Research paper

Experiments on energy balance and thermal efficiency of pulse detonation turbine engine

Tomoaki Takahashi*, Akihiko Mitsunobu*, Yudai Ogawa*, Shin-ichi Kato*, Hiroyuki Yokoyama*, Akio Susa*, and Takuma Endo*†

*Department of Mechanical Systems Engineering, Hiroshima University,
1−4−1 Kagamiyama, Higashi−Hiroshima, Hiroshima 739−8527, JAPAN
Phone : +81−82−424−7567
† Corresponding address : takumaendo@hiroshima-u.ac.jp

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Abstract

A water-cooled single-tube pulse detonation turbine engine (PDTE) system was constructed using a turbocharger designed for automobiles, where the output of the system was the compression work done by the turbocharger. The PDTE system was operated for more than 10 minutes at the frequencies of 10 and 60 Hz using hydrogen as the fuel and air as the oxidizer. In the experiments, the isentropic efficiency of the turbocharger and the energy balance of the system were evaluated by comprehensive diagnostics. The isentropic efficiency of the turbocharger was about 0.6 at the operation frequencies of both 10 and 60 Hz, which was almost the same as the value in the case that only air steadily flowed through the rig. The evaluation of the energy balance showed that about four out of ten of the input energy was lost in the operation. The values of the thermal efficiency evaluated in the PDTE experiments at the frequencies of 10 and 60 Hz were 0.1 and 0.09, respectively. These values were consistent with the theoretical predictions taking account of the evaluated isentropic efficiency of the turbocharger and energy balance of the system.

Keywords : detonation, engine, turbine, efficiency, energy balance

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1. Introduction

A pulse detonation engine (PDE) is an internal combustion engine (ICE), in which fuel is intermittently burned as self-sustained detonations[1], and more than five decades have passed since the first journal paper on PDE experiments[2]. Because the theoretical thermal efficiency of a PDE is higher than that of a conventional gas turbine engine based on isobaric combustion[3], a PDE is expected to be a next-generation high-efficiency ICE.

PDEs have been developed mainly for propulsion applications so far. Especially, the performance of a simplified PDE has been clarified in detail[4)−6]. Here we call a straight detonation tube with fixed cross section, one end of which is closed and the other end open, a simplified PDE. However since preliminary experiments on a turbine driven by pulsed detonations performed in Air Force Research Laboratory[7, 8], the concept of a pulse detonation turbine engine (PDTE) has drawn attention[9)−12]. A PDTE is such an engine that steady-flow isobaric combustion in a conventional gas turbine engine is replaced with pulsed detonations. That is, the model thermodynamic cycle of a PDTE is the following. The working gas is isentropically compressed first. After the initial compression, the working gas is burned as a self-sustained detonation. Finally, the burned gas is isentropically expanded to the initial pressure.

Although the theoretical thermal efficiency of the model thermodynamic cycle of a PDTE is higher than that of the Brayton cycle, which is the model thermodynamic cycle of a conventional gas turbine engine, the measured values of the thermal efficiency of PDTEs in the early-stage experiments were lower than the theoretical predictions by more than one order[8, 11]. Recently, Rasheed *et al*. reported interesting experimental results[12]. In their experiments, the isentropic efficiency of a turbine driven by pulsed detonations was comparable to that driven by steady flow, and the overall rig efficiency of the PDTE system was higher than that in the case where the same rig was operated as a steady burner turbine engine at corresponding turbine conditions. Although the results reported by Rasheed *et al*. are encouraging, the thermal efficiency of their PDTE system seems still lower than that of the model thermodynamic cycle.

In this paper, we report experimental results on the energy balance of a PDTE system as well as the isentropic efficiency of a turbine. We prepared a water-cooled single-tube PDTE system and operated it at the frequencies of 10 and 60 Hz for more than 10 minutes. At the final stage of the operation, the output of the PDTE system was almost in steady state, and we measured output, various losses, and conditions of the working gas at the inlet and outlet of the turbine. The values of the isentropic efficiency of the turbine measured at both operation frequencies were almost the same, and comparable to that in the case where the turbine was driven by steady air flows through the same rig. The measured values of the thermal efficiency of the PDTE system based on the lower heating value of the fuel was about 9% and 10% at the operation frequencies of 60 and 10 Hz, respectively. Taking account of the measured losses and turbine efficiency, the measured values of the thermal efficiency was found to be consistent with the theoretical predictions.

