ANALYSIS, MODIFICATION AND IMPROVEMENT OF PERFORMANCE
OF A CLOSED, REGENERATIVE, RECIPROCATING BRAYTON CYCLE ENGINE

by

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B.S., Seoul National University
(1972)
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(1974)

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Signature of Author

Department of Mechanical Engineering, May 15, 1978

Certified by

Thesis Supervisor

Accepted by

Chairman, Departmental Committee on Graduate Students

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Submitted to the Department of Mechanical Engineering in May, 1978 in partial fulfillment of the requirements for the Degree of Doctor of Philosophy

ABSTRACT

The VHGE is a closed, regenerative, reciprocating Brayton cycle engine using helium as the working gas. Preliminary analysis shows that this engine is competitive with the Stirling engine in terms of low pollution, high efficiency, and power density. The prototype one cylinder engine was built in 1972, and tested at low power. The causes of the low efficiencies achieved in the tests have been under investigation since 1972.

Piston ring leakage was first suspected as the cause, but the tests and analyses reported in this thesis show that cyclic heat transfer between the gas and the cylinder walls is the dominant cause of the low efficiency. Compressor and expander performance have been correlated with the magnitude of the
cyclic heat transfer. Modifications of the compressor configuration to decrease the cyclic heat transfer improved the compressor efficiency from 60% to 83% and the indicated engine efficiency from 21% to 33%.

The results indicate that indicated engine efficiency could approach the expected 47% by means of further reduction of cyclic heat transfer in both the expander and the compressor.

Thesis Supervisor

Joseph L. Smith, Jr.

Professor of Mechanical Engineering
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## GLOSSARY OF SYMBOLS

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<tr>
<td>A</td>
<td>Heat transfer area</td>
</tr>
<tr>
<td>$A_c$</td>
<td>Compressor volume per unit displacement</td>
</tr>
<tr>
<td>$A_w$</td>
<td>Cross-section area of the thermocouple wire</td>
</tr>
<tr>
<td>$b$</td>
<td>$h/k$</td>
</tr>
<tr>
<td>$C$</td>
<td>Discharge coefficient for the orifice ; Celsius</td>
</tr>
<tr>
<td>$c$</td>
<td>Specific heat of the plate</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Drag coefficient</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat of gas at constant pressure</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Specific heat of gas at constant volume</td>
</tr>
<tr>
<td>$C_0$</td>
<td>$\left( \gamma - 1 \right)/\gamma$</td>
</tr>
<tr>
<td>$C_1$</td>
<td>$C_0/\xi_c$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>$C_0/\xi_e$</td>
</tr>
<tr>
<td>c.v.</td>
<td>Control volume</td>
</tr>
<tr>
<td>$D$</td>
<td>Thermocouple bead diameter ; tube diameter</td>
</tr>
<tr>
<td>$d$</td>
<td>Thermocouple wire diameter ; diameter of the orifice</td>
</tr>
<tr>
<td>$E$</td>
<td>Total energy of the gas</td>
</tr>
<tr>
<td>$E_r$</td>
<td>Expansion factor of the gas due to compressibility</td>
</tr>
<tr>
<td>$F$</td>
<td>Radiation view factor</td>
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<td>$F_a$</td>
<td>Thermal expansion factor of the orifice metal</td>
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<td>$F_{\text{bead}}$</td>
<td>Drag force due to thermocouple bead</td>
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<tr>
<td>$F_d$</td>
<td>Drag force</td>
</tr>
<tr>
<td>$F_{\text{wire}}$</td>
<td>Drag force due to thermocouple wire</td>
</tr>
<tr>
<td>$g_c$</td>
<td>Gravitational proportionality constant</td>
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H  Total enthalpy; heat stored in half a cycle
h  Specific enthalpy; heat transfer coefficient
ΔH Change of total enthalpy
I  Moment of inertia
I.D. Inner diameter
j  $\sqrt{-1}$
K  Kelvin
k  Thermal conductivity; wavelength constant
$k_s$ Average thermal conductivity of thermocouple wires
L  Length of the thermocouple wire exposed to the gas
$L_c$ Connecting rod length
L.P. Low pressure
M  Mach No.
m  Mass of gas
$m$ Mass flow rate
$m_f$ Mass flow rate of the engine
$M_{\text{max}}$ Maximum bending moment
$N_u$ Nusselt No.
$n_1, n_2$ Polytropic index
O.D. Outer diameter
P  Pressure
p  Acoustic pressure
$p(x,t)$ Acoustic pressure as a function of $x, t$
$\Delta P$ Pressure difference
$P_d$ Mean discharge pressure
$P_h$ Expander inlet pressure
$P_i$ Mean intake pressure

$P_L$ Compressor inlet pressure

$P_r$ The ratio $P_h/P_L$

$Pr_g$ Prandtl No. for the gas

$P_{ref.}$ Reference pressure for entropy

$q$ Heat transfer rate

$q(x,t)$ Acoustic volume flow rate

$q_c$ Net heat transfer rate for the compressor

$q_e$ Net heat transfer rate for the expander

$q_{cooler}$ Cooling water heat removal rate of the cooler

$q_{cyclic}$ Magnitude of cyclic heat transfer rate

$q_{environ,i}$ Rate or heat loss to the environment from control volume $i$

$q_{heater}$ Rate of electrical heat input to the heater

$q_{net}$ Net heat input to the compressor per cycle

$q_{pm}$ Rate of heat transfer during process $p$ for mass portion $m$

$q_{water}$ Cooling water heat removal rate

$q_+$ Rate of heat transfer into the gas

$q_-$ Rate of heat transfer out of the gas

$R$ Gas constant ($PV/\text{mT}$)

$r$ One half the engine stroke

$Re_d$ Reynolds No. based on $d$

$R.H.Ex.$ Regenerative heat exchanger

$RPS$ Engine revolution per second

$S$ Specific entropy

$S_1, S_2$ Cross-sectional area of the pressure tap chambers
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<td>$T$</td>
<td>Temperature of the gas</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
</tr>
<tr>
<td>T.C.</td>
<td>Thermocouple</td>
</tr>
<tr>
<td>$T_d$</td>
<td>Mean discharge temperature of the compressor</td>
</tr>
<tr>
<td>$T_{ei}$</td>
<td>Effective intake temperature</td>
</tr>
<tr>
<td>$T_{ed}$</td>
<td>Effective discharge temperature</td>
</tr>
<tr>
<td>$T_h$</td>
<td>Expander intake temperature</td>
</tr>
<tr>
<td>$T_{in}$</td>
<td>Inlet temperature</td>
</tr>
<tr>
<td>$T_j$</td>
<td>Thermocouple junction temperature</td>
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<tr>
<td>$T_L$</td>
<td>Compressor inlet temperature</td>
</tr>
<tr>
<td>$T_{out}$</td>
<td>Outlet temperature</td>
</tr>
<tr>
<td>$T_{ref.}$</td>
<td>Reference temperature for entropy</td>
</tr>
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<td>$T_w$</td>
<td>Wall temperature</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature difference</td>
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<td>$T_c$</td>
<td>Conduction error for thermocouple</td>
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<td>$T_{comp.}$</td>
<td>Temperature difference between inlet and outlet compressor</td>
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<tr>
<td>$T_{exp.}$</td>
<td>Temperature difference between inlet and outlet expander</td>
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<td>$T_r$</td>
<td>Radiation error for thermocouple</td>
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<td>$T_t$</td>
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<td>$T_v$</td>
<td>Velocity error for thermocouple</td>
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<td>$V$</td>
<td>Volume of the gas</td>
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<td>$V_d$</td>
<td>Discharge displacement</td>
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<td>$V_i$</td>
<td>Intake displacement</td>
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<tr>
<td>$v_{p,\text{max}}$</td>
<td>Maximum piston velocity</td>
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\[ W_{\text{brake}} \quad \text{Brake engine power} \]
\[ W_c \quad \text{Compressor indicated power} \]
\[ W_{\text{cv}} \quad \text{Rate of work transfer for control volume} \]
\[ W_{\text{linkage}} \quad \text{Linkage power loss of the engine} \]
\[ W_{nc} \quad \text{Reversible polytropic compressor power} \]
\[ W_{ne} \quad \text{Reversible polytropic expander power} \]
\[ W_{pm} \quad \text{Rate of work transfer during process p for mass portion m} \]
\[ W_{sc} \quad \text{Isentropic compressor power} \]
\[ W_{se} \quad \text{Isentropic expander power} \]
\[ \alpha \quad \text{Thermal diffusivity} \]
\[ \beta \quad \text{Ratio orifice diameter d/ pipe diameter D} \]
\[ \gamma \quad \text{Ratio of specific heats } c_p/c_v \]
\[ \varepsilon \quad \text{Emissivity} \]
\[ \eta \quad \text{Efficiency} \]
\[ \eta_c \quad \text{Adiabatic efficiency of the compressor} \]
\[ \eta_e \quad \text{Adiabatic efficiency of the expander} \]
\[ \eta_i \quad \text{Indicated engine efficiency} \]
\[ \eta_{nc} \quad \text{Polytropic efficiency of the compressor} \]
\[ \eta_{ne} \quad \text{Polytropic efficiency of the expander} \]
\[ \eta_R \quad \text{Regenerative heat exchanger effectiveness} \]
\[ \eta_v \quad \text{Actual volumetric efficiency} \]
\[ \theta \quad \text{Temperature above average plate temperature} \]
\[ \vartheta \quad \text{Crank angle} \]
\[ \vartheta_{ma} \quad \text{Temperature amplitude of the medium} \]
\[ \xi_c \quad \text{Polytropic coefficient of the compressor} \]
\( \xi_e \quad \text{Polytropic coefficient of the expander} \\
\rho \quad \text{Density of plate} \\
\rho_g \quad \text{Density of gas} \\
\sigma \quad \text{Normal stress} \\
\sigma_{\text{max}} \quad \text{Maximum normal stress} \\
\tau \quad \text{Shear stress, time} \\
\tau_{\text{max}} \quad \text{Maximum shear stress} \\
\tau_0 \quad \text{Characteristic response time for a thermocouple} \\
\omega \quad \text{Angular speed}
CHAPTER I. INTRODUCTION

1. History, Description of the Engine

The new valved hot-gas engine (VHGE) was constructed (1) at the Cryogenics Laboratory, M.I.T. in 1972 as a byproduct of many years of work in the laboratory on closed cycle engines and refrigerators. Studies on VHGE have been presented as theses (1,2,3,4) and a paper (5). The VHGE concept evolved from the idea of adding valves to the Stirling engine to overcome some of its practical limitations.*

The engine consists of a reciprocating expander/compressor, a heater, a cooler, and a regenerative heat exchanger as shown in Fig. I-1.** The compressor has spring-loaded check valves and the expander has cam-operated poppet valves.

The working gas undergoes a thermodynamic cycle close to ideal Brayton cycle. Ideally, the gas gets heated in the heater under constant-pressure followed by isentropic expansion in the expander, constant pressure regenerative cooling in the regenerative heat exchanger. The gas gets further cooled in the cooler under constant pressure, followed by isentropic compression and constant-pressure regenerative heating in the regenerative heat exchanger before the gas enters the heater to reach the maximum temperature (see Fig. I-2).

* For comparison between the VHGE and the Stirling engine, refer to reference 1.

** Taken from reference 1.
Fig. 1-1 Prototype configuration of VHGE
Fig. I-2  T-s, P-v diagram of VHGE
*s denotes isentropic process.
2. Objectives

The ultimate objective of this engine project is to realize the potential of the VHGE concept as a power plant for transportation vehicles or other energy conversion systems.

The immediate objective is to find the causes of low performance of the first generation VHGE and to improve the efficiency sufficiently to warrant further research on this type of external combustion engine, which seems to have a potential equal to the Stirling engine. It is hoped that careful documentation of the problems encountered in the work on this engine will be of benefit to others working on similar equipment.

3. Problems and Approaches

The low-power tests (12kW) showed consistently low efficiency* for a range of speeds, mean pressure, and temperature (2). The following symptoms were generally observed: (a) About 10% more heater power than expected was required to reach the designed operating conditions; (b) the compressor power from the indicator diagram was about 1.8 times that expected; and (c) the compressor clearance volume had to be set at its minimum in order to reach the designed pressure ratio of 2:1. Further analysis of the data revealed the following: (d) about 10% of total heater power input could not be accounted for in the energy balance of the engine and (e) the "apparent mass

* $\eta_1 = 20.8\%$ based on net heat input to the engine (Test #1)
flow rates" through the engine components differed from one another significantly as shown in Table I-1.

Table I-1  Apparent Mass Flow Rates in terms of Cooler Flow Rate ( Test #1, (1) )

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Expander intake</td>
<td>99%</td>
<td>Compressor intake</td>
</tr>
<tr>
<td>Expander discharge</td>
<td>98%</td>
<td>Compressor discharge</td>
</tr>
<tr>
<td>R.H.Ex. (H.P.)</td>
<td>111%</td>
<td>Heater</td>
</tr>
<tr>
<td>R.H.Ex. (L.P.)</td>
<td>107%</td>
<td></td>
</tr>
</tbody>
</table>

These mass flow rates are called as such because they are not actual flow rates but the flow rates calculated using average temperatures, pressures measured external to the components, and for the expander and the compressor the volume changes as well during intake and discharge. In the beginning, the piston–ring leakage between the expander and the compressor was suspected to be the main cause of the discrepancies in apparent mass flow rates, and also of the low efficiency. But a series of leakage tests showed that the leakage alone did not account for the engine performance (1,3,4).

Measurements of the transient gas temperatures inside cylinder showed significant spatial variation in the temperature. The compressor head was cooled by a water jacket to show the effect of internal heat transfer. The significant changes in compressor and expander performance point to the internal heat
transfer as a very important factor.

In order to analyze the effect of heat transfer separately, the piston-ring leakage was eliminated by means of a temporary rubber O-ring piston seal. The zero-piston-ring-leakage test did not show any significant increase in compressor indicated efficiency. The compressor was found to be far from adiabatic, with a cyclic heat transfer at a rate on the order of the compressor power. In the expander, the cyclic heat-transfer rate was about half that of the compressor.

On the basis of this evidence, the compressor was modified to decrease the cyclic heat transfer inside compressor. The result was a drastic increase in compressor efficiency. Instantaneous mass flow rates were also measured at the compressor inlet and outlet during tests of the modified compressor. These more complete and accurate tests also corrected the energy balance and resolved the discrepancies in the apparent mass flow rates.
Chapter II PERFORMANCE EVALUATION AND IMPROVEMENT OF THE ENGINE

1. Experiments Performed

1) Leakage measurement

The aforementioned discrepancy in the apparent mass flow rates through engine components implied a significant gas leakage between expander and compressor past the polyimide piston rings. Static leakage rates past piston rings were measured to be about 5% of the engine mass flow rate (3). In order to simulate the reversal of the pressure difference during normal engine operation, the sign of ΔP across the piston was changed abruptly by pressurizing or depressurizing either the compressor or expander (4). The dynamic-leakage rate, estimated from the pressure-time trace, was on the order of 10% of the engine mass flow rate. Direct applicability of the result to the actual engine was questionable, however, because the pressure reversal occurs much faster during engine operation causing the piston rings to move rapidly across the grooves.

2) Transient gas-temperature measurement

After the inconclusive leakage tests, it was decided to take transient gas-temperature data inside the cylinder. Fast-response thermocouple probes of special design were fabricated using 25.4 μm O.D. chromel alumel wires, insulated with alumina ceramic tubes and protected by Inconel tubes. They were design-
ed to minimize the inherent errors due to conduction, radiation, velocity, and transients effect. The response time was less than a millisecond based on the flow conditions during the intake and discharge period. At first, four probes were inserted into the cylinder: one inside each expander port, and two inside the annular space of the compressor near each valve. (T.C. 1, T.C. 2 in Fig. II-1. (a)). Expander temperature data thus obtained show a significant temperature drop during the intake process, indicating significant negative heat transfer. Curiously, average compressor-inlet-and-exhaust temperature data do not show significant deviation from the isentropic case. However, it was the excessive compressor power (1.8 times the expected (1)), that was responsible for the low efficiency rather than the expander. Later tests with two additional probes (T.C. 3, T.C. 4 in Fig. II-1. (a)) into the edge of the main compressor space, revealed a significant spatial temperature variation inside the compressor (Fig. II-2) and a much higher temperature fluctuation in the main space than the annular space where the original two thermocouples were located. This information on cylinder gas temperature motivated further study of the internal heat-transfer problem.

3) Cooled-compressor test (Test #2)

During the measurements of the transient gas temperatures, additional performance data were obtained with a cooling-water jacket around the compressor block (Fig. II-1). The increased
Fig. II-1  Compressor of VHGE before (a) and after (b) modification

* denotes location of thermocouples
Fig. II-2 Temperature measurement at 4 locations inside compressor before modification (Test #3)

* See Fig. II-1.
compressor cooling increased the overall indicated engine efficiency from $20.8\%$ (Test #1) to $25.9\%$. The compressor adiabatic efficiency increased from $59\%$ (Test #1) to $70\%$ due to cooling, whereas, the expander adiabatic efficiency fell from $97\%$ (Test #1) to $88\%$. The unusually high expander efficiency of test #1 was attributed to a favorable effect from the leakage into the expander. However, in Test #2, the temperature of the leaking gas dropped due to increased cooling of compressor which probably caused the drop in expander efficiency.

4) Zero-piston-ring-leakage test (Test #3)

One of the observation from Test #2 was that the compressor efficiency increased by $11\%$ as a result of cooling the wall even though piston-ring leakage was present. In subsequent tests, the piston-ring leakage was eliminated by the use of a rubber O-ring to study the heat-transfer phenomena inside the cylinder without uncertain leakage. One of the four grooves for the polymide rings was filled with fiber-glass-reinforced epoxy up to appropriate depth to accommodate a rubber O-ring. A felt O-ring soaked into O-ring lubricant was placed in the adjacent groove. After the installation, the piston seal was thoroughly checked for possible leakage by both static and dynamic test, and proved to be quite satisfactory. One problem with the O-ring was that the O-ring swept area had to be maintained relatively cool to prevent early failure; thus the
cylinder water jacket was extended closer to the expander section of the cylinder. The increased cooling of the expander limited the improvement so that the increase in expander efficiency for no leakage was moderate (88.1% to 90.5%). The compressor efficiency remained virtually the same (69.7% to 69.8%), and the overall efficiency increased from 25.9% to 27.7%. The slight changes in efficiency implies that once the compressor is cooled sufficiently, leakage was not an important parameter but the internal cyclic heat transfer was. This point will be treated in detail in a later section on cyclic heat transfer.

5) Modified compressor test (Test #4, #5, #6)

The results of the internal cyclic heat-transfer analysis (Fig. II-7, Fig. II-8) show that an increase in the efficiency requires a decrease in the magnitude of the cyclic heat transfer. The following modifications of the compressor were made to reduce the cyclic heat transfer (Fig. II-1): (a) Removal of the piston-rod-seal assembly. Elimination of the annular space. Replacement of the six polyimide piston-rod seals with a rubber O-ring; (b) Removal of the separate clearance-volume adjusting cylinder which extended horizontally and replacement with a movable compressor head complete with cooling water circuits and a 0 ring piston rod seal; (c) Insulation of the bottom face of the piston with a 3 cm layer of phenolic Micarta. The piston bottom face had been
suspected as one of the hot spots inside the compressor; and (d) Replacement of the low carbon steel compressor block with a newly fabricated brass block which had water cooling near the inner surface and valve ports (see Fig. II-3). Because of the new arrangement, it was possible to insert transient temperature probes well inside the compressor volume. Thus, the thermocouples recorded temperatures close to the bulk gas temperature. The compressor adiabatic efficiency increased significantly from 69.8% to 82.9% (Test #6), and the clearance volume of compressor for the 2:1 pressure ratio increased from $2.950 \times 10^{-4} \text{ m}^3$ to $4.542 \times 10^{-4} \text{ m}^3$ which was quite close to the $5.244 \times 10^{-4} \text{ m}^3$ for isentropic compression. Expander efficiency decreased somewhat (90.5% to 86.7%) which could be attributed to the following: (a) Lower intake temperature and (b) More heat loss to the cylinder jacket. The indicated engine efficiency based on net heat input to the engine was 33.3% in Test #6.

