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OFF-DESIGN PERFORMANCE COMPARISON BETWEEN SINGLE AND TWO-SHAFT ENGINES PART 1 – FIXED GEOMETRY

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ABSTRACT

This paper describes an investigation into the off-design performance comparison of single and two-shaft gas turbine engines. A question that has been asked for a long time which gas turbine delivers a better thermal efficiency at part load. The authors, notwithstanding their intensive searches, were unable to find a comprehensive answer to this question. A detailed investigation was carried out using a state of the art performance evaluation method and the answer was found to be: It depends!

In this work, the performance of two engine configurations is assessed. In the first one, the single-shaft gas turbine operates at constant shaft rotational speed. Thus, the shape of the compressor map rotational speed line will have an important influence on the performance of the engine. To explore the implications of the shape of the speed line, two single-shaft cases are examined. The first case is when the speed line is curved and as the compressor pressure ratio falls, the non-dimensional mass flow increases. The second case is when the speed line is vertical and as the compressor pressure ratio falls, the non-dimensional mass flow remains constant.

In the second configuration, the two-shaft engine, the two-shafts can be controlled to operate at different rotational speeds and also varying relationships between the rotational speeds. The part-load operation is characterized by a reduction in the gas generator rotational speed.

The tool, which was used in this study, is a 0-D whole engine simulation tool, named Turbomatch. It was developed at Cranfield and it is based on mass and energy balance, carried out through an iterative method, which is based on component maps. These generic, experimentally derived maps are scaled to match the design point of a particular engine before an off-design calculation is performed. The code has been validated against experimental data elsewhere, it has been used extensively for academic purposes and the research activities that have taken place at Cranfield University.

For an ideal cycle, the single-shaft engine was found to be a clear winner in terms of part-load thermal efficiency. However, this picture changed when realistic component maps were utilized. The basic cycle and the shape of component maps had a profound influence on the outcome.

The authors explored the influence of speed line shapes, levels of component efficiencies and the variation of these component efficiencies within the operating range. This paper describes how each one of these factors, individually, influences the outcome.

Keywords: gas turbine performance, single-shaft, two-shaft, thermal efficiency, specific work.

NOMENCLATURE

CDP	Compressor Delivery Pressure
OPR	Overall Pressure Ratio
SSC	Single-shaft – Curved speed line
SSV	Single-shaft – Vertical speed line
SW	Specific Work
2S	Two-shaft engine
TET	Turbine Entry Temperature
η_p	Polytropic Efficiency
η_{th}, ETA_{th}	Thermal Efficiency
DP	Design Point

INTRODUCTION

Gas turbines are often required to operate for significant periods at part load. This is happening more and more when gas turbines are used in electricity generation and have to 'follow load' given the variable and intermittent electricity arising from the extensive use of renewable energy systems. Very frequently legislation requires that renewable energy systems have "first access" to the grid. In these circumstances other power systems, including gas turbines have to operate in a load following mode. This requirement of part load operation is not restricted to land based gas turbines. In naval applications large power variations are, normally, needed and this has resulted in complex designs of Combined Gas and/or Gas or Combined Gas and/or Diesel plant. In some aircraft applications, such as long range maritime patrol, large power variations are also needed between different phases of a flight. In such applications, frequently the "loiter" phase of the mission is carried out with one or more engines shut down; this will enable the gas turbines that remain in operation to work at high power and a lower specific fuel consumption. When the "loiter" phase of the mission is finished, the gas turbines that have been shut down are restarted for the cruise.

The anatomy of a gas turbine and its control philosophy will influence its performance at off-design conditions. Two frequently used simple cycle engine anatomies are assessed here: the single-shaft and the two-shaft gas turbine. There is a power range of between 10 to 100 MW and there is a choice between these two engine anatomies. The largest two-shaft gas turbines are currently capable of delivering 30 MW while three shaft engines can deliver up to 100MW. Above 100 MW, currently, only single-shaft gas turbines are available.

The single-shaft gas turbine can operate in a variety of modes. In the present investigation operation at constant shaft rotational speed is examined and the shape of the rotational speed line in the compressor map will have an important influence on the performance of the engine. To explore the implications of the shape of the speed line, two single-shaft cases are examined. The first case is SSC, here the speed line is curved and as compressor OPR falls, the non-dimensional or corrected mass flow increases. The second case is SSV, here the speed line is vertical and as compressor OPR falls, the non-dimensional mass flow remains constant. The resulting operating lines are shown in fig. 1.

