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Thermodynamic Study of a Cooled Micro Gas Turbine for a Range Extended Electric Vehicle

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Carbon dioxide released by the transportation sector has a significant impact on the environment. Nowadays, the research efforts are thoroughly investing in the capability of series hybrid electric vehicles in reducing CO_2 and NO_X emissions. Many regulations are encouraging automakers to substitute internal combustion engines with electric vehicles. Although the latter are zero-emissions energy sources, they are expensive and have a limited autonomy range. Therefore, recent studies are interested in gas turbines (GTs) in series hybrid electric vehicles (SHEV). A gas turbine (GT) coupled to an alternator is considered as an auxiliary power unit capable of charging the battery of the electric vehicle once depleted. The microturbine is generally composed of a centrifugal compressor and a radial turbine mounted on the same shaft with a recuperator and air bearings. Based on previous researches and thermodynamic analysis, the most suitable gas turbine cycle with regard to the efficiency and the net specific work is composed of two compression stages with an intercooler, a regenerator, and two expansion stages with a reheater. It can be deduced that by increasing the turbine inlet temperature, the system efficiency increases. However, the inlet temperature is limited by turbine materials constraints and specifications. The present paper focuses on developing a more efficient GT cycle by reaching higher inlet temperatures by cooling the turbine blade. It consists of using compressed gas from the compressor and introduce them into the turbine blades. This technology takes into consideration the influence of the extracted mass of compressed air, the effectiveness of the coolant, and the turbine blade temperature on thermal efficiency. An increase of the cycle efficiency of 4.78 points is obtained for a turbine inlet temperature of 1,450 °C.

1. Introduction

The vast growth in population and its respective needs has led the temperature of the planet to rise significantly. The reduction of harmful emissions and fuel consumption of road vehicles becomes a necessity. Recent studies have been interested in gas turbines (GTs) in series hybrid electric vehicles (SHEV) as potential substitutes for internal combustion engines (ICE) (Bou Nader, 2019). GTs are suitable for different applications due to their various benefits. They are lightweight compact engines, known for their high reliability due to the small number of rotating parts and the low vibration noise. GTs offer multi-fuel capability, high power-weight ratio, and low emissions compared to ICE. GTs are integrated into SHEV to overcome the disadvantages of electric vehicles (low battery energy density, limited autonomy range, long charging time, expensive...).
Recent studies are interested in GTs in SHEV, which combine the GT cycle with an electric powertrain. GTs coupled to an alternator are considered as a power unit (APU) capable of charging the battery of the electric vehicle once depleted. The range extender system is then totally decoupled from the vehicle load (Ji et al, 2020). This work suggests a new configuration of a GT cycle to be used in SHEV. First, an assessment for different GT configurations in the literature is conducted to determine the potential one to be used in SHEV. The different GTs cycles are evaluated through a thermodynamic analysis using REFPROP to identify the one with higher efficiency. Then, the new GT cycle is presented. Increasing the turbine inlet temperature leads to an increase in the cycle efficiency, but it is limited by the material melting point. The new suggested configuration of the GT cycle is composed of a cooled turbine in such a way to reach higher turbine inlet temperatures and higher cycle

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efficiency. Internal cooling is applied and a thermodynamic analysis is carried out to test if this new configuration is with higher efficiency than the cycles proposed before.

2. Micro Gas Turbine configuration

This section presents the different configurations and components that can be added to the gas turbine cycle to improve its performance. A simple gas turbine cycle operates according to the Brayton GT cycle as seen in Figure 1a, and is composed of a compressor, a combustion chamber, and a turbine (Bhargava et al., 2007).