2. Mass Flow Rate and Energy Balance

Referring back to section I.3, the apparent mass flow rates through engine components were quite different from one another. The apparent mass flow rates for the heat exchange components were calculated from the estimated heat fluxes and the temperature changes across the component. The energy balance for the entire engine showed that about 10% of the total heat input was unaccountable. The consistency with which the
Fig. II-3 (a) New, effectively cooled compressor block (cross-sectional top view)
Fig. 19(b) New, effectively cooled compressor block (cross-sectional sideview)
apparent mass flow rate of the heater exceeded the apparent mass flow rate of the cooler indicated the heater heat leak into the environment was under-estimated. In addition, the temperature drop for the low-pressure stream of the regenerative heat exchanger always exceeded the temperature increase for the high-pressure stream. This was first attributed to an imbalance of the mass flow because of friction and leakage; however, the results did not change when the leakage was eliminated.

The other possibility was that the external heat leak for the regenerative heat exchanger had been underestimated. For Test #3, the gas temperatures indicated a 0.62 kW heat leak while conduction and convection calculations for the external insulation predicted only 0.15 kW. An error in estimating the external heat leak of both the heater and the regenerative heat exchanger also proved to be the cause of the error in the energy balance and the difference in calculated heater and cooler flow rates. This conclusion was proved by two tests, a heat-transfer flow test of the engine and a direct measurement of the mass flow rate.

1) Heat-transfer flow test*

In this test, the expander valves were blocked open so that the pressure inside the entire engine was nearly constant except for small flow-friction losses. The expander-inlet temperature was set at the average of the intake and discharge

* See Appendix A. 5 for detail.
temperature for normal operation, and the pressure was close to normal low pressure. With the expander and compressor powers essentially zero, the heat leaks of the engine could be observed directly and the external heat leaks were verified, 0.35 kW for the regenerative heat exchanger and 0.88 kW for the heater. However, a direct measurement was still needed to confirm the actual engine flow rate.

2) Instantaneous mass flow measurement

Two square-edged orifices were installed, one before and one after the compressor. The connecting tubes were rearranged to provide adequate straight runs before and after the orifices. Although orifices are not usually recommended for pulsating flow, the actual gas flow through the orifice was nearly quasistatic at any given time. The instantaneous flow was accurately indicated because the natural frequency of the differential-pressure transducer (6000 Hz) and of the transducer pressure taps (1400 Hz) were much higher than the flow pulsating frequencies (10 Hz due to engine speed, 300 Hz due to valve motion). The error due to pulsation was estimated to be at most 1-2%. The instantaneous mass flow rate was calculated from the ΔP data of Test #5 and integrated in time to yield 7.60 x 10^{-3} kg/sec for intake and 7.71 x 10^{-3} kg/sec for discharge, while the cooler flow rate was 7.56 x 10^{-3} kg/sec. Since the pulsating error is positive (6), the use of the cooler-flow-rate calculation in preference to the heater-
flow-rate calculations is justified. For more detail on mass flow measurement, refer to section III.2.

3. Cyclic Heat Transfer Inside Cylinder

1) Evidences of cyclic heat transfer

The importance of internal heat transfer was well demonstrated by the marked changes in performance when the compressor block was first cooled (Test #2). The compressor adiabatic efficiency increased from 58.9% to 69.7%, and indicated engine efficiency increased from 20.8% to 25.9%.

The observed gas-temperature differences of up to 30°C (Fig. II-2) between different locations within the compressor indicate the reason for the continued low compressor efficiency. The lower temperature was in the annular space near the valves (Fig. II-2) and the higher temperature was at the edge of the compressor swept volume. The high apparent mass flow of the compressor* compared to actual flow indicated an even higher gas temperature in the swept volume where thermocouples could not be located. The implication of the temperature gradient in the gas inside the compressor was that a significant internal heat transfer was taking place between the gas and the walls of the compressor. The nature of the heat transfer was cyclic in time: heating during reexpansion and intake, cooling during compression and discharge. The temperature data indicated a similar heat transfer in the expander with the incoming gas cooled by as much as 100°C during intake. However, the

* See Table I-1.
average discharge temperature recorded outside the expander was very close to the reversible adiabatic temperature, indicating a significant reheating of the gas during expansion and discharge.

2) Mechanism of cyclic heat transfer

The simplest explanation of the cyclic heat transfer is that the inside walls of the compressor and expander are operating as a thermal regenerator, extracting heat from the gas in the cylinder when the gas is hotter than the wall, storing the heat in the wall, and returning the heat to the gas when the gas is colder than the wall. The simplest analysis of this action is to consider an infinitely thick plate in contact with a fluid experiencing a periodic change of temperature.

The unsteady conduction equation is \( \frac{\partial \Theta}{\partial \tau} = \alpha \frac{\partial^2 \Theta}{\partial x^2} \) and the boundary condition at the surface is \( k \frac{\partial \Theta}{\partial x} \bigg|_s = h \Theta_s \). The heat energy stored in half a cycle can be expressed as (7),

\[
H = \frac{1}{\omega} \rho c k A \Theta_{ma} y,
\]

(II - 1)

where \( \Theta_{ma} \) is the temperature amplitude of the medium, \( \Theta \) is the temperature above the average plate-surface temperature which is assumed to be equal to the average of medium temperature, \( \tau \) is time, \( \alpha \) is the thermal diffusivity, \( k \) is the thermal conductivity of plate, \( h \) is the heat-transfer coefficient, \( \omega \) is the angular speed of the temperature pulsation, and \( \rho c \) is the thermal capacity of plate. The ratio of temperature amplitude of the plate surface to that of the medium, \( Y \), can be expressed
\[
Y = (1 + 2a/b + 2(a/b)^2)^{-0.5};
\]
\[
a \equiv (a/\omega)^{0.5}; \quad b \equiv h/k.
\]

For the actual processes, equation II-1 cannot be applied directly because the wall temperature at an instant of time is not uniform throughout and usually the local time-average wall temperature is not the same as the local average gas temperature. Furthermore, the gas flows by different parts of the cylinder so that the gas sees a certain wall temperature distribution along the flow path. Flow past a wall with a temperature distribution but with little change in local temperature (i.e. low \(Y\)) can create the effect in the gas of a wall of uniform temperature that has high surface-temperature amplitude (i.e. high \(Y\)) facing a quiescent gas undergoing temperature fluctuation. This means that the wall-temperature distribution can enhance or inhibit heat transfer.

The above argument and equation II-1 show how to design a compressor and expander to decrease the internal cyclic heat transfer: (a) Decrease heat-transfer surface area \(A\), by decreasing the surface-to-volume ratio; (b) use a material with a low \(\kappa\) for the inside wall; (c) maintain proper wall temperature distribution; and (d) decrease the heat transfer coefficient \(h\) by reducing the turbulence level. This involves change in flow pattern, proper choice of the shape of the chamber, and location, direction and size of the valves.
3) Effective intake temperature, Effective discharge temperature

In both expander and compressor, inlet density or outlet density, measured external to the cylinder, can be combined with the volume changes during intake or discharge to calculate "apparent mass flow rates". Apparent mass flow rates thus calculated are significantly different from actual mass flow rate. In view of this, it was appropriate to introduce new temperatures which will better represent the conditions inside the cylinder during intake or discharge. The effective intake temperature $T_{ei}$ is defined as the mean gas temperature which will give correct average mass flow rate during intake. The effective discharge temperature $T_{ed}$ is defined similarly.

$$T_{ei} = \frac{\bar{P}_i \Delta V_i}{m_f R}, \quad T_{ed} = \frac{\bar{P}_d \Delta V_d}{m_f R} \quad (\text{II-2})$$

where $\bar{P}_i$ is mean intake pressure; $\bar{P}_d$ is mean discharge pressure; $\Delta V_i$ is intake displacement; $\Delta V_d$ is discharge displacement; $m_f$ is mass flow/cycle; $R$ is gas constant.

These new definitions are related to the effect of heat transfer on volumetric efficiency, $\eta_v$. Take the intake process for example:

$$\eta_v = \frac{\text{actual mass drawn in}}{\text{displacement x inlet density}} = \frac{\Delta V_i}{\Delta V_d} \frac{\bar{P}_i}{P_i} \frac{T_i}{T_{ei}} \quad (\text{III-3})$$
where $T_i$, $P_i$ are inlet temperature, and inlet pressure respectively. The ratio $\Delta V_i/V_D$ is the effect of re-expansion. The ratio $\bar{P}_i/P_i$ is the effect of valve pressure drop and $T_i/T_{ei}$ is the effect of heat transfer during intake. $T_{ei}$ and $T_{ed}$ for the engine tests are given in Table II-1 along with measured inlet temperature $T_i$, and discharge temperature $T_d$.

Table II-1 Effective Intake and Discharge Temperatures of Compressor

<table>
<thead>
<tr>
<th></th>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{ei}/T_i$</td>
<td>412/296</td>
<td>402/297</td>
<td>400/293</td>
<td>317/288</td>
<td>314/287</td>
<td>302/280</td>
</tr>
<tr>
<td>$T_{ed}/T_d$</td>
<td>634/415</td>
<td>582/409</td>
<td>549/397</td>
<td>453/367</td>
<td>436/369</td>
<td>447/360</td>
</tr>
</tbody>
</table>

4) Cyclic heat-transfer estimate

Two methods will be used for estimating the amount of cyclic heat transfer for the compressor or expander, the equivalent isentropic method and the equivalent reversible polytropic method. The isentropic method is a very simple approach, requires minimum input data, but somewhat lacks the accuracy of the polytropic method which requires additional data on mass, pressure and volume.

(1) Equivalent isentropic method

This method models the observed heat-transfer effects
inside the cylinder by equivalent steadyflow processes. The compressor is modeled as a preheater, an isentropic steady-flow compressor and an aftercooler. The expander is modeled as a precooler, an isentropic expander, and an afterheater. Fig. II-4 and Fig. II-5 show the states and the model for the engine.

The cyclic heat transfer is calculated from measured values of $P_h/P_L$, $T_1$, $T_2$ and the actual indicated compressor power $\dot{W}_c$, where the states refer to Fig. II-4. The inlet temperature $T_{1i}$ for the equivalent isentropic compression is adjusted so that the steady-flow isentropic compressor power $\dot{W}_{1i-2i} = \dot{W}_c$. The magnitude of cyclic heat transfer $\overline{Q}_{cyclic}$ is the average of the magnitude of heat transfer in preheater $|\dot{Q}_+|$ and the magnitude of heat transfer in the aftercooler $|\dot{Q}_-|$. Table II-2 lists the values of $\overline{Q}_{cyclic}$ for the compressor and the expander in non-dimensional form $X = \overline{Q}_{cyclic}/\Delta H$, where total-enthalpy change $\Delta H = H_2 - H_1$ for compressor, and $\Delta H = H_4 - H_5$ for expander. For examples of calculation using Equivalent Isentropic method, see Appendix C.

Table II-2 Magnitude of Cyclic Heat Transfer

(Equivalent Isentropic Method)

<table>
<thead>
<tr>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$X_c$</td>
<td>2.01</td>
<td>1.31</td>
<td>1.69</td>
<td>1.46</td>
<td>0.97</td>
</tr>
<tr>
<td>$X_e$</td>
<td>0.12</td>
<td>0.37</td>
<td>0.27</td>
<td>0.22</td>
<td>0.46</td>
</tr>
</tbody>
</table>

41
Fig. II-4  T-s diagram of VHGE for equivalent isentropic method

\[ Q_+ = m_f c_p (T_{1i} - T_1) \]

pre-heater 1i

\[ Q_- = m_f c_p (T_2 - T_{2i}) \]

after-cooler 2i

\[ Q_- = m_f c_p (T_{4i} - T_4) \]

pre-cooler 4i

\[ Q_- = m_f c_p (T_5 - T_{5i}) \]

after-heater 5i

Fig. II-5  Equivalent isentropic approximation
(2) Equivalent reversible polytropic method

This method requires indicated P-V data, mass flow rate \( \dot{m}_f \), and inlet and outlet temperatures. This matches the measured indicated power with reversible-polytropic compression and reexpansion processes, and constant-pressure intake and discharge process. Fig. II-6 shows the flow diagram for the model of the processes of the compressor. \( \dot{W}_{pm} \) and \( \dot{Q}_{pm} \) in Fig. II-6 denote power and heat-transfer rate during the process \( p \) for mass portion \( m \) where \( p = i, c, d, r \), denoting intake, compression, discharge and reexpansion respectively; and, \( m = f, c, t \), denoting flow mass, clearance mass, and total mass respectively.

The sum of the powers \( \sum_{m,p} \dot{W}_{pm} \) is equated to the measured indicated power \( \dot{W}_c \). Heat-transfer rate into the gas inside the compressor \( \dot{Q}_+ \), and out of the gas \( \dot{Q}_- \) are given as

\[
\dot{Q}_+ = \dot{Q}_{if} + \dot{Q}_{ic} + \dot{Q}_{rc} \\
\dot{Q}_- = \dot{Q}_{df} + \dot{Q}_{dc} + \dot{Q}_{ct}
\]  

(II-4)

The heat-transfer rates \( \dot{Q}_+ \) and \( \dot{Q}_- \) are expressed in terms of the effective intake temperature \( T_{ei} \), and the effective discharge temperature \( T_{ed} \).

\[
\dot{Q}_+ = \dot{m}_f c_p (T_{ei} - T_{in}) + \dot{Q}_{rc} \\
\dot{Q}_- = \dot{m}_f c_p (T_{out} - T_{ed}) + \dot{Q}_{ct}
\]

(II-5)

As before, the non-dimensional magnitude of cyclic heat transfer of compressor \( X_C \) is as follows.
Fig. II-6 Equivalent reversible polytropic approximation for compressor
\[ x_c = \left( \frac{|q_+| + |q_-|}{2 \times \Delta H} \right), \quad (\text{II-6}) \]

where \( \Delta H = \dot{m}_f c_p (T_{out} - T_{in}) \). Table II-3 lists \( x_c \) for engine tests using Equivalent Reversible Polytropic method. Appendix C shows the examples of calculation.

Table II-3  Magnitude of Cyclic Heat Transfer (Equivalent Reversible Polytropic Method)

<table>
<thead>
<tr>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>( x_c )</td>
<td>1.71</td>
<td>1.45</td>
<td>1.69</td>
<td>1.47</td>
<td>0.99</td>
</tr>
</tbody>
</table>

The accuracy of equivalent-reversible-polytropic method was confirmed by a first-law integral analysis to be described in detail in section III.3. This analysis uses instantaneous mass flow rate data together with instantaneous pressure and volume data to calculate the instantaneous mixed mean temperature of the gas in the cylinder. The heat transfer is calculated by numerical integration of the first law. For the compressor data of Test #5, the difference between the reversible polytropic method and the first-law integral analysis was less than 1%. The data of Table II-2 and II-3 show that the much simpler Equivalent Isentropic method gives almost as accurate a result as the Equivalent Reversible Polytropic method, except for Test #1 and #2 with piston ring leakage.
4. Influence of Cyclic Heat Transfer on VHGE Performance

1) Adiabatic efficiency vs. polytropic efficiency

Conventionally, compressors are modeled as adiabatic or at least overall-adiabatic. Thus, adiabatic efficiency $\eta_c$ is usually defined as

$$\eta_c = \frac{\dot{W}_{sc}}{\dot{W}_c} \quad \text{(Isentropic compressor power)}$$

where $\dot{W}_{sc}$ and $\dot{W}_c$ are for the same mass flow rate, inlet state, discharge pressure referring to Fig. II-4. For an ideal gas with constant specific heats,

$$\eta_c = \frac{h_2 - h_1}{h_2 - h_1} = \frac{T_2 - T_1}{T_2 - T_1} \quad \text{(II-8)}$$

where $h$ is specific enthalpy and $s$ denotes isentropic process. However, when the compressor has net heat transfer $\dot{Q}_c$, equation II-8 must be modified to include $\dot{Q}_c$.

$$\eta_c = \frac{h_2 - h_1}{(h_2 - h_1) - \dot{Q}_c/\dot{m}_f} \quad \text{(II-9)}$$

Therefore, in a non-adiabatic case for given initial state and compression pressure ratio, $\eta_c$ can give the value of $\dot{W}_c$, but $\dot{Q}_c$ is also required to fix the discharge state.
The polytropic efficiency of a compressor $\eta_{nc}$, and the polytropic coefficient $\xi_c$ are introduced to describe the non-adiabatic compressor.

The adiabatic efficiency of a compressor may be separated into two parts,

$$\eta_c = \frac{\dot{W}_{sc}}{\dot{W}_c} = \frac{\dot{W}_{sc}}{\dot{W}_{nc}} \frac{\dot{W}_{nc}}{\dot{W}_c} = \frac{(1-P_r^{C_1})}{\xi_c(1-P_r^{C_0})} \eta_{nc}$$  \hspace{1cm} (II-10)

where $\xi_c \equiv$ polytropic coefficient of compressor $= \frac{\gamma-1}{\gamma} \frac{\log P_r}{\log T_{rc}}$,

$\eta_{nc} \equiv$ polytropic efficiency of compressor $= (\dot{W}_{nc}/\dot{W}_c)$,

$\dot{W}_{nc} \equiv$ reversible polytropic compressor power

$\dot{W}_{sc} = \dot{W}_c \xi_c \dot{m}_f c_p T_L (1-P_r^{C_1})$.

$P_r \equiv P_h/P_L$ ; $T_{rc} \equiv T_2/T_L$

$C_0 \equiv (\gamma-1)/\gamma$ ; $C_1 \equiv C_0/\xi_c$.

The first term, $\dot{W}_{sc}/\dot{W}_{nc}$, reflects the effect of reversible net heat transfer following the path 1-2 in Fig. II-4. The second term, $\dot{W}_{nc}/\dot{W}_c = \eta_{nc}$ reflects the effect of irreversible cyclic heat transfer. The net heat transfer rate $\dot{Q}_c$ is given in terms of $\xi_c$ and $\eta_{nc}$ as,

$$\dot{Q}_c = (\xi_c/\eta_{nc} - 1) \dot{m}_f c_p T_1 (1-P_r^{C_1}).$$  \hspace{1cm} (II-11)
When $\dot{Q}_c = 0$, i.e. at least overall adiabatic, $\xi_c$ is equal to the polytropic efficiency $\eta_{nc}$. In fact, $\xi_c$ is called the polytropic efficiency in adiabatic turbines or compressors (8). In non-adiabatic case, the newly defined $\eta_{nc}$, not $\xi_c$, is more appropriate to be called polytropic efficiency. Thus, prescribing $\xi_c$ and $\eta_{nc}$ gives both $\dot{W}_c$ and the end state for a given initial state and pressure ratio.

Polytropic efficiency of expander $\eta_{ne}$ and polytropic coefficient of expander $\xi_e$ are similarly defined:

$$\eta_e = \frac{\dot{W}_e}{\dot{W}_{se}} = \frac{\dot{W}_{ne}}{\dot{W}_{ne}} \cdot \xi_e \frac{(1-P_r^{C_2})}{(1-P_r^{C_0})} \cdot \eta_{ne} \quad (\Pi - 12)$$

where $\xi_e = \frac{(\gamma - 1)}{\gamma} \cdot \frac{\log P_r}{\log T_{re}}$; $T_{re} = T_h / T_5$.

$$\eta_{ne} = \frac{\dot{W}_e}{\dot{W}_{ne}} \quad ; \quad C_2 = C_0 / \xi_e,$$

$\dot{W}_{ne} = \text{reversible polytropic expander power}$

$$= \xi_e \dot{m}_c P T_h (1-P_r^{C_2}).$$

The net heat transfer of expander can be given in terms of $\xi_e$ and $\eta_{ne}$ as,

$$\dot{Q}_e = (\xi_e \eta_{ne} - 1) \dot{m}_c P T_4 (1-P_r^{C_2}). \quad (\Pi - 13)$$

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2) Simple Cycle Analysis of VHGE

Fig. II-4 shows the thermodynamic processes inside the engine as the working gas circulates through the components of the engine. Indicated engine efficiency, $\eta_i$, is defined as $\eta_i = \frac{\text{net indicated power of engine}}{\text{net heat input rate of engine}}$. Net indicated power of the engine is the sum of compressor indicated power $\dot{W}_c$ and expander indicated power $\dot{W}_e$.

