Introduction to **Gas Turbine Theory**

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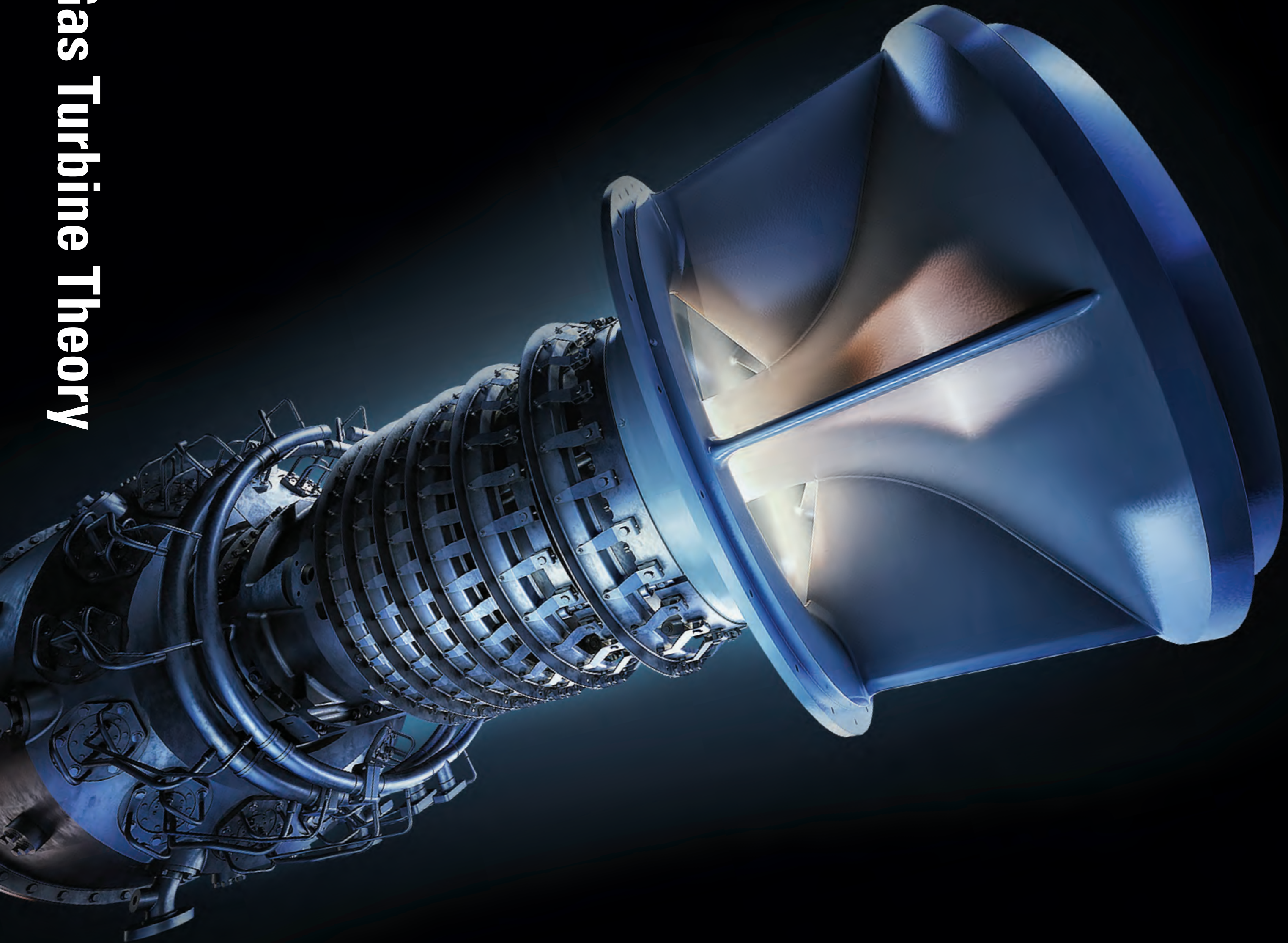

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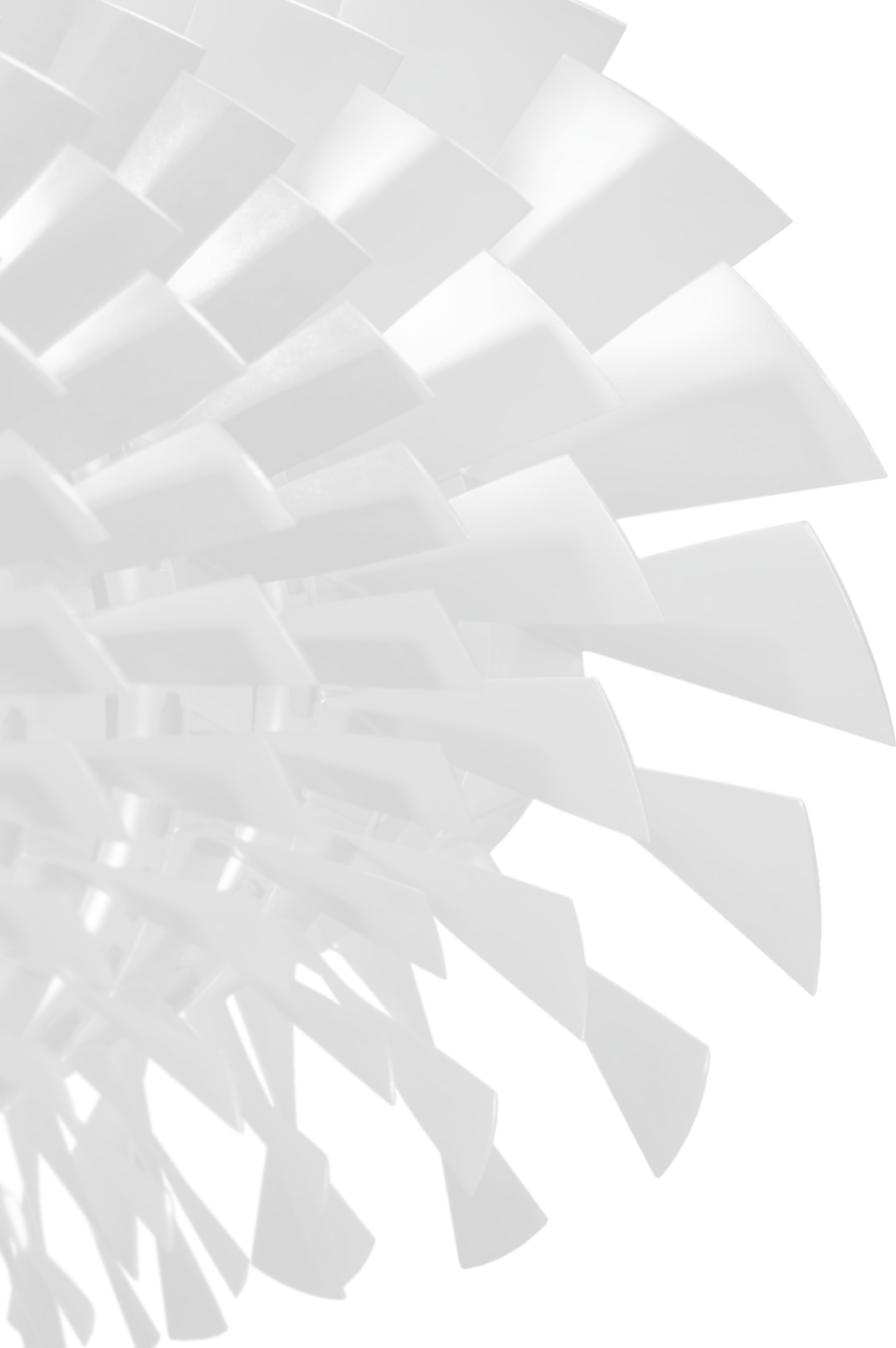

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ABOUT THE AUTHORS

Klaus Brun – Dr. Brun is the Director of the Machinery Program at Southwest Research Institute. His experience includes positions in engineering, project management, and management at Solar Turbines, General Electric, and Alstom. He holds seven patents, authored over 150 papers, and published two textbooks on gas turbines. Dr. Brun won an R&D 100 award in 2007 for his Semi-Active Valve invention and ASME Oil & Gas Committee Best Paper/Tutorial awards in 1998, 2000, 2005, 2009, 2010, 2012, and 2014. He was chosen to the "40 under 40" by the San Antonio Business Journal. He is the past chair of the ASME-IGTI Board of Directors and the past Chairman of the ASME Oil & Gas Applications Committee. He is also a member of the API 616 and 692 Task Forces, the Middle East and Far East Turbomachinery Symposiums, the Fan Conference Advisory Committee, and the Supercritical CO_2 Conference Advisory Committee. Dr. Brun is the Executive Correspondent of Turbomachinery International Magazine and an Associate Editor of the *ASME Journal of Gas Turbines for Power*.

Rainer Kurz – Dr. Kurz is Manager of Systems Analysis and Field Testing for Solar Turbines Incorporated in San Diego, California. His organization is responsible for predicting gas compressor and gas turbine performance, as well as conducting application studies and field performance tests on gas compressor and generator packages. Dr. Kurz attended the University of the Federal Armed Forces in Hamburg, Germany where he received the degree of a Dipl.-Ing. and, in 1991, the degree of a Dr.-Ing. He has authored over 100 publications in the field of turbomachinery and fluid dynamics, holds two patents, and was named an ASME Fellow in 2003. He is a member and former chair of the ASME Oil and Gas Applications Committee, a member of the Turbomachinery Symposium Advisory Committee, the Gas Machinery Conference Organizing Committee, the GMRC Project Supervisory Committee, and the SDSU Aerospace Engineering Advisory Committee. Many of his publications are recognized as being of archival quality, and he received numerous Best Paper awards, as well as the ASME Industrial Gas Turbine Award in 2013.

DEDICATION AND ACKNOWLEDGMENTS

We dedicate this book to our loving and supportive wives.

Marisol & Renee

This text would not have been possible without the contributions of many people. First of all, we would like to thank our colleagues at Solar Turbines and Southwest Research Institute for their editorial, as well as technical suggestions. Finally, the completion of this work stands on the shoulders of many colleagues and friends:

To you, a simple thanks is offered!

PREFACE

This textbook was developed directly from a series of Solar Turbines Incorporated internal short courses that were presented to an audience with a wide range of technical backgrounds, not necessarily related to turbomachinery. Thus, functional principles and physical understanding are emphasized, rather than the derivation of complicated mathematical equations. The aim of this short course on gas turbine theory is not to provide an in-depth knowledge of gas turbine aerodynamics or thermodynamics, nor is it intended to make the reader an expert in the field of turbomachinery.

The course is, however, intended as a summary and brief overview of the many different topics and theories that pertain to the subject matter. Throughout the text, the reader is provided with simplified explanations of complex physical theories and is then, if interested, encouraged to further investigate a topic by studying more specific literature about the subject matter. Also, by following this text, the reader will hopefully develop an appreciation of the many disciplines of engineering that are involved in the design and analysis of gas turbines.

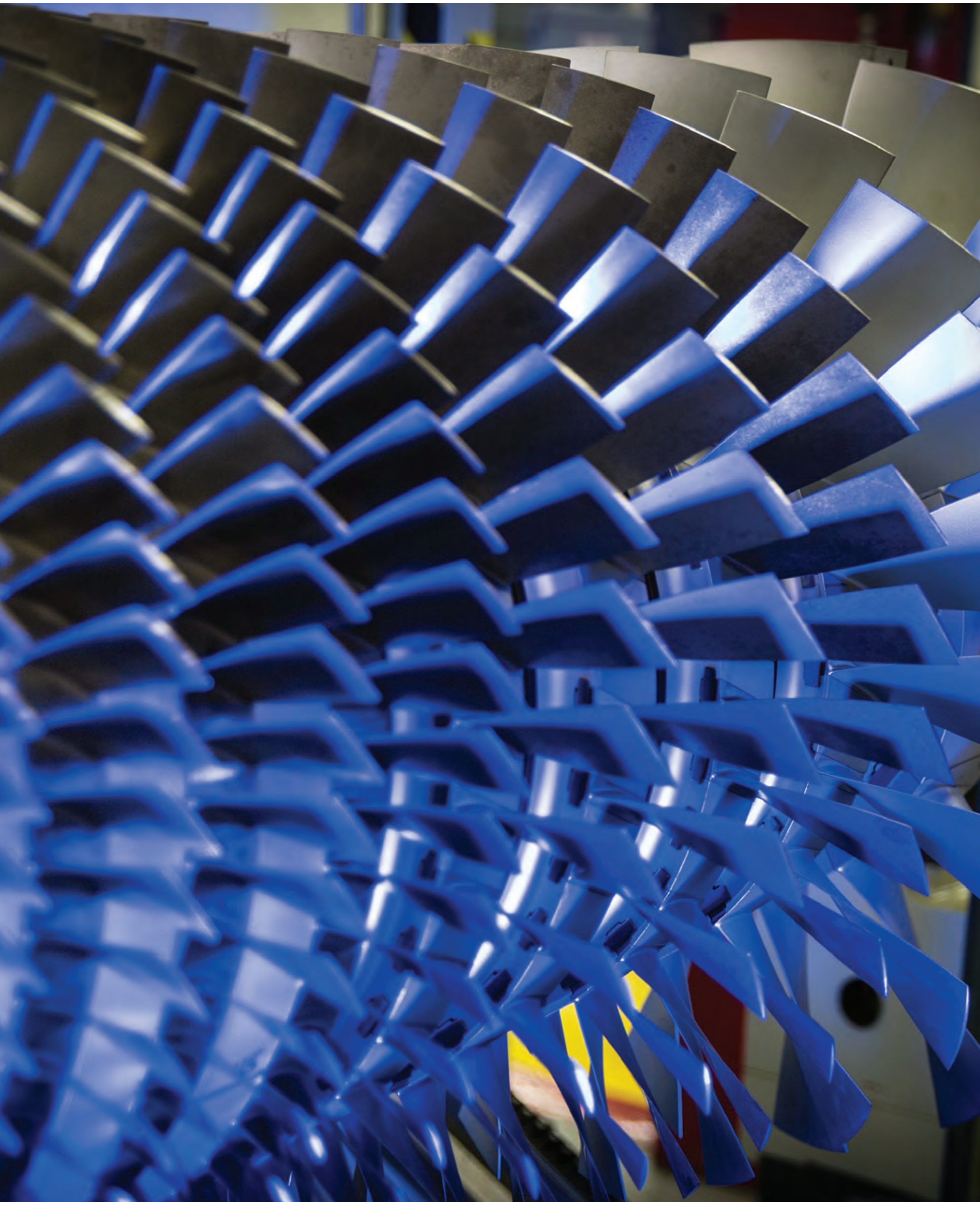
The adjacent nomenclature reference shows symbols, acronyms and subscripts commonly used in discussing gas turbine thermodynamics.

NOMENCLATURE

A	throughflow area
a	mechanical acceleration
a	speed of sound
c	mechanical damping
c	velocity vector in stationary frame
c_p	specific heat
E	Young's modulus
E	energy
e	eccentricity
$[F]$	unbalanced force vector
f	force
H	head
h	enthalpy
I	moment of inertia
k	mechanical stiffness
L	length
M	Mach number
M_n	machine Mach number
m	mass
N	rotational speed in rpm
n	number of stages
P	power
p	stagnation pressure
Q	volumetric flow rate
$[Q]$	influence coefficient matrix
q	heat flow
q_R	lower heating value
R	specific gas constant
Re	Reynolds number
R_m	universal gas constant
RMU	rotor mass unbalance
r	blade radius
SM	separation margin
s	blade span
s	entropy
T	temperature
U	blade velocity
u	tangential direction
v	velocity
v	specific volume
W	weight
W	mass flow rate
w	velocity vector in rotating frame
X	work
$[X]$	unbalanced response vector
x	displacement
Z	force response
α	velocity vector angle in stationary frame
β	velocity vector angle in rotating frame
δ	referenced pressure $= p/pref$
γ	specific heat ratio $= c_p/c_v$
Δ	difference
η	efficiency
θ	Temperature Ratio $= T/T_{ref}$
μ	viscosity
ν	kinematic viscosity
ζ	damping factor
π	pressure ratio
ρ	density
τ	torque
ϕ	phase angle
ω	rotational speed in rad/sec $= 2\pi N/60$
ω_N	natural frequency
ω	frequency of vibration

Subscripts

0	ambient
1	at engine inlet
2	at engine compressor exit
3	at turbine inlet
5	at power turbine inlet
7	at engine exit
a	axial
c	compressor
f	fuel
M	mechanical
t	turbine
u	circumferential



INTRODUCTION

This book is divided into two sections. Chapters 1 through 5 are intended to provide the reader with a very basic introduction to the analysis, behavior and design of industrial gas turbines. Chapters 6-12 cover a selected list of significantly more complex subjects on gas turbine performance and are intended for readers with a more advanced knowledge of turbomachinery systems.

INTRODUCTION TO GAS TURBINE THEORY

The basic introduction is divided into five chapters: Gas Turbine Thermodynamics and Aerodynamics, Turbomachinery Component Basics, Modern Analysis Methods, Rotordynamics and Vibrations, and Ancillary and Auxiliary Systems.

Gas Turbine Thermodynamics introduces some of the basic thermodynamics of the simple gas turbine open cycle, also referred to as the Brayton cycle. Studying the Brayton cycle thermodynamics provides you with an overall picture of gas turbine performance and an understanding of the function and principal design of each individual gas turbine mechanical component, including: the inlet system, compressor, combustor or burner, gas producer turbine, power turbine, and exhaust. Some thermodynamic concepts such as conservation of energy, entropy, enthalpy, T-s/P-v diagrams, isentropic and isobaric processes and adiabatic efficiencies will be introduced. The basics of Aerodynamics are discussed in this section, as well.

Once thermodynamic design and function are established, more detailed explanations are provided for each gas turbine component, and the internal flow aerodynamics, combustion chemistries and mechanical/operational limitations will be studied. Some classic mathematical methods, such as Euler's momentum equation, velocity polygons and combustion stoichiometry will be described. These methods are often used in the conceptual gas turbine design process, and some simplified examples are provided. Gas turbine rotordynamics and vibrations are also discussed.

After understanding the challenges of gas turbine component design and performance, some of the modern technologies that are being employed to improve overall gas turbine performance will be briefly discussed. The focus of this introductory chapter is the application of computational fluid dynamics analysis to turbomachinery flows.

This is followed by a brief overview of the fundamental concepts of rotordynamics.

The packaged systems for an industrial gas turbine are designed to provide the installation with its utility requirements (air, fuel, oil, and water), control the operation of the unit and process, and ensure safety.

ADVANCED TOPICS OF GAS TURBINE PERFORMANCE

While chapters 1 to 5 provide the foundation of gas turbine physics; chapters 6-12 describe in more detail, methods and equations that are employed to determine the actual performance of gas turbines under varying operating conditions. Also, gas turbine performance maps, design characteristics and combustion emissions are discussed in significant detail. Some important issues involving the operation of gas turbines — such as air filtration, fuel capability, and performance degradation — are also discussed in greater detail.

Finally, an outlook on advanced gas turbine cycles is provided.

GAS TURBINES – A BRIEF HISTORY

The invention and early development of gas turbines were clearly driven by airplane aeronautics, jet engine propulsion research and the development of industrial power machinery. Employing a jet engine as a prime mover for compressors, pumps and generators was initially a secondary application to the development of the jet engine for flight. Therefore, when looking at the history of the gas turbine, we must also study the history of the jet engine.

In 200 B.C., the Egyptian philosopher Hero of Alexandria invented a device that demonstrated the physical principles of a simple turbine. Leonardo Da Vinci, the multi-talented medieval Italian philosopher, sketched a "smoke stack" gas turbine in 1550. The first actual patent for a type of gas turbine engine design was given to John Barber of England in 1791. It contained all elements of a modern gas turbine including a reciprocating compressor, combustor, and turbine (Figure 0-1).

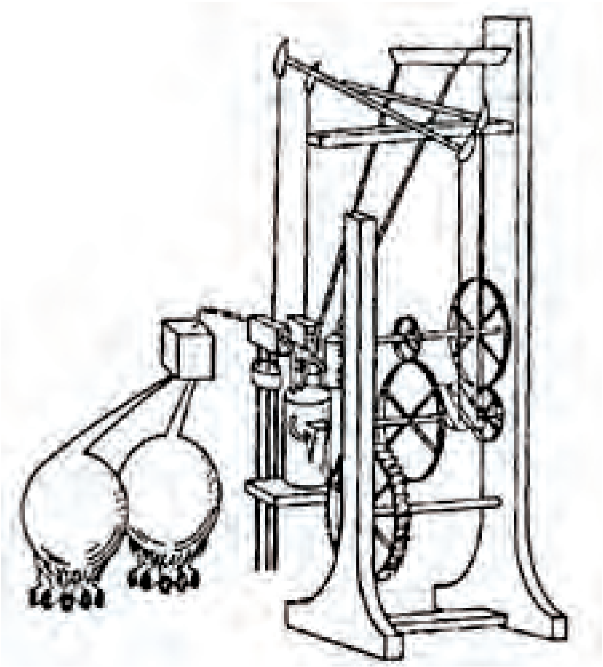


Figure 0-1. *John Barber's patent for a gas turbine engine from 1791.*

The Brayton or gas turbine cycle involves compression of air (or another working gas), the subsequent heating of this gas (either by injecting and burning a fuel or by indirectly heating the gas) without a change in pressure, followed by the expansion of the hot, pressurized gas. The compression process consumes power, while the expansion process extracts power from the gas. Some of the power from the expansion process can be used to drive the compression process. If the compression and expansion process is performed efficiently enough, the process will produce useable power output. This principle is used for any gas turbine, from early concepts by Bresson (1837); F. J. Stolze (in 1899); C.G. Curtiss (in 1895); S. Moss (in 1900); Lemale and Armengaud (in 1901); to today's jet engines and industrial gas turbines:

In 1837 M. Bresson patented a machine that essentially resembles the layout of modern gas turbines. Similarly, Franz Stolze of Germany designed, but failed to build, a gas turbine in 1872. At that time, neither the technology, manufacturing capability, nor materials were available to actually build a functioning jet engine. In the first decade of the 20th century Franz Stolze in Germany and R. Armengaud and C. Lemane in France built industrial gas turbines based on design technologies developed for steam turbines. Although both machines ran, neither was capable of producing net positive output power.

Several significant technology improvements for combustion and axial compressors followed until the 1930s. For example, in 1913, Rene Lorin designed a jet engine that closely resembles today's RAM-jet. Lorin also failed to construct a functioning prototype. During that time, other significant designs and patents for industrial gas turbines came from Charles Parsons (1884); Charles Curtiss (1895); Sanford Moss (1900); Charles Lemal (1901); and Tom R. Sawyer. Clearly, the idea of a gas turbine is not a new concept, but development had to wait until satisfactory materials could be obtained and manufacturing techniques developed.

The first truly functioning industrial gas turbine power plant was built in 1936 by Brown Boveri Company in Switzerland and then installed at Markus Hook refinery near Philadelphia to drive a process gas compressor. It utilized many technologies from steam turbines, but its essential layout is that of a modern industrial gas turbine with both axial compressor and turbine chapters. The first power plant to utilize a gas turbine was built and tested by Brown Boveri in 1939 and later installed near Neuchatel, Switzerland. It produced net 4,000 kW of electric output power at an efficiency of 18 percent.

The real technology thrust that made today's gas turbines possible came from jet engine development during the Second World War. Sir Frank Whittle of England and Dr. Hans von Ohain of Germany are both credited with the real invention of the jet engine in the 1930s. While Whittle patented the first feasible jet engine design in January 1930 and built a working model in 1935, it was von Ohain who, independently and completely without the knowledge of Whittle's work, built a similar engine and employed it for actual aircraft propulsion on the Heinkel 178 jet airplane in August 1939 (Figure 0-2).

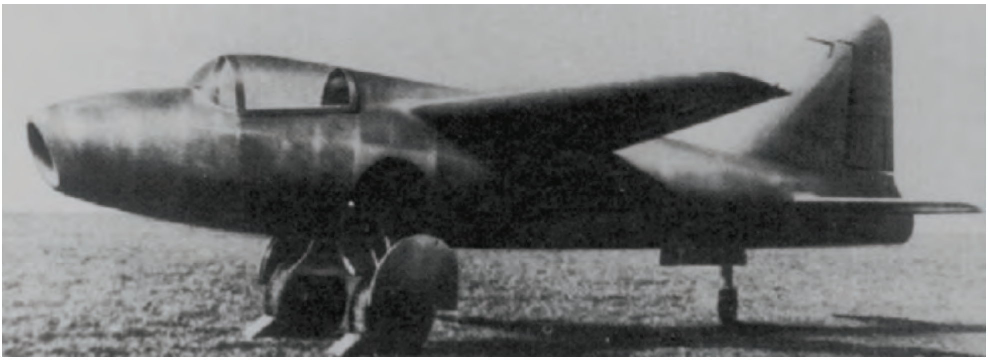


Figure 0-2. *Heinkel 178 Airplane*

Whittle's and Ohain's developments were primarily driven by the need for faster flight of military aircraft during World War II. America's first jet engine was also developed during the war in 1942 and was a relatively small engine with only 1300 lb. thrust. After World War II, the aerospace industry continued to lead the research and development of jet engines; however, a secondary industry — employing the jet engine as a gas turbine prime mover — arose quickly. Since then, jet engine and gas turbine research have continued in parallel.

The first prime mover gas turbines were what are commonly called "aeroderivatives." These engines were developed from existing jet engines, but with slightly modified designs. It is difficult to determine which company manufactured the first prime mover gas turbine, but early aeroderivative and industrial gas turbines were commercially available in the 1950s from several engine manufacturers. Soon thereafter in the 1960s, Solar Turbines Incorporated began building gas turbines that were not derived from a jet engine, but were a completely new design dedicated solely to prime mover applications (Figure 0-3). This was found to be an inherently better approach, since jet engines are designed for lightweight and short life span while industrial gas turbine prime movers require ruggedness and a long life. Since then, Solar has continued to lead in the development of small to mid-size industrial gas turbines.

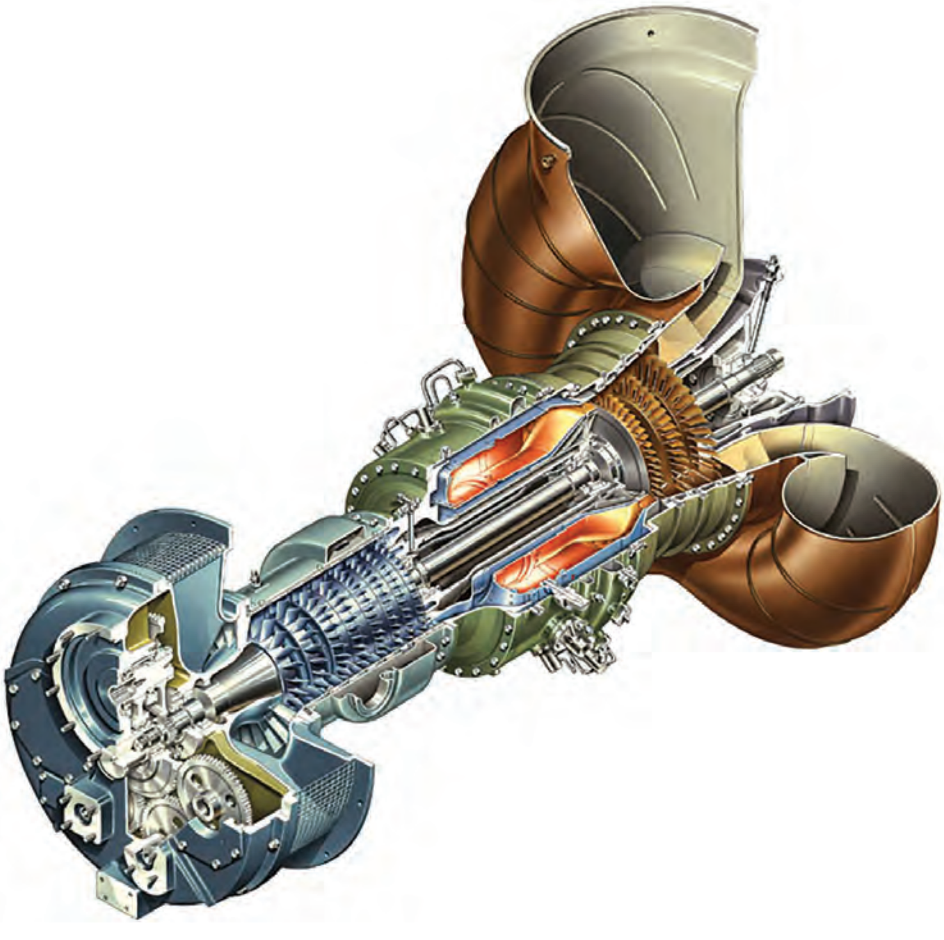
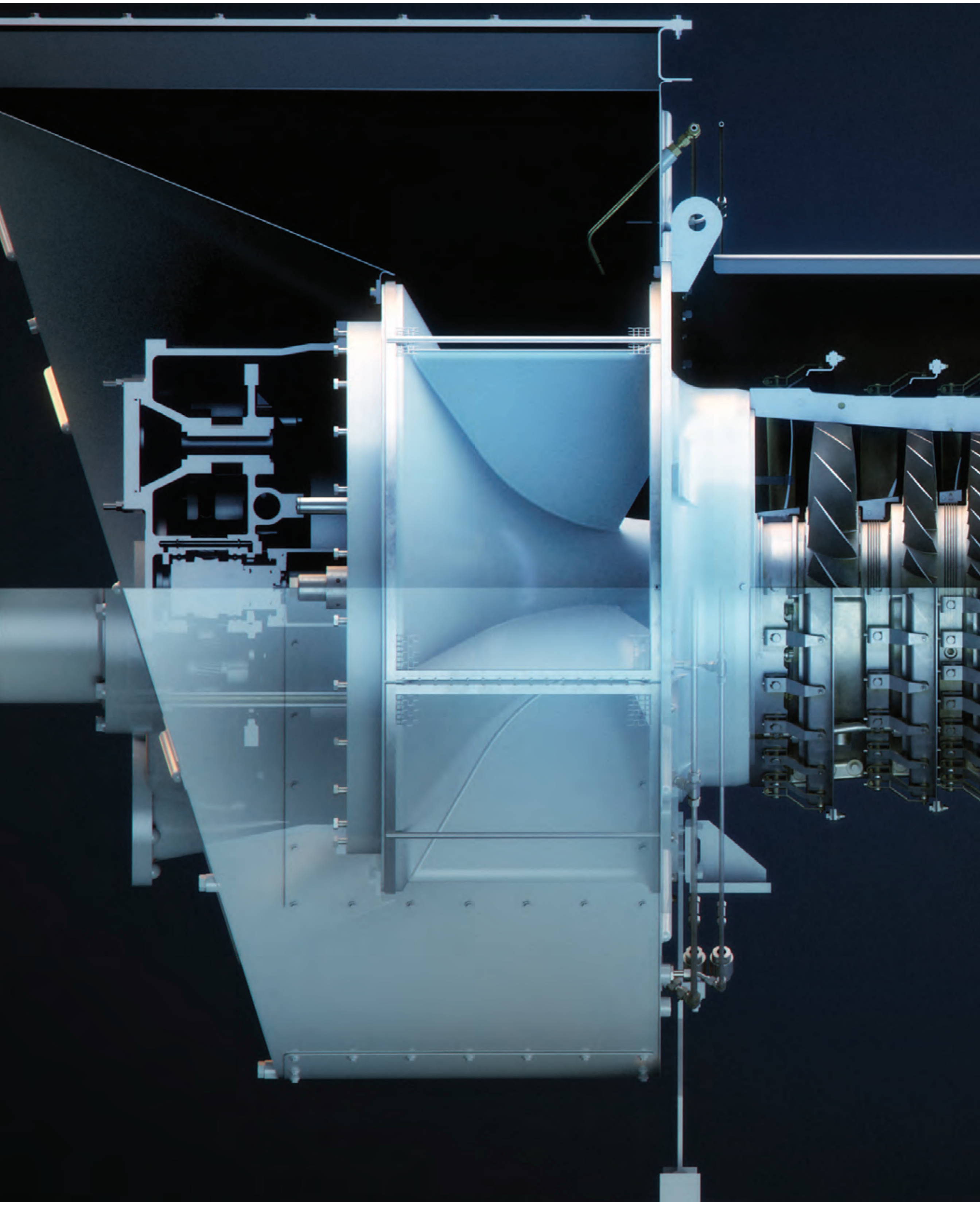


Figure 0-3. *Saturn Gas Turbine*



CHAPTER 1

GAS TURBINE THERMODYNAMICS

WHAT IS A GAS TURBINE?

Before providing detailed information about turbomachinery thermodynamics, it is necessary to develop a basic definition of a gas turbine. *Figure 1-1* presents an illustration from a 1945 textbook about gas and steam turbines. Essentially, the illustration shows an electric fan blowing air; the fan's air passes through a firebox (combustor) and the exhaust air from the firebox is driving another fan (turbine). The author's intent was to demonstrate a "gas turbine" process that is supposed to be conceptually easy to understand. Unfortunately, what is actually shown in the illustration is over simplistic, incorrect and does not demonstrate an actual gas turbine process.

To design a gas turbine, it is not enough to simply drive a turbine by blowing hot air on it. By definition, the thermodynamic process that has to take place to make the above simple aerodynamic process into a real thermodynamic gas turbine cycle is a density change of the working fluid, i.e., compression and expansion of the air. Let us analyze this statement in more detail.

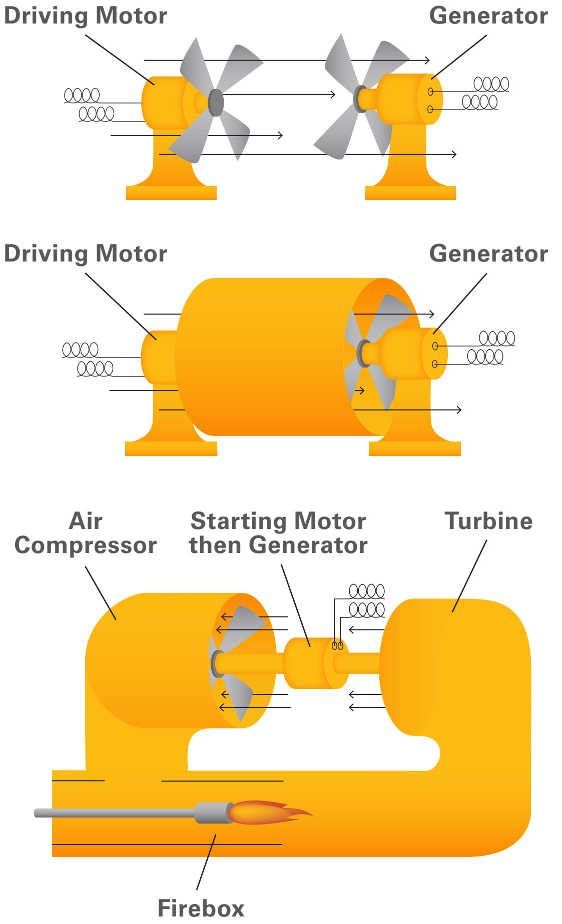


Figure 1-1. *Early Demonstration of Gas Turbine Operation (R. Tom Sawyer, "The Modern Gas Turbine" 1945)*

A gas turbine's function is to provide mechanical shaft output power to drive a pump, compressor or an electric generator. Within the gas turbine, chemical energy (fuel) is converted to heat energy, which in turn is converted to rotational shaft (mechanical) energy. The working fluid that is employed for the conversion of heat energy to mechanical output energy is simply the air that flows through the gas turbine. Air is ingested into the gas turbine by a compressor, heated in the combustor and expanded to drive a turbine. If this air maintains a constant density throughout the process, as well as a constant volume, then the maximum energy a turbine can extract from the air is exactly the same energy that was added by the compressor, regardless of how much the air is heated. Thus, no additional shaft output energy can be gained and, effectively, this would just be a very inefficient air-heater design.

On the other hand, if there is a density change of the air in the gas turbine process, the volume of air that drives the turbine can be greater than the volume that was compressed by the compressor; therefore, more energy can be extracted from the air in the turbine than was introduced by the compressor. Intuitively, we expect the increase in energy of the air between the compressor and turbine to be somehow related to the combustor heat addition. This concept will be discussed in more detail, but it is important to understand the following:

A gas turbine process must, by definition, involve density changes (compression and expansion) of the working fluid (typically air).

SIMPLE GAS TURBINE SYSTEM

Keeping in mind what was said previously about compression and expansion, let us take a quick look at a functional schematic of the most commonly employed gas turbine process *(Figure 1-2)*. This thermodynamic process is called the simple, open gas turbine cycle, often referred to as the Brayton cycle. In this part, we will focus only on the Brayton cycle. More advanced thermodynamic process cycles beyond the simple Brayton cycle are only briefly discussed in later chapters. One should note that more than 95% of all small to mid-size gas turbines deployed in industrial applications are based on a simple open Brayton cycle design. The study of non-Brayton advanced cycles is discussed in Chapter 12.

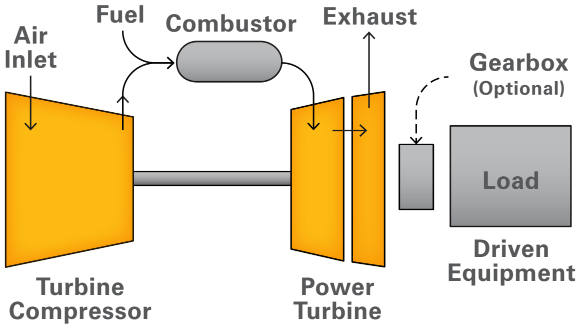


Figure 1-2. *Simple Brayton Cycle*

The Brayton gas turbine cycle consists of three principal processes: fluid compression, heat addition (combustion) and expansion.

In a typical industrial Brayton cycle, gas turbine application ambient air, approximately 300°K, enters into an axial compressor and is compressed (density increase) by a pressure ratio (P_{out}/P_{in}) typically between 5 and 30. This compression of the air also increases the air's temperature to between 500°K and 900°K. (The temperature and pressure of air are related via the isentropic relationship, $P_1/P_2=(T_1/T_2)^{\gamma/\gamma-1}$, which will be briefly covered later in this text.)

The compressed and hotter air then enters into the axial combustion chamber or burner. In the combustion chamber, chemically stored fuel energy is converted to thermal energy (heat), which is used to further increase the air temperature to around 1,300°K to 1,800°K. In most gas turbine combustors, natural gas (a mix of methane, propane, butane, pentane, carbon dioxide or nitrogen) or a liquid hydrocarbon fuel such as diesel or kerosene is burned with the process air. The very hot and compressed air exits the combustor and then expands (density decrease) to drive the axial turbine. Thus, the energy that was added to the air in the compressor and combustor is taken out in the turbine.

As can be seen in *Figure 1-2*, the turbine and compressor are mounted on the same shaft. Some of the energy that the turbine extracts from the expanding process air is being used to drive the compressor. The remaining energy the turbine extracts from the expanding air is effectively the gas turbine shaft output power. One would expect that since the turbine and compressor are on the same shaft, by a simple balance of power, the output shaft power must be proportional to the temperature increase of the air in the combustor; i.e., the combustor heat addition.

Figure 1-3 demonstrates how the individual components are incorporated in an actual gas turbine. Principal gas turbine functional components i.e., the multi-stage axial compressor, annular-axial combustor and multi-stage axial turbines are easily identifiable. The term "axial" indicates that the airflow enters and exits the compressor, combustor and turbine primarily along the axial (gas turbine shaft) direction.

Several gas turbine designs employ centrifugal compressors rather than axial compressors, and radial in-flow turbines rather than an axial turbine. Some of these component design variations will be discussed and compared later; however, regardless of the component design, the basic gas turbine Brayton simple-cycle physics remain the same.

Air Compression - Heat Addition - Air Expansion

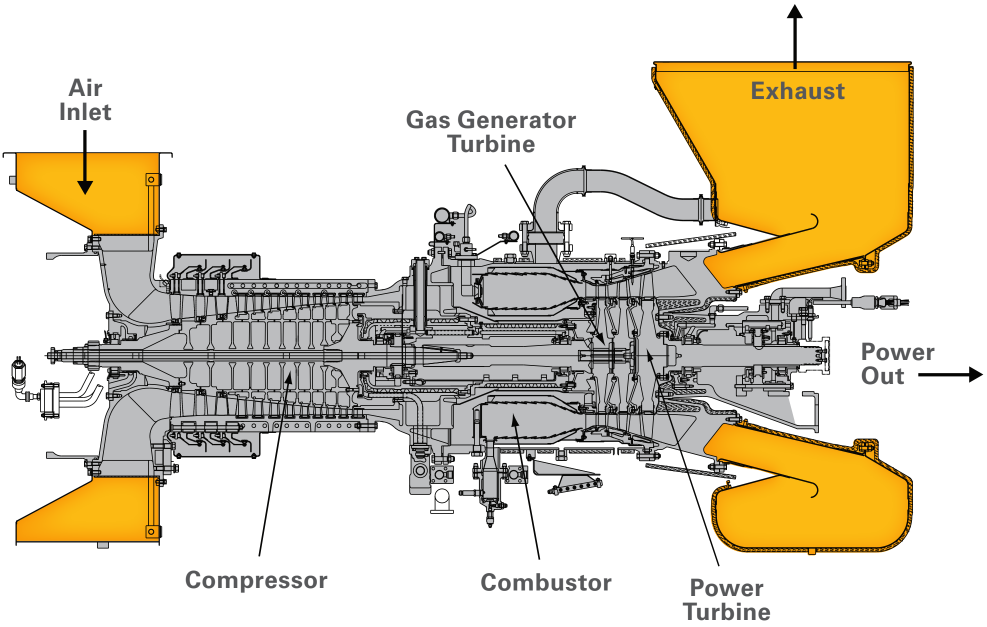


Figure 1-3. *Simple Brayton Cycle Axial Flow Gas Turbine*

FIRST LAW OF THERMODYNAMICS

Before analyzing gas turbine component mechanical intricacies and aerodynamic details, let us return to basic thermodynamics and treat the gas turbine as a very simple black box. We will apply the first law of thermodynamics (energy is conserved) to this black box, and see if we can develop a useful and hopefully simple expression for gas turbine performance.

The first law of thermodynamics for open systems under steady-state conditions simply states that the energy flowing into a system equals the energy flowing out of the system. While the energy is thus conserved, it may show up in different forms. Such forms are chemical energy, kinetic energy, potential energy, mechanical work, or heat.

Another way of writing this is by introducing enthalpy h, velocity w, and elevation z, heat q and mechanical work W_t:

$$\left(h_2+\frac{w_2^2}{2}+gz_2\right)-\left(h_1+\frac{w_1^2}{2}+gz_1\right)=q_{12}+W_{t,12}$$

For many systems, we can neglect elevation, and introduce the total (or stagnation) enthalpy. Stagnation value means that the pressure or temperature measurement is assumed to have been taken after the flow has been decelerated to zero velocity. On the other hand, static value means that the measurement has been taken as if the measuring probe is following the flow. Throughout this text, stagnation temperatures and static temperatures are assumed to be equal. Due to the low, relative flow velocities, this is a fair assumption for most turbomachinery:

$$h_t = h + \frac{w^2}{2}$$

and get:

$$\Delta h_{t,12} = q_{12} + W_{t,12}$$

Unless noted, we will not specifically distinguish total enthalpy and enthalpy in this text, because in many instances, the velocities are relatively low.

Writing this equation in terms of power, we simply multiply the equation above by the mass flow W, to get:

$$W\Delta h_{12} = W \cdot q_{12} + P_{t,12}$$

We do not try to give a thermodynamic explanation of what enthalpy is. Most textbooks about thermodynamics provide adequate, detailed derivations. Rather, we note that enthalpy is a convenient concept to express thermodynamic relationships, and that under certain, idealized circumstances, enthalpy and temperature of a gas are related by:

$$\Delta h = cp\Delta T$$

Lastly, we want to introduce the concept of an adiabatic process. An adiabatic process does not exchange heat with the environment. In an adiabatic process, the first law of thermodynamics then becomes:

$$\Delta h_{t,12} = W_{t,12}$$

In other words, in an adiabatic system (a gas compressor without intercooling, for example), the mechanical work a system is subjected to will be observed by an increase in enthalpy.

If enthalpy (J/kg) is multiplied by the mass of a fixed quantity of gas (kg) one obtains its total energy (Joule). Furthermore, if enthalpy is multiplied by mass flow (kg/s), one obtains power (J/s=W) or work per time. From an engineering analysis perspective, enthalpy is simply the useable energy per unit mass of the working fluid (air). Later, we will show that enthalpy is a quantity that is very convenient for the analysis of flow processes in turbomachinery.

The first law of thermodynamics states that the energy that goes into a domain has to equal the energy that comes out; i.e., a simple energy balance *(Figure 1-4)*. Therefore, let us first take a look at all the energy that enters into the gas turbine: fuel flow and cold air that are drawn in. The power of the fuel flow can be expressed mathematically as the mass flow of the fuel, multiplied by its heating value multiplied by some burner efficiency. Similarly, the power of the cold air going into the gas turbine can be expressed as the mass flow of the air multiplied by its enthalpy, where enthalpy is defined as the air temperature multiplied by its specific heat.

Now let us analyze the energy or power that exits the gas turbine: hot exhaust fuel/air mixture and shaft power. The hot exhaust gas is essentially a loss (unused energy) and can be expressed as the mass flow out multiplied by its enthalpy. The outgoing mass flow can be simplified further as the mass flow of the fuel plus the mass flow of the cold air in (mass must also be conserved). Finally, shaft output power is the usable end product of the thermodynamic process, which can also be expressed as the shaft torque (angular momentum) multiplied by the angular (rotational) speed of the shaft.

To keep things simple, we do not count the energy flows from radiated heat, and heat removed with the lube oil. We also neglect the fact that the fuel gas not only enters chemical energy, but also heat energy due to the fact that it has an enthalpy different from the ambient air. We further (for now) also neglect the complication that chemical reactions may not be complete, thus not all chemical energy is converted into heat. We therefore do not look at the fuel flow, but simply the heat that burning the fuel adds to the system.

Let us apply the first law of thermodynamics and sum up all of the above terms. What we are plainly looking for is some type of expression for the gas turbine shaft output power as a function of measurable variables, such as mass flows (volume flow multiplied by density), temperatures and gas properties *(Figure 1-4)*.

Hence,

$$P_{Shaft\ Ideal} = P_4 = P_1 + P_2 - P_3$$

Thus,

$$P_{Shaft\ Ideal} = W_{air}\, c_{p,in}\ (T_{in} - T_{ref}) + W_{fuel}\, q_R\ - (W_{air} + W_{fuel})\, c_{p,out}\ \ (T_{out} - T_{ref})$$

The above expression is a simplified and neglects heat radiation and mechanical (bearing, seal, accessory and gear friction) losses. To account for mechanical losses, mechanical efficiency is defined and introduced into the energy equation:

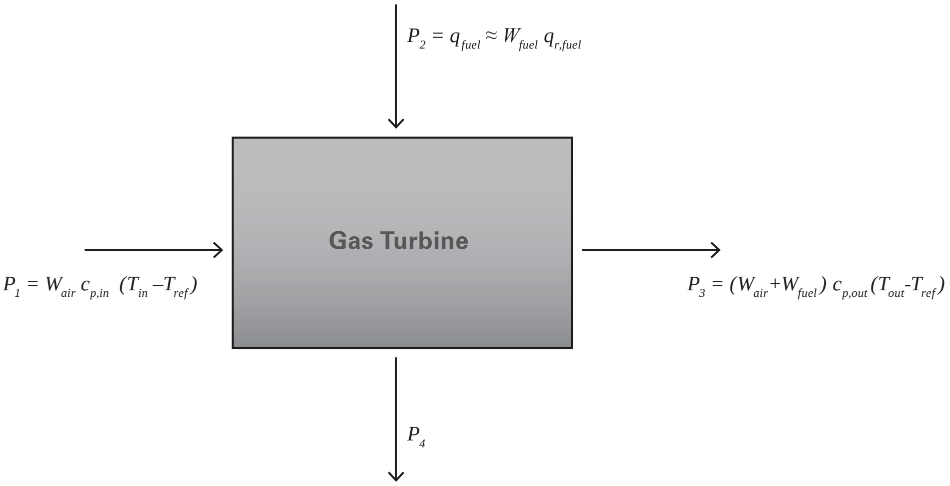

$P_2 = q_{fuel} \approx W_{fuel}\, q_{r,fuel}$
Gas Turbine
$P_1 = W_{air}\, c_{p,in}\ (T_{in} - T_{ref})$
$P_3 = (W_{air} + W_{fuel})\, c_{p,out}(T_{out} - T_{ref})$
P_4

Figure 1-4. *First Law of Thermodynamics: Energy In = Energy Out*

$$\eta_M = \frac{P_{Shaft\ Actual}}{P_{Shaft\ Ideal}}$$

As a result, we get:

$$P_{Shaft} = \eta_M\, [W_{air}\, c_{p,in}\ (T_{in} - T_{ref}) + W_{fuel}\, q_R\ -(W_{air} + W_{fuel})\, c_{p,out}\ \ (T_{out} - T_{ref})]$$

This simple equation still neglects some heat radiation losses, but is otherwise an accurate representation of the thermodynamic performance of a gas turbine and, typically, does not deviate more than 1% to 2% from reality. Thus, by applying energy conservation, we managed to derive an equation that can be used to assess the shaft output power of a gas turbine without having even studied the gas turbine components in any detail.

CARNOT EFFICIENCY

The efficiency of any heat engine, such as a gas turbine, is limited by its ability to convert heat into mechanical output shaft work. Any heat that is not converted into mechanical work will exhaust the heat engine and is generally considered a loss. The ideal limit of this heat to mechanical work efficiency is called Carnot Efficiency, based on the ideal cycle proposed by Nicolas Carnot in 1821. A simplified approach to determine this efficiency limit can be determined by again looking at a heat engine—or gas turbine— as a black box as shown in *Figure 1-5*:

It shows the energy flow into a generic heat engine. Here the Work In is the fuel flow energy that is converted into heat via combustion, the Work Out is the net shaft output work, and the Work Lost is the exhaust heat. If a basic efficiency is defined as the ratio of shaft output work divided by fuel flow energy, we get:

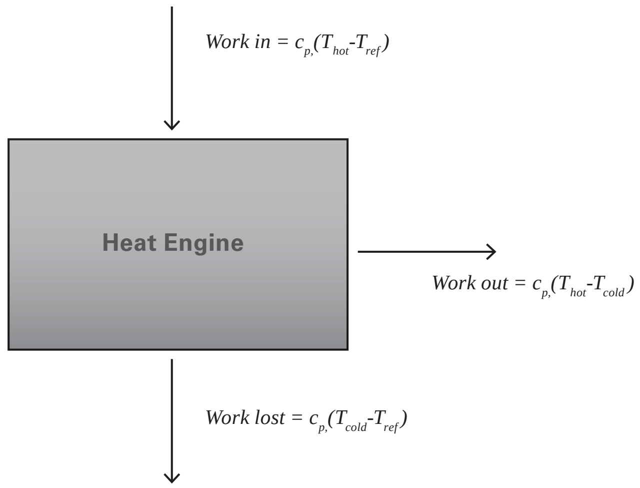

Work in = $c_{p,}(T_{hot}-T_{ref})$
Heat Engine
Work out = $c_{p,}(T_{hot}-T_{cold})$
Work lost = $c_{p,}(T_{cold}-T_{ref})$

Figure 1-5. *Energy Balance of a General Heat Engine*

$$Efficiency = \frac{Work \rightarrow Out}{Work \rightarrow In} = \frac{c_p T_{Hot} - c_p T_{Cold}}{c_p T_{Hot}} = 1 - \frac{T_{Cold}}{T_{Hot}}$$

This equation provides a basic definition of the maximum achievable efficiency of any heat engine, the aforementioned Carnot Efficiency. No heat engine, such as a gas turbine, can ever exceed this efficiency and in reality most heat engines do not reach efficiencies anywhere near the ideal Carnot Efficiency. Nonetheless, the definition of Carnot Efficiency provides us with valuable information regarding gas turbine behavior. For a simple heat engine the T_{Hot} is the combustion peak firing temperature and the T_{Cold} is the ambient air temperature. Thus, from the equation we can easily see:

- The engine efficiency must always be less than 100% (2nd law of thermodynamics).
- The engine efficiency can be improved by either increasing its combustion firing temperature (T_{Hot}) or reducing its inlet air temperature (T_{Cold}).
- If there is no difference between T_{Hot} and T_{Cold}, i.e., there is no firing or heat input, then the efficiency is zero and no output power is produced.
- T_{Cold} cannot be zero (3rd law of thermodynamics) or T_{Hot} cannot be infinite.

GAS TURBINE COMPONENTS

In our previous review the gas turbine was viewed as a black box and the first law of thermodynamics applied. We will continue to use energy concepts; however, now we will begin analyzing the individual gas turbine components (compressor, combustor, turbine), rather than the system as a whole.

We start by dividing the gas turbine process into individual steps or states as shown in *Figure 1-6*. State 1 is the air at ambient conditions before it enters the gas turbine; State 2 is the compressed air exiting the compressor; State 3 is heated air exiting the combustor; and State 7 is air exhausted back into the atmosphere after driving the turbine. By following the condition of the air, pressure, temperature and gas properties, as it flows from State 1 through States 2, 3 and 7 and then back to State 1, the entire simple gas turbine cycle can be modeled. This method is called a one-dimensional (1-D) cycle analysis and will be studied in more detail in the upcoming sections. For now, we will only look at the energy enthalpy (energy per unit mass) of the air as it flows through the gas turbine and see if we can develop a physical understanding of the process.

The enthalpy of the air at State 1 (h_1) is $c_{p1}T_1$ and the enthalpy of the air at State 2 (h_2) is $c_{p2}T_2$. Thus, the energy per unit mass that is added to the air as it passes through the compressor is the enthalpy difference h_2-h_1 or $c_p(T_2$-$T_1)$, assuming for now that the specific heat remains constant; i.e., c_{p1}=c_{p2}. This enthalpy difference is also often referred to as "head."

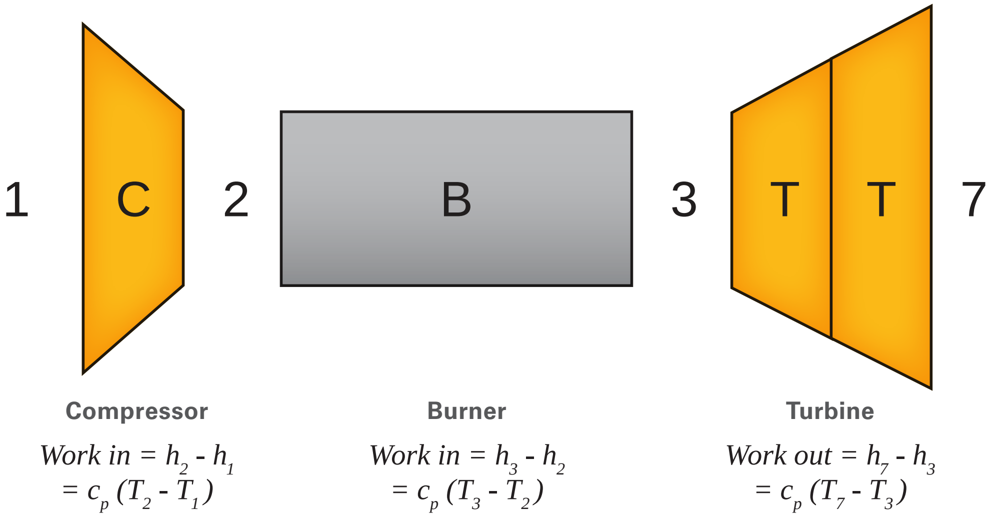


Figure 1-6. *Gas Turbine Components*

Similarly, the energy added by the combustor and the energy extracted by the turbine can be expressed as enthalpy differences h_3 - h_2 and h_7 - h_3, respectively. Thus, by observing gas turbine temperatures, we can model how energy is added to the airflow in the compressor and combustor and taken out of the airflow in the turbine. In multiplying the enthalpy difference (head) with the gas turbine mass flow, the total work done by the compressor, combustor or turbine can be determined. For example, the total work done by the compressor on the air is $W\bullet\ (h_2-h_1)$.

This means, that the work output of the gas turbine will be the difference between the power the turbine extracts from the gas, and the compressor consumes to compress the gas:

Work Output = Work Turbine-Work Compressor = $c_p(T_3-T_7) - c_p(T_2-T_1)$

Assuming c_p doesn't change (in the real gas turbine, it will to some degree, because c_p is a function of temperature and gas composition), we get:

Work Output = $c_p(T_3–T_7-T_2+T_1)$, or, after rearranging:

Work Output = $c_p(T_3-T_2)+(T_1-T_7)= c_p(T_1-T_7)$ + work combustor

In other words, the total output depends on the amount of work addition in the combustor (i.e., the firing temperature T_3), but the less of this work addition that can be recovered (i.e., the higher T_7 gets), the lower the output becomes. The recovery is a function of the pressure ratio available. Therefore, the work output (or the power density) increases with pressure ratio and firing temperature.

BRAYTON EFFICIENCY

Next, let us define a gas turbine efficiency based on energies (or work); i.e., gas turbine efficiency equals energy-out (shaft output) divided by energy-in (fuel flow):

$$\eta = \frac{\textit{Work Output}}{\textit{Work Input}}$$

This efficiency usually is referred to as thermal efficiency and, for this specific thermodynamic cycle, it is also called the Brayton Efficiency. The Brayton Efficiency will tell us the ideal, maximum theoretically possible, efficiency of this simple gas turbine cycle.

ENERGY-IN: The chemical energy put into the gas turbine by the fuel/air combustion process, which is the enthalpy difference across the combustor: (h_3-h_2).

ENERGY-OUT: Since the compressor and the turbine are on a single shaft, the total gas turbine output energy is simply the turbine energy minus the compressor energy: $(h_3-h_7)-(h_2-h_1)$.

Hence, we obtain:

$$\eta = \frac{\textit{Work Output}}{\textit{Work Input}} = \frac{C_p(T_3 - T_7) - C_p(T_2 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{T_7 - T_1}{T_3 - T_2}$$

This general expression for Brayton Efficiency is more specific than the previously discussed Carnot Efficiency, but demonstrates similar physical trends. The cold temperature (T_{Cold}) in a gas turbine is effectively the difference between the ambient and the exhaust temperature (T_7-T_1, the heat not utilized in the cycle). Similarly, the hot temperature (T_{Hot}) is the heat added in the cycle, which is the temperature rise across the combustor (T_3-T_2).

By introducing the isentropic relationship between pressure and temperature for an ideal gas, $T_2/T_1=(P_2/P_1)^{\gamma-1/\gamma}$, a simple expression for the maximum theoretically possible Carnot efficiency of a simple open cycle gas turbine is obtained:

$$\eta_{maximum} = 1 - \frac{T_2}{T_1} = 1 - \frac{T_3}{T_7} = 1 - \left(\frac{P_2}{P_1}\right)^{\frac{1-\gamma}{\gamma}}$$

NOTE: Isentropic means that the entropy (s) remains constant; i.e., an ideal process with no thermodynamic losses. Entropy will be discussed in further detail in later sections; however, it is beyond the scope of this text to derive the isentropic relationships, which can be found in most thermodynamics textbooks.

This expression shows that a thermodynamic cycle efficiency is either limited by its pressure ratio or by its temperature ratio. But since in all real machines, such as a gas turbine, there is unused thermal energy in the form of exhaust heat, the real limit of efficiency is always the pressure ratio, rather than the temperature ratio. Thus, it is important to recognize, based on this simple relationship, the ideal efficiency of a gas turbine depends only on the compression ratio (P_2/P_1) of the axial compressor. Later in this text, we will discuss how the actual gas turbine efficiency deviates from the ideal efficiency, but for now, two conclusions should be emphasized:

- Simple-cycle gas turbine efficiency strongly depends on the compression ratio of the gas turbine.
- Simple-cycle gas turbine actual efficiency must always be lower than its ideal Brayton Efficiency; i.e., $\eta_{Actual} \leq \eta_{Ideal}$

MAXIMUM THEORETICAL EFFICIENCY

Now let us apply some real numbers to the relationship between ideal efficiency and compression ratio and learn about actual gas turbine performance. In *Figure 1-7*, the general trend of ideal efficiency versus gas turbine compression ratio is plotted. The trend indicates that gas turbine efficiency is a strong function of pressure ratio; i.e., increasing the pressure ratio across the axial compressor of the gas turbine will increase the overall efficiency of the gas turbine. *Table 1-1* compares maximum theoretical efficiency as calculated from the above equation versus actual efficiency. The reasons for this difference are mechanical losses (bearings, seals and gearbox friction), aerodynamic losses (air boundary layer friction, inlet/exhaust pressure drops, aerodynamic stall and blockage) and heat losses (casing heat losses and bearing losses).

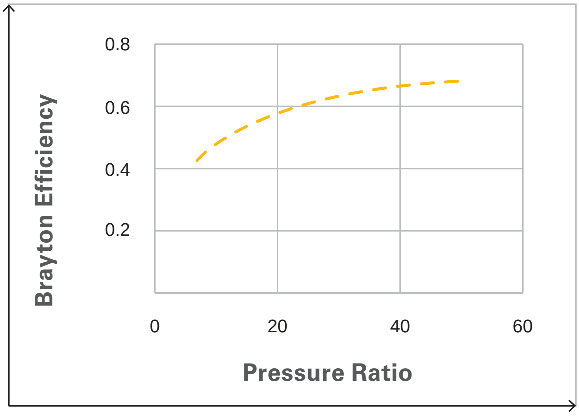


Figure 1-7. *Simple Brayton Cycle Maximum Theoretical Efficiency*

	Saturn 20	**Centaur 40**	**Taurus 60**	**Taurus 70**	**Mars 100**	**Titan 130**	**Titan 250**
$\eta_{Theoretical}$, %	41	48	51	56	56	56	60
η_{Actual}, %	24.5	28	32	35.	34.5	36	40
Power, hp	1,600	4,700	7,800	10,800	16,000	20,500	30,000

Table 1-1. *Comparison of Maximum Theoretical versus Actual Efficiency*

Hence, we have established that a gas turbine using the simple open Brayton cycle is limited in thermodynamic efficiency and can never reach a 100% thermal efficiency. For realistic design purposes, the simple-cycle gas turbine thermal efficiency in actuality can never exceed 55%. It will be shown later that the primary reason for this efficiency limitation is that the air exhausting from the gas turbine is always significantly hotter than the ambient air. That is, the gas turbine cannot recover all the energy that was put into the air by the combustor.

Smaller gas turbines have a lower airflow, volume-to-wall surface ratio and larger boundary layer pressure losses; i.e., they are less efficient. Especially the blades in the last compressor stages become very small.

Also, *Table 1-1* indicates a direct correlation between power and efficiency. Why would gas turbine efficiency increase with output power? There is no intuitively obvious reason from thermodynamics. Nonetheless, this apparent trend can be explained as follows:

We can also use these correlations for another assessment:

Since the turbine power is, neglecting mechanical losses, assuming negligible contributions of the fuel flow to the overall mass flow, and assuming constant heat capacity:

$$P_{Shaft} = Wc_p\,(T_{in} - T_{ref}) + W_{fuel}\,q_R - W\,c_p\,(T_{out} - T_{ref}) = Wc_p(T_{in} - T_{out}) + Q_{in}$$

or, in other words (with $Q_{23} = q_r W_{fuel}$ and $Q_{71} = Wc_p(T_{out} - T_{in})$

$$P_{shaft} = Q_{23} - Q_{71}$$

With Q_{23} the heat introduced to the process via the combustion process, and Q_{71} the exhaust heat lost, we get:

$$P = Wc_p(T_3 - T_2)(1-(T_7 - T_1)/(T_3 - T_2))$$

For the assumed isentropic compressor, and isentropic turbine, we get:

$$\frac{T_1}{T_2} = \frac{T_7}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma - 1}{\gamma}}$$

In this case, as discussed above, thermal efficiency is a function of the cycle pressure ratio p_2/p_1, thus one might want to build a gas turbine with a very high pressure ratio.

However, increasing the pressure ratio leads to an increase in absorbed compressor power, but the higher T_3 gets, the more power is produced. We find the available power:

$$P = Wc_pT_1\left[\frac{T_3}{T_1} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]\left[1 - \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

We can now, by differentiating, find the optimum pressure ratio that gives the best power density for a given achievable firing temperature T_3:

$$\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]_{opt} = \sqrt{T_3/T_1}$$

If, for a given achievable firing temperature T_3 and a given mass flow W, the pressure ratio is higher or lower than the optimum calculated above, the available gas turbine output is reduced. This means that a gas turbine with the best power density will require different cycle parameters than a gas turbine with the best possible efficiency.

SECOND LAW OF THERMODYNAMICS

The term entropy was mentioned previously without providing a definition. The textbook thermodynamics definition of heat transfer entropy is:

ds≥dQ/T = entropy flux is greater or equal than heat flux divided by temperature

This also is called the Second Law of Thermodynamics. While the expression may be useful to the thermodynamicist, within the context of this basic text on fundamental principles, it is not. However, it should at least be mentioned that this very simple statement, when properly analyzed, is an extremely powerful engineering tool. Another, more physically intuitive definition is:

Entropy indicates losses in the fluid (air) due to molecular disorder.

Entropy is created whenever heat transfer or fluid friction occurs.

Thus, entropy is essentially a quantitative measurement of a non-recoverable energy loss. Any real thermodynamic process involves an increase in entropy. However, if we assume that a thermodynamic process is ideal (no losses), then the entropy will be maintained constant (s=constant); i.e., we have an isentropic process. In this case, a set of equations called the Isentropic Relationships for Ideal Gasses can be employed:

$$\frac{P_1}{P_2} = \left(\frac{\rho_1}{\rho_2}\right)^{\gamma} = \left(\frac{v_2}{v_1}\right)^{\gamma} = \left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma-1}}$$

The derivation of the isentropic relationships is beyond the scope of this text, but can be found in most thermodynamics textbooks. Nonetheless, the above isentropic relationships are extremely important equations since they allow us to relate the gas turbine air pressure, temperature, density and volume to each other throughout the thermodynamic cycle. Later, we will see that we can relax the assumption that the process has to be ideal and still use the isentropic relationships by introducing multiplication factors or efficiencies into the equations.

One particular result of these relationships is the expression for isentropic work as a function of compression ratio:

$$\Delta h_s = c_p T_1 \left[\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

PROCESS DIAGRAMS (PRESSURE-VOLUME)

Previously, the simple Brayton gas turbine cycle was studied by following the gas as it passes through the individual gas turbine components *(Figure 1-8)*. Let us repeat this approach; however, this time the simple gas turbine thermodynamic cycle will be presented graphically. We will plot the relevant process variables such as pressure, specific volume, temperature, enthalpy and entropy as the air passes from State 1 - ambient to State 2 - compressor exit to State 3 - combustor exit to State 7 - turbine exit through the gas turbine *(Figure 1-9)*. This type of map is called a process diagram. The two most common process diagrams are pressure-volume (P-v) and enthalpy-entropy (h-s). (The h-s diagram is also often presented as a temperature-entropy (T-s) diagram.) For now, we will assume no losses; i.e., an ideal thermodynamic process.

We will begin by plotting the pressure versus specific volume (P-v diagram) as the air passes through a simple Brayton cycle gas turbine. The specific volume (v) is defined as the reciprocal of the density (ρ).

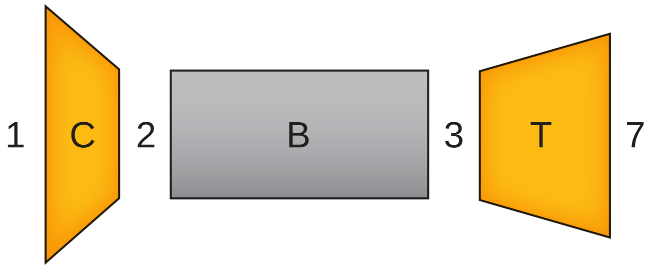


Figure 1-8. *Simple Open Brayton Cycle Process Components*

State 1 - Ambient air has not entered gas turbine and is at atmospheric condition:

$P = 1.0 \times 10^5$ Pa, $v = 0.82$ m^3/kg

Process 1 to 2: Air is being compressed as it passes through the axial compressor. Since the assumption is an ideal process, this compression is considered to be isentropic, constant entropy; i.e., no losses. As the air is compressed, its pressure (P) increases, density (ρ) increases and specific volume (v) decreases.

State 2 - Air between compressor exit and combustor inlet:

$P = 15.0 \times 10^5$ Pa, $v = 0.13$ m^3/kg

Process 2 to 3: Air passes through the combustor and is heated. Since the assumption is an ideal process, the air pressure remains constant, no pressure drop isobaric process, and only the temperature and specific volume increase. From the equation of state, the gas density decreases almost linearly with temperature; i.e., the gas expands, density (ρ) decreases and specific volume (v) increases.

State 3 - Air between combustor exit and turbine inlet:

$P = 15.0 \times 10^5$ Pa, $v = 0.75$ m^3/kg

Process 3 to 4: Air is being expanded as it passes through the axial turbine. Since the assumption is an ideal process, this expansion is again considered to be isentropic. As the air is expanded, its pressure (p) decreases, density (ρ) decreases and specific volume (v) increases.

State 7 - Air exiting the turbine into ambient:

P = 1.0×10^5 Pa, v = 1.5 m^3/kg

Figure 1-9 shows the actual p-v diagram for the process just described. There are many direct conclusions one can draw from this diagram, but the most important is the following realization:

WORK = FORCE x DISTANCE = PRESSURE x VOLUME

The area under the curve in the P-v diagram represents the work output of the gas turbine. This type of diagram is very important for the gas turbine designer, since based on simple graphics, the gas turbine output power can be assessed as a function of the process parameters.

For example, if the firing temperature of the gas turbine is increased, the line from State 2 to State 3 becomes longer; thus, the area under the P-v curve becomes proportionally larger. This is consistent with our earlier conclusion that gas turbine output power is directly proportional to combustor firing temperature. Also, if the pressure ratio increases, State 1 to 2 and 3 to 7 become longer and the area under the curve increases, more output work is obtained without adding more work input (temperature increase), which means that the process efficiency must be increasing.

This confirms another earlier observation that gas turbine efficiency improves with pressure ratio. Finally, if the ambient temperature decreases, the line from State 7 to 1 is lowered, which increases the area inside the curve; i.e., the gas turbine produces more power at lower ambient temperatures.

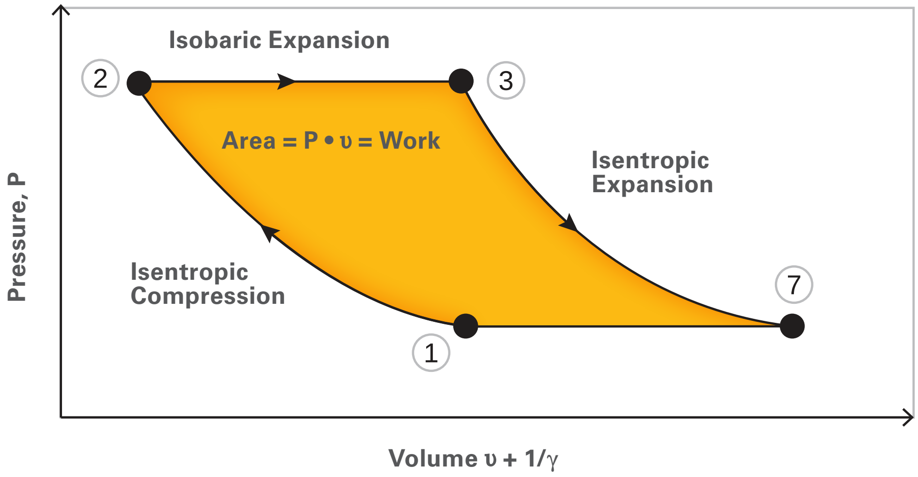


***Figure 1-9.** Pressure-Volume Process Diagram*

PROCESS DIAGRAMS (TEMPERATURE-ENTROPY)

Let us again analyze the ideal simple Brayton cycle graphically, this time plotting temperature versus entropy *(Figure 1-10)*. Note that the temperature (T) on this diagram can be exchanged with enthalpy (h) since they are related by:

$$h = c_p \cdot T$$

The specific heat, c_p, is assumed to stay constant throughout the process. The temperature versus entropy (T-s) diagram allows us to quantitatively assess the efficiency of a thermodynamic process, since an entropy increase indicates thermodynamic non-recoverable energy losses. Thus, following the air as it passes through the previously defined states of a simple Brayton cycle:

State 1 - Ambient air has not entered gas turbine and is at atmospheric condition:

T = 300°K, s = 0 J/mol · K

Process 1 to 2: Air is being compressed as it passes through the axial compressor. Since the assumption is an ideal process, this compression is considered to be isentropic. As the air is compressed, its pressure (P) increases and temperature (T) and enthalpy (h) also increase.

State 2 - Air between compressor exit and combustor inlet:

T = 700°K, s = 0 J/mol · K

Process 2 to 3: Air passes through the combustor and is heated. Based on the definition of entropy (ds≥dQ/T), we know that for any heat transfer the entropy has to increase. Therefore, temperature (T), enthalpy (h) and entropy(s) increase.

State 3 - Air between combustor exit and turbine inlet:

T_1 = 1,400°K, s = 800 J/mol · K

Process 3 to 7: Air is being expanded as it passes through the axial turbine. Since the assumption is an ideal process, this expansion is considered to be isentropic. As the air expands, its pressure (P) decreases and temperature (T) and enthalpy (h) also decrease. As shown in *(Figure 1-10)*.

State 7 - Air exiting the turbine into ambient:

T_1=500°K, s = 800 J/mol · K

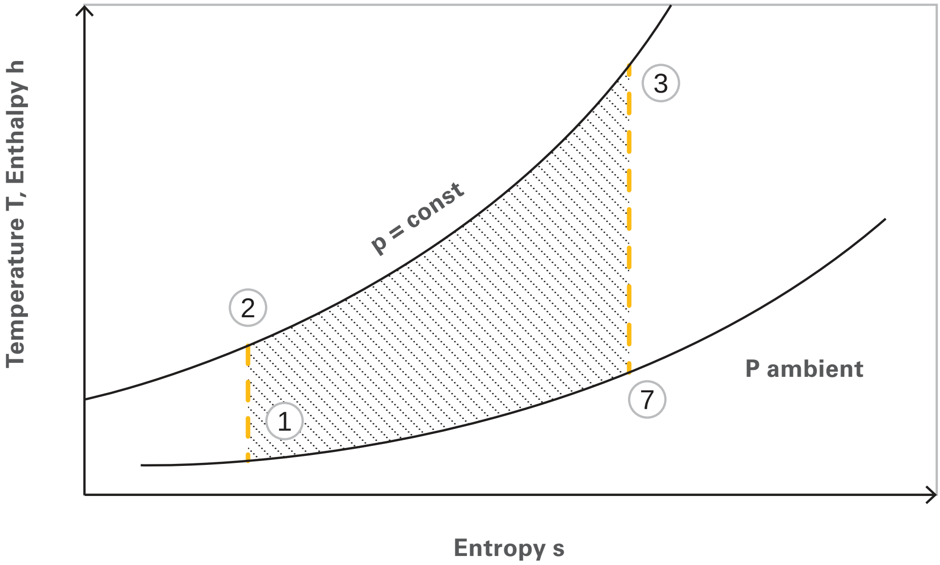

Temperature T, Enthalpy h
p = const
3
2
1
7
P ambient
Entropy s

Figure 1-10. *Temperature vs Entropy Process Diagram*

IDEAL SIMPLE-CYCLE ANALYSIS

Let us formalize some of the concepts we have employed for the simple Brayton cycle over the previous pages and try to make them into useful engineering tools for gas turbine performance analysis. For now, we will continue to assume that the gas turbine cycle is an ideal process with no component losses, but we will model a more realistic gas turbine by including inlet and exhaust systems and separating the gas producer and power turbines. *Figure 1-11* shows the resulting subdivision of the gas turbine into States: 0=ambient, 1=compressor inlet, 2=combustor inlet, 3=turbine inlet, 5=power turbine inlet, 7=exhaust inlet, and 0=exhaust outlet.

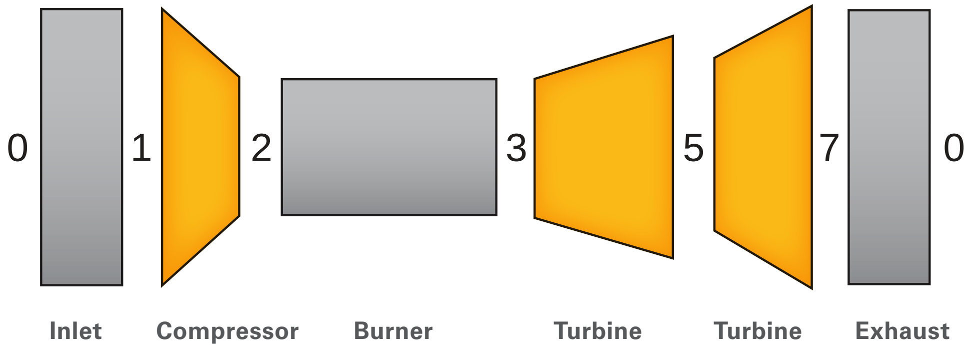


Figure 1-11. *Components*

For the simple-cycle analysis, we will follow the air as it passes through each of the gas turbine components and determine temperature, pressure, and density at each of the 7 States. Later, we will show that, based on the resulting information, total gas turbine power, efficiency, fuel flow, and exhaust gas temperature can be determined; i.e., we can model and predict gas turbine performance. We will begin by summarizing the primary assumptions for the ideal cycle analysis:

1. The physical properties of the gas are constant:
   - specific heat ratio: γ = constant
   - specific heat: c_p = constant
2. No losses due to fluid friction, therefore, isentropic compression and expansion.
3. Air velocities are very low, so that stagnation and static pressures and temperatures can be considered to be identical.

The above assumptions essentially state that each of the gas turbine components has a thermal efficiency of 100% (Energy Out/Energy In = 1.0). However, we previously showed that even for an ideal simple Brayton cycle the total gas turbine thermal efficiency is significantly lower than 100%, typically between 28 and 40%. How can a gas turbine where each component has 100% efficiency have a thermal efficiency of around 30%?

Each gas turbine component may have 100% efficiency, but the exhaust air exiting the gas turbine is significantly hotter than the inlet air; i.e., hot exhaust air (energy) is blown into the atmosphere and lost. Thus, the usable output shaft energy of the gas turbine must be lower than the input fuel energy: Energy Out/Energy In < 1.0. Intuitively it is expected that the energy lost by a gas turbine due to hot exhaust gas is to be approximately as follows, assuming for example a 30% thermal efficiency:

(Shaft Output Energy + Hot Exhaust Energy)/Fuel Inflow Energy = 1.0,

or

Shaft Output Energy/Fuel Inflow Energy = 0.3,

thus,

Hot Exhaust Energy = 0.7 Fuel Inflow Energy.

Later in this text, the actual amount of energy that is lost due to unused hot exhaust air will be calculated. Also, some technologies that are employed to capture this lost energy such as combined or advanced cycles will be briefly discussed.

