Wilson TurboPower Inc.

Phone and FAX: 781 729 2203; E-mail: dgw@wilsonturbopower.com
Friday, November 22, 2002

THE BASIS FOR THE PREDICTION OF HIGH THERMAL EFFICIENCY IN W.T.P.I. GAS-TURBINE ENGINES

David Gordon Wilson

SUMMARY

A high thermal efficiency for an engine requires that a cycle capable of producing high efficiency is first chosen, and that components that can operate at high efficiencies are used. The choices made in both areas are discussed below. The "enabling" technology is a high-temperature, high-effectiveness (0.975) regenerator, with a low pressure drop. This requires only a low cycle pressure ratio (about 2.5:1) for optimum performance. The component efficiencies of compressors and turbines of low pressure ratio are intrinsically high. W.T.P.I. also divides this low pressure ratio among three compressor stages and three turbine stages. Doing so greatly reduces the kinetic energy that is usually lost from single stages, and thus substantially improves the component efficiencies. By these means a shaft-power efficiency of 54% can be confidently predicted for the 300-kW engine, which when coupled to a high-efficiency electrical generator would lead to an electrical efficiency of about 50%.

INTRODUCTION

The shaft-output efficiencies just quoted for the W.T.P.I. 300-kW gas-turbine engine, about 54%, and an electrical efficiency of 50%, are double those of a principal competitor, and considerably higher than those attained in Japan's long-duration M.I.T.I. program. In this program a goal was set for a shaft thermal efficiency of 42% for the three 300-kW engines, the development of which was funded. The manufacturers involved have claimed that the goal was eventually reached after an extension of the program.

It is natural that there should be some skepticism that a new small company should be able to exceed the efforts of many established organizations. The arguments below are aimed at allaying this understandable uncertainty.

The design and performance predictions for the W.T.P.I. engine have been subjected to a critical analysis by Dr. Peter E. Jenkins, dean of the college of engineering and applied science, University of Colorado at Denver, who affirmed the claimed performance[E1]. The approach has been treated at length in Wilson and Korakianitis (1998)[E2], and in greater detail and specificity in Wilson (1997)[E3], and in Wilson and Pfahnl (1997)[E4], from which much of the following will be taken.

As each choice is revealed, the directions taken by other companies will be briefly discussed.

CHOICE OF THERMODYNAMIC CYCLE CAPABLE OF HIGH EFFICIENCY

Temperature ratio

All heat-engine cycles are governed by the second law of thermodynamics, embodied in the Carnot cycle. This cycle has the maximum efficiency that a heat engine can attain, and is expressed by $\{1 - (T_1/T_2)\}$. Here T_1 is the temperature of heat rejection by the engine, and T_2 the temperature of heat addition. For a high efficiency, therefore, the temperature ratio T_2/T_1 (referred to below as T') must be high. In a gas-turbine engine, the surrogates for these temperatures are those of the air entering the compressor and of the gas entering the turbine.

There isn't much choice for the temperature of the air entering the compressor because it is normally taken in directly from the atmosphere (although it has become popular to use gas turbines' waste heat to run absorption chillers and thereby to cool the compressor inlet air. This will not affect our argument here.)

The fundamental method of increasing the temperature ratio T' is to increase the turbine-inlet temperature. In large engines, including most aircraft engines and industrial turbines used for central generating plant, metal turbine blades incorporating sophisticated air- or steam-cooling systems are used, allowing turbine-inlet temperatures up to 1600 C (for which T' approaches 6.5). When used in combined-cycle plants, such temperatures allow electrical efficiencies of 60% to be reached and even exceeded.

However, with currently available technology, turbine blades used in low-power engines such as 300 kW are too small to incorporate cooling of this type. Some limited cooling could be developed for small turbines, but it would be prohibitively expensive unless a major production run were foreseen. All small turbines known to this writer have blades that are uncooled. Therefore, for metal blades that cannot be cooled, the turbine-inlet temperature cannot exceed about 1000 C (when T' is about 4.25). The consequence of this limited temperature is reduced efficiency.

An increased temperature could be reached if nonmetallic blades could use materials such as ceramics, some of which have maximum-use temperatures listed as 1400 - 1600 C (T' from 5.6 - 6.5). There have been several attempts to do so. A prominent example is the engine that started as the Garrett-Ford AGT-101 engine. Its turbine-inlet temperature was about 1375 C. Unfortunately, the designers chose to use a silicon-nitride radial-inflow turbine wheel rotating with a peripheral speed of over 700 m/s. The centrifugal stresses were extremely high. Moreover, for a single grain of sand to pass through the turbine would have required that it attain a centripetal acceleration of well over half-a-million 'g's almost entirely by aerodynamic drag forces. Sand and other particles could not attain this acceleration, and were trapped between the nozzle exits and the turbine-wheel inlet, hitting both with enormous energy until the wheel was destroyed.

One successful demonstration of an axial-flow ceramic turbine was accomplished late in the AGT-100 program by Allison, then a division of General Motors. Phil Haley, the Allison program manager, reported to this writer that the turbine reached full speed and full temperature on the first run. This was remarkable after the years of reporting attempts at reaching full speed even at low temperature in radial-inflow turbines. Axial-flow turbines can pass small particles without problem. We are using axial-flow ceramic turbines.

Silicon nitride (and silicon carbide) have been found to suffer attack by water vapor at high temperatures. This has turned out to be a serious problem when designers have attempted to use these materials for microturbine blades. Many environmental barrier coatings have been tested to solve this problem. One being tested is mullite.

It happens that the first choice of W.T.P.I. for its turbine blades is mullite, a fully oxidized ceramic, and is thus more resistant to the oxidizing effects of the oxygen ions in high-temperature water vapor. Mullite has a lower strength than silicon nitride, and therefore cannot be used in place of silicon nitride in microturbines with typical high tip speeds. However, mullite is quite viable in an axial-flow turbine having the very low peripheral speed of 275 m/s, achieved by the methods described below. Thus centrifugal and foreign-object-impact stresses are low enough to give a high factor of safety even with this commonplace ceramic. The turbine-inlet temperature chosen is 1230 C, well below the maximum-use temperature of 1600 C given for this class of mullite, and so producing an even larger factor of safety.

Pressure ratio

While the Carnot cycle can be characterized simply by a temperature ratio, all practical heat engines running on gaseous fluids (including steam) also use a pressure ratio, r_c, between appropriate processes in the cycle. In a Brayton cycle, each value of temperature ratio has associated with it an optimum pressure ratio (one which is affected by component efficiencies also). Modern jet engines have pressure ratios of over 30:1, and values of 50 and even 100:1 are foreseen.

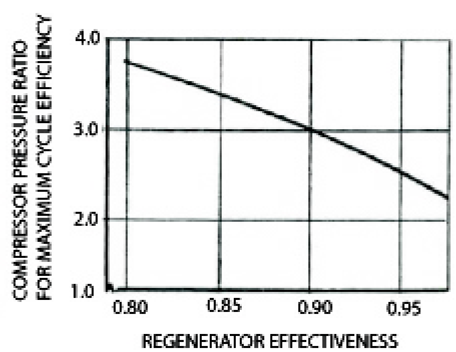

Figure 1: Optimum compressor pressure ratio falls with increase of regenerator effectiveness.

Such high pressure ratios are out of the question for small engines. High pressure ratios are handled by using long blades at entrance to a compressor, and very short blades at exit, so that the peripheral speeds can be kept high and the axial through-flow velocity can also stay high. In a low-power (and thus small) engine the use of a high pressure ratio would lead to compressor-outlet blade lengths that would be smaller than the blade-to-casing clearance that is required to handle rotor excursions, and temperature and pressure loads, for both axial- and radial-flow compressors. This situation would degrade the compressor efficiency to the point at which the engine could not produce positive power.

There are, fortunately, many possible thermodynamic cycles on which a gas-turbine engine can operate. The most-relevant to the small-turbine problem is the heat-exchanger cycle. The hot exhaust leaving the turbine passes through one 'side' of a heat exchanger, and the cool air leaving the compressor goes through the other side, undergoing heating and thus reducing the quantity of fuel required to bring the air to the chosen turbine-inlet temperature. When a heat exchanger is used, a high pressure ratio would increase the temperature of the compressor-outlet air and would decrease the temperature of the turbine-outlet gas, thus reducing the amount of heat that could be transferred. Thereby a remarkable switch in the optimum pressure ratio is brought about. For a cycle with no heat exchanger but a high temperature ratio, a high pressure ratio is optimum. However, for a cycle with a heat exchanger, a low pressure ratio is optimum. Moreover, the more efficient the heat exchanger (in a heat exchanger, efficiency is defined as 'effectiveness') the lower is the optimum pressure ratio (figure 1). Consequently, all small turbines designed for high efficiencies use heat exchangers.

Normally, the use of heat exchangers leads to compromises, for the following reason. If a high-effectiveness heat exchanger is used, the optimum pressure ratio is low, and the temperature drop through a turbine of low pressure (or more strictly expansion) ratio is also low. The resulting temperature is usually too high for a metallic recuperator, for which the maximum inlet temperature is about 700 C (figure 2). The obvious step to take is to use a ceramic heat exchanger. There have been many attempts at producing a ceramic recuperator (defined as a steady-flow tubular or plate-fin heat exchanger), none so far successful. Many experimental ceramic-honeycomb rotary regenerators (in which the flow is switched to transfer heat) have been tried, but all known trials have produced fast seal wear and high leakage (e.g., 15% of the compressor air[E5]). Accordingly, the pressure ratio is usually set at higher-than-optimum values, and the heat-exchanger at below-optimum effectiveness, so reducing the potential cycle thermal efficiency.

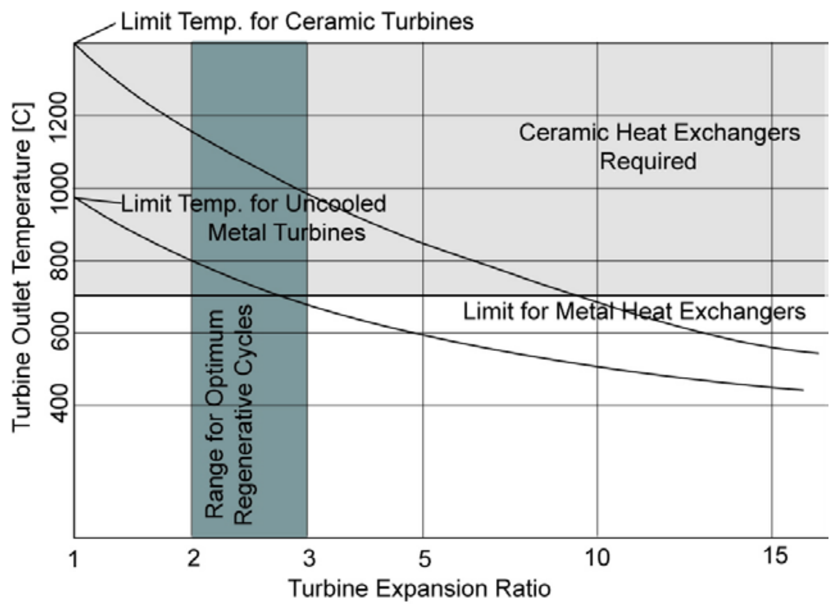

Figure 2: Effect of compressor pressure ratio (turbine expansion ratio) on heat-exchanger inlet temperature (turbine outlet temperature).

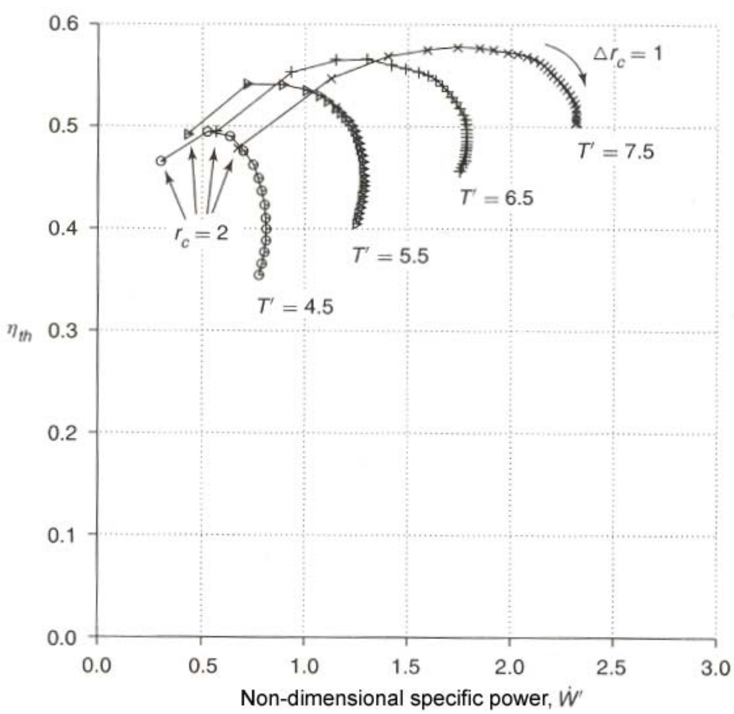

Figure3:High regenerator effectiveness (here it is 0.975) and low pressure ratio gives high cycle thermal efficiency.

The W.T.P.I. technology includes a patent on a new variation of the rotary regenerator in which both leakage and wear are greatly reduced[E6]. It is also relatively easy to design for a very high effectiveness, 95% and even 97.5% coupled with a low exhaust-side pressure drop (the air-side pressure drop is normally very low). The regenerator core is also of mullite, having a temperature capability allowing an optimum, low, pressure ratio to be used. This combination of high effectiveness and low pressure ratio leads to the potential of high cycle efficiency (figure 3).

COMPONENT EFFICIENCIES

As stated above, it is necessary to have both a cycle with the potential of giving a high thermal efficiency, and high efficiencies for the components. It has been claimed that the new regenerator technology allows the use of very high effectiveness, low pressure drops and low leakage. This low pressure drop intrinsically results in somewhat higher efficiencies for the compressor and turbine. The writer has correlated data from all published results of tests on axial- and radial-flow compressors and turbines in Wilson and Korakianitis (1998), and the relevant plots are in figures 4 & 5. Both the pressure-ratio and the size effects on maximum efficiency are shown.

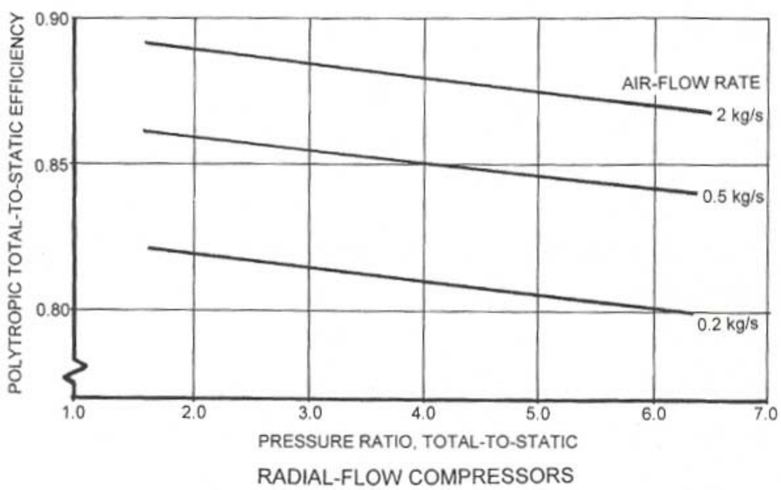

Figure 4: Best efficiencies of radial-flow compressors as function of pressure ratio and mass-flow rate.

It is less generally recognized that the kinetic energy at the outlet of single-stage compressors and turbines can be very high, of the order of 10 - 15% of the enthalpy change across the machine. Turbomachinery designers try to recover as much kinetic energy as possible by passing the flow into a diffuser (which ideally converts the dynamic pressure of the flow into a static-pressure rise), but the disturbed flow at the outlet of compressors and turbines is not conducive to ideal diffusion. The effects on the useful efficiency of an axial-flow turbine are shown in figure 6. If the turbine blading has a (polytropic) efficiency of 90% and no diffuser is used, the useful efficiency (the so-called total-to-static polytropic efficiency) is only 70%. A diffuser with a pressure-recovery coefficient (C_{pr}) of 0.4, a typical value, would increase the useful efficiency to 77%.

There is a dramatic effect of using two, three or even four stages (combinations of stator blade rows and rotor blade rows) in this axial turbine. With three stages, the number chosen for the W.T.P.I. engine, both in the compressor and the turbine) and the typical diffuser, the useful efficiency is 86%.

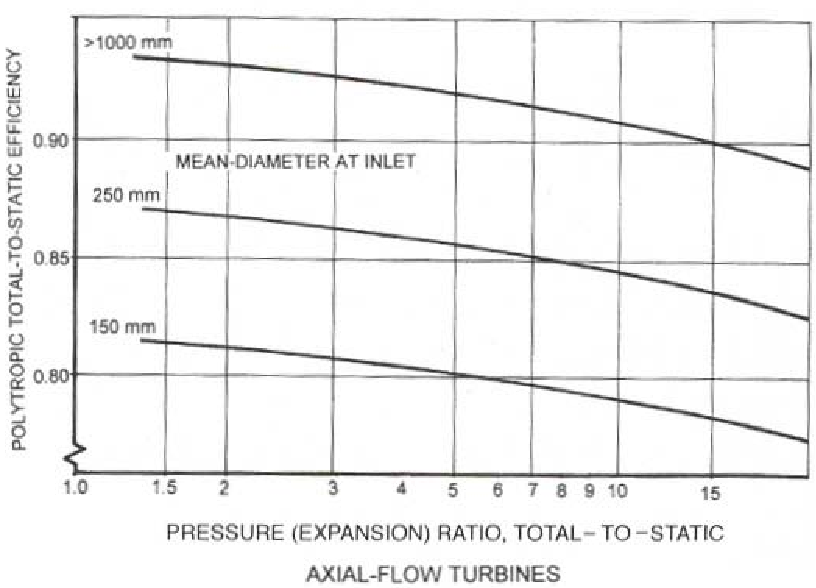

Figure 5: Best efficiencies of axial-flow turbines as function of pressure (expansion) ratio and size.

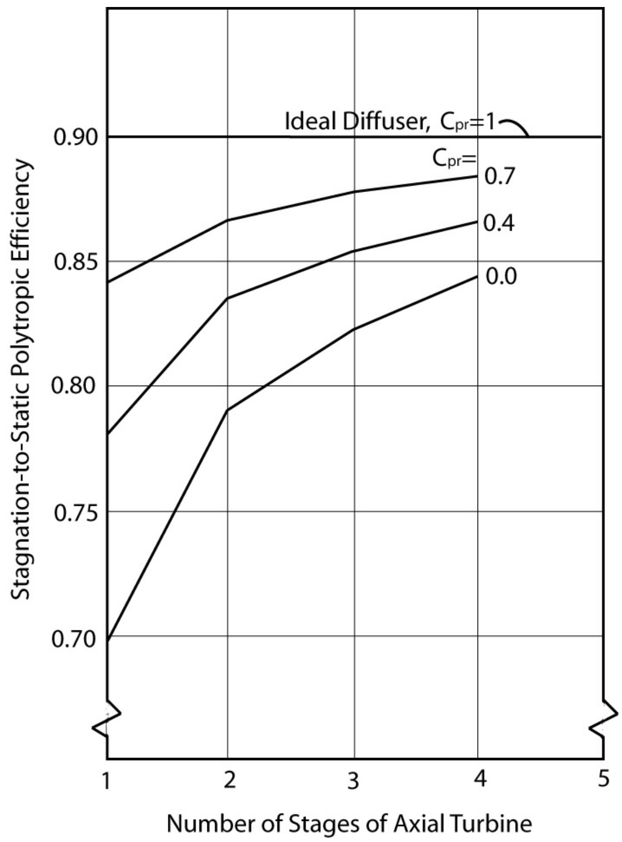

Figure 6: Effect of number of stages and diffuser quality on the useful efficiency of an axial turbine (having an efficiency with a "perfect" diffuser of 0.90).

In a typical centrifugal compressor (which W.T.P.I. plans to use) there is normally a radial-flow diffuser after the impeller, and there is considerable kinetic energy left after that diffuser. Therefore the same remarks hold as for the axial-flow turbine. Having three stages instead of one means that each stage has one-third the enthalpy rise, one third the enthalpy in the kinetic energy leaving the stage, and, if this kinetic energy can be fed into the following stage and thus used, the potential loss of kinetic energy is that after the last stage, which is again one-third that of a single-stage compressor.

Moreover, the polytropic or useful efficiency of each stage is higher than that of a single stage because the pressure ratio is much reduced.

These factors contribute greatly to the predicted high overall efficiency of the engine (although they have not been fully taken advantage of in the analysis.) The intrinsically low loading also gives the designer the freedom to choose higher-efficiency blade (and 'velocity-diagram') configurations

ADDED ADVANTAGES OF LOW-SPEED MULTISTAGE DESIGNS

The following are irrelevant to the arguments for high engine efficiency, but should be mentioned because otherwise the reader could be excused from wondering "why use three stages when one will do?"

1. Very low blade speeds lead to extremely low stresses (which are, other things being equal, proportional to the square of the blade speed.) The designer has the freedom, therefore, to consider using a low-cost high-temperature ceramic like mullite in the turbine rather than an extremely expensive type of silicon nitride. In the compressor, magnesium die-casting or injection-molded fiberglass-resin construction can be considered. These production methods have higher accuracy and lower costs than current manufacturing methods (such as precision casting). The overall manufacturing costs should be considerably lower for our three-stage design than for the one-stage design typical of other microturbines.

2. Foreign-object damage is reckoned to apply only to blade speeds above 485 m/s by one prominent researcher. Our tip speed is about 275 m/s.

3. Noise increases (and decreases) as the fourth or fifth power of the blade speed. Therefore noise

will be just about negligible, an important consideration in the market place.

4. The stored kinetic energy will be relatively small, so that a thin casing to contain rotor bursts will be adequate, giving an additional cost saving.

5. The rotational kinetic energy of three rotors going at slow speed is much less that that of one rotor going at a high speed. Hence rotor acceleration can be higher and load following will be enhanced.

6. The shaft RPM will be greatly reduced, for instance 16,000 RPM for the 300-kW engine vs. 45,000 RPM for a single-stage design. This low speed would reduce the need for two-stage gearsets in mechanical-drive applications, leading to further cost savings.

REFERENCES

[1] Jenkins, Peter E. (2001). “Review of the design analysis of the WTPI 300 kW engine,” June 20, 2001.
[2] Wilson, David Gordon and T.P. Korakianitis (1998). "The design of high-efficiency turbomachinery and gas turbines" second edition, Prentice-Hall, Upper Saddle River, NJ.
[3] Wilson, David Gordon (1997). "A new approach to low-cost high-efficiency automotive gas turbines," SAE paper 970234, Warrendale, PA
[4] Wilson, David Gordon and Andreas Carl Pfahnl (1997). "A look at the automotive-turbine regenerator system and proposals to improve performance and reduce cost," SAE paper 9270237, Warrendale, PA.
[5] Shimada, Kazuaki, et al (1993). "Advanced ceramics technology developed for industrial 300-kW CGT (ceramic gas turbine) research and development project in Japan," Paper no. 93-GT-188. ASME, NY, NY.
[6] Wilson, D. G. (1993). "Heat exchanger containing a component capable of discontinuous movement," US patent no 5,259,444, re-issued as US RE37,134 E in 2001, assigned to MIT, Cambridge, MA.