From definitions of $\xi_c$, $\xi_e$, $\eta_{nc}$, $\eta_{ne}$, $\dot{W}_c$ and $\dot{W}_e$,

$$
\dot{W}_c = \dot{W}_{nc} / \eta_{nc} = (\xi_c / \eta_{nc}) \dot{m}_{fc} c_p T_L (1 - P_r^C) \tag{II-14}
$$

$$
\dot{W}_e = \dot{W}_{ne} / \eta_{ne} = (\xi_e \eta_{ne}) \dot{m}_{fc} c_p T_h (1 - P_r^C) \tag{II-14}
$$

Net heat input rate, $\dot{Q}_{\text{net}} = \dot{m}_{fc} c_p (T_h - T_3)$. After some algebraic manipulations,

$$
\dot{Q}_{\text{net}} = \dot{m}_{fc} c_p (T_h - T_L P_r^C + \eta_R (T_L P_r^C - T_h P_r^C)) \tag{II-15}
$$

The indicated efficiency $\eta_i$ is,

$$
\eta_i = \frac{(\xi_c / \eta_{nc}) (1 - P_r^C) + \xi_e \eta_{ne} T_r (1 - P_r^C)}{T_r - P_r^C + \eta_R (P_r^C - T_r P_r^C)} \tag{II-16}
$$

where $T_r = T_h / T_L$; $\eta_R \equiv$ regenerative heat exchanger effectiveness.

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3) Engine performance vs. cyclic heat transfer

Table II-4 lists the polytropic coefficient of the compressor $\xi_c$ and of the expander $\xi_e$, the polytropic efficiency of the compressor $\eta_c$, of the expander $\eta_e$, and indicated engine efficiency $\eta_i$ based on net input power to the engine for Tests #1 through #6.

Table II-4 Polytropic Coefficients ($\xi_c$, $\xi_e$), Polytropic Efficiencies ($\eta_{nc}$, $\eta_{ne}$), and Indicated Engine Efficiency $\eta_i$ for Engine Tests

<table>
<thead>
<tr>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\xi_c$</td>
<td>0.924</td>
<td>0.925</td>
<td>0.999</td>
<td>1.120</td>
<td>1.140</td>
</tr>
<tr>
<td>$\xi_e$</td>
<td>1.041</td>
<td>0.906</td>
<td>0.933</td>
<td>0.926</td>
<td>0.938</td>
</tr>
<tr>
<td>$\eta_{nc}$</td>
<td>0.597</td>
<td>0.706</td>
<td>0.698</td>
<td>0.740</td>
<td>0.804</td>
</tr>
<tr>
<td>$\eta_{ne}$</td>
<td>0.963</td>
<td>0.887</td>
<td>0.914</td>
<td>0.932</td>
<td>0.860</td>
</tr>
<tr>
<td>$\eta_i$</td>
<td>20.8</td>
<td>25.9</td>
<td>27.7</td>
<td>33.1</td>
<td>29.8</td>
</tr>
</tbody>
</table>

From Table II-2, II-3, and II-4, polytropic efficiencies $\eta_c$ and $\eta_e$ can be plotted with respect to the non-dimensional magnitude of the cyclic heat transfer of the compressor $X_c$ and of the expander $X_e$ as shown in Fig. II-7. By definition, $\eta_{nc}$ and $\eta_{ne}$ become 100% when cyclic heat transfer goes to zero. In the expander case, polytropic efficiency $\eta_{ne}$ is related to
Fig. II-7 Polytropic efficiencies of compressor and expander vs. Cyclic heat transfer.
the non-dimensional magnitude of cyclic heat transfer \( X_e \) by the linear-regression relation

\[ \eta_{ne} = 1 - 0.310 \, X_e. \]  

(II-17)

In the compressor case, polytropic efficiency \( \eta_{nc} \) is related to the non-dimensional magnitude of cyclic heat transfer \( X_c \) by the linear regression relation

\[ \eta_{nc} = 1 - 0.177 \, X_c. \]  

(II-18)

The significant deviation of Test #1 and Test #2 from the linear-regression relations is believed to be due to the effect of piston-ring leakage on compressor performance. The serious effect of cyclic heat transfer on compressor and expander efficiencies is clear in Fig. II-7.

The influence of cyclic heat transfer on the indicated engine efficiency \( \eta_i \) is shown in Fig. II-8, which was plotted from equation II-16, with the aid of the linear-regression relations of equations II-17, and II-18.
Fig. II-8 Indicated engine efficiency vs. cyclic heat transfer
Chapter III  DETAILED STUDY AND MEASUREMENT OF INTERNAL STATES INSIDE COMPRESSOR

The need for more information on internal states arose when we could not determine the cause of low efficiency from indicated P-V diagram and average temperatures measured outside the cylinder. For complete specification of the thermodynamic state of cylinder at a given time, temperature and mass of cylinder gas were also needed. Transient gas temperature was measured by fast-response thermocouples inserted into the cylinder. Although the information thus obtained was very useful in understanding internal states changes, there was a question as to how closely the local temperatures read by thermocouples approximate the mixed mean temperature of the gas inside cylinder. Most of the time, the spatial variation of gas temperature inside cylinder makes it difficult to accept the thermocouple readings even after corrections for errors, unless there are enough of them distributed throughout the cylinder space. Since this is not practical for a reciprocating engine due to sweeping motion of the piston, an indirect method utilizing instantaneous mass flow rate was used. The measurement of instantaneous mass flow rate serves two purposes: (a) confirmation of cooler flow rate as the engine flow rate, (b) calculation of the mixed mean temperature of cylinder gas.
1. Transient Gas Temperature Measurement

1) Thermocouple-probe design

To measure the temperature fluctuation during each cycle inside the cylinder, it is necessary to design a special thermocouple probe which can withstand up to 3.8 MPa, 800 K, and maximum gas velocity of 50m/sec. Chromel-alumel wire was chosen since it has suitable temperature-emf characteristics and durability in such an environment. Since the characteristic thermal response time $\tau$ of a thermocouple decreases as the diameter of the wire gets smaller, 25.4 $\mu$m O.D. wire was used for the hot end of the thermocouple junction. Lead wire of 254$\mu$m O.D. was welded to the junction wire. The two wires were separated by a two-hole alumina insulation tube which was inserted into 4.76 mm O.D. Inconel 600 protection tube. At the hot end of the thermocouple, high-temperature cement was used to support the thermocouple junction. To prevent the working gas from leaking through the thermocouple probe, a compression fitting with a Teflon seal was soldered to the external end of the protection tube and cooled to protect the Teflon seal from being overheated. The final assembly of the probe is shown in Fig. III-1.

There are four kinds of error to be considered in thermocouple design: velocity, conduction, radiation, and transient error. If the environmental effects are small enough so that one effect has little influence on others, i.e. if they are
Fig. III-1 Thermocouple probe assembly
Drawing not to Scale
linear to a first-order approximation, we can estimate the errors separately and add them together \((9)\). The velocity error is given by

\[
\Delta T_v = T_g \frac{(1-\alpha)}{(\gamma-1)M^2/ (2+ (\gamma-1)M^2)}, \quad (\text{III-1})
\]

where \(\alpha\) \equiv \text{recovery factor} and \(\gamma \equiv c_p/c_v\), \(M \equiv \text{Mach number}\). For a given \(\alpha\), as long as \(M\) is low, \(\Delta T_v\) remains small. The conduction error is given by

\[
\Delta T_c = \frac{T_g - T_m}{\cosh (2(L/d) (hd/k_s)^{0.5})}, \quad (\text{III-2})
\]

where \(L = \text{length of the wire exposed to gas stream}\), \(d = \text{wire diameter}\), \(k_s = \text{average thermal conductivity of the two wires}\), which means, \(\Delta T_c\) will decrease as \(L/d\) and \((hd/k_s)\) increases. As will be shown later, \(L/d\) is limited by the stiffness of the wire. Convective-heat-transfer coefficient \(h\) is given by \((10)\):

\[
Nu \equiv \frac{(hd/k_g)}{Pr_g} = (Pr_g)^{0.3} \left( 0.35 + 0.56 (Re_d)^{0.52} \right), \quad (\text{III-3})
\]

where subscript \(g\) denotes the gas. The radiation error is given by

\[
\Delta T_r = (Fg/h) \left( T_j^4 - T_w^4 \right) \quad (\text{III-4})
\]

where the radiation view factor \(F\) is, in this case, the emissivity \(\varepsilon\) of the thermocouple junction, \(T_j\) is the junction temperature, and \(T_w\) is the temperature of the surrounding walls.
The only practical method of reducing the radiation error is to choose a wire of the smallest possible diameter, which gives a high value of \( h \). Transient error is given by

\[
\Delta T_t = \zeta_c \frac{dT_j}{dt}
\]  

(III-5)

For a junction with a bead of diameter \( D \),

\[
\zeta_c = (D/d)^{0.375} \tau_0
\]  

(III-6)

where \( \tau_0 = (\rho cv/hA) = (\rho cd/4h) \) (for a butt-welded junction).

The drag force exerted by the gas on the thermocouple is given by

\[
F_d = C_d A_d \rho_g v^2 / 2g_c
\]  

(III-7)

where \( A_d = (\pi D^2/4) \) for a sphere (bead), \( A_d = d \times L \) for a wire, \( C_d \) = drag coefficient, which is a function of Reynolds number \( R_{ed} \) and the shape of the object, \( \rho_g \) is the density of the gas. Maximum bending stress occurs at the foot of the thermocouple wire and is given by

\[
\sigma_{max} = M_{max} d / (2I)
\]  

(III-8)

where maximum bending moment \( M_{max} = (F_{bead} + 0.5F_{wire}) L \), \( L \) = length of the thermocouple exposed to the gas stream, \( d \) = diameter of the wire, \( I \) = moment of inertia of the cross section at the foot of the wire. Maximum shear stress also
occurs at the foot of the thermocouple and is given by

$$\tau_{\text{max}} = \left(\frac{4}{3A}\right) (F_d \text{ wire} + F_d \text{ bead})$$

where $A_w$ is the cross section area of the wire.

Since the gas velocity and the temperature at the expander inlet valve are the highest among the four thermocouple location, we use the condition at that point. Given the average mass flow rate of $7.56 \times 10^{-3}$ kg/sec, flow area $4.55 \times 10^{-4}$ m$^2$, the speed 10 rev/sec, temperature 900 C, and pressure 3.79MPa, the velocity, and radiation errors are negligible. Conduction error also becomes negligible when L/d is about 20. In this case, the maximum bending stress, $\sigma_{\text{max}}$, is about 0.6 MPa, and the maximum shear stress $\tau_{\text{max}}$ is about 0.14 MPa. According to Mises yield criterion (11), these stress levels are low enough for the thermocouple to withstand the drag force exerted by the gas stream at moderate temperatures. Later in the tests, expander-inlet-side thermocouples were bent over when the inlet temperature exceeded 950 K. The characteristic time was calculated to be about 1m-sec at the expander inlet valve side when the valve was wide open, which is short enough for our purpose.

Drilling holes to insert the thermocouple probes into the engine required a special care because there was a possibility of dropping the chips into the cylinder. For the expander intake valve port, a plexiglass plug was inserted through the
valve-access port. The plexiglass plug was pressed against the wall, and a hole was drilled through the engine wall into the plug, leaving the chips inside the plug. For the expander discharge side, a hole was drilled through the closure plug of the discharge-valve access port.

Before the modification of compressor, thermocouple ports into the compression chamber were provided by drilling axially along the wall of the cylinder used for controlling the clearance volume. Plexiglass plugs were again used to prevent chips from entering the cylinder. The thermocouples are located right downstream of the inlet valves and right upstream of the discharge valves. The results are given in App. A for Tests #1 through #6. The result from Test #5 will be later compared to the mixed mean temperature obtained indirectly by measuring instantaneous mass flow rate (see section III.3).
2. Instantaneous-Mass-Flow-Rate Measurement

1) Flow-meter design

Two square-edged orifice plates were made of S.S.300. The diameters of them were properly sized* so that at maximum-flow condition, the ΔP across the orifice would not exceed 0.1 MPa. Two±0.1 MPa differential-pressure transducers (unbonded-strain-gauge type) were used to record the differential pressure across the orifice. Special adapters for the transducers were made of S.S.300 as shown in Fig. III-2. The tubings before and after the orifices were arranged to ensure enough straight runs satisfying the requirement prescribed in ref. 6 (see Fig. III-3, and III-4).

As mentioned in section II. 2.2), accurate measurement of gas flow using orifice meter requires that the flow be quasi-steady. According to reference 6, when the flow pulsating frequency is less than one fifth that of the flow-metering device, the pulsation is slow enough to be considered quasisteady.

Fig. III-5 and III-6 shows ΔP from intake orifice and discharge orifice with respect to crank angle in Test #5. The high-frequency noise at about 1400 Hz is thought to be due to resonance of the adapter system. The flow pulsation around 300 Hz is thought to be due to the movement of compressor valves. Flow-pulsation frequency due to engine speed is 10 Hz. Considering the natural frequency of the pressure transducer

* See Appendix D.
Fig. III-2 Differential-pressure transducer and its adapter for orifice metering (schematic)
Fig. III-4  Gas line arrangement for the orifice meter (discharge side)
Fig. III-5 Pressure drop across intake orifice vs. crank angle
Fig. III-6 Pressure drop across discharge orifice vs. crank angle
was 6000 Hz, the flow condition can be assumed quasisteady. The error due to pulsation is estimated to be at most 1~2% (see Appendix D).

One difficulty with the orifice-ΔP data was the zero shift during the measurement as shown in Fig. III-5 and Fig. III-6. The zero shift of this kind in much larger scale had been previously observed during compression-expansion test of the expander without any gas flow through the valves.

Assuming that when the compressor valves are closed, the gas flow across the orifice is almost negligible, or at least the net flow is negligible, the zero reference for the orifice-pressure-drop data was drawn using line segments as shown in Fig. III-5 and Fig. III-6. Orifice-ΔP data thus obtained were used in the computer program to calculate instantaneous mass flow rate and mixed mean gas temperature of the compressor in section III. 3.

2) Results

The mass flow rate was first calculated using the orifice-ΔP data as it was (Fig. III-5, III-6), and then recalculated using the data after smoothing out 1400 Hz noise (due to adapter resonance) numerically (Fig. E-4, Fig. E-5). Both methods gave almost the same mass flow rate of $7.60 \times 10^{-3}$ kg/sec during the intake process, and $7.71 \times 10^{-3}$ kg/sec during the discharge process.

* See Appendix A.
Considering the error caused by pulsation and temperature change across the orifice, the intake-side flow rate is more accurate than the discharge-side flow rate to be regarded as engine flow rate. The calculated cooler flow rate had been regarded as engine flow rate from energy-balance considerations. It turned out to be very close to the measured engine flow rate (99.5% of intake orifice measurement).

3. First-Law Integral Analysis

1) Formulation

Ignoring changes in kinetic energy and potential energy associated with mass fluxes, the First Law of thermodynamics for control volume (compressor space) is

\[
\frac{\partial E}{\partial t}_{c.v.} = \dot{Q}_{c.v.} - \dot{W}_{c.v.} + \sum_{\text{in}} \dot{m}_{\text{h}} - \sum_{\text{out}} \dot{m}_{\text{h}},
\]

(III-10)

where \(\dot{Q}_{c.v.}\), \(\dot{W}_{c.v.}\), \(\frac{\partial E}{\partial t}_{c.v.}\) are the rate of heat transfer, work transfer, and change in internal energy of the control volume respectively; \(\sum_{\text{in}} \dot{m}_{\text{h}}\), \(\sum_{\text{out}} \dot{m}_{\text{h}}\) are enthalpy-flux terms due to mass in-flow and out-flow respectively. It is assumed that mass inside the tube spaces between the compressor and orifices remain constant with respect to time. Thus, with appropriate phase lag adjustment, the mass flow rates through the compressor valves and orifices are the same.

In finite difference form between time \(t_i\) and \(t_{i+1}\), the above equation becomes as follows:
\[ E_{i+1} - E_i = Q_{i+1,i} - W_{i+1,i} + (\Delta H_{i+1,i})_{\text{in}} - (\Delta H_{i+1,i})_{\text{out}} \]

where \( \Delta H_{i+1,i} = \int_{t_i}^{t_{i+1}} m \cdot h \, dt \);
\[ W_{i+1,i} = \frac{(P_i + P_{i+1})(V_{i+1} - V_i)}{2} \]

(III-11)

If the working gas (helium) is assumed to be an ideal gas with constant specific heats,

\[ E_{i+1} - E_i = m_{i+1} c_v T_{i+1} - m_i c_v T_i, \]

(III-12)

\[ \Delta H_{i+1,i} = \int_{t_i}^{t_{i+1}} m \cdot c_p \cdot T \, dt, \]

where temperatures \((T, T_i, T_{i+1})\) are mixed mean temperatures.

During the intake process, assuming intake temperature is constant \(T_{\text{in}}\),

\[ (\Delta H_{i+1,i})_{\text{in}} = (m_{i+1} - m_i) c_p T_{\text{in}} \]

(III-13)

\[ (\Delta H_{i+1,i})_{\text{out}} = 0. \]

During the discharge process, assuming discharge temperature is equal to the mixed mean temperature,

\[ (\Delta H_{i+1,i})_{\text{in}} = 0 \]

(III-14)

\[ (\Delta H_{i+1,i})_{\text{out}} = (m_{i+1} - m_i) c_p (T_{i+1} + T_i) / 2 \]

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During the compression and reexpansion processes,

\[ \Delta H_{i+1,i}^\text{in} = 0, \]

\[ \Delta H_{i+1,i}^\text{out} = 0. \]  \hspace{1cm} (III-15)

The equation of state for ideal gas \( PV = nRT \) is used to relate \( P, V \), and mixed mean temperature \( T \) of compressor gas. Specific entropy of the gas is approximated by

\[ s = s_{\text{ref.}} + c_p \log \left( \frac{T}{T_{\text{ref.}}} \right) - R \log \left( \frac{P}{P_{\text{ref.}}} \right) \]  \hspace{1cm} (III-16)

where \( \text{ref.} \) means reference state.

2) Implementation

Pressure and volume in equation III-11 are given from the pressure, volume - crank angle data (Fig. A-8) recorded by oscillograph. The instantaneous mass flow rates measured by orifice meters give the changes in cylinder mass \( m_{i+1} - m_i \) during intake and discharge processes.

From the equation of state and equation III-11, knowing \( P_i, P_{i+1}, V_i, V_{i+1}, m_{i+1} - m_i, T_{\text{in}} \) and \( T_{\text{out}} \), the heat transfer \( Q_{i+1,i} \) can be calculated. In order to calculate the mixed mean temperature also, one more information is needed either on \( m_i \) or \( T_i \) at any time \( t_i \). Specifying either one would fix \( m_i \) and \( T_i \) for the entire cycle, because \( m_i T_i \) is equal to \( P_i V_i / R \), which
is given by pressure volume data.

It was decided to use one data point from the transient gas temperature measurement. The temperature at the start of compression (end of intake) was chosen from Fig. A-14 as the mixed mean gas temperature at that instant. It was thought that spatial temperature variation inside the compressor would be minimum at this instant being preceded by a mixing period during intake process.

The resulting mixed mean temperature is shown in Fig. E-6, (curve A). Fig. E-10 shows the heat-transfer rate (kW) vs. crank angle calculated from equation using the mixed mean temperature of curve A in Fig. E-6.