2. Experimental arrangement and methods

Figure 1 shows the experimental arrangement. The PDTE system consisted of a detonation tube, a buffer chamber, an adapter pipe, and a turbocharger which was for 660-cc automobiles: HITACHI 1390075F52. The detonation tube was a straight tube made of stainless steel whose inner diameter was 16 mm, thickness was 8 mm, and length was 642 mm. The detonation tube was cooled by water as shown in Figure 1. For measurement of the energy balance, the flow rate of the cooling water was measured by a vortex flowmeter, and the temperature of the cooling water was measured at the inlet and outlet of the water-cooling system by thermocouples. Also, we measured temperature at several portions of the system by thermocouples. Hydrogen as fuel and air as oxidizer were separately injected into the detonation tube, and mixed inside it. The hydrogen was supplied to the detonation tube from a steel cylinder through a regulator and a magnetic valve. The air was supplied to the detonation tube from an external air compressor driven by a diesel engine through a dryer, a large plenum chamber, and a magnetic valve. The air was also used for purging the residual hot burned gas before injecting the fresh fuel. The flow rate of each gas was adjusted by the supply pressure and by an orifice or a flow-rate adjuster. The flow-rate adjuster is an implement, which is inserted midway in the gas-feed pipe, and by changing whose inner diameter the flow rate of the gas is adjusted. The detonable gas mixture was ignited by using two spark plugs. For shortening the deflagration-to-detonation-transition distance, 14 obstacle pipes, which were orthogonal alternately and with 12.5mm separation, were installed downstream the spark plugs. The obstacle pipes were inclined 45 degrees from the vertical so that cooling water flowed inside the obstacle pipes by natural convection. The detonation initiation in each cycle was confirmed by using three ionization probes installed on the side wall of the detonation tube.

In order to protect the turbine blades against destructive shock waves from the detonation tube, we installed the buffer chamber, whose inner diameter was 120 mm, thickness was 10 mm, and length was 298 mm, downstream the detonation tube. And in order to protect the turbine blades against overheat, we installed the adapter pipe, whose inner diameter was 25 mm, thickness was 12.5 mm, and length was 200 mm, between the buffer chamber and the turbine inlet, and injected secondary air continuously to the system as shown in Figure 1 by bifurcating the air-feed piping. By this secondary-air injection, the time-averaged turbine-inlet temperature was sufficiently lower than the tolerable temperature which

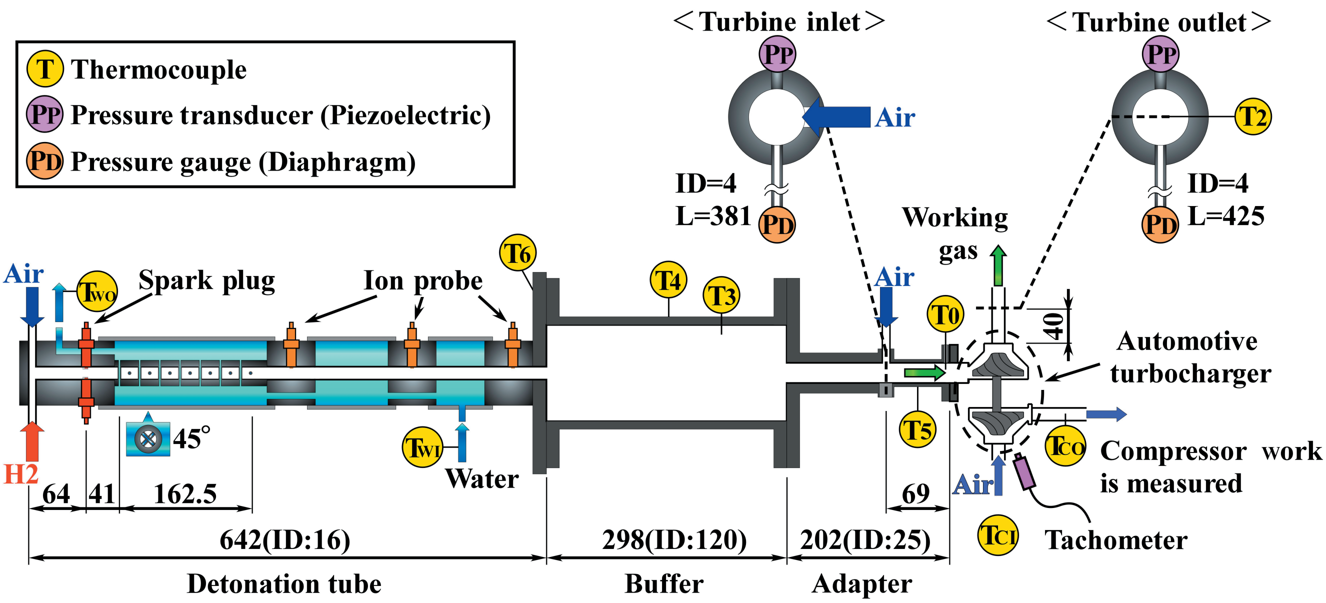

Figure 1 Experimental arrangement.

