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# Active Control of Twin-Pulse Combustors

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**The pressure losses of steady-flow combustors can be overcome with so-called pressure gain or pulse combustors. Higher combustion efficiency and low $NO_x$ are the well-known advantages that outweigh the disadvantages of noise, combustor control, and lower operational flexibility. Inlets with variable cross section enable an active control of the combustor as gas generator for turbines. Therefore, a special emphasis is given to the design of the inlet, its control capacity, and its impact on combustor and turbine performance.**

## Nomenclature

| | |
|---|---|
| $A$ | = area |
| $c$ | = speed of sound |
| $D$ | = damping |
| $F$ | = transfer function |
| $f$ | = frequency |
| $H$ | = specific enthalpy |
| $K$ | = constant |
| $L_T$ | = length of the tailpipe |
| $\dot{m}$ | = flow rate |
| $n$ | = rotational speed |
| $P$ | = pressure |
| $P_\infty$ | = ambient pressure |
| $p$ | = mean pressure |
| $q$ | = heat |
| $R$ | = gas constant |
| $S$ | = entropy |
| $s$ | = Laplace operator |
| $T$ | = temperature |
| $T_w$ | = lag time |
| $t$ | = time |
| $V$ | = volume |
| $v$ | = velocity |
| $x$ | = coordinates |
| $\gamma$ | = ratio of specific heats |
| $\Delta P$ | = half of pressure amplitude |
| $\rho$ | = density |
| $\omega$ | = circular frequency |

*Subscripts*

| | |
|---|---|
| $A$ | = air |
| $c$ | = combustion chamber |
| $E$ | = evacuation |
| $f$ | = fuel |
| in | = inflow |
| $K$ | = compression |
| $N$ | = nozzle exit |
| out | = outflow |
| pr | = pressure |
| $SP$ | = settling chamber |
| trg | = trigger |
| 0 | = total |
| $\infty$ | = ambient |

## Introduction

BECAUSE of the limited capacity of natural resources and the necessity to protect the atmosphere against $NO_x$ pollutions,[1] the development of advanced combustors for power plants or jet engines with higher thermal efficiencies has gained a renewed interest in the current research. Steady-flow combustors based on the constant pressure cycle generate pressure losses due to heat transfer and friction, thus reducing the total efficiency of power plant cycles. Pulse combustion provides a certain capacity to overcome these losses and moreover offers a pressure gain caused by compression waves.[2] Karavodine (see Ref. 3) was the first who combined such a combustor with a turbine. Even the turbine could be operated with a single jet; the first attempts were done to enhance the performance by using two combustors in parallel with a common nozzle and independent or combined inlets. Such developments were not continued, and most of the actual proposals for the design of pulse combustors originate from that time. Theoretical studies on the overall performance of twin combustors by Reynst[4] have shown that the efficiency can be enhanced by 10%, but it could not yet be verified experimentally. Contemporary twin burners were investigated by Saito et al.,[5] who reported strong pressure instabilities during the engine run caused by the inlet. Kentfield,[6] in his notable research work, propelled turbines with conventional aerovalved pulse combustors equipped with a rectifier and modified the system later to an ejector-type combustor. The operation of the turbine with the unsteady flow ejector-type thrust augmenter led to a total pressure gain of about 4%, and his newer studies have shown that up to 7% is possible for large size units. The pumping capacity of twin combustors investigated by Kentfield and Read[7] was found to be lower than the one of conventional jet systems because a high portion of the energy remains within the combustor system.

Only a small amount of research work deals with combined pulse combustor-turbine systems. Therefore, a survey of the system performance and of the capacity of an inlet with variable cross section for an active control is presented here. The combustor performance

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depends on the compression capacity of the gas column in the tailpipe. The compression due to the acceleration of the gas column is controlled with the inlet valve. Cause–effect relations governed by differential equations were derived from experiments and theoretical approaches proposed by Barr et al.[8] In contrast to the standard methods using characteristic cycles as presented in the literature, here the off-design operation of the combustor and the behavior of reacting and nonreacting flows were analyzed to detect the influence of several design parameters. Several rough simplifications of the thermodynamic and fluid mechanical processes are necessary to enable a design of the control system.

## Principles of the Twin-Pulse Combustor

For detailed descriptions of the conventional pulse combustor operation in contemporary literature, the reader is referred to Refs. 9 and 10 because the basic behavior of each individual combustor was found to be similar to that of conventional pulse combustors. Here, only a brief summary of the principles will be given because that is essential for the understanding of the twin-pulse combustors. A twin-pulse combustor as gas generator for a turbine has the advantage of a high combustion efficiency and the supply of a constant pressure gas to the turbine. Although the influence of unsteady flow on turbine efficiency investigated by Porter[11] was found to be likely negligible, the high mechanical load to the turbine blade rather than the efficiency loss is the reason that a steady flow should be supplied to the turbine. A twin combustor with its alternating combustion is the simplest method to compensate for the pressure fluctuations in the settling chamber. The incorporation of a convergent nozzle prevents disturbances propagating to the turbine.