In Fig. E-10, the sudden change in heat-transfer direction during discharge and intake of curve A is artificially caused by the mismatch between the slope of input-pressure data and the mass flow rate. The First Law during intake and discharge (Equation III-17 and III-18) shows that the heat-transfer rate depends on the slope of pressure and temperature.

\[
\dot{Q}_{\text{discharge}}^* = -V \frac{\partial P}{\partial \theta} + m c_p \frac{\partial T}{\partial \theta} \tag{III-17}
\]

\[
\dot{Q}_{\text{intake}} = -V \frac{\partial P}{\partial \theta} + \frac{\partial}{\partial \theta} \left( m c_p (T - T_{\text{in}}) \right) \tag{III-18}
\]

The specific entropy s of the gas from equation III-16 will be used to reduce the mismatch mentioned above. Fig. E-9 shows the specific entropy s vs. crank angle before and after the * assume \( T_{\text{discharge}} = \) mixed mean temperature

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numerical curve smoothing. Using the specific entropy $s$ of the smoothed curve, the new mixed mean temperature can be calculated from equation III-16 which is shown in Fig. E-6 (curve B) along with the thermocouple recordings without any corrections for errors. Taking into account the transient error due to finite response time (minimum 1 m sec during intake, section III.1), the agreement between the measured local temperatures and the calculated mixed mean temperature seems to reasonable. The new mixed mean temperature decreases the degree of mismatch between $P,V$ and $T$ data, and resulting heat transfer (curve B, Fig. E-10) bears this out. The flow chart in Fig. E-1 is the summary of the procedures in this section.

3) Results

Data from Test #5 were used in the computer program listed in Appendix E. Some of the results are discussed here.

a) The engine mass flow rate from the orifice-meter measurement ($7.60 \times 10^{-3}$ kg/sec) confirmed the accuracy of the calculated cooler flow rate ($7.56 \times 10^{-3}$ kg/sec) as the engine flow rate. This is very important for energy balance of the engine (Appendix B) and gives more reliability to cyclic heat-transfer estimates from both the compressor and the expander.

b) Step-by-step integration of the First Law for the Compressor gives cyclic heat-transfer rate of 3.24 kW which is very close to 3.23 kW from Equivalent Reversible Polytropic method.

c) The following is observed from heat-transfer rate vs. crank angle plot (Fig. E-10). The fact that the heat transfer is
out of the gas into the wall throughout the compression process is quite unusual considering the fact that the wall temperature is supposed to be higher than the mixed mean gas temperature in the beginning of compression. This fact can be explained as follows. At the end of intake, the gas layer near the wall is in thermal equilibrium with the wall. So, as soon as compression starts, the temperature of the gas layer starts increasing above the wall temperature. This causes heat transfer from the gas layer into the wall, even though the mixed mean gas temperature at that moment is lower than wall temperature.

During reexpansion (about 100° crank angle duration), it takes a while (about 20° crank angle, 5.5 msec) for the temperature of the gas layer near the wall to decrease below wall temperature due to expansion causing heat transfer into the gas. The 5.5 msec delay indicated that at the end of discharge the gas layer near the wall was not in thermal equilibrium with the wall.

From above observations, we can say that the mixed mean temperature is not a good representative gas temperature in calculating the instantaneous heat transfer inside cylinder when there is a significant spatial temperature variation, especially in the radial direction.
CONCLUSIONS

The low performance of the VHGE has been primarily the result of irreversible cyclic heat transfer between the gas and the cylinder walls inside the expander and compressor. This heat transfer has a large effect on the mass per cycle and a small effect on the indicated P-V diagram and the indicated work per cycle.

The observed loss \((1-\eta)\) in the expander and the compressor are linearly related to the observed magnitude of the cyclic heat transfer. The cyclic heat transfer in the VHGE compressor was significantly reduced by changes in the wall-temperature distribution and by changes in the internal geometric configuration of the compressor and valves.

The cyclic heat transfer in a reciprocating expander or compressor can be calculated from observation of inlet temperature, discharge temperature, the mass per cycle and the indicated P-V diagram.

Further improvement in the performance of the VHGE will require additional reduction in the cyclic heat transfer in both the compressor and the expander. A reduction to about one half the present level will be required to achieve the performance levels originally projected (about 47% indicated efficiency).
REFERENCES


5. B.C. Fryer and J.L. Smith, Jr., "Design Construction and Testing of a New Valved Hot-Gas Engine", Proc. of 8th Inter-


10. W.M. Rohsenow and H.Y. Choi, "Heat, Mass and Momentum

Appendix A  ENGINE TEST DATA

The engine tests are identified as follows:

Test #1. Brent Fryer's Test (ref.1, 1972)
Test #2. First Cooled Compressor Head Test (1975)
Test #3. First Rubber O Ring Test (1976)
Test #4. After Compressor Modification (1977)
Test #5. After Compressor Modification (1977)
Test #6. After Compressor Modification (1978)

1. Pressure and Volume Data

Cylinder pressures are measured by two unbonded strain
gauge type pressure transducers, and recorded on oscilloscope
pictures and/or oscillograph recording paper.

Silicon rubber was coated (about 1 mm) on the diaphragm of
pressure transducers to decrease excessive zero shift due to
temperature. In the expander case, the pressure transducer
was water-cooled. The volume was measured by a displacement
transducer whose plunger was in contact with a cam that was
mounted on dynamometer axis and was shaped to convert the crank
angle into the piston displacement.*

Fig. A-1, 2, 3, 4, 5 show indicated P-v diagrams of the
compressor and the expander for Tests #2, #3, #4, #5, #6 respec-
tively. Fig. A-6, 7, 8, 9 show pressure-crank angle plot of
the compressor and expander for Tests #3, #4, #5, #6 respectively.
* See reference 3.

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**Compressor Volume** $(10^{-4} \text{ m}^3)$

<table>
<thead>
<tr>
<th></th>
<th>8.741</th>
<th>2.950</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>3.55</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Expander</td>
<td></td>
<td></td>
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<tr>
<td>Pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(MPa)</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>0.876</th>
<th>6.992</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1.70</strong></td>
<td></td>
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<tr>
<td>Expander</td>
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<td></td>
</tr>
<tr>
<td>Pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(MPa)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Expander Volume** $(10^{-4} \text{ m}^3)$

Fig. A-1  Indicated P-V diagrams (Test #2)
Fig. A-2 Indicated P-V diagrams (Test #3)
Compressor Volume \( (10^{-4} m^3) \)

10.216  4.425

Pressure (MPa)

3.24

1.56

0.876  6.992

Expander Volume \( (10^{-4} m^3) \)

Fig. A-3  Indicated P-V diagrams (Test #4)
Fig. A-4  Indicated P-V diagrams (Test #5)
Fig. A-5 Indicated P-V diagrams (Test #6)
Fig. A-6 P-θ diagram for expander and compressor - Test #3
Fig. A-7  P-0 diagram for expander and compressor - Test #4
Fig. A-8  P-θ diagram for expander and compressor - Test #5
2. Temperature Data

1) Average temperatures

Multi-channel recorders were used to record average gas temperatures outside the cylinder and average surface temperatures of engine components. Fig. A-10 shows the locations of thermocouples recorded by multi-channel recorders. Table A-1 lists the recorded temperatures from Test #2 to #6.

2) Transient temperature data

The Chromel-Alumel thermocouples were used for the compressor and Platinum-Rhodium thermocouples were used inside expander. Of the two thermocouples inside the expander, the one in the intake port was bent over during Test #3. However, it gave us valuable information about heat transfer during the intake process. Fig. A-11 is the temperature volume plot of Test #2. Fig. A-12, 13, 14, 15 show the temperature fluctuation inside the compressor with respect to crank angle for Tests #3, 4, 5, 6 respectively. Fig. A-16 shows the significant temperature difference between the readings from the intake side and the discharge side thermocouple of the expander before the intake side thermocouple failed in Test #3. Fig. A-17, 18, 19, 20 show the expander temperature fluctuation with respect to crank angle measured by discharge side thermocouple for Tests #3, 4, 5, 6 respectively.
Fig. A-10 Locations of thermocouples connected to multirecorder
(a) Compressor Discharge and Intake

(b) Expander Discharge

Fig. A-11 Transient gas temperatures (Test #2)
Fig. A-12 Compressor gas temperature vs. crank angle
(Test #3)
Fig. A-13 Compressor gas temperature vs. crank angle
(Test #4)
Fig. A-14  Compressor gas temperature vs. crank angle
(Test #5)
Fig. A-15 Compressor gas temperature vs. crank angle (Test #6)
Fig. A-16  Comparison of the temperature readings from expander intake side and discharge side thermocouple (Test #3)
Fig. A-18  Expander gas temperature vs. crank angle
(Test #4)
Fig. A-19  Expander gas temperature vs. crank angle
(Test #5)
Fig. A-20  Expander gas temperature vs. crank angle
(Test #6)
Table A-1  Average Temperatures (°C) from Multi-channel Recorders

<table>
<thead>
<tr>
<th>T.C.No</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>150</td>
<td>137</td>
<td>116</td>
<td>120</td>
<td>112</td>
</tr>
<tr>
<td>2</td>
<td>124</td>
<td>112</td>
<td>86</td>
<td>89</td>
<td>81</td>
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<tr>
<td>3</td>
<td>459</td>
<td>452</td>
<td>451</td>
<td>430</td>
<td>439</td>
</tr>
<tr>
<td>4</td>
<td>422</td>
<td>411</td>
<td>408</td>
<td>389</td>
<td>398</td>
</tr>
<tr>
<td>5</td>
<td>136</td>
<td>124</td>
<td>94</td>
<td>96</td>
<td>87</td>
</tr>
<tr>
<td>6</td>
<td>655</td>
<td>685</td>
<td>687</td>
<td>651</td>
<td>668</td>
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<tr>
<td>7</td>
<td>686</td>
<td>687</td>
<td>698</td>
<td>653</td>
<td>675</td>
</tr>
<tr>
<td>8</td>
<td>724</td>
<td>731</td>
<td>720</td>
<td>684</td>
<td>699</td>
</tr>
<tr>
<td>9</td>
<td>613</td>
<td>602</td>
<td>604</td>
<td>574</td>
<td>586</td>
</tr>
<tr>
<td>10</td>
<td>463</td>
<td>433</td>
<td>433</td>
<td>416</td>
<td>416</td>
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<tr>
<td>11</td>
<td>552</td>
<td>532</td>
<td>522</td>
<td>505</td>
<td>516</td>
</tr>
<tr>
<td>12</td>
<td>446</td>
<td>379</td>
<td>351</td>
<td>356</td>
<td>363</td>
</tr>
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<td>13</td>
<td>155</td>
<td>153</td>
<td>149</td>
<td>-</td>
<td>-</td>
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<tr>
<td>14</td>
<td>72</td>
<td>68</td>
<td>27</td>
<td>38</td>
<td>31</td>
</tr>
<tr>
<td>15</td>
<td>38</td>
<td>42</td>
<td>18</td>
<td>18</td>
<td>12</td>
</tr>
<tr>
<td>16</td>
<td>136</td>
<td>127</td>
<td>-</td>
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<td>-</td>
</tr>
<tr>
<td>17</td>
<td>618</td>
<td>649</td>
<td>639</td>
<td>616</td>
<td>627</td>
</tr>
<tr>
<td>18</td>
<td>531</td>
<td>462</td>
<td>464</td>
<td>439</td>
<td>451</td>
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<td>19</td>
<td>24</td>
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<tr>
<td>20</td>
<td>422</td>
<td>407</td>
<td>411</td>
<td>392</td>
<td>402</td>
</tr>
<tr>
<td>21</td>
<td>150</td>
<td>141</td>
<td>120</td>
<td>123</td>
<td>116</td>
</tr>
</tbody>
</table>
3. Cooling Water Data

Up to Test #2, thermometers of low resolution (maximum resolution error 0.5 C when water temperature drop in the cooler was about 5 C). were used. From Test #3, high resolution thermometers (maximum resolution error 0.05 C) were used. In order to increase the reliability of the water temperature measurement further, special receptacles (Fig. A-21), which will provide enough immersion depth, were fabricated and used in the measurements to minimize the thermometer error due to conduction through the thermometer stem.

Table A-2, 3 list the cooling water heat removal rates in Test #2, #3 respectively. Table A-4 lists the cooling water heat removal rates for Tests #4, #5, and #6.

<table>
<thead>
<tr>
<th>Table A-2  Cooling Water Heat Removal Rates (kW) - Test #2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooler</td>
</tr>
<tr>
<td>Piston Rod Seal &amp; Cyl. Jacket</td>
</tr>
<tr>
<td>Compressor Block (Jacket)</td>
</tr>
<tr>
<td>Expander Head</td>
</tr>
<tr>
<td>Heater</td>
</tr>
<tr>
<td>Total</td>
</tr>
</tbody>
</table>
Fig. A-21 Receptacle for a thermometer (schematic)
### Table A-3  Cooling Water Heat Removal Rates (kW) - Test #3

<table>
<thead>
<tr>
<th>Component</th>
<th>Rate (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooler</td>
<td>4.37</td>
</tr>
<tr>
<td>Cylinder Jacket</td>
<td>1.42</td>
</tr>
<tr>
<td>Compressor Block</td>
<td>0.92</td>
</tr>
<tr>
<td>Piston Rod Seal</td>
<td>1.07</td>
</tr>
<tr>
<td>Expander Head</td>
<td>0.43</td>
</tr>
<tr>
<td>Heater</td>
<td>0.05</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>8.26</strong></td>
</tr>
</tbody>
</table>

### Table A-4  Cooling Water Heat Removal Rates (kW) - Test #4, 5, 6

<table>
<thead>
<tr>
<th>Component</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooler</td>
<td>3.14</td>
<td>4.19</td>
<td>4.12</td>
</tr>
<tr>
<td>Cylinder Jacket</td>
<td>1.85</td>
<td>2.08</td>
<td>2.15</td>
</tr>
<tr>
<td>Compressor Block (Bolt holes)</td>
<td>0.37</td>
<td>0.40</td>
<td>0.49</td>
</tr>
<tr>
<td>Compressor Block (inner wall &amp; valve ports)</td>
<td>0.65</td>
<td>0.76</td>
<td>0.75</td>
</tr>
<tr>
<td>Piston Rod Seal</td>
<td>0.16</td>
<td>0.19</td>
<td>0.22</td>
</tr>
<tr>
<td>Expander Head</td>
<td>0.36</td>
<td>0.37</td>
<td>0.44</td>
</tr>
<tr>
<td>Heater</td>
<td>0.09</td>
<td>0.09</td>
<td>0.13</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>6.62</strong></td>
<td><strong>8.08</strong></td>
<td><strong>8.30</strong></td>
</tr>
</tbody>
</table>
4. Power Measurement

1) Heater input

Electrical power input to the heater is calculated from

\[ Q \text{ (kW)} = \text{Voltage across heater (V)} \times \frac{100 \text{ (Amp)}}{50 \text{ (mV)}} \times \frac{\text{shunt Voltage(mV)}}{1000} \]

Table A-5 lists data for Tests #2 through #6.

Table A-5  Heater Electrical Input Data

<table>
<thead>
<tr>
<th>Test</th>
<th>#2</th>
<th>#3</th>
<th>#4</th>
<th>#5</th>
<th>#6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage across heater(V)</td>
<td>245</td>
<td>257</td>
<td>242</td>
<td>257</td>
<td>258</td>
</tr>
<tr>
<td>Shunt Voltage (mV)</td>
<td>24.02</td>
<td>25.04</td>
<td>23.20</td>
<td>24.60</td>
<td>24.95</td>
</tr>
<tr>
<td>Input Power (kW)</td>
<td>11.77</td>
<td>12.87</td>
<td>11.23</td>
<td>12.64</td>
<td>12.87</td>
</tr>
</tbody>
</table>

2) Brake power

The brake power is measured by a dynamometer, load cell, and manometer set up. It is calculated from the following equation.

\[ W_{\text{brake}} \text{ (kW)} = 0.3491 \times \text{manometer reading above zero (m)} \times \text{rotational frequency (sec}^{-1}) \]

Table A-6 lists data for Tests #2 through #6.
Table A-6  Brake Power Data

<table>
<thead>
<tr>
<th></th>
<th>Test #2</th>
<th>#3</th>
<th>#4</th>
<th>#5</th>
<th>#6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manometer reading(m)</td>
<td>0.557</td>
<td>0.406</td>
<td>0.617</td>
<td>0.605</td>
<td>0.668</td>
</tr>
<tr>
<td>Rot. frequency (sec⁻¹)</td>
<td>9.07</td>
<td>9.02</td>
<td>9.00</td>
<td>10.00</td>
<td>9.97</td>
</tr>
<tr>
<td>(\dot{W}_{\text{brake}}) (kW)</td>
<td>1.76</td>
<td>1.28</td>
<td>1.94</td>
<td>2.11</td>
<td>2.32</td>
</tr>
</tbody>
</table>

3) Compressor indicated power \(\dot{W}_c\), and expander indicated power \(\dot{W}_e\).

The areas of the P-V diagram (Fig. A-1 to 5) were measured by a planimeter and then multiplied by proper calibration factors to yield indicated powers:

\[
\dot{W} \text{ (kW)} = \text{area of P-V diagram (m}^2\) \times \text{vertical calibration (kPa/m)} \\
\times \text{horizontal calibration (m}^{-3}\text{)} \times \text{rotational frequency (sec}^{-1}\)
\]

Table A-7 lists the compressor data and Table A-8 lists the expander data.

Table A-7  Indicated Power Data - Compressor

<table>
<thead>
<tr>
<th></th>
<th>Test #2</th>
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<th>#4</th>
<th>#5</th>
<th>#6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area of P-V diagram((10^{-4}\text{m}^2))</td>
<td>6.23</td>
<td>5.86</td>
<td>12.62</td>
<td>13.18</td>
<td>12.66</td>
</tr>
<tr>
<td>Vertical Calibration(kPa/m)</td>
<td>73020</td>
<td>68330</td>
<td>33380</td>
<td>33330</td>
<td>34240</td>
</tr>
<tr>
<td>Horizontal Calibration ((10^{-2}\text{ m}^3/\text{m}))</td>
<td>1.30</td>
<td>1.55</td>
<td>1.05</td>
<td>1.05</td>
<td>1.06</td>
</tr>
<tr>
<td>(-\dot{W}_c) (kW)</td>
<td>5.37</td>
<td>5.58</td>
<td>3.98</td>
<td>4.61</td>
<td>4.57</td>
</tr>
</tbody>
</table>

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Table A-8  Indicated Power Data - Expander

<table>
<thead>
<tr>
<th></th>
<th>Test #2</th>
<th>#3</th>
<th>#4</th>
<th>#5</th>
<th>#6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area of P-V diagram (10^{-4} \text{m}^2)</td>
<td>9.42</td>
<td>8.85</td>
<td>21.51</td>
<td>21.77</td>
<td>22.26</td>
</tr>
<tr>
<td>Vertical Calibration (\text{kPa/m})</td>
<td>73020</td>
<td>68330</td>
<td>33380</td>
<td>33330</td>
<td>34240</td>
</tr>
<tr>
<td>Horizontal Calibration (10^{-2} \text{m}^3/\text{m})</td>
<td>1.38</td>
<td>1.63</td>
<td>1.11</td>
<td>1.11</td>
<td>1.12</td>
</tr>
<tr>
<td>(\dot{W}_e \text{ (kW)})</td>
<td>8.25</td>
<td>8.90</td>
<td>7.17</td>
<td>8.04</td>
<td>8.49</td>
</tr>
</tbody>
</table>

5. Miscellaneous Tests

In addition to the regular tests runs of the engine, the following tests were conducted: Expander compression-expansion test, heat transfer flow test, and a motoring test.

1) Expander compression-expansion test

(1) Purpose and procedure

The purpose of the test was to see the effect of heat transfer inside expander during expansion or compression with a fixed mass of gas. Both the intake and the discharge valves were closed. The intake valve was clamped in closed position using a bolt through the expander head plate and intake valve access port. This arrangement was necessary because there was a considerable amount of leakage past the intake valve when the high pressure of the intake tube was used to keep the intake valve closed. The leakage would have made it impossible to maintain a fixed amount of gas inside the expander. Gas was supplied through the thermocouple port. A throttle valve
and a pressure gauge was fitted to the gas supply line near the expander.

