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Design of Waste Heat Recovery System based on ORC for a Locomotive Gas Turbine

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ABSTRACT

Gas turbines are widely and conveniently used in myriads of applications from aerospace to industrial and marine to power generation. However, their application in the field of locomotion is still not as widespread as it could be. The present study attempts to showcase feasibility to overcome some of the major limitations that restricted the application of gas turbines in this field in the first place. The aim is to showcase that a higher efficiency design is possible with gas turbines while taking care of all the constraints put by their application in locomotives. In the present study, initially a gas turbine cycle was designed with a target power output of 9 MW. The basic cycle that was designed had a thermal efficiency of 34.5%. To increase the efficiency of the system, an Organic Rankine Cycle based system was employed as the bottoming cycle which extracted the heat from the compressed air (intercooler) as well as from the exhaust. The pressure ratios of the compressors and the operating pressures and temperatures in the ORC system were optimized using parametric studies. The addition of intercooling and recuperation to this basic system and the incorporation of the bottoming cycle resulted in increasing the efficiency to about 47.94%. Since the component efficiencies were assumed during the initial cycle, preliminary design for the turbines and compressors for both gas turbine and ORC unit was performed. Data from the preliminary component designs were used in the cycle calculation along with the provision of required component cooling to arrive at the final combined cycle performance. With actual efficiencies of the components (as obtained from the preliminary design) were included into the cycle, the final efficiency obtained is 46.49% for the gas turbine unit with bottoming cycle.

INTRODUCTION

Gas turbines find applications in aerospace, marine, power generation and many other fields. Recently there has been a renewed interest in gas turbines for locomotives. (Herbst et al., 2003) Though gas turbines were first used in locomotives in 1950 – 1960's, the rising fuel cost made them uneconomical for commercial operation and almost all of them were taken out of service. The diesel locomotives gained popularity and presently locomotives are operated by diesel engines and electric motors. The emission levels in diesel locomotives have raised concerns among the environmentalists, leading to stringent emission norms in recent years. One of the solutions to reduce emission for these locomotives is to switch to LNG fuel which requires huge investment in upgrading the engines to operate with LNG. The other alternative is Gas Turbine based locomotives and this has gained renewed interest with RZD and Sinara Group of Russia successfully operating LNG based Gas Turbine-electric locomotives. Fig. 1 shows the GT1-001 freight GTEL from Russia, introduced in 2007. It runs on liquefied natural gas and has a maximum power output of 8,300 kW (11,100 hp). Presently, this locomotive holds the Guinness record for being the largest gas turbine electric locomotive (Source: http://www.guinnessworldrecords.com). Though there have been a lot of improvements in gas turbines, the thermal efficiency is still not very high unless the exhaust heat is efficiently utilized by a bottoming cycle.

Fig. 1 Russian GT1_001 gas turbine locomotive

Converting the gas turbine into a combined cycle unit, with a bottoming steam cycle, is employed in case of several land-based and marine applications; however, such an option is not practical in a locomotive gas turbine due to the requirements of steam generators, steam turbines and other auxiliaries. The next best alternatives are to utilize either an organic Rankine cycle (ORC) or a supercritical carbon dioxide cycle (sCO_2) to extract heat from the exhaust of the gas turbine and convert it into useable energy in the bottoming cycle (Rudenko et al., 2015; Moroz et al., 2015a; Moroz et al., 2015b; Nassar et al., 2014; Moroz et al., 2014). Supercritical carbon dioxide cycles, operating in a closed-loop Brayton cycle, are still in research phase. There is not much practical experience in deploying an sCO_2 unit for propulsion gas turbines even though there is considerable research currently in progress. Hence, the obvious choice is to incorporate an ORC based system which is compact, modular and easy to operate. The same concept can also be implemented in any gas turbine application, be it a land-based, power generation, or marine application.

As the first step in this study, a gas turbine cycle was designed with a target power output of 9 MW to be used as Gas Turbine Electric Locomotive (GTEL). The base cycle was designed using a simple open cycle configuration which was subsequently modified to incorporate a recuperator to increase the overall thermal efficiency

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of the cycle. It is well known that using intercoolers between compression stages can result in reduced work for compressor thus resulting in improved cycle efficiency. At the same time, it requires additional work for extracting heat from the compressed air. (Cohen et al., 1996). In addition to the use of intercoolers and recuperators, the authors in the present work propose the use of a waste heat recovery system (WHRS) with the locomotive gas turbine to increase the power output as well as the overall efficiency of the system. The waste heat available in this cycle is at two locations, one at the exhaust and the other between the compressors (intercooler). Here, an ORC based system was proposed as the bottoming cycle which extracted heat from the compressed air and also from the exhaust after recuperation. Multivariable parametric studies were performed to optimize the pressure ratios of the compressors on either side of the intercooler. The operating pressures and temperatures of the ORC system were also determined by such studies. To keep the overall target power output to about 9 MW, the gas turbine was resized based on the additional power that could be generated in the bottoming cycle.

THERMODYNAMIC CYCLE OF THE GAS TURBINE

The conceptual thermodynamic cycle for the basic gas turbine unit based on the Brayton cycle was developed using commercial software AxCYCLE™. The software provides flexibility in building the conceptual cycle, by using appropriate available components, as well as in choosing the parameters to be assigned as fixed constraints in the boundary conditions without the need for any geometric information. The boundary conditions, used for the present design, are shown in Table 1. The cycle layout is shown in Fig. 2a. The overall efficiency of the cycle is 34.5%, which can be considered as idealistic, since the effect of cooling in turbines, as well as inlet and outlet pressure losses were not considered.

Table1: Boundary conditions and parameters assigned for the components

Sl. No	Parameter	Value	Units
1	Compressor inlet pressure	1.01	bar
2	Compressor inlet temperature	20	Celsius
3	Compressor outlet pressure	17.8	bar
4	Combustor outlet temperature	1200	Celsius
5	Turbine outlet pressure	1.02	bar
6	Generator terminal power	9000	kW
7	Turbine Efficiency	0.90	-
8	Compressor internal efficiency	0.79	-
9	Compressor mechanical efficiency	0.98	-
10	Combustor efficiency	0.98	-
11	Generator efficiency	0.98	-

The cycle was then modified to incorporate the recuperator. In the modified cycle, the compressed air from the compressor passes through a recuperator and gets heated by the exhaust gases from the turbine. This resulted in efficiency of the cycle increasing to 35.56% (Fig.2b). The boundary conditions and component efficiencies were all retained same as in the open cycle. A pinch type recuperator model was considered with temperature difference of 10 degrees at pinch point. Pressure losses in the recuperator were specified as 0.1 bar on the hot side and twice that value on the cold side. The value of heat recovery coefficient was estimated to be 0.9. It was found that the use of recuperator reduces the fuel flow to 0.522 kg/s compared to that of 0.538 kg/s in the base cycle, for the same power output, resulting in increase of cycle efficiency with recuperator.

After the addition of the recuperator in the base cycle, it was found that the exhaust temperature after recuperator is 483^0C. Since the compressor outlet temperature itself is high, not much heat is transferred in the recuperator to the compressed air. If the compressed air temperature at the outlet of compressor is reduced, retaining the same compressor pressure ratio, the possibility of higher heat transfer in the recuperator is possible which can possibly improve the cycle efficiency further. This is precisely what the addition of an intercooler does with an added benefit of lower work input to the compressor after intercooling. Thus, two opportunities opened up to increase the power output as well as the efficiency in the cycle. The first was to modify the base cycle, by addition of an intercooler, to allow more heat transfer in the recuperator resulting in lower exhaust temperature; and the other was to utilize a part of the remaining heat from the recuperator exhaust in a waste heat recovery system. Both modifications were systematically applied in combination in this work. As the next step, the intercooler was combined into a waste heat recovery system to make full use of the waste heat from all the sources.

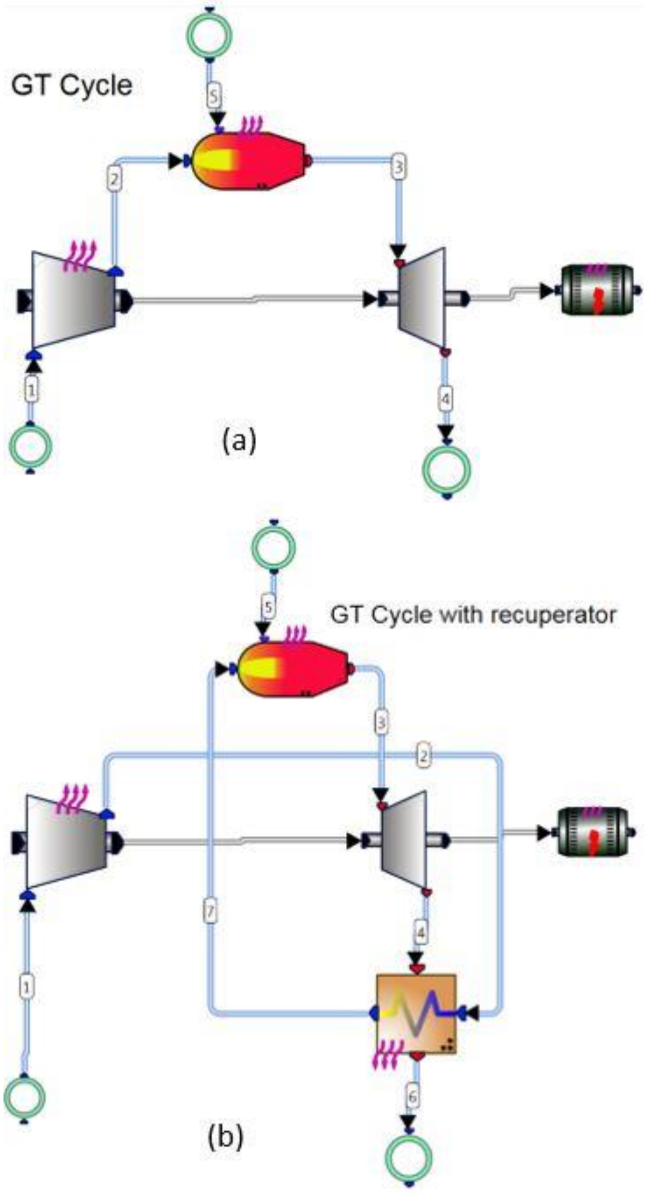


Fig. 2: Gas turbine cycle without (a) and with (b) recuperator for 9 MW power output

First, to reduce the overall compression work and the temperature of the compressed air, an intercooler was added and the compression was divided in two parts with the temperature of air after intercooling fixed to 100^{0}C. To utilize energy from the recuperator exhaust, an ORC system consisting of a turbine, an air-cooled condenser, a pump and a heat exchanger was added (Fig. 3). Pressure drops on the hot and cold sides of the heat exchanger were assumed to be 0.1 bar and 0.2 bar, respectively. The fluid used in the ORC cycle was R245fa based on previous experience of Moroz et al. (2013) in working with this fluid. The ORC system was designed for a maximum working pressure of 32 bar and the turbine inlet temperature was 200^{0}C. Since our objective was to maintain the total power at about 9 MW, the power output from the gas turbine was iteratively reduced to about 8.21 MW. At this rating of the gas turbine, the ORC system produced about 0.839 MW to give the total net power desired. The overall system efficiency had increased to 45.6%.

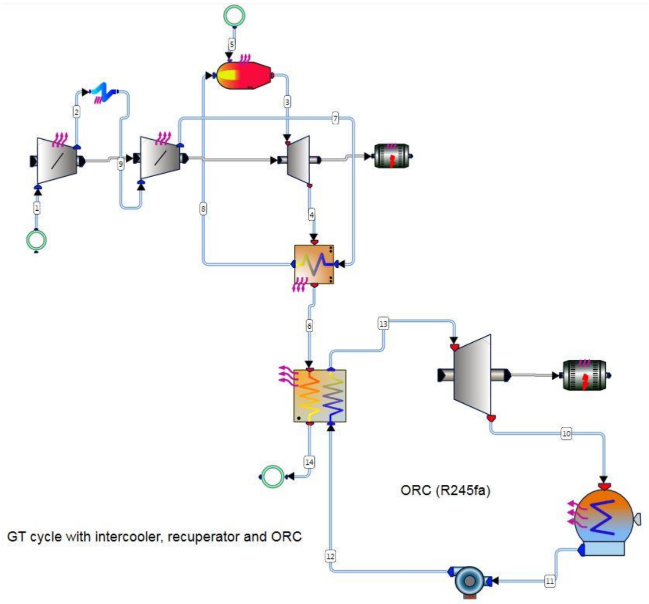


Fig. 3: Gas turbine cycle with an added intercooler, recuperator and ORC cycle

After solving the cycle of Fig. 3, it was found that considerable amount of heat was transferred through the intercooler at a fairly high temperature. This heat is wasted in the form of temperature rise in the cooling circuit. Also, some additional work is utilized to run the cooling flow to the intercooler (not simulated here) which is a penalty to the total output from the system. Here, instead of using a cooling circuit to remove heat from the compression stages, this heat was directed to the ORC system such that it could utilize heat from both the exhaust as well as the intercooler. Fig. 4 shows the thermodynamic cycle of the gas turbine unit with such a configuration for the ORC system.

As shown in Fig. 5, a parametric study was performed to obtain the ideal values of pressure ratios for the two compressors as well as the output power of the gas turbine cycle. The feature of generating solutions from multiple variables using Low Discrepancy Sequence (LDS) model which is built inside AxCYCLE™ was used to perform this study, wherein the pressure ratios for both compressors and the generator power from the gas turbine cycle were chosen as variables for the study. Net power production and cycle thermal efficiency were the chosen objective functions. Two additional parameters were also assessed during the study, viz. turbine outlet temperature in GT cycle and ORC mass flow rate. The pressure ratios were varied between 2 to 7 for both compressors and the gas turbine electrical power output was allowed to change between 7.5 to 7.9 MW. 400 combinations of variable values were chosen for calculation by a quasi-random Sobol sequence, within the specified variable ranges and the calculated efficiency are shown in the design space (Fig. 5a).

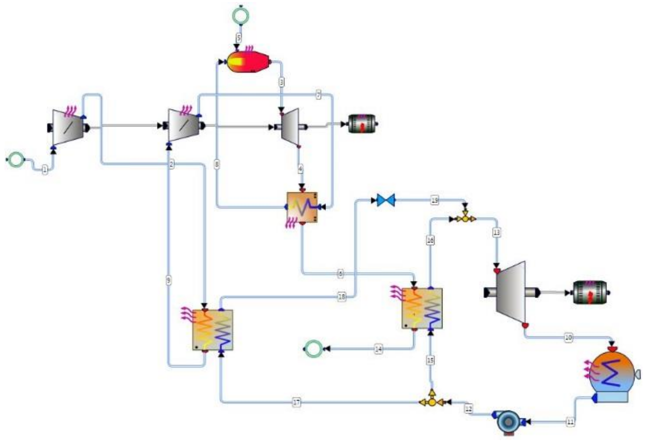


Fig. 4: Gas turbine cycle with an ORC cycle extracting heat from the exhaust and from the intercooler

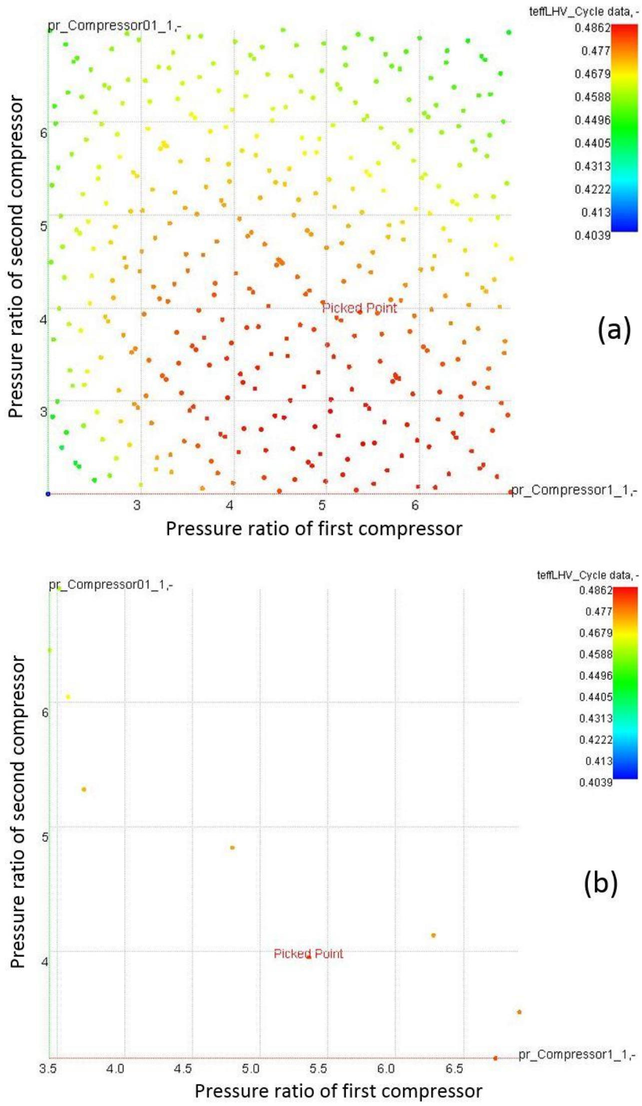


Fig. 5: Selection of compressor pressure ratios based on thermal efficiency (colour contours) and required net power

The calculated results show a wide range of output parameters.

A refinement was done by filtering out the solutions which gave net power production lower than 9 MW and higher than 9.05 MW. An additional filter was applied to the calculated values of the turbine outlet temperature in gas turbine system. Solutions with the turbine outlet temperatures lower than 500^{0}C and higher than 600^{0}C were also filtered out. After the refinement, all but 9 out of the 400 solutions were filtered out (Fig. 5b). Out of the remaining 9 combinations of the variable values, the one that gave the best thermal efficiency was chosen for further study. This particular design had the first compressor pressure ratio as 5.36, the second compressor pressure ratio as 3.95 and a gas turbine electric power production of 7.89 MW. The cycle in Fig. 4 was recalculated with these values of the three variables. The calculated efficiency, in this case, was 47.94% with the compressed air temperature at the inlet of the intercooler as 229.56^{0}C.

PRELIMINARY DESIGN OF THE COMPRESSORS

The efficiencies and component performances at this stage of cycle design were based on assumed values and previous experiences. However, to get realistic values of the performance, preliminary design needed to be performed and this was carried out using the commercial software AxSTREAM® as it allows designing the complete turbomachinery from conceptual stage with very few inputs. The preliminary design module of AxSTREAM®, based on an inverse task solver, generates thousands of designs allowing to choose the optimal yet feasible solution by considering multiple objective parameters, including manufacturability in a short span of time.

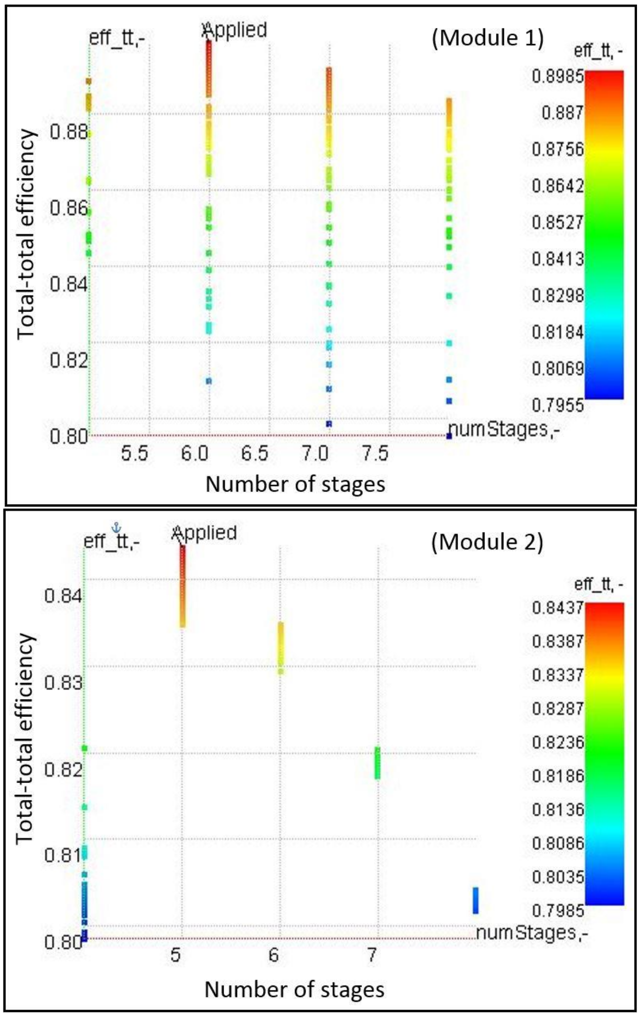


Fig. 6: Preliminary designs of compressor modules and comparison of various designs

Here, both compressors were designed simultaneously, considering them as two different modules and taking in to account the pressure drop and temperature change in between the two modules due to the presence of the intercooler. Fig. 6 shows the comparison of different geometries created for both compressors in the preliminary design module. The best design consisted of a total of 11 stages (6 stages in the first compressor module and 5 in the second) with total-total efficiencies of 89.85% and 84.37% for the modules, respectively. The compressors generated a combined pressure ratio of 20.79 with a mass flow rate of 20.65 kg/s at an optimal speed of 14300 rpm. Various designs generated during this phase were further evaluated, and additional constraints were specified to filter the solutions and to choose the optimal flow path that met both geometric and performance criteria.

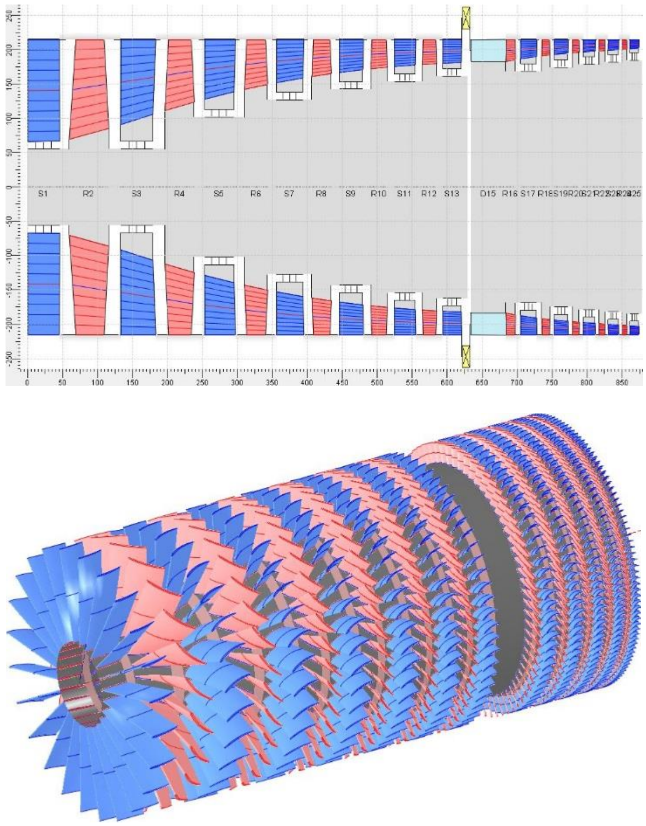
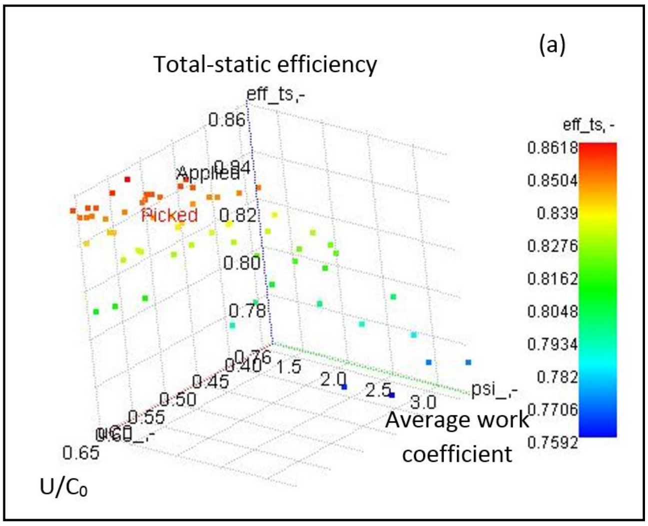


Fig. 7: Flow path of the designed compressor.

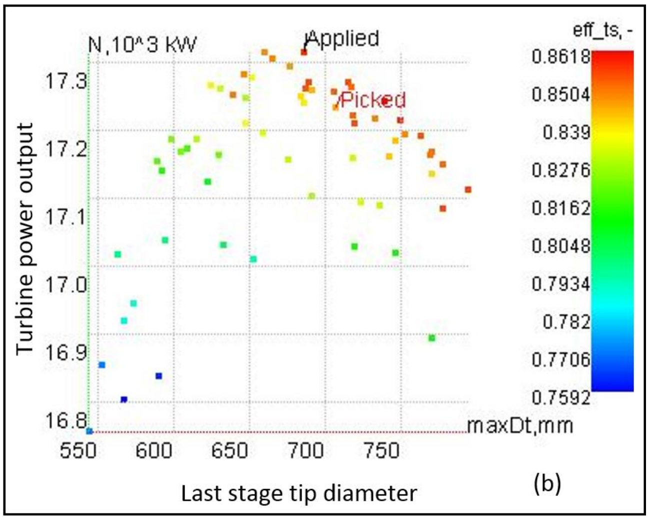
The selected design was analysed in the meanline solver and blade twist was incorporated for the first four stages based on the diameter to length ratio (D/L < 10). Streamline calculations were performed after reassigning the pressure loss in the intercooler and maintaining the cooler outlet temperature as per the cycle parameters. After detailed design, the total-to-total efficiencies for both compressors modules were increased to 90.64% and 87.05%, respectively, at the design point. Fig. 7 shows the compressor flow path with the intercooler. The total power consumption in the process of compression is about 8.5 MW out of which the first compressor requires 4.17 MW whereas the second compressor after the intercooler consumes 4.41 MW.

PRELIMINARY DESIGN OF THE GAS TURBINE

The boundary conditions for the turbine were similarly taken from the cycle calculation in AxCYCLE™ and preliminary design was performed using AxSTREAM®. Various designs generated at this stage were evaluated and filtered based on different constraints, and the optimal design was selected for detailed designing. Fig. 8(a) shows the comparison of total-static efficiency as a function of work coefficient and isentropic expansion ratio (U/C0).



(a) Effect of isentropic velocity ratio and average work coefficient on turbine total-to-static efficiency



(b) Effect of maximum tip diameter on turbine power output

Fig. 8: Turbine preliminary designs and comparison of various designs

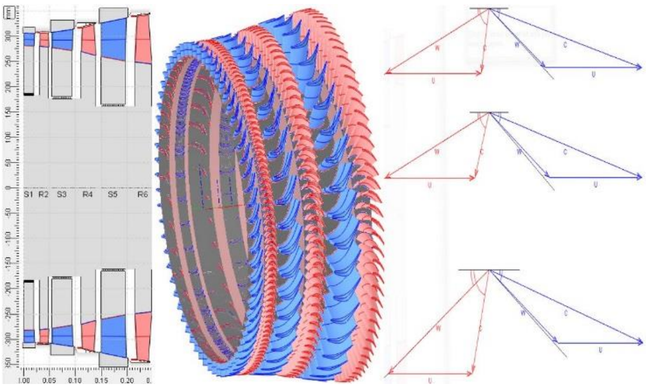


Fig. 9: Flow path of the designed turbine and velocity triangles at mean section shown from first stage (top) to last stage (bottom)

The effect of maximum tip diameter on the power output of the turbine is shown in Fig. 8(b). The selected design is composed of 3 stages and produces a total power of 17.4 MW with total to static pressure ratio of 19.46 and mass flow rate of 21.04 kg/s. The surplus power from the turbine, after supplying required power to the compressors, is about 8.82 MW. This surplus power is adequate to satisfy our design goal of 7.89 MW of net electrical power from the gas turbine even after considering all the losses such as shaft losses, generator losses etc. Detailed design was performed on the selected turbine configuration and the final flow path and velocity triangle were obtained as shown in Fig. 9.

PRELIMINARY DESIGN OF THE ORC TURBINE

The ORC turbine was also designed in a similar manner as the main gas turbine, by obtaining the boundary conditions from the cycle calculation. The working fluid for this design was R245fa with wide enough range to envelope the operating conditions of our ORC turbine. Various designs generated in the preliminary design module of AxSTREAM® were evaluated for both geometric and performance criteria and one final design was chosen for detailed study. The designed turbine and its velocity triangles after the mean-line calculation are shown in Fig. 10.

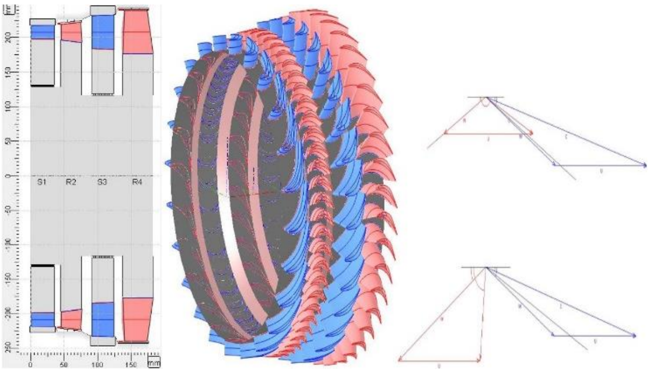


Fig. 10: Flow path of the designed ORC turbine and velocity triangles at mean section shown for the first stage (top) and for the second stage (bottom)

The design has two stages with the power output of 1.13 MW. The total-to-static pressure ratio of the turbine is 15.9 with a mass flow rate of 23.53 kg/s. The power generated by the ORC turbine is adequate enough to supply to the ORC pump and still produce enough output electrical power through the generator to satisfy our overall goal of 9 MW net output power.

THERMODYNAMIC CYCLE CONSIDERING COOLING IN THE GAS TURBINE

Since the turbine inlet temperature was 1200^0C, the turbine first stage nozzles and blades needed to be cooled. The next step was to incorporate cooling. This was done as an iterative process from the cycle to the components, as it involved extraction of air from the compressors and passing that air through the turbine vanes and blades to maintain reasonable blade metal temperature. Since it involved significant changes in the boundary conditions and flow path, the thermodynamic cycle was recalculated considering the cooling flow requirements. The cycle was modified by replacing the first turbine stage with a cooled turbine stage model. The downstream turbine stages were calculated as a separate turbine by adopting the new inlet conditions directly from the outlet of the cooled stage. (Fig. 11)

Only the first stage of the turbine required cooling as per the initial estimate. Usually the turbine cooling mass flow is determined during the detailed design phase, considering cooling methods employed and the temperatures on the surface of nozzles and blades. Based on prior experience, the initial approximation was to extract 4% of the total compressed air flow for cooling, before it entered

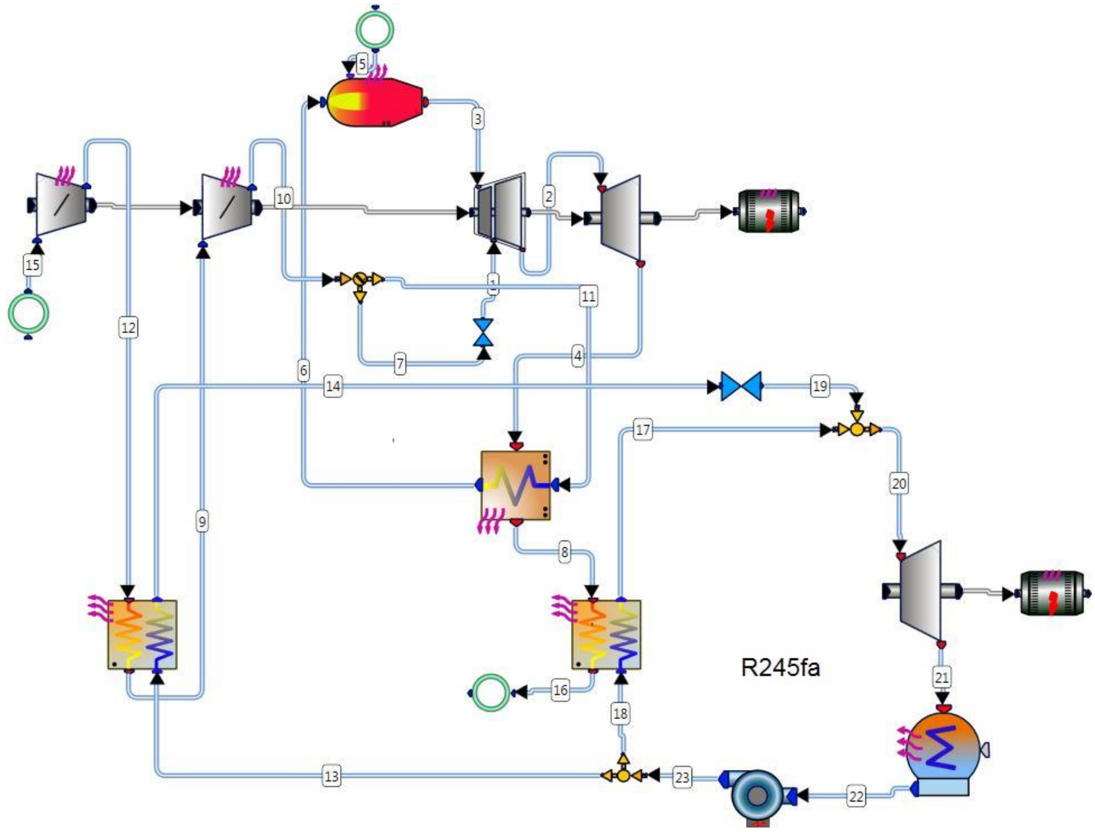


Fig. 11: Gas turbine cycle with cooling and ORC system

the combustor. The assumed efficiencies in the cycles were corrected for the turbines and the compressors considering the cooling flows in turbine and bleed in the compressor. The overall thermal efficiency of the system had slightly reduced to 46.3% compared to the earlier efficiency of 47.94%, with a net power production of slightly above 9 MW. The gas turbine produced 7.89 MW and the ORC turbine produced 1.18 MW of power. This shows that the application of a waste heat recovery system, in this case, leads to about 15% of additional power generation and a significant increase in overall thermal efficiency of the system.

CONCLUSION

In this research, the authors have studied the role of a bottoming ORC system in increasing the overall thermal efficiency of the gas turbine unit for application in locomotives.

- The thermodynamic cycle of a gas turbine was modelled using commercial software AxCYCLE™. The simple cycle had an overall thermal efficiency of 34.5%.
- The potential to recover large part of the exhaust heat was identified and a recuperator was added to the system, which resulted in increasing efficiency to 35.56%.
- A bottoming cycle, based on ORC, was added to extract heat from the exhaust after recuperator and also from the compressor intercooler using R245fa as the working fluid. This resulted in an increased thermal efficiency of 47.94% and reduced gas turbine size of 7.89 MW compared to the earlier rating of 9 MW. This emphasizes the advantage of using a waste heat recovery system to further leverage on the efficient use of wasted heat at higher temperatures.
- The compressors and turbines were designed based on the technical specifications obtained during the thermodynamic cycle development. Turbine cooling was incorporated in the cycle based on the prevailing thermodynamics in the cycle. The inclusion of cooling dropped the overall thermal efficiency to 46.3%. This additional step showcases that, even after incorporating all the possible sources of losses and performing the detailed component designs, the end result is still way more promising than the almost idealistic performance of a standalone plant.
- The flexibility in use of multiple fluids offered by gas turbines, combined with the higher performances obtained by incorporating waste heat recovery systems, make them a very promising alternative for locomotive propulsion while coping with the stringent environmental regulations. Although the application presented here is for locomotive gas turbines, the same can be implemented in any other application of gas turbines to improve the thermal efficiency.

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