IDEAL CYCLE ANALYSIS METHOD

The step-by-step method to determine the performance of an ideal simple Brayton cycle is presented below:

State 0 to 1 - Inlet System

Assuming no losses, the entropy through the inlet system remains constant:

$$h_0 = h_1$$

$$c_{p0} T_0 = c_{p1} T_1$$

Assuming that the specific heat, c_p, remains constant:

$$T_0 = T_2$$

from the isentropic relationship, $p_1/p_2 = (T_1/T_2)^{\gamma/\gamma\text{-}1}$, thus:

$$P_0 = P_1 = P_{atmosphere}$$

and from the equation of state, we can derive the air density:

$$\rho_0 = P_0/RT_0 = P_1/RT_1 = \rho_1$$

State 1 to 2 - Axial Compressor

Assuming isentropic compression ($\Delta s=0$) and for a given compressor pressure ratio (π_c), we can determine the compressor exhaust temperature (T_3) and pressure (P_3) from:

$$P_2/P_1 = \pi_c$$

$$T_2/T_1 = \pi_c^{\frac{\gamma}{\gamma - 1}}$$

and from the equation of state, we determine the air density:

$$\rho_2 = P_2(R \cdot T_2)$$

State 2 to 3 - Combustor

Assuming no pressure drop in the combustor:

$$P_2 = P_3$$

The firing temperature (T_4) of the gas turbine is known and depends typically only on blade material limitations. If the gas turbine air volume flow Q is known, which is usually a compressor/turbine design specification, the air mass flow W_{air} can be determined:

$$W_{air} = \rho_2 \cdot Q$$

Furthermore, assuming no energy losses, for the temperature increase from burning the fuel, we know that:

$$Work\ Out_{\ heat} = Work\ In_{\ fuel}$$

thus,

$$W_{air} \cdot (h_3 - h_2) = W_{air} \cdot c_p \cdot (T_3 - T_2) = W_{fuel}\, q_R$$

Where q_R is the heating value (J/kg) of the fuel. The above equation allows us to determine the fuel flow W_{fuel} into the gas turbine to achieve a certain firing temperature T_4. From the equation of state, we can then get the air density:

$$\rho_3 = P_3/(R \cdot T_3)$$

State 3 to 5 - Gas Producer Turbine

The gas producer turbine and axial compressor are mounted on the same shaft; i.e., their powers have to be identical:

$$Work_{compressor} = Work_{gas\ producer\ turbine}$$

Thus,

$$W_{air} \cdot (h_2 - h_1) = [W_{air} + W_{fuel}] \cdot (h_5 - h_3)$$

or

$$W_{air} \cdot c_p \cdot (T_2\text{-}T_1) = [W_{air} + W_{fuel}] \cdot c_p \cdot (T_5\text{-}T_3)$$

This expression can be solved to obtain the gas producer turbine exhaust temperature (T_5). Assuming isentropic expansion (Δs=0), the gas producer turbine exhaust pressure (P_5) is then determined from:

$$P_5/P_3 = (T_5/T_3)^{\frac{\gamma}{\gamma - 1}}$$

Density is again obtained directly from the equation of state:

$$\rho_5 = P_3/(R \cdot T_3)$$

State 5 to 7 - Power Turbine

Assuming isentropic expansion (Δs=0) and knowing the power turbine exhaust pressure (P_7) from the subsequent State 7 to 0, the power turbine exhaust temperature (T_7) is determined from:

$$T_7 /T_5 = (P_7 /P_5)^{\frac{\gamma}{\gamma - 1}}$$

Air density is obtained from:

$$\rho_7 = P_7/(R \cdot T_7)$$

State 7 to 0^1 - Exhaust System

Assuming no losses, the entropy through the exhaust system remains constant:

$$h_7 = h_{0^1}$$

$$c_{p7}T_7 = c_{p0^1}\, T_{0^1}$$

Assuming that specific heat (c_p) remains constant:

$$T_7 = T_{0^1}$$

And therefore:

$$P_6 = P_{0^1} = P_{atmosphere}$$

Air density is finally obtained from:

$$\rho_{0^1} = P_{0^1} / RT_{0^1})$$

The gas turbine shaft output power equals the work absorbed by the power turbine and can therefore be obtained from:

$$P_{output} = [W_{air} + W_{fuel}] \cdot (h_7\text{-}h_5)$$

or

$$P_{output} = [W_{air} + W_{fuel}] \cdot c_p \cdot (T_7\text{-}T_5)$$

Also, the gas turbine thermal efficiency is determined from:

$$\eta = Power\ Ouput\ /\ Power\ Input = P_{output}\ /\ [W_{fuel} \cdot q_R]$$

or

$$\eta = P_{output}\ /\ [W_{air} \cdot c_p \cdot (T_3\text{-}T_2)]$$

Using the above equations, ideal gas turbine performance can be determined. This simplified mathematical model can be used to perform parametric and optimization studies, which are required early on in the design process of a new gas turbine. However, to predict real gas turbine performance, the model needs to be refined to include actual component losses and efficiencies. This more advanced type of model is called the non-ideal or real simple Brayton cycle analysis and is described in the following section.

NON-IDEAL SIMPLE-CYCLE ANALYSIS

The non-ideal or real simple-cycle analysis is based on the equations of the previously described ideal analysis; however, component efficiencies and pressure losses are introduced to model the non-ideal nature of the actual gas turbine cycle. For example, instead of assuming that the pressure is constant throughout the inlet/exhaust system and combustor, we assume a small pressure drop expressed in the form of a pressure ratio smaller than unity. Also, instead of assuming that compression and expansion are isentropic processes, the isentropic efficiency, based on the work ratio, is introduced to account for energy losses. Finally, the previous assumptions that physical gas properties are constant and that stagnation and static pressures/temperatures are identical are relaxed. In summary:

1. The physical properties of the gas vary with temperature and pressure:
   - specific heat ratio:
   $$\gamma = \gamma(T,P)$$
   - specific heat:
   $$c_p = c_p(T,P)$$
2. Fluid friction losses are expressed in the form of:
   - pressure drops or pressure ratios:
   $$\pi = P_{out}/P_{in}\ (<1.0)$$
   - isentropic component efficiencies:

   $$\eta = Work\ In/Work\ Out = Ideal\ (Isentropic)\ Work/Actual\ Work <1.0$$

3. Static pressure and temperature values are corrected using the actual flow Mach number to determine stagnation values:

$$T_t = T \cdot \left(1 + \frac{\gamma - 1}{2} \cdot M^2 \right)$$

$$P_t = P \cdot \left(1 + \frac{\gamma - 1}{2} \cdot M^2 \right)^{\frac{\gamma - 1}{\gamma}}$$

The derivation of these pressure/temperature Mach number corrections can be found in most aerodynamics textbooks.

NON-IDEAL CYCLE ANALYSIS METHOD

Based on the above definitions, we can formulate the following terms for the gas turbine component efficiencies and pressure ratios:

State 0 to 1 - Inlet System

Pressure ratio across the inlet system (inlet filter, silencer, ducting, diffuser):

$\pi_0 = P_1/P_2$

State 1 to 2 - Compressor

Isentropic efficiency of the compressor based on:

$$\eta_c = \frac{\text{Ideal (Isentropic) Work}}{\text{Actual Work}} = \frac{h_2^* - h_1}{h_2 - h_1}$$

State 2 to 3 - Combustor

Pressure ratio across the combustor and combustion efficiency:

$\pi_b = P_3/P_2$

$$\eta_b = \frac{\text{Energy Out}}{\text{Fuel Energy}} = \frac{W_{air} \cdot c_p \cdot (h_3 - h_2)}{W_{Fuel} \cdot q_R}$$

where q_R is the heating value (J/kg) of the gas turbine fuel.

State 3 to 5 - Gas Producer Turbine

Isentropic efficiency of the gas producer turbine based on:

$$\eta_t = \frac{Actual\ Work}{Ideal\ (Isentropic)\ Work} = \frac{h_5 - h_3}{h_5^* - h_3}$$

State 5 to 7 - Power Turbine

Isentropic efficiency of the power turbine based on:

$$\eta_t = \frac{Actual\ Work}{Ideal\ (Isentropic)\ Work} = \frac{h_7 - h_5}{h_7^* - h_5}$$

State 7 to 0 - Exhaust System

Pressure ratio across the exhaust system (exhaust silencer, ducting, expander):

$\pi_e = P_7/P_0$

Thus, there is a system of equations that consists of the equations presented earlier for the ideal analysis, the efficiency and pressure ratio expression, the gas property equations, and the Mach number corrections. By solving this system of equations, the actual (real) performance (power and efficiency) at any operating condition of the gas turbine can be determined.

The somewhat difficult numerical solution for this system of equations must be obtained using an iterative or a matrix solver approach, such as a Gauss-Seidel matrix inversion method. It is beyond the scope of this text to present a detailed numerical analysis, which can be found in most turbomachinery textbooks. However, a T-s diagram for a typical 20,000-hp gas turbine example is shown in *Figure 1-12*. The non-ideal nature of the process can be seen in the form of entropy increases from State 1 to 3 and from State 4 to 7.

Turbomachinery manufacturers frequently employ this very powerful mathematical analysis method for the design of new turbines and to predict performance for actual gas turbine applications. Nonetheless, the non-ideal cycle analysis still requires prior knowledge of compressor and turbine isentropic efficiencies, the inlet, exhaust, and combustor pressure ratios, and combustion efficiency. These values are usually obtained from experimental test stand data and/or computational fluid dynamics (CFD) analysis. We will briefly discuss the CFD analysis approach later in this text.

Some typical values for component efficiencies and pressure ratios of modern gas turbines are as follows:

Inlet System: $\pi_i > 0.99$
Axial Compressor: $\eta_c \approx 0.90$
Combustor: $\eta_b > 0.98$ and $0.80 < \pi_b < 0.95$ (strongly depending on basic design)
Gas Producer Turbine: $\eta_{gt} \approx 0.90$
Power Turbine: ηpt ≈ 0.90
Exhaust System: $\pi_e > 0.99$

In reality, these efficiencies are not constants, but functions of many variables such as ambient temperature, ambient pressure, humidity, shaft rotational speed and many more.

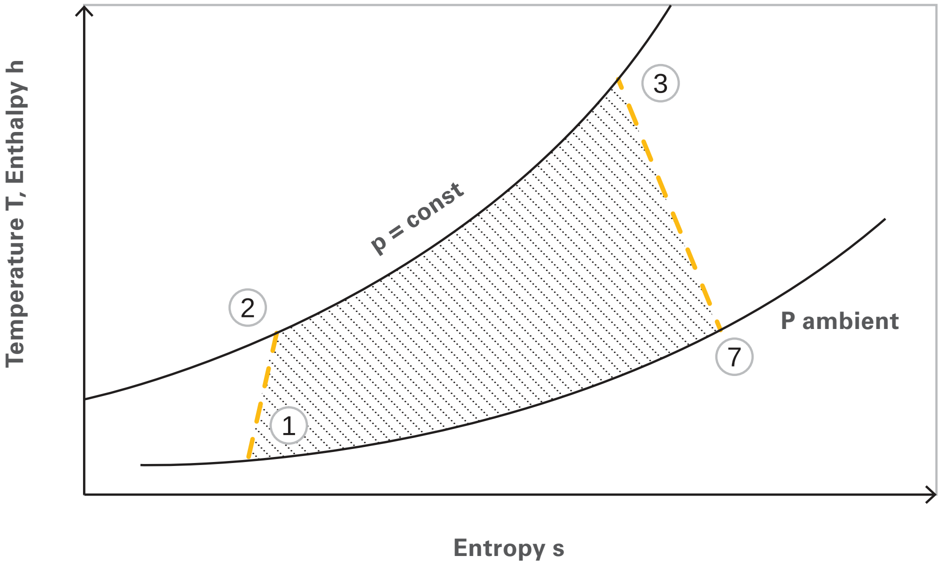


Figure 1-12. *Non-Ideal Simple Cycle Temperature vs Entropy Process Diagram*

SUMMARY

The following is a summary of what has been learned thus far from thermodynamics about the function and design objective of each of the relevant gas turbine components:

Inlet System. Collects and directs air into the gas turbine. Often, an air cleaner and silencer are part of the inlet system. It is designed for a minimum pressure drop, while maximizing clean airflow into the gas turbine.

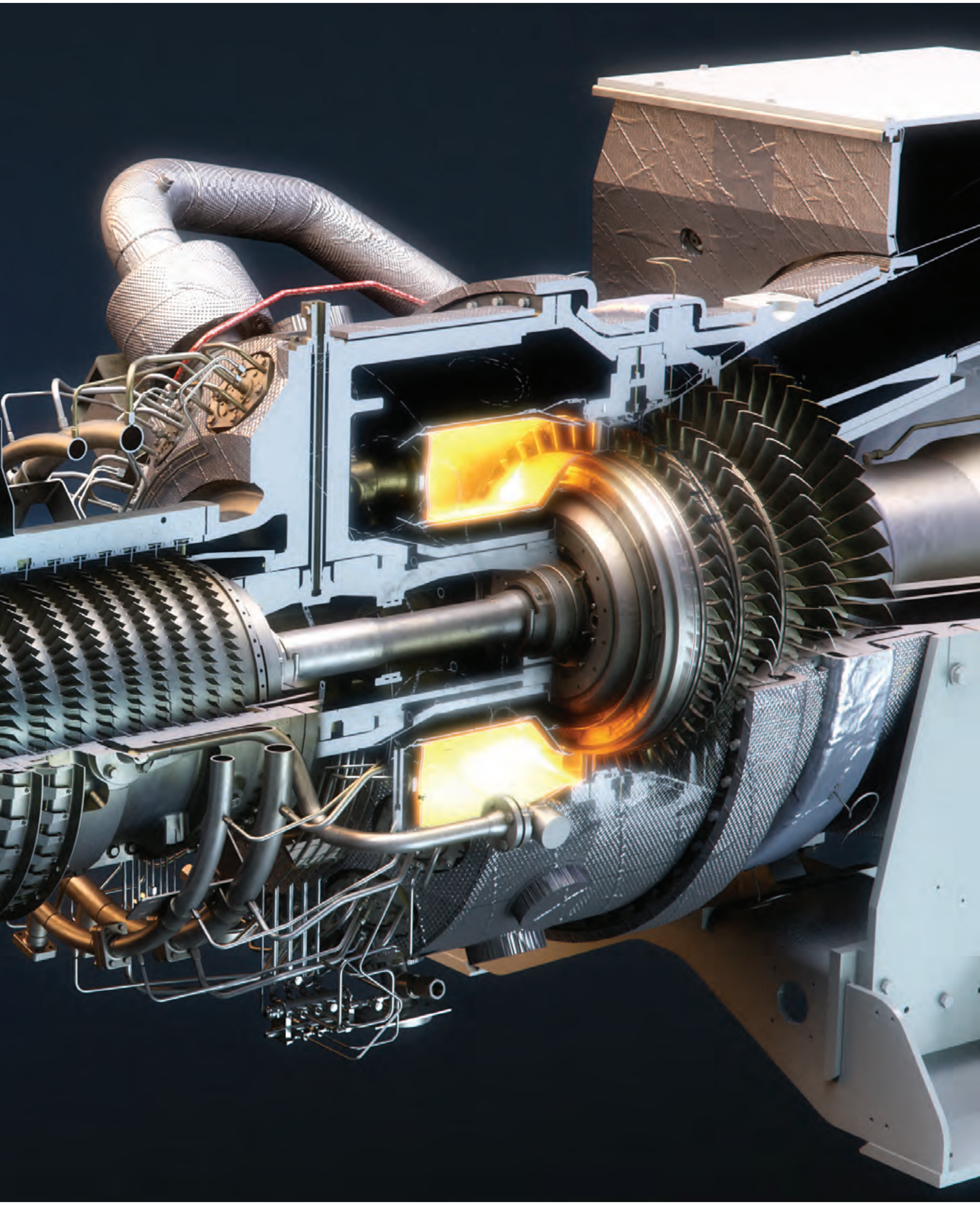
Compressor. Provides compression and, thus, increases the air density for the combustion process. The higher the compression ratio, the higher the total gas turbine efficiency. Low compressor efficiencies result in high compressor discharge temperatures, therefore, lower gas turbine output power.

Combustor (Burner). Adds heat energy to the airflow. The output power of the gas turbine is directly related to the combustor firing temperature; i.e., the combustor is designed to increase the air temperature up to the material limits of the gas turbine, while maintaining a reasonable pressure drop.

Gas Producer Turbine. Expands the air and absorbs just enough energy from the flow to drive the compressor. The higher the gas producer discharge temperature and pressure, the more energy is available to drive the power turbine, therefore, creating shaft work.

Power Turbine. Converts the remaining flow energy from the gas producer into useful shaft output work. The higher the temperature difference across the power turbine, the more shaft output power is available.

Exhaust System. Directs exhaust flow away from the gas turbine inlet. Often a silencer is part of the exhaust system. Similar to the inlet system, the exhaust system is designed for minimum pressure losses.



CHAPTER 2

TURBOMACHINERY COMPONENT BASICS

In Chapter 1, simple gas turbine cycle thermodynamics were discussed, focusing on gas turbine component design objectives. In this chapter, the actual function and internal aero/combustion dynamics of the compressor, combustor and turbine will be discussed. Some engineering concepts such as Euler's formula, velocity polygons, and combustion stoichiometry, all necessary for designing these gas turbine components will also be examined. We will begin by looking at the components in the order as the airflow passes through a gas turbine *(Figure 2-1)*.

COMPRESSOR

As seen previously, the gas turbine air compressor function is to increase pressure and density of the airflow. Based on isentropic relations, as the air pressure increases, the air temperature also increases. On all simple-cycle gas turbines, we know that this compression process is accomplished using either an axial or centrifugal compressor. Other machine designs employed to compress gas include the piston, reciprocating and rotary-screw compressors. These compressor designs are commonly referred to as positive-displacement compressors. The authors are not aware of any simple-cycle gas turbines that employ a positive displacement air compressor.

Figure 2-2 shows a typical axial compressor. In an axial compressor, the airflow follows primarily along the direction of the gas turbine axis, while in a centrifugal compressor the airflow is turned and directed radially outward. Later, we will discuss in more detail the design advantages and disadvantages of the two designs. For starters, the focus will be placed on understanding the compression process by asking a simple, but immensely important question:

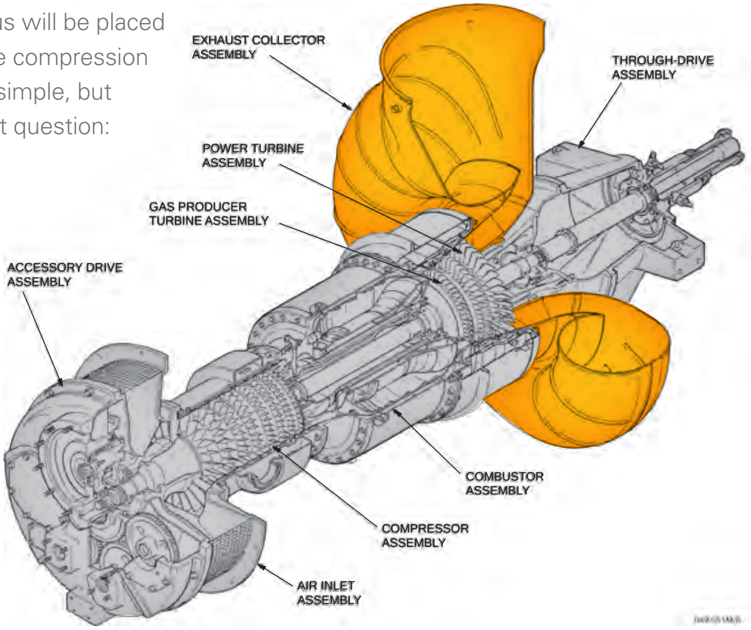


Figure 2-1. *Gas Turbine Components*

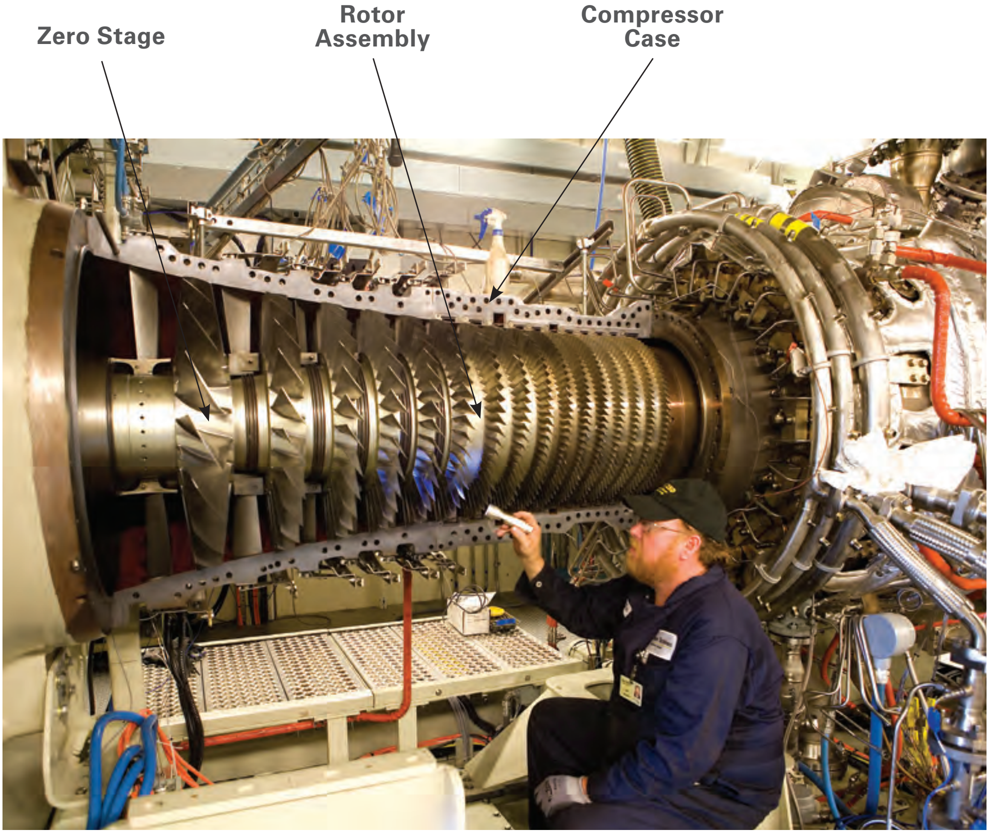


Figure 2-2. *Axial Compressor Rotor and Stator*

How is energy imparted to the airflow by a compressor?

The aerodynamic flow path of the compressor consists essentially of rotating blades and stationary blades (or diffuser vanes). Rotating (rotor) blades do not, at least directly, change the temperature or pressure of the airflow. Physically, then, the only function a rotating compressor blade can perform is to increase the tangential velocity (or angular momentum) of the airflow; i.e., change the air's kinetic energy. Thus:

A rotating compressor blade passage (stage) imparts energy to the fluid (air) by increasing the fluid's angular momentum (torque) *(Figure 2-3)*.

On the other hand, stationary (stator) blades cannot impart energy to the flow. The only function stationary blades can perform is to redirect and/or decelerate the airflow. However, by decelerating the tangential velocity of the airflow, angular momentum is converted to pressure; i.e., kinetic angular energy is converted to potential pressure energy:

A stationary blade compresses the fluid (air) by decelerating it *(Figure 2-3)*.

Hence, a simplified explanation of how a gas turbine compressor works is as follows:

1. Air passes through the compressor rotating blades and is accelerated in the tangential direction; energy is added.

2. Airflow with a high tangential velocity component exits the rotating blade passage and enters the stationary blade passage. Both the kinetic energy and the static pressure have increased.

3. The stationary blades decelerate the tangential velocity component and the air's angular momentum is converted to pressure; i.e., kinetic energy is converted to potential energy.

In a multi-stage compressor, this three-step compression process is repeated until a desired compression ratio is achieved *(Figure 2-3)*. Multi-stage compressors will be discussed in more detail later in this section.

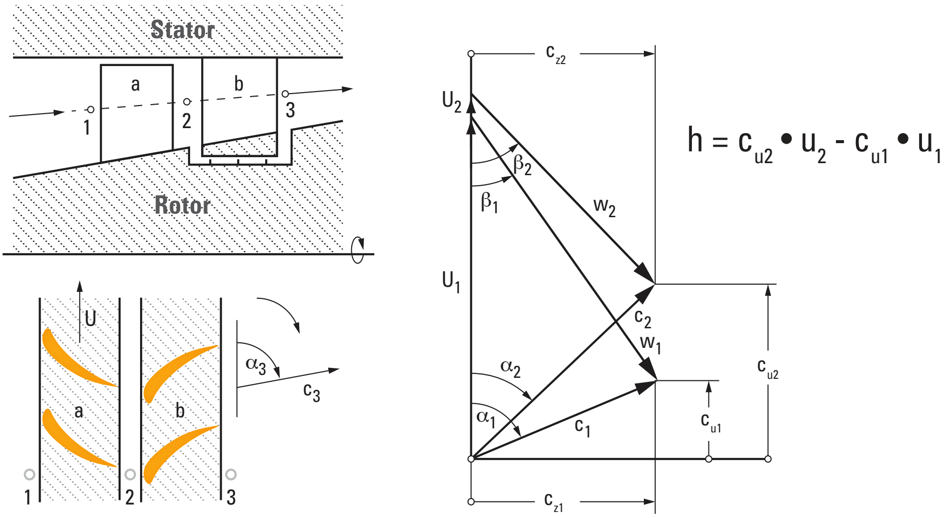


Figure 2-3. *Velocities in a Typical Compressor Stage. Mechanical work h transferred to the air is determined by the change in circumferential momentum of the air*

EULER'S ANGULAR MOMENTUM EQUATION

We just claimed that we can relate the tangential air velocity (cu), also called swirl, in the compressor blade passage to the air pressure increase. This is significant since it means that if blade geometry and compressor rotational speed are known, we should be able to estimate the blade passage compression ratio. Let's restate what we already know: "a rotating compressor blade passage (stage) imparts energy to the fluid (air) by increasing the fluid's angular momentum (torque)." We can express this statement in mathematical terms:

The increase in the tangential velocity component of the air is simply:

$$\Delta c_{u2} = c_{u2} - c_{u1}$$

The mathematical expression for angular momentum (torque) from basic physics is:

$$\tau = W \cdot r \cdot c_u$$

Since the blade imparts or extracts torque from the working fluid, the torque must equal the change of angular momentum of the working fluid. Thus, the change of angular momentum of the fluid can be written as:

$$\Delta\tau = W \cdot \Delta(r \cdot c_u) = W \cdot (r_2 \cdot c_{u,2} - r_1 \, c_{u,1})$$

Power (work/time) is defined as torque multiplied by the angular speed:

$$P = \omega \cdot \Delta\tau = \omega \cdot W \cdot \Delta(r \cdot c_u) = \omega \cdot W \cdot (r_2 \, c_{u,2} - r_1 \, c_{u,1})$$

This equation is called Euler's Angular Momentum equation and it is very important. The expression explains that we can determine the power required to drive a compressor from the tangential velocities of the flow in the compressor blade passages. Later, we will demonstrate how the tangential velocities can be calculated directly from the geometry of the compressor blades. From Euler's equation, we can also determine the head (enthalpy difference) per stage as follows:

$$H = h_2 - h_1 = \frac{P}{W} = \omega \cdot W \cdot (r_2 c_{u,2} - r_1 c_{u,1})$$

Since h = c_pT, we can estimate the temperature increase across the compressor stage:

$$T_2 = \frac{\omega}{c_p} \cdot \omega \cdot W \cdot (r_2 c_{u,2} - r_1 c_{u,1}) + T_1$$

Because this is a compressor, we are really more interested in the pressure increase than the temperature increase per stage. The isentropic relationships provide a function between pressure and temperature ratio; however, when employing the isentropic relationships in this context they apply for ideal (no losses) thermodynamic processes. In this case, the process is not ideal; thus, a compressor efficiency has to be introduced:

$$\eta_c = \frac{Ideal\ (Isentropic)\ Head}{Actual\ Head} = \frac{h_2^* - h_1}{h_2 - h_1} = \frac{T_2^* - T_1}{T_2 - T_1}$$

When combining the above efficiency with the expression for temperature and the isentropic relationship for temperature and pressure ($T_2/T_1 = (P_2/P_1)^{(\gamma/\gamma-1)}$), we obtain after some algebra:

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta\omega}{c_p T_1} \cdot \omega \cdot W \cdot (r_2 c_{u,2} - r_1 c_{u,1}) \right)^{\frac{\gamma}{\gamma - 1}}$$

This expression gives us the pressure ratio per stage as a function of the tangential air velocity. Using the expression for head, we can also rewrite this as follows (to relate head to pressure ratio):

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta}{c_p \cdot T_1} \cdot H\right)^{\frac{\gamma}{\gamma-1}}$$

The above equations are extremely powerful engineering tools. From basic knowledge of the tangential flow velocities inside the compressor rotating blade passage, we can determine the head, required power, temperature ratio and pressure ratio across a compressor stage. The obvious next question then is: How do we determine the tangential velocity components?

VELOCITY VECTOR POLYGONS

To determine the tangential velocity component or swirl of the airflow in the compressor, some simple rules from basic geometry and trigonometry are employed. Let us demonstrate using the axial compressor rotor/stator blade passage (stage) example presented below.

Axial Compressor Example

First, a typical rotor/stator blade stage of an axial compressor is separated into three states: flow before it enters the rotor cascade; flow exiting the rotor cascade and before entering the stator cascade; and flow exiting the stator cascade *(Figure 2-4)*. Next, the flow vector (direction and magnitude) is analyzed at each state, assuming that the flow follows the blades perfectly; i.e., no slip or a slip factor of unity. (**NOTE:** The slip factor is defined as a parameter describing how much the rotor exit flow angle deviates from the actual blade angle).

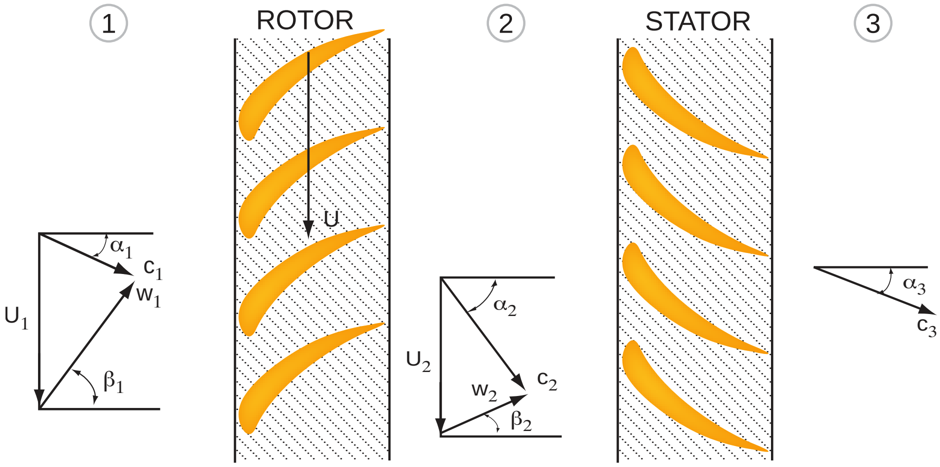


Figure 2-4. *Velocity Vector Polygons for Rotor-Stator Blade Passage*

State 1 - Flow Entering Rotor Cascade

Typically, the known information or boundary conditions are the inlet velocity vector, the blade rotating speed and the geometry of the blades:

Given: Inlet flow magnitude - c_1

Inlet flow angle - α_1

Blade radius - r_1

Blade angular speed - ω

The inlet flow into the compressor either enters with none or only a very small tangential velocity component (pre-swirl); hence, a_1 is very small. The blade speed (U) is determined from:

$$U = r_1 \cdot \omega$$

We can determine what the velocity vector from the stationary frame (c_1,α1) looks like when translated into the relative rotating frame (relative to the rotating blade) by using simple vector addition (U is a vector pointing in the tangential direction). A velocity vector triangle is obtained as shown above where the velocity vector in the rotating frame (w_1,β_1) is calculated using basic trigonometry. This method is called Velocity Vector Polygons. Note that velocity vectors relative to the stationary frame are denoted using symbols c,α (magnitude, direction), while velocity vectors relative to the rotating blade frame are denoted by w,β (magnitude, direction). For now, we are only interested in the tangential component of the through-flow velocity relative to the stationary frame, which is simply determined from:

$$V_{\theta 1} = c_1 \cdot \sin(\alpha_1)$$

State 2 - Flow Exiting the Rotor Cascade before Entering the Stator Cascade

Since we assumed there is no slip, the flow exiting the rotor blades must be perfectly aligned with the blades; i.e., the exit flow angle (β_2) is the rotor blade angle in the rotating frame. Also, the magnitude of the through flow vector (w_2) can be determined by applying the conservation of mass (continuity) equation:

$$W = \rho VA = constant$$

Thus, to determine the tangential component of the through flow exiting the rotor blades, the known rotating vector (w_2,β_2) has to be translated to the stationary frame to obtain the vector (c_2,α_2). As previously mentioned, this is done using vector addition and trigonometry. Once the vector (c_2,α_2) is known, the tangential component relative to the stationary frame is again obtained from:

$$c_{u,2} = c_2 \cdot \sin(\alpha_2)$$

State 3 - Flow Exiting the Stator Cascade

The flow exiting the stator cascade is assumed to follow the blades; therefore, the stator blade angle is the flow exit angle (α_2). The vector magnitude (c_3) is again obtained from the continuity equation. However, since the stator cascade does not add energy to the flow, there is no need to evaluate the tangential through-flow velocity component. The stator exit flow vector is determined only to obtain the inlet flow vector into the next rotor blade cascade.

Since the tangential velocity components of the flow inside the axial compressor have been calculated, Euler's Angular Momentum equation can be used to determine the pressure ratio, temperature increase, head and energy per stage as previously shown. For example, the equation for pressure ratio per stage becomes:

$$\frac{p_2}{p_1} = \left(1 + \frac{\eta\omega}{c_p T_1} \cdot (r_2 c_2 \sin(\alpha_2) - r_1 c_1 \sin(\alpha_1) \right)^{\frac{\gamma}{\gamma-1}}$$

MULTI-STAGE AXIAL COMPRESSOR

In a multi-stage compressor, the air passes through a number of rotor/stator cascades or stages, each of which increases the overall air pressure to achieve a desired total compression ratio. *Figure 2-5* shows a typical gas turbine axial compressor with all relevant flow elements. Usually, there are between 10 to 17 stages in a gas turbine axial compressor with each stage achieving a pressure ratio of 1.1 to 1.4. The total compression ratio is calculated by simply multiplying the individual stage compression ratios:

$$\frac{P_n}{P_1} = \frac{P_2}{P_1} \cdot \frac{P_3}{P_2} \cdot \frac{P_4}{P_3} \cdots \frac{P_n}{P_{n-1}}$$

where the subscript index is the stage number and n is the total number of stages.

The more stages a compressor has, the higher the total achievable pressure ratio. We previously determined that the gas turbine thermal efficiency is directly related to the pressure ratio: as the pressure ratio increases, so does the gas turbine efficiency. So why not just add more and more stages to the compressor to improve its efficiency?

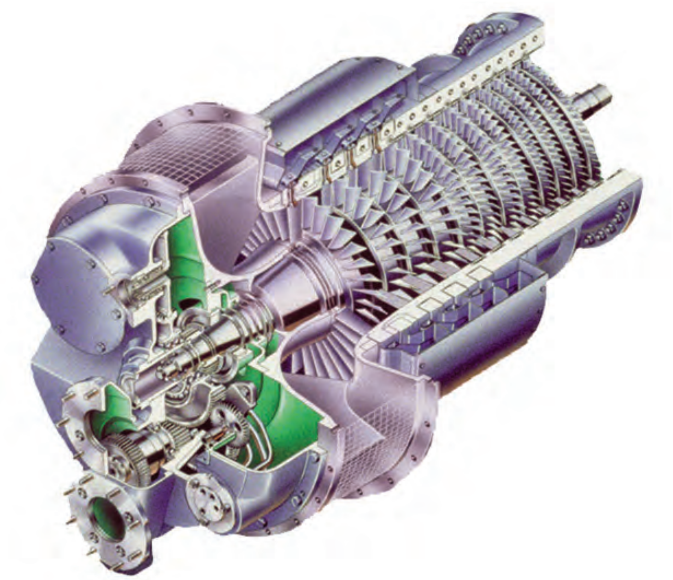


Figure 2-5. *Multi-Stage Axial Compressor*

There are a number of good answers; however, the primary reason is that gas turbine efficiency increases with pressure ratio, but not necessarily the total output power. As the compression ratio increases, the gas producer turbine must absorb more energy from the combustor exhaust flow, and less energy remains to drive the power turbine and create useful shaft output power. Consequently, for every gas turbine design there must be an optimum design with an optimum number of compressor stages. It is the design engineer's difficult job to evaluate the multitude of variables that affect gas turbine performance and obtain an optimal compressor stage combination for a particular gas turbine design.

MULTI-STAGE AXIAL COMPRESSOR FLOW

To reinforce the understanding of some of the above concepts, pressure and velocity of the air as it passes through an actual axial compressor is plotted. A cross-sectional view of the axial compressor with the corresponding pressure/velocity trend plot is shown in *Figure 2-6*. Within the inlet section (inlet guide vanes), the air pressure decreases slightly while the velocity increases; namely, the flow is redirected to match the rotor blade angle and is slightly accelerated.

As the flow passes through the rotor blade passage, both the velocity and pressure increase, adding energy to the flow. In the subsequent stator blade passage, the air velocity is converted to pressure: air is decelerated and pressure increases. This process is repeated by each rotor/stator stage such that at the compressor exit the desired pressure is reached, while the total velocity remains low.

Note that the compressor blade span (its radial length) decreases along the axial direction of the compressor; thus, the compressor blades get shorter and the through-flow area smaller, the further the air passes through the compressor. The reason this occurs is that as the air is compressed by the rotor/stator stages and it requires less volume and less through-flow area, the blade span must decrease. This is another factor that can limit the pressure ratio, since the rear stages eventually become so short that efficiency (and pressure ratio pre stage) are impacted.

It is important to clarify and emphasize this cause/effect relationship. The flow is not being compressed because the through-flow area decreases. On the contrary, the through-flow area must be designed to decrease because the flow is being compressed by the rotor/stator blade cascades and thus requires less volume.

MULTI-STAGE AXIAL COMPRESSOR EXAMPLE

Since the concept of head and pressure ratio in multi-stage compressors is very important, we will further demonstrate it using a simple mathematical example. Previously in this text, we defined head as simply an enthalpy (energy) difference: $H = \Delta h = c_p \Delta T$. We also mentioned earlier that each stage in a multi-stage compressor increases the air pressure by a certain pressure ratio; i.e., each stage increases the air's enthalpy and thus, produces head. The total head (H) of the air exiting the multi-stage compressor is obtained by adding all individual stage heads ($H = H_1 + H_2 + H_3 \ldots + H_n$). Let us employ Euler's Angular Momentum equation to determine total head and pressure ratio for a simplified multi-stage compressor example.

Multi-Stage Axial Compressor Example

We will assume that all rotor and stator blades are perfectly axial and the flow again follows the blades perfectly (no slip). A simplified schematic of the compressor blades and the rotor/stator blade rows divided into states is shown in *Figure 2-7*. Now we can analyze the flow and total head at each state.

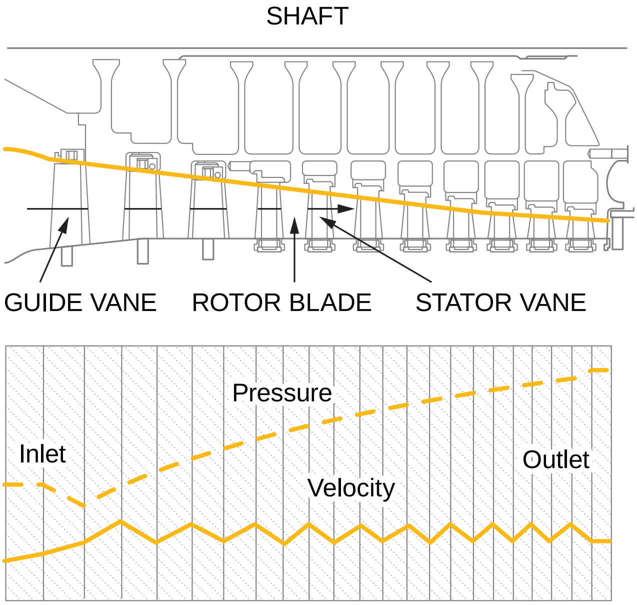


Figure 2-6. *Pressure and Velocity in a Multi-Stage Axial Compressor*

State 1: The flow enters the compressor axially and thus, has no tangential velocity component; i.e.:

$Cu_1 = 0$

therefore, the head is also zero, $H_1 = 0$

State 2: Since the flow follows the rotor blades perfectly, the tangential velocity component of the flow exiting the rotor must be equal to the rotor blade velocity (U); i.e.,

$cu_2 = U \approx \omega \cdot r$

therefore, the new head is calculated from Euler's Angular Momentum equation:

$H_2 = \omega \cdot (r_2 cu_2 - r_1 cu_1) = \omega \cdot (r_2 cu_2 - 0) \approx \omega^2 r^2$

The total head is the previous head plus the new head:

$H = H_1 + H_2 \approx \omega^2 r^2$

State 3: The flow's tangential velocity component is decelerated to zero in the stator blades such that:

$cu_3 = 0$, thus, $H_3 = 0$

The new, total head is determined by adding the previous heads:

$H = H_1 + H_2 + H_3 \approx \omega^2 r^2$

State 4: The flow exits the second rotor/stator stage, thus (identical to state 2):

$cu_4 = U = \omega \cdot r, H_4 = \omega^2 r^2$

The total head is the previous head plus the new head:

$H = H_1 + H_2 + H_3 + H_4 = 2\omega^2 r^2$

State 5: $cu_5 = 0$, thus, $H_5 = 0$

$H = H_1 + H_2 + H_3 + H_4 + H_5 = 2\omega^2 r^2$

State 6: $cu_6 = U = \omega r,$

$H_6 = \omega^2 r^2$

$H = H_1 + H_2 + H_3 + H_4 + H_5 + H_6 = 3\omega^2 r^2$

State 7: $cu_7 = 0$, thus, $H_7 = 0$

$H = H_1 + H_2 + H_3 + H_4 + H_5 + H_6 + H_7 = 3\omega^2 r^2$

This calculation is repeated until the total number of the axial compressor stages has been reached. The above expressions for total head can also be simplified by rewriting them as:

$H = H_2 + H_4 + H_6 + \dots H_n = n \times \omega^2 r^2$

The total pressure ratio can be determined from:

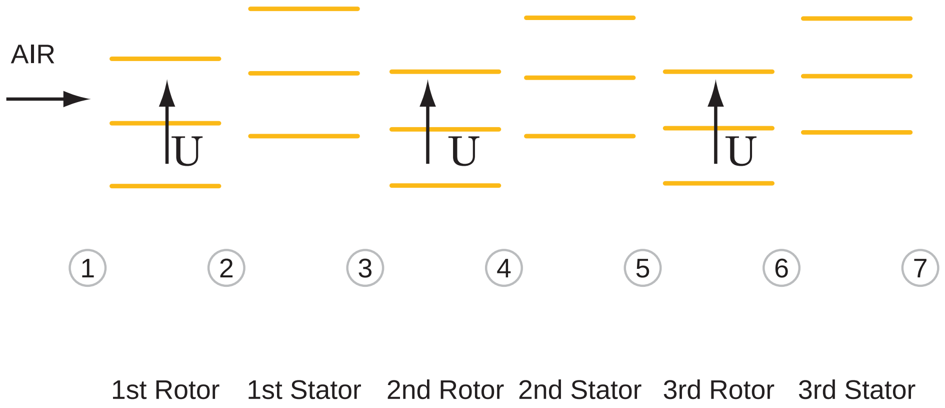


Figure 2-7. Simplified Multi-Stage Axial Compressor Example

$$\frac{P_n}{P_1} = \left(1 + \frac{\eta}{c_p \cdot T_1} \cdot H\right)^{\frac{\gamma}{\gamma-1}} = \left(1 + \frac{\eta}{T_1} \cdot n \frac{\omega^2 r^2}{c_p}\right)^{\frac{\gamma}{\gamma-1}}$$

Here the subscript index is the stage number and n is the total number of stages. Some real numbers should be applied to the above equation to evaluate how accurate this expression is. For example, a typical 15,000-hp gas turbine has approximately the following operational and design parameters:

Compressor Efficiency: $\eta = 0.85$

Inlet Temperature: $T_1 = 100$ F = 311 K

Number of Stages: n = 15

Angular Speed: $\omega = 600$ 1/s

Average Blade Radius: r = 0.4 m

Specific Heat: $c_p = 1007$ J/ kg · K

Specific Heat Ratio: $\gamma = 1.4$

Plugging these numbers into the above equation:

$$\frac{P_{15}}{P_1} = \left(1 + \frac{0.85}{311} \cdot 15 \cdot \frac{600^2 \cdot 0.4^2}{1007}\right)^{\frac{1.4}{0.4}} = 16.9$$

The actual compression ratio of this 15,000-hp gas turbine compressor is $P_{15}/P_1 = 17.1$, compared to 16.9 calculated above. Clearly, the simple equation derived in the above example is adequate for rough estimates of gas turbine compressor performance.

COMPRESSOR BLADE SPAN

As mentioned previously, the span of the compressor blades decreases in the axial direction *(Figure 2-8)*. The reason for this is, that as the air is being compressed it requires less volume, thus less through-flow area, and the blade span must decrease. Based on the equation for the pressure ratio per stage from the previous simplified example, an expression can be developed that defines how much the blade span and through-flow area must decrease per stage.

Remember that the pressure ratio per stage is:

$$\frac{P_N}{P_1} = \left(1 + \frac{\eta}{T_1} \cdot \frac{\omega^2 r^2}{c_p}\right)^{\frac{\gamma}{\gamma-1}}$$

We also know from the isentropic expressions that pressure ratio and density ratio are related by:

$$\frac{P_2}{P_1} = \left(\frac{\rho_2}{\rho_1}\right)^{\gamma}$$

Combining the two equations, the result is:

$$\frac{\rho_2}{\rho_1} = \left(1 + \frac{\eta}{T_1} \cdot \frac{\omega^2 r^2}{c_p}\right)^{\frac{1}{\gamma-1}}$$

Finally, in the gas turbine compressor, mass is conserved (continuity equation):

$$W = \rho \cdot A \cdot V = constant$$

It can be rewritten as:

$$\frac{\rho_2}{\rho_1} = \frac{W_2}{A_2 V_2} \cdot \frac{A_1 V_1}{W_1}$$

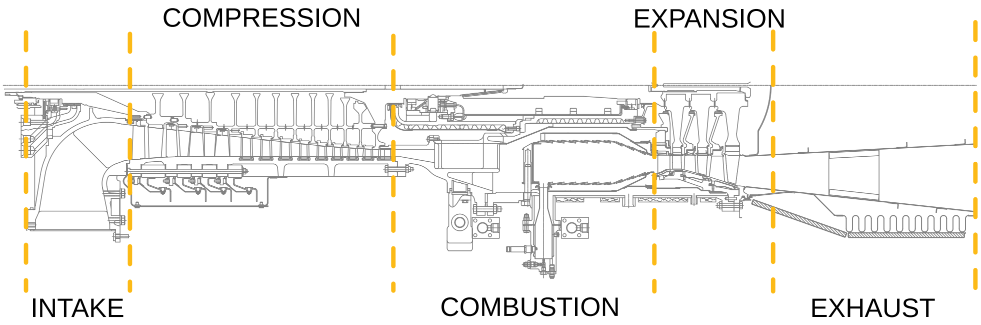


__Figure 2-8.__ Blade Span: Compressor Decreasing, Turbine Increasing

We already established that the through-flow velocity in the compressor remains approximately constant, V2»V1, and the mass flow is constant, W1 = W2 , thus:

$$\frac{\rho_2}{\rho_1} \approx \frac{A_1}{A_2}$$

The through-flow area (A) is related to the blade span (s) using simple geometry by $A \propto \pi s^2$. Combining the above equations, the following is obtained for the through flow area ratio:

$$\frac{A_2}{A_1} \approx \left(1 + \frac{\eta}{T_1} \cdot \frac{\omega^2 r^2}{c_p} \right)^{\gamma\text{-}1}$$

or for the blade span ratio:

$$\frac{s_2}{s_1} \approx \left(1 + \frac{\eta}{T_1} \cdot \frac{\omega^2 r^2}{c_p} \right)^{2\cdot(\gamma\text{-}1)}$$

The previous expression tells us how much the through-flow area and blade span must decrease per stage in an axial compressor. This expression is a higher order polynomial, which means that the ideal flow path would be curved; i.e., non-linear in the functional form of $s = c^{2\cdot(\gamma-1)}$. However, to simplify manufacturing, in all actual gas turbines the flow path is shaped as one or multiple straight-line segments as shown in *Figure 2-8*.

AXIAL VERSUS CENTRIFUGAL COMPRESSORS

Most modern gas turbines use axial compressors; however, on some smaller gas turbine designs, centrifugal compressors are employed. The applicability, advantages and disadvantages of each design will be discussed briefly.

While an axial compressor can be recognized by rows of airfoil-like blades, the centrifugal compressor has wheels that have an axial inlet and, more or less, radial exit *(Figure 2-9)*. In analyzing Euler's Angular Momentum equation:

$$H = \omega \cdot (r_2 c_{u,2} - r_1 c_{u,1})$$

We realize that there are two different methods to augment airflow head passing through a compressor rotor: by increasing the tangential component of the velocity (swirl - cu_2); or increasing the radius (r_2) of the rotor (impeller) exit. The outcome is principally the same—the tangential momentum of the airflow is increased between the inlet and exit of the rotor. However, while the axial compressor is only able to alter the swirl, a centrifugal compressor, which has inlet and outlet at different radii, takes advantage of both effects *(Figure 2-10)*.

Thus:

Per stage, a centrifugal compressor creates more head than an axial compressor.

On the other hand, the flow channels, through which the flow has to be compressed, are much narrower in a centrifugal compressor than in an axial compressor. Thus, the axial compressor has a larger through-flow area and can compress a larger flow volume for a given frontal area than a centrifugal compressor. For the same reason, an axial compressor incurs less friction losses than a centrifugal compressor. Thus:

An axial compressor allows higher through flow than a centrifugal compressor.

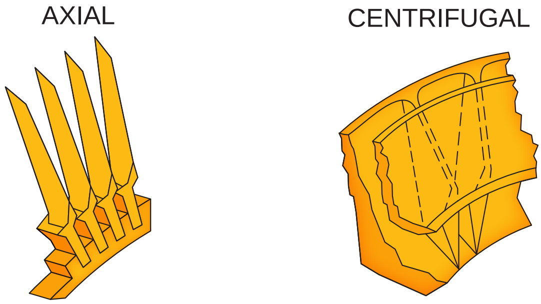


Figure 2-9. *Axial vs Centrifugal Compressor*

The previous two statements indicate the applicability of each compressor design. Centrifugal compressors are typically found in low-flow and high-pressure ratio applications, such as in gas pipelines, gas storage, oil and gas well service, chemical processing and gas refrigeration. Axial compressors are typically found in high-flow and low-stage pressure ratio applications where there is also a greater emphasis on efficiency. On virtually all industrial gas turbines, with the exception of some very small engines, axial compressors are employed.

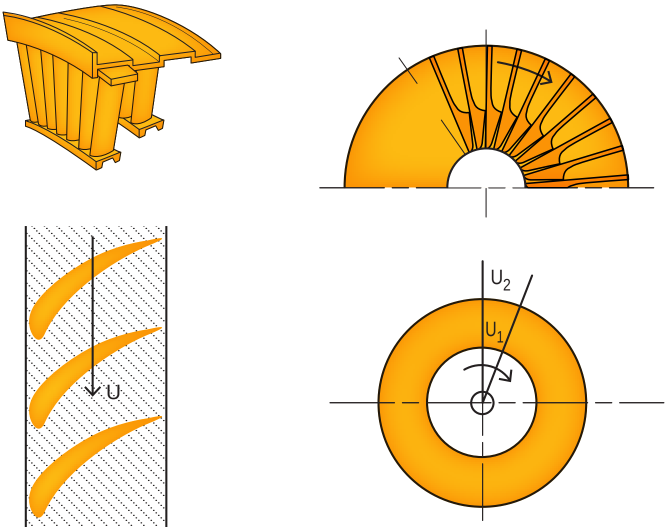


Figure 2-10. *Function of Axial and Centrifugal Compressors*

COMBUSTOR

The next principal aerodynamic component the airflow encounters after exiting the compressor is the combustor, sometimes referred to as the burner. The function of the combustor is to add heat energy to the flow, thus, increasing the temperature of the air passing through the combustor.

A typical, annular combustor design with its internal airflow patterns is shown in *Figure 2-11*. In this section, combustor aerodynamics, combustion chemistry, stoichiometry and design will be discussed in some detail.

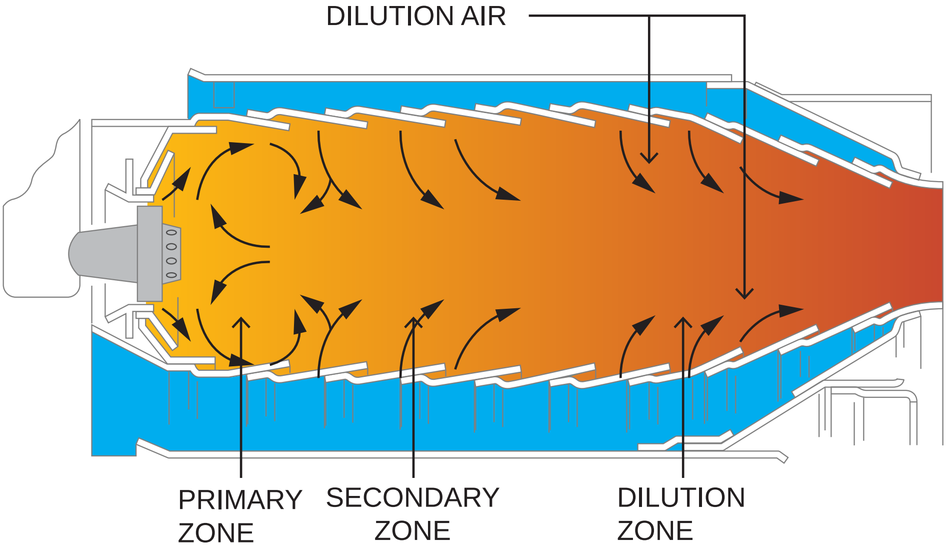


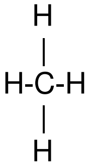
Figure 2-11. *Vortex-Flame Combustor*

COMBUSTION REACTION

Gas turbine combustion is a steady-state flow process in which a hydrocarbon fuel is burned with air to achieve a desired combustor firing temperature. Hydrocarbon fuels typically employed in this process are either a natural gas or liquid fuel such as diesel or kerosene.

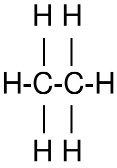
Most hydrocarbons burn in the form of an exothermic chemical reaction: heat is released during the combustion reaction. The airflow through the combustor provides the required oxygen for this reaction (air consists of approximately 23% oxygen and 76% nitrogen by weight). Let us briefly discuss the reaction chemistry of natural gas, the most commonly employed gas turbine fuel for industrial applications.

Natural gas consists of a combination of hydrocarbon gases and occurs as a natural underground deposit in differing compositions throughout the world. Natural gas usually contains the following primary combustibles: methane (CH_4), ethane (C_2H_6), propane (C_3H_8), butane (C_4H_{10}), pentane (C_5H_{12}); basically gases made up of carbon (C) and hydrogen (H) in the chemical form of $C_nH_{(2+2n)}$. Natural gas also often contains some nitrogen (N_2), carbon dioxide (CO_2), water (H_2O) and contaminants such as hydrogen sulfide (H_2S). In most cases, the CH_4, C_2H_6, and C_3H_8 content of natural gas is above 75%, making it an excellent combustible gas.



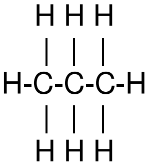
CH_4

Methane



C_2H_6

Ethane



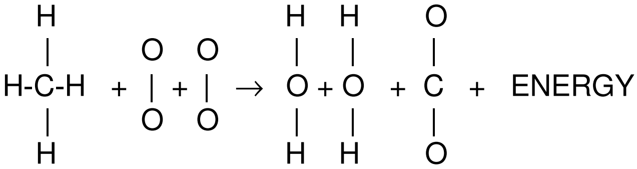
C_3H_8

Propane

NOTE: Theoretically, most hydrocarbon gases could be converted into liquid fuel by binding OH molecules, changing them into alcohols. For example, methane (CH_4) becomes liquid methanol (CH_3OH), which also is called methyl alcohol when adding OH. Similarly, ethane (C_2H_6) is converted to ethanol (C_2H_5OH) also called ethyl alcohol. Both methanol and ethanol are sometimes employed as liquid fuels for gas turbines; however, the conversion of natural gas to alcohols requires a forced chemical reaction that is complicated and rather costly.

The combustion process is a chemical oxidation reaction involving natural gas (or liquid fuel) and air. However, to correctly design the combustor, the actual proportion of air and fuel required needs to be known. What is the air-to-fuel ratio required to obtain a complete reaction between the fuel gas and the airflow? Let us first analyze this problem using a simplified example of methane/air combustion; the results will then be generalized to apply for any natural gas.

Assuming the natural gas fuel is made up only of methane (CH_4) and that it will react only with the oxygen (O_2) in the air, we can easily deduct from basic chemistry:



Methane Oxygen Water Carbon Dioxide

or in equation form:

$CH_4 + 2\ O_2 \rightarrow 2\ H_2O + CO_2 + \text{ENERGY}$

This is an exothermic reaction since heat is released (heat absorption reactions are called endothermic). The molecular weights of CH_4 and O_2 are:

$CH_4 = 16$

$O_2 = 32$

Hence, the weight ratio of O_2 to CH_4 for a theoretically ideal and complete combustion reaction, also called stoichiometric combustion, must be:

$$\left(\frac{Oxygen}{Methane}\right)_{weight} = \frac{2 \cdot 32}{16} = 4.0$$

Since, approximately 23% of air is O_2, the ideal stoichiometric combustion air-to-fuel ratio by weight becomes:

$$\left(\frac{Air}{Fuel}\right)_{weight} = \frac{4.0}{0.23} = 17.4$$

Thus, for one kilogram of CH_4 fuel flowing into the combustor, a minimum of 17.4 kilograms of airflow is required to achieve stoichiometric combustion.

COMBUSTION CHEMISTRY

Based on the previous example, a more generalized combustion equation for natural gas and air can be assumed. The combustion equation expresses the conservation of mass in molecular terms; namely, the equation emulates the rearrangement of the molecules that occurs during the reaction. Thus, for any natural gas that consists of a $C_nH_{(2+2n)}$ composition, the O_2 required for ideal stoichiometric combustion can be found from the general combustion equation:

$$C_uH_m + (u+m/4)\ O_2 \rightarrow u\ CO_2 + (m/2)H_2O + Energy$$

Using the above equation, the amount of O_2 required for stoichiometric combustion can be determined for any natural gas. For example, for a 50% CH4 and 50% C2H6 natural gas mixture, the following is obtained:

Methane/Oxygen Reaction: $CH_4 + 2\ O_2 \rightarrow 2\ H_2O + CO_2 + ENERGY$
Ethane/Oxygen Reaction: $2\ C_2H_6 + 7\ O_2 \rightarrow 6\ H_2O + 4\ CO_2 + ENERGY$

The molecular weights are: $CH_4 = 16$, $C_2H_6 = 30$, $O_2 = 32$. Hence, the fuel-oxygen ratios are:

$$\left(\frac{Oxygen}{Methane}\right)_{weight} = \frac{2 \cdot 32}{16} = 4.0$$

$$\left(\frac{Oxygen}{Ethane}\right)_{weight} = \frac{7 \cdot 32}{2 \cdot 30} = 3.73$$

The resulting air-to-fuel ratio for a 50% CH_4 and 50% C_2H_6 gas mixture combustion is:

$$\left(\frac{Air}{Fuel}\right)_{weight} = \frac{0.5\,(4.0 + 3.73)}{0.23} = 16.8$$

Based on this simple method the air-to-fuel ratio for any natural gas stoichiometric combustion reaction can be determined. We see that the 50% CH_4 and 50% C_2H_6 mixture air-to-fuel ratio is lower than the one for pure CH_4 fuel gas. Generally, if the content of higher order $C_nH_{(2+2n)}$ (heavier) components in the natural gas increases, the required stoichiometric air-to-fuel ratio decreases.

COMBUSTION BY-PRODUCTS

Since air contains 76% nitrogen and 23% oxygen, each kilogram of oxygen must be accompanied by 76/23 = 3.3 kilograms of nitrogen, which is ideally considered to be inert and should appear unchanged in the exhaust gas. However, in extremely high temperature regions of the combustor primary zone, small amounts of oxides of nitrogen (NO_x), which are considered to be harmful pollutants, may be formed. Also, in the above stoichiometric combustion equation, complete reaction of the carbon (C) to carbon dioxide (CO_2) is assumed. However, incomplete combustion can result in small amounts of carbon monoxide (CO) and unburned hydrocarbons (UHC) also being present in the exhaust gas. Depending on the relative humidity, the combustion air also contains a certain amount of water. Finally, the gas turbine provides a large quantity of excess air, resulting in a considerable amount of unburned oxygen in the exhaust air.

Consequently, the exhaust gas of any gas turbine consists primarily of CO_2, H_2O, unburned O_2 and N_2. CO_2 is considered a greenhouse gas. Unlike other pollutants like CO and NO_x, CO_2 is an unavoidable by-product of the combustion reaction. The only method of reducing CO_2 is by increasing the thermal efficiency of the gas turbine.

Beyond the desired natural gas/oxygen exothermic reaction, a number of undesired sub-reactions may occur in the combustor. Undesired sub-reactions are reactions that involve either a heat loss (endothermic reactions) and/or the creation of pollutants such as NO_x and CO. To summarize these undesired sub-reactions:

Endothermic Reactions (heat loss only):

$O_2 + Energy \rightarrow 2\,O$

$N_2 + Energy \rightarrow 2\,N$

Pollution and Endothermic Reaction (NO_x creation):

$N_2 + O_2 + Energy \rightarrow 2\,NO$

$N_2 + 2\,O_2 + Energy \rightarrow 2\,NO_2$

Pollution and Endothermic Reaction (CO creation):

$2\,CO_2 + Energy \rightarrow 2\,CO + O_2$

These sub-reactions only occur in very high temperature regions of the combustor. Thus, to limit these undesirable reactions, a uniformly distributed temperature combustion process is desired. Over the next couple of pages, we will discuss how this and other combustion requirements are achieved.

COMBUSTION PROPERTIES

We previously described how to determine the stoichiometric air-to-fuel ratio (by weight) using concepts from basic chemistry. In a real gas turbine, significant amounts of excess air are available from the axial compressor. While the stoichiometric combustion air-to-fuel ratio is usually below 20 for natural gas, a real gas turbine has approximately three to six times more compressed air available than the stoichiometric requirements; i.e., the air-to-fuel ratio based on the total airflow through the combustor (also called total air-to-fuel ratio) is typically between 50 and 100. This excess air is not employed in the primary combustion process but is ducted into the combustion chamber to cool the hot combustion product gases and/or is bypassed directly into the turbine section.

There are two different definitions related to combustor temperatures:

Firing Temperature (Turbine Inlet Temperature, Combustor Exit Temperature): The firing temperature is the average temperature of all the gas leaving the combustion chamber and entering the turbine section. Knowledge of the firing temperature is critical for the analysis of gas turbine performance and turbine blade design.

Flame Temperature: The flame temperature is the peak temperature in the actual primary combustion zone. This temperature is required to evaluate the chemical disassociation reactions that occur in the primary combustion zone.

Both the flame and firing temperatures can be estimated from air-to-fuel ratios. The stoichiometric air-to-fuel ratio is employed to estimate the flame temperature, while total air-to-fuel ratio can be employed to evaluate the firing temperature. Let us demonstrate:

Firing Temperature
From the previous thermodynamics discussion, we know that the energy across the combustor must balance, namely:

$$W_{fuel} \cdot q_R = W_{air} \cdot (h_3 - h_2)\eta_B$$

W_{air}/Wfuel is the air-to-fuel ratio (by weight). We can re-write this equation to:

$$h_3 = c_{p3} T_3 = \frac{q_R}{(Air / Fuel)_{weight} \cdot \eta_B} + c_{p2} \cdot T_2$$

And further simplifying this expression by assuming that η_B = 1.0 and $c_{p3} = c_{p4}$, we get:

$$T_3 = \frac{q_R}{(Air / Fuel)_{weight} \cdot c_p} + T_2$$

where T_3 is the compressor exit air temperature.

This simple expression allows the estimation of the combustor firing temperature (T_4). For example, assuming a total air-to-fuel ratio of 65 and pure methane combustion (lower heating value of methane is q_R = 48,000 kJ/kg, specific heat of the exhaust gas 1.3 kJ/kgK), the above simple expression results for a typical gas turbine in a firing temperature of:

$$T_3 \approx \frac{48{,}000\ kJ/kg}{65 \cdot 1.3\ kJ/kgk} + 700K \approx 1270K$$

Hence, for this example the air/gas mixture that is exiting the combustor and entering the turbine has an average temperature of 1270 K (1830°F).

Flame Temperature

If we employ the stoichiometric air-to-fuel ratio in the above equation, the approximate flame temperature is determined. Let us assume pure methane combustion, which we previously showed to have a stoichiometric air-to-fuel ratio of 17.6:

$$T_3 \approx \frac{48{,}000\ kJ/kg}{17.6 \cdot 1.34\ kJ/kgk} + 700K \approx 2700K$$

The peak flame temperature in the primary combustion zone is thus approximately 2700 K. This temperature is very close to the flame temperatures seen in conventional combustion systems with diffusion flames, where the flame temperature is about at 2200-2500 K (3500-4000°F). Clearly, this flame temperature is significantly hotter than the firing temperature. The temperature obtained by the previous expression assumes an adiabatic flame and does not take the effects of dissociation and heat transfer into account. Note that the specific heat (c_p = 1.34 kJ/kgK) is greater than in the example for the firing temperature because the specific heat is a function of the temperature and the gas composition. It is very important to design the combustion chamber in such a way that this extremely hot flame does not come into direct contact with any part of the combustor; the flame touching the combustor liner would create a hot spot that can rapidly cause combustor liner degradation and subsequent material failure.

NOTE: For dry low NO_x combustors, the flame temperature is typically limited to below 1900 K (using higher primary zone air-to-fuel ratios) to reduce the formation of nitrous oxides (NO_x pollution). This is achieved by burning a mixture with a lower than stoichiometric fuel-to-air ratio. However, the firing temperature in dry low NO_x combustors is approximately the same as in conventional combustors. This will be discussed later in this book.

Flame Speed and Auto-ignition Delay Time

For complete combustion to occur, the mixture of air and fuel must stay in the combustion chamber at high temperatures long enough, so that all fuel can react with the available oxygen in the air. The combustion transient time must be sufficient, so that all fuel is completely burned out. Otherwise combustion and heat release may occur downstream in the turbine section, causing damage to the turbine nozzles and blades. Also, if the temperature of the reaction is dropped too quickly by the cooling air entering the combustion chamber in the secondary or dilution zones, the reaction may be quenched and the results would be unburned hydrocarbons and CO gases in the gas turbines exhaust mixture. On the other hand, if the fuel and air mixture reacts too quickly, then heat may be released near the combustor inlet, which could result in damage to the injector, air-premixer, or combustor liner. Thus, proper placement of the flame in the center of the combustor is critical for a durable burner design.

The air-fuel mixture's properties that primarily determine the combustor flame placement are the mixture's flame speed and the auto-ignition delay time. Flame speed is simply the velocity at which a flame front travels inside a combustible mixture and is a function of local temperature, pressure, and gas mixture composition. In the gas turbine's combustor, the flame front will be stable at a place where the mixture's bulk flow velocity coincides with its flame speed. Auto-ignition delay time defines the time that it takes a combustible mixture to start to react once it is exposed to ignition temperatures. Auto-ignition delay is also a function of pressure, temperature and mixture composition, but tends to be short for light combustible gases, such as hydrogen, and high for liquids fuels in droplet form. The combustor design must take both flame speed and auto-ignition delay time into consideration, so that the flame front is stable and placed at a location in the middle of the combustor, away from any walls, fuel injectors, or nozzles.

REQUIREMENTS FOR COMBUSTION

As previously mentioned, a uniform temperature distribution is desirable for complete stoichiometric combustion and to minimize pollutant formation. The following are additional requirements for complete combustion:

To ensure that hydrocarbon fuels in the combustor are completely burned, all regions of the flame have to be saturated with excess oxygen. This is achieved by injecting large quantities of air into the combustion chamber and providing proper fuel/air mixing.

Complete combustion requires time. Hence, the fuel/air mixture must have sufficient transient time passing through the combustor to completely react. To achieve this, the airflow through the combustor is slowed down by expanding (widening) the through-flow area. In a typical industrial gas turbine, the combustor through-flow (cross-sectional) area is approximately twice as large as that of its turbine or compressor.

The above can be summarized into the three principal gas turbine combustion requirements:

1. Proper fuel/air mixing with excess available air.
2. Uniform temperature distribution.
3. Sufficient transient chemical reaction time.

Some other requirements for proper combustor design include:

- Prevention of heat damage to the turbine blades. The temperature of the gases exiting the combustor must be considerably lower than the actual combustor flame temperature.

- Maintaining a stable flame. The airstream velocity around the combustor nozzles must be maintained steady with no significant flow fluctuations.
- Preventing damage to the combustor lining. The core of the flame must be directed away from the walls toward the center of the combustor.
- Achieving a high overall gas turbine efficiency. The airflow pressure drop across the combustor has to be kept low; the pressure ratio (B_b) should be close to unity.

The gas turbine combustion engineer must integrate design requirements that are sometimes contrary in nature. For example, achieving a uniform temperature distribution in the combustor is nearly impossible, if the airflow is non-uniform. However, by actively mixing the fuel/air flows, a non-uniform flow is created. Similarly, the aerodynamic mixing of the fuel/air flows results in a pressure drop across the combustor, as previously noted was undesirable. Consequently, the design of a gas turbine combustion system is a difficult challenge, which often requires engineering compromises between many design objectives. In the following pages, one commonly employed gas turbine combustion design, the vortex-flame annular combustor, will be described.

VORTEX-FLAME ANNULAR COMBUSTOR

The principal idea behind the vortex-flame annular combustor is that a large airflow vortex is maintained in the combustor core, which results in a stable flame, good mixing of the airflow with the fuel and an increased transient time of the air in the combustor *(Figure 2-12)*.

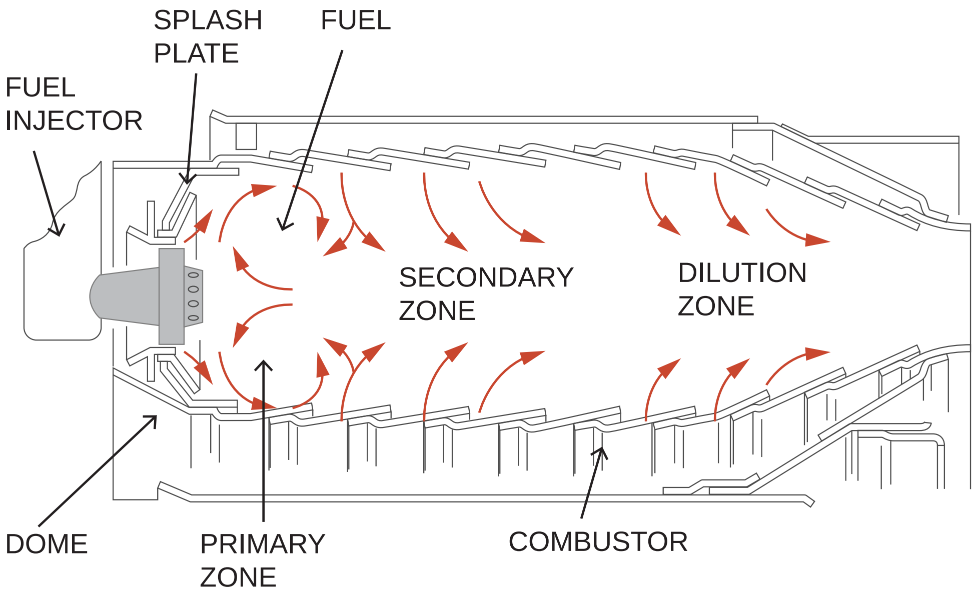


Figure 2-12. *Vortex-Flame Annular Combustor*

Primary air is the combustion air introduced through the dome, head plate or injector and through the first row of liner air holes. This air mixes with the incoming fuel and is ignited. Intermediate air is introduced through a second row of liner holes to complete the reaction in the secondary zone. Finally, in a downstream set of holes, dilution air is added to lower the combustor exhaust temperature.

The key feature of the primary zone is the flow recirculation or vortex region. Air entering the combustor moves an order of magnitude faster than the theoretical flame speed, thus, no stable combustion is possible. However, by using a recirculating-flow vortex, which redirects some of the burning mixture back towards the incoming fuel-air flow, the total flow moves slower, slow enough to allow for a stable combustion process. Obviously, the recirculation vortex also enhances the fuel-air mixing and increases the mixture's transient time in the combustor. A recirculation vortex can be created either by introducing the combustion air with a pre-swirl using swirl vanes, which cause a region of low pressure along the axis of the resulting vortex, or by radially injecting air through holes downstream of the flame *(Figure 2-13)*.

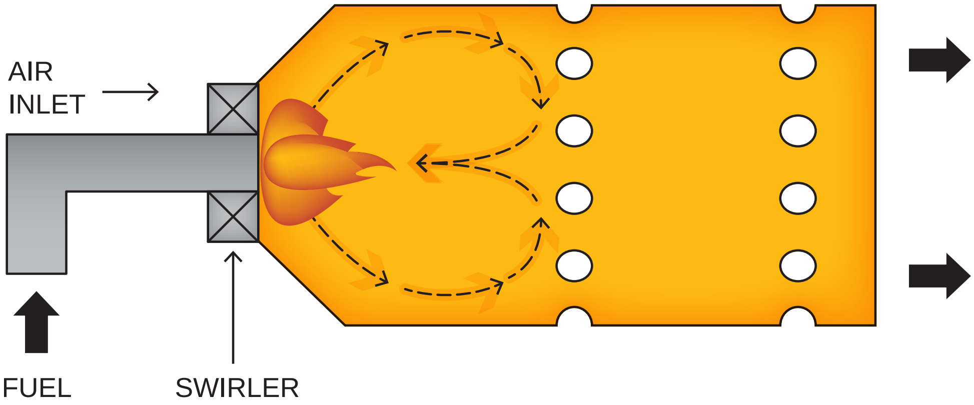


Figure 2-13. *Vortex Flow Pattern in Gas Turbine Combustor*

Most of the chemical combustion reactions take place in the primary zone. In a gas turbine combustor, the temperature of the hot gases in this zone can reach to above 2200°C (4000°F), depending upon the compressor discharge air temperature, pressure, fuel type and fuel/air ratio. The hot gases from the combustor primary zone enter a secondary zone where the temperature is reduced to approximately 1550°C (2822°F) by the gradual addition of compressor discharge air. This zone serves primarily to enhance lower temperature combustion sub-reactions that were not completed in the primary zone. Some chemical reactions require temperatures lower than those found in the primary zone. For example, the chemical re-association of CO to CO_2 ($2\ CO + O_2 \rightarrow 2\ CO_2 + \text{Energy}$) can only occur in lower temperature combustion regions. In some combustors with lower primary zone temperatures, this secondary zone is entirely omitted. Also, UHCs are significantly reduced in the secondary combustion zone.

Downstream of the secondary zone, the hot gases enter the dilution zone where they mix with the remaining compressor discharge air. This zone's function is to control the turbine inlet temperature requirements: average and local gas temperatures have to be lowered to acceptable levels. Significant heat damage can result in the turbine if the combustor exhaust temperature is not properly controlled.

COMBUSTOR EFFICIENCY

The overall performance of a combustor is gauged by the combustion efficiency (η_b) and the combustor pressure ratio (π_b). While the combustor pressure ratio is calculated from the exhaust over the inlet pressure (p_e/p_i), the combustion efficiency is a slightly more complicated expression that describes the completeness of the conversion of chemical fuel energy into heat energy ($(dm/dt)_{fuel} \cdot q_R$):

Combustor Pressure Ratio:

$$\pi_b = P_e/P_i$$

Combustion Efficiency:

$$\eta_b = \frac{Energy\ Out}{Fuel\ Energy} = \frac{W_{air} \cdot c_p \cdot (T_e - T_i)}{W_{fuel} \cdot q_R}$$

The combustor pressure ratio (π_b) is indicative of the pressure loss of the flow passing through the combustor. Ideally, this pressure ratio should be unity (no pressure losses). However, to achieve complete stoichiometric combustion requires expanding, mixing and recirculating the flow, all of which will create significant pressure losses. Typically, the pressure ratio for a vortex-flame annular combustor is between 0.90 and 0.95. However, other designs, such as the multiple-can combustor, can have pressure ratios as low as 0.80.

Combustion efficiency (η_b) expresses the ability of the combustor to convert chemical fuel energy into heat. Most modern gas turbine combustors have combustion efficiencies of 0.99 or above; i.e., more than 99% of the fuel's chemical energy is converted into heat. Clearly, the combustion efficiency must be a function of many parameters, including aerodynamics, gas composition, mixing, fuel/air ratio and geometry. It is beyond the scope of this text to discuss all the factors that can affect the combustion efficiency; however, the following important observations can be made:

- Because of the contrary nature of combustion efficiency and pressure ratio, typically the higher the combustion efficiency the lower the pressure ratio.
- Space and weight limitations often force designers to accept higher combustor pressure losses.

Clearly, designing a combustor is often a trade-off between high combustion efficiency and space/weight limitations versus high-pressure ratio.

GAS PRODUCER TURBINE

The gas mixture exiting the combustor contains potential energy mainly in two forms: heat and pressure. The function of the gas turbine is to convert these fluid energies into mechanical shaft energy.

As discussed previously, a gas turbine always contains a gas producer turbine and power turbine. The function of the gas producer is to absorb only enough energy from the flow to drive the compressor, while the power turbine is supposed to convert the remaining flow energy into shaft rotational output energy.

AXIAL-FLOW TURBINES

Except for some very small gas turbines that employ radial in-flow turbines, almost all modern gas turbines incorporate axial-flow turbines (flow enters and exits in the axial direction). Thus, only the axial flow turbine design will be discussed. The function of a turbine is almost exactly opposite to that of an axial compressor. Instead of adding energy to the flow, the axial flow turbine absorbs flow energy and converts it to shaft rotation and torque ($P = \tau \cdot \omega$). Consequently, all flow parameters that increased in the compressor (head, density, temperature and pressure) decrease in the axial turbine.

Similar to the axial compressor, the gas producer axial-flow turbine consists of a number of stator and rotor blade passages or stages. While the axial compressor usually has more than 10 stages, a typical gas producer turbine will have less than three rotor/stator stages. Thus, the pressure ratio per stage must be much greater for an axial-flow turbine than for an axial compressor. Below is a brief functional description of an axial-flow turbine.

Flow entering the gas producer turbine first passes through a row of inlet guide vanes (nozzles), where the flow expands and is tangentially redirected, converting pressure and temperature into tangential flow velocity. The gas flow exits the nozzle and impinges on the rotor blades, accelerating the turbine; i.e., converting flow kinetic energy into mechanical shaft rotation *(Figure 2-14)*. In the rotor stage, the gas flow further expands and is tangentially redirected by the converging blade passages. The accelerating gas mixture passing through the blade passage adds further tangential reaction acceleration to the turbine blades *(Figure 2-15)*.

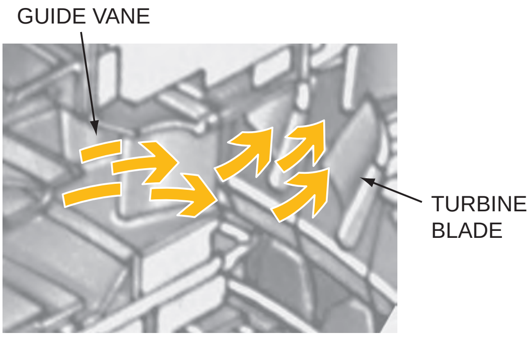


Figure 2-14. *Gas Flow of Inlet Guide Vanes and Turbine Blade*

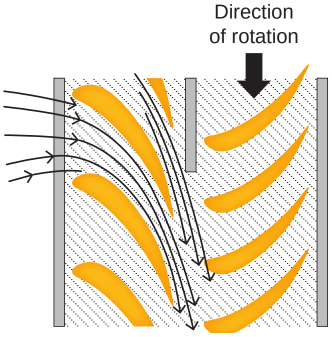
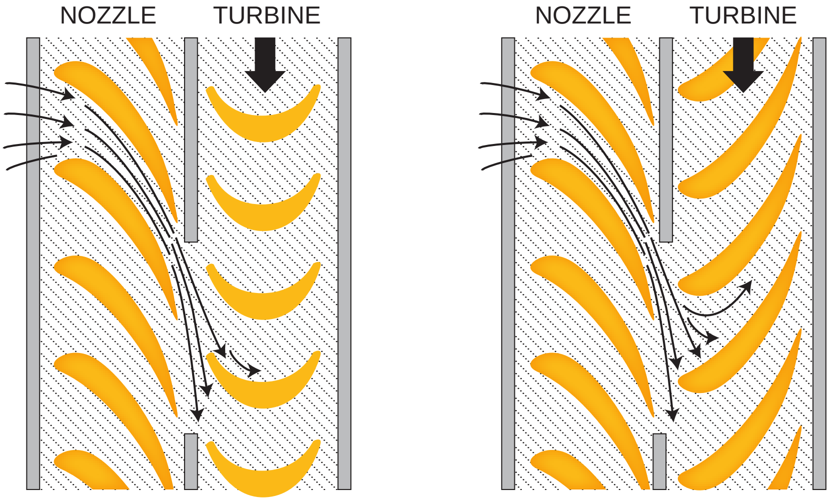


Figure 2-15. *Flow-through Axial Turbine*

The previous process repeats in the subsequent downstream rotor/stator stages. Thus, each gas producer turbine rotor/stator stage converts pressure and temperature into shaft rotational energy until enough rotational energy is created to drive the axial compressor.

IMPULSE AND REACTION TURBINES

There are two important types of axial turbine designs: impulse and reaction type turbines *(Figure 2-16)*. Although their basic designs are similar, they function based on somewhat different physical principles.



Turbine driven by the impulse of the gas flow only.

Turbine driven by the impulse of the gas flow and its subsequent reaction as it accelerates through the converging blade passage.

Figure 2-16. *Impulse versus Reaction Turbine*

Impulse Turbine

The turbine is driven by the impulse of the gas flow impinging on the rotating blades. All conversion of pressure to velocity occurs in the stator. The rotor blades only redirect the flow. In this case, the turbine blades are "bucket" shaped to absorb the optimum amount of tangential kinetic flow energy.

Reaction Turbine

The turbine is driven by the impulse of the impinging gas flow, as well as, its subsequent reaction as the gas is expanded through the converging turbine blade passage. For the reaction turbine design, the blade passages are converging, i.e., passages which expand the gas mixture. In a reaction turbine, the pressure reduction is split between rotor and stator. A reaction turbine is generally more efficient than an impulse turbine.

Some turbines found in older industrial gas turbine applications are hybrid designs, incorporating a mix of both impulse and reaction turbines. The initial stages in such gas turbines employ the impulse blade design, while the subsequent stages are closer to a reaction blade design.

AXIAL-FLOW TURBINE PERFORMANCE

Since the gas producer turbine and axial air compressor are mounted on the same shaft, their rotational speed (ω) and torque (τ) have to be identical (assuming no mechanical losses):

$$\tau_{compressor} = \tau_{turbine}$$

$$\omega_{compressor} = \omega_{turbine}$$

And since power equals torque multiplied by rotational speed ($P = \omega \cdot \tau$), the following is obtained:

$$P_{compressor} = P_{turbine}$$

Hence, the power of the gas producer must always equal the axial compressor's power.

It is important to remember that in functional principle, the axial-flow turbine is not much more than a reversed axial compressor. The mathematical tools previously developed for the axial compressor, such as Euler's equation, velocity vector polygons and the multi-stage total head estimate method, are similarly applicable to an axial-flow turbine. For example, Euler's equation for the turbine becomes *(Figure 2-17):*

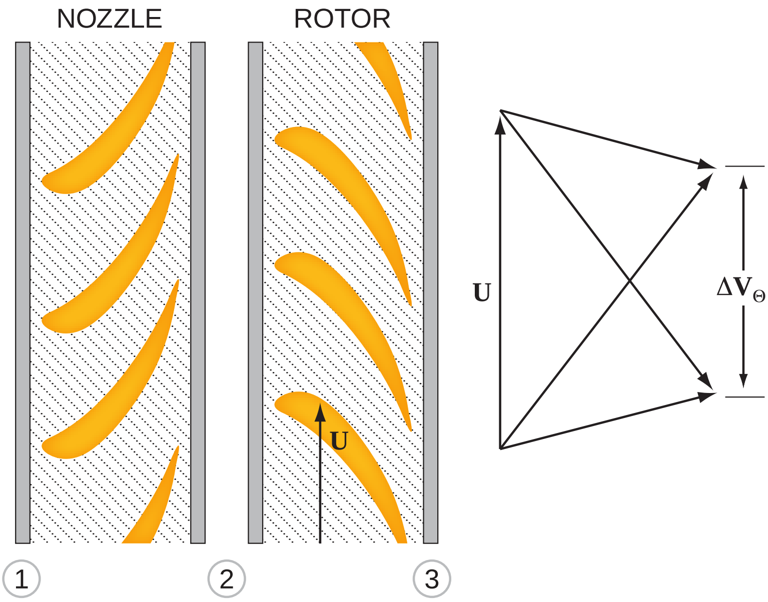


Figure 2-17. *Axial Flow Turbine Performance*

Turbine output power per stage:

$$P = \omega \cdot \Delta\tau = \omega \cdot W \cdot \Delta(r \cdot c_u) = \omega \cdot W \cdot (r_2 c_{u,2} - r_1 c_{u,1})$$

This expression can be simplified assuming that the mean blade radius

remains constant, $r_1 = r_2$, and that there is no flow slip:

$$P = U \cdot W \cdot \Delta c_u$$

Similarly, the previously derived axial compressor equations can be applied to determine head, pressure ratio and temperature for the axial flow turbine:

Turbine head decrease per stage:

$$H = h_2 - h_1 = \frac{P}{W} = \omega \cdot (r_2 c_{u,2} - r_1 c_{u,1})$$

Turbine temperature decrease per stage:

$$T_2 = \frac{\omega}{c_p} \cdot (r_2 c_{u,2} - r_1 c_{u,1}) + T_1$$

Turbine pressure ratio per stage:

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta\omega}{c_p T_1} \cdot (r_2 c_{u,2} - r_1 c_{u,1})\right)^{\frac{\gamma}{\gamma-1}}$$

or

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta}{c_p \cdot T_1} \cdot H\right)^{\frac{\gamma}{\gamma-1}}$$

or

$$\frac{P_2}{P_1} = \left(1 + \frac{\eta\omega}{c_p T_1} \cdot (r_2 c_2 \sin(\alpha_2) - r_1 c_1 \sin(\alpha_1))\right)^{\frac{\gamma}{\gamma-1}}$$

These axial-flow turbine equations are identical to the equations that were previously derived for the axial compressor. The only expression that must be different for the axial-flow turbine is the definition of its isentropic efficiency, which is inversed:

efficiency = actual work output / ideal work output

$$\eta_{gt} = \frac{Actual\ Head}{Ideal\ (Isentropic)\ Head} = \frac{h_2 - h_1}{h_2^* - h_1} = \frac{T_2 - T_1}{T_2^* - T_1}$$

Typical axial flow turbine isentropic efficiencies are around 90%, but can be as high as 94%.

POWER TURBINE

The gas turbine elements upstream of the power turbine are commonly referred to as the "gas producer." In principle, the gas producer of a gas turbine and an aircraft jet engine are very similar: both produce a hot, high-pressure exhaust fuel/air gas mixture *(Figures 2-18 and 2-19)*. However, while in the jet engine these gases are employed to generate aerodynamic thrust, the gas turbine employs them to drive a power turbine. Thus, it is the function of the power turbine to recover as much energy as possible from the exhaust mixture of the gas producer and convert it into mechanical shaft output energy.

Although the gas producer and power turbine each provide for a different function in the gas turbine, their aerodynamic designs are very similar. The functional description and mathematical equations that were previously presented for the gas producer turbine are similarly applicable to the power turbine.

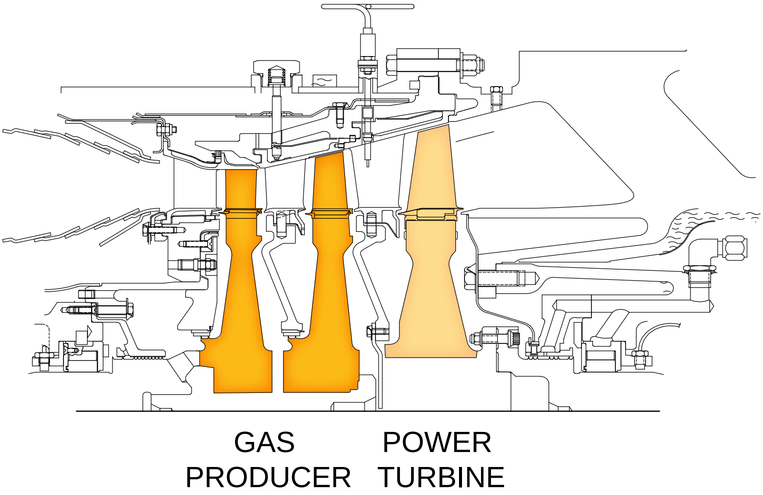


Figure 2-18. *Gas Producer and Power Turbine of an Industrial Gas Turbine*

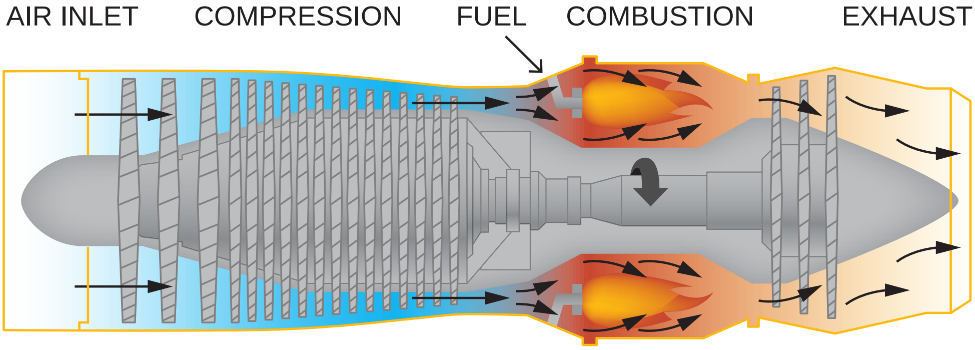


Figure 2-19. *Gas Producer of a jet engine*

SHAFT ARRANGEMENT

As previously noted, there are two common types of industrial gas turbine shaft arrangements: two-shaft and single-shaft *(Figures 2-20 and 2-21).*

Two-Shaft Gas Turbine

A two-shaft gas turbine has a free power turbine design. In two-shaft gas turbines, the power turbine is independently supported on its own bearings and allowed to freely rotate. Thus, two-shaft gas turbines allow for variable shaft speed output power:

$$\omega_{gp} \neq \omega_{pt}$$

Single-Shaft Gas Turbine

On the other hand, if the power turbine and gas producer turbine are connected to one integral shaft (rotating at the same speed), as in a single-shaft gas turbine, then the gas turbine shaft output speed must be more or less fixed. Thus, for a single-shaft gas turbine:

$$\omega_{gp} = \omega_{pt}$$

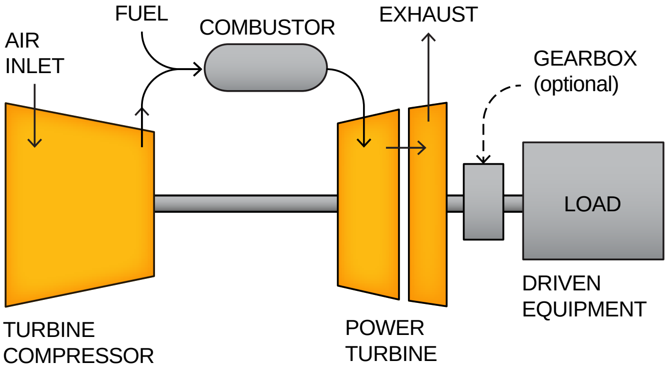

FUEL
AIR INLET
COMBUSTOR
EXHAUST
GEARBOX (optional)
LOAD
DRIVEN EQUIPMENT
TURBINE COMPRESSOR
POWER TURBINE

Figure 2-20. *Two-Shaft Gas Turbine (Hot-End Drive)*

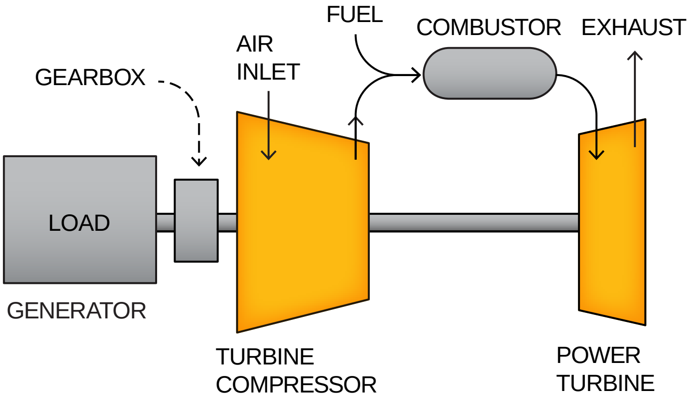

FUEL
AIR INLET
COMBUSTOR
EXHAUST
GEARBOX
LOAD
GENERATOR
TURBINE COMPRESSOR
POWER TURBINE

Figure 2-21. *Single-Shaft Gas Turbine (Cold-End Drive)*

Most compressor, pump and blower applications require two-shaft gas turbines, while most electric generator applications employ single-shaft gas turbines.

TURBINE LIMITATIONS

We already established that the total power of both gas producer and power turbine can be determined from:

$$P = W \cdot H = W \cdot c_p \cdot (T_{combustor} - T_{exit})$$

The above equation indicates that the power of the turbine is strongly dependent on the gas turbine firing temperature ($P \alpha T_{combustor}$), namely:

The higher the combustion firing temperature, the more shaft output power the gas turbine will provide.

What are the limiting factors of the firing temperature? We obviously would like to fire the gas turbine as hot as possible to generate the maximum output power. The simple answer is that combustor/blade materials and design put physical limitations on the maximum allowable firing temperature. Namely, the firing temperature is limited by how much heat the combustor and the first-stage turbine blades can take before melting or degrading severely.

To overcome this limitation, the main focus of modern gas turbine research lies in the development and testing of blade and combustor materials that tolerate higher temperatures. Combustor liners and first- or second-stage gas producer turbine stages are often manufactured from exotic materials that have melting points much higher than those of more common metals such as stainless steel or titanium.

Another method being employed to allow for higher firing temperatures is the use of active cooling of the combustor liner and/or the gas producer turbine's first- or second-stage blades.

It should be noted, however, that the amount of temperature reduction is limited by the pressure ratio available. In other words, for a fixed pressure ratio, increasing the firing temperature would simply increase the exhaust temperature.

TURBINE BLADE COOLING

To cool the turbine blades, a fraction of the axial compressor discharge air is bled and forced through minute cooling passages within the first- or second-stage gas producer turbine blades *(Figure 2-22)*. There are several sophisticated schemes that are used for blade cooling, such as single-pass internal cooling, single-pass internal cooling with film cooling or multiple-pass internal cooling with extensive film cooling. Blade cooling techniques will be explained in more detail later

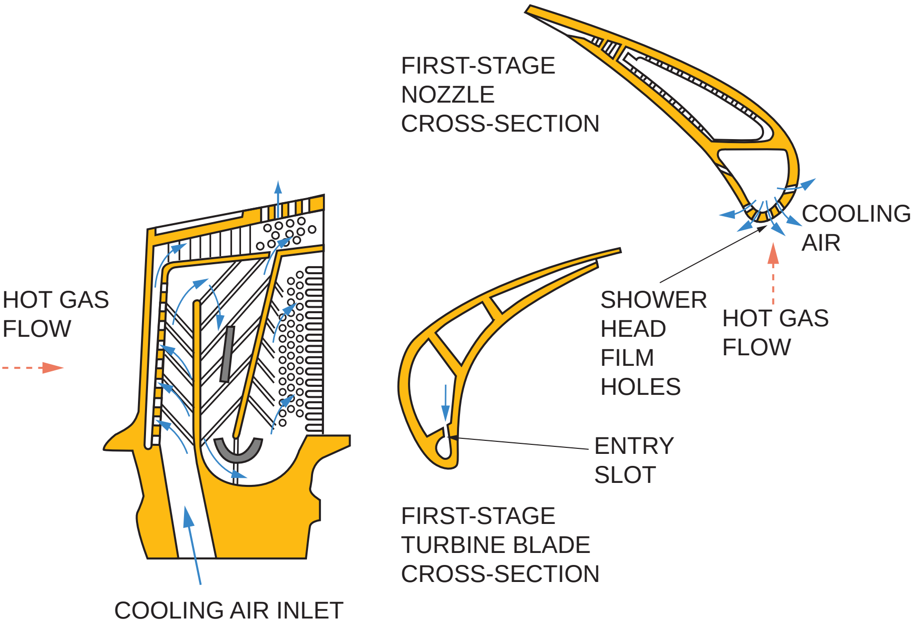

FIRST-STAGE NOZZLE CROSS-SECTION
HOT GAS FLOW
COOLING AIR
SHOWER HEAD FILM HOLES
HOT GAS FLOW
ENTRY SLOT
FIRST-STAGE TURBINE BLADE CROSS-SECTION
COOLING AIR INLET

Figure 2-22. *Various Schemes for Turbine Blade Cooling*

Combustor Liner Cooling

Combustor liner cooling was briefly discussed in the previous combustor section. The cooling of the combustor liner is accomplished by redirecting the flow away from the liner walls using dilution air and passing cooling air over the outside liner walls. Typically, between 40% and 60% of the airflow entering the combustion chamber is being employed for cooling purposes *(Figure 2-23)*.

As a reference, peak firing temperatures for some Solar® gas turbine models are provided in *Table 2-1.* Obviously, more modern gas turbines are designed for higher firing temperatures than older models.

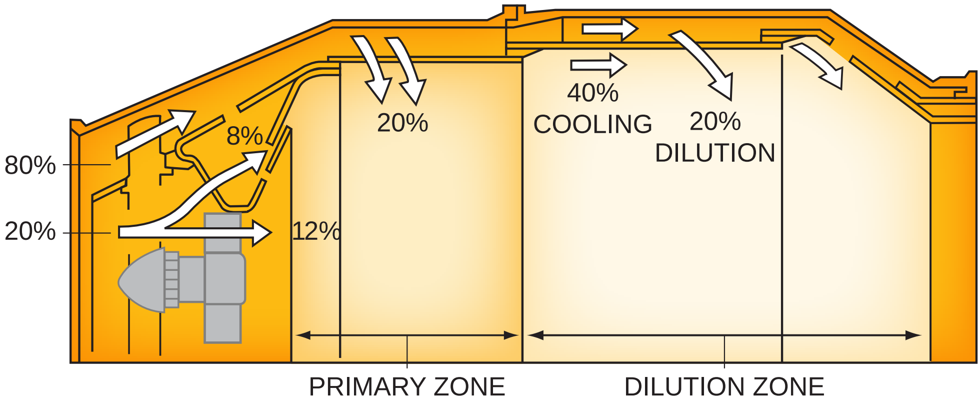


Figure 2-23. *Typical Flow Distribution for Combustor Liner Cooling (Diffusion Flame)*

Gas Turbine Model (Two-Shaft)	Firing Temperature, °C (°F)
Saturn 20	**≈ 900 (1660)**
Centaur 40	**≈ 900 (1660)**
Taurus 60	**≈1065 (1950)**
Mars 100	**≈1140 (2080)**
Titan 130	**≈1180 (2150)**
Titan 250	**≈1200 (2200)**

Table 2-1. *Solar Gas Turbine Firing Temperatures*

GAS TURBINE EXHAUST

Ideally, one would desire a power turbine that recovers all remaining energy from the exhaust mixture of the gas producer so that the air exiting the power turbine has a temperature and pressure equal to ambient temperature:

$$P_{\text{Gas Turbine Exhaust}} = P_{\text{Gas Turbine Intake}} = P_{\text{Ambient}}$$
$$T_{\text{Gas Turbine Exhaust}} = T_{\text{Gas Turbine Intake}} = T_{\text{Ambient}}$$

In an industrial gas turbine, the exhaust flow is always subsonic; therefore, the exhaust air pressure must be equal to atmospheric pressure:

$$P_{\text{Exhaust}} = P_{\text{ambient.}}$$

However, in any gas turbine, the gas mixture exhausts into the atmosphere at a temperature significantly higher than ambient temperature. This means that the gas exiting the gas turbine contains residual heat energy that is not being converted to useful mechanical shaft energy. By being exhausted to the atmosphere, this residual heat energy is lost. Unfortunately, this is an unavoidable physical fact of all simple-cycle gas turbines.

Most simple-cycle industrial gas turbines have exhaust temperatures between 451 and 534°C (844-994°F) as shown in *Table 2-2*.

Let us calculate how much heat power is actually exhausted to the atmosphere by a Mars 100 gas turbine. Remember, the energy contained per unit air volume is defined as the entropy ($h = c_pT$) and that power can be calculated by multiplying entropy with the mass flow ($P = W{\cdot}h$). In a Mars 100 gas turbine, the mass flow is approximately 36 kg/s and, assuming an ambient temperature of 39°C (102°F), the temperature difference between exhaust and ambient (ΔT) is 463°C (865°F). Thus, the total energy that the exhaust air contains is:

Gas Turbine Model (Two-Shaft)	Exhaust Temperature, °C (°F)
Saturn 20	520 (970)
Centaur 40	450 (840)
Taurus 60	510 (950)
Mars 100	480 (900)
Titan 130	480 (895)
Titan 250	455 (850)

Table 2-2. *Simple-Cycle Gas Turbine Exhaust Temperatures*

$$P = W \cdot c_p \cdot (T_{Exhaust} - T_{Ambient}) = 36\,\frac{kg}{s} \cdot 1003\,\frac{J}{kg\,°C} \cdot 463°C$$

$$= 16{,}718kW \ = 22{,}350hp$$

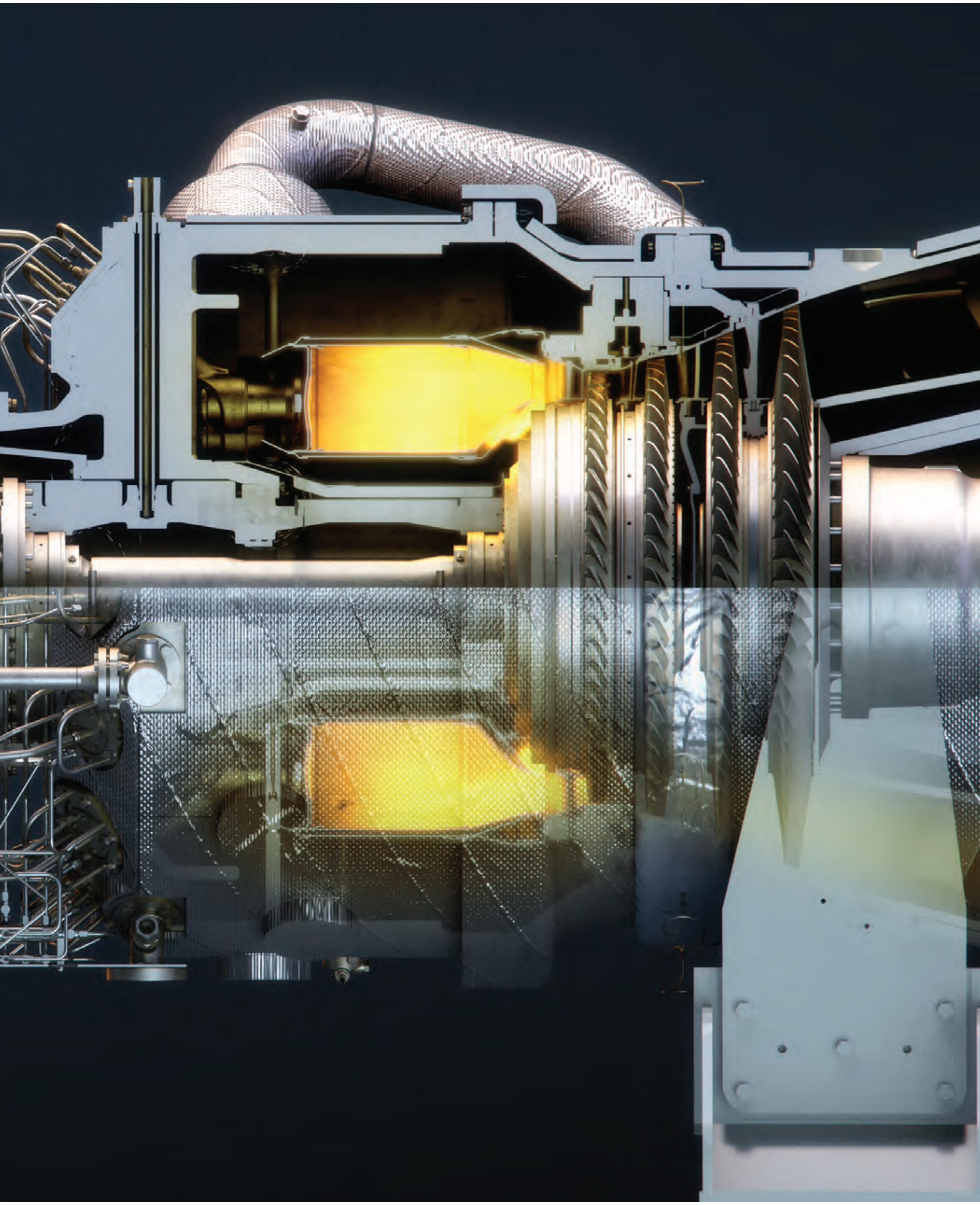
Interestingly, a Mars 100 gas turbine, rated at 15,000-hp shaft output power, produces approximately 22,350 hp in hot exhaust air; therefore, more energy is lost in exhaust heat than is created in shaft output power. This does not mean that the Mars 100 gas turbine was poorly designed, but rather shows the thermodynamic efficiency limitation of the simple Brayton gas turbine cycle.

Here also lies the explanation for the level of peak efficiencies of any simple Brayton cycle gas turbine. The simple Brayton cycle gas turbine cannot recover the residual exhaust heat energy and, which limits its efficiency. The efficiency of the gas turbine can be estimated:

$$\eta = \frac{Useful\ Output}{Total\ Output} = \frac{Shaft\ Output\ Power}{Shaft\ Output\ Power + Exhaust\ Air\ Power}$$

$$= \frac{15{,}000}{15{,}000 + 22{,}350} = 40\%$$

The actual peak efficiency of a Mars 100 gas turbine is 34%. Sometimes the hot exhaust air can be recovered using waste heat recovery and heat exchanger units for heating, drying or steam generation applications. In these cases, the overall system efficiency is typically significantly better than that of a simple Brayton cycle. Some of these advanced cycles will be discussed in a later chapter.



CHAPTER 3

MODERN ANALYSIS METHODS

COMPUTER-AIDED ENGINEERING

The following is a brief discussion of modern computational fluid dynamic (CFD) technologies that are being developed to analyze and improve gas turbine performance. This introduction is intended to give the reader an appreciation of the efforts undertaken by the gas turbine industry to develop better, higher efficiency and cleaner gas turbines for the future *(Figure 3-1)*.

The gas turbine engineer's goal is to be able to completely design a gas turbine, optimize its performance, and test it without actually building a working prototype; i.e., completely computer model it. Computer software packages that are being employed in the design process can be divided into two types: Computer Aided Design (CAD) and Computer Aided Engineering (CAE). CAD software packages are used for design drafting, dimensional layouts, architectural/engineering drawings and solid body modeling. On the other hand, CAE software is employed for the actual computational solution of engineering problems. CAE software includes numerical solutions for structural, vibration, rotor dynamic and/or fluid dynamics behavior predictions. There is often significant overlap between CAE and CAD software applications and features.

Although CAE software for vibration, rotordynamics and structural analysis are used extensively in the gas turbine industry, the focus in this section will be only on fluid dynamics analysis to stay within the general subject matter of this text. Previously discussed was the non-ideal, simple-cycle analysis application to predict and study gas turbine performance. The simple-cycle thermodynamic analysis can also be called a one-dimensional (1-D) fluid dynamics analysis since it describes the behavior of the fluid (air mixture) passing through the gas turbine as one-dimensional. This analysis assumes that the gas at any point in the gas turbine is radially and circumferentially uniform in temperature, pressure, density and velocity. Obviously, this is an oversimplification of the real three-dimensional (3-D) gas turbine flow field, which is highly non-uniform. Nonetheless, non-ideal simple-cycle analysis is a first step in gas turbine computational fluid analysis.

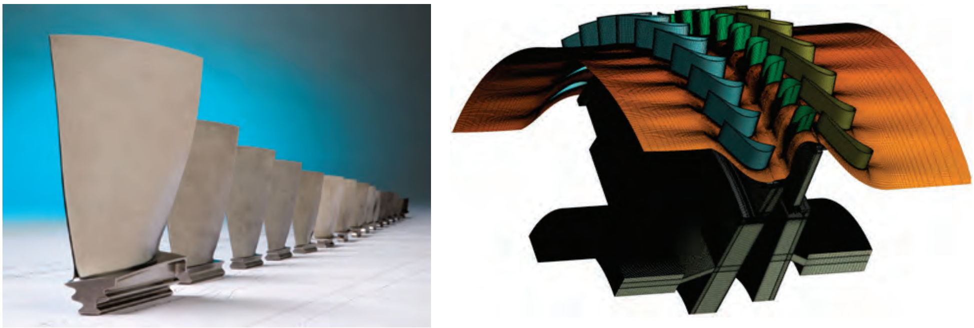


Figure 3-1. *Rotor blades for the compressor of a modern industrial gas turbine and sector model for the turbine section*

To expand this type of analysis into three dimensions (axial, radial and circumferential) requires a significantly more complicated analysis that is very computationally intensive. As of this date, the technology is still limited to accurately model the 3-D flow in a complete gas turbine. However, modern computers have reached a level where it is feasible to model gas turbine components, such as the compressor or turbine, individually and achieve satisfactory results. This type of analysis can be very useful and will be discussed over the next few pages.

DESIGN GOALS

As shown earlier, when using the non-ideal, simple-cycle analysis method, gas turbine performance can be completely modeled and predicted, if the component efficiencies are known. If we can determine, either experimentally or mathematically:

- **Inlet Pressure Ratio**
- **Axial Compressor Efficiency, Flow and Pressure Ratio**
- **Combustor Efficiency and Pressure Ratio**
- **Gas Producer Turbine Efficiency, Flow and Pressure Ratio**
- **Power Turbine Efficiency, Flow and Pressure Ratio**
- **Exhaust System Pressure Ratio**

Gas turbine output power and efficiency can be calculated at any ambient condition. It is important to realize that the above efficiencies and pressure ratios are not constants, but depend upon a number of variables, such as inlet temperature and pressure, shaft speed, elevation, mass flow, air humidity and others. Further, to predict the life of a gas turbine and its components, stress levels and temperature levels need to be calculated. Emissions predictions require the calculations of chemical reactions, and how they interact with local flow and temperature fields.

Traditionally, the most often employed method to obtain these data relies solely on experimental test stand data. Component efficiencies are either measured directly in a dedicated component test stand or derived from overall gas turbine performance. Empirical functions are developed for each component’s efficiency and pressure ratio from the test stand data. These empirical efficiency functions can then be integrated into the non-ideal, simple-cycle analysis program.

Major drawbacks of this method are: that a prototype of the component has to be built and tested before accurate performance predictions can be run; if the component design is significantly altered, the empirical functions will no longer be valid; and testing can become very expensive and time consuming. Thus, a mathematical/theoretical method to determine these component efficiencies would be preferable.

COMPUTATIONAL FLUID DYNAMICS

Over the last 30 years, computers have become more sophisticated, enabling the gas turbine designer to perform some limited theoretical performance and efficiency calculations. Hundreds of different numerical methods have been developed, but in principle most of them can be separated into two classes:

Streamline balance and Computational fluid dynamics

Streamline balance methods were developed in the early 1960s and employed until the mid-1970s. These were based on the concept that the streamline locations in an internal flow field of the gas turbine can be determined from the interactions of the centrifugal, Coriolis and inertial forces on the fluid (air). However, experience has shown that results from the streamline balance method are not very accurate. These methods do not allow for correct performance predictions of new gas turbine component designs.

Since the early 1970s, significant efforts have been made by a large number of researchers to develop numerical methods to solve theoretical equations that describe the actual behavior and dynamics of a fluid. These equations are called the Navier-Stokes equations and, generally, are applicable to any fluid and boundary condition *(Figure 3-2)*. The Navier-Stokes equation set consists of: continuity equation - conservation of mass; x,y,z equations of motion - F = m · a (Newton's Second Law for a fluid); and an energy equation – first law of thermodynamics (energy is conserved).

The Navier-Stokes equations were originally derived by M. Navier in 1827 and S.D. Poisson in 1831. Historically, however, useful analysis of the equations is mostly associated with the work of L. Prandtl, T. v. Karman, H. Blasius, and H. Schlichting of the University of Gottingen in Germany between 1900 and 1930. Solutions to these equations are difficult since they are non-linear, non-homogenous and coupled partial differential equations. Only for a limited number of very simple cases can exact analytic solutions be found. Thus, until modern computers became available in the 1970s, the Navier-Stokes equations were of very limited practical use for engineering applications. However, with the advent of high-speed computers, more complicated mathematical equations became solvable using a numerical rather than an analytical approach.

$$\underline{\partial u_\theta} \qquad \frac{1}{R}\frac{\partial}{\partial \mathrm{R}}(Ru_R)+\frac{1}{\mathrm{R}}\frac{\partial u_\theta}{\partial \theta}+\frac{\partial u_z}{\partial \mathrm{z}}=0$$

$$\rho\left(\frac{\partial u_R}{\partial \mathrm{t}}+u_R\frac{\partial u_R}{\partial \mathrm{R}}+\frac{u_\theta}{\mathrm{R}}\frac{\partial u_R}{\partial \theta}-\frac{u_\theta^2}{\mathrm{R}}+u_z\frac{\partial u_R}{\partial \mathrm{z}}\right)=-\frac{\partial \rho}{\partial \mathrm{R}}+\mu\left(\nabla^2 u_R-\frac{u_R}{\mathrm{R}^2}-\frac{2}{\mathrm{R}^2}\frac{\partial u_\theta}{\partial \theta}\right)+\rho f_R$$

$$\rho\left(\frac{\partial u_\theta}{\partial \mathrm{t}}+u_R\frac{\partial u_\theta}{\partial \mathrm{R}}+\frac{u_\theta}{\mathrm{R}}\frac{\partial u_\theta}{\partial \theta}+\frac{u_R u_\theta}{\mathrm{R}}+u_z\frac{\partial u_\theta}{\partial \mathrm{z}}\right)=-\frac{1}{R}\frac{\partial \rho}{\partial \mathrm{R}}+\mu\left(\nabla^2 u_\theta-\frac{u_\theta}{\mathrm{R}^2}+\frac{2}{\mathrm{R}^2}\frac{\partial u_R}{\partial \theta}\right)+\rho f\theta$$

$$\rho\left(\frac{\partial u_z}{\partial \mathrm{t}}+u_R\frac{\partial u_z}{\partial \mathrm{R}}+\frac{u_\theta}{\mathrm{R}}\frac{\partial u_z}{\partial \theta}+u_z\frac{\partial u_z}{\partial \mathrm{z}}\right)=-\frac{\partial \rho}{\partial \mathrm{z}}+\mu\nabla^2 u_z+\rho f_{\mathrm{z}}$$

$$\rho C_v\left(\frac{\partial \mathrm{T}}{\partial \mathrm{t}}+u_R\frac{\partial \mathrm{T}}{\partial \mathrm{R}}+\frac{u_\theta}{\mathrm{R}}\frac{\partial \mathrm{T}}{\partial \theta}+u_z\frac{\partial \mathrm{T}}{\partial \mathrm{z}}\right)$$

$$=\kappa\nabla^2\mathrm{T}+2\mu\left\{\left(\frac{\partial u_R}{\partial R}\right)^2+\left[\frac{1}{R}\left(\frac{\partial u_\theta}{\partial \theta}+u_R\right)\right]^2+\left(\frac{\partial u_z}{\partial z}\right)^2\right\}$$

$$+\mu\left\{\left(\frac{\partial u_\theta}{\partial z}+\frac{1}{R}\frac{\partial u_z}{\partial \theta}\right)^2+\left(\frac{\partial u_z}{\partial R}+\frac{\partial u_R}{\partial z}\right)^2+\left[\frac{1}{R}\frac{\partial u_R}{\partial \theta}+\frac{\partial}{\partial R}\left(\frac{u_\theta}{\mathrm{R}}\right)\right]^2\right\}$$

$$\nabla^2=\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial}{\partial R}\right)+\frac{1}{R^2}\frac{\partial^2}{\partial \theta^2}+\frac{\partial^2}{\partial z^2}$$

Figure 3-2. *Navier-Stokes Equations in Cylindrical Coordinates*

NUMERICAL APPROACH

Analytical solutions of the Navier-Stokes equations cannot be found for complex turbomachinery geometries and inlet/outlet boundary conditions. However, analytic solutions for simple geometries, such as cubes or wedges, if they have simple, uniform boundary conditions can be calculated. Thus, a complex-turbomachinery geometry is divided into thousands of simple geometries (cubes, bricks, wedges or tetrahedrals) that are individually solvable *(Figure 3-3 shows this for a two-dimensional airfoil)*. By matching the boundary conditions of each geometry in a sophisticated iterative-stepping process, a solution of the complete complex-geometry flow field can be found.

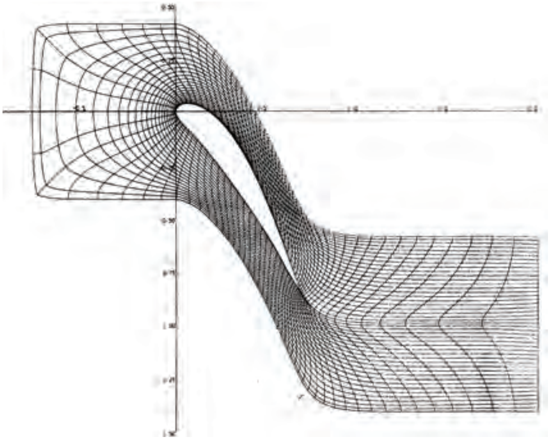


Figure 3-3. *Computational Grid for a Two-Dimensional Flow Problem*

This is a computationally very intensive method that enables us to analyze the internal flow field of many complicated turbomachinery component geometries.

We are still several developmental years away from accurately modeling the gas turbine as a whole. Nonetheless, once the internal flow field of a turbomachinery component is known, the component's efficiency and pressure drop can be calculated, and the results can be easily integrated into the non-ideal, simple-cycle analysis. Similar numerical methods exist for the analysis of rotordynamic and vibration applications.

STATE-OF-THE-ART

Let us briefly discuss some of the commonly employed CFD methods, their assumptions, limitations and applicability. Any type of CFD analysis is an approximation of the real world. The real flow is approximated by more-or-less accurate mathematical models. These mathematical models usually do not lend themselves to analytical solutions, but have to be solved by more-or-less accurate numerical methods.

Potential Flow Method (1930-1965)

Potential flow assumes that the fluid is irrotational, inviscid and incompressible. Turbulence, boundary layers or flow unsteadiness cannot be directly modeled. This type of method is limited to large Reynold's number gas flows, typically external flows such as subsonic flow around an airplane wing or propeller. Frequently, to account for the boundary layer losses, the method is coupled with a simple, empirical boundary displacement function. Since the method assumes incompressible flow, no density changes can be modeled, which makes it useless for compressor or turbine applications. For internal gas turbine component flow analysis, this method is not adequate.

Euler Method (1970-1985)

Euler flow solves the Navier-Stokes equations, but neglects the terms that account for viscosity; i.e., the fluid is inviscid. Since turbulence and boundary layer are both viscosity functions, the Euler method cannot account for either. However, since the Euler code does not assume irrotational flow and allows for density changes, it is significantly more accurate than the Potential Flow Method for compressible flow. Euler codes are often employed to analyze internal turbomachinery flows, however, because viscosity is not modeled, it can be inaccurate especially for turbulent flows with fluid separation and recirculation.

Full Navier-Stokes Solvers (1980-Present)

The Navier-Stokes solver evaluates the full set of fluid dynamics equations. Since the length and time scales of turbulence are too small to be properly modeled, Navier-Stokes solvers still typically employ simplified methods to account for the effects of turbulence (so called turbulence models). Navier-Stokes solvers can be very accurate but are extremely computationally intensive.

In recent years, Navier-Stokes solvers have become a standard tool in the turbomachinery industry. Nonetheless, because of the many assumptions for boundary conditions and turbulence modeling that have to be made, even with the full Navier-Stokes solver, computational results need to be carefully vetted with actual tests.

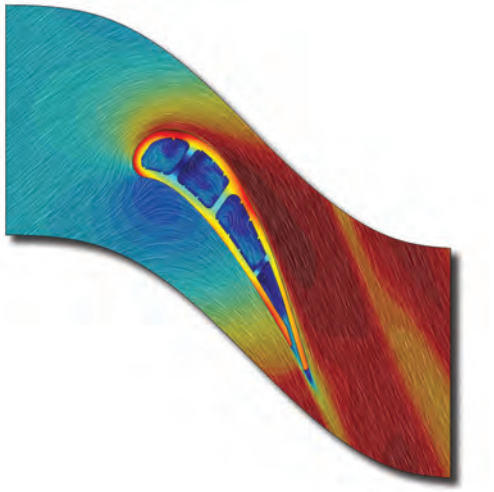
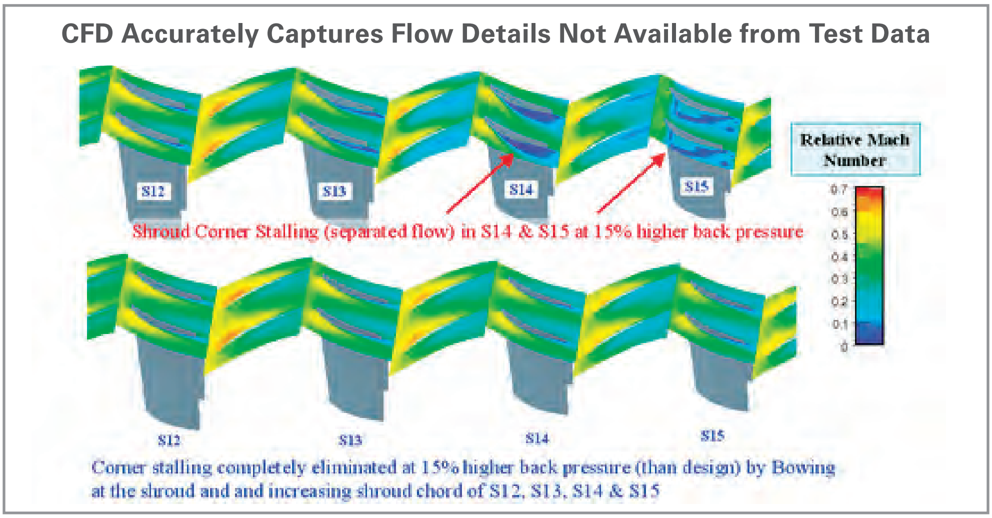


Figure 3-4. *Computational Fluid Dynamic (CFD) Analysis of the unsteady flow through a turbine rotor*

Today, it is possible to model entire compressors, and resolving unsteady flow *(Figure 3-4)*. Modeling entire turbomachines (for example, the entire compressor of a gas turbine, *Figure 3-5)* creates the difficulty that some components rotate, while others are stationary. For the rotating components, the flow from the stationary components appears unsteady, and for the stationary components, the flow from the rotating components appears unsteady. In other words, the code has to deal with unsteady flow *(Figure 3-4)*. The capability to perform conjugate heat transfer calculations, that can be used to calculate the heat transfer between a body (like an airfoil) and the gas flow, have great impact in optimizing engine life, cooling air usage, and engine performance. Other classes of problems are also solvable with great accuracy including flows where chemical reactions occur, such is in the combustor of a gas turbine *(Figure 3-6)*, or gas flows with solid or liquid particles.

SUMMARY

Many techniques are available to model the flow inside a gas turbine. In the future, we expect to be able to completely design gas turbines and optimize them solely on the



CFD Accurately Captures Flow Details Not Available from Test Data

Shroud Corner Stalling (separated flow) in S14 & S15 at 15% higher back pressure

Corner stalling completely eliminated at 15% higher back pressure (than design) by Bowing at the shroud and and increasing shroud chord of S12, S13, S14 & S15

Figure 3-5. *3-D CFD Analysis of complete axial compressor*

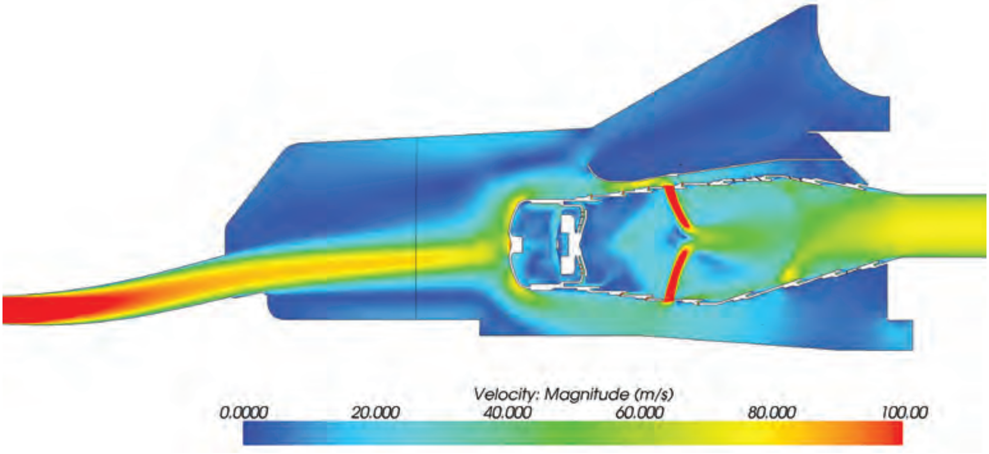


Figure 3-6. *Flow in a gas turbine combustor*

computer, without ever having to build a prototype. Currently however, the state-of-the-art of CFD technology is:

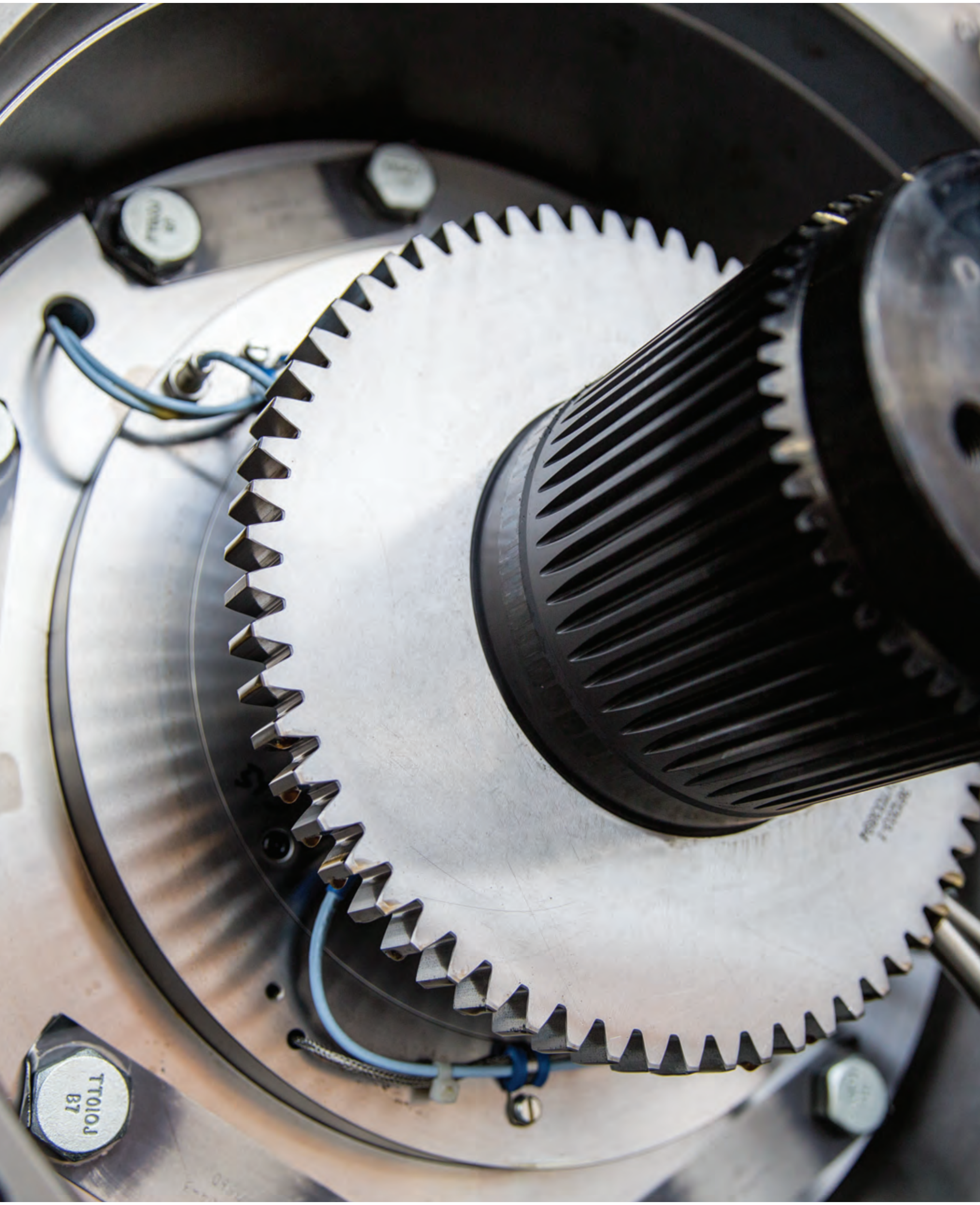
Computational results are often inaccurate and need test validation.

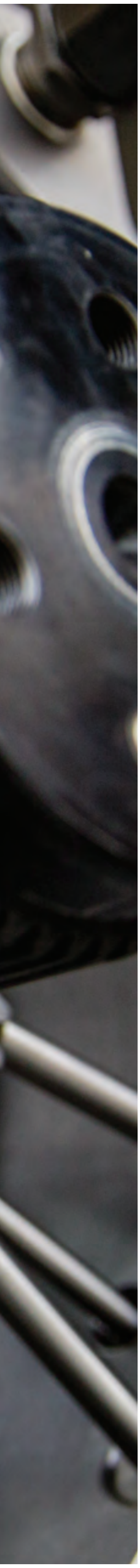
CFD modeling and analysis can be expensive and time intensive, depending on the number of components that have to be analyzed, and the accuracy of the results required. Transient, non-steady-state calculations exponentially increase the effort.

Typically, CFD models provide good insights into flow structures, but may encounter problems supplying accurate predictions for bulk characteristics such as stage mass flow or stage efficiency. It also must be stressed that in order to achieve correct results, the user has to be quite experienced in using the code and interpreting the results. Ubiquitous inaccuracies are caused by modeling errors (i.e., the numerical model is not identical with the physical reality), numerical errors (the solution of the programmed equations is not accurate), convergence errors (calculations are stopped after too few iterations to save time), application uncertainties (inlet or exit conditions, or geometry are not precisely known), user errors (use of the wrong model, neglect of structures that influence the flow, etc.), and code errors ("bugs"). Detail problems, such as the transition from laminar to turbulent flow, are still not entirely resolved. All calculations need, therefore, to be calibrated against reliable test data.

However, the detailed insight in flow structures has yielded numerous detail improvements, leading to higher engine efficiencies, increased engine life, and improved reliability.

Thus, a combination of experimental results with CFD is currently the most sensible design approach for gas turbines. The gas turbine engineer should never base a design decision solely on CFD results.





CHAPTER 4

ROTORDYNAMICS & VIBRATIONS

ROTORDYNAMICS AND VIBRATIONS

The principal mechanical moving components of a gas turbine are the gas producer and power turbine shafts. On small to mid-size gas turbines, these shafts can weigh between 2,268 and 13,608 kg (5,000 and 30,000 lb) and rotate at angular speeds between 5,000 and 25,000 revolutions per minute. The gas producer shaft is typically supported by three to five journal bearings to restrict the radial movements and one or two thrust bearings to limit the axial movement. Similarly, the power turbine shaft typically has two journal bearings and one thrust bearing. In most modern industrial gas turbines, tilting pad or other fluid film bearings are employed; older industrial and aero derivative gas turbines often use rolling element bearings. Fluid film bearings employ petroleum based or synthetic oil to act as a lubricant and coolant between the moving shaft and the stationary bearing surfaces. Bearings function to transmit the static loads and dynamic vibration forces.

When the gas turbine shafts rotate at high angular speeds, small imbalances or eccentricities in the shafts due to manufacturing imperfection or mechanical/thermal stresses are amplified and can create significant forces on the gas turbine's bearings and bending of the shaft. If these forces are not adequately controlled, gas turbine catastrophic failure may be the consequence. Hence, it is of the utmost importance for the gas turbine designer to understand and predict the shaft and bearing rotordynamic forces of a gas turbine. Over the next couple of pages, we will briefly present the basic mechanic and dynamic theory of gas turbine rotordynamics. The discussion will be confined to lateral rotordynamics, and torsional behavior will not be covered in this text.

Simple Mass-Spring-Damper System (Single Degree of Freedom)

Let us start by analyzing a significantly simplified mechanical model. The assumption is that we can simulate shaft vibrations by modeling the shaft as a single stationary mass (m) with one fixed support (bearing). Naturally, this single support bearing would possess a certain physical stiffness (k) and damping (c) characteristic. *Figure 4-1* shows a schematic of this simple mass-spring-damper system.

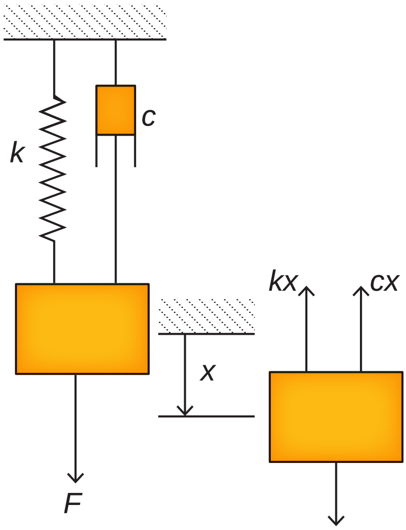


Figure 4-1. *Simple Mass-Spring-Damper System*

After applying Newton's second law (f=ma), we get the following equation of motion for this single degree of freedom system:

$$ma + cv + kx = f$$

where a is the acceleration, v is velocity, x is the displacement, and f is an external force. The above equation can easily be rewritten as an ordinary differential equation:

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = f$$

For now we will neglect the damping (c = 0) and the external force (f = 0). Thus, the ordinary differential equation simplifies to:

$$m\frac{d^2x}{dt^2} + kx = 0$$

It is beyond the scope of this text to demonstrate the solution for this equation, which can be found in most differential equations textbook. But the general solution of this equation is:

$$x = A\sin\sqrt{\frac{k}{m}}t + B\cos\sqrt{\frac{k}{m}}t$$

where A and B are constants that are evaluated from the initial conditions. This solution is also sometimes called the Eigenvalue analysis. The above solution shows that if the system is allowed to vibrate freely, the frequency at which it would vibrate is:

$$\omega_N = \sqrt{\frac{k}{m}}$$

ω_N is called the natural frequency or Eigenvalue of the system. The natural frequency is very important in rotordynamics since it also corresponds to the resonance frequency of a system; namely:

If an excitation force is applied to a system at its natural frequency, the system will resonate.

Now assume that a regular periodic (excitation) force is applied to the mass in the form of a sinusoidal function,

$$f = F\sin(\omega t)$$

where ω is the excitation frequency, and F is the amplitude of the excitation force. To generalize further, the damping (c) will not be neglected. The equation of motion thus becomes:

$$m\frac{dx^2}{d^2t} + c\frac{dx}{dt} + kx = F\sin(\omega t)$$

The steady-state “particular” solution to the above ordinary differential equation is:

$$x = X \sin(\omega t - \varphi)$$

where

$$X = \frac{F}{\sqrt{\left(k - m\omega^2\right)^2 + \left(c\omega\right)^2}}$$

NOTE: There is also a “complementary” transient term of the functional form x=Ce^{-Dt} sin(Et+α), which is part of the complete solution to the above ordinary differential equation. However, because of the exponential multiplication factor, this transient term decreases with time and will approach zero for most real systems. Thus, for a steady-state rotordynamic analysis the complementary term can often be neglected.

$$\varphi = \tan^{-1}\left(\frac{c\omega}{k - m\omega^2}\right)$$

The above steady-state solution shows that the system vibrates at the excitation frequency (ω) but experiences a certain lag (φ). This lag is called the phase lag or phase angle of the system. We can non-dimensionalize the above equation and rewrite it as follows:

$$Z = \frac{Xk}{F} = \frac{1}{\sqrt{\left(1 - \left(\frac{\omega}{\omega_N}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_N}\right)\right)^2}}$$

$$\tan\varphi = \frac{2\zeta\left(\frac{\omega}{\omega_N}\right)}{1 - \left(\frac{\omega}{\omega_N}\right)^2}$$

where ζ is defined as the damping factor and Z is called the force response.

Damping Factor:

$$\zeta = \frac{c}{2m\omega_N}$$

Force Response:

$$Z = \frac{Xk}{F}$$

The force response represents the non-dimensional vibration amplitude of the system. Hence, the peak vibration amplitude of the system is:

$$X = \frac{F}{2\zeta k} = \frac{F}{c\omega_N}$$

If we plot Z versus ω/ω_N (non-dimensional excitation frequency), we get *Figure 4-2*. This type of plot, also called a force response plot, is commonly employed for vibration analysis since it allows for the determination of the vibration peak amplitude for any frequency at which the system may be excited. The force response plot shows asymptotic behavior at $\omega/\omega_N = 1.0$; i.e., when the forcing frequency reaches the system's natural frequency. Namely, as the frequency of the sinusoidal force approaches the system's natural frequency, there is a possibility that the system's resonance may become unstable if the system is not adequately damped.

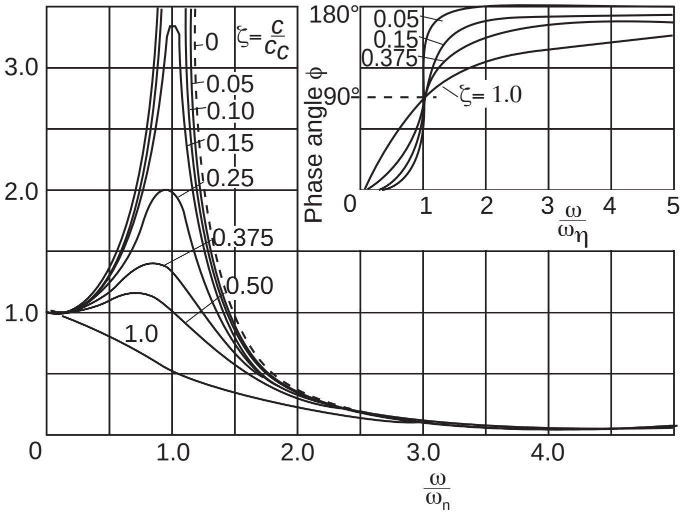


Figure 4-2. *Force Response Plot*

As can be seen from *Figure 4-2*, the system's force response significantly depends on the level of damping on the system. For example, if the system is damped beyond unity ($\zeta > 1.0$), the plot shows no peak response and the stable system is called overdamped. On the other hand, if the damping is below unity ($\zeta < 1.0$), the system force response (Z) will exceed unity at the natural frequency and the system is called underdamped. The damping value at which the system is just in between over and underdamped ($\zeta = 1.0$), is called the critical damping. It is important to realize that even systems that are underdamped ($\zeta < 1.0$) may be considered to be stable and acceptable for a particular application as long as the force response stays within the allowable engineering design parameters of the system.

Supported Rotor System (Two Degrees-of-Freedom)

So far we have studied a single degree of freedom mass-spring-damper system. Let us now develop a more realistic, but still somewhat simplified, mechanical model of an actual rotor. Consider a single rotor with a concentrated mass (m), supported on two bearings, as shown in *Figure 4-3*. The bearings are considered infinitely stiff; i.e., they do not allow for any displacement. However, the shaft itself still has certain stiffness and damping characteristics; if the shaft experiences sufficiently unbalance force at its mass center, it will bend or vibrate.

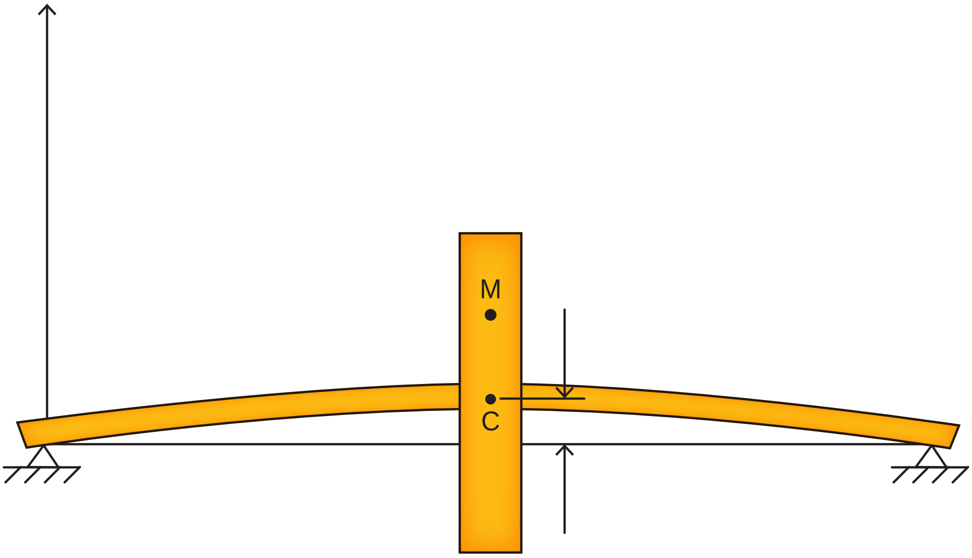


Figure 4-3. *Simple Rotor Model*

NOTE: Clearly, a real rotor never has a mass concentrated at one single point. Thus, for a more advanced rotordynamic analysis the modal mass, which accounts for the rotor's weight distribution, rather than the actual mass of the rotor is used. Typically, the modal mass of a rotor is between 50% and 80% of its actual mass.

An unbalance force occurs when the rotor's center of gravity is not perfectly aligned with the axis of rotation of the rotor. This effect is called rotor mass unbalance (RMU) or shaft bow. The rotordynamic definition of rotor mass unbalance is:

$$RMU = W_u \cdot R_u$$

where W_u (unbalance weight) is a small weight located a distance R_u (unbalance radius) away from the axis of rotation. The eccentricity (e) of a rotor is thus:

$$e = \frac{RMU}{W}$$

where W is the weight of the rotor. To determine the actual unbalance force magnitude on a rotor, we multiply the rotor's unbalance mass by its centrifugal acceleration (f=ma):

$$F = m_u\ \omega^2 = m\ e\ \omega^2$$

Since the rotor is spinning at a fixed angular speed, the unbalance force acts on the shaft in the form of a periodic sinusoidal function:

$$f_u = F\sin(\omega t) = m\ e\ \omega^2 \sin(\omega\, t)$$

For example, if a small 1000-hp gas turbine shaft rotates at 20,000 rpm (=2094 rad/sec) and has an unbalance weight of 100 grams located 5 mm away from the shaft center, the resulting unbalance force is 2190 N (approximately 100 lbf).

The equations of motion for the two degrees-of-freedom supported rotor *(as shown in Figure 4-3)* are again derived using Newton's second law. When analyzing the model, we realize that there are two degrees-of-freedom rather than one as in the previous example; namely, the system can vibrate in the x and y directions. The equations of motion for this system are:

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = m\ e\ \omega^2 \cos(\omega t)$$

$$m\frac{d^2y}{dt^2} + c\frac{dy}{dt} + ky = m\ e\ \omega^2 \sin(\omega t)$$

For now we will assume that the damping is very small and can be neglected. The stiffness of the shaft from basic material properties is:

$$k = -48\frac{EI}{L^3}$$

where E is Young's Modulus, I is the area moment of inertia, and L is the rotor length. We can determine the natural frequencies of the system if we neglect the unbalance force and solve for the Eigenvalues:

$$m\frac{d^2x}{dt^2} - 48\frac{EI}{L^3}x = 0$$

$$m\frac{d^2y}{dt^2} - 48\frac{EI}{L^3}y = 0$$

Thus,

$$\omega_{Nx} = \sqrt{\frac{k}{m}} = \sqrt{48\frac{EI}{mL^3}}$$

$$\omega_{Ny} = \sqrt{\frac{k}{m}} = \sqrt{48\frac{EI}{mL^3}}$$

In rotordynamics, these natural frequencies are also called the undamped critical speeds. If the rotor operates at any undamped critical speed or a multiple thereof, the system is excited at one of its natural frequencies and may resonate.

Clearly, it is desirable to operate the gas turbine shafts away from these critical speeds. One criteria that is often employed to evaluate a gas turbine's rotordynamic adequacy is the critical speed margin, which is the difference between the shaft operating speed and the nearest critical speed:

Critical Speed Margin:

$$\omega_{m\arg in} = \omega_{operating} - \omega_{critical}$$

The critical speed margin is also often expressed as the separation margin:

Separation Margin:

$$SM = \frac{\omega_{critical} - \omega_{operating}}{\omega_{critical}}$$

For rotordynamic design purposes a, 15% separation margin, SM =15%, is desirable and usually adequate. However, in some cases a gas turbine shaft is required to pass a critical speed during startup and shutdown. In these cases, the rotor system must be sufficiently damped to prevent large vibrations while traversing the critical speeds. If the vibrations amplitudes exceed the design clearances, rubbing between the gas turbine's rotor and housing or even catastrophic gas turbine failure may be the result.

The steady-state "particular" solution for the above shaft equations of motion is identical in form to the solution for the previously analyzed simple mass-spring-damper system, except that two degrees of freedom must now be considered and that the excitation force is F= $me\omega^2$. Thus, the solution is expressed in the form of:

$$x = X\sin(\omega t - \varphi)$$

$$\varphi = \tan^{-1}\left(\frac{c\omega}{k - m\omega^2}\right)$$

$$y = Y\cos(\omega t - \varphi)$$

where:

$$X = \frac{m\,e\,\omega^2}{\sqrt{\left(k - m\omega^2\right)^2 + (c\omega)^2}}$$

$$Y = \frac{m\,e\,\omega^2}{\sqrt{\left(k - m\omega^2\right)^2 + (c\omega)^2}}$$

These equations are made arranged to:

$$Z_x = \frac{Xk}{F} = \frac{m\,e\,\omega^2}{\sqrt{\left(1-\left(\frac{\omega}{\omega_N}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_N}\right)\right)^2}}$$

$$Z_y = \frac{Yk}{F} = \frac{m\,e\,\omega^2}{\sqrt{\left(1-\left(\frac{\omega}{\omega_N}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_N}\right)\right)^2}}$$

In rotordynamics the force response (Z) is also called unbalance response. The peak vibration amplitudes are:

$$X = \frac{m\,e\,\omega^2}{2\zeta k} = \frac{m\,e\,\omega^2}{c\ \omega_N}$$

Using the above results, we can plot an unbalance response plot (Z versus ω/ω_N) as shown in *Figure 4-4*. An unbalance response plot is crucial for gas turbine design since it defines and limits the speed ranges at which the gas turbine shaft may operate. Clearly, the unbalance response magnitude at the critical speed (ω/ω_N =1.0) is again seen to be strongly dependent on the damping factor.

The unbalance response plot *(Figure 4-4)* also shows that for increased damping ratios, the peak amplitude of vibration occurs at frequencies slightly above the undamped critical

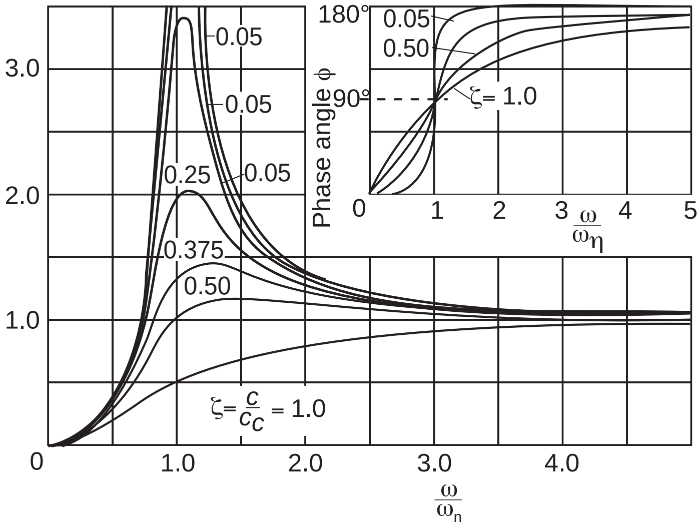


***Figure 4-4.** Unbalance Response Plot*

speed. This small critical speed deviation from the natural frequency can easily be derived from the above unbalance response equation. Consequently, the frequency correction for the undamped critical speed is:

$$\omega_{ucr} = \frac{\omega_N}{\sqrt{1-2\zeta^2}}$$

The corrected undamped critical speed is usually called the unbalance critical speed. Although this appears to be only a quantitatively small deviation from the undamped critical speed, the unbalance critical speed correction is important for the accurate determination of a rotor's critical speeds, especially for well damped systems.

In some cases, if a rotor system is not adequately damped, the transient "complementary" solution to the equations of motion may also affect the rotordynamic performance and lead to rotor instability. Without going into too much detail, we should state that from the "complementary" solution, another set of Eigenvalues can be derived, which are called the damped critical speeds.

A transient analysis of the equations of motion shows that the damped critical speeds are related to the undamped critical speeds by:

$$\omega_{dcr} = \omega_N \Big/ \sqrt{1-2\zeta^2}$$

Theoretically, a rotor system remains stable at the damped critical frequency as long as the damping factor (ζ) is positive. However, in any real rotordynamic application, experience has shown that the damping factor at the damped critical speeds should exceed 0.1 ($\zeta>0.1$). Care must be taken when performing the damped critical speed analysis since bearing damping will significantly affect the overall system damping.

Multi-Degrees of Freedom System

So far we have analyzed a simple rotor by assuming that the rotor has a concentrated mass and bending movement in only two directions (x,y); i.e., a two degrees of freedom system. In reality we know that this is an over-simplification. Namely, a real rotor can bend in different shapes and the bearings allow for some limited vibration of the shaft. A real rotor thus has multiple degrees of freedom. The degrees of freedom can be divided into three categories which are descriptive of the types (modes) of vibrations the rotor experiences: rigid body modes, lateral bending modes, and torsional modes.

Rigid Body Modes: Rigid body modes are the vibrations the rotor undergoes if it were perfectly stiff and not allowed to bend. A rigid rotor can move in one axial direction (z) and two radial directions (x,y).

Lateral Bending Modes: As previously shown, a typical rotor is not infinitely stiff and thus can bend laterally. There are theoretically an infinite number of possible bending shapes; however, for most gas turbine rotordynamic analysis, it is adequate to study the first four bending modes only. The actual shape of the bending modes is strongly dependent on the bearing locations, shaft bow, bearing and shaft stiffness.

It must be noted that the stiffness of the bearings involved impacts the mode shapes *(Figure 4-5 and 4-6)*. For example the first mode is a true rigid body mode if the bearings are not very stiff. Stiff bearings limit the movement of the shaft at the bearings, thus causing the shaft to bend.

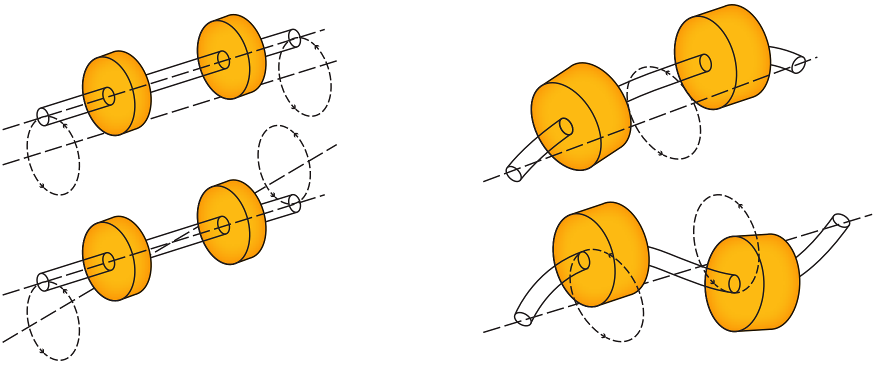


Figure 4-5. *Rigid Body (left) and Bending Modes (right)*

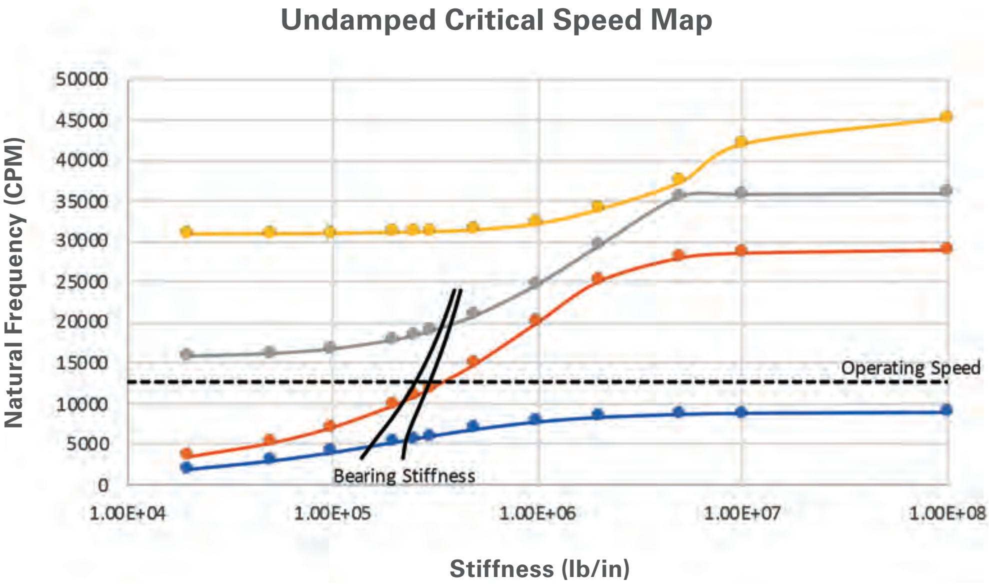


Figure 4-6. *Undamped Critical speed map, showing critical speeds for the first modes of a rotor depending on the bearing stiffness. For a known bearing stiffness, the chart allows to determine the undamped critical speed of the rotor system. Depending on the stiffness, the modes can be rigid body modes (low stiffness), or bending modes (high stiffness)*

Torsional Modes: Due to the torque on the shaft, there also can be torsional twisting. On a single shaft, there is typically only one relevant torsional mode: simple rotor twist.

The equations of motion for a multi-degrees of freedom system are still derived from Newton's second law (f=ma). However, one equation of motion is required for each degree of freedom. This results in a rather large system of equations, even for a simple shaft model. To simplify the mathematical syntax, this system of ordinary differential equations is usually expressed in matrix form:

$$\begin{vmatrix} m_{11}\ m_{12}\ m_{13} \dots m_{1n} \\ m_{21}\ m_{22}\ m_{23} \dots m_{2n} \\ m_{31}\ m_{32}\ m_{33} \dots m_{3n} \\ . \quad . \quad . \quad . \\ . \quad . \quad . \quad . \\ m_{n1}\ m_{n2}\ m_{n3} \dots m_{nn} \end{vmatrix} \begin{vmatrix} x_1'' \\ x_2'' \\ x_3'' \\ . \\ . \\ x_n'' \end{vmatrix} + \begin{vmatrix} c_{11}\ c_{12}\ c_{13} \dots c_{1n} \\ c_{11}\ c_{12}\ c_{13} \dots c_{1n} \\ c_{11}\ c_{12}\ c_{13} \dots c_{1n} \\ . \quad . \quad . \quad . \\ . \quad . \quad . \quad . \\ c_{11}\ c_{12}\ c_{13} \dots c_{nn} \end{vmatrix} \begin{vmatrix} x_1' \\ x_2' \\ x_3' \\ . \\ . \\ x_n' \end{vmatrix} + \begin{vmatrix} k_{11}\ k_{12}\ k_{13} \dots k_{1n} \\ k_{11}\ k_{12}\ k_{13} \dots k_{1n} \\ k_{11}\ k_{12}\ k_{13} \dots k_{1n} \\ . \quad . \quad . \quad . \\ . \quad . \quad . \quad . \\ k_{11}\ k_{12}\ k_{13} \dots k_{nn} \end{vmatrix} \begin{vmatrix} x_1 \\ x_2 \\ x_3 \\ . \\ . \\ x_n \end{vmatrix} = \begin{vmatrix} f_1 \\ f_2 \\ f_3 \\ . \\ . \\ f_n \end{vmatrix}$$

Here x_i is the displacement variable and n is the number of degrees of freedom. This can also be written as:

$$[M]x''+[C]x'+[K]x=[F]$$

where M is the mass matrix, C is the damping matrix, K is the stiffness matrix, and F is the force vector. For more complicated systems, which may include multiple bearings, coupled shafts, and/or flexible support structures, the mathematical model may become very complex, with matrices easily reaching fifty or more degrees of freedom. The matrix equation can be solved for the Eigenvalues using a matrix solver, such as a Gauss-Seidel matrix inversion method. It is beyond this text to describe a detailed analysis, which can be found in most rotordynamics textbooks. However, it is important to note the following:

Each degree of freedom can have a distinct undamped and damped critical rotor speed. Additional resonances can also occur at any multiple of the frequency of the original damped/undamped critical speed.

NOTE: Since the equations of motion for the torsional vibrations are not coupled physically and mathematically with the lateral bending and rigid body equations of motion, the torsional critical speeds are typically calculated in a separate and independent Eigenvalue analysis.

Rotordynamic Plots

It is often convenient to present a gas turbine's rotordynamic behavior in graphical form. The most common rotordynamic graph is called the frequency response plot, which is very similar to the unbalance response plot previously presented. The frequency response plot is generated by either experimentally or numerically determining the shaft unbalance response for a range of rotor running speeds and plotting the results in the form of vibration amplitude (mils) versus shaft speed (rpm or Hz). For example, *Figure 4-7* shows a typical compressor shaft speed response plot from 0 to 12,000 rpm. The first, second, and third critical speeds are easily identifiable by the asymptotic response behavior on the frequency response plot.

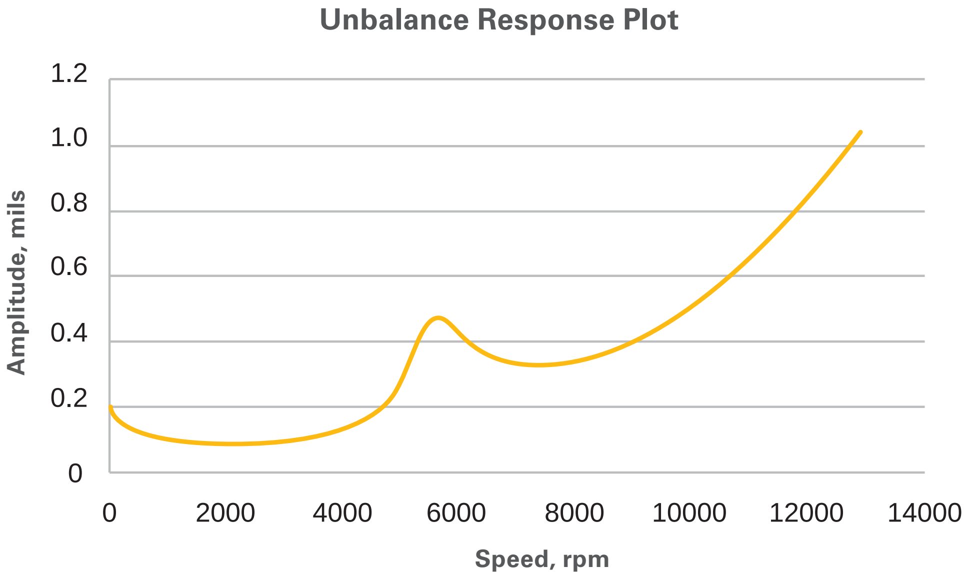


Figure 4-7. *Response Plot*

Figure 4-8 shows typical mode shapes that correspond to critical speeds. It is important to remember that the critical speed and mode shape are system specific; i.e., they cannot easily be generalized to any system. Hence, to accurately determine the critical speeds and mode shapes of a given system, an Eigenvalue analysis must always be performed. However, in general we can state the following:

Typically, the first two critical speeds (lowest frequencies) corresponds to rigid body mode, the next higher frequency critical speed is lateral bending mode.

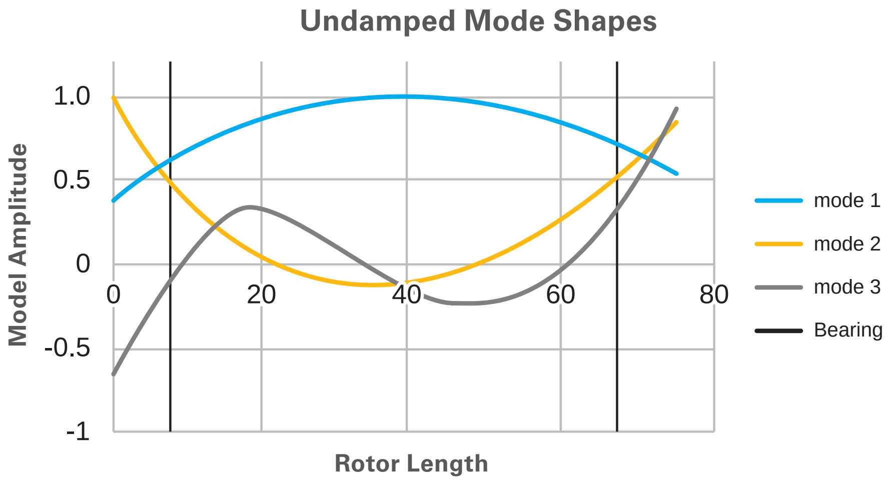


Figure 4-8. *Mode Shapes*

Critical speed analysis results are also often plotted in the form of a Campbell diagram as shown in *Figure 4-9*. The Campbell diagram presents the individual critical speeds of a rotor as a function of the rotational speed. Rotor physical characteristics such as damping and stiffness are not system constants, but can vary with the angular speed and/or operating conditions of the system. Hence, a plot showing the locations of the critical speeds as a function of rotor speed is essential for the gas turbine designer to determine allowable shaft operating ranges.

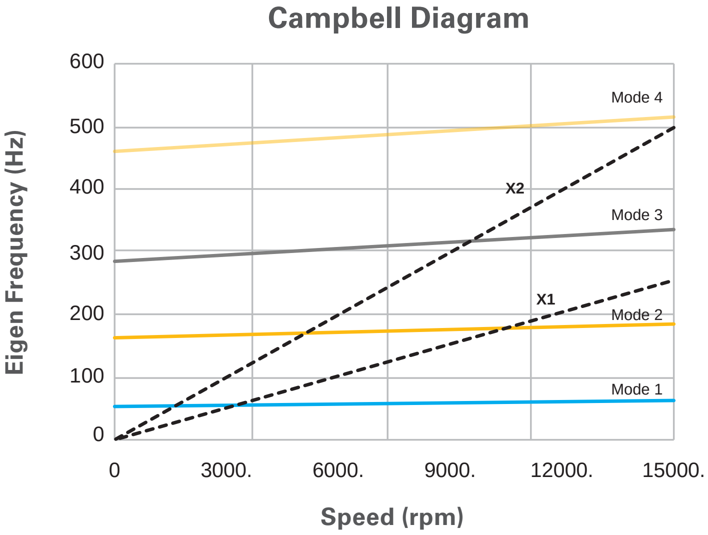


Figure 4-9. *Campbell Diagram*

Rotordynamics Testing and Instrumentation

To experimentally determine rotordynamic performance such as critical speeds and unbalance response of a gas turbine, some standard tests can be performed. The two most common of these tests are the ping (or wrap) and the rundown test. For the ping test the rotor is vibration isolated and then excited with an extremely short but high amplitude impulse force. For example, the rotor is hung on a long, thin wire to mechanically isolate it and then hit with a metal hammer to apply the impulse. This short, high frequency impulse effectively excites the rotor simultaneously at all possible frequencies. By measuring the frequencies of the peak amplitudes of the resulting rotor vibrations, the natural frequencies and forced response of the rotor can be identified. On the other hand, the rundown test is performed by simply accelerating a gas turbine shaft to its maximum operating speed, decoupling it and then carefully measuring its vibration amplitudes and frequencies as it decelerates. Clearly, a rundown test can only performed if the gas turbine's rotordynamic stability and safety is already well established.

There are three types of transducers that are commonly employed to measure rotordynamic vibrations on gas turbines packages. These are:

Proximity Probes: Proximity probes measure the actual rotor displacement (x) relative to a fixed position. Modern proximity probes are usually either magnetic reluctance, eddy current, or optical pickups. Most gas turbines employ a set of orthogonal radial eddy current proximity probes for each radial bearing and two axial eddy current proximity probes to monitor shaft vibrations relative to the gas turbine casing. These proximity probes can typically measure vibrations frequencies between 0 and 10 kHz.

Velocity Transducers: Velocity transducers measure the first derivative (dx/dt) of the displacement (i.e., the actual free movement velocity) using a miniature piezo-electric or piezo-resistive mass-spring system. These transducers are typically employed to measure lower frequency free vibrations (100-1000 Hz) such as the gas turbine case, skid and other subsynchronous vibrations.

Accelerometers: Accelerometers typically employ a miniature piezo-electric or piezo resistive mass-spring system to measure the second derivative of the displacement (dx^2/dt^2); i.e., the actual free movement acceleration. Accelerometers are most often used to measure high frequency vibrations (1-100 kHz) such as the gas turbine gear meshing and blade interaction frequencies.

Most gas turbines incorporate a combination of these three types of transducers to accurately monitor and diagnose the shaft's rotordynamic behavior. For safety reasons, no gas turbine should ever be operated without properly functioning vibration instrumentation and associated package alarm/shutdown switches.

Often, a transducer is combined with a spectrum analyzer to examine the rotordynamic behavior. A spectrum analyzer performs a mathematical transformation of the time/vibration signal into the frequency domain called a Laplace or Fourier transform (also sometimes referred to as an FFT or Fast Fourier Transform). The output from the spectrum analyzer FFT provides a plot of vibration amplitude versus frequency which is very similar to the previously presented frequency response plot.

Shaft Balancing

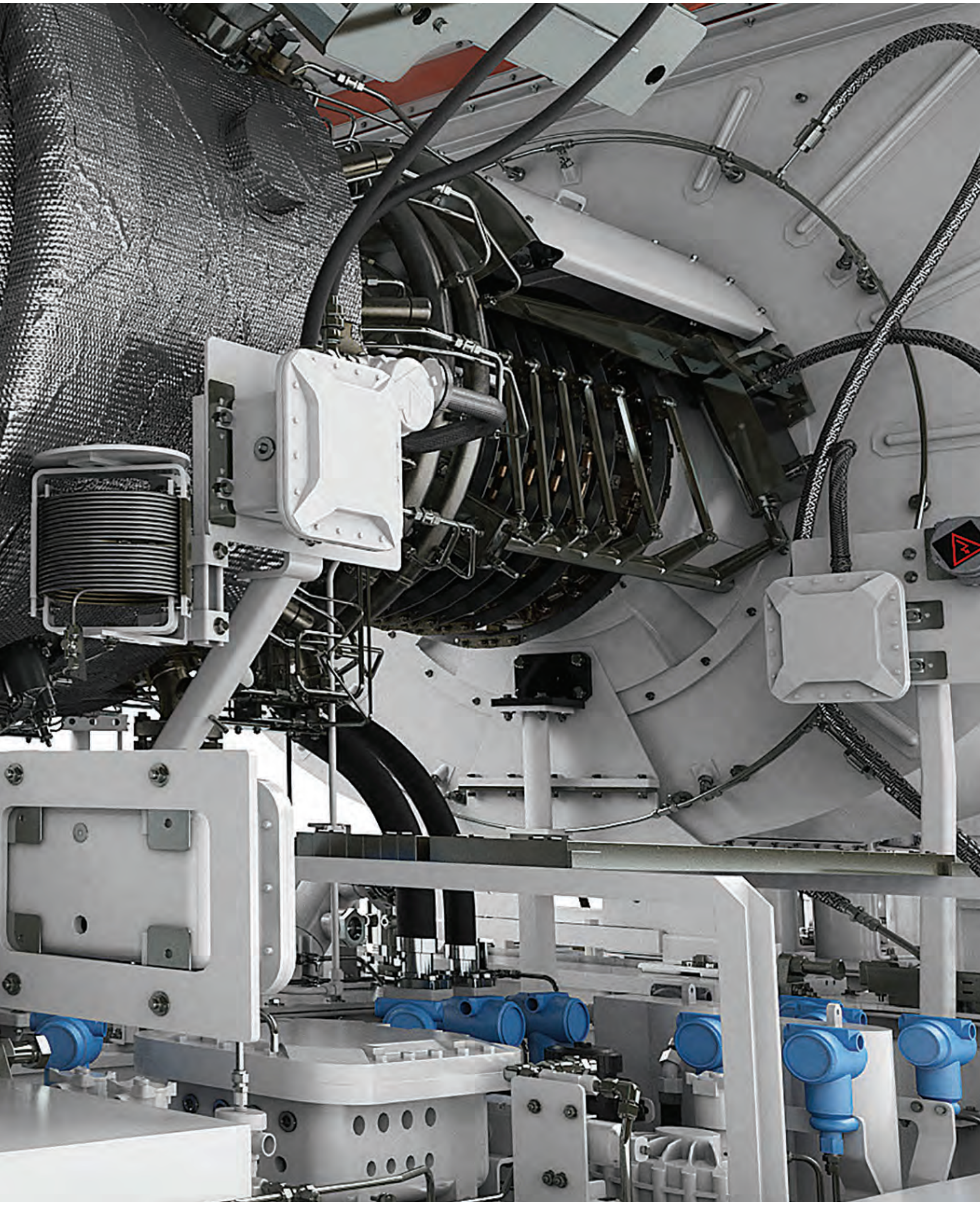
To minimize vibrations, a gas turbine's shaft must be mechanically balanced; i.e., eccentricities must be physically eliminated. This is typically achieved by applying counterweights and/or by removing material from the rotor at strategic radial locations. Rather than removing material from the rotor, which may permanently damage the shaft, gas turbines are typically balanced using counterweights. This method is also called trim balancing. Gas turbines usually have several special access points along the shaft to allow for easy application/removal of the balancing weights. Balancing weights range from fractions of one gram up to one hundred grams. Several mathematical procedures have been derived to determine the correct mass and radial location of the counterweights for proper trim balancing.

To maintain proper rotor balancing, the above trim balancing procedure has to be repeated if any gas turbine blade, disk, shaft, seal, and bearing elements are exchanged or repaired. Also, cyclical/thermal loads and fatigue stress may affect the rotor balance, so periodic trim balancing is required for some applications. For the above reasons a gas turbine should always be designed to allow for easy access, removal/installation and site balancing of the rotor.

SUMMARY

In the preceding chapter, a short analysis of the compressor, combustor, gas producer turbine and power turbine functions was discussed. Also examined was the application of some simple engineering concepts, such as Euler's formula, velocity polygons, rotordynamics and combustion stoichiometry, to these components.

The authors hope that the reader of this text understands that the preceding chapter represents just a very brief and simplistic overview; there are many books and research papers solely dedicated to each of the presented topics. We encourage the reader to study these topics in more detail.



CHAPTER 5

ANCILLARY & AUXILIARY SYSTEMS

The packaging of gas turbine driven compressors and generators, especially for oil and gas applications, presents some very special challenges, and gas turbine manufacturers have developed specialized packaging options for these different types of services. Fundamentally, the package systems are designed to provide the gas turbine with its utility requirements (air, fuel, oil, and water), control the operation of the unit and process, and assure safety. The gas turbine, the gas compressor, and, if necessary, the gearbox are usually mounted on a skid, together with most of the systems described below *(Figure 5-1)*. Thus, the primary gas turbine package systems are:

Starting

Lube oil

Fuel

Seal gas (if dry gas seals are utilized)

Fire/gas detection and fire fighting

Inlet and exhaust air

Enclosure (Figure 5-2)

Control and instrumentation

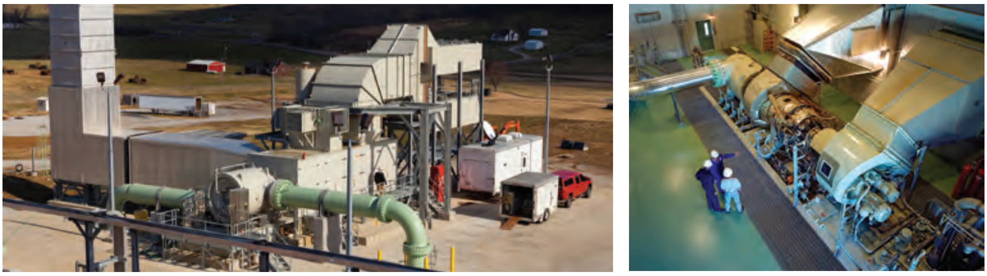


Figure 5-1. *Gas turbine package driving a centrifugal compressor. Left: Fully enclosed, Right: Installed in a building*

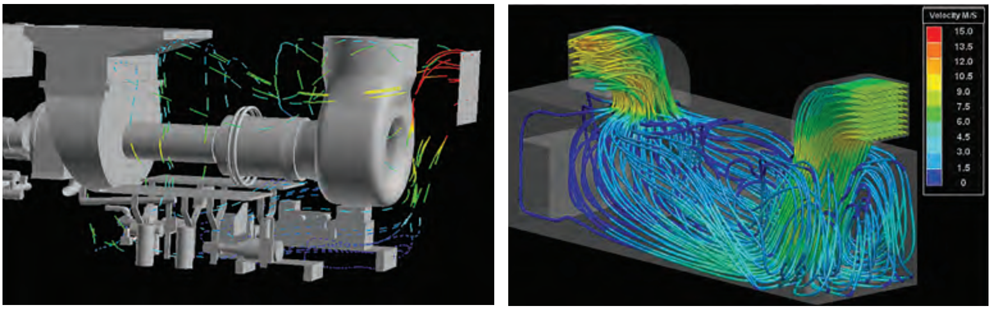


Figure 5-2. *Analysis of the Air Flow in a Gas Turbine Enclosure*

Each of these package systems has a number of subsystems and components. Generally, package systems are classified into on-skid and off-skid systems. On-skid classification generally corresponds to all equipment inside the gas turbine enclosure, while off-skid or outside the package refers to equipment connecting to the gas turbine package, such as ancillaries and auxiliaries. The main turbo compressor's ancillary equipment, classified by this definition, is listed below:

On-Skid – Inside the Package

Fuel system and spark igniter

Natural gas (control valves, filter)

Liquid (pumps, valves

Bearing lube oil system[1,2]

Tank (integral)

Filter (simple, duplex)

Pumps (main, pre/post, backup)

Accessory gears[3]

Fire/gas detection system

Starter/helper drive

Pneumatic, hydraulic, or variable speed AC starter motor

Controls and instrumentation (on-skid, off-skid)

Seal gas/seal oil system (compressors)

Off-Skid – Outside the Package

Enclosure with lifting devices and fire protection system[4]

Enclosure ventilation

Inlet system

Air-filter (self-cleaning, barrier, inertial, demister, screen)

Silencer

Inlet fogger/cooler

Anti-icing system

Exhaust system

Silencer

Stack

Lube oil cooler (water, air)

Fuel filter/control valve skid

Off skid control system

Motor control center

Switchgear, neutral ground resistor

Yard valves (load, recycle, anti-surge)

Turbine cleaning system (on-line, on-crank)

Documentation

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[1] *Some packages have two separate lube oil systems.*

[2] *Lube oil systems/lube oil tanks can also be separate skids.*

[3] *The accessory gear is used by the starter but can also drive the main lube oil pump and hydraulic pumps.*

[4] *Enclosures can be mounted on the skid, or they can be of the drop-over type.*

MACHINERY CONTROL AND PROTECTION

As gas turbines are complex mechanical devices, they require electronic and mechanical controls, as well as, instrumentation. The gas turbine must be protected from operational upsets and process excursions. The control system has to be designed to prevent the gas turbine, and its driven equipment, from operating in unsafe modes—be it overspeed, excessive operating temperatures or unsafe vibration levels. Problems with lube oil supply, fuel supply, air supply, sensors, and the control system itself have to be detected, and evaluated. Thus, the gas turbine's unit control system principal design function is to protect the turbomachinery train from these types of upsets.

The unit control system relies on auxiliary measurements (flow, temperature, and pressure) for reaching set alarm points and/or shutdown levels, and it initiates auxiliary automatic valves (surge, loading, isolation, blowdown, etc.) action to avoid or mitigate a flow upset condition. An off-skid backup battery rack and battery charger are employed to bring the unit to a safe shutdown upon a significant operational upset.

At a minimum, the control system of a gas turbine driven compressor must provide the following functions through the use of a Human Machine Interface (HMI):

Machinery Monitoring and Protection

- Equipment startup, shutdown and protective sequencing
- Stable equipment operation
- Alarm, shutdown logic
- Backup (relay) shutdown

Driven load regulation

- Fuel/speed control
- Process control
- Surge control
- Communication (SCADA) interface

UTILITIES

Usually, the quality of only three or four utility fluids can affect a gas turbine's life and performance. These are the lube oil, the combustion fuel, the inlet air, and possible water used for washing or performance enhancement. To protect the machine, the air, fuel, and oil utility streams must be maintained within the manufacturer's specifications, which are primarily achieved by proper filtration.

Inlet Air Filters

The selection of an inlet filtration system for a new or existing gas turbine installation is often under appreciated and handled as one of the lower priority decisions in the overall equipment purchasing process. Considering that the inlet system only constitutes a small fraction of the total purchase price for a gas turbine package, this may appear appropriate.

However, the proper selection of the inlet filtration system can have a substantial impact on the overall lifecycle cost of the gas turbine installation. While many operators only consider the filter replacement and sometimes the gas turbine's output power loss associated with the inlet pressure drop in their cost analysis, other factors that can affect the overall

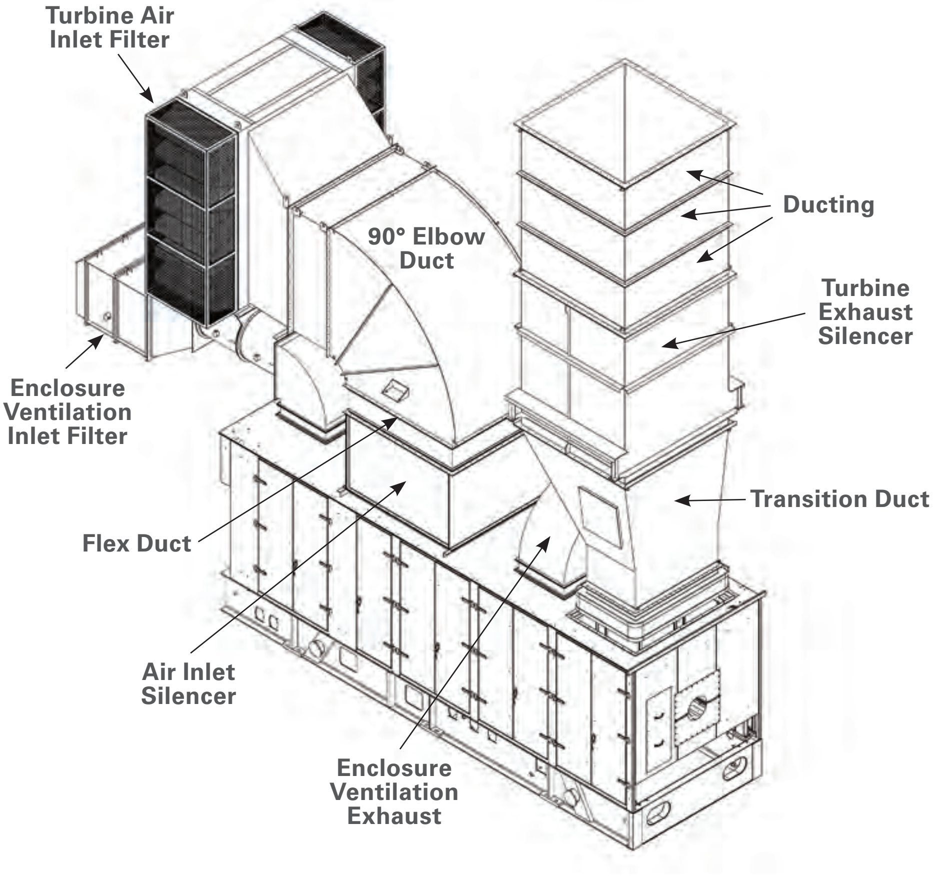

Turbine Air Inlet Filter
Ducting
90° Elbow Duct
Turbine Exhaust Silencer
Enclosure Ventilation Inlet Filter
Transition Duct
Flex Duct
Air Inlet Silencer
Enclosure Ventilation Exhaust

Figure 5-3. *Fully enclosed gas turbine package with inlet air filters and exhaust ducting*

operation and maintenance cost of a plant include: downtime associated with offline water washing, reduced time-between-overhaul, and startup reliability. All of these can be directly attributed to filter selection. *Figures 5-1 and 5-3* show fully enclosed mid-size gas turbine packages with inlet filter and exhaust stack all mounted on a single skid. In Chapter 10, we will discuss this topic in greater detail.

Lube Oil Filtration

Contamination, such as particles, water, and process fluids in lubricant streams can lead to increased maintenance costs and reduce equipment service life on lubrication systems, such as the turbine bearing lube and hydraulic control systems. Gas turbine manufacturers have very strict guidelines for lube oil quality, and failure to maintain the lube oil to these standards is often sufficient reason for manufacturers to reject warranty claims.

Most modern gas turbines employ duplex lube oil filters with a continuous-flow transfer valve. The media inside these filters can range from paper cardboard to synthetic matrix fibers. Multi-layer filter cartridges with a high flow, high-efficiency media are typically provided by the manufacturer when the units are shipped. Differential pressure transducers to monitor the pressure drop across the filter media should also be mounted to the unit control system, with electronic signals and alarms for trending purposes.

Fuel Gas Quality

"Clean" fuel is essential at today's elevated gas turbine firing temperatures, since modern engines are more sensitive to high temperature corrosion if there are impurities present in the fuel and/or in the combustion air. The fuel must be free of chemical impurities and solids as these either stick to the fuel passages and blades of the turbine or damage the components in a turbine that operates at high temperatures. Older engines may be less sensitive because of lower temperature operation, however, contaminated fuel and/or combustion air can still lead to performance degradation, premature maintenance, and operating difficulties. Keeping fuels free of contaminants such as water, compressor oils, and dirt are important to efficient operation and low maintenance. Filter and coalescers for liquid/gas separation are available to remove impurities as small as 5 microns.

Fuel gas heating is often employed to ensure that 100% of the fuel that enters the gas turbine is in the gaseous state. To ensure this, the fuel must be at a temperature above its dew point. However, the natural gas' dew point is a strong function of its hydrocarbon composition; the higher the percentage of heavy hydrocarbons in the fuel, the higher the dew point. Consequently, swings in the composition of the natural gas can easily move the average dew point of the pipeline's gas significantly. If the fuel constituents in the gas turbine combustion system vary significantly from the design point, liquids may form in the gas turbine's fuel system and combustor damage will rapidly result.

A fuel heating system should be directly interfaced with either the unit control system or the station control system to maintain a constant fuel gas temperature above the dew point, and to be automatically activated during significant fuel quality swings. Also, electric heat tracing of the fuel supply lines, rather then natural gas burners, is a safer and more controllable means to maintain the fuel gas temperature. Fuel gas related issues are discussed in detail in Chapter 9.

OFF-SHORE APPLICATIONS

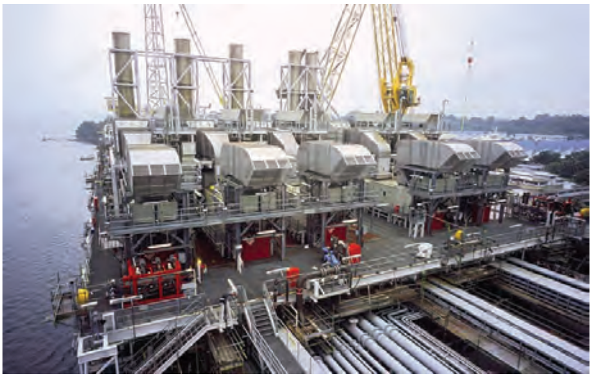


Figure 5-4. *Gas Turbine Installation on an FPSO*

Gas turbines are preferred prime movers for power generation, as well as, drivers for compressors and pumps for offshore installations *(Figure 5-4)*. Many offshore projects, especially in deep waters, use Floating Production Storage and Off-Loading Systems (FPSO) as the equipment platform. This type of system, as well as, semi-submersible platforms, SPAR's, and TLP's, exhibit significant deck movement as a result of wave and wind action, as well as, possible deck deflections. This requires specific design considerations for the turbomachinery packages employed to achieve the highest possible availability and reliability. There are various types of structures used for oil and gas exploration and production, either bottom-supported structures or vertically moored structures, i.e., Fixed Leg Platform (FLP), Compliant Tower (CT), Tension Leg Platform (TLP) and Mini Tension Leg Platform (Mini TLP)] and the Floating Production and Sub-Sea category consisting of Spar, Semi-Submersible (SS) and Floating Production Storage and Off-Loading (FPSO).

Offshore platforms and Floating Production Systems perform complex operations in a compact space. As a result, there is a high value placed on the installed size of offshore equipment. In addition to the installed footprint, successful design for operation in offshore applications must address a variety of load cases. A prime initiator of many of these load cases is the environment. Wind and its resulting effects are key contributors to sea conditions causing wide variations from the relatively calm condition to the very severe condition as evidenced, for example, in the Gulf of Mexico during hurricane Katrina. At one stage, a Category 5 hurricane, Katrina ultimately made landfall in Louisiana and Mississippi, at Category 4 strength.

In addition to considering the effects on the environment, code compliance to assure safe, reliable equipment has a role to play. The framework of directives and standards, such as ATEX and CSA, are one piece of the puzzle. There are other international, national, and local rules and regulations that must also be satisfied. These other rules and regulations are represented by two groupings of authority. International Authority, represented by an International Class Society, may represent a country's government's interest and also have the capability to enforce the laws. Additionally, each country may have a local authority. In the United States, the USCG Code of Federal Regulations (CFR) serves this function. Each Class Society has individually developed rules for building and classification of vessels. Depending on the type of machinery, and its usage, several options are available.

Unlike a land-based application, movements in all six degrees-of-freedom are considered when designing the machinery for floating applications *(Figure 5-5)*. It is critical to understand the applicable class requirement, the type of vessel, the expected deck deflection, and where the equipment will be located on the vessel.

This information is then translated into the design conditions of:

- Type of mounting method used to attach the package to the deck
- Acceleration for structural analysis
- Static and dynamic angles for fluid management

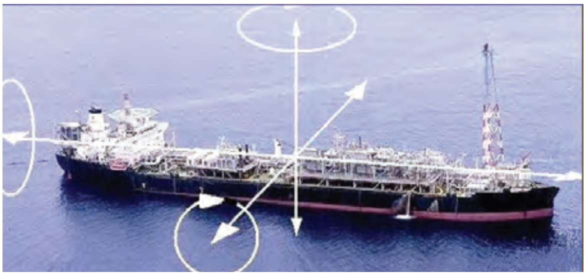


Figure 5-5. *Degrees-of-Freedom for Dynamic Movement and Static Displacement*

Wind and Wave action cause stresses and deflections in the Floating Production System, resulting in hogging, sagging and twisting of the vessel's hull. Apart from the external connections between the package and site interfaces, the machine's drivetrain alignment requires close attention.

Figure 5-6 illustrates how, if not properly addressed, the effect of the deck deflection can translate through the base-plate and compromise the machinery's alignment. Insufficient stiffness in the machinery's supporting structure, or improper attachment to the deck, can result in unwanted vibration levels. *Figure 5-6* illustrates two methods of attaching a turbine package to a vessel's deck. The four-point tie-down graphic on the left does not decouple the vessel's structural deflections from the package. Although acceptable under certain conditions, in this mounting arrangement, the package will see more of an effect from deck twist. On the other hand, the three-point tie-down graphic on the right allows the vessel's structural deflections to become decoupled from the package. With less effect from deck twist being transmitted to the package, this is generally considered to be the preferred arrangement for offshore applications.

Understanding the benefit of three-point mounting is only one aspect. The success of three-point mounting the equipment is dependent on what is used to connect the package to the deck. Two common mounting methods are utilized to transition from the turbine package to the deck.

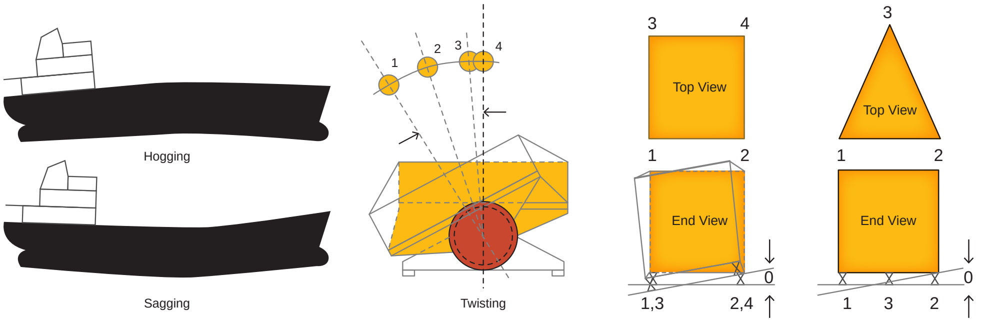
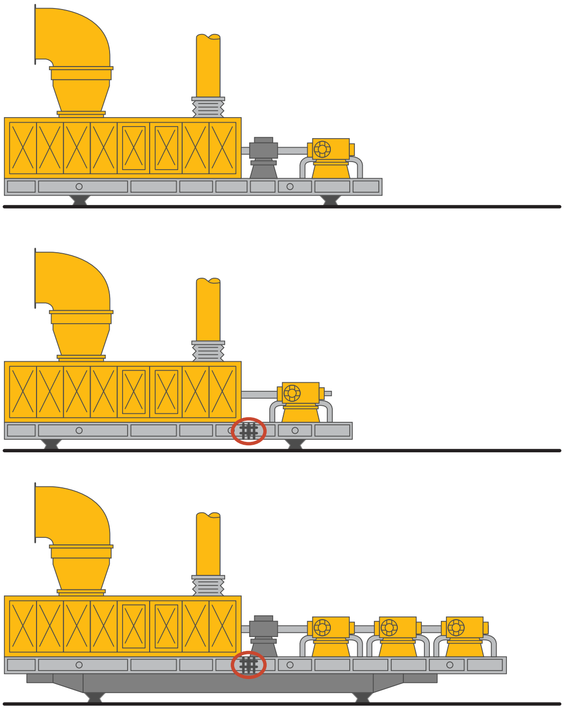


Figure 5-6. *Deck Deflections and their Impact on a Turbomachinery Package for 4-Point and 3-Point Mounts*

One method is using an Anti-Vibration Mount, commonly referred to as an AVM. It is located between the package and the deck in sets of three. AVMs are positioned to most effectively carry the package's operational load requirements. The AVM affords a level of protection to the package and drivetrain alignment from deck movement, but less than that provided by a gimbal design. AVMs are used when there is a need to isolate the turbomachinery package from deck vibration; or, when installation near onboard living quarters mandates a level that cannot be achieved without them. AVM selection is project specific and depends on the package weight, center of gravity location, and application (i.e., moderate or severe duty).

The second method commonly used is the gimbal mount. The gimbal mount is also located between the package and the deck in sets of three. Like the AVM, gimbal mounts are positioned on the package to most effectively carry its operational load requirements. A gimbal is designed with a spherical bearing that allows the two mounting surfaces to move independent of one another. This, in effect, isolates the package and the drivetrain alignment from the deck's constant movement. The gimbal is chosen when the package will be exposed to excessive deck deflection, or the deck deflection data is not available. Although three-point mounting decouples the turbomachinery from the vessel's structural deflections, it does not alleviate the requirement to react to the dynamic forces applied to the components supported by the package structure. Structural requirements, transportation needs, and handling all play a role in the design of the base-plate resulting in three basic methods *(Figure 5-7).*



Single-Piece Base Plate

Two-Piece Base Plate

Two-Piece Base Plate with Sub-Base

Figure 5-7. *Base-Plate Design*

- Single-piece minimizes the number of on-site connections, but depending on factory needs and/or transportability to site, this is not always a practical solution.

- Two-piece is an appropriate solution when transportability, manufacturing, and non-operational requirements take precedence. This allows the machine to be easily moved and transported in smaller modules, and then assembled either dockside for single lift or assembled on the deck.

- Two-piece with a sub-base is appropriate for large machines with long drive trains. More easily managed in the factory and during transport to site in smaller modules, these long base-plates alone do not provide enough structural rigidity for the severe environmental conditions of an FPS. As a result, a sub-base structure under the entire machine, designed to maintain the powertrain alignment, is required.

To develop and predict how the machinery will respond to the various load cases, finite element models are used to evaluate the base-plate designs. With the environmental loads applied to the model, deflection and stresses can be determined. Using this information, structural design of the base-plate can be compared with established design criteria to ensure it meets the application requirements. In addition to the base-plate analysis performed in the as-installed condition, analysis is also performed for the offshore lift scenarios. In offshore applications, the engine-handling kits are designed with an integrated braking system for safety. A chain is welded along the underside of the beams, and a sprocket is incorporated into the hoist.

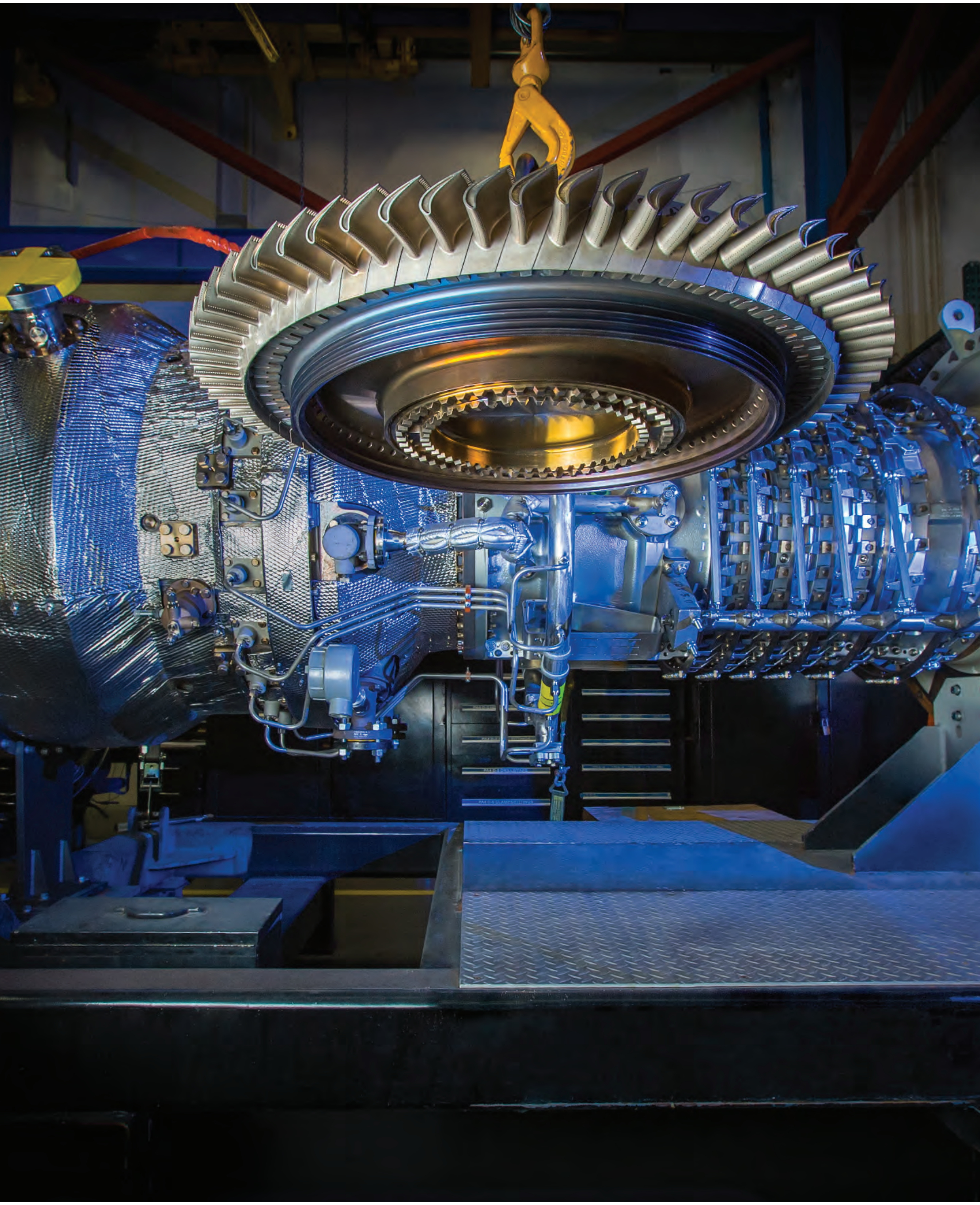
In the same way that the base-plates are analyzed using finite element analysis, the enclosure and ancillary structures are also evaluated using 3D beam analysis.

Fluid management is another important aspect of ensuring successful operation of the equipment. Knowing the installed package orientation relative to the vessel’s centerline is a key data point. Installation perpendicular to the vessel’s centerline is a structurally simpler design, but is not as prevalent on FPSs. This is because perpendicular orientation provides a level of difficulty in designing for fluid management of the lube oil system. By far the most common and recommended installation option is parallel with the vessel’s centerline. While this option presents difficulty in structural design, it is the simplest orientation when considering fluid management of the package lube oil system.

CFD tools are also used to assist the engineer in efficiently addressing fluid management within the package. Regardless of the location of the equipment, accurately monitoring the nominal oil level is critical to successful operation. With the constant motion of the lube oil within the tank, monitoring the lube oil level within the tank must be handled differently than in a non-floating application. Installing the level indicator inside a stilling tube to dampen the effect of the oil movement can assist in accomplishing this vital operational feature. Due to the constant motion offshore, if such a simple feature is not incorporated, even the most efficient placement of baffles within the oil tank can cause false low and/or high level alarms to be sounded.

As mentioned earlier, the severity of the dynamic or static inclinations experienced may exceed the baffle and gate valve capability, and/or cause the lube oil drain lines to no longer slope downwards. In these instances, a scavenge system is required. To ensure oil is drawn out of the turbine bearing cavity regardless of the inclination, the addition of a dedicated scavenge pump and drain line to return lube oil back to the tank may be the only turbine modification required. The package scavenge system ensures that all other lube oil drain lines return oil back to the oil tank efficiently. There are two scenarios that can drive the need for a package-scavenge system. These are defined not only by the magnitude of movement, but also the package length. Where the package length is such that even slight inclinations cause the lube oil drains to no longer slope downwards back to the tank, a second tank system is added to the driven base-plate. This is dedicated to ensure effective bearing drainage of the driven equipment. When the inclinations seen by the package are significant, a completely separate oil tank system located beneath the package may be required. Locating the tank system below the package provides the additional insurance that, regardless of inclination, the drains will always have a downward slope to the tank.

Other environmental considerations taken into account that are vital to the longevity of the machinery are corrosion due to chemical reactions with oxygen, water and sulfur; erosion as a result of wind, sand and salt; and electrolysis, resulting from contact of dissimilar materials. These elements are addressed with the type of attachments, consideration for galvanic bonding, material selection, and coatings.



CHAPTER 6
GAS TURBINE COMPONENTS

To understand in more detail how a gas turbine works, explanations of the working principles of a gas turbine have to start with the thermodynamic principles of the Brayton cycle, which essentially defines the requirements for the gas turbine components *(Figure 6-1)*. Since the major components of a gas turbine perform based on aerodynamic principles, we will explain these in this chapter, too.

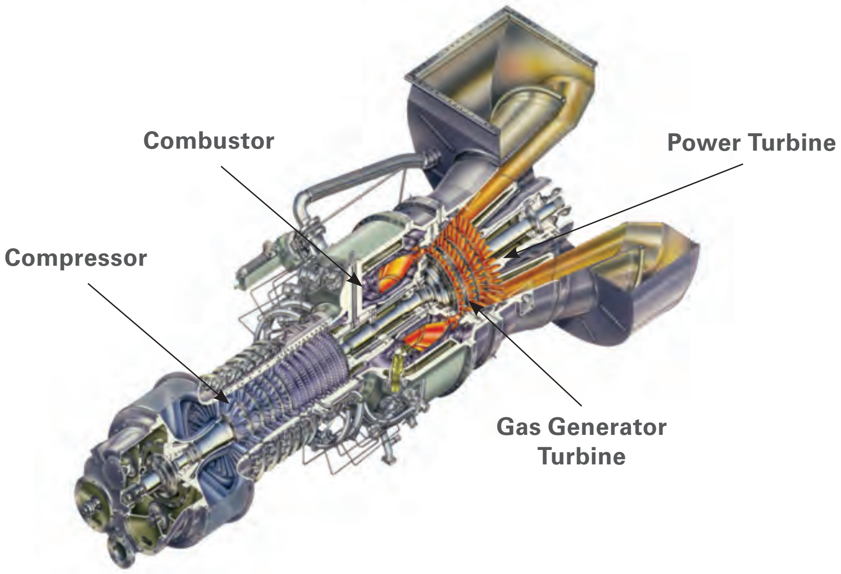


Figure 6-1. *Typical Industrial Gas Turbine*

AERODYNAMICS

Any gas turbine consists of several turbo machines. First, there is an air compressor, and after the combustion has taken place, there is a turbine chapter. Depending on the design of the gas turbine, the turbine chapter may consist either of a gas generator turbine, which operates on the same shaft as the air compressor, and a power turbine which is on a separate shaft (two-shaft design) or only of a gas generator turbine (single shaft) *(Figure 6-2)*.

The ***energy conversion*** from mechanical work into the gas (in the compressor) and from energy in the gas back to mechanical energy (in the turbine) is performed by the means of aerodynamics, by appropriately manipulating gas flows. Leonard Euler (in 1754) equated the torque produced by a turbine wheel to the change of circumferential momentum of a working fluid passing through the wheel. Somewhat earlier (in 1738), Daniel Bernoulli stated the principle that (in inviscid, subsonic flow) an increase in flow velocity is always accompanied by a reduction in static pressure and vice versa, as long as no external energy is introduced. While Euler's equation applies Newton's principles of action and reaction,

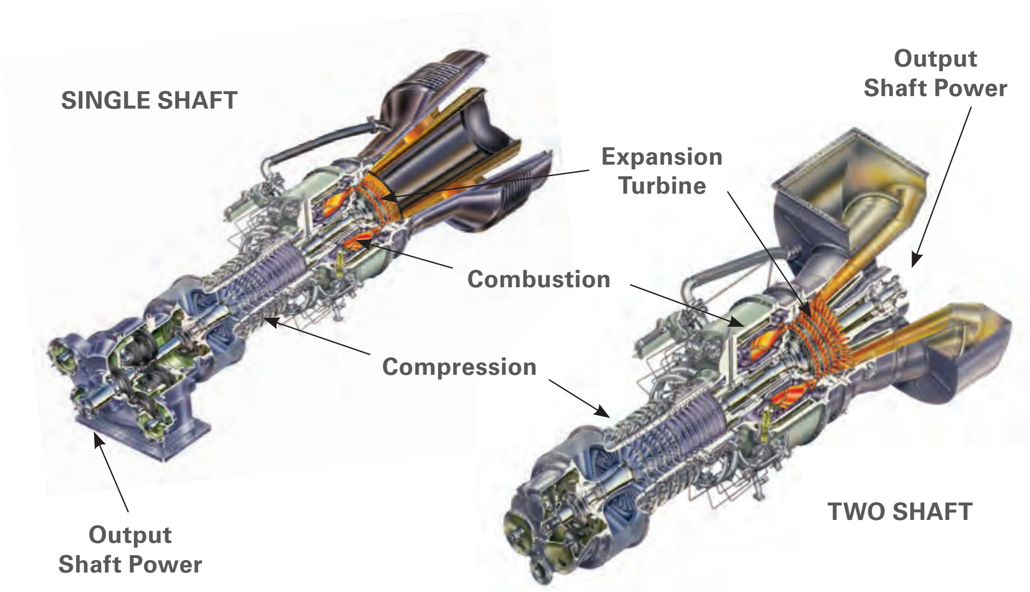


Figure 6-2. *Single and Two shaft gas turbines*

Bernoulli's law is an application of the conservation of energy. These two principles explain the energy transfer in a turbomachinery stage *(Figure 6-3)*. We had discussed some basic principles in Chapter 2.

Turbomachines in general consist of stationary and rotating airfoils. How do airfoils interact with a fluid such as air?

There are two important mechanisms:

- Changing the velocity of the fluid
- Changing the direction of the fluid

Changing the velocity of the fluid: A flowing fluid (such as air) is subject to the conservation of mass and energy.

Bernoulli's law (which is strictly true only for incompressible flows, but which can be modified for the subsonic compressible flows, describes the interchangeability of two forms of energy : static pressure and velocity.

The incompressible formulation of Bernoulli's law for a frictionless, stationary, adiabatic flow without any work input is:

$$p_t = p + \frac{\rho}{2}c^2 = const$$

For compressible flows, the equation becomes

$$h_t = h + \frac{c^2}{2} = const$$

Another requirement is, that mass cannot appear or disappear, thus for any flow from a point 1 to a point 2

$$\dot{m}_1 = \rho_1 Q_1 = \dot{m}_2 = \rho_2 Q_2$$
$$\rho \cdot Q = \rho \cdot c \cdot A$$

This requirement is valid for compressible and incompressible flows, with the caveat that for compressible flows the density is a function of pressure and temperatures, and thus ultimately a function of the velocity.

These two concepts explain the working principles of the vanes and diffusers used. Due to requirement for mass conservation, any flow channel that has a wider flow area at its inlet and a smaller flow area at its exit will require a velocity increase from inlet to exit. If no energy is introduced to the system, Bernoulli's law requires a drop in static pressure *(Figure 6-3a)*. Examples for flow channels like this are turbine blades and nozzles, inlet vanes in compressors and others *(Figure 6-3b)*. Conversely, any flow channel that has a smaller flow area A at its inlet and a larger flow area at its exit will require a velocity decrease from inlet to exit. If no energy is introduced to the system, Bernoulli's law requires a increase in static pressure *(Figure 6-3c)*. Examples for flow channels like this are vaned or vaneless diffusers, flow channels in impellers, rotor and stator blades of axial compressors volutes and others *(Figure 6-3d)*.

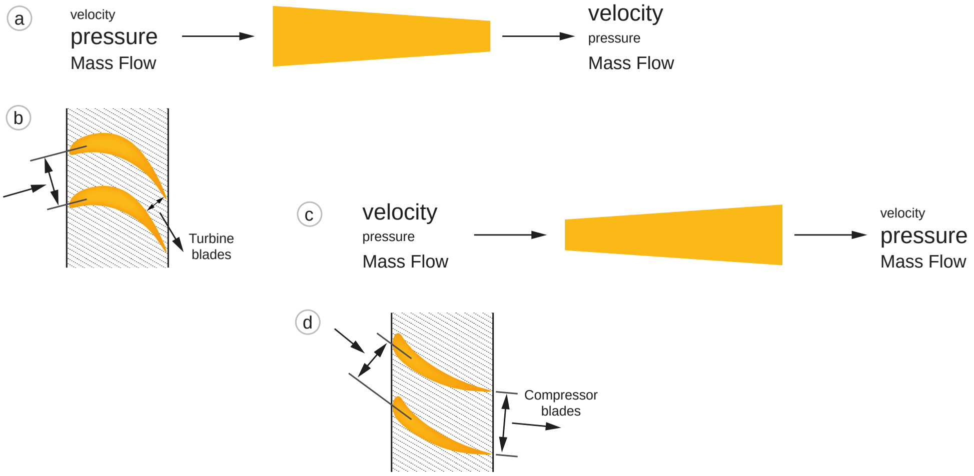


Figure 6-3a-d. *Acceleration and Diffusion*

If these flow channels are in a rotating system (for example in a turbine or compressor rotor), mechanical energy is added to or removed from the system. Nevertheless, if the velocities are considered in a rotating system of coordinates, above principles are applicable as well.

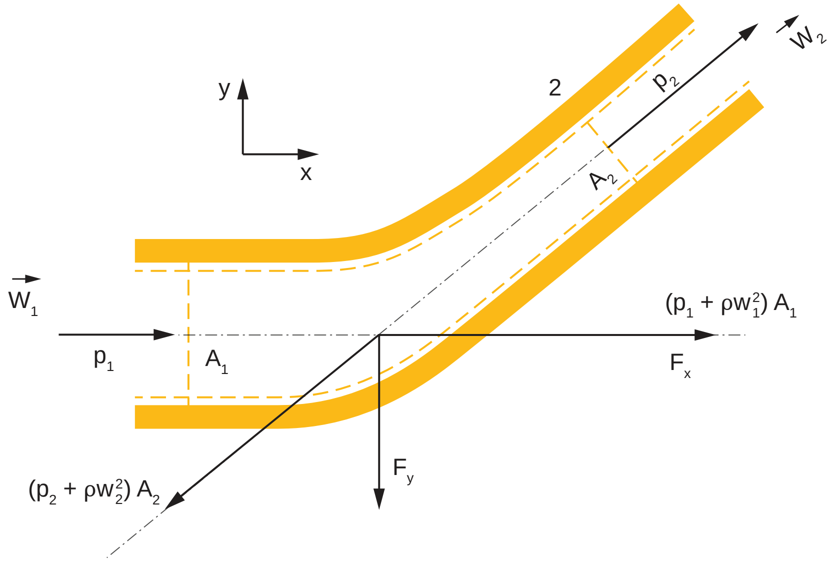


Figure 6-4. *Conservation of Momentum*

Changing the direction of the fluid

When we force a fluid to change its flow direction, we will have to act with a force upon this fluid. This is also known as the conservation of momentum. The change in momentum M of gas flowing from a point 1 to a point 2 is its mass times the change in its velocity (m c), and is also the sum of all forces F acting. The change in momentum is

$$\frac{d\vec{M}}{dt} = \dot{m}\left(\vec{c}_2 - \vec{c}_1\right) = \vec{F}$$

To change the momentum of this gas, either by changing the velocity or the direction of the gas (or both), a force is necessary. *Figure 6-4* outlines this concept for the case of a bent, conical pipe. The gas flows in through the area A_1 with w_1, p_1, and out through the flow area A_2 with w_2, p_2. The differences in the force due the pressure (p_1A_1 and p_2A_2, respectively), and the fact that a certain mass flow of gas is forced to change its direction generates a reaction force F_R. Split into x and y coordinates, and considering that

$$\dot{m} = \rho_1 A_1 w_1 = \rho_2 A_2 w_2$$

we get (due to the choice of coordinates, $w_{1y}=0$)

$$\begin{aligned} x: &\quad \rho A_1 w_1\left(w_{2x} - w_1\right) = p_1 A_1 - (p_2 A_2)_x + F_{Rx} \\ y: &\quad \rho A_1 w_1\left(w_{2y}\right) = -(p_2 A_2)_y + F_{Ry} \end{aligned}$$

It should also be noted that this formulation is also valid for viscous flows, because the friction forces become internal forces. For a rotating row of vanes in order to change the velocity of the gas, the vanes have to exert a force upon the gas. This is fundamentally the same force that F_{Ry} that acts in the previous example for the pipe. This force has to act in direction of the circumferential rotation of the vanes in order to do work on the gas. According to the conservation of momentum, the force that the blades exert is balanced by the change in circumferential velocity times the associated mass of the gas. This relationship is often referred to as Euler's Law, described in Chapter 2.

The importance of Euler's law lies in the fact that it connects aerodynamic considerations (i.e the velocities involved) with the thermodynamics of the compression process.

Mach number

To make things more complicated, the density of the gas flowing over airfoils or through channels is not constant. In other words, the gas is compressible. Form the discourse above we know, that when we conserve the mass flow, then the volumetric flow, and with it all velocities, will change, if the density changes. An indicator of the severity of the impact of these density changes is the Mach number Ma which compares flow velocities c to the speed of sound a:

$$Ma = c/a$$

If the flow velocities through the entire section is below the speed of sound, we talk about sub sonic flow, and for really small Mach numbers, we can actually consider the flow incompressible, in other words we don't need to consider the density changes. Flow velocities above the speed of sound are called supersonic. If the flow changes within a component form subsonic to supersonic or vice versa, the flow is call transonic. An example for a transonic compressor stage is shown in *Figure 6-5*, where the flow enters at speeds above the speed of sound, and is decelerated through a shock to subsonic velocities.

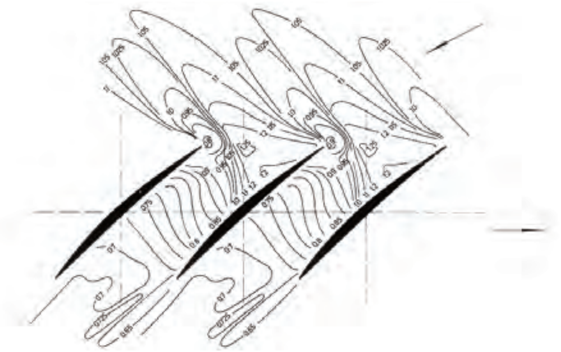


Figure 6-5. *Mach Number distribution for typical transonic compressor blades. The flow enters at supersonic speeds, and is decelerated to subsonic speeds at the exit (Schodl, 1977). The shock, where the supersonic flow is decelerated to subsonic speeds immediately upstream of the leading edge is clearly visible.*

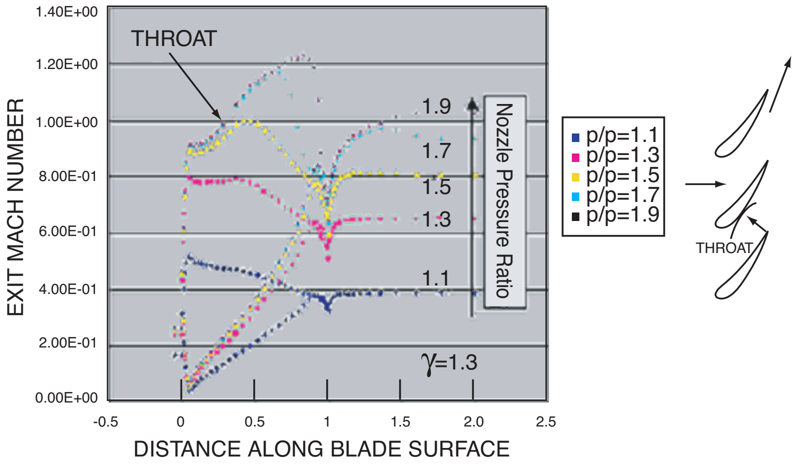


Figure 6-6. *Velocity distribution (isentropic Mach Number Mis along the surface coordinates) in a turbine nozzle at different pressure ratios*

As soon as the maximum local flow velocity exceeds Mach 1 (at a pressure ratio of 1.5 in this example), the inlet flow can no longer be increased (Kurz, 1991).

Figure 6-6 shows the situation of a turbine nozzle that is operated at different pressure ratios. We see that not only the levels of velocity increases with increased pressure ratio, but the velocity distribution actually changes its shape- an example for the profound changes of aerodynamic behavior with the change in Mach number. At a certain pressure ratio (in this case 1.5) the velocity at the suction side of the nozzle just reaches the speed of sound. For higher pressure ratios, the flow is actually accelerated past the speed of sound. A further increase of the pressure ratio yields higher velocities downstream of the throat, but the flow, which is proportional to the velocity at the inlet into the nozzle, can no longer be increased.

If we look at the volumetric flow, represented by the flow velocity entering the nozzle *(Figure 6-7)* we find that once we exceed the pressure ratio where we first reached the speed of sound, the flow cannot be increased any more: the nozzle is choked. In other words, even an increase in pressure ratio does not yield more flow.

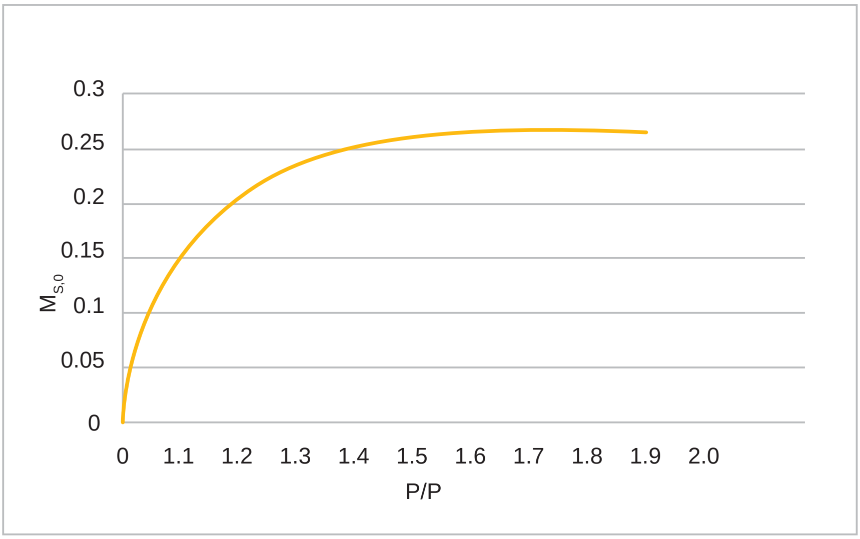


Figure 6-7. *Flow through a Nozzle as a Function of Pressure Ratio*

Any aerodynamic component performance will change if the characteristic Mach numbers are changed.

Not surprisingly, the Mach number has a strong influence on losses and enthalpy rise or decrease for a given blade row. *Figure 6-8* shows how the losses increase for an axial compressor stage, while the range is reduced with a rising Mach number.

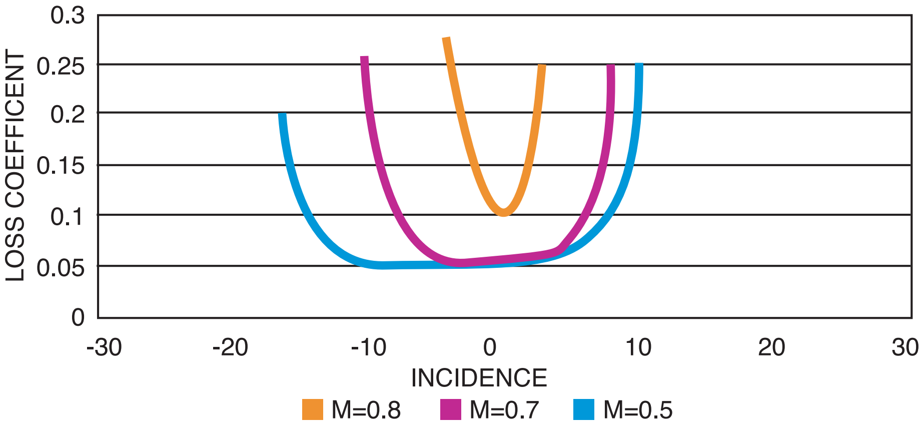


Figure 6-8. *Effect of Mach Number on Losses and Range of an Axial Compressor Airfoil*

The practical conclusion is that the performance of any aerodynamic component will change if the characteristic Mach number changes. On the other hand, if characteristic Mach numbers of a component are held constant, then the performance of said component will remain the same (except for changes in Reynolds number, gas composition etc.).

The aerodynamic behavior of a turbine or compressor is thus significantly influenced by the Mach number of the flow. The same turbine or compressor will show significant differences in operating range (flow range between stall and choke), pressure ratio and efficiency.

In a modern gas turbine, the compressor front stages are transonic, which means that the relative flow velocity into the rotor is higher than the speed of sound, while the flow velocity leaving the rotor is below the speed of sound. Turbine stages usually see subsonic inlet velocities, but the velocities within the blade channels can become locally supersonic.

Component performance maps show a significant sensitivity to changes in Mach numbers. There is a strong dependency of losses, enthalpy rise or decrease, and flow range for a given blade row on the characteristic Mach number.

Reynolds number

While the Mach number essentially accounts for the compressibility effects of the working gas, the Reynolds number describes the relative importance of friction effects. In industrial gas turbines, where neither the working temperatures, nor the working pressures change as dramatically as in the operation of aircraft engine, the effects of changes in the Reynolds number are typically not very pronounced. A change in the ambient temperature from 0° F to 100° F changes the Reynolds number of the first compressor stage by about 40%. The typical operating Reynolds numbers of compressor blades and turbine blades are above the levels where the effect of changing the Reynolds number is significant.

The Reynolds number is defined by:

$$\mathrm{Re} = \left[\frac{w \cdot L}{v} \right]$$

The kinematic viscosity (υ) depends on the gas, temperature and pressure. In industrial gas turbines, where neither the working temperatures nor the working pressures change as dramatically as in the operation of aircraft engines, the effects of changes in the Reynolds number are typically not very pronounced. A change in the ambient temperature from -18 to 38°C (0 to 100°F) changes the Reynolds number of the first compressor stage by about 40%. The Reynolds number dependency of the losses of a typical compressor or turbine airfoil are shown in *Figure 6-9*.

The typical operating Reynolds numbers of compressor and turbine blades are above the levels where the effect of changing the Reynolds number is significant.

However, in all instances the Reynolds numbers are rather large (usually in the range over several hundred thousands or millions). This means, that only areas close to surfaces such as the blades or the casing are affected by frictional effects. These areas are usually called boundary layers *(Figure 6-10)*.

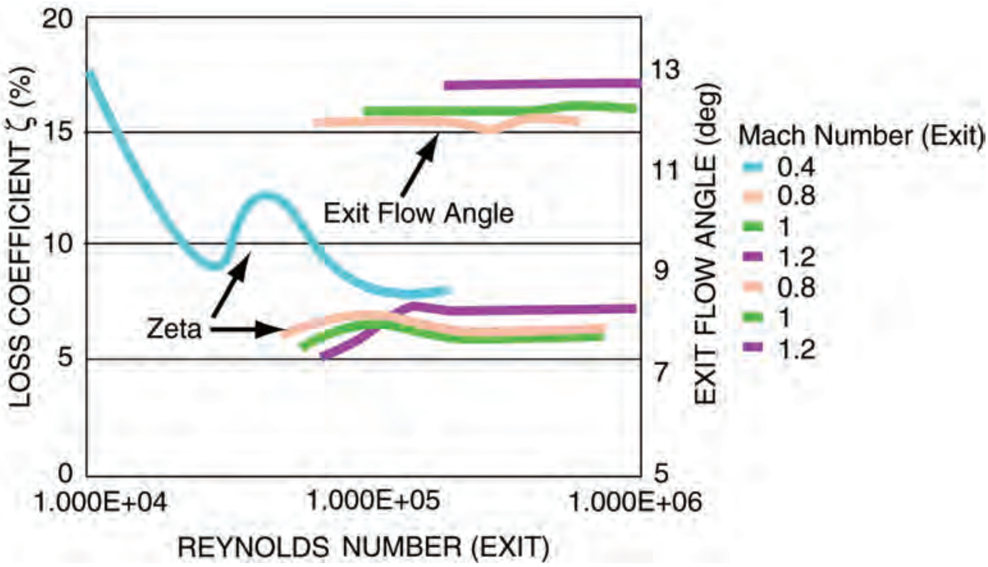


__Figure 6-9.__ Losses and Exit Flow Angle for a Turbine Nozzle as a Reynolds Number Function

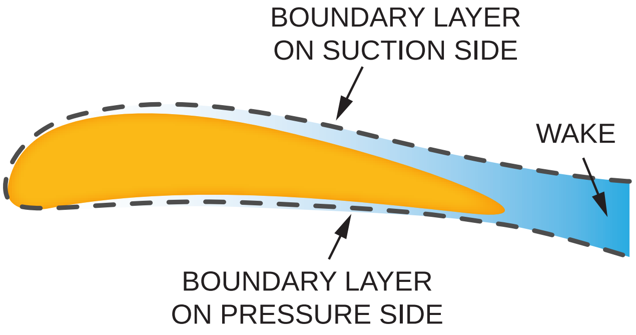

BOUNDARY LAYER ON SUCTION SIDE
WAKE
BOUNDARY LAYER ON PRESSURE SIDE

__Figure 6-10.__ Boundary Layer on an Airfoil

The practical implication is, that the state of the flow in these areas, be it laminar, turbulent, separated or transitional, has a significant impact on friction losses, mixing losses and heat transfer. For example, turbulent flow leads to higher friction losses and increased heat transfer, but it also tends to be more stable versus the risk of flow separation. Laminar flow causes lower friction, but tends to transition smoothly, or via a separation bubble, into turbulent flow. Laminar flow is also more prone to flow separation. The structure of the boundary layer does not disappear downstream of a blade. It rather, after the suction and pressure side boundary layer mix at the trailing edge of the blade forms a wake *(Figure 6-10)*. This means that the flow entering the downstream blade rows is always non-uniform.

SECONDARY FLOWS

An important problem that arises in the design and performance analysis of modern turbomachinery is the understanding and prediction of the nature and influence of secondary flows. Secondary flows can be defined as those portions of the total flow that behave significantly differently from the main flow. Thus, any flow that deviates from the average flow path is considered a secondary flow *(Figure 6-11)*.

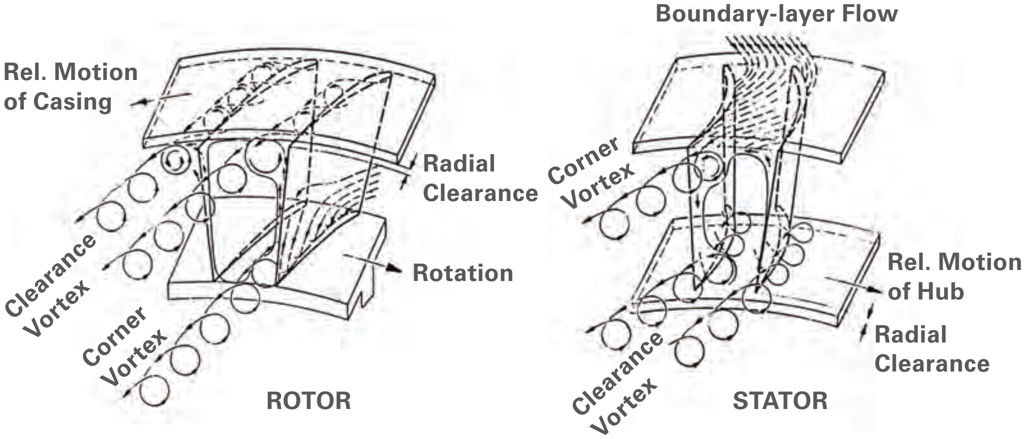

Boundary-layer Flow
Rel. Motion of Casing
Radial Clearance
Corner Vortex
Clearance Vortex
Rotation
Corner Vortex
ROTOR
Rel. Motion of Hub
Radial Clearance
Clearance Vortex
STATOR

Figure 6-11. *Secondary Flow Regions in rotor and stator of an axial flow compressor by Fottner (1989)*

Strong secondary flows occur when the gas flow is forced to change its flow direction in an airfoil. For this change in direction from incoming to exiting direction, the interaction between the boundary layers of the blade and the casing or the hub lead to systems of vortices. The flow in these vortices clearly does not follow the intended direction of the flow path and can have a detrimental effect on the performance of the airfoil.

Figure 6-12 shows streamlines of the flow close to the side wall in a cascade of turbine blades. Clearly, the flow in this region does not follow the path prescribed by the airfoil geometry.

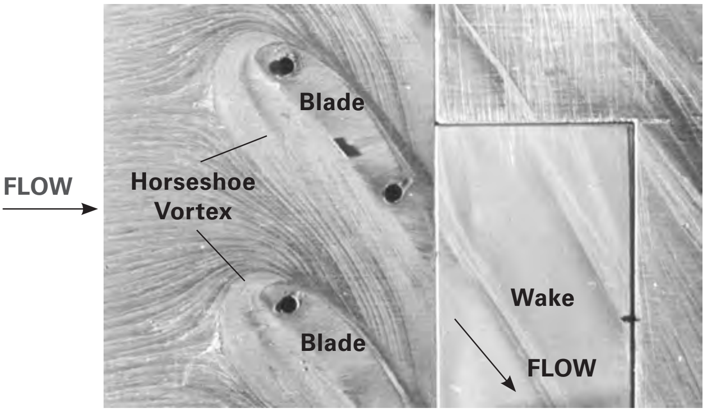

Blade
FLOW
Horseshoe Vortex
Blade
Wake
FLOW

Figure 6-12. *Flow Lines Created by Secondary Flow Vortex Structures on the side wall of a Turbine Blade Cascade*

Secondary flows in turbine blades tend to be more prominent than in compressor blades, due to the stronger turning. Various efforts have been made to reduce the impact of secondary flows. Earlier efforts included boundary layer fences and slots. More recently, blades are sculpted in 3 dimensions, thus addressing the flow structures created by secondary flows *(Figure 6-13)*.

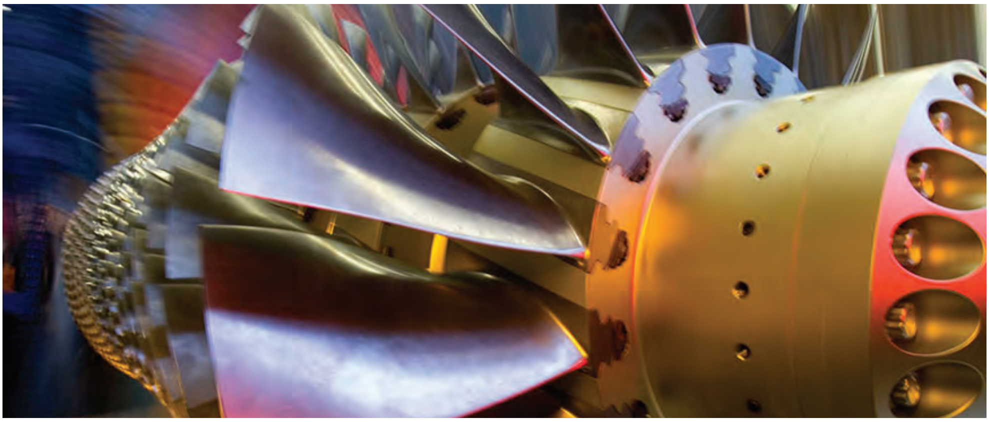


***Figure 6-13.** Sculpted blades*

COMPONENT PERFORMANCE

In following paragraphs, we will discuss the relationships of flow, pressure ratio and speed that are expressed in the performance maps for compressors and turbines. As expected, the influence of Mach numbers is very visible.

Figure 6-14 with the compressor maps for typical gas turbine compressors show in particular the narrowing of the operating range with an increase in Mach number, which in this case is due to increasing compressor speed NGP.

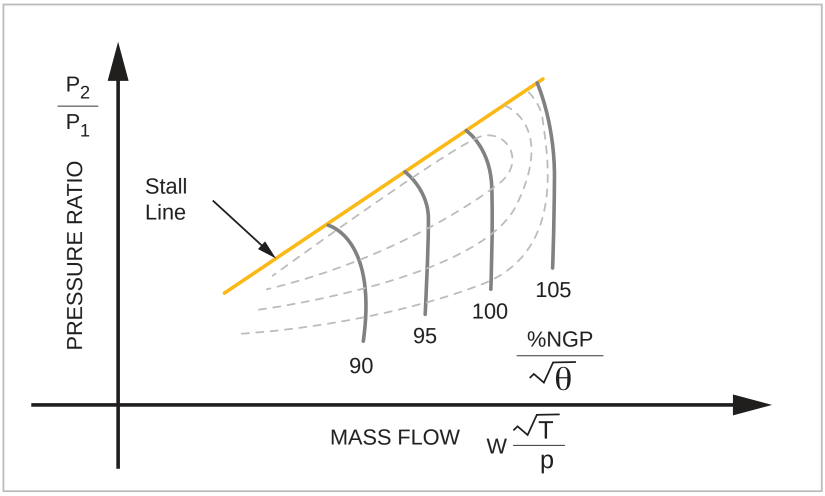


***Figure 6-14.** Typical compressor performance map*

For turbine nozzles, one of the effects connected with the Mach number is the limit to the maximum flow that can pass through a nozzle. Beyond a certain pressure ratio, the amount of actual flow Q that can pass through the nozzle can no longer be increased by increasing the pressure ratio. As demonstrated with the turbine map in *Figure 6-15*, which shows the flow Q in a turbine nozzle for increasing pressure ratios, the velocity or Mach number levels in the nozzle become higher and higher, until the speed of sound (= Mach 1) is reached in the throat, as discussed in Chapter 2. A further increase of the pressure ratio yields higher velocities downstream of the throat, but the through flow (which is proportional to the velocity at the inlet into the nozzle) can no longer be increased.

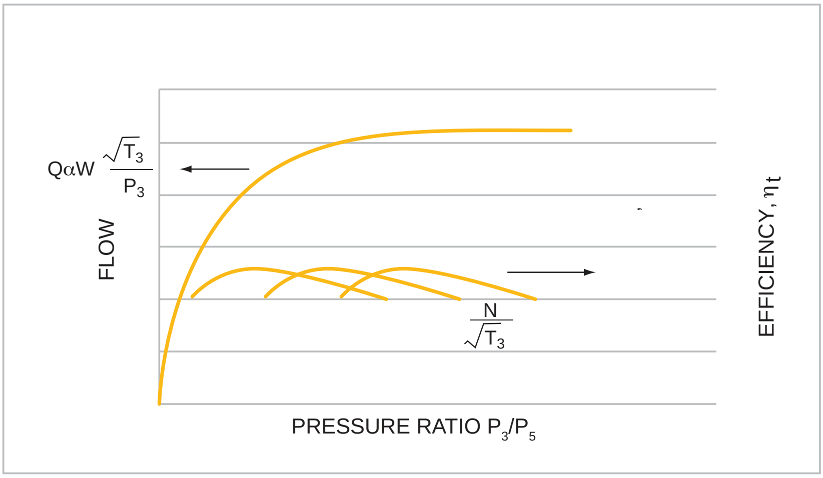


Figure 6-15. *Typical turbine map*

The Mach number increases with increasing flow velocity , and decreasing Temperature T. It also depends on the gas composition, which determines the ratio of specific heats γ and the gas constant R. To characterize the level the Mach number of a turbo machine, the 'Machine' Mach number M_n is frequently used. Mn does not refer to a gas velocity, but to the circumferential speed u of a component, for example a blade tip at the diameter D:

$$M_n = \frac{u}{\sqrt{\gamma RT}} = \frac{2\pi DN}{\sqrt{\gamma RT}}$$

This points to the fact that the Mach number of the component in question will increase once the speed N is increased. The consequences for the operation of the gas turbine are, that

- The engine compressor Mach number depends on its speed , the ambient temperature and the relative humidity.
- The gas generator turbine Mach number depends on its speed, the firing temperature, and the exhaust gas composition (thus, the load, the fuel and the relative humidity)
- The power turbine Mach number depends on its speed, the power turbine inlet temperature and the exhaust gas composition.

For a given geometry, the reference diameter will always be the same. Thus, we can define the Machine Mach number also in terms of a speed, for example the gas generator speed, and get the so called corrected gas generator speed:

$$N_{corr} = \frac{N}{\sqrt{T/T_{ref}}}$$

Even though N_{corr} is not dimensionless, it is a convenient way of writing the machine Mach number of the component. In the following text, we will also use this simplified expression N/√T, which is based on the above explanations.

Similarly, a relations ship for mass flow and volumetric flow can be defined as an 'axial Mach number':

$$Q = \frac{W}{\rho} = \frac{W\ R\ T}{p}$$

and, for some reference through flow area A

$$M_{ax} = \frac{c_z}{a} = \frac{Q}{aA} = \frac{W\ R\ T}{A\ p\sqrt{RT}} = \frac{W\sqrt{R\ T}}{A\ p} \propto \frac{W\sqrt{T}}{p}$$

Because each gas turbine consists of several aerodynamic components, the Mach number of each of these components would have to be kept constant in order to achieve a similar operating condition for the overall machine. While the characteristic temperature for the engine compressor is the ambient temperature, the characteristic temperature for the gas generator turbine and the power turbine is the firing temperature T_3 and the power turbine inlet temperature T_5 respectively. Therefore, if two operating points (op1 and op2) yield the same machine Mach numbers for the gas compressor and the gas generator turbine, and both operating points are at the respective optimum power turbine speed, then the thermal efficiencies for both operating points will be the same – as long as second order effects, such as Reynolds number variations, effects of gaps and clearances etc. are not considered.

The requirement to maintain the machine Mach number for compressor and gas generator turbine can be expressed by N_{GPcorr}= constant (which leads to identical Mach numbers for the compressor):

$$\frac{N_{GP,op1}}{\sqrt{T_{1,op1}}} = \frac{N_{GP,op2}}{\sqrt{T_{1,op2}}}$$

and, in order to maintain at the same time the same mach number for the gas generator turbine, which rotates at the same speed as the compressor, we require for the firing temperature:

$$\frac{T_{3,op1}}{T_{1,op1}} = \frac{T_{3,op2}}{T_{1,op2}}$$

In this case, the fact that the volumetric flow through the turbine section is determined by the nozzle geometry, also enforces (approximately) identical head and flow coefficients for compressor and turbine.

Therefore, the engine heat rate will remain constant, while the engine power will by changed proportional to the change in inlet density. This approach does not take effects like Reynolds number changes, changes in clearances with temperature, changes in gas characteristics or the effect of accessory loads into account. This approach also finds its limitations in mechanical and temperature limits of an actual engine, that restrict actual speeds and firing temperatures.

Compressor

In previous chapters, the concept of energy transfer in compressors and turbines was explained. Here we want to extend this concept to understand the behavior of these components when they are operated at different operating conditions. In other words, we want to understand component performance maps.

The general behavior of any gas compressor can be described by some additional, fundamental relationships: The vanes of the rotating impeller ‘see’ the gas in a coordinate system that rotates with the impeller. The transformation of velocity coordinates from an absolute frame of reference (c) to a frame of reference rotating with a velocity u is by:

$$\vec{w} = \vec{c} - \vec{u}$$

where, for any diameter D and speed N of the stage $u=\pi DN$.

The importance of Euler’s law lies in the fact that it connects aerodynamic considerations (i.e., the velocities involved) with the thermodynamics of the compression process.

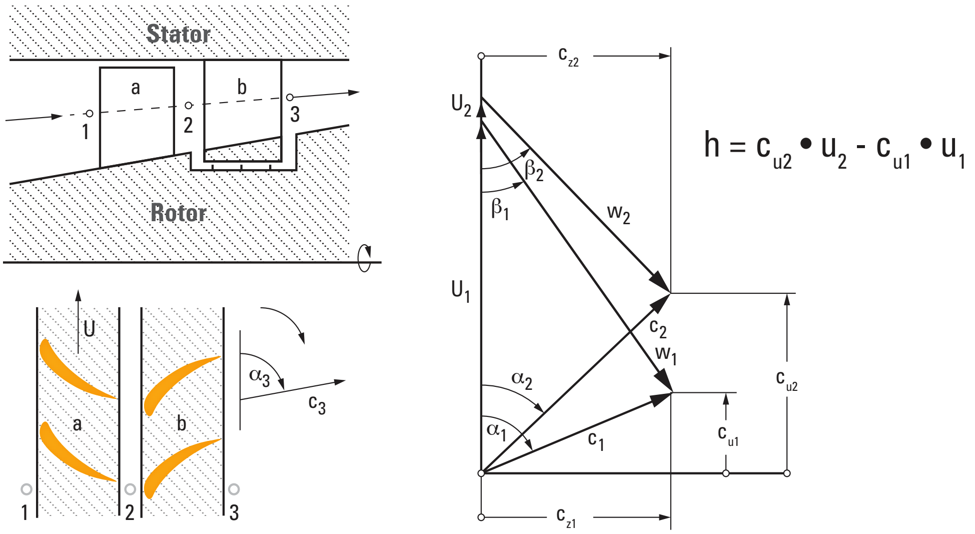


Figure 6-16. *Velocities in a Typical Compressor Stage. Mechanical Work h Transferred to the Air is Determined by the Change in Circumferential Momentum of the Air.*

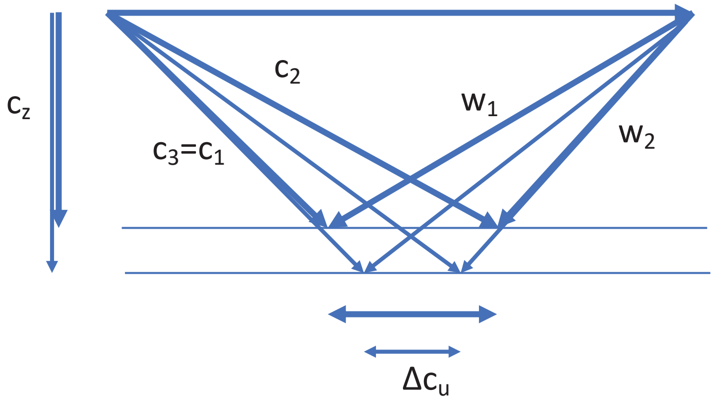

c_z
c_2
w_1
w_2
$c_3 = c_1$
Δc_u

Figure 6-17. *Velocity vectors in an axial compressor at constant speed and changing flow*

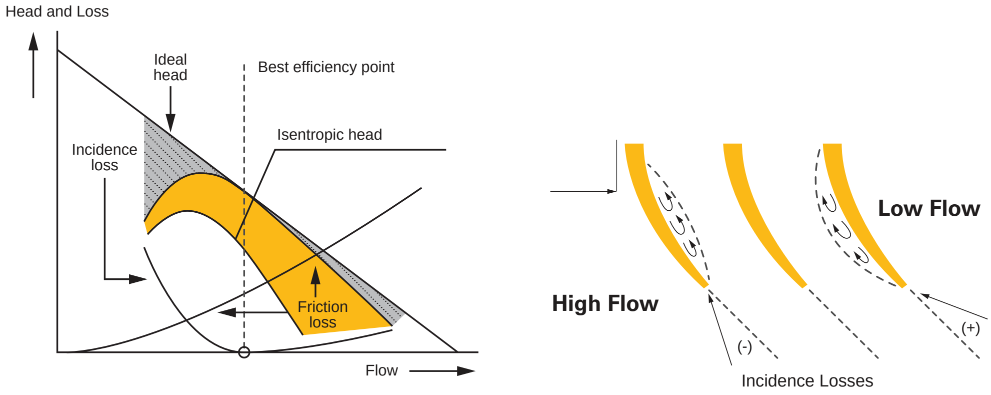

Head and Loss
Ideal head
Best efficiency point
Isentropic head
Incidence loss
Friction loss
Flow
High Flow
Low Flow
(-)
(+)
Incidence Losses

Figure 6-18. *Head (or pressure ratio) versus flow relationship at constant speed*

The rotor exit geometry determines the direction of the relative velocity w_2 at the rotor blade exit. The basic 'ideal' slope of head (or pressure ratio)vs. flow is dictated by the kinematic flow relationship of the compressor, in particular exit angle of the blades. Any increase in flow at constant speed *(Figure 6-17)* causes a reduction of the circumferential component of the absolute exit velocity (c_{u2}). It follows from Euler's equation above, that this causes a reduction in head. Adding the influence of various losses to this basic relationship shape the head-flow-efficiency characteristic of a compressor *(Figure 6-18)*.

Whenever the flow deviates from the flow the stage was designed for, the components of the stage operate at lower efficiency. This is due to incidence losses. *Figure 6-19* illustrates this, using an airfoil as an example: At the 'design flow' the air follows the contours of the airfoil. If the direction of the incoming air is changed, increasing zones occur where the airflow ceases to follow the contours of the airfoil, and create increasing losses. Furthermore, the higher the flow, the higher the velocities and, thus, the friction losses.

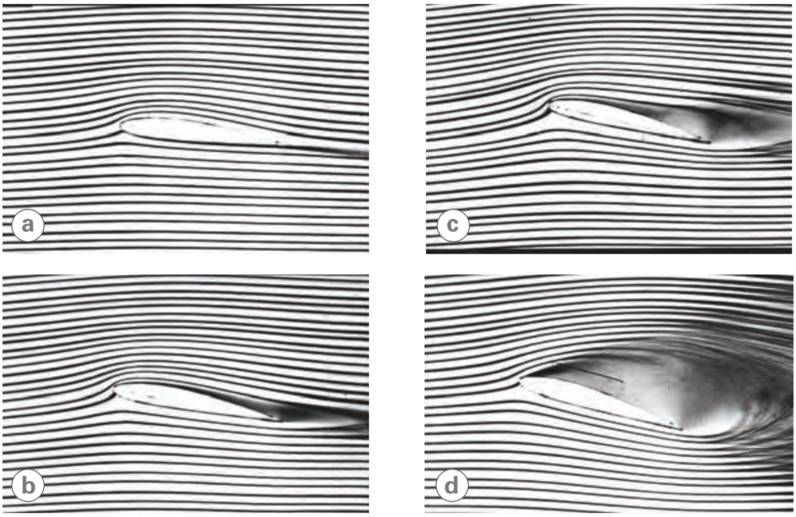


Figure 6-19. *Unseparated (a, b), partially separated (c), and fully separated (d) flow over an airfoil at increasing angle of attack (Nakayama, 1988)*

A compressor, operated at constant speed, is operated at its best efficiency point ('Best Efficiency Point' in *Figure 6-18*. If, without changing the speed, the flow through the compressor is reduced (for example, because the discharge pressure that the compressor has to overcome is increased), then the compressor efficiency decreases as well. At a certain flow, stall, probably in the form of rotating stall, in one or more of the compressor components will occur. At further flow reduction, the compressor will eventually reach its stability limit, and go into surge.

If, again starting from the best efficiency point, the flow is increased, then the efficiency is also reduced, accompanied by a reduction in head. Eventually the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke.

Stall

If the flow through a compressor at constant speed is reduced, the losses in all aerodynamic components will increase. Eventually the flow in one of the aerodynamic components will separate *(Figure 6-19 shows such a flow separation for an airfoil)*.

Stall in the rotor or stator blades is due to the fact that the direction of the incoming flow (relative to the rotating blade) changes with the flow rate through the compressor. Therefore, a reduction in flow will lead to an increased mismatch between the direction of the incoming flow the blades or vanes were designed for and the actual direction of the incoming flow. At one point this mismatch becomes so significant that the flow through the blades breaks down.

Flow separation can take on the characteristics of a rotating stall. Rotating stall occurs if the regions of flow separation are not stationary, but move in the direction of the rotor. Rotating stall can often be detected from increasing vibration signatures in the sub-synchronous region.

Choke

At high flow the head and efficiency will drop steeply, until the compressor will not produce any head at all. This operating scenario is called choke. The efficiency starts to drop off at higher flows, because a higher flow causes higher internal velocities, and thus higher friction losses. The increased mismatch between design and actual incidence further increases the losses. The head reduction is a result of both the increased losses and the basic kinematic relationships in a compressor *(Figure 6-18)*. 'Choke' and 'Stonewall' are different terms for the same phenomenon.

Surge

At flows lower than the stability limit, practical operation of the compressor is not possible. At flows on the left of the stability limit, the compressor cannot produce the same head as at the stability limit. It is therefore no longer able to overcome the pressure differential between suction and discharge side. Because the gas volumes upstream (at discharge pressure) is now at a higher pressure than the compressor can achieve, the gas will follow its natural tendency to flow from the higher to the lower pressure: The flow through the compressor is reversed. Due to the flow reversal, the system pressure at the discharge side will be reduced over time, and eventually the compressor will be able to overcome the pressure on the discharge side again. If no corrective action is taken, the compressor will again operate to the left of the stability limit and the above described cycle is repeated: The compressor is in surge. The observer will detect strong oscillations of pressure and flow in the compression system.

It must be noted that the violence, frequency and the onset of surge are a function of the interaction between the compressor and the inlet system, and discharge volumes.

If we operate a compressor at changing speeds, each constant speed curves forms part of an overall map *(Figure 6-20)*.

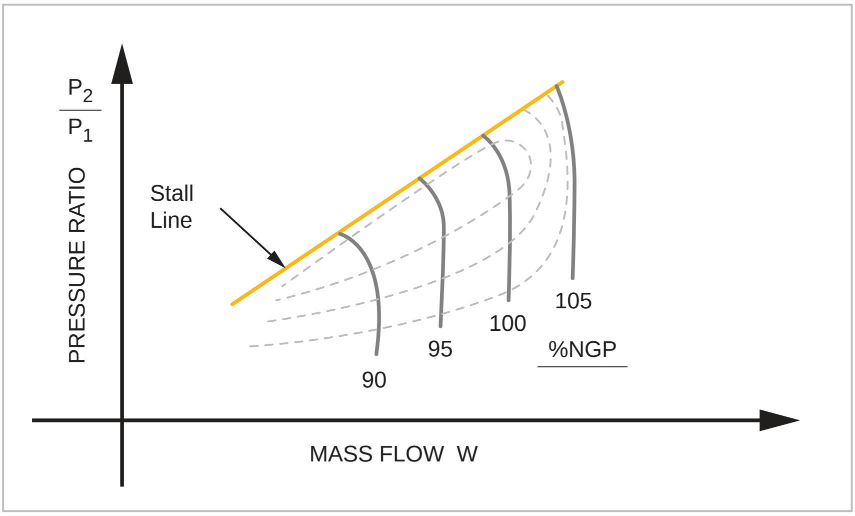


Figure 6-20. *Typical compressor performance map*

Variable Inlet and Stator Vanes

Many modern gas turbines use variable inlet guide vanes and variable stator vanes in the engine compressor. Adjustable vanes allow to alter the stage characteristics of compressor stages (see explanation on Euler Equations) because they change the head-making capability of the stage by increasing or reducing the pre-swirl contribution. This means that for a prescribed pressure ratio they also alter the flow through the compressor *(Figure 6-21)*. It is therefore possible to change the flow through the compressor without altering its speed. There are three important applications:

1. During startup of the engine it is possible to keep the compressor from operating in surge

2. The airflow can be controlled to maintain a constant fuel-to-air ratio in the combustor for dry low NO_x applications on single-shaft machines.

3. Two-shaft engines can be kept from dropping in gas generator speed at ambient temperatures higher than the match temperature, i.e. the gas generator turbine will continue to operate at its highest efficiency.

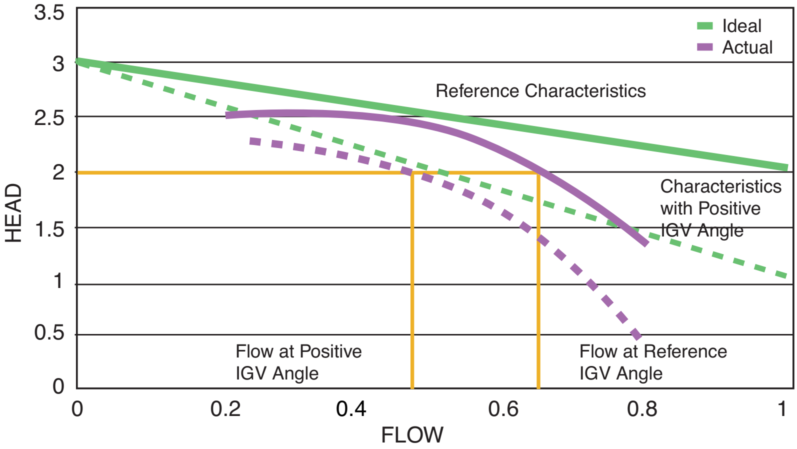


Figure 6-21. *Characteristics of an Axial Compressor Stage with Variable Guide Vanes*

Turbine

We have described earlier *(Figure 6-9)* the characteristic a turbine exhibits regarding the flow through of said turbine versus the pressure ratio. A choked turbine nozzle therefore limits the flow a turbine can accommodate.

To understand the effects of changing the power turbine speed, we have to consider the velocity polygons. *Figure 6-22* shows velocity polygons for a reaction turbine. Leaving the flow area and flow constant, since we do not change the flow condition from the gas producer, the axial portion of the velocity remains unchanged. Also, the exit flow angles of both rotor (in the absolute frame) and stator (in the rotating frame) will not change, since they are set by the geometry of the blades. The tip speed will change from u to u* due to the change in speed.

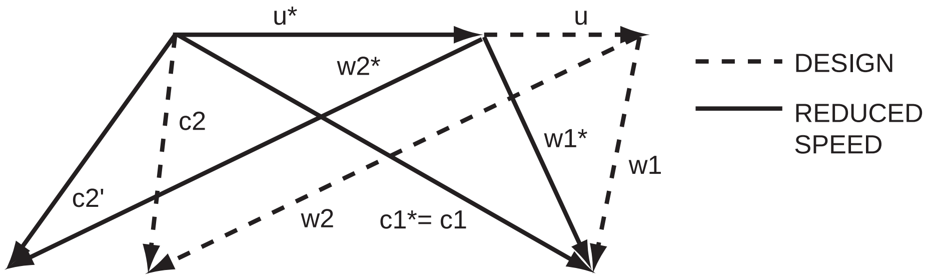


Figure 6-22. *Velocity Polygons for a Turbine Stage*

We can see that for the design point:

$$c_{u2} \cong 0$$
$$c_{u1} \cong u$$
$$h \cong u^2$$

The change in speed from N to N* yields:

$$c_{u2} \cong u^* - u$$
$$c_{u1} \cong u$$
$$h \cong u^*(c_{u1}{}^* - c_{u2}{}^*) \cong u^*(2u - u^*)$$

And with P= W • h, we get:

$$\frac{P^*}{P} = \frac{h^*}{h} = 2\frac{N^*}{N} - [\frac{N^*}{N}]^2$$

This correlation yields the speed/power relationship for the turbine stage and is plotted in *Figure 6-23*. The drop in power for off-optimum speeds is, in reality, somewhat larger since we neglected the reduction of the turbine efficiency due to higher incidence angles and higher swirl in the exit flow. The pressure ratio over the turbine, therefore, remains almost constant for a wide flow range.

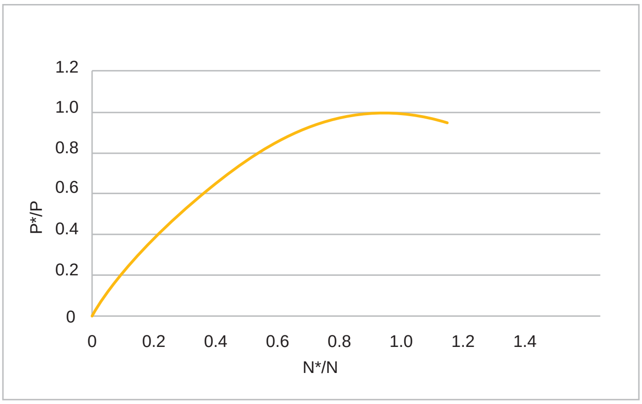


Figure 6-23. *Resulting Speed/Power Characteristics for a Turbine Stage (assuming a constant actual inflow)*

ZERO STAGING AND SCALING

The use of proven components in the development or uprating existing engines has many advantages regarding development risk and performance prediction. Two design strategies foster this goal: zero staging and scaling.

Zero staging is used extensively for engine compressors *(Figure 6-24)*. By adding a front stage to an existing compressor, the existing compressor will see the same flow as before, but at a higher pressure level *(Figure 6-25)*. Therefore, its performance remains unchanged. However, the addition of the front or zero-stage increases both the pressure ratio and mass flow of the compressor, thus increasing engine output.

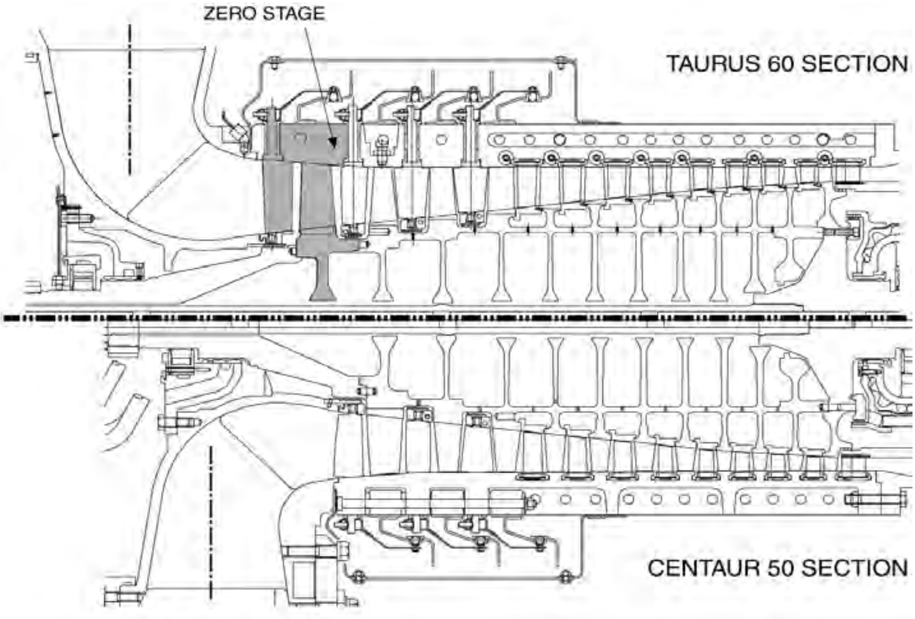


Figure 6-24. *Zero-Staged Engine Compressor*

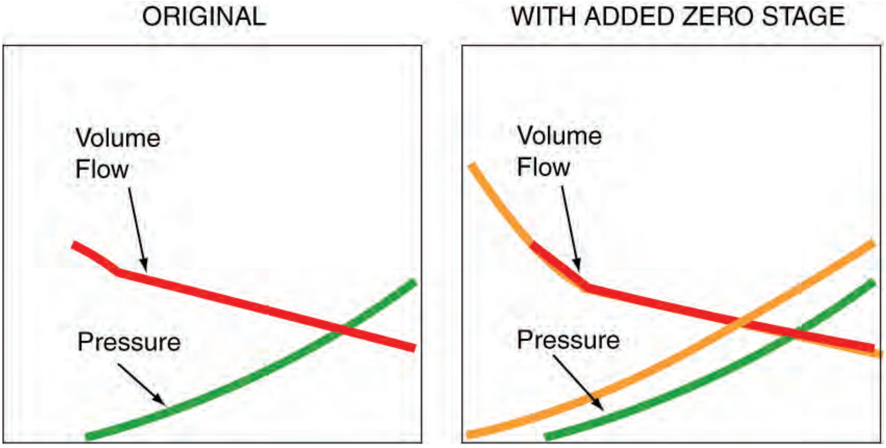


Figure 6-25. *Zero Staging of Existing Compressor to Improve Flow Capacity and Pressure Ratio*

The idea behind scaling is that if the engine or engine component is scaled linearly while reducing the speed by the same amount, and the cycle parameters (such as pressures and temperatures) are left unchanged, then the aerodynamic characteristics, the stress and temperature load levels and, thus, the component life remain the same. However, the power output will increase by the square of the scale factor. A larger engine can therefore be developed very rapidly and with little risk, since it is based on proven technology *(Figure 6-26)*.

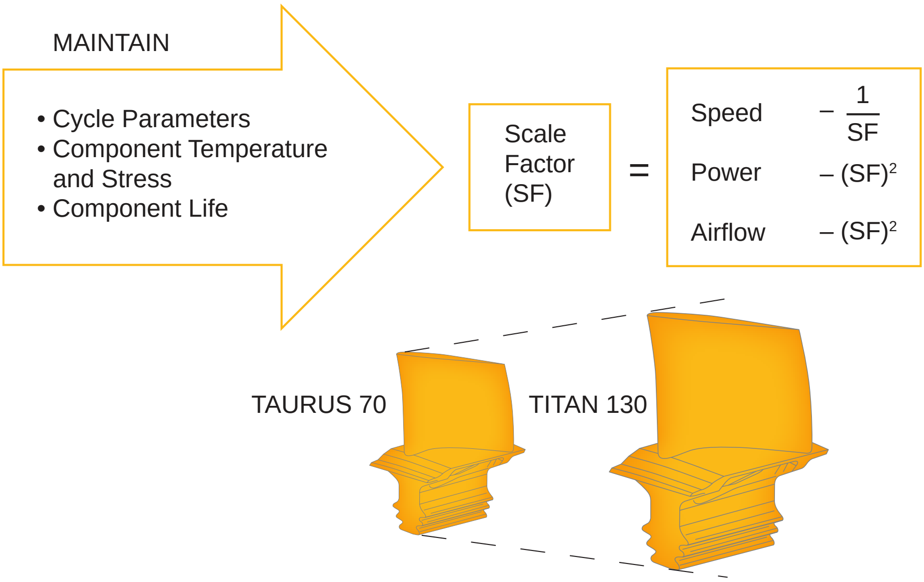


Figure 6-26. *Gas Turbine Component Scaling*

Combustion

The gas turbine combustor is the place where fuel is injected (through fuel injectors) into the air previously compressed in the engine compressor. The released fuel energy E_f causes the temperature to rise from T_2 to T_3:

$$\tilde{c}_{pe}(T_3 - T_2) = E_f W_f / W$$

The heat capacity $\tilde{c}_{pe}$ in the equation above is a suitable average heat capacity. Modern combustors convert the energy stored in the fuel almost completely into heat. (Typical combustion efficiencies for natural gas burning engines range above 99.9%). This is also evident from the fact that the results of incomplete combustion, namely CO and unburned hydrocarbons, are emitted only in the parts per million level. Only some part of the compressed air participates directly in the combustion, while the remaining air is later mixed into the gas stream for cooling purposes. The temperature profile in a typical combustor is shown in *Figure 6-27*. The local temperatures are highest in the flame zone, also known as primary zone. Cooling of the combustor liner, and the addition of air into the combustor bring the gas temperature down to an acceptable combustor exit temperature.

The flow will also incur a pressure loss due to friction and mixing. Today's industrial gas turbines use either diffusion flame or lean premix combustions systems *(Figure 6-28)*. In a diffusion flame combustor, the fuel is injected into the combustor via the injector. It mixes with combustion air in the combustor, and the flame burns where it finds stoichiometric conditions. Stoichiometric conditions mean, that for each carbon and each hydrogen atom in the fuel, there is just enough oxygen available to form CO_2 and water. In a lean premix combustor, the fuel is mixed with air in the injector barrel prior to entering the combustor. This allows creation of a lean mixture, at a prescribed fuel-to-air ratio, where there is a surplus of oxygen for the hydrogen and carbon atoms in the fuel.

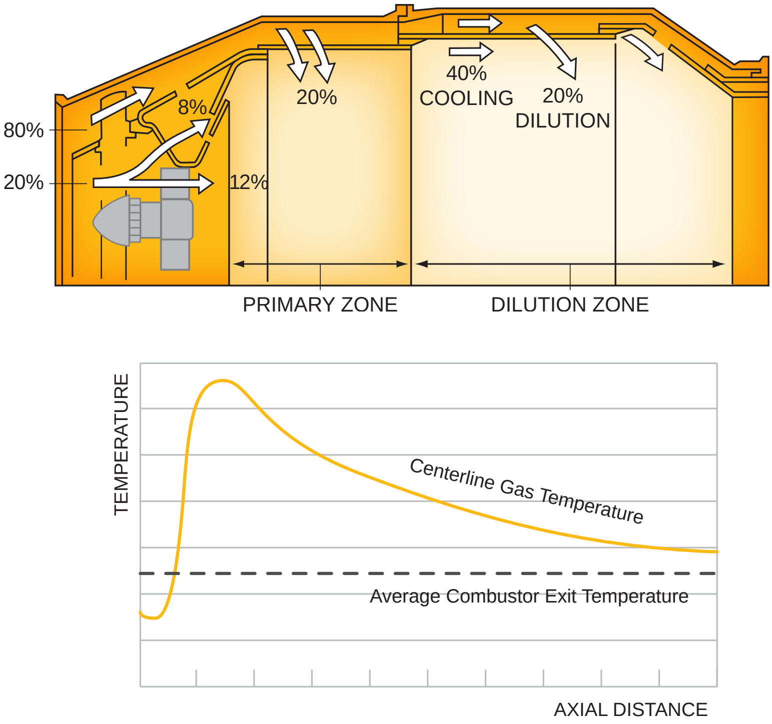

80%
20%
8%
12%
20%
40%
COOLING
20%
DILUTION
PRIMARY ZONE
DILUTION ZONE
TEMPERATURE
Centerline Gas Temperature
Average Combustor Exit Temperature
AXIAL DISTANCE

***Figure 6-27.** Axial temperature distribution in a combustor*

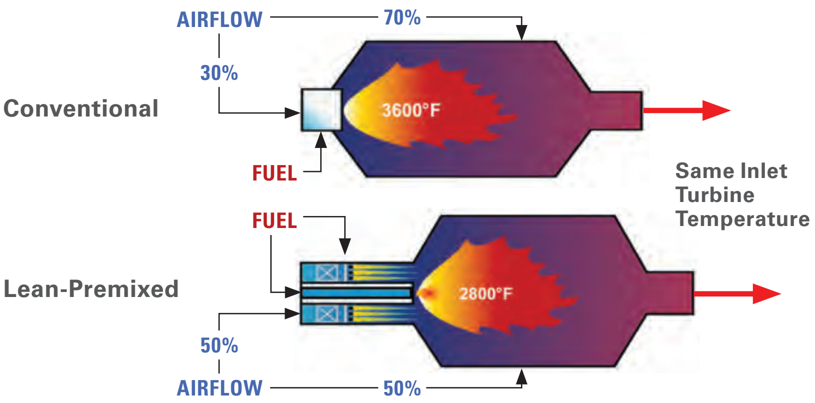

AIRFLOW
70%
30%
Conventional
3600°F
FUEL
Same Inlet
Turbine
Temperature
FUEL
Lean-Premixed
2800°F
50%
AIRFLOW
50%

***Figure 6-28.** Conventional and Lean-Premix Combustion Systems*

Blade Cooling

As pressure ratios and firing temperatures are increased to improve simple-cycle efficiencies, the role of turbine section cooling technology also becomes a critical issue. TRIT above 1100°C (2000°F) are now common in industrial gas turbines, while acceptable metal temperatures are still limited at about 900 to 1000°C (1600 to 1800°F). The blade temperature impacts mechanical strength (creep, fatigue) as well as oxidation and corrosion resistance, and thus life. As firing temperatures increase, more of the stationary nozzles and rotating blades must be actively cooled; and as pressure ratios increase, the temperature of the cooling air, typically bled from the compressor, also increases. Increases in cooling airflow cause losses in cycle thermal efficiency. For a modern industrial turbine, the blade-cooling airflow is about 10 to 15% of the total compressor flow. A major design challenge in achieving high turbine efficiency is to minimize turbine-cooling airflow with the best utilization of its cooling potential.

The cooling air is taken from the engine compressor, and is guided to a variety of cooling locations in the turbine section as shown in *Figure 6-29*.

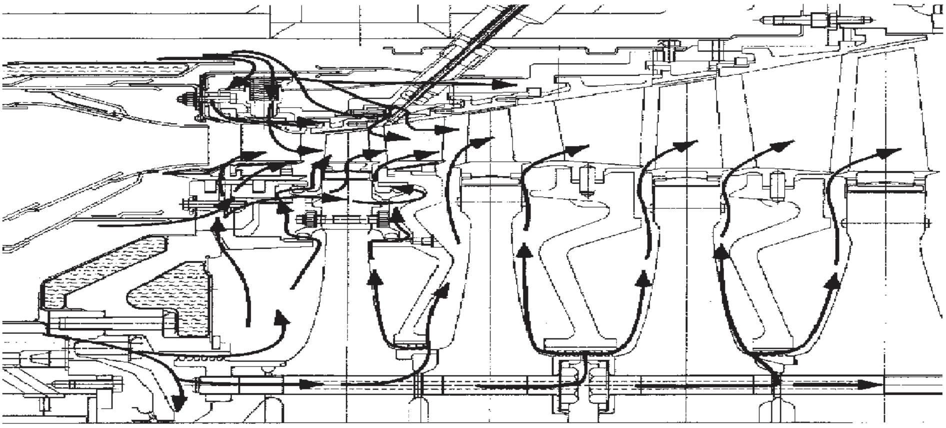


Figure 6-29. *Turbine Air Cooling Circuits*

In some designs, steam is used rather than air. There are a number of different ways the cooling is accomplished *(Figure 6-30 and 6-31)*: The air is pushed through the inside of the blade with the goal to remove as much heat from the blade surface as possible. To this end, ribs are used to increase the turbulence, and thus the heat transfer (convection cooling), and jets of air are blown though small holes to impinge on the blade inside (impingement cooling). Another

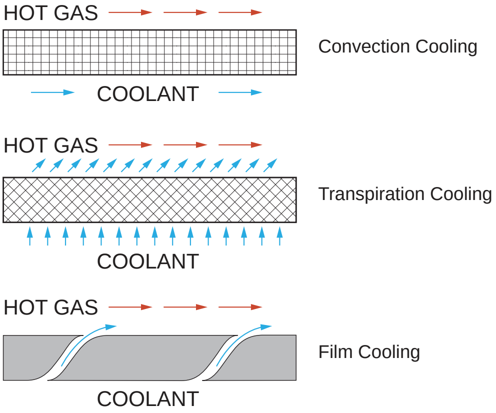


Figure 6-30. *Overview of Various Cooling Techniques*

design brings cold air from the inside of the blade through small holes to the outer blade surface, this generating a thin layer of cooler air between the blade surface and the hot gas (film cooling). The amount of air used impacts the performance of the gas turbine, because some (but not all) of the work to compress the air is lost if air is used for cooling purposes.

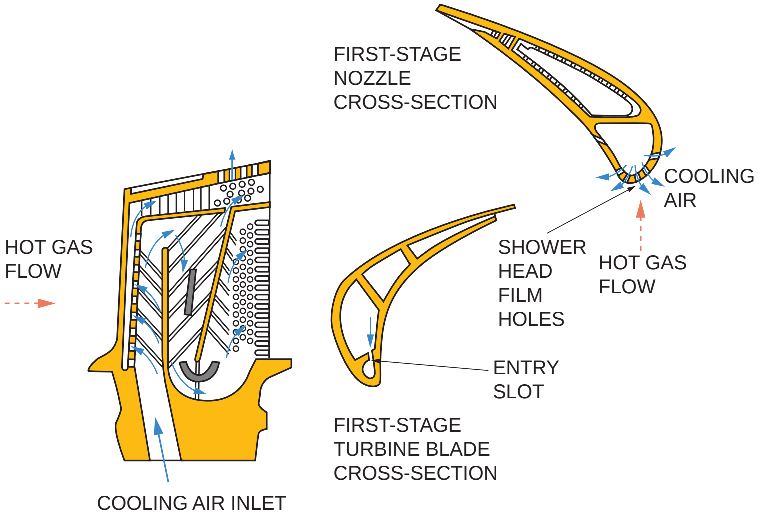


Figure 6-31. *Practical Examples of Blade Cooling Concepts*

Blade and nozzle cooling techniques have advanced considerably. Common features of gas turbine blade cooling systems include:

- Cooled Stage 1 and Stage 2 nozzle vanes (additional vanes may be cooled depending on TRIT)
- Cooled Stage 1 blades (additional blades may be cooled depending on TRIT)
- Blade cooling air delivery system based on a pre-swirler that accelerates air in the tangential direction in order to reduce air temperature when it enters the cooled blades
- Cooled nozzle and tip shroud support structures (nozzle case, diaphragms)
- Cooled turbine rotor assemblies

In addition to component temperature reduction, another important role for the cooling system is to control the relative position between rotor and stator, maintaining turbine blade tip clearances. Tip clearance and corresponding gas turbine performance degradation during long-term operation are concerns among all gas turbine users and manufacturers. Design concepts to address this challenge include the highly effective cooling of the stationary components supporting the nozzles and tip shrouds. Evidence from direct tip clearance measurements performed during instrumented turbine tests at various transient and steady-state operating conditions as well as extensive field experience proves a remarkable stability in blade tip clearances and associated turbine performance.

First-stage nozzle vanes and blades operate at the highest cycle gas temperatures and experience large transient thermal gradients. Accordingly, cooling them presents the most difficult task in turbine cooling system design. Typical thermal loads for the blades, such as the thermal boundary conditions on external surfaces, depends on the local heat transfer coefficients and the gas temperature, which in turn depends on TRIT. The heat transfer from the hot gas to the blade surface depends on the velocity distribution around the blade and, in particular, on the state of the blade boundary layers. How much heat is transferred from the hot exhaust gas to the blade surface depends on the state of the boundary layer, and whether it is laminar, turbulent, transitional or separated.

Internal airfoil cooling is often employed to counterbalance thermal loads to keep the metal temperatures at an acceptable level. *Figures 6-30 and 6-31* show various cooling techniques for advanced airfoils, including:

- Convective cooling with internal heat transfer augmenters (turbulence promoting devices such as pin fins and trip strips), where cooling air is circulated inside the blade.
- Impingement cooling, where cooling jets impinge on surfaces inside the blade, which is highly effective for the leading edge.
- Film cooling through discrete holes for the entire airfoil and particularly for the leading edge vicinity, which is also known as "shower head" cooling.
- Transpiration cooling is another form of film cooling. Instead of discrete holes, the cooling air is transpired through highly porous blade surfaces.
- Trailing edge slots. Cooling air is brought in at the root or tip of the airfoil and then discharged at the other end of the airfoil or through the trailing edge or film ejecting holes.

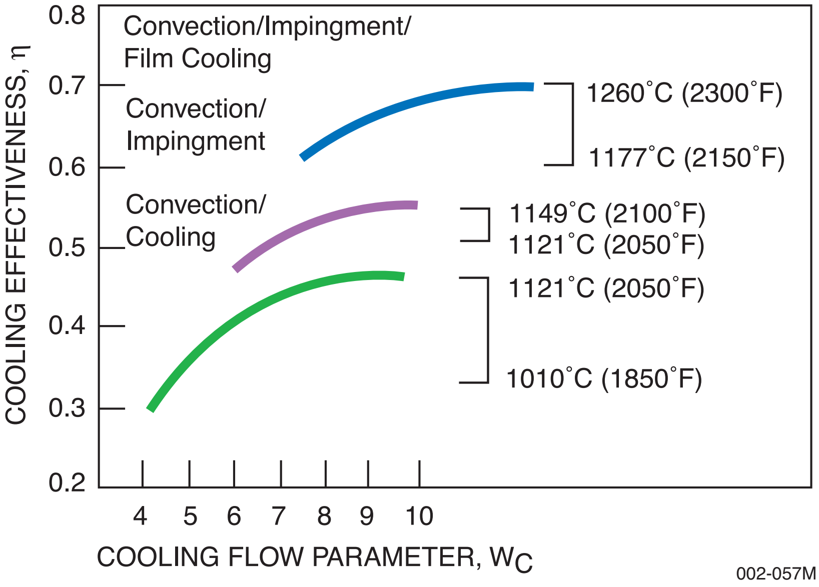


Figure 6-32. *Blade Cooling Effectiveness*

The three latter methods will alter the flow around the airfoil, thus potentially causing additional losses. Cooling effectiveness is defined by:

$$\eta = \frac{T_{gas} - T_{l.e.,met}}{T_{gas} - T_{cool}}$$

where:

T_{gas} = Mainstream gas temperature

$T_{l.e.,met}$= Leading edge metal temperature

T_{cool} = Cooling air temperature

and, from the amount of cooling air necessary:

$$\delta = \frac{W_{cool}}{W}$$

Different types of cooling can be used to increase the cooling effectiveness, thus either allowing higher gas temperatures for a given amount of cooling air, or reduced amount of cooling air for a design gas temperature *(Figure 6-32)*.

COMBUSTOR LINER COOLING

Just as the turbine blades and nozzles, the combustor liner has to survive at temperatures that are higher than the acceptable material temperature. It is also subject to intense thermal cycles as the temperatures change when the gas turbine is started, changes load or is shut down.

A variety of technologies are used to protect the combustor liner from excessive temperatures caused by the high convective and radiative heat loads in the combustor. Cooling air is either introduced to serve as a blanket between the combustion gas and the metal surface (louver, film, transpiration and effusion cooling) or to remove heat from the liner quickly by cooling its backside (impingement and augmented-backside cooling, often in connection with thermal barrier coatings on the liner). As future combustor exit temperature requirements increase, the percentage of available air for cooling decreases. This mandates an increase in cooling effectiveness.

Earlier combustor liners *(Figure 6-33)* used louvers to bring cooling air jets between the combustor liner and the combustion gases. The method leads to a very non-uniform temperature distribution on the line, which causes a number of issues: It requires relatively large amounts of cooling air to keep all metal surfaces at an acceptable temperature, and the colder areas tend to quench local combustion reactions, thus potentially increasing CO and unburned hydrocarbon emissions. The development of effusion-air cooling has enhanced combustor liner life while reducing cooling air requirements and, therefore, the formation of CO in low NO_x combustion systems. In effusion cooling, the cooling air is introduced through small holes in the combustor wall and emerges as closely spaced small jets on the hot gas side. These jets coalesce to form a protective layer of coolant over the entire surface, thereby reducing convective heat transfer to the combustor wall *(Figure 6-34)*.

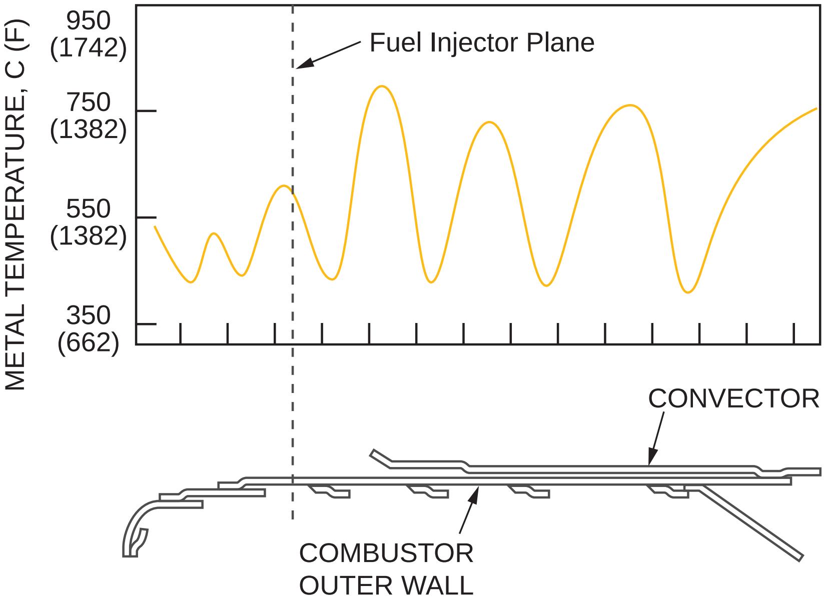


Figure 6-33. *Non-Uniform Liner Temperatures with Louver Cooling*

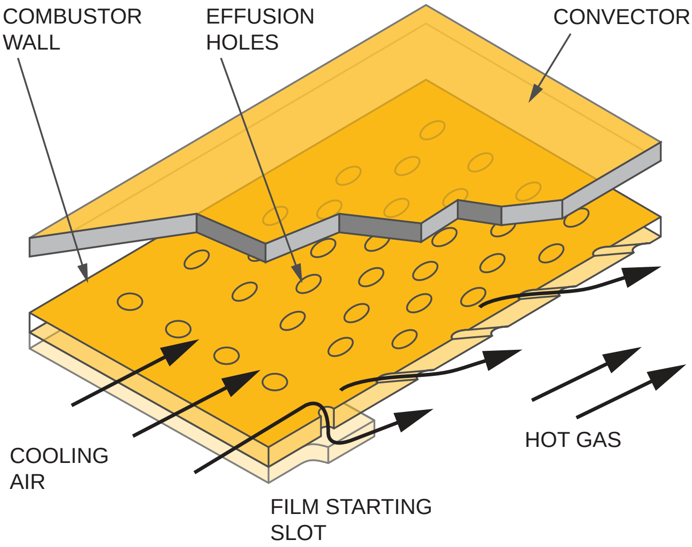

COMBUSTOR
WALL
EFFUSION
HOLES
CONVECTOR
COOLING
AIR
HOT GAS
FILM STARTING
SLOT

Figure 6-34. *Effusion-Cooled Liner Concept*

Augmented backside-cooled liners use convective cooling of the liner's outer surface with internal heat transfer augmenters (turbulence promoting devices such as pin fins and trip strips) to avoid cold spots on the liner wall, thus reducing CO and UHC production *(Figure 6-35)*. However, the typically higher wall temperatures require thermal-barrier coating.

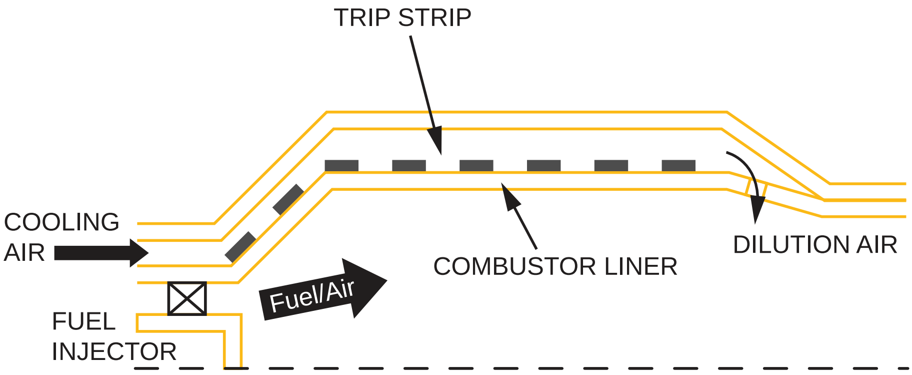
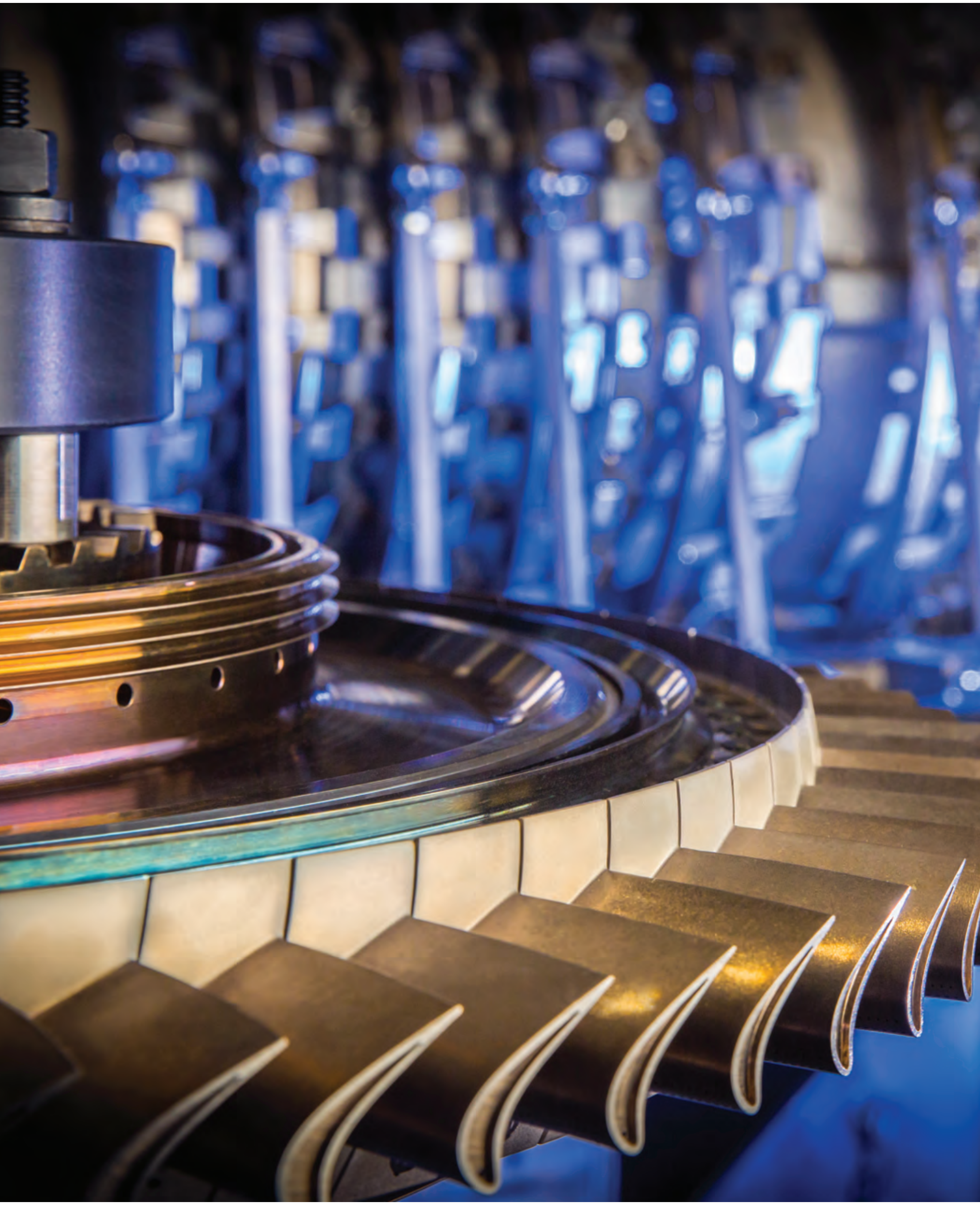

TRIP STRIP
COOLING AIR
COMBUSTOR LINER
DILUTION AIR
Fuel/Air
FUEL INJECTOR

Figure 6-35. Augmented Backside-Cooled Liner



CHAPTER 7

THE GAS TURBINE AS A SYSTEM

The *major components* of a gas turbine include the compressor, the combustor, and the turbine.

The compressor (usually an axial flow compressor, but some smaller gas turbines also use centrifugal compressors) compresses the air to several times atmospheric pressure. In the combustor, fuel is injected into the pressurized air from the compressor and burned, thus increasing the temperature. In the turbine section, energy is extracted from the hot pressurized gas, thus reducing pressure and temperature. A significant part of the turbine's energy (from 50 - 60 percent) is used to power the compressor, and the remaining power can be used to drive generators or mechanical equipment (gas compressors and pumps). Industrial gas turbines are built with a number of different arrangements for the major components:

- Single-shaft gas turbines have all compressor and turbine stages running on the same shaft *(Figure 7-1)*
- Two-shaft gas turbines consist of two sections: the gas producer (or gas generator) with the gas turbine compressor, the combustor, and the high pressure portion of the turbine on one shaft and a power turbine on a second shaft *(Figure 7-1)*. In this configuration, the high pressure or gas producer turbine only drives the compressor, while the low pressure or power turbine, working on a separate shaft at speeds independent of the gas producer, can drive mechanical equipment.
- Multiple spool engines: Industrial gas turbines derived from aircraft engines sometimes have two compressor sections (the HP and the LP compressor), each driven by a

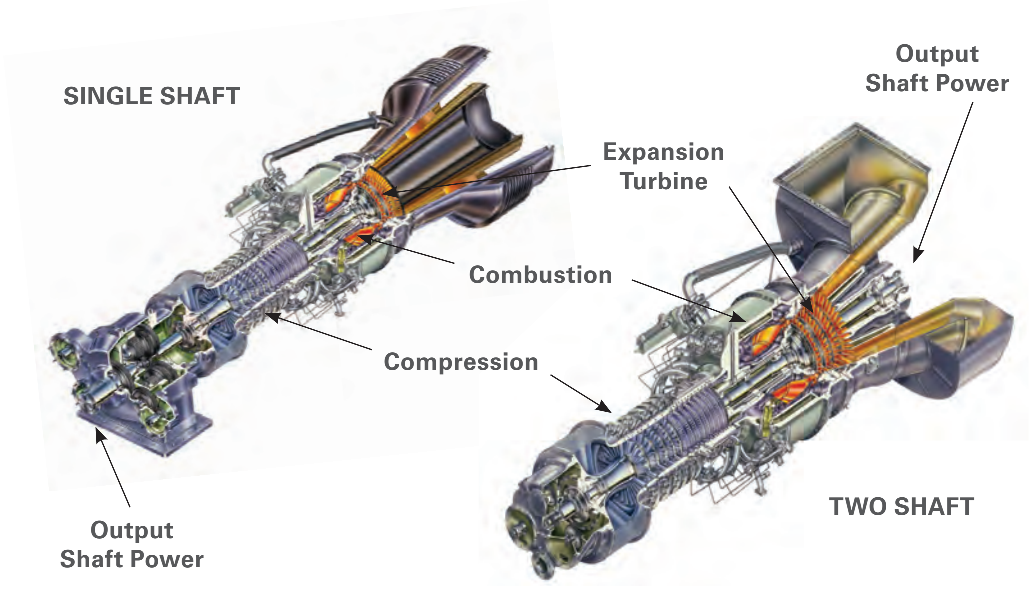


Figure 7-1. *Single-and Two-shaft gas turbines*

separate turbine section (the LP compressor is driven by an LP turbine by a shaft that rotates concentric within the shaft that is used for the HP turbine to drive the HP compressor), and running at different speeds. The energy left in the gas after this process is used to drive a power turbine (on a third, separate shaft), or the LP shaft is used as output shaft. In general, all the operating characteristics described below also apply to this type of engines.

The Brayton or gas turbine cycle *(Figure 7-2)* involves compression of air (or another working gas), the subsequent heating of this gas (either by injecting and burning a fuel or by indirectly heating the gas) without a change in pressure, followed by the expansion of the hot, pressurized gas. This was described in Chapter 1. The compression process consumes power, while the expansion process extracts power from the gas. Some of the power from the expansion process can be used to drive the compression process. If the compression and expansion process are performed efficiently enough, the process will produce useable power output. The process is thus substantially different from a steam turbine (Rankine) cycle that does not require the compression process, but derives the pressure increase from external heating. The process is similar to processes used in Diesel or Otto reciprocating engines that also involve compression, combustion, and expansion. However, in a reciprocating engine, compression, combustion, and expansion occur at the same place (the cylinder), but sequentially, in a gas turbine, they occur in dedicated components, but all at the same time.

The compressed air from the compressor enters the gas turbine combustor. Here, the fuel (natural gas, natural gas mixtures, hydrogen mixtures, diesel, kerosene and many others) is injected into the pressurized air and burns in a continuous flame. The flame temperature is usually so high that any direct contact between the combustor material and the flame has to be avoided, and the combustor has to be cooled, using air from the engine compressor. Additional air from the engine compressor is mixed into the combustion products for further cooling. An important topic is the combustion process and emissions control: Unlike in reciprocating engines, the gas turbine combustion is continuous. This has the advantage that the combustion process can be made very efficient, with very low levels of products of incomplete combustion like carbon monoxide (CO) or unburned hydrocarbons (UHC). Lowering the amount of NO_x, a reaction product occurring at the high temperatures in flame can also successfully accomplished.

The conversion of heat released by burning fuel into mechanical energy in a gas turbine is achieved by first compressing air in an air compressor, then injecting and burning fuel at (ideally) constant pressure, and then expanding the hot gas in turbine *(Figure 7-2, Brayton Cycle)*. The turbine provides the necessary power to operate the compressor. Whatever power is left is used as the mechanical output of the engine. This thermodynamic cycle can be displayed in an enthalpy-entropy (h-s) diagram *(Figure 7-2)*. The air is compressed in the engine compressor from state 1 to state 2. The heat added in the combustor brings the cycle from 2 to 3. The hot gas is then expanded. In a single-shaft turbine, the expansion is from 3 to 7, while in a two-shaft engine, the gas is expanded from 3 to 5 in the gas generator turbine and afterwards from 5 to 7 in the power turbine. The difference between Lines 1-2 and 3-7 describes the work output of the turbine, i.e. most of the work generated by the expansion 3-7 is used to provide the work 1-2 to drive the compressor.

In a two-shaft engine, the distances from 1 to 2 and from 3 to 5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 5-7 describes the work output of the power turbine.

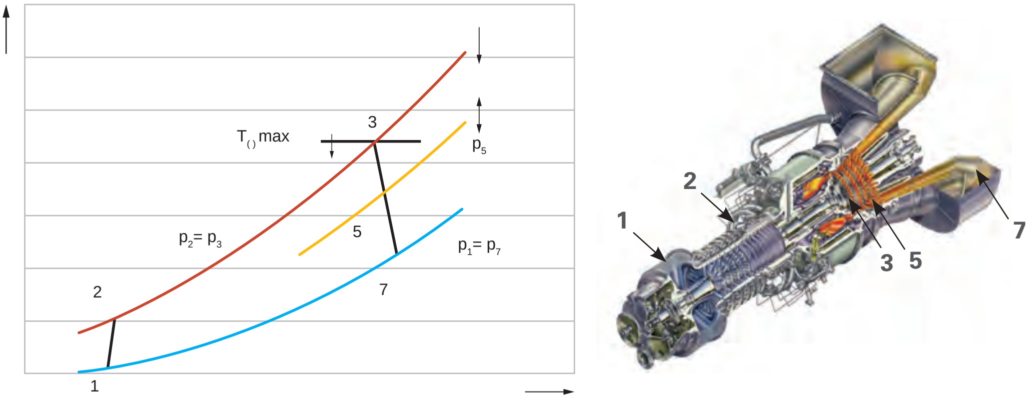


Figure 7-2. *Brayton Cycle*

For a perfect gas, enthalpy and temperature are related by

$$\Delta h = c_p \Delta T$$

For the actual process, the enthalpy change Δh for any step can be related to a temperature rise ΔT by a suitable choice of a heat capacity c_p for each of the steps.

We can thus describe the entire process (assuming that the mass flow is the same in the entire machine, i.e. neglecting the fuel mass flow and bleed flows, and further assuming that the respective heat capacities c_p, $c_{p,e}$ (for the inlet air) $c_{p,a}$ (for the exhaust gas) are suitable averages.

$$-c_{p,a}(T_2 - T_1) + c_{pe}(T_3 - T_7) = P / W$$

$$c_{p,e}(T_3 - T_2) = E_f W_f / W$$

In this equation, the first term is the work input by the compressor, and the third term describes is the work extracted by the turbine section. The second term the temperature increase from burning the fuel in the combustor.

For two-shaft engines, where the gas generator turbine has to balance the power requirements of the compressor, and the useful power output is generated by the power turbine, we can re-arrange the equation above to find:

$$-c_{p,a}(T_2 - T_1) = c_{pe}(T_3 - T_5)$$

$$c_{p,e}(T_5 - T_7) = P / W$$

This relationship neglects mechanical losses and the difference between the gas flow into the compressor and into the turbine due to the addition of fuel mass flow. However, the resulting inaccuracies are small, and don't add to the understanding of the general principles.

The compressor and the turbine sections of the engine follow the thermodynamic relationships between pressure increase and work input, which are for the compressor

$$P = \frac{\Delta h}{W} = \frac{c_{p,a}}{W}\left(T_2 - T_1\right) = \frac{c_p}{W}\frac{T_1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

and the turbine

$$P = \frac{\Delta h}{W} = \frac{c_{p,e}}{W}\left(T_3 - T_7\right) = \frac{c_p}{W}\cdot\eta_t\cdot T_3\left[1 - \left(\frac{p_7}{p_3}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

The efficiency of a gas turbine is defined by comparing the amount of power contained in the fuel fed into the engine with the amount of power yielded. The thermal efficiency is thus

$$\eta_{th} = \frac{P}{W_f E_f}$$

and the heat rate is

$$HR = \frac{1}{\eta_{th}} = \frac{W_f E_f}{P}$$

In this discussion, T_3, TIT and T_{RIT} will be (loosely) referenced as firing temperatures. The differences, which lie simply in fact that temperatures upstream of the first turbine nozzle (TIT) are different from the temperatures downstream of the first nozzle (T_{RIT}) due to the cooling of the nozzles, are not important for the understanding of the topic. Appendix A shows an example for a typical GT cycle.

The specific heat and ratios of specific heats, which determine the performance of the turbine section depend on the fuel-to-air ratio, the fuel composition and the relative humidity of the air. The main constituents of the exhaust gas are Nitrogen, Oxygen, Carbon dioxide and Water (see part 1). The specific heat (c_p) and gas constant (R) of all these constituents are known, so it is easy to calculate the overall c_p and γ, once the mole fractions of the constituents m_i are known.

The specific heat and ratios of specific heats, which determine the performance of the turbine section, depend on the fuel-to-air ratio, the fuel composition and the relative humidity of the air.

$$c_p = \frac{1}{100}\sum c_{p,i}\, m_i$$

$$R = \frac{1}{100}\sum R_i\, m_i$$

$$\gamma = \frac{c_p}{c_p - R}$$

where m_i is the mole fraction of the individual component.

The following *Table 7-1* gives γ, c_p and R of the above substances

at 10 bar, 800°C:

	c_p (J/kgK)	R(J/kgK)	$\gamma = 1/(1-R/c_p)$
Air	**1156**	**287**	**1.33**
CO_2	**1255**	**189**	**1.18**
H_2O	**2352**	**462**	**1.24**

and 1 bar, 500°C

	c_p (J/kgK)	R(J/kgK)	$\gamma = 1/(1-R/c_p)$
Air	**1093**	**287**	**1.36**
CO_2	**1158**	**189**	**1.19**
H_2O	**2132**	**462**	**1.28**

Table 7-1. *Specific heat (c_p), ratio of specific heats γ and gas constants (R) for some components of exhaust gas*

Assuming an expansion over a constant pressure ratio, and constant efficiency , then

$$\Delta T = \frac{\Delta h}{c_{p,e}} = \frac{\eta_t \cdot \Delta h_s}{c_{p,e}} = \eta_t T_3 \left[1 - \left(\frac{p_5}{p_3} \right)^{\frac{\gamma-1}{\gamma}} \right]$$

Thus, the temperature differential over the turbine for a given pressure ratio depends on the fuel gas composition, and the water content of the inlet air. The practical consequence of the above is the fact that most engines measure T_5 rather than T_3 for control purposes. The ratio T_5/T_3 is often assumed constant. However, in reality it is dependent on the fuel-to-air ratio, the fuel gas and the water content of the air (i.e. the relative humidity and the ambient temperature).

Another effect that must be considered is the fact that the relationship between the pressure ratio and the Mach number depend on γ. That means that the maximum volumetric flow through the 1st stage GP nozzle and the first stage PT nozzle also depend on the exhaust gas composition, which means that different fuel compositions (if the differences are very large) can influence the engine match.

Key cycle parameters for any gas turbine are its specific work and thermal efficiency which are related to the cycle pressure ratio and turbine inlet temperature. The general qualitative relationships between PR and TIT (Turbine Inlet Temperature) are indicated in *Figure 7-3* (Meher-Homji et al., 2009).

1. The salient relationships of this figure may be conveniently summarized as follows:
2. For a given TIT, gas turbine specific work increases with pressure ratio, reaching a maximum and then decreasing with further pressure ratio increase.
3. The specific work increases with increasing turbine inlet temperature.
4. The maximum specific work as the TIT is increased occurs at increasing pressure ratios.

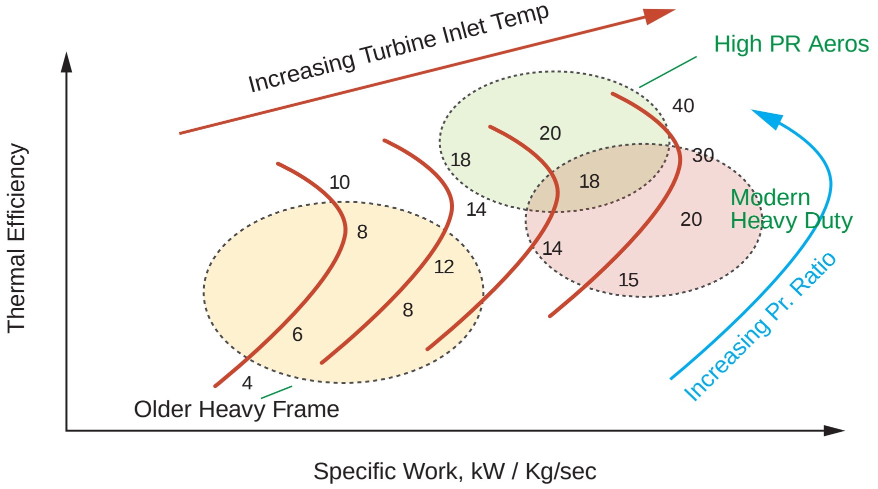

Increasing Turbine Inlet Temp
High PR Aeros
Thermal Efficiency
40
20
30
18
18
10
14
8
20
12
14
8
15
6
4
Modern
Heavy Duty
Increasing Pr. Ratio
Older Heavy Frame
Specific Work, kW / Kg/sec

Figure 7-3. *Key Cycle Parameters*

Pressure ratios and TIT parameters of a wide range of industrial gas turbines are shown in *Figure 7-4*.

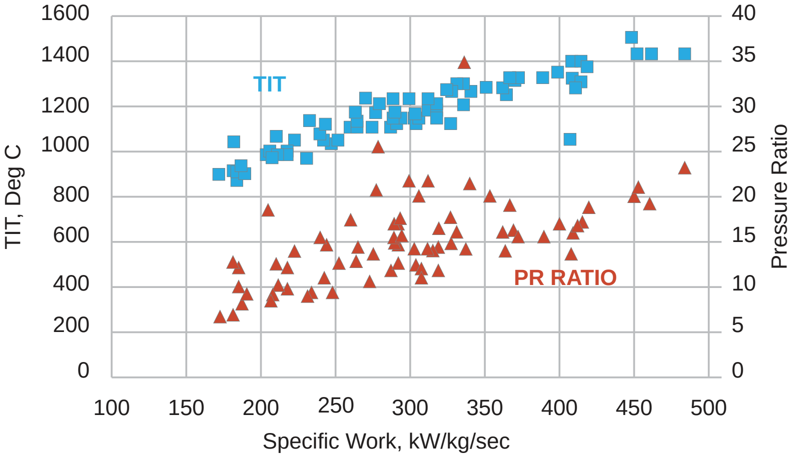


Figure 7-4. *Pressure Ratios and Firing Temperature Parameters of a Wide Range of Industrial Gas Turbines*

We will see that the gas turbine power output is a function of the speed, the firing temperature, as well as the position of certain secondary control elements, like adjustable compressor vanes, bleed valves, and in rare cases, adjustable power turbine vanes. The output is primarily controlled by the amount of fuel injected into the combustor. Most single-shaft gas turbines run at constant speed when they drive generators. In this case, the control system modifies fuel flow (and secondary controls) to keep the speed constant, independent of generator load. Higher load will, in general, lead to higher firing temperatures. Two-shaft machines are preferably used to drive mechanical equipment, because being able to vary the power turbine speed allows for a very elegant way to adjust the driven equipment to process conditions. Again, the power output is controlled by fuel flow (and secondary controls), and higher load will lead to higher gas producer speeds and higher firing temperatures.

Key to understand gas turbine behavior is the fact that the different components influence each other:

When the compressor, the gas generator turbine, and the power turbine (if applicable) are combined in a gas turbine, the operation of each component experiences operating constraints that are caused by the interaction (matching) between the components *(Figure 7-5)* (Texam, 2014).

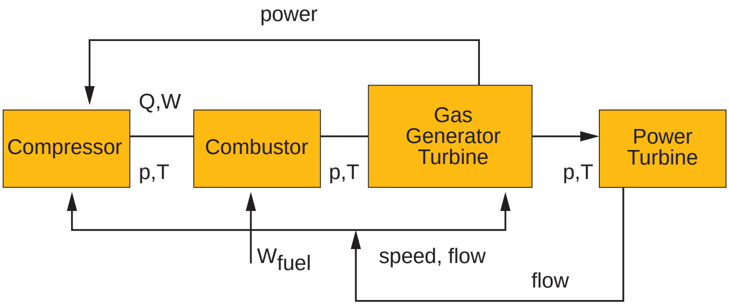


Figure 7-5. *Component Interaction*

For example, the engine compressor will compress a certain mass flow, which in turn dictates the compressor discharge pressure necessary to force the mass flow through the turbine section.

On a two-shaft engine, the gas generator turbine provides the power for the compressor. This determines the gas generator speed, where the compressor-absorbed power and the gas generator turbine-produced power are at equilibrium.

On a single-shaft engine, where the gas generator speed is held constant, the firing temperature has to be at a level where the power from the turbine, after the necessary power for the compressor is subtracted, satisfies the load demand of the driven electric generator. The firing temperature influences both the power that the turbines can produce, but it also impacts the discharge pressure necessary from the compressor.

In the following text, we will analyze the interaction of the gas turbine components.

SINGLE SHAFT

A single-shaft engine consists of an air compressor, a combustor and a turbine. The air compressor generates air at a high pressure, which is fed into to the combustor, where the fuel is burnt. The combustion products and excess air leave the combustor at high pressure and high temperature. This gas is expanded in the gas generator turbine, which provides the power to turn the air compressor. The excess power is used to drive the load. Most single-shaft turbines are used to drive electric generators at constant speeds. We will not consider the rare case of single-shaft turbines driving mechanical loads at varying speeds. The operation of the components require the following compatibility conditions:

1. Compressor speed = Gas generator Turbine speed
2. Mass flow through turbine = Mass flow through compressor - Bleed flows + Fuel mass flow
3. Compressor power < Turbine power

Typical compressor and turbine maps are shown in *Figure 7-6 and 7-7*, respectively.

The fact that the gas turbine operates at constant speed means any operating point of the engine compressor (for given ambient conditions) lies on a single speed line. Load increases are initiated by increasing the fuel flow, which in turn increases the firing temperature. Due to the fact that the first turbine nozzle is usually choked, the compressor operating point moves to a higher pressure ratio, to compensate for the reduced density (from the higher firing temperature). The possible operating points of the compressor depending on the load running are also shown in the compressor maps *(Figure 7-6).*

In the case of the single-shaft engine driving a generator, reduction in output power results in only minute changes in compressor mass flow as well as some reduction in compressor pressure ratio.

A single-shaft engine has no unique matching temperature. Used as a generator drive, it will operate at a single speed, and can be temperature topped at any ambient temperature as long as the load is large enough

The relationships between the work input and pressure ratio of turbine and compressor (we assume $p_7 = p_1$ and $W_c = W_t = W$ to make the example simpler) are given by:

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$\frac{\Delta T_t}{T_3} = \eta_t\left[1 - \left(\frac{p_7}{p_3}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

These relationships are a consequence of the relationships between temperature rise and pressure ratio in isentropic systems. The introduction of the component efficiencies then bridges the gap between the isentropic and real process.

The gas producer turbine and the compressor operate on the same shaft, thus:

$$\frac{N}{\sqrt{T_3}} = \frac{N}{\sqrt{T_1}} \cdot \sqrt{\frac{T_1}{T_3}}$$

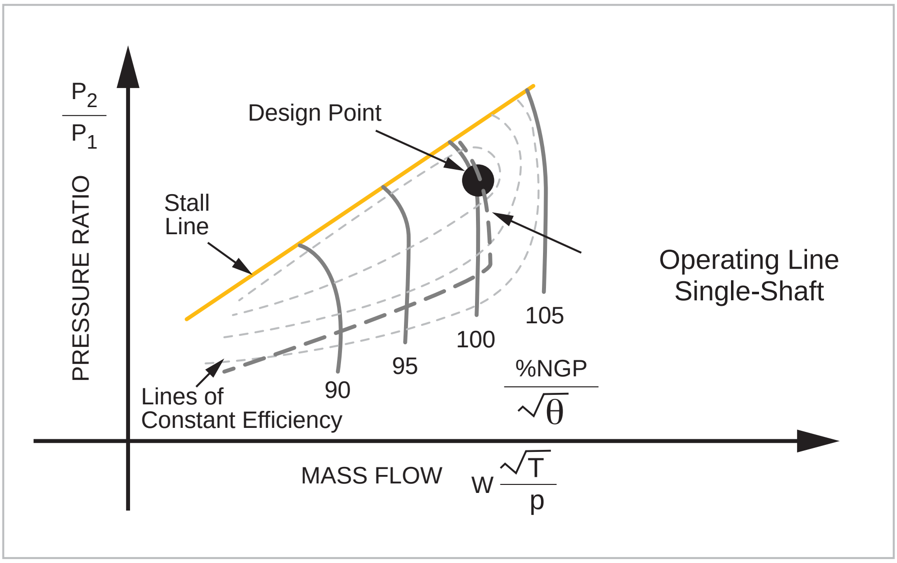

$\frac{P_2}{P_1}$
PRESSURE RATIO
Design Point
Stall Line
Operating Line Single-Shaft
105
100
95
90
$\frac{\%NGP}{\sqrt{\theta}}$
Lines of Constant Efficiency
MASS FLOW
$W\frac{\sqrt{T}}{p}$

Figure 7-6. *Typical Map of Axial Compressor – Single-Shaft Engine (Constant mechanical speed)*

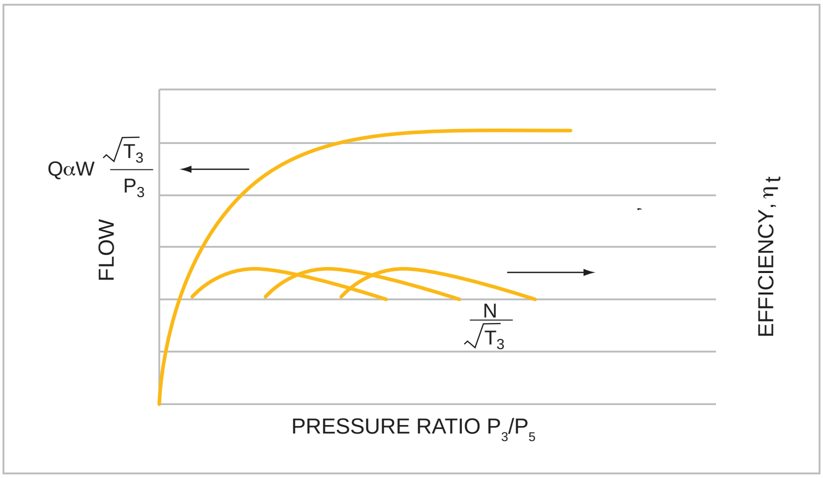


Figure 7-7. *Typical Map of Axial Turbine – Single-Shaft Engine*

The preceding equations are all coupled by the firing temperature ratio (T_3/T_1). This ratio has to be determined by trial and error for operation at any arbitrary point of the compressor characteristic using the following method:

1. On the compressor map *(Figure 7-6)*, choose any point on the operating ($N/\sqrt{T_1}$) line. The characteristics then yield p_2/p_1, $W\sqrt{T_1}/p_1$ and $\Delta T_c/T_1$.
2. Estimate a value for p_3/p_5. The turbine characteristic *(Figure 7-7)* then gives $W\sqrt{T_3}/p_3$ and T_3/T_1 can be obtained from the second equation; furthermore, it allows us to calculate $N/\sqrt{T_3}$.
3. With the turbine efficiency from the map *(Figure 7-7)* $\Delta T_t/T_3$ can be calculated and, using the power balance equation, another ratio (T_3/T_1) results.
4. Because the ratio T_3/T_1 normally is not the same as the initial one, a new ratio (p_3/p_5) has to be estimated, until the same ratio T_3/T_1 is obtained. At this point, a compatible operating point is found if the resulting T_3 is within the allowed operating limits of the engine.
5. The power output finally becomes the difference between turbine power and compressor absorbed power.

Using the above method, we can determine the performance of a single-shaft gas turbine at any operating point.

TWO SHAFT

A two-shaft gas turbine consists of an air compressor, a combustor, a gas generator turbine and a power turbine. The air compressor generates air at a high pressure, which is fed into to the combustor, where the fuel is burnt. The combustion products and excess air leave the combustor at high pressure and high temperature. This gas is expanded in the gas generator turbine, which has the sole task of providing power to turn the air compressor. After leaving the gas generator turbine, the gas still has a high pressure and a high temperature. It is now further expanded in the power turbine. The power turbine is connected to the driven equipment. It must be noted at this point, that the power turbine (together with the driven equipment) can and will run at a speed that is independent of the speed of the gas generator portion of the gas turbine (i.e., the air compressor and the gas generator turbine).

The gas generator is controlled by the amount of fuel that is supplied to the combustor. Its two operating constraints are the firing temperature and the maximum gas generator speed (on some engines, torque limits may also constrain the operation at low ambient temperatures). If the fuel flow is increased, both firing temperature and gas generator speed increase, until one of the two operating limits is reached. Variable stator vanes at the engine compressor are frequently used, however, not for the purpose of controlling the airflow, but rather to optimize the gas producer speed. In two-shaft engines, the airflow is controlled by the flow capacities of the gas generator turbine and power turbine nozzles.

Increasing the speed and temperature of the gas generator provides the power turbine with gas at a higher energy (i.e., higher pressure, higher temperature and higher mass flow), which allows the power turbine to produce more power. If the power supplied by the power turbine is greater than the power absorbed by the load, the power turbine together with the driven compressor will accelerate until equilibrium is reached.

The operation of the components requires the following compatibility conditions:

1. Compressor speed = Gas generator Turbine speed
2. Mass flow through turbine = Mass flow through compressor - Bleed flows + Fuel mass flow
3. Compressor power = Gas generator turbine power (- mechanical losses)
4. The subsequent free power turbine adds the requirement that the pressure after the GP turbine has to be high enough to force the flow through the power turbine.

Typical compressor and turbine maps are shown in *Figures 7-8 and 7-9*, respectively. The gas generator for a two-shaft engine adapts to different load requirements (and , accordingly different fuel flow) by changing both speed and firing temperature. Note, that the compressor operating points are very different between a single-shaft and a two-shaft engine *(Figure 7-6 and 7-8)*.

Again, we can write the relationships between the work input and the pressure ratio of components. To make the example easier to follow, we again assume $p_7 = p_1$ and $W_c = W_t = W$.

$$\frac{\Delta T_c}{T_1} = \frac{1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$\frac{\Delta T_t}{T_3} = \eta_t\left[1 - \left(\frac{p_5}{p_3}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

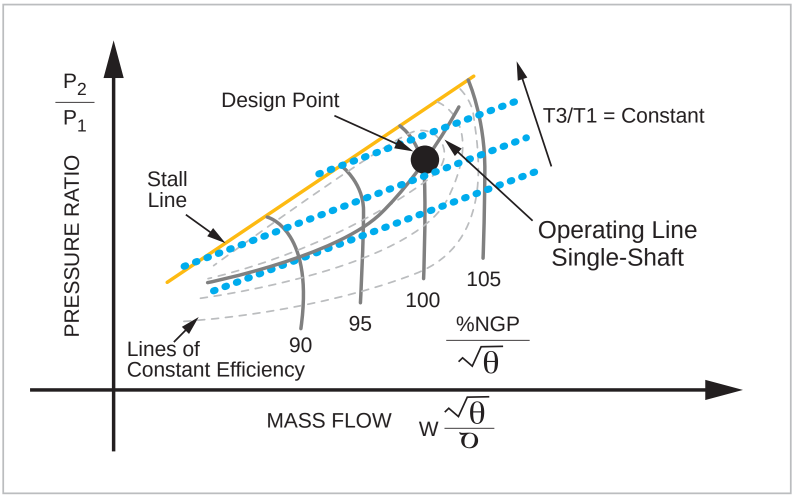


Figure 7-8. *Typical Map of Axial Compressor – Two-Shaft Engine*

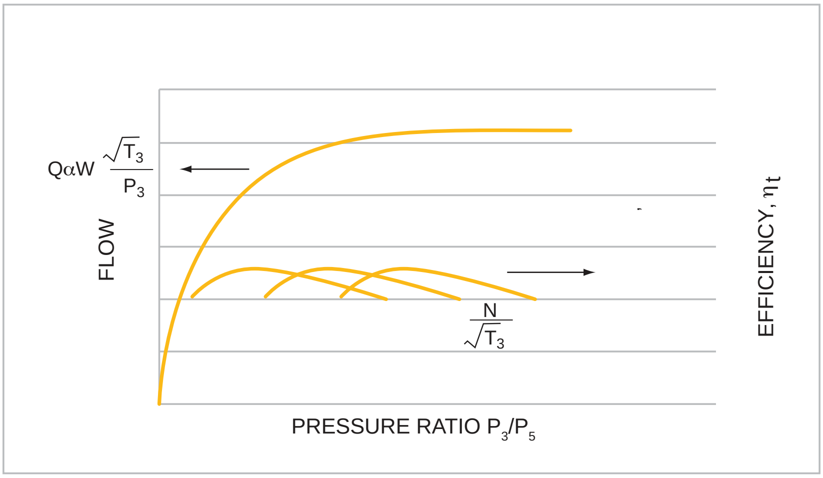


Figure 7-9. *Typical Map of Axial Turbine – Two-Shaft Engine*

Unlike for the single-shaft engine, the pressure ratio for the turbine (p_3/p_5) is not known in advance. However, we find an additional condition because the absorbed compressor power must be provided by the gas producer turbine:

$$c_{p,a}\cdot\frac{\Delta T_c}{T_1}\cdot\frac{1}{\eta_m}=c_{p,e}\cdot\frac{\Delta T_t}{T_3}\frac{T_3}{T_1}$$

Also, the gas producer turbine and compressor operate on the same shaft, thus:

$$\frac{N}{\sqrt{T_3}}=\frac{N}{\sqrt{T_1}}\cdot\sqrt{\frac{T_1}{T_3}}$$

The preceding equations are all coupled by the firing temperature ratio (T_3/T_1). This ratio can be determined by trial and error for operation at any arbitrary point of the compressor characteristic, using the following method:

1. On the compressor map *(Figure 7-8)*, choose any point on the operating ($N/\sqrt{T_1}$) line. The characteristics then yield p_2/p_1, $W\sqrt{T_3}/p_3$ and $\Delta T_c/T_1$.
2. Estimate a value for p_3/p_5. The turbine characteristic gives $W\sqrt{T_3}/p_3$ and T_3/T_1 can be obtained from the second equation. Furthermore, this characteristic allows us to calculate N/T_3 using the same speed (N) as previously used for the compressor.
3. With the turbine efficiency from the map *(Figure 7-9)*, $\Delta T_t/T_3$ can be calculated and, using the power balance equation, another ratio T_3/T_1 results. Because the ratio T_3/T_1 normally is not the same as the initial one, a new ratio (p_3/p_5) has to be estimated, until the same ratio T_3/T_1 is obtained. At this point, a compatible operating point is found if the resulting T_3 is within the allowed operating limits of the engine.
4. For the power turbine, the following relations apply:

$$\frac{W\sqrt{T_5}}{p_5} = \frac{W\sqrt{T_1}}{p_1} \cdot \frac{p_1}{p_5} \cdot \sqrt{\frac{T_5}{T_1}}$$

$$\frac{\Delta T_{pt}}{T_5} = \eta_{pt}[1 - (\frac{p_a}{p_5})^{\frac{\gamma-1}{\gamma}}]$$

This condition adds another constraint, because the available pressure ratio (p_5/p_1) has to be sufficient to move the mass flow (W) through the power turbine section. If the available pressure ratio is too low, another operating point at a lower mass flow, thus a lower gas producer speed, has to be selected.

5. Combining the relationships for the gas producer turbine and the power turbine yields *(Figure 7-10)*:

$$\frac{W\sqrt{T_5}}{p_5} = \frac{W\sqrt{T_3}}{p_3} \cdot \frac{p_3}{p_5} \cdot \sqrt{\frac{T_5}{T_3}}$$

$$\sqrt{\frac{T_5}{T_3}} = \sqrt{1 - \eta_t[1 - (\frac{p_5}{p_3})^{\frac{\gamma-1}{\gamma}}]}$$

$$\frac{p_5}{p_a} = \frac{p_2}{p_a} \cdot \frac{p_3}{p_2} \cdot \frac{p_5}{p_3}$$

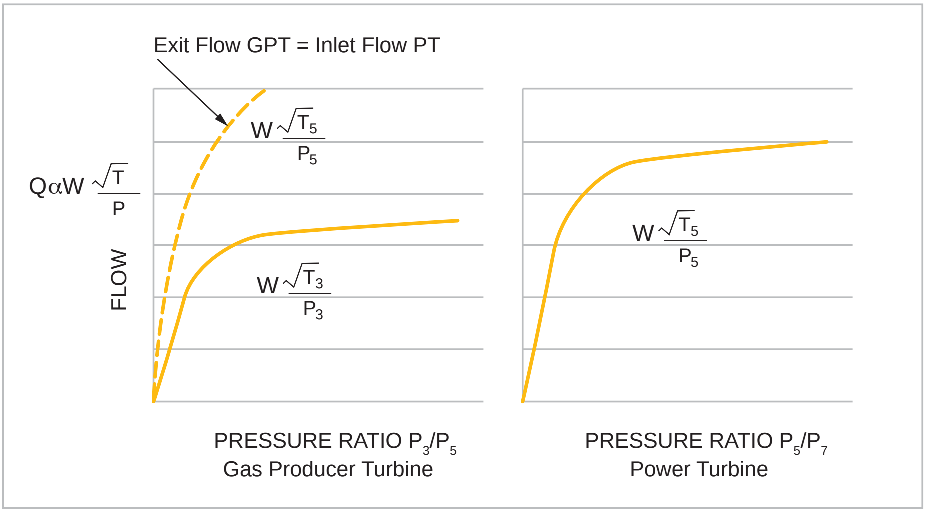


Figure 7-10. *Turbine Characteristic for Gas Producer and Power Turbine*

6. Finally, the engine output is the power that the power turbine generates at the available gas producer exit temperature and pressure ratio over the power turbine:

$$P_{pt} = c_p \cdot \Delta T_{pt} \cdot W = c_p T_5 \eta_{pt} [1 - (\frac{p_7}{p_5})^{\frac{\gamma-1}{\gamma}}]W$$

Note that the power turbine efficiency depends strongly on the actual power turbine speed deviation from its optimum speed.

Two-shaft engines operate with the gas generator turbine and the power turbine in series. The power turbine pressure ratio p_5/p_a is thus related to the compressor pressure ratio p_2/p_a by the identity:

$$\frac{p_5}{p_a} = \frac{p_2}{p_a}\frac{p_3}{p_2}\frac{p_5}{p_3}$$

The pressure drop in the gas generator turbine p_5/p_3 and the pressure increase in the compressor p_2/p_a are related insofar as the gas generator turbine has to provide enough power to drive the compressor.

The maximum possible pressure ratio p_5/p_a is controlled by the flow capacity Q_5 of the power turbine. In particular, if the power turbine is choked, it will cause the gas generator turbine to operate at one fixed point *(Figure 7-10)*. In many cases, both the gas generator turbine first-stage nozzle and the power turbine nozzle operate at or near choked flow conditions. In this case, the actual flow Q_3 through the gas generator turbine nozzle is practically constant. The mass flow is then only dependent on the combustor exit pressure p_3, the firing temperature T_3 , the gas composition (which determines γ, and thus the volume increase during the expansion), and the geometry of the nozzle, which determines the through flow area (in reality, it is determined by the critical nozzle area, the clearance area and the effective bleed valve area).

Gas Turbine Performance Control

Gas turbines are primarily controlled by the amount of fuel supplied. There are some secondary control options, such as adjustable vanes, or bleeding air from the compressor directly to the exhaust. The control system will limit the gas turbine operation within the limits of maximum gas generator speed and maximum firing temperature.

A single-shaft engine, is controlled by the amount of fuel that is supplied to the combustor, such that the speed of the engine stays constant even if the load changes. Its other operating constraint is the firing temperature. Variable stator vanes at the engine compressor are frequently used, for the purpose of controlling the airflow. Further, the airflow is constrained by the flow capacities of the gas generator turbine nozzles.

In a two-shaft engine, the gas generator operation is controlled by the amount of fuel that is supplied to the combustor. Its two operating constraints are the firing temperature and the maximum gas generator speed (on some engines, torque limits or corrected core speed limit may also constrain the operation at low ambient temperatures). If the fuel flow is increased, both firing temperature and gas generator speed increase, until one of the two operating limits is reached. Variable stator vanes at the engine compressor are frequently used, however, not for the purpose of controlling the airflow, but rather to optimize the gas producer speed. In two-shaft engines, the airflow is controlled by the flow capacities of the gas generator turbine and power turbine nozzles.

Increasing the speed and temperature of the gas generator provides the power turbine with gas at a higher energy (i.e., higher pressure, higher temperature, and higher mass flow), which allows the power turbine to produce more power. If the power supplied by the power turbine is greater than the power absorbed by the load, the power turbine together with the driven compressor will accelerate until equilibrium is reached.

One of the two operating limits of a gas turbine is the turbine rotor inlet temperature (TRIT or T_3). Unfortunately, it is not possible to measure this temperature directly – a temperature probe would only last for a few hours at temperatures that high. Therefore, the inlet temperature into the power turbine (T_5) is measured instead. The ratio between T_3 and T_5 is determined during the factory test, where T_5 is measured and T_3 is determined from a thermodynamic energy balance. This energy balance requires the accurate determination of output power and air flow, and can therefore be performed best during the factory test.

It must be noted, that both T_3 and T_5 are circumferentially and radially very non-uniformly distributed in the reference planes (i.e., at the combustor exit, at the rotor inlet, at the power turbine inlet). Performance calculations use a thermodynamic average temperature. This is not exactly the temperature one would measure as the average of a number of circumferentially distributed temperature probes.

Rather than controlling T_3 the control system limits engine operations to the T_5 that corresponds to the rated T_3 . However, the ratio between T_3 and T_5 is not always constant, but varies with the ambient temperature: The ratio T_3 / T_5 is reduced at higher ambient temperatures. Modern control algorithms can take this into account.

Engines can also be controlled by their exhaust temperature (T_7). For single-shaft engines, measuring T_7 or T_5 are equivalent choices. For two-shaft engines, measuring T_7 instead of

T_5 adds the complication that the T_7 control temperature additionally depends on the power turbine speed, while the relationship between T_3 and T_5 does not depend on the power turbine speed.

GAS TURBINE PERFORMANCE CHARACTERISTICS

The gas turbine power output is a function of the speed, the firing temperature, as well as the position of certain secondary control elements, like adjustable compressor vanes, bleed valves, and in rare cases, adjustable power turbine vanes. The output is primarily controlled by the amount of fuel injected into the combustor. Most single-shaft gas turbines run at constant speed when they drive generators. In this case, the control system modifies fuel flow (and secondary controls) to keep the speed constant, independent of generator load. Higher load will, in general, lead to higher firing temperatures. Two-shaft machines are preferably used to drive mechanical equipment, because being able to vary the power turbine speed allows for a very elegant way to adjust the driven equipment to process conditions. Again, the power output is controlled by fuel flow (and secondary controls), and higher load will lead to higher gas producer speeds and higher firing temperatures.

Figure 7-11 shows the influence of ambient pressure and ambient temperature on gas turbine power and heat rate, as well as the impact of part load and power turbine speed. The influence of ambient temperature on gas turbine performance is very distinct. Any industrial gas turbine in production will produce more power when the inlet temperature is lower and less power when the ambient temperature gets higher. The rate of change cannot be generalized and is different for different gas turbine models. Full-load gas turbine power output is typically limited by the constraints of maximum firing temperature and maximum gas producer speed (or, in twin-spool engines, by one of the gas producer speeds). Gas turbine efficiency is less impacted by the ambient temperature than the power.

Lower ambient pressure (for example, due to higher-site elevation) will lead to lower power output, but has practically no impact on efficiency. It must be noted that the pressure drop due to the inlet and exhaust systems impact power and efficiency negatively with the inlet pressure drop having a more severe impact.

Gas turbines operated in part load will generally loose some efficiency. Again, the reduction in efficiency with part load is very model-specific. Most gas turbines show a very small drop in efficiency for at least the first 10% of drop in load.

In two-shaft engines, the power turbine speed impacts available power and efficiency. For any load and ambient temperature, there is an optimum power turbine speed. Usually, lowering the load (or increasing the ambient temperature) will lower the optimum power turbine speed. Small deviations from the optimum (by say +/- 10%) have very little impact on power output and efficiency.

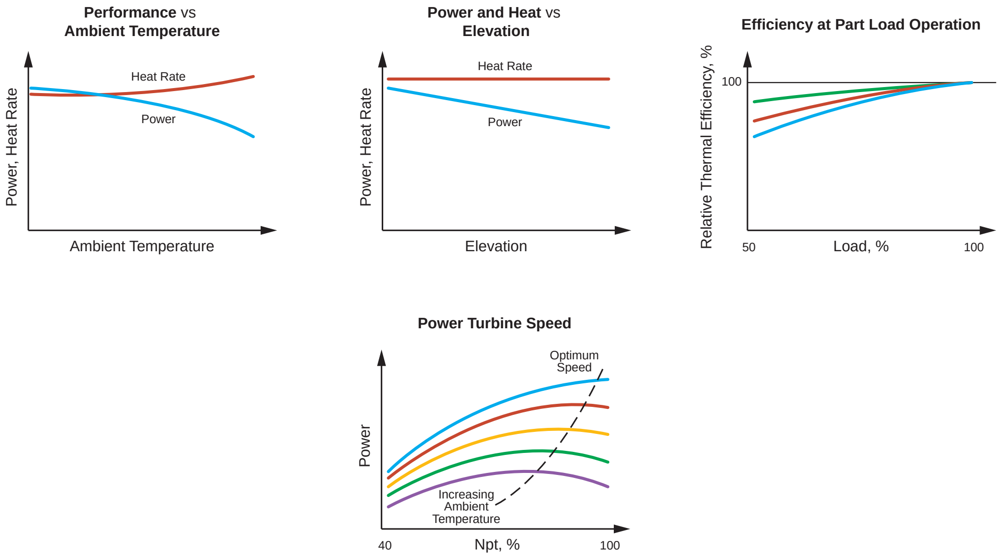


Figure 7-11. Performance Characteristics

The humidity does impact power output, but to a small degree, (generally, not more than 1 to 3%, even on hot days). The impact of humidity tends to increase at higher ambient conditions.

After this general overview, we will discuss the influence factors in more detail:

Load

Any gas turbine will experience a reduced efficiency at part load *(Figure 7-11)*. The reduction in efficiency with part load differs from design to design. In particular, DLN engines show different part-load efficiencies than their conventional combustion counterparts. The necessity to control the fuel-to-air ratio closely yields different part load behavior when comparing gas turbines with DLN combustors and with conventional combustors. A typical way of controlling these engines is by controlling the airflow into the combustor, thus keeping the combustor primary zone temperature within narrow limits. The part load behavior of single shaft and two shaft, Standard Combustion and Dry Low No_x concepts is fundamentally different. This is due both to the different aerodynamic configuration, and the requirements of keeping the fuel-to-air ratio within a narrow window for DLN engines.

To operate the engine at part load, the fuel flow is adjusted, until some control parameter (for example the flow through the driven compressor, or a certain kW value for a generator) is satisfied. Other adjustments, such as guide vane settings or bypassing combustion air may be necessary.

For a single-shaft engine, which has to operate at constant gas generator speed (to keep the generator frequency constant) this means that the firing temperature will be changed with load. A governor will keep the speed constant and will increase the fuel flow with increasing load, thus increasing the firing temperature, until the control limit is reached. Due to the constant speed, the airflow through the engine will not vary greatly between full load and part load. This means, that the fuel-to-air ratio drops significantly at part load and the combustor exit temperature drops significantly from full load to part load. Therefore, most single-shaft dry-low NO_x engines use variable stator vanes on the engine compressor to vary the airflow, and thus keep the fuel-to-air ratio relatively constant.

For a given mass flow, any increase in firing temperature would increase the volume flow through the turbine section. Therefore the pressure supplied from the compressor has to be increased, because the turbine nozzle is at or near choked conditions. Since the compressor also operates at constant speed, the result is a reduction of mass flow until equilibrium is reached. A typical performance map for a single-shaft engine shows this increase in compressor discharge pressure at increased load. If the engine is equipped with VIGV's to keep the fuel-to-air ratio in the combustor constant, a reduction in load will require a closing of the VIGV's to reduce the airflow. Closing the VIGV's also reduces the pressure ratio of the compressor at constant speed.

For single-shaft engines as described above the part load efficiencies of gas turbines with DLN combustion and conventional combustion are very similar. The need to bleed combustion air for two-shaft engines typically leads to a lower part load efficiency for engines equipped with DLN combustors.

For a two-shaft engine, both gas generator speed and firing temperature change with load.

If the output at the power turbine has to be increased, the fuel flow has to increase. Because the gas generator is not mechanically coupled with the power turbine, it will accelerate, thus increasing airflow, compressor discharge pressure and mass flow. The increase in gas generator speed means, that the compressor now operates at a higher Mach number. At the same time the increased fuel flow will also increase the firing temperature. The relative increase is governed by the fact that the power turbine requires a certain pressure ratio to allow a given amount of airflow pass. This forces an equilibrium where the following requirements have to be met:

1. The compressor power equals gas generator turbine power. This determines the available pressure upstream of PT.

2. The available pressure ratio at the power turbine is sufficient to allow the airflow to be forced through the power turbine.

Depending on the ambient temperature relative to the engine match temperature, the fuel flow into the engine will either be limited by reaching the maximum firing temperature or the maximum gas generator speed. The ambient temperature, where both control limits are reached at the same time is called engine match temperature.

Variable stator vanes at the engine compressor are frequently used, however, not for the purpose of controlling the airflow. This is due to the fact that in two-shaft engines, the airflow is controlled by the flow capacities of the gas generator turbine and power turbine nozzles. To control the fuel-to-air ratio in the combustor, another control feature has to added for two-shaft engines with Dry Low No_x combustors: Usually, a certain amount of air is bled from the compressor exit directly into the exhaust duct. This leads to the fact that while the airflow for two-shaft engines is reduced at part load, the reduction in airflow is larger for an engine with a standard combustion system. Like in single-shaft engines, the combustor exit temperature at part load drops significantly for engines with standard combustion, while it stays relatively high for DLN engines. The drop in combustor temperature in engines with standard combustion, which indicates the leaner fuel-to-air ratio, automatically leads to NO_x emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction, because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel-to-air ratio.

Ambient Temperature

Changes in ambient temperature have an impact on full-load power and heat rate, but also on part-load performance and optimum power turbine speed *(Figure 7-11)* Manufacturers typically provide performance data that describe these relationships for ISO conditions. These curves are the result of the interaction between the various rotating components and the control system. This is particularly true for DLN engines.

If the ambient temperature changes, the engine is subject to the following effects:

1. The air density changes. Increased ambient temperature lowers the density of the inlet air, thus reducing the mass flow through the turbine, and therefore reduces the power output (which is proportional to the mass flow) even further. At constant speed, where the volume flow remains approximately constant, the mass flow will increase with decreasing temperature and will decrease with increasing temperature.
2. The pressure ratio of the compressor at constant speed gets smaller with increasing temperature. This can be determined from a Mollier diagram, showing that the higher the inlet temperature is, the more work (or head) is required to achieve a certain pressure ratio. The increased work has to be provided by the gas generator turbine, and is thus lost for the power turbine, as can be seen in the Enthalpy-Entropy Diagram.

At the same time N_{Ggcorr} (i.e., the machine Mach number) at constant speed is reduced at higher ambient temperature. As explained previously, the inlet mach number of the engine compressor will increase for a given speed, if the ambient temperature is reduced. The gas generator Mach number will increase for reduced firing temperature at constant gas generator speed.

The Enthalpy-Entropy Diagram *(Figure 7-12)* describes the Brayton cycle for a two-shaft gas turbine. Lines 1-2 and 3-5 must be approximately equal, because the compressor work has to be provided by the gas generator turbine work output. Line 5-7 describes the work output of the power turbine. At higher ambient temperatures, the starting point 1 moves to a higher temperature. To make the same pressure ratio, the compressor will consume more work, the gas producer turbine has to generate more work, and therefore the power turbine inlet pressure p_5 has to drop. Looking at the combustion process 2-3, with a higher compressor discharge temperature and considering that the firing temperature T_3 is limited to T_{RITmax}, we see that less heat input is possible, i.e., less fuel will be consumed. The expansion process has, due to the lower p_5, less pressure ratio available or a larger part of the available expansion work is being used up in the gas generator turbine, leaving less work available for the power turbine.

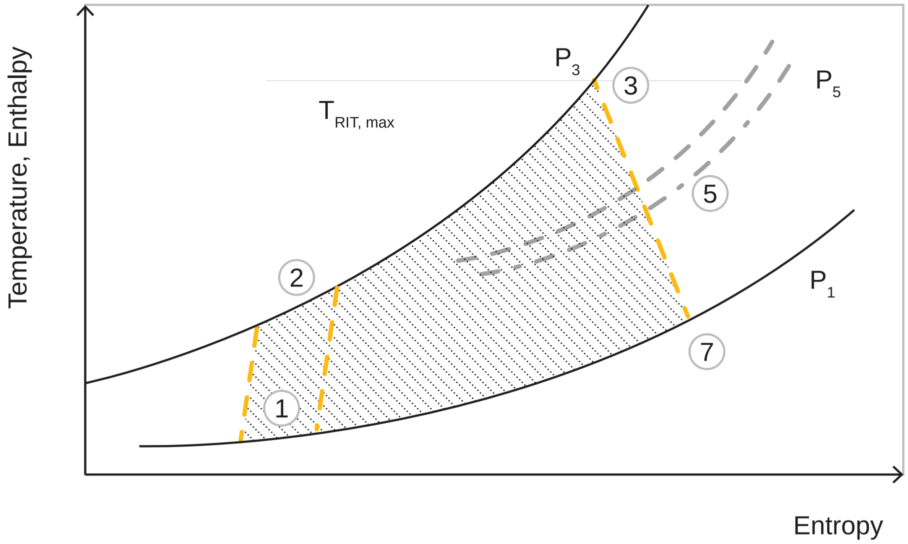


Figure 7-12. *Brayton Cycle for Different Ambient Temperatures*

On two-shaft engines, a reduction in gas generator speed occurs at high ambient temperatures. This is due to the fact that the equilibrium condition between the power requirement of the compressor (which increases at high ambient temperatures if the pressure ratio must be maintained) and the power production by the gas generator turbine (which is not directly influenced by the ambient temperature as long as compressor discharge pressure and firing temperature remain) will be satisfied at a lower speed.

The lower speed often leads to a reduction of turbine efficiency: The inlet volumetric flow into the gas generator turbine is determined by the first stage turbine nozzle, and the Q3/NGG ratio (i.e., the operating point of the gas generator turbine) therefore moves away from the optimum. Variable compressor guide vanes allow to keep the gas generator speed constant at higher ambient temperatures, thus avoiding efficiency penalties.

In a single-shaft, constant-speed gas turbine one would see a constant head (because the head stays roughly constant for a constant compressor speed), and thus a reduced pressure ratio. Because the flow capacity of the turbine section determines the pressure-flow-firing temperature relationship, an equilibrium will be found at a lower flow, and a lower pressure ratio, thus a reduced power output.

3. The compressor discharge temperature at constant speed increases with increasing temperature. Thus, the amount of heat that can be added to the gas at a given maximum firing temperature is reduced.

4. The relevant Reynolds number changes

At full load, single-shaft engines will run a temperature topping at all ambient temperatures, while two-shaft engines will run either at temperature topping (at ambient temperatures higher than the match temperature) or at speed topping (at ambient temperatures lower than the match temperature). At speed topping, the engine will not reach its full firing temperature, while at temperature topping, the engine will not reach its maximum speed.

The net effect of higher ambient temperatures is an increase in heat rate and a reduction in power. The impact of ambient temperature is usually less pronounced for the heat rate than for the power output, because changes in the ambient temperature impact less the component efficiencies than the overall cycle output.

Humidity

The impact of humidity on engine performance would be better described by the water content of the air (say, in mole%) or in terms of the specific humidity ($kg_{H20}/kg_{dry\ air}$). *Figure 7-13* illustrates this, relating relative humidity for a range of temperatures with the specific humidity.

Since the water concentration in the air for the same relative humidity increases with increasing temperature, the effects on engine performance are negligible for low ambient temperatures and fairly small (in the range of 1 or 2%) even at high temperatures of 38°C (100°F). Since the water content changes the thermodynamic properties of air (such as density and heat capacity), it causes a variety of changes in the engine, such that on some engines performance is increased with increased humidity, while other engines show reduced performance at increased humidity.

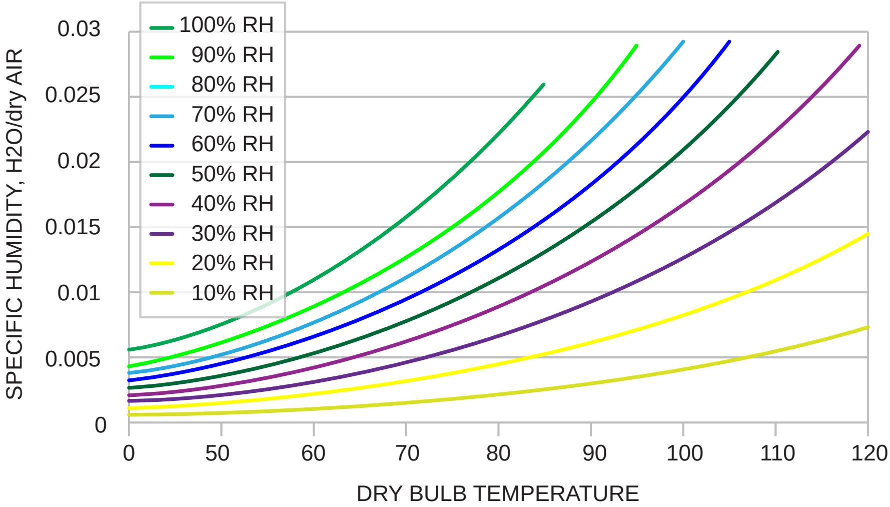


***Figure 7-13.** Specific and Relative Humidity as a Function of Temperature*

The main properties of concern that are affected by humidity changes are density, specific heat, and enthalpy. Because the molecular weight of water (18 g/mol) is less than dry air (28 g/mol), density of ambient air actually decreases with increasing humidity. When the density of the ambient air decreases the total mass flow will decrease, which then will decrease thermal efficiency and output power.

Performance of the combustor and turbines as a function of humidity is dominated by the changes in specific heat and enthalpy. Increases in water content will decrease temperatures during and after combustion (the same reason water is injected into the fuel to reduce NO_x levels).

Since the water concentration in the air for the same relative humidity increases with increasing temperature, the effects on engine performance are negligible for low ambient temperatures and fairly small (in the range of 1 or 2%) even at high temperatures of 38°C (100°F). The water content changes the thermodynamic properties of air (such as density and heat capacity) and thus causes a variety of changes in the engine.

For single-shaft engines, increasing humidity will decrease temperatures at the compressor exit. Humidity also causes decreased flame temperatures at a given fuel-air ratio. As a result T_2, combustor exit temperature, T_{RIT} and T_5 all decrease with an increase in humidity. Since the speed is set in single-shaft engines, the controls system will increase fuel flow in order to get T_5 temperature up to the topping set point. Despite the increase in fuel flow, the total exhaust flow still decreases due to the decrease in airflow. Output power increases throughout the range of temperatures and humidity experienced by the engines, which shows that the increased fuel energy input has a greater influence on output power than does the decreased total flow.

In two-shaft engines, we have to distinguish whether the engine runs at maximum speed (NGP topped), or at maximum firing temperature (T_5 topped). Increasing humidity will decrease air density and mass flow when running NGP topped, which will decrease output power. This is the general trend in output power noticed in all two-shaft engines when running NGP topped. As previously discussed, increased humidity causes lower T_2, Flame temperature, T_{RIT}, and T_5 temperatures. When running T_5 topped, the trend in output power reverses due to the engine increasing fuel flow to increase temperatures, and results in increased output power. So for two-shaft engines, output power will be seen to increase when running T_5 topped, and to decrease when running NGP topped.

Wobbe Index, Fuel Supply Pressure

While the influence of the fuel composition on performance is rather complex, fortunately the effect on performance is rather small if the fuel is natural gas. Fuel gas with a large amount of inert components (such as CO_2 or N_2) have a low Wobbe index, while substances with a large amount of heavier hydrocarbons have a high Wobbe index. Pure methane has a Wobbe index of about 1220.

In general, engines will provide slightly more power if the Wobbe Index

$$WI = \frac{LHV}{\sqrt{SG}}$$

is reduced. This is due to the fact that the amount of fuel mass flow increases for a given amount of fuel energy when the Wobbe index is reduced. This increases the mass flow though the turbine section, which increases the output of the turbine. This effect is to some degree counteracted by the fact that the compressor pressure ratio has to increase to push the additional flow through the flow restricted turbine. In order to do this, the compressor will absorb somewhat more power. The compressor will also operate closer to its stall margin. The above is valid whether the engine is a two-shaft or single-shaft engine.

The fuel gas pressure at skid edge has to be high enough to overcome all pressure losses in the fuel system and the combustor pressure, which is roughly equal to the compressor discharge pressure p_2. The compressor discharge pressure at full load changes with the ambient temperature, and therefore, a fuel gas pressure that is too low for the engine to reach full load at low ambient temperature may be sufficient if the ambient temperature increases.

If the fuel supply pressure is not sufficient, single and two-shaft engines show distinctly different behavior, namely:

A two-shaft engine will run slower, such that the pressure in the combustor can be overcome by the fuel pressure *(Figure 7-14)*. If the driven equipment is a gas compressor (and the process gas can be used as fuel gas), 'bootstrapping' is often possible: The fuel gas is supplied from the gas compressor discharge side. If the initial fuel pressure is sufficient to start the engine and to operate the gas compressor, the gas compressor will increase the fuel gas pressure. Thus, the engine can produce more power which in turn will allow the gas compressor to increase the fuel pressure even more, until the fuel gas pressure necessary for full load is available.

A single-shaft engine, which has to run at constant speed, will experience a severe reduction in possible firing temperature and significant loss in power output, unless it uses VIGV's. With VIGV's, the compressor exit pressure, and thus the combustor pressure can also be influenced by the position of the VIGV's, thus leading to less power loss *(Figure 7-14)*.

Without VIGV's, the only way to reduce PCD pressure is by moving the operating point of the compressor on its map. This can be done by reducing the back pressure from the turbine, which requires a reduction in volume flow. Since the speed is fixed, only a reduction in firing temperature — which reduces the volume flow through the gas generator if everything else remains unchanged — can achieve this. A reduced volume flow will reduce the pressure drop required for the gas generator turbine.

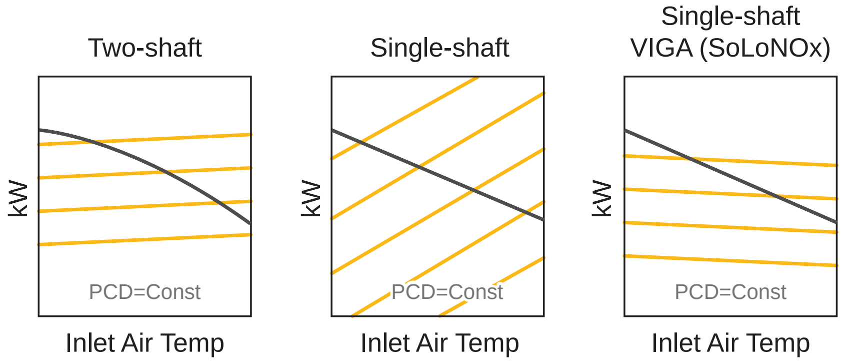


Figure 7-14. *Effect of low fuel gas pressure on different engine designs. PCD is the pressure in the combustor.*

Elevation/Ambient pressure

The impact of operating the engine at lower ambient pressures (for example, due to site elevation or simply due to changing atmospheric conditions) is that of a reduced air density *(Figure 7-11)*. The engine, thus, sees a lower mass flow (while the volumetric flow is unchanged). The changed density only impacts the power output, but not the efficiency of the engine. However, if the engine drives accessory equipment through the gas generator, this is no longer true because the ratio between gas generator work and required accessory power (which is independent of changes in the ambient conditions) is affected .

The impact is universal for any engine, except for the result of some secondary effects such as accessory loads. If the ambient pressure is known, the performance correction can be easily accomplished by:

$$\delta = \frac{p_{ambient}}{p_{sea\ level}}$$

If only the site elevation is known, the ambient pressure at normal conditions is:

$$p_{ambient} = p_{sea\ level} \cdot e^{-\frac{elevation\,(ft)}{27200}}$$

Inlet/Exhaust Loss/Accessory Loads

Any gas turbine needs an inlet and exhaust system to operate. The inlet system consists of one or several filtration systems, a silencer, ducting and possibly de-icing, fogging, evaporative cooling and other systems. The exhaust system may include a silencer, ducting, and waste heat recovery systems. All these systems will cause pressure drops, i.e., the engine will actually see an inlet pressure that is lower than ambient pressure, and will exhaust against a pressure that is higher than the ambient pressure. These inevitable pressure losses in the inlet and exhaust system cause a reduction in power and cycle efficiency of the engine. The reduction in power, compared to an engine at ISO conditions, can be described by simple correction curves, which are usually supplied by the manufacturer. The ones shown in *Figure 7-15* describe the power reduction for every inch (or millimeter) of water pressure loss. These curves can be easily approximated by second order polynomials. The impact on heat rate is easily calculated by taking the fuel flow from ISO conditions and dividing it by the reduced power.

Accessory loads are due to mechanically driven lube oil or hydraulic pumps. While the accessory load can be treated fairly easily in a single-shaft engine — its power requirement is subtracted from the gross engine output — this is somewhat more complicated in a two-shaft machine.

In a two-shaft gas turbine, the accessory load is typically taken from the gas generator. In order to satisfy the equilibrium conditions, the gas generator will have to run hotter than without the load. This could lead to more power output at conditions that are not temperature limited. When the firing temperature is limited (i.e., for ambient temperatures above the match point), the power output will fall off more rapidly than without the load. That means, that an accessory load of 50hp may lead to power losses at the power turbine

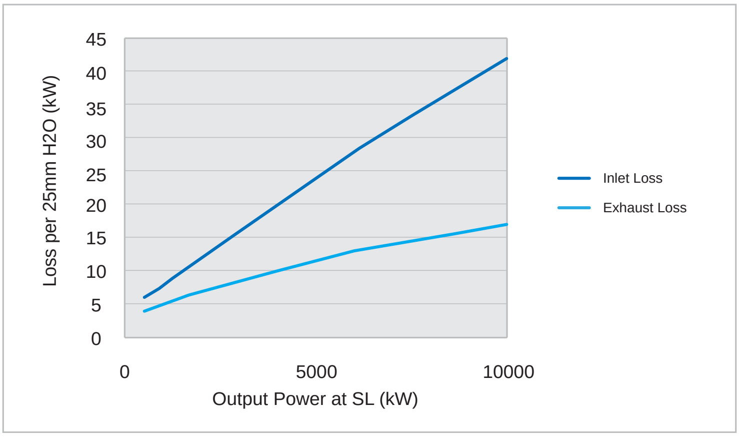


Figure 7-15. *Impact of Inlet and Exhaust Pressure losses*

of 100 or more hp at higher ambient temperatures. The heat rate will increase due to accessory loads at all ambient temperatures. The net effect of accessory loads can also be described as a move of the match point to lower ambient temperatures.

Power Turbine Speed

For any operating condition of the gas generator, there is an optimum power turbine speed at which the power turbine operates at its highest efficiency, and thus produces the highest amount of power for a given gas generator operating point. Aerodynamically, this optimum point is characterized by a given ratio of tip speed (u) and axial velocity (c_{ax}). The axial component of the velocity (c_{ax}) is proportional to the volume flow at the power turbine inlet Q_5, while the tip speed is related to the rotating speed N_{PT}. The volumetric flow depends on the ambient temperature and the load. This explains why the optimum power turbine speed is a function of ambient temperature and load.

If the power turbine does not operate at this the optimum power turbine speed, the power output and the efficiency of the power turbine will be lower *(Figure 7-16)*. The impact of changing the power turbine speed is easily described by:

$$\frac{P}{P_{opt}} = 2 \cdot \frac{N_{PT}}{N_{PT,opt}} - \left(\frac{N_{PT}}{N_{PT,opt}} \right)^2$$

This equation can be derived from basic relationships (see Chapter 6) and is pretty accurate for any arbitrary power turbine:

When using this relationship, it must be considered that the optimum power turbine speed ($N_{pt,opt}$) depends on the gas generator load and the ambient temperature (Kurz and Brun, 2001). In general, the optimum power turbine speed is reduced for increasing ambient temperatures and lower load *(Figure 7-11)*. The heat rate becomes for a constant gas generator operating point:

$$\frac{HR}{HR_{opt}} = \frac{P_{opt}}{P}$$

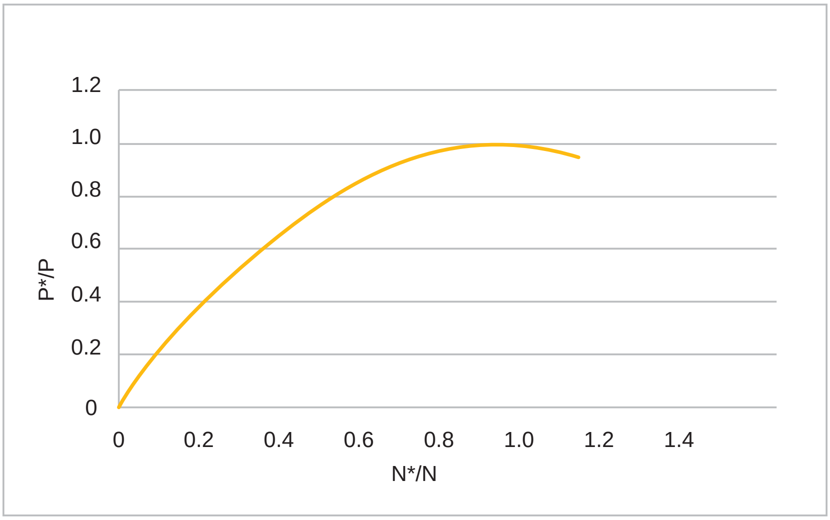


Figure 7-16. *Resulting Speed/Power Characteristics for a Turbine Stage (assuming a constant actual inflow)*

Off optimum speed of the power turbine reduces the efficiency and the ability to extract head from the flow. Even if N_{GG} (and the fuel flow) do not change, the amount of power that is produced by the PT is reduced. Also, because of the unchanged fuel flow, the engine heat rate increases and the exhaust temperature increases accordingly. Theoretically, any engine would reach its maximum exhaust temperature at high ambient, full load and locked PT.

Another interesting result of the above is the torque behavior of the power turbine *(Figure 7-17)*, considering that torque is power divided by speed :

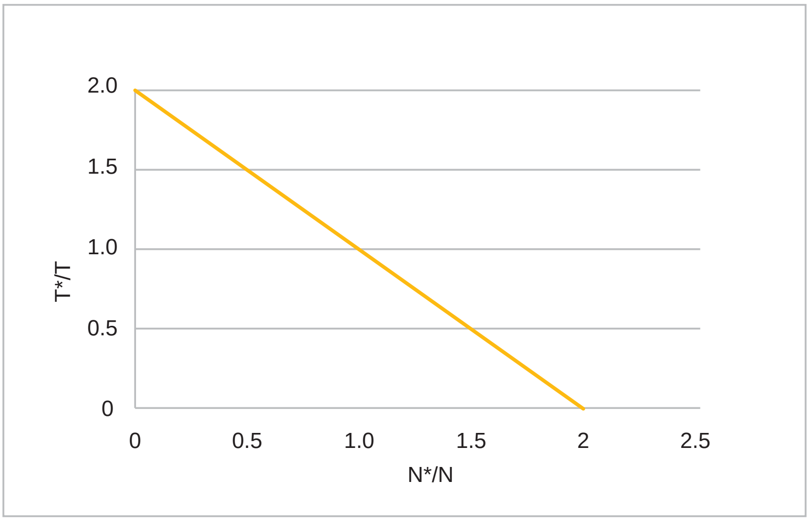


Figure 7-17. *Resulting Speed/Torque Characteristic for a Turbine Stage*

$$\frac{\tau}{\tau_{opt}} = 2 - \left(\frac{N_{PT}}{N_{PT,opt}} \right)$$

The torque is thus a linear function of the speed, with the maximum torque at the lowest speed This explains one of the great attractions of a free power turbine: To provide the necessary torque to start the driven equipment is usually not difficult (compared to electric motor drives or reciprocating engines) because the highest torque is already available at low speeds of the power turbine. As stated earlier, the optimum power turbine speed is not a constant, but changes with ambient temperature and load *(Figure 7-18)*.

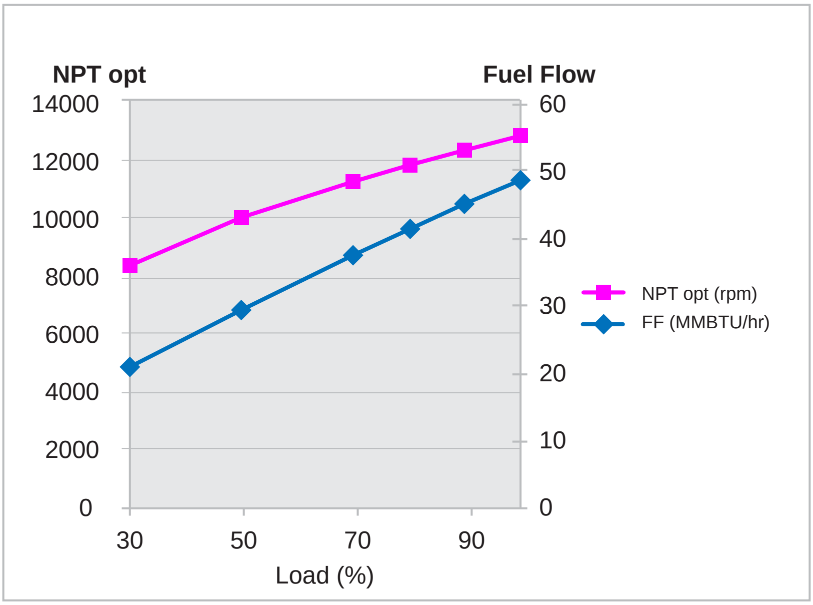


Figure 7-18. *Fuel Flow and Optimum Power Turbine Speed as a Function of Load*

Emission Control

All emission control technologies that use lean-premix combustion require a precise management of the fuel-to-air ratio in the primary zone of the combustor (i.e., where the initial combustion takes place) as well as the a precise distribution of combustor liner cooling and dilution flows. Deviations in these areas can lead to increased NO_x production, higher CO or UHC levels, or flame-out (Greenwood, 2000).

Lean-premix combustion achieves reduction in NO_x emissions by lowering the flame temperature. The flame temperature is determined by the fuel-to-air ratio in the combustion zone. A stoichiometric fuel-to-air ratio (such as in conventional combustors) leads to high flame temperatures, while a lean fuel-to-air ratio can lower the flame temperature significantly. However, a lean fuel-to-air mixture also means, that the combustor is operating close to the lean flame-out limit.

Any part load operation will cause reduction of fuel-to-air ratio, because the reduction in air flow is smaller than the reduction in fuel flow.

Several different approaches to control the fuel-to-air ratio are possible to avoid flame-out at part load or transient situations, for example:

- bleeding air overboard
- using variable inlet guide vanes
- managing the ratio between fuel burned in lean-premix mode and in a diffusion flame to name a few.

Obviously, all these approaches can have an effect on the part load performance characteristics of the gas turbine.

Matching the Gas Producer and the Power Turbine (two-shaft only)

Depending on

- the ambient temperature
- the accessory load
- the engine geometry (in particular the first power turbine nozzle)

the engine will reach one of the two limits first. At ambient temperatures below the match temperature, the engine will be operating at its maximum gas generator speed, but below its maximum firing temperature (speed topping). At ambient temperatures above the match temperature, the engine will operate at its maximum firing temperature, but not at its maximum gas generator speed (temperature topping). The match temperature is thus the ambient temperature at which the engine reaches both limits at the same time *(Figure 7-19)*.

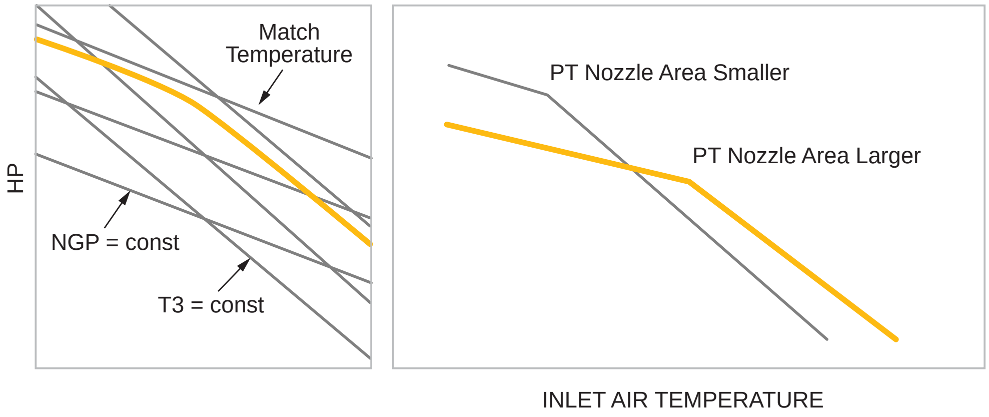


Figure 7-19. *Matching*

Because the first power turbine nozzle determines the amount of pressure ratio needed by the power turbine to allow a certain gas flow it also determines the available pressure ratio for the gas generator turbine. If the pressure ratio available for the gas generator does not allow to balance the power requirement of the engine compressor (see enthalpy-entropy diagram), the gas generator will have to slow down, thus reducing the gas flow through the power turbine. This will reduce the pressure ratio necessary over the power turbine, thus leaving more head for the gas generator to satisfy the compressor power requirements.

Some effects can cause the gas turbine to exhibit an altered match temperature:

Gas fuel with a low heating value, or water injection increase the mass flow through the turbine relative to the compressor mass flow. The temperature topping will thus be shifted to higher ambient temperatures. Dual-fuel engines, that are matched on gas will top early on liquid fuel. This is caused by the change in the thermodynamic properties of the combustion product due to the different Carbon to Hydrogen ratio of the fuels. The matching equations indicate that a reduction in compressor efficiency (due to fouling, inlet distortions) or turbine efficiency (increased tip clearance, excessive internal leaks, corrosion) will also cause early topping. Accessory loads also have the effect of leading to earlier topping.

Single-shaft Versus Two-shaft Engines

The choice of whether to use a single-shaft or two-shaft power plant is largely determined by the characteristics of the driven load. If the load speed is constant, as in the case of an electric generator, a single-shaft unit is often specified; an engine specifically designed for electric power generation would make use of a single-shaft configuration. An alternative, however, is the use of a two-shaft engine. If the load needs to be driven with varying speeds (compressors, pumps), two-shaft engines are advantageous.

The two types have different characteristics regarding the supply of exhaust heat to a cogeneration or combined cycle plant, primarily due to the differences in exhaust flow as load is reduced; the essentially constant air flow and compressor power in a single-shaft unit results in a larger decrease of exhaust temperature for a given reduction in power, which might necessitate the burning of supplementary fuel in the waste heat boiler under operating conditions where it would be unnecessary with a two-shaft. In both cases, the exhaust temperature may be increased by the use of variable-inlet guide vanes. Cogeneration systems have been successfully built using both single-shaft and two-shaft units.

The torque characteristics, are very different and the variation of torque with output speed at a given power may well determine the engine's suitability for certain applications. The compressor of a single-shaft engine is constrained to turn at some multiple of the load speed, fixed by the transmission gear ratio, so that a reduction in load speed implies a reduction in compressor speed. This results in a reduction in mass flow, hence of output of torque. This type of turbine is only of limited use for mechanical drive purposes. The two-shaft unit, having a free power turbine, however, has a very favorable torque characteristic. For a constant fuel flow, and constant gas generator speed, the free-power turbine can provide relatively constant power for a wide speed range. This is due to the fact that the compressor can supply an essentially constant flow at a given compressor speed, irrespective of the free turbine speed. Also, at fixed gas generator operating conditions, reduction in output speed results in an increase in torque. It is quite possible to obtain a stall torque of twice the torque delivered at full speed.

Starting

When a gas turbine is started, a purge cycle is initiated first, to avoid the presence of combustible gas anywhere in the engine or the exhaust system. To accomplish this, the starter motor will rotate the engine gas producer at low speeds for a preset amount of time. The starting process is outlined in *Figure 7-20*: Following the purge cycle, the starter accelerates the gas producer, fuel gas is injected and ignited by a torch (Light-Off). The starter continues to accelerate the gas producer to a point where the gas producer can operate self sustaining, and operates at idle speed. Along this path, compressor surge, as well as excessive combustor temperatures have to be avoided. Once the machine operates at idle, it is ready to be loaded, that is, the fuel flow can be increased, to bring the engine to full load. The maximum acceleration rate is limited by the maximum allowed firing temperature. For a single-shaft engine, the starter motor has to accelerate the gas turbine and the driven load. For a two-shaft engine, the starter motor only has to accelerate the gas producer section of the gas turbine.

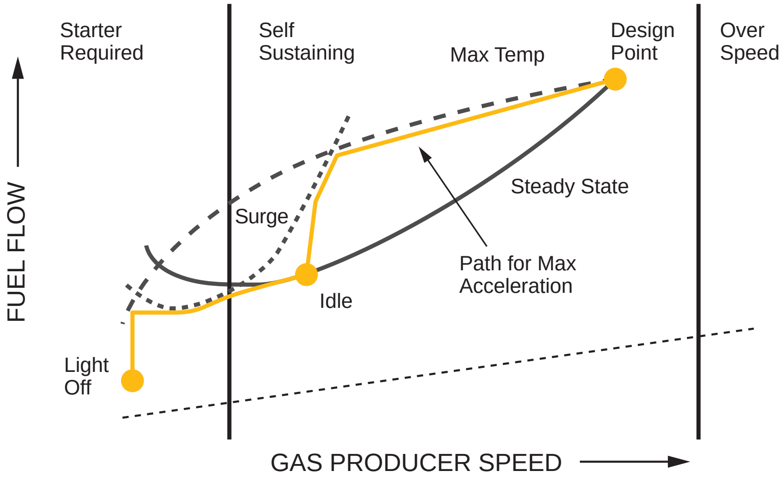


Figure 7-20. *Starting of a Gas Turbine and Acceleration Limits*

Gas Turbine Maps

Figures 7-21 and 7-22a-b show operating maps for single and two-shaft gas turbines. They describe the behavior of power, fuel flow, exhaust flow and temperature, compressor discharge pressure and, for two-shaft machines, gas producer speed, optimum power

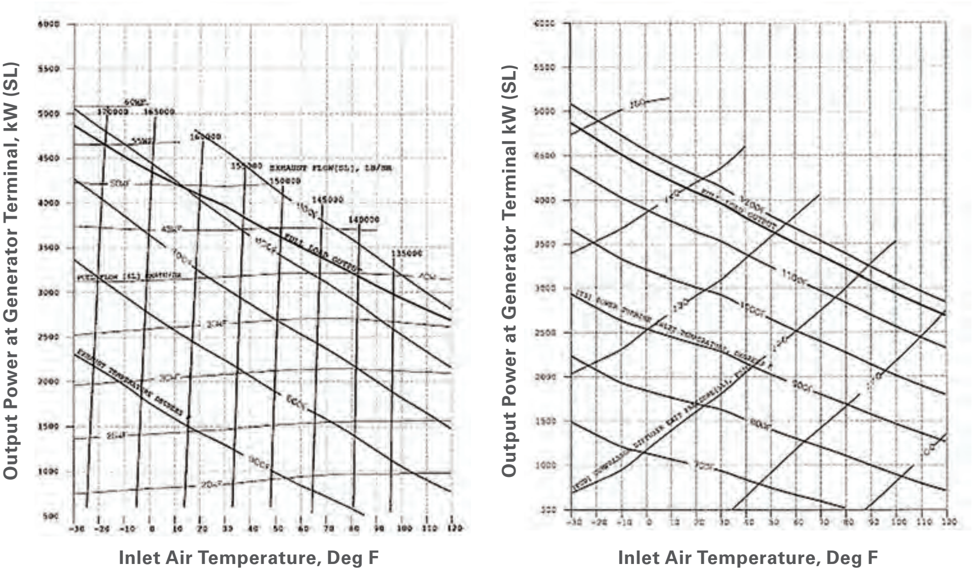


Figure 7-21. Maps for Typical Single-shaft Gas Turbines.

turbine speed, and power turbine output versus power turbine speed curves, all, as a function of ambient temperature. Some of the component behavior described above can be identified in these maps.

For example, in the single-shaft engine maps *(Figure 7-18)*, the exhaust flow is almost vertical, indicating the operating lines of the compressor displayed in *Figure 7-3*.

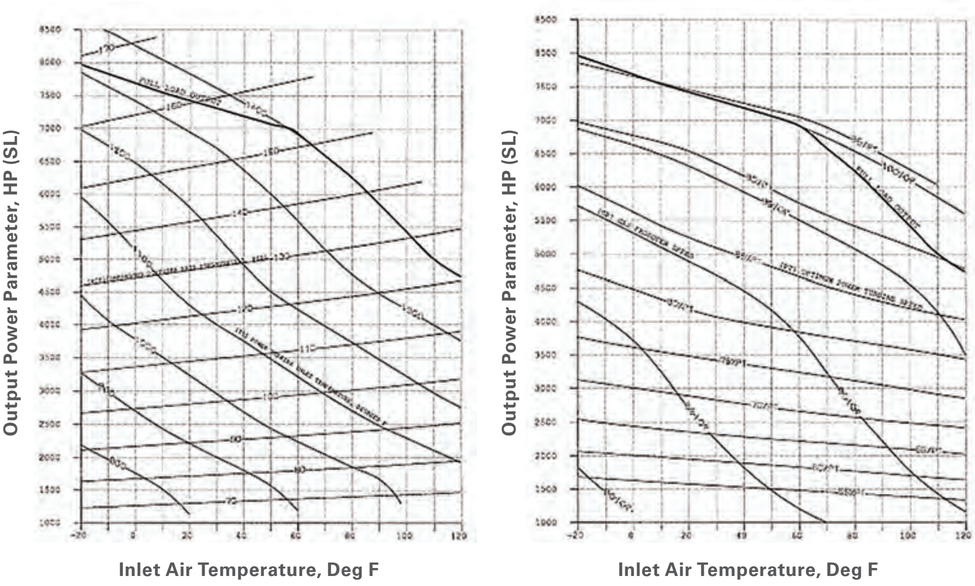


Figure 7-22a. Typical Maps for Two-shaft Gas Turbines

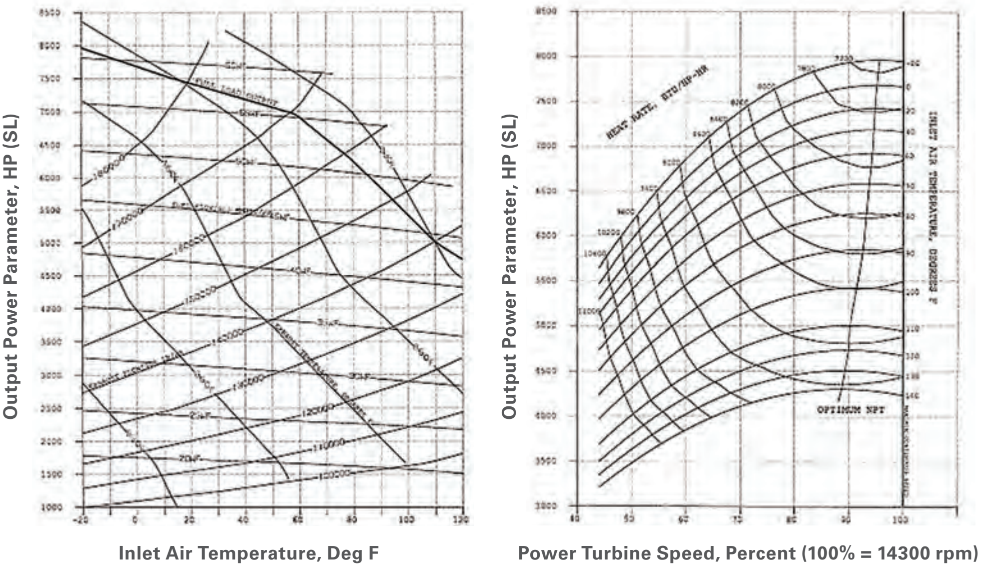


Figure 7-22b. *Typical Maps for two-shaft gas turbines*

In the two-shaft engine, we see the shape of the power turbine curves, as expected from the discussion leading to *Figure 7-16*. The optimum power turbine speed is indeed load and ambient temperature dependent. We can also see that the engine is matched at 59°F, because the full load power line follows a line of constant firing temperature at high ambient temperatures, and a line of maximum gas producer speed at low ambient temperatures.

APPENDIX A: GAS TURBINE CYCLE CALCULATION

A gas turbine may be designed for the following parameters for the compressor:

p_a=14.73 psia (1.013bara) p_2=147.3 psia (10.13bara) W= 100lbs/s (45 kg/s) η_c=85%
T_a=100F(37.8°C) = 560R (311K)

and the turbine (neglecting the fuel mass flow and the combustor pressure drop)

p_a=14.73psia (1.013bara) p_3=147.3 psia (10.13bara) W= 100lbs/s (45 kg/s) η_c =85%
T_3=1600F(870°C) = 2060R (1144K)

We use the relationships for work H and power P:

$$H = c_p \Delta T$$
$$P = W \cdot H$$

and the gas properties for air: c_p=0.24BTU/lbR (1.007kJ/kgK) ; γ=1.4 (this is a simplified assumption, because the gas properties of the exhaust gas are somewhat different from air).

The compressor temperature rise is

$$T_2 - T_1 = \frac{T_1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 613R \qquad (341K)$$

and the compressor discharge temperature is therefore

560R + 613R = 713°F (379°C).

This indicates, that the compressor consumed the work

H = .24 x 613 = 147 BTU/lb (= 1.007 x 341 = 344kJ/kg),

and the power

P= 100 lbs/s x 147BTU/lb=14700BTU/s= 20800hp (= 45 kg/s x 344kJ/kg =15480 kJ/s = 15480 kW)

The power extraction of the turbine causes a temperature drop of

$$T_3 - T_7 = \eta_t \cdot T_3\left[1 - \left(\frac{p_7}{p_3}\right)^{\frac{\gamma-1}{\gamma}}\right] = 844R \qquad (469K)$$

and thus an exhaust temperature of

T_7=1600-844= 756°F (=1144-469= 402°C)

Thereby extracting the work

H =.24 x 844 = 202 BTU/lb (=1.007 x 469 = 472 kJ/kg),

producing a power of

P=100 lbs/s x202BTU/lb=20200BTU/s = 28583hp (=45 kg/s x 472kJ/kg =21240 kJ/s = 21240 kW)

The net engine output is the difference between the power produced by the turbine and absorbed by the compressor:

P_{net}= 28583hp-20800hp = 7783hp (= 21240kW-15480kW = 5760kW)

With a compressor exit temperature of 713°F (379°C) and a turbine inlet temperature of 1600F (870°C), we need to add heat to bring the gas from 713°F (379°C) to 1600F (870°C):

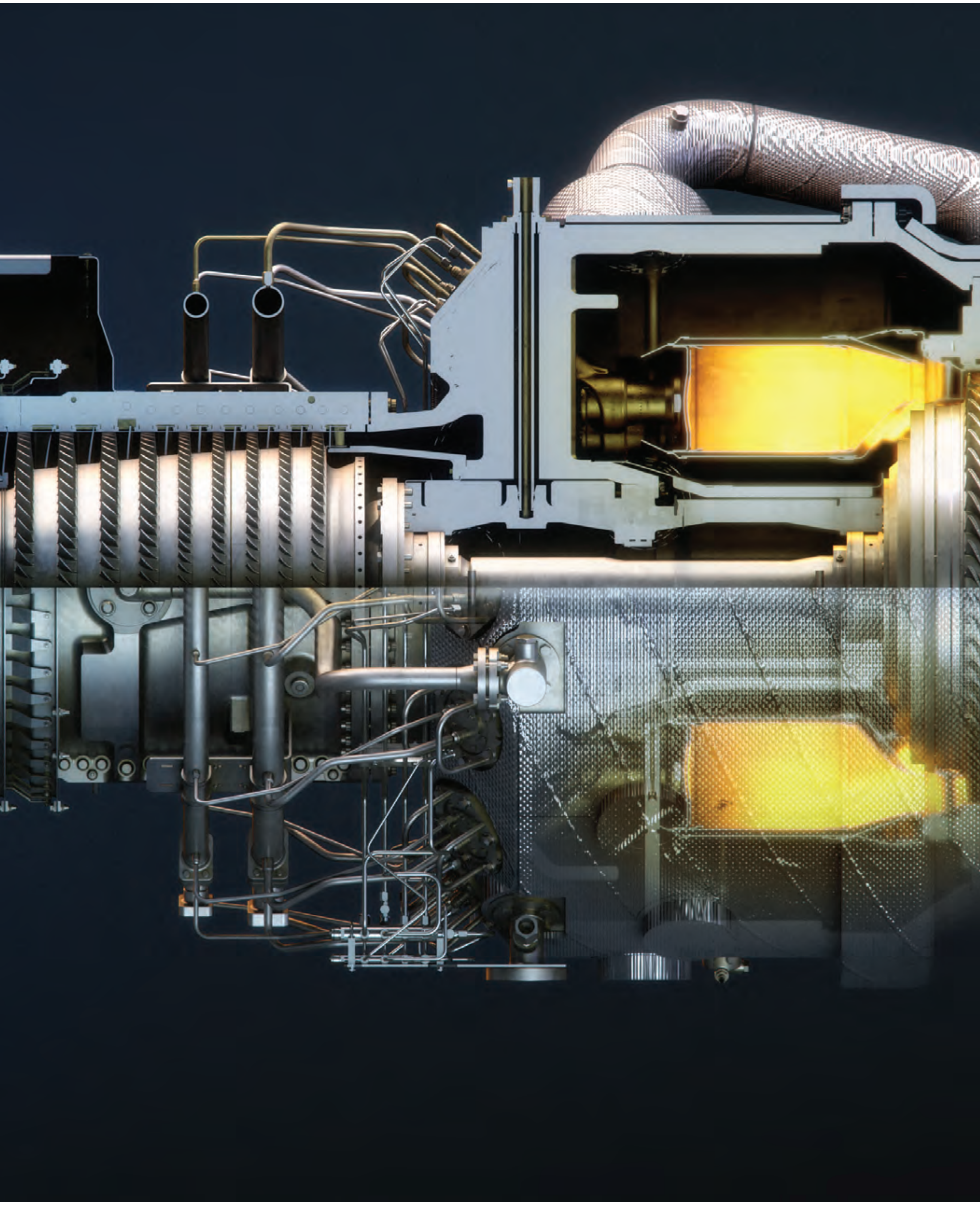
Q = W cp ΔT = 100 x 0.24 x (1600-713) = 21300BTU/s = 76.7MMBTU/hr (= 45 x 1.007 x (870-379) = 22250 kJ/s = 80.1 GJ/hr)

With this, the engine heat rate can be calculated from

HR = 76.7/ 7783 = 9850 BTU/hphr (= 80.1/5760 = 13900 kJ/kWh),

and the thermal efficiency is

η_{th} =5760kW/22250kJ/s=25.9%.



CHAPTER 8

COMBUSTION & EMISSIONS

The engine combustor is the place where fuel is injected (through fuel injectors) into the air previously compressed in the engine compressor. The released fuel energy causes the temperature to rise:

$$\tilde{c}_{pe}(T_3 - T_2) = E_f W_f / W$$

The heat capacity $\tilde{c}_{pe}$ in the equation above is a suitable average heat capacity. In practice, both the large change in temperature and the change in gas composition due to the combustion products have to be considered.

Modern combustors convert the energy stored in the fuel almost completely into heat (Typical combustion efficiencies for natural gas burning engines range above 99.9%). This is also evident from the fact that the results of incomplete combustion, namely CO and unburned hydrocarbons, are emitted only in the parts per million level.

Only a part of the compressed air participates directly in the combustion, while the remaining air is later mixed into the gas stream for cooling purposes. The temperature profile in a typical combustor is shown in *Figure 8-1.*

The local temperatures are highest in the flame zone. Cooling of the combustor liner, and subsequent addition of air bring the gas temperature down to an acceptable combustor exit temperature. The flow will also incur a pressure loss due to friction and mixing.

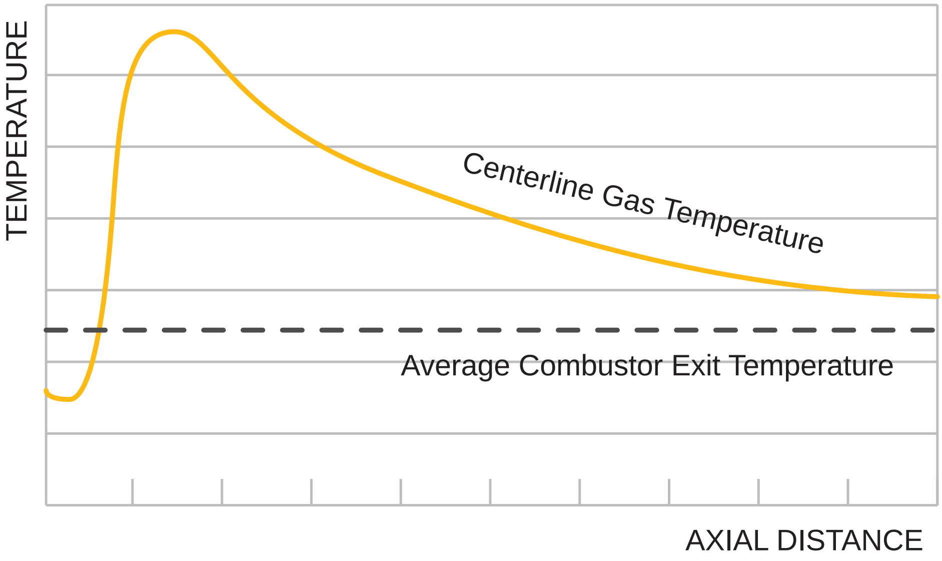


Figure 8-1. *Combustor Axial Temperature Distribution*

Most industrial gas turbines today use either conventional combustors where the fuel is injected into the combustor, and mixes with the air present in the combustor, to burn in a more or less stoichiometric flame , or lean-premix combustors, where the fuel is premixed with air in the injector before it enters the combustor *(Figure 8-2)*

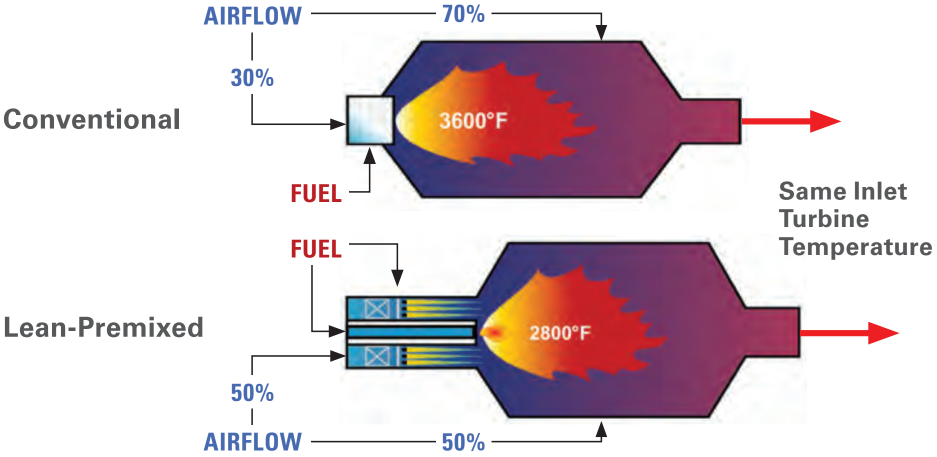


Figure 8-2. *Conventional and Lean-Premix Combustion Systems*

Unlike in reciprocating engines, the gas turbine combustion is continuous. This has the advantage that the combustion process can be made very efficient, with very low levels of products of incomplete combustion like carbon monoxide (CO) or unburned hydrocarbons (UHC). The other major emissions component, oxides of nitrogen (NO_x), is not related to combustion efficiency, but strictly to the temperature levels in the flame and the amount of nitrogen in the fuel. The solution to NO_x emissions, therefore, lies in the lowering the flame temperature. Initially, this was accomplished by injecting massive amounts of steam or water in the flame zone, thus 'cooling' the flame. This approach has significant drawbacks, not the least the requirement to provide large amounts (fuel-to-water ratios are approximately around 1) of extremely clean water. Since the 1990's, combustion technology has focused on systems often referred to as dry low NO_x combustion, or lean-premix combustion *(Figure 8-2)*.

CONVENTIONAL (DIFFUSION FLAME) COMBUSTORS

Diffusion-flame combustion, used in Solar's conventional combustor, is characterized by the fact that fuel and air are not homogeneously mixed prior to entering the combustion zone *(Figure 8-2 and 8-3)*. Fuel and air are separate entities in the initial stages of combustion; they come together in the reaction (or primary) zone through molecular and turbulent diffusion.

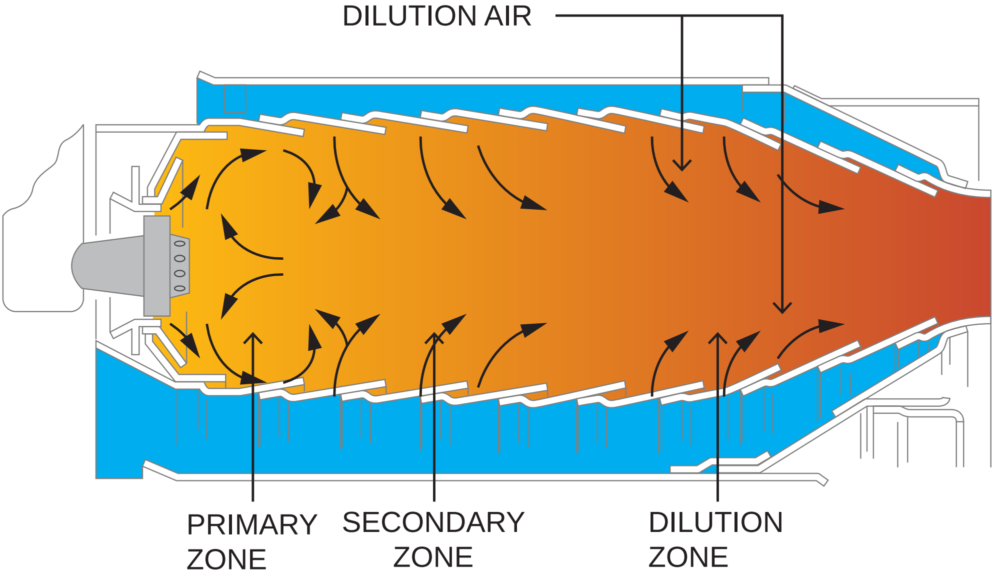


Figure 8-3. *Diffusion-Flame Combustor*

Combustion occurs over a range of local fuel-to-air ratios that may range from the lean limit to the stoichiometric mixture and further on to the rich limit. This leads to locally different flame temperatures and reaction rates. The burning rate is controlled by the mixing time, not the chemical reaction rate. The flame typically has a white, yellow or orange color.

LOW EMISSION COMBUSTION SYSTEMS

While several low emission combustion system designs are possible, two basic approaches are used. The first employs a "lean-premixed" primary zone and the second employs a rich primary zone followed by rapid quenching and subsequent dilution of the reacting gases to the required turbine inlet temperature. The second approach is known as a "rich-lean" combustion system. In both approaches, the local and average temperatures in the primary zone are controlled to levels considerably less than the level corresponding to the stoichiometric fuel/air ratio in order to minimize formation of NO_x.

The lean-premixed approach, also known as dry low NO_x, or SoLoNox, is currently the most widely used method to reduce emissions *(Figures 8-2 and 8-4)*. Unlike the diffusion flame, where the local fuel-to-air ratio varies strongly, the idea behind lean-premixed combustion is to create a uniform and lean fuel-to-air ratio in the primary zone. To achieve this, fuel and air are thoroughly mixed prior to entering the primary zone. Therefore, an overall low flame temperature can be achieved, which in turn reduces the formation of NO_x. The burning rate is controlled by the ability to transfer heat and mass from the flame to the unburned gas (and only to a small degree by the chemical reaction rate). The flame typically has a blue appearance.

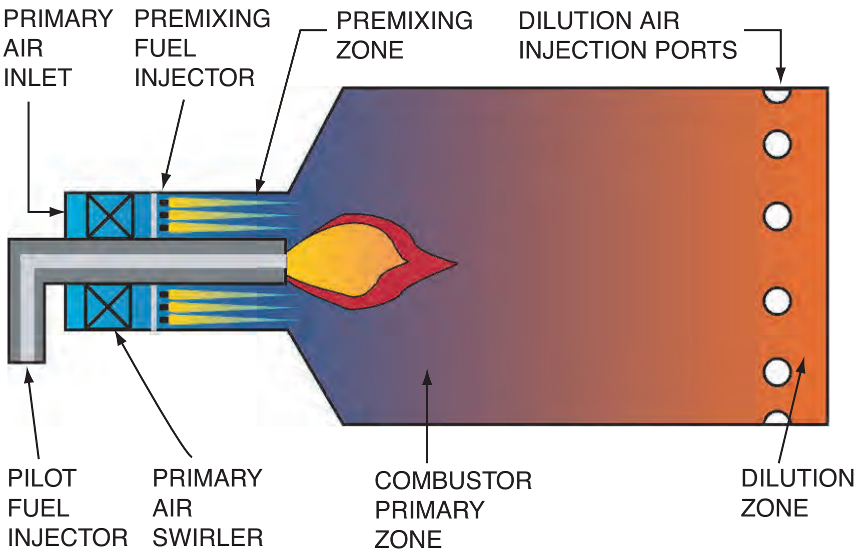

PRIMARY AIR INLET
PREMIXING FUEL INJECTOR
PREMIXING ZONE
DILUTION AIR INJECTION PORTS
PILOT FUEL INJECTOR
PRIMARY AIR SWIRLER
COMBUSTOR PRIMARY ZONE
DILUTION ZONE

Figure 8-4. *Lean-Premixed Combustor Concept*

The general idea behind any Lean-Premix combustor currently in service is to generate a thoroughly mixed lean fuel and air mixture prior to entering the combustor of the gas turbine (Greenwood, 2000). The lean mixture is responsible for a low flame temperature *(Figure 8-5)*, which in turn yields lower rates of NO_x production *(Figure 8-6)*. Because the mixture is very lean, in fact fairly close to the lean extinction limit, the fuel-to-air ratio has to be kept constant within fairly narrow limits. This is also necessary due to another constraint: The lower combustion temperatures tend to lead to a higher amount of products related to incomplete combustion, such as CO and unburned hydrocarbons (UHC). The dry low NO_x combustion system uses a fuel/air mixing zone where fuel is injected into the airstream to achieve complete vaporization (for liquid fuel) and uniform mixing of fuel with air prior to combustion. The fuel/air ratio in this zone at the design point is controlled to be significantly less than stoichiometric *(Figure 8-4)*. The premixed gases enter the primary zone of the combustor where they are allowed to react by providing a recirculation zone to stabilize the combustion process. The gases from the primary zone enter the dilution zone where they are mixed with dilution air to meet gas turbine inlet temperature requirements.

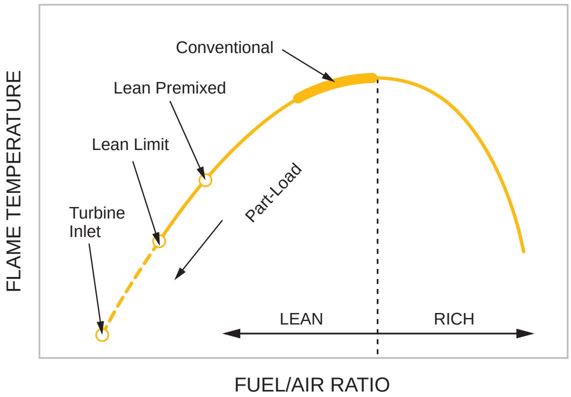


Figure 8-5. *Flame Temperature and Fuel-to-Air Ratio*

The necessity to control the fuel-to-air ratio closely yields different part-load behavior when comparing gas turbines with conventional combustors and DLN engines.[1] While for conventional combustion systems, the firing temperature drops when the engine is operated at reduced load, in a lean-premix combustor the firing temperature has to be kept high to avoid combustion inefficiency and increased CO and UHC levels. To accomplish this, the air flow into the combustor has to be reduced at part load. In single-shaft engines, the task can be accomplished by modulating the compressor inlet guide vanes, while two-shaft engines usually bleed air from the compressor directly into the exhaust. Therefore, while the airflow for any two-shaft engines is reduced at part load, the reduction in airflow is greater for a conventional combustion engine than for a DLN engine. This sounds paradoxical because the amount of air available at the combustor in part-load operation has to be less for a DLN engine (to maintain the fuel-to-air ratio) than for an engine with conventional combustion. However, due to the bleeding of air in a DLN engine, the flow capacity of the turbine section is artificially increased by the bleeding duct. The combustor exit temperature at part load drops significantly for engines with conventional combustion, while it stays high for DLN engines. Once the bleed valve opens, the part-load efficiency of a DLN engine drops faster than for an engine with conventional combustion. Since the opening of the bleed valve is driven by emissions considerations, it is not directly influenced by the load. Regarding emissions, the drop in combustor temperature in engines with conventional combustion, leading to a leaner fuel-to-air ratio, automatically leads to NO_x emissions that are lower at part load than at full load. In DLN engines, there is virtually no such reduction because the requirement to limit CO and UHC emissions limits the (theoretically possible) reduction in fuel-to-air ratio. However, the NO_x emissions levels of DLN engines are always lower than for engines with conventional combustion.

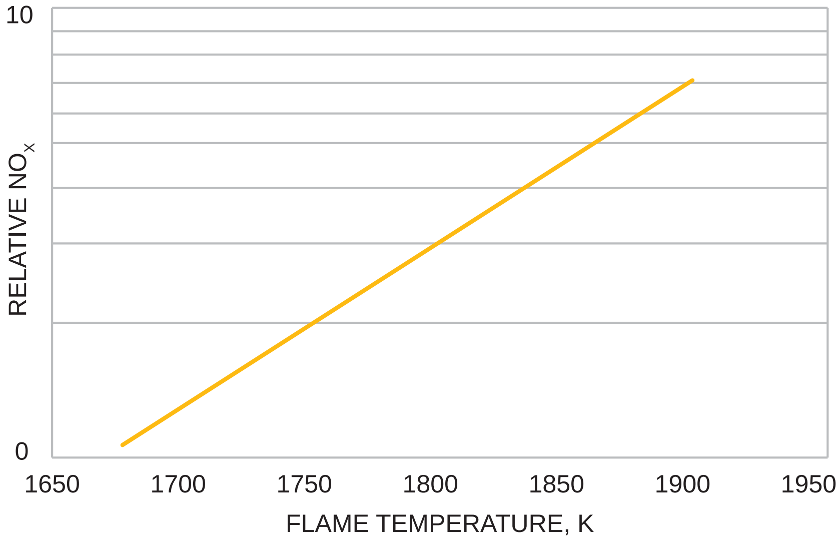


Figure 8-6. *Effect of Flame Temperature on* NO_x *Emissions*

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[1] Regarding the requirements for DLN engines, multi-spool engines show no fundamental differences from single-spool engines.

The impact of design considerations on NO_x emissions needs to be considered: Fortunately, there is no evidence that pressure ratio influences NO_x production rate (on a ppm basis) in DLN systems. There might be some compromises necessary for engines with high firing temperatures regarding the cooling air usage. But this is a secondary effect, because the combustor exit temperature and the flame temperature are not directly related. Some aeroderivative engines, which tend to have high pressure ratios, have space limits for the combustion system, thus might be at a disadvantage. But this is not primarily due to the high pressure ratio, but rather due to the specific design of the engine. A limiting factor for lowering NO_x emissions is often driven by the onset of combustor oscillations. Again, there is no evidence that the operating windows that allow operation without oscillations are influenced by operating pressure or firing temperature. They rather seem to depend far more on the specific engine design.

Combustion systems using this approach generally do not have a secondary zone since all the air that participates in the combustion is directed through the injector. Furthermore, the temperature of the primary zone gases is low enough to prevent any significant dissociation of CO_2 to CO. The temperature in the primary zone of a lean-premixed combustor is uniform and significantly lower than the average temperature in the primary zone of a conventional combustor. The lower temperature in turn leads to reduced NO_x levels *(Figure 8-6).*

Due to the low equivalence ratio and uniformity of the fuel/air ratio in the primary zone, such a combustion system operates close to its weak flame extinction limit (equivalence ratio is defined as the fuel/air ratio divided by the stoichiometric fuel/air ratio). Therefore, a very accurate control of the primary zone fuel/air ratio is required to prevent flame-outs during off-load transients and for part-load operation.

To accommodate part-load operation, the fuel-to-air ratio in the primary zone can be maintained in an acceptable range either by adjusting the airflow by means of adjustable IGVs, air bleed, variable air management systems, adjustable power turbine nozzles or through staged combustion.

Operational Constraints

Due to operation relatively close to the lean limit, dry low NO_x combustion systems face certain operational constraints:

- **Stability.** The locus of points in the flame region needs to exhibit a burning velocity equal to the flow velocity; otherwise, as shown in *Figure 8-7*, the result is flame-out.
- **Dynamics.** The energy release in the flame may amplify or dampen pressure waves. Combustor oscillations are a result of "resonance" between heat and pressure waves.
- **NO_x emissions.** Nitrogen oxides form in the hot regions of the combustor.
- **CO and UHC emissions.** Carbon monoxide and unburned hydrocarbons result from local cold spots (typically at over-cooled combustor liners). DLN systems typically exhibit temperatures too low to force dissociation of CO_2.

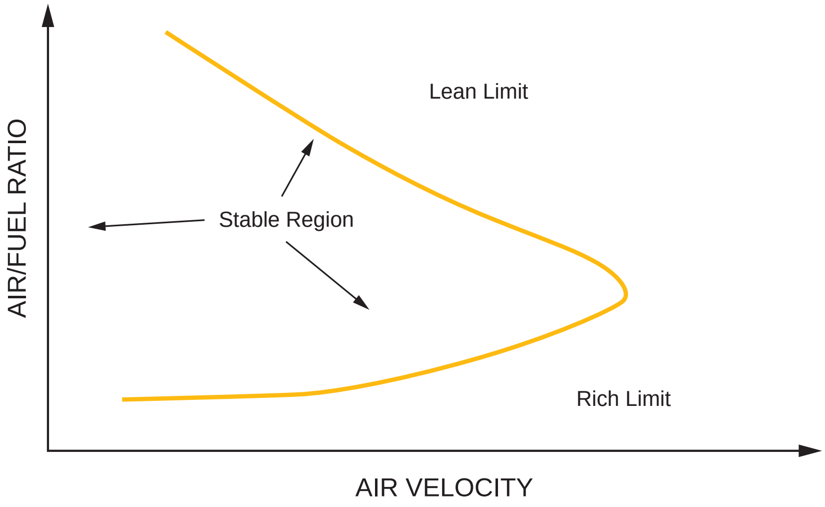


Figure 8-7. *Combustion Stability Limits*

For any combustion chamber, there are both rich and lean limits to the fuel/air ratio beyond which the flame is unstable. Aside from the chemical composition of the fuel, these limits depend on the air velocity in the combustor *(Figure 8-7)*. The velocity of the air leaving the compressor has to be reduced in a diffuser. This is required to assure stable combustion, as well as to reduce the combustor pressure loss.

It must be noted that these stability limits not only have to be met at all loads, but in particular during acceleration and deceleration of the engine.

To accelerate the engine, the fuel flow will increase rapidly, while, due to the inertia of the gas producer (in a two-shaft engine), the airflow will lag behind, thus creating a very rich mixture. Control systems, therefore, usually limit the rate of change of fuel flow, not only to avoid flame-out, but also to avoid high transient temperatures in the turbine.

The reverse situation occurs during deceleration. The fuel flow is reduced very quickly, while the airflow remains high for a short period of time, thus creating a very lean mixture at the continuously high air velocity of the combustor.

COMBUSTOR OSCILLATIONS

An unfortunate side effect of lean-premixed combustion, is the fact that the heat release and local pressures can be the basis of a system capable of oscillations. In gas turbines that use lean-premix-combustion for emissions reduction, a common and key feature is that fuel and air are premixed prior to entering the combustor. This is necessary to control the local fuel-to-air ratio in the combustor to a lean mixture to maintain a reduced flame temperature, and consequently low production of NO_x.

If the fuel nozzles or air nozzles are not choked, the flow rate through these nozzles is dependent on fluctuations of the pressure and temperature field in the combustor. Any pressure disturbance in the combustor will, after a delay determined by the speed of sound along its path, cause a pressure disturbance immediately downstream of the fuel nozzle (or air nozzle). If the fuel (and/or air) flow is modulated as a result of these fluctuations in the combustor, the local fuel-to-air ratio in the injector, and subsequently in the combustor, will fluctuate. There is, thus, a time delay between the time when the fuel or air enters the mixing zone in the injector and the time when it burns in the combustor. In this case, this time is determined by flow velocities in the injector and combustor. Because the local flame temperature is also determined by the local fuel-to-air ratio, the fluctuations in the local fuel-to-air ratio lead to fluctuations in the pressure and temperature field in the combustor. This is a feedback mechanism that can cause measurable pressure fluctuations if the time constants involved excite gas in the combustor at one of its natural frequency modes. These can cause resonant oscillations strong enough to initiate mechanical damage to the engine and, thus, every effort is made to prevent them. One mechanism of these oscillations becomes clear from *Figures 8-8 and 8-9*. The oscillation modes can be axial, radial or circumferential depending on the combustor geometry *(Figure 8-9)*.

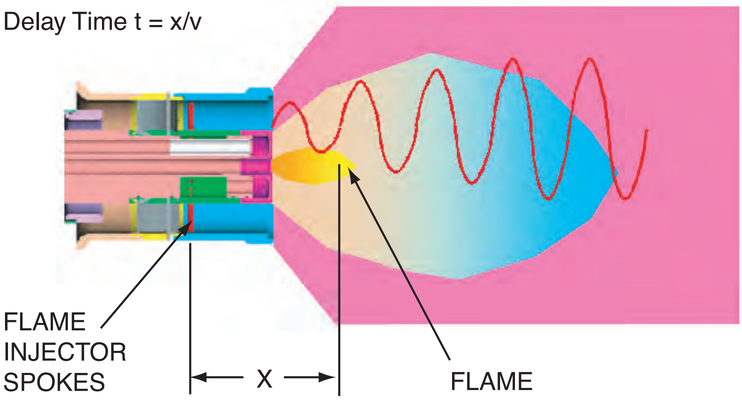


Figure 8-8. *Combustor Oscillations: Delay Time and Rayleigh Coupling. There is a time lag between the release of a fuel particle at the fuel spokes and the actual combustion of this fuel particle in the primary zone*

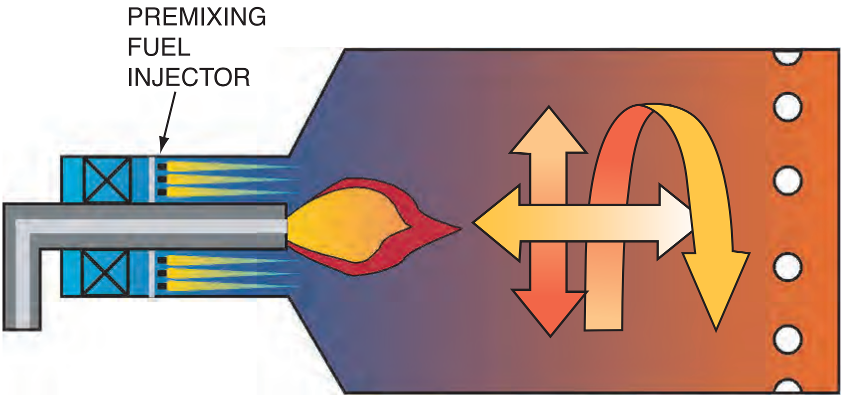


Figure 8-9. *Combustor Oscillation Modes*

GAS TURBINE EMISSIONS

The combustion process in a gas turbine is essentially a clean and efficient process and for many years there was little concern about emissions, with the exception of the need to eliminate smoke from the exhaust when burning liquid fuels. Recently, however, control of emissions has become probably the most important factor in the design of industrial gas turbines, as the causes and effects of industrial pollution become better understood and the population of gas turbines increases.

The pollutants appearing in the exhaust include oxides of nitrogen (NO and NO_2), carbon monoxide (CO) and unburned hydrocarbons (UHC); any sulphur in the fuel results in oxides of sulphur (SO_x), the most common of which is SO_2. Although these all represent a very small proportion of the exhaust flow (typically in the 5 to 300 ppm range), the large flow of exhaust gases produces significant quantities of pollutants that can become concentrated in the area close to the emission source. Nitrogen oxides can react in the presence of sunlight to produce "smog," which can be seen as a brownish cloud. This problem was originally identified in the Los Angeles area, where the combination of vehicle exhausts, strong sunlight, local geography and temperature inversions resulted in severe smog.

The problem of controlling emissions is complicated by the fact that gas turbines may be operated over a wide range of power and ambient conditions. As a simple example, power changes on a single-shaft gas turbine driving a generator will occur with the compressor operating at constant speed and approximately constant airflow, while an engine with a free-power turbine must operate at different compressor speeds and, hence, airflows as the power setting is changed. Variable geometry compressor systems introduce further variables, but also introduce the means of controlling the combustion process.

In the past, the only function of the engine control system was to provide the correct amount of fuel for all operating conditions, both steady-state and transient while emissions control was a function of the combustor design. With modern engines, however, the control system plays a major role in adjusting fuel/air ratios to minimize emissions over a large portion of the operating range. This was only possible with the advent of sophisticated digital fuel control systems.

POLLUTANT FORMATION

The initial goal of combustion engineers was the design and development of high efficiency combustors that were rugged and durable. Later, relatively simple solutions to the problem of smoke were developed. When the requirements for emissions control emerged, much basic research became necessary to establish the fundamentals of pollutant formation.

Regarding the generation of pollutants, there are different classes :

- Pollutants generated due to the high temperature environment: NO_x
- Pollutants resulting from incomplete combustion: CO, UHC
- Pollutants resulting from complete combustion of fuel contaminants: SO_x

- Pollutants that are carried through the engine, coming from air bound contaminants: Most particles.

The single most important factor affecting the formation of NO_x is flame temperature; it is theoretically at maximum at stoichiometric conditions and will fall off at both rich and lean mixtures *(Figure 8-10)*. Unfortunately, while NO_x can be reduced by operating well away from stoichiometric conditions, this results in an increasing formation of both CO and UHC. The rate of formation of NO_x, varies exponentially with the flame temperature, so the key to reducing NO_x is reduction of the flame temperature *(Figure 8-6)*. The formation of NO_x is also slightly dependent on the residence time of the fluid in the combustor, decreasing in a linear fashion as residence time is reduced. An increase in residence time, however, has a favorable effect on reducing both CO and UHC emissions, because it allows for the time necessary to complete these reactions. Increasing the residence time implies an increase in combustor cross-sectional area or volume. This is the reason low NO_x combustors are so much larger than conventional combustors *(Figure 8-11)*.

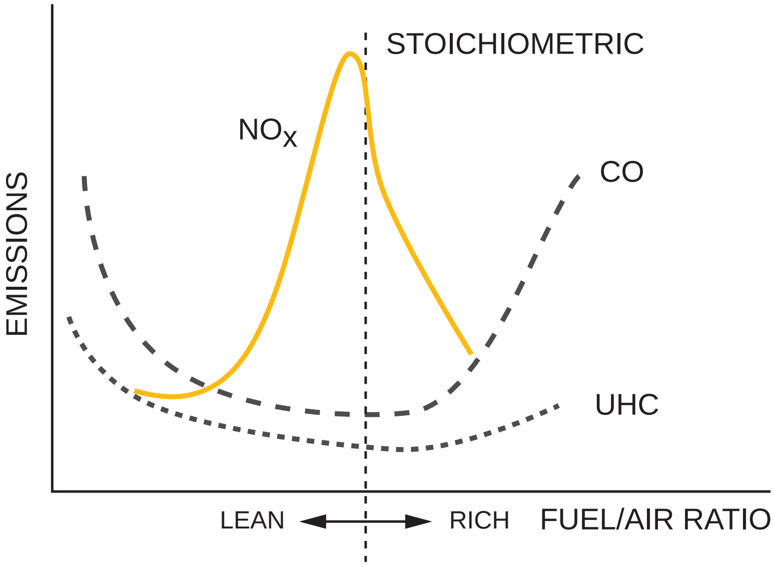


Figure 8-10. *Emissions as a Function of Fuel-to-Air Ratio*

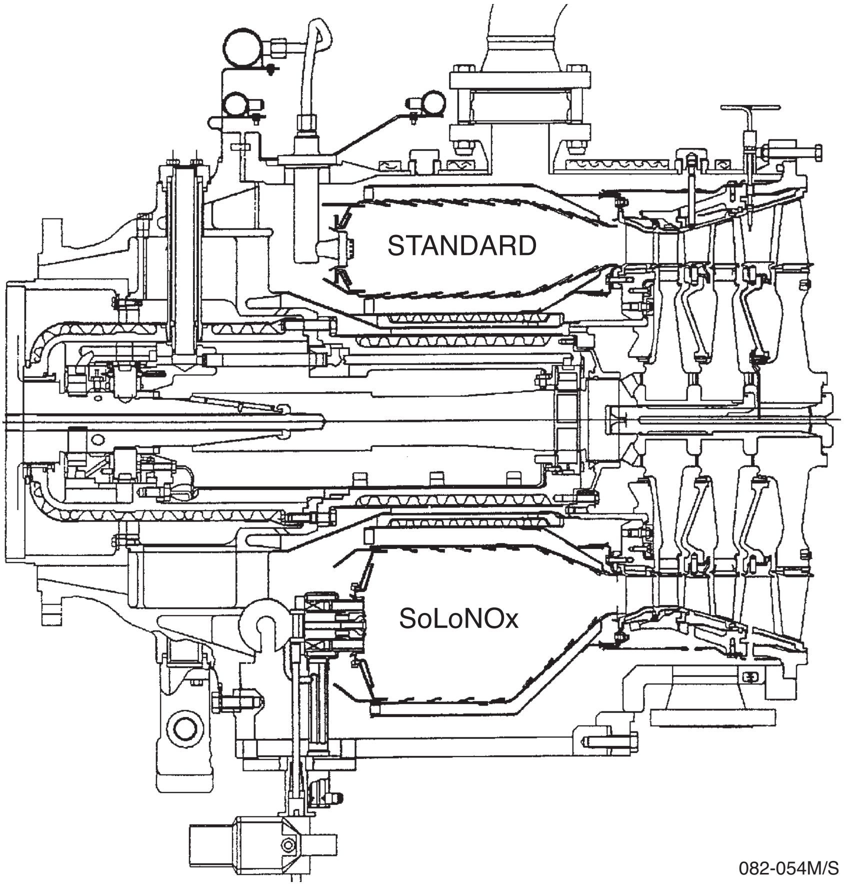


Figure 8-11. *Comparison of Dry Low* NO_x *and Standard Combustors*

It is important to understand the relationship between emissions and the key cycle parameters of pressure ratio and turbine inlet temperature. The Zeldovich reactions of formation of thermal nitrogen oxides are as follows:

$$N + O_2 \leftrightarrow NO + O$$
$$O + N_2 \leftrightarrow NO + N$$
$$N + OH \leftrightarrow NO + H$$

The initial rate of formation of NO_x is:

$$\frac{d(NO)}{dt} = C_1 T^{-1/2} \exp(-\frac{69090}{T})(O_2)^{1/2}(N_2)$$

The reaction rates become significant above 1650°C (3000°F) flame temperature. They roughly double in magnitude for each 20°C (36°F). Note that the adiabatic flame temperature in a combustor for natural gas is approximately 2000-2300°C (3500-4200°F), while dry low NO_x systems may operate with flame temperatures of about 1538°C

(2800°F). Since alternative fuels, such as hydrogen, have much higher flame temperatures (about 140°C or 250°F higher), they would tend to create higher NO_x levels than for natural gas.

Once combustion systems designers began to understand the problem and incorporated the necessary measures to minimize NO_x, it was found that cycle pressure ratio did not have a major effect: it is the flame temperature that is important. NO_x emissions can be more than halved by reducing the flame temperature from 1900°K to 1800°K.

This insight inspired the design of lean-premixed combustion systems that operate at a less than stoichiometric fuel-to-air ratio in the combustion zone, thus achieving a lower flame temperature.

Lean-premixed combustion is the basis of Solar's current pollution-prevention, dry low NO_x emission technology called SoLoNOx™.

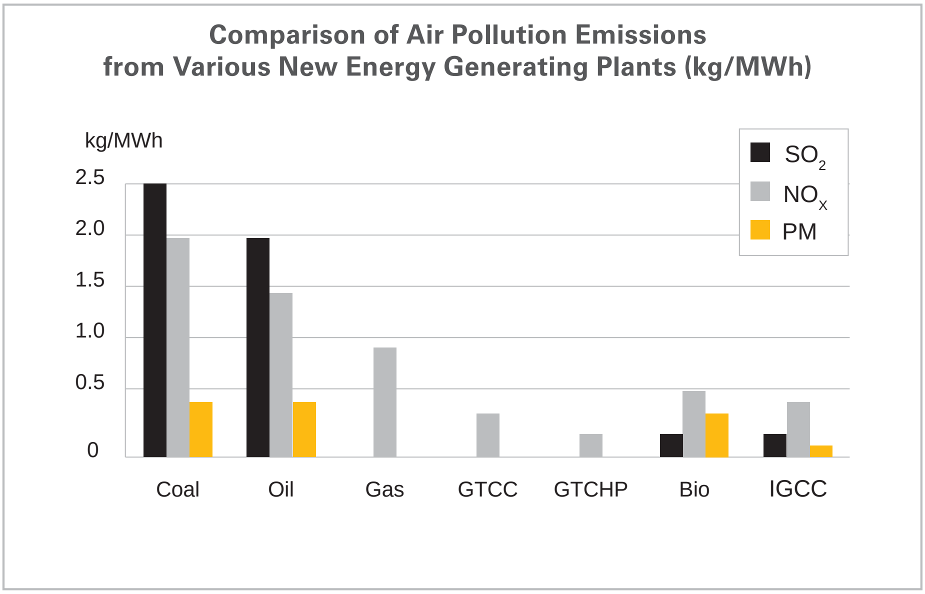
Successful development of ultra lean-premixed (ULP) combustion will further decrease emission levels by reducing the temperature in the primary zone of the combustor. ULP requires a fuel injector that provides very thorough mixing of the fuel and air. To obtain stable combustion in a lean-premixed system at part load, the fuel/air mixture ratio must be held nearly constant as gas turbine load (fuel flow rate) is reduced. This can be accomplished either with air bleed from the compressor at part load or via a variable-geometry system that reduces airflow to the combustor as fuel flow is reduced.

Also, carbon monoxide (CO) generation can be caused by the combustor liner film-cooling air passing through the liner and entering the reaction zone before the CO has oxidized to carbon dioxide (CO_2). One way to prevent this from occurring is to avoid internal film cooling. An alternative to film cooling is, for example, back-side cooling of the combustor liner. (Refer also to the discussion on cooling technologies in Chapter 6)

Comparison with other Fossil Fuel-Burning Alternatives

It may be preferable to compare emissions from various alternatives in terms of "mass of pollutant per quantity of useful product output", in this case on a kg/MWh basis (Klein and Kurz). *Figure 8-12* provides an approximate emissions comparison of typical modern coal, oil and gas-fired Rankine Cycle boiler systems, with GT Combined Cycle and Combined Heat & Power, Wood waste and Gasification plants (GTCC, GT CHP, Biomass, IGCC). Note that older coal and oil-fired plants, with less controls, have much higher emissions of acid gases than noted in the figure.

The use of natural gas in a high-efficiency cycle with a DLN based gas turbine results in much lower emissions of all acid gases, thus accomplishing goals consistent with a comprehensive pollution prevention strategy. About 60 percent of the energy value of natural gas is from its hydrogen content, thus CO_2 emissions from high-efficiency natural gas combustion are a fraction of those for coal-fired systems. Particulates, associated toxic trace elements, and thermal impacts from cooling water, are also minimized with gas plants. As shown in *Figure 8-12*, the gas turbine based facilities can reduce overall pollution from average coal and oil plants by 90-95 percent .The issue is then: Why constrain this cleaner energy choice by forcing regulatory NO_x concentration standards that are so low as to prevent effective plant operation.



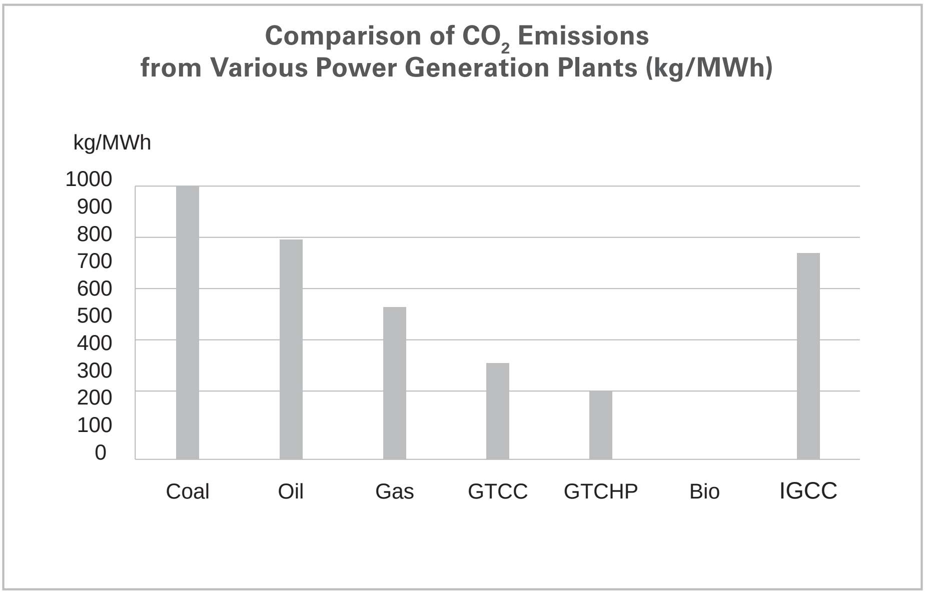
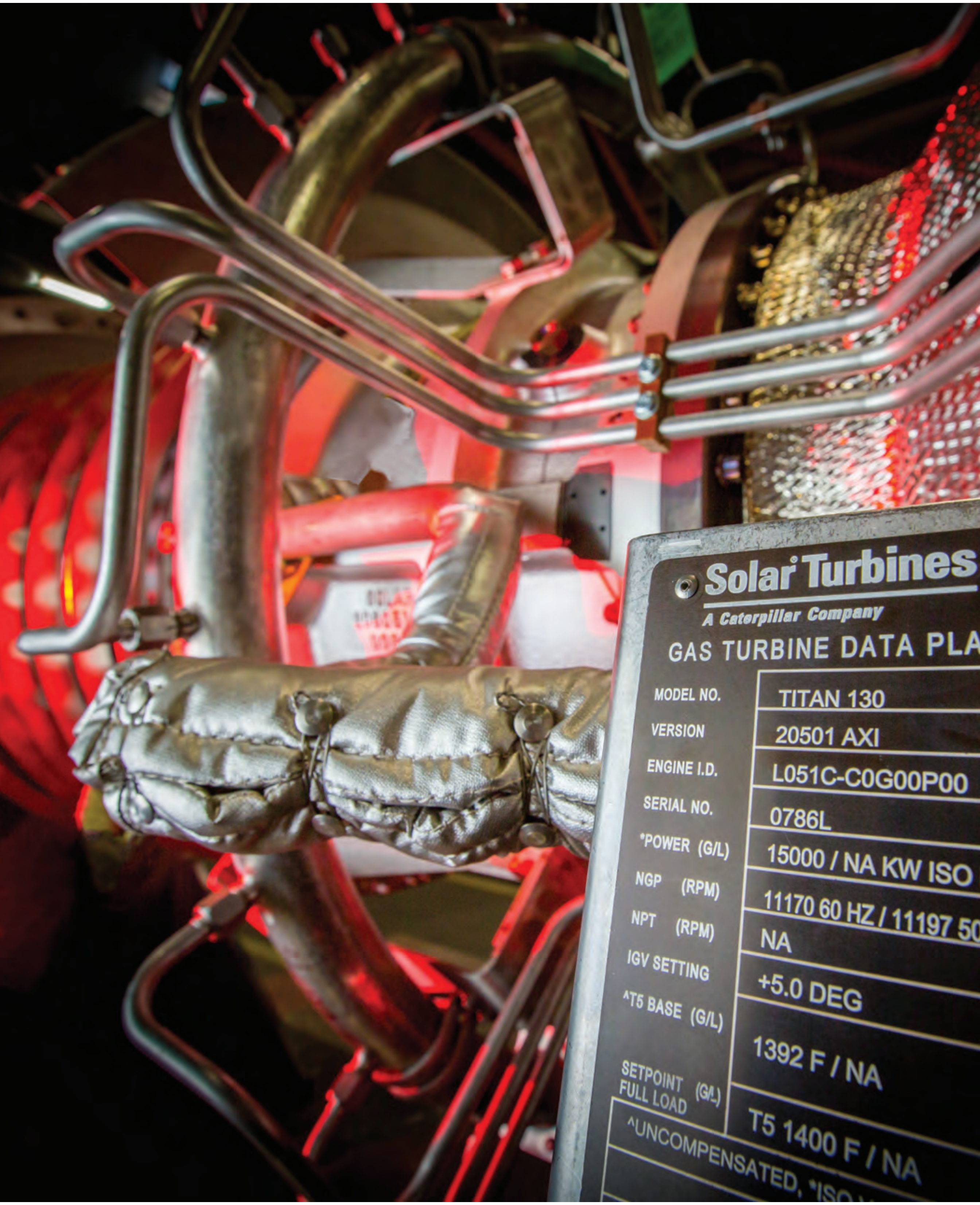


Figure 8-12. *CO_2, and other pollutant emissions of different power plants*


Solar Turbines
A Caterpillar Company
GAS TURBINE DATA PLA
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TITAN 130
VERSION
20501 AXI
ENGINE I.D.
L051C-C0G00P00
SERIAL NO.
0786L
*POWER (G/L)
15000 / NA KW ISO
NGP (RPM)
11170 60 HZ / 11197 50
NPT (RPM)
NA
IGV SETTING
+5.0 DEG
^T5 BASE (G/L)
1392 F / NA
SETPOINT FULL LOAD (G/L)
T5 1400 F / NA
^UNCOMPENSATED, *ISO

CHAPTER 9

FUEL & FUEL TREATMENT

Industrial gas turbines enable operation with a wide variety of gaseous and liquid fuels, while maintaining very low emissions. Lean-premix combustion systems, widely used for emissions control, require particular considerations regarding the fuel quality and fuel composition. To determine the suitability for operation with a gas fuel system, various physical parameters of the proposed fuel need to be considered: heating value, dew point, Wobbe Index and others. Attention is given to the impact of fuel constituents on combustion characteristics and the problem of determining the dew point of the potential fuel gas at various pressure levels. The impact of fuel properties on emissions, and the proper operation of the combustion system are discussed. Suggestions about how to approach fuel suitability questions during the project development and construction phase, as well as in operation are made (Kurz et al., 2006).

Gas Fuels

The quality and composition of fuel burned in a gas turbine impacts the life of the turbine, particularly its combustion system and turbine section. The fuel specified for a given application is usually based on availability and price. Natural gas is a typical fuel of choice for gas turbines due to its low (but increasing) cost, widespread availability, and low resulting emissions. However, the composition of fuel gas can widely vary, from gas with significant amounts of heavier hydrocarbons[1] (butane and heavier), to pipeline-quality gas consisting mostly of methane, to fuel gas with significant amounts of noncombustible gases (such as nitrogen or carbon dioxide).

The fuel composition impacts the entire gas turbine system. For example, fuels with a large amount of noncombustible components can move the operating point of the gas turbine compressor closer to its stall limit. They also have an impact on the power and efficiency of the gas turbine. Fuel choices impact the pollutant emissions levels. Durability issues, especially of the hot section, due to corrosive compounds or the formation of liquids in gaseous fuels, have to be addressed. Combustor operability (and durability) issues, especially for lean-premix combustion systems depend on the choice of fuel. Certain fuel types pose safety challenges for the gas turbine package due to their flammability, containment (hydrogen is an example), and/or toxicity (carbon monoxide, hydrogen sulfide) issues.

Current dry low emissions technology primarily focuses on burning natural gas, a fuel that is mainly composed of methane. However, interest in utilizing other energy resources, in reducing pollutant emissions, as well as concern about energy security, have motivated interest in utilizing associated gas and raw natural gas, coal-derived syngas or fuels from other sources, such as biomass, landfill gas, or process gas. For many applications, dry, low-emissions systems also have been designed to burn liquid fuels like diesel.

Gas fuels for gas turbines are combustible gases, or mixtures of combustible and inert gases with a variety of compositions covering a wide range of heating values and densities. The combustible components can consist of methane and other low molecular weight hydrocarbons, hydrogen and carbon monoxide. The major inert components are nitrogen, carbon dioxide and water vapor. It is generally accepted that this type of fuel has to be completely gaseous at the entry to the fuel gas system, and at all points downstream to the fuel nozzle.

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[1] Hydrocarbons in fuel gas are usually alkanes, with the summary chemical formula C_nH_{2n+2}

The amount of energy that will be released during the combustion process for a specific gaseous fuel composition is determined by the heating value. For simple cycle gas turbines, the lower heating value is generally used since the latent energy of the steam is not recovered. Gaseous fuels can be categorized by the lower heating value, but another parameter is used more commonly in the gas turbine industry. That parameter is the Wobbe Index. From the lower heating value (LHV) in Btu/scf [kJ/Nm^3] and the specific gravity (SG), the Wobbe Index (WI) of the gas can be calculated:

$$WI = \frac{LHV}{\sqrt{SG}}$$

Because the fuel supply temperature T_f has an impact on the actual volumetric fuel flow, a temperature corrected Wobbe Index is often used, where the reference temperature T_{ref} is usually 520°R or 288°K:

$$WI = \frac{LHV}{\sqrt{SG}} \cdot \sqrt{\frac{T_{ref}}{T_f}}$$

If two different fuel gas compositions have the same Wobbe Index, the pressure drop in a given fuel system will be the same for both gases and in general, direct substitution is possible and no change to the fuel system is required. The Wobbe Index is, thus, an indication of energy flow in the system at the same gas pressure and pressure drop.

The Wobbe Index is used as a parameter to indicate the ability of the overall fuel handling and injection system to accommodate the fuel composition. If the Wobbe Index varies too far from the design value, changes to the fuel system need to be made. A high Wobbe Index often indicates the presence of heavier hydrocarbons in the fuel, while a low Wobbe Index is often caused by the presence of significant amounts of noncombustible fuel components or by the presence of significant amounts of (highly combustible) hydrogen or carbon monoxide.

Because a lower Wobbe Index means, that the fuel mass flow, relative to the engine air mass flow increases, the power output of an engine increases slightly *(Figure 9-1)* when the Wobbe Index is lowered.

However, the Wobbe Index does not capture the effects of other fuel properties, such as dew point, flame speed and combustion chemistry. Therefore, the entire fuel composition must also be considered and, if more reactive species such as hydrogen, Alkanes and carbon monoxide are present in significant quantities, additional changes to the fuel system may be required. A good design criterion is that gases having a Wobbe Index within ±10% can be substituted without making adjustments to the fuel control system or injector flow area. If the fuel injector flow area has to be changed, the gas Wobbe Index should be inversely proportional to the effective controlling area of the injector.

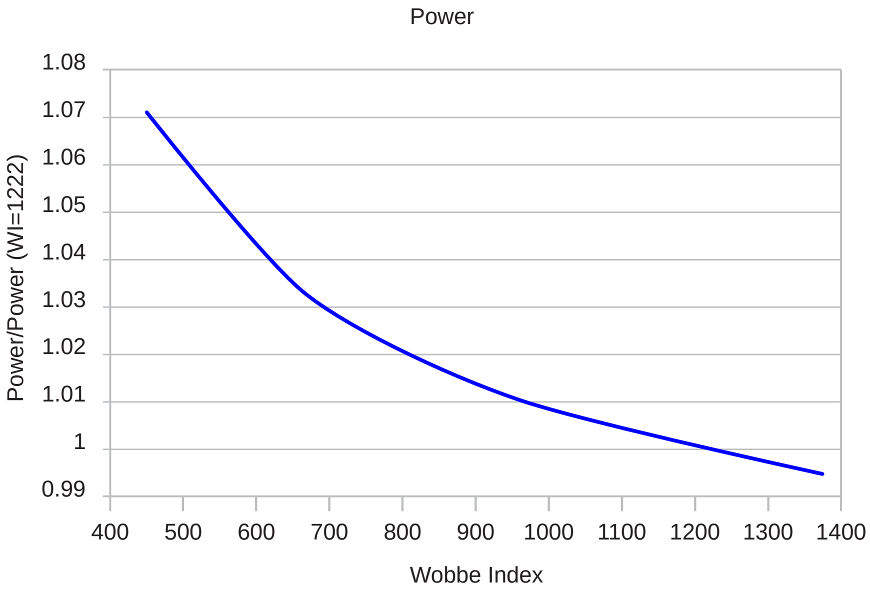


Figure 9-1. *Impact of Wobbe Index on Gas Turbine Power Output (Typical)*

Figure 9-1: For a typical landfill gas, the Wobbe Index is one-third the value of standard pipeline-quality natural gas. The controlling orifices on the injectors must be enlarged to three times their previous area. This allows the fuel flow rate of the landfill gas to have an equivalent pressure drop across the injector at full-load condition. This will provide stable, high-efficiency combustion, with the desired turbine inlet temperature distribution, for long combustor and blade life.

As the fuel heating value decreases below standard levels, the torch igniter and the combustion system may require standard natural gas or liquid fuel for startup or shutdown, as well as possible restrictions on turbine transient load operation.

Gaseous fuels can be classified based on changes required to the injector, fuel or control systems as specified in *Table 9-1*. The table includes special start-up requirements and limitations in loading based on the class of gas fuel. Both Lean-Premix and conventional combustion gas turbines can be designed to use the standard fuel with Wobbe Index range of 1220 ±10%. This range would be typical of pipeline-quality natural gas. High heating value fuels are accommodated with minor changes to the control systems and can be used in conventional combustion and, to a lesser degree, in gas turbines with lean-premix combustors. Very high heating value fuels usually require that the fuel temperature be monitored closely to prevent two-phase (gas and liquid) flow. For medium heating value fuels, conventional combustion may require a change in injector flow area. Low-heating value fuels require even more extensive changes to the injector and fuel system. The last category of very low-heating values has limited application for conventional combustion only. DLN combustion systems in industrial gas turbines are usually designed for standard natural gas. The use with other gases (either high BTU or medium BTU) usually requires careful evaluation and testing.

Gaseous fuels can be divided into four distinct categories as indicated in *Figure 9-2*. These categories are broadly based on the fuel source.

Fuel Gases in Oil and Gas Upstream and Midstream Applications

Gas fuels produced from oil and natural gas reserves include: raw natural gas, associated gas, pipeline-quality natural gas, liquefied natural gas (LNG), natural gas liquids (NGL) and liquefied petroleum gas (LPG).

Fuel Energy Density	Wobbe Index BTU/scf (MJ/Nm^3)
Very High (NGL & LPG)	>1600 (>59. 6)
High	1342 – 1600 (50. 0 – 59. 6)
Standard (Pipeline)	1098 – 1342 (40. 9 – 50. 0)
Medium	700 – 1098 (26. 1 – 40. 9)
Low	350 – 700 (13. 0 – 26. 1)
Very Low	<350 (<13. 0)

Table 9-1. *Classification of Gaseous Fuels*

Natural gas is extracted from the ground either with crude oil or from natural gas reservoirs. It is primarily a mixture of naturally occurring paraffin hydrocarbons including methane (CH_4), ethane (C_2H_6), propane (C_3H_8) butane (C_4H_{10}), pentane (C_5H_{12}), hexane (C_6H_{14}), and higher molecular weight compounds. Hydrocarbons heavier than propane form isomers, such as iso-butane (iC_4H_{10}) and iso-pentane (iC_5H_{12}) that have different properties than their 'normal' counterparts. Additionally, other gases such as carbon dioxide, hydrogen sulfide, mercaptanes, water vapor, and nitrogen can be present.

Associated Gas is extracted with crude oil and found either dissolved in the oil or as a cap gas above the oil in the reservoir. Historically, associated gas has been treated as a by-product of oil extraction and has either been flared or re-injected to extract more oil. Burning associated gas in the gas turbines used to extract the crude oil has been recognized in the past two decades as a more economical and environmentally sensitive approach. The majority of associated gas is produced offshore, complicating further efforts to recover and utilize these gases. The composition of associated gas depends on the type of reservoir from which it originated.

Raw Natural Gas or wellhead gas is extracted from gas wells. The gas reserves are generally classified as either dry or wet. Dry reserves do not contain appreciable condensable heavier hydrocarbons (pentane+), while wet reserves do. Natural gas containing hydrogen sulfide (H_2S) is referred to as sour gas. Excessive H_2S creates extensive corrosion problems on fuel-wetted parts; thus, stainless steel and/or coatings are often prescribed for equipment handling sour gas.

Pipeline-Quality Natural Gas that is transported and distributed for industrial and residential use is a subset of raw natural gas. Pipeline gas has been processed to remove

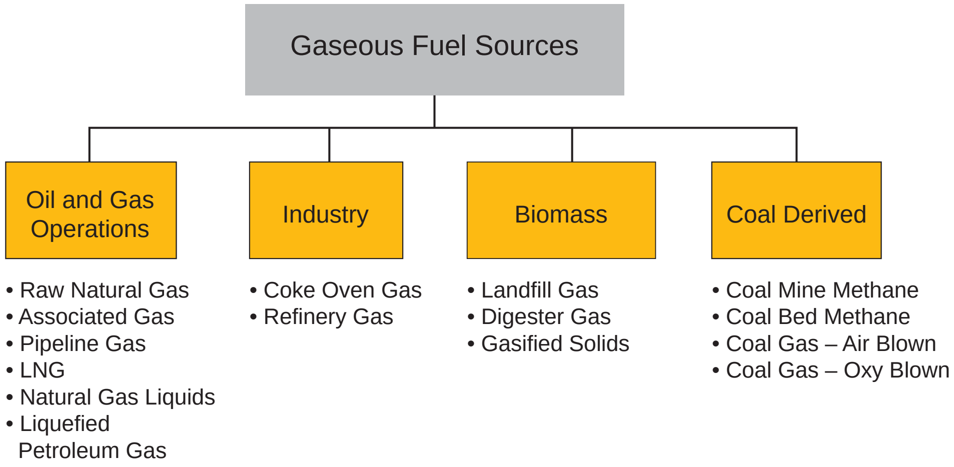

Gaseous Fuel Sources
Oil and Gas Operations
Industry
Biomass
Coal Derived
• Raw Natural Gas
• Associated Gas
• Pipeline Gas
• LNG
• Natural Gas Liquids
• Liquefied Petroleum Gas
• Coke Oven Gas
• Refinery Gas
• Landfill Gas
• Digester Gas
• Gasified Solids
• Coal Mine Methane
• Coal Bed Methane
• Coal Gas – Air Blown
• Coal Gas – Oxy Blown

Figure 9-2. *Gas Turbine Gaseous Fuel Categories*

the heavier hydrocarbons and adjust the heating value of the fuel within prescribed limits. Methane is the lightest combustible component of natural gas and generally referred to interchangeably with pipeline-quality gas (>90%) because it is the major constituent. Mercaptans are typically added to pipeline gas to give a distinctive odor that facilitates gas leakage detection.

Liquified Natural Gas (LNG) is pipeline-quality natural gas extracted remotely, chilled to a liquid (-260°F/-162°C), transported via tanker, heated, revaporized, processed as required and injected into an existing natural gas transmission and distribution network. LNG has a similar composition to pipeline-natural gas.

Natural Gas Liquid (NGL) is a general classification for those paraffin hydrocarbons heavier than methane, which can be transported and distributed in liquid form. At atmospheric pressure, NGL is in the gaseous state. As a gas, NGL is separated from methane by liquefying it through the process of pressure, absorption, or a combination of both. Because of the relatively high dew point of this gas compared to natural gas, NGL is usually handled as a liquid fuel for most industrial gas turbine applications. A determination must be made for each source of NGL based on the fuel analysis as to whether the fuel should be handled in the gaseous or liquid phase.

Liquified Petroleum Gas (LPG) is a sub-classification of NGL referring specifically to propane and butane mixtures offered commercially. LPG is always transported in liquid form and as with NGL, LPG is usually handled as a liquid fuel in gas turbine applications.

Fuel Gases Produced in Industrial Applications

Several industrial and chemical processes generate significant quantities of gas by-products and waste streams with acceptable fuel composition and adequate calorific value for use in gas turbines without extensive equipment modifications. Primary industries producing such fuels include steel mills and chemical and refining plants.

Refinery Gas is a by-product of gasoline production. This gas contains high hydrogen (H_2) content, along with several other reactive species including ethylene and propylene, mixed with methane, ethane and propane. Refinery gas is corrosive and hydrogen has a great tendency to leak and is explosive, therefore, safety is the main concern. With the proper safety considerations incorporated into design modifications and minor fuel control adjustments, refinery waste gas has been successfully used to fuel industrial gas turbines. Hydrogen has a relatively low molecular weight and high flame speed compared to natural gas.

Coke Oven Gas is generated during the production of coal and petroleum coke for iron and steel production. It contains methane, hydrogen and some carbon monoxide and has a lower heating value than pipeline gas. Coke oven gas composition is quite variable and requires significant treatment to remove contaminants for use in a gas turbine combustion that produces gases with high carbon monoxide and hydrogen content.

Anaerobic Digestion Gas is the gas recovered from the decomposition of organic matter by bacteria in the absence of oxygen. Anaerobic digestion occurs in:

- Liquid sewage
- Residues from fruit and vegetable canneries
- Animal and crop wastes
- Solid wastes disposed in sanitary landfills
- Marine plants, including macro-algae, water hyacinth and sea kelp

This process produces gas consisting of methane, nitrogen, carbon dioxide and air with a low heating value. Large variations in gas composition are common from different landfill and digester sites.

Fuel Gases Derived from Coal

Coal-derived gases include coal gas produced through several different gasification processes and gases that are collected during coal mining.

Coal Gas produced by gasification contains hydrogen, methane, and carbon monoxide as their combustible components, with large proportions of carbon dioxide and nitrogen as inerts. Depending on the type of gasification process, air-blown processes generate an LHV of from 3. 0 to 7. 9 MJ/Nm^3 (75 to 200 Btu/scf), while oxygen-blown processes yield medium BTU gases with a LHV of from 7. 9 to 21. 2 MJ/ Nm^3 (200 to 570 Btu/scf).

Coal Mine Gas is generally extracted in two forms: diluted with air as Coal Mine Methane (CMM) or as a high quality natural gas as Coal Bed Methane (CBM). In a gaseous coal seam, methane concentration can be up to 94-98% with 1-5% carbon dioxide. For safety reasons, this gas has to be drained from underground coal seams, before mining, in the form of CBM. To prevent explosions in coal mines, extensive ventilation is required to extract gaseous methane trapped between coal seams and the surrounding rock. In practice, the gas/air mixture from the coal mine is further diluted with air during extraction and the resultant CMM gas contains 30 to 70% methane by volume. This gas/air mixture can be compressed and directly used as a fuel in a gas turbine. Since the methane fuel and its air oxidant are premixed in the fuel gas, potential explosion hazards need to be addressed. The methane content in the gas/air mixture should always be kept with a sufficient margin above the upper flammability limit, at the highest anticipated pressure and temperature.

Fuel Gases from Biomass

Biomass can be converted into potential gas fuels for industrial gas turbines through thermal gasification and anaerobic digestion.

Thermal Gasification is a process of heating wood and other biological substances without combustion that produces gases with high carbon monoxide and hydrogen content.

Contaminants

In many systems, the gas composition and quality may be subject to variations (Newbound et al., 2003). Typically, the major contaminants within these fuels are:

- Solids
- Water
- Heavy gases present as liquids
- Oils typical of compressor oils
- Hydrogen sulfide (H_2S)
- Hydrogen (H_2)
- Carbon monoxide (CO)
- Carbon dioxide (CO_2)
- Siloxanes

Other factors that will affect turbine or combustion system life and performance include lower heating value (LHV), specific gravity (SG), fuel temperature, and ambient temperature.

Some of these constituents and contaminants may co-exist and be interrelated. For instance, water, heavy gases present as liquids, and leakage of machinery lubricating oils, may be a problem for turbine operators at the end of a distribution or branch line, or at a low point in a fuel supply line.

Water in the gas may combine with other small molecules to produce a hydrate — a solid with an ice-like appearance. Hydrate production is influenced, in turn, by gas composition, gas temperature, gas pressure and pressure drops in the gas fuel system. Liquid water in the presence of H_2S or CO_2 will form acids that can attack fuel supply lines and components. Free water can also cause turbine flameouts or operating instability if ingested in the combustor or fuel control components.

Heavy hydrocarbon gases present as liquids provide many times the heating value per unit volume than they would as a gas. Since turbine fuel systems meter the fuel based on the fuel being a gas, this creates a safety problem, especially during the engine start-up sequence when the supply line to the turbine still may be cold. Hydrocarbon liquids can cause:

- Turbine overfueling, which can cause an explosion or severe turbine damage.
- Fuel control stability problems, because the system gain will vary as liquid slugs or droplets move through the control system.
- Combustor hot streaks and subsequent engine hot section damage.
- Overfueling the bottom section of the combustor when liquids gravitate towards the bottom of the manifold
- Internal injector blockage over time, when trapped liquids pyrolyze in the hot gas passages.

Liquid carryover is a known cause for rapid degradation of the hot gas path components in a turbine.

The condition of the combustor components also has a strong influence and fuel nozzles that have accumulated pipeline contaminants that block internal passageways will probably be more likely to miss desired performance or emission targets. Thus, it follows that more maintenance attention may be necessary to assure that combustion components are in premium condition. This may require that fuel nozzles be inspected and cleaned at more regular intervals or that improved fuel filtration components be installed.

With a known gas composition, it is possible to predict dew point temperatures for water and hydrocarbons. However, the prediction methods for dew points may not always be accurate. In fact, it is known that different equations of state will yield different calculated dew points under otherwise identical conditions. Furthermore, the temperature in an unheated fuel line will drop because the pressure drop due to valves and orifices in the fuel line causes a temperature drop in the gas *(Figure 9-3)*. This effect is known as the Joule-

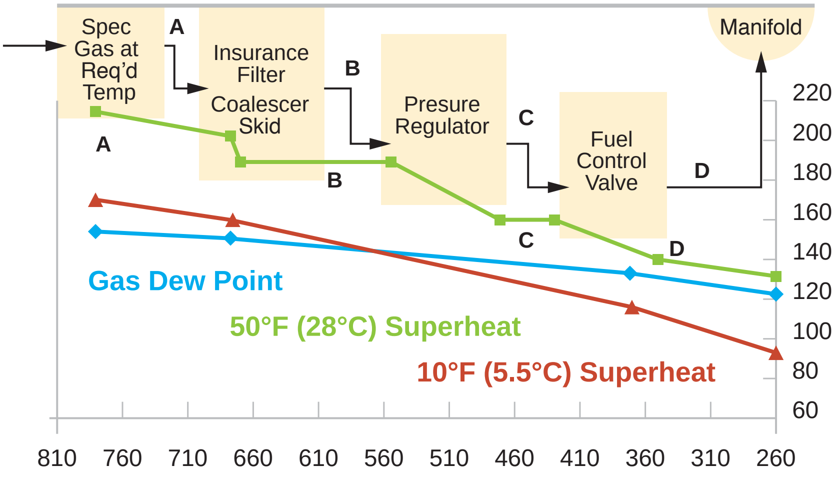


***Figure 9-3.** Pressure and Temperature drop in the fuel system*

Thompson effect. Most fuel gases (except hydrogen) will exhibit a reduction in temperature during an adiabatic throttling. Hydrogen, on the other hand, actually shows an increased temperature when the pressure drops, which creates a potential explosion hazard.

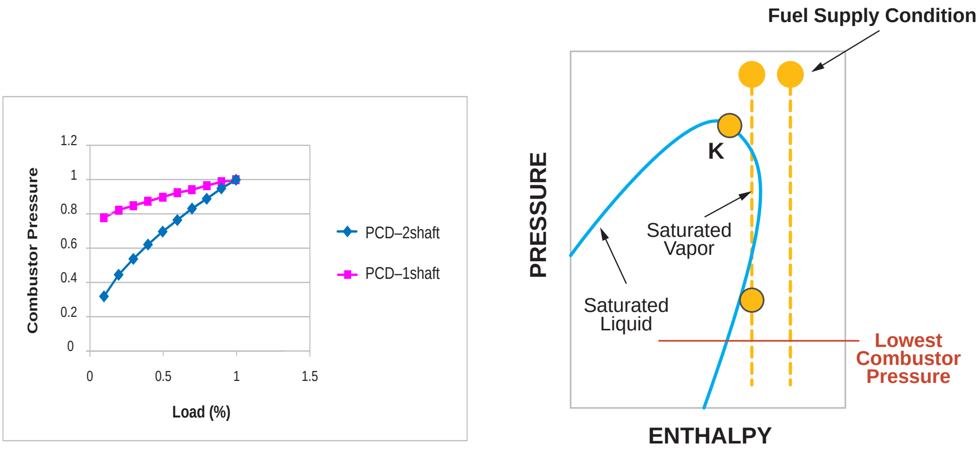


Figure 9-4. *Pressure drop and Two-Phase region*

Important Fuel Properties

Dew point of a gas is a function of gas composition, pressure and temperature. It describes the boundary between the single-phase gas and the two-phase (gas and liquid) state of a fluid (Elliott et al, 2002) *(Figure 9-4)*. Usually, the water dew point of a fuel gas is reported separately from the hydrocarbon dew point. One of the issues is that heavy hydrocarbons, even if they are only present in traces, have a large impact on the dew point. In many instances, a gas analysis lumps all the heavier hydrocarbons into one number (e.g., C6+). Since the dew point depends on the distribution of the heavy hydrocarbons, estimates can be made based on characterization methods (Campbell, 2000). However, these estimates are often not very accurate, and direct dew point measurements may be the preferred method.

In particular, there is no reproducible relationship between heating value and dew point. Because most gases will see a reduction in temperature during isenthalpic expansion (this is the Joule-Thompson Effect), it is possible that even a dry gas can develop liquids if it is subject to the pressure drop in a typical fuel supply system. It is, therefore, necessary to provide fuel gas sufficiently superheated. Values of 28°C/50°F of superheat at turbine skid edge are frequently used as a requirement, but the appropriate amount (which can be higher or lower) of superheat can be determined by a detailed dew point analysis. The amount of superheat needs to include allowances for uncertainty in the fuel gas composition at present and in the future, as well as the potential for heat loss of the fuel system.

Blowout refers to a situation where the flame becomes detached from the location where it is anchored and is physically "blown out" of the combustor. Blowout is often referred to as the “static stability” limit of the combustor. Blowout occurs when the time required for chemical reaction becomes longer than the combustion zone residence time. It can be an issue because the chemical kinetic rates and flame propagation speeds vary widely with fuel composition. For example, many candidate fuels have similar heating values but also have chemical kinetic times that vary by an order of magnitude (Lieuwen et al., 2006).

The opposite problem is flashback, where the flame physically propagates upstream of the region where it is supposed to anchor and into premixing passages that are not designed for high temperatures. Similar to blowout, flashback is an issue because of the widely varying flame speeds of candidate fuels. Again, fuels with similar heating values but with flame speeds that vary by factors of five or more are common (Lieuwen et al., 2006). Flashback occurs when the turbulent flame speed exceeds the flow velocity along some streamline, allowing the flame to propagate upstream into the premixing section. Flashback often occurs in the flow boundary layer, since this is the point of lowest flow velocity. As such, the effect of fuel composition variations upon flashback depend upon the corresponding change in turbulent flame speed. Flame speed is a propagation of the flame front moving in the combustion zone. Changes in the fuel composition, fuel-to-air ratio and inlet temperature affect the flame speed.

The laminar flame speed, also called flame velocity, or burning velocity *(Figure 9-5)*, is defined as the velocity at which unburned gases move through the combustion wave in the direction normal to the wave surface. A key point here is that the flame speed does not vary linearly between the respective pure values of the mixture constituents. For example, the addition of H_2 to CH_4 does not have a significant impact upon the flame speed until H_2 is the dominant constituent of the mixture. The impact of adding diluents, like CO_2, is, that the flame speed is still lower than the non-diluted mixtures, even if the temperature is maintained by increasing the equivalence ratio.

However, most issues are related to the turbulent flame speed, which depends, besides the laminar flame speed, also on the turbulence levels of the gases in question. In particular, data show that, as the turbulence intensity increases, the turbulent flame speed initially increases, then asymptotes to a constant value, and then, at very high turbulence intensities, begins to decrease. The most obvious effect of fuel properties on the turbulent flame speed is through the laminar flame speed. For example, for a given turbulence intensity and a given burner, fuels with higher laminar-flame speeds should have higher turbulent-flame speeds. However, turbulence intensity and laminar-flame speed alone do not capture many important characteristics of the turbulent-flame speed. Two different fuel mixtures having the same laminar-flame speed, turbulence intensity and burner can have appreciably different turbulent-flame speeds depending on the diffusion characteristics of the species involved.

Flame-propagation velocity is also strongly influenced by the fuel/air mixture ratio; the leaner the mixture the lower the velocity. If the flow velocity exceeds the flame-propagation velocity, then flameout could occur. If the flame-propagation velocity exceeds the flow velocity, then flashback within the premixing injectors could occur that can cause damage by overheating the injector tips and walls. To maintain flame stability at a point, the velocity of the fuel/air mixture must be within the flame-propagation speed to prevent flashback.

Flame flashback from the combustion chamber into the premixing zone is one of the inherent reliability problems of lean-premixed combustion. The flame speed is one of the most important parameters governing flashback. High flame speeds occur, for example, in associated gases containing high percentages of propane or butane *(Figure 9-5)*.

Autoignition is a process where a combustible mixture spontaneously reacts and releases heat in absence of any concentrated source of ignition such as a spark or a flame (Lefebvre, 1998). Rather than the flame propagating upstream into the premixing section, autoignition involves spontaneous ignition of the mixture in the premixing section *(Figure 9-6)*. Similar to flashback, it results in chemical reactions and hot gases in premixing sections, but its physical origins are quite different from those of flashback. In lean-premix combustors, or in general, in any combustor where fuel and air are premixed prior to combustion, this

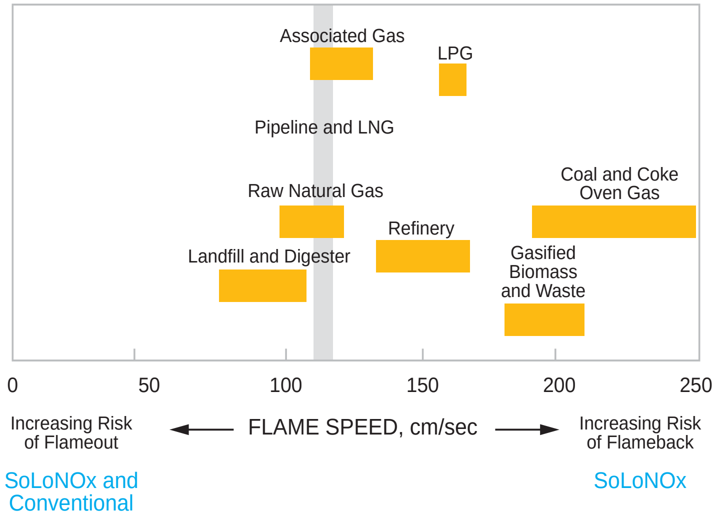


Figure 9-5. *Fuel Type Effects on Flame Speed*

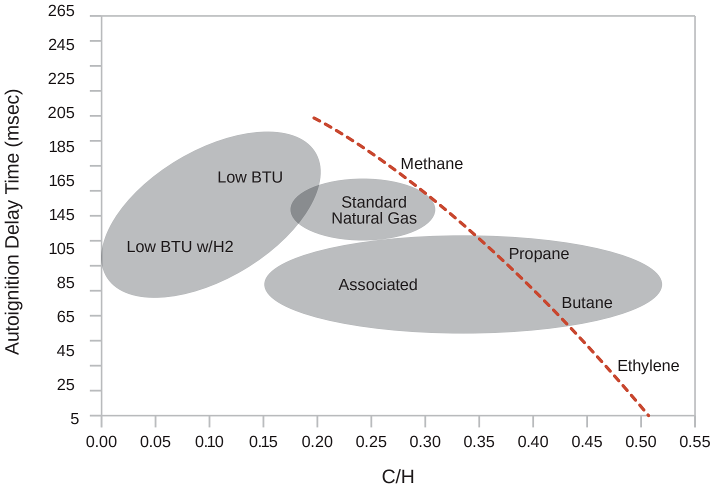


Figure 9-6. *Autoignition Delay Time Depending on Fuel Constituents*

spontaneous ignition inside the injector barrel has to be avoided, because it can damage combustor components, and yields high pollutant emissions. The autoignition delay time of a fuel is the time required for a mixture to spontaneously ignite at some given condition. This parameter is a function of the fuel composition, the fuel-to-air ratio, the pressure, and the mixture temperature. Ignition delay time is of importance to the combustion specialist because it is a direct indication of potential autoignition in the mixing barrel, and it is a useful parameter that defines the chemical kinetic time scale at any temperature, such as in the main burner, thus, playing an important role in the position of the flame relative to the injector tip.

Leaner mixtures tend to have a longer delay time, while higher mixture temperatures and higher pressures tend to shorten the delay time. In a lean-premix injector, the flow velocities thus have to be high enough to avoid autoignition inside the injector at the prevailing temperatures. Increasing the content of heavier hydrocarbons in an associated gas leads to a decrease of delay time. This is mainly caused by the non-symmetry of all higher hydrocarbons: Heavy hydrocarbons can be attacked much easier than methane molecules, resulting in reduced ignition delay times *(Figure 9-6)*.

In general, the characteristic kinetic times decrease with the addition of hydrogen, with the lowest times (and hence faster chemical kinetics) corresponding to mixtures of CO and H_2. For the ignition times (given in milliseconds) for 1000 K temperatures, the longest times (and hence slower chemical kinetics) are attributed to the mixtures containing mostly methane. Similar conclusions can be drawn for the higher temperature (1400 K) conditions. For example, CH_4 only mixtures ignite at approximately 800 μs at 1400 K, while most H_2- and CO-based mixtures ignite in less than 10 μs. In general, few data are available for specific mixtures, and engine specific tests are often necessary to avoid problems.

Another important parameter is the ratio of flammability limits. In the combustor, the fuel and air must be continually burned to keep the engine running. When the flame in the combustor is extinguished it is called a flameout or blowout. The fuel-to-air ratio changes with the engine load, as described earlier. In order to prevent flameout the combustor must support combustion over a range of fuel-to-air ratios. Each fuel composition has its own flammability range (Ratio of Flammability Limits). If the engine required fuel-to-air ratio range is equal to or larger than the fuel flammability range, then, at some point, the engine will experience flameout and will not be able to operate at that point. Knowing the ratio of flammability limits enables a decision as to whether or not the fuel composition has a broad enough flammability range to support combustion for all operating points of the engine. The ratio of flammability limits is defined as the upper flammability limit divided by the lower flammability limit. The upper flammability limit is the maximum fuel percentage (volumetric) mixed with air that will still light and burn when exposed to a spark or other ignition source. The lower is the minimum fuel percentage to sustain combustion. Different gases have different ranges of flammability. Hydrogen, for example, will burn with as little as 4 percent fuel and 96 percent air (lower limit) and as much as 75 percent fuel and 25 percent air (upper limit). Outside of this range (less than 4 percent or more than 75 percent fuel) the hydrogen-air mixture will not burn. Therefore, hydrogen has a ratio of flammability limits of 75/4 equal to 18. On the other hand, a typical coal gas has a ratio of 13.5/5.3 equal to 2.5. Coal gas typically contains methane, CO_2, and CO. CO_2 is not combustible. Therefore, if the coal gas contains too much CO_2, the flammability range will decrease and this ratio of flammability limits will decrease as well.

The stoichiometric flame temperature impacts the amount of NO_x emissions. It is also a parameter to help verify that a given fuel composition can be burned at all gas turbine operating loads and idle. Across the flammability range, the mixtures of fuel and air will burn at different temperatures. As the fuel-to-air ratio is increased from the lower flammability limit, the flame temperature will increase. Upon further increase in the ratio, a point will be reached where the amount of fuel and air will be perfectly matched so that all the oxygen in the air is reacted with all the fuel — this is called the stoichiometric fuel-to-air ratio. It also corresponds to the maximum flame temperature. As the fuel-to-air ratio is increased further still, the flame temperature starts to decrease and continues to decrease until the upper flammability limit is reached. In standard combustion systems, with a very heterogeneous mixture, the flame temperature is close to the stoichiometric flame temperature. The flame temperature has a significant impact on the NO_x production rate *(Figure 9-7)*. Fuel gas with a high amount of non-combustibles will usually cause low flame temperatures, while fuel gas containing amounts of hydrogen, carbon monoxide, or heavier hydrocarbons, will exhibit higher flame temperatures. Therefore, a fuel gas with a high amount of diluents will yield lower NO_x emissions even in standard combustion systems.

In order to understand how variations in fuel composition affect the phase difference between pressure and heat release fluctuations, it is necessary to consider the specific mechanism responsible for the instability. Two mechanisms known to be particularly significant in premixed systems are fuel/air ratio oscillations and vortex shedding. In the former mechanism, acoustic oscillations in the premixer section cause fluctuations in the fuel and/or air supply rates, thus, producing a reactive mixture whose equivalence ratio varies periodically in time. The resulting mixture fluctuation is convected to the flame where it produces heat-release oscillations that drive the instability. The coupling of the premixer

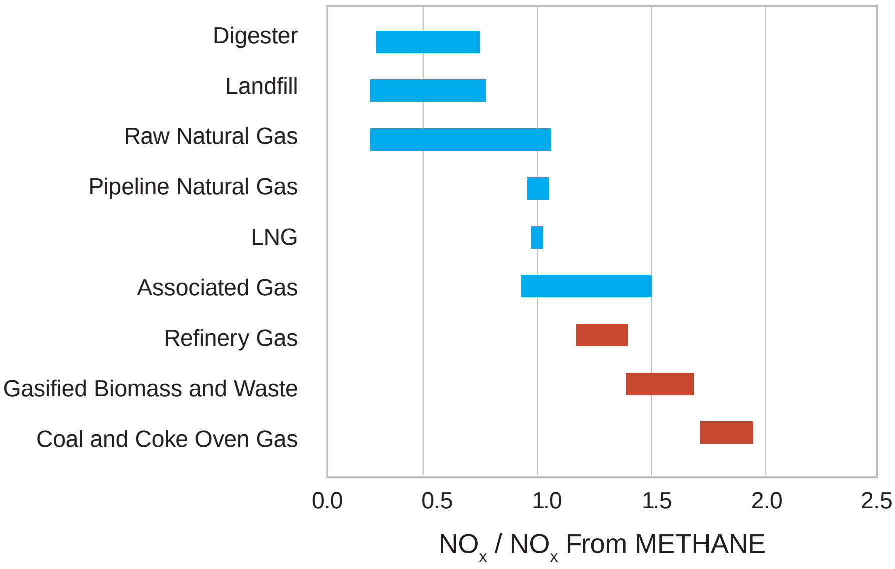


Figure 9-7. *Impact of Fuel Gas on NO_x Emissions*

acoustics with the fuel system is affected by the pressure drop across the fuel injector. The vortex shedding mechanism, as its name suggests, is due to large scale, coherent vortical structures. These structures are the result of flow separation from flameholders and rapid expansions, as well as vortex breakdown in swirling flows. They are convected by the flow to the flame where they distort the flame front and thereby cause the rate of heat release to change. Fuel/air ratio oscillations and vortex shedding become important when the resulting heat-release perturbation is in phase with the pressure fluctuation.

Liquid Fuels

Industrial gas turbines can also burn liquid fuels. The source of liquid fuels is usually petroleum liquids, but liquid petroleum gas, natural gas liquids, and alcohols have also been used.

Petroleum liquids originate from processed crude oil. They may be true distillates (diesel, kerosene) or ash-forming fuels. Natural gas liquids (NGL) are paraffin hydrocarbons other than methane found in natural gas, which can be extracted and subsequently handled as liquid fuels. They can also include LPG (liquid petroleum gas). LPG fuels are primarily mixtures of propane and butane.

In certain regions of the world natural gas is either not available or in short supply, and in these situations the local power producers must turn to alternative liquid fuels with qualities ranging from light distillates to heavy residual-grade oils or un-refined crude oils. Liquid distillates such as diesel oil or kerosene are also widely used as secondary back-up fuels in dual-fuel applications, which is often a contractual obligation in case of disruptions to natural gas supplies. Dual-fuel capability may also enable the primary fuel (i.e., natural gas) to be purchased at a more favorable price (Meher-Homji et al., 2010).

Fuels should be appropriate for the intended application and, especially in the case of low-grade fuels, must also be suitable for the particular gas turbine model. Residual-grade oils and crude oils will require additional balance of plant and on-site treatment programs, and these items also need to be carefully considered during the initial planning and design stages. The use of residual oil and crude oil in gas turbine applications and the underlying requirements for fuel treatment, BOP design considerations and the importance of realistic fuel specifications for heavy fuel projects are discussed in Kurz et al, 2012.

Distillates and residual-grade fuels originate in the oil refinery, and are produced from the fractionation of crude oil. The lower boiling point distillates are relatively free of sulfur and trace metals, but contaminant concentrations increase in the higher boiling point fractions toward the bottom of the distillation column. In fact, these components become concentrated in the “residuum”, so that residual-grade oils contain higher contaminant levels than the crude oil from which they were produced. This is an important consideration for gas turbine applications, because trace metal impurities create ash deposits during combustion and, unless treated, are responsible for various high-temperature corrosion mechanisms. Other properties such as density and viscosity also increase on moving down the distillation column.

Distillates are classified as “clean” fuels because of the absence of ash-forming trace metal impurities. Combustion characteristics are similar to natural gas and, without considering price, they are the most logical fuel alternative.

Several different grades of distillate are available and those most commonly used as gas turbine fuels are No. 2 distillate (also known as No. 2 diesel oil, No. 2 fuel oil or heating oil) and kerosene — which in Britain is often called paraffin. Naphtha is also gaining popularity as a gas turbine fuel, and is actually the generic name for a group of volatile and low-boiling point distillates normally used as a feedstock for gasoline production. However, depending on the desired product mix at a particular refinery, portions of the naphtha fraction can be sold directly as fuel. Due to the high volatility of naphtha, explosion-proof, fuel-handling systems are needed and have been developed for this purpose, and many successful gas turbine applications now exist around the world including the U. S. Probably the largest use of naphtha is in India, where it is burned as the primary fuel in almost all gas turbines in that country.

It also should be noted that the fuels listed under No. 2 distillate may contain different amounts of sulfur. U.S. Highway Diesel No. 2 contains 15ppm sulfur, while many other fuels in this category contain significantly higher levels of sulfur.

Distillates normally do not require additional treatment at the gas turbine power plant, but certain fuel properties can influence reliable operation. For example, some diesel oils and kerosenes have relatively high wax content and (more importantly) high wax-melting temperatures, so fuel-heating systems must be designed to prevent wax crystallization and filter plugging. Also, naphtha and certain kerosenes are known to have poor lubricating properties, and lubricity-improvement fuel additives are often needed to protect critical components such as fuel pumps and flow dividers.

The combustion of liquid fuels in gas turbines is dependent on effective fuel atomization to increase the specific surface area of the fuel, thus resulting in sufficient fuel/air mixing and evaporation rates. A smaller droplet size leads to higher volumetric heat release rates, easier ignition, a wider burning range, and lower concentration of pollutant emissions. The combustion reactions typically occur near the surface of the liquid droplets as the partially vaporized fuel mixes with the adjacent air (Lefebvre, 1998). Ignition performance is affected by two factors: the fuel volatility, as indicated by the Reid vapor pressure or the ASTM evaporated temperature, and the total surface area of the fuel spray, which is directly related to the spray Sauter Mean Diameter (SMD), and hence the fuel viscosity.

One of the main differences in the combustion between liquid and gas fuel is the presence of free carbon particles (soot), which determine the degree of luminosity in the flame. The presence of carbon particles at high temperatures is highly significant and luminous radiation from the liquid fuel dominates, whereas the non-luminous radiation of gas fuels is less important. The amount of radiation impacts the combustor liner temperature, and the heavier the fuel, the greater the emissivity and the resulting wall temperatures become.

Further issues involve the storage and transport of the liquid fuel: Low viscosity fuel (such as LPG) requires special pumps to avoid problems due to poor lubrication of the pump. Foaming and formation of solids and waxes have to be avoided (Reid vapor pressure, cloud point). The fuel has to be warm enough to be still capable to flow under gravity (pour point), but not too warm to exceed is flash point temperature (flash point). The fuel also has to be evaluated for its capability to corrode components of the fuel system (copper strip corrosion). The ash content has to be limited to avoid fuel system erosion. Excessive content of Olefins and Diolefins can cause fuel decomposition and plugging of fuel system components. If the carbon residue of the distillate is too high, carbon deposits may form in the combustion system. Solids in the fuel can cause clogging of the fuel system, in particular the very fine flow passages in injectors.

Lastly, certain contaminants that can cause corrosion of the hot section of the engine have to be limited. These contaminants include Sodium, Potassium, Calcium, Vanadium, Lead and Sulfur. Sodium is often found in fuels that are transported in barges, due to the contamination with salt water.

For very volatile liquid fuels, such as propane, it is important to prevent these fuels from flashing or bubble formation in the fuel line. The fuel pressure in the system has to be high enough to keep the fuel above its bubble point pressure at the highest anticipated temperature. The bubble point for a multi-component fuel is defined by the temperature at which the first bubble of gas appears. The bubble point has to be considered for several reasons:

- To set the minimum fuel supply pressure for a liquid system. The pressure must be high enough to allow for fuel turndown ratio, and the pressure drops in the fuel system including the injectors.
- To determine the size of the metering orifices.
- If the bubble point is below 0°C (32°F) at any combustor operating condition, ice formation around the fuel injector is possible. This can distort the combustor spray pattern, resulting in a change in the combustor temperature profile.

If the fuel temperature is above its critical temperature, it cannot be liquefied by compression alone.

The fuel parameters that are of importance are thus:

Density and Lower Heating Value.

Reid Vapor Pressure:

- Bubble Point
- Cloud and Pour Point
- Flash Point
- Distillation End points (ASTM D86)
- Aromatics Content
- Content of Olefins and Diolefins
- Carbon Residue (ASTM D524)
- Ash content

Water and Sediment Contents:

- Copper Strip Corrosion
- Corrosive Contaminants

FUEL AND EMISSIONS

The fuel used impacts obviously the constituents in the exhaust gas. If the fuel yields a high flame temperature, as it is the case with heavier hydrocarbons, as well as hydrogen, carbon monoxide and some others, it will usually yield a higher amount of NO_x. On the other hand, fuels of this type often have a wide flammability range. Fuels with a low Wobbe Index due to a large amount of dilutants will yield low flame temperatures, thus low NO_x levels, but can cause problems for start-up and load transients due to a limited flammability range.

Another aspect should be considered: The amount of carbon dioxide produced in the combustion process depends. Besides the thermal efficiency of the engine, only on the amount carbon in the fuel. While methane has four hydrogen atoms for each carbon atom, ethane has only three, and octane only a little over two hydrogen atoms per carbon atom. Thus, burning a methane molecule generates four water molecules, but only one CO_2 molecule. Burning hydrogen causes no CO_2 emissions at all (however, most methods of generating hydrogen do). Burning coal or CO will yield combustion products consisting entirely of CO_2. Therefore, typical coal-fired power plants will emit 1000kg/MWh of CO_2, oil-fired plants yield 800kg/MWh, while natural gas-fired plants produce 600kg/MWhr or less *(Figure 9-8)*.

It is often argued that electric motor drives (for example for compressor or pump applications) do not generate any carbon dioxide, while the drivers that use natural gas as a fuel do. However, the argument becomes less clear once we consider that the electricity to drive the motor has to be generated somewhere, and has to be transported to the compression site using transmission lines.

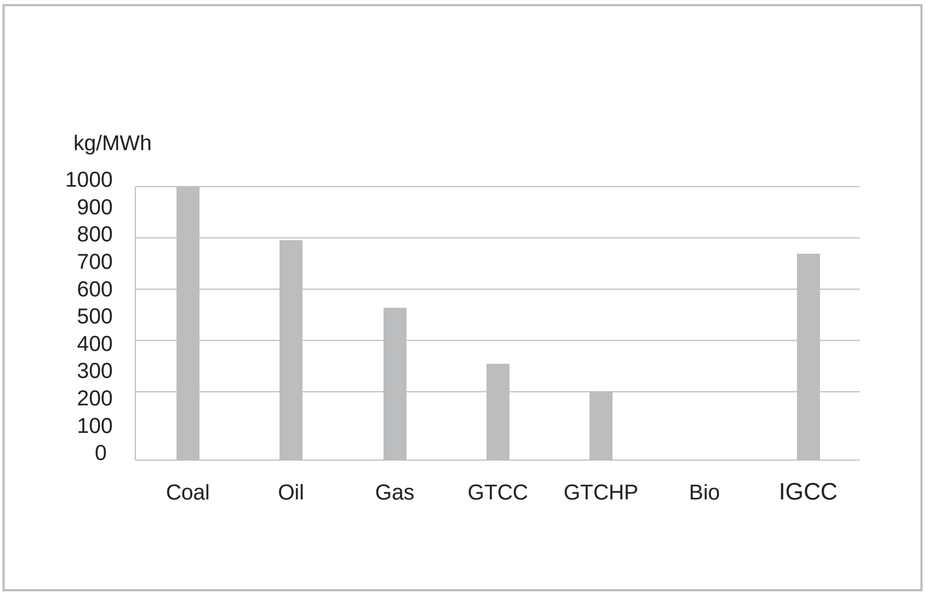


***Figure 9-8.** Comparison of CO_2 Emissions from Various Power Generation Plants*

This in turn means, that a gas turbine-driven compressor station may generate less emissions (both in terms of CO_2, as well as in terms of NO_x, S_{ox} and others) than an electric motor-driven compressor station, if the emissions produced when the electricity was generated are taken into account. *Figure 9-9* shows a comparison.

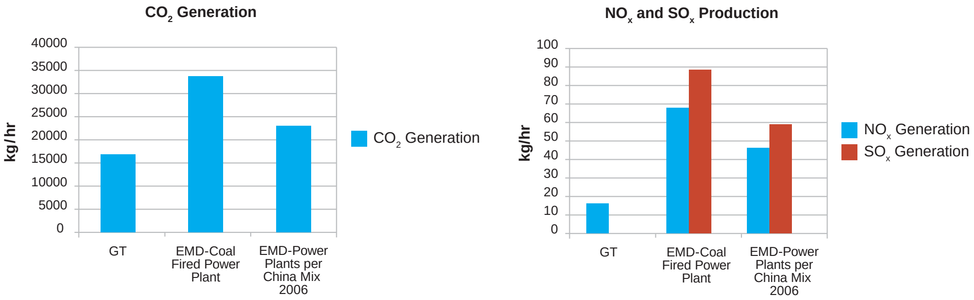


***Figure 9-9.** Emissions Footprints: Comparison for a 30MW Compressor Station with either two 15MW Gas turbine Drivers or two 15MW Electric Drives. Assumptions: Dry Low No_x Gas Turbine, power plant emissions per Figure 9-8, 93% efficiency for electric drive train, 95% transmission efficiency for electric power, and a 65% Coal, 3% Gas, 2% Oil, 30% Nuclear and Hydro Power Plant Mix (China, 2006).*

CHAPTER 10

AIR FILTRATION & AIR INLET SYSTEMS

Gas turbines ingest a large amount of ambient air during operation.

A normal mid-size (10 MW) gas turbine ingests about 100 million cubic feet (3 million cubic meters) of air per day. At that rate, even 100 ppb of foulant such as dirt, oils, or minerals, contained in the airflow, will result in almost one pound of foulant entering the gas turbine per day. Axial compressor fouling due to blade dirt deposits, leading/ trailing edge erosion, or surface corrosion, can reduce a gas turbine's output power and efficiency by up to 10%.

Because of this, the quality of the air entering the turbine is a significant factor in the performance and life of the gas turbine. A filtration system is used to control the quality of the air by removing harmful contaminants that are present. The system should be selected based on the operational philosophy and goals for the turbine, the contaminants present in the ambient air, and expected changes in the contaminants in the future due to temporary emission sources or seasonal changes.

Components used in gas turbine air cleaning applications with the purpose of removing solid particles and liquid droplets from the airstream include:

- Barrier filter elements capable of removing solid particles and repelling water
- Self-cleaning filter elements capable of removing solid particles and repelling water
- Vanes mist separators capable of removing water droplets
- Screens (insect or trash) capable of removing large contaminants
- Weather hoods capable of avoiding direct rain or snow ingestion

Gas turbine applications typically require air cleaners that use combinations of different filter types. Often, the face velocity is used to distinguish high velocity and medium/low velocity systems (Wilcox, et al., 2011).

During gas turbine operation the air cleaners are exposed to a variety of contaminants such as large and small particles, as well as liquids. Particles can be removed from the air stream by various types of filtration materials. The type of the solid or liquid contaminants depend on the location and environmental conditions. Water (and salts dissolved in the water) can be removed either by inertial separation (i.e., vane separators), or by filtration materials that repel water.

For off-shore applications, where salt water ingestion is a major concern, two filtration concepts have emerged: High Velocity Systems and Medium/Low Velocity Systems. These systems are also often used in coastal areas. For land-based on shore applications, most applications use either barrier filters or self-cleaning filter systems. Self-cleaning filter systems are essentially barrier filters that allow the occasional use of pressurized air to pulse-clean the filter material.

Applications also may have to deal with issues like heavy rain, cold and freezing temperatures, snow or sand storms, swarming insects, very high or very low relative humidity and combinations thereof.

Filtration Mechanisms

Filters in the filtration system use many different mechanisms to remove particles from the air. The filter media, fiber size, packing density of the media, particle size, and electrostatic charge influence how the filter removes particles. Each filter typically has various different mechanisms working together to remove the particles. Four filtration mechanisms are shown in *Figure 10-1a-d*.

The first filtration mechanism is inertial impaction. This type of filtration is applicable to particles larger than 1 micron in diameter. The inertia of the large heavy particles in the flow stream causes the particles to continue on a straight path as the flow stream moves around a filter fiber. The particulate then impacts and is attached to the filter media and held in place as shown in the top picture of *(a)*. This type of filtration mechanism is effective in high-velocity filtration systems.

The next filtration mechanism, diffusion, is effective for very small particles typically less than 0.5 micron in size. Effectiveness increases with lower-flow velocities *(b)*. Small particles interact with nearby particles and gas molecules. Especially in turbulent flow, the path of small particles fluctuates randomly about the main stream flow. As these particles diffuse in the flow stream, they collide with the fiber and are captured. The smaller a particle and the lower the flow rate through the filter media, the higher probability that the particle will be captured.

The next two filtration mechanisms are interception *(c)* and sieving *(d)*. Interception occurs with medium-sized particles that are not large enough to leave the flow path due to inertia or not small enough to diffuse. The particles will follow the flow stream where they will touch a fiber in the filter media and be trapped and held. Straining is the situation where the space between the filter fibers is smaller than the particle itself, which causes the particle to be captured and contained.

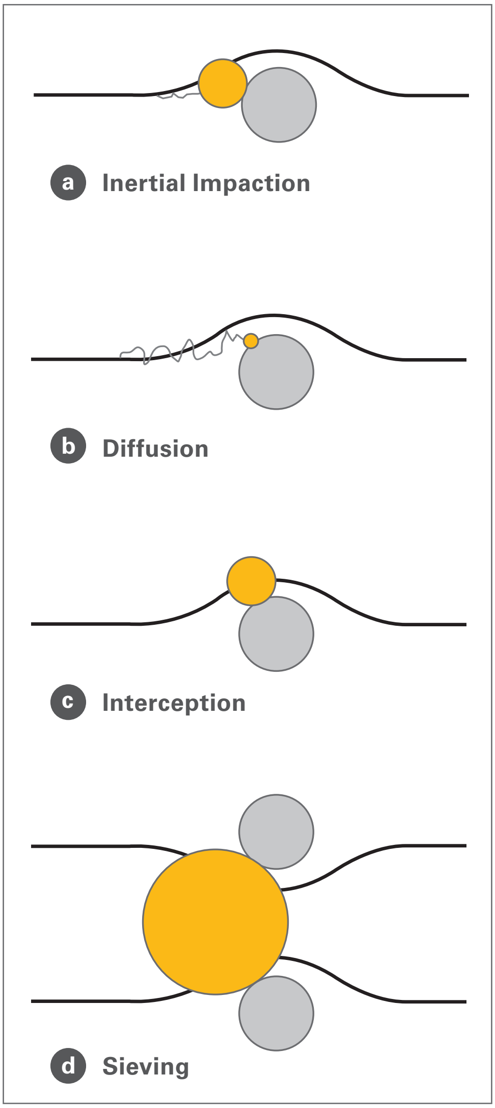


Figure 10-1a-d. *Common Filtration Mechanisms*

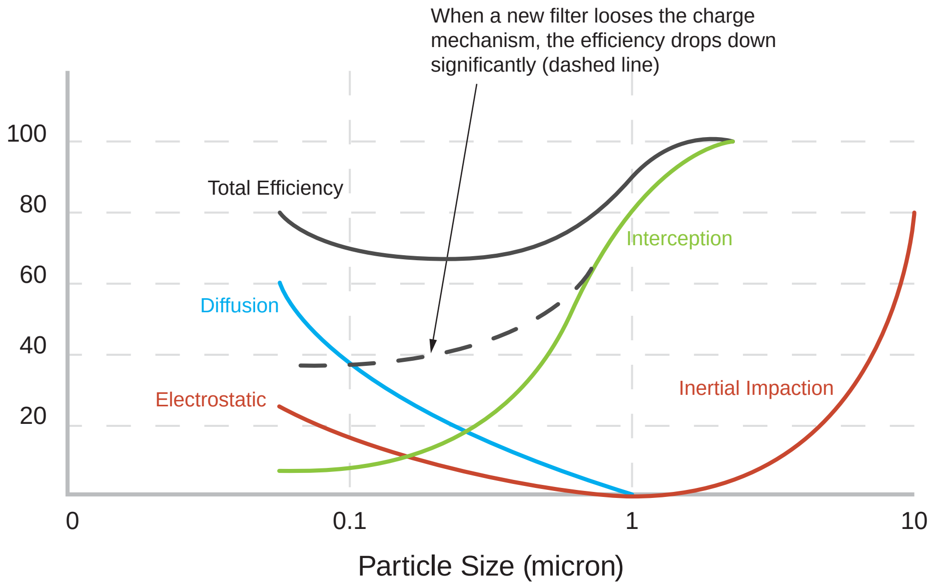


Figure 10-2. *Combination of Filtration Mechanisms to Obtain Filter Efficiency at Various Particle Sizes*

Electrostatic charged filters are effective for particles in the 0.01 to 2 micron-size range *(Figure 10-2)*. This filtration mechanism works through the attraction of charged particles to a charged filter. In gas turbine applications, the charge is applied to the filter before installation as a result of the manufacturing process. Filters always lose their electrostatic charge over time because the particles captured on their surface occupy charged sites, thereby neutralizing their electrostatic charge. As the charge is lost, the filter efficiency for small particles will decrease. On the other hand, as the filter is loaded, the filtration efficiency increases, thus counteracting the effect of lost charge to some extent. This will offset some of the loss of filtration efficiency due to the lost charge. *Figure 10-2* shows a comparison of a filter's total efficiency based on the various filtration mechanisms that are applied. The figure shows the difference between the filter's efficiency curve before and after the charge is lost. The performance of the filter should be based on the discharged condition.

Different filter systems show different effectiveness for different dry particle sizes *(Figure 10-3)*. The paragraph above outlines methods to remove solid particles from the airstream. However, as outlined earlier, ingestion of liquids is also an issue. Liquid water can contain dissolved salts and can penetrate conventional filter materials regardless of their capability to filter solid particles. At high humidity, water can enter the filter system. Water can dissolve dirt already captured in the filtration material and seep through the filter. Therefore, water removal is a crucial feature for air filtration systems especially in off-shore, marine and coastal environments. There are two fundamental methods to remove water: either with vanes upstream and downstream of the filter system or with special

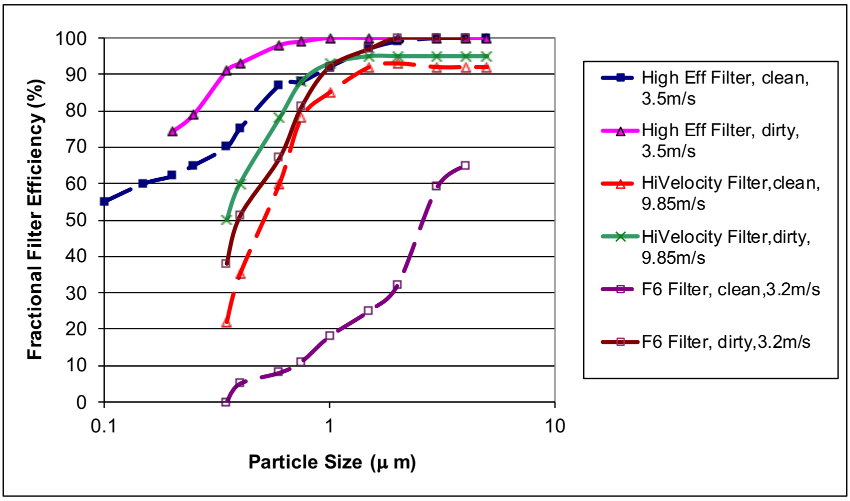


Figure 10-3. *Comparison of Fractional Efficiency for Filter Elements from Different Suppliers and Different Face Velocities in New and Dirty Conditions (Brekke and Bakken, 2010)*

hydrophobic filter material that cannot be penetrated by water, typically in the last, high-efficiency, filter stage. Water removal with vanes favors high air-flow velocities, and thus, high-velocity filtration systems. Water removal with special high-efficiency filter material favors low/medium velocity systems. Both methods, if applied properly, are capable of preventing liquid water from entering the gas turbine. However, as can be determined from *Figure 10-3*, high-velocity systems tend to have a lower filtration efficiency for solid particles than can be achieved with low/medium velocity systems.

All gas turbine inlet filters are designed to deal with the common types of contaminants such as: sand, dust and soot. Operation in the marine environment brings its own additional contaminant challenges that must be addressed by the filtration system to maintain a reliable and available machine, the main ones being discussed below (Wilcox et al, 2012 and Orhon et al, 2015).

Salt

Salt aerosol is the mixture of very small particles or droplets of salt with the surrounding air. This is formed when waves trap air as bubbles, which rise through the sea to its surface where they burst expelling small droplets into the atmosphere. The amount and make-up of these aerosols is a function of the wind and sea states. Strong winds lead to more and larger droplets. The resulting form of the salt within the aerosol when it eventually reaches the gas turbine inlet is directly related to the ambient relative humidity.

At relative humidities (RH) below ~40%, salt is crystal and can be considered as a dry particulate and can therefore be filtered just like any other dust-type particle with traditional filtration methods. However, the salt particles may be very small, so the filter efficiency for small particles is important.

Complications arise when the relative humidity changes as salt crystals are hygroscopic in nature, i.e., they have an affinity for water and like to absorb moisture from the surrounding air *(Figure 10-4)*.

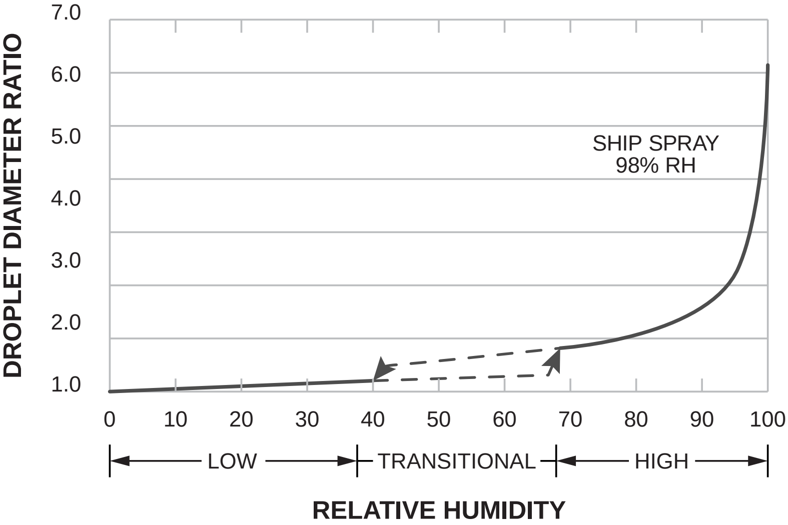


Figure 10-4. *Change in Salt State with Humidity (Gas Turbine Design Handbook, 1983)*

At a certain relative humidity (~70% RH, *(Figure 10-4)*, the salt crystal will continue to absorb moisture (and swell accordingly) until it reaches super saturation, at which time it deliquesces. Deliquescence is the change undergone by certain substances, which become damp and finally liquefy when exposed to the air, owing to the very low vapour pressure of their saturated solution. Salt is such a substance. At this critical RH the salt crystal becomes a saline droplet, which requires different filtration methods to capture and retain it.

At a relative humidity between 40% and 70%, the salt crystal is neither completely 'wet' nor completely 'dry' and can be considered as dynamic or 'sticky'. The relative humidity near the surface of the sea approaches 100% and has a vertical distribution with height which is dependent upon wind velocity. Offshore operators have recorded frequently and regularly varying relative humidity, often below the 70% RH level where salt begins to change from wet droplet to dry crystalline. Consequently, a gas turbine inlet air filter system design for coastal, marine and offshore installations must be able to handle salt in its wet, dry and dynamic phases *(Figure 10-4)*. In other words, it has to be capable of removing liquid water as well as fine salt dust.

When salt reaches a gas turbine, it can foul and corrode the compressor section, but more importantly, the sodium in the salt combines with the sulphur in the fuel (if present) to cause accelerated corrosion in the hot section of the gas turbine.

For any gas turbine installation, the trade-off between filter effectiveness, filter size, weight and cost, and the pressure loss due to the filter have to be very carefully analyzed based on site conditions *(Figure 10-5)*. The selection of the filter system will impact both initial cost as well as the cost to operate the equipment.

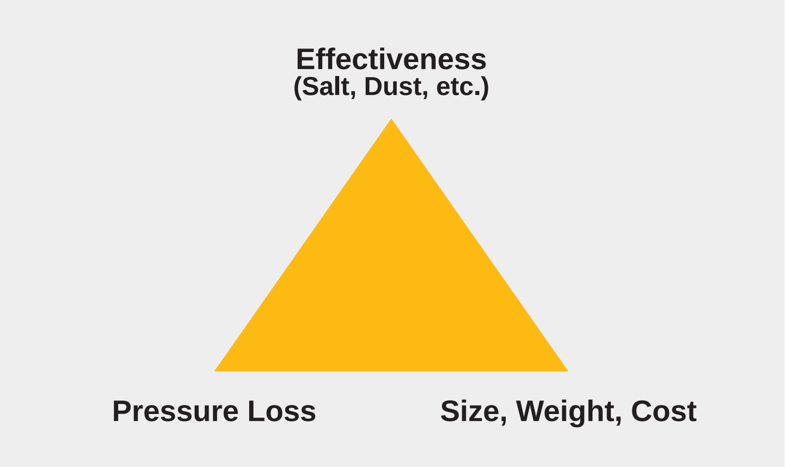
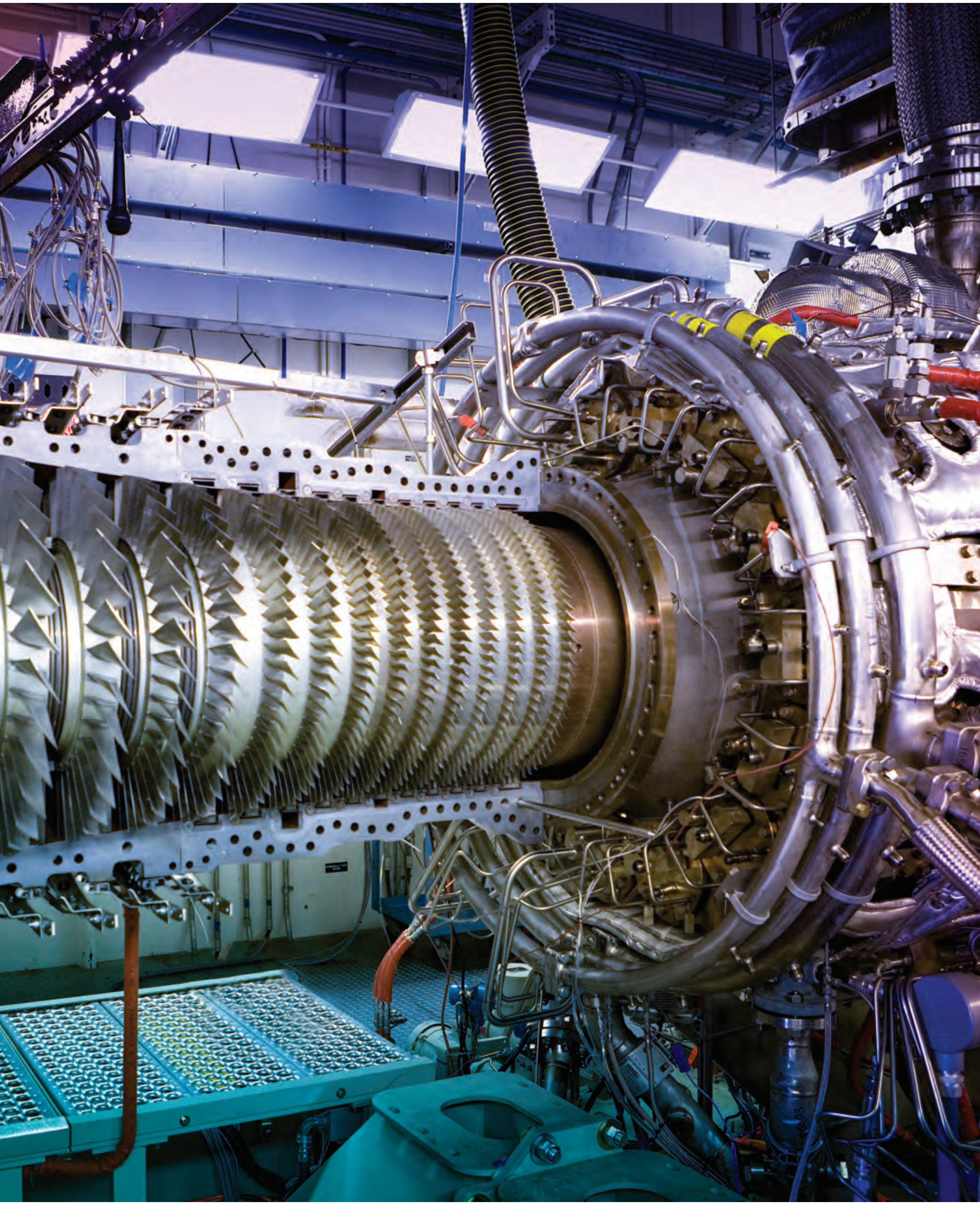


Figure 10-5. *Air Filtration Trade-off*

(Figure 10-5). For a given gas turbine airflow, the pressure drop is a function of the filter material, the amount of dirt the filter has already caught, and in particular the face velocity. Thus, a bigger filter, will have a lower face velocity, thus a lower pressure loss. But it also will be heavier, and probably more expensive. Lower face velocity in general allows for a higher filter efficiency for fine dust *(Figure 10-3)*.



CHAPTER 11

DEGRADATION IN GAS TURBINE SYSTEMS

Any prime mover exhibits the effects of wear and tear over time. The function of a gas turbine and of a gas turbine package is the result of the fine-tuned cooperation of many different components. Any of these parts can show wear and tear over the lifetime of the package and, thus, can adversely affect the operation of the system. In particular, the aerodynamic components, such as the engine compressor, the turbines, and the driven compressor have to operate in an environment that will invariably degrade their performance. Degradation of individual engine components has a compounded effect on engine performance, because the change in component performance characteristics leads to a mismatch of these components on the engine level, as well as on the component level. The impact of individual component degradation is also influenced by the control system and the control modes of the engine. Single-shaft engines, operating at constant speed, will show different degradation behavior than two-shaft engines. The impact of degradation on two-shaft engines depends on the control mode they are in, i.e., whether the gas generator speed or the firing temperature are the limiting factors. Also, the method and location of measuring the control temperature will determine the behavior of the engine in degraded conditions (Kurz et al., 2008).

Causes for engine degradation include:

1. changes in blade surfaces due to erosion or fouling and the effect on the blade aerodynamics. This includes airfoil (blade) fouling and pitting (loss of flow profile caused by corrosion and surface deposits), and leading-and trailing-edge erosion (caused by mechanical solid and liquid impacts on the rotor blades, as well as by erosion and hot corrosion). Other flow path components (such as casing walls, etc.) are affected in similar ways.
2. changes in blade clearances, seal geometries and the effect on parasitic flows. Opening of blade clearances is typically caused by softening bearings and increased rotor imbalance, thus a result of mechanical degradation.
3. changes in the combustion system (e.g., which result in different pattern factors).
4. mechanical degradation.

While some of these effects can be reversed by cleaning or washing the engine, others require the adjustment, repair or replacement of components.

It should be noted that the determination of the exact amount of performance degradation in the field is rather difficult. Test uncertainties are typically significant, especially if package instrumentation, as opposed to a calibrated test facility is used. Even trending involves significant uncertainties, because in all cases the engine performance has to be corrected from datum conditions to a reference condition.

The overall effect of degradation on an engine compressor yields added losses and lower capability of generating head. Typically, a degraded compressor also will have a reduced surge or stall margin. This will not have any significant effect on the steady state operation, as long as other effects that lower the stall margin (such as water or steam injection) are avoided (Brun et al, 2005). For a given speed of a degraded compressor, each subsequent stage will see lower Mach numbers (because of the higher temperature) and an increased axial velocity component (because $\rho = p/RT$, where p is reduced, T is increased, thus the density gets reduced).

When the engine compressor operates at design aerodynamic speed (i.e., Machine Mach Number), the stages in the axial compressor are usually fairly equally loaded. At low aerodynamic speed (i.e., low mechanical speed, or high ambient temperature) the front stages become higher loaded than the rear stages and vice versa for high aerodynamic speed (i.e., high mechanical speed or low ambient temperature) (Williams, 2008, Kappis and Guidati, 2012). The impact of fouling in the compressor can also change the stage loading across the compressor. The net effect will be, that while in the new gas turbine compressor all stages were working at their optimum efficiency point at design surge margins, the degradation will force all stages after the first one to work at off-optimum surge margins and lower-than-design efficiency. This will not only lower the overall efficiency and the pressure ratio that can be achieved, but also the operating range.

Calculations for a typical axial compressor (Kurz and Brun, 2001) reveal that the combined effects of airfoil fouling and increased clearances lead to loss of pressure ratio, loss of efficiency and loss of range or stall margin. In particular, the increased clearances cause choke to occur at a lower flow, in other words, the maximum flow through the compressor is reduced.

For the turbine section, increased roughness may increase the boundary layer thickness, thus reducing the flow capacity. This would also impact the operating point of the compressor (see Chapter 7). On the other hand, erosion of the trailing edge would actually increase the flow capacity of the turbine, again also affecting the compressor operating point. Both effects also will reduce the amount of work the turbine can extract from the gas stream, and reduce its efficiency. The airflow through the engine is typically controlled by the fact that the turbine nozzle is choked. Thus, any geometry change or change in boundary layer thickness will directly impact the airflow.

Several mechanisms cause the degradation of engines (Kurz et al., 2008):

- Fouling
- Corrosion, hot corrosion, and oxidation
- Erosion and abrasion
- Particle fusing
- Mechanical degradation

Fouling

Fouling is caused by the adherence of particles to airfoils and annulus surfaces. The adherence is increased by oil or water mists. The result is a build-up of material that causes increased surface roughness and, to some degree, changes the shape of the airfoil. Particles that cause fouling are typically smaller than 2 to 10 μm. Smoke, oil mists, carbon, and sea salt are common examples. Fouling can be controlled by an appropriate air filtration system. Fouling can often be reversed to some degree by detergent washing of components. Fouling in the colder sections of the engine can normally be eliminated by cleaning. Components operating at elevated temperatures are also subject to fouling, but the dirt will often be baked to the surfaces, and can usually not be removed by cleaning processes available in the field.

There are generally four different types of fouling deposits that can be found inside an axial compressor: salts, heavy hydrocarbons (oils and waxes), carbon dirt, and other dirt. A gas turbine compressor acts like a very effective air filter in that it collects and deposits a significant percentage of any solids that are carried by the ambient air and are ingested into the gas turbine inlet. Also, due to the heat of compression, the temperature of the air passing through the compressor will increase to above 600°F such that any solids or chemicals dissolved in the water will drop out and deposit on the compressor blades.

Salts: Salts (and other chlorides) in combination with moisture are primarily responsible for metal surface pitting in gas turbine compressors. As one of the most common deposits, salts lead to the formation of the deposit of chlorides in microscopic surface cracks and subsequent subsurface corrosion (evidenced as characteristic pits (small holes) on the surface of the blades). Salts do not enter the gas turbine from the ambient air (as is typical in offshore applications) but would be introduced by the plant water (i.e., fogging and water wash).

Oils and Waxes: Oils and waxes are usually residues from compressor washing or ambient air contamination. Generally, the components do not cause significant damage but can act as binding agents for dirt or sand in the compressor and, thus, can contribute to fouling.

Carbon: Carbon (or coke) deposits on compressor blades indicate that exhaust gases from the gas turbine or other internal combustion engines are entering the axial compressor. Coking inside the rotor can also be caused by local overheating or lube oil leakage. Deposits will bind with oils to form a surface deposit on the blades, which is very hard to remove with online and offline cleaning. Most will deposit in the first couple of stages of the compressor and will not be carried downstream.

Dirt: Sand and other types of dust (for example from farming, industrial processes, drilling, etc.) are primarily introduced into the gas turbine through the inlet filter and are an indication of inadequate inlet filtration or filter dirt saturation. The likely source of any sand or dirt particles entering the gas turbine at subject installation is from the inlet air as any water carrying sand or dirt would have been captured by the high-efficiency, fogging-water filter.

Engine fouling is caused by the buildup of contaminants on internal surfaces. Adhesion of particles on blade surfaces alters the profiles and results in a loss of engine power and an increase in fuel consumption. Fouling in intercoolers reduces compression heat removal. Air contamination can be significantly reduced by the use of more efficient air filtration systems, especially those that reduce the quantity of smaller particles ingested. Therefore, it is essential that the air filtration system be optimized to remove small particles, while also being capable of protecting against larger contaminants. Many improved media filter elements can be used as replacement elements during filter change-out. Filter companies have developed media to capture more dust particles in the 0.1 to 2 μm size range, while still offering satisfactory filtration of larger particles and acceptable service life. However, generally speaking, the more efficient the filtration efficiency of small particles, the higher the pressure loss across the inlet system, or the larger the filtration system becomes.

Composition of the particles is important in determining the rate of fouling. Because of very high mass flows, even very low levels of ingested foreign material can produce a substantial input of contaminants through gas turbine engines (see discussion above). For example, 1 ppmw of impurities entering the gas turbine results in over 10,000 lbs. of ingested material for a gas turbine over an 8,000-hour operating period. Even after several stages of cleanup (including barrier filters) of ambient inlet air, deposits commonly form in gas turbine compressors due to ingested particulate so that compressor washing is periodically needed to restore efficiency by removing deposits. Details on compressor fouling may be found in (Meher-Homji et al, 2013).

Corrosion, Hot Corrosion, and Oxidation

Corrosion is the wearing away of surface metals due to a chemical reaction of the metal with the environment (unlike erosion or fretting which are mechanisms of the wearing away of surface metal due to mechanical surface action). Usually the metal reacts with oxygen in the air but there are many other chemical reactions that can contribute to the different corrosion mechanisms. Certain types of corrosion (such as oxidation, sulfidation, and hot corrosion) are mainly attacking the hot section of the gas turbine. Other types, such as crevice corrosion and pitting, are rather found in the compressor section of the gas turbine. For gas turbine applications, the most important corrosion mechanisms are (Brun and Kurz, 2010): Oxidation, type 1 hot corrosion, type 2 hot corrosion, corrosion pitting, and crevice corrosion. These mechanisms are relevant for gas turbines and can impact their life and performance.

Corrosion can be caused both by inlet air contaminants and by fuel, water, or combustion-derived contaminants. Fuel-side corrosion is more typically noted and severe with heavy-fuel oils and distillates than with natural gas because of impurities and additives in the liquid fuels that leave aggressive deposits after combustion. Salt-laden air can also cause corrosion on unprotected engine parts.

Oxidation of a metal is the loss of one or more electrons, causing the metal to go from a neutral state to a positively charged ion. This results in the formation of a surface metal oxide. Rusting is a typical oxidation process. The oxide layer on the surface can either be beneficial, as a protective barrier (also called a passivation film) over the metal, or detrimental, when the corrosion continues at a rate that leads to a rapid reduction of the mechanical properties of the metal. Oxidation is the chemical reaction between a component and the oxygen in its surrounding gaseous environment. Oxidation of turbine section components is relatively easy to predict and measures can be taken to control it since it primarily involves relatively simple metal/oxygen reactions. The oxidation rate increases with temperature. Metal loss due to oxidation can be reduced by the formation of protective oxide scales. Due to the high temperatures, significant oxidation is often observed in the combustor or on the hot section turbine blades; this should be monitored carefully as it can weaken these highly stressed parts. It is important not to confuse this oxidation with type 1 and 2 hot corrosion, which will be discussed next, or with basic metal melting caused by a local hot spot.

Hot Corrosion, also called high-temperature corrosion requires the interaction of the metal surface with another chemical substance at elevated temperatures. Hot corrosion is a form of accelerated oxidation that is produced by the chemical reaction between a component and molten salts deposited on its surface. Hot corrosion comprises a complex series of chemical reactions, making corrosion rates very difficult to predict. It is the accelerated oxidation of alloys caused by the deposit of salts (e.g., Na_2SO_4). Type I or high temperature hot corrosion, occurs at a temperature range of 730 to 950°C. Type II or low temperature hot corrosion occurs at a temperature range of 550 to 730°C. Inside gas turbines, sulfidation, and vanadium-assisted hot corrosion are the most potent causes for premature metal weakening and failure. For example, hot stress-corrosion cracking of turbine blades, which is effectively the continuous weakening of the blade metal surface from hot corrosion and the subsequent growth of subsurface cracks under mechanical stresses, has been the recognized root cause for many aircraft and ground-based gas turbine engine failures. Hot corrosion requires the interaction of the metal surface with sodium sulfate or potassium sulfate, salts that can form in gas turbines from the reaction of sulfur oxides, water, and sodium chloride (salt) or potassium chloride, respectively. Hot corrosion is caused by the diffusion of sulfur from the molten sodium sulfate into the metal substrate which prevents the formation of the protective oxidation film and results in rapid removal of surface metal. For hot corrosion to occur, both sulfur and salt (e.g., sodium chloride or potassium chloride) have to be present in the very hot gas stream in and downstream of the combustor. Sulfur and salt can come from the inlet air, from the fuel, or water (if water is injected).

Vanadium and lead are found in certain types of liquid fuels. Vanadium and lead-assisted hot corrosion mechanisms are different in that molten vanadates (vanadium oxides such as vanadium-pentoxide) or lead slag dissolve (flux) the protective metal surface oxide layer and allow the diffusion of new oxygen into the metal substrates. i.e., the protective oxide layer is destroyed, leaving the underneath metal unprotected against oxidation. As this process continues as long as either vanadates or lead is present, the surface corrosion can be very rapid. Most liquid fuels contain small traces of vanadium, potassium, and sometimes lead (from accidental mixing with leaded gasoline during transport). It is critically important to maintain the gas turbine’s fuel quality within the OEM specifications for these trace elements, treat/wash the fuel, or use vanadium inhibitors if necessary.

Sulfidation is the reaction between a metal and a sulfur and oxygen-containing atmosphere to form sulfides and/or oxides. This means in particular, that sulfidation only requires the presence of sulfur in the combustion air or the fuel, but it does not require the presence of sodium or potassium. In essence, sulfidation attack is a form of accelerated oxidation resulting in rapid degradation of the substrate material due to loss of corrosion protection. Whereas during oxidation protective oxide scales can form, the metallic sulfides formed are not protective. This accounts for the rapid rate of degradation produced by sulfidation attack.

Corrosion of compressor components, which operates at much lower temperatures, is unlikely during engine operation because the compressor is dry. However, during shutdowns where cold surfaces can condense water, chemical species such as salts or sulfur compounds can be absorbed in the water producing an acidic, corrosive liquid. This liquid phase can result in aqueous corrosion of compressor components through a variety of mechanisms, e.g., generalized, pitting and crevice corrosion, and stress corrosion cracking. Compressor coatings, where used, are very effective in preventing these types of corrosion.

Corrosion Pitting, also simply called "pitting," this is a localized corrosion mechanism that leads to the formation of small but deep holes in the metal surface. As these holes are often not detected because the remainder of the metal part may appear completely clean, shiny, and polished, they represent a significant danger for unexpected failures. Pitting is often found on gas turbine compressor blades and it is caused by the intrusion of conductive impurities, such as salt water, into small surface cracks in the metal surface. As the water evaporates, the concentration of the impurity (usually sodium, sulfate, or chloride) increases, which results in highly localized corrosion and consequent deepening of the cracks. This process continues until a deep crack forms, metal pieces break off, and a pit (or hole) is formed. The pit can severely weaken the blade and also cause stress concentrations. Pitting can be initiated by very small microscopic surface defects or scratches, which are often not detectable visually. Most often, pitting on compressor blades occurs in gas turbine applications where salt is ingested into the compressor (from the inlet air) and if the unit's operation is highly cyclic with many starts and stops. During any prolonged shutdown, water condensate forms on the blades, dissolves the blade salt deposits, and then enters into microscopic surface cracks to initiate the pitting process. To limit compressor blade pitting, anti-corrosive blade coatings can be utilized, and the axial compressor should be thoroughly water-washed before any extended shutdown period (to remove salt deposits), and the inlet filtration system should be designed to minimize salt intrusion.

The physical process of crevice corrosion is similar to pitting, but rather than inside metal surface cracks, crevice corrosion occurs in pre-existing tight gaps, such as contact areas between parts, underneath grout, seals, and gaskets, or below hardened dirt or blade foulant. Concentration factors of impurities in these crevices can reach several millions. Because these areas cannot easily be inspected without disassembly, crevice corrosion presents a substantial risk of catastrophic failure. In gas turbines, crevice corrosion at the highly stressed mating surfaces between rotor blade base and disk slots can go undetected for years until either the blades are disassembled (which is usually not done unless the gas turbine is repaired/overhauled) or the blade structural support fails and they are liberated into the gas path (resulting in domestic object damage).

Other corrosion mechanisms, such as weld corrosion, microbial corrosion, galvanic corrosion, and green rot are also possible mechanisms in gas turbines and their ancillary systems, but these are less common. Hydrogen embrittlement is also sometimes observed in gas turbine fuel systems if the fuel composition contains elemental hydrogen but this is not usually considered a classical corrosion mechanism.

Since all types of corrosion can be either enabled or accelerated by the presence of certain contaminants in fuel, air or water supplied to the gas turbine, inspection, maintenance and proper fuel, air and water treatment are important factors in the prevention of corrosion. Ingested contaminants can result in corrosion to the compressor, combustion, and turbine sections of gas turbine engines.

Corrosion can be controlled by proper maintenance procedures, good air filtration, attention to fuel and water. It is important to note that corrosion processes are often self-propagating, and will continue even after the source is removed or abated.

To avoid hot section corrosion, it is very important to minimize the amount of Na+K entering the machine. By definition these limits apply to "total Na+K" entering the hot gas path from all sources; i.e., from fuel, air and any water or steam that may be injected for NO_x control or power augmentation. As can be seen, these limits are very severe.

EROSION

Erosion is the abrasive removal of material from the flow path by hard or incompressible particles impinging on flow surfaces. These particles typically have to be larger than 10 µm in diameter to cause erosion by impact. Erosion is more a problem for aircraft engines, because state of the art filtration systems used for industrial applications will typically eliminate the bulk of the larger particles. Erosion can become a problem for engines using water droplets for inlet cooling or water washing.

Abrasive solid particles attack rotating parts. Collisions between high-speed rotating blades and airborne particles result in metal fragments being ejected from blade surfaces. Particles as small as ten microns in diameter can cause erosion. Particle composition and shape can significantly affect erosion rates. Blade profiles are so carefully designed that even minor abrasions can alter the profiles to an extent that engine performance is affected. Erosion is an expensive problem, since it causes permanent damage, eventually requiring parts refurbishment or replacement. Erosion is proportional to particle concentration and, in severe service with poor filtration, can significantly reduce engine life.

Damage may also be caused by foreign objects striking the flow path components. These objects may enter the engine with the inlet air or the gas compressor with the gas stream or are the result of broken off pieces of the engine itself. Pieces of certain types of ice breaking off the inlet or carbon build up breaking off from fuel nozzles can also cause damage.

ABRASION

Abrasion is caused when a rotating surface rubs on a stationary surface. Many engines use abradable surfaces, where a certain amount of rubbing is allowed during the run-in of the engine, in order to establish proper clearances. The material removal will typically increase seal or tip gaps. Part of this is also age related, as bearings tend to become softer (reduction in stiffness) due to an increase in clearance over time that causes an increase in journal orbital amplitude.

While some of the effects of fouling can be reversed by cleaning or washing the engine, most other types of damage require the adjustment, repair, or replacement of components. It is thus common to distinguish between recoverable and non-recoverable degradation. Any degradation mechanisms that can be reversed by online and offline water washing are considered recoverable degradation. Degradation mechanisms that require the replacement of parts are considered non-recoverable, because they usually require an engine overhaul. There are some grey areas, because some degradation effects can be recovered by control system adjustments (that are, however, difficult to perform in the field due to limited capabilities to measure mass flow and performance). It should be noted, that the determination of the exact amount of performance degradation in the field is rather difficult.

Test uncertainties are typically significant, especially if package instrumentation as opposed to a calibrated test facility is used. Even trending involves some uncertainties, because in all cases, the engine performance has to be corrected from datum conditions to a reference condition.

Particle Fusing

The fusion of particles on hot surfaces leads to another source of problems. While dry, 2 to 10 µm-size particles could pass through older engines, causing little or no damage. However, these particles can cause problems in new generation, hotter-running engines. If the fusion temperature of the particles is lower than the turbine operating temperature, the particles will melt and stick to hot metal surfaces. This can cause severe problems since the resultant molten mass can block cooling passages, alter surface shape, and severely interfere with heat transfer, often leading to thermal fatigue. Affected surfaces are usually permanently damaged and will eventually need replacement.

Mechanical Degradation

Causes of mechanical degradation include wear in bearings and seals, coupling problems, excessive vibration and noise, or problems in the lube oil system. Mechanical degradation includes:

Gas turbine component creep or thermal ratcheting. The creep deformation of a nozzle can cause aerodynamic problems and performance changes:

- Bearing wear and increased losses
- Gearbox losses
- Coupling problems
- Excessive misalignment causing higher bearing loads and losses
- Combustor nozzle coking or mechanical damage
- Excessive rotor imbalance
- Excessive parasitic loads in auxiliary systems due to malfunctions
- Excessive mechanical losses in the driven equipment (generator or compressor)

Probably the most common indicator of mechanical degradation has been vibration. It is most important to note that several problems that manifest themselves as vibration may in fact have underlying causes that are aerodynamic (or performance) related in nature. Combustor fuel nozzles can, at times, plug up. There can be several causes for this such as coking, erosion, and incorrect assembly. Temperature distortions can create a host of problems in the hot section. Severe temperature distortions can create serious dynamic loads on blading, possibly inducing fatigue problems. The pattern of the EGT spreads can be monitored during transient conditions to indicate nozzle problems. Blade failures can be induced by performance and other factors. Specific issues and deterioration problems related to gas and liquid fuels are detailed in (Meher-Homji and Bromley, 2010)

Recoverable and Non-Recoverable Degradation

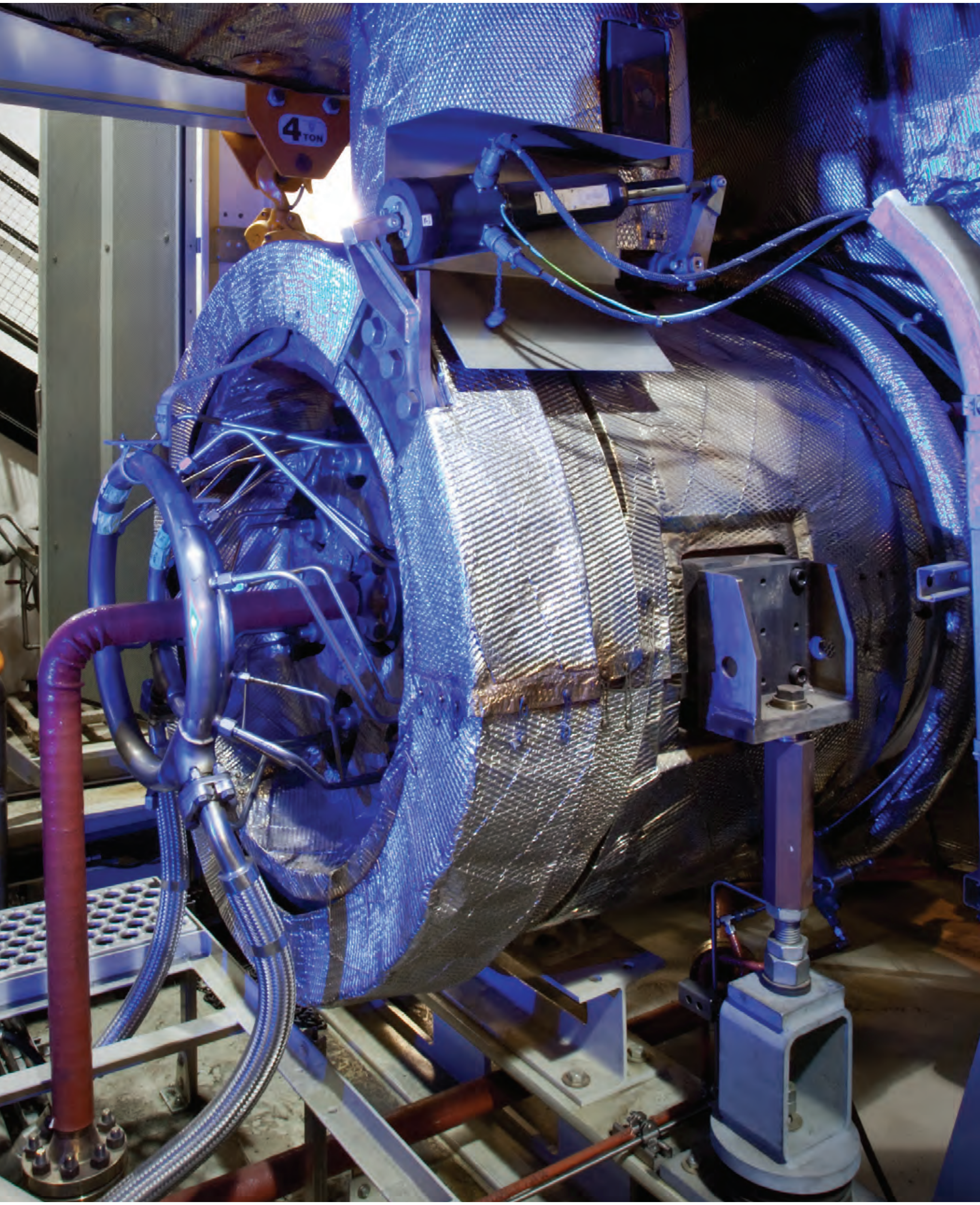
The distinction between recoverable and non-recoverable degradation is somewhat misleading. The majority of degradation is recoverable, however the effort is very different depending on the type of degradation. The recovery effort maybe as small as on-line washing, or on-crank washing. The degradation recovery, by any means of washing, is usually referred to as recoverable degradation. However, a significant amount of degradation can be recovered by engine adjustments (such as resetting variable geometry). Last, but not least, various degrees of component replacement in overhaul can bring the system performance back to as-new conditions.

Protection against Degradation

While engine degradation cannot be entirely avoided, certain precautions can clearly slow down the effects on performance. These precautions include the careful selection and maintenance of the air filtration equipment, and the careful treatment of fuel, steam or water that are injected into the combustor, as well as the careful treatment of water used for evaporative cooling or fogging purposes. It also includes carefully following manufacturers' recommendations regarding shut-down and restarting procedures.

The site location and environment conditions, which dictate airborne contaminants, their size, concentration and composition, need to be considered in the selection of air filtration. Atmospheric conditions, such as humidity, smog, precipitation, mist, fog, dust, oil fumes, industrial exhausts, will primarily affect the engine compressor. Fuel quality will impact the hot section. The cleanliness of the process gas, entrained particles or liquids, will affect the driven equipment performance. Given all these variables, the rate of degradation is impossible to predict with reasonable accuracy.

Thorough on-crank washing can remove deposits from the engine compressor blades, and is an effective means for recovering degradation of the engine compressor. The engine has to be shut down, and allowed to cool down prior to applying detergent to the engine compressor while it rotates at slow speed. On-line cleaning, when detergent is sprayed into the engine running at load, can extend the periods between on-crank washing, but it cannot replace it. If the compressor blades can be accessed with moderate effort (for example, when the compressor casing is horizontally split), hand cleaning of the blades can be very effective.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CHAPTER 12

ADVANCED CYCLES & PERFORMANCE AUGMENTATION

In the previous sections, we discussed gas turbines operating in the simple Brayton cycle. This leads to a relatively low cost, low-complexity installation with a very high power density. However, the Brayton cycle has some inherent disadvantages:

- Insufficient utilization of exhaust heat
- Limited efficiency
- High sensitivity to ambient conditions
- Efficiency increase by increasing firing temperature
  - Emissions issues
  - Materials issues

Improvements to the Brayton cycle can be achieved; however, they typically increase the complexity and cost of an installation. We will describe some of the possible improvements to the Brayton cycle, such as:

- Cogeneration
- Combined cycles
  - Brayton and Rankine Cycles
  - Cheng Cycle (Steam-Injected Gas Turbine)
- Recuperation
- Intercooling
- HAT/CHAT
- Inlet cooling, storage
- Fuel cells

COGENERATION

Cogeneration is the simultaneous production of two useful energy forms from the same source. Examples would be a gas turbine driven generator that produces electricity, while the gas turbine exhaust gas is passed through a heat recovery boiler to produce steam. In conjunction with a gas turbine, cogeneration is typically applied in a “topping cycle”, where the fuel is burned primarily to produce electricity, while the exhaust heat is used to satisfy process heat requirements. In a “bottoming cycle”, the fuel is burned first to satisfy the thermal requirements of the process, while the excess heat rejected from the process is used to generate electricity.

COMBINED CYCLES

One of the characteristics of the processes using the Brayton cycle is relatively high exhaust temperature. The available pressure ratio of the gas turbine limits the amount of heat that can be converted into work. If the available exhaust heat can be used, significant improvements in the overall efficiency are possible. A boiler and potentially another heat source (e.g., a duct burner) to increase the exhaust temperature are used to generate steam *(Figure 12-1)*.

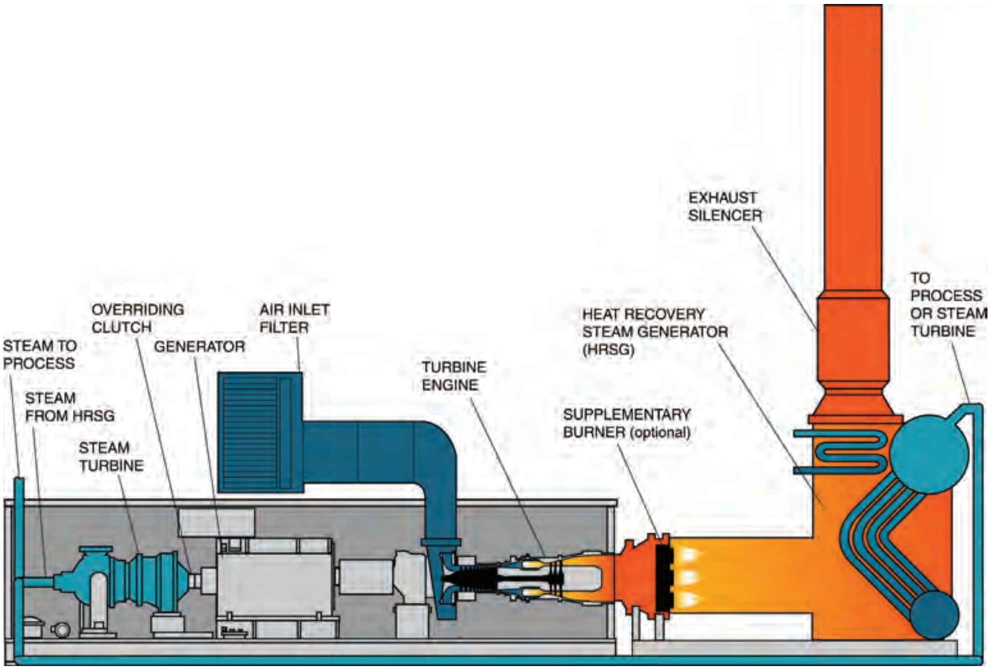

EXHAUST SILENCER
TO PROCESS OR STEAM TURBINE
OVERRIDING CLUTCH
AIR INLET FILTER
HEAT RECOVERY STEAM GENERATOR (HRSG)
STEAM TO PROCESS
GENERATOR
TURBINE ENGINE
STEAM FROM HRSG
SUPPLEMENTARY BURNER (optional)
STEAM TURBINE

Figure 12-1. *Gas Turbine and Steam Turbine Driving a Generator*

The steam is then used to drive a steam turbine. *Figure 12-2* shows the temperature/ entropy diagram for this combination of a Brayton cycle (for the gas turbine) and a Rankine cycle (for the steam turbine). The Cycle 1-2-3-4 is the gas turbine cycle as discussed in Chapter 1. The exhaust heat from Point 4 is subsequently used to heat the feed water (A-B), evaporate the water (B-C) and superheat the steam (C-D). The steam is expanded through a steam turbine (D-E) and afterwards condensed in a condenser (E-A).

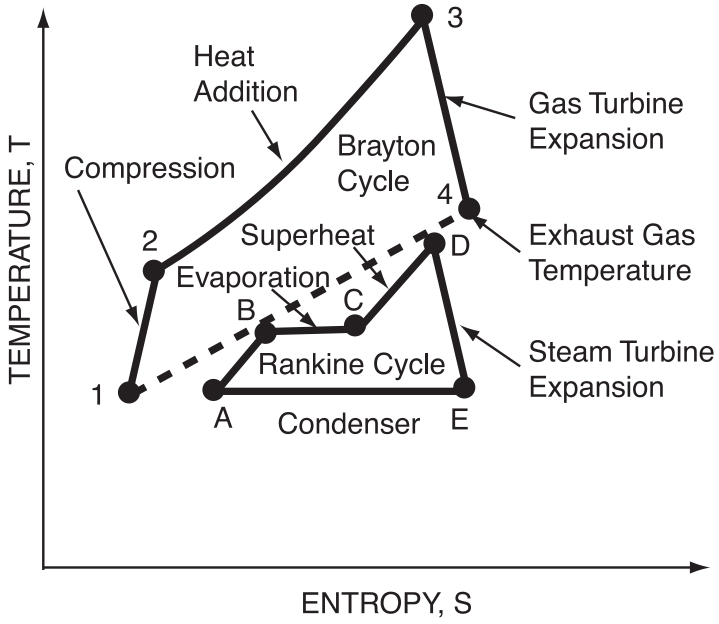

3
Heat
Addition
Gas Turbine
Expansion
Brayton
Cycle
Compression
TEMPERATURE, T
4
Exhaust Gas
Temperature
2
Superheat
D
Evaporation
C
B
Steam Turbine
Expansion
Rankine Cycle
1
A
Condenser
E
ENTROPY, S

***Figure 12-2.** Combined Cycle Gas Turbines – Temperature/Entropy Diagram*

The steam is used to operate a steam turbine that either drives a separate generator or the same generator as the gas turbine, as in a Steam Turbine Assisted Cogeneration (STAC) system *(Figure 12-1)*. Flexibility is gained by adding a clutch between the generator and the steam turbine. The STAC system is optimized around flexibility, i.e., the system can be used to vary the electrical power output based on the needs of the process steam load.

The amount and quality of steam produced depends not only on the available exhaust heat (H_{ex}):

$$H_{ex} = c_p \cdot W_{ex} \cdot (T_7 - T_{sink})$$

but, also on the temperature (T_7) at which the exhaust gases are available. Heat transfer in a heat exchanger follows:

$$H = k \cdot A \cdot \Delta T$$

with k being the heat transfer coefficient, A the surface area and ΔT the average temperature difference between the exhaust gas and the coolant. The term kA can be considered constant once the boiler geometry is defined. The average temperature difference ΔT depends also on the heat exchanger geometry. In a counterflow arrangement, ΔT becomes:

$$\Delta T = \frac{\left(T_{exh,in} - T_{w,out}\right) - \left(T_{exh,out} - T_{w,in}\right)}{1n\left(\frac{T_{exh,in} - T_{w,out}}{T_{exh,out} - T_{w,in}}\right)}$$

Assuming we know the steam temperature ($T_{w,out}$), the feedwater temperature ($T_{w,in}$) and the exhaust temperature ($T_7=T_{exh,in}$) as well as the exhaust flow (W_{ex}) and additionally kA from the boiler geometry, then we can calculate the stack temperature ($T_{exh,out}$) from:

$$\frac{W_{exh} c_{p,exh}}{kA} = \frac{\left(T_{exh,in} - T_{w,out}\right) - \left(T_{exh,out} - T_{w,in}\right)}{1n\left(\dfrac{T_{exh,in} - T_{w,out}}{T_{exh,out} - T_{w,in}}\right)}$$

Knowing the stack temperature, we can now calculate the average temperature difference between the exhaust gas and the water/steam (ΔT). Then we can calculate the transferred heat (H), as well as the resulting steam flow (W_w) (if the steam enthalpy is defined) or the steam enthalpy (Δhw) (if the steam flow is defined) from:

$$W_w \cdot \Delta h_w = c_{p,ex} \cdot W_{ex} \cdot (T_{exh,in} - T_{exh,out}) = c_{p,ex} \cdot W_{ex} \cdot (T_7 - T_{stack}) = H$$

If the temperature difference drops (as T_7 drops), less heat can be transferred to the steam. On single-shaft engines without variable inlet geometry, T_7 will drop at part load while W_{ex} will stay roughly constant. With variable inlet guide vanes, T_7 will stay roughly constant, but W_{ex} will drop *(Figure 12-3)*.

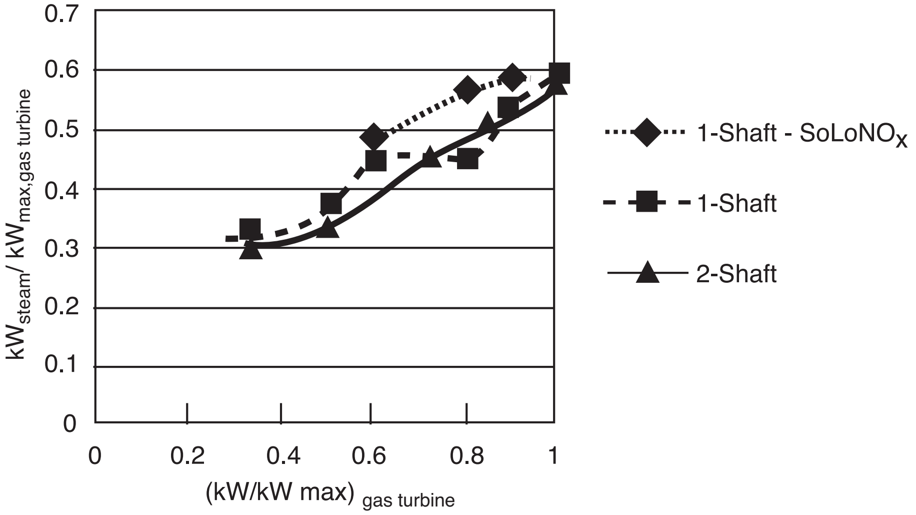


Figure 12-3. *Steam Power at Part Load for Three Different Engine Concepts: Single-Shaft with VIGVs (for Dry Low NO_x Control), Conventional Single-Shaft, Two-Shaft*

In a Steam-Injected Gas Turbine (STIG) cycle or Cheng cycle, the steam is injected into the gas turbine *(Figure 12-4)*. This approach eliminates the need for a steam turbine and also allows some flexibility whether the steam is used to generate electricity or process steam. However, since the gas turbine section has to act both as steam and as gas turbine (and has to swallow the added steam mass flow), the flexibility is limited because the process steam conditions may not match the steam injection conditions in the turbine. Also, the turbine usually requires a higher feed water quality than a steam turbine due to the higher process temperatures.

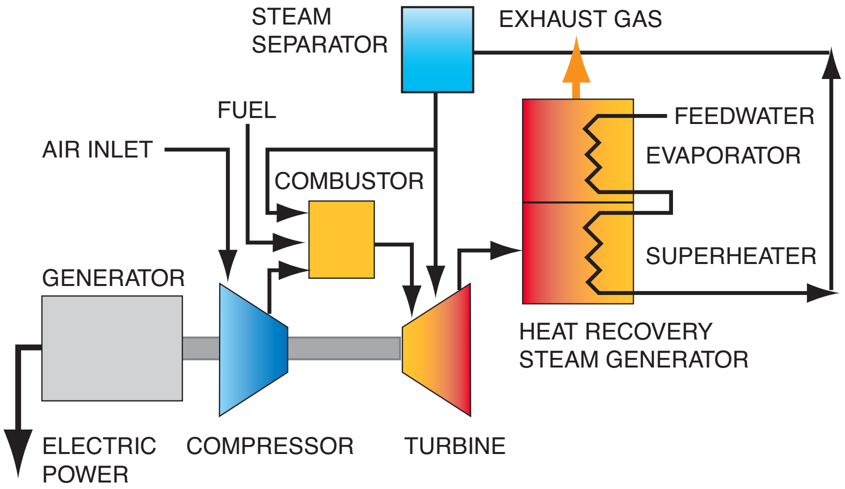

STEAM
SEPARATOR
EXHAUST GAS
FUEL
AIR INLET
COMBUSTOR
FEEDWATER
EVAPORATOR
SUPERHEATER
GENERATOR
HEAT RECOVERY
STEAM GENERATOR
ELECTRIC
POWER
COMPRESSOR
TURBINE

Figure 12-4. *Steam Injected Gas Turbine Cycle (STIG, Cheng Cycle)*

RECUPERATED CYCLE

Another method uses a heat exchanger to preheat the compressed air from the air compressor before it enters the combustor *(Figures 12-5 and 12-6)*. Therefore, the heat input from 2 to 2' can be accomplished without using fuel. Obviously, this is more effective the larger the temperature difference becomes between the exhaust gas and the compressor discharge gas (T_7-T_2). An engine that is recuperated is typically designed for

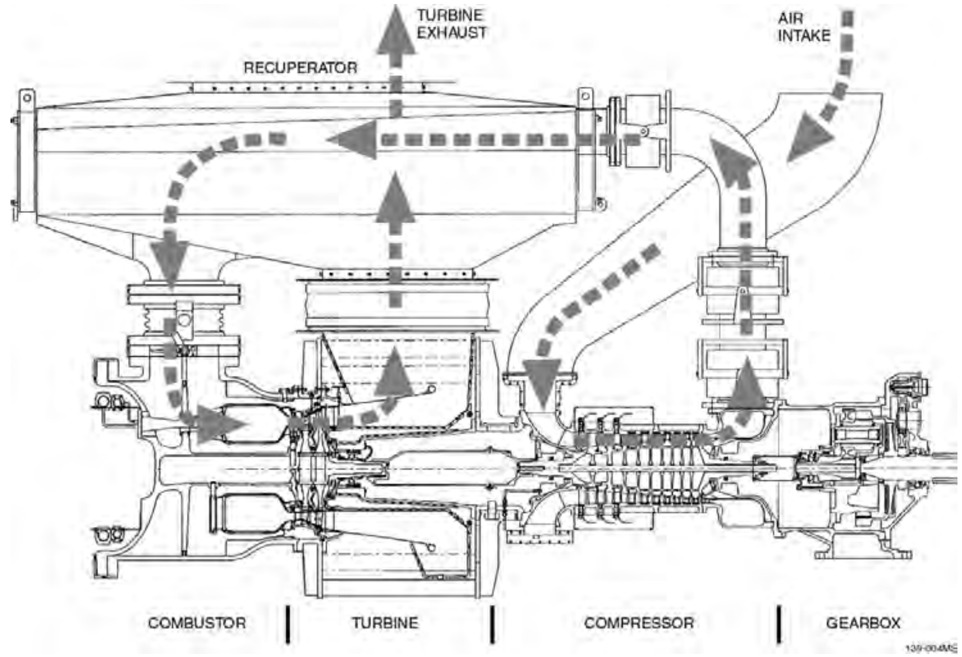

TURBINE
EXHAUST
AIR
INTAKE
RECUPERATOR
COMBUSTOR
TURBINE
COMPRESSOR
GEARBOX
139-004MS

Figure 12-5. *Recuperated Cycle Gas Turbine*

lower pressure ratios than a non-recuperated engine, since a lower pressure ratio leads to lower compressor discharge temperatures, which increases the potential efficiency improvements *(Figure 12-6)*.

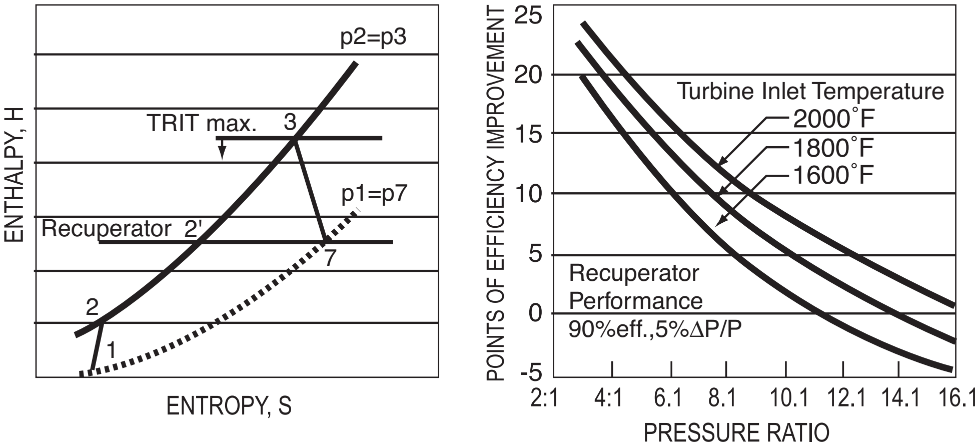
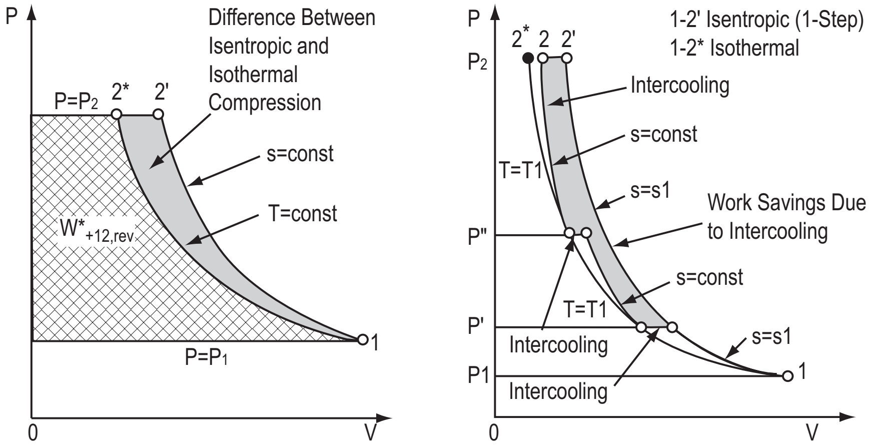


Figure 12-6. *Recuperated Cycle*

INTERCOOLING

The power requirement for the engine compressor to achieve a certain pressure ratio can be significantly reduced by introducing a gas cooler after some stages of compression. As *Figure 12-7* shows, a given compression requirement can be achieved with the least amount of work if an isothermal process can be achieved. Intercooled compression allows the compression process to be closer to an isothermal process, thus saving power.


P
Difference Between
Isentropic and
Isothermal
Compression
P=P2
2*
2'
s=const
T=const
$W^*_{+12,rev}$
P=P1
1
0
V
P
2*
2
2'
P2
1-2' Isentropic (1-Step)
1-2* Isothermal
Intercooling
s=const
T=T1
s=s1
Work Savings Due
to Intercooling
P"
s=const
T=T1
P'
Intercooling
s=s1
P1
1
Intercooling
0
V

Comparison of isentropic and isothermal processes in the p-v diagram

Savings in power compression in 3 steps with intercooling compared to compression without intercooling

Figure 12-7. *Effect of Intercooling on Compression Power Requirement*

Because the compression power requirement is lost for the power output of the engine, a significant increase in power output and efficiency could be realized.

However, the use of intercoolers yields pressure losses due to the flow losses in the cooler and the necessary volutes and piping. Also, unless cooling water is available, air-cooled gas coolers require a significant amount of power for driving the cooler fans.

INLET COOLING

Simple- and combined-cycle gas turbine concepts exhibit a significant loss of output power with increasing ambient temperatures (see also Chapter 7). This can actually lead to acute power shortages on hot days. Many efforts therefore, are, made to reduce the actual engine inlet temperature below the ambient temperature.

A number of different gas turbine inlet cooling power augmentation technologies are currently available commercially. They can be generally classified into two categories:

1. ***Evaporative Cooling:*** These include wetted media, fogging, wet compression, overspray, and interstage injection.
2. ***Chillers:*** Mechanical and absorption chillers with or without thermal energy storage.

In general, chillers only affect the gas turbine by directly cooling the inlet air temperature. This increases the gas turbine's output power but otherwise has minimal effects on the gas turbine's internal aerodynamics. Inlet chillers are generally very large, expensive to operate, and rarely offer more than marginal economic benefits. Thus, they are not widely employed. On the other hand, evaporative inlet cooling relies on injecting water droplets or vapor into the gas turbine's compressor (either upstream or interstage). Two different principles are employed: evaporative coolers utilize a wetted media that is exposed to the inlet air flow while inlet fogging systems spray water mist into the gas turbine upstream inlet system. Both systems are normally designed to avoid liquid water carryover into the engine inlet but fogging systems can sometimes (unintentionally or intentionally) "overspray," which results in water droplets entering the gas turbine compressor.

The primary difference between the various commercially-available evaporative cooling technologies is the quantity of water (percent air saturation), the water droplet size, and the location of the water injection ports into the gas turbine. However, the basic functional principle of all inlet water-augmentation technologies is that they effectively reduce the gas turbine's inlet air temperature from the air's dry bulb temperature to the wet bulb temperature of the ambient air. This effective temperature difference depends on the ambient air's relative humidity and temperature, as well as the evaporative cooler's efficiency as shown on a basic psychiometric chart for water *(Figure 12-8)*.

Several arrangements for inlet water cooling are possible but the most common is to place fine mist, water-atomizing nozzles just downstream of the gas turbine's inlet filter. *Figure 12-9* shows a schematic of a typical gas turbine inlet cooling fogging system. *Figure 12-10* summarizes typical output power net gains that can be achieved with various inlet cooling techniques.

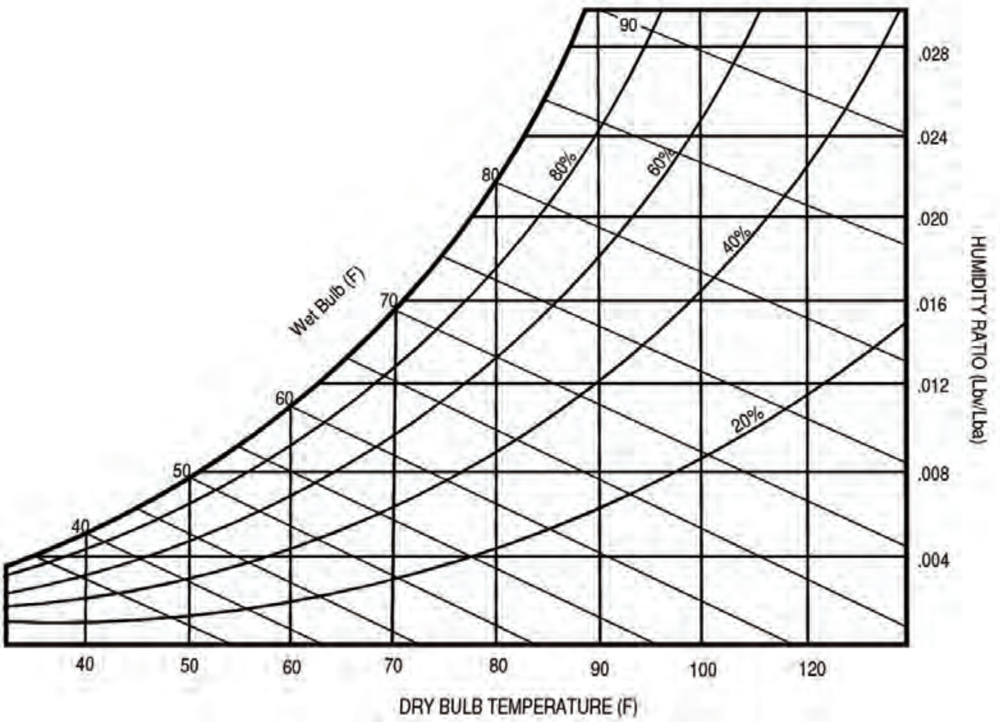


Figure 12-8. *Water Psychiometric Chart Relates Dry Bulb and Wet Bulb Temperatures*

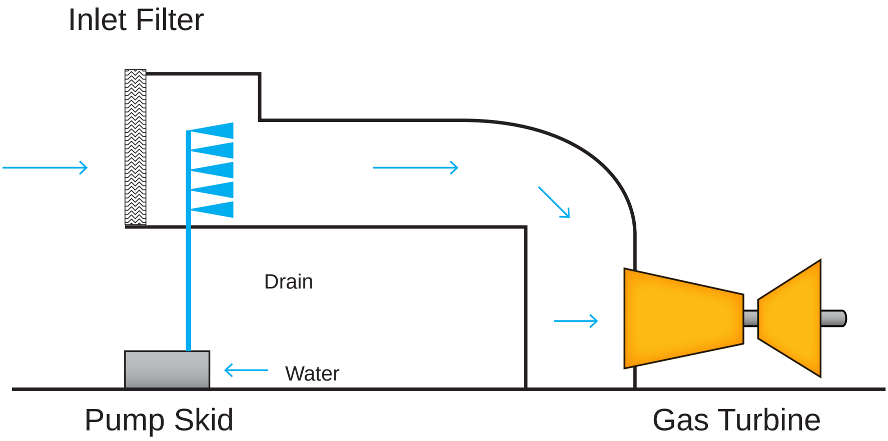


Figure 12-9. *Gas Turbine Inlet Cooling Using Atomizing Water-Fogging Nozzles*

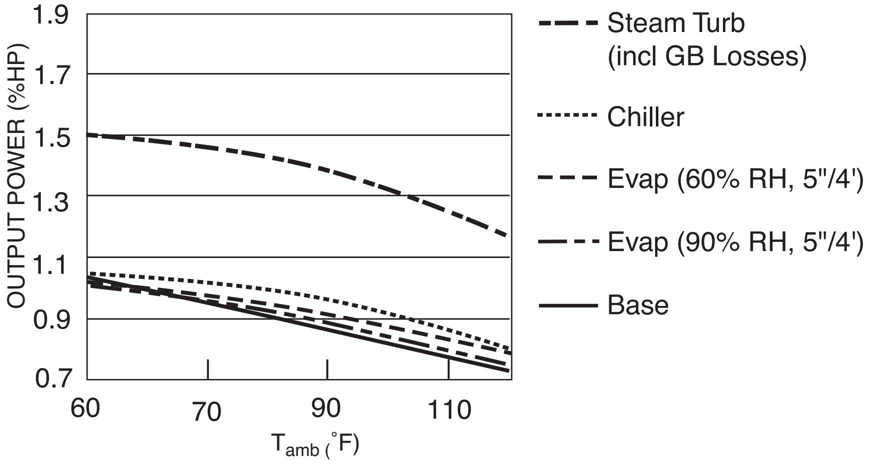


Figure 12-10. *Cooling, Inlet Chilling, and Combined-Cycle Application*

FUEL CELLS

Fuel cells offer a highly-efficient way of converting chemical energy into electricity, without an intermediate combustion process, with subsequent conversion of mechanical energy into electricity *(Figure 12-11)*. Having said this, one may wonder why fuel cells are discussed in a book about gas turbines. The reason is that there are cycle concepts that integrate a gas turbine into the fuel cell cycle.

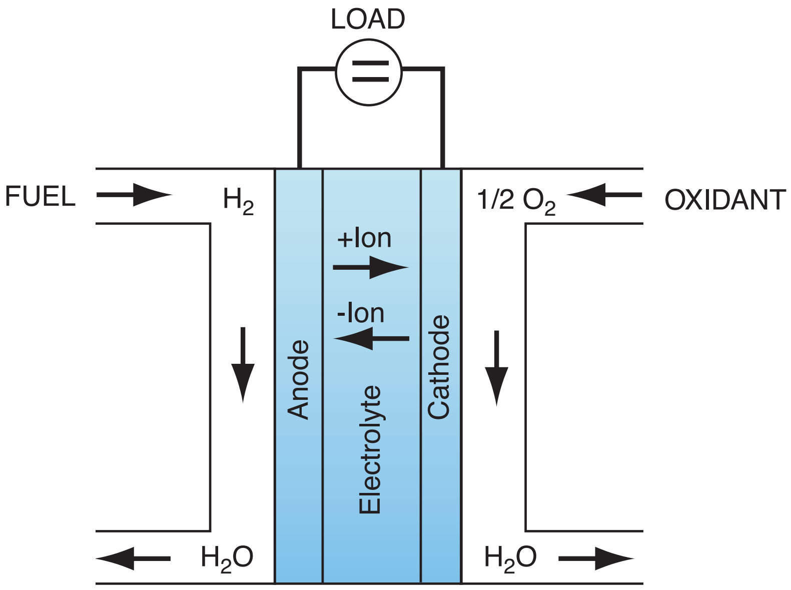

LOAD
FUEL
H_2
1/2 O_2
OXIDANT
+Ion
-Ion
Anode
Electrolyte
Cathode
H_2O
H_2O

Figure 12-11. *Fuel Cell*

Fuel cells, in particular Solid Oxide Fuel Cells (SOFC), exhibit a significant increase in power density and efficiency when operated at elevated temperatures and pressures. Further, the conversion process in a fuel cell is exothermic; i.e., it generates heat. So, a possible integration method would use a gas turbine compressor to produce air at elevated temperature and pressure (at a pressure ratio of around 6). This air is used as the oxidant in the fuel cell. The exhaust gas from the fuel cell is then expanded through the turbine part of the gas turbine. The remaining heat can be used in a fuel reformer (*Figure 12-12),* which is typically endothermic. Depending on the cycle design, the fuel cell may be the sole source of electrical energy, while the turbine section is used only to drive the compressor, or both the fuel cell and a turbine-driven generator produce electricity.

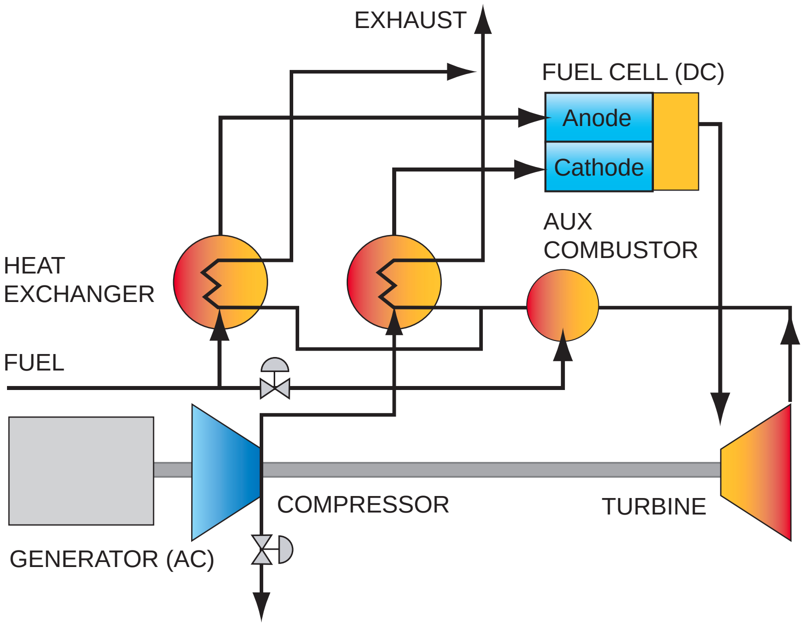


Figure 12-12. *Gas Turbine Integrated in a Fuel Cell Cycle*

Fuel cell pressurization increases the performance of the fuel cell and system at the cost of providing the pressurization. Fundamentally, the question of pressurization is a trade-off between the improved performance (and/or reduced cell area) and the reduced piping volume, insulation, and heat loss compared to the increased parasitic load and capital cost of the compressor and related equipment.

For SOFCs, a high temperature of approximately 1000°C (1830°F) is required in order to have sufficiently high ionic conductivities with the existing materials and configurations.

ORGANIC RANKINE AND SUPERCRITICAL CO_2 BOTTOMING CYCLES

Similar to the previously discussed steam Rankine combined-cycle concept, other closed power cycles can be utilized to efficiently convert gas turbine exhaust heat into mechanical or electrical energy. These cycles are of special interest in gas turbine applications where water is not available, such as in remote pipeline compression or oil and gas production facilities. Instead of utilizing water and steam, organic fluids such as Pentene or inert gases such as Helium or supercritical CO_2 are utilized in a closed cycle as the process fluid. These cycles can be fairly simple with a single compressor or pump, heat exchanger, and turbine in a simple loop, as shown in *Figure 12-13*, or much more complex with multiple loops of recuperators, re-compressors, and staged turbines.

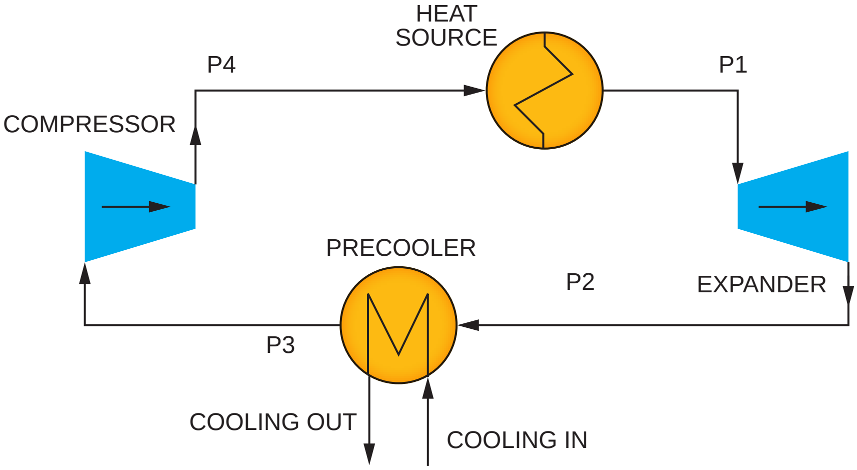


Figure 12-13. *Simple Closed Brayton Power Cycle for Waste Heat Applications*

Depending on the application, the gas turbine exhaust temperature, and the mass flow, the bottoming cycle can thus be optimized for the highest efficiency, the smallest footprint and weight, or the least complexity. For example, *Figure 12-14* shows a recompression supercritical CO_2 power cycle optimized for highest efficiency in gas turbine waste heat-recovery applications. Nonetheless, all these cycles function basically the same in that they utilize the hot exhaust air from the gas turbine in a heat exchanger to heat the compressed process fluid and then convert the heat energy into mechanical shaft output power in one or multiple-turbine sections. The type of cycle can involve a vapor-liquid phase change (Rankine), operate purely within the vapor phase (Brayton), or function as a mix of the two (hybrid). Organic Rankine cycles tend to be preferred for applications where the waste heat temperature is below 400 degrees Celsius while supercritical CO_2 cycles have the best efficiencies for combined cycle temperatures between 450 and 650 degrees Celsius. Supercritical CO_2 Brayton cycles, because of the high-energy density of the CO_2 working fluid, also tend to be more compact and have slightly higher efficiencies than Rankine cycles.

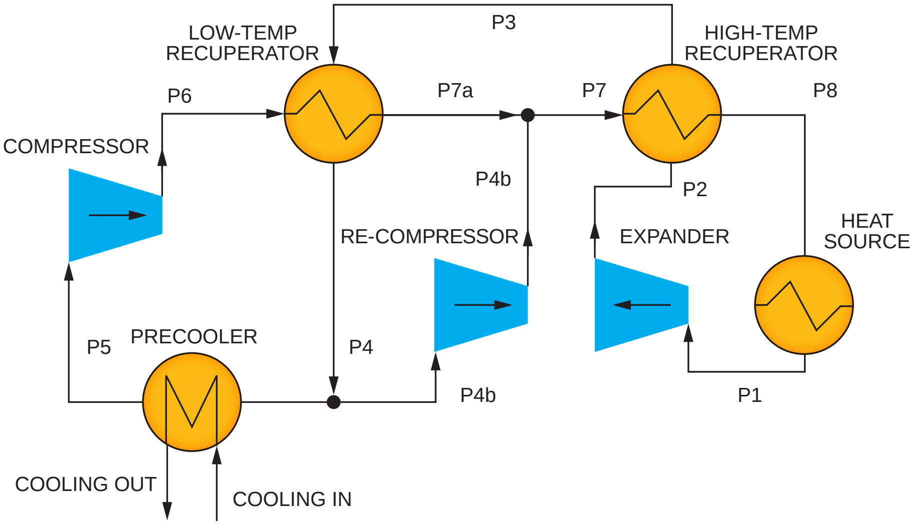

LOW-TEMP
RECUPERATOR
P3
HIGH-TEMP
RECUPERATOR
P6
P7a
P7
P8
COMPRESSOR
P4b
P2
RE-COMPRESSOR
EXPANDER
HEAT
SOURCE
P5
PRECOOLER
P4
P4b
P1
COOLING OUT
COOLING IN

Figure 12-14. *Complex Recompression, Reheat CO_2 Power Cycle Schematic*

The analysis of advanced power cycles can be very complex. Specialized-process analysis software is often used for this purpose. This software links and models the individual cycle components, such as heat exchangers, compressors, turbines, valves, etc. and allows for the optimization of a power cycle for a specific application.

SUMMARY

The Brayton cycle can be improved in many different ways. Some of the technologies described are presently in use, while others (such as the fuel cell and supercritical C) are still in the experimental stage. None of these advanced cycles can improve on the simplicity and the high-power density of the simple-cycle gas turbine, but they are able to improve on some of the inherent short falls of Brayton cycle engines.

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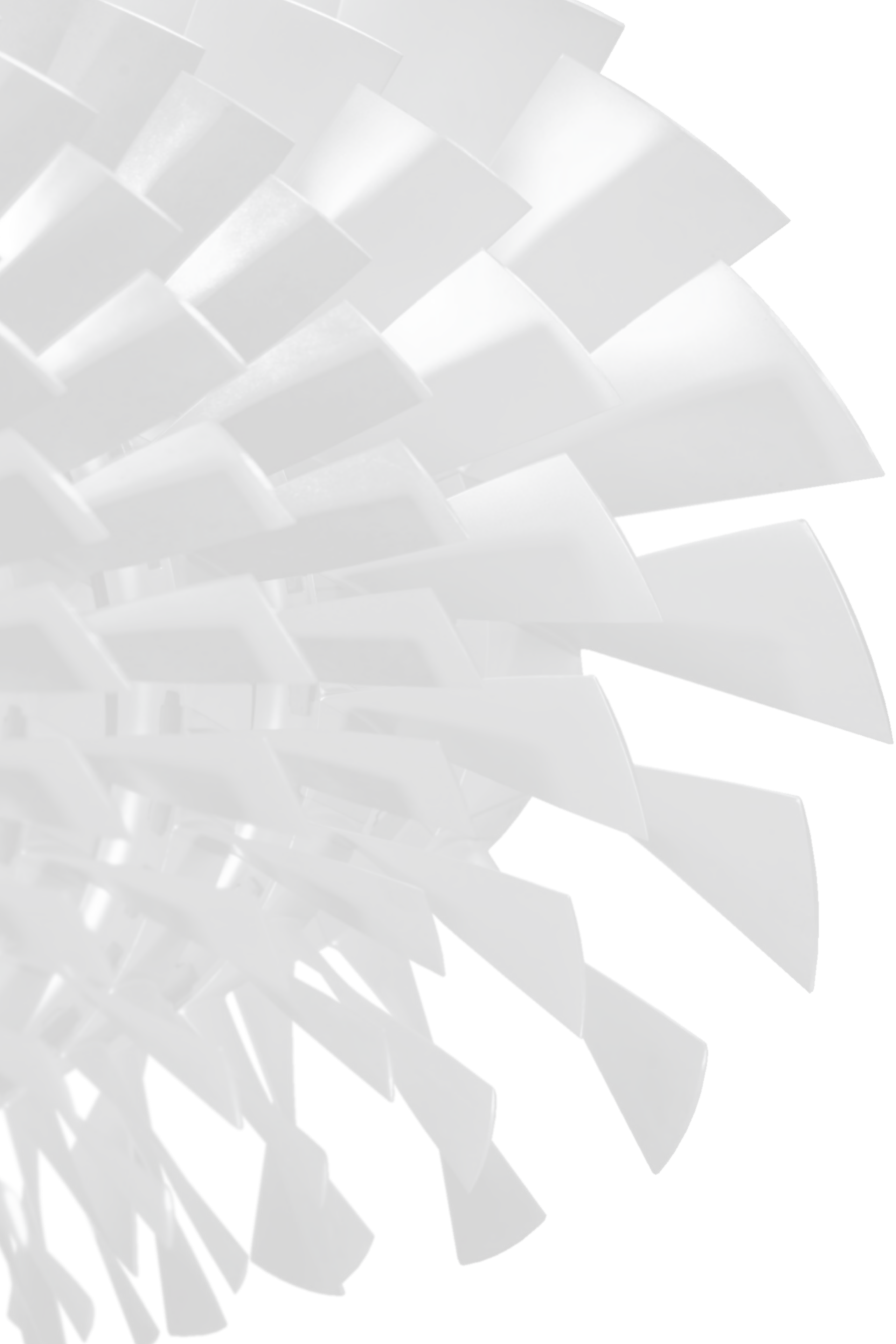

Solar® Turbines
A Caterpillar Company