**MEASUREMENTS OF IN-CYLINDER HEAT TRANSFER
WITH INFLOW-PRODUCED TURBULENCE**

by

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(ABSTRACT)

Heat transfer in cylinder spaces is important to the performance of many reciprocating energy conversion machines, such as gas compressors and Stirling machines. Work over the past 10 years has shown that heat transfer driven by oscillating pressure differs from steady state heat transfer in magnitude and in phase; in-cylinder heat transfer under this oscillating condition can be out of phase with the temperature difference. For studies with closed piston-cylinder gas springs, this heat transfer phase shift has been successfully predicted with the use of a complex Nusselt number; since a complex number has both a magnitude and a phase, a complex Nusselt number can describe the phase shift between temperature and heat transfer. Quasi-steady heat transfer models, such as Newton's Law of Cooling, do not predict this phase shift.

This project studied the problem of in-cylinder heat transfer with inflow-produced turbulence. Initial tests were conducted without the generated turbulence; this enabled the researchers to compare the results of this apparatus to previous work. Then, an orifice plate was added to the apparatus to generate simulated inflow-produced turbulence. The tests from this configuration were compared to the previous set, without the turbulence, to see how inflow-produced turbulence affected heat transfer and the heat transfer related cyclic lost work. The complex Nusselt number, which had been used in previous studies to model non-turbulent in-cylinder heat transfer, was applied to the turbulent data as well.

The tests conducted without generated turbulence (one space experiments) matched previous results and also extended their range to lower volume ratios and higher oscillating speeds. These tests also demonstrated that an analytical heat transfer model based on low volume ratios (approaching 1.0) was valid over the range from 1 to 2.

The tests conducted with the generated turbulence (two space experiments) were

compared against the results from the one space experiments. These results indicated that the in-cylinder heat transfer was increased due to the generated turbulence. The magnitude of the complex Nusselt number compared favorably to an analytical model of in-cylinder heat transfer with inflow-produced turbulence.

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CHAPTER 1

INTRODUCTION

This work describes the initial experimental efforts at Virginia Polytechnic Institute & State University in the area of Stirling engines. The particular purpose of these experiments was to study cyclic heat transfer to and from cylinder walls and to see how gas inflow and outflow would affect this heat transfer. Also of importance was the study of heat transfer related hysteresis loss. Heat transfer related loss is known to decrease engine performance. However, no studies have been conducted to see how gas inflow and outflow may affect this particular thermodynamic loss.

1.1 IN-CYLINDER HEAT TRANSFER

For many years the engine performance of Stirling engines has been lower than what design codes have predicted. It had been thought that this discrepancy was due to the underestimation of fluid viscous losses; thus, it was common to use pressure drop weighing factors to calibrate codes against engine data. However, when this was done it was found that the code which was calibrated for one machine did not accurately predict the performance of another machine (Tew, et al. 1990). In recent years, researchers have begun to look at the importance of heat transfer when predicting engine performance.

In machines with oscillating flow and oscillating pressure, heat is transferred to and from the wall, known as in-cylinder heat transfer, resulting in a thermodynamic loss. Currently, heat transfer related hysteresis loss is seen as an important factor effecting engine performance. There currently exist many models of heat transfer and of heat transfer hysteresis loss. The more that is understood in these two areas, the better engine designers can minimize the heat transfer hysteresis loss to improve engine performance.

1.2 MOTIVATION/OBJECTIVE

In-cylinder heat transfer is an important factor affecting Stirling engine performance.

Previous work has resulted in experimentally verified models for heat transfer and heat transfer related loss in gas springs. Models have been developed to extend these results to in-cylinder heat transfer with inflow and outflow. Experiments have not been conducted to test the validity of these newer models.

The objectives of this research project are as follows:

1. To measure averaged in-cylinder heat transfer and heat transfer hysteresis loss in a gas spring and compare these results to previous work.
2. To measure surface averaged in-cylinder heat transfer and heat transfer hysteresis loss in a cylinder with inflow and outflow. The cylinder should be designed to simulate Stirling engines in both engine geometry and appropriate non-dimensional parameters.
3. To compare the experimental measurements for oscillating flow with inflow and outflow against existing heat transfer models.

1.3 EXPERIMENTAL CONCEPT

A simplified schematic of the apparatus built to measure in-cylinder heat transfer is shown in Figure 1.1. The apparatus is a piston-cylinder arrangement, with the reciprocating motion of the piston causing an oscillating pressure flow. The oscillating volume causes the pressure in the cylinder to vary approximately sinusoidally with time. Due to the oscillating pressure flow, the gas temperature, the cylinder wall temperature, and the wall to gas heat transfer also vary approximately sinusoidally. Turbulence is induced in the flow due to a change in cross-sectional area at the placement of an orifice plate located just above the top center position of the piston.

Figure 1.1:
Oscillating Pressure Apparatus

1.4 BACKGROUND: STIRLING ENGINES

A Stirling engine is a regenerative, reciprocating engine with a gas-phase working fluid and external heat source. A Stirling engine is divided into an expansion space with a heater, a compression space with a cooler, and a regenerator between the two spaces. The expansion space is often referred to as the hot end because the heating takes place there, while the compression space may be referred to as the cold end. The piston in a Stirling engine is driven by the pressure difference between these two spaces.

Stirling engines have many advantages over other engines. For any given heat input temperature and heat rejection temperature, the ideal Stirling cycle has the highest possible efficiency. Because they do not have valves or high speed flows, Stirling engines are less noisy than other engines. Since the combustion takes place at low pressures external to the engine, the Stirling engine also has better emission properties than internal combustion engines.

