ABSTRACT

A description is given of, what is believed to be the first test ever made of a gas turbine in which a valveless pulsed combustor replaced the conventional steady flow combustor. It is explained that the main incentive for using a pulsed combustor in a gas turbine is to achieve a net stagnation pressure gain between the compressor outlet and the turbine inlet. Brief descriptions are given of the pulsed combustor and the adaptation of the small gas turbine, which was of the gas generator type, to receive the pulsating combustion system. Results are presented which show that the gas turbine operated successfully using the pulsed combustor and that a very small net stagnation-pressure gain was achieved. An indication is given of possible future developments which should result in improved performance.

NOMENCLATURE

- $C_p$ = Specific heat at constant pressure
- $P$ = Mass flow
- $N$ = Gas turbine rotor speed
- $P_{amb}$ = Ambient pressure
- $P_o$ = Stagnation pressure
- $S$ = Entropy
- $T$ = Stagnation temperature
- $T_{amb}$ = Ambient temperature
- $T_{out}$ = Outflow stagnation temperature from combustor (= $T_3$ in gas turbine case)
- $X$ = Adjustable length duct sections
- $Y$ = Ratio of specific heats
- $\Delta P_{cp}$ = Combustor pressure loss (−ve) or pressure gain (+ve)
- $\eta$ = Isentropic efficiency
- $\theta$ = Combustor temperature ratio
- $\gamma$ = Cycle temperature ratio

Subscripts

1, 2, 3, 4 = Stations within the gas turbine (see Fig. 7)

C, T = Compressor, turbine

A = Air

f = Fuel

INTRODUCTION

The work described in this paper represents a logical continuation of earlier work [1, 2] dealing with the possibility of substituting a valveless pulsating combustor for a conventional steady-flow combustion system normally found in a gas turbine. The paramount incentive for using a pulsed combustor as a substitute for a steady flow combustor is to obtain the benefit of an increase in stagnation pressure, as distinct from the customary pressure loss, between the combustor inlet (i.e. compressor delivery) and combustor outlet (i.e. turbine inlet).

The application of an experimental, laboratory, valveless pulsed combustor to a small educational, gas generator, gas turbine constitutes the basis of the work reported here. So far as is known to the writers the present work represents the very first actual application of a pulsed combustor to a gas turbine as an alternative to a conventional steady flow combustor although this concept, in another form, appears to have been originated by Reynst [3]. The Reynst concept should not be confused with the yet earlier work of Holzworth [4], an attempt to achieve a constant volume cycle gas turbine. The results of a more recent
attempt to achieve a constant-volume cycle gas turbine
have been reported by Catchpole and Runacres [5].
The prime aim of the work described was to investi-
gate the compatibility of the valveless pulsating
combustor with the compressor-turbine unit and to
compare machine operating characteristics with both the
conventional steady flow combustor, supplied as an
integral component of the gas turbine, and the alterna-
tive pulsating combustor. The main concern was the
compatibility, or otherwise, of the pulsed combustor
with the turbo-machine rather than the converse. It is
well known, from turbo-charger work, that turbo-
machinery can operate well when coupled to non-steady
flow systems. The demonstration of a combustor
generated net stagnation-pressure gain, whilst desir-
able, was viewed as less important than the primary
objective.

Although the fuel used for the experiments was
propane, the fuel specified by the makers for the gas
turbine unit when operating with the conventional
steady flow combustor, other fuels can be used with
pulsed combustors. The capability of pulsed combustors
to handle a wide range of gaseous, liquid and even
solid fuels has been established previously by experi-
ment, and subsequently reported, by various workers
[6,7,8]. The range of fuels investigated extends from
methane to pulverised coal and appears to be of suffi-
cient breadth to cover the demands of most gas tur-
brines.