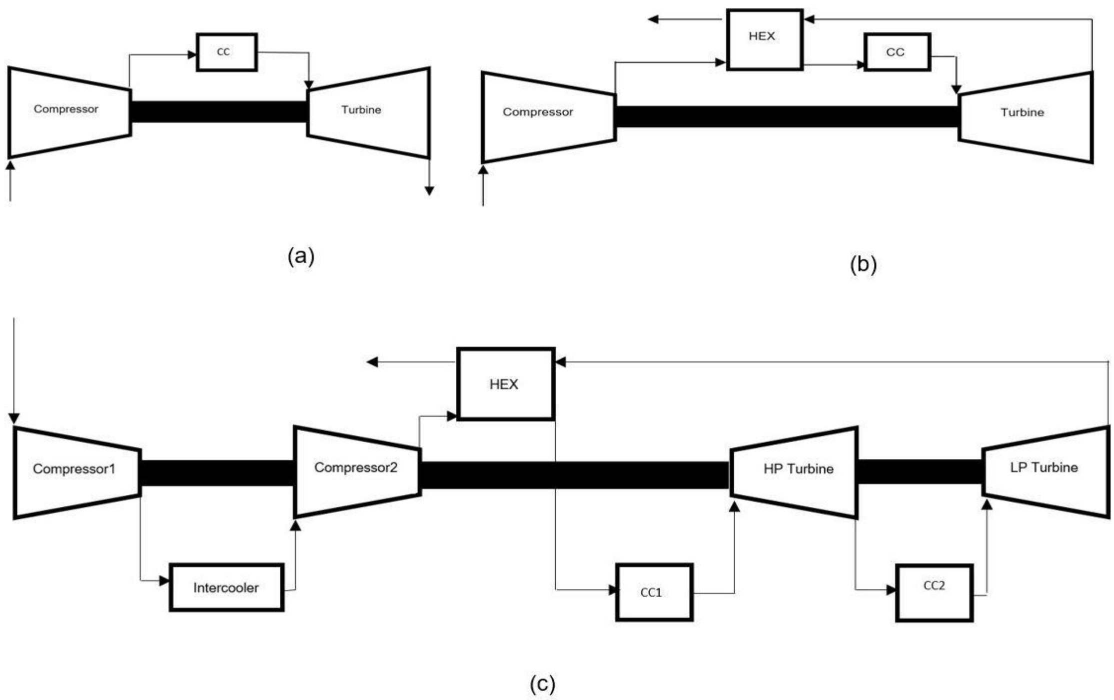

Figure 1: GTs configurations: (a) Brayton cycle (standard cycle), (b) Recuperated gas turbine cycle (RGT), (c) Intercooled recuperated reheated gas turbine cycle (IRReGT)

Microturbines are energy generators with a capacity between 15 kW and 300 kW. They operate on a simple GT cycle. Generally, they are composed of a centrifugal compressor and a radial turbine mounted on the same shaft with a generator, rotating at high speeds between 40,000 and 120,000 rpm. Air bearings or oil bearings are usually used (Breeze, 2016).

2.1 Recuperated Gas Turbine cycle

In order to increase the efficiency of the system, the heat from the exhaust gas can be recovered using an air-to-air recuperator as presented in Figure 1b. The recuperator preheats the air at the outlet of the compressor using the hot exhaust gas to reduce the fuel consumption in the combustion chamber. Adding a recuperator can lead to an increase in the system efficiency from 17 % to 30 % (Breeze, 2016).

2.2 Reheating and intercooling

Increasing the GT net work can be achieved either by decreasing the compressor work or by increasing the turbine work. Cooling the fluid between the compressor stages allows the reduction of the compressed air volume and reduces the work of the compressor for a given pressure, leading to lower energy consumption. By adding compression stages, the compression gets closer to the isothermal transformation, which matches the minimum work.
Reheating consists of dividing the turbine into several stages and adding combustion chambers between them. Thus, a reheat cycle leads to a higher turbine outlet temperature and enhances the potential for regeneration (Moran and Shapiro, 2006) . Figure 1c depicts an intercooled recuperated reheated gas turbine cycle.
Ibrahim and Rahman (2013) developed a model to compare GT configurations using Matlab. They concluded that reheating leads to higher power output, and regenerating leads to higher thermal efficiency. Salpingidou et

al. (2016) investigated the performance of different GT configurations with recuperators and reheating for aero-engines.

2.3 Comparison between GT configurations

A comparison between the suggested GT configurations is depicted in Figure 2. The efficiency of the Brayton and the RGT cycle are calculated using Eq(1), and the efficiency of the IRReGT cycle is calculated by Eq(2).

$$\eta = \frac{W_{Turbine} - W_{compressor}}{q_{cc}} \tag{1}$$

$$\eta = \frac{(W_{HP-Turbine} + W_{LP-Turbine}) - (W_{compressor1} + W_{compressor2})}{q_{cc1} + q_{cc2}} \tag{2}$$

For the same turbine inlet temperature and maximum pressure, the IRReGT cycle presents the highest efficiency (46.9 %) compared to the Brayton cycle (28 %) and the RGT cycle (38.67 %). The results obtained are in good agreement with (Bou Nader, 2019) that suggested IRReGT as a prioritized cycle offering higher efficiency and power density as well as reduced fuel consumption compared to the other GT configurations.