A review of the overall performance of the experimentally investigated systems showed that a pressure gain of 20 kPa is obtained for combustors with reed valves and 5–10 kPa for aerovalved systems. Considering the efforts made throughout this century, it is doubtful whether a further enhancement of the performance can be achieved from conventional pulse combustors. Moreover, the operating range of self-aspirated pulse combustors is limited by the pressure difference between inlet and nozzle (pumping capacity). Therefore, the tasks of an active control system are to provide a stable engine run and to extend the operating range without bigger changes in the geometry of the combustor. Noteworthy higher operational flexibility of the combustor can be obtained with a variable inlet cross section. Two overlapped porous plates shown in Fig. 1 enhance the operational flexibility and allow for an on-line optimization of the inlet

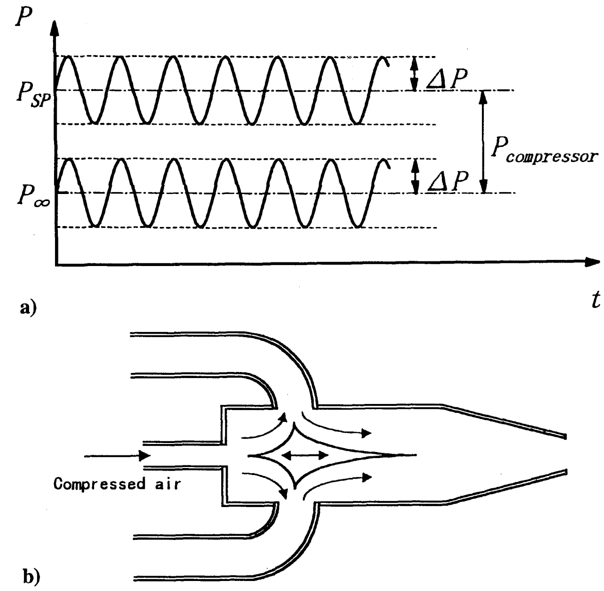



**Fig. 2 a) Operating range of self-aspirated and charged pulse combustors and b) proposed charging system for the settling chamber.**

geometry of the running combustor. The basic idea of this concept is that the maximum compression of the mixture in the combustor is done by the compression wave in the tailpipe whose strength depends on the tailpipe length and the inlet cross section. The valving process influences not only the pressure and wave characteristics due to the charged mass flow but also the time characteristics of the cycle. This impact is used to smooth the pressure in the settling chamber at the combustor exit and to adjust the nozzle and inlet pressure conditions. If the stagnation pressure in front of the intake becomes too high or the ambient pressure at the pipe exit drops too much, a steady flow will be established. To overcome these effects, Diedrich[12] proposed shrouded versions, thus creating an artificial higher ambient pressure at the nozzle exit. Schultz-Grunow[13] invented a cap in front of the inlet to reduce the flow rate into the chamber. The pulse combustion can also be sustained by the influence of an increasing inlet pressure. Choking the nozzle flow or feeding bypass gas into the settling chamber creates a higher back pressure. The pressure oscillation is set on a higher potential, as schematically shown in Fig. 2.

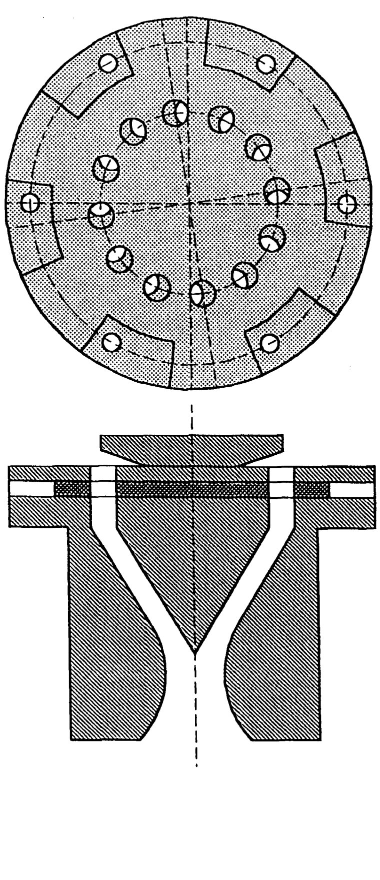


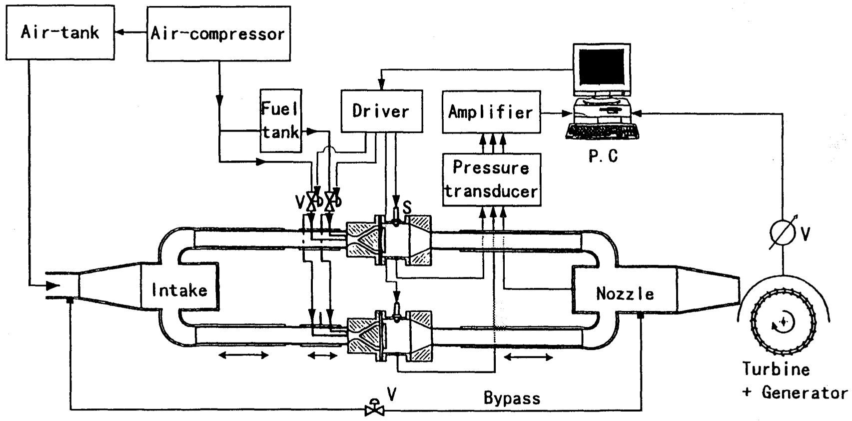
**Fig. 1 Inlet with overlapped porous plates for airflow regulation.**

## Experimental Setup

A theory for conventional twin-pulse combustors is not yet available, so that the development is done using trial and error. Therefore, it was an emphasis of the design of the setup shown in Fig. 3 to make possible a simple parameter variation of the geometry.