BENEFITS OF COMBUSTOR PRESSURE-GAIN

This topic has been reviewed previously in the
literature [1,2] consequently it will only be summa-
rised here briefly. Figure 1 shows, in a subjective
manner, on the temperature (T) entropy (S) plane the
desirability of achieving a (stagnation) pressure rise,
or gain, during combustion. Several analyses have been
carried out quantifying the benefits of combustion
pressure gain. Examples are the analyses of Porter
[9], Marchal [10], Kentfield [11], Servanty [6], and
Marzouk and Kentfield [12]. Generally for simple-cycle
gas turbines the advantages of pressure-gain combustion
increase with decreasing cycle temperature ratio and
for any prescribed temperature ratio there is a cycle
pressure ratio where the benefit is at a minimum. The
benefit attainable is strongly dependent upon details
of the cycle. Typically, for simple cycles the specif-
cic output increase, and specific fuel consumption is
reduced, by 1-2% per percent reduction in pressure loss
across the combustor. Figure 2 presents the results of
an elementary thermodynamic analysis of the Brayton gas
turbine cycle showing, for the component efficiencies
etc. listed in Fig. 2, predicted increase in power
output per percent increase of combustor pressure gain
or percent reduction of combustor pressure loss.
It can be seen intuitively, and immediately
without rigorous proof, that a pulsed combustor is
inherently capable of generating a stagnation pressure
increase between air inlet and exhaust outlet when it
is recalled that a classical pulse-jet at rest inhales
intake air the stagnation pressure of which is equal to
the ambient pressure whilst the discharge occurs under
conditions where the static pressure is equal to the
ambient pressure. Normally the performance of a
strongly non-steady flow device such as a pulse-jet,
and many other pulsed combustors, is predicted by means
of the method-of-characteristics [13] after modifica-
tion to suit numerical computation. It is beyond the
scope of the work reported here to include a summary of
progress in the computational performance prediction
area.
region. The secondary flow passes to the right hand end of the system, mixed with the inlet backflow, via the two secondary flow ducts. The fluid leaving the secondary flow ducts finally combines with the combustor tailpipe flow in the combining zone prior to exiting the combustion system.

![Diagram](https://example.com/diagram.png)

**FIG. 3** **DIAGRAMMATIC ILLUSTRATION OF PROTOTYPE PULSED PRESSURE-GAIN COMBUSTOR**

It can be seen from Fig. 3, and Fig. 4 and 5 photographs of the actual laboratory combustor, that no attempt was made to utilise the secondary flow to cool the "primary zone" or pulsed combustor proper. Clearly such a development is possible and will be discussed later. The reason this cooling was omitted from the laboratory demonstrator unit was to allow the flow-path length of the secondary ducts to be easily tuned to permit pulses passing through the secondary flow ducts to arrive in the combining zone in antiphase to those emitted from the combustor tailpipe and to permit other component changes to be implemented (Fig. 5). The desirability of achieving this situation has been discussed previously [1].

It is, of course, apparent that the system shown in Fig. 3, 4 and 5 must suffer a significant heat loss to the surroundings, and overheating of the primary zone, which could both be avoided by redesigning the unit. It was decided, however, that the laboratory combustor, despite obvious shortcomings, would serve as a substitute for a conventional steady flow combustor constituting an integral component of a small, available, educational gas turbine of the gas generator type. Fortuitously the gas turbine had a throughput approximately matching that of the laboratory pulsed combustor when the latter was assumed to operate at conditions comparable to those of the gas turbine.

Results of tests of the laboratory pulsed combustor, which had a combustion zone diameter of 73 mm (2.875"), are presented in Fig. 6. The tests were carried out with ambient pressure and temperature constituting the stagnation conditions at the combustor air inlet. The inlet air was drawn directly from the laboratory without the provision of an air intake plenum; an intake plenum is, of course, required when the combustor is used in a gas turbine. Similarly the combustor exhaust discharged directly into the laboratory without the presence of a turbine or turbine inlet duct. The static pressure at the exhaust outlet was therefore, also that of the ambient. It can be seen from Fig. 6 that the maximum stagnation pressure gain was approximately 6% of the combustor inlet stagnation pressure at a combustor temperature ratio of approximately 2.7.

**FIG. 4** **PROTOTYPE PULSED PRESSURE-GAIN COMBUSTOR.**

**COMBUSTION ZONE DIAMETER 73 mm (2.875 in)**

**OPERATIONAL FREQUENCY APPROXIMATELY 190 Hz (190 CYCLE/S).**

The emission of pollutants from pulsed combustors of the type incorporated in the laboratory prototype pressure-gain unit have been shown, by experiment, to be generally small or negligible. The strongest concentration has been that of CO where 30 parts per million (CO emission index approximately $3 \times 10^{-3}$) has been noted at the tailpipe of the core pulsed combustor [1]. Consequently, for the preliminary tests reported here, no provision was made to perform gas analysis within, or at the exhaust of, the pulsed combustor equipped gas turbine.

In order to adapt the laboratory pulsed pressure-gain combustor to the gas turbine it was only necessary to construct a plenum type enclosure at the inlet end of the combustor into which the delivery of the gas turbine compressor was ducted. The exhaust end of the combustor was attached, via a conical adapter, to the turbine inlet.
FIG. 5  PROTOTYPE PULSED, PRESSURE GAIN, COMBUSTOR OF FIG. 4 EQUIPPED WITH ALTERNATIVE SECONDARY FLOW DUCTS AND COMBINING ZONE; ADDITIONAL ALTERNATIVE EXHAUST OUTLET COMPONENTS ARE ALSO SHOWN.

FIG. 6  STAGNATION PRESSURE GAIN PERFORMANCE OF UNIT SHOWN IN FIG. 4. NOTE: TEST DATA UNCORRECTED FOR AMBIENT CONDITIONS, ALL TESTS CARRIED OUT AT 1096 m (3600 ft) ALTITUDE. AMBIENT TEMPERATURE = 292-298 K (67-77°F), AMBIENT PRESSURE = 88.0-88.7 kN/m$^2$ (12.8-12.9 lb/in$^2$).

ADAPTATION OF THE GAS-TURBINE

The gas turbine used for the experiments was a Cussons P.9000, propane fuelled, educational gas generator unit the turbo-compressor portion of which is based on a C.A.V. turbo-charger featuring plain journal bearings, a centrifugal compressor and a centripetal (radial inflow) turbine. The gas turbine was built by G. Cussons Limited, 102 Great Clowes Street, Manchester, England M7 9HN. The basic configuration of the system is shown in Fig. 7.

FIG. 7  DIAGRAMMATIC ILLUSTRATION OF CUSSONS P.9000 GAS GENERATOR WITH CONVENTIONAL, STEADY FLOW, COMBUSTOR

Apart from the construction of a special plenum chamber to accommodate the air inlet portion of the pulsed combustor the only other changes made to the gas turbine were as follows:

i) the entire gas generator unit was rotated about the rotor-shaft axis to permit, for convenience, the pulsed pressure-gain combustor to lie in the horizontal plane. Originally the steady flow combustor was arranged vertically,

ii) the original fuel flow meter provided with the P.9000 unit was of the rotameter type. This was replaced with a choked-jet type flow meter to minimise the risk of erroneous fuel flow measurements when operating with the pulsed combustor. The latter can generate pressure waves in the fuel supply line which can propagate upstream towards the flow meter,

iii) a Validyne strain gauge type differential pressure transducer was installed. This was arranged to record, in the differential mode, the difference in static pressure between stations 2 and 3 (see Fig. 7) and in the non-differential mode, the static pressures at those stations.

Figure 8 is a diagram of the pulsed combustor installation. The tapered diffuser-duct connected to the compressor outlet is common to both the steady flow and pulsed combustion systems and is a piece of the original equipment of the P.9000 gas turbine. The P.9000 is started by blowing air, from an electric-motor driven blower, into the compressor intake. During this process the normal air inlet is shut off. When the unit is running in a self sustaining mode the flow metering air intake is opened. The starting air blower is then shut down and the blower intake is closed. For other than no load operation the butterfly type back pressure valve in the turbine exhaust duct is set to generate the required back pressure at the turbine outlet. Normally a fixed setting would be used to simulate the back pressure of a power turbine.

RESULTS

The most basic measurements obtained during the tests, namely those of air and fuel mass flows, are

\[
\frac{m_A}{m_F} = \frac{T_{out}}{T_{in}} \approx 2.5
\]

\[
\frac{\Delta P}{P_{AM}} = \frac{\Delta m}{m_F} \approx 3.0
\]

\[
\frac{T_{out}}{T_{in}} \approx 2.5
\]

\[
\frac{\Delta m}{m_F} = \frac{\Delta P}{P_{AM}} \approx 3.0
\]

\[
GROSS FUEL CONSUMPTION/COMBUSTION ZONE
\]