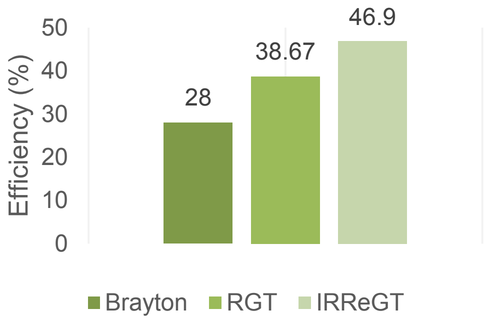

Figure 2: Comparison of the efficiency of Brayton cycle, RGT, and IRReGT

3. Blade cooling technology

The gas turbine thermal efficiency is related to its inlet temperature. Higher turbine inlet temperature (TIT) leads to higher turbine efficiency. In addition to high temperatures, turbines operate at high rotational speed, leading to huge centrifugal forces on their blades. The latter can be damaged by resultant creep rupture and fatigue failure, in addition to high resonance and vibrations (Breeze, 2016). As a result, turbine blade materials should have high melting points, good oxidation/corrosion resistance, and a high temperature strength (Soumikh et al., 2018).

A new method to increase the TIT in a gas turbine cycle without affecting the turbine blade material is by cooling them. There are two types of cooling techniques: internal cooling and external cooling. The first consists of compressed air flowing inside the channel of the turbine blade. The heat transfer occurs between the turbine blade and the coolant air. The latter consists of injecting cooling air through holes on the surface of the turbine blade. The coolant air forms a layer between the turbine surface and the hot gas.

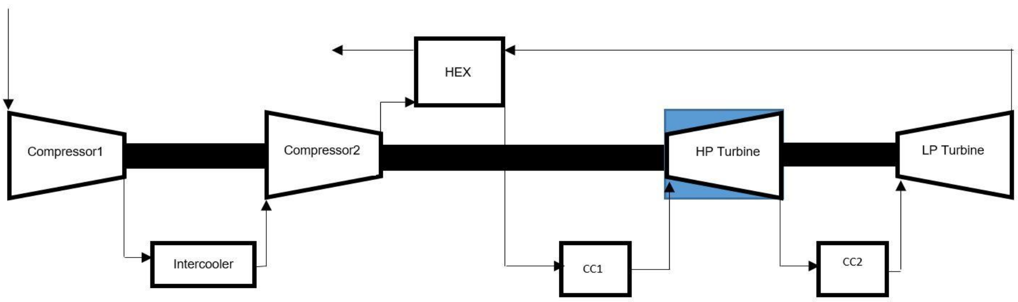

Figure 3: IRReGT cycle with cooled HP Turbine

Salpingidou et al. (2017) developed a methodology to evaluate the cooling technique with film cooling based on the theory of Young and Wilcock. However, this study assesses the internal cooling technique of the HP turbine and the thermodynamic analysis is a refinement of the different approach developed by (Bala Prasad et al., 2016).

3.1 Thermodynamic analysis

This section presents a thermodynamic analysis of the current cycle with a cooled HP turbine presented in Figure 3. This study aims to determine the influence of blade cooling on cycle efficiency. A mass fraction from the compressed air of the first compressor is extracted and inserted at the root of the HP turbine blade. The air circulates through passages reaching the trailing edge, where it exits and mixes with the mainstream. The expansion in the HP turbine is only achieved by the remaining mass of compressed air. The coolant air passes through each blade of the HP turbine reaching the trailing edge, then exits the blades and mixes with the main flow before entering the second combustion chamber (CC2). Exiting the CC2 with a high temperature, the flow is then expanded in the LP turbine.
A thermodynamic analysis is conducted to compare the proposed cycle with a cooled HP turbine with the three GT configurations presented before. The performance parameters are as follows:

- The temperature and pressure of the air at the compressor inlet are the ambient temperature 25 °C and the atmospheric pressure 1.013 bar
- The compression ratio of each compressor1 and compressor2 is respectively 3.2 and 3
- The expansion ratio of HP turbine and LP turbine is respectively 3.2 and 3
- The temperature of the compressed air after the intercooler is 35 °C
- TIT at the uncooled LP turbine is 1,100 °C, TIT is limited by the material constraints
- The isentropic efficiencies of the compressor, the turbine, and the regenerator are respectively 0.8, 0.85, and 0.8 (Bou Nader, 2019)

The calculation methodology of the power output and the efficiency of the proposed cycle is presented below. The turbine blades in this case are subjected to internal convection cooling.
The internally cooled turbine is treated as a heat exchanger operating at a constant metal temperature (Tw) and the coolant exit temperature (Tc,o) is expressed in function of the heat exchanger effectiveness as proposed in (Bala Prasad et al., 2016). Figure 4 is a schematic presentation of the cooled turbine.

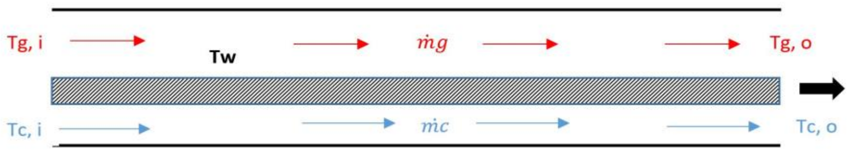

Figure 4 : A schematic presentation of the cooled turbine as a heat exchanger

The enthalpy at the exit of the HP turbine is calculated using energy balance as presented in Eq(3):

$$\dot{m}_g h_g + \dot{m}_c h_c = \dot{m}_{total\ air}\, h_{mix} \tag{3}$$

The coolant air from the compressor's first stage exit is determined by a mass fraction ratio as suggested by Eq(4):

$$X = \frac{\dot{m}_c}{\dot{m}_{total\ air}} \tag{4}$$

The blade heat effectiveness is expressed by:

$$\varepsilon = \frac{T_{c,o} - T_{c,i}}{T_w - T_{c,i}} \tag{5}$$

The blade heat effectiveness depends on several parameters: the cooling air temperature, the heat transfer area, coefficient, and the temperature difference. The effectiveness is assumed to be between 0.3 (film cooling) and 0.5 (internal cooling) (Bala Prasad et al., 2016).
The heat flux between the hot gas and the coolant air is calculated by:

$$Q(kJ) = \dot{m}_c Cp_c (T_{c,o} - T_{c,i}) \tag{6}$$

The output net specific work of the cycle is calculated by Eq(7). The mass of air in the different components is not the same and it depends on the mass of the coolant.

$$W_{net} = [(1-X).W_{HP-turbine} + W_{LP-turbine}] - [W_{compressor1} + (1-X).W_{compressor2}] \quad (7)$$

Finally, Eq(8) represents the efficiency of the IRReGT with a cooled HP turbine.

$$\eta = \frac{W_{net}}{(1-X).q_{cc1} + q_{cc2}} \quad (8)$$

3.2 Results and discussion

As mentioned in the section before, the coolant effectiveness depends on the internal heat transfer area and the heat transfer coefficient of the blade. The variation of the coolant effectiveness from 0.1 to 0.9 decreases the coolant mass fraction from 0.0659 to 0.0068 and increases the cycle efficiency by 2.32 points (Figure 5). The result is expected because increasing coolant effectiveness means that the heat transfer coefficient is higher; and, less coolant mass fraction is needed. By increasing the coolant effectiveness, the temperature of the coolant at the exit of the HP turbine is higher leading to a higher mixture temperature before entering the CC2. Fixing the TIT2 to 1,100 °C, less energy consumption is needed in the CC2.

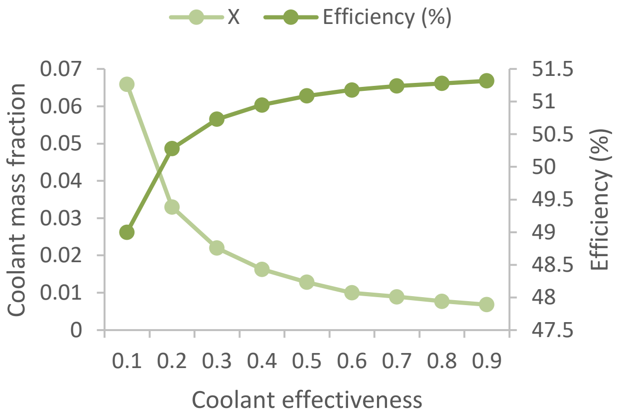

Figure 5-Effect of cooling effectiveness on coolant mass fraction and cycle efficiency

The cycle optimum inlet temperature and efficiency depend on the turbine blade operating temperature as depicted in Figure 6. Comparing the cycle efficiency in Figures 6a and 6b, it is shown that the increase in blade temperature increases the overall cycle efficiency. For HP TIT =1,300 °C, the cycle efficiency is 34.4 % for Tw = 226.85 °C and 50.43 % for Tw = 1,100 °C.