The inlet pipe holds the ejector whose position could be varied within 25 mm relative to the throat of the Venturi-type inlet for an adjustment of the mixer geometry. Larger distances between ejector and inlet throat are favorable for the mixing, whereas the combustor start capability is improved if the ejector is positioned close to the inlet throat. The throat cross section could be varied with the porous plates to adjust the total flow rates to the cycle demands. The achieved improvement of the combustor performance can be determined from the combustion pressure and the rotational speed of the turbine providing an additional measure for the kinetic energy of the flue gas at the combustor exit. The turbine propels a small generator, and the revolutions per minute can be derived from the generated ac voltage signal. All modifications of the combustor will be compared with the reference case that is given with the steady flow condition measured in each test run for cold and hot flows as well.

To collect some data on the overall performance and to trace the energy transfer from the chemical energy to the mechanical energy of the turbine, characteristic flow rates, pressures, and temperatures were measured. Because of the heavy heat load, the transducers could not be positioned in the hot flow, and so only the static pressures acting at the walls were measured. The pressures




I:Injector S:Spark plug V:Valve

**Fig. 3 Experimental setup.**

in the combustion chambers and settling chamber were measured with water-cooled piezo pressure transducers that have an accuracy of 2%, depending on temperature effects. The mean pressure was measured with a U-type manometer to confirm the pressure gain.

## Results and Discussion

### Operation of Twin-Pulse Combustors

Numerous parameters and their interactions complicate a comparison of the characteristics of steady and unsteady, reacting and nonreacting flows produced in single or twin combustors. The complex experimental data can be reduced, with the turbine speed and fuel consumption giving a survey of the energy transfer. The pressure histories for both combustion chambers with a phase difference $\pi$ indicate an accurate operation of the combustor. Only during the start, or if the fuel rates are too different, is the operation unstable. Although the higher amplitude and the mean pressure of combustors with reed valves are favorable, heat and mechanical load often lead to a valve destruction, and so aerovalved systems were preferred.

Characteristic pressure histories depending on the nozzle are shown in Fig. 4a for a straight nozzle with $A_N/A_{SP} = 1$ and in Fig. 4b for a convergent nozzle $A_N/A_{SP} = 0.25$. The amplitude of the pressure in the combustor remains almost constant, but the absolute amount of the pressure will be obviously reduced from 91–114 kPa for $A_N/A_{SP} = 1$ to 91–109 kPa for $A_N/A_{SP} = 0.25$. Because of the acceleration of the gas in a convergent nozzle with an exit diameter of 40 mm, which is still larger than the pipe diameter of 35 mm, the pressure drops in the settling and the combustion chamber as well. The pressure in the settling chamber labeled with $SP$ in Fig. 4 has a maximum amplitude of about 4 kPa and a phase difference $\pi/2$ for $A_N/A_{SP} = 1$, whereas the amplitude is reduced to 1 kPa for $A_N/A_{SP} = 0.25$. These pressure fluctuations shown in Fig. 4c could be reduced with a redesigned version of the twin-pulse combustor. The diameter ratio of the tailpipe and combustor was changed from 0.45 to 0.625.

Some overall performance data are summarized in Table 1. The mean pressure shown here is defined with $p = 1/t \int P \, \mathrm{d}t$. The performance improvement is amazingly high but in accordance with measurements made by Kentfield,[6] who found a performance increase of about 40% in comparison to the steady-flow system. In the actual experiments, it was not possible to operate the combustor with the same fuel rate alternatively in steady or pulsed modes. A steady combustion could be sustained only if the mixture was enriched.

An onset of pressure fluctuations can be obeyed if the combustor is operated at off-design conditions or if any irregularities (labeled with IR in Fig. 4b) due to mixing or ignition occur. The nonfrictional portion of the damping capacity of the settling chamber equipped with a convergent nozzle scales with the characteristic value $A_N/V_{SP}$. The first law of thermodynamics for unsteady flows can be rewritten with

$$\frac{m(t)\,\mathrm{d}T_c(t)}{\gamma T_c(t)} + \frac{\mathrm{d}m(t)}{\gamma} + \mathrm{d}m_{\mathrm{out}} - \frac{T_{\mathrm{in}}(t)}{T_c(t)}\,\mathrm{d}m_{\mathrm{in}}(t) = 0 \qquad (1)$$

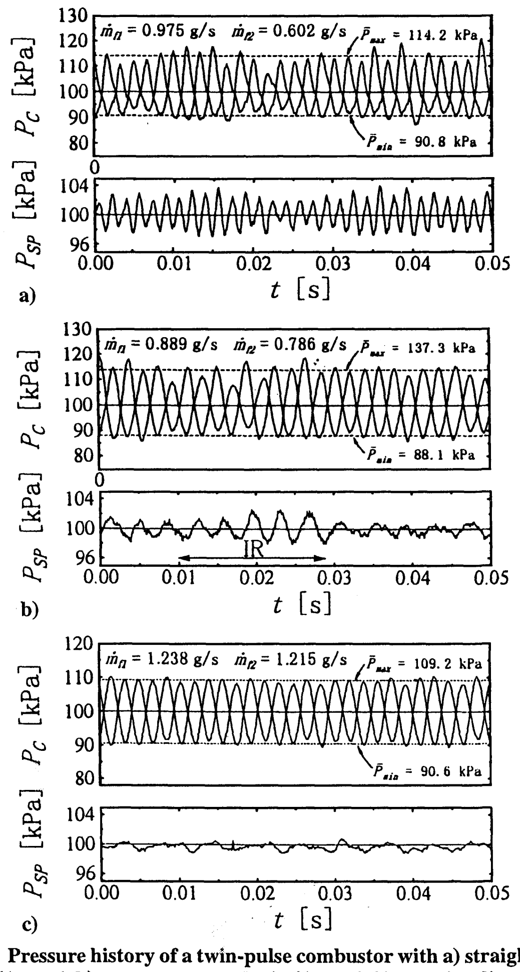



**Fig. 4 Pressure history of a twin-pulse combustor with a) straight nozzle $A_N/A_{SP} = 1$, b) convergent nozzle $A_N/A_{SP} = 0.25$ and pipe diameter of 35 mm (IR: irregularities in combustor operation), and c) pipe diameter of 50 mm.**

and assuming perfect gas, the pressure time derivative is obtained from

$$\frac{\mathrm{d}p(t)}{\mathrm{d}t} = \frac{\gamma R}{V}\left\{T_{\mathrm{in}}(t)\frac{\mathrm{d}[m(t)]_{\mathrm{in}}}{\mathrm{d}t} - T_c(t)\frac{\mathrm{d}[m(t)]_{\mathrm{out}}}{\mathrm{d}t}\right\} \qquad (2)$$

The residual air mass in the plenum and the pressure can be calculated with the mass flow equation and (+ denotes filling and − denotes emptying)

$$\frac{m(t)}{m(t=0)} = \frac{m(t=0) \pm \int \dot{m}(t)\,\mathrm{d}t}{m(t=0)} = \left[\frac{T(t_0)}{T(t_0)}\right]^{1/(\kappa-1)} = \left[\frac{P(t)}{P(t=0)}\right]^{1/\kappa} \qquad (3)$$

**Table 1 Performance of the twin combustor**

| Mode | $A_N/A_{SP}$ | $n$, rpm | $\dot{m}_{F1}$, g/s | $\dot{m}_{F2}$, g/s | $p1$, 100 kPa | $p2$, 100 kPa | $SP$, 100 kPa |
|---|---|---|---|---|---|---|---|
| Steady | 1 | 640 | 1.327 | 1.072 | 0.993 | 1.039 | 0.972 |
| Pulsed | 1 | 880 | 0.930 | 0.601 | 0.988 | 1.026 | 0.966 |
| Steady | 0.25 | 1225 | 1.439 | 1.283 | 0.984 | 1.021 | 0.989 |
| Pulsed | 0.25 | 1621 | 0.621 | 0.736 | 1.039 | 1.003 | 0.965 |
| Pulsed | 0.25 | 1970 | 0.748 | 0.784 | 0.956 | 0.999 | 0.990 |
| Pulsed | 0.25 | 2076 | 0.887 | 0.745 | 0.990 | 1.018 | 0.993 |

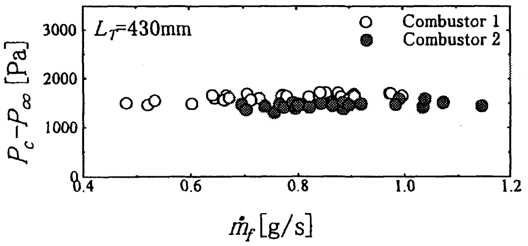



**Fig. 5 Pressure gain of a twin-pulse combustor.**

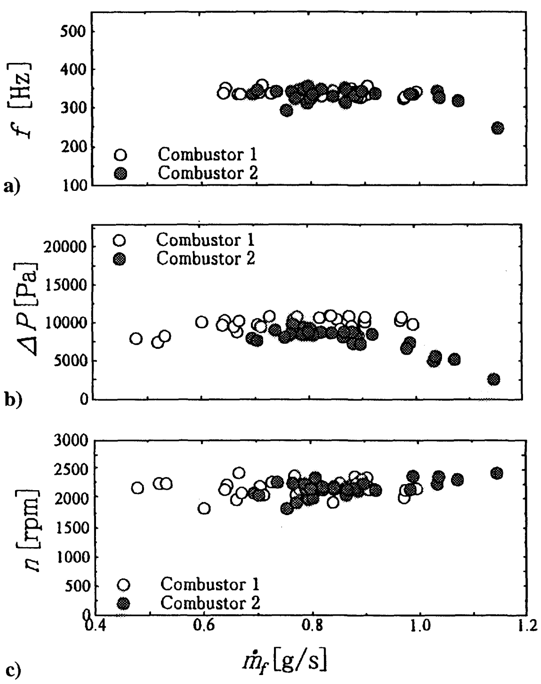



**Fig. 6 a) Frequency, b) pressure amplitude, and c) rotational speed of a twin combustor, depending on fuel rate.**

The obtainable pressure gain shown in Fig. 5 of the twin-pulse combustor amounts to about 1.5 kPa and is almost independent of the tailpipe length, varied between 380 and 630 mm. The total performance of the twin-pulse combustor remains higher than that of steady-flow combustors. Tanaka and Shimamoto[14] found that the pressure oscillations within the flow are responsible for the minor transfer of chemical energy to dissipative energy. A portion of $(\kappa - 1)/\kappa$ remains in the form of mechanical energy within the flow due to its unsteadiness. This energy can only be stored up in unsteady flows in the form of pressure waves and can be governed with the Rayleigh equation

$$\frac{\mathrm{d}H_0}{\mathrm{d}t} = \frac{1}{\rho}\frac{\delta p}{\delta t} + T\frac{\mathrm{d}S}{\mathrm{d}t} = \frac{1}{\rho}\frac{\delta p}{\delta t} + \frac{\delta q}{\delta t} + V\frac{\delta q}{\delta x} \tag{4}$$

The one-dimensional analysis of Tanaka and Shimamoto[14] showed that for $\kappa = 1.33$ about 25% of the chemical energy is used to establish the wave system and to overcome the damping losses.

Characteristic amplitudes, frequencies, and rotational speeds shown in Fig. 6 weakly depend on the fuel rate, in contrast to the strong parabolic characteristic of the conventional combustor.

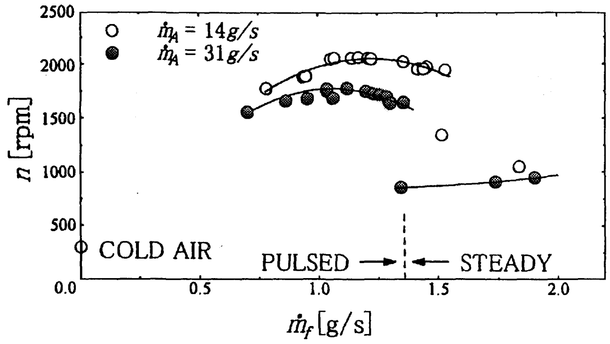



**Fig. 7 Rotational speed depending on operating mode and fuel rate.**

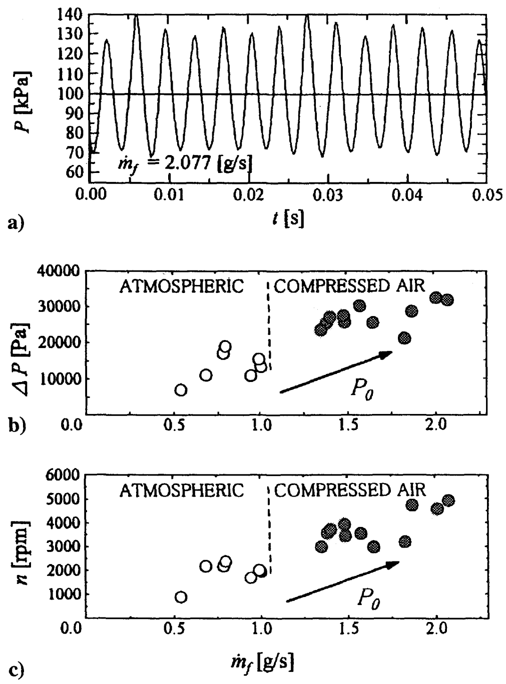



**Fig. 8 a) Pressure history of a charged combustor, b) pressure amplitude, and c) rotational speed depending on fuel rate and feeding pressure.**

Maximum rotational speeds can be obtained with an equivalence ratio as lean as possible. The total fuel rate amounts to about $\dot{m}_f = 1.2$ g/s, which is comparable to the rate of a conventional combustor.

The performance of a conventional pulse combustor, depending on the fuel rate, is depicted in Fig. 7. A turbine speed of 300 rpm is generated by the ejector air and raised by a steady-flow combustion to about 1000 rpm. In agreement with the results of Kentfield,[6] maximum speeds of about 2000 rpm are obtained for pulse combustion, although the fuel rate is reduced. An enrichment of the mixture causes a steady flow with lower turbine speeds.

The charge of compressed air into the inlet settling chamber enhances the combustor performance. A characteristic pressure history is exemplified in Fig. 8a. The pressure amplitude increases from 10 to about 30 kPa (Fig. 8b) and the rotational speed from 2400 to 4900 rpm (Fig. 8c) if the charging pressure increases to 110 kPa. For higher stagnation pressures, a pulse combustion can be sustained only with a smaller inlet cross section to keep the airflow rate into the combustor constant. If the inlet stagnation pressure or the fuel rate exceeds its maximum, the flow changes to a steady one with lower rotational speed.

### Coupling of Fluid Mechanics and Combustion

The phenomena occurring in a pulse jet are usually handled with the method of characteristics. Considering the advances that the automotive engine development has made because of the employment of electronically operated fuel ejector and control systems, the objective of the theoretical approach as presented here is to make possible a design of such controllers for pulse combustors. Therefore, it is necessary to comprise the complicated interaction of mixing, combustion, and ignition in differential equations that are as simple as possible. It led to the idea of implementing thermodynamics and fluid dynamics in the control theory. This theoretical approach is a continuation of the excellent work done by Barr et al.[8] and will consist of three parts, with the control theory as the main frame. The transport phenomena are no longer negligible for pulse combustors and must be included in the theory with lag times. The equations are transformed from the time to the frequency domain with the Laplace transformation, and the corresponding transfer function $F$ expressing the wave propagation can be written as

$$F = e^{-T_w s} K \qquad \text{with} \qquad T_w = L/C \tag{5}$$

The characteristic parameters for the modeling can be obtained from a combustor operated under off-design conditions. The actual simulation is still done in a so-called open-loop operation to obtain the influence of individual parameters on the combustor run and will later be accomplished by a closed-loop simulation as illustrated in Fig. 9. The ignition can be simulated with an impulsive signal under ideal conditions. Any irregularities, as, for example, those shown in Fig. 4b, are described with a probability, thus including the varying mixing and ignition effects. The combustor itself is controlled with the inlet valve.

**Valve Design**

Kentfield[6] reports the importance of a moderate back-pressure ratio ($P_\infty/P_c$) to guarantee an efficient compression by the subsequent wave. The control of the twin combustor with the inlet valve and pipe system has a strong impact on the compression and the combustor performance. Highest efficiency is obtained from a combustor running under design condition. A control system enhances the flexibility but also reduces this efficiency, which is discussed in the following. The pressure gain obtained for several inlet tailpipe systems as they are incorporated in a twin-pulse combustor is shown in Fig. 10 and amounts to about 1–3%, depending on the fuel rate and

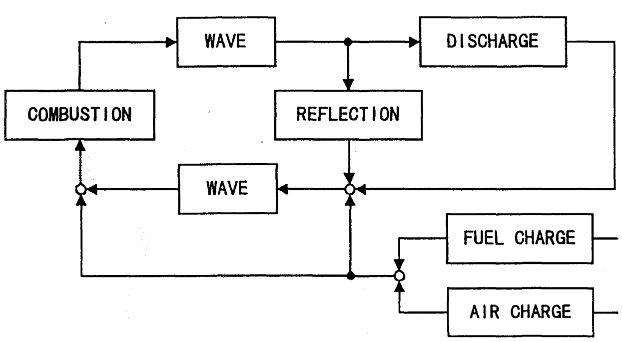



**Fig. 9 Simplified scheme of the pulse jet operation in the control theory.**

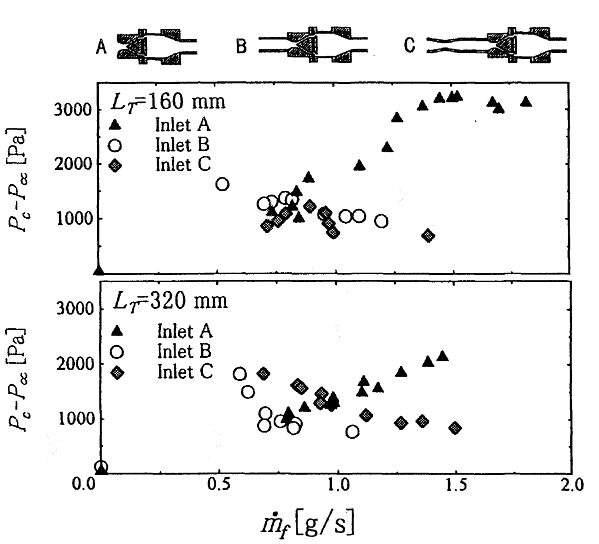



**Fig. 10 Pressure gain depending on fuel rate, inlet, and tailpipe length.**

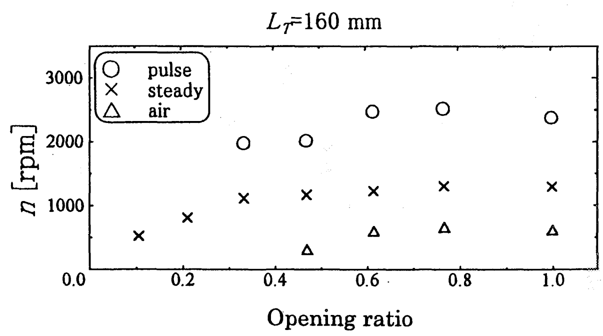



**Fig. 11 Rotational speed depending on inlet opening ratio.**

the tailpipe length. In general, any inlet pipes reduce the obtainable absolute maximum pressure gain by about 1%.

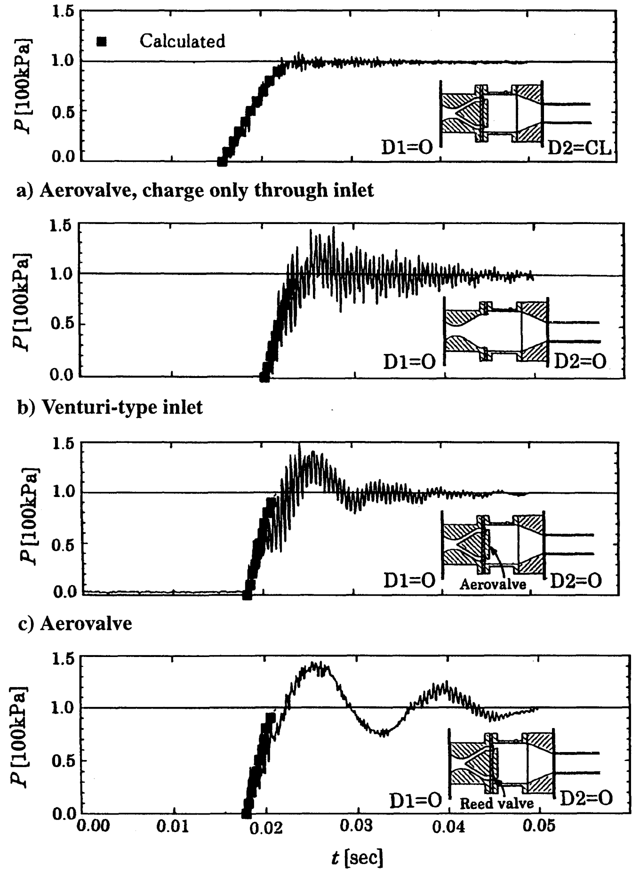
The combustor operation and the effectiveness depend on the inlet throat geometry. Using a Venturi-type inlet without inlet cone, the resistance is too low to prevent efficiently a gas outflow through the inlet. That might be the reason for the unstable combustor operation reported by Saito et al.[5] because the emitted waves interact with the charging of the other combustor. Our experiments using these inlets have shown that, in general, both combustion chambers are working in phase. An alternating combustion could be established only if the charging process is attenuated by choking the flow rate. The attenuation can be optimized with the overlapped porous plates shown in Fig. 1. The impact of the inlet cross section on the turbine speed is presented in Fig. 11 for nonreacting and reacting flows in combustors with different tailpipe lengths. The maximum speed is obtained for an inlet opening ratio of 0.75.

To show the impact of the individual waves on the total compression and to obtain the discharge coefficients for inlet and tailpipe, the combustor was equipped with diaphragms at the inlet and exit. The pipe diaphragm is attached to a needle, so that a small pressure increase after the rupture of the inlet diaphragm is sufficient to break the tailpipe diaphragm immediately. The acceleration of the gas is too low for a compression by inertia if the combustor is charged only through the inlet (Fig. 12a). The pressure and the flow rate history can be calculated with the continuity equations for quasisteady flows under these conditions. The discharge coefficient was determined for the sonic state and kept constant throughout the charge process. The maximum compression capacity depending on the ratio of the total pressure $P_0$ to the ambient pressure $P_\infty$ for a given combustor geometry can be obtained if the combustion chamber is fully evacuated to vacuum. The initial chamber pressure was for all test cases $P_c(t=0) = 100$ Pa. Typical values for a conventional Venturi-type inlet with the lowest compression capacity amount to only $P_{max} = 1.1 P_\infty$ (Fig. 12b). The gas will be discharged too fast through the inlet because the inertia of the gas in the tailpipe is much higher than that in the inlet. Inlets with a cone produce peak pressures of $P_{max} = 1.25 P_\infty$ (Fig. 12c) and those equipped with a reed valve a maximum of about $P_{max} = 1.4 P_\infty$, as illustrated in Fig. 12d.

The compression capacity of the cold gas is almost identical to that of the hot gas for the combustor type under investigation here. The aerovalved engine delivers a pressure of about 1.1–1.2$P_\infty$ and the version with a reed valve about 1.5$P_\infty$. As an important result, the compression is substantially done by the acceleration of the gas column in the tailpipe, which depends on the minimum pressure of the previous cycle. This pressure increases if gas is charged to the combustor through the inlet, depending on the throat cross section.

This relationship can be seen in Fig. 13, where the impact of the reed valve on the time characteristics is shown in comparison with the impact of an aerovalve. The gas expands from the combustion chamber until the valve begins to open, so that new gas is fed into the combustor, raising the pressure. The inflow of gas through the inlet and the compression wave confine the obtainable minimum pressure to 88 kPa, whereas the maximum emptying capacity and minimum pressure of 82 kPa are obtained for a fully closed inlet.

A survey of the total performance of the compression and evacuation capacity is given in Fig. 14. The capacity depends on the initial pressure ratio $P_c/P_\infty$; for example, the combustor can be evacuated to about 75 kPa with a maximum initial pressure of about 210 kPa




a) Aerovalve, charge only through inlet

b) Venturi-type inlet

c) Aerovalve

d) Reed valve

**Fig. 12 Pressure rise and maximum compression capacity depending on inlet geometry: D1 and D2, diaphragm; O, opened; and Cl, closed.**

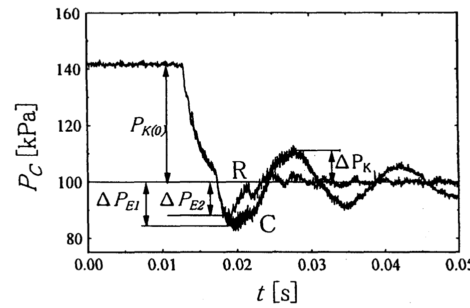



**Fig. 13 Pressure history of a combustor with closed inlet (C) and reed valve (R).**

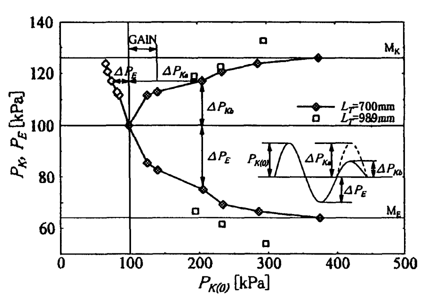



**Fig. 14 Charging performance of the combustor: index a, cold flow, and b, hot flow.**

that permits a subsequent compression of about 120 kPa for nonreacting flows. The pressure must be increased to the initial value by the combustion for a continuous operation. A realistic pressure range is between 90 and 110 kPa for aerovalved combustors, resulting in a pressure gain of about 3 kPa, in agreement with the measured mean pressures shown in Fig. 10. The high total pressure loss in the first period must be overcome with the energy taken from the combustion to regain the previous initial condition. The pressure in the combustion chamber is a measure for the nondissipative energy.

The dynamic behavior of the gas in the tailpipe can be obtained if this gas column is excited by a known event, as can be quite easily realized in experiments with a combustor operated under off-design conditions. A known event can be a pressure pulse or a single ignition (open-loop simulation). Although the first periods are always dominated by transport phenomena, the oscillation becomes harmonic from the second or third period. The characteristics can be described by a second-order differential equation for the Helmholtz condition with

$$\ddot{P}_c + \omega^2 P_c = 0 \tag{6}$$

and including the damping coefficients, the Helmholtz equation is extended to a more general form with

$$\ddot{P}_c + D\dot{P}_c + \omega^2 P_c = D_N \dot{P}_N + \omega_N^2 P_N \tag{7}$$

The Helmholtz theory does not consider damping and is based on the assumption of short throats and isentropic flows. The theory cannot be applied to pulse combustor flows because transport phenomena are neglected. The damping includes a frictional and a nonfrictional term determined by the combustor geometry $A_N/V_c$. Because of the expansion of the gas in the nozzle, a time derivative of the nozzle exit pressure must be considered in Eq. (8). The coefficients in this equation are constant for supersonic flows with the special cases of 1) steady flow with cross section change, $\ddot{P}_c$ and $D = D_N = 0$, and 2) quasisteady emptying or filling of a reservoir, $\ddot{P}_c$ and $D_N = 0$.

The flow within the combustor is subsonic; that means the nonconstant flow coefficients must be linearized or calculated numerically and the whole friction is comprised in a constant damping coefficient. The validity of such a strong simplification is shown in Fig. 15 for nonreacting flows and in Fig. 16 for reacting flows. Although the properties of the combustor can be calculated in fair agreement with a second-order differential equation, the start process due to the more complicated interaction between exciting waves and excited gas requires further efforts. This interaction, which can be seen from the pressure history in Fig. 15, is labeled with $P$. The concept of the open-loop simulation is illustrated at the top of Fig. 16, and the transfer function can be written as

$$F = F_{\mathrm{trg}} F_{\mathrm{pr}} = \frac{e^{-T_{\mathrm{trg}} s} K}{s^2 + Ds + \omega^2} \tag{8}$$

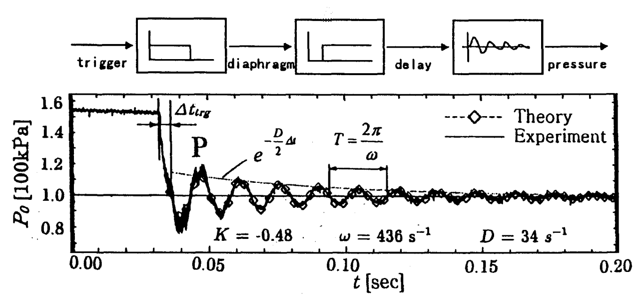



**Fig. 15 Pressure history for nonreacting flow, comparison with theory; P: impact of exciting wave on pressure history.**

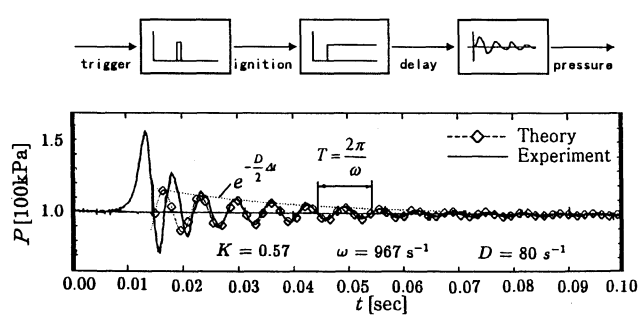



**Fig. 16 Pressure history for reacting flow, comparison with theory.**

The transfer function for reacting flows is

$$F = F_{\mathrm{trg}} F_{\mathrm{pr}} = \frac{e^{-T_{\mathrm{trg}} s} K s}{s^2 + Ds + \omega^2} \tag{9}$$

The pressure history for an exciting of the gas column by ignition is shown in Fig. 16. The expansion of the nonreacting flow can be simulated with a step function, but an impulsive function is employed for reacting flows, as illustrated at the top of Fig. 16. In contrast to the frequency, the damping kept constant in the calculation is only weakly affected by the temperature change in the tailpipe. That means it is possible to comprise friction effects in a constant damping coefficient. The discharge of the hot flue gases causes a temperature drop of 5 K/ms that must be considered in the closed-loop simulation in the cycle frequency.

## Conclusions

Conventional pulse combustors were set in parallel, and the flue gas was supplied to a turbine. The pressure balancing with a settling chamber equipped with a convergent nozzle could be successfully demonstrated. Even though the comparison of the efficiency of pulse combustors and steady-flow combustors is quite difficult to carry out due to the number of interactions, the superiority of the pulse combustion could be shown for all test cases. Fuel savings of more than 20% could be achieved in the test cases shown here. The combustor operation depends strongly on the compression that is substantially done by the gas column in the tailpipe. The task of the inlet valve is the active control of the strength and the timing of the gas in the tailpipe. The operational flexibility could be extended with an inlet consisting of two overlapped porous plates. Several characteristics of the dynamic behavior of pulse combustors and their control system could be detected by studying the off-design conditions.

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