As shown in Fig. A-22, the P-V diagram does not open up, indicating that the process is nearly reversible. However, the fact that there is little loss in this test does not guarantee that the same is true with the actual running, in which case there are considerable T between the gas and wall temperature, and more turbulence.

(2) Pressure transducer zero shift

An important by-product of this test was the discovery of the unexpected pressure transducer behavior. At first, it seemed that the compression ratio was too small (about 3:1), which was far below the isothermal compression ratio (8:1) and the isentropic compression ratio (32:1). The compression ratio was based on the information in the specification sheet that the pressure transducer zero shift was negligible in the temperature range of 55 C to 120 C. Also, we had anticipated that the gas temperature near the pressure transducer diaphragm would be within this range.

The following method was used to detect a considerable zero shift. After expander went through a number of cycles to warm up, the engine was stopped and expander was vented immediately to read atmospheric pressure. It was observed that the pressure transducer output was still well above the original zero point even after the expander was vented. The output voltage dropped slowly for about 30 seconds until it

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Fig. A-22 P-V diagram of compression-expansion test of expander
reached the original zero point. It was decided that the gas near the pressure transducer diaphragm reached a high temperature outside the compensated range. Taking into account the zero shift, the actual compression ratio 22:1 was reasonable.

As a cure to this problem, the transducer diaphragm was coated with RTV Silicon rubber to prevent the direct exposure to high temperature gas. The result was that the zero shift became significantly smaller, although it persisted. Therefore, it was decided to use the readings from two pressure gauges located outside the cylinder to decide the maximum and the minimum pressure during a cycle.

It was recommended that the gas near the pressure transducer be cooled more effectively in order to decrease the zero shift.

(3) Dynamic piston ring leakage

During the expander compression-expansion test, the compressor pressure was monitored to see if there was any leakage past the piston seal. Even when the peak pressure of the expander went up to 6.90 MPa, there was no sign of pressure increase in the compressor, which indicated the O ring was sealing perfectly.
2) Heat transfer flow test

The purpose of this test was to back up the assumptions concerning energy balance of the engine (section II.2 & Appendix B). The assumptions were as follows: a) Calculated cooler gas flow rate is the gas flow rate of the engine. b) Heat leak from the heater and the regenerative heat exchanger is much more than from natural convection.

As briefly explained in section II.2.1), the working gas was circulated through the engine components with both of the expander valves wide open. The pressure difference along the path was only due to flow friction and compressor check valve springs, which was very small compared to mean pressure of 1.38 MPa, making the expander and compressor power negligible. Table A-9 lists the test data on heater input power, shaft power, pressure, and cooling water. Table A-10 gives the average temperatures from the multi-channel recorder.

3) Motoring test

The purpose of the test was to determine the friction loss due to seals by measuring the torque required to turn the engine with dc powered dynamometer. Cooling water flow rates and temperatures were measured to calculate how much heat was generated due to friction.

Table A-11 lists the data from the motoring test. Of the measured motoring power 1.77 kW, about 0.44 kW is the linkage
| **Table A-9** Heat Transfer Flow Test Data  
(Power, pressure, and cooling water) |
<table>
<thead>
<tr>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Heater input power</strong></td>
</tr>
<tr>
<td>Voltage across heater (V)</td>
</tr>
<tr>
<td>Shunt voltage (mV)</td>
</tr>
<tr>
<td>Heater electrical input (kW)</td>
</tr>
<tr>
<td><strong>Shaft power due to friction</strong></td>
</tr>
<tr>
<td>Manometer reading (m Hg)</td>
</tr>
<tr>
<td>Rotational frequency (sec$^{-1}$)</td>
</tr>
<tr>
<td>$W_{\text{shaft}}$ (kW)</td>
</tr>
<tr>
<td><strong>Mean pressure (MPa)</strong></td>
</tr>
<tr>
<td><strong>Cooling water heat removal rate (kW)</strong></td>
</tr>
<tr>
<td>Cooler</td>
</tr>
<tr>
<td>Cylinder jacket</td>
</tr>
<tr>
<td>Compressor block</td>
</tr>
<tr>
<td>Piston rod</td>
</tr>
<tr>
<td>Expander head</td>
</tr>
<tr>
<td>Heater</td>
</tr>
<tr>
<td>Total</td>
</tr>
</tbody>
</table>
Table A-10  Average Temperatures Measured by the Multi-channel Recorder(Heat Transfer Flow Test)

<table>
<thead>
<tr>
<th>T.C. No.*</th>
<th>Location of Thermocouple</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cooler Inlet (from R.H. Ex.)</td>
<td>62</td>
</tr>
<tr>
<td>2</td>
<td>R.H. Ex. Inlet (from Comp. Disch.)</td>
<td>13</td>
</tr>
<tr>
<td>3</td>
<td>R.H. Ex. Inlet (from Exp. Disch.)</td>
<td>500</td>
</tr>
<tr>
<td>4</td>
<td>R.H. Ex. Outlet (to Heater)</td>
<td>443</td>
</tr>
<tr>
<td>5</td>
<td>Comp. Discharge</td>
<td>13</td>
</tr>
<tr>
<td>6</td>
<td>Exp. Inlet Wall</td>
<td>489</td>
</tr>
<tr>
<td>7</td>
<td>Heater Wall</td>
<td>508</td>
</tr>
<tr>
<td>8</td>
<td>Exp. Inlet (Tube T.C.)</td>
<td>520</td>
</tr>
<tr>
<td>9</td>
<td>Cylinder Wall</td>
<td>503</td>
</tr>
<tr>
<td>10</td>
<td>Cylinder Wall</td>
<td>391</td>
</tr>
<tr>
<td>11</td>
<td>Cylinder Wall</td>
<td>479</td>
</tr>
<tr>
<td>12</td>
<td>Cylinder Wall</td>
<td>343</td>
</tr>
<tr>
<td>13</td>
<td>Cylinder Wall</td>
<td>134</td>
</tr>
<tr>
<td>14</td>
<td>Comp. Block (Discharge Side)</td>
<td>13</td>
</tr>
<tr>
<td>15</td>
<td>Comp. Block (Inlet Side)</td>
<td>6</td>
</tr>
<tr>
<td>16</td>
<td>Cylinder Wall</td>
<td>110</td>
</tr>
<tr>
<td>17</td>
<td>Exp. Head (Inlet Side)</td>
<td>509</td>
</tr>
<tr>
<td>18</td>
<td>Exp. Head (Discharge Side)</td>
<td>485</td>
</tr>
<tr>
<td>19</td>
<td>Comp. Inlet</td>
<td>8</td>
</tr>
<tr>
<td>20</td>
<td>R.H. Ex. Wall (Upper part)</td>
<td>456</td>
</tr>
<tr>
<td>21</td>
<td>R.H. Ex. Wall (Lower part)</td>
<td>62</td>
</tr>
</tbody>
</table>

* See Fig. A-10
loss due to crankcase friction (3). The remaining 1.33 kW is composed of the total of cooling water heat removal rates 1.20 kW and unmeasured heat leak from the engine components to the environment.

Table A-11 Motoring Test Data

<table>
<thead>
<tr>
<th>Motoring power (kW)</th>
<th>1.77</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling water heat removal rates (kW)</td>
<td></td>
</tr>
<tr>
<td>Cooler</td>
<td>0.20</td>
</tr>
<tr>
<td>Cylinder jacket</td>
<td>0.70</td>
</tr>
<tr>
<td>Compressor block</td>
<td>0.12</td>
</tr>
<tr>
<td>Piston rod seal</td>
<td>0.12</td>
</tr>
<tr>
<td>Expander head</td>
<td>0.05</td>
</tr>
<tr>
<td>Heater</td>
<td>0.01</td>
</tr>
<tr>
<td>Total</td>
<td>1.20</td>
</tr>
</tbody>
</table>
Appendix B  ENERGY BALANCE CALCULATION

The First law for control volume (equation III-10) will be used for control volumes drawn around engine components, and around the entire engine as shown in Fig. B-1. For temperatures, average temperature data from the multichannel recorders (Table A-1) will be used. For pressures, average pressures measured by pressure gauges (Appendix A.1) will be used. Apparent mass flow rate of cooler, which was confirmed to be very close to the actual engine flow rate by orifice metering (section III.2.2) will be used as the mass flow rate.

Assuming steady operation, $\frac{\partial E}{\partial t}$ will be identical to zero for each control volume.

(a) Control volume 1 (Entire engine)

Knowing the electrical heat input to the heater $\dot{Q}_{\text{heater}}$, cooling water heat removal rates $\dot{Q}_{\text{cooler}}$, $\dot{Q}_{\text{other sinks}}$, brake engine power $\dot{W}_{\text{brake}}$, linkage power loss $\dot{W}_{\text{linkage}}$, the energy balance for the control volume 1 gives the rate of heat loss to the environment from the engine components $\dot{Q}_{\text{environ,1}}$. Table B-1 summarizes the energy balance calculations for the first zero piston ring leakage test (Test #3), the after compressor modification test (Test #5), and the heat transfer flow test (H.T.F. Test).

113
Fig. B-1  Control volumes for energy balance
Table B-1 Energy Balance for the Entire Engine

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F. Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{Q}_{\text{heater}}$ (kW)</td>
<td>12.87</td>
<td>12.64</td>
<td>5.53</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{cooling water}}$ (kW)</td>
<td>8.26</td>
<td>8.08</td>
<td>6.14</td>
</tr>
<tr>
<td>$\dot{W}_{\text{brake}}$ (kW)</td>
<td>1.28</td>
<td>2.11</td>
<td>-2.30</td>
</tr>
<tr>
<td>$\dot{W}_{\text{linkage}}$ (kW)</td>
<td>0.44</td>
<td>0.44</td>
<td>0.44</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ,1}}$ (kW)</td>
<td>2.89</td>
<td>2.01</td>
<td>1.25</td>
</tr>
</tbody>
</table>

$\dot{Q}_{\text{environ,1}}$ can also be estimated from component surface temperatures and natural convection law. Hankard (3) thus estimated $\dot{Q}_{\text{environ,1}}$ to be about 1.20 kW for Test #1 which is similar to Test #3 and #5 in terms of surface temperatures of the engine components. The so-called unaccounted-for-energy mentioned in reference 1 and 3 is the underestimated portion of $\dot{Q}_{\text{environ,1}}$. As we check the energy balance of each components, it will be shown that most of the underestimation is from the heater and the regenerative heat exchanger.

(b) Control volume 2 (Cooler)

As mentioned earlier, the apparent mass flow rate of the cooler can be obtained from the cooler heat removal rate $\dot{Q}_{\text{cooler}}$, and gas temperature change $\Delta T$ between inlet and outlet. In the calculation, the heat loss to the environment from the cooler $\dot{Q}_{\text{environ,2}}$ was neglected because the temperature between
the cooler surface and the environment was negligible. Table B-2 summarizes the energy balance calculation for the cooler.

Table B-2  Cooler Energy Balance Calculations

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F.Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{Q}_{\text{cooler}}$ (kW)</td>
<td>4.37</td>
<td>4.19</td>
<td>3.42</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ,2}}$ (kW)</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>$\Delta T$ (C)</td>
<td>117</td>
<td>107</td>
<td>54</td>
</tr>
<tr>
<td>$\dot{m}_f$ (g/sec)</td>
<td>7.22</td>
<td>7.56</td>
<td>12.22</td>
</tr>
</tbody>
</table>

(c) Control volume 3 (Heater)

Given the mass flow rate $\dot{m}_f$ of Table B-2, gas temperature increase $\Delta T$, electrical input to the heater $\dot{Q}_{\text{heater}}$, and the cooling water data for electrical feed through $\dot{Q}_{\text{water}}$, the heat loss to the environment from the heater $\dot{Q}_{\text{environ,3}}$ is calculated as shown in Table B-3.

Table B-3  Heater Energy Balance Calculations

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F.Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{Q}_{\text{heater}}$ (kW)</td>
<td>12.87</td>
<td>12.64</td>
<td>5.53</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{water}}$ (kW)</td>
<td>0.05</td>
<td>0.09</td>
<td>0.08</td>
</tr>
<tr>
<td>$\Delta T$ (C)</td>
<td>320</td>
<td>295</td>
<td>77</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ}}$ (kW)</td>
<td>0.91</td>
<td>1.00</td>
<td>0.55</td>
</tr>
</tbody>
</table>

Hankard (3) estimated $\dot{Q}_{\text{environ,3}}$ to be 0.14 kW from
natural convection law and surface temperatures, which is far less than the values in Table B-3.

(d) Control volume 4 (Regenerative heat exchanger)

Given the temperature changes ($\Delta T$) of both high pressure and low pressure stream, and the identical mass flow rate $\dot{m}_f$ for both stream, the heat leak to the environment from the regenerative heat exchanger $\dot{Q}_{\text{environ},4}$ can be calculated as shown in Table B-4.

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F. Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T_{\text{H.P.}}$ (°C)</td>
<td>299</td>
<td>300</td>
<td>430</td>
</tr>
<tr>
<td>$\Delta T_{\text{L.P.}}$ (°C)</td>
<td>315</td>
<td>310</td>
<td>438</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ},4}$ (kW)</td>
<td>0.62</td>
<td>0.39</td>
<td>0.53</td>
</tr>
</tbody>
</table>

Hankard's (3) estimate for $\dot{Q}_{\text{environ},4}$ from natural convection law was only 0.07 kW.

(e) Control volume 5 (Cylinder)

Knowing the brake engine power $\dot{W}_{\text{brake}}$, linkage loss $\dot{W}_{\text{linkage}}$, the average mass flow rate $\dot{m}_f$, temperature changes through compressor $\Delta T_{\text{comp}}$ and expander $\Delta T_{\text{exp}}$, cooling water heat removal rate $\dot{Q}_{\text{water}}$, the energy balance for the cylinder gives the rate of cylinder heat loss to the environ-
ment $Q_{\text{environ},5}$. Table B-5 summarizes the calculations.

Table B-5  Cylinder Energy Balance Calculations

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F. Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T_{\text{comp.}} (\degree C)$</td>
<td>104</td>
<td>83</td>
<td>5</td>
</tr>
<tr>
<td>$\Delta T_{\text{exp.}} (\degree C)$</td>
<td>279</td>
<td>254</td>
<td>20</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{water}} (\text{kW})$</td>
<td>3.84</td>
<td>3.80</td>
<td>2.64</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ},5} (\text{kW})$</td>
<td>0.92</td>
<td>0.37</td>
<td>0.18</td>
</tr>
</tbody>
</table>

(f) Control volume 6 (Tubings between components)

Tubing heat loss is mostly from the tubing between compressor discharge and regenerative heat exchanger. It can be estimated from the mass flow rate $\dot{m}_f$ and the gas temperature drop $\Delta T$ along the tubing. Table B-6 shows the results.

Table B-6  Heat Loss Rate from Tubings

<table>
<thead>
<tr>
<th></th>
<th>Test #3</th>
<th>Test #5</th>
<th>H.T.F. Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T (\degree C)$</td>
<td>12</td>
<td>7</td>
<td>negligible</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{environ},6} (\text{kW})$</td>
<td>0.44</td>
<td>0.26</td>
<td>negligible</td>
</tr>
</tbody>
</table>

If all the test data are reasonably correct, the heat loss from the entire engine $\dot{Q}_{\text{environ},1}$ should match the sum of $\dot{Q}_{\text{environ}}$'s of the control volumes 2 to 6. It turned out that
\[ \dot{Q}_{\text{environ},1} \text{ and } \frac{6}{\sum_{i=2}^{5} \dot{Q}_{\text{environ},i}} \text{ is within 1\% of each other.} \]

which strengthens the energy balance argument in section II.2.
Appendix C  EXAMPLES OF CYCLIC HEAT TRANSFER CALCULATION

1. Equivalent Isentropic Method

As explained in section II . 3.4), this method estimates the magnitude of cyclic heat transfer by approximating compressor (or expander) processes with steady flow isentropic compressor (or expander). Referring to Fig. II - 3 , the equations for compressor side are as follows.

Inlet temperature for steady flow compressor

\[ T_{\text{in}} = \frac{\dot{W}}{\dot{m}_f c_p (1 - P_C)} \]

Outlet temperature for steady flow compressor

\[ T_{\text{out}} = T_{\text{in}} P_C \frac{c_o}{r} \quad ; \quad c_o = \frac{(\gamma - 1)}{\gamma} \]

Heat transfer rate into the gas

\[ \dot{Q}_+ = \dot{m}_f c_p \left( T_{\text{in}} - T_1 \right) \]

Heat transfer rate out of the gas

\[ \dot{Q}_- = \dot{m}_f c_p \left( T_2 - T_{\text{out}} \right) \]

Non-dimensional magnitude of cyclic heat transfer for compressor

\[ X_c = \frac{\left| \dot{Q}_+ \right| + \left| \dot{Q}_- \right| }{2 \Delta H} \]

Total enthalpy change

\[ \Delta H = \dot{m}_f c_p \left( T_2 - T_1 \right) \]
The equations for expander side are as follows.

Inlet temperature for steady flow expander

\[ T_{4i} = \frac{\dot{W}_e}{\dot{m}_f c_p} (1-P_r^c) \]

Outlet temperature for steady flow expander

\[ T_{5i} = T_4 P_r^c \]

Heat transfer rate into the gas

\[ \dot{Q}_+ = \dot{m}_f c_p (T_5 - T_{5i}) \]

Heat transfer rate out of the gas

\[ \dot{Q}_- = \dot{m}_f c_p (T_{4i} - T_4) \]

Non-dimensional magnitude of cyclic heat transfer for expander

\[ X_c = (|\dot{Q}_+| + |\dot{Q}_-|) / 2\Delta H \]

Total enthalpy change

\[ \Delta H = \dot{m}_f c_p (T_4 - T_5) \]

Table C-1 shows the input data and results of Equivalent Isentropic Method for Test #1 through #6.
Table C-1  Input Data and Results of Equivalent Isentropic Method

<table>
<thead>
<tr>
<th></th>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_h/P_l$</td>
<td>2.18</td>
<td>2.09</td>
<td>2.14</td>
<td>2.08</td>
<td>2.07</td>
<td>2.10</td>
</tr>
<tr>
<td>$T_1$(K)</td>
<td>296</td>
<td>297</td>
<td>293</td>
<td>288</td>
<td>287</td>
<td>280</td>
</tr>
<tr>
<td>$T_2$(K)</td>
<td>415</td>
<td>409</td>
<td>386</td>
<td>367</td>
<td>369</td>
<td>361</td>
</tr>
<tr>
<td>$T_4$(K)</td>
<td>925</td>
<td>998</td>
<td>1004</td>
<td>993</td>
<td>957</td>
<td>973</td>
</tr>
<tr>
<td>$T_5$(K)</td>
<td>686</td>
<td>733</td>
<td>726</td>
<td>724</td>
<td>703</td>
<td>713</td>
</tr>
<tr>
<td>$\dot{m}_f$ (Kg/sec)</td>
<td>8.06</td>
<td>7.07</td>
<td>7.22</td>
<td>5.95</td>
<td>7.56</td>
<td>7.53</td>
</tr>
<tr>
<td>$\dot{W}_c$(KW)</td>
<td>-7.71</td>
<td>-5.37</td>
<td>-5.58</td>
<td>-3.98</td>
<td>-4.61</td>
<td>-4.57</td>
</tr>
<tr>
<td>$\dot{W}_e$(KW)</td>
<td>10.05</td>
<td>8.25</td>
<td>8.90</td>
<td>7.17</td>
<td>8.04</td>
<td>8.49</td>
</tr>
<tr>
<td>$T_{1i}$(K)</td>
<td>503</td>
<td>426</td>
<td>421</td>
<td>379</td>
<td>350</td>
<td>337</td>
</tr>
<tr>
<td>$T_{2i}$(K)</td>
<td>687</td>
<td>573</td>
<td>569</td>
<td>507</td>
<td>467</td>
<td>454</td>
</tr>
<tr>
<td>$T_{4i}$(K)</td>
<td>896</td>
<td>879</td>
<td>908</td>
<td>915</td>
<td>816</td>
<td>844</td>
</tr>
<tr>
<td>$T_{5i}$(K)</td>
<td>656</td>
<td>654</td>
<td>671</td>
<td>683</td>
<td>611</td>
<td>627</td>
</tr>
<tr>
<td>$\eta_c$(%)</td>
<td>58.9</td>
<td>69.7</td>
<td>69.8</td>
<td>75.9</td>
<td>81.9</td>
<td>83.0</td>
</tr>
<tr>
<td>$\eta_e$(%)</td>
<td>96.8</td>
<td>88.1</td>
<td>90.5</td>
<td>92.1</td>
<td>85.2</td>
<td>86.7</td>
</tr>
<tr>
<td>$X_c$</td>
<td>2.01</td>
<td>1.31</td>
<td>1.69</td>
<td>1.46</td>
<td>0.97</td>
<td>0.93</td>
</tr>
<tr>
<td>$X_e$</td>
<td>0.12</td>
<td>0.37</td>
<td>0.27</td>
<td>0.22</td>
<td>0.46</td>
<td>0.41</td>
</tr>
</tbody>
</table>

* In $10^{-3}$ Kg/sec
2. Equivalent Reversible Polytropic Method

As mentioned in section 2.3.4, this method estimates the cyclic heat transfer by approximating compressor processes with reversible compression, reexpansion process, and constant pressure intake, discharge process. Referring to Fig. 2-5 the equations used are:

\[
\dot{Q}_+ = \dot{m}_f c_p (T_{ei} - T_{in}) + \dot{Q}_{rc}
\]

\[
\dot{Q}_- = \dot{m}_f c_p (T_{out} - T_{ed}) + \dot{Q}_{ct}
\]

From equation 2-2, ignoring valve pressure drops, the effective intake temperature

\[
T_{ei} = \frac{P_L (V_1 - V_4)}{\frac{\dot{m}_f}{RPS} R}
\]

the effective discharge temperature

\[
T_{ed} = \frac{P_h (V_2 - V_3)}{\frac{\dot{m}_f}{RPS} R}
\]

where RPS is the rotational frequency. The heat transfer rates \(\dot{Q}_{ct}\) and \(\dot{Q}_{rc}\) are

\[
\dot{Q}_{ct} = (\frac{1}{1-n_1} - \frac{1}{1-\gamma}) (P_h V_2 - P_L V_1) \text{ RPS}
\]
\[
\dot{Q}_{rc} = \left( \frac{1}{1-n_2} - \frac{1}{1-n_1} \right) (P_LV_4 - P_hV_3) \text{ RPS}
\]

where \( n_1 = -\log \left( \frac{P_h}{P_L} \right) / \log \left( \frac{V_2}{V_1} \right) \),
\[
n_2 = -\log \left( \frac{P_L}{P_h} \right) / \log \left( \frac{V_4}{V_3} \right).
\]

Non-dimensional cyclic heat transfer of the compressor \( X_c \) is defined as in the previous section. Table C-2 shows input data and results for Test #1 through #6.