1.4.1 STIRLING CYCLE

The ideal Stirling cycle is a reversible process. P-V and T-s diagrams are shown in Figure 1.1 with a diagram showing piston position at the extreme points in the cycle (Urieli and Berchowitz, 1984). The external heat source is applied to the expansion space, while the compression space is cooled. The processes are:

1-2 : Isothermal Compression Stroke. Heat is removed from the engine at a constant temperature, the cold sink temperature, and work is added to perform the compression.

2-3 : Constant Volume Displacement. A constant volume process in which the working fluid passes from the cool end to the hot end across the regenerator. The working fluid picks up the heat that was deposited in the regenerator during the process 2-3.

3-4 : Isothermal Expansion Stroke. Isothermal expansion process in which heat is added to the engine at the hot source temperature. Work, greater than that in process 1-2, is obtained.

4-1 : Constant Volume Displacement. A constant volume process in which the hot fluid, from the hot end, is passed through the regenerator. The working fluid is cooled by giving up heat to the regenerator.

Due to the regenerator, the amount of heat that needs to be added by the external heat source is minimized. Even though a great deal of heat is being transferred in processes 2-3 and 4-1, externally the engine is adiabatic during these processes.

The regenerator is an integral part of the Stirling cycle. The regenerator is a heat exchanger and a heat store, and is usually made by stacking together fine wire screens to form a kind of metallic sponge (West, 1986). As mentioned above, the working fluid passes through the regenerator twice during the cycle. The regenerator stores heat when the working fluid is hot and gives the heat back to the working fluid when the fluid is cold.

Another part of the Stirling cycle which can not be overlooked is the external heat source. The heat source is applied to the hot end to add heat to the gas to return it to the hot source temperature. Heat sources used have been (Ross, 1988):

- solar
- nuclear
- biomass combustion
- liquid-fuel combustion
- solid-fuel combustion
- exothermic chemical reactions
- thermal energy storage

1.4.2 KINEMATIC/ FREE PISTON MACHINES

Stirling engines either have free piston or kinematic drives (Urieli and Berchowitz, 1984). A kinematic drive employs a series of mechanical elements (cranks, connecting rods, flywheels) which move together to vary the working spaces in a prescribed manner. This is the conventional manner in which reciprocating engines transfer their thermodynamic work to an output shaft. For a kinematic drive, the efficiency is determined by the operating conditions and the way that the volume space changes with time.

A free piston drive uses the variations in the pressure of the working fluid to move the reciprocating elements. Work is removed by devices such as linear alternators or hydraulic pumps. In the free piston drives the thermodynamics and the mechanical dynamics are strongly coupled.

Figure 1.2: Stirling Cycle

1.5 APPLICATIONS FOR STIRLING ENGINES

Stirling engines have been considered for a great number of applications. The Stirling engine provides high efficiency, lower emissions, and less noise than its counterparts. However, very few Stirling machines are currently on the market. Unfortunately, start up costs are very great and the development of modern Stirling technology is rather recent. More established engines (IC engine, Rankine cycle, and Brayton cycle) have not been displaced by the Stirling engine.

1.5.1 AUTOMOTIVE

Automobile engines typically use fossil fuels as a fuel source; however, fossil fuels are limited, and global governments desire to eliminate the pollution created from burning them. The Stirling engine would be a suitable replacement since it can run on many sources of heat and causes little pollution when burning fossil fuels.

The Automotive Stirling Engine Program, funded by NASA, began in 1978 and finished in 1989. The program resulted in the MOD I and MOD II Stirling engines, both of which were placed in vehicles and road tested. The MOD I was placed in both a van and a pickup truck and accumulated 30,500 road miles, while the MOD II engine was placed in a US Postal vehicle and accumulated 9,400 miles (Ross and Dudenhoefer 1991). The MOD II engine had a fuel economy of between 37 and 41 mpg, which was better than the US fleet average of 27 mpg (Tabata, 1989). However, domestic auto manufacturers seem to be reluctant to use Stirling technology; this reluctance has been attributed to the high initial costs of retooling the production plants for mass production.

In 1981 the Japanese also initiated a program to study the feasibility of using Stirling engines in automobiles. They tested a prototype Stirling engine up to speeds of 81 mph in a Toyota Corolla. They found the noise level, power response, and speed control were comparable to that of an IC engine. Once again, the monetary cost was cited as a setback to developing the Stirling engine on a commercial basis (Kondoh and Katoh, 1988).

1.5.2 SPACE APPLICATIONS

In 1977 NASA-Lewis started working on a project using a free piston Stirling engine

to provide power in space. At that time, space power was supplied by photovoltaic and electrochemical storage systems. It was estimated that future power requirements (needed for orbiting space platforms) would reach 100 kW_e, which current power units could not supply. Stirling engines were thought to be a better source of power since the engine could be powered by solar energy or by nuclear energy.

In 1989 the SPRE-100 engine was developed which ran at a peak temperature of 1050 K. As of 1989, NASA was initiating a program to build another Stirling engine to provide the power requirements and which would operate at the design goal of 1300 K (Slaby, 1989).

Between 1985 and 1991 seven space missions contained Stirling equipment, with the Stirling cryocooler being the most predominant application (Dudenhoefer and Winter 1991). The space station Freedom design includes several Stirling cycle units on board for various freezing purposes. Stirling refrigeration units provide the facilities for two cryogenic freezers (for freezing and storage of specimens) operating at -196 °C, a freeze drier for on-orbit preservation of research samples, and a freezer for general storage operating at -70 °C (Smith, Dean, and Angeli, 1990).

1.5.3 MARINE APPLICATIONS

Large submarines have been powered by nuclear power, since only nuclear powered vessels have had the ability to stay fully submerged for days or weeks. However, nuclear power is both expensive and is seen by the public, at large, to be unsafe.

