experimental level, first, at Vrije Universiteit Brussel (VUB), a Turbec T100 mGT has been transformed into a mHAT by adding a spray saturation tower [18] in the cycle. Experimental results showed that the electrical efficiency of the mHAT increased by up to 4.2% due to the water injection at a fixed rotational speed [19]. Later, Zhang et al. [20] designed and installed a pressurized packing humidifier in a 100 kW mHAT system at Shanghai Jiao Tong University and developed a two-phase temperature and humidity measurement. Their study investigated the influence of water-air ratio and inlet temperature on the dynamic performance of the humidifier. In another study of this university [21], a simplified two-shaft HAT cycle gas turbine was built to experimentally investigate the performance of the HAT cycle at low *TIT* and low-pressure ratios. In their research, two different cases were considered for testing. In one case, the fuel flow rate was kept constant at 57 kg/h, while in the other case, the *TIT* was fixed at 665°C. In the first case, when the water-air ratio increased from 30 g/kg to 54 g/kg of dry air, the output power increased by 3 kW_e and *TIT* decreased by 20°C. In the latter case, increasing the air humidity ratio from 48 g/kg to 57 g/kg dry air increased the power output by approximately 10 kW_e. An mHAT cycle facility with a novel ceramic foam packing humidifier was built, showing that electrical efficiencies of around 31% could be achieved for an electrical output power of 80 kW [22].

Nevertheless, previously cited works on mGT humidification remained limited to single-shaft engines, exploiting the simple recuperated Brayton cycle layout. More advanced systems (e.g., two-shaft intercooled and reheated engines) have been excluded from these studies. Such a two-stage mGT is a type of gas turbine with two compression and expansion stages. The two-stage design leads to higher efficiency and power output than a single-stage turbine. This two-shaft configuration separates the compressor and turbine into two separate shafts, allowing for more precise control of the compressor and turbine speed and leading to superior part load performance. More specifically, given the existence of 2 compressor and turbine stages, intercooling and reheat become an option, enhancing the performance further. Finally, considering the still relatively small scale (<500 kWe), pressure ratios remain limited (typically below 8), still requiring the implementation of a recuperator for improved cycle performance. Like single shaft mGT, the two-stage mGT can be used in various applications, such as a generator only or as a combined heat and power production unit.

However, only a few studies are available on the performance assessment of a two-shaft micro gas turbine. Malkamaki et al. [23] introduced and optimized a 500 kW_e two-shaft mGT for a realistic combination of TITs, recuperation rates, and pressure ratios. The suggested mGT design aimed to achieve significantly increased performance within the range of mGTs and even compete with the efficiencies achieved in large industrial gas turbines. Further, Gaitanis et al. [24] developed a specific operating strategy for such a 2-spool system when being operated with different fuels. Nevertheless, so far, no studies have been performed on the feasibility of humidification using a saturation tower in this type of mGT.

The present article aims to fill this void by assessing the impact of humidification on a two-shaft, intercooled, recuperated and reheat mGT and evaluating its performance in terms of energy analysis. Due to their specific features compared to bigger scale gas turbines (radial turbomachinery, inclusion of a recuperator, limited turbine inlet temperature, and variable rotational speed), but also compared to single shaft mGTs (two pressure levels, allowing for alternative humidification method), they require dedicated investigation. Indeed, since most of the available literature focuses on single-shaft engines, we present a thorough study of the main thermodynamic properties at all points of the two-shaft mGT and mHAT engines in this paper. Furthermore, a comprehensive analysis of the energy flows between the components of the two-shaft mGT that uses a humidification unit is presented. Such analysis plays a crucial role in understanding the thermodynamic mechanisms that lead to an increase in electrical efficiency when water injection takes place, as indicated previously by Montero Carrero et al. [16] for the single spool engine. Hence, the goal of the study presented in this paper is to evaluate the advantages of water injection from an energy perspective to fully comprehend the effect that water injection brings in the two-shaft mGT cycle. To perform such analysis, the present paper introduces Sankey (enthalpy flow) diagrams, which visually summarizes the volume and direction of the energy flows in a process or cycle. Certainly, Grassmann diagram (exergy flow) and cost perspectives along with Sankey diagram could be useful for a detail investigation but these topics are out of scope of this paper and indeed would be addressed later.

The paper is structured as follows: first, the two-shaft mGT cycle is presented, followed by the mHAT cycle for this layout. Next, the used modelling approach is detailed, after which the results are presented and discussed. Finally, the conclusions about the impact of converting a two-shaft mGT into a mHAT are given.

2. TWO-SHAFT MGT CYCLE

The mGT and mHAT cycles investigated in this article are based on the MITIS micro-20 mGT. The micro-20 mGT cycle is a typical Brayton cycle with recuperation. It is characterized by two stages of compression with intercooling and two stages of turbines with a reheat (second combustor) between the high-pressure (HP) and low-pressure (LP) stages. The implementation of such a cycle by MITIS is schematically shown in Fig. 1.

