ON THE UTILIZATION OF THE KINETIC ENERGY OF PULSED DETONATION PRODUCTS

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The possibility of utilizing the kinetic energy of detonation products by a pulse turbine of the simplest water-wheel-like design during the implementation of the Zel'dovich thermodynamic cycle with pulse detonation combustion of fuel is investigated computationally and experimentally. The coefficients of utilization of the momentum and kinetic energy of detonation products in the pulse turbine with unoptimized mass and dimensions are found to be as low as 8%–16%. To improve the efficiency of the pulse turbine, it is necessary to take measures for eliminating unnecesary reflections of shock waves (SWs), to select the optimal mass and dimensions of the turbine rotor and the number of blades, to profile the blades and to select the optimal angle of attack, to optimize the size of the lateral gap between the rotor and the housing, and to select the optimum location of the exhaust duct. It is expected that the efficiency of a combined cycle including the optimized pulse turbine and conventional gas and/or steam turbines attached to the exhaust duct could be higher than the efficiency of a conventional hybrid cycle by at most 9%.

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Introduction

Modern power plants, in which, as a rule, operation cycles are used with the deposition of heat due to deflagrative combustion of various combustible mixtures, have reached a high degree of perfection. Further increase in their energy efficiency can be ensured by changing the combustion mode. By its thermodynamic efficiency, the most attractive combustion mode is detonation [1]. In the traveling detonation wave (DW), the maximum intensity of chemical energy release is reached: the energy is released in the self-ignition mode at very high local density and temperature in a thin layer of a shock-compressed combustible mixture. Unlike products of slow combustion (deflagration), products of supersonic combustion (detonation), in addition to thermal energy, also possess a large kinetic energy, which can be utilized to perform additional useful work. Consequently, the replacement of conventional combustion by a detonative combustion, i. e., the transition from a combustion cycle at constant pressure $P = const$ to the Zel'dovich cycle should ensure an increase in the thermodynamic efficiency of the power plant. At present, the Zel'dovich cycle is mainly considered in connection with the development of thrust (see [2] and references therein). As for energy devices with conversion of the chemical energy of fuel into mechanical work, they are acutely concerned with the question: how to maximize utilization of the kinetic energy of detonation products?

In the literature, it was suggested to utilize the kinetic energy of detonation products obtained with the help of pulse detonation tubes for powder coatings [3, 4], for ultrafine aerodynamic fragmentation of viscous liquids and coal-in-water emulsions [5, 6], for grinding and melting the snow–ice mass [7], for detonation-assisted stamping of housing parts [8], for land-based electric power generation [9], etc. In [9–14], kinetic energy of detonation products was suggested to be utilized by directing SWs and high-speed jets to a standard (axial or centrifugal) gas turbine. To minimize the action of SWs on the gas turbine blades, it is proposed to place an intermediate damping volume between a pulse detonation tube and the gas turbine, which dissipates the kinetic energy of the detonation products. Mechanical energy from a gas turbine can be transmitted to consumers, an electric generator, and/or a compressor that directs compressed air into the air supply system of the pulse detonation tube. Hot detonation products expanded in the turbine could be utilized for thermal and chemical preparation of fuel and for further energy conversion. However, it is noted that thermomechanical loads on the gas turbine and other structural elements of the power plant caused by SWs can significantly exceed the maximum permissible values and lead to resource reduction.

The issues discussed above for pulse detonation devices seem to be less relevant to the integration of the conventional axial or centrifugal turbines with so-called stationary (or "rotating") detonation combustors [15, 16]. However, various imperfections like unsteady flow of detonation products with cyclic variations of velocity, pressure, and temperature are still the issue as well as the necessity of cooling the products ahead of the turbine.

In view of it, the question arises: what kind of a device should one use to efficiently utilize the kinetic energy of detonation products generated by a pulse detonation combustor? It is implied that after utilization of the kinetic energy of detonation products, one can apply conventional axial or centrifugal gas and/or steam turbines to further utilize the thermal energy of the products. By other words, one can consider a three-stage combined thermodynamic cycle with successive utilization of the kinetic energy and thermal energy of detonation products in different devices.

The objective of this work is to evaluate the efficiency of utilization of the kinetic energy of detonation products in a pulse turbine of the simplest water-wheel-like design and to determine ways to increase this efficiency.