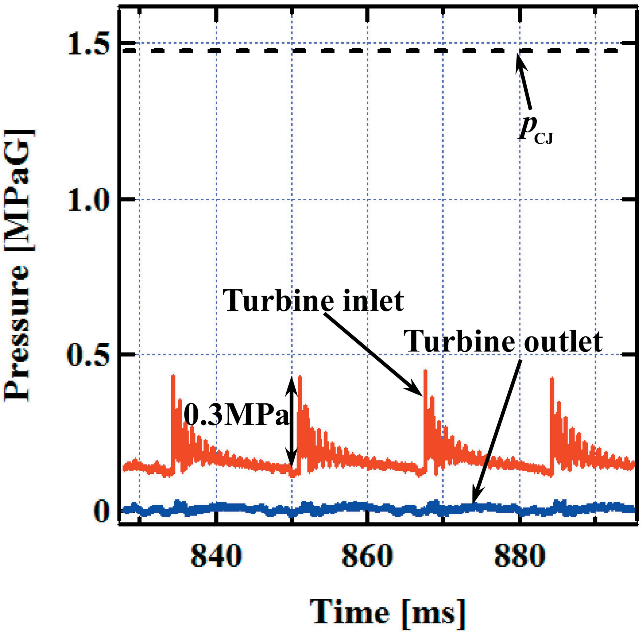

Figure 2 Pressure histories at the turbine inlet and outlet (operation frequency: 60 Hz).

Table 1 Experimental conditions.

Operation frequency [Hz]	Operation time [min]	Gas-supply pressure [MPaG]		Equivalence ratio	Fill fraction of the detonable gas in the detonation tube
		Air	H_2		
60	10	0.54	0.32	1.1 ± 0.2	< 0.74
10	20	0.54	0.25	1.0 ± 0.1	0.74 - 0.93

was approximately 900 °C.

In order to diagnose the conditions of the working gas at the inlet and outlet of the turbine, we installed piezoelectric pressure transducers, diaphragm-seal-type pressure gauges (DSPGs), and thermocouples as shown in Figure 1. The DSPGs were installed to the system through the extension pipes as shown in Figure 1 because they stand up poorly to heat. Although DSPGs are suited to accurate measurement of absolute pressure, their response time is long. On the other hand, although piezoelectric pressure transducers are not suited to accurate measurement of absolute pressure, their response time is short. Therefore, we corrected the pressure histories measured by the piezoelectric pressure transducers by using the pressure values measured by the DSPGs assuming that the time-averaged pressure values measured by both sensors are the same. Figure 2 shows examples of the corrected pressure histories at the turbine inlet and outlet, where the operation frequency of the detonation tube was 60 Hz. In Figure 2, p_{CJ}, which denotes the Chapman-Jouguet detonation pressure for the initial conditions of ambient pressure and temperature, is plotted as a reference. Because the tolerable shock pressure for the turbine blades was about 0.6 MPa, it was found that the shock waves were attenuated enough by the buffer chamber we installed. As the output of the PDTE system, we evaluated the compression work done by the turbocharger. For its evaluation, we measured the flow rate and total-enthalpy gain of air passing through the compressor part of the turbocharger. For the measurement of these quantities, we used an orifice flowmeter, a thermocouple, and a DSPG installed downstream the compressor. The revolving speed of the turbocharger was measured by using an optical tachometer.

In the experiments, we operated the PDTE system at the frequencies of 10 and 60 Hz in order to investigate the dependence of the system characteristics on the operation frequency. Table 1 summarizes the experimental conditions. The equivalence ratio was estimated by comparing the detonation speed measured by the ion probes with the Chapman-Jouguet detonation speeds calculated for various equivalence ratios. The fill fraction of the detonable gas in the detonation tube was estimated from the positions of the ion probes which sensed detonations.