The shape of the design speed line of the compressor depends on the velocity of the flow through the compressor, with a vertical speed line corresponding to choked flow at the exit of the compressor. This is typically present in high performance compressors and is the result of a relatively high overall compressor pressure ratio per stage. As the maximum OPR that can be achieved in a stage is limited, stacking more stages is required to increase the OPR of the compressor. The addition of a stage turns the characteristic towards the vertical direction, increasing the pressure ratio for the same mass flow rate. The increase of the stagger angles of the blades above 70° is also typically employed in order to achieve a high pressure ratio for the stage.

On the other hand a design speed line where the mass flow rate increases as the OPR falls corresponds to relatively slow flow velocities through the compressor. This arises from a larger number of stages, or a lower loading of the individual stages (i.e. a higher de Haller number) that can be chosen to reduce aerodynamic loss or improve the surge margin. Due to the lower velocities the flow areas will be larger.

In a two-shaft gas turbine, one shaft is used to drive the compressor and the second shaft connects the free turbine with the external load. The two-shafts can be controlled to operate at different rotational speeds and also varying relationships between the rotational speeds. Part load operation is characterised by a reduction in gas generator rotational speed. The resulting operating line is also shown in Figure 1 and labelled 2S.

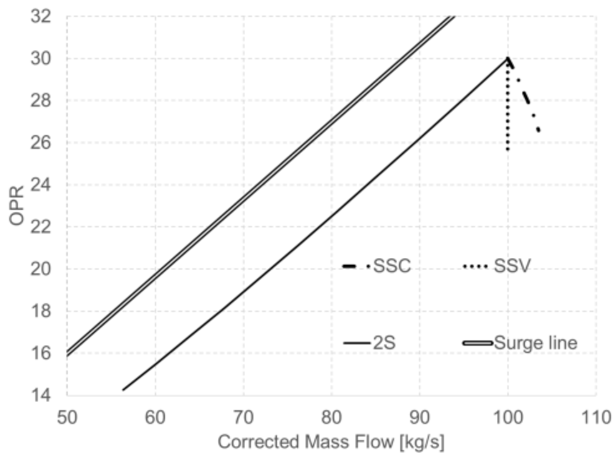

FIGURE 1: Operating lines of the three cases examined

A question that has been frequently asked is how the part-load thermal efficiency varies as the operating point changes. The authors were intrigued that they could not find a detailed comparison of the engine anatomies outlined here in the public domain, apart from the preliminary deductions carried out by one of the authors (p 493, Saravanamuttoo et al, 2017). Therefore, in the study described here, a reference list has not been produced, instead a bibliography that may be helpful to the reader is provided.

Given the absence of such a study, the authors decided to launch their own. To carry out the investigation using Turbomatch, the Cranfield gas turbine performance code was used. It was configured to model the three cases examined here. A range of compressor maps was used to represent the speed lines indicated above. The outcome of the investigation is dependent on the shape of the map so this range incorporated some "artificial" and some realistic maps. Turbine Entry Temperature (TET) was used to control power output and the power range examined was such that the turbines were always choked. All evaluations were carried out at standard conditions of engine inlet temperature and pressure. Real gas properties have been used throughout the investigation.

1 IDEAL CYCLE

The first examination was that of the ideal (or quasi-ideal given the combustor pressure loss) simple cycle. In the present investigation the cycle characteristics are as shown in Table 1.

Table 1: Cycle Parameters

Parameter	Value
Turbine Entry Temperature (TET)	1700 K
Compressor Design OPR	30
Blade cooling mass flow fraction	Constant 20% compressor flow
Combustor pressure loss	Constant 5% of CDP
Combustor efficiency	100%
Compressor inlet flow	100 kg/s
Compressor & Turbine Polytropic efficiencies	100%

A few key results of this analysis are shown in Figures 2a, b and c. The results have been plotted on the basis of fractional power output (part load power/design power). This fractional power output will be used throughout the investigation to make the comparison on the basis of useful work. Fig 2a shows that the part-load thermal efficiency of the SSC is the highest, SSV is slightly lower and 2S is significantly lower. This is a result of the OPR of each cycle (Fig 2b). This is the outcome of the combination of TET, mass flow and OPR needed to produce the constant non-dimensional mass flow at the inlet to the turbine section because the turbine is choked. In the case of the SSC the mass flow increases as the TET reduces with reduced power output (Fig. 1). This requires a lower pressure reduction to achieve the required value of turbine inlet non-dimensional flow i.e. the engine will operate at higher OPR. The SSV is experiencing a constant mass flow (Fig. 1), so the turbine inlet non-dimensional flow can be achieved with a slightly higher pressure reduction than the SSC i.e. the engine will operate at lower OPR (Fig. 2b). The 2S cycle experiences a reduction in mass flow as the gas generator shaft speed falls. Thus the resulting OPR is much lower. Figure 2c shows the resulting TETs. Here the opposite effect is observed because to deliver the useful power, the cycles with a lower mass flow will need the higher TETs. In summary as power output falls, single-shaft engines will experience smaller reductions in OPR, while two-shaft engines will experience smaller reductions in TET.

This pattern of change of TET and OPR, arising from the need to achieve the constant value of non-dimensional flow of the choked turbine, is the same across the many cases examined here, so it is shown only once. In the case of the ideal cycle, OPR determines the thermal efficiency of the whole cycle. Consequently, the thermal efficiency pattern of fig. 2a follows the OPR pattern of fig 2b.

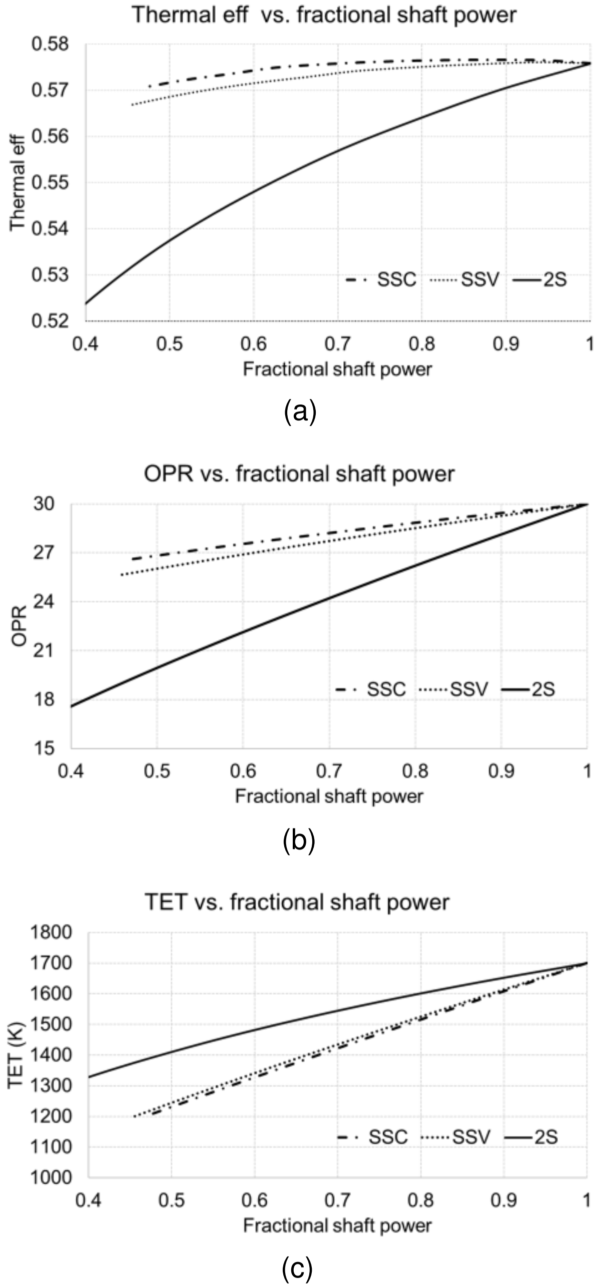

FIGURE 2: (a) Thermal efficiency (b) OPR and (c) TET (K) vs. fractional power output = Power/(Design Power)

The results are shown for fractional power because this shows results at the same power output for the different engine anatomies. It also enables, in later sections of the paper, the comparison across different polytropic efficiencies, where the design power output changes. The pattern in fig. 2, shows that for the same fractional power output, single-shaft engines exhibit higher OPR, whereas two-shaft engines exhibit higher TETs. This pattern is a key factor underpinning this investigation. It repeats itself across all the evaluations carried out below. It will be referred to below but not illustrated again.

2 CYCLES WITH CONSTANT TURBOMACHINERY EFFICIENCIES

In real cycles the TET will influence the thermal efficiency. This is because the TET exerts a major influence in the specific power of a cycle and the sensitivity of thermal efficiency to component losses is lower for cycles with high specific power. The authors felt it pertinent then to examine this effect. This was done by selecting different baselines, some for high thermal efficiency and some for high specific power. A Design Point performance analysis was carried out using the parameters of Table 1 as a baseline. Three cases with turbomachinery polytropic efficiencies of 0.85, 0.90 and 0.95 were arbitrarily chosen to assess a suitably wide range appropriate to large gas turbines; hopefully bounding state of the art component technologies.

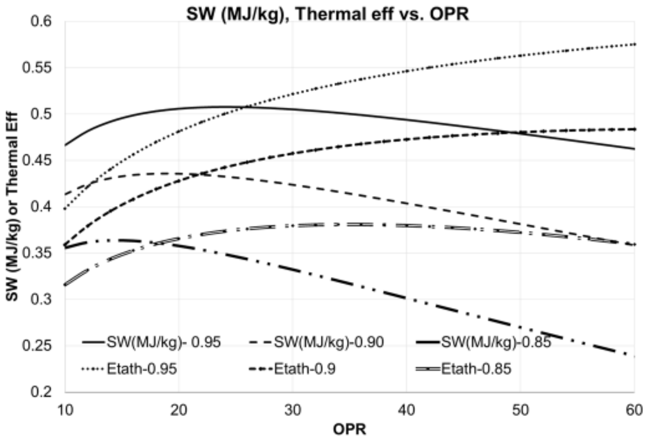

FIGURE 3: Design Point analysis for a range of polytropic efficiencies (0.85, 0.90 and 0.95 referred to in the figure). This plot shows Specific Work (in MJ/kg) and thermal efficiency vs. OPR. This figure is the basis for the selection of the cycles shown in Table 2.

Figure 3 shows the result of this analysis, where specific work and thermal efficiency are plotted against Design OPR. Given that these are design point evaluations, the results are the same for the three engine anatomies under investigation. Figure 3 shows the usual conflict where a choice has to be made between specific work (requiring moderate design OPRs) and thermal efficiency (requiring high design OPRs). From these the appropriate baseline cycles were selected, as shown in Table 2. The OPRs selected are not always the ones delivering the highest value of thermal efficiency, but are considered to be comfortably achievable with the present state of the art. Compressors with a design OPR of 35 are used in some single-shaft engines, while a design OPR of 25 is used in the high pressure shaft of some two-shaft engines and over 50 in two and three shaft aero engines. In both cases variable geometry is employed to ensure adequate stability through the power range.

To enable the separation of cycle effects from component effects, this part of the investigation was carried out assuming that the polytropic efficiency of the components was constant throughout the operating range. The assumption is made here that for the 2S engines, no variable geometry is needed to ensure the compressor does not surge. This impractical choice is justified here because the objective is to evaluate cycles, not to design a gas turbine engine. When the calculations were carried out, as expected, the pattern of fig 1 repeated itself. Single-shaft (SS) engines exhibited higher OPRs and two-shaft engines (2S) exhibited higher TETs for the same fractional power output.

Table 2: Baseline Cycles Design OPR

η_p	OPR for High SW	OPR for High n_{th}
0.85	14	40
0.90	18	40
0.95	20	40

2.1 HIGH SPECIFIC WORK CYCLES

High specific work cycles are not just a thermodynamic curiosity. When gas turbines are designed for integrated operation with steam turbines in a combined cycle arrangement, the gas turbine is designed with a moderate OPR so as to exhibit a high exhaust gas temperature to enable an efficient steam cycle. These cycles are designed to be very near the maximum specific power pressure ratio. Many of these gas turbines have been installed independently, without steam cycles.

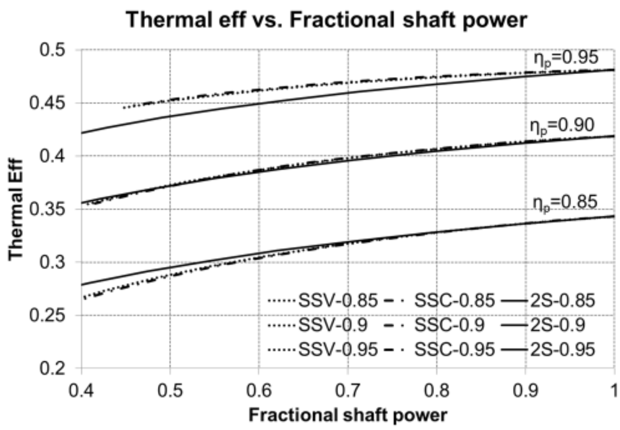

FIGURE 4: High Specific Work Cycles: Thermal Efficiency vs Fractional Power output for different values of η_p.

Cycles with high specific power have a lower sensitivity to component losses. Thus they will derive more thermal efficiency benefit from higher OPRs. So in cycles with high specific power, the single-shaft engine will benefit from the smaller reductions in OPR. Only when the component efficiencies are low the higher relative OPR will be a disadvantage. This is visible in fig. 4. It shows that when the polytropic efficiency of compressors and turbines is 0.85, the cycle is very sensitive to reductions in

TET (i.e. part load conditions), so the two-shaft engine exhibits a higher thermal efficiency at part load. When polytropic efficiencies are 0.9 the part load efficiency is very similar. When polytropic efficiency is very high (η_p =0.95), then the one shaft engines have the advantage.

2.2 HIGH THERMAL EFFICIENCY CYCLES

Cycles with high thermal efficiency have a lower specific power; this is clearly visible in fig. 3. Thus they have a higher sensitivity to component losses. Consequently they will derive less of a thermal efficiency benefit from higher OPRs. So in cycles with lower specific power, the single-shaft engine will not benefit from the smaller reductions in OPR. This is visible in fig. 5. It shows that when the polytropic efficiency of compressors and turbines is 0.85, the cycle is very sensitive to reductions in TET, so the two-shaft engine (2S) exhibits a significantly higher thermal efficiency at part load.

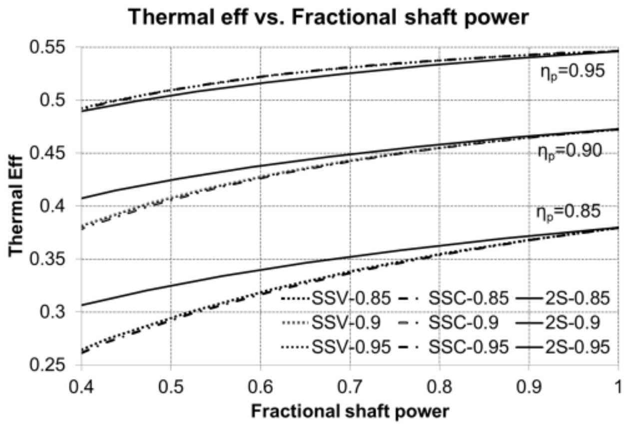

FIGURE 5: High Thermal Efficiency Cycles: Thermal Efficiency vs Fractional Power output for different values of η_{p}.

When $\eta_{p.}$ are 0.9 this difference becomes smaller. When polytropic efficiency is very high (η_p=0.95), the part load efficiency of all the cycles is very similar. In this case (fig. 5, $\eta_{p.}$=0.95) it is of interest to observe that for the higher power settings, the single-shaft engines exhibit a small thermal efficiency advantage that gradually disappears at the lower power setting. Again, this is due to the faster reduction in specific work of the one shaft engines, experiencing high mass flows. Initially, the higher OPR gives one shaft engines an advantage. As specific power falls, then the two-shaft engine's higher TET and consequent higher specific work give this layout the advantage in thermal efficiency. This causes the higher curvature of the single-shaft engine efficiency lines.

3 CYCLES WITH VARYING TURBOMACHINERY EFFICIENCIES

The exploration of cycles with compressors with constant polytropic efficiencies is very useful but it is artificial. The next step is the evaluation of the shapes of the compressor maps on the performance of the gas turbines. Four maps were used, the combination of vertical and curved speed lines with two polytropic efficiency patterns. In the first pattern the polytropic efficiency exhibits a large change with mass flow but a relatively modest change with OPR; the second pattern exhibits the opposite variation. Two of these maps are shown in fig. 6: one with a large polytropic efficiency change along the OPR axis and curved speed lines (fig. 6a) and the other with a large polytropic efficiency change along the Mass Flow axis and vertical speed lines (fig. 6b). Figure 6 shows the unscaled compressor maps. Turbomatch scaled the maps as necessary during the calculations.

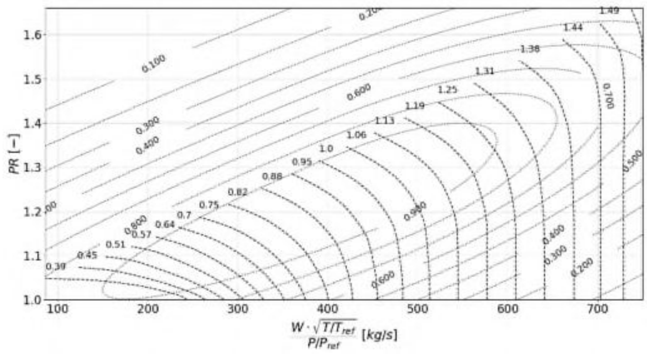

FIGURE 6A: Compressor map - large vertical variation of η_p

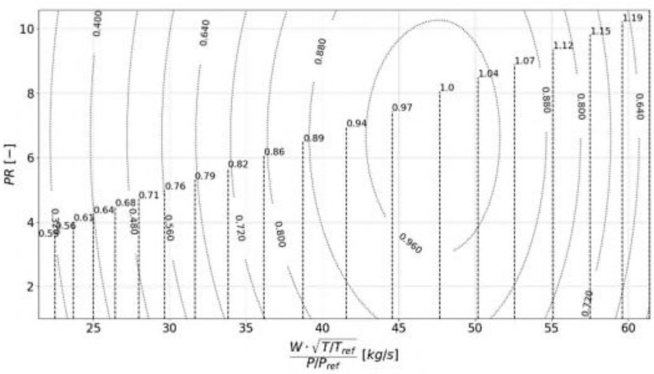

FIGURE 6B: Compressor map - large horizontal variation of η_p.

The isentropic efficiency of a compressor is driven by whether the aim of the design is to provide a robust design for off-design operation or to maximize the design efficiency. The former is more suitable for aircraft engines that typically operate at off-design conditions and are required to have a high efficiency in a multitude of operating points, resulting in an efficiency curve that is relatively flat for different mass flow rates. The latter is suitable for power generation engines if they were to constantly operate at their design point, where peak power is preferred over versatility. It can also be suitable for an aero engine in a hybrid-electric configuration, where the coupling of the gas turbine with a battery and electric motor can allow it to operate as a power generation gas turbine at its high efficiency design point. In terms of compressor design, the main direction of the efficiency contours is closely linked to the design of the blades and the incidence angle, as aerodynamic loss is coupled to compressor efficiency. A blade designed for a unique incidence angle will provide peak efficiency at the design flow conditions and will have very high losses at off-design. A blade with a more rounded leading edge will allow a higher range of positive and negative incidence angles at a lower aerodynamic cost, but sacrificing peak efficiency. The choice of the design incidence angle for the blades also determines the operation of the compressor at part load. A design incidence that corresponds to the minimum pressure loss will provide an efficiency curve that has a strong peak and a compressor map like in fig. 6b. A design incidence can be chosen to optimize part load performance. In this case the compressor map looks like the one in fig. 6a.

3.1 HIGH SPECIFIC WORK CYCLES

Two different cases for high specific work cycles were evaluated. Figure 7 shows the results of the first case, an exploration using compressor maps with a large horizontal variation of η_p.

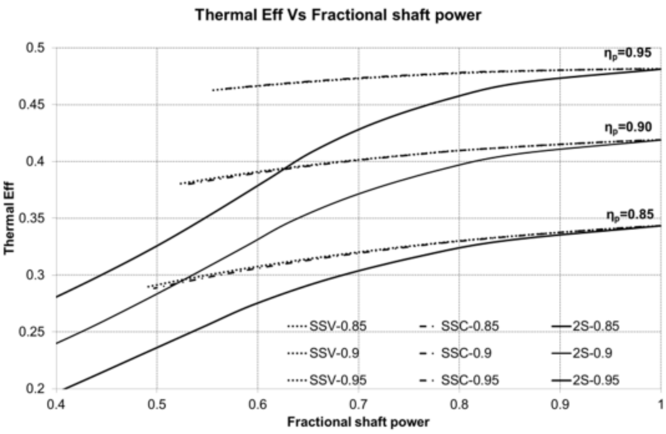

FIGURE 7: Thermal eff vs. fractional shaft power for a large horizontal variation of η_p

The map with vertical speed lines was used for the SSV case and the map with curved speed lines was used for the SSC case and for the two-shaft engine throughout these evaluations. Also in the following evaluations the value of η_p that is shown in figures 7-10 corresponds to the design point. As the operating point moves to lower output, the compressor efficiency changes based on the maps used.

Figure 7 shows that the two-shaft engine suffers significantly from the large polytropic efficiency reduction as mass flow falls. This results in a very large drop of thermal efficiency as power falls. For 50% part load, there is a 31% reduction in thermal efficiency (relative to its DP value) for η_p= 0.95 while it is 26% for η_p=0.85 (fig. 7). The compressors of the one shaft engines experience a much smaller reduction in polytropic efficiency so their thermal efficiency remains higher. Due to the shape of the map, the thermal efficiencies of the single-shaft engines in fig. 7 are very similar to those of fig. 4 because there is very little change in the polytropic efficiency of the compressors. Also the curved or vertical speed lines have a very small effect of the thermal efficiency at a single-shaft engine.

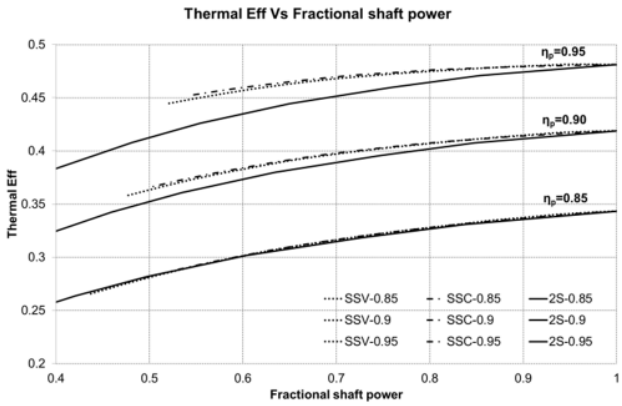

FIGURE 8: Thermal eff vs. fractional shaft power for large η_p vertical variation

Figure 8 shows the results of the second case, an exploration using compressor maps with a large vertical variation of η_p. Comparing the results with those of fig. 7 it is quite obvious that there is a small reduction in efficiency of the one shaft engines. This is due to the relatively small change in OPR. On the other hand, the two-shaft engine is now faring much better. This compressor map does not penalize so much this engine anatomy.

3.2 HIGH THERMAL EFFICIENCY CYCLES

As in section 3.1, two cases for high thermal efficiency cycles were examined, one each for each type of compressor map. Figure 9 shows the results of the first case, an exploration using compressor maps with a large horizontal variation of η_p. The results are similar to those of fig. 7, where the two-shaft engine exhibits a dramatic drop in thermal efficiency. As expected the large drop in polytropic efficiency with reduced mass flow has a dramatic impact on the overall efficiency of the two-shaft engine. The one shaft engines, on the other hand do not exhibit a visible change. The combination of the shape of the map with the small change in OPR (compared to the design point) yield a very small change of thermal efficiency. This is evident when fig. 9 is compared to fig. 5, where compressor polytropic efficiency does not change. For 50% part load, there is a 36% reduction in thermal efficiency (relative to its DP value) for η_p= 0.95 for the 2S while for SS is about only 5% (fig. 9).

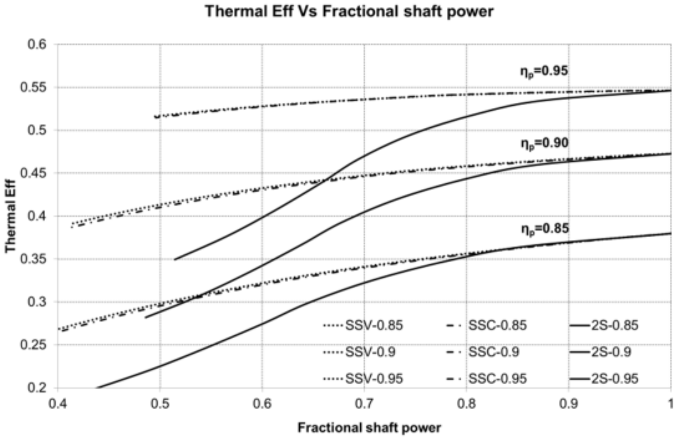

FIGURE 9: Thermal efficiency vs. fractional shaft power for a large horizontal variation of η_p

Figure 10 shows the results of the second case, an exploration using compressor maps with a large vertical variation of η_p. Comparing the results with those of fig. 9 it is quite obvious that there is a small reduction in efficiency of the one shaft engines. This is due to the relatively small change in OPR. On the other hand, the two-shaft engine is now faring much better. When η_p=0.95 at DP, at 50% part load the thermal efficiency drops about 13% in two-shaft engine (fig. 10). This compressor map does not penalize so much this engine anatomy.

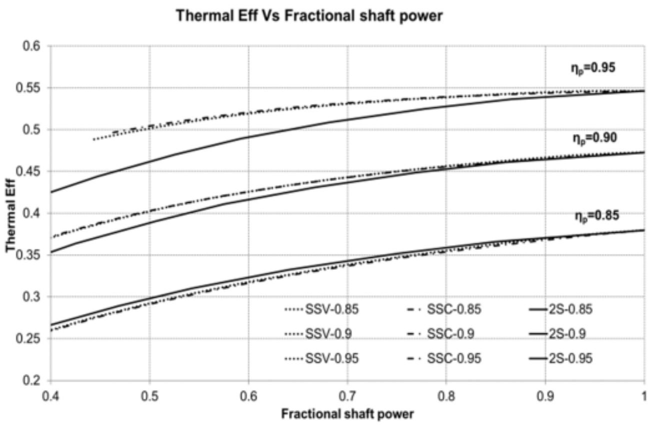

FIGURE 10: Thermal efficiency vs. fractional shaft power for a large vertical variation of η_p

Observing the results, as the compressor polytropic efficiency is lower at reduced power, the reduction in OPR becomes progressively smaller. This happens because to overcome the inefficiencies of the compressor the TET needs to be higher. This higher TET forces a higher OPR given the restriction of the turbine NGV area. Table 3 illustrates this.

Table 3: OPR (SSC) for 50% part load and different η_p

Fractional shaft power	η_p	OPR
1.0 (DP)	0.95	40.0
0.5	0.95	36.4
0.5	0.90	37.0
0.5	0.85	37.7

CONCLUSION

A thermodynamic performance model for different cases for each engine anatomy was produced using Turbomatch, to deliver an analysis that the authors have not seen in the public domain. This study shows the significant insights that can be obtained through the judicious use of a reliable gas turbine performance model.

The primary observation is that at part load, for the same fractional power output, the OPR of single-shaft engines (in the case where shaft speed is kept constant) falls much more slowly, while in two-shaft engines the TET falls more slowly. The latter arises from the reduction in mass flow associated with the reduction of rotational speed of the gas generator of two-shaft engines.

In a single-shaft engine operating at constant speed the mass flow does not fall at reduced power (in the absence of variable geometry). For this flow to pass through the turbine nozzle guide vanes, the gas pressure has to remain high as power falls. In the case of the two-shaft engine, the reduction in mass flow through the turbine requires a larger reduction in pressure ratio. At the same time to deliver the required fractional power output, with a lower mass flow, the TET will decline more slowly. Thus the reduction in power of the single-shaft engine is achieved entirely through a reduction in specific power, while in the two-shaft engine the reduction in power is achieved through a reduction in mass flow and a (smaller) reduction in specific power. Consequently, for high thermal efficiency cycles with constant turbomachinery efficiencies, the part load thermal efficiency of single-shaft engines reduces at higher rate than two-shaft engines, which is clearly visible in fig. 5 (η_p=0.85).

From the perspective of the ideal cycle, the single-shaft engine has the clear advantage at part load. Thermal efficiency is only a function of OPR, so for a given fractional power output, the OPR of the single-shaft engine is higher. This yields a higher thermal efficiency than the two-shaft engine.

However in real cycles, TET influences the thermal efficiency and the outcome depends on the component efficiencies. The lower the component efficiency, the larger the influence of the TET will be. Then for the lower power setting the level of polytropic efficiency and the shape of the compressor map is of primary importance.

In conclusion, the paper has explored the different conditions, which affect the performance of single and two-shaft engines and explained why the answer on the question set at the beginning is: it depends!

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