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CYCLE ANALYSIS OF ISOTHERMAL EXPANSION COMBUSTION GAS TURBINE

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ABSTRACT In practical gas turbines, high compression is unfavorable because it causes large irreversible losses in the adiabatic processes. Though, for improvement of the thermal efficiency in the region of a high pressure ratio, the turbine inlet temperature must be increased, it is restricted by the endurance of the materials. This study proposes a gas-turbine cycle consisting of isothermal expansion combustion which produces mechanical energy rather than thermal energy. Analysis of the cycle analysis indicates that, compared with the conventional gas-turbine cycle, the thermal efficiency of this isothermal expansion combustion gas-turbine cycle is very high in the case of a high pressure ratio without maximum temperature ascent. In addition, the methods and problems of isothermal expansion combustion are discussed.

***Keywords**: Gas turbine, Isothermal expansion combustion, Thermal efficiency.*

1. INTRODUCTION

The Carnot cycle is a well known cycle, because it is the most desirable in terms of the second law of thermodynamics. However, it is very difficult to realize the Carnot cycle, and thus it is generally regarded as conceptual and is not considered in the development of practical thermal cycles.

One of the characteristics of the Carnot cycle is that it has isothermal processes. Under the first law of thermodynamics ($\delta Q = dU + \delta W$), if dU = 0 is assumed, we have δQ=δW; this indicates that energy added in an isothermal process is converted to mechanical energy instead of thermal energy. If combustion is controlled in order to keep the temperature constant, the chemical energy of the original fuels can be converted directly to mechanical energy; however, it is difficult to establish such a combustion field. [1]

The maximum temperature of heat engines cannot be increased easily, because it is restricted by the endurance of materials. If isothermal combustion is available for internal combustion engines, it will be possible to compose a high-efficiency cycle without increasing the maximum temperature.

This study proposes a gas-turbine cycle with an isothermal expansion combustion process. [2] This cycle is expected to achieve high thermal efficiency, particularly in the region of high pressure ratio. In addition, the methods and problems of isothermal expansion combustion in gas turbines are discussed.

2. NOMENCLATURE

A = cross-sectional area [m^2]

c_p = specific heat at constant pressure [kJ/(kg·K)]

h = enthalpy [kJ]

M = Mach number

p = pressure [Pa, atm]

R = gas constant [kJ/(kg·K)]

Q = heat [kJ]
r = pressure ratio ($=p_2/p_1$)
s = specific entropy [kJ/(kg·K)]
T = temperature [K]
U = internal energy [kJ]
W = mechanical work [kJ]
w = velocity [m/s]
x = pressure parameter
ε = compression ratio ($=v_1/v_2$)
γ = density [kg/ m^3]
η = thermal efficiency
η_c = adiabatic efficiency of compressor
η_t = adiabatic efficiency of turbine
η_{th} = thermal efficiency of Brayton cycle
κ = adiabatic exponent

3. CONVENTIONAL GAS-TURBINE CYCLE

The thermal efficiency of the Brayton cycle (the basic cycle of a gas turbine) shown in Fig.1 is expressed as the following equation.

$$\eta_{th} = 1 - \frac{1}{r^{(\kappa-1)/\kappa}} = 1 - \frac{1}{\varepsilon^{\kappa-1}} = 1 - \frac{T_1}{T_2} \qquad (1)$$

This equation indicates that the efficiency of the Brayton cycle is determined only by the compression process; the maximum temperature T_3 does not affect the efficiency. In spite of these theoretical relationships, high compression is not generally employed to improve the thermal efficiency in actual gas turbines.

However it is well known that, considering the adiabatic efficiency of compressor η_c and that of turbine η_t, the maximum temperature T_3 appears in the equation of thermal efficiency:

$$\eta = \left(1 - \frac{T_1}{T_2}\right) \frac{\eta_c \eta_t T_3 - T_2}{\eta_c T_3 - \{\eta_c + (1-\eta_c)\eta_{th}\} T_2} \ . \qquad (2)$$

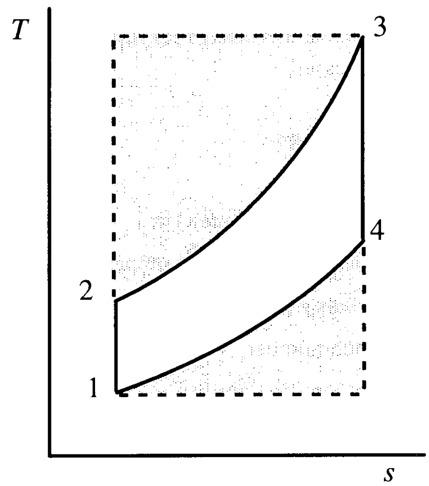

Figure 1 T-s diagram of Brayton cycle.