\[
\begin{align*}
\text{GROSS FUEL CONSUMPTION/COMBUSTION ZONE} &= \text{c.s.a.} \\
\text{GROSS FUEL CONSUMPTION/COMBUSTION ZONE} &= \text{c.s.a.} \\
\text{GROSS FUEL CONSUMPTION/COMBUSTION ZONE} &= \text{c.s.a.} \\
\end{align*}
\]
presented in the usual so-called dimensionless form in Fig. 9, versus temperature-corrected rotor speed, for no load operation that is with the turbine back-pressure control valve fully open. As can be seen from the diagram the dimensionless air mass flow is about 3% lower when using the resonant combustor. The approximately 20% greater fuel flow rate, which was to be expected, with the resonant combustor can be shown by simple heat loss analysis, to be a consequence of the exposed central pulsed combustor constituting the primary zone of the prototype, pulsed, pressure-gain combustion system.

Fig. 10 shows, also for the no-load case with an abscissa of temperature-corrected rotor speed, compressor and turbine pressure ratios and the cycle overall temperature ratio. A difficulty was experienced in measuring, experimentally, the turbine inlet temperature, $T_3$, when using the steady flow combustor. The problem was due to lateral temperature gradients within the turbine inlet duct. Consequently two temperature ratio curves are presented for the steady flow case. One curve was obtained experimentally, the other theoretically from an energy balance requiring knowledge of the compressor delivery temperature. Figure 10 shows that a higher turbine inlet temperature ratio was obtained with the resonant combustor. It can be seen from Fig. 10 that the temperature ratio with the resonant combustor was typically about 6% greater than the theoretical, energy balance derived, curve of $T_3/T_1$ for the steady flow case. The increase in $T_3/T_1$ with the resonant combustor is probably due to the combined effects of a small drop in compressor efficiency, due to the small decrease in compressor dimensionless mass flow, and radiative heat transfer to the compressor casing from the nearby exposed combustor primary zone. A simple heat transfer analysis indicates that it only requires the absorption, by the compressor casing, of 7% of the energy radiated by the combustor primary zone only, an amount which could be readily absorbed, to account, entirely, for a drop in compressor efficiency sufficient to result in the observed 6% increase in $T_3/T_1$ when employing the resonant combustor.

There may also be a small drop in turbine efficiency due mainly to the re-matching situation. According to the work of both Porter [9] and Cronje [15] the use of the pulsed combustion system should, in its own right, have a negligible influence on turbine efficiency which can be expected, under the worst circumstances, to be reduced by about 0.2%.

Due to better mixing in the turbine inlet duct with the resonant combustor there was little or no ambiguity, corresponding to the situation with the steady flow combustor, in the experimentally measured values.
values of $T_1/T_2$ for the resonant combustor. The maximum $N/T$, would have been restricted, due to the relatively high values of $T_1$ occurring with the pulsed combustor, had not a fuel system limitation, which it is later hoped to alleviate, imposed a yet lower limit on the maximum fuel flow rate and hence on rotor speed. Evaluations of combustor pressure loss, or gain, for both the steady flow and resonant combustors are presented in Fig. 11 for no load operation. It can be seen that the stagnation pressure loss with the steady flow combustor varies from approximately 0.8% to 2% over the speed range of the unit. For the resonant combustor there is a corresponding increase of stagnation pressure varying from 0.5% at low rotor speeds to approximately 0.2% near the upper end of the speed range. It would appear, when allowances are made for unavoidable pressure losses in the additional (non-common) duct work associated with the installation of the resonant combustor, a stagnation pressure gain from about 0.8% to 0.7% should be achievable as indicated by the dotted curve of Fig. 11.

FIG. 11 COMBUSTOR PRESSURE RATIOS WITH CONVENTIONAL AND RESONANT COMBUSTORS (NO-LOAD OPERATION)

