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Some Aerothermodynamic Anomalies Concerning the Design and Applications of Turbomachines

1. ABSTRACT

Aero derivative large gas turbines are often preferred as the low cost alternative to industrial gas turbine engines for a range of applications such as electrical power generation, driving compressors to pump gas in gas pipelines, water pumping stations and desalination. However, applications engineers often find it difficult to appreciate the full significance of terms such as adiabatic, polytropic and isentropic efficiencies of compression and expansion processes, variations of specific heats and the ratio of specific heats of the working fluid with temperature on the performance of gas turbine engines. This paper is an attempt to show the significance of these terms and their influence on the performance of gas turbine engines. The paper should be of particular interest to applications engineers who tend to be affected more seriously by the obsolescence of their acquired knowledge of the fundamentals of engineering than their counterparts in research, design and development of advanced gas turbine engines.

Key words: turbomachines, gas turbine engines, engineering, aerothermodynamics, efficiency.

2. INTRODUCTION

The gas turbine engine has been the centre of attention for continuous development through fundamental research since its inception. This was due to its importance for military applications as well as the rapid expansion of civil aviation. Therefore, it is not surprising that many papers have been written on the whole range of advanced topics such as the aerothermodynamics of the compressible flow through cascades of blades, structural analysis, combustion and heat transfer, high temperature materials, control system and avionics. Consequently, the design of the gas turbine engine has advanced from simple single spool configurations to very advanced multi-spool engines in less than a century. Now the aero derivative gas turbine engine offers a cost-effective alternative to the heavier and bulkier industrial gas turbine engine for such applications as electrical power generation, gas pipelines, water pumping stations and desalination. These applications have created highly competitive yet expanding new markets for aero derivative gas turbines. In order to take advantage of these opportunities, the applications engineer must have a sound knowledge of the fundamental operating principles and the performance of gas turbine engines.

It should be noted that due to rapid advances in science and technology the knowledge of the fundamentals that a fresh graduate acquires through formal degree-level education diminishes to half its value during the first three years[1,2,3,4]. The decay of acquired knowledge and the growth of practical experience are shown qualitatively in Figure 1.

In order to compensate for the obsolescence of acquired science and technology knowledge, practising engineers should take advantage of every opportunity to attend short courses at regular intervals and participate actively in professional conferences and seminars.

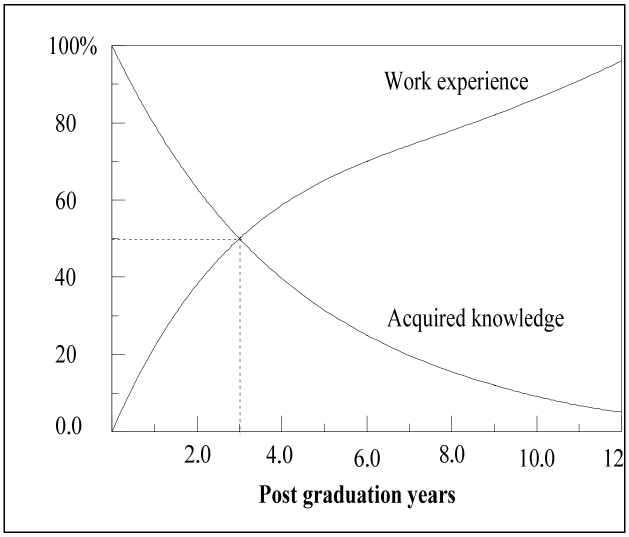

Figure 1: Decay of acquired engineering knowledge and growth of experience with time.

Not surprisingly, one sad consequence of the obsolescence of acquired scientific and technological knowledge is that we tend to misinterpret or misuse some of the basic concepts related to our particular field of interest. Therefore, occasionally it is necessary to pause and question the validity of these concepts and understand definitions in the light of the relevant physical principles.

Looking at our own area of interest, the gas turbine engine has made spectacular advances during the last seventy odd years since Sir Frank Whittle successfully demonstrated the gas turbine engine for aircraft propulsion. Since then maximum cycle temperatures have increased almost two and a half fold; percentage efficiencies have increased from low twenties to almost as high as mid forties. With rapid advances like these, engineers can no longer rely solely on their work experience; they must also have up-to-date knowledge of the fundamental principles of science and technology in order to use that experience profitably. In the remaining sections of this paper some of the basic concepts that are important for performance and applications are discussed, as far as possible, from first principles.

3. THE GAS TURBINE CYCLE

The gas turbine engine is an internal combustion heat engine which operates on the Joule-Brayton cycle. The practical cycle is shown qualitatively in the temperature-entropy diagram[5] in Figure 2.