The air is first compressed in a low-pressure radial compressor and, after passing through an intercooler and rejecting its heat (used for cogeneration applications), compressed further in a high-pressure compressor. Intercooling is essential to improve efficiency and performance. The compressed air from the high-pressure compressor is fed into the recuperator, where it is preheated by the exhaust gases from the low-pressure turbine before entering the natural gas-fired combustion chamber. The combustion products expand in a high-pressure radial turbine (driving the high-pressure compressor and a first generator). The exhaust gas from this high-pressure turbine is heated in the reheater by burning extra fuel to raise the temperature of the gases to a specified *TIT* for the low-pressure turbine. Noted that, the term ‘reheat’ in the ‘IRRGT (Intercooler, Regenerative, Reheat Gas Turbine)’ concept, relates to the fact that the flue gases are reheated after a first expansion, which is done through a combustion chamber. Since the combustion occurs lean in the combustor before the HP turbine, a significant oxygen fraction is left in the flue gases, and can be used as oxidizer in the reheat chamber. Hence, no additional air should be added in this reheater. Additional power is extracted from the flue gas in the low-pressure turbine, to drive the low-pressure compressor and a second generator. Finally, the remaining heat content in the exhaust gases leaving the low-pressure turbine is extracted in two components: first in the already-mentioned recuperator, to preheat the high-pressure compressed air, and second, in an economizer, where water is heated up to meet the heat demand of the CHP consumer.

3. PROPOSED MHAT CYCLE

To change the mGT into an mHAT cycle, a saturation unit is introduced and placed after the high-pressure compressor. This saturation unit allows the recovery of the available heat from the economizer, using humidification, when the consumer’s heat demand is low, leading to an increase in electrical efficiency. The mHAT investigated in the present paper is depicted in Fig. 2, where the traditional components of the mGT are displayed in grey. The water—previously heated in the economizer—is sprayed over the high-pressure compressed air in the saturation tower. As air goes up through the saturator, its relative humidity increases: during this process, both air enthalpy and air mass flow rise while heat is removed from the circulating water below boiling temperature. The original economizer is kept in the mHAT cycle; therefore, the water flow rate is fixed at 176 g/s. However, to fully humidify the air, only 6.57 g/s are required to evaporate in the tower. Hence, the remaining water is collected at the bottom of the saturator and re-routed towards the economizer.

It is important to note that for the study presented in this paper, we considered the simplified mHAT cycle and not the complete mHAT cycle, as presented by Parente et al. [16]. In Parente’s cycle, an aftercooler is added to the cycle for enhanced operation of the saturation tower. We opted not to include this component, considering the significant cost and pressure loss, significantly reducing the economic potential of the application. Furthermore, although the heat from the intercooler could also be used to heat the water in the saturation tower, as is the case for the HAT cycle, we opted not to do this at this stage. However, a better energy integration is foreseen in future work.