In the case that the operation frequency was 10 Hz, the magnetic valve for the air injection was kept opened all through the operation time, and only the hydrogen injection was controlled by a magnetic valve. In this operation mode, the air supply is stopped when a detonation is initiated in the detonation tube due to its pressure higher than the air-supply pressure, and restarts when the pressure inside the detonation tube relaxes below the air-supply pressure. The quantities of the supplied air for the purge of the residual hot burned gas and the supplied detonable gas are controlled mainly by the open and close timings of the magnetic valve for the hydrogen injection, respectively. In the case that the operation frequency was 60 Hz, both of the magnetic valves for the injection of the air and hydrogen were kept opened all through the operation time. And the pulsed operation of the combustor was controlled by the spark plugs only. We call this operation mode the valveless mode[13]. In this operation mode, the supply pressure of the fuel is set lower than that of the air. Due to this difference in the supply pressures between the air and the fuel, the restart timing of the fuel supply after an explosion in the detonation tube is delayed against that of the air supply. By this delay, the supplied air preceding the fuel automatically purges the residual hot burned gas. In this operation mode, the quantities of the supplied air for the purge of the residual hot burned gas and the supplied detonable gas are controlled by the combination of four parameters: the supply pressures and the flow rates of the air and the fuel. Although the determination of the controlling parameters for stable operation is not so easy in this operation mode, this operation mode is suited for long operation because the system is free from moving components except for the turbocharger.

3. Results and discussions

3.1 Isentropic efficiency of turbocharger

The isentropic efficiency of a turbine η_T is defined by

$$\eta_T = \frac{L}{\left(h_0 + \frac{u_0^2}{2}\right) - h_{2s}}, \tag{1}$$

where L is the work actually done by the turbine per unit mass of the working gas, h_0 and u_0 are the specific enthalpy and flow speed, respectively, of the working gas at the turbine inlet, and h_{2s} is the specific enthalpy of the working gas at the turbine outlet in the case that the flow in the turbine is isentropic. Assuming that the working gas is a calorically perfect gas, the denominator of the above formula is evaluated by

$$\left(h_0 + \frac{u_0^2}{2}\right) - h_{2s} = c_{p\_\mathrm{WG}} T_0 \left[1 - \left(\frac{p_2}{p_0}\right)^{\frac{R_\mathrm{WG}}{c_{p\_\mathrm{WG}}}}\right] + \frac{u_0^2}{2}, \tag{2}$$

where T_0 and p_0 are the temperature and pressure, respectively, of the working gas at the turbine inlet, p_2 is the pressure of the working gas at the turbine outlet, and c_{p_WG} and R_WG are the specific heat at constant pressure and specific gas constant, respectively, of the working gas. In the data reduction, we used the time-averaged values for T_0, p_0, and p_2, where the value of T_0 was measured by the thermocouple T_0 in Figure 1. The values of c_{p_WG} and R_WG were estimated as their mass-based averaged values for the working gas, namely the burned gas, which was assumed to be isentropically expanded to the ambient pressure after detonation, and the room-temperature air which was used for the purge of the residual hot burned gas and for cooling the turbine blades. The value of u_0 was evaluated from the cross-sectional area of the turbine inlet and the time-averaged volume flow rate of the working gas. The specific work L can be expressed as $L = \dot{W}/\dot{m}_\mathrm{WG}$, where $\dot{W}$ is the work actually done by the turbine per unit time and $\dot{m}_\mathrm{WG}$ is the mass flow rate of the working gas. The value of $\dot{W}$ was measured as the compression work actually done by the turbocharger per unit time, and evaluated by

$$\dot{W} = \dot{m}_\mathrm{Air} \left[c_{p\_\mathrm{Air}} (T_\mathrm{CO} - T_\mathrm{CI}) + \left(\frac{u_\mathrm{CO}{}^2}{2} - \frac{u_\mathrm{CI}{}^2}{2}\right)\right], \tag{3}$$

where $\dot{m}_\mathrm{Air}$, c_{p_Air}, T, and u are the mass flow rate, specific heat at constant pressure, temperature, and flow speed, respectively, of the air passing through the compressor, and subscripts CO and CI denote the downstream and upstream of the compressor. The values of $\dot{m}_\mathrm{Air}$ and u_CO were evaluated by using an orifice flowmeter. The value of T_CO was measured by the thermocouple T_CO in Figure 1. The values of c_{p_Air} and T_CI were of the ambient room-temperature air, and u_CI was approximated as $u_\mathrm{CI} \approx 0$. The value of T_CI was measured by the thermocouple T_CI in Figure 1, which was actually placed far from the rig and sensed the room temperature. This method for evaluation of $\dot{W}$ means that the evaluated isentropic efficiency is of rather a turbocharger than a turbine. Here, we introduce a quantity c_ϕ called the spouting velocity, which is the maximum available velocity for expansion through the turbine, defined by

Table 2 Measured conditions of the turbine-drive experiments.