Calculations

Computational studies are aimed at understanding the physical processes in the pulse turbine of the water-wheel type used to convert the kinetic energy of the detonation products, periodically ejected from the pulse detonation tube, into the rotational energy of the turbine rotor. Figure 1 presents a pulse turbine of the water wheel type of the simplest design. The pulse turbine is a sealed cylindrical housing with a coaxial cylindrical blade wheel (rotor) on the shaft. The rotor

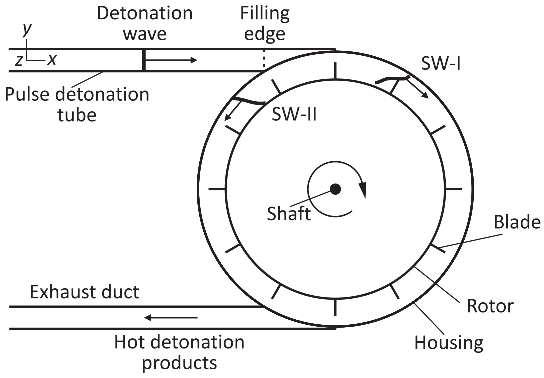


Figure 1 Schematic of a pulse turbine for converting the kinetic energy of detonation products into rotational energy of a turbine rotor

blades are made in the form of plates partially blocking the annular gap between the housing and the rotor. The blockage factor δ is defined as the ratio of the height of the blade to the height of the annular gap.

The operation principle of the pulse turbine is as follows. A pulse detonation tube, operating, e. g., on the principle set forth in [8, 17], periodically generates a DW and a high-speed jet of detonation products. The primary SW and the jet of detonation products are guided from the pulse detonation tube tangentially to the pulse turbine (see Fig. 1). When the SW enters the annular gap, it diffracts with the formation of two SWs: SW-I, traveling clockwise, and SW-II traveling in the opposite direction. Both SWs propagating in the annular gap will be weakened by interaction with the rotor blades. In the schematic of Fig. 1, the intensity of the SW-I will be greater than the intensity of the SW-II at the same distance from the inlet to the turbine, due to the predominant tangential motion of the detonation products clockwise. Because of multiple reflections of SW-I and SW-II from the rotor blades and due to the transfer of momentum from the high-speed jet of detonation products to the rotor, the energy of directed motion of SWs and a high-speed jet of detonation products is partially converted into energy of directional rotational motion of the rotor and partially dissipated due to dissipative processes. The mechanical energy of the rotor can be transmitted through the shaft to a consumer of mechanical energy and hot detonation products can be directed through the exhaust duct for further utilization of thermal energy.

The question we studied with the help of two-dimensional numerical calculations is the effect of the blockage factor δ on the degree of utilization of the momentum and kinetic energy of the detonation products and on the torque on the rotor shaft. For the sake of definiteness, the pulse turbine had the following characteristics: a diameter of the central body of the rotor of 0.4 m; height of the annular gap of 0.05 m; number of blades of 12; and blockage factor of $\delta = 0.1$, 0.2, 0.4, 0.6, 0.8, and 0.9. The DW was assumed to enter the pulse turbine from the pulse detonation tube, a channel 0.04 m high and 0.6 m long, filled with a stoichiometric propane–air mixture to the filling edge (see Fig. 1). Propagation of the DW in the tube was initiated by a short-time (5 ms) injection of hot gaseous combustion products through the left end of the tube with the velocity of the Chapman–Jouguet detonation products causing the formation of a detonation front. The calculations were based on the overall kinetic mechanism of self-ignition of the propane–air mixture [18], which provides an acceptable (in terms of concentrations of main chemical species) equilibrium composition of the detonation products at a detonation velocity of $D \approx 1860$ m/s. The calculations described below were performed for a quiescent rotor with the duration of the operation cycle in the pulse detonation tube $\Delta t_c = 3$ ms. The calculation procedure was the same as in [19].

To determine whether the rotation of the rotor affects the flow pattern, consider Fig. 2. Figure 2 compares the calculated instantaneous pressure fields in the annular gap of the pulse turbine in the case of a stationary rotor (Fig. 2*a*) and a moving rotor (Fig. 2*b*) at the time instants when the SW-I reached the first rotor blade. The calculation with a moving rotor is performed using a moving mesh for conditions where the rotor rotates at a constant speed of 4000 rpm. When the SW-I is reflected from the stationary blade, the maximum overpressure on the leeward side of the blade reaches 6.5 MPa, whereas when the SW-I is reflected from the moving blade, it is much lower: 4 MPa. Note that at the moment of initiation of the DW, the blade in Fig. 2*b* was in the same position as in Fig. 2*a*. These significant differences in

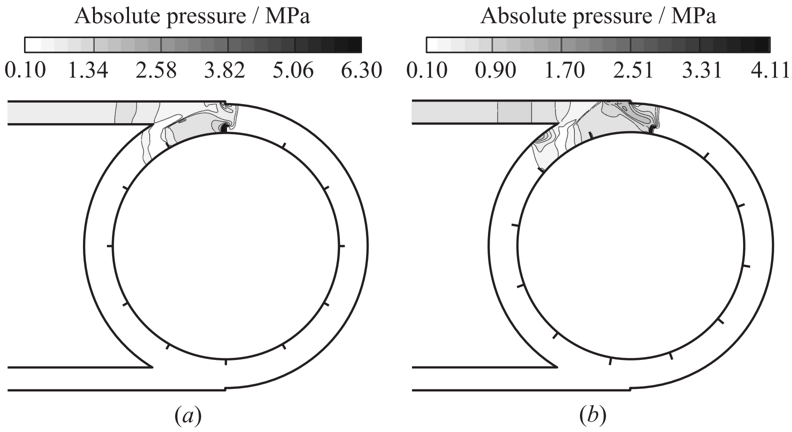


Figure 2 Examples of calculated instantaneous pressure fields in the annular gap of a pulse turbine at the time when the SW entering the turbine reached the first rotor blade in the case of a stationary (a) and moving (b) rotor (rotation speed 4000 rpm) with a blockage ratio $\delta = 0.2$

the pressure fields mean that at high rotor speeds, it is necessary to consider the effect of the rotation speed on the resulting torque acting on the rotor from the side of the SW and the flow of the detonation products.

Let us estimate the fraction of the momentum that is transferred by the detonation products to the turbine rotor. First, let us estimate the momentum of the detonation products at the inlet to the turbine. We will characterize the flow of detonation products by the local instantaneous value of the flow momentum i:

$$\vec{i} = P\vec{n} + \left(\vec{U} \cdot \vec{n}\right) \left(\rho\vec{U}\right)$$

where P is the absolute pressure; ρ is the density; U is the gas velocity; and $\vec{n}$ is the normal vector. By integrating over time the flux $\vec{i}$ through the cross section of the pulse detonation tube located at the inlet to the turbine, one obtains the total momentum of the detonation products, I_D, at the inlet to the turbine:

$$I_D = S_D \int\limits_0^t \left(\vec{i} \cdot \vec{n}\right) \, dt \tag{1}$$

where S_D is the cross-section area of the pulse detonation tube. The resultant dependence $I_D(t)$ is presented in Fig. 3. The initial linear portion of the curve $I_D(t)$ in the time interval from 0 to t_1 is formed due to the initial pressure of the combustible mixture. At $t_1 = 0.3$ ms, a DW arrives at the selected section and then detonation products flow through this section. Over a time interval from t_1 to $t_2 \approx 0.44$ ms, the integral (1) changes rapidly to a value of ~ 0.13 kg·m/s, caused both by the dynamic pressure of the detonation products and by their pressure. The further growth of the curve $I_D(t)$ is mainly determined by the pressure of the detonation products with an insignificant role of the velocity head. Within $\Delta t_c = 3$ ms (operation cycle), the total momentum of detonation products at the inlet to the turbine reaches a value of ~ 0.57 kg·m/s.

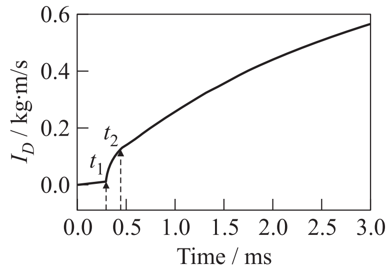


Figure 3 The calculated temporal dependence of the total momentum of the detonation products at the inlet to the turbine

Now, let us estimate the momentum transferred to the rotor by the detonation products. The instantaneous tangential force acting on the rotor blade can be determined from the ratio

$$F_\tau = \int\limits_{S_w} P_w \left(\vec{n} \cdot \vec{\tau}\right) \, ds$$

where S_w is the surface area of the blade; P_w is the static pressure at the blade surface; $\vec{n}$ is the normal vector to the blade surface; and $\vec{\tau}$ is the vector orthogonal to the radius-vector $\vec{r}$ originating in the rotor center. The total momentum transferred to the rotor is determined by the integral:

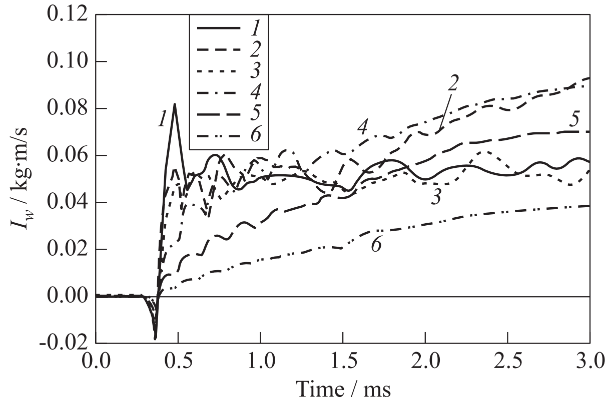


Figure 4 The calculated temporal dependencies of the momentum transferred to the rotor from the detonation products for different values of the blockage factor: *1* — $\delta = 0.9$; *2* — 0.8; *3* — 0.6; *4* — 0.4; *5* — 0.2, and *6* — 0.1

$$I_w(t) = \int_0^t \left(\sum_{i=1}^{N} F_\tau \right) dt$$

where N is the number of blades on the rotor. Figure 4 shows the calculated dependencies $I_w(t)$ for 6 values of the blockage factor: $\delta = 0.1$, 0.2, 0.4, 0.6, 0.8, and 0.9. The momentum transferred to the rotor by detonation products during a single operation cycle is seen to be maximal at $\delta = 0.4$ and 0.8 and reaches a value of 0.09 kg·m/s. When the blockage factor is large ($\delta = 0.9$), the momentum transferred to the rotor reaches very quickly the value of 0.08 kg·m/s when SW-I is reflected from the first blade, and then decreases and levels out at 0.05–0.06 kg·m/s. When the blockage factor is small ($\delta = 0.1$), the momentum transferred to the rotor gradually reaches a value of 0.04 kg·m/s.

It is interesting to compare the momentum transferred to the rotor by the detonation products and the products of combustion of the same fuel–air mixture at constant volume ($V = const$) and at constant pressure ($P = const$). For such a comparison, calculations were made in which a combustible mixture in a pulse detonation tube was burned at a constant volume and in a laminar flame propagating

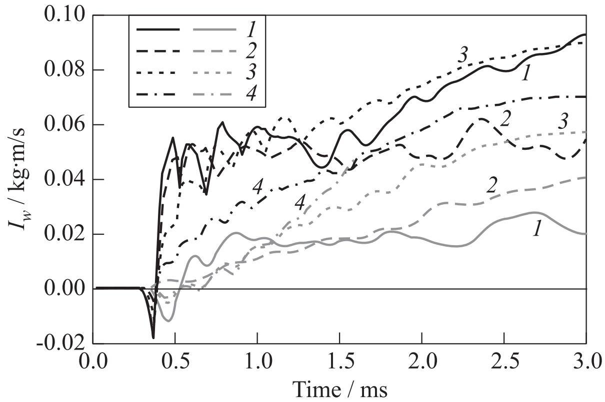


Figure 5 Comparison of the calculated temporal dependencies of the momentum transferred to the rotor from the detonation products (black curves) and from the products of combustion at $V = const$ (grey curves) for different values of the blockage factor $\delta = 0.8$ (*1*), 0.6 (*2*), 0.4 (*3*), and 0.2 (*4*)

from the filling edge to the closed (left) end of the tube. Figure 5 presents the results of calculations for $\delta = 0.2$, 0.4, 0.6, and 0.8. The black curves reproduce the corresponding curves in Fig. 4. The grey curves refer to the case $V = const$. The curves for the case $P = const$ virtually merge with the zero line, i. e., the momentum transferred to the rotor in this case is negligible. As expected, the detonation products transfer more momentum to the rotor than the combustion products at $V = const$. The maximum difference in the results (exceeding a factor of 3) is observed for $\delta = 0.8$.

Let us define the coefficient of momentum utilization in one operation cycle, k_i, as the ratio of the momentum transferred to the rotor to the momentum of the detonation products at the inlet to the turbine. Dividing the value of 0.09 kg·m/s by 0.57 kg·m/s, one obtains that in this example, no more than 16% of the momentum of the detonation products is transferred to the rotor during the operation cycle, i. e., $k_i < 0.16$. The low efficiency of the momentum transfer is obviously associated with the dissipation of the kinetic energy of the detonation products mainly on the rotor blades.

Now, let us estimate the torque arising on the rotor blades. The instantaneous total torque M acting on the turbine rotor can be determined by the relation

$$M = \sum_{i=1}^{N} \left| \left[\vec{F} \times \vec{r} \right]_i \right|$$

where $\vec{F}$ is the instantaneous force acting on the rotor blade:

$$\vec{F} = \int_{S_w} P_w \vec{n} \, ds\,.$$

The torque is considered positive if it is directed clockwise. Given the instantaneous torque, one can determine the time-averaged torque using the formula:

$$\overline{M} = \frac{1}{t} \int_0^t M \, dt\,.$$

Figure 6 shows the calculated dependencies $M(t)$ (curve *1*) and $\overline{M}(t)$ (curve *2*) for a turbine rotor with $\delta = 0.4$ during one operation

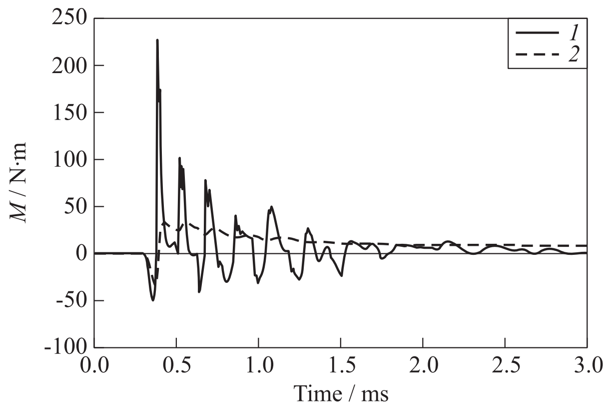


Figure 6 An example of calculation of the instantaneous (*1*) and time-averaged (*2*) torque acting on the rotor of a pulse turbine in one operation cycle of a pulse detonation tube ($\delta = 0.4$)

cycle. The curve $M(t)$ shows oscillations caused by reflections of SW-I and SW-II from the rotor blades. Since SW-I and SW-II contribute to the torque in different directions, the total torque alternately takes positive and negative values. Nevertheless, the time-averaged torque is always positive except for a short initial period associated with the advanced reflection of SW-II from the nearest blade.

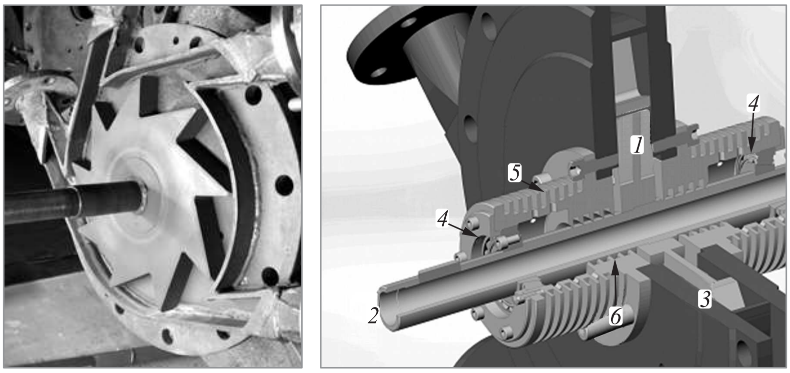
The oscillations of the torque are damped with time, which is caused by the weakening of the SWs as they move along the annular gap with obstacles in the form of rotor blades. The curves in Fig. 6 indicate that the total positive torque is created as a result of the first three or four reflections of SW-I from the rotor blades. All other reflections of SW-I are compensated by reflections of SW-II. The time-averaged torque is 10–30 N·m and acts on the rotor during the entire operation cycle, although the maximum instantaneous torque is considerably larger (220 N·m).

Another important implication of the computational results is that the flow oscillations at the inlet to the exhaust duct of the pulse turbine are getting small. This means that the thermal energy of the detonation products could be further utilized by applying conventional gas and/or steam turbines.

Experiment

To verify the obtained estimates, we developed an experimental setup with a pulse turbine (Fig. 7).

The turbine consists of a cylindrical leaky housing with an enclosed rotor on the shaft. The rotor is a thermally and mechanically durable monolithic steel disk 30 mm thick and 315 mm in diameter with 9 blades in the form of triangular teeth partially blocking the annular gap between the housing and the rotor. The walls of the housing are cut from steel sheets with a thickness of 10 mm and are joined by electric welding. The rotor shaft is fixed to the housing by means of two tapered bearings pressed into the holder with labyrinth seals. The holder is provided with cooling ribs. Weight of the rotor is 11 kg. The blockage factor δ of the annular gap by the rotor blades is 0.8 (the side gaps between the rotor and the housing are not taken into account).



(*a*) (*b*)

Figure 7 Experimental sample of a pulse turbine: (*a*) photograph, (*b*) mount of the rotor; *1* — rotor with blades; *2* — shaft; *3* — housing; *4* — bearings; *5* — holders with cooling fins; and *6* — seals

A pulse detonation tube with a diameter of 50 mm and a length of 2.5 m as well as an exhaust duct are attached to the turbine through the flanges. There are three ports for attaching the exhaust duct (Fig. 8). The pulse detonation tube has been used by the present authors earlier in other studies and its characteristics are reported elsewhere (see, e. g., [17]). It uses a mixture of hydrocarbons mainly consisting of propane and butane (liquified propane gas, LPG) as a fuel and air as an oxidizer. The heat of combustion of the fuel is approximately 46 MJ/kg. Fuel con-

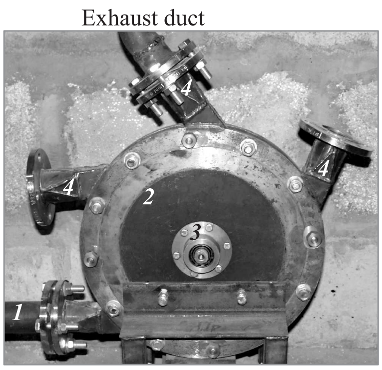


Figure 8 Photograph of a pulse turbine connected to a pulse detonation tube and exhaust duct: *1* — pulse detonation tube; *2* — turbine housing; *3* — turbine shaft; and *4* — exhaust ports

sumption in the pulse detonation tube was determined in two ways: based on the characteristics of the fuel injector at a given pressure and based on the level of liquid fuel in the fuel tank. At a detonation pulse frequency of 10 Hz, the fuel consumption was $\sim$ 3 g/s. A simple recalculation gives an estimate for the thermal power, W, of the pulse detonation tube when operating at a frequency of 10 Hz: $W \approx 140$ kW.

In experiments with a pulsed turbine, we measured the rotor speed using a noncontact Hall sensor mounted on the turbine shaft. The signal of the Hall sensor was registered with a tachometer TX-01. The tachometer readings were recorded using a videocamera with further digitization of the readings. Figure 9 shows an example of the temporal dependence of the measured rotation speed of the pulse-turbine rotor at a detonation pulse frequency of 10 Hz.

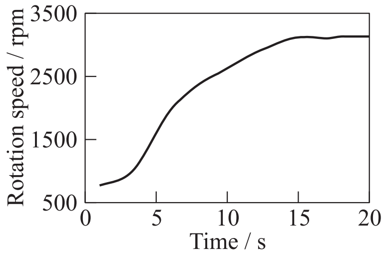


Figure 9 Temporal dependence of the rotation speed of a pulse-turbine rotor at a detonation pulse frequency of 10 Hz

Before the experiment, the exhaust duct was connected to one of the three exhaust ports of the turbine. The remaining exhaust ports were closed with a plug. The experiment began with blowing the pulse detonation tube with air and boosting the turbine to the rotation speed of $\sim$ 500 rpm. Then, the control automatics was turned on and the pulse detonation tube began to operate in the preset mode with a fixed frequency of detonation pulses. At a detonation pulse frequency of 12 Hz, the rotation speed of the turbine reached $\sim$ 4000 rpm. A further increase in speed was prevented by losses due to friction in the bearings, leakage of the turbine housing, as well as unoptimized mass and dimensions of the rotor and the shape of the blades.

