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TOWARDS HIGHLY-FLEXIBLE CARBON-CLEAN POWER PRODUCTION USING GAS TURBINES: EXHAUST GAS RECIRCULATION AND CYCLE HUMIDIFICATION

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ABSTRACT

The current shift in electrical power generation towards more renewable production strengthens the need for highly-flexible, preferably carbon-clean, production units. These units need to provide the flexibility to the system necessary to compensate for the intermittent nature of the renewable energy production. Of all fossil-based power production units, Gas Turbines (GTs) are the only units capable of offering this flexibility, but they still require carbon capture to reduce the CO_2 exhaust, resulting in rather low efficiencies. More advanced GT cycles, i.e. cycle humidification possibly in combination with Exhaust Gas Recirculation (EGR), offer a solution; however, their performance is not yet fully identified and experimental data is still lacking. Applying these concepts first on small-scale, using micro Gas Turbines (mGTs), to show the potential of these cycle concepts, has some advantages; however, again numerical and especially experimental data are still missing. In this paper, two models predicting the potential of EGR and cycle humidification applied on a typical mGT, the Turbec T100, with post-combustion carbon capture have been compared and experimentally validated. Simulation results indicated that the impact of the advanced cycles on the mGT performance of both models is predicted in a similar way. In addition, numerical results show little difference with experimental data, validating both models. Finally, both numerical analyses highlighted that combining EGR with the micro Humid Air Turbine (mHAT) concept results in most efficient cycle layout, suggesting this as a promising application.

NOMENCLATURE

AHAT	Advanced Humid Air Turbine
CC	Carbon Capture
CCGT	Combined Cycle Gas Turbine
EGR	Exhaust Gas Recirculation
GT	Gas Turbine
HAT	Humid Air Turbine
mGT	micro Gas Turbine
mHAT	micro Humid Air Turbine
NG	Natural Gas
TIT	Turbine Inlet Temperature
TOT	Turbine Outlet Temperature

INTRODUCTION

As a measure to limit the global warming, the emission of greenhouse gasses and of CO_2 in particular, has to be reduced drastically. The electrical power generation is shifting towards more renewable production.

However, given the high intermittent production of renewables, like solar and wind energy, fossil fuelled units are still needed in the energy mix to provide the necessary flexibility to the energy system to balance these renewables and avoid possible blackouts. Natural Gas (NG) fuelled plants, like Gas Turbine (GTs) and Combined Cycle Gas Turbines (CCGTs), are the most suitable among the fossil-based options (due to quick startup, ramping, etc.). Even though natural gas has the lowest CO_2 emissions of all fossil solutions (compared to e.g. coal), there is still a need for development of low/zero emission technologies including carbon capture to meet the long-term climate targets (2°C). Post-combustion capture, where the CO_2 is captured from the flue gasses, is currently the most mature technology. In addition, it can be applied to the current installations, allowing to significantly reduce the emissions of these unit once deployed at large scale in the short term.

Like any carbon capture technique, post-combustion capture introduces an energy penalty to the electricity production plant. The low CO_2 concentration in the exhaust from NG fuelled GT plants, which was seen before as an advantage, results in a post-combustion capture context in a more severe energy penalty. Use of innovative solutions i.e. cycle humidification in combination with Exhaust Gas Recirculation (EGR) to improve the performance and increase the CO_2 content of the exhaust seems to be a promising solution.

EGR in GT cycles offers three major advantages: higher CO_2 concentration in the exhaust gases, lower flue gas mass flow rate and lower NO_x emissions. The higher CO_2 concentration in the exhaust gases in combination with a lower flue gas mass flow rate makes post-combustion amine-based carbon capture more economically feasible in CCGT [1]. In the field of EGR applied to large scale GTs, several studies are available in literature, showing the positive effects of EGR. E.g. Li et al. have shown that, compared to a cycle without EGR, a recirculation ratio of 50% could increase the CO_2 concentration from 3.8 to 7.9 mol% and reduce the mass flow rate of flue gases, fed to the absorber, by 51% [2]–[4]. The third positive effect of Dry-EGR, the reduction of NO_x emissions, has recently been studied in mGT combustors by means of CFD [5]–[7] and experiments [8]. Both experiments and simulations indicated the potential to control NO_x emissions by EGR; however, to the knowledge of the authors, EGR is not yet implemented in commercial applications.

The positive effects of humidification of the GT cycle, being also NO_x reduction, but more importantly a significant waste heat recovery leading to higher specific power output and electrical efficiency, has also been widely studied [9]. Among the different humidification methods, the Humid Air Turbine (HAT), first proposed by Rao [10], has been identified as optimal solution [9] and tested experimentally at Lund University [11], but has not been implemented on industrial scale. The Advanced Humid Air Turbine (AHAT) of Hitachi is the most advanced humidified GT cycle that has been tested [12], [13], but as mentioned before, is not yet commercially available.