Figure 2 shows the efficiencies of the Brayton cycle and the cycle including the adiabatic efficiencies, as obtained by Eq. (1) and (2), respectively. When the pressure ratio becomes excessively large, the efficiency of the Brayton cycle η_{th} is improved; on the other hand, those of the cycle including adiabatic efficiencies are reduced because the loss of mechanical work increases. In order to improve the thermal efficiency of an actual gas turbine by applying high compression, the maximum temperature must necessarily be higher. However, it is not easy to increase the turbine inlet temperature (TIT), which is the maximum temperature in a gas-turbine system, because of the limitation of material endurance.

Under the condition that the maximum temperature is restricted, the shaded portions in the T-s diagram shown in Fig. 1 would be filled for improvement in thermal efficiency. The lower portion could be filled according to a bottoming cycle such as the Rankine cycle, and the reheat cycle and intercooled gas-turbine cycle can utilize the upper space effectively.

4. ISOTHERMAL EXPANSION COMBUSTION GAS-TURBINE (ITECT) CYCLE

4.1 Concept of Isothermal Expansion Combustion

Combustion is used mainly as a process for the storage of thermal energy. In heat engines, first, the chemical

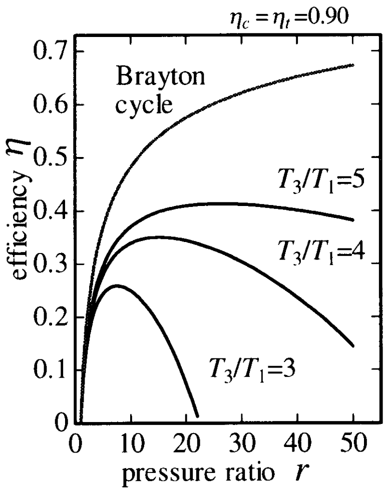

Figure 2 Efficiencies of Brayton cycle and gas-turbine cycle considering adiabatic efficiencies.

energy is converted to thermal energy through a combustion process and then sequentially to mechanical energy (work) by an adiabatic expansion process. In general, the condition of combustion is constant pressure or constant volume, with the result that the temperature rises. On the contrary, if combustion is controlled just to compensate for the temperature drop in the expansion process, the isothermal combustion is realized [1].

If the isothermal expansion combustion is available for internal combustion engines, a high-efficiency cycle can be composed without increasing the maximum temperature which is limited by material endurance. Though, as mentioned in the previous chapter, excessively high compression is not preferable in actual gas turbines, it is expected to improve the gas-turbine cycle incorporated with the isothermal expansion combustion. Here the problem arises of where the isothermal expansion combustion should occur in a gas-turbine. The expansion process generally occurs in the turbine; however, with respect to attempting to perform the isothermal expansion combustion in the turbine, it is difficult to complete the combustion within a short passage such as that in a stator or the rotor blades.

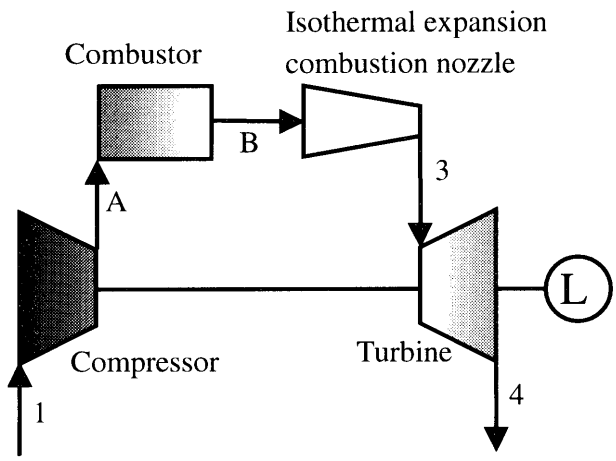

Figure 3 Schematic of isothermal expansion combustion gas turbine.