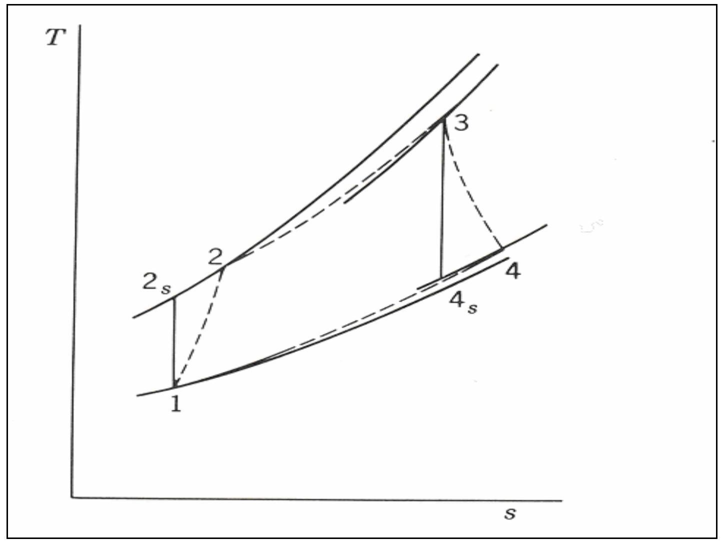

Figure 2: Thermodynamic operations in gas turbine engine.

Atmospheric air is compressed from state 1 to state 2 by the compressor. From state 2 to sate 3 the compressed air is heated in the combustion chamber by using the thermal energy released by the fuel as a result of combustion reaction. During combustion the stagnation pressure drops from P_{02} to P_{03}. High-pressure combustion gas at elevated temperature is then expanded in the turbine down to point 4. Engine output is represented by the difference between the expansion work and the compression work.

The cycle analysis is usually based on a number of simplifying assumptions but the following two are perhaps most significant:

a. The combustion and the expansion processes are adiabatic but not reversible. Since the through-flow velocities of the fluid in the compressor and the turbine are of the order of 100 m/s, the residence time for the fluid particles per metre length of either the compressor or the turbine will be approximately 10 milliseconds. This means that the heat transfer either to or from the fluid should be small. Hence, the assumption of adiabatic compression and expansion can be justified.
b. Specific heat c_p at constant pressure and the ratio of specific heats γ for air and combustion gas are assumed to be invariant with temperature. This is a highly

controversial issue. Many will argue that ignoring the variation of C_p and γ with temperature has only a minor effect on the results. Others say that for better accuracy it is very important to take this variation into account.

The effects of these two assumptions will be examined critically in the following section.

4. COMPRESSION AND EXPANSION EFFICIENCIES

Quite often one finds a great deal of confusion in the use of the terms adiabatic, isentropic and polytropic in connection with the efficiencies of the compression and the expansion processes. The thermodynamic definitions are given below.

For the compression process the isentropic efficiency is given by the following equation:

$$\eta_c = \frac{\left\{\left(\dfrac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}}{\dfrac{T_{02}}{T_{01}} - 1} \qquad (1)$$

The actual as well as the ideal compression processes are assumed to be adiabatic; the former is irreversible and the latter reversible. The reversible adiabatic, ie isentropic, process is used as the yardstick. Therefore the efficiency must be quantified with reference to the isentropic process and should be termed as the isentropic efficiency. The term adiabatic, although quite often used in old literature, is fundamentally wrong and its use should be discouraged.

Similarly for the turbine, the actual expansion process is assumed to be adiabatic but irreversible and the ideal expansion process as reversible. Therefore its efficiency should be calculated by comparing it with the isentropic process defined as follows:

$$\eta_t = \frac{(\Delta h_0)_{actual}}{(\Delta h_0)_{ideal}} = \frac{1 - \dfrac{T_{04}}{T_{03}}}{1 - (\dfrac{P_{04}}{P_{03}})^{\frac{\gamma-1}{\gamma}}} \qquad (2)$$

In the case of axial flow compressors, since the pressure ratio per stage is small, of the order of 1.18:1, quite often polytropic efficiency (also know as small stage efficiency) is used because it is independent of the number of stages. The polytropic efficiency is defined below.

For the compression process:

$$\eta_\infty = \frac{dh_0'}{dh_0} = \frac{c_p dT_0'}{c_p dT_0}$$

Rearranging this expression we get:

$$dT_0 = \frac{1}{\eta_\infty} dT_0'$$

Integrating this expression logarithmically between states (1) and (2):

$$\ln\frac{T_{02}}{T_{01}} = \frac{1}{\eta_\infty}\ln\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{or} \quad \frac{T_{02}}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\eta_\infty\gamma}}$$

Finally, the relationship between the polytropic and the isentropic efficiencies can be written as follows:

$$\eta_c = \frac{\dfrac{P_{02}}{P_{01}}^{\frac{\gamma-1}{\gamma}} - 1}{\dfrac{P_{02}}{P_{01}}^{\frac{\gamma-1}{\gamma\eta_\infty}} - 1} \qquad (3a)$$

Similarly, for the expansion process the polytropic efficiency can be defined as follows:

$$\eta_\infty = \frac{dT_0}{dT_0'}$$

$$or$$

$$\frac{dT_0}{T_0} = \eta_\infty \frac{dT_0'}{T_0}$$

As before, integrating this expression logarithmically between states (3) and (4):

$$\frac{T_{04}}{T_{03}} = \left\{\frac{P_{04}}{P_{03}}\right\}^{\frac{\eta_\infty(\gamma-1)}{\gamma}}$$

Finally, the relationship between the polytropic and the isentropic efficiencies can be written as follows:

$$\eta_t = \frac{1 - \dfrac{P_{04}}{P_{03}}^{\eta_\infty(\gamma-1/\gamma)}}{1 - \dfrac{P_{04}}{P_{03}}^{\gamma-1/\gamma}} \qquad \text{(3b)}$$

Sometimes polytropic and isentropic efficiencies are quoted as if they were synonymous terms. The buyers of turbomachines should take note that the differences in the polytrophic and isentropic efficiencies are quite significant as shown in Figures 3a and 3b.