Despite the proven potential of both EGR and cycle humidification, no commercial application is already available. On top, the combination of both has not yet been studied on large scale. So, even though this advanced GT cycle, i.e. cycle humidification possibly in combination with EGR, offers the solution to make post-combustion carbon capture more economic, their performance is not yet fully identified and experimental data is still lacking. Applying these concepts first on small-scale, using micro Gas Turbines (mGTs), to show the potential of these cycle concepts, has some advantages, however, again numerical and especially experimental data are still missing.

On the smaller micro Gas Turbine (mGT) scale, the concept of cycle humidification has been studied before, by both research groups involved in this paper and by others, both numerically and experimentally (see overview paper [14]). Additionally, the authors of this paper already studied the impact of EGR on the mGT performance, both in dry and wet operation [15]–[19]. Next to the research efforts of the authors of this paper, several works performed on the subject from the University of Sheffield are also available in literature [20]–[24]. Each of the papers indicate the same positive impacts of EGR and cycle humidification as can be found in larger machines; however, most of the data remains unvalidated and the possibility for scaling up remains still unanswered.

In this paper, two models predicting the potential of EGR and cycle humidification applied on a typical mGT, the Turbec T100, with post-combustion carbon capture have been compared and experimentally validated. The final aim of this paper is thus applying EGR and cycle humification on small-scale, using mGTs, to show the potential of these concepts on large scale GTs.

METHODOLOGY

In the methodology section, first the mGT model and the considered case studies are discussed, followed by the presentation of the numerical models to simulate these cycles, where a special subsection is dedicated to the comparison of the two simulation models. Finally, the experimental test rig used for model validation is presented.

mGT cycle and case study description

As reference case, we considered the Turbec T100 mGT with a nominal power output of 100 kW_e. The T100 like most mGTs, is a typically recuperated Brayton cycle (Figure 1). The main components of the cycle include thus: a variable speed radial compressor (1) to increase the pressure; a recuperator (2) to preheat the compressed air before entering the combustion chamber, using the heat available in the flue gasses; a counter-flow can burner (3) in which natural gas is burned to increase the temperature

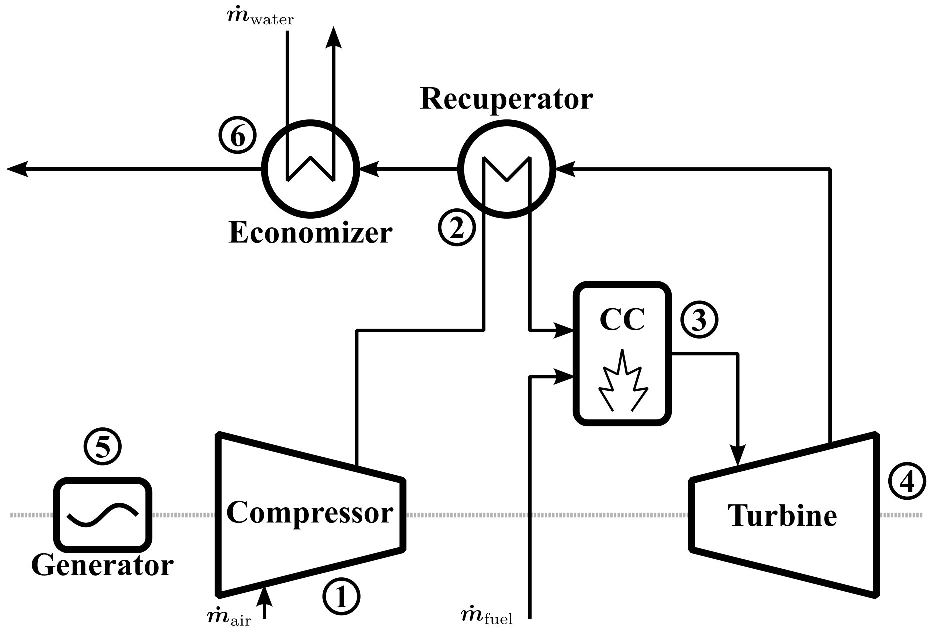

Figure 1: The Turbec T100 mGT, consisting in a typical recuperated Brayton cycle, was used as based case.

of the working fluid before entering the turbine; a radial turbine (4) in which the working fluid expands, delivering power to the shaft to drive the compressor and the generator (5) for power production and finally an economizer (6) to heat up water for heating purpose, since the mGT is typically used in Combined Heat and Power (CHP) applications.

Like most mGTs, the T100 operates at constant power output and Turbine Outlet Temperature (TOT) to ensure a high efficiency. To keep the power production constant, the rotational speed of the compressor shaft is varied. This variable speed operation is possible since the power is produced in a high-speed generator, which is linked with power electronics. The variable speed operation has an additional advantage that the unit can operated at constant TOT operation of 645°C, also ensuring high electrical efficiency. The TOT is kept constant by controlling the fuel injection in the combustion chamber.

Applying post-combustion carbon capture on mGT or in general GT applications always leads to an energy penalty. This penalty strongly depends on the used capture technology, but also on the concentration of the CO_2 in the flue gasses and the total amount of these flue gasses. In the field of mGT, several possible improvements for reduced energy penalty exists, focussing on the increase of flue gas CO_2 content and cycle performance improvement. The options considered in these papers are: EGR, cycle humidification by converting the unit in a micro Humid Air Turbine (mHAT) unit and finally a combination of mHAT with EGR.

The mGT can be converted into a mHAT by adding a saturation tower in the cycle between the compressor and the recuperator (Figure 2). This mHAT cycle was previously identified as the optimal cycle humidification option, taking into account cycle complexity [25], [26]. In this cycle, waste heat is recovered from the flue gasses by preheating water in the economizer and reinjecting it in the cycle in the saturation tower. In this saturation tower, the hot water will evaporate and while doing so, transferring the waste heat to the cycle working fluid (under the form of latent heat). Due to the evaporation of water in the compressor air, the mass flow rate passing through the turbine increases, resulting in higher power production in combination with an enhanced heat recovery in the recuperator, leading to significant fuel savings and thus reducing the CC penalty.

mGTs typically operated with a very lean combustion (air-fuel ratio in the range of 120 for the T100) to limit the inlet temperature of the turbine. The combustion in very lean conditions has as result a rather low CO_2 and high O_2 concentration in the exhaust. This high O_2 content allows recirculating part of the exhaust gases and by doing so, increasing the CO_2 content of the flue gas and thus again reducing the CC penalty (so-called EGR, Figure 2). Part of the exhaust gases is recovered after leaving the recuperator and cooled down, to avoid a too high compressor inlet air temperature, which affects strongly negative the performance of the unit. the part of the water in the flue gases, entered the cycle in the saturation tower or produced during the combustion process, that has condensated during the cooling process is separated during the cooling process is drained from the EGR loop. A blower is installed to ensure the necessary pressure difference to drive the EGR stream and finally a filter is installed to remove possible impurities from the cycle.

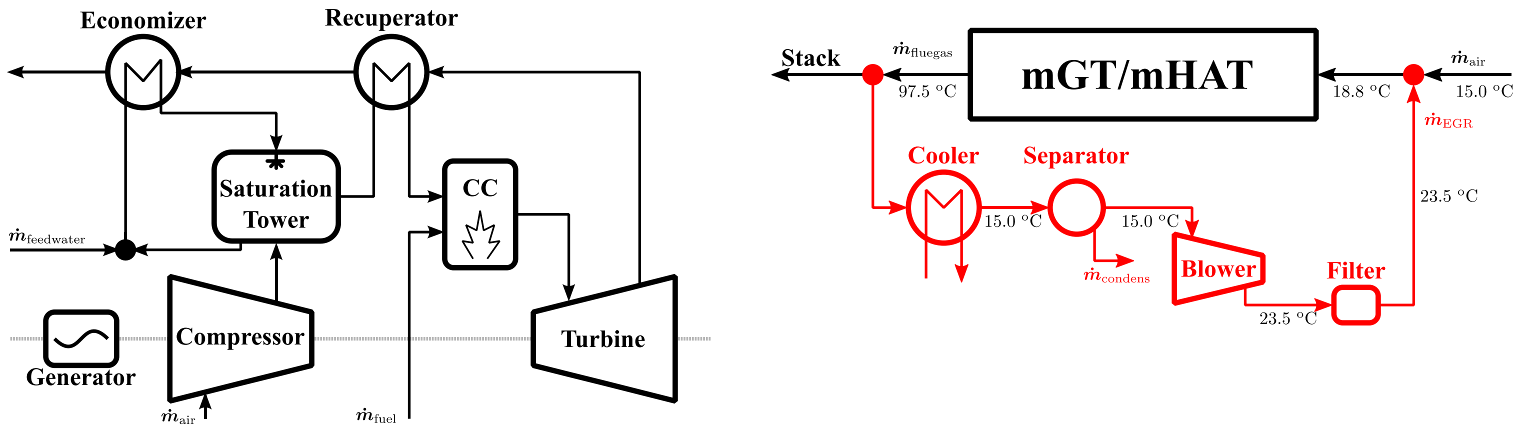

Figure 2: To reduce the energy penalty linked to the post-combustion carbon capture, two advanced mGT cycle concepts: the micro Humid Air Turbine (mHAT, left) and Exhaust Gas Recirculation (EGR) have been considered in this paper.

Finally, to capture the remaining CO_2 from the exhaust gasses, a post-combustion carbon capture facility needs to be installed. Given the rather low CO_2 content of the flue gasses (even after applying EGR to the cycle), amine-based carbon capture was identified in the past as most suitable solution [16]. The coupling between the amine-based plant and the mGT has already been studied before by several authors [16], [21], but will not be considered in this paper.

System modelling

In this subsection, first the Aspen models of the mGT, mHAT and EGR cycle, developed at the BURN joint research group of VUB and ULB are presented. Secondly, the IPSEpro models of the same cycles, developed at UiS are presented. Finally, an in-depth comparison of the both models is performed, highlighting possible differences and similarities.

Aspen Plus model

The first model is constructed in the commercial software Aspen plus process simulator (V9.0) [27].

The compressor was modelled using the operating map provided by the manufacturer. Both constant speed lines and efficiency areas were introduced in the compressor model. Rather than using the map to simulate the turbine performance, the model was simplified by assuming a constant turbine outlet pressure and isentropic efficiency (in dry mode), since turbine efficiency and outlet pressure remained constant over a large variety of parameters [28]. A turbine outlet pressure of 1.05 bar is assumed, which allows the exhaust gases to overcome the head losses in the recuperator, economizer and stack. An isentropic efficiency of 85% is used in dry operating mode. The turbine is assumed to be choked. For both turbine and compressor, a mechanical efficiency of 99% has been used. The recuperator is simulated as a counterflow heat exchanger, where the surface is adapted to correct for the cross flow in- and outlet sections of the component. A pressure loss of 5% over the cold side was assumed. The combustion chamber is modelled using a Gibbs reactor, assuming complete combustion and 5% pressure loss. The different pressure losses used in the Aspen model are based on experimental data and available information in literature on the recuperator [29] and were fined tuned using model optimization based on experimental results. The losses in the generator and power electronics were combined, leading to a total efficiency of the electrical part of 94%.

The control system of the mGT was implemented in the Aspen model using two *Design Specs*. In a first *Spec*, the power output is kept constant by controlling the rotational speed, while in the second *Spec*, the TOT is kept constant at 645°C by adjusting the fuel flow rate. Finally, as property method, the Redlich-Kwong-Soave cubic equation of state with Boston-Mathias alpha function (RKS-BM) method was used. Previous simulations have indicated that this property method has some difficulties assessing the dew point [26], however this is less crucial for the simulations presented in this paper.

For the mHAT operation, the saturation tower was simulated using the *Radfrac* module, available in Aspen, according to the recommendations of Queiroz et al. [30], assumed to introduce an additional pressure loss 0.5% to the cycle. The compressor model has not been changed compared to the dry model. For the turbine modelling in humidified operation, the isentropic efficiency is corrected for the changing composition of the working fluid according the recommendations of Parente et al. [31]. The choking constant of the turbine is, similar to the efficiency, corrected for the changing working fluid composition in wet operation. The compressor model remained unchanged; however, since the turbine is choked, during water injection, the compressor operating point will shift closer to the surge limit. Part of the air mass flow rate is thus replaced by water vapor, leading to a reduction in the air mass flow rate passing through the compressor. Previous simulations have indicated that this surge margin reduction is limited when using natural gas as fuel, given the variable speed operation of the mGT [32]. The recuperator model and combustion chamber models remained unchanged. Finally, both fuel and power control were implemented in mHAT model, similarly to the implementation in the dry mode.

Finally, to simulate the effect of EGR on the dry mGT and the humidified mHAT performance, part of the exhaust gases is rerouted to the compressor inlet using the EGR loop presented in Figure 2. The EGR stream is simulated in Aspen by splitting part of the exhaust gases, cooling the gases down using a generic heater block, separating the condensed water in a Separator block and finally a blower with an isentropic efficiency of 85% to provide the necessary pressure increase for EGR. Due to the increasing content of CO_2 and to a lesser extend H_2O in the inlet air, the thermodynamic properties will change, which will affect the performance of the compressor [33]. Since the amount of CO_2 in the compressor inlet air remained limited, the compressor map is not changed (effect of additional CO_2 and H_2O on heat capacity ratio is less than 0.26% for an EGR ratio of 0.55 at nominal power output). The turbine—which is still assumed to be choke—is adjusted, by considering the actual gas properties for calculation of the choking constant (see before). The recuperator, economizer, combustion chamber and saturation tower are kept unchanged. Next to the two control loops for power output and TOT, a third loop, changing the recirculation ratio to obtain a certain O_2 content in the combustion inlet air, was added.

<u>IPSEpro model</u>

The second model is developed using IPSEpro that is a commercial heat and mass balance software tool [34]. The properties of gas components are calculated with polynomials derived from the JANAF (i.e. Joint Army Navy Air Force) Thermodynamic Tables [34]. In IPSEpro, all calculations are performed assuming that the gas components are ideal gases. IPSEpro provides access to the source code of models and their underlying assumptions, enabling the development of new components or the modification of existing ones. A comprehensive model library, which has been developed as a result of several simulation projects inside the research group in UiS, was employed for this study.

Characteristics maps that were made available for the research group at UiS were used to model the compressor and the turbine. The values in the maps were defined by non-dimensional and corrected parameters, namely pressure ratio, corrected air mass flow rate and corrected speed. For the definition of corrected parameters, readers are referred to [35]. These parameters are used to reduce the number of variables required to define each operating point of the compressor. To see how much the operating point deviates from the design value, pressure ratio and corrected variables were normalized with their design values, representing the relative parameters. Maps with normalized values were then implemented in the model.

The recuperator effectiveness was calculated as $(T_{air,out}-T_{air,in})/(T_{gas,in}-T_{air,in})$ using the temperature measurements obtained during experiments carried out at an existing test facility in Norway. The calculations based on experimental results showed that it does not change significantly with operational condition and it almost remains unchanged around 0.91. This value was then kept fixed in the model for design and off-design conditions. The economizer as a counter-current heat exchanger was used to preheat the water before entering the saturation tower. In the economizer model, the pressure loss over the cold side was assumed negligible. In the combustion chamber model, the composition of the gas stream exiting the combustor is calculated by assuming complete combustion. The combustion model calculates the pressure loss due to heat increase as a function of mass flow rate, pressure and temperature, which was further tuned for the mGT using the design point data. To calculate the pressure loss in the air path from the compressor outlet to the combustion chamber, a simple pipe model was added at upstream of the compressor and recuperator, in which the coefficient losses were tuned using the design data [36]. Other model details such as mechanical efficiencies and losses in the generator were adjusted in a way to reach a good agreement with performance data published in the manufacturer's technical document and available in the literature [37].

The model can be operated at either constant power output or constant rotational speed, which is set by the user. The model has two constraints, namely the TOT and Turbine Inlet Temperature (TIT) with maximum allowable limit of 645°C and 950°C, respectively. If TOT (TIT) limit is reached, the mass flow rate and TIT (TOT) are regulated to avoid exceeding the limit.

The mGT model was validated both in design and off-design conditions using data obtained from experiments run by UiS [36]. The generality of the model has also been tested against experimental work conducted at Pilot-Scale Advanced Carbon-Capture Technology (PACT) National Core Facilities in the UK [38].

The mGT model was used as the baseline and additional components were added to simulate mHAT and EGR cycles. For the mHAT cycle, it was assumed that pressure drop in the gas flow through the saturation tower is low. This was verified by experiments in the pilot plant in Lund [39]. Moreover, the humidified air exiting the saturation tower is saturated according to Lindquist [39]. The humidified exhaust gas exiting the humidifier is assumed to behave like an ideal gas. To simulate the EGR cycle, part of the exhaust gas leaving the economizer is recirculated to be mixed with the air at the compressor inlet as shown in Figure 2. The exhaust gas first passes

Table 1: **Comparison between the modelling of the main mGT components in the Aspen and IPSEpro models.**

Component	Aspen	IPSEpro
Compressor	Operating map provided by the manufacturer	
Recuperator	Counterflow model using surface and heat exchange coefficient, based on literature and experimental data.	In-house model, defining effectiveness based on experimental results, resulting in a constant effectiveness of 91%.
Combustion chamber	Gibbs reactor with global energy balance calculation using combustion efficiency of 100% and 5% pressure loss.	Calculations assume complete combustion with pressure loss function of mass flow rate, temperature and pressure.
Turbine	Assumed to be choked with isentropic efficiency corrected for the changing working fluid composition. Values are based on the map provided by the manufacturer	Operating map provided by the manufacturer
Property method	RKS-BM method	JANAF Thermodynamic Tables
Control system	Fuel flow rate control to keep TOT and TIT constant at 645°C and 950°C respectively. Constant electrical power output control by adjusting the rotational speed	

through an exhaust gas condenser, where it is cooled down by cooling water, before entering the compressor. An exhaust fan is used to overcome the pressure drop in the EGC.

Model comparison

Since the aim of this paper is to compare and validate two different models for EGR and cycle humidification in an mGT, a comparison between both models, discussing the main similarities and differences, is presented in Table 1.

Test rig description

For the validation of the steam injection models, experiments have been performed on the humidified Turbec T100 mGT test rig of the VUB (Figure 3). This test rig consists of a Turbec T100 Series 2 mGT equipped with a steam injection line to study the impact of steam injection on the cycle performance [40], [41], a saturation tower to convert the mGT into a mHAT [42], [43] and most recently with a CO_2 injection line to emulate the impact of EGR on the cycle performance. To capture the different impacts of the cycle modifications on the mGT performance, the test rig is equipped with several sensor and a data acquisition system, as described in [42]. This test rig has been used to validate the dry performance prediction of the Aspen model in the past [44], while the dry IPSEpro model was validated on a different machine [18].

(a) (b)

Figure 3: Pictures of the integration of the saturation tower in the mGT cycle in between the compressor outlet and recuperator inlet (a) and of the saturation tower with circulation pump, valves and sensors (b).

The humidified tests have been performed according to the start-up and shut-down procedures described in [42], which means that in wet operation mode, water is already injected during startup of the engine to avoid flameout when water would be injected during dry run. For each wet test, the engine is preheated by performing at least 1 hour dry test and each wet test is followed by a dry run of again at least 1 hour which serves as dry reference for the wet test. When performing water injection, a small amount of the compressor air was bled off, to prevent the compressor from going into surge due to the shift in operation point because of the additional pressure loss over the humidification unit and the additional water mass flow going through the turbine, lowering the compressor air flow rate as a result of the turbine choking. Although the mGT runs in standard dry mode at constant power output, for the validation, wet tests have been performed at constant rotational speed by slightly modifying the control system. These tests were necessary, since at constant power output, due to the limitations of the control system, the mGT could not reach the requested power output when running without water injection, but with the saturation

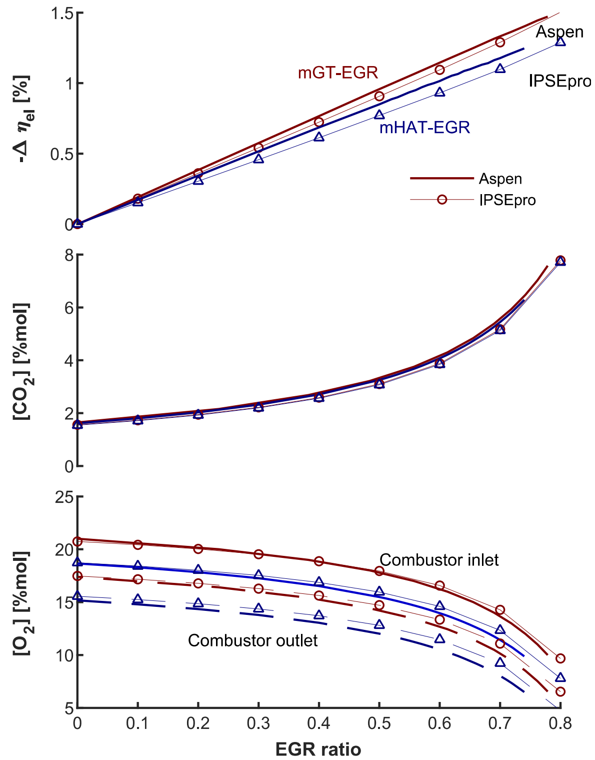

Figure 4: The CO2 content of the flue gases can be increased significantly for both dry and wet cycles by performing EGR. EGR however has a strong negative effect on the electric efficiency

tower in the cycle and a small bleed stream. Therefore, tests were performed at constant rotational speed (for a more in-depth discussion, we refer to previous work [45]).

Finally, CO_2 injection experiments have been performed using the procedure described by Simone et al. [46]. However, due to a limitation on the maximal CO_2 amount that could be injected (CO_2 was provided using bottles, which had a limited maximal flow rate), the injection rate was limited to 0.3% addition, which is far below the envisaged 3% injection to emulate the additional CO_2 when performing EGR. However, this small amount already allows to assess the impact of extra CO_2 on the compressor and turbine performance and also on the impact on the combustion process (possible change in emissions).

RESULTS AND DISCUSSION

In this section, first the prediction of the impact of transforming the mGT into a humidified cycle (mHAT) and the EGR of both models are presented and compared, followed by an experiment validation of the wet model and the EGR emulations.

Numerical model comparison

Increasing the EGR fraction has a negative effect on both the mGT and mHAT performance (Figure 4). The negative impact is a result of the power consumed by the blower to provide the necessary pressure difference to ensure the EGR stream. Due to the smaller total air mass flow rate in humidified operation, the negative impact of the blower on the performance is lower for the mHAT with EGR compared to the mGT. By increasing the EGR flow rate, the amount of CO_2 in the exhaust gasses can be increased. At low EGR rates, the increase is rather linear; however, starting from an EGR rate of 0.6, the CO_2 concentration in the exhaust gas increases exponentially. For both dry and wet operation, we see a similar CO_2 concentration in the exhaust gas for similar EGR rates. However, when looking at the O_2 concentration in the combustor inlet and outlet, it is clear to notice that for the humidified cycle, the O_2 concentration is lower, due to the presence of the water in the air. This is important, given that combustion in traditional premixed swirl combustion cannot be sustained below 16%vol O_2. Below this 16%vol O_2, the CO emissions increase very rapidly, leading to very low combustion efficiencies [47].

Comparison of the numerical results of the impact of EGR on the mGT and mHAT performance, obtained using both Aspen and IPSEpro shows good correspondence for the different EGR rates simulated (Figure 4). The predicted impact of the EGR loop on the electrical efficiency for both mGT and mHAT operation is almost identical for both the Aspen and IPSpro simulations. The small remaining differences can be explained by slight difference in the modelling (see Table 1). For both the CO_2 and O_2 content in the exhaust gases and the combustion inlet air, we see again good agreement between both models, which indicates both models can predict the impact of EGR and can be used for future development of carbon clean mGT and GT cycles. At very high EGR-ratios (starting from 0.7), we see a slight deviation between the results for the composition of the flue gases, which can be explained by the slightly different combustion chamber modeling.

The negative impact of EGR on the electrical increases slightly when going to part load when considering a minimal O_2 concentration of 16%vol at the combustor inlet for both mGT and mHAT (Figure 5). The larger impact can be explained by the lower relative fuel consumption at part load. Indeed, at part load, the relative injected fuel mass flow rate compared to the air flow rate is lower, resulting lower CO_2 concentration in the exhaust gasses. To keep the O_2 concentration at 16%vol in the combustor inlet, the EGR fraction must be increased (from 63% at nominal power output to 65% at 70 kW_e power output), which is reflected in the higher consumption of the blower and thus the more negative impact.

Comparison of the numerical results obtained with the two simulation tools show again good agreement (Figure 5). For the electrical efficiency, it can be observed that the predicted efficiencies are higher than the values obtained in with the Aspen model (±2%), but this difference is consistent for all models (dry/wet and no EGR/EGR). The moderate discrepancies between both models can be explained by the different implementation of the turbine characteristic maps in both numerical models (see Table 1), resulting in slightly different operation of both turbine and compressor and by the fact that the model parameters of both models (especially the recuperator parameters) have been tuned using experimental data of two different versions of the T100. Nevertheless, as mentioned before, these differences are limited, and similar trends can be observed when going from full to part load operation. Similar conclusion can be made for CO_2 content in the exhaust gases and the necessary EGR rate to achieve a concentration of 16% of O_2 in the combustor inlet.

<ins>Wet operation experimental validation</ins>

In addition to the comparison of the numerical data, an experimental validation of the mHAT results was performed. As mentioned in the model description, due to the additional pressure loss introduced by the saturation tower, model validation at constant power production was

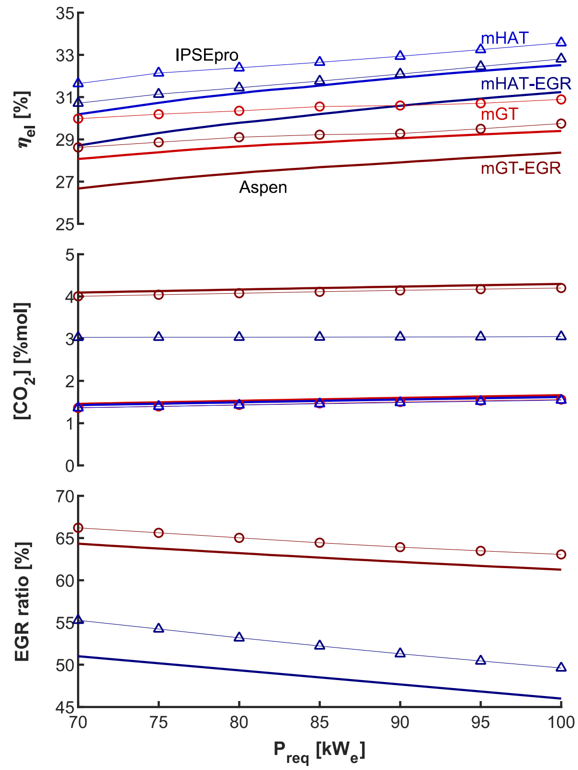

Figure 5: Performing EGR on both mGT and mHAT cycle has a severe negative effect on the total efficiency of the cycle. The efficiency of the mHAT cycle with EGR is however still higher than the base efficiency of the standard mGT cycle for nominal and part load operation.

not possible. Therefore, numerical and experimental results at constant rotational speed where compared.

Comparison between the numerical and experimental results at constant rotation speed indicate that the measured efficiency increase is higher than the numerical predictions (Figure 6). The difference between both can be explained by the bleeding. Indeed, during the experiments, part of the compressor air needed to be bled of to ensure a safe surge margin to avoid possible compressor surge. Because of the constraints of the facility it was not possible to implement flow measurement techniques; therefore, we cannot determine how much air is actually bled during the tests. It is important to bear in mind that air bleeding leads to higher electrical efficiency increases when comparing dry and wet operating points; this is why in Figure 6, the experimental results appear to be better than simulations. Nonetheless, it is obvious that water injection does lead to an electrical efficiency increase. In constant rotational speed mode, the T100 mHAT at VUB allows assessing this increase, which ranges between 3.6±0.5% and 4.2±0.5%. Additionally, we can conclude that the both models of the mHAT are validated. For a more in-depth discussion on the wet validation, we refer to [45].

Table 2: Comparison between physical parameter of the mGT with and without CO_2 injection captured during several preliminary experiments with CO_2 injection in the compressor inlet. The effect of the slight dilution of the inlet air on the various physical quantities is negligible.

			Test #1		Test #2	
Parameter	**Error**	**Unit**	**No Inj**	**Inj**	**No Inj**	**Inj**
$\dot{m}_{CO_2,inj}$	-	[kg/h]	0.0	6.5	0.0	6.5
P_{req}	-	[kW]	80	80	70	70
P_{gen}	1%	[kW]	80.0	80.0	70.0	70.0
n	0.01%	[krpm]	65.31	64.92	63.52	63.23
$\dot{V}_{fuel}$	1%	[Nm³/h]	31.2	31.3	27.9	28.0
η_{el}	$\pm$0.9%	%	27.2	27.2	26.7	26.6
$[CO_2]_{exh}$	5%	[%vol]	1.46	1.69	1.46	1.65

EGR operation experimental validation

Measurements during an injection of 6.5 kg/h have been performed at two power outputs (70 and 80 kW_e) and the key parameters have been monitored to identify the effect of the injection on the mGT operation. The limited amount of CO_2 that could be injected is a result of the limitations on the maximal flow rate available from the CO_2 bottles. The amount of 6.5 kg/h of CO_2 injection corresponds, as mentioned before, to a CO_2 concentration in the compressor inlet air of 0.3%, which is still far below the envisaged 3%. (see Figure 4). All tests were performed using a similar experimental procedure, during which CO_2 was injected after a dry run of approximately 1 hour, for 15 minutes and was followed again by 1 hour dry run to serve as reference. Theoretically, there should be no impact on the mGT performance, since the decrease in electrical efficiency is fully due to the consumption of the blower in the EGR loop (Figure 4).

As expected, the limited amount of CO_2 injected had a very little impact on the compressor operation. Both rotational speed and compression ratio remained unaffected when CO_2 injection was started. Similarly, the mGT control system did not have to adjust the fuel mass flow rate to obtain a constant TOT, while the electrical power output was remained constant.

The comparison between experimental results of the mGT with and without CO_2 injection clearly shows the negligible effect of the dilution (Table 2). While a mass flow rate of about 6.5 kg/h of CO_2 has been injected, almost all physical parameters of the turbine cycle showed a very little fluctuation from their previous value without injection. The only sensible variation is the increase of concentration of CO_2 in the exhaust gas which went from 1.46%vol to 1.69%vol (at 80 kW_e) and from 1.46%vol to 1.65%vol (at 70kW_e).

CONCLUSION

In this paper, two models predicting the potential of EGR and cycle humidification applied on a typical mGT, the Turbec T100, with post-combustion carbon capture have been compared and experimentally validated. The final aim of the paper was applying EGR and cycle humification on small-scale applications, using mGTs, to show the potential of these concepts on large-scale GTs.

Comparison between both models, one performed in Aspen Plus and the second one in IPSEpro, showed that both models predict the impact of humidification and EGR in a similar way. Both models indicated that cycle humidification allows to increase the efficiency of the cycle, which is beneficial for compensating the energy penalty introduced on one hand. On the other hand, both models indicated a significant increase in CO_2 content of the exhaust gases and a reduction of the total mass flow rate of these flue gases, having a beneficial effect on this energy penalty of the CC plant. Combining both EGR with cycle humidification shows best performance due to the beneficial effect of both measures. Finally, both models were successfully validated using experimental results of humidified mGT operation and operation under CO_2 injection conditions, showing the potential to apply these technologies on small-scale applications.

These promising results on small-scale applications can be used for future development of cycle innovations on large-scale GTs, to reduce further the energy penalty related to carbon capture and make carbon clean energy production from GTs economically viable. A more in-depth analysis of the carbon capture cycles coupled with the GT/mGT remains however still necessary, as well as a full economic analysis.

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