Table C-2 Input Data and Results of Equivalent Reversible Polytropic Method

<table>
<thead>
<tr>
<th>Test #1</th>
<th>Test #2</th>
<th>Test #3</th>
<th>Test #4</th>
<th>Test #5</th>
<th>Test #6</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_h ) (MPa)</td>
<td>3.69</td>
<td>3.55</td>
<td>3.83</td>
<td>3.24</td>
<td>3.43</td>
</tr>
<tr>
<td>( P_L ) (MPa)</td>
<td>1.69</td>
<td>1.70</td>
<td>1.79</td>
<td>1.56</td>
<td>1.66</td>
</tr>
<tr>
<td>( V_1 )</td>
<td>8.741</td>
<td>8.741</td>
<td>8.741</td>
<td>10.22</td>
<td>10.33</td>
</tr>
<tr>
<td>( V_2 )</td>
<td>5.788</td>
<td>5.563</td>
<td>5.303</td>
<td>6.332</td>
<td>6.522</td>
</tr>
<tr>
<td>( V_3 )</td>
<td>2.950</td>
<td>2.950</td>
<td>2.950</td>
<td>4.425</td>
<td>4.542</td>
</tr>
<tr>
<td>( V_4 )</td>
<td>4.621</td>
<td>4.905</td>
<td>5.003</td>
<td>7.384</td>
<td>7.330</td>
</tr>
<tr>
<td>( m_f ) **</td>
<td>8.06</td>
<td>7.07</td>
<td>7.22</td>
<td>5.95</td>
<td>7.56</td>
</tr>
<tr>
<td>( T_{in} ) (K)</td>
<td>296</td>
<td>297</td>
<td>293</td>
<td>288</td>
<td>287</td>
</tr>
<tr>
<td>( T_{out} ) (K)</td>
<td>415</td>
<td>409</td>
<td>386</td>
<td>367</td>
<td>369</td>
</tr>
<tr>
<td>( \dot{Q}_+ ) (kW)</td>
<td>7.38</td>
<td>5.22</td>
<td>5.67</td>
<td>3.00</td>
<td>2.58</td>
</tr>
<tr>
<td>( \dot{Q}_- ) (kW)</td>
<td>-9.63</td>
<td>-6.65</td>
<td>-7.48</td>
<td>-4.21</td>
<td>-3.89</td>
</tr>
<tr>
<td>( \Delta H ) (kW)</td>
<td>4.98</td>
<td>4.10</td>
<td>3.89</td>
<td>2.45</td>
<td>3.25</td>
</tr>
<tr>
<td>( X_c )</td>
<td>1.71</td>
<td>1.45</td>
<td>1.69</td>
<td>1.47</td>
<td>0.99</td>
</tr>
</tbody>
</table>

* volumes in 10^-4 m^3. ** in 10^-3 kg/sec
Appendix D  FLOWMETER CALCULATIONS

1. Orifice Sizing

The two square edged orifices made of stainless steel 300 for measuring instantaneous mass flow rates during compressor intake and discharge had to be properly sized in order not to introduce too much pressure drop in the engine, and to limit the error due to non-linearity of orifice meter. Flange taps were used for pressure taps: the centers of the inlet and outlet pressure taps are 2.54 cm from the inlet and outlet faces of the orifice plates respectively (see Fig. III-2).

The equations for calculation of gas flow rate \( \dot{m} \) (kg/sec) are as follows (6):

\[
\dot{m} = \frac{\pi d^2}{4} \frac{Fa E_f}{C \left( \frac{1}{(1-\beta f)^2} \sqrt{2 \epsilon_c \rho_1 \left( P_1 - P_2 \right)} \right)} \tag{D-1}
\]

where 
\( d \) = diameter of the orifice (m),
\( Fa \) = thermal expansion factor of the orifice metal,
\( C \) = discharge coefficient,
\( \beta \) = orifice diameter \( d \) (m) / tube diameter \( D \) (m),
\( E_f \) = expansion factor due to compressibility,
\( \epsilon_c \) = gravitational proportionality constant,
\( \rho_1, \rho_2 \) = density of the gas (kg / m\(^3\))
\( P_1, P_2 \) = Pressure of the gas (Pa).

Subscript 1 refers to the upstream section, and subscript 2 refers to the downstream section.
The discharge coefficient \( C \) is computed from the following relations:

\[
C = Ke \sqrt{1 - \beta^4} \left(1 + \frac{Z}{R_d}\right) \left(\frac{10^6 d}{10^6 d + 15 A}\right)
\]  
(D-2)

where \( Ke = 0.5993 + 1.778 \times 10^{-4} / D + (0.364 + 4.769 \times 10^{-1} / D)\beta^4 \)

\[
+ 0.4 \left(1.6 - 2.54 \times 10^{-2} / D\right)^5 \left(0.07 + 1.27 \times 10^{-2} / D - \right)^2 \cdot 5
\]

\[
- (0.009 + 8.636 \times 10^{-4} / D) (0.5 - \beta)^1.5
\]

\[
+ (1.008 \times 10^5 + 3) (\beta - 0.7)^1.5,
\]

\[Z = 1000d \left(32.68 - 196.9\beta + 354.3\beta^2 - 165.4\beta^3 + 3.326 / D\right).
\]

The expansion factor \( E \) is evaluated from

\[
E_f = 1 - (0.41 + 0.35 \beta^4) \cdot \frac{\Delta P}{P_1} \cdot \frac{1}{\gamma} ; \Delta P = P_1 - P_2
\]  
(D-3)

when the reference pressure is measured at the inlet pressure tap, and from

\[
E_f = \sqrt{1 + \left(\frac{\Delta P}{P_2}\right)^2 - (0.41 + 0.35 \beta^4) \frac{(\Delta P/P_2)^2}{\gamma} \cdot \frac{1}{\gamma}} \frac{1}{1 + (\Delta P/P_2)^2}
\]  
(D-4)

when the reference pressure is measured at the outlet pressure tap.

The maximum flow rate during intake or discharge can be estimated from the piston velocity \( v_p,_{\text{max}} \) and inlet or outlet density.
\[ \dot{m}_{\text{max}} = A_c v_{p, \text{max}}, \]  \hspace{1cm} (D-5)

where \( A_c \) is compressor volume per unit displacement. From geometry, the piston velocity \( v_p \) is given by

\[ v_p = \frac{d}{dt} \left( -r \cos \theta - L_c \sqrt{1-\left(\frac{r}{L_c \sin \theta}\right)^2} \right) \approx r \sin (1 + (\cos \theta) \frac{r}{L_c}) \]  \hspace{1cm} (D-6)

where \( \theta \) is crank angle, \( L_c \) is connecting rod length, and \( r \) is half the engine stroke. The maximum piston velocity \( v_{p, \text{max}} \) is approximated by \( v_{p, \text{max}} = rw \) within reasonable accuracy. Substituting values \( r = 5.715 \text{ cm} \), \( L_c = 20.32 \text{ cm} \), and \( \omega = 20 \text{ radian/sec} \), \( v_{p, \text{max}} = 3.6 \text{ m/sec} \).

From equation D-5, typical inlet density (Test #5) 2.789 \( \text{kg/m}^3 \), and outlet density 4.463 \( \text{kg/m}^3 \) give the maximum mass flow rate of \( 5.10 \times 10^{-2} \text{ kg/sec} \) during intake and \( 7.69 \times 10^{-2} \text{ kg/sec} \) during discharge. In the temperature range being considered (15 \( \text{C} \sim 120 \text{ C} \)) the thermal expansion factor \( F_a \) in equation D-1 can be regarded as 1.0 for stainless steel 300.

The diameter of the orifice plate \( d \) can be obtained from equation D-1 by solving for \( d \) after substituting appropriate values for \( \rho_1, P_1, P_2, F_a, D, \) and \( \dot{m} \). For intake side,

\( \rho_1 = 2.789 \text{ kg/m}^3 \), \( \dot{m} = 5.10 \times 10^{-2} \text{ kg/sec} \). For discharge side,

\( \rho_1 = 4.463 \text{ kg/m}^3 \), \( \dot{m} = 7.69 \times 10^{-2} \text{ kg/sec} \). \( D = 3.25 \times 10^{-2} \text{ m}, \)

\( F_a = 1.0 \).
To be used with the $\pm 0.1$ MPa differential pressure transducers, orifice diameters were sized so that $\beta = 0.40$ (d=1.3 cm), $\Delta P_{\text{max}} = 0.068$ kPa for the intake side, and $\beta = 0.45$ (d=1.46 cm), $\Delta P_{\text{max}} = 0.065$ kPa for the discharge side. The maximum pressure drops were designed to be less than 0.1 MPa to provide enough room in case the pressure drops were under-estimated.

2. Resonance Frequency of Pressure Taps

As shown in Fig. III-2, the differential pressure transducer adapter has almost equal void volume at each side. This was necessary to match the acoustic characteristics of the inlet pressure tap and outlet pressure tap. In order to estimate the resonance frequency of the pressure taps, it is assumed that the wave form inside the pressure tap is the plane wave. Referring to Fig. D-1, the acoustic pressure $p$ and volume velocity $q$ is expressed as follows (12).

\[
p_1(x_1, t) = j \omega \rho c (A_1 e^{-j k x_1} + B_1 e^{j k x_1}) e^{j \omega t}
\]

\[
p_2(x_2, t) = j \omega \rho c (A_2 e^{-j k x_2} + B_2 e^{j k x_2}) e^{j \omega t}
\]

\[
q_1(x_1, t) = j \omega S_1 (A_1 e^{-j k x_1} - B_1 e^{j k x_1}) e^{j \omega t}
\]

\[
q_2(x_2, t) = j \omega S_2 (A_2 e^{-j k x_2} - B_2 e^{j k x_2}) e^{j \omega t},
\]

where coordinates $x_1$ and $x_2$ are as shown in Fig. D-1, $S_1$ and $S_2$ are the cross-sectional areas of the chamber, and the tube
Fig. D-1 Pressure tap for differential pressure transducer (schematic)
respectively, \( \omega \) is the angular speed, A and B are the coefficients to be decided by boundary conditions, t is time, and k is the wavelength constant defined by

\[
k = \frac{\omega}{c} = \frac{2\pi f}{c} ; \quad c = \text{speed of sound}
\]

(D-8)

The boundary conditions are the continuity of acoustic pressure and volume velocity at the junction between the chamber and the tube, zero volume velocity at the end of chamber, and zero acoustic pressure at the tube entrance:

\[
p_1 (L_1, t) - p_2 (0, t) = 0
\]

\[
q_1 (L_1, t) - q_2 (0, t) = 0
\]

\[
q_1 (0, t) = 0
\]

\[
p_2 (L_2, t) = 0
\]

(D-9)

From equations D-6 and D-7, the boundary condition becomes

\[
[D] = \begin{bmatrix}
A_1 \\
B_1 \\
A_2 \\
B_2 \\
\end{bmatrix} = \begin{bmatrix}
f(L_1) & f(-L_1) & -1 & -1 \\
S_1 f(-L_1) & -S_1 f(L_1) & -S_2 & -S_2 \\
0 & 0 & f(-L_2) & f(L_2) \\
1 & -1 & 0 & 0 \\
\end{bmatrix} \begin{bmatrix}
A_1 \\
B_1 \\
A_2 \\
B_2 \\
\end{bmatrix} = 0
\]

(D-10)

where \( f(x) = e^{jkx} \). For non-trivial solution, the determinant \( |D| \) has to be equal to zero. This condition leads to the following relation:
\[ \tan k L_1 \cdot \tan k L_2 = \frac{S_2}{S_1}. \quad (D-11) \]

Substituting values \( L_1 = 1.59 \text{ cm}, \ L_2 = 10 \text{ cm}, \ S_1 = 2.52 \times 10^{-2} \text{ cm}^2, \ S_2 = 8.23 \times 10^{-3} \text{ cm}^2, \) the wavelength constant \( k \) can be calculated from equation D-11. Then from equation D-5, the resonance frequency is calculated to be about 1400 Hz. The orifice pressure drop data in Fig. III-5, and III-6 show the presence of 1400 Hz resonance.
Appendix E  COMPUTER CODES FOR EVALUATION OF COMPRESSOR PERFORMANCE
THIS IS THE COMPUTER PROGRAM FOR THE FIRST LAW INTEGRAL ANALYSIS.
IT COMPUTES MASS FLOW RATE OF THE ENGINE FROM THE ORIFICE PRESSURE
DROP DATA. USING THE COMPUTED MASS FLOW RATE, AND PRESSURE VOLUME
DATA, THE MIXED MEAN TEMPERATURE, HEAT TRANSFER RATE, AND SPECIFIC
ENTROPY ETC. ARE CALCULATED.

ORADL, ORADH, ORPDL, ORPDH, ORAIL, ORAITH, ORPIL, ORPIH ARE ORIFICE
FLOW RATE MEASUREMENT DATA. THIRD LETTERS A AND P DENOTE CRANK
ANGLE IN DEGREES AND ORIFICE PRESSURE DROP IN INCHES RESPECTIVELY.
THE FOURTH LETTERS D AND I DENOTE DISCHARGE SIDE AND INTAKE SIDE
RESPECTIVELY. THE FIFTH LETTERS L AND H MEAN THE DATA WAS FROM
THE LOWER ENVELOPE AND THE HIGHER ENVELOPE OF THE ORIFICE PRESSURE
DROP CURVE RESPECTIVELY.

PINH, AIH, PIL, AIL, PDH, ADH, PDL ARE COMPRESSOR PRESSURE VS.
CRANK ANGLE DATA. THE FIRST LETTERS P AND A DENOTE COMPRESSOR
PRESSURE IN INCHES AND CRANK ANGLE RESPECTIVELY. THE SECOND
LETTERS I AND D DENOTE INTAKE AND DISCHARGE PERIOD. ORFIL, ORFIH
ARE ARRAYS OF ORIFICE DATA FOR PLOTTING.

SENLD, SENHD, SENTD, SENLI, SENHI; SENTI ARE ARRAYS FOR SMOOTHING
OUT ENTROPY CURVE. THE FOURTH LETTERS L AND H MEAN THE LOWER AND
THE HIGHER ENVELOPE OF THE ENTROPY CURVE. THE FOURTH LETTER T
MEANS CUMULATIVE ENTROPY. THE FIFTH LETTERS D AND I MEAN
DISCHARGE AND INTAKE RESPECTIVELY.
TREF, AND PREF ARE THE REFERENCE TEMPERATURE AND PRESSURE FOR
ENTROPY CALCULATION. PCALD, PCALI ARE THE CONVERSION FACTORS FOR
ORIFICE DATA INTO PSL FOR DISCHARGE AND INTAKE SIDE RESPECTIVELY.

PH IS THE EXPANDER INTAKE PRESSURE AND PL IS THE COMPRESSOR INTAKE PRESSURE. YSPA IS THE CONVERSION FACTOR FOR COMPRESSOR PRESSURE DATA. CFLR IS THE APPARENT MASS FLOW RATE OF COOLER. VCL IS THE CLEARANCE VOLUME OF THE COMPRESSOR. CSV, CSP ARE SPECIFIC HEAT OF THE GAS AT CONSTANT VOLUME AND CONSTANT PRESSURE RESPECTIVELY.

R IS THE GAS CONSTANT. TINT AND TDIS ARE THE AVERAGE INTAKE AND DISCHARGE TEMPERATURE OF COMPRESSOR. TROC IS THE TEMPERATURE AT THE BEGINNING OF COMPRESSION. ZI AND ZD ARE THE CRANK ANGLE AT THE BEGINNING OF INTAKE AND DISCHARGE. NDIV IS THE NO. OF OUTPUT DATA PER CYCLE. RPM IS ENGINE REVOLUTION PER MINUTE.

IN THIS PROGRAM, ALL THE COMPUTATION WILL BE DONE IN BRITISH UNITS AND THE RESULTS WILL BE CONVERTED TO S.I. UNITS AT THE END.