General Motors and NV Phillips of Holland both investigated the use of Stirling engines for submarine and torpedo applications in the 1960's and early 1970's. The engines were powered by liquid-metal combustion or with thermal storage. The liquid-metal combustion engine had a rated output of several hundred kW, while the thermal storage Stirling only 4.5 kW. The projects were dropped due to unsafe heat transfer from the heat source units (with violent chemical reactions from the liquid-metal combustion) and from lack of funding. In the 1980's, France, Britain, and the US once again looked at Stirling engine powered torpedoes. However, the projects were dropped because the liquid-combustion Stirling systems were heavier than existing turbine systems (Reader, Walker, Hawley 1990). The prospect of a Stirling driven submarine is still a very attractive prospect to many countries. Since there is no combustive explosion or the opening and closing of

valves in the Stirling engines, a Stirling system would run much quieter. In military operations a quieter engine would make a submarine harder to locate and more effective.

Sweden currently employs Stirling engines, developed by Kockmus AB, in some of its smaller submarines. Two 75 kW Stirling engines which use Helium as the working fluid are used to power the submarines. The submarines have liquid oxygen on board, which is combined with recirculated exhaust gas, to provide combustion gas for the diesel fuel. The submarine can submerge to depths of 800 ft. where the excess exhaust gas can be discharged. The vessel can be expected to stay submerged for up to 3 weeks (Ross, 1988).

1.5.4 HEAT PUMPS/ CRYOCOOLERS

1.5.4.1 HEAT PUMPS

Stirling cycle machines are also being looked at in place of vapor compression systems, most of which use chlorofluorocarbons (CFC's) as refrigerants. There currently exist concerns that these refrigerants, when exposed to the atmosphere, destroy the ozone layer. Stirling cycle machines employ no such refrigerants. The Stirling heat pump can operate over much higher temperature lifts than similarly sized vapor compression cycles which eliminates the need for auxiliary electric heaters in cold climate applications.

One of the few commercially available Stirling cycle machines is a Vuilleumier heat pump. The Japanese developed a gas fired Direct Drive Heat Pump which was available in 1991. The unit was intended to provide heating/cooling and hot water for medium sized shops and offices. The heating COP was 1.5 and the cooling COP was 0.8 (Ross, 1989).

As of 1990 Stirling Thermal Motors have also been developing a heat pump for commercial applications. Eventually, the STM4 series of machines are expected to have COP's comparable with the CFC vapor compression systems (Kahlili & Meijer, 1990).

1.5.4.2 CRYOCOOLERS

Stirling refrigerators capable of achieving cryogenic temperatures (cryocoolers) have been available since 1954. The Stirling cryocoolers have had two established commercial applications:

- the military uses miniature refrigerators, achieving cryogenic temperatures, on their infrared night vision and missile guidance systems
- cryogenic refrigerators are used for the liquefaction of air or nitrogen for laboratory purposes, for metallurgical and food processing, and for specialty applications.

Stirling cryocoolers have the ability to deliver temperatures of 120 K or less. One of the future applications of the cryocoolers would be the cooling of superconducting equipment. Near superconductivity is achieved at 80 K (Walker, et al, 1988).

1.5.5 MEDICAL APPLICATIONS

A research project started in the mid 1960's by the National Institute of Health was devoted to the development of an implantable, Stirling powered artificial heart. The researchers achieved a module operating at 5 Watts that demonstrated a continuous, maintenance free life of 4 years. The heat source for the Stirling engine was either a rechargeable thermal store or radio-isotope decay (West, 1986). Later work concentrated on decreasing the size of the module. The current Stirling implants were expensive due to the necessary biocompatibility and because of the necessary effective insulation to protect the patient from the hot end of the engine (Ross, 1988).

1.5.6 SOLAR

Stirling engines have been used to produce electric power from solar power. These engines have shown greater efficiency than the traditional photovoltaic methods. The concept of using Stirling engines in power conversion have been studied for terrestrial and space applications.

The dish Stirling electric system consists of a Stirling engine mounted to a parabolic dish solar concentrator. The advantage of using the Stirling engine is that the solar energy can be concentrated directly onto the heater of the engine (West, 1986). Dish electric systems using Stirling engines were successfully demonstrated by Advanco Corp. and McDonnell Douglas Corp. between 1984 and 1986. Both of the efforts showed the potential for high efficiency (Andraka & Diver 1988).

Studies have also been conducted to evaluate orbiting solar heat receivers with Stirling engine cycles. One of the promising studies involved a Stirling heat pipe receiver.

In this concept, the sunlight strikes the unit and melts a fluid within the heat pipe. Condensation of the heat pipe fluid occurs in a thermal storage device, which melts a phase change material. The latent heat of the phase change material is then absorbed by the working fluid in the Stirling engine heater. Initial analysis indicated an efficiency of 29% (Strumpf, Coombs, Lacy 1988). However, a closed Brayton cycle was later chosen over the free piston Stirling engine, because the Brayton cycle was a proven technology and seen to be more reliable. This choice was made even though the Stirling engine was proven to be more efficient and had the characteristics of lower weight and decreased volume (Friefeld & Wallin, 1989).

1.6 LITERATURE REVIEW

There exists a small body of previous work on heat transfer and loss in oscillating pressures and oscillating flows. The work presented here was research conducted on Stirling engines and gas springs (a volume of gas confined by a fixed cylinder and a moving piston). Some related studies pertaining to internal combustion engines will not be presented.

In-cylinder heat transfer is a major source of inefficiency in Stirling machines and gas springs. This heat transfer is seen as an irreversible thermodynamic process which causes lost work, sometimes called hysteresis loss. Most of the work in Stirling machines has been devoted to studying the loss and trying to improve engine efficiency. Later work in heat transfer in oscillating pressures and flows has been concerned with the phase difference between heat transfer and temperature which occurs under these conditions. Some researchers have looked into the use of a complex Nusselt number, or complex heat transfer coefficient, to predict the heat transfer in oscillating flows.

1.6.1 LOSS STUDIES

If the compression and expansion processes in a gas spring were truly adiabatic then there would be no cyclic lost work and $\oint PdV = 0$. All of the work done on the gas spring during compression would be recovered during expansion; however, this never occurs. When the gas in the cylinder is hotter than the cylinder walls, heat is transferred to the wall. When the gas is cooler than the wall, heat is transferred to the gas. This is an

irreversible thermodynamic process which causes work to be lost, $\oint PdV < 0$.

URIELI AND BERCHOWITZ

Urieli and Berchowitz (1984) analyzed gas spring hysteresis loss. They found that the loss was a function of temperature gradient effects, and may have a serious affect on free-piston machines. In free-piston machines the gas spring hysteresis loss could increase damping which would tend to alter the phase angle and the amplitude of the piston, which would strongly affect the power and efficiency of the gas spring. They developed the following expression for the loss based on a heat transfer analysis close to the wall,

$$\bar{W}_{ir} = \frac{k}{4} \sqrt{\frac{\omega}{2\alpha}} \gamma (\gamma - 1) T_w A_w \left(\frac{\Delta V}{V_B} \right)^2 \quad 1.1$$

where:

α = thermal diffusivity of the gas evaluated at mean pressure,

T_w = wall temperature,

V_B = mean volume,

k = thermal conductivity,

ΔV = volume amplitude, and

A_w = wetted area.

They determined that, in order to minimize losses, the volume ratio and the wetted area should be kept small and the wall temperature should be kept low.

BRECKENRIDGE, HEUCHLING, AND MOORE

Breckenridge, et al (1971) studied losses based on conductive heat transfer in gas springs. They estimated a "lower limit" magnitude of the losses based on a simplified model of the processes which occur in gas springs. They also compared their experimental model against experimentally obtained loss results. In the model they assumed that 1) the bulk gas temperature varied sinusoidally, 2) the wall temperature was constant, and 3) the heat transfer to the wall was by conduction only. Their analysis resulted in expressions for energy lost from the gas spring during the cycle, energy stored in the gas spring during the

cycle, and the gas spring power loss:

$$EL = -\pi\sqrt{2} \frac{\gamma}{\gamma - 1} \frac{AP_o A_o^2}{T_w^2} \sqrt{\frac{\alpha}{\omega}} \quad 1.2$$

$$ES = -\frac{2\gamma}{(\gamma - 1)^2} \frac{P_o V_o}{T_w^2} A_o^2 \quad 1.3$$

$$GSPL = \frac{\omega}{2\pi} EL \quad 1.4$$

They compared their predicted values of efficiency ($\eta = 1 - EL/ES$) against experimental results of gas springs, and found that the predicted efficiencies were approximately 2.6 times less than the actual efficiency. They thought that the difference was due to heat transfer by convection and from second order effects neither of which were included in the model. The resulting empirically corrected efficiency was given by:

$$\eta = 1 - \frac{EL}{ES} = 2.6 \frac{\pi}{\sqrt{2}} \frac{(\gamma - 1) A}{V_o} \sqrt{\frac{\alpha}{\omega}} \quad 1.5$$

where in equations 1.2 through 1.5

A = surface area,

P_o = mean pressure,

A_o = amplitude of temperature variations,

α = thermal diffusivity of the gas, and

V_o = mean volume.

From their analysis they concluded that:

- 1) losses were proportional to the ratio of surface area to volume, and in order to minimize losses the area of the surfaces in contact with the gas should be minimized,
- 2) low reciprocating speeds favor lower losses, and
- 3) gases with higher values of α and γ would produce higher amounts of loss.

COOKE-YARBOROUGH AND RYDEN

Cooke-Yarborough and Ryden (1985) calculated losses due to imperfect heat transfer when a gas was expanded and compressed in a Thermo-Mechanical Generator (an application of a Stirling engine). They modelled the process as gas flowing in a parallel plane geometry with the cylinder walls at a constant temperature. Because the volume changed sinusoidally, the gas pressure change was treated as being sinusoidal, and was given by:

$$P = P_o + P_a e^{i\omega t} \quad 1.6$$

For the parallel plane geometry, the energy equation simplified to

$$\rho_o c_p \frac{\partial T}{\partial t} = k \frac{\partial^2 T}{\partial x^2} + \frac{\partial P}{\partial t} \quad 1.7$$

The energy equation was solved with the expression for oscillating pressure to solve for the temperature

$$T = T_o + \frac{T_o P_a}{c_p P_o} \left(1 - \frac{2}{\beta h} \tan \frac{\beta h}{2} \right) e^{i\omega t} \quad 1.8$$

where:

T_o = wall temperature,

$\beta = (1 - i) (\omega/2\alpha_o)^{1/2}$,

α_o = thermal diffusivity of the gas calculated from mean gas properties, and

h = gap between the isothermal walls.

The expressions for temperature and pressure and the ideal gas law were used to calculate the loss from $\int PdV$. The final expression for power loss due to imperfect heat transfer was given by:

$$\bar{E} = \frac{P_a^2 \sqrt{2\alpha_o \omega}}{P_o h} \frac{\gamma - 1}{\gamma} \left[\frac{\sinh(\beta h) - \sin(\beta h)}{\cosh(\beta h) + \cos(\beta h)} \right] \quad 1.9$$

where:

P_o = mean pressure and

$z = (\omega/2a_o)^{1/2}$.

FAULKNER AND SMITH

Faulkner and Smith (1983) measured the loss due to the cyclic heat transfer within a closed cylinder, without combustion, with the gas being compressed and expanded. They measured loss from pressure and volume data and presented loss in a non-dimensional form,

$$L_{nd} = \frac{\oint_{cy} P dV}{\oint_c P dV} \quad 1.10$$

where the cyclic lost work was non-dimensionalized against the lost work measured during the compression stroke.

They measured the non-dimensional loss for different engine geometries and working fluids, and plotted this loss against the Reynolds number which was based on mean piston speed and mean gas properties. They found that, at low Re, loss increased monatomically to a point of maximum loss. After this point of maximum loss, they found that loss decreased monatomically with Re. They also found that, for a common Re in the range of maximum loss, points of high speed-low pressure had a lower loss than points of low speed-high pressure. They found that the point of maximum loss was usually centered around $Re = 100$ and that the value of maximum loss increased with increasing specific heat ratio. They also found that the overlap region, where there were different losses measured for different Re, occurred at different Re for different gases.

LEE

Lee (1983) developed a simplistic model of in-cylinder heat transfer. This model was essentially identical to the model developed by Cooke-Yarborough and Ryden, but the two models were developed independently. Lee modelled the flow in the cylinder as a layer

of an ideal gas between two parallel plates at a temperature T_o which oscillated in the transverse direction against each other. He developed the following expression for loss:

$$\Delta \dot{W} = -mRf \int_0^\tau \frac{T_B}{P} \frac{dP}{dt} dt \quad 1.11$$

where T_B , the bulk temperature (which was calculated from the energy equation), and P were defined as:

$$T_B = T_o + T_1 \epsilon \cos(\omega t + \phi_T) \quad 1.12$$

$$P = P_o + P_1 \cos(\omega t) \quad 1.13$$

where in equations 1.11 through 1.13:

m = mass of gas,

R = gas constant,

f = frequency of oscillations,

τ = period of oscillations,

P_1, T_1 = amplitude of pressure and temperature,

ϵ = temperature wave amplitude modification factor, and

ϕ_T = phase angle of bulk gas temperature.

The final form of the power loss was given by:

$$\Delta \dot{W} = -\frac{\omega}{4} P_1 V_o \frac{\gamma - 1}{\gamma} \frac{P_1}{P_o} \left[\frac{\cosh(y) \sinh(y) - \sin(y) \cos(y)}{y(\cosh^2(y) \cos^2(y) + \sinh^2(y) \sin^2(y))} \right] \quad 1.14$$

where:

$$y = D_h \omega^{1/2} / 4(2\alpha)^{1/2},$$

D_h = hydraulic diameter ($4 \cdot V/A$),

α = thermal diffusivity of the gas, and

V_o = mean volume.

KORNHAUSER AND SMITH

Kornhauser and Smith (1987) conducted experiments on a modified version of the apparatus used by Faulkner and Smith (1983). They measured loss and non-dimensionalized it with the adiabatic compression work, and then plotted the data against the cyclic averaged Peclet number. The non-dimensional loss and the cyclic averaged Peclet number were given by:

$$L_{nd} = \frac{\oint PdV}{W_{adiab}} = \frac{\oint PdV}{\frac{P_{max}V_{min} - P_{min}V_{max}}{\gamma - 1}} \quad 1.15$$

$$Pe = \frac{V_p D}{\alpha_o} \quad 1.16$$

where:

V_p = mean piston speed,

D = cylinder diameter, and

α_o = gas thermal diffusivity calculated from mean properties.

These non-dimensional loss measurements agreed well with those of Faulkner and Smith (1983). At low Pe , the loss increased with increasing Pe . They found that this increase was proportional to Pe . Peak loss occurred approximately at $Pe = 70$; after this point, loss decreased proportionally to $Pe^{-1/2}$ with increasing Pe . As with the results of Faulkner and Smith, there existed a breakdown region near maximum loss.

Kornhauser and Smith (1993) extended the data set to different volume ratios, gases, and bore-stroke ratios. Based on the expressions of Lee (1983) and Cooke-Yarborough and Ryden (1971), Kornhauser used a different term to non-dimensionalize the loss than the term used by Kornhauser and Smith (1987). Kornhauser defined the non-dimensional loss as:

$$LOSS_{nd} = \frac{\oint PdV}{P_o V_o \left(\frac{P_a}{P_o}\right)^2 \frac{\gamma - 1}{\gamma}} \quad 1.17$$

He then plotted the non-dimensional loss against the oscillating Peclet number, as opposed to the cyclic averaged Peclet number, which he defined as:

$$Pe_{\omega} = \frac{\rho_o c_p \omega D_h^2}{4k} \quad 1.18$$

where in equations 1.17 and 1.18:

$$P_a/P_o = (P_{\max} - P_{\min})/(P_{\max} + P_{\min}),$$

P_o , V_o , ρ_o = mean pressure, volume, and density,

c_p = specific heat at constant pressure,

D_h = hydraulic diameter = $4 \cdot \text{Volume}/\text{Surface Area}$, and

k = thermal conductivity.

The results from the expanded data set matched the results of Kornhauser and Smith as far as the shape of the loss curve, although they were scaled differently due to the different expressions for non-dimensional loss. Once again, the loss increased proportionally to Pe_{ω} with increasing Pe_{ω} to a point of maximum loss, which occurred at a Pe_{ω} approximately of 30. After this point, the loss decreased proportionally to $Pe_{\omega}^{-1/2}$ as Pe_{ω} was increased. As with the previous results, the correlation broke down in the region of maximum loss. The correlation brought together the data made at different bore-stroke ratios and the data taken with different gases; however, the data taken with different gases and at different volume ratios did not correlate well. The loss at higher volume ratios was higher than lower volume ratios.

RECKTENWALD, RAMSEY, AND PATANKAR

Recktenwald, Ramsey, and Patankar (1990) developed a computational model to study oscillating flow within a gas spring. The goals of the model were to simulate the experiments of Kornhauser and Smith, in order to validate the model, and to provide detailed information on the flow and the heat transfer occurring within the gas spring. The hysteresis loss predictions of the computer model compared favorably to the experimental measurements of Kornhauser and Smith; however, the model did not predict the breakdown region at intermediate Pe_{ω} , where higher loss was measured for points of high speed-low

pressure had lower loss than points of low speed-high pressure.

1.6.2 HEAT TRANSFER STUDIES

Several projects have been conducted to study the heat transfer within Stirling engines and gas springs. Since in-cylinder heat transfer is a known cause of thermodynamic loss, some of the studies presented in the previous section also studied heat transfer. These particular studies will also be presented in this section. There are other studies which only looked at in-cylinder heat transfer. Of particular importance to this project are the studies of the heat transfer in oscillating pressures and oscillating flows with inflow and outflow.

Many studies of in-cylinder heat transfer revealed that heat transfer was out of phase with the temperature difference. Therefore, heat transfer expressions which do not incorporate this phase difference, such as Newton's law of cooling, are not applicable to the study of heat transfer in oscillating flows. Some researchers have begun to investigate the use of a complex Nusselt number, or a complex heat transfer coefficient, to study heat transfer under these conditions, since these complex Nusselt number models incorporate the phase difference between heat transfer and temperature.

1.6.2.1 HEAT TRANSFER IN OSCILLATING FLOW

WENDLAND

Wendland (1968) studied unsteady heat transfer from a close gas system to a plane metal surface, with the gas experiencing periodic pressure and temperature fluctuations. He measured wall surface heat flux and then compared that to the results of a model he developed, called the adiabatic plane model, which simulated the flow based on the following assumptions:

- 1) the system was one-dimensional,
- 2) the head surface temperature was constant,
- 3) there existed a plane midway between the piston and the head at which point the gradient of the gas temperature was zero, and
- 4) the fluid pressure was a function only of time.

The model was solved with a finite difference technique.

Wendland found that surface heat flux increased with increasing frequency and that, due to the compression and expansion of the gas, the surface heat flux led the surface temperature by 9 to 11°. He also showed that, with a positive temperature difference between the gas and the wall, heat could be transferred from the wall to the gas. He concluded that using Newton's law of cooling was doubtful since the heat transfer coefficient assumed negative, zero, and infinite values.

LEVINE AND OTIS

Levine and Otis (1980) conducted experiments on a 2.5 liter piston-cylinder enclosure containing nitrogen at pressures of 50 to 200 bar. The gas was compressed and expanded by sinusoidal piston motion.

Levine and Otis found that the heat flow varied periodically, changing from positive (heat outflow) to negative (heat inflow). The heat inflow occurred near the bottom of the cycle (where the gas temperature was lower than the wall temperature) and the heat outflow near the top of the cycle. They also found that the flow within the cylinder showed a transition from laminar to turbulent during each cycle. The turbulence was more prominent at lower frequencies even though the flow at higher frequencies had a higher Rayleigh number, which would indicate a higher level of turbulence at these frequencies. The researchers credited the decrease in turbulence at high frequencies to the decreased time for a complete cycle to occur. Levine and Otis argued that, at high frequencies, the duration of the cycle was too quick for turbulence to develop.

CHONG AND WATSON

Chong and Watson (1976) presented a numerical procedure for predicting heat transfer in cylinders of slow speed reciprocating air compressors. The predictions were made for a machine with a bore of 8.23 cm and a stroke of 12.7 cm ($D/L_s = 0.648$), $T_o = 373$ K, and $\omega = 1.05$ and 62.8 rad/sec. The temperature pressure, heat flux rates, and velocities within the cylinder were predicted for the compression stroke.

The numerical model predicted that, for both of the angular velocities listed, the heat flux was a maximum prior to the point where the temperature difference was a

maximum. The model also predicted differences between the heat flux at the two angular velocities. They found that the convective component of heat transfer was suppressed at higher angular velocities ($\omega \geq 10.5$ rad/sec) and that the heat transfer could be considered as a conduction only problem; but, the convective component of heat transfer could not be neglected at lower angular velocities. They also found that at $\omega = 62.8$ rad/sec the peak pressure was only 7% greater than at $\omega = 1.05$ rad/sec, but the peak heat flux increased by 30%.

1.6.2.2 COMPLEX NUSSELT NUMBER STUDIES

PFRIEM

Pfriem (1943) presented a one-dimensional analysis of cyclic in-cylinder gas to wall heat transfer. He solved the one-dimensional energy equation for an ideal gas with oscillating pressure change. He assumed that the cyclic pressure variations were small enough to consider density a constant:

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} + \frac{1}{\rho c_p} \frac{\partial P}{\partial t} \quad 1.19$$

To solve the energy equation, Pfriem used a complex pressure in the form

$$P_c = P_o + \Delta P e^{i\omega t} \quad 1.20$$

Pfriem used the energy equation and the complex pressure as a basis to solve two different oscillating pressure conditions.

In the first case, he solved the problem of a large gas space where the gas varied harmonically with time. He solved for the heat transfer when a very thick laminar boundary layer existed at the wall. His analysis resulted in an expression for the temperature difference between the gas and the wall and for heat transfer,

$$T_g - T_w = \frac{\gamma - 1}{\gamma} \frac{P_a}{P_o} T_o e^{i\omega t} \quad 1.21$$

$$\dot{q}''(t) = P_a \sqrt{\omega \alpha_o} [e^{(\omega t + \frac{\pi}{4})i}] \quad 1.22$$

where:

P_a = amplitude of pressure oscillations,

ω = natural frequency of the pressure oscillation,

α_o = thermal diffusivity calculated from mean properties, and

T_o, P_o = mean temperature and mean pressure.

Pfriem defined a heat transfer coefficient by dividing the complex heat transfer expression by the complex temperature difference which yielded

$$h_c = \frac{\gamma}{\gamma - 1} \frac{P_o}{T_o} \sqrt{\omega \alpha_o} [e^{i\frac{\pi}{4}}] \quad 1.23$$

Pfriem's analysis indicated that, for an oscillating flow with a thick laminar boundary layer, the heat flux would lead the temperature difference by 45° , regardless of the frequency of the pressure variation. He also noted that conventionally solving for a heat transfer coefficient, real heat transfer divided by real temperature difference, would result in values of h varying from $-\infty$ to ∞ .

For the second case, Pfriem divided the gas space into two distinct regions. The boundary layer flow was replaced by an idealized boundary layer, creating a laminar flow parallel to the wall surface, and beyond that a turbulent core. The temperature in the turbulent core was assumed to be uniform throughout. The results for this case were as follows:

$$\begin{aligned}
0 < x < \delta \quad T(x, t) = T_o + T_a \left[1 - \frac{\sinh(\delta - x)\psi + \sinh(x\psi)}{\sinh(\delta\psi)} \right] e^{i\omega t} \\
+ T_{a\infty} \frac{\sinh(x\psi)}{\sinh(\delta\psi)} e^{i\omega t}
\end{aligned} \tag{1.24}$$

$$x > \delta \quad T(x, t) = T_o + T_{a\infty} e^{i\omega t} \tag{1.25}$$

where:

δ = boundary layer thickness,

$\psi = (i\omega/\alpha)^{1/2}$

T_a = temperature amplitude for adiabatic compression,

= $(\gamma - 1)/\gamma (T_o P_a/P_o)$

$T_{a\infty}$ = temperature amplitude in the turbulent gas,

= $T_a [s \psi \sinh(\delta\psi) + \sinh^2(\delta\psi/2)] / [s \psi \sinh(\delta\psi) + \cosh(\delta\psi)]$

s = volume/surface of the turbulent gas.

Differentiating to get wall heat flux and obtaining h_c as above gave

$$\dot{q}'' = T_a \sqrt{i\omega\alpha} \frac{P_o}{T_o} \left(\frac{\gamma}{\gamma - 1} \right) \frac{\sinh(\delta\psi) + s\psi \cosh(\delta\psi)}{s\psi \sinh(\delta\psi) + \cosh(\delta\psi)} e^{i\omega t} \tag{1.26}$$

$$h_c = k\psi \frac{\sinh(\delta\psi) + s\psi \cosh(\delta\psi)}{s\psi \sinh(\delta\psi) + \sinh^2(\frac{\delta\psi}{2})} \tag{1.27}$$

For the case of the turbulent core, Pfriem's analysis indicated that at low frequencies the heat transfer and temperature were in phase. As the frequency was increased, the heat of compression produced within the boundary layer was more important, and the phase angle (between heat transfer and temperature) would increase to the limiting value of 45°.

Note: Equations 1.22 - 1.27 were put into the form shown by Kornhauser (1989).

LEE

Lee (1983) modelled the cyclic heat transfer in an oscillating flow; this was the same model which he used to develop an expression for hysteresis loss. This model consisted of an ideal gas between two parallel walls at temperature T_0 which oscillated in the transverse direction against each other. He solved the 1-D energy equation with a complex pressure term:

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} + \frac{1}{\rho C_p} \frac{\partial P}{\partial t} \quad 1.28$$

where

$$P = P_0 + P_a e^{i\omega t} \quad 1.29$$

The energy equation was solved with the pressure expression for a complex temperature distribution, which was given by

$$T(x, t) = T_0 + T_1 [1 - a_1 e^{(1+i)x/\lambda} - a_2 e^{-(1+i)x/\lambda}] e^{i\omega t} \quad 1.30$$

where:

$$T_1 = T_0 (P_a/P_0) (\gamma - 1)/\gamma,$$

$$P_a = \text{pressure amplitude,}$$

$$P_0 = \text{mean pressure,}$$

$$a_1 = e^{-z}/(e^z + e^{-z}),$$

$$a_2 = e^z/(e^z + e^{-z}),$$

$$z = (1+i)\gamma,$$

$$\gamma = r_h/\lambda, \text{ and}$$

$$\lambda = (2\alpha/\omega)^{1/2}.$$

The complex heat flux, due to the complex temperature difference, was given by

$$\dot{q}'' = -\left(\frac{kAT_1}{\lambda}\right) (1+i) e^{i\omega t} \tanh(z) \quad 1.31$$

The complex heat transfer coefficient was based on the difference between the wall temperature T_o and the complex bulk temperature, where

$$T_b(t) = T_o + T_1 \left(1 - \frac{\tanh(z)}{z}\right) e^{i\omega t} \quad 1.32$$

Thus,

$$h_c = \left(\frac{k}{\lambda}\right) (1+i) \frac{\tanh(z)}{1 - \frac{\tanh(z)}{z}} \quad 1.33$$

Lee's model showed that a phase difference existed between the heat transfer and the temperature difference. At values of $r/\lambda \leq 0.1$ (near isothermal conditions) the heat transfer process was in phase with the temperature difference, while at $r/\lambda \geq 100$ (near adiabatic conditions) the heat transfer would lead the temperature difference by 45° . In between those two values of r/λ the phase difference would gradually increase from 0° to 45° .

KORNHAUSER AND SMITH

Kornhauser and Smith (1988) measured heat flux, from the same pressure and volume data that they used to measure non-dimensional loss, in a piston-cylinder gas spring over a wide range of engine geometries and working fluids. They also predicted heat flux using a complex Nusselt number model. They used the model to predict the phase difference between the heat flux and the temperature difference.

The complex Nusselt number model was based on complex expressions for the heat flux and the temperature

$$\dot{q}''_c = \frac{k}{D_h} Nu_c (T_c - T_w) \quad 1.34$$

where:

D_h was the hydraulic diameter defined as (4 Volume)/(total surface area) and

$$T_c = T_o + T_a \cos(\omega t) + iT_a \sin(\omega t) \quad 1.35$$

The expression to predict the heat flux was then given by:

$$\dot{q}'' = \frac{k}{D_h} [Nu_r (T - T_w) + \frac{Nu_i}{\omega} \frac{dT}{dt}] \quad 1.36$$

where:

Nu_r = the real portion of the complex Nusselt number and

Nu_i = the imaginary portion of the complex Nusselt number.

The predicted heat flux matched the measured heat flux in phase angle and magnitude. The heat flux was out of phase with the temperature difference, as Pfriem's and Lee's work had suggested. At low values of PeD_h/L_s (where Pe was the averaged cyclic Peclet number and L_s was the stroke length) the real part of the complex nusselt number was much larger than the imaginary part, which indicated that the heat flux was nearly in phase with the temperature difference. At high values of PeD_h/L_s , Nu_r and Nu_i were nearly equal, which indicated that the heat flux was out of phase with the temperature difference by nearly 45° . At high PeD_h/L_s , where most Stirling engines operate, a power law expression was fitted to the Nu_c data taken at a volume ratio of 2.0. This expression was given as

$$Nu_r = Nu_i = 0.98 \left(\frac{PeD_h}{L_s} \right)^{0.59} \quad 1.37$$

1.6.2.3 OSCILLATING FLOW WITH INFLOW/OUTFLOW

GEDEON

Gedeon (1989) formulated a model for oscillating heat transfer with turbulence. He derived an analytical expression for the film heat transfer coefficient in a gas undergoing compression and expansion between parallel walls. Gedeon followed Lee's analysis (1983) for oscillating flow with in-cylinder heat transfer, but he replaced the molecular conductivity with a turbulent conductivity term which he assumed would increase linearly with distance from the wall. He calculated the complex Nusselt number, under these flow conditions, to be

$$Nu \approx \frac{0.2R_t}{\ln \left[\frac{0.2R_t}{2\sqrt{2}} \frac{\lambda}{d} \right]} \quad 1.38$$

where:

$$\lambda/d = (2Pe_w)^{1/2} \text{ and}$$

$$R_t = \{\rho [3/8(V_{in})^{1/2}]^{1/2} D_h\}/\mu,$$

D_h = the hydraulic diameter, and

V_{in} = the inlet velocity.

KORNHAUSER

Kornhauser (1992) developed a model to study in-cylinder heat transfer with inflow-produced turbulence. The model extended the work of Pfriem (1943), in which the cylinder space was divided into a laminar boundary layer and a core turbulent region. In the model, Kornhauser assumed that the boundary layer was a function of inflowing gas velocity.

Kornhauser took Pfriem's expression for the complex Nusselt number

$$Nu_c = z \left[\frac{1 - e^{-2\frac{\delta}{D}z} + z\left(\frac{1}{4} - \frac{\delta}{D}\right)(1 + e^{-2\frac{\delta}{D}z})}{(1 - e^{-\frac{\delta}{D}z})^2 + z\left(\frac{1}{4} - \frac{\delta}{D}\right)(1 - e^{-2\frac{\delta}{D}z})} \right] \quad 1.39$$

where:

$$z = (1 + i)(2Pe_w)^{1/2},$$

D = the hydraulic diameter, and

δ/D = a non-dimensional boundary layer thickness.

He then determined the quantity δ/D by using a correlation for the mean Nusselt number on the trailing surface of a cylinder in crossflow. The expression, if the Prandtl number was not assumed to be a function of temperature, was given by

$$Nu = 0.071Re^{0.073}Pr^{0.4} \quad 1.40$$

Next, by assuming that all of the thermal resistance was in the laminar wall layer, he found that for a temperature difference ΔT and thermal conductivity k

$$\dot{q}'' = \frac{NuK}{D}\Delta T = \frac{k}{\delta}\Delta T \quad 1.41$$

For a Prandtl number of 0.67 he solved the two expressions above for δ/D

$$\frac{\delta}{D} = 16.5Re_{in}^{-0.73} \quad 1.42$$

In a parallel plate geometry, the boundary layers growing on the opposite walls will meet when $\delta/D = 1/4$. Therefore, when calculating the thickness of the boundary layer in Pfreim's expression for Nu_c , Kornhauser used the smaller of 1/4 or equation 1.42 for the non-dimensional boundary layer thickness.

Kornhauser then calculated Nu_c for laminar ($Re_{in} \leq 310$, which would set δ/D in equation 1.42 to 0.25) and highly turbulent conditions (large Re_{in}):

$$Nu_{cl} = z \frac{1 + e^{-\frac{z}{4}}}{1 - e^{-\frac{z}{4}}} \quad 1.43$$

$$Nu_{ct} \approx \frac{D}{\delta} = 0.0606 Re_{in}^{0.73} \quad 1.44$$

The model predicted a larger heat transfer magnitude and smaller heat transfer phase shift for turbulent flow than for laminar flow. Kornhauser's model predicted larger heat transfer phase shift and smaller heat transfer magnitude than Gedeon's model.

IBