Theoretical study on performance of a combined gas and steam turbine propulsion system for road transport

K Uzuneanu
Department of Thermal Systems & Environmental Engineering, “Dunarea de Jos” University of Galati, Galati, Romania
E-mail: kristina.uzuneanu@ugal.ro

Abstract. Depletion of fossil fuels and environmental impact of their combustion currently represents a great concern. That is why research in energy field, where transportation represents a significant sector, is focused on finding solutions to match the increasing energy demand by consuming less fossil fuel and generating less emissions. In this approach, a combined gas and steam turbine propulsion system is proposed as alternative solution to reciprocating internal combustion engines (ICE) in road transport. Performance of this system is analysed in the paper. There are studied two configurations – with and without supplementary firing combuster after the gas turbine. The study indicates higher performance of the analysed system compared to the most advanced reciprocating ICE currently used in road transport sector. Besides, it reveals that supplementary firing should be avoided in typical operating conditions since increases specific fuel consumption and reduces efficiency.

1. Introduction
The global warming currently represents a subject of great interest due to its potential effects on human life. According to the recent studies, it is estimated to increase with 1…3.7 °C over the 21st century [1]. Several analyses strongly suggest that this warming is the result of CO₂ and other greenhouse gases concentration increase in atmosphere. They are also suggest that CO₂ concentration increasing is associated with fossil fuel burning [2] and an important contributor is the transport sector, especially transport on the road. Thus, 30.5 % of the globally greenhouse gas emissions of EU countries in 2014 was accounted in transport sector and the share of the road transport was 72.8 % [3]. It is estimated that CO₂ global emissions in transport sector will be at least doubled from 2014 to 2050 [4].

Beside the environmental impact, another concern related to the fossil fuel is depletion. There were studied financial strategies, such as an export revenue tax, to leave unextracted an optimal fuel stock into the ground [5]. On the other hand, energy consumption is continuously increasing and this requires more resources, so they are more and more perceived as unsatisfactory. The scenario in [6] assesses that electricity demands fulfilling will become a very serious problem in the next two decades, until 2040 [6]. In this approach, research in energy sector – where transportation have a significant share – concentrate their efforts to find solutions to match the increasing energy demand by assuming less fossil fuel dependency and less emissions. Hybridization is currently assumed as the most promising solution to this questions especially for urban zones, where low emissions is a more stringent requirement [7]. Besides, hybrid electric vehicles (HEV) are indicated by recent studies [8] as more energy efficient than electric-only vehicles. The analysis in [9] based on real-world data from
Brazil indicate that hybrid electric city buses use 30% less well-to-wheel fossil energy compared with conventional buses, with internal combustion engines (ICE). It should be noted that hybridization improves considerably the management of ICE operation but do not avoid the use of ICE. Thus, even in most advanced HEV, ICE are the first option as main source for power supply [10].

An alternative more cleaner to ICE in transportation is hydrogen fuel cell. The great advantage of fuel cells is that moving parts are missing. There are studies referring to hydrogen fuel cell as stand-alone propulsion system [11] as well as power supply source in hybrid systems [12]. The main drawback of fuel cell is the reduced number of hydrogen stations. For example, in Japan, where fuel cell technology is one of the most advanced – there were only 81 operative hydrogen stations in January 2017 [13]. A great advantage in what concerns flexibility to the fuel type is offered by Stirling engines, capable to run on almost any fuel, fossil or renewable. A Stirling engine version proper for propulsion is variable displacement alpha type, analyzed in [14].

Due to the lower pollutant emissions, lower size and lower mass, gas turbine (GT) engines were also analyzed as potential alternative solution to ICE in road transport but only as range extenders in hybrid configurations [15]; the study performed in [16] reveals that driving range ensured by GT is quite similar to reciprocating ICE. In stand-alone operation, efficiency of GT is lower than efficiency of ICE at this level of power. The perspective changes when use an additional steam turbine (ST) to convert the waste energy of GT in supplemental power. Performances of a combined gas and steam turbine (CGST) system for road transport is presented in [17]. This system includes supplementary firing in GT exhaust. A CGST configuration more complex, with two pressure levels in steam cycle, is analyzed as performance in [18] while size and mass are investigated in [19]. Studies on performances and dimensions of a CGST in hybrid configuration are performed in [20] and [21], respectively. This configuration is based on a GT with heat exchanger and also has two pressure levels in steam cycle. It should be noted that studies performed on CGST systems mentioned above indicate higher performance than ICE. But a comparative analysis of these configurations is not possible yet because each study was performed in peculiar conditions. Aiming to investigate the effects of supplementary firing on a CGST for road transport, a comparative analysis on performance of two CGST configurations – with and without supplemental firing – was performed in the present paper. Both configurations have one pressure level in steam cycle.

2. Analyzed configurations of combined gas and steam turbine system

The diagram in figure 1 presents the both analyzed CGST configurations – without supplementary firing (configuration 1) and with supplementary firing (configuration 2). Thus, the supplementary firing combustor – emphasized by the grey square – makes the difference.

![Diagram of CGST system](image_url)
An inconvenient feature of the conventional steam power station is water loss, which is induced by the cooling tower. A combined gas and steam power station faces the same problem since contains a steam power unit. This would be a major drawback in the case of a propulsion system for road transport. In order to avoid it, in studies developed so far [17]...[21] a condensing system with subcooled condensate spraying in ST exhaust steam was used instead of the conventional condensing solution. A similar condensing system, based on injection of condensate in ST exhaust steam, is used in the present study. The condensate injected in condenser is previously cooled by ambient air (see figure 1). Condensing pressure and temperature are 0.38 bar and 75 °C. Mechanical efficiency and isentropic efficiency of steam turbine were assumed \( \eta_{ma} = 99 \% \) and \( \eta_{ia} = 86 \% \).

As consequence of the fact that CGST is analyzed as alternative solution of reciprocating ICE in road transport, it was considered that CGST operates with gasoline. According to [22] and [23], the following properties were assumed for gasoline: carbon and hydrogen mass contents – \( g_c = 85.56 \% \), \( g_H = 13.01 \% \); lower heating value – \( LHV = 42840 \) kJ/kg; density – \( \rho_g = 0.752 \) kg/m\(^3\); stoichiometric air-fuel ratio – \( AFR = 14.46 \) kg air/kg fuel. The air compressor delivers an air mass flow rate \( D_a = 0.3 \) kg/s (with reference to the compressor inlet) in the combustion chamber of the gas turbine engine, where the temperature of flue gas rises to the gas turbine inlet temperature of 1800 K. It should be noted that current materials and turbine technology make possible turbine inlet temperature up to 1900 K [24]. Heat conservation efficiency of the combustion chamber is \( \eta_{co} = 98 \% \) while heat conservation efficiency of the supplementary firing combustor is \( \eta_{sf} = 97 \% \). After combustion chamber, flue gas expends in the gas turbine, than passes the heat recovery steam generator (HRSG) and gets to the stack. Similar to the steam turbine, mechanical efficiency and isentropic efficiency of the gas turbine were assumed \( \eta_{mg} = 99 \% \) and \( \eta_{ig} = 86 \% \). Flue gas temperature at stack was admitted \( T_{fg} = 383 \) K.

### 3. Analytical method

The comparative study on performances of the two CGST configurations was performed by quantifying their power outputs, fuel consumptions, specific fuel consumptions and efficiencies. These parameters are calculated as presented below.

Output power of CGST is the sum of the output powers provided by gas turbine engine and steam turbine. The output power of the gas turbine engine is calculated by subtracting the power consumption of the compressor from output power of the gas turbine. It results

\[
P_{CGST} = \left( P_{gt} - P_c \right) + P_{st} = \left( \eta_{mg} \cdot D_{gsp} \cdot w_{gt} - D_a \cdot w_c \right) + \left( D_i \cdot \Delta h_{st} - D_\text{es} \cdot \Delta h_{es} \right) \quad [kW],
\]

where:

- \( P_{gt} \) and \( P_{st} \) are output powers of the gas turbine and steam turbine, kW
- \( P_c \) is compressor power consumption, kW
- \( w_{gt} \) is specific work of the gas turbine, kJ/kg
- \( w_c \) is compressor specific consumption, kJ/kg
- \( D_i \) and \( D_\text{es} \) are HRSG steam mass flow rate and extracted steam mass flow rate, kg/s
- \( \Delta h_{st} \) and \( \Delta h_{es} \) are the entire change of steam specific enthalpy in steam turbine and change of steam specific enthalpy in steam turbine from extraction point to condenser, kJ/kg
- \( D_{gsp} \) is gas mass flow rate of the gas turbine, expressed as

\[
D_{gsp} = G_a \left( 1 - \chi_a \right) \left[ 1 + \left( \lambda \cdot AFR \right)^{-1} \right] \quad [kg / s],
\]

where \( \chi_a \) is fraction of air flow rate used as cooling air in gas turbine while \( \lambda \) is air excess ratio in the combustion chamber of the gas turbine.

Fuel consumption of the gas turbine engine is calculated as

\[
FC_{gt} = D_a \left( 1 - \chi_a \right) \left( \lambda \cdot AFR \right)^{-1} \quad [kg / s],
\]

while overall fuel consumption of CGST is given by
where \( \lambda \) is air excess ratio in the supplementary firing combustor.

Specific fuel consumption of CGST is calculated as

\[
SFC_{\text{CGST}} = FC_{\text{CGST}} P^{-1} \quad \text{[kg / kg h]},
\]

Efficiency of the gas turbine engine is expressed as [17]

\[
\eta_g = \left(1 - \chi_a\right) \frac{(1 + \lambda^{-1} AFR^{-1}) w' - w'' - w' - \chi_a}{(1 - \chi_a)^{-1} \left[ (1 + \lambda^{-1} AFR^{-1}) h_i - h_t \right]},
\]

while efficiency of the steam turbine engine is given by

\[
\eta_s = P_s \left( D_{f_t} \Delta h_{f_t} \right)^{-1} = \left( \Delta h_{f_t} - D_{f_t} \Delta h_{f_t} \right) \left( D_{f_t} \Delta h_{f_t} \right)^{-1}.
\]

In formulas (6) and (7) the following notations were made:

- \( w_{ar} \) is specific consumption for compression of the gas turbine cooling air, kJ/kg
- \( h_i \) is gas turbine inlet specific enthalpy, kJ/kg
- \( h_t \) is compressor discharge specific enthalpy, kJ/kg
- \( \Delta h_{f_t} \) is flue gas specific enthalpy change in HRSG, kJ/kg
- \( D_{f_t} \) is gas mass flow rate in HRSG, expressed as

\[
D_{f_t} = G_u \left( 1 - \chi_a \right) \left[ 1 + \left( \lambda^{-1} \cdot AFR \right)^{-1} \right] \quad \text{[kg / s]}.
\]

The formula expressing overall efficiency of CGST is [17]

\[
\eta_{\text{CGST}} = \eta_{\text{cg}} FC_{\text{cg}} \left( \eta_{\text{cg}} + \eta_s \left( 1 + \lambda AFR \right) h' \right) \left( \eta_{\text{cg}} \cdot LHV \right)^{-1} \left[ \eta_{\text{cg}} FC_{\text{cg}} \left( \eta_{\text{cg}} - \eta_s \right) FC_{\text{cg}} \right]^{-1}.
\]

The parameters of performance described above were analysed function by compressor compression ratio, \( r_p \), and HRSG gas inlet temperature, \( T_{HRSG} \).

### 4. Results of the analysis and discussions

The performance parameters of CGST, namely \( P_{CGST}, FC_{CGST}, SFC_{CGST} \) and \( \eta_{CGST} \), were analysed for both configurations function by the compressor compression ratio, \( r_p \). In the case of configuration 2, performance of CGST was also analysed function by HRSG gas inlet temperature (gas temperature after the supplementary firing combustor), \( T_{HRSG} \).

Results of the analysis are graphically presented in figure 2. The dashed curves are associated with configuration 1 while solid curves are associated with configuration 2. These curves represent variation of performance parameters with \( r_p \) in the range 5…25. In the case of configuration 2 they were drawn for \( T_{HRSG} \) in the range 1050…1400 K.

It can be observed that only the curves described by \( T_{HRSG} = 1400 \) K are covering entire range of \( r_p \), from 5 to 25 in the case of configuration 2. In other words, \( T_{HRSG} \leq 1350 \) K is not possible when \( r_p = 5 \). The explanation is that gas turbine outlet temperature is higher than 1350 K in these cases, so supplementary firing is meaningless. When \( r_p \) gradually increases, gas turbine outlet temperature gradually decreases. Thus, supplementary firing can be associated with lower values of \( T_{HRSG} \).

Supplementary firing brings additional power (see figure 2a) but induces higher fuel consumption (see figure 2b). The higher \( T_{HRSG} \), the higher \( P_{CGST} \) and \( FC_{CGST} \) are. In what concerns \( r_p \) influence, it can be observed that is an optimum value with strict reference to output power. Regardless of \( T_{HRSG} \), this value is \( r_p = 19 \) and is associated with maximum \( P_{CGST} \); in order to emphasize it, notation \( r_{pp} \) is used in this case. \( FC_{CGST} \) increases continuously with \( r_p \).

As figures 2c and 2d indicate, configuration 1 offers lower specific fuel consumption and higher efficiency than configuration 2 on the entire range of \( r_p \). Lowest \( SFC_{CGST} \), of 0.158 kg/kWh, and
Figure 2. Variation of performance parameters with $r_p$ for both CGST configurations.
highest $\eta_{CGST}$, of 0.53, are achieved at $r_p = 25$ – the upper limit of $r_p$ range. This is assumed as optimum compressor compression ratio for CGST. Efficiency value mentioned above (0.53) indicate that CGST in configuration 1 is significantly more performant than current reciprocating ICE used in vehicles propulsion, which are described by efficiencies lower than 0.42 [25].

In spite of a lower efficiency compared to configuration 1, configuration 2 is still more efficient than current reciprocating ICE used in vehicles propulsion since efficiency is higher than 0.44 on the entire ranges of $r_p$ and $T_{HRSG}$. It can be observed that the higher $T_{HRSG}$ (the higher level of supplementary firing heat input), the higher $SFC_{CGST}$ is while $\eta_{CGST}$ is lower. This indicates that supplementary firing is justified only in overload operating regimes. In typical operation it should be avoided since induces increasing of $SFC_{CGST}$ and reducing of $\eta_{CGST}$. In the case of configuration 2 and for $T_{HRSG} \geq 1100$ K, minimum specific fuel consumption and maximum efficiency are achieved when $r_p = 16$; notation $r_pT$ is used in this case. It can be assumed that $r_pT$ is the optimum compression ratio if intend to use the supplemental firing by default.

5. Conclusions
The analyzed CGST system represents a small scale application for propulsion of the combined gas and steam turbine power systems, which currently are the most advanced solutions for power generation at high and medium power level.

The output power of CGST system, up to 232 kW in configuration 1 and up to 267 kW in configuration 2, recommend it for propulsion of medium-size vehicles like mini-trucks or minibuses.

Efficiency of CGST, up to 0.53 in configuration 1, indicate it as an attractive alternative to reciprocating ICE in road transport propulsion field.

Highest efficiency and minimum specific fuel consumption of configuration 1, namely 0.53 and 0.158 kg/kWh, are achieved when $r_p = 25$. This is assumed as the optimum compressor compression ratio for CGST.

Supplemental firing – characteristic to configuration 2 – is justified only by overload requirements. The use of supplemental firing should be avoided in typical operating conditions for economic reasons. It can be concluded that optimum solution is to include the supplementary firing combustor in CGST scheme but to use it only in special circumstances, like peak power demand.

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