Drive condition	Turbine-inlet temperature [°C]	Turbine-inlet pressure [MPaG]	Turbine-outlet pressure [MPaG]	Revolving speed [rpm]
Pulsed detonations (60 Hz)	306.2 ± 1.7	0.167 ± 0.007	0.004 ± 0.006	159000
Pulsed detonations (10 Hz)	104.4 ± 0.6	0.116 ± 0.037	0.006 ± 0.005	129000
Steady air flow	31.0 ± 0.1	0.101 ± 0.003	0.006 ± 0.003	118000

$$\frac{c_\phi^2}{2} = \left(h_0 + \frac{u_0^2}{2}\right) - h_{2s}. \tag{4}$$

The dimensionless parameter U/c_ϕ, where U is the peripheral velocity of the rotor blades, is known as one of the basic dimensionless parameters governing the performance of a turbine.

For investigation of the dependence of the isentropic efficiency of the turbocharger η_T on the operation frequency, we experimentally evaluated η_T in three cases: (1) the turbocharger was driven by pulsed detonations at 60 Hz, (2) the turbocharger was driven by pulsed detonations at 10 Hz, and (3) the turbocharger was driven by steady air flow. Table 2 summarizes the measured conditions of the turbine-drive experiments. The operation conditions of the case (3) corresponded to those in the idling phase of the cases (1) and (2), and the operation time of the case (3) was 5 minutes. The values in Table 2 were the time-averaged ones in the aftermost 10 seconds of the corresponding operation. The results on η_T are summarized in Figure 3, where the experimentally-evaluated η_T is plotted against the velocity ratio U/c_ϕ. The isentropic efficiency of the turbocharger η_T was evaluated by using time-averaged quantities in the aftermost 10 seconds of the corresponding operation. The squares show the values of η_T in the case that only air steadily flowed through the rig. As shown in Figure 3, the values of the isentropic efficiency of the turbocharger are almost the same in all three cases. This result is similar to that obtained by Rasheed *et al*[12]. The reason why the isentropic efficiency of the turbocharger was not lowered in the case of pulsed gas flow as shown in Figure 2 has not been clarified in detail yet. However, it may be due to the characteristics of the turbocharger specially designed for automobiles. That is, it is possible that the turbocharger we used was specially designed intending allowable performance in relatively wide turbine-drive conditions at sacrifice of peak efficiency. Hence, similar investigation by using a state-of-the-art turbine designed for steady gas flow intending highest efficiency is an important issue in the future.

3.2 Energy balance of PDTE system

As stated earlier, the output per unit time of the PDTE

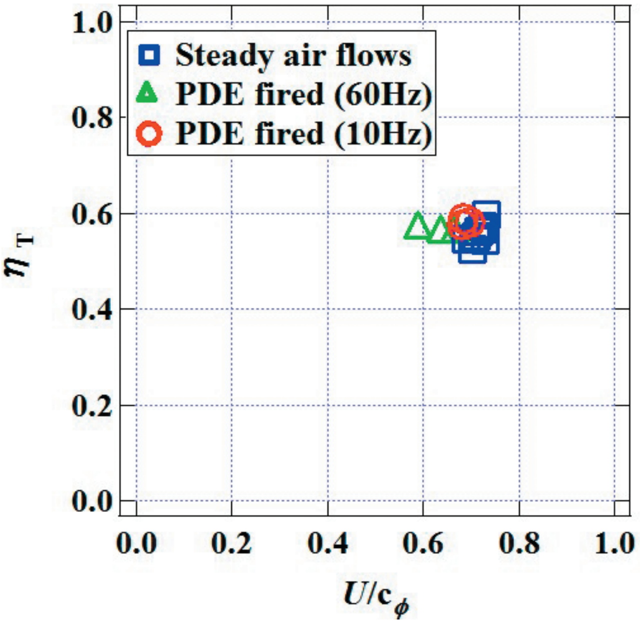

Figure 3 Isentropic efficiency of the turbocharger.