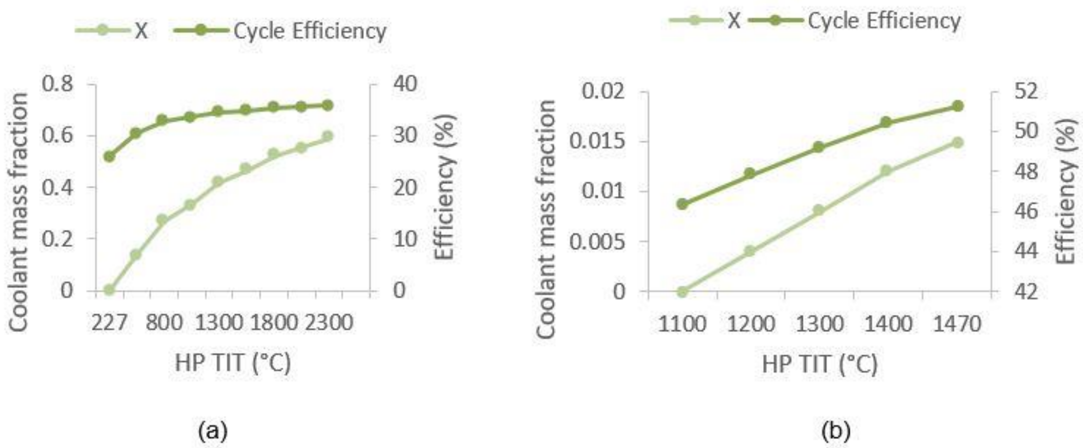

Figure 6-Effect of HP TIT on coolant mass fraction and cycle efficiency for (a) Tw=226.85 °C, (b) Tw=1,100 °C

For a given blade temperature Tw = 1,100 °C and coolant effectiveness of 0.5, the coolant fraction increases to 0.0128 while increasing TIT from 1,200 °C to 1,450 °C (Figure 6b). The result obtained is coherent because as TIT increases, more coolant requirements are necessarily leading to higher air coolant flow. Increasing TIT from 1,100 °C to 1,450 °C increases the cycle efficiency by 4.78 points. Although, higher coolant mass is related to lower mass quantity expanded in the first turbine, the effect of higher TIT is more relevant.

4. Conclusions

The GT cycle is a potential range extender for SHEV by recharging its battery. The aim of this work is to suggest a new configuration for GT cycle with a higher efficiency than the ones studied in the literature. A novel GT cycle representing an IRReGT with a cooled HP turbine is investigated. Convection cooling is applied in order to reach higher turbine inlet temperature and higher cycle efficiency. The thermodynamic analysis outlines the impact of cooling the turbine blade on the cycle efficiency compared to the standard IRReGT cycle. The cycle efficiency is increased by 4.78 points comparing to the IRReGT cycle for an HP TIT = 1,450 °C and a coolant effectiveness of 0.5. This study shows as well that a further increase in the TIT for the blade cooled cycle leads to a further increase in efficiency and the ability to remove the second combustion chamber. Thus, the cycle will be with less complexity. Possible future developments of this research work can be taken into consideration such as the effect of increasing the TIT on the NOx emissions, the assessment of a new GT cycle with a HP cooled turbine but without a second combustion chamber, and economic implications of the system.

Nomenclature

h – Specific enthalpy, kJ/kg
ṁ – Mass flow rate, kg/s
q – Heat flux, kJ/kg
W – Net specific work, kJ/kg
η – Cycle efficiency
c – Coolant flow
CC – Combustion Chamber
g – hot gas flow
GT – Gas turbine
GTs – Gas turbines
HEX – Heat exchanger
HP turbine – High Pressure turbine
ICE – Internal Combustion Engine
i –inlet
IRReGT – Intercooled Recuperated Reheated gas turbine cycle
LP – Low Pressure turbine
mix – mixture of coolant and mainstream flow
o – outlet
RGT – Recuperated Gas turbine cycle
w – wall
X – Extracted mass ratio

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