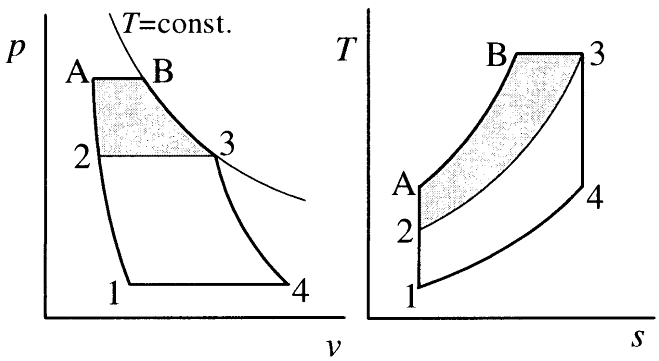

Figure 4 *p-v* and *T-s* diagrams of ITECT cycle.

Accordingly, a converging nozzle configured to the isothermal expansion combustion is arranged between the combustion chamber and the turbine, as shown in Fig. 3. This isothermal expansion combustion nozzle enables the conversion process from chemical energy to kinetic energy which means that the flow is accelerated. In the isothermal expansion combustion nozzle, the velocity increases with the temperature constant and the pressure and density decreased. The flow accelerated in isothermal expansion combustion nozzle is introduced into the guide vane which changes the direction of flow and do not reduce the static enthalpy. Next, the flow goes into the rotor in which the static enthalpy is not reduced like a rotor of the zero reaction stage and here the kinetic energy is converted to the work. It appears that cooling must be higher because of increase in total enthalpy; however, the relative flow velocity at the rotor is small so that the requirement of cooling is not so severe.[3][4]

4.2 Composition of Isothermal Expansion Combustion Gas-Turbine Cycle

Incorporating the above isothermal expansion combustion, the novel gas turbine shown in Fig. 4 is proposed: we name it the Isothermal Expansion Combustion Gas-Turbine cycle (ITECT cycle) and it has the following composition [2].

1 → [adiabatic compression] → A → [isobaric combustion] → B → [isothermal expansion combustion] → 3 → [adiabatic expansion] → 4 → [exhaust] → 1.

From the point of view of thermal efficiency, only adiabatic compression for attaining the maximum temperature is ideal, although such a compressor is currently impossible. Therefore, isobaric combustion is used after the adiabatic compression in which the pressure ratio is higher than that of conventional gas turbines.

The efficiency of the ITECT cycle η is derived:

$$\eta = \left(-\frac{1}{\eta_c} T_1\left(\varepsilon^{\kappa-1}-1\right)+\frac{\kappa-1}{\kappa}\cdot T_3 \ln\frac{\varepsilon^{\kappa}}{\varepsilon^{\kappa}(1-x)+x} + \eta_t T_3\left[1-\left\{\varepsilon^{\kappa}(1-x)+x\right\}^{\frac{1-\kappa}{\kappa}}\right]\right) \Big/ \left\{\left(T_3-T_1\varepsilon^{\kappa-1}\right)+\frac{\kappa-1}{\kappa}\cdot T_3 \ln\frac{\varepsilon^{\kappa}}{\varepsilon^{\kappa}(1-x)+x}\right\} \quad (3)$$

where $x \equiv (p_B - p_3)/(p_B - p_1)$. The pressure parameter x represents the magnitude of the isothermal expansion combustion; x=0 corresponds to the cycle without isothermal expansion combustion (that is, the Brayton cycle), and x=1 the cycle without adiabatic expansion.

It is assumed that the ITECT cycle has the benefit indicated by the shaded portion in Fig.4, in comparison with the Brayton cycle.

5. CYCLE ANALYSIS

Here we discuss the result obtained from Eq. (3). In this analysis, the maximum temperature is fixed at 1500[K] and the adiabatic efficiency of both compressor η_c and turbine η_t, is 0.9.

Figure 5 shows the efficiencies of the ITECT cycle with variation of x. The pressure ratios are 20, which is the ordinary level of conventional gas turbines, and 40. With increasing x, i.e. increasing the proportion of the isothermal expansion combustion, the efficiencies of the ITECT cycle increase because the unfavorable effect of the adiabatic efficiencies is relatively reduced. However, when the x becomes larger than about 0.8, the efficiencies decline because the exhaust temperature becomes high, with the result that the extracted work decreases.

Figure 6 shows the efficiency of the ITECT cycle compared with that of the conventional gas turbine cycle. In the ITECT cycle on this calculation, T_4 is fixed at 850[K] because it is sufficient for using the exhaust gas to the bottoming Rankine cycle and any higher temperature is ineffective. Under the conditions of T_3=1500[K] and T_4=850[K], we obtain 0.68 and 0.84 as the pressure parameters x for r=20 and 40, respectively. In Fig.5, it is found that these are close to the points at which the thermal efficiencies are optimum. The efficiency of the ITECT cycle does not decrease in the case of higher pressure ratios unlike the conventional cycle; in other words, by adopting the isothermal expansion combustion, the gas turbine can be improved without involving a temperature ascent. Furthermore, in spite of high compression, the temperature of the exhaust gas of the ITECT cycle can be kept relatively high due to the isothermal expansion combustion, with the result that this cycle is suitable for the topping of a combined cycle.

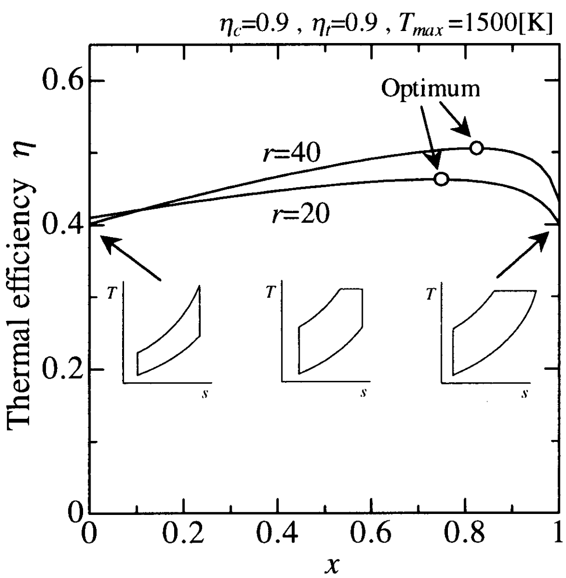

Figure 5　Relationship between efficiencies of isothermal expansion combustion cycle and pressure parameter x.

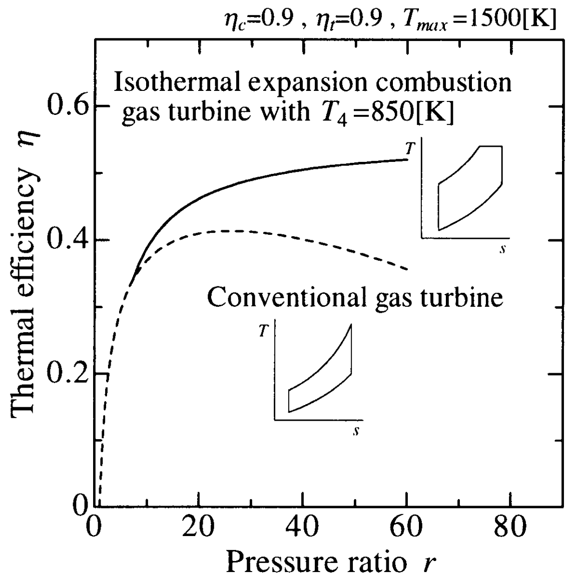

Figure 6　Comparison of efficiencies between isothermal expansion combustion gas turbine and conventional gas turbine cycle.

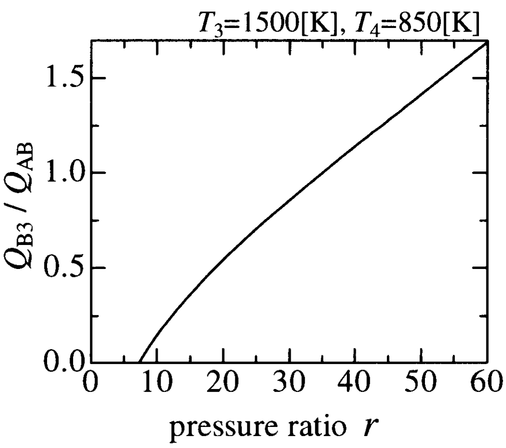

Figure 7　Ratio of amount of heat released from isothermal expansion combustion Q_{B3} to that from isobaric combustion Q_{AB}.

Figure 7 shows the ratio of an amount of heat released from isothermal expansion combustion Q_{B3} to that from isobaric combustion Q_{AB} under the condition of T_3=1500[K] and T_4=850[K]. When the pressure ratio increases, the isothermal expansion combustion forms a larger part of energy conversion process. This is one of the reasons why the thermal efficiency of the ITECT cycle is improved in the region of high pressure ratio, because the isothermal expansion combustion has the potential to produce mechanical energy instead of thermal energy.

6. ADDITIONAL CONSIDERATION OF ISOTHERMAL EXPANSION COMBUSTION

There are some points in applying isothermal expansion combustion in gas-turbine systems; here we consider two of them.

6.1 Restriction by Mach Number

In an isothermal expansion combustion nozzle, working gas is accelerated, involving pressure descent; however, the pressure descent in a single stage has a limitation in that the Mach number should be less than unity. Here we consider the characteristics of the isothermal expansion combustion nozzle by means of basic equations for a one-dimensional compressible flow [5].

Equation (4) to (8) express the changes of velocity, pressure, density, temperature and Mach number in terms of changes in cross-sectional area and enthalpy, respectively.

$$\frac{dw}{w} = -\frac{1}{1-M^2}\frac{dA}{A} + \frac{1}{1-M^2}\frac{dh}{c_pT}, \tag{4}$$

$$\frac{d\gamma}{\gamma} = \frac{M^2}{1-M^2}\frac{dA}{A} - \frac{1}{1-M^2}\frac{dh}{c_pT}, \tag{5}$$

$$\frac{dp}{p} = \frac{\kappa M^2}{1-M^2}\frac{dA}{A} - \frac{\kappa M^2}{1-M^2}\frac{dh}{c_pT}, \tag{6}$$

$$\frac{dT}{T} = \frac{(\kappa-1)M^2}{1-M^2}\frac{dA}{A} + \frac{1-\kappa M^2}{1-M^2}\frac{dh}{c_pT}, \tag{7}$$

$$\frac{dM}{M} = -\frac{1+\frac{\kappa-1}{2}M^2}{1-M^2}\frac{dA}{A} + \frac{1+\kappa M^2}{2(1-M^2)}\frac{dh}{c_pT}. \tag{8}$$

First, in the conventional adiabatic expansion nozzle with dA<0 and dh =0 in equation(4) to (8), the velocity and Mach number increase and the temperature, pressure and density decrease.

Next, we consider the isothermal expansion combustion nozzle. If the temperature drop due to the expansion is only compensated for by energy released from the combustion reaction, isothermal expansion combustion is realized; in the case of dT=0 in Eq.(7), we have

$$-\frac{(\kappa-1)M^2}{1-M^2}\frac{dA}{A} = \frac{1-\kappa M^2}{1-M^2}\frac{dh}{c_pT}. \tag{9}$$

By substituting Eq.(9) and the definition of the Mach number for Eq.(4), the following relationship is obtained:

$$wdw = dh, \tag{10}$$

which indicates that all energy released from the combustion is converted to kinetic energy (this is more easily derived from the first law of thermodynamics).

Substituting Eq.(9) for equation(4) to (8), we obtain the expressions in terms of cross-sectional area under the condition of dT=0, as follows:

$$\frac{dw}{w} = -\frac{1}{1-\kappa M^2}\frac{dA}{A}, \tag{11}$$

$$\frac{d\gamma}{\gamma} = \frac{\kappa M^2}{1-\kappa M^2}\frac{dA}{A}, \tag{12}$$

$$\frac{dp}{p} = \frac{\kappa M^2}{1-\kappa M^2}\frac{dA}{A}, \tag{13}$$

$$\frac{dM}{M} = -\frac{1}{1-M^2}\frac{dA}{A}. \tag{14}$$

Substituting Eq. (13) for Eq. (14) and performing the integration, we obtain the following relationship between pressures and Mach numbers:

$$\frac{p_2}{p_1} = \exp\left\{\frac{\kappa\left(M_1{}^2 - M_2{}^2\right)}{2}\right\}. \tag{15}$$

Figure 8 shows the static pressure drop in the isothermal process with the inlet Mach number M_1=0.

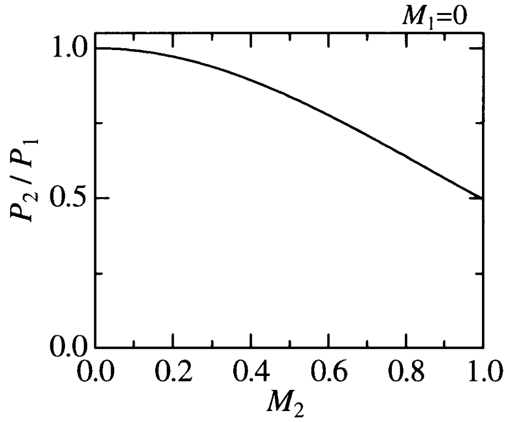

Figure 8 Ratio of static pressures at outlet to that at inlet in the case of dT=0

In the case of M_1=0 and M_2=1, the value of p_2/p_1 is 0.497, which means that, in the isothermal expansion combustion nozzle, pressure is reduced by half at the most.

For example, we consider the isothermal system in which static pressure is changed from 40[atm] to 10[atm]. Here we assume the dynamic pressure at 40[atm] could be zero. At first, the static pressure is decreased to 20[atm] in the isothermal expansion combustion nozzle, and next an zero-reaction turbine is used in order to retard the flow and extract the work. Further, the same combination of nozzle and turbine reduces the pressure from 20[atm] to 10[atm]. Therefore, two pairs of isothermal expansion combustion nozzle and impulse turbine are needed.

6.2 Lean Combustion

High-temperature gas has a much larger thermal energy than kinetic energy, so that the temperature descent accompanying the flow acceleration is not large. Assuming that the condition is adiabatic (i.e., dh=0) and substituting Eq.(8) for Eq.(7), the integration yields the following relationship:

$$\frac{T_2}{T_1} = \frac{1+\dfrac{\kappa-1}{2}M_1^{\ 2}}{1+\dfrac{\kappa-1}{2}M_2^{\ 2}} \ . \qquad (16)$$

In the case that the temperature fall is maximum with M_1=0 and M_2=1, we obtain T_2/T_1=1/1.2; even if still gas at 1500[K] is introduced into the adiabatic nozzle and accelerated to M_2=1, the temperature drop is only 250[K]. In other words, the degree of temperature descent due to flow acceleration is so small that lean combustion is necessary for a single isothermal expansion combustion nozzle.

Here, we consider a proportion of heat released from a single stage of isothermal expansion combustion to that from the entire combustion process. Using Eq. (15), energy added in an isothermal process is represented as

$$Q = RT\ln\left(\frac{p_1}{p_2}\right) = \frac{\kappa RT(M_2 - M_1)}{2} \ , \qquad (17)$$

where we obtain

$$Q = \frac{\kappa RT}{2} \qquad (18)$$

in the case that Mach number increases from 0 to unity. In the ITECT cycle including two stages of isothermal expansion combustion similar to the previous section, the ratio of energy estimated from Eq. (18) to that from the entire combustion process is about 0.24; in other words, a quarter of combustion could be executed in a single isothermal expansion combustion nozzle. In summary, although lean combustion is necessary for a single stage of isothermal expansion combustion, the effect of isothermal expansion combustion on the whole system is not small.

Although the fuel injected into the isothermal expansion combustion nozzle must be thinner than that in a conventional combustor, we consider that the ignition is possible because the gas has a very high temperature.

7. CONCLUDING REMARKS

Isothermal expansion combustion has the potential to produce mechanical energy instead of thermal energy. The gas-turbine cycle which includes the isothermal expansion combustion process, is proposed. The following characteristics were obtained through consideration of this ITECT cycle:

(1) Compared with the conventional gas-turbine cycle, the ITECT cycle has high efficiency particularly in the high pressure region.

(2) The ITECT cycle is suitable for the topping of a combined cycle because the temperature of the exhaust gas is relatively high.

(3) In the isothermal expansion combustion nozzle, the pressure drop is restricted by the condition that the Mach number must be less than unity, so that pressure is reduced by half at the most.

(4) Lean combustion is necessary for the isothermal expansion combustion nozzle; however the proportion of the isothermal expansion combustion to the entire combustion process is not small.

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