system $\dot{W}$ was measured as the compression work actually done by the turbocharger per unit time and evaluated by Eq. (3). By using a measured value of $\dot{W}$, the thermal efficiency of the PDTE system η_{th} was evaluated by

$$\eta_{\mathrm{th}} = \frac{\dot{W}}{q_{\mathrm{F}} \dot{m}_{\mathrm{F}}}, \tag{5}$$

where q_{F} and $\dot{m}_{\mathrm{F}}$ are the specific lower heating value and mass flow rate, respectively, of the fuel. The mass flow rate of the fuel $\dot{m}_{\mathrm{F}}$ was evaluated from the operation time and the pressure difference of the hydrogen cylinder between before and after the operation as the time-averaged one.

The energy exhausted with the working gas per unit time $\dot{Q}_{\mathrm{exhaust}}$ was evaluated by

$$\dot{Q}_{\mathrm{exhaust}} = \dot{m}_{\mathrm{WG}} \left(c_{p\_\mathrm{WG}} T_2 + \frac{u_2^2}{2} \right), \tag{6}$$

where T_2 and u_2 are the temperature and flow speed, respectively, of the working gas at the turbine outlet. The value of T_2 was measured by the thermocouple T_2 in Figure 1. The value of u_2 was evaluated from the cross-sectional area of the turbine outlet, and the time-averaged volume flow rate of the working gas. The heat loss to the cooling water per unit time $\dot{Q}_{\mathrm{water}}$ was evaluated by

$$\dot{Q}_{\mathrm{water}} = \dot{m}_{\mathrm{water}} c_{p\_\mathrm{water}} (T_{\mathrm{water\_out}} - T_{\mathrm{water\_in}}), \tag{7}$$

where $\dot{m}_{\mathrm{water}}$ is the mass flow rate of the cooling water, c_{p_water} is the specific heat at constant pressure of water, and $T_{\mathrm{water\_in}}$ and $T_{\mathrm{water\_out}}$ are the temperatures of the cooling water at the inlet and outlet, respectively, of the water-cooling system. The values of $T_{\mathrm{water\_in}}$ and $T_{\mathrm{water\_out}}$ were measured by the thermocouples T_{WI} and T_{WO} in Figure 1, respectively. The energy loss at the turbocharger per unit time $\dot{Q}_{\mathrm{turbo}}$ was evaluated by

$$\dot{Q}_{\mathrm{turbo}} = \dot{m}_{\mathrm{WG}} \left[c_{p\_\mathrm{WG}} (T_0 - T_2) + \left(\frac{u_0^2}{2} - \frac{u_2^2}{2} \right) \right] - \dot{W}. \tag{8}$$

It should be noted that the energy loss at the turbocharger per unit time $\dot{Q}_{\mathrm{turbo}}$ includes not only heat loss but also mechanical loss because the turbocharger has moving components. The heat loss to the ambient air from the outer surface of the buffer chamber or adapter pipe per unit time $\dot{Q}_{\mathrm{air}}$ was evaluated by

$$\dot{Q}_{\mathrm{air}} = \frac{Nu_d \lambda}{d} (T_{\mathrm{wall}} - T_\infty) \pi d L, \tag{9}$$

where λ is the thermal conductivity of the ambient air, d and L are the outer diameter and length, respectively, of the corresponding component, T_{wall} is the temperature of the outer surface of the corresponding component, T_∞ is the temperature of the ambient air far from the PDTE system, and Nu_d is the Nusselt number based on the diameter d, which is empirically given by[14]

$$Nu_d = \left(0.60 + 0.387 \left\{ \frac{Gr_d Pr}{[1 + (0.559/Pr)^{9/16}]^{16/9}} \right\}^{1/6} \right)^2. \tag{10}$$

In the above formula, Pr is the Prandtl number of the ambient air, and Gr_d is the Grashof Number of the ambient air based on the diameter d, which is given by

$$Gr_d = \frac{d^3 g \beta (T_{\mathrm{wall}} - T_\infty)}{\nu^2}, \tag{11}$$

where g, β, and ν are, respectively, the gravitational acceleration, the coefficient of thermal expansion of the ambient air, and the kinematic viscosity of the ambient air. The material properties of the ambient air were evaluated assuming that the temperature was $(T_{\mathrm{wall}} + T_\infty)/2$. The value of T_∞ was measured by the thermocouple T_{CI} in Figure 1. The values of T_{wall} were measured by the thermocouple T_4 in Figure 1 for the buffer chamber and by the thermocouple T_5 in Figure 1 for the adapter pipe. The empirical formula (10) is valid in the range $10^{-6} < Ra < 10^9$, where $Ra = Gr_d Pr$ is the Rayleigh number. The Rayleigh number estimated for the present experiments was between 3×10^5 and 2×10^7. Therefore, eq. (10) is valid for our experimental conditions. When the PDTE system was not completely in thermal steady state, the heat loss to the solid components per unit time $\dot{Q}_{\mathrm{solid}}$ was evaluated by

