AN INSIGHT ON REHEAT GAS TURBINE FOR COMBINED CYCLES

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Summary. *The design point performance of reheat gas turbines for combined cycle is investigated in this work. A comparison of reheat and simple cycle results is used to draw the main conclusions. Some conclusions available in the literature were confirmed; such as the fact that reheat increases the specific power and reduces the efficiency of the gas turbines. The latter is mainly a consequence of the increase of the amount of energy (heat) lost in the exhaust, represented by an increase in turbine exhaust temperature. Thus, the reheat cycle gas turbine can be considered well suitable to combined cycle. In this case the reheat combined cycle efficiency increases, compared to simple cycle, in spite of the reduction in efficiency of the gas turbine reheat cycle, with the advantage of a significant increase in specific power.*

Keywords: *Gas turbine, Combined cycle, Reheat*

1. INTRODUCTION

In the past, most power plants for electricity generation were based on steam turbines. More recently, the gas turbines became more popular since their efficiencies have increased significantly due to the borrowed developments of the aero gas turbines. Besides, gas turbines for industrial application have the largest power density, becoming very good option when space is at a premium. Other important advantages that: the period between the ordering of the equipment and its commissioning is much shorter than for other power plants; they do not need plenty of water for operation and can be loaded after a short period following start. The greatest disadvantage of industrial gas turbines is the poor efficiency at part-load, so that they are in general adopted for base load or when the period of operation is relatively short, such as peak operation. The adoption of complex cycle to improve the engine performance, like intercooling or regeneration, is usually considered. In this case they loose one of their advantage since very bulky pieces of equipment are added to the engines, increasing the initial costs, size and weight. Engine efficiency is increased mainly at part-load operation. The relatively poor gas turbine efficiency is due mainly to the waste of exhaust gases energy.

However, a better solution, at least for large power plants, is a combination of the Brayton and the Rankine cycles – the combined cycle – that is, the association of a bottoming cycle (steam turbines Rankine cycle) to the gas turbine. The gas turbine exhaust heat is recovered to in the boiler of the steam cycle. This solution is of course very expensive and more than double the size of the power plant. The efficiency is however significantly increased. While large high technology simple cycle gas turbine efficiency is in the order of 40%, the combined cycle efficiency (gas turbine plus steam turbine) is approaching 60%. Hence, the increase in initial costs is easily compensated by the fuel economy during the operational life of the engine. See Horlock (1995), for example, for more details.

In fact, this type of cycle has the highest efficiency that a thermal power station can achieve currently. Therefore it is the most important alternative presently considered to save fuel, and of course to meet the increasingly stringent legislation for environment protection.

In a combined cycle power plant the gas turbine generates around 2/3 of the total power, and in most cases it burns all the fuel delivered to the plant. Hence, it is the plant most important equipment and deserves special attention. The steam turbine cycle produces the remaining 1/3 of power and operates as a slave. Any changes in the gas turbine exhaust temperature alter the steam turbine output. The steam leg major concern is how to adequately recover the exhaust energy to generate steam in conditions to be used by steam turbine.

The gas turbine for operation in a combined cycle is somehow different from the gas turbine working alone. For the same turbine entry temperature (TET), the overall pressure ratio (OPR) for a simple cycle gas turbine operating in a combined cycle is much lower than the pressure ratio of a gas turbine working alone. Roughly speaking, this happens because with a lower compressor pressure ratio the turbine expansion ratio is reduced and the exhaust temperature increased. This brings some benefits to steam cycle since the exhaust temperature increases, which causes the reduction of the gas turbine efficiency. However, the combined cycle overall efficiency is increased. This explains why the combined cycle optimisation must consider both cycles, paying attention to the level of the gas turbine exhaust temperature.

An alternative very common to increase the gas turbine exhaust temperature is the supplementary heating, with additional fuel burnt in the heat recovery unit (steam generator). If the supplementary heating occurred in some point during the expansion in the turbine - reheat cycle gas turbine- the efficiency could be improved, although complications are expected. A second combustor between turbine stages allows the gas temperature to be increased to the level of the main combustor TET. It has been adopted with the main purpose of increasing the gas turbine specific power. This is not a new idea (Cohen et al. (1987)). Because it reduces the gas turbine efficiency it was not accepted until recently. The gas turbine efficiency reduction, in this case, occurs due to the increase in the amount of energy lost in the exhaust. However it represents by no means a problem when the combined cycle is considered, since part of this energy is used for steam generation purpose. Hence, reheat gas turbine seems to be well appropriated to be used in combined cycle.

The great advantage of using reheat is the lower TET for the same specific power, compared to a simple cycle, allowing reduction of the costs of expensive super alloys, and probably less compressor air bleed for cooling, preventing degradation of the engine performance. Additionally, lower grade fuel (less expensive) can be utilised when TET is relatively low as in this case, including the indirect firing suggested by Huang and Wang (1987). In the other hand, there is increase in the specific fuel consumption, since additional fuel is burnt at a lower pressure in the second combustor. In fact, other aspects as a longer engine life and lower costs could compensate this. Depending on the application, this inconvenience causes no concern at all, since for combined cycle or cogeneration, heat excess in the exhaust is not actually lost.

The work of Rice (1980) is probably the first important investigation addressing the combined cycle based on a reheat gas turbine. According to his estimates, reheat gas turbine experience 40% increase in specific power at the cost of 6% reduction in efficiency. He based his analysis on a two-shaft gas turbine, considering that architecture more appropriate for reheat. He suggested that it is much easier to introduce the reheat between the gas generator and power turbine. It is important to point out that in spite of reheat gas turbine having lower efficiency, the performance of the combined cycle is improved with a reheat gas turbine.

More recently, Polyzakis (1995) carried out a similar analysis to define the best gas turbine cycle for combined cycle. He also based his investigation on a two-shaft engine. He concluded that a reheat gas turbine is the better option for a combined cycle, and pointed out the benefits of reheat in the performance of a combined cycle.

Although both works preferred to concentrate their analysis on a two-shaft gas turbine, since this layout seems to be more appropriated for the reheater location, it does not seem to be a good solution in terms of the reheat gas turbine efficiency, which of course reflects in the performance of the combined cycle.

Alves et al. (2000) investigated the best position for the reheater in terms of best efficiency for a reheat single shaft gas turbine. The conclusion is that the reheater must be positioned after the early stages of the turbine, preferably after the first stage. In other words, the reheating must happen at the highest pressure possible, with insignificant efficiency degradation. When the reheater is installed after the second stages the efficiency drops but a further increase in the specific power is observed. The use of reheater at the last stages does not seem a good idea even in terms of specific power. In addition, problems related to blade integrity due to stress level (longer blades and high temperature) become a major issue at these stages. Also the volume of the reheater is bigger, since the pressure is lower. Thus, the two-shaft gas turbine with reheat does not seem to be a feasible alternative for the moment. The fact that the only reheat gas turbine available in the market is a single shaft engine may be an indication of this reasoning. Analysis of combined cycles based on this engine is appearing in the literature.

Dechamps (1998) combined this reheat gas turbine with several steam cycles, to analyse advanced combined cycles, because of its very high exhaust temperature. The investigation showed that the introduction of reheat in the steam cycle is a good alternative for fuel economy, mainly when a triple pressure combined cycle is adopted.

This work intends to carry out a parametric analysis of the reheat-combined cycle based on a single shaft gas turbine, considering the findings of Alves et al. (2000). Attention is given to the relative merits of reheat gas turbine compared to the simple cycle, and its beneficial effect on the combined cycle. The same parameters adopted by Alves et al. (2000), like compressor and turbine polytropic efficiency, are adopted, in order to bring the computational model as close as possible to the actual case. Compressor air bleed for blade cooling purpose is considered. This is an important aspect that must be considered in the analysis since one of the greatest drawbacks of reheating is the high air mass flow needed for blade cooling.

2. SIMULATION

The computer code DESTUR (Alves (1997)) carries out the analysis of the gas turbine cycle. Polytropic efficiency is used to facilitate the analysis when either the compressor or the turbine is split. In this work, the compressors and turbines polytropic efficiencies are 90% and 86%, respectively; main combustor and the reheat combustor pressure losses 5%; and the maximum reheat temperature is assumed to be the same as the TET. It is also considered that the engine is operating at ISA sea level static conditions. The effect of blade cooling is considered in the results. The variation of the gas to coolant mass flow ratio, for each cooled blade row, with TET and compressor outlet temperature is considered. The convection heat transfer assessment between the blade metal and cooling flow and its parameters like cooling efficiency (η_{cool}) and effectiveness (ε) are based on Rubini (1998). The cooling efficiency definition in this context is $\eta_{cool} = (T_2 - T_1)/(T_b - T_1)$, where T_1 and T_2 are the cooling air temperatures entering and leaving the blade passages, respectively; T_b is the metal blade temperature. Typical adopted values are 1123 K for T_b and 50% for the cooling efficiency.

The steam turbine cycle is simulated by a computer code based on the work of Dechamps (1997) that applies to a single pressure steam cycle. Table 1 gives the design parameters.

Table 1: Some parameters adopted in the steam cycle simulation.

Parameter	Value
Superheater pinch point difference	30 K
Superheater pressure	80 / 40 bar
Evaporator pinch point difference	20 K
Approach point difference	9 K
Minimum stack temperature	348 K
Pump efficiency	80 %
Pressure in the condenser	0.12 bar
Steam turbine efficiency	90 %

Two superheater pressures were adopted for comparison purposes. Below 40 bar the cycle efficiency reduction is more noticeable and over 80 bar problems related to the steam quality become more pronounced. A lower limit of steam quality is fixed at 88 %, as suggested by Dechamps (1997). The steam temperature is also limited to 833 K, the maximum conditions the steam turbine can stand currently, in most cases. The minimum Superheater pinch point difference is kept at 30 K for steam temperature below 833 K. If the exhaust gas temperature is higher than 863 K the pinch point difference is increased in order to guarantee that the steam temperature is not higher than 833 K. High pinch point difference allows reduction in the heat recovery unity size. Low stack temperature is avoided, in order to prevent corrosion caused by SO_2 in the heat recovery system. The limit value of 348 K is typical for sulphur-free natural gas, Dechamps (1998).

No attempt to optimise the costs of the heat recovery system is made, since the objective of this work is to compare the performance of combined cycle based on reheat and simple cycle gas turbines. Nevertheless, appropriate steam cycle constraints, related to economic aspects, as pinch point differences, shown previously in Table 1, are used.

3. RESULT S

3.1 Simple Cycle

Simple cycle performance curves are shown in Fig. 1, for OPR varying from 10 to 60 and TET from 1300 K to 1800 K. They are presented for the sake of comparison with the results of the reheat cycles analyses. It can be noted in these curves the pressure ratios for maximum efficiency and specific power are indicated on full and dotted lines, respectively. For a given value of TET, the OPR must be between these two extremes, its position depending on the engine application. A commercial engine, identified as GT1, is placed on this chart to give an indication of the accuracy of the model developed. The efficiency and specific power adopted are the ones published by the manufacturer (Rolls-Royce, 1996).

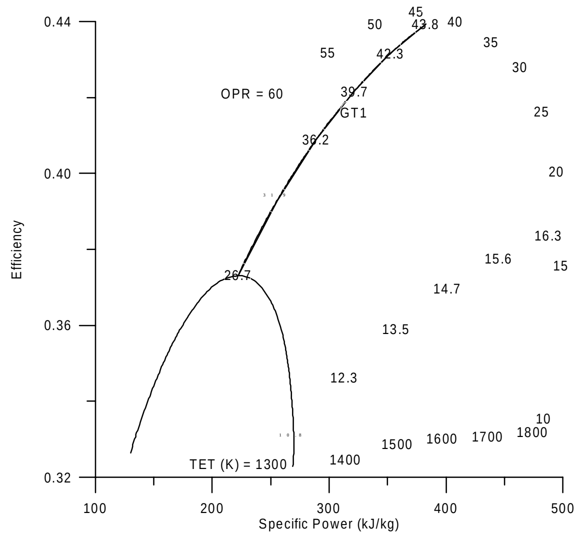

Figure 1: Simple cycle performance curves

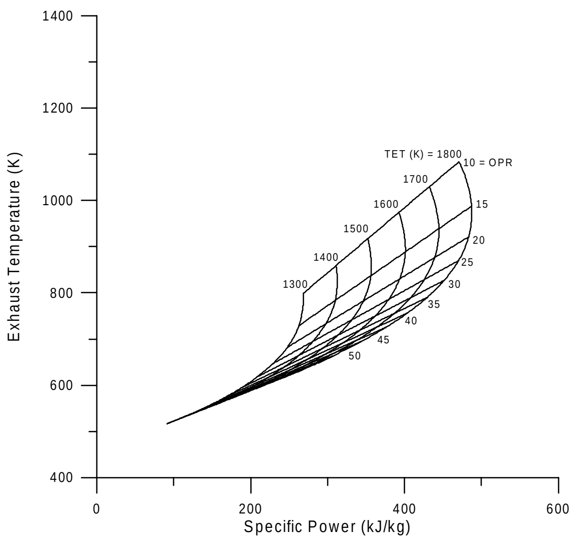

Figure 2: Exhaust gas temperature versus specific power for simple cycle.

Besides the parameters in Fig. 1, the exhaust gas temperature is an important parameter for combined cycle and cogeneration. Figures 2 and 3 show the variation of exhaust gas temperature with engine efficiency and specific power, respectively. Combining the information of Figs. 1 – 3 a useful analysis be carried out in order to define the best engine for combined cycle or cogeneration. Note that the gas turbine exhaust temperature increases as OPR decreases, for the same value of TET. The exhaust temperature also increases with TET for the same OPR. Thus, the exhaust temperature is strongly a function of both OPR and TET. For a given value of OPR both efficiency and specific power increase with TET. For efficiency, this tendency is more pronounced for higher values of OPR. The reverse is true for specific power. The results of these charts can be used for the gas turbine cycle selection.

3.2 Reheat Cycle

Figures. 4 – 6 are the reheat counterpart for the Figs. 1 – 3, respectively. This simulation considers all losses effects and also the blade cooling. The reheat gas turbines performance curves are presented in Fig. 4. They were obtained considering the practical expansion ratio of 2.0 for the high pressure turbine (before the reheater), as suggested by Alves et al. (2000), instead of using the values of expansion ratio that give the best efficiency with reheat, since these values are too small to be considered in practice for a turbine stage. Thus, the results of Fig. 4 were obtained considering that the reheat takes place at a pressure level that gives almost the best efficiency. The values on full line are OPR corresponding to maximum efficiency for given TET, which are significantly greater than the OPR for maximum specific power. This fact disagrees with the observation made by Rice (1980) that the OPRs for optimum efficiency and specific power are the same. A possible explanation is that the gas generator turbine expansion ratio he adopted is certainly higher. According to the discussion presented by Alves et al. (2000), the reheat gas turbine cycle efficiency based on gas generators is lower at a higher specific power. This may bring the two optimum points closer. Rice (1987) suggested also, in his investigation on reheat for two-shaft engines, that the best reheat cycles are those whose OPR is equal to the more efficient simple cycle OPR, for the same TET. This does not seem to apply to this case.

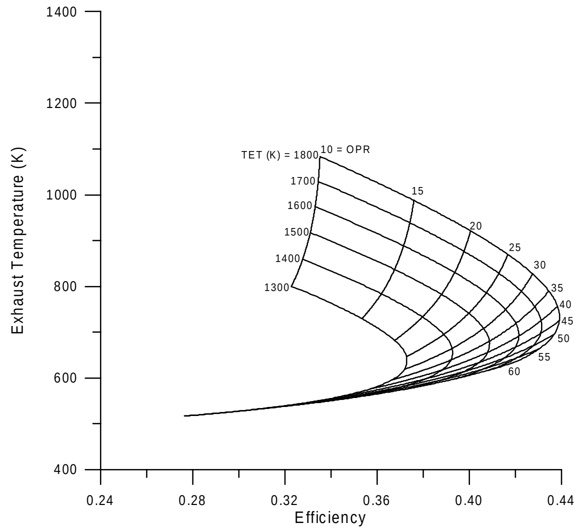

Figure 3: Exhaust gas temperature versus efficiency for simple cycle.

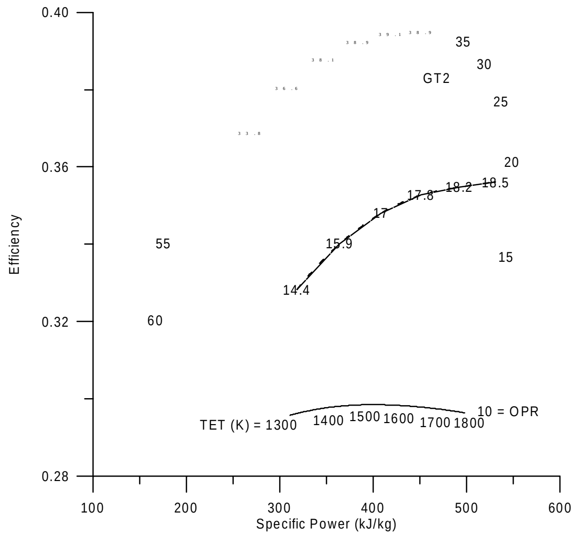

Figure 4: Performance curves for a reheat cycle obtained with high-pressure turbine expansion ratio equal to 2.

In fact, a direct comparison of the results presented in this work with the results obtained by Rice (1980, 1987) for reheat is not appropriated, because of the position of the reheat in the expansion and the use of a two-shaft gas turbine. Some of Rice's (1980) conclusions, such as that mentioned previously and that OPR influences little on the power turbine exhaust gas temperature, for a fixed value of TET, were not confirmed in this work. The explanation for the latter is obvious since the power turbine does not drive the compressor and the TET for the reheater is constant.

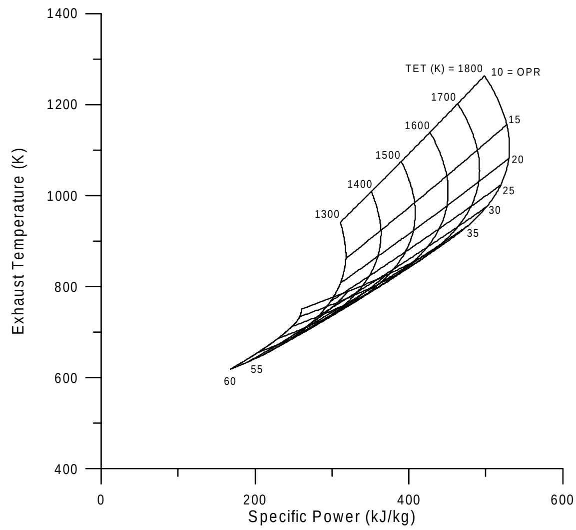

Figure 5: Exhaust gas temperature versus specific power for reheat cycle.

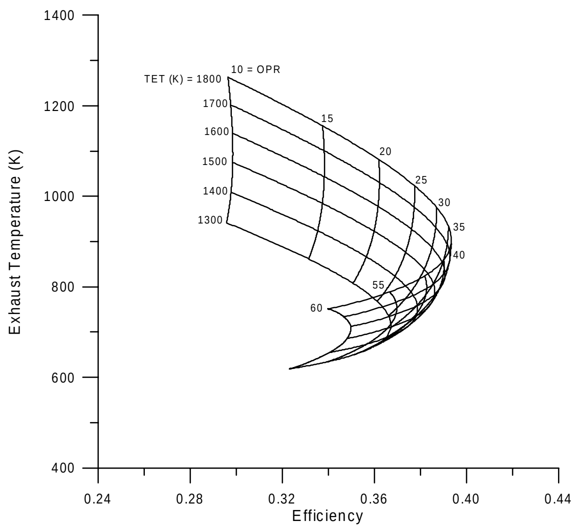

Figure 6: Exhaust gas temperature versus efficiency for reheat cycle.

An overview of Fig. 4 shows that the reheat increases substantially the specific power, at a penalty in efficiency, compared to the simple cycle. This is so because the heat addition to the working fluid occurs at lower pressure. On a T-S diagram it is possible to see that the proportional increase in heat addition becomes greater than the proportional increase in the specific power output, as the reheat pressure is reduced (the expansion ratio of the high-pressure turbine is increasing). Hence, the efficiency drops. An ABB reheat gas turbine was positioned in this chart (GT2) using data from Joos et al. (1998).

A comparison of Figs. 1 and 4 shows that the OPR values for both maximum specific power and efficiency are greater for reheat cycles compared to simple cycle. The OPR for maximum specific power is also closer to the one corresponding to maximum efficiency.

The shapes of Figs. 5 and 6 are similar to Figs. 2 and 3, for simple cycle, respectively. Thus, the same conclusions as before apply. It is crystal clear, however, that the exhaust temperature increases significantly when reheat is considered. This characteristic of reheat cycle is useful for combined cycle.

3.3 Combined Cycle

The combined cycle total efficiency and specific power calculation, based on the gas turbine mass flow, for some values of gas turbine cycle parameters, is presented in Figs. 7 – 10. They show the variation of the combined cycle total efficiency with the specific power, for constant OPR and TET values. The results for simple and reheat gas turbine cycle analysis presented previously were used. These results were input to the computer code written for the combined cycle steam leg, and refer to a single pressure steam cycle. Note that some points on the charts were deleted, since they refer to steam cycles in which the steam quality falls below the limiting value of 88 %, which are not of practical interest, or when the stack temperature is below the limiting value. The former affects the left side of these charts, high OPR, and the latter the right side that corresponds to low value of OPR.

It is important to point out the OPR effect on the combined cycle performance. For fixed TET, it is possible to see that the specific power increases continuously as OPR is reduced. The observed peak values on Figs. 1 and 4 do not seem to appear in this case, at least for the practical OPR range. On the other hand, there is a value of OPR corresponding to the maximum efficiency like what happens in the gas turbine charts (Figs. 1 and 4). As it was mentioned in the introduction, it is possible to see that the OPR to attain the maximum efficiency is less than the value for the corresponding gas turbine cycle best efficiency. Considering for instance the results of Figs. 8 and 10, for TET = 1800 K, the OPR for the combined cycle maximum efficiency is around 25.0 while for the simple and reheat cycle gas turbines are 43.8 and 38.9, respectively, as shown on Figs. 1 and 4. The OPR values for maximum specific power for gas turbine cycles, on the other hand, are closer to the OPR corresponding to the combined cycle maximum efficiency. Hence, it seems more adequate to optimise the gas turbine cycle to maximum specific power for combined cycle application.

TET seems to affect equally both the combined cycle efficiency and specific power, which increase significantly with TET. This tendency is not observed in the results presented in Figs. 1 and 4, for low OPRs. For the simple cycle, TET increase affects mostly the specific power with little effect on the efficiency for low OPRs. The effect on efficiency becomes more significant when OPR is increased. For the reheat, TET effect on specific power is more pronounced but its effect on efficiency is not noticeable. A significant influence is observed only for gas turbines with very high OPR.

These observations show that the best gas turbine cycle for an engine working alone does not necessarily produce the best conditions for combined cycle application.

Figures 7 and 8 refer to simple cycle gas turbine in a combined cycle working with superheater pressure of 80 and 40 bar, respectively. Comparing Figs. 7 and 8 with Fig. 1, it is seen promptly that the specific power produced by combined cycle is significantly higher than the specific power of simple cycle gas turbine. The combined cycle efficiency is also higher. For TET = 1800 K, for example, the simple gas turbine maximum efficiency is around 44% while for combined cycle is about 58 %, for a steam cycle pressure of 80 bar. The specific power, on the other hand, increases from about 350 kJ/kg to around 700 kJ/kg, that is, it is the double. The effect of the steam cycle superheater pressure on the results can be observed comparing Figs. 7 and 8. The main conclusion is that the cycle efficiency suffers a noticeable increase with the superheater pressure, while the specific power experiences no significant change. The superheater pressure also affects strongly the steam quality.

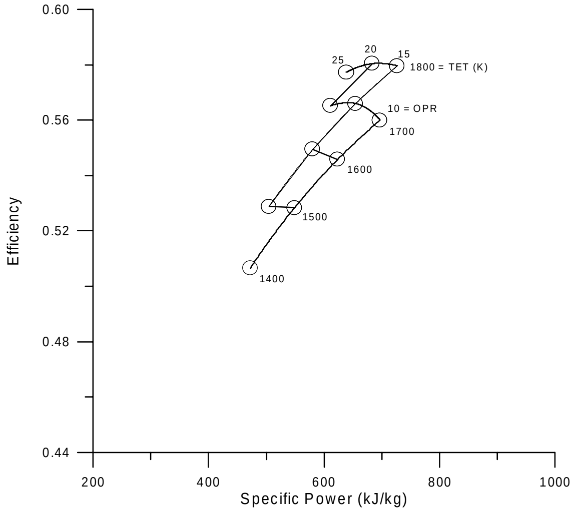

Figure 7: Performance of a combined cycle based on simple cycle gas turbine and superheater - pressure = 80 bar.

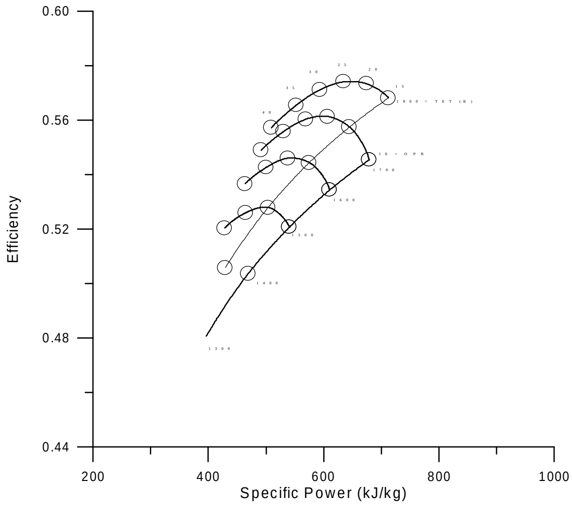

Figure 8: Performance of a combined cycle based on simple cycle gas turbine and superheater - pressure = 40 bar.

The results for reheat cycle are presented in Figs. 9 and 10, for superheater pressure of 80 and 40 bars, respectively. It is possible to see that the combined cycle performance is improved with the inclusion of reheat cycle, compared with simple cycle gas turbine. For TET = 1800 K and superheater pressure = 80 bar, for example, the maximum cycle efficiency is just under 59 % while the specific power is around 800 kJ/kg. These figures show an improvement compared to the results presented previously for combined cycle based on simple cycle gas turbine. It is worth mentioning that the efficiency of combined cycle with reheat gas turbine is improved, in spite of the fact that, for the reheat gas turbine, the efficiency suffers a reduction compared to simple cycle gas turbine.

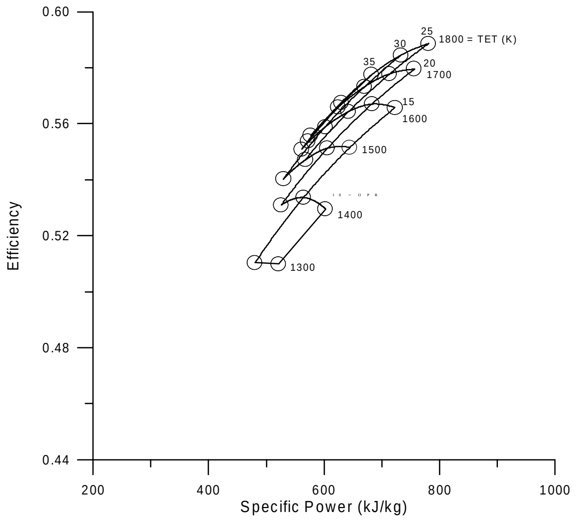

Figure 9 - Combined cycle based on reheat cycle (superheater - pressure = 80 bar).

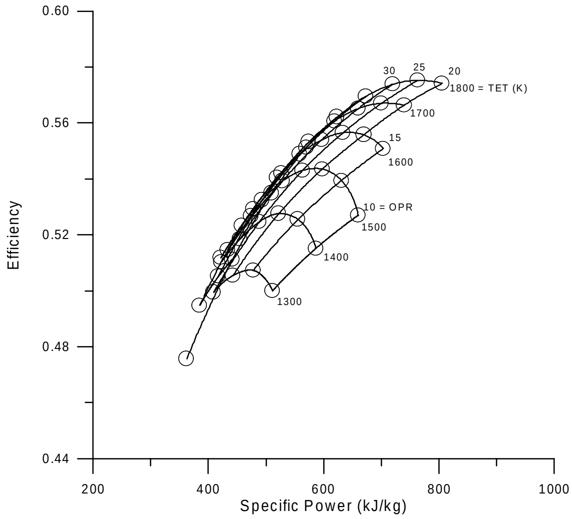

Figure 10 - Combined cycle based on reheat cycle (superheater - pressure = 40 bar).

It is important to point out the fact that the TET for the simple cycle to produce the same specific power as the reheat cycle, both with the same OPR, is significantly higher, as expected. Comparison of Figs. 8 and 10 is used to give estimate of the increase in TET for simple cycle. From Fig. 8, for simple cycle, the specific power obtained with OPR = 15 and TET = 1800 K is basically the same as that of a reheat cycle with the same OPR and TET = 1600 K. Hence, the reheat can be used as an alternative to reduce the firing temperature of a gas turbine, reflecting in the technology involved in its design and its life.

A quick analysis of Figs. 2 and 3, for simple cycle, and Figs. 5 and 6, for reheat cycle, shows that the regions of Figs. 7 – 10, corresponding to low OPR and high TET, have a limited importance for combined cycle, since the resulting stack temperature is very low, below the limit of 348 K. Some of those points were not included in Figs. 7 – 10. One possible way to overcome this problem is to increase the evaporator pinch point difference, through the reduction the heat recovery heat exchanger surface. This is a beneficial aspect, in economic terms, of having a high pinch point difference. However, the heat recovery unity efficiency decreases in this situation, reducing cycle efficiency. On the other hand, the region corresponding to the gas turbine values of high OPR and low TET is also unimportant since the total efficiency is very low. This is mainly because the gas turbine does not release much heat with the exhaust gases so that the steam cycle efficiency is very low. Most parts of that region were consequently not shown on Figs. 7 – 10 because of reduced efficiency, and on top of that because the steam quality falls below the limit fixed. Therefore, a rather narrow region is left for the designer to make the appropriated choice for the gas turbine cycle parameters.

At the light of these results, it is important to point out that although high OPR leads to more efficient gas turbine, it not meant that the combined cycle global efficiency is directly improved. In fact, the best gas turbine to be used in a combined cycle is the one that works in the medium-to-low OPR range, compared to a gas turbine optimised as prime mover.

The advantage of a gas turbine with low OPR is that an unsophisticated compressor, based on a proved technology, and therefore cheaper, can be used. Moreover, as usually the size is not relevant for this kind of application, a more efficient unity can be used in the gas turbine design. On the other hand, it is expected the top technology to be adopted in the hot section because, for combined cycle, high TET means more efficiency and power, as it is seen from the figures, due to the increase in gas turbine output and exhaust gas temperature.

4. FINAL REMARKS AND CONCLUSION

There are two additional aspects of reheat that must be better explored. One is related to the part-load power control and the other represents an interesting alternative for the reheat cycle performance improvement, via reduction of compressor bleed air for cooling.

The combined cycle power control is achieved by varying the fuel flow to the gas turbine, that is, varying TET. The gas turbines can be equipped with other devices to improve part-load performance (off-design operation). Turbine variable geometry is one of the most important. The reheat cycle has the additional flexibility of transferring part of the engine control for part-load operation to the second combustor. This would be a better option to variable geometry. This subject is currently under investigation by the authors. Rice (1980) also mentioned the advantage of the second combustor as an alternative for NOx emission reduction.

The need to increase the blade cooling airflow due the introduction of a second combustor (reheating) is one of the main inconveniences of the reheat cycle. It is responsible for part of the engine performance degradation. However, this can be attenuated if a small amount of the combined cycle steam is used to cool the gas turbine hot parts. The benefits were not investigated in this work but the literature addresses this subject (Johansson, 1993).

In conclusion, the reheat combined cycle performance was analysed in this work, considering a single shaft gas turbine. A parametric study of simple and reheat cycle gas turbine was carried out. The results confirm that reheat causes a significant increase in the gas turbine specific power at expense of the gas turbine efficiency degradation. The important aspect is that the gas turbine exhaust temperature increases significantly with reheat, thus reducing the gas turbine efficiency but making it more adequate for use in combined cycle.

The combined cycle analysis confirmed the expectation that the combined cycle performance is much better than the performance of the corresponding gas turbine. Most importantly, it was shown that reheat combined cycle has as good performance as the performance of an engine based on a simple cycle gas turbine.

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REFERENCES

Alves, M. A. C.; Carneiro, H. F. F. M.; Barbosa, J. R.; Travieso, L. E.; Pilidis, P.; and Ramsden, K. W., An Insight on Intercooling and Reheat Gas Turbine Cycles, Submitted for publication in the IMechE, 2000.

Alves, M. A. C., DESTUR- Scheme for Aero/Industrial Gas Turbine Engine Design/Off-design Point Performance Calculation, CTA-IAE-ASAP, Internal Report (unpublished), 1997 (in Portuguese).

Cohen, H.; Rogers, G. F. C. and Saravanamuttoo, H. I. N., Gas Turbine Theory, Longman - Scientific & Technical, 4th edition, Longman, 1997.

Dechamps, P. J., Combined Cycle, Cranfield University, Lecture Notes, SME / PEP, 1997.

Dechamps, P. J., Advanced Combined Cycle Alternatives with the Latest Gas Turbines, Journal of Eng. for Gas. Turb. and Power, ASME, Vol. 120, n° 2, 1998, pp. 350-357.

Joos, F.; Brunner, P.; Stalder, M. and Tschirren, S., Field Experience with the sequential Combustion System of the GT24/GT26 Gas Turbine Family, ABB Review Article, Issue 5/98, M477, 1998, pp. 12-20.

Johansson, O., Influence of Blade Cooling Technology on the efficiency of a Combined Cycle Power Plant, Cranfield University, MSc. Thesis, SME, 1993.

Horlock, J. H., Combined Power Plants – Past, Present, and Future, Journal of Eng. for Gas. Turb. and Power, ASME, Vol. 117, n° 4, 1995, pp. 608-616.

Huang, F. F. and Wang, L., Thermodynamic Study of an Indirect Fired Air Turbine Cogeneration System with Reheat, ASME, J. Eng. for Gas Turb. and Power, Vol. 109, 1987, pp. 17-21.

Polyzakis, A., Industrial Gas Turbine for Combined Cycle Plant, Cranfield University, MSc. Thesis, 1995.

Rice, I. G., The Combined Reheat Gas Turbine/Steam Turbine Cycle: Part I – A Critical Analysis of the Combined Reheat Gas Turbine/Steam Turbine Cycle, ASME, J. of Eng. for Power, Vol. 102, 1980, pp. 35-41.

Rice, I. G., Thermodynamic Evaluation of Gas Turbine Cogeneration Cycles: Part II – Complex Cycle Analysis, ASME, J. Eng. for Gas Turb. and Power, Vol. 109, 1987, pp. 8-15.

Rolls-Royce (Industrial &Marine Gas Turbine Limited), Industrial Trent Power Generation, Brochure IMG/9605/001/1-1, 1996.

Rubini, P., Blade Cooling: Single Pass Gas Cooling, Cranfield University, Lecture Notes, SME/2187(e)/AM/MEI, 1998.