It is worth noting that the dotted curve of Fig. 11 shows pressure gains much lower than those which would be expected on the basis of the test results shown in Fig. 6. From Fig. 6 the indication is that a pressure gain of between 3% and 5% should be achievable for the applicable gas turbine combustor temperature ratios, $T_1/T_2$, which correspond to $T_1/T_2$ of Fig. 6. Probable reasons for this apparent discrepancy are interference effects in the gas turbine installation due to the presence of the pulsed combustor intake plenum chamber and possibly also the turbine inlet nozzle. Both the intake plenum and turbine inlet nozzle may serve to modify wave reflection conditions at what would otherwise have been open ends but for the presence of these components. It is also apparent, from a comparison of Fig. 6 and 9, that the fuel flow rates and combustor temperature ratios of Fig. 9 do not correspond to those of Fig. 6 when due allowance is made for the different combustor inlet densities in the two cases. The implication is, therefore, that the operating conditions applicable to Fig. 6 and 9 differ considerably and are not due merely to differences in combustor inlet density. Further work is required to clarify the situation in order that remedial action can be undertaken. Future work should also include testing under loaded conditions, that is with the turbine back pressure valve actuated to simulate the presence of a power turbine.

POSSIBLE FUTURE DEVELOPMENT

The first priority for continuing the work is to restructure the pulsed combustion system in such a way that the secondary flow is arranged to flow over the primary zone and thereby reduce heat losses whilst providing a measure of convective cooling. Figure 12 shows at least one way in which this can be done without increasing the overall length of the combustor. However, the geometry depicted in Fig. 12 is complicated and is likely to be very costly for a single unit due, in large measure, to the spiral baffles needed to ensure that the secondary flow path is of the required length. At the expense of increasing, slightly, the overall length of the combustor the simplified configuration shown in Fig. 13 should be easier and cheaper to build. It provides the same advantages as the configuration of 12, namely convective cooling of the primary zone and virtual elimination of the very significant primary zone heat loss occurring in the prototype unit.

FIG. 12 DIAGRAMMATIC ILLUSTRATION OF AN ADVANCED VERSION OF A PULSED PRESSURE-GAIN COMBUSTOR

Another factor which should receive attention during redesign of the pulsed combustor, and installation on the gas turbine, are the elimination of unnecessary pressure losses both in the installation duct work and within the combustor itself. The overall consequences of ducting the secondary flow over the pulsed combustor and also reducing pressure losses should be a major reduction in fuel consumption and an increase in pressure gain such that, at worst without other improvements, the latter should approach the dotted curve of Fig. 11.

Additionally, for the next stage of the work it would be useful, and an important design guide, to add thermocouples to the highest temperature components of the pulsed pressure-gain combustor. Also exhaust gas analyses should be carried out and, so far as possible, correlated with earlier emission data for the basic pulsed combustor operating at atmospheric pressure [1].
FIG. 13 PROPOSED SECOND GENERATION PULSED, PRESSURE GAIN, COMBUSTOR FOR CUSSONS P.9000 GAS GENERATOR. X INDICATES EQUAL, REPLACEABLE, LENGTHS OF RECTANGULAR CROSS-SECTION DUCTING FOR TUNING SECONDARY FLOW PATH LENGTH (DIAGRAMMATIC).

It would also be useful, if circumstances permit, to advance to a larger system by employing a larger gas turbine than the Cussons P.9000. The influence of size on the potential pressure gain performance of pulsed pressure-gain combustors has been studied previously indicating significant performance benefits accruing from an increase of scale [2]. Acoustic measurements should also be made in future tests. The resonant combustor is inherently more noisy than many steady flow combustors, however expansion through the turbine serves to reduce, substantially, the characteristic noise associated with a highly loaded pulsed combustor. This observation is consistent with that often made in relation to turbo-chargers where the exhaust noise is partially muffled by the turbine. Ideally, rematching of the compressor and turbine performance characteristics is required to maximise the benefits obtainable from the use of a pressure-gain combustor. Such rematching is not practical in the present circumstances but should be born in mind in relation to future work.

CONCLUSIONS

The following conclusions can be drawn from the work:

i) a simple gas turbine employing a resonant combustion system, as a substitute for a conventional steady flow combustor, has been demonstrated in a very embryonic form,

ii) it would appear that a resonant combustor can have sufficient operational flexibility to serve as a gas-turbine combustor,

iii) a resonant, pulsed, pressure-gain combustor has been shown to be capable of eliminating combustor pressure losses and producing a small combustion generated stagnation-pressure gain,

iv) it is not necessary to introduce any additional major moving parts when a valveless resonant combustion system is employed in a gas turbine,

v) what is, in effect, a turbo-charged valveless pulsed-combustor was demonstrated.

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