Chapter 18

Thermodynamic Power Cycles for Pump-Fed Liquid Rocket Engines

Randy C. Parsley*
Pratt and Whitney, United Technologies Corporation, West Palm Beach, Florida

and

Baojiong Zhang†
Shanghai Bureau of Astronautics, Shanghai, People's Republic of China

I. Introduction

THE objective of this chapter is to provide an overview of the thermodynamics that influence the configuration selection of pump-fed liquid propellant rocket cycles. The intent is to provide insight into the fundamental differences and inherent advantages of different cycle approaches. To simplify the explanations, first-order calculations are presented with many secondary influences neglected or simplified by assumption. Expander, gas generator, and staged combustion cycles are explained and compared with special emphasis on the thermodynamic implications of including oxidizer-rich combustion for the turbopump drive cycle. Also discussed are the interactions of the cycle thermodynamics with the engine component stress limitations, thermal limitations, and efficiency trends.

A. Cycle Types and Configurations

Pump-fed liquid rocket cycles are defined by two configuration variables. The first cycle configuration variable is the energy source for the turbine drive. Energy for the turbine can come from an auxiliary combustion device such as

---

*Manager, Propulsion System Analysis and Integration. Associate Fellow AIAA.

†Research Professor.

a preburner or a gas generator, or from the main combustion chamber, either directly by the extraction of combusted propellants or indirectly by heat transfer through the chamber walls. The second cycle configuration variable is the turbine discharge location. Historically, there are two options for the turbine discharge flow. If the turbine discharge is to a high-pressure source, specifically the main combustion chamber, the cycle is referred to as a closed cycle. If the turbine discharge flow is to a low-pressure source, generally overboard or into the nozzle skirt, the cycle is referred to as an open cycle. A third turbine discharge source exists but has no development history—turbine discharge flow into an intermediate-pressure combustion device referred to as an afterburner.

Figure 1 summarizes eight possible cycle configuration options based on these two configuration variables and identifies the common names associated with each cycle. Also included are the turbine gas compositional options, propellant limitations, and examples of operational engines associated with each cycle type. All of the eight cycle options, except the afterburning cycles, have been developed into operational engines. This chapter examines advantages, disadvantages, and fundamental performance limitations of each cycle. For simplicity, the supporting engine schematics do not include boost pumps and are examined with separate turbopumps for the fuel and oxidizer. The schematics include the minimum valve complement required for engine startup and control.

In an open cycle, a minimum fraction of the engine propellant is expanded through the turbines to provide power to the pumps. The turbine flow is discharged either overboard or into the divergent section of the primary nozzle, which allows for a turbine pressure ratio of 5 or greater. Because the propellants for an open cycle are pressurized to only slightly above chamber pressure, the pump work is minimized. For a closed cycle, the turbine drive flow is discharged into the main chamber, which is at a relatively high pressure. This generally

		Turbine Energy Source		
		Main Combustor		Auxiliary Combustor
		Chamber Coolant Heat Limited to Hydrogen, Methane, Propane Fuels	Chamber Combustion Gas All Propellants are Compatible	Gas Generator or Preburner All Propellants are Compatible
Turbine Discharge Pressure Sink	High Pressure: Main Combustor	Closed Expander Fuel Cooling Fuel Cooling with Regeneration Fuel & Oxidizer Cooling (Full Flow) Operational Engine Examples: (RL10)	(No Flow Potential) (None Possible)	Staged Combustion Fuel-Rich Oxidizer-Rich Fuel & Oxidizer-Rich (Full Flow) (SSME, LE-7, RD-170, RD-0120, NK-33)
	Intermediate Pressure: A/B Combustor	Afterburning Open Expander Fuel Cooling (None)	Afterburning Tapoff Fuel-Rich (None)	Afterburning Gas Generator Oxidizer-Rich Fuel-Rich (None)
	Low Pressure: Overboard or in Nozzle Skirt	Open Expander Fuel Cooling (LE-5A)	Tapoff Fuel-Rich (J-2S)	Gas Generator Oxidizer-Rich Fuel-Rich Fuel & Oxidizer-Rich (Full Flow) (F-1, J-2, Vulcain, RS-27, LR87-AJ, YF-20)

Fig. 1 Clarification of liquid rocket power cycles by turbopump power options.

limits the turbine pressure ratio to 2 or less to avoid excessive pump discharge pressures.

For either the open or closed cycle approach, energy must be provided to the turbine working fluid prior to its expansion through the turbine. Depending on the option selected to provide this turbine energy, the cycle definition is different. The three common thermodynamic cycles for liquid rocket engines are *expander*, *gas generator*, and *staged combustion*.

The expander cycle is an open or closed cycle in which the energy to drive the turbine comes from the thermal energy absorbed by propellant used to regeneratively cool the thrust chamber and nozzle. In the closed cycle, this turbine flow is discharged into the main chamber. Consequently, all the pump propellants are combusted in the main chamber and expanded through the primary nozzle. For an open cycle the turbine flow is discharged overboard or into the divergent sector of the nozzle. The energy available for the expander cycle is limited by the thrust chamber and nozzle heat transfer, which limits potential chamber pressure to $\sim$10 MPa.

The gas generator cycle is an open cycle in which the energy to drive the turbine is supplied by combustion of a minimal fraction of propellants in a gas generator combustion device. Because of the high expansion ratio, the pressure of the turbine discharge flow is below the main combustion chamber pressure, and therefore the discharge must bypass main chamber combustion. The chemical energy released during combustion in the gas generator is influenced by the temperature limit of the turbine. Chamber pressure for a gas generator cycle is selected to optimize total engine performance, which includes both the higher performance main engine flow and the lower performance turbine discharge flow. This performance optimum generally occurs at 10–15 MPa of chamber pressure, depending on propellant selection, with an overboard flow generally less than 4% of the total engine flow. For simplification, the turbine exhaust gases are usually expanded through a separate nozzle to provide some thrust. A more complex but higher performing option is to dump this gas into the divergent section of the primary nozzle. A third option would be an intermediate pressure afterburner downstream of the gas generator and turbine discharge. The afterburning option has been investigated because of the potential to offset the performance loss due to the main chamber mixture ratio shift that occurs for open cycle configurations. Although this option increases performance significantly relative to the conventional gas generator, the weight and complexity of the afterburner must be considered and optimized for the intended application.

The staged combustion cycle is a closed cycle in which major portions of the propellants are burned, in preburner combustion devices, to provide energy to drive the turbines. The energy released in the preburners minimizes turbine pressure ratio, allowing chamber pressure to be maximized. The energy released in the preburners is influenced by the turbine temperature limit and the percentage of the propellants included in the combustion process. The performance of a staged combustion cycle generally begins to become hardware limited between 20 and 25 MPa chamber pressure.

Once the propellants and engine mixture ratio have been selected, the performance of the various cycles is influenced by only a few parameters. For the closed cycles (expander and staged combustion), these parameters are engine impulse efficiency, engine chamber pressure, and engine nozzle exit pressure. Engine impulse efficiency is the product of the main chamber combustion energy release efficiency and the primary nozzle expansion efficiency. For oxygen and hydrogen propellants, the main chamber combustion energy release efficiency for a well-designed system is generally 98–99%, whereas for oxygen and kerosene propellants the practical limit has been 95–96%. Typical nozzle expansion efficiencies are generally from 98 to 99%, depending on several factors, including operating nozzle pressure ratio and the design area ratio.

First-order performance trends, as functions of chamber pressure and nozzle exit pressure, are shown in Fig. 2a, for hydrogen and Fig. 2b for kerosene fuels. The theoretical specific impulse was generated using NASA's chemical equilibrium program for rocket performance.[1] This theoretical performance was adjusted according to the indicated constant values of combustion energy release efficiency and nozzle expansion efficiency. The performance is shown for three nozzle discharge pressures p_{exit}. The $p_{\text{exit}} = 1.0$ bar line represents the nozzle expansion ratios generally required to maximize sea-level thrust, which is desirable and typical for booster applications. The $p_{\text{exit}} = 0.3$ bar line represents the maximum sea-level expansion ratio that can typically be sustained without nozzle separation, used for engines that must start at sea level but also operate at high altitudes. The $p_{\text{exit}} = 0.1$ bar line is representative of upper-stage expansion ratios that balance performance, weight, and engine geometric size. These performance estimates are presented for initial screening activities only. The secondary effects of changes in combustion energy release efficiency and nozzle efficiency depending on each individual design are important and should be investigated and optimized for each individual application.

B. Pump-Fed Powerhead Power Balance

In a powerhead device, propellants are increased in pressure using single- or multiple-stage pumps. Power to drive the pumps is supplied by single- or multiple-stage turbines. The work relationship for pumping propellants is given by Eq. (1). The required work is related to flowrate $\dot{m}_p$, liquid propellant density ρ, pressure rise Δp, and pump efficiency η_p:

$$W_p = \frac{\dot{m}_p \Delta p}{\rho \eta_p} \tag{1}$$

The delivered turbine work is given by Eq. (2). The delivered power is related to flowrate $\dot{m}_t$, turbine pressure ratio TPR, turbine inlet temperature T, gas constant R, gas specific heat ratio γ, and turbine efficiency η_t:

$$W_t = \dot{m}_t \left(\frac{\gamma}{\gamma - 1} \right) RT \left[1 - \left(\frac{1}{TPR} \right)^{\frac{\gamma-1}{\gamma}} \right] \eta_t \tag{2}$$

Rocket cycle performance is generally improved by increasing the cycle operating pressure (i.e., pump discharge pressure). This trend continues until component thermal or structural limits are reached or until component efficiencies

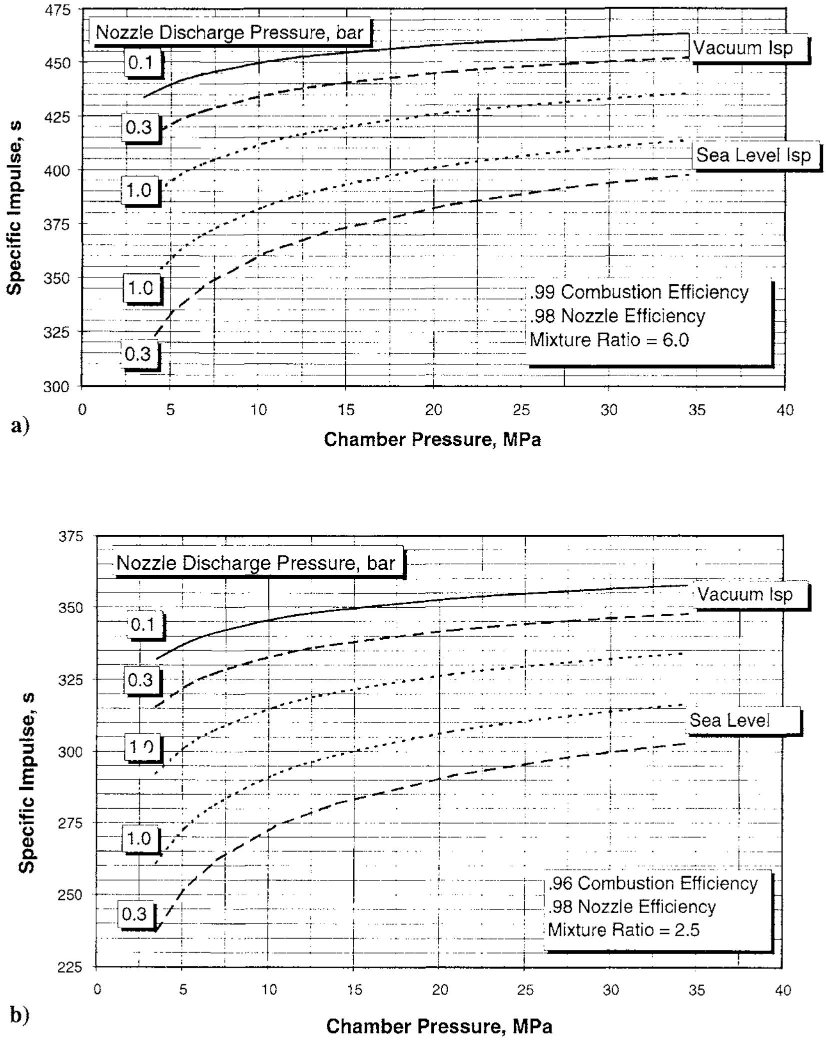

Fig. 2 Performance trends for a) hydrogen/oxygen and b) kerosene/oxygen.

begin to drop. Pump and turbine efficiencies generally decrease as cycle pressure is increased, because the components are forced to operate at increased work levels. This component efficiency loss can be mitigated by increasing the number of pump and turbine stages, but only at the expense of engine hardware complexity, engine cost, and engine weight. Although neglected in the simplified discussion in this chapter, turbine power must be greater than the pumping power requirements by the amount of the connecting shaft friction losses (windage) and the turbine power control margin needed to achieve practical steady-state operation. The sum of these real-world effects can be as much as 10% of the total turbine power.

C. Thermodynamic and Hardware Interactions

Optimization of liquid rocket engines requires closure between the cycle thermodynamics and the component hardware configuration. The engine hardware must be startable and controllable, and weight must be minimized. The thermodynamic cycle and configuration must be compatible with a development program that considers cost, technical risk, and schedule. For expendable applications the acquisition cost of the hardware is important, whereas for reusable applications the hardware durability is the primary consideration. Four of the most common hardware limitations that influence the overall cycle optimization are turbine temperature limits, thrust chamber cooling requirements, turbopump rotational speed limits, and pump discharge pressure limits.

1. Turbine Temperature Limits

For most rocket engine cycles, it is desirable to maximize turbine temperature within practical limits. For the gas generator cycle, this minimizes the overboard flow and maximizes performance. For staged combustion cycles, increasing temperature allows a reduced turbine pressure ratio at a given pump discharge pressure, enabling an increase in chamber pressure, and therefore, as shown in Fig. 2, an increase in performance.

However, increased turbine temperatures can significantly increase the cost of the hardware and can have an adverse effect on hardware durability. For most applications it is desirable to maintain turbine temperatures at or below 900 K. However, with an increased degree of difficulty and cost, turbine temperatures up to 1200 K can be achieved, using state-of-the-art turbine materials, while maintaining durability and reliability margin.[2]

2. Thrust Chamber Cooling

For most propellant combinations, the fuel has superior heat-transfer characteristics relative to the oxidizer; therefore, a portion of the fuel is used to cool the main combustion chamber. For fuel-rich closed staged combustion cycles, the chamber coolant flow rate should be minimized, which allows more fuel to be available for preburner combustion to drive the turbopumps. The percentage of the fuel required for most cycles is generally less than 20%. For oxidizer-rich staged combustion gas generator, and closed expander cycles, most or all of the fuel is used in the chamber cooling process. Because it is desirable to maximize the chamber heat transfer, yet minimize the chamber coolant pressure drop, the optimum trade will depend on the chosen cycle details and on the size of the chamber and the engine. For the gas generator cycle and open expander cycle, minimum pressure drop will minimize pump discharge pressure, improving the cost and weight characteristics of the engine.

For high-pressure kerosene cycles, significantly more than 20% cooling flow may be required. This is a contributing factor to the use of oxidizer-rich staged combustion cycles for this propellant. Because only a small amount of kerosene is needed for the preburner supply, over 90% of the kerosene is available for chamber cooling.

3. *Turbopump Rotational Speed*

Turbopump rotational speed must be optimized to minimize the pump diameter and improve the pump and turbine efficiencies. In many cases, especially for oxygen and hydrogen propellants, the rotational speed should be maximized. Turbopump rotational speed is generally limited by one of four parameters: pump inlet cavitation, bearing rotational speeds, rotordynamics, or turbine blade root-stress.

Turbopump cavitation occurs when the local static pressure of the propellant, as it accelerates around the pump impeller inlet blading, drops below the vaporization pressure. The result is the formation of small pockets of vapor in the low-pressure regions. For a constant inlet flow rate, increased pump speed will increase these local accelerations due to increasingly severe inlet incident angles between the propellant and the blading. Cavitation causes loss of pumping capability and can result in physical damage to the pump hardware. Low-speed boost pumps are often used to increase the main pump inlet pressures to avoid cavitation and allow main pump rotational speed to be increased.

The bearing rotational speed for conventional rotor support systems becomes limited as internal stress in the bearing increases. The bearing stress level is proportional to the product of the inner diameter of the bearing D in mm, and the bearing rotational speed N in revolutions per min. The practical DN limit for state-of-the-art rolling element bearings is $DN \sim 2.0 \times 10^6$. Attempts to reduce the DN by decreasing the diameter will result in increased flexibility of the shaft, which can lead to rotodynamic vibrations. Fluid film bearings are being developed to eliminate this constraint.[3,4] Fluid film bearings support the rotor on high-pressure propellant, eliminating internal surface-to-surface contact during high-speed operation.

Rotordynamic vibrations can occur when the natural bending or natural vibration frequency of the pump rotor falls within the operating speed range. If these natural vibration frequencies "couple" with the rotational energy of the rotor, the pump can fail mechanically. To avoid these vibrations, the pump must operate at a rotational speed below the natural vibration frequency or must transition quickly through the natural frequency modes at a very low speed, where the rotational energy is low. High-speed rotor damping, developed in Russia over the last two decades, has been shown to preclude many of the historical problems with rotordynamic vibration.

The turbine blade root-stress is proportional to the product of the turbine rotor exit annulus area A in cm^2, and the turbine rotational speed squared N^2, in revolutions per min. The practical limit for this parameter is influenced by the operating temperature because the turbine blade material strength drops as temperature increases. For temperatures of 1000 K the practical limit is $AN^2 \sim 3.0 \times 10^{11}$ mm^{2}rpm^2.

4. *Pump Discharge Pressure Limitations*

Staged combustion cycle performance is generally limited by pump discharge pressure. The fuel is generally the limiting propellant because of the lower density of fuel, relative to oxidizer. Lower density results in increased power requirements. Current state-of-the-art limits for pumping hydrogen is ~50 MPa, and for kerosene ~100 MPa. These limits result from the combination of allowable impeller tip speed and allowable number of pump stages. The allowable tip speed of ~700 m/s allows acceptable stresses to be maintained for the impeller. The maximum number of stages is generally limited to three to avoid pump integration concerns such as rotor vibrations, rotor thrust balance, and total power transmitted between the turbine and impellers.

D. Fuel-Rich vs Oxidizer-Rich Combustion for Turbine Drive

Material temperatures are the limiting features in turbines when two propellants are burned to release chemical energy for powerhead work requirements. These temperature limits can be achieved by selecting propellant mixture ratios that are either fuel rich or oxidizer rich. Figure 3 illustrates the relationship of combustion temperature to mixture ratio for oxygen with either hydrogen or kerosene fuel.

Figure 3 shows that 900 K can be achieved for kerosene and hydrogen at fuel-rich mixture ratios of 0.055 and 0.775, respectively, but also at oxidizer-rich mixture ratios of 41.5 and 115.0, respectively. Selection of fuel-rich or oxidizer-rich preburner combustion for a particular application depends on the cycle configuration and propellant selections. There are fundamental thermodynamic differences in energy release potential and net powerhead work potential

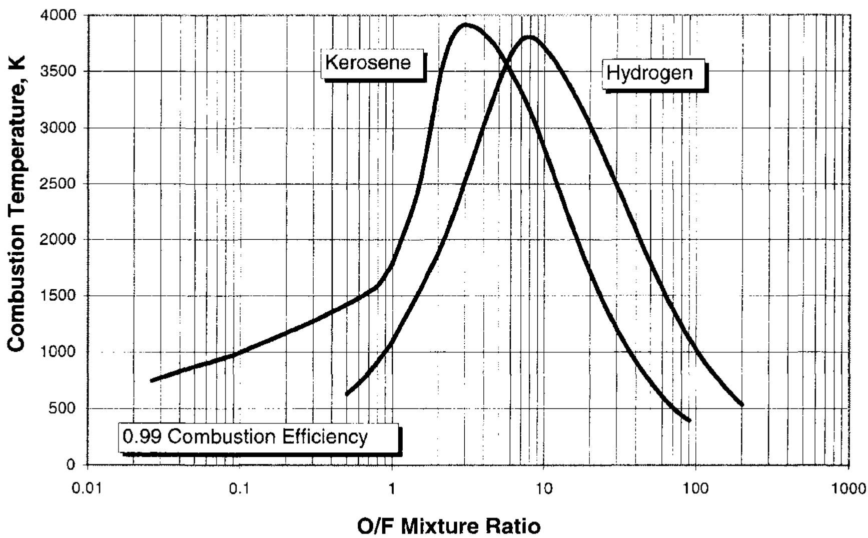

Fig. 3 Combustion temperature vs mixture ratio.

between the various fuel-rich and oxidizer-rich approaches, as discussed in the following sections.

1. Energy Release Potential

The propellant combustion process is a complicated series of mostly exothermic chemical reactions. When fuel and oxidizer are burned, there is a net positive enthalpy release, the magnitude of which is represented by the lower heating value (LHV) of the fuel. More specifically, LHV is the net positive energy, per unit mass of fuel, that is released at constant temperature when the fuel is completely combined with oxygen. This net positive enthalpy release is then absorbed by the combustion products, which increases the resulting energy state of the combustion mixture.

Depending on the selected propellants and mixture ratios, the energy will be absorbed in different ways. For a very oxidizer-rich combustion process, the combustion products are nearly 90% oxygen. In this case the majority of the net positive enthalpy will be used to vaporize and heat the excess oxygen that surrounds the local combustion processes in particular abundance. For a fuel-rich hydrocarbon mixture, the net positive enthalpy will be mostly absorbed by the "cracking" of the excess unburned fuel molecules and then heating of the resulting constituents. For a fuel-rich hydrogen mixture, the positive enthalpy will vaporize and heat the excess flow. In either case the absorption of the liberated LHV increases the net energy state of the combustion products, making them more useful to provide turbine work.

For closed cycle engines with specified inlet flow rates of fuel and oxidizer, the total absolute level of energy that is released and absorbed by the combustion products should be maximized while an acceptable turbine inlet temperature is maintained. This creates the maximum energy state for the turbine drive fluid. In some cases, as for the oxidizer-rich cycles, extra flow must be pressurized and delivered to the preburner to achieve the maximum level of energy release. The penalty of the extra horsepower required to accomplish this increased preburner flow rate is addressed in Section I.D.2.

Table 1 Hypothetical oxygen/kerosene staged combustion engine parameters

Engine parameters	Valve
Engine inlet kerosene flow, kg/s	100
Engine inlet oxygen flow, kg/s (O/F mixture ratio = 2.5)	250
Powerhead turbine temperature limit, K (Either fuel-rich or oxidizer-rich)	900
Kerosene LHV, MJ/kg	43.15
Engine main chamber pressure, MPa	25
Preburner pressure, MPa (Simplified turbine pressure ratio = 2.0)	50

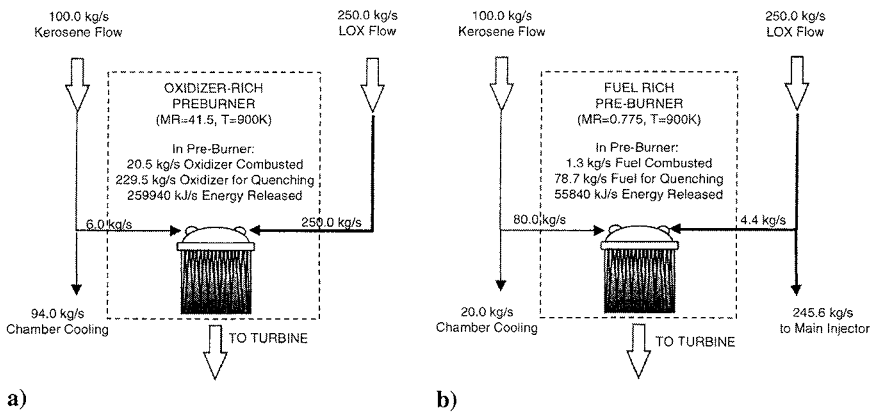

Fig. 4 Comparison of a) oxidizer-rich and b) fuel-rich powerhead propellant distribution.

The following simplified example compares the energy release potential for hypothetical fuel-rich and oxidizer-rich oxygen/kerosene staged combustion engines with the characteristics shown in Table 1. The simplified propellant distribution schematics for both powerhead options are shown in Fig. 4.

Assume that the engine is configured with an oxygen-rich powerhead (Fig. 4a) and that all of the oxygen (250 kg/s) is available for combustion in the preburners. To maintain 900 K temperature, the preburner kerosene flow is limited to 6 kg/s because of the mixture ratio limit of 41.5 (from Fig. 3). Because the stoichiometric mixture ratio for kerosene and oxygen is approximately 3.4, the 6 kg/s of preburner kerosene flow can combine chemically with only 20.4 kg/s of the preburner oxygen flow to release its stored chemical energy. The remaining 229.6 kg/s of preburner oxygen flow effectively serves only to quench the entire combustion mixture down to the 900 K temperature limit. The total energy release is, therefore, the 6 kg/s of kerosene multiplied by 43.15 MJ/kg LHV, equaling 259 MJ/s. The remaining 94 kg/s of kerosene flow, not used in the preburner, is available for chamber cooling.

For the fuel-rich powerhead option (Fig. 4b), a very aggressive level of 20 kg/s of fuel is assumed to be required for chamber cooling. This leaves 80 kg/s of kerosene available for combustion in the preburners. To maintain 900 K temperature, the preburner oxygen flow is limited to 4.4 kg/s because of the mixture ratio limit of 0.055 (from Fig. 3). Because the stoichiometric mixture ratio for kerosene and oxygen is 3.4, only 1.3 kg/s of the preburner kerosene flow can chemically combine with the oxygen to release its stored chemical energy. The remaining kerosene does not release chemical energy but effectively serves only to quench the entire preburner combustion mixture down to the 900 K temperature limit. Therefore, the total energy release is 1.3 kg/s of kerosene multiplied by 43.15 MJ/kg LHV, which equals 56 MJ/s. The remaining 245.6 kg/s of the oxygen, not used in the preburners, is delivered to the main injector.

For oxygen and kerosene propellants, the energy release of the oxidizer-rich approach is over 400% greater than for the fuel-rich approach, within the same turbine temperature limits. The important factor, however, is the potential for this energy release to be converted into useful turbine work, which is discussed next.

2. *Net Powerhead Work Potential*

To estimate the work potential previously discussed, the thermodynamic characteristics of the propellants and the resulting products of combustion must be understood. These characteristics, which are consistent with the previous example, are given in Table 2. The component efficiencies are assumed to be conservatively low, because this simplified example ignores all of the system pressure losses, including the chamber coolant pressure loss.

The work required by the pumps is different for the oxidizer-rich and the fuel-rich approaches. The oxidizer-rich approach requires that 250 kg/s of oxygen and 6.0 kg/s of kerosene be pumped to preburner pressures of 50 MPa. The remaining 94 kg/s of kerosene needs to be pumped to only chamber pressure, which is 25 MPa. By using Eq. (1) for the individual propellant discharge pressures and flow rates, the estimated total work required for the oxygen-rich pumping combination is ~28.2 MW. The fuel-rich powerhead approach requires that 80 kg/s of kerosene and 4.4 kg/s of oxygen be pumped to preburner pressure of 50 MPa. The remaining 245.6 kg/s of oxygen flow and 20 kg/s of kerosene cooling flow needs to be pumped only to chamber pressure, which is 25 MPa. Again, by using Eq. (1), the fuel-rich pumping combination will require ~23.3 MW. This is less than the oxidizer-rich approach, but the available turbine work must also be compared.

The work delivered by the turbines for each case can be calculated using the information in Tables 1 and 2, and substituting into Eq. (2). Most of the parameters used in Eq. (2) are similar except for the turbine flow rate, which is three times larger for the oxygen-rich approach. Equation (2) indicates a delivered turbine work for the oxygen-rich approach of ~25 MW and for the fuel-rich approach of ~9 MW. For the oxidizer-rich cycle, the turbine work available

Table 2 Powerhead parameters for hypothetical oxygen/kerosene staged combustion engine

Powerhead parameter	Oxygen-rich	Fuel-rich
Powerhead mixture ratio	41.5	0.055
Powerhead temperature, K[a]	900	900
Powerhead total flow rate, kg/s	256	84.4
Gas constant, R, kJ/kg K[a]	0.261	0.275
Ratio of specific heats[a]	1.304	1.107
Assumed pump efficiency, %	50	50
Assumed turbine efficiency, %	65	65
Oxygen density, kg/m^3	1140	1140
Kerosene density, kg/m^3	810	810

[a]As given per Ref. 1 for respective mixture ratios.

is only slightly less than the required pump work, indicating that chamber pressure will drop slightly below the 25 MPa level. For the fuel-rich cycle, however, the turbine work available falls far short of that required by the pumps; therefore, chamber pressure must be reduced significantly.

Iterating these calculations for each approach shows that the chamber pressure for the oxidizer-rich cycle balances at ~23 MPa, whereas the fuel-rich cycle balances at ~12.3 MPa. Therefore, considering equal system temperatures and peak system pressures, the oxygen-rich cycle provides a chamber pressure that is 87% higher. In practice, this ratio could be even higher because the selection of 20% of the kerosene to cool the fuel-rich cycle main chamber was very aggressive.

3. *Propellant Selection Influences*

The previous example illustrates the fundamental difference of oxidizer-rich and fuel-rich thermodynamics for oxygen and kerosene propellants. The 87% increase in chamber pressure, assuming equal turbine temperatures and pump discharge pressures, provides the rationale for the selection of the oxidizer-rich cycle as the preferred approach for oxygen and kerosene staged combustion engines. However, for oxygen and hydrogen propellants, a similar iterative investigation would show that a hydrogen-rich cycle is superior to the oxygen-rich cycle approach, largely because of the differences in thermodynamic parameters that strongly influence the pump and turbine work calculations.

For a hydrogen cycle, the flow rate of hydrogen is a smaller percentage of the total flow compared with that in kerosene engines. Although the fuel percentage is lower (~15% for hydrogen vs ~30% for kerosene), the reduced density of hydrogen (71 kg/m^3 for hydrogen vs 810 kg/m^3 for kerosene) results in a substantial increase in the required pump work. However, this increased pump work is more than offset by the profound increase in available turbine work, because of the thermodynamic properties of hydrogen fuel.

The energy release potential of hydrogen, represented by the LHV of hydrogen at 117.7 MJ/kg, is more than 2.5 times larger than that of kerosene, at 43.15 MJ/kg. The turbine work potential is directly influenced by the gas constant of the combustion products, which for oxygen-rich combustion using hydrogen fuel is 0.275 kJ/kg-K. For hydrogen-rich combustion the gas constant of the combustion products is nearly an order of magnitude higher at 2.32 kJ/kg-K.

The combined effect of improved energy release and improved turbine work potential more than offsets a moderate increase in required pump work, and results in a fuel-rich preference for staged combustion cycles if hydrogen is the fuel. An alternative hydrogen-based staged combustion cycle that operates with two preburners (one fuel-rich and one oxygen-rich) is addressed in Section VIII.B.

II. Expander Cycles

As described in Fig. 1, expander engines can use open or closed cycles. All of the energy to support the cycle is supplied by heat transfer to the propellant, generally the fuel, accomplished by using the propellant to regeneratively cool the thrust chamber and nozzle. The heat added to the propellant is used as work potential to power the turbines. An operative example of a closed expander cycle is the RL10.[5]

A. General Cycle Discussion

A schematic of the basic closed expander cycle engine is shown in Fig. 5. The fuel enters a single- or multiple-stage pump and is increased in pressure before cooling the thrust chamber and nozzle. The separate chamber and nozzle flows are vaporized by the energy that is added during this cooling function, and then mixed before being directed to the turbines. For a hydrogen expander engine, the mixed coolant flow temperature is typically between 200 and 300 K. The gaseous fuel flow is expanded through the turbines before being directed to the main injector and chamber for combustion and expansion in the primary nozzle.

The selection of fuels that are compatible with the expander cycle is limited. The fuel must have a high heat capacity and adequate heat-transfer properties, and it must vaporize easily. Generally, fuels are limited to hydrogen, methane, or propane. The selection of oxidizers for an expander engine is not limited, but it must be compatible with the selected fuel.

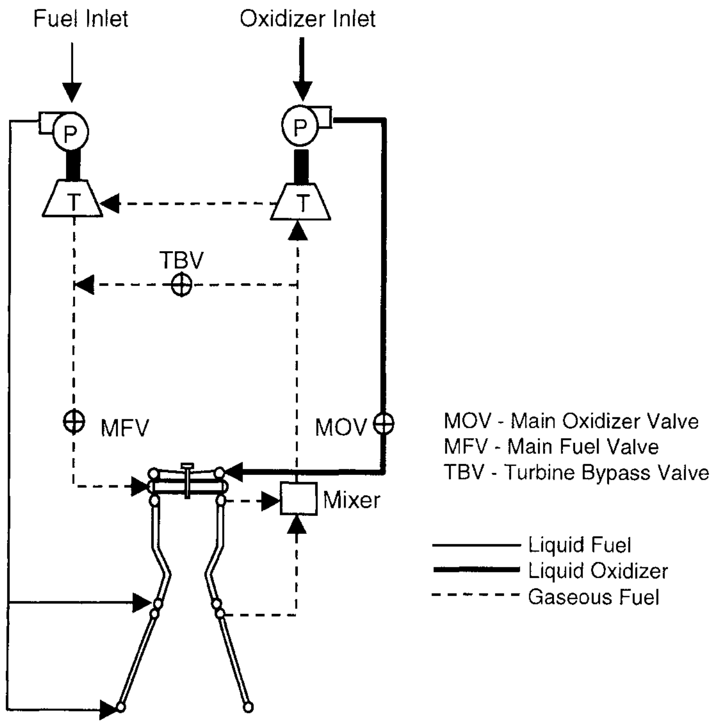

Fig. 5 Basic closed expander cycle.

The fuel side flow rate is controlled by the main fuel valve (MFV) located just upstream of the injector. Configuring the cycle with the MFV at this location has several advantages. The first advantage is that prior to start, with the MFV closed, the coolant passages and turbopump turbine housings will fill with hydrogen. This trapped hydrogen is no longer cryogenic as it will have been vaporized by the latent heat of the metal and the ambient surroundings of the metal. The gaseous fuel will provide an initial acceleration to the turbopumps, prior to main chamber ignition, that allows for a fast, smooth start or restart to begin immediately upon opening of the MFV. The dynamics and stability concerns of powerhead combustion are minimized. A second advantage is that upon engine shutdown, closing the MFV results in immediate main chamber fuel interruption while maintaining trapped propellant in the coolant passages for additional heat capacity to accommodate shutdown thermal transients. Finally, a third advantage is that prior to launch and during launch, the purges are minimized because the MFV isolates the fuel side of the powerhead from ambient conditions.

The power level of the expander cycle is controlled by the turbine bypass valve (TBV). To decrease engine power, the TBV is opened to allow flow to bypass both turbines. This reduces pump speed and flow, and therefore reduces chamber pressure. The oxidizer flow rate is controlled by the main oxidizer valve (MOV), located just upstream of the main injector. This valve is used to control engine mixture ratio and provide shutdown of the oxidizer at engine cutoff. Loading the MOV just upstream of the injector isolates the oxidizer powerhead from ambient contamination prior to engine start or restart, in a manner similar to the MFV on the fuel side.

The influence of cycle operating pressure (i.e., fuel pump discharge pressure) and component efficiencies is important for the expander cycle. Figure 6 shows an example of predicted chamber pressure for an expander cycle configured with different numbers of fuel pump stages, from a detailed oxygen/hydrogen expander engine design study at 1100 kN thrust level.[6] This study also included the influence of a copper chamber liner, which is significantly better than a steel chamber because of the higher thermal conductivity of copper. For the single-stage configuration, the chamber pressure increases as pump discharge pressure increases before reaching a peak. This peak results from the interaction between decreasing main fuel pump efficiency and cycle efficiency improvements with increased cycle pressures. By increasing the number of fuel stages to two, the pump efficiency is improved because of the reduced work per stage, which allows chamber pressure to increase. With a three-stage configuration, there is a small but diminishing gain.

Although not used in this manner historically, expander cycle engines can be considered for booster applications. The reduced cost and weight of the powerhead can offset the reduced performance from the moderately low chamber pressure. Because there are no preburners and the turbine temperatures are low, the engine cost can be affordable for expendable applications. Expander cycle engines are currently used for upper stages and space transfer applications, in which ease of starting and multiple restarts are required and the engine can restart on space-ambient heat capacity. Also, upon restart the concern of

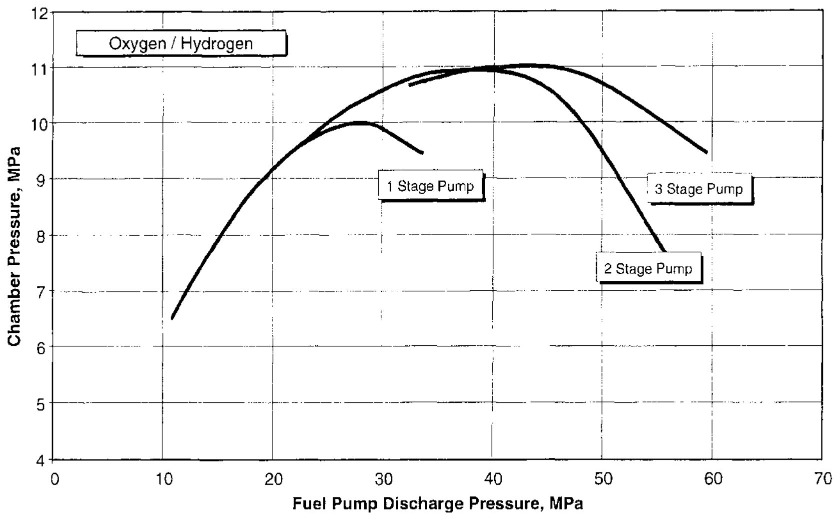

Fig. 6 Typical expander cycle chamber pressure vs fuel pump discharge pressure.

freezing the residual moisture that can result from propellant precombustion is avoided.

B. Configuration Options

There are many possible configurations for expander cycle engines. The configuration selection, however, does very little to change the total energy available to the cycle. Generally the energy available is determined by the chamber and nozzle surface area and combustion-side heat transfer conditions, and it is relatively unaffected by different coolant-side configurations or by the coolant-side pressure drop. The primary objective of the coolant-side design is to maintain adequate chamber and nozzle material temperatures, required for durability, and to minimize coolant pressure drop, which will maximize chamber pressure and cycle performance.

1. Expander Cycle with Regenerator

The basic expander cycle can be enhanced by the addition of a regenerator (heat exchanger) between the turbine discharge and main injector inlet. The regenerator recaptures energy from the turbine discharge flow to provide additional turbine work capability. However, because the turbine discharge flow is approximately 250 K while the liquid hydrogen temperature is approximately 20 K, only moderate levels of energy can be recaptured before the weight of the regenerator becomes excessive.

2. *Full-Flow Expander Cycle*

In a full-flow expander cycle, the fuel is used to cool the thrust chamber and only a portion of the nozzle, and then used exclusively to drive the fuel pump. The oxidizer is used to cool the remainder of the nozzle, and the energy absorbed is used to provide power exclusively to the oxidizer pump. This cycle requires an additional coolant manifold section to the nozzle and an oxidizer turbine bypass valve (OTBV) on the oxidizer turbine line to control and throttle the oxidizer side of the cycle.

The performance for the full-flow expander cycle will be less than for the conventional expander. The total energy available to the cycle remains the same because the total chamber and nozzle heat transfer is unaffected by this alternate pump configuration, but the total pump horsepower requirement is increased. This occurs because the oxidizer pump discharge pressure must be increased to provide the oxidizer turbine pressure ratio potential that is now required. However, one advantage is that this cycle would eliminate the need for an interpropellant seal (IPS) purge. The IPS purge is required for booster applications to protect against mixing and possible ignition of propellants in the oxygen turbopump. The IPS purge is not required for upper-stage applications because the interpropellant location can be vented to a low ambient pressure that is below the pressure required to allow propellant ignition and combustion. In general there are not many advantages for this full-flow expander cycle as compared with the traditional approach.

C. Expander Thrust Scaling Trends and Issues

All expander cycles are dependent on chamber and nozzle heat transfer to provide energy to the cycle. For expander cycles, as with all cycles, the required propellant pumping power scales proportionally to engine thrust level (neglecting efficiency differences, etc.). The energy available to drive the expander cycle does not scale proportionally to thrust level, however, which results in a reduced chamber pressure as thrust size increases. The following discussion will provide insight into these scaling trends.

Chamber heat transfer does not scale proportionally with thrust. If engine thrust size is increased, then required pump work and throat area will increase proportionally. If combustion chamber length is approximately constant, however, the chamber surface area, and therefore the chamber heat transfer, scales proportionally to the square root of the throat area. This relative loss of available energy can be partially offset by increasing chamber length, but this will increase engine length and weight and increase the chamber coolant pressure losses, which will limit chamber pressure increase.

Unlike chamber heat transfer, nozzle heat transfer does scale proportionally with thrust. For a constant nozzle expansion ratio, the nozzle surface area is proportional to the throat area. Therefore, as throat area increases proportional to thrust, the nozzle heat transfer area also increases proportional to thrust.

Therefore, because total heat transfer (chamber plus nozzle) does not scale proportionally with increased thrust, the energy available is increasing more slowly than the work requirement, and the chamber pressure decreases as thrust size is increased. This trend is illustrated in Fig. 7, indicating the

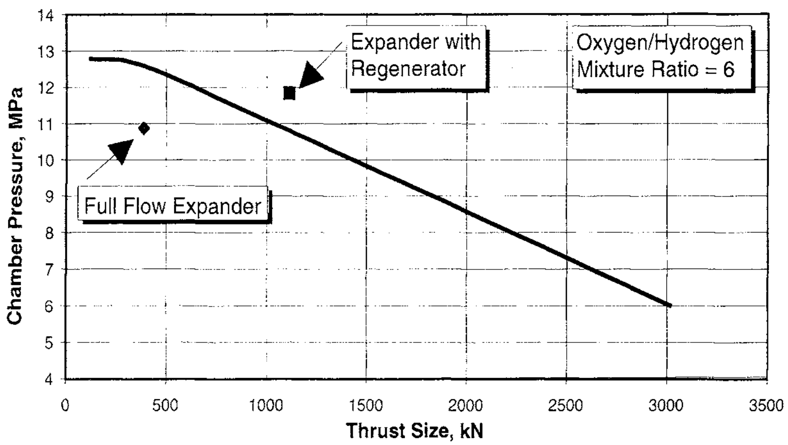

Fig. 7 Thrust scaling trends for oxygen/hydrogen expander engines.

chamber pressure drops to ~8 MPa at a thrust level of 2225 kN. Figure 7 also shows the approximate performance benefit of adding a regenerator and the approximate performance penalty for the full flow expander configuration.

Because expander cycles operate at lower chamber pressures than gas generator or staged combustion cycles, the geometric size of the engine can be an issue, as thrust size is increased. Size issues can be mitigated, however, depending on the application. For upper stages the important dimension is generally the interstage length required for engine packaging. Packaged engine length can be shortened by use of a retracted nozzle skirt that is stowed at liftoff and translates prior to engine start. For booster applications, the overall engine geometry is nearly unaffected by chamber pressure because reduced chamber pressure forces the selection of a reduced nozzle expansion ratio to maximize sea-level thrust or to avoid nozzle flow separation. However, a smaller nozzle expansion ratio will reduce engine performance.

III. Gas Generator Cycles

Gas generator cycle engines are open cycle engines. An operative example of a gas generator cycle is the Vulcain.[7,8] The energy to support the cycle is supplied by the combustion of a portion of the propellants in a "gas-generator" combustion device. This turbine drive flow is configured in parallel with the main chamber flow and is discharged overboard or into the divergent section of the primary nozzle, providing a minimal contribution to thrust. For a fuel-rich configuration, the mixture ratio of the gas generator flow, selected to maintain acceptable turbine temperature limits, is much lower than the mixture ratio of the main chamber. To maintain a nominal engine inlet mixture ratio, the main chamber must operate above the nominal mixture ratio to compensate for the lower mixture ratio in the gas generator. Similarly, for an oxidizer-rich configuration, the mixture ratio of the main chamber must operate below the nominal mixture ratio, because the mixture ratio of the gas generator is higher. Thus, for both configurations main chamber performance is reduced because the mixture ratio is further from the peak performance value. The gas generator flow should be minimized to avoid the thrust loss of the outer performance gas generator stream and to minimize the main chamber mixture ratio shift. To minimize the impact of this mixture ratio shift, the vehicle tanks can be designed to operate at a "system" or engine inlet mixture ratio that is somewhat reduced. This works better for nonhydrogen fueled engines, where the fuel density is nearer to the density of the oxidizer and mixture ratio selection has minimal impact on the stage tank weight and volume.

Another remedy for the mixture ratio shift would be to include an afterburner located downstream of the turbines but prior to expansion through a secondary expansion nozzle. For the afterburning approach the performance loss of an overboard gas generator flow is mitigated, and the mixture ratio shift of the primary combustion chamber is avoided, although now a portion of the main chamber flow is combusted and expanded from a lower chamber pressure. The gas generation afterburning cycle is discussed further in Section VIIB.

A. General Cycle Discussion

A conventional fuel-rich gas generator cycle is shown in Fig. 8. The fuel enters a single- or multiple-stage pump and is increased to a pressure adequate to cool

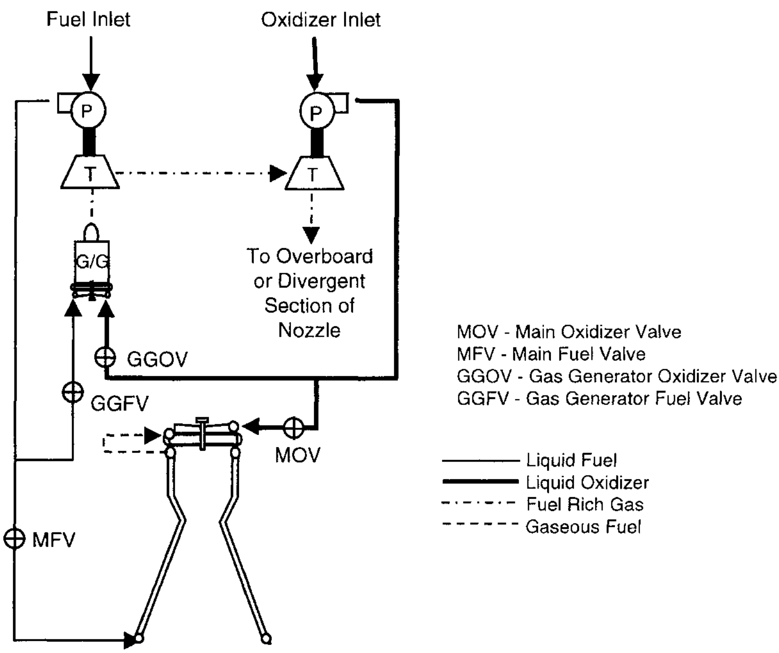

Fig. 8 Fuel-rich gas generator cycle.

the thrust chamber and nozzle. A small percentage of the fuel is directed to the gas generator combustion device. The pump discharge pressure must be adequate to overcome coolant pressure losses prior to fuel entry into the main injector and chamber. The fuel flow is controlled by the MFV located just upstream of the chamber and nozzle coolant circuit. The gas generator fuel flow is controlled by the gas generator fuel valve (GGFV).

The oxidizer flow is pressurized to moderate pressures by a single- or multiple-stage oxidizer pump. The oxidizer flow is controlled by the MOV located just upstream of the main injector, and the gas generator oxidizer valve (GGOV) upstream of the gas generator. The desired engine power level is achieved and maintained by using the GGFV and GGOV to control the gas generator mixture ratio. Engine mixture ratio is controlled by the MOV.

To minimize the overboard flow, the gas generator temperature should be maximized. A moderate temperature limit of 900 K is typical for a low-cost turbine because gas generator cycles are generally configured for expendable applications.

B. Configuration Options

To recover performance from the use of a gas generator, the turbine discharge flow can be directed to an afterburner combustion device for additional combustion and expansion through a secondary nozzle. Although either a fuel-rich or oxidizer-rich gas generator could be considered upstream of the afterburner, the oxidizer-rich option was selected for two reasons. First, the oxidizer-rich combustion products contain much less carbon and are less likely to result in soot buildup in the turbines and the injector passages of the afterburner injector. Second, the fuel dedicated to the afterburner is a more appropriate propellant to use in cooling of the afterburner combustion liner.

An oxidizer-rich gas generator cycle with an afterburner is shown in Fig. 9. The oxidizer-rich turbine discharge is burned with fuel diverted from the fuel pump discharge or the chamber coolant discharge. The afterburner mixture ratio is maintained at the nominal engine level by using the afterburner fuel valve (ABFV); therefore the performance loss due to main chamber mixture ratio shift is avoided. The performance penalty normally associated with overboard gas generator flow is mitigated by the additional combustion in the afterburner, although at a lower pressure than in the main chamber.

The performance of the oxidizer-rich gas generator cycle with afterburning is shown in Fig. 10 for oxygen and kerosene propellants and is compared with a conventional fuel-rich gas generator cycle and an oxidizer-rich staged combustion cycle. For this comparison the specific impulse is plotted vs gas generator (or preburner) pressure, not chamber pressure as is the conventional comparison. This is a more valid comparison, as the staged combustion cycle would require significantly higher pump discharge pressure than the gas generator cycles for the same chamber pressure. Figure 10 shows that the performance of the afterburning cycle is higher than that of the conventional gas generator cycle and of the staged combustion cycle for low to moderate pump discharge pressures. Only at higher pump discharge pressures does the staged combustion cycle begin to provide performance benefit.

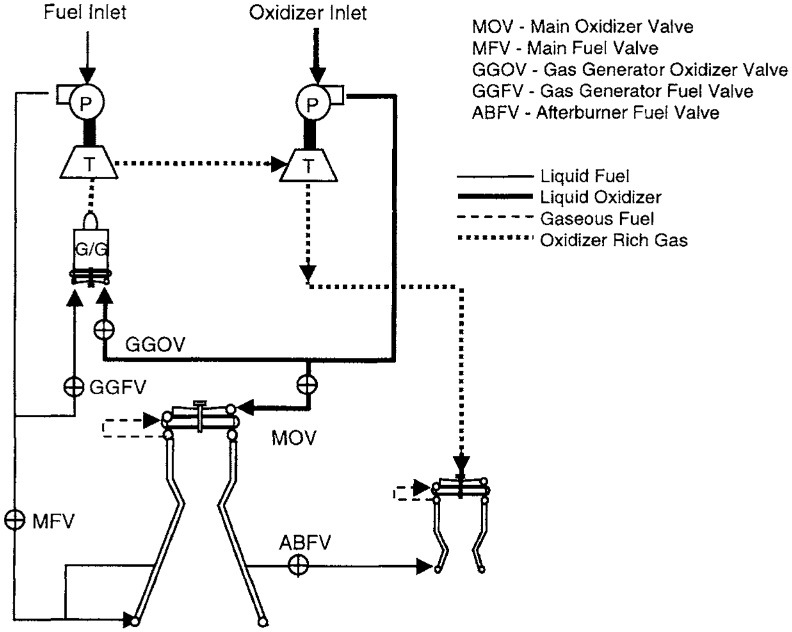

Fig. 9 Oxidizer-rich gas generator cycle with afterburner.

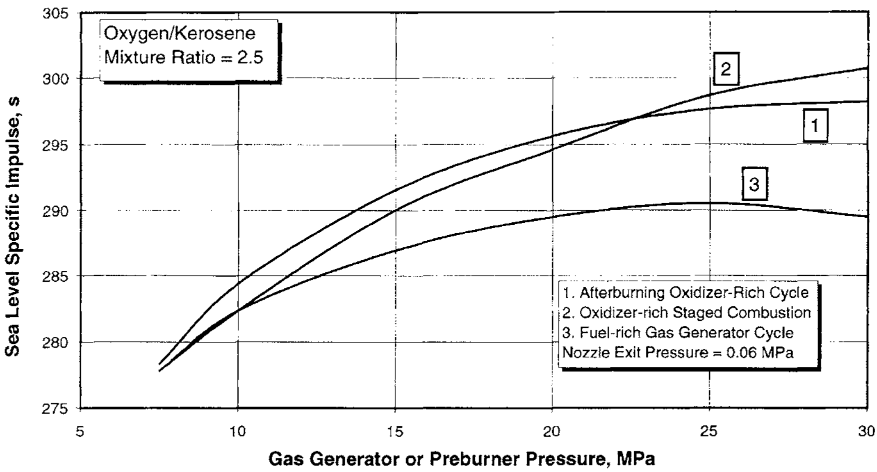

Fig. 10 Performance comparison of afterburning oxidizer-rich gas generator cycle.

The addition of the afterburning hardware will increase complexity as compared with a single chamber design. However, large booster applications may require multiple individually gimbaled thrust chambers, the complexity could be reduced as compared with a staged combustion cycle by dedicating one of the multiple chambers as the afterburner.

C. Gas Generator Thrust Scaling Trends and Issues

The thermodynamics of gas generator cycles generally scale very easily to high and low thrust levels. There are some secondary effects on the thermodynamics because of main chamber cooling and component efficiencies, both of which improve the thermodynamics as thrust size is increased. Because normalized chamber heat transfer is reduced as thrust size increases (see Section VI.B), the chamber coolant flow pressure drop can be decreased. Also, as the pump size is increased the relative internal clearances can be improved, which provides a secondary improvement in component efficiency.

IV. Staged Combustion Cycles

Staged combustion engines are closed-cycle engines. Nearly all of the energy to support the cycle is provided by chemical energy released in a preburning combustion device. Hence, these cycles are sometimes referred to as preburner cycles. The turbine drive flow is configured in series with the main chamber, so that all propellant is combusted in the main chamber and expanded in the primary nozzle. Staged combustion cycles can be used for either booster or upper-stage applications depending on the cost vs performance trades for the particular application.

A. General Cycle Discussion

The flow schematic of a fuel-rich staged combustion cycle engine with dual preburners representative of the space shuttle main engine[8] is shown in Fig. 11. The chemical energy to support the cycle is transported to the preburners by the fuel; the amount of available energy is represented by the LHV of the fuel. To maximize the energy release, the fuel flow provided to the preburners should be maximized. Some of the energy to the staged combustion cycle is supplied by nozzle heat transfer similar to the expander cycles described in Section VI. Unlike in the expander cycle, however, chamber coolant flow cannot be used in the preburner because the increased chamber pressures for the staged combustion cycles result in higher chamber heat flux and higher coolant pressure drop, and therefore chamber coolant discharge pressure is too low to return to the preburners.

For the staged combustion cycle, the fuel enters a single- or multiple-stage pump and is pressurized to high pressure. The current state-of-the-art limit for pumping hydrogen is approximately 70 MPa, because of the high power requirements resulting from the low density of hydrogen. A minimum amount of the high pressure fuel is diverted for main chamber cooling, generally less than 20% of the total fuel flow. The remainder of the fuel is vaporized while

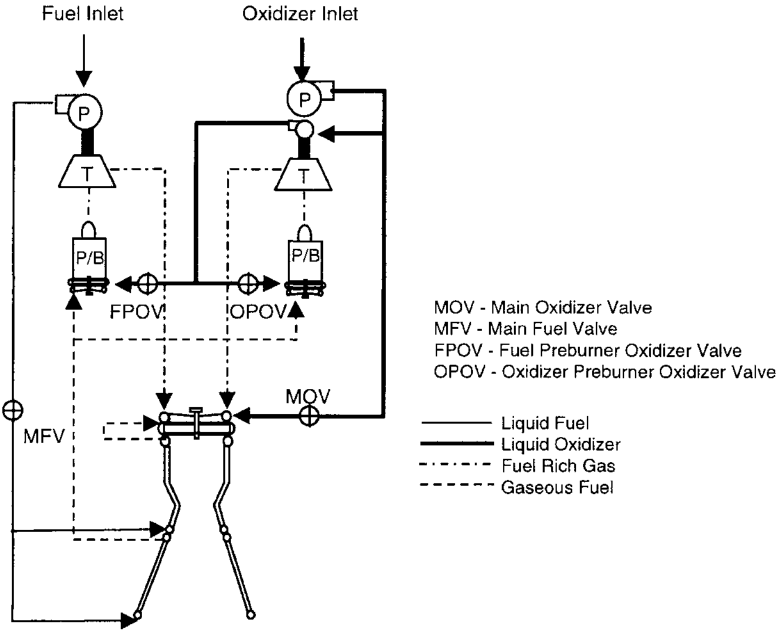

Fig. 11 Fuel-rich staged combustion.

cooling the nozzle, then delivered to the preburners for precombustion to provide power to the turbines. The total engine fuel flow is controlled by the MFV. The fuel distribution between the preburners is set by the respective preburner flow areas to ensure a split relative to the work requirement of each turbopump.

The oxidizer flow is pressurized to moderate pressure by a single- or multiple-stage main oxidizer pump. The pressure must be sufficient to deliver oxidizer to the main chamber but not to the preburners; a kick-stage is used to pressurize a fraction of the oxidizer to preburner pressure levels. The total engine oxidizer flow is controlled by the MOV. A fuel preburner oxidizer valve (FPOV) and an oxidizer preburner oxidizer valve (OPOV) are required to control the mixture ratio for each preburner. The desired engine power level is achieved and maintained by controlling the preburner mixture ratios.

The use of separate preburners has advantages for increased flexibility for component matching and throttling, but it can be a severe disadvantage during starting. The dynamic interaction of the preburners must be tailored to the inertia of each pump during the transition to full power, which must be accomplished without overspeeding the pumps and while avoiding transient preburner mixture ratios that may damage the turbines. Staged combustion cycles can also be configured with a single preburner. In this configuration, the preburner discharge flow must either power the turbines in series or be split to the respective turbines, or a single turbine must be configured to power both the fuel and oxidizer pumps on a single shaft. An operative example for a single preburner staged combustion oxygen/hydrogen engine is the Russian RD-0120.[9] The differences in the potential performance of this thermodynamic cycle are not strongly influenced by the alternate turbine or preburner configurations.

B. Configuration Options

Configuration options for staged combustion cycles are primarily influenced by the selection of a fuel-rich or an oxidizer-rich preburner configuration. As previously discussed in Section V, the fuel-rich powerhead approach is preferred for hydrogen-fueled cycles, and the oxygen-rich powerhead approach is preferred for hydrocarbon-fueled cycles. An additional option that can be considered for a dual preburner configuration is to operate one preburner fuel rich to drive the fuel pump and the other preburner oxidizer rich to drive the oxidizer pump. This is commonly referred to as a full-flow staged combustion cycle because nearly all of the flow is used in the powerhead combustion process. This configuration is also referred to as a gas-gas cycle because both propellants are gaseous entering the main injector. A technology demonstrator for this full-flow staged combustion cycle approach, is Integrated Powerhead Demonstrator.[10]

1. Full-Flow Powerhead

The use of both an oxygen-rich and a hydrogen-rich preburner for a staged combustion cycle has a profound effect on the cycle thermodynamics. A schematic of this approach is shown in Fig. 12. The fuel supply configuration is

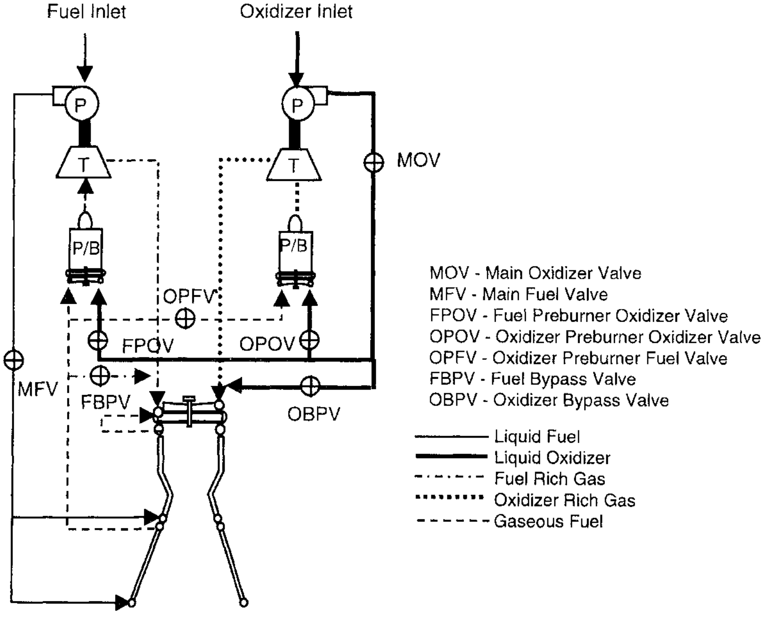

Fig. 12 Full-flow staged combustion.

similar to the conventional fuel-rich approach except that the majority of the fuel is directed to the fuel-rich preburner while a small portion of the fuel flow is directed to the oxidizer-rich preburner through the oxidizer preburner fuel valve (OPFV). The oxidizer side of this configuration requires that all of the oxygen be pumped to the preburner pressure level. Although this increases the total pump work, the pump configuration is simplified by the elimination of the oxidizer kick-stage. The majority of the main oxidizer flow is directed to the oxidizer-rich preburner, while a small portion is directed to the fuel-rich preburner through the FPOV.

A significant control constraint differentiates the conventional fuel rich and full-flow cycle approaches. The conventional fuel-rich approach has the flexibility to freely split the preburner flow rate to match the work requirements for the fuel and oxidizer pumps during both steady-state and transient operation. Also, the mixture ratio of the separate preburners can be controlled in a relatively independent fashion, especially during original design point selection. For the full-flow cycle, there are constraints to allocate only fuel-rich preburner flow to the fuel pump and only oxidizer-rich preburner flow to the oxidizer pump, as well as to hold total powerhead flow equal to the total flow being delivered by the pump. Also, as with the conventional fuel-rich cycle, the turbine pressure ratio of the fuel and oxidizer sides must be equal. The net result of these constraints for the full-flow cycle is that, without bypassing propellant around the preburners, only one of the preburner temperatures can be specified; the other preburner temperature will be determined by the remaining available propellants. This constraint applies for both steady-state and transient conditions. These constraints result in extra valves to control the full-flow cycle during startup, shutdown, and throttling, as indicated by comparison of Figs. 11 and 12.

The thermodynamic advantage of the full-flow staged combustion cycle is that it provides a significant increase in powerhead energy release within the same turbine temperature limits as the conventional fuel-rich approach. The proof of this increase can be shown by examining the energy release potential and the net powerhead work potential in a simplified example similar to that described in Section V. Rather than repeat the calculation, the results from a hypothetical oxygen/hydrogen staged combustion engine with characteristics as shown in Table 3 will be summarized. The simplified propellant distribution schematics

Table 3 Hypothetical oxygen/hydrogen staged combustion engine parameters

Engine parameters	Valve
Engine inlet hydrogen flow, kg/s	100
Engine inlet oxygen flow, kg/s	600
Powerhead turbine temperature limit, K	900 (both fuel-rich and oxidizer-rich)
Hydrogen LHV, MJ/kg	117.7
Engine main chamber pressure, MPa	20
Preburner pressure, MPa	35

for both the conventional fuel-rich and full-flow powerhead approaches are shown in Fig. 13.

From the propellant distributions shown in Fig. 13a, the preburner in the conventional fuel-rich cycle will produce an energy release of 912 MJ/s, given a hydrogen LHV of 117.7 MJ/kg. From Fig. 13b, the fuel-rich preburner in the full-flow cycle will provide an energy release of 859 MJ/s, while the oxidizer-rich preburner will produce 553 MJ/s, for a total of 1412 MJ/s. Thus, the full-flow cycle provides a 55% increase in total energy release that can be used to increase turbine work potential.

Using the characteristics shown in Table 4, the net powerhead work potential is examined next. The component efficiencies in Table 4 are assumed conservatively low because this simplified example ignores all system pressure losses including the chamber coolant pressure loss. Pump work is calculated using Eq. (1), and work delivered by the turbines is calculated using Eq. (2). Results of the noniterated power balance are shown in Table 5.

Table 5 shows that, although the total turbopump work requirement for the full-flow cycle is ~13% higher than the conventional fuel-rich cycle (37 MW + 98 MW vs 120 MW), the available turbine work is more than 40% greater (46 MW + 95 MW vs 100 MW). This comparison alone indicates a higher performance potential for the full-flow cycle. For the conventional fuel-rich cycle the delivered turbine work (100 MW) is less than the required pump work (120 MW), therefore, the chamber pressure will be less than the assumed 20 MPa. An iterative calculation indicates that chamber pressure for the fuel-rich conventional cycle would drop to 14.5 MPa.

For the full-flow cycle, the turbine work for the hydrogen side turbine (95 MW) is nearly equal to the requirement for the hydrogen pumps, indicating that the hydrogen side of the full-flow cycle is approximately balanced at the 20 MPa chamber pressure level. The turbine work for the oxidizer side turbine (46 MW) is greater than the oxidizer pump requirement (37 MW). Because the

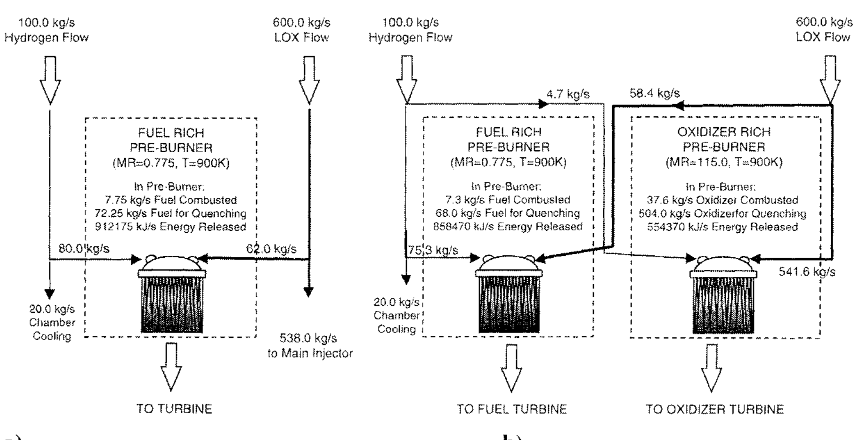

Fig. 13 Powerhead propellant distribution: a) conventional vs b) full-flow option.

Table 4 Powerhead parameters for hypothetical oxygen/hydrogen staged combustion engine

	Conventional	Full-flow	
Powerhead parameters	Fuel-rich	Oxidizer-rich	Fuel-rich
Preburner mixture ratios	0.775	115	0.775
Preburner temperatures, K[a]	900	900	900
Preburner flow rates, kg/s	142	546.3	133.7
Preburner gas constant, R, kJ/kg K[a]	2.323	0.275	2.323
Preburner ration of specific heats[a]	1.368	1.309	1.368
Assumed pump efficiency, %	50	50	50
Assumed turbine efficiency, %	65	65	65
Oxygen density, kg/m^3	1140	1140	1140
Hydrogen density, kg/m^3	71	71	71

[a]As given per Ref. 1 for respective mixture ratios.

chamber pressure is limited by the hydrogen side work balance, the oxidizer side must be balanced by reducing the temperature to ~700 K. Thus, the full-flow cycle can deliver the full assumed 20 MPa. The 38% improvement in chamber pressure over the conventional dual fuel-rich preburner cycle provides the rationale for consideration of the full-flow cycle approach for oxygen and hydrogen staged combustion rocket engines.

Both the conventional and full-flow cycles are strongly influenced by component efficiencies. As pump discharge pressure is increased, chamber pressure for both cycles will increase until component structural or thermal limitations are reached or until component efficiencies begin to decrease more rapidly. An example is shown in Fig. 14. In much the same fashion as shown previously in Fig. 6, local peaks in chamber pressure result from increasing cycle efficiency being offset by accelerating reductions in component efficiencies as their work

Table 5 Power balance parameters for hypothetical oxygen/hydrogen staged combustion engine

	Conventional	Full-flow	
Power balance parameters	Total	Oxidizer-side	Fuel-side
Preburner energy release, MJ/s	912	553	859
Delivered turbine work, MW	100	46	95
Fuel flow pumped to 35 MPa, kg/s	100	0	100
Oxidizer flow pumped to 35 MPa, kg/s	62	600	0
Fuel flow pumped to 20 MPa, kg/s	0	0	0
Oxidizer flow pumped to 20 MPa, kg/s	538	0	0
Turbopump work requirement, MW	120	37	98

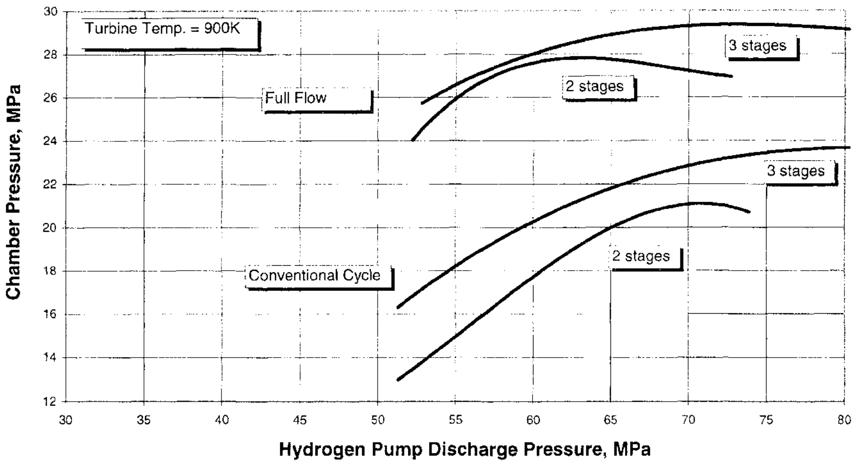

Fig. 14 Staged combustion cycle hydrogen pump discharge pressure.

levels are increased. The effect of reduced fuel pump work load, by increasing from two to three stages, is evident in the increased chamber pressure for a given fuel pump discharge pressure.

2. *Single-Shaft Oxidizer-Rich Powerhead*

As previously discussed in Section V, an oxidizer-rich powerhead is thermodynamically desirable for staged combustion cycles using kerosene propellants. The simplification of a common shaft for the fuel and oxidizer turbomachinery is feasible because the combination of propellant densities and discharge pressures results in similar desired rotational speeds of the fuel and oxidizer impellers. This configuration with a single preburner and single shaft turbopump allows the engine to be controlled with a minimum complement of valves. The starting and throttling of the engine is more controllable because the dynamic complications of separate preburners and separate turbopumps is avoided. Operative examples of oxidizer-rich, oxygen/kerosene, single-shaft staged combustion engines are the NK-33[11] and RD-120.[12]

C. Staged Combustion Thrust Scaling Trends and Issues

The thermodynamics of staged combustion cycles generally scale very easily, in the same fashion as gas generator cycles. As thrust size increases, the required percentage of main chamber coolant flow can be decreased, instead of or in addition to a reduction in coolant flow pressure drop.

V. Summary

This chapter provides an overview of the thermodynamics that influence the cycle configuration selection of liquid rocket engines. An overview of the interaction of the cycle thermodynamics with the engine hardware stress limitations, thermal limitations, and component efficiency trends is addressed. Thermodynamic cycle comparisons are presented for expander cycle, gas generator cycle, and staged combustion cycles. For the expander cycles the thermodynamic issues are presented relative to operation with a regenerator as well as scaling trends. For the staged combustion cycles, the thermodynamic implications of oxidizer-rich combustion and the influence on liquid rocket engine cycles are specifically considered. For the gas generator cycle, the thermodynamic implications of using an afterburner downstream of an oxygen-rich gas generator and turbine drive are addressed.

References

[1]McBride, B. J., and Gordon, S., "Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications," NASA RP-1311, 1996.

[2]Beckman, B., Chilton, J., and Jue, F., "Block II SSME Operability Improvements," AIAA Paper 97-2819, June 1997.

[3]Minick, A., and Peery, S., "Design and Development of an Advanced Liquid Hydrogen Turbopump," AIAA Paper 98-3681, July 1998.

[4]Ohta, T., Kitamura, A., and Ogata, H., "LH_2 Turbopump Test with Hydrostatic Bearing," AIAA Paper 99-2195, June 1999.

[5]Santiago, J., "Evolution of the RL10 Liquid Rocket Engine for a New Upper Stage Application," AIAA Paper 96-3013, July 1996.

[6]Parsley, R., and Crocker, A., "The RL200: Incorporating A Balanced Approach for Reusable Propulsion Safety," AIAA Paper 2000-3287, July 2000.

[7]Brossel, P., et al., "Development Status of the Vulcain Engine," AIAA Paper 94-2751, June 1994.

[8]Bradley, M., "Space Shuttle Main Engine Off-Nominal Low Power Level Operation," AIAA Paper 97-2685, July 1997.

[9]Rachuk, V., Martinenko, Y. A., Gontcharov, N. S., and Sciorelli, F., "Design, Development, and History of the CADB RD-0120 Engine," AIAA Paper 95-2540, 1995.

[10]Vilja, J., "Rocketdyne Advanced Propulsion Systems Overview," AIAA Paper 97-3309, July 1997.

[11]Kuznetsov, N. D., "Closed Cycle Liquid Propellant Rocket Engines," AIAA Paper 93-1956, June 1993.

[12]Fatuev, I., "RD-120M High Performance Engine for Booster Applications," AIAA Paper 95-3144, July 1995.