$$\dot{Q}_{\mathrm{solid}} = \sum_i m_{\mathrm{solid},i} c_{p\_\mathrm{solid},i} \frac{dT_{\mathrm{solid},i}}{dt}, \tag{12}$$

where $m_{\mathrm{solid},i}$, $c_{p\_\mathrm{solid},i}$, and $T_{\mathrm{solid},i}$ are the mass, specific heat at constant pressure, and temperature, respectively, of the solid component i of the PDTE system. In the experiments, we approximately evaluated $\dot{Q}_{\mathrm{solid}}$ taking account of a large flange only and measuring the local surface temperature at a point only. That is, the value of T_{solid} was measured by the thermocouple T_6 in Figure 1. This might result in underestimation of $\dot{Q}_{\mathrm{solid}}$.

Because we wanted to know the energy balance against the input based on the lower heating value of the fuel, the output $\dot{W}$, the energy exhausted with the working gas

$\dot{Q}_{\mathrm{exhaust}}$, and the energy loss at the turbocharger $\dot{Q}_{\mathrm{turbo}}$ obtained in the idling phase preceding each PDTE operation, which were produced by air flow induced by the external compressor, were subtracted from the corresponding quantities obtained in the aftermost 10 seconds of the PDTE operation.

Figure 4 summarizes the experimental results on the energy balance. As shown in Figures 4(a) and 4(b), the output (the compression work done by the turbocharger) was almost in steady state in the final stage of the PDTE operation, although the PDTE system was not completely in thermal steady state even in the final stage of the PDTE operation because about 5% of the input energy was lost to the solid component as shown in Figure 4(c). As stated earlier, the heat loss to the solid component $\dot{Q}_{\mathrm{solid}}$ was possibly underestimated, the missing energy expressed as "Missing" in Figure 4(c) might be lost to the solid components. It is remarkable that about four out of ten of the input energy was lost as shown in Figure 4(c). This suggests that the reduction of the energy loss in the PDTE system is an extremely important issue for the development of high-efficiency PDTE systems. The energy loss to the cooling water in the 10-Hz case was much smaller than that in the 60-Hz case. This is because the flow-rate ratio of the burned gas to the air for purge, which significantly affects the time-averaged temperature of the working gas, in the 10-Hz case was much smaller than that in the 60-Hz case.

The experimentally-obtained thermal efficiency of the PDTE system was 0.09-0.1 as shown in Figure 4. The theoretical thermal efficiency corresponding to the experiments was about 0.35 considering that the turbine-inlet pressure was about twice the ambient pressure when only air steadily flowed through the rig as shown in Table 2. Taking account of the measured energy balance, the expected thermal efficiency is lowered to 0.19-0.22. Furthermore, taking account of the measured isentropic efficiency of the turbocharger, the expected thermal efficiency is further lowered to 0.11-0.13. Although this is a crude estimation, we conclude that the obtained results are self-consistent.

4. Conclusions

A water-cooled single-tube pulse detonation turbine engine (PDTE) system was operated at the frequencies of 10 and 60 Hz for more than 10 minutes using hydrogen as the fuel and air as the oxidizer. The isentropic efficiency of the turbocharger and the energy balance of the PDTE system were evaluated by comprehensive diagnostics. The isentropic efficiency of the turbocharger was about 0.6 at the operation frequencies of both 10 and 60 Hz, which was almost the same as the value in the case that only air steadily flowed through the same rig. Because the turbine isentropic efficiency of about 0.6 for steady air flow is rather low compared with that of a state-of-the-art turbine, experiments with better-performance turbines are needed in the future. The thermal efficiency based on the lower heating value of the fuel was 0.09 at the operation frequency of 60 Hz, and 0.1 at the operation frequency of 10 Hz. These values were consistent with the theoretical predictions taking account of the evaluated energy balance and isentropic efficiency of the turbocharger. The experimentally-evaluated energy balance suggests that the reduction of the energy loss in the PDTE system is an extremely important issue for the development of high-efficiency PDTE systems because about four out of ten of the input energy was lost in the operation. When the heat loss to the cooling water was reduced by lowering the time-averaged temperature of the working gas via the alteration of the operation frequency from 60 to 10 Hz, the thermal efficiency was slightly increased. This shows the possibility that the

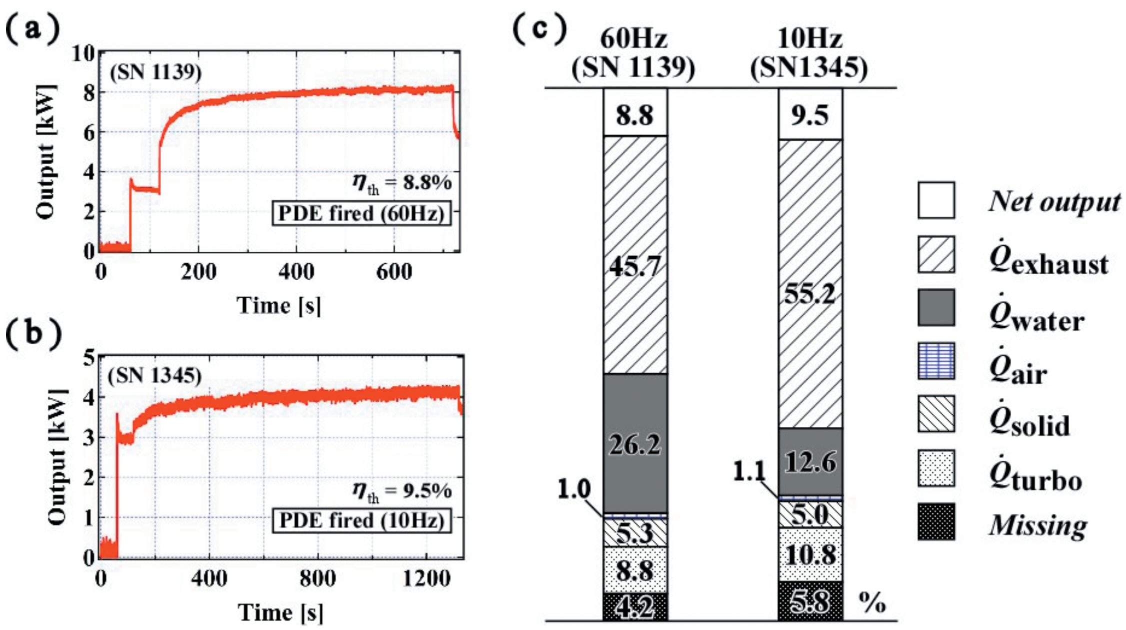

Figure 4 Experimental results on the energy balance.
(a) Time history of the output for the 60-Hz case.
(b) Time history of the output for the 10-Hz case.
(c) Energy balance.

thermal efficiency will be increased by the reduction of heat loss.

Acknowledgment

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パルスデトネーションタービンエンジンの熱効率とエネルギーバランスに関する実験

高橋友彰*，光延昭彦*，小川雄大*，加藤槇一*，横山裕之*，須佐秋生*，遠藤琢磨*†

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自動車用に設計されたターボチャージャーを用い，水冷型単気筒のパルスデトネーションタービンエンジン（PDTE）システムを構築した。このシステムの出力はターボチャージャーが行う圧縮仕事である。燃料に水素を，酸化剤に空気を用い，PDTEシステムを周波数10Hzおよび60Hzで10分間以上運転した。実験では，包括的な診断により，ターボチャージャーの等エントロピー効率およびシステムのエネルギーバランスを実験的に評価した。ターボチャージャーの等エントロピー効率は10Hzおよび60Hzいずれの運転周波数においても約0.6であり，この値は実験装置に空気のみを定常的に流した場合の値とほぼ同じであった。また，エネルギーバランスの評価により，入力エネルギーの約４割が運転中に失われていることがわかった。PDTE運転時の熱効率は，運転周波数10Hzの場合0.1，運転周波数60Hzの場合0.09であった。これらの値は，評価されたターボチャージャーの等エントロピー効率およびシステムのエネルギーバランスを考慮すると理論値と矛盾しないものであった。

*広島大学 機械システム工学専攻　〒739−8527 広島県東広島市鏡山1−4−1
Phone：082−424−7567
†Corresponding address：takumaendo@hiroshima−u.ac.jp