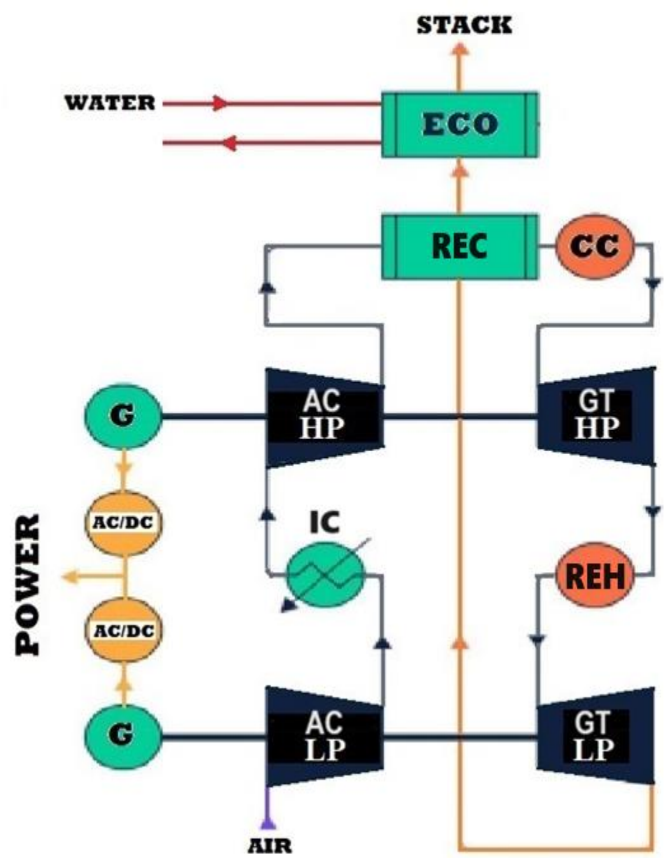

FIGURE 1: MITIS IMPLEMENTATION OF A TWO-SHAFT MGT

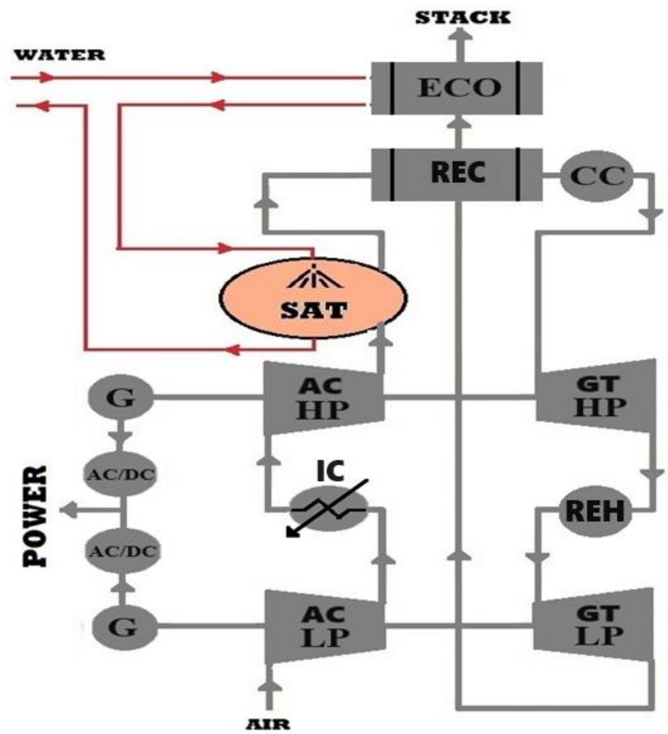

FIGURE 2: TRANSFORMATION OF TWO-SHAFT MGT (IN GREY) INTO A MHAT CYCLE THROUGH A SATURATION TOWER

4. CALCULATION MODEL AND ASSUMPTION

Both mGT and mHAT cycles have been simulated in Aspen Plus by adapting the models developed and validated by the last authors' research group [13,16]. The main assumptions considered for the simulation are described in this section and given in Table 1. The performance of the compressor is evaluated using a compressor map. This map allows for the calculation of the pressure ratio (P_r) and the isentropic efficiency (η) for both compressors as a function of the corrected air mass flow ($\dot{m}_{corr}$) and corrected rotational speed (N_{corr}):

$$P_r = \mathrm{f}_1(\dot{m}_{corr}, N_{corr}) \tag{1}$$

$$\eta = \mathrm{f}_2(\dot{m}_{corr}, N_{corr}) \tag{2}$$

Such functions could be numerically evaluated through a two-dimensional interpolation on the digitized maps. The corrected mass flow and corrected rotational speed are expressed in Eqs. (3) and (4), respectively:

$$\dot{m}_{corr} = \frac{\dot{\mathrm{m}}\sqrt{\frac{T}{T_{ref}}}}{\frac{P}{P_{ref}}} \tag{3}$$

$$N_{corr} = \frac{\mathrm{N}}{\sqrt{\frac{T}{T_{ref}}}} \tag{4}$$

TABLE 1. MAIN ASSUMPTION APPLIED FOR THE SIMULATION AND RESULTS OF CYCLE PARAMETERS

	mGT	mHAT
LP Compressor isentropic efficiency (%)	78.71	77.31
LP Compressor isentropic efficiency (%)	77.19	77.17
Compressors mechanical efficiency(%)	99	99
LP Turbine isentropic efficiency (%)	82.5	82.47
HP Turbine isentropic efficiency (%)	80	78.5
Turbine mechanical efficiency (%)	99	99
Pressure loss in intercooler (%)	2	2
Approach temperature difference in recuperator (C)	50	50
Pressure loss in recuperator (airside) (bar)	0.1	0.1
Pressure loss in recuperator (gas side) (bar)	0.04	0.04
Combustion/reheater pressure loss (%)	3	3
Saturation tower pressure loss (%)	-	0.5
Methane LHV (kJ/kg)	50035	50035

where T_{ref} and P_{ref} are the reference temperature and pressure, generally at ISO conditions of 15ºC and 1.013 bar. The overall turbine performance could be evaluated using the same approach. For this study, the original micro-20 mGT turbomachinery components are included in the model: The actual map of both the compressor (LP & HP) and the LP turbine are introduced in Aspen Plus (HP map was not available regarding both pressure ratio and isentropic efficiency and thus simulated in a different way, as indicated below). These maps calculate the isentropic efficiency for compressors and high-pressure turbine as indicated in Table 1 in such way that the net electrical power output is 27 kWe for both mGT and mHAT case. The mechanical efficiency for both compressors and turbines was set to 99%. Since the HP turbine map was unavailable, this turbine is assumed to be choked and considered to have an isentropic efficiency of 80% and a mechanical efficiency of 99%. For the mHAT case, the addition of water has a substantial effect on the properties of the working fluid, particularly on the heat capacity ratio (k), thus affecting the turbine choking constant, which is corrected according to the following equation [25]:

$$\frac{\dot{m}_{GT}\sqrt{TIT}}{TIP} = A\sqrt{\frac{k_{GT}}{R}\left(\frac{2}{k_{GT}+1}\right)^{\frac{k_{GT}+1}{k_{GT}-1}}} = cte \tag{5}$$

Identically, the isentropic efficiency of the high-pressure turbine is also corrected for the humidified cycles using a correction method. This is done with Eq. (6), where the apex (*) refers to the properties of standard dry air—as proposed by Parente et al. [16]. The final values of compressor and turbine isentropic efficiencies for the mGT and mHAT cycles are collected in Table 1.

$$\frac{\eta_{is}}{\eta_{is}^*} = \frac{k-1}{k^*-1}\sqrt{\frac{k^*+1}{k+1}\left(\frac{1-\left(\frac{1}{\pi}\right)^{k^*}}{1-\left(\frac{1}{\pi}\right)^{k}}\right)} \tag{6}$$

The combustion chamber is modeled as a classical Gibbs reactor with a 3% pressure loss. The fuel for both cycles is methane with an *LHV* of 50035 kJ/kg. A combustion efficiency of 100% (complete combustion) is assumed. Combustion instabilities related to the water injection have not been included in the model since the water fraction in the combustion air is limited to 4.7 wt%, well below the 30% limit for premixed combustion at which CO levels become too high to ensure stable and complete combustion [26]. Given the negligible effect of water injection in the combustion process, the pressure loss in the combustors is presumed to be the same for the mGT and mHAT cycles.

The recuperator is simulated as a counter-current gas-gas heat exchanger. In order to model the recuperator, it is needed to set a specification constraining the heat exchange in this component. In this case, we opted to fix the temperature difference between the hot inlet and the cold outlet of the recuperator, since there the minimal temperature difference is expected (considering the differences in mass flow and heat capacity, the temperature difference is the smallest between hot inlet and cold outlet, compared to the hot outlet and cold inlet temperature difference) . Its value has been set to 50°C, which is a typical value for gas-gas heat exchangers. Moreover, this value is in agreement with the maximum effectiveness of about 85% for the recuperator, which the manufacturer (MITIS) can achieve [12].

The main effect of water injection on the recuperator relates to the change in the heat capacity of the flow, a fact that is considered in the study. Considering pressure loss, there is less volumetric flow due to the decrease in the inlet temperature on the cold side of the recuperator, which translates to a slightly higher density. As the high-pressure turbine is choked, the flow rate will not change substantially; therefore, the pressure losses over the recuperator would slightly decrease with water injection. However, the current model selects a conservative approach with the 0.1 bar pressure loss over the air side and 0.04 bar over the gas side for both mGT and mHAT cycle to avoid overestimating the performance.

As proposed by Queiroz et al. [27], the saturation tower is modeled using the RadFrac module. It is assumed that the saturator introduces a pressure drop of 0.5%, which was the value De Paepe et al. determined when designing this component for the experimental mHAT unit placed at VUB laboratories [18]. Experiments in the T100 mHAT unit in the University of Brussels (VUB) laboratories confirm that the pressure loss over the saturator fits this design value [28].

In the simulation models, three control loops are implemented: Turbines Inlet Temperature (*TIT*) control, rotational shaft speed control, and water injection control. The first control loop adjusts the fuel flow rate going into the combustion chamber and reheater for specified *TITs* in the mGT cycle or mHAT cycle. The second control loop ensures mGT operation at constant power, by adjusting the setpoints for the *TITs* of the high-pressure and low-pressure turbines. This means that when water is introduced into the cycle behind the HP compressor, more power will be available on the shaft due to the mass imbalance between the compressor and the turbine, which leads to a power output increase. Therefore, the *TITs* will be lowered by the control system to keep the electrical power output constant for two-shaft mGT and mHAT at 27 kW_e. To keep the high-pressure turbine choked, inlet air mass flow rate should be decreased, which requires reducing the rotational speed of both shafts (both shafts are controlled individually). The final control loop sets the amount of introduced water into the cycle. The control system increases the feedwater flow rate till a maximum is reached. This maximal amount of water corresponds to the point where the economizer has reached its minimal pinch temperature (5°C), but all the other boundary conditions from Table 1 are still respected.

5. RESULTS AND DISCUSSION

The resulting temperature, pressure, mass flow rate, and enthalpy of each stream in the mGT and mHAT cycles, as well as the compressors, turbines, and generators' net power for both LP and HP shafts, are calculated based on the assumption given in Table 1 and displayed in Figs. 3 and 4, respectively. For these results, the simulated mGT and mHAT engines were assumed to operate, as indicated before, at constant electrical power output of 27 kWe for both mGT and mHAT, by altering the rotational shaft speeds independently, TITs for the HP and LP turbine, and air and injected water mass flow rates. For the dry mGT, this operation corresponds to a thermal power output from the economizer of 24 $kW_{th,}$ while no remaining thermal power was available in the mHAT case.

To analyze the state properties and results of both mGT and mHAT cycles, Sankey diagrams, which illustrate the enthalpy flows between components, are drawn. Sankey diagrams are a powerful tool in thermodynamics for visualizing the energy transfers and identifying areas of efficiency and waste flow of an energy system where the width of each arrow is proportional to the amount of flow. The thicker the line or arrow in the Sankey diagram, the greater the amount of energy involved. They provide a clear and intuitive way to understand how energy is conserved and transformed within a system and help locate the most important contributions to a flow [16].

The Sankey diagram of the two-shaft mGT cycle working as a CHP unit is shown in Fig. 5. The unit produces an electrical power output of 27 kW_e and a thermal power output of 43 kW_{th} (economizer and intercooler combined). By generating 37.65 kW and 27.49 kW, the LP turbine and HP turbine drive the LP compressor and HP compressor through the LP and HP shaft, which consume 24.13 kW and 13.99 kW, respectively. The major role of the recuperator is evident in this figure: it allows for recovering 70.66 kW from the exhaust gases, representing about 92% of the energy the fuel provides (77 kW). Eventually, 5.81 kW is lost through the exhaust gases in the stack after delivering the energy for heating purposes.

Water injected flow rate calculated in Fig.4 have been obtained as results from the simulations, where the algorithm aims to satisfy all the specific conditions mentioned in the previous section. Indeed, in our modelling, through the selection of design parameters of the different components (e.g.:

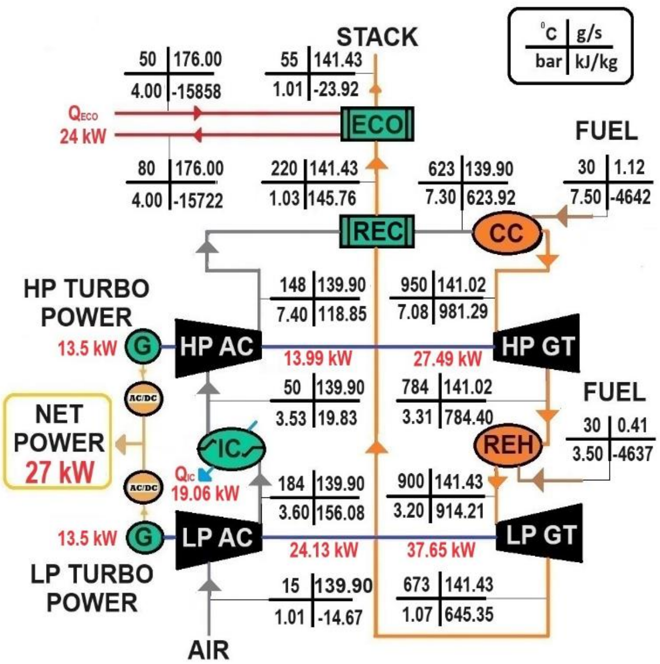

FIGURE 3: RESULTING PROPERTIES INCLUDING TEMPERATURE, PRESSURE, MASS FLOW RATE, AND ENTHALPIY AT ALL STATES OF THE TWO-SHAFT MGT CYCLE AS WELL AS COMPRESSORS, TURBINES, AND GENERATED NET POWER

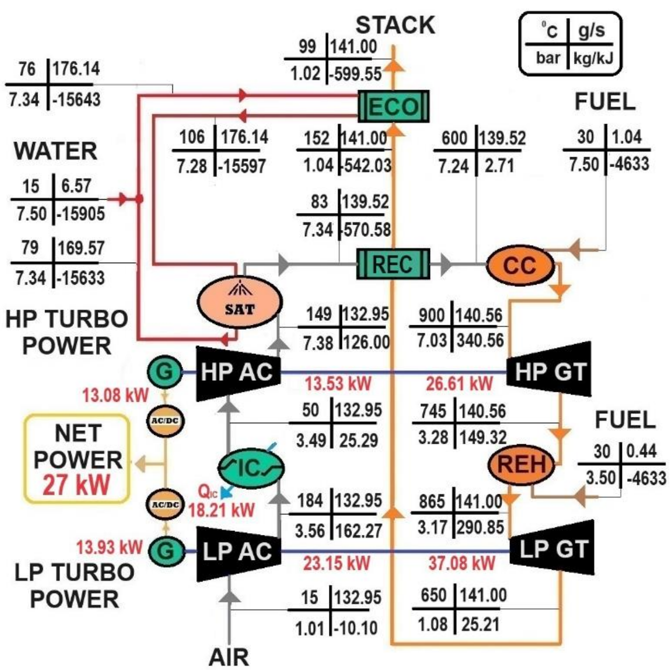

FIGURE 4: RESULTING PROPERTIES INCLUDING TEMPERATURE, PRESSURE, MASS FLOW RATE, AND ENTHALPIY AT ALL STATES OF THE TWO-SHAFT MHAT CYCLE AS WELL AS COMPRESSORS, TURBINES, AND GENERATED NET POWER

compressor maps, turbine choking, TITs, constant power output and pinch points in the heat exchangers), we limit the degrees of freedom of the system, forcing the control to maximize the water injecting, still respecting first law of thermodynamics. 6.57 g/s injected water in mHAT cycle is thus the maximal amount of water that can be injected, considering the selected operating conditions. These operating conditions are partially specific for the engine (two-shaft MITIS mGT), but also based on limitations set based on previous experience of the group on what is thermodynamically feasible.

The Sankey diagram of the two-shaft mHAT cycle is depicted in Fig. 6. The controller keeps the electrical power output constant at 27 kW_e: the effect of water injection is therefore appreciated in a lower total fuel input in the combustion chamber and reheater, 74.4 kW as opposed to 77 kW for the dry two-shaft mGT (3.37% reduction). Of the 24 kW_{th} corresponding to the heat output of the two-shaft mGT working as a CHP unit, 8.14 kW is recovered in the saturation tower through water injection. In contrast, 15.86 kW is further regained in the recuperator. Therefore, the saturation tower also work as an after-cooler. As seen in Figs. 3 and 4, the saturation tower can work as an aftercooler since the compressed air enters at a temperature of 149°C while the water is sprayed at 106°C. Through the humidification process, the air-water mixture's temperature lowers to 83°C. That is, with water injection, the inlet temperature of the air on the cold side of the recuperator is substantially reduced compared to the dry mGT, where air enters at 148ºC. This is possible due to the aftercooler effect of the saturation tower. Hence, since the approach temperature difference in the recuperator is kept constant at 50°C and the heat capacity of the air-vapor mixture is larger than the one of air, more heat can be exchanged in the recuperator of the mHAT: 79.98 kW compared to the 70.66 kW of the dry case. This means that in the mHAT cycle, more heat is recovered in the recuperator than energy is provided by the fuel. Finally, 28.19 kW are lost in the mHAT stack.

About the effect of water injection on the flow entering the combustion chamber and reheater, both temperature, and pressure are slightly lower for the mHAT cycle compared to the mGT (600ºC vs. 623ºC and 7.24 bar vs. 7.3 bar, respectively, for the combustion chamber and 745 ºC vs 784ºC and 3.28 bar vs 3.31 bar respectively for the reheater, see Figs. 3 and 4). As indicated before, it should be pointed out that the humidification of the combustion air in a mGT affects combustion stability, efficiency, and exhaust gas emissions. This can lead to a non-stable, incomplete combustion, which will affect the global efficiency negatively. Additionally, CO emissions will increase. Nevertheless, the humidity levels achieved here are low (4.7%) and, as indicated in literature, no significant impact on emissions can be expected [26].

It is worth noting that both the high-pressure turbine choking constant and the high-pressure turbine isentropic efficiency is reduced due to the injection of water (according to Eqs. (5) and (6)): from 6.95 to 6.84 for the former and from 80.0% to 78.5% for the later. Nevertheless, the heat capacity of the turbine flow is higher in the mHAT cycle due to its more

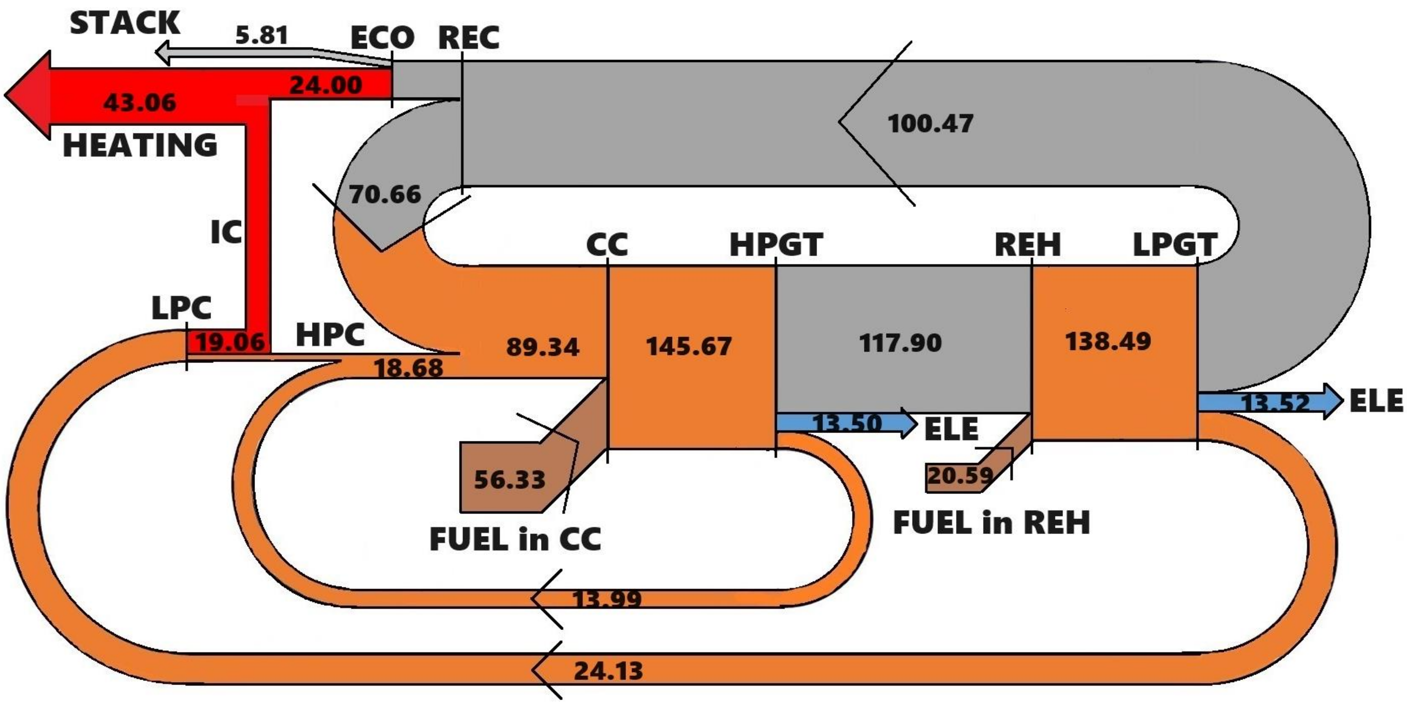

FIGURE 5: SANKEY DIAGRAM OF THE TWO-SHAFT MGT CYCLE WORKING AS CHP, WITH AN ELECTRICAL EFFICIENCY OF 35.12% AND A TOTAL EFFICIENCY OF 91.10%. ENERGY FLOWS ARE EXPRESSED IN KW.

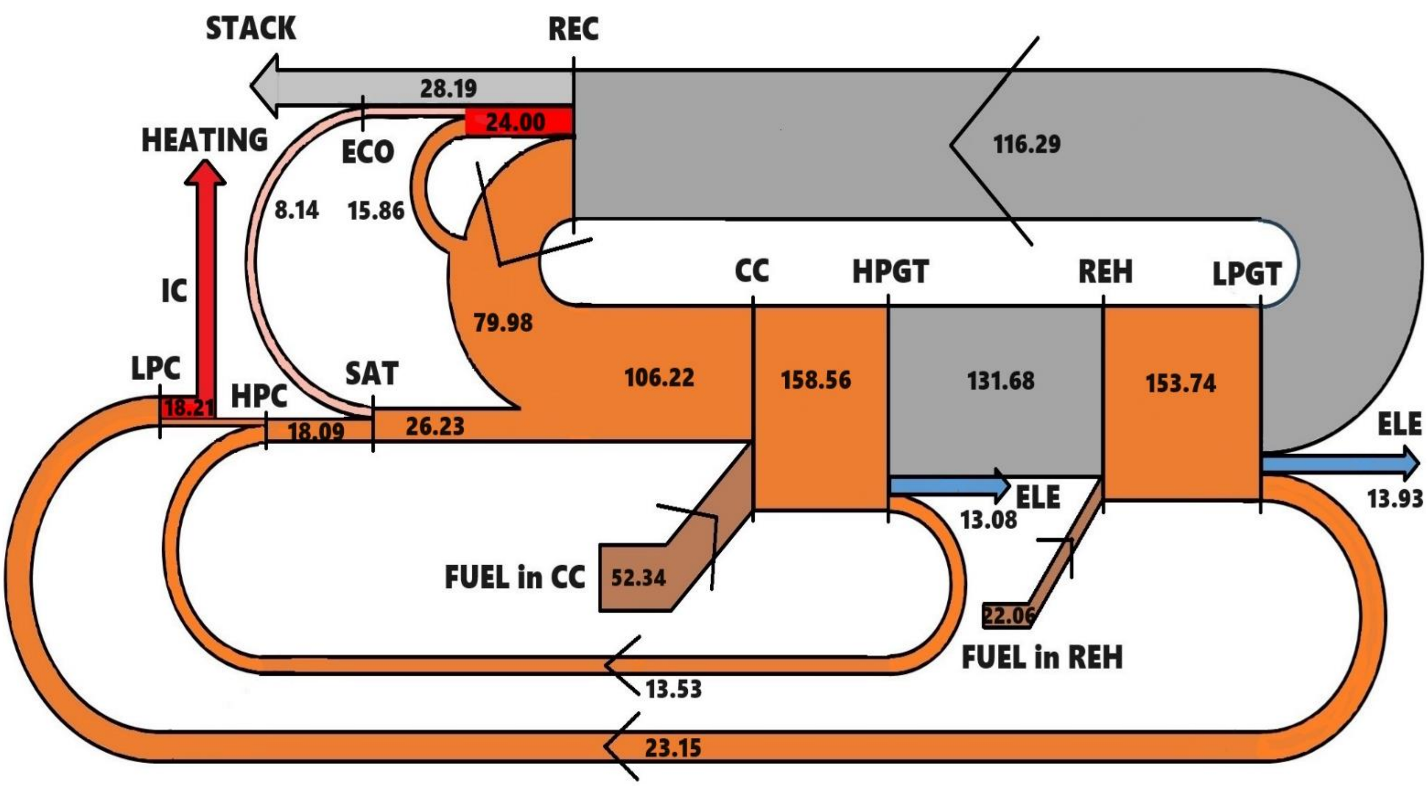

FIGURE 6: SANKEY DIAGRAM OF THE TWO-SHAFT MHAT CYCLE WITH AN ELECTRICAL EFFICIENCY OF 36.31% AND A TOTAL EFFICIENCY OF 60.77%. ENERGY FLOWS ARE EXPRESSED IN KW.

extensive water content: 1.25 kJ/kg K compared to 1.20 kJ/kg K for the dry mGT. All in all, the increase in heat capacity leads to the decrease in choking constant in such a way that to keep the power output constant at 27 kW_e (because the turbo generators are not able to produce more), the *TITs* required in the mHAT cycle is lower: 900ºC and 865ºC in the mHAT cycle as opposed to 950ºC and 900ºC for the mGT (see Figs. 3 and 4). Advantages of this reduction in *TITs* is that the fuel input required by the CHP mGT unit, amounting 77 kW in the dry case, is now restricted to 74.4 kW for the mHAT for both the combustion chamber and the reheater, resulting in increased efficiency. Moreover, a positive side effect is that the lower *TIT* reduces the thermal stress on the turbine blades, which can extend the life of the blades and reduce the risk of high-temperature corrosion (which is however less relevant in our energy analysis). The water evaporated in the saturation tower of the mHAT cycle replaces part of the air that would otherwise flow through the turbine since this component is choked. This means the required airflow through the compressor is lower than for the dry cycle. Hence, the shaft rotational speed is reduced with humidification, and, in turn, the net power demanded by the compressors is curtailed (24.13 kW vs. 23.15 kW for LP compressors and 13.99 kW vs. 13.53 kW for HP compressor). The ultimate effect of water injection is thus appreciated in increased electrical efficiency, from 35.12% for the mGT to 36.30% for the mHAT.

As remark, it should be stated that despite the general advantages and proven thermodynamic potential, humidified mGTs have not yet been commercially developed due to some limitations such as combustion stability, cycle layout limitation and material and recuperator constraints, as discussed in detailed in a review paper on humidification of single shaft mGTs [2]. Humidifying the mGT cycle will have, as discussed before, an influence on the combustion stability. Moreover, since the mGT operates at a fairly low pressure ratio, introducing new component like saturation tower make the additional pressure loss in the system and will have a strong negative impact on the performance of the mGT. Furthermore, on the material side, the presence of the water can cause some corrosion problems, especially on the hot side of the recuperator. By adding water to the mGT cycle, the lifetime of the recuperator is significantly reduced. In summary, humidification of mGT cycles offers great potential for enhancing the cycle's electrical efficiency and flexibility, but further research is necessary to make the technology commercially available.

6. CONCLUSION

In the present study, a novel two-shaft mGT cycle, working as a CHP unit, is transformed into an mHAT cycle and thoroughly evaluated from the view of thermodynamics. For both cycles, the Sankey diagrams are drawn to show the enthalpy flows between the different components of the system. Results show the positive effect of water injection on increasing electrical efficiency. The main benefits of water injection from a first-law perspective consist of (1) increased energy recovery in the saturation tower through the injection of water previously heated up by the exhaust gases, (2) due to the aftercooler effect of the saturator and the higher heat capacity of the flow, more energy (79.98 kW) can be recovered in the recuperator than to dry mGT cycle with 70.66 kW heat recovery in the recuperator, (3) by replacing part of the incoming air to compressors with the water evaporated in the saturation tower cause the required airflow through the compressor is lower than for the dry cycle. Hence, the shaft rotational speed is reduced with humidification, and, consequently the net power demanded by the compressors is reduced from 24.13 kW to 23.15 kW for LP compressors and 13.99 kW vs. 13.53 kW for HP compressor, (4) the increase in heat capacity of the flow through the turbine results in a lower required *TIT* for LP and HP turbine. Finally, the fuel flow is reduced from 77 kW to 74.4 kW for a constant electrical power output, increasing electrical efficiency from 35.12% to 36.31%.

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