The time of turbine boosting from 1000 to 3000 rpm in Fig. 9 is $\approx$ 10 s. This allows one to estimate the mechanical power extracted by the turbine. Thus, the kinetic energy of rotation of the rotor, E_k, can be estimated from the formula:

$$E_k = \frac{I_z \omega^2}{2}$$

where $I_z = mR^2/2$ is the moment of inertia of a solid cylindrical body with R being the radius of the body and m being its mass; and ω is the angular velocity of rotation in radians. At rotation speeds of 1000 and 4000 rpm (104 and 418 rad/s, respectively), the kinetic energy of rotational motion of a rotor with a radius of 0.16 m and a mass of 11 kg is 3.1 and 12.3 kJ, respectively. The turbine mechanical power is thus ~ 1 kW.

Let us define the coefficient of utilization of the kinetic energy of detonation products in one operation cycle, k_e, as the ratio of the turbine mechanical power in one cycle to the change in the kinetic energy of the detonation products at the turbine inlet during the cycle time. Dividing the mechanical power of the turbine (~ 1 kW) by the thermal power of the pulse detonation tube (~ 140 kW), one may conclude that only about 0.7% of the thermal power of the pulse detonation tube is extracted by the experimental sample of a pulse turbine. The ratio of the kinetic energy of the detonation products in the fixed coordinate system to their enthalpy at the Chapman–Jouguet point is approximately $(\gamma - 1)/(2\gamma^2) \approx 0.09$ at a ratio of specific heats $\gamma \approx 1.3$. In reality, no more than $0.007/0.09 \approx 8\%$ of the kinetic energy of the detonation products is utilized in the pulse turbine. By other words, the coefficient of utilization of the kinetic energy of detonation products in one operation cycle is $k_e < 0.08$. Note that the utilization of all kinetic energy of detonation products could potentially increase the Zel'dovich cycle efficiency by $\sim 9\%$.

Intuitively, the coefficients of utilization of the momentum, k_i, and the kinetic energy, k_e, should be of the same order. The calculated value of k_i ($k_i < 0.16$) and the experimental estimate of k_e ($k_e < 0.08$) are quite close to each other, indeed. The fact that the obtained values of k_i and k_e are much less than unity means that the pulse turbines of the water-wheel type considered herein are imperfect. To increase the efficiency of a pulse turbine, it is necessary to exclude the formation of SW-II, to select the optimum mass and dimensions of the rotor for reducing dissipation of kinetic energy, to select the optimum number of blades, to profile the blades and choose their optimum angle of attack, as well as to minimize the size of the lateral gap between the rotor and the housing.

Concluding Remarks

On the basis of calculated and experimental studies of the processes occurring during the implementation of the Zel'dovich thermodynamic cycle with pulse detonation combustion of fuel, the efficiency of utilization of the kinetic energy of detonation products in a pulse turbine of the simplest water-wheel type was estimated. Several configurations of the pulse turbine were considered with different values of the blade blockage factor. The following results are worth mentioning:

(1) the coefficients of utilization of the momentum and kinetic energy of the detonation products in such a pulse turbine with unoptimized mass and dimensions were shown to be less than 8%–16%. In all cases, a positive time-averaged torque on the turbine rotor was obtained;

(2) calculations reveal significant differences in the pressure fields near turbine blades at high rotation speeds and, therefore, in the predicted torque when the effect of rotor rotation is considered;

(3) there exists the optimal value of the blade blockage factor of the turbine rotor providing the maximum torque;

(4) the torque obtained in a single cycle is created as a result of several reflections of the lead shock wave from the rotor blades. All subsequent reflections do not contribute much to the torque;

(5) flow oscillations in the exhaust duct of the pulse turbine are small due to weakening of the lead shock wave in the turbine path. This means that the thermal energy of the detonation products could be further utilized by applying conventional gas and/or steam turbines downstream of the exhaust duct;

(6) to improve the efficiency of the pulse turbine, it is necessary to take measures for eliminating unnecessary reflections of shock waves, selecting the optimal mass and dimensions of the rotor and the number of blades, profiling the blades and selecting their optimal angle of attack, optimizing the size of the lateral gap between the rotor and the housing, and selecting the optimum location of the exhaust duct; and

(7) once the efficiency of the pulse turbine is improved, one could consider the possibility of realizing a three-stage combined thermodynamic cycle with the utilization of the kinetic energy in a pulse turbine, followed by the utilization of the thermal energy of hot detonation products in a conventional axial or centrifugal turbine and in a steam turbine. It is expected that the efficiency of such a combined cycle could be higher than the efficiency of a conventional gas–steam hybrid cycle by at most 9%.

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