INTEGER*2 XLAB(40), DIMENSION ORADL(29), ORADH(29), ORPDL(29), ORPDH(29), ORAIL(38), ORAIH(41), ORPIH(41), ORPIL(2,38), XSCL(4), PIH(12), AIH(12), PIL(14), AIL(14), PHD(11), ADH(11), PDL(14), SNIK(2,7), SNHD(2,6), SNTD(2,21), NEHD(6), NFDL(7), SFNLI(2,7), SFNH(2,12), SNTI(2,22), NEVI(12), NERI(7), ADL(14), ORFIH(2,41), ARRAY(12,91), ORPDL(2,29), ORPDH(2,29) COMMON DELTA, ADJD, ADJI, ORAIL, ORPIL, ORATH, ORPIL, ORADI, ORPDH, ORADL, ORPDH, ARRAY
DATA PCALD, PCALI /2.1871, 2.3438/
DATA PH, PL, YSPA, CFLR /497.241, 2.945, 60./
DATA VCL, CSV, CSP, R, TINT, TDIS /27.72, 7462, 1.2405, 4943, 516., 665./
DATA TBOC, ZI, ZD, NDIV, NETS/531.276.100.2, 360.91/
DATA RPM, PCALI, PCALD /600.2, 3438.2, 1871/
DATA PH/0.0,-.075,-.07.,-.078,-.086,-.07,-.012,.0,.0,.0,.0,.0/0/.0
DATA AIH/276., 279., 282.6, 297., 311.4, 322.2, 343.8, 347.4, 351.0, 354.6, 358.2, 360.0/
C -0.060,-.055,0.0,0.0,0.0/0.0
DATA AIL/276., 279., 286.2, 289.8, 293.4, 304.2, 315.8, 325.8, 329.4, 333.4, 336.6, 354.6, 358.2, 360.0/
C 2.945,3.055,3.055,3.039,2.980,2.973,2.965,2.965,2.969,2.961,2.945/
DATA ADH/100.2, 102.6, 109.8, 113.4, 120.6, 124.2, 145.7, 149.4, 174.6, 178.2, 180.0/
C 2.945,3.055,3.055,3.039,2.977,2.949,2.945,2.945,2.947,2.944,2.943,2.946,2.946,2.945/
DATA ADL/100.2, 102.6, 109.8, 113.4, 117.1, 127.8, 135.1, 138.6, 142.2, 153.2, 156.6, 163.8, 167.4, 180.0/
C 590.0, 40.0, 45.0/
DATA TREF, TREF/537.1, 14.7/
DATA NEHI /1,4,5,6,8,9,11,12,13,14,20,22/
DATA NELI /1,3,7,10,17,18,22/
DATA NEHD /1,4,8,15,20,21/
DATA NELD /1,2,3,12,18,19,21/
ADJ=1.
ADJD=1.

READ DATA FROM ORIFICE MEASUREMENT

READ(8,2) (ORADL(I),I=1,29)
READ(8,2) (ORADH(I),I=1,29)
READ(8,3) (ORPD(I),I=1,29)
READ(8,3) (ORPDH(I),I=1,29)
READ(8,2) (ORAIL(I),I=1,38)
READ(8,2)  (ORAIH(I),I=1,41)
READ(8,3)  (ORPIL(I),I=1,38)
READ(8,3)  (ORPIH(I),I=1,41)
2 FORMAT((12F6.1))
3 FORMAT((12F6.3))
4 FORMAT((10F7.1))

C
SET UP COMPRESSOR PRESSURE ARRAY(1,J1)

DO 500 TI=1,12
DO 500 JJ=1,NPTS
500 ARRAY(TI, JJ)=0.0
COIL=CFLR/(RPM*60.)
DELTA=360./(FLOAT(NPTS)-1.)
PCAL=(PH-PL)/YSPA
I1=IFIX(ADH(1)/DELTA)+1
I2=IFIX(ADH(11)/DELTA)+1
I3=IFIX(AIH(1)/DELTA)+1
I4=IFIX(AIH(12)/DELTA)+1

C
.. Compressor

100 DO 150 I=1,I1
FI=FLOAT(I-1)
ANG=FI*DELTA

C
.. P**1.57291=EXP(12.0032)
150 ARRAY(I,1)=163276.*/CYLY(ANG,VCL)**1.57291

C
.. Reexpansion

C.. P**1.512610082=EXP(B)
C.. B=11.2472733-9034283305*SIN((.05+(X-3.32215)*.9/.478506)*PI)**.45

C
200 DO 250 I=I2,I3
  FI=FLOAT(I-1)
  ANG=FI*DELT
  VX=CYLV(ANG,VCL)
  BX1=(.05+(ALOG(VX)-3.32215)*.9/.478506)*3.1415926
  BY2=3.4283305E-02*SIN(BX1)**.45
  BX3=11.2472733-AX2
  250 ARRAY(1,I)=EXP(BX3)/VX**1.512610082
C
C....DISCHARGE
C
  KDL=14
  KDLP1=KDL-1
  KDH=11
  KDHP1=KDH-1
  I1P1=I1+1
  DO 320 J=I1P1,I2
  JP1=J-1
  THETA=DELT*FLOAT(JP1)
  DO 300 I=1,KDLP1
  IP1=I+1
  IF(THETA .GE. ADL(I).AND.THETA .LT. ADL(IP1)) GO TO 305
300 CONTINUE
305 SLOPE=(PDL(IP1)-PDL(I))/(ADL(IP1)-ADL(I))
  TPI=(PDL(I)+SLOPE*(THETA-ADL(I)))*PCAL+PL
  DO 310 I=1,KDHP1
  IP1=I+1
  IF(THETA .GE. ADH(I).AND.THETA .LT. ADH(IP1)) GO TO 315
310 CONTINUE
315 SLOPE=(PDH(IP1)-PDH(I))/(ADH(IP1)-ADH(I))
  TPH=(PDH(I)+SLOPE*(THETA-ADH(I)))*PCAL+PL
320 ARRAY(1,J)=(TPL+TPH)/2.
C..INTAKE

K1=14
K1,F1=K1-1
K1=12
K1,F1=K1-1
I3P1=I3+1
DO 350 J=I3P1,14
IF1=J-1
THETA=DELTA*FLOAT(IF1)
DO 330 I=1,K1,F1
IP1=I+1
IF(THETA.GE.AI(I).AND.THETA.LT.AI(IP1)) GO TO 335

330 CONTINUE
335 SLOPE=(PI(I,IP1)-PI(I))/((AI(IP1)-AI(I))
TP1=(PI(I)+SLOPE*(THETA-AI(I)))*PCAL+PL
DO 340 I=1,K1,F1
IP1=I+1
IF(THETA.GE.AI(I).AND.THETA.LT.AI(IP1)) GO TO 345

340 CONTINUE
345 SLOPE=(PI(I,IP1)-PI(I))/((AIHP1)-AIH(I))
TPH1=(PIH(I)+SLOPE*(THETA-AIH(I)))*PCAL+PL
350 ARRAY(1,J)=(TPH1)/2.

C..CALCULATE CLEARANCE VOLUME MASS CLMS

G=3.625*TRMC*12.
TOMA=PL*CYLV(0.0,VCL)/G
CLMS=TOMA-COOL

C..CONSTRUCT A PART OF COMPRESSOR MASS ARRAY(2,I)
I1=FIX(ZD/DELTA)+1
I2=FIX(180./DELTA)+1
I3=IFIX(ZI/DELTA)+1
DO 2000 I=I,I1
2000 ARRAY(2,I)=TOMA
DO 2200 I=I2,I3
2200 ARRAY(2,I)=CLMS

C
C     CALCULATE INTAKE AND DISCHARGE MASS FLOW RATE. COMPARE WITH COOL.
C
C     DECIDE ADJUSTING CONSTANS ADJI, ADJD.
C
C
C
C     XX=360./FLOAT(YDIV)
XX=1.0
K=IFIX(DELTA/XX)
MM=IFIX((360.-ZI)/XX)
NN=IFIX((180.-ZD)/XX)
WPTTE(5,112) I1,I2,I3,X,MM,NN

112 FORMAT(1H ,6I10)
CALL MASI(ZI,MM)
MI=MM/K+I3
ADJI=(ARRAY(2,MI)-ARRAY(2,I3))/COOL
CALL MASD(ZD,NN)
MD=NN/K+I1
ADJD=-(ARRAY(2,MD)-ARRAY(2,I1))/COOL

C
WRITE(5,222) ADJI,ADJD
222 FORMAT(1H ,2E10.4)

C
C     COMPLETE COMPRESSOR MASS ARRAY(2,I)
C
C
C     CALL MASI(ZI,MM)
C     CALL MASD(ZD,NN)
C
C
C     CONSTRUCT VOLUME AND TEMPERATURE ARRAYS
DO 2300 L=1,NPTS
D = (FLOAT(L) - 1.) * DELTA
ARRAY(3, L) = CYLV(D, VCL)
ARRAY(4, L) = ARRAY(1, L) * ARRAY(3, L) * 2.1575E-04 / ARRAY(2, L)
CONTINUE

C THE FIRST LAW OF THERMODYNAMICS FOR THE COMPRESSOR
DO 5000 IS = 1, 2
DO 2350 I = 1, NPTS
D = FLOAT(I - 1) * DELTA
2350 ARRAY(5, I) = HTRA(D)
PDV = 0.0
NPT = NPTS - 1
QSUM = 0.0
XMULT = RPM * 50. * 360. / DELTA
K1 = FIX(IZD / DELTA) + 1
K2 = FIX(180. / DELTA) + 1
K3 = FIX(IZT / DELTA) + 1
DO 3000 I = 1, NPTS
3000 IF (I .EQ. 1) GO TO 3000
IF1 = I - 1
DFCV = (ARRAY(2, I) * ARRAY(4, I) - ARRAY(2, IF1) * ARRAY(4, IF1)) * CSV
DWCY = (ARRAY(1, I) + ARRAY(1, IF1)) * (ARRAY(3, I) - ARRAY(3, IF1)) / (2. * 777.9
C * 12.)
IF (I .LT. K1) GO TO 2400
IF (I .LT. K2) GO TO 2500
IF (I .LT. K3) GO TO 2400
GO TO 2600
2400 DMF = 0.0
DMD = 0.0
GO TO 2700
2500 DMF = 0.0
DMD = ARRAY(2, IF1) - ARRAY(2, I)
GO TO 2700
2600 DMF = ARRAY(2, I) - ARRAY(2, IF1)
2700 DMD=0.0
HICV=DMI*CSP*TINT
HOCV=DMD*CSP*TDIS
PDV=PDV+DWCV

C
SET UP ARRAYS FOR DQ, UA, SUM OF DQ
C
XX=DECV+DWCV-HICV+HOCV
ARRAY(6,I)=XX*XMULT
ARRAY(7,I)=ARRAY(6,I)/ARRAY(5,I)
QSUM=QSUM+XX*RPM*60.
ARRAY(6,I)=QSUM

3000 CONTINUE

C
THF SECOND LAW OF THERMODYNAMICS FOR THE COMPRESSOR
C
ENTS=0.0
DO 4000 I=1,NPTS
IF1=I-1
IF(1.EQ.1) GO TO 3950
IF1=I-1
DSC1=ARRAY(2,IF1)*(CSP*ALOG(ARRAY(4,IF1)/TREF)-R*ALOG(ARRAY(1,IF1)/
C /TREF))
DSC2=ARRAY(2,I)*(CSP*ALOG(ARRAY(4,I)/TREF)-R*ALOG(ARRAY(1,I)/
C /TREF))
DSCV=DSC2-DSC1
FI1=FLOAT(I)-1.
X=FI1*DELTA
IF(I.LT.K1) TW=540.+X*100./100.2
IF(I.GE.K1.AND.I.LT.K2) TW=640.
IF(I.GE.K2.AND.I.LT.K3) TW=640.-(X-180.)*180./96.
IF(I.GE.K3.AND.I.LE.NPTS) TW=540.
QSCV=ARRAY(6,I)/(TW*XMULT)
IF(I.LT.K1) GO TO 3100
IF(I.LT.K2) GO TO 3200
IF(I.LT.K3) GO TO 3100
GO TO 3300
3100 DMI=0.0
DMD=0.0
GO TO 3400
3200 DMI=0.0
DMD=ARRAY(2,IF1)-ARRAY(2,I)
GO TO 3400
3300 DMI=ARRAY(2,I)-ARRAY(2,IF1)
DMD=0.0
3400 AVT=(ARRAY(4,I)+ARRAY(4,IF1))/2.
AVP=(ARRAY(1,I)+ARRAY(1,IF1))/2.
AVM=.5*(ARRAY(2,I)+ARRAY(2,IF1))
SICV=(CSP*ALOG(TINT/TFEF)-R*ALOG(AVP/PREF))*DMI
SOCV=(CSP*ALOG(AVT/TFEF)-R*ALOG(AVP/PREF))*DMD

C
XX=DSCV-QSCV-SICV+SOCV
ARRAY(5,I)=ARRAY(6,I)/XMULT/(XX+QSCV)
ARRAY(10,I)=XX/AVM
ARRAY(11,I)=ARRAY(11,I)+ARRAY(10,I)
3950 ARRAY(9,I)=CSP*ALOG(ARRAY(4,I)/TFEF)-R*ALOG(ARRAY(1,I)/PREF)
4000 CONTINUE
ARRAY(5,1)=ARRAY(5,NPTS)

C
SET UP ARRAYS FOR CRANK ANGLE
DO 4500 I=1,NPTS
IF1=I-1
4500 ARRAY(12,I)=FLOAT(IF1)*DELT
IF(IS.EQ.1) GO TO 4890

C
ARRAY(1,I)=PRESSURE ARRAY
C  ARRAY(2,I)=MASS ARRAY
C  ARRAY(3,I)=VOLUME ARRAY
C  ARRAY(4,I)=TEMPERATURE ARRAY
C  ARRAY(5,I)=HEAT TRANSFER AREA ARRAY
C  ARRAY(6,I)=HEAT TRANSFER RATE (BTU/HR)
C  ARRAY(7,I)=HEAT TRANSFER RATE (BTU/SF-HP)
C  ARRAY(8,I)=CUMULATIVE HEAT TRANSFER (BTU)
C  ARRAY(9,I)=SPECIFIC ENTROPY (BTU/R-LB)
C  ARRAY(10,I)=SPECIFIC INTERNAL ENTROPY GENERATION (BTU/R-LB)
C  ARRAY(11,I)=CUMULATIVE INT. ENT. GENERATION (BTU/R-LB)
C  ARRAY(12,I)=CRANK ANGLE ARRAY
C
CONVERSION TO S.I. UNITS

DO 4550 I=1,NPTS
  ARRAY(1,I)=ARRAY(1,I)*1.01325E-01
  ARRAY(2,I)=ARRAY(2,I)*0.4536
  ARRAY(3,I)=ARRAY(3,I)*1.6387E-05
  ARRAY(4,I)=ARRAY(4,I)/1.8
  ARRAY(6,I)=ARRAY(6,I)*2.9307E-04
  ARRAY(7,I)=ARRAY(7,I)*3.1546
  ARRAY(8,I)=ARRAY(8,I)*1.0551

4550 ARRAY(9,I)=ARRAY(9,I)*4.1868

C

WRITE(5,4600)
WRITE(5,4650)
WRITE(5,4700) (ARRAY(12,J),(ARRAY(I,J),I=1,4),J=1,NPTS)
WRITE(5,4800)
WRITE(5,4850)
WRITE(5,4700) (ARRAY(12,J),(ARRAY(I,J),I=6,9),J=1,NPTS)
4600 FORMAT(1H1, 2X,'CRANK ANGLE', 3X,'PRESSURE', 8X,'MASS', 9X,'VOLUME',
        9X,'TEMP')
4650 FORMAT(1H1, 4X,'DEGREE', 6X,'MPA', 9X,'KG', 9X,'CU-M', 7X,
        C' DEGREE K') '//
4700 FORMAT((3X,F7.0,2X,4E14.5))
4800 FORMAT(1H1,2X,'CRANK ANGLE',3X,'H.T. RATE',6X,'H.T. COEFF',4X
C,'H.T. SUM',6X,'SPEC. ENTRPY')
4850 FORMAT(1H1,'4X,'(DEGREE)',5X,'(KW)',8X,'(KW/SQ-M)',7X,'(KJ)',8X,
C,'(KJ/KG-K)')

C PLOT INPUT DATA AND OUTPUT

4855 DO 4860 I=1,29
   ORFDL(1,I)=ORFDL(I)*PCALD
   ORFDL(2,I)=ORADL(I)
   ORFDH(1,I)=ORFDH(I)*PCALD
4860 ORFDH(2,I)=ORADH(I)
   DO 4865 I=1,38
   ORFIL(1,I)=ORFIL(I)*PCALI
4865 ORFIL(2,I)=ORAIL(I)
   DO 4870 I=1,41
   ORFTH(1,I)=ORFTH(I)*PCALI
4870 ORFTH(2,I)=ORAHL(I)
   READ(A,1) XLAB
   CALL PICTR(ORFDH,2,XLAB,XSCL,1,29,2,-1,1004,2,0,0,1)
   CALL PICTR(ORFDL,2,XLAB,XSCL,1,29,2,0,1000,-2,0,0,1)
   DELA=(ORFDH(2,29)-ORFDH(2,1))/28.
   AA=ORFDH(2,1)
   DO 4875 I=1,29
   THETA=AA+DEL*FLOAT(I-1)
   ORFDH(1,I)=PDIST(THETA)
4875 ORFDH(2,I)=THETA
   CALL PICTR(ORFDH,2,XLAB,XSCL,1,29,2,0,0,-2,0,0,1)
   PUSE
   READ(A,1) XLAB
   CALL PICTR(ORFTH,2,XLAB,XSCL,1,41,2,-1,1004,2,0,0,1)
   CALL PICTR(ORFIL,2,XLAB,XSCL,1,38,2,0,1000,-2,0,0,1)
   DELA=(ORFTH(2,41)-ORFTH(2,1))/40.
AA=ORPH(2,1)
DO 4880 I=1,41
THETA=AA+DELA*FLOAT(I-1)
ORPH(1,I)=PI(THETA)
4880 ORPH(2,I)=THETA
CALL PICTR(ORPH,1,1.1,1.1,200,00,0,0,0,0,1)
PAUSE
MOVE=-1
LOOK=1
LABEL=4

C

COMPRESSOR PRESSURE VS. VOLUME
C
READ(8,1) XLAB
CALL QPictR(ARRAY,12,91,XY(1),QX(3),QISCL(1),QLABEL(LABEL),
CQXLAB(XLAB),QLOOK(LOOK),QMOVE(MOVE))
PAUSE
CALL QPictR(ARRAY,12,91,XY(1),QX(3),QISCL(31),QLABEL(LABEL),
CQXLAB(XLAB),QLOOK(LOOK),QMOVE(MOVE))
PAUSE
C

COMPRESSOR TEMPERATURE VS. VOLUME
C
4882 READ(8,1) XLAB
CALL QPictR(ARRAY,12,91,XY(4),QX(3),QISCL(1),QLABEL(LABEL),QXIAB
C(XLAB),QLOOK(LOOK),QMOVE(MOVE))
PAUSE
C