Also it is worth observing that for the compression process the polytropic efficiency is greater than the isentropic efficiency and for the expansion process it is smaller. The reason is simply that for a constant γ the polytropic efficiency is a function of the polytropic index only. Since for the compression process the polytropic index is greater than γ, polytropic efficiency is greater than the isentropic efficiency and for the expansion process (polytropic index is less than γ), the polytropic efficiency is less than the isentropic efficiency.

In the case of the compression process, as pressure ratio is increased the isentropic efficiency reduces quite sharply at low values of the polytropic efficiency. In the case of the expansion process the isentropic efficiency increases at constant polytropic efficiency and increasing pressure ratio. However, the difference in both cases becomes gradually smaller with increasing values of the polytropic efficiency.

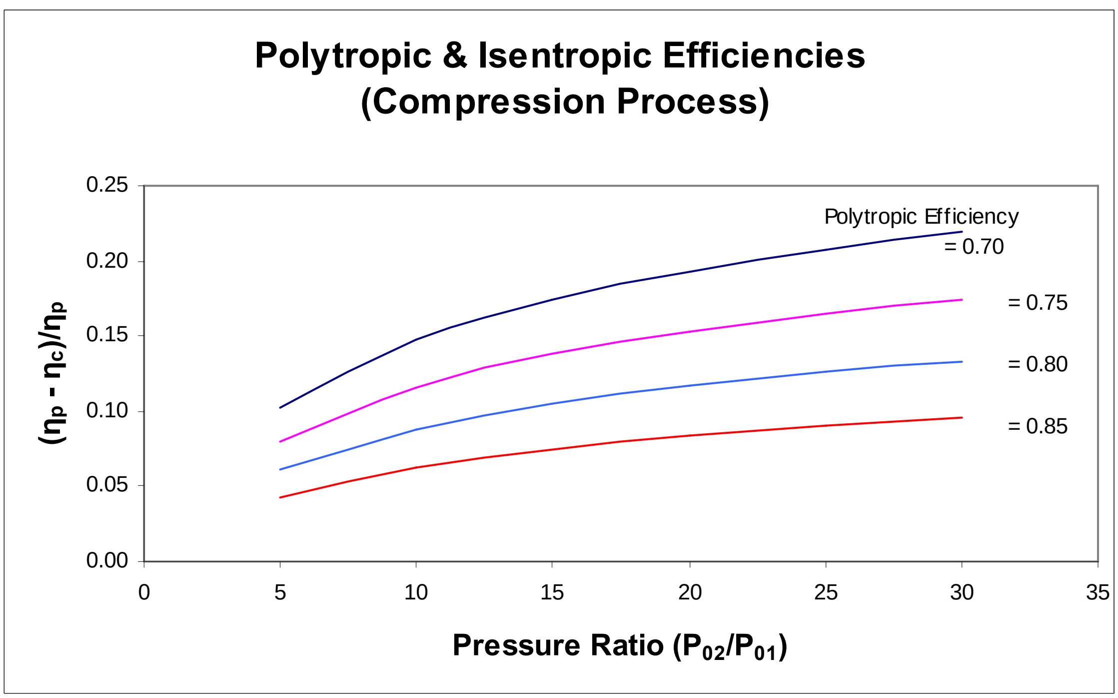

Figure 3a: Isentropic efficiency versus polytropic efficiency (compression).

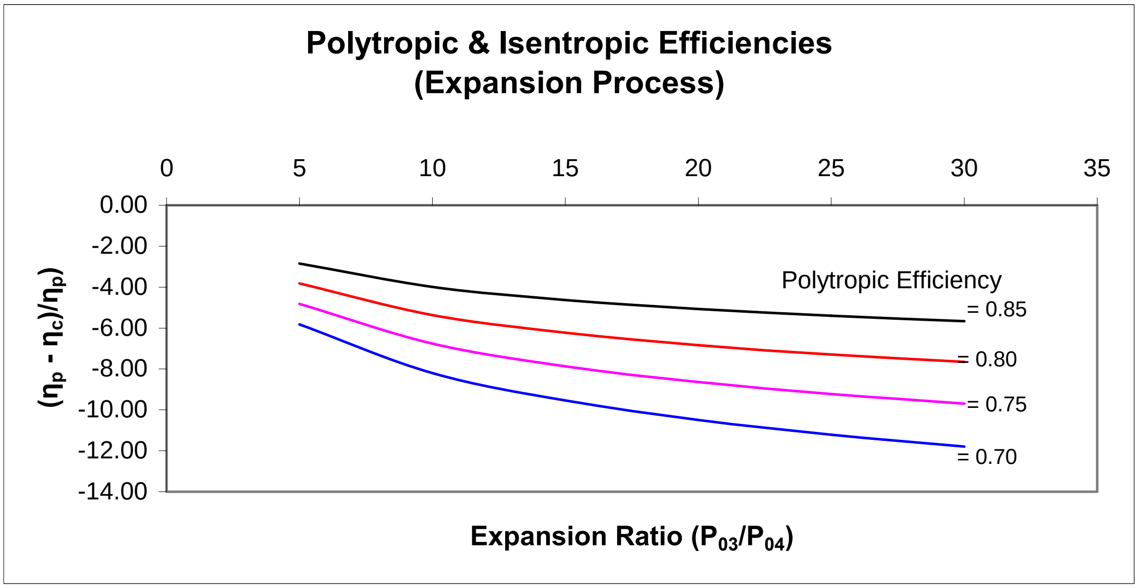

Figure 3b: Isentropic efficiency versus polytropic efficiency (expansion).

5. VARIATIONS OF THERMODYNAMIC PROPERTIES

The two most important thermodynamic properties of the working fluid for cycle calculations are the specific heat at constant pressure (C_p) and the ratio of specific heats (γ). For an ideal gas, C_p can be expressed in terms of γ by using the following simple relationships:

Since $R = C_p - C_v\,,\; \gamma = \dfrac{C_p}{C_v}$

and $R = \dfrac{\bar{R}}{M}\,;\; C_p = \dfrac{\bar{R}}{M}\left(\dfrac{\gamma}{\gamma - 1}\right)$ (4)

The variations of C_p and γ with temperature for air are shown in Figure 4a and those for combustion gas in Figure 4b.

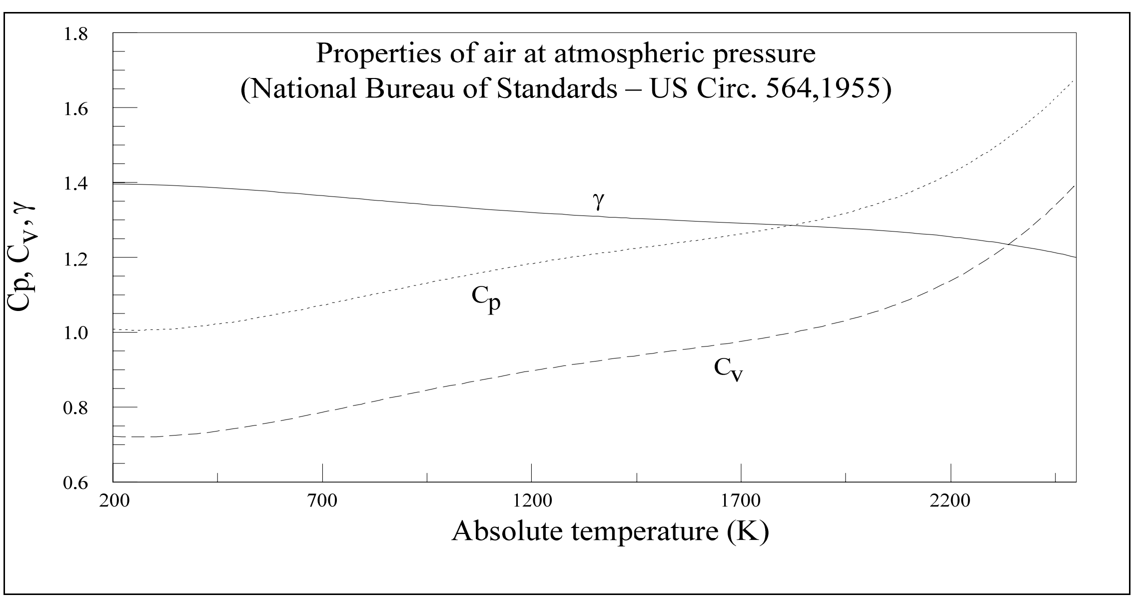

Figure 4a: Variations of C_p, C_v (kJ/kg K) and γ of dry air with temperature.

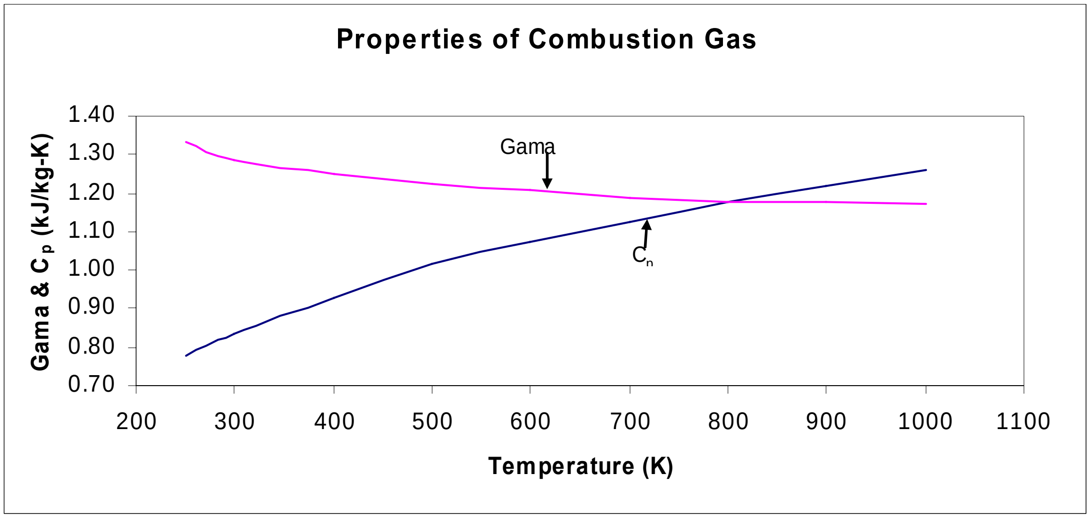

Figure 4b: Variation of C_p and γ of combustion gas with temperature.

It should be mentioned that both C_p and γ of combustion products depend on the air fuel ratio used (typical fuel air ratio value given on page 11). Figures 4a and 4b show clearly that variations of C_p as well as of γ with temperature are quite significant and should be taken into account when calculating the performances either of the complete engine or of compressors and turbines. Therefore it should not be surprising to find that numerous papers have been written on the methods of calculating C_p and γ and still this topic continues to attract the attention of research workers and theoreticians.

In gas turbine practice the operational maximum cycle pressure ratio seldom exceeds 25:1, but the designs of some modern industrial gas turbines aim for higher pressure ratios, of the order of 33:1. At present the average running pressure ratio is approximately 17:1. This corresponds to temperature ratio about 2.25. It should be noted further that the values of C_p and γ are assumed to be functions of temperature only; they are quite accurate over a wide range of pressures. The effect of variations of C_p and γ with temperature on pressure ratio and specific work output for the compression process are shown in Figures 5a and 5b respectively.

It is evident from Figure 5a that up to the compressor discharge temperature of approximately 600K and corresponding pressure ratio of approximately 15, the effect on pressure ratio as well as specific work input is hardly noticeable.

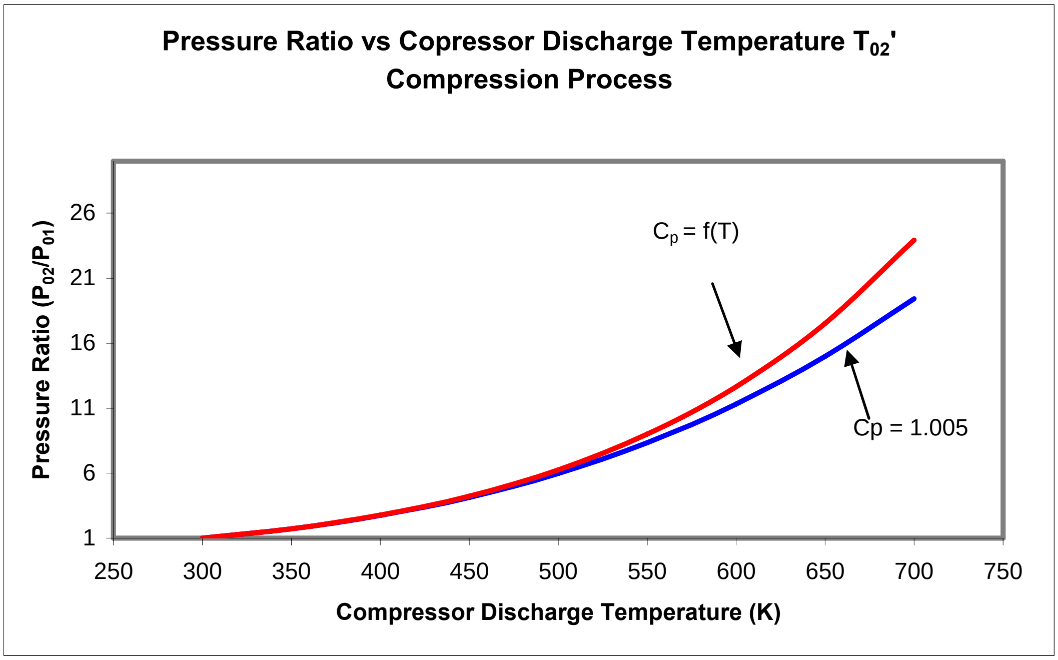

Figure 5a: Effect of varying C_p and γ on compressor pressure ratio.

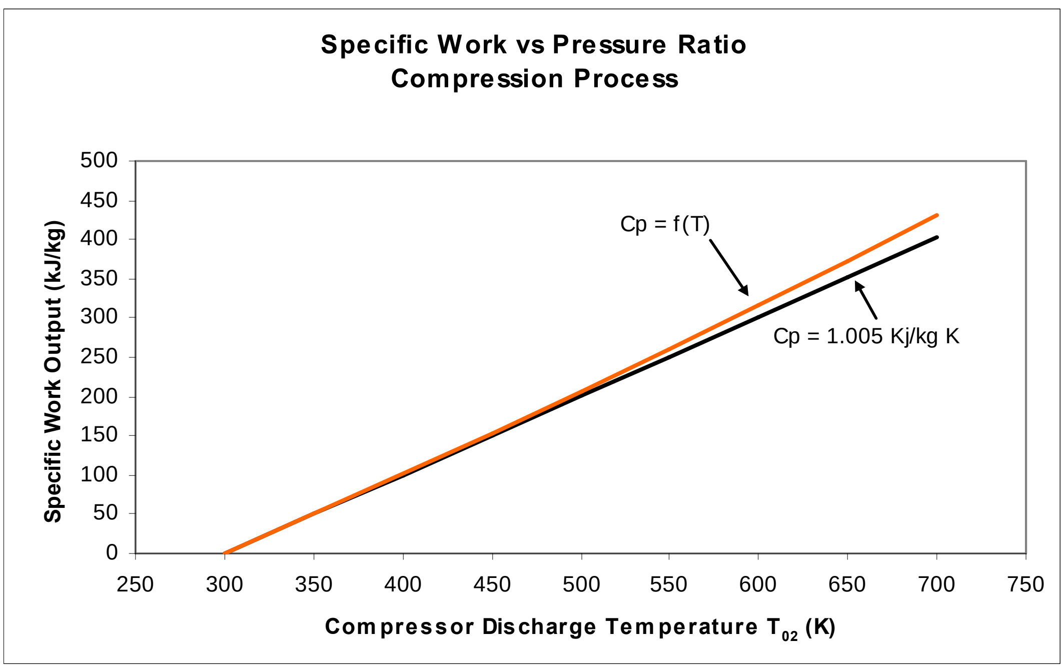

Figure 5b: Effect of varying C_p and γ on the compressor specific work.

Figures 5c and 5d show the effects of the variations of C_p and γ of combustion products with temperature on the turbine discharge temperature and the specific work output. Here again the effect is negligible over the practical range of turbine entry temperatures. These diagrams were plotted for the fixed expansion ratio of 10:1. However, calculations were also made for expansions ratios of 16:1 and 24:1. The results have not been presented in this paper for the sake of brevity but the observations from those results were not significantly different from what has been presented here.

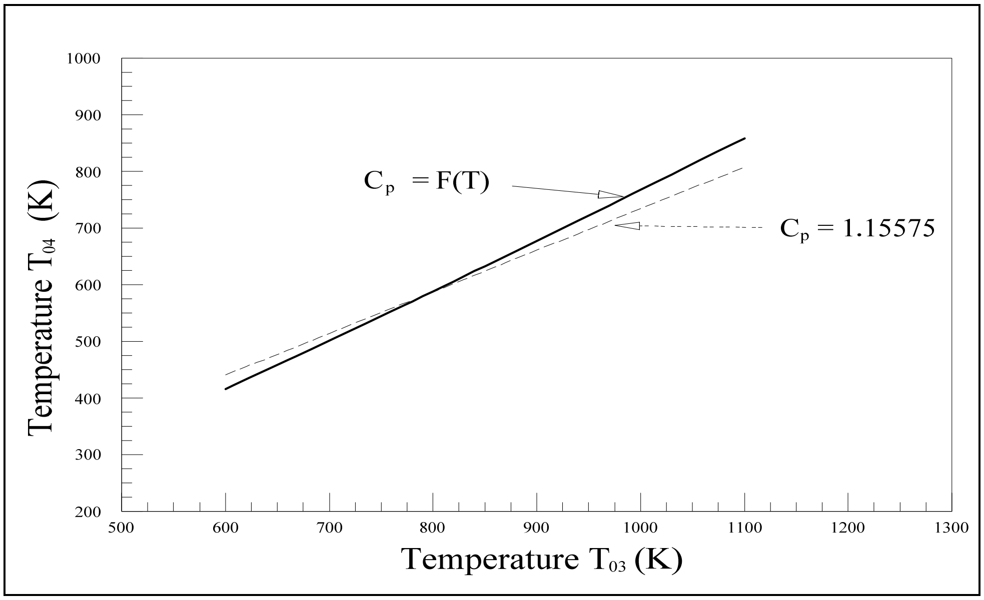

Figure 5c: Effect of varying C_p on turbine discharge temperature at the expansion ratio of 10:1.

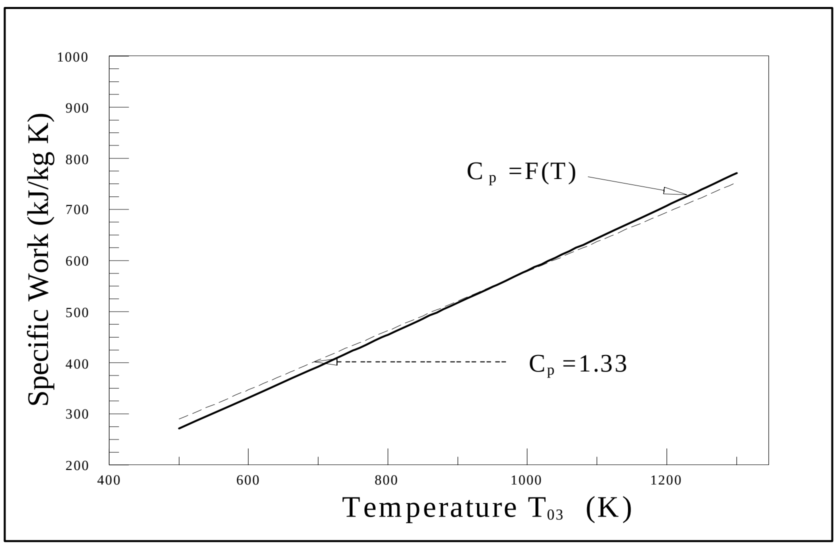

Figure 5d: Effect of varying C_p on turbine specific work output at the expansion ratio of 10:1.

These are simple but clear examples of ends not justifying the means. Perhaps it would not be wrong to say that performing gas turbine cycle calculations using elaborate computer programmes to evaluate the variation of C_p and γ with temperature are only for the sake of showing academic rigour. However, often analyses which take into account such insignificant factors as the influence of variations of C_p and γ with temperature on performance receive more acclaim than highly meritorious works of immense practical value, which ignore such factors but may concentrate on demonstrating the effects of geometrical variables on the performance of turbomachines.

6. MODELLING AND SIMULATION

Testing of either the components or the complete gas turbine engine for the purpose of obtaining performance characteristics can be a laborious and expensive exercise. Modelling is a very convenient method of testing the behaviour of large engineering systems of various levels of complexity. Two types of modelling techniques are widely used; they are:

a. Modelling one physical system by another physical system that is designed to satisfy certain similarity criteria. The later is a scaled-down replica of the former, for example, a scaled model of an aircraft, which is tested in the wind tunnel. The science and the art of scaling and testing of models are well developed, therefore the test data can be corrected for effects of scale, blockage, end wall boundary layer etc.
b. Modelling a physical system by using experimental data in order to simulate the physical system. Curve fitting techniques are used to derive mathematical equations to describe the performance of a system in terms of either its design variables or operating conditions. These equations are then programmed to run either on analogue or a digital computer in order to simulate the physical system.

Although modelling by scaling down physical systems has been in use for a long time, computer modelling is relatively recent and has become very popular during the last 40 to 50 years. However, it should be noted that the quality of simulation depends on the reliability of the experimental data that were used to develop the analytical model of the system.

Computer models of engineering systems such as the aero and industrial gas turbine engines are developed mostly by mathematicians and computer software specialists who, quite often, may lack proper understanding of the physical characteristics of the systems they are trying to model. Such models are semi-empirical and they be based on a limited amount of experimental data. The source of data quite often may not be traceable; for example, modelling the fuel system for a gas turbine engine from a single fuel delivery curve in order to develop the fuel control system. Quite often models developed on such unsound foundations are then used for optimising the real plant.

The gas turbine engine may be claimed to be the epitome of simplicity compared with its counterpart, the reciprocating engine, nevertheless the fluid mechanics of its components is by no means simple. Hence, reliable prediction of the performance characteristics by analytical means is very difficult. Moreover, it is not practical to represent the measured characteristics, particularly those of centrifugal or axial flow compressors, by a single line. A convenient method to overcome this difficulty is to model the complete engines by using the actual performance characteristics of the compressor and the turbine to prepare look-up tables. This approach has been used with reasonable success[6,7,8]; however, it must be mentioned that measured performance characteristics of engine components are not always readily available.

Simulation deals with using the model to reproduce the true characteristics of the physical system that has been modelled. Quite often, models developed from sparse experimental data are used to simulate real engines. Such simulation exercises only bring this valuable tool into disrepute. Reference must be made to extensive validation in order to ensure that simulation of the real system by means of a computer model is reliable.

7. CONTROL SYSTEMS FOR GAS TURBINE ENGINES

The control and monitoring systems for gas turbine engines would be designed to meet either all or a selection of the following conditions:

a. safety of operators
b. safety of plant
c. economy of operation
d. proper monitoring and control of exhaust gas pollutants
e. condition monitoring (continuous monitoring of the state of the engine) for the purpose of diagnostics and preventive maintenance

Functions a, b and d are mandatory and functions c and e may be optional. However, the gas turbine industry is under pressure to make these functions available in standard control equipment for gas turbine engines.

The control systems for gas turbine engines, as for any other control system, comprise three basic elements, namely:

a. the sensing elements, ie transducers
b. the decision-making element which functions in accordance with a set of stored criteria
c. actuating elements, eg stepping motors, which physically move the control elements

The sensors and actuators are mostly electro-mechanical devices and often incorporate certain moving parts. Therefore these elements are prone to develop faults more often than the electronics hardware, the so-called nerve centre or the brain of the control system. However, much of the research, design and development efforts have tended to concentrate on electronics hardware. Consequently duplex and triplex controllers have emerged as the industry's answer to achieving high values of mean time between failures (MTBF). Duplicating most if not all the transducers without affecting the structural strength of engine parts may not be practical. Better MTBF figures may be achieved by improving the design and construction of transducers to achieve greater precision of measurements or by significantly reducing human error and thus achieving leap-frog reliability similar to the aviation safety improvements achieved over the past century (see Figure 6)[9].

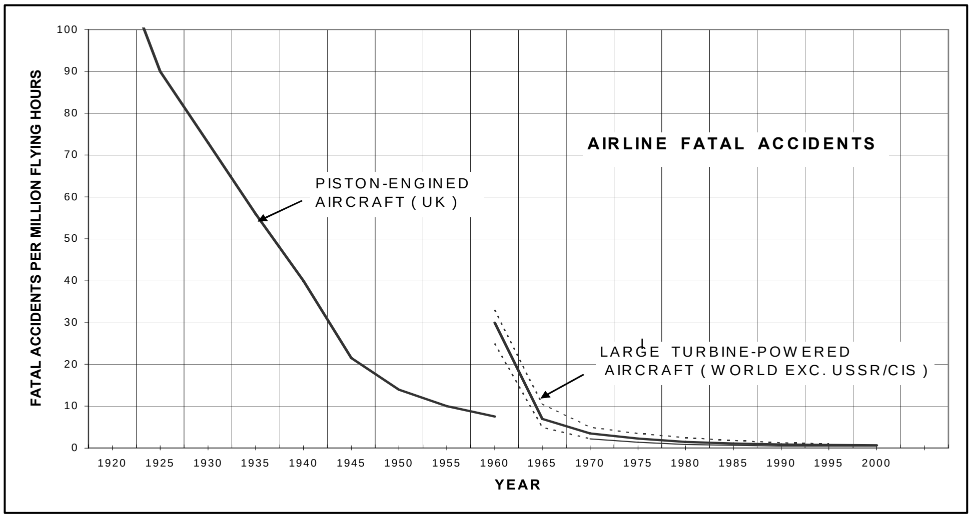

Figure 6: Accident rates per 10^6 flying hours, 1920–2000.

8. THERMAL EFFICIENCY

The precision with which the turbine entry temperature is controlled can have a significant effect on the thermal efficiency of the engine, which can be defined by referring to Figure 2 as follows:

$$\eta_{th} = \frac{\dot{m}_g C_{p_g}(T_{03} - T_{04}) - \dot{m}_a C_{P_a}(T_{02} - T_{01})}{\dot{m}_f \eta_b CV} \quad (5)$$

The left equation can be rewritten in a more convenient form for computation as follows:

$$\eta_{th} = \frac{C_{pa} T_{01}\left\{(1 + F/A)\frac{C_{pg}}{C_{pa}}\frac{T_{03}}{T_{01}}\eta_t\left[1 - \frac{P_{04}}{P_{03}}^{\frac{\gamma_g - 1}{\gamma_g}}\right] - \frac{1}{\eta_c}\left[\frac{P_{02}}{P_{01}}^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right]\right\}}{(F/A)\eta_b CV} \quad (6)$$

Equation (6) has been plotted in Figure 7 by assuming standard values for the following parameters:

C_{pa}=1.005	η_b= 98%	P_{02}/P_{01}=10:1
C_{pg}=1.155	η_c= 85%	P_{03}=0.95 P_{02}
γ_a=1.4	A/F= 70	P_{04}=P_{01}
γ_g=1.33	CV=43100 kJ/kg	T_{01}=300K

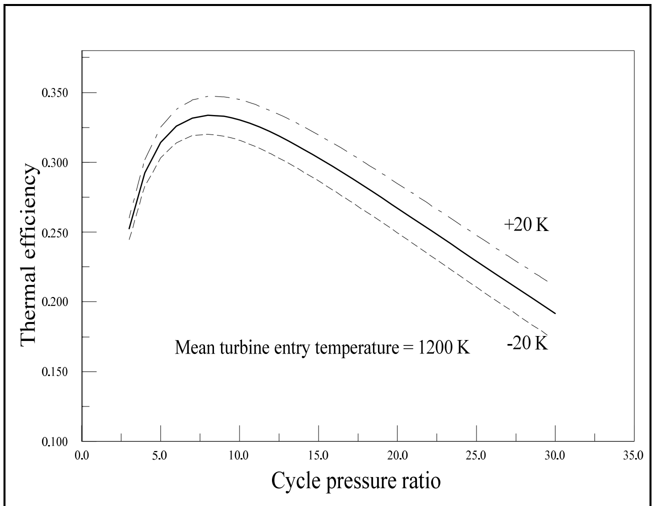

Figure 7: Effect of turbine entry temperature tolerance on the thermal efficiency of the engine.

It can be seen that thermal efficiency excursions can be quite significant for the turbine entry temperature tolerance of $\pm 20^0 C$ for a nominal temperature of 1200K. A tolerance of $\pm 20^0 C$ would be acceptable in the case of the analogue controllers. Digital systems can control turbine entry temperatures to much closer tolerances. If fuel economy was the prime consideration, which is the case these days, then the digital versus analogue debate must weigh heavily in favour of the former because it can guarantee the operating thermal efficiency of the engine close to the desired nominal value.

9. CONCLUDING REMARKS

1. In order to guard against malpractices resulting from such anomalies as referred to in the paper, it is very important to update fundamental knowledge on a regular basis, preferably at three-yearly intervals, through continuing education. This is necessary to maintain the scientific credibility of the practical experience.
2. There are many sources of information available to keep engineers current; all that it takes is the desire to learn, professional investment or employer support by following a policy of investing in people.
3. The obsolescence of engineering knowledge, which is the primary cause of certain anomalies that tend to creep in the design and analysis practice of gas turbine engines, might be avoided if every so often recourse was made to refresh the fundamental principles of the important fields engineering.

10. NOTATION

C_p	specific heat of fluid at constant pressure (kJ/kg K)
C_v	specific heat at constant of a fluid at constant volume (kJ/kg K)
F/A	fuel air ratio
Δh_0	change in stagnation enthalpy of a fluid (kJ/kg)
$\bar{M}$	molecular weight of a fluid
MTBF	mean time between failures
P	static pressure (bar on kN/m^2)
P_0	stagnation pressure (bar on kN/m^2)
R	gas constant (kJ/kg K)
$\bar{R}$	universal gas constant (kJ/kg K)
s	specific entropy (kJ/kg K)
T	static temperature (K)
T_0	stagnation pressure (K)
γ	ratio of specific heats
η_c	isentropic efficiency of compression process
η_t	isentropic efficiency of expansion process
η_p	polytropic efficiency

Subscripts

a	air
f	fuel
g	gas
gama	ratio of specific heats

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