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Investigation of the Performance of Different Recuperative Cycles for Gas Turbines/aero Engine Applications

Christina Salpingidou[a], Dimitrios Misirlis*[c], Zinon Vlahostergios[a], Stefan Donnerhack[b], Michael Flouros[b], Apostolos Goulas[a], Kyros Yakinthos[a]

[a]Laboratory of Fluid Mechanics & Turbomachinery, Department of Mechanical Engineering, Aristotle University of Thessaloniki, Thessaloniki, 54124
[b]MTU Aero Engines AG, Dachauer Strasse 665, Munich, Germany,
[c]Technological Educational Institute (TEI) of Central Macedonia, Serres, Greece,
misirlis@eng.auth.gr

This work presents an investigation of the performance of three recuperative cycles for gas turbines, with a particular interest for aero engine applications. The first configuration under investigation is the conventional recuperative cycle, in which a heat exchanger placed after the last turbine (low pressure or power turbine). In the second configuration, referred to as alternative recuperative cycle, a heat exchanger is placed between the high pressure and low pressure turbine, while in the third configuration, referred to as staged heat recovery cycle, two heat exchangers are employed, the primary one between the high and low pressure turbines and the secondary downstream the last turbine. At the first part of the present work, a parametric conceptual analysis was conducted using available literature data in order to investigate the impact of heat exchanger effectiveness and overall pressure ratio on cycle performance. The results show that the conventional recuperative cycle presents superior performance in relation to the alternative recuperative cycle for low overall pressure ratio values, while for higher values the alternative recuperative cycle outperforms. In addition, for the staged heat recovery cycle, the selection and combination of the effectiveness of the primary and secondary heat exchangers affects significantly the cycle efficiency. The second part of this work was focused on the assessment of practical issues regarding the implementation feasibility of the alternative recuperative and the staged heat recovery concepts in a recuperative aero engine. For the analysis, the advanced MTU-developed and designed intercooled recuperated thermodynamic cycle was used. The heat exchangers of the recuperation system in the intercooled recuperative cycle consist of specially profiled elliptic tubes placed in a 4/3/4 staggered arrangement. For the sizing of the recuperators, the GasTurb11 aero engines geometrical data was used as reference in order to design a recuperator which would be mountable in the limited available space between the intermediate pressure turbine and the low pressure turbine. In the analysis various recuperator scenarios were examined taking into consideration different axial lengths and tube core arrangements (5/4/5, 6/5/6 etc.) keeping always as basis the MTU-heat exchanger core geometry. For the determination of the recuperator inner/outer pressure losses and effectiveness, results from previously performed CFD computations, experimental measurements and from the ε-NTU method were used. The recuperator effectiveness and pressure losses for each scenario were included and assessed with the use of thermodynamic cycle models of the recuperative aero engine, which were developed in CAPE-OPEN/COFE software. The performance analysis of the recuperative aero engine cycles showed the existence of significant optimization potential which can be further increased when combined with more flexible aero engine geometry architectures and supported by the improvement of the endurance of recuperator candidate materials and alloys.

1. Introduction

Fuel consumption and increased pollutants emissions of gas turbines are important factors that an engineer should take into account for both environmental and economic reasons. Towards this direction, the