PLOT COMPRESSOR PRESSURE, MASS, VOLUME, TEMPERATURE, HEAT TRANSFER
C
RATES (KW), HEAT TRANSFER RATE (KW/SQ-M), CUMULATIVE HEAT TRANSFER
C
(K), SPECIFIC ENTROPY VS. CRANK ANGLE
C
C
DO 4885 LI=1,9
IF(LL.EQ.3) GO TO 4885
IF(LL.EQ.5) GO TO 4885
READ(8,1) XLAB
1 FORMAT(40A2)
C
CALL QPICTR(ARRAY,12,91,QX(IL),QX(12),QISCL(1),QLABEL(LABEL),
CQXIAR(XLAB),QLOOK(LOOK),QMOVE(MOVE))
PAUSE
4885 CONTINUE
IF(IS.EQ.2) GO TO 5010
C
C SMOOTHING OF SPECIFIC ENTROPY CURVE
4890 DO 4900 II=1,6
     I=NHND(II)
     IF1=I-1
     IIIF1=I1+JF1
     SNHD(1,II)=ARRAY(9,IIIF1)
4900 SFHD(2,II)=ARRAY(12,II1)+FLOAT(IF1)*DEITA
     DO 4905 II=1,7
     I=NHLD(II)
     IF1=I-1
     IIIF1=I1+JF1
     SNHD(1,II)=ARRAY(9,IIIF1)
4905 SFHD(2,II)=ARRAY(12,II1)+FLOAT(IF1)*DEITA
     DO 4950 J=I1P1,12
     JF1=J-1
     JJ=J-1
     THETA=DEITA*FLOAT(JF1)
     DO 4930 I=1,5
     IP1=I+1
     IF(THETA.GE.SNHD(2,I).AND.THETA.LT.SNHD(2,IP1)) GO TO 4935
4930 CONTINUE
4935 SMPR=(SNND(1,IP1)-SNHD(1,I))/(SNND(2,IP1)-SNHD(2,I))
TSH=SNHD(1,I)+SMPR*(THETA-SNHD(2,I))
     DO 4940 I=1,6
IP1=I+1
IF(THETA.GE.SENLD(2,I).AND.THETA.LT.SENLD(2,IP1)) GO TO 4975
4940 CONTINUE
4945 SLOPE=(SENLD(1,IP1)-SENLD(1,I))/(SENLD(2,IP1)-SENLD(2,I))
       TSL=SENLD(1,I)+SLOPE*(THETA-SENLD(2,I))
       TS=(TSH+TSL)/2.
       SENLD(1,JJ)=TS
       SENLD(2,JJ)=THETA
       ARRAY(9,J)=TS
       VY=(TS+R*ALOGARRAY(1,J)/PREF))/CSP
       ARRAY(4,J)=TREF*EXP(VX)
4950 ARRAY(2,J)=ARRAY(1,J)*ARRAY(3,J)/(386.25*ARRAY(4,J)*12.)
       DO 4960 JJ=1,12
            I=NIGHT(JJ)
            IF1=I-1
            I3IF1=I3+IF1
            SENHI(1,I)=ARRAY(9,13IF1)
        4960 SENHI(2,I)=ARRAY(12,I3)+FLOAT(IF1)*DELTA
       DO 4965 IT=1,7
            I=WELL(I)
            IF1=I-1
            I3IP1=I3+IF1
            SENLI(1,II)=ARRAY(9,13IF1)
        4965 SENLI(2,II)= ARRAY(12,I3)+FLOAT(IF1)*DELTA
       DO 4990 J=I3IP1,I4
            JF1=J-1
            JJ=J-I3
            THETA=DELTA*FLOAT(JF1)
       DO 4970 I=1,11
            IP1=I+1
            IF(THETA.GE.SENHI(2,I).AND.THETA.LT.SENHI(2,IP1)) GO TO 4975
        4970 CONTINUE
        4975 SLOPE=(SENHI(1,IP1)-SENHI(1,I))/(SENHI(2,IP1)-SENHI(2,I))
TSH = SENH(1, I) + SLOPE * (THETA - SENH(2, I))
DO 4980 I = 1, 6
IP1 = I + 1
IF (THETA .GE. SENLI(2, I) .AND. THETA .LT. SENLI(2, IP1)) GO TO 4985
4980 CONTINUE
4985 SLOPE = (SENLI(1, IP1) - SENLI(1, I)) / (SENLI(2, IP1) - SENLI(2, I))
TSL = SENLI(1, I) + SLOPE * (THETA - SENLI(2, I))
TS = (TSH + TSL) / 2.
SNTI(1, JJ) = TS
SNTI(2, JJ) = THETA
ARRAY(9, JJ) = TS
VX = (TS + R*PI*LOG (ARRAY(1, J)/PREF)) / CSP
ARRAY(4, J) = TREF*EXP(VX)
4990 ARRAY(2, J) = ARRAY(1, J) * ARRAY(3, J) / (386.25 * ARRAY(4, J) * 12.)
5000 CONTINUE
WRITP(5, 4835) ADJI, ADJD, PDV
4835 FORMAT (1H0, 10X, 3E15.6)
5010 CONTINUE
END

PROGRAM *MAIN* HAS NO ERRORS
COMPRESSOR PRESSURE FUNCTION WRT THETA

FUNCTION CYLP(THETA)
DIMENSION ORAIL(38), ORPIL(38), ORAIH(41), ORPIH(41), ORADL(29),
  ORPDL(29), ORADH(29), ORPDH(29), ARRAY(12, 91)
COMMON DELTA, ADJD, ADJI, ORAIL, ORPIL, ORAIH, ORPIH, ORADL, ORPDL,
  ORADH, ORPDH, ARRAY
M=IFIX(THETA/DELTA)
FM=FLOAT(M)
MP1=M+1
SLOPE=(ARRAY(1,MP1)-ARRAY(1,M))/DELTA
CYLP=ARRAY(1,M)+SLOPE*(THETA-DELTA*(FM-1.))
RETURN
END

PROGRAM CYLP WAS NO ERRORS
COMPRESSOR VOLUME IN TERMS OF CRANK ANGLE AND CLEARANCE VOL VCI
FUNCTION CYLV(THETA, VCI)
THETR = THETA * 1.745329E-02
Z1 = 28125 * SIN(THETR)
Z2 = SIGN(1. - Z1^2)
Z3 = 2.25 * (1 + COS(THETR))
Z4 = 8 * (-1 + Z2)
CYLV = VCL + 7.85398 * (Z3 + Z4)
RETURN
END

PROGRAM CYLV HAS NO ERRORS
HEAT TRANSFER AREA

FUNCTION HTRA(THETA)
DATA XR,XL,BORE,PISD/2.25,8.,3.25,.75/
DATA GAP/2.9325/
THETR=THETA*3.14159/180.
AO=3.14159*(BORE*BORE-PISD*PISD)/4.+GAP*2.*3.14159*(BORE+PISD)
X1=XR*SIN(THETR)/XL
X2=SQRT(1.-X1*X1)
X3=XR*(1.+COS(THETR))
X4=XL*(-1.+X2)
HTFA=(AO+2.*3.14159*(X3+X4)*(BORE+PISD))/144.
RETURN
END

PROGRAM HTRA HAS NO ERRORS
INTAKE ORIFICE PRESSURE DATA INPUT

FUNCTION PINT(THETA)

DIMENSION ORAIL(38), ORPIL(38), ORAIH(41), ORPIH(41), ORADL(29),
C ORPDL(29), ORADH(29), ORPDH(29), ARRAY(12, 91)
COMMON DELTA, ADJ1, ADJ2, ORAIL, ORPIL, ORAIH, ORPIH, ORADL, ORPDL,
C ORADH, ORPDH, ARRAY

PCALI=2.3438
KL=38
KLF1=KL-1
DO 100 I=1, KLF1
TP1=I+1
IF(THETA.GE.ORAIL(I).AND.THETA.LT.ORAIL(IP1)) GO TO 150
100 CONTINUE

150 SLOPE=(ORPIL(TP1)-ORPIL(I))/(ORAIL(IP1)-ORAIL(I))
PINTL=(ORPIL(I)+SLOPE*(THETA-ORAIL(I)))*PCALI
KH=41
KHF1=KH-1
DO 200 I=1, KHF1
IP1=I+1
IY(THETA.GE.ORAIH(I).AND.THETA.LT.ORAIH(IP1)) GO TO 250
200 CONTINUE

250 SLOPE=(ORPIH(IP1)-ORPIH(I))/(ORAIH(IP1)-ORAIH(I))
PINTH=(ORPIH(I)+SLOPE*(THETA-ORAIH(I)))*PCALI
PINT=(PINTL+PINTH)/2.
RETURN
END

PROGRAM PINT HAS NO ERRORS
DISCHARGE ORIFICE PRESSURE

FUNCTION PDIS(THETA)

DIMENSION ORAIL(38), ORPIL(38), ORAIIH(41), ORPIH(41), ORADL(29),
C
ORPDL(29), ORADH(29), ORPDH(29), ARRAY(12,91)
COMMON DELTA, A, D, ADJ, ORAIL, ORPIL, ORAIIH, ORPIH, ORADL, ORPDL,
C
ORADH, ORPDH, ARRAY

PCALD=2.1971
K=29
KF1=K-1
DO 100 I=1, KF1
IP1=I+1
IF(THETA.GE.ORADL(I).AND.THETA.LT.ORADL(IP1)) GO TO 150
100 CONTINUE
150 SLOPE=(ORPDL(IP1)-ORPDL(I))/(ORADL(IP1)-ORADL(I))
PDISL=(ORPDL(I)+SLOPE*(THETA-ORADL(I)))*PCALD
DO 200 I=1, KF1
IP1=I+1
IF(THETA.GE.ORADH(I).AND.THETA.LT.ORADH(IP1)) GO TO 250
200 CONTINUE
250 SLOPE=(ORPDH(IP1)-ORPDH(I))/(ORADH(IP1)-ORADH(I))
PDISH=(ORPDH(I)+SLOPE*(THETA-ORADH(I)))*PCALD
PDIS=(PDISL+PDISH)/2.0
RETURN
END

PROGRAM PDIS HAS NO ERRORS
C INTAKE ORIFICE MASS FLOW RATE (LB/SEC)
FUNCTION DMTI(THETA)
DIMENSION ORAIL(38), ORPI(38), ORPIH(41), ORPIH(41), ORADL(29),
C ORPDL(29), ORADH(29), ORPDL(29), ARRAY(12,91)
COMMON DELTA, ADJL, ADJH, ORAIL, ORPIL, ORAIH, ORPIH, ORADL, ORPDL,
C ORADH, ORPDL, ARRAY
BETA = 4.038
YK = 1.662
CNSTI = (41 + 35*BETA**4) / XK
IF(PINT(THETA),LT,0.0) GO TO 100
FY = 1 + CNSTI*PINT(THETA)/CYLP(THETA)
DMTI = FY*129.1*SORT(PINT(THETA))/3600./ADJI
GO TO 200
100 FY = 1 - CNSTI*PINT(THETA)/CYLP(THETA)
DMTI = FY*129.1*SORT(-PINT(THETA))/3600./ADJI
200 RETURN
END

PROGRAM DMTI HAS NO ERRORS
C INTEGRATED MASS FLOW RATE(INTAKE)
SUBROUTINE MASI(ZI,MM)
DIMENSION ORAIL(38),ORPIL(38),ORAIH(41),ORPIH(41),ORADL(29),
C ORPDL(29),ORADH(29),ORPDH(29),ARRAY(12,91)
COMMON DELTA,ADJD,ADJT,ORAIL,ORPIL,ORAIH,ORPIH,ORADL,ORPDL,
C ORADH,ORPDH,ARRAY
NDIV=360
C XX=360.*FLOAT(NDIV)
XX=1.
IJ=IFIX(ZI/DELTA)+1
K=IFIX(DELTA/XX)
TCMI=ARRAY(2,II)
DO 1000 I=1,MM
DELT=XX*FLOAT(I)
AVDM=(DMTI(ZI+DELT)+DMTI(ZI+DELT-XX))/2.
TCMI=TCMI+AVDM*XX/3600.
IF(I.EQ.(I/K)*K) GO TO 990
GO TO 1000
990 IY=I/K+II
ARRAY(2,IX)=TCMI
1000 Continue
Return
End
Program MASI HAS NO ERRORS
C DISCHARGE ORIFICE MASS FLOW RATE(LB/SEC)
FUNCTION DMTD(THETA)
DIMENSION ORAIL(38),ORPIL(38),ORAIH(41),ORPIH(41),ORADL(29),
   ORPD(29),ORADH(29),ORPDH(29),ARRAY(12,91)
C
   COMMON DELTA,ADJD,ADJI,ORAIL,ORPIL,ORAIH,ORPIH,ORADL,ORPD,
   ORADH,ORPDH,ARRAY
RETAD=.45
XK=1.667
CNSTD=-(.41+.35*RETAD**4)/XK
IF(PDIS(THETA).LT.0.0) GO TO 100
FY=1.+CNSTD*PDIS(THETA)/CYLP(THETA)
DMTD=FY*205.35*SQR(PDIS(THETA))/3600./ADJD
GO TO 200
100 FY=1.-CNSTD*PDIS(THETA)/CYLP(THETA)
DMTD=-FY*205.35*SQR(-PDIS(THETA))/3600./ADJD
200 RETURN
END

PROGRAM DMTD HAS NO ERRORS
C INTEGRATED MASS FLOW RATE (DISCHARGE) PFR CYCLE
SUBROUTINE MASD(ZD,NN)
DIMENSION ORAIL(38), ORPIL(38), ORAIH(41), ORPIH(41), ORADL(29),
C ORPDH(29), ORPDH(29), ORPDH(29), ARRAY(12,91)
COMMON DELTA, ADJ, ADJI, ORAIL, ORPIL, ORAIH, ORPIH, ORADL, ORPDH,
C ORADU, ORPDH, ARRAY
NDIV=360
C XX=36C/FLOAT(NDIV)
XX=1.0
II=IFIX(ZD/DELTA)+1
TCMD=ARRAY(2,II)
K=IFIX(DELTA/XX)
DO 1000 I=1,NN
DELT=XX*FLOAT(I)
AVDM=(DMTD(ZD+DELT)+DMTD(ZD+DELT-XX))/2.
TCMD=TCMD-AVDM*XX/3600.
IF (I.EQ. (T/K)*K) GO TO 990
GO TO 1000
990 IX=I/K+II
ARRAY(2,IX)=TCMD
1000 CONTINUE
RETURN
END

PROGRAM MASD HAS NO ERRORS
<table>
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<tr>
<th>CRANK ANGLE (DEGREE)</th>
<th>PRESSURE (MPA)</th>
<th>MASS (KG)</th>
<th>VOLUME (CU-FT)</th>
<th>TEMP (DEGREE K)</th>
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<td>-1.12504E+00</td>
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| 184 | -0.30040E+01 | -0.57024E+05 | -0.13884E+05 | -0.62253E+05 |
| 188 | -0.56892E+00 | -0.10777E+05 | -0.13906E+05 | -0.62262E+05 |
| 192 | -0.74989E+00 | -0.14098E+05 | -0.13935E+05 | -0.62273E+05 |
| 196 | -0.69583E+00 | -0.12372E+05 | -0.13964E+05 | -0.62294E+05 |
| 200 | -0.43490E+00 | -0.80218E+04 | -0.13982E+05 | -0.62292E+05 |
| 204 | -0.29961E-02 | -0.54556E+02 | -0.13992E+05 | -0.62293E+05 |
| 208 | 0.52926E+00 | -0.94932E+04 | -0.13961E+05 | -0.62286E+05 |
| 212 | 0.11571E+01 | -0.20404E+05 | -0.13914E+05 | -0.62270E+05 |
| 216 | 0.18409E+01 | -0.31850E+05 | -0.13841E+05 | -0.62243E+05 |
| 220 | 0.25621E+01 | -0.43414E+05 | -0.13738E+05 | -0.62204E+05 |
| 224 | 0.33379E+01 | -0.55305E+05 | -0.13605E+05 | -0.62153E+05 |
| 228 | 0.41321E+01 | -0.66839E+05 | -0.13439E+05 | -0.62086E+05 |
| 232 | 0.49277E+01 | -0.77090E+05 | -0.13242E+05 | -0.62009E+05 |
| 236 | 0.57634E+01 | -0.88800E+05 | -0.13012E+05 | -0.61916E+05 |
| 240 | 0.65861E+01 | -0.98372E+05 | -0.12748E+05 | -0.61807E+05 |
| 244 | 0.74214E+01 | -0.10772E+06 | -0.12451E+05 | -0.61683E+05 |
| 248 | 0.82541E+01 | -0.11635E+06 | -0.12121E+05 | -0.61543E+05 |
| 252 | 0.90750E+01 | -0.12415E+06 | -0.11758E+05 | -0.61386E+05 |
| 256 | 0.99053E+01 | -0.13146E+06 | -0.11362E+05 | -0.61212E+05 |
| 260 | 0.10743E+02 | -0.13829E+06 | -0.10932E+05 | -0.61021E+05 |
| 264 | 0.11645E+02 | -0.14537E+06 | -0.10466E+05 | -0.60811E+05 |
| 268 | 0.12688E+02 | -0.15362E+06 | -0.99589E+05 | -0.60579E+05 |
| 272 | 0.14043E+02 | -0.16493E+06 | -0.93972E+05 | -0.60319E+05 |
| 276 | 0.16535E+02 | -0.18847E+06 | -0.87357E+05 | -0.60009E+05 |
| 280 | 0.82686E+02 | -0.91508E+05 | -0.84050E+05 | -0.59853E+05 |
| 284 | 0.79949E+02 | -0.85966E+05 | -0.80852E+05 | -0.59696E+05 |
| 288 | 0.68518E+02 | -0.71637E+05 | -0.78111E+05 | -0.59567E+05 |
| 292 | 0.65954E+02 | -0.67069E+05 | -0.75477E+05 | -0.59447E+05 |
| 296 | 0.46193E+02 | -0.45790E+05 | -0.73629E+05 | -0.59366E+05 |
| 300 | 0.45529E+02 | -0.44017E+05 | -0.71800E+05 | -0.59290E+05 |
| 304 | 0.31892E+02 | -0.30168E+05 | -0.70532E+05 | -0.59239E+05 |
| 308 | 0.34811E+02 | -0.32133E+05 | -0.69139E+05 | -0.59186E+05 |
| 312 | 0.32722E+02 | -0.29574E+05 | -0.67830E+05 | -0.59139E+05 |
| 316 | 0.29143E+02 | -0.25226E+05 | -0.66665E+05 | -0.59098E+05 |
| 320 | 0.36716E+02 | -0.31952E+05 | -0.65196E+05 | -0.59048E+05 |
| 324 | 0.37024E+02 | -0.31591E+05 | -0.63715E+05 | -0.59001E+05 |
| 328 | 0.35144E+02 | -0.29636E+05 | -0.62309E+05 | -0.58958E+05 |
| 332 | 0.39645E+02 | -0.32989E+05 | -0.60723E+05 | -0.58910E+05 |
| 336 | 0.40343E+02 | -0.33183E+05 | -0.59115E+05 | -0.58862E+05 |
| 340 | 0.41223E+02 | -0.33575E+05 | -0.57461E+05 | -0.58814E+05 |
| 344 | 0.53638E+02 | -0.43334E+05 | -0.55315E+05 | -0.58750E+05 |
| 348 | 0.62984E+02 | -0.50564E+05 | -0.52796E+05 | -0.58672E+05 |
| 352 | 0.61328E+02 | -0.49013E+05 | -0.50342E+05 | -0.58595E+05 |
| 356 | 0.49644E+02 | -0.39567E+05 | -0.48356E+05 | -0.58531E+05 |
| 360 | 0.46931E+02 | -0.37371E+05 | -0.46479E+05 | -0.58467E+05 |
Input data $P(\theta)$, $V(\theta)$, TBOM

$\Delta P(\theta)$ of orifices

Calculate mass flow rate using orifice equation

$\dot{m}(\theta) = f(\Delta P(\theta))$

Integrate $\dot{m}(\theta)$ during intake and discharge to get mass flow per cycle $m_f$ from

$m_f(\theta) = \int \dot{m}(\theta) \, d\theta / \omega$

Calculate clearance mass $m_c$ from

$m_c = PV / (R \cdot TBOM) - m_f(\theta_o)$

Calculate mixed mean temperature $T(\theta)$ from

$T(\theta) = P(\theta) \cdot V(\theta) / M(\theta)$

Calculate specific entropy $s$ of the gas from

$s = s_{ref.} + c_p \log \left( \frac{T}{T_{ref.}} \right) - R \log \left( \frac{P}{P_{ref.}} \right)$

Smoothe out the entropy curve
In order to decrease the mismatch among $P, V, m, T$ data

Calculate new mixed mean temperature heat transfer rate etc., from the new entropy valve

Fig. E-1 Flow chart for the First Law Integral Analysis program
Fig. E-3 Compressor pressure vs. volume - input data
Intake orifice pressure drop (kPa)

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Fig. E-4 Intake orifice pressure drop vs. crank angle
- input data
Fig. E-5 Discharge orifice pressure drop vs. crank angle - input data
Fig. E-6 Comparison of calculated mixed mean temperature and thermocouple recordings.

* See Appendix E
Fig. E-8 Mass of the gas vs. crank angle for compressor
Fig. E-9 Specific entropy of the gas vs. crank angle for compressor
Fig. E-10 Heat transfer rate vs. crank angle for compressor
Cumulative heat transfer per hour (kJ)

- after entropy curve smoothing
- before entropy curve smoothing

Fig. E-11 Cumulative heat transfer vs. crank angle for compressor
BIOGRAPHICAL NOTE

The author was born in Kyunggi do, Korea on January 9, 1945. He attended Kyunggi Middle School, and Kyunggi High School in Seoul, Korea.

In 1964, he entered Seoul National University, and received B.S. in mechanical engineering in 1972. Between 1965 and 1968, he served in the army of the Republic of Korea.

In 1972, he came to the United States and entered Clarkson College of Technology, Potsdam, N.Y., and took graduate courses for one year. He entered M.I.T. in 1973, and received S.M. degree in 1974, specializing in Acoustics. He held research assistantships for four and half years at M.I.T.

The following paper was co-authored by him: