Part Load Performance of the Intercooled Two-Shaft Gas Turbine with Power Output at Constant Speed on the High-Pressure Shaft

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ABSTRACT

The general analysis of the part load performance of gas turbines indicates that the intercooled cycle with two shafts and power output at constant speed on the high-pressure shaft can have a good part load efficiency. Calculations with fixed geometry of the turbomachines show an intolerable increase of the turbine inlet temperature above the permissible level. By introducing variable geometry in the turbomachines, this disadvantage can be overcome. With variable inlet guide vanes at the high-pressure compressor an excellent part load performance is achieved. Further improvements are possible by adding an internal heat exchanger.

A GENERAL THEORY OF THE PART LOAD PERFORMANCE

The simple equation

\[ P = \dot{m} W \]

relates the power output \( P \), the fluid mass flow \( \dot{m} \), and the specific work \( W \) of the gas turbine cycle. It is easy to imagine, that according to this expression, there can be two boundaries:

1) With changing the power output the fluid mass flow remains constant

\[ \dot{m} = \text{const} \]

but the specific work must accordingly be changed directly proportional to the power output

\[ W = P \]

2) Another boundary is defined by a constant specific work

\[ W = \text{const} \]

at all levels of the power output, therefore the mass flow becomes directly proportional to the power output

\[ \dot{m} = P \]

In Fig. 1 the full lines represent the case 1), and the dash-dotted lines the case 2).

In the gas turbine practice all open cycle single shaft gas turbines for electricity production with constant rotational speed of the shaft correspond very nearly to the case 1) because in the compressor characteristic maps the lines of the constant rotational speed are practically vertical, i.e., that along these curves there is only a very slight dependence of the mass flow with the pressure ratio of the compressor.

The closed cycle gas turbines approximate the case 2). The fluid inventory within the cycle can be

NOMENCLATURE

\( \dot{m} \) mass flow
\( n \) rotational speed
\( P \) power output
\( p \) pressure
\( p_{\text{amb}} \) ambient pressure
\( Q \) heat input
\( s \) entropy
\( T \) temperature
\( W \) specific work of the gas turbine cycle
\( \eta \) thermal efficiency of the gas turbine cycle
\( \eta_{\text{is}} \) isentropic efficiency
\( \mu \) reduced mass flow
\( \nu \) reduced rotational speed
\( E \) pressure ratio
\( \Pi_{\text{tot}} \) total pressure ratio of the gas turbine cycle

Subscripts

\( \text{DP} \) design point
\( \text{HPC} \) high-pressure compressor
\( \text{HPS} \) high-pressure shaft
\( \text{LPC} \) low-pressure compressor
\( \text{LPS} \) low-pressure shaft
\( \text{RP} \) reference point
\( t \) total condition of the fluid
\( 1 \) through 7 stations of the gas turbine cycle

(see Fig. 2 and 3)
Fig. 1
Relative fluid mass flow $\dot{m}/\dot{m}_{DP}$, relative specific work $W/W_{DP}$, and the relative thermal efficiency $n/n_{DP}$ of the gas turbine cycle in relation to the relative power output $P/P_{DP}$

Full lines:
Single shaft constant speed gas turbine
Dash-dotted lines:
Closed cycle gas turbine

changed by influencing the pressures: The mass flow is directly proportional to the pressure level.

The operator of a gas turbine power plant is interested specifically in the dependence of the thermal cycle efficiency (resp. specific fuel consumption) with the power output.

In the case 1) the change of the specific work can be performed only by influencing the turbine inlet temperature (resp. fuel mass flow): Lower specific work means lower turbine inlet temperature. When the influence of the highest cycle temperature in the Carnot-Cycle is considered, it is evident that the part load performance of such gas turbines is unsatisfactory. If the thermal efficiency of the cycle is written as

$$n = \frac{W}{Q}$$
then there exists the relation

$$n/n_{DP} > P/P_{DP}$$
see Fig. 1. This can be explained as follows: With lower power output, and lower turbine inlet temperature, there is also a smaller heat input $Q$ to the cycle.

In closed cycle gas turbines which represent the case 2), the temperatures at different stations within the cycle remain unchanged: Therefore the Carnot efficiency is constant. However, in practice, at very low power output levels, according to Fig. 1, there exists a steep decline in efficiency. That is mainly the influence of the power consumption of the auxiliaries.

All other types of gas turbines, for example, aero-derived gas generators with attached constant speed power turbines have part load performance somewhere between these two cases, because the mass flow changes with the power output.

With the aid of this analysis there should be made an attempt to find a suitable configuration of a gas turbine cycle with the following characteristics:
- open cycle with air as fluid.
- mass flow dependence with the power output like that of closed cycle gas turbines (case 2)) which means a cycle where the specific work and the turbine inlet temperature remain as constant as possible.
- the power plant should not be too complicated which implies no multiple stages of intercooling and/or reheating. For some inherent problems with reheating, even only one stage of reheating should be avoided.
- the mutual arrangement of the different parts (turbomachines, combustors, etc.) within the power plant must conform with the existing technology.
- with respect to the high development costs of a new gas turbine power plant, it would be very favourable if some already existing elements (especially compressors) could be used in the contemplated cycle.

From the operator's point of view, such a power plant would have an excellent load following capability. This fact offers the possibility to open new markets for the gas turbines in electricity production.

**THE TWO-SHAFT INTERCOOLED CYCLE**

For the moment, let us forget the part load performance of a gas turbine power plant. The gas turbine manufacturers are constantly looking for means to increase the efficiency and the specific work (i.e. to reduce the specific fuel consumption, and to reduce the fluid mass flow for a given power output). There exist, generally speaking, the following possibilities: intercooling, reheating, internal heat exchange, and the usage of the gas turbine as a topping unit for a steam power plant.

One stage of intercooling, see Fig. 2, increases the efficiency and the specific work. With this scheme, it is important to consider the influence of the overall pressure ratio and also the division of this overall pressure ratio between the low-pressure and the high-pressure compressor (1).

Fig. 3 shows the heat flow scheme of such a gas turbine, where two shafts are used, and the power output with constant rotational speed is located at the high-pressure shaft. If the power output would be with the low-pressure shaft, there would be the same unsatisfactory part load performance as with the single shaft simple cycle gas turbine (case 1): $\dot{m} = \text{const}$.

With the power output at the high-pressure shaft, there can exist some positive influences by using the following analysis:

*Fig. 2 Temperature-entropy diagram of an intercooled gas turbine cycle. The stations 1 through 7 correspond to the heat flow scheme in Fig. 3*
The rotational speed of the high-pressure shaft is constant \( n_{HpS} = \text{const} \). Due to the intercooling, we can say, that the temperature \( T_{t3} \) before the high-pressure compressor will remain approximately constant. W.r.t. the reduced rotational speed, that means

\[
\frac{n_{HPS}}{\sqrt{\frac{T_{t3}}{P_{t3}}} = \text{const}
\]

and also

\[
\frac{\dot{n}_{HPC}}{\sqrt{\frac{T_{t3}}{P_{t3}}}} = \text{const}
\]

just as with the compressor of a single shaft engine. But, if the pressure \( P_{t3} \) can be changed, the mass flow can also be changed

\[
\dot{n}_{HPC} = P_{t3} \cdot \dot{n}_{HPS}
\]

Because of the negligible pressure losses in the intercooler we can write

\[
P_{t3} = P_{t2}
\]

or

\[
\dot{n}_{HPC} = \dot{n}_{LPC} = \dot{n}
\]

With the continuity

\[
\dot{n}_{HPC} = \dot{n}_{LPC} = \dot{n}
\]

it follows

\[
\dot{n}/\dot{n}_{DP} = P_{t2}/P_{t2,DP} = \text{const}
\]

The rotational speed of the low-pressure shaft \( n_{LPS} \) is variable. Therefore, it can be expected that with the part load, the mass flow will be low, the specific work will be high, resulting in a favourable efficiency.

The cycle according to the heat flow scheme in Fig. 3 can be transformed into a practical gas turbine power plant as shown in Fig. 4. The critical parts, which are exposed to high temperatures and pressures, i.e. combustor and both turbines, can be built within one casing, thereby avoiding any lengthy ducts between those parts. The duct between the intercooler and the high-pressure compressor is cold and is exposed only to some medium pressure. Therefore, it is no point of concern.

**CALCULATIONS: FIXED GEOMETRY OF THE TURBOMACHINES**

With the assumptions as given in the appendix, Fig. 5 shows the design point calculations for the thermal efficiency. The importance of the ratio \( \frac{\dot{n}_{HPC}}{\dot{n}_{LPC}} \) can be clearly seen. High overall pressure ratio \( n_{tot} = P_{t4}/P_{t1} \) is advantageous, but the necessary pressure ratio \( n_{HPC} \) must be considered. If this pressure ratio is too high, there exist similar limitations as with the single shaft gas turbine. Assuming that the state of the art is limited to approximately \( n_{HPC} = 12 \), the values above the dash-dotted line cannot be realized.

With fixed geometry of the turbomachines there have been made calculations of the part load performance for the design point \( n_{tot} = 36.3, n_{LPC} = 5.547, \) and \( n_{HPC} = 6.675 \). For other parameters, see the appendix. The real behaviour of the turbomachines have been taken into consideration with the aid of synthetically generated characteristics maps (2, 3).

Fig. 6 shows the thermal efficiency \( n \), the turbine inlet temperature \( T_{t5} \), and the relative air mass flow \( \dot{m}_{air} \) in relation to the relative power output \( P/P_{DP} \). At first glance it is obvious, that the temperature-"overshot" \( T_{t5} > T_{t5,DP} \) in the extent up to 70 deg K cannot be tolerated. In this figure the relative air mass flow shows a favourable dependence with the relative power output (see above the theoretical case 2).

The temperature-"overshot" can be explained with the temperature-entropy diagram as shown in Fig. 7. The
Fig. 6 Thermal efficiency \( n \), the turbine inlet temperature \( T_{t5} \), and the relative air mass flow \( \dot{m}/\dot{m}_{DP} \) in relation to the relative power output \( P/P_{DP} \).

three lines correspond to

\[
\begin{align*}
P/P_{DP} &= 100\% \quad \text{(design point)} \\
P/P_{DP} &= 42\% \quad \text{(maximum of the temperature-'overshot')} \\
P/P_{DP} &= 25\% \quad \text{(part load where } T_{t5} = T_{t5,DP} \text{)}.
\end{align*}
\]

According to this figure, at part load the overall pressure ratio is reduced to such an extent that for \( T_{t5} \leq T_{t5,DP} \) the specific work of the cycle becomes too small. This loss must be compensated by the increase of the turbine inlet temperature \( T_{t5} \) (see Fig. 6).

The change of the operating conditions in the low-pressure compressor characteristics map (line OPL) is shown in Fig. 8. The reference point RP of the compressor is defined as the point of the maximum isentropic efficiency. With the reduced rotational speed \( \dot{n}/\sqrt{1-t_{1}} \) and the reduced mass flow \( \dot{m}/\dot{m}_{t1/P_{t1}} \), the reference point is valid \( \dot{m}/\dot{m}_{RP} = 1 \), and \( P/P_{RP} = 1 \).

Fig. 7 The temperature-entropy diagram for the intercooled cycle with fixed geometry

Full line: \( P/P_{DP} = 100\% \) (design point = DP)
Dash-dotted line: \( P/P_{DP} = 42\% \) (maximum of \( T_{t5} \))
Dashed line: \( P/P_{DP} = 25\% \) (\( T_{t5} = T_{t5,DP} \))

Fig. 8 Characteristics map of the low-pressure compressor with the operating line (OPL) at part load

DP design point
RP reference point (\( \eta_{IS} = \eta_{IS,max} \))

According to this figure, the design point DP is not identical with the reference point RP. The operating line (OPL) follows the expression \( \dot{m}/\dot{m}_{DP} = 0/\dot{m}_{DP} \).

VARIEABLE GEOMETRY

From the foregoing results it is clear that the intercooled cycle with power output at the high-pressure shaft and with only one way to govern the power output (fuel flow), is not feasible until some other means are introduced which prevent the temperature-'overshot'. There is a possible way: variable geometry must be used.

Looking at the investigated cycle, there are three places where variable geometry can be applied using already established low-risk, low-cost technology. There are some examples of gas turbines where variable inlet guide vanes in the compressor or variable first stage vanes of a low-pressure turbine are used. In a particular case the variable geometry can be applied as variable inlet guide vanes in the low-pressure compressor (VLPC), in the high-pressure compressor (VHPC), and as variable first stage vanes of the low-pressure turbine (VLPT). The variable geometry is symbolically indicated in Fig. 4. In total there are seven combinations at disposal:

1) VLPC
2) VHPC
3) VLPT
4) VLPC plus VHPC
5) VLPC plus VLPT
6) VHPC plus VLPT
7) VLPC plus VHPC plus VLPT.

VARIABLE INLET GUIDE VANES IN THE LOW-PRESSURE COMPRESSOR (VLPC)

Calculations with synthetic characteristics maps (2, 4) show that for this case the temperature-'overshot' cannot be avoided. By opening the vanes, the air mass flow will be increased. This increase cannot compensate the loss of specific work. The 'overshot' is smaller, but nevertheless, at part load it still exists, similar to that in Fig. 6. It is unnecessary to discuss these negative results at length.

VARIABLE INLET GUIDE VANES IN THE HIGH-PRESSURE COMPRESSOR (VHPC)

With variable inlet guide vanes in the high-pressure compressor the condition \( T_{t5} \leq T_{t5,DP} \) can be fulfilled. For different angles \( \alpha_{VHPC} \) within some ranges
of the relative power output $P/P_{DP}$ there is no temperature-"overshot", see Fig. 9. Corresponding thermal efficiencies $n$ are shown in Fig. 10.

There are no significant differences for the thermal efficiencies. In Fig. 10 the results are summarized and appear as a narrow strip. The upper boundary is given by $\Delta \omega_{HPC} = 0^\circ$ up to $P/P_{DP} = 25\%$, and further from $P/P_{DP} = 33\%$ up to $100\%$. In between, the angle is changed according the prescription $\omega_{HPC} = \omega_{HPC,DP}$ (see Fig. 9). The lower boundary is given by $\Delta \omega_{HPC} = 7^\circ$.

In Fig. 10 the scale for the efficiency is exaggerated, so that exact values can be seen. The small built-in figure $n/\eta_{DP} = f(P/P_{DP})$ gives the proper overall impression of the excellent part load performance approaching that of the closed cycle gas turbine.

There are small differences in the efficiency between, for example, the angles $\Delta \omega_{HPC} = 0^\circ$ and $7^\circ$ though there can be temperature differences in the turbine inlet temperature $T_{t5}$ as high as 50 degrees (Fig. 9). This can be explained by the higher total pressure ratio $\Pi_{tot}$ which is for $\Delta \omega_{HPC} = 7^\circ$, see Fig. 11. The ratio $\Pi_{HPC}/\Pi_{LPC}$ remains practically the same. Nevertheless, this small difference is in favour of $\Delta \omega_{HPC} = 7^\circ$. In the same figure we see the values of $\Pi_{tot}$ and $\Pi_{HPC}/\Pi_{LPC}$ for $\Delta \omega_{HPC} = 0^\circ$, i.e. fixed geometry, with temperature-"overshot" according to Fig. 6.

Looking at Fig. 9 and 10 for angles $\Delta \omega_{HPC}$ higher than $0^\circ$ the design point load $P/P_{DP} = 100\%$ cannot be reached. This limit is given by the design point speed of the low-pressure compressor ($\gamma_{LPC}/\gamma_{HPC,DP} = 1.05$).

In connection with Fig. 9 and 10 a change of $\Delta \omega_{HPC}$ from $6^\circ$ to $0^\circ$, as shown in Fig. 12, results in values of $T_{t5}$ and $n$ as indicated in the same figure.

The same pattern of variation of the angle $\Delta \omega_{HPC}$ delivers values of $\Pi_{HPC}/\Pi_{LPC,DP}$, $\Pi_{LPC}/\Pi_{HPC,DP}$, $\gamma_{HPC}/\gamma_{LPC,DP}$, and $\gamma_{LPC}/\gamma_{HPC,DP}$ as shown in Fig. 13 (dashed lines). Curves for $\Delta \omega_{HPC} = 0^\circ$ (fixed geometry with temperature-"overshot": full lines) can also be seen. At part load the pressure ratio of the high-pressure compressor increases only slightly: there is no danger that surging occur.

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![Fig. 9](image-url) Turbine inlet temperature $T_{t5}$ for different inlet guide vane angles $\Delta \omega_{HPC}$ of the high-pressure compressor in relation to the relative power output $P/P_{DP}$.

![Fig. 10](image-url) Thermal efficiency $n$ for different inlet guide vane angles $\Delta \omega_{HPC}$ of the high-pressure compressor, as shown in Fig. 9, in relation to the relative power output $P/P_{DP}$. The small figure in the righthand corner shows the relative efficiency $n/\eta_{DP}$ in relation to the relative power output (note: 0% to 100% on both scales).

![Fig. 11](image-url) The total pressure ratio $\Pi_{tot}$ and the ratio $\Pi_{HPC}/\Pi_{LPC}$ of the compressor pressure ratios for the inlet guide vane angles $\Delta \omega_{HPC} = 0^\circ$ and $7^\circ$. $\Delta \omega_{HPC} = 0^\circ$ correspond to the fixed geometry with temperature-"overshot" (see Fig. 6).

![Fig. 12](image-url) The efficiency $n$ and the turbine inlet temperature $T_{t5}$ in relation to the relative power output $P/P_{DP}$ when the inlet guide vane angles $\Delta \omega_{HPC}$ are changed as indicated.
Fig. 13 Relative values of the rotational speed \(v/v_{\text{RP}}\), and relative pressure ratio \(\Pi/\Pi_{\text{DP}}\) for the high- and low-pressure compressor in relation to the relative power output \(P/P_{\text{DP}}\).

For the sake of completeness, Fig. 14 shows the temperature \(T_{16}\) at the inlet, and the temperature \(T_{17}\) at the outlet of the low-pressure compressor.

The governing of the high-pressure compressor inlet guide vanes is shown in Fig. 12. As a signal, the rotational speed of the low-pressure shaft can be used. A second way can be discussed with the aid of Fig. 9. Up to \(P/P_{\text{DP}} = 93\%\) the angle \(\Delta \alpha_{\text{HPC}}\) is fixed to 6°. For higher power outputs until \(P/P_{\text{DP}} = 100\%\), it becomes smaller until 0°.

For quick load following modes of operation, the first possibility is advantageous, for slower load changes the second one.

VARIABLE GEOMETRY: THE REMAINING PART OF THE POSSIBILITIES

The case of variable first stage vanes of the low-pressure turbine (VLPT) delivers no better part load efficiencies than the already examined case VHPC. It is preferable to use variable vanes at low temperatures (approx. 290 K) than at high temperatures (approx. up to 810 K). Therefore, it is omitted to show the results for the case VLPT.

The case VLPC plus VHPC delivers better efficiency of only some 0.001 points in the load range between 60 and 80%. This gain in too small to justify introducing an additional variable geometry in the power plant.

The results of the cases VLPC plus VLPT and VHPC plus VLPT have been also disappointing. Therefore, the expectations of the most complicated possibility VLPC plus VHPC plus VLPT have not been assumed as promising, so the calculations have not be done.

INTERCOOLED CYCLE WITH REGENERATION

By introducing a heat exchanger into the cycle, the design point efficiency can be enhanced significantly. Looking at the case VHPC with variable inlet guide vanes of the high-pressure compressor, the following statements concerning part load behaviour can be made:

- The temperature \(T_{14}\) at the outlet of the high-pressure compressor remains practically constant (see the ratio \(\Pi_{\text{HPC}}/\Pi_{\text{DP}}\) in Fig. 13).
- Reducing the load, the temperature \(T_{17}\) at the outlet of the low-pressure turbine increases considerably until \(P/P_{\text{DP}} = 30\%\) and thereafter decreases (see Fig. 14).
- The turbine inlet temperature \(T_{15}\) remains nearly constant between \(P/P_{\text{DP}} = 40\%\) through 100% (see Fig. 12).

From these facts it can be concluded, that by decreasing the load in a broad range, the temperature of the air at the outlet of the heat exchanger, resp. the temperature at the inlet of the combustor increases, thereby reducing the specific heat input \(Q\), with the overall result of an unprecedented part load efficiency, as shown in Fig. 15. At full load the efficiency increases from 33.2% to 38.4%. Comparing this figure with Fig. 10, there exists a loss of power output due to pressure losses in the heat exchanger.

From already performed calculations (1) it is known, that by additional regeneration in an intercooled cycle, the overall pressure ratio should be smaller than in an intercooled cycle without regeneration (see Fig. 5). The chosen overall pressure ratio \(\Pi_{\text{DP}}\) of 36.3 is not very favourable for the examined cycle. Nor is the chosen division of this total pressure ratio between the high-pressure and the low-pressure compressor favourable. This data, however, would be perfectly suited for a regenerated cycle.
In the search for more efficient gas turbine power plants, there should be asked: How can the already existing gas turbines be adapted for use in a cycle with intercooling and regeneration? This question makes sense w.r.t. the excessive development costs of a new power plant. The manufacturer, and also the operator, will be glad if some already proven parts can be used again, thereby minimizing the risk of introducing a new power plant. (In this idea, there is some analogy to the development of the intercooled and regenerated Spey marine propulsion unit by Rolls-Royce.)

Looking at the existing gas turbine power plants, some twin-spool aero-derived gas generators can be very conveniently adapted for intercooling and regeneration. This change will increase the air flow. The power output will rise as a result of increases in air flow and specific work (because of intercooling). A redesign of the turbines is necessary.

CONCLUSIONS

The intercooled two-shaft gas turbine with power output at constant speed on high-pressure shaft has an excellent part load performance approaching that of closed cycle gas turbines.

With fixed-geometry turbomachines there is at part load an intolerable increase of the turbine inlet temperature above the permissible level at the design point. By introducing variable inlet guide vanes at the high-pressure compressor, this disadvantage can be overcome. Calculations of other possibilities of variable geometries did not offer any significant improvement in part load efficiency.

Some of the already existing gas turbine power plants can be adapted to the proposed configuration, thereby increasing the power output and the efficiency. The addition of an internal heat exchanger (regeneration) can further improve the design point efficiency, and in general the part load performance.

This unprecedented excellent part load performance offers for this type of gas turbine the possibility of a real load following capability.

REFERENCES


APPENDIX: ASSUMPTIONS FOR THE CALCULATIONS

Temperatures:
- design point temperature before the high-pressure turbine $T_{1h,DP} = 1373$ K
- ambient temperature $T = 288$ K

Pressure ratios:
- low-pressure compressor $\Pi_{PC,DP} = 5.547$
- high-pressure compressor $\Pi_{HPC,DP} = 6.675$

Pressure losses:
- intercooler = 0.98
- combustor = 0.99
- no pressure losses at the inlet and the outlet of the gas turbine power plant have been assumed (= function of cleaning and noise)

Isentropic efficiency at the design point (DP):
- low-pressure compressor $= 0.875$
- high-pressure compressor $= 0.88$ (not 0.872)
- high-pressure turbine $= 0.870$
- low-pressure turbine $= 0.870$

Isentropic efficiency at the reference point (RP):
- low-pressure compressor $= 0.8784$
- high-pressure compressor $= 0.8755$
- high-pressure turbine $= 0.870$
- low-pressure turbine $= 0.870$

Heat exchanger effectiveness $= 0.80$
Heat loss in the combustor $= 0.98$
Real behaviour of the air and combustion gases have been considered

Pressure losses:
- intercooler $= 0.98$
- combustor $= 0.99$
- no pressure losses at the inlet and the outlet of the gas turbine power plant have been assumed (= function of cleaning and noise)