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exploitation of waste heat energy in gas turbines by integration of heat exchangers can be of significant value (McDonald, 1990). The conventional integration of heat exchangers (HEXs) in gas turbines is typically performed with the installation of a system of heat exchangers right after the last turbine (low pressure (LPT) or power turbine (PT)) exit in order to exploit the hot-gas high thermal energy content. The heat exchangers employed in these setups are usually following a cross/counter flow configuration setup, with the compressor discharge air flowing inside the heat exchangers tubes/channels and the hot-gas flowing on the tubes/channels external side. As heat is transferred from the hot-gas to the compressor (C) discharge air, the latter enters the combustion chamber (CC) with higher enthalpy and thus, the cycle fuel demand is reduced leading to increased cycle thermal efficiency. This selection of heat exchangers installation (corresponding to conventional recuperative (CR) cycle) is guided by the available space downstream the LPT and the overall relative simplicity in relation to alternative recuperation concepts. However, apart from this conventional approach, presented in Figure 1, some researchers have investigated adaptations in the conventional recuperative cycle by altering the positioning (Dellenback, 2002) or/and the number of heat exchangers in gas turbine applications (Dellenback, 2006), including helicopter engines, (Shapiro and Levy, 1990). In the present work the two new configurations of the recuperative thermodynamic cycle proposed by Dellenback (2006) are investigated in detail. The first configuration presented is referred as alternative recuperative (AR) cycle, with the heat exchanger being placed between the high pressure (HPT) and power turbine, as shown in Figure 2. In this concept, the heat exchangers preheat the compressor discharge air with high-temperature hot-gas, before the latter is fully expanded across the power turbine. Dellenback (2006) proposed an additional configuration that combines both CR and AR and is referred as ‘Staged Heat Recovery’ (SHR), which is presented in Figure 3. In the SHR configuration the number of recuperators is increased. Two heat exchangers are used, the first one (primary) is placed between the HPT and PT, at the same position as in the AR configuration, and the second one (secondary) at the gas turbine exhaust similarly to the CR configuration. In the last part the realisability of the AR and SHR cycles was investigated. Therefore, data from the intercooled-recuperative (IR) aero engine cycle of MTU were used, (Goulas et. al, 2015). In order to investigate the impact of the implementation of AR and SHR recuperative concepts on MTU aero engine cycle performance, a heat exchanger suitable to be mounted between the LPT and the IPT was designed. In the analysis the MTU heat exchanger design (Schonenborn, 2004) was used as the reference point having elliptically shaped tubes in a specially designed staggered arrangement. Various heat exchanger scenarios were examined taking into considerations different axial lengths and tube core arrangements. Already existing data from CFD computations and experimental measurements (Yakinthos et. al, 2015) combined with ε-NTU method (Kays and London, 1984) were used for the determination of pressure losses and effectiveness. Thermodynamic models for the AR and SHR cycle implemented in the MTU IR aero-engine concept were created and implemented in CAPE-OPEN/COFE flowsheet environment with COCO (CAPE-OPEN to CAPE-OPEN) simulator software, COCO(2016) and the performance of each case was examined. The comparison was based on the thermal efficiency, the specific fuel consumption and the complexity of the installation. The selection of possible recuperator materials was also taken into consideration due to the presented high temperature values.

2. Conceptual design

2.1 Thermodynamic model design

The first step for the analysis was the development of the computational thermodynamic models for each cycle. The models were implemented in the softwares: CAPE-OPEN/COFE and GasTurb11 (Kurzke, 2011). At the first stage of the investigation, a simple Brayton cycle without recuperation was designed and used as the reference case. The cycle is shown in Figure 1 and consists of one compressor and two turbines. The first turbine (HPT) is used to drive the compressor; therefore the work produced by the HPT is equal to the work consumed by the compressor, Eq(1). The last turbine (PT) is used for the work output, as shown in Eq(2)

$$\dot{W}_{compressor} = \dot{W}_{HPT} => \dot{m}c_{p12}(T_2 - T_1) = \dot{m}c_{p34}(T_4 - T_3) \tag{1}$$

$$\dot{W}_{net} = \dot{m}c_{p45}(T_5 - T_4) \tag{2}$$

where: c_p is the specific heat capacity, $\dot{W}$ the work, T the temperature, $\dot{m}$ the mass flow. The numbering is based on Figure 1. At the next steps, the thermodynamic models of the CR, AR and SHR recuperative cycles were developed, presented in Figures 2, 3 and 4. The cycle thermal efficiency was chosen for comparison of the performance of the configurations under investigation and its definition is given in Eq(3) where the network output is the work produced by the power turbine and in case of a simple cycle is given by Eq(2).

$$\eta_{thermal} = \text{net work output/heat input} \tag{3}$$

The heat input in the combustion chamber (following the numbering of Figure 1) is given in Eq(4) where $\dot{Q}_{in}$ is the heat input.

$$\dot{Q}_{in} = \dot{m}c_{p23}(T_3 - T_2) \tag{4}$$

One of the most critical parameters of the cycle is the turbine inlet temperature (TIT), which corresponds to the temperature of the gas as it leaves the combustion chamber. TIT was kept constant at 1,500 °C similarly to the work of Dellenback (2006).The most important parameter that is discussed in detail is the heat exchanger effectiveness (ε), defined in Eq(5), where $\dot{Q}_{achieved}$ is the achieved heat transfer and $\dot{Q}_{max}$ is the maximum possible heat transfer. The investigated effectiveness range is between 0.3-0.9.

$$\varepsilon = \dot{Q}_{achieved}/\dot{Q}_{max} \tag{5}$$

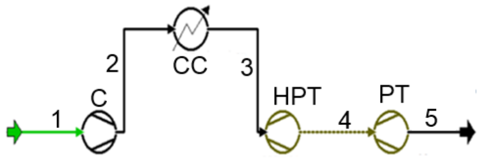

Figure 1: Brayton cycle model

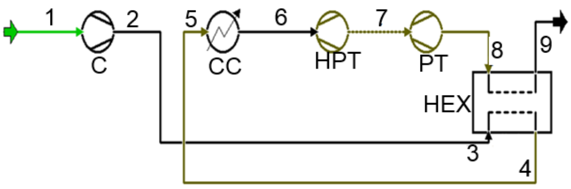

Figure 2: Conventional recuperative(CR) cycle

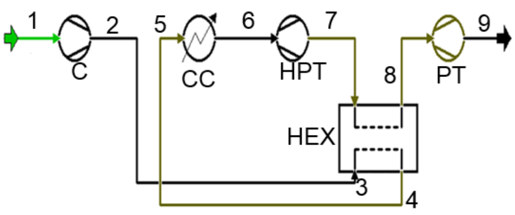

Figure 3: Alternative recuperative (AR) cycle

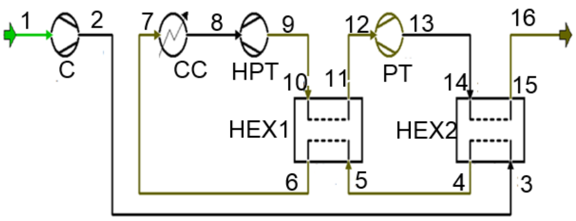

Figure 4: Staged heat recovery (SHR) cycle

Following the work of Dellenback (2006), the pressure losses for the hot (outer) and cold (inner) sides were set constant at 2 % and were assumed as a percentage of the pressure at the inlet of the heat exchanger, Eq(6).

$$DP_{losses}\% = (P_{in} - P_{out})/P_{in} \text{ (for both hot and cold flow)} \tag{6}$$

For the complete development of each model, three additional parameters were specified; the polytropic efficiency of the compressor, η_c=90 %, the polytropic efficiency of the turbine, η_t=87%, and the pressure loss percentage through the combustion chamber, DP/P=3 % of the inlet flow pressure, following the values used in the work of Dellenback (2006). Concerning the working fluid inlet conditions, air at 1atm and 15°C and the specific heat capacity of air (c_p), was provided as a function of temperature with its values based on CHEMSEP (CHEMSEP, 2016) and the Peng Robinson equation of state.

2.2 Results

Figure 5 shows the thermal efficiency of all cycles as a function of the overall pressure ratio and the variation of the performance as the heat exchanger effectiveness changes from 0.5 to 0.9. Beginning with the AR cycle, as OPR increases, the thermal efficiency of the cycle also gets higher and presents the same trend for all the heat exchanger effectiveness values. However, regarding the CR cycle as OPR increases the thermal efficiency increases but at a certain point (depending on effectiveness value) it starts to drop. The higher the heat exchanger effectiveness is, the lower the OPR value at which the decrease of the efficiency starts. Comparative results of the cycles show that the increase of heat exchanger effectiveness has a positive impact on both cycles. However, in case of low effectiveness the AR cycle starts to outperform at OPR equal to 21 whereas in case of very high value of effectiveness the AR cycle exceeds the CR at low OPR, at almost 17. These results lead us to the conclusion that the AR cycle is preferable to be used at high OPR, especially in case of low effectiveness values. The shaded area in Figure 5 shows the results range for the SHR cycle. The SHR efficiency values which are plotted correspond to various combinations of the heat exchangers effectiveness, ε_1 and ε_2. The maximum SHR efficiency values (indicated with thick line in Figure 5) are achieved for different combinations of the primary and secondary heat exchanger effectiveness, as shown in Figure 5. For low OPR values (OPR<~20) the SHR performance is maximized for a combination of ε_1=0.3 and ε_2=0.9, which corresponds to a setup more closely adapted to the CR cycle (low effectiveness value for the primary heat exchanger and high value for the secondary one). On the other hand, for high OPR values (OPR>~30) the SHR performance is maximized for a combination of ε_1=0.9 and ε_2=0.3, which corresponds to a setup more closely adapted to the AR cycle (high effectiveness value for the primary heat exchanger and low value for the secondary one). Additionally, the selection of the effectiveness for both the primary and secondary heat exchanger is of crucial importance since it can lead to a cycle efficiency variation of more than 20 % in absolute values, almost independently of the OPR.

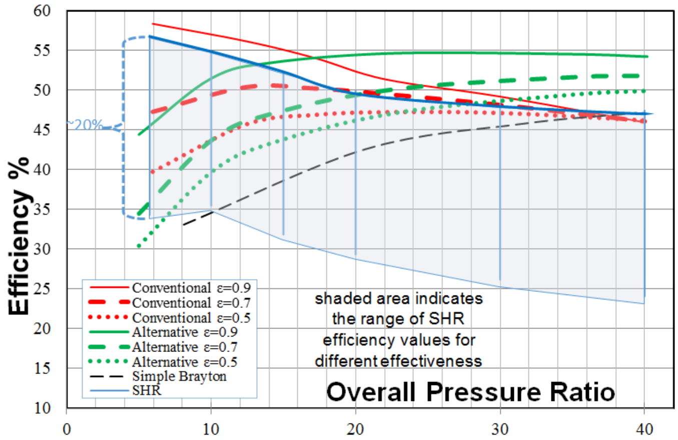

Figure 5: Efficiency of all cycles as a function of non-dimensional overall pressure ratio-OPR (SHR maximum values are achieved for ε_1=0.3 and ε_2=0.9 for 5.75≤OPR≤20 and ε_1=0.9 and ε_2=0.3 for 20<OPR≤40)

3. Realisability

3.1 Engine application

At this stage, a design study was carried-out to evaluate the feasibility of introducing the AR and SHR thermodynamic cycle into the IR turbofan aero engine design developed by MTU, shown in Figure 6. The IR engine is a three-spool configuration with a heat exchanger, which is shown in Figure 7, installed at the exhaust nozzle downstream the LPT (similar to a conventional recuperation setup). Details regarding the IR concept can be found in the work of Wilfert et al. (2007). Initially, a thermodynamic cycle model of the IR engine (corresponding to a setup closer to conventional recuperation cycle) was created in CAPE-OPEN/COFE and a performance analysis for average cruise conditions was carried out. Additional details about the selected average cruise conditions can be found in Goulas et al. (2015) and Schonenborn et al. (2004).

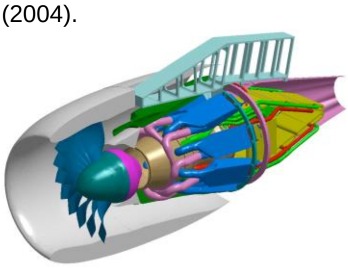

Figure 6: The IR aero engine concept

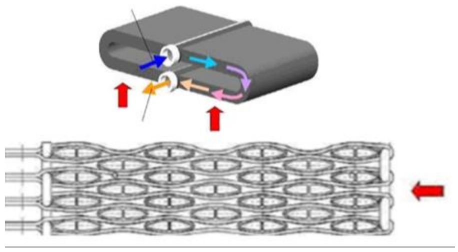

Figure 7:The MTU-heat exchanger (4/3/4 core)

At the next step, in order to examine the feasibility of SHR and AR cycles in IR engine, a heat exchanger suitable to be placed between the LPT and IPT duct was designed taking always into consideration the engine geometrical constraints. The first step in this direction was the identification of the available space between the IPT and LPT turbines. For this reason GasTurb 11 reference engine geometrical data were used and various heat exchangers were examined having always as reference the developed by MTU heat exchanger. This heat exchanger consists of specially profiled elliptic tubes placed in a 4/3/4 staggered arrangement, aiming to achieve maximum heat transfer rates and minimum pressure drop. Different axial lengths and tube core arrangements (3/2/3, 4/3/4, 5/4/5, 6/5/6 etc.) were taken into consideration. Due to the limited available space between the IPT and LPT, heat exchangers corresponding only to relatively low effectiveness values were realizable. In addition, the investigated heat exchangers were based on a radial distribution of elliptic tubes so as to be more properly mounted in the limited space between IPT and LPT. For every heat exchanger the heat transfer coefficients, the effectiveness and the pressure drop losses were calculated based on a combination of previously performed experimental measurements, CFD computations and heat transfer analysis, Yakinthos et al. (2015), and the ε-NTU method for cross flow HEX (Kays and London, 1984). The pressure losses were calculated using previously derived correlations which describe the macroscopic HEX performance and were used for the development of a HEX heat transfer and pressure losses porosity model, as presented in Yakinthos et al. (2015). More specifically, the outer and inner pressure losses are described by Eq(7) and Eq(8).

$$DP/L = [(a_0 + a_1 v)\mu U + (b_0 + b_1 v + b_2 v^2)\rho U^2]/L \quad (7)$$

$$f = DP_{static}/(\frac{l}{D}\frac{\rho U^2}{2}) \quad (8)$$

where U is the flow velocity, v is the kinematic viscosity, μ is the dynamic viscosity, ρ is the density, L is the HEX thickness, D is the tube hydraulic diameter, l is the tubes length, a_0, a_1, b_0, b_1, b_2 the viscous and inertial pressure loss coefficients and f the friction coefficient which is a function of Reynolds number Eq(9)

$$f = C_1 Re^{C_2} \quad (9)$$

With these data the complete model of the aero-engine was developed in CAPE-OPEN/COFE software and the performance of IR engine derivatives using alternative recuperation (HEX between LPT and IPT) and staged heat recovery (primary HEX between LPT and IPT and secondary HEX downstream LPT inside at the exhaust nozzle) was investigated. A parametric study was conducted since for each case all possible (yet mountable) heat exchanger core arrangement scenarios were examined (i.e.3/2/3, 4/3/4, 5/4/5, 6/5/6, 7/6/7, 8/7/8, 9/8/9 and10/9/10). Regarding the heat exchanger material selection for the conventional IR cycle, nickel-chromium alloys such as Inconel alloy 625, Inconel 617, Haynes 214 or Haynes 230 are sufficient to be used since (relatively) lower HEX inlet temperatures are met (<700 ˚C). On the other hand, for the cases where a heat exchanger is placed between IPT and LPT, in the AR and SHR IR cycles, it is necessary to use more temperature resistant materials, such as ceramic materials, advanced carbon and silicon carbide composites or superalloys due to the increased temperatures (>1,000 ºC) which are presented, as reported in McDonald and Rodgers (2005). This more stringent materials selection must be considered in relation to current and new manufacturing methods for the proper recuperator integration in gas turbines.

4. Results

In order to understand the effect of SHR and AR cycles on engine performance two parameters were assessed, the thermal efficiency $\eta_{thermal}$, which is the increase of the kinetic energy by the amount of the heat added by the fuel, and the specific fuel consumption TSFC, Eq. (10) where $\dot{m}_f$ is the consumed fuel mass flow.

$$TSFC = \frac{\dot{m}_f}{Thrust} \quad (10)$$

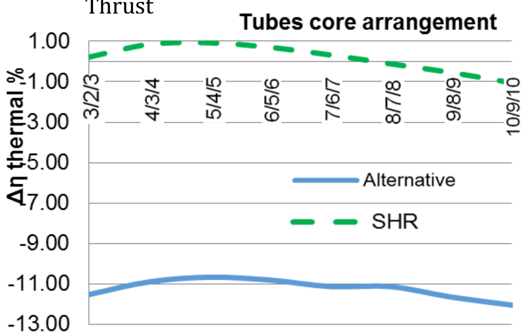

Figure 8: Comparison of AR and SHR cycles thermal efficiency in relation to IR cycle (CR)

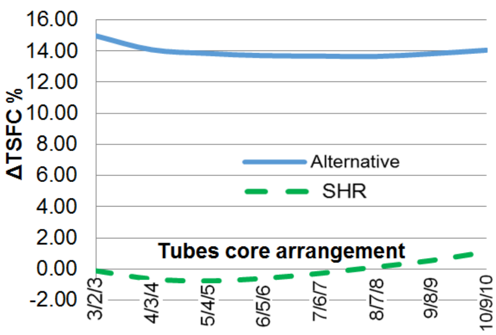

Figure 9: Comparison of AR and SHR cycles TSFC in relation to CRIR cycle (CR)

The results were compared with those of the (conventional) MTU IR engine, in which the heat exchanger is placed downstream the LPT in the exhaust nozzle. In Figures 8 and 9 the results regarding the thermal efficiency and TSFC of the AR and SHR cycles are presented in relation to CR. In all figures the relative differences are presented and not absolute values. Starting with the AR cycle, the installation of a HEX between the turbines has a negative impact on the efficiency, since for all arrangements the efficiency is much lower (~11 %) than in the reference case. Moreover, the AR cycle has increased specific fuel consumption (~14 %). This behaviour is linked to the strong effect of the limited available space between IPT and LPT which results in HEX designs of low effectiveness values and high pressure losses (due to high outer and inner flow velocities). However, the SHR cycle proved to be most promising, since for various arrangements it outperforms the reference IR cycle. Not only the thermal efficiency in most of the cases is higher but the TSFC is also lower. More specifically, in case of 5/4/5 arrangement the thermal efficiency has the highest efficiency by being 1 % higher than the reference case and the TSFC is ~1 % less. Moreover, also other configurations such as 4/3/4, 6/5/6 and 7/8/7 led to improved cycle performance. Further increase of the axial length (9/8/9) has a negative impact on the performance and TSFC. It must be mentioned that for the SHR cycle small geometrical adaptations (always compatible with the current aero engine architectures) in the space between IPT and LPT were adopted to improve the performance of the heat exchanger between IPT and LPT mainly through the pressure losses decrease.

Conclusions

1. When geometrical constraints are not strongly affecting the heat exchanger design:
   i. the performance of the CR cycle is preferable than the one of the AR cycle for low OPR values. For higher OPR, the AR performance can be better than the one of CR, especially when geometrical constraints in the mounting of the recuperator can be properly addressed. The OPR value at which the AR cycle outperforms the CR cycle, shifts to higher values as the effectiveness of the heat exchanger is reduced.
   ii. regarding the SHR cycle, the maximum efficiency values are achieved for different combinations of the primary and secondary heat exchanger effectiveness. For low OPR the SHR performance is maximized for a specific combination of ε_1 and ε_2, more closely adapted to the CR. For high OPR the performance of the SHR cycle is maximized for a specific combination of ε_1 and ε_2, more closely adapted to the AR cycle.
2. When strong geometrical constraints are presented, as in aero engines, in the design of a heat exchanger between IPT and LPT, the cycles comparative performance is significantly affected. More specifically:
   i. for the AR cycle, the limited available space between IPT and LPT results in low heat exchanger effectiveness values combined with increased pressure losses for both outer and inner flow. As an outcome, the AR configuration has a negative impact both on the thermal efficiency and on TSFC.
   ii. on the other hand, the SHR cycle performance presents promising trends when the heat exchanger design is combined with small geometrical adaptations in the space between IPT and LPT in order to improve the performance of the heat exchanger mainly through the decrease of the pressure losses. The results show that with a careful HEX design the SHR cycle presents improved performance since not only the thermal efficiency is increased but also the TSFC is reduced. As a result, a higher degree of flexibility in the aero engine geometrical constraints would result in further improvements in the cycle efficiency and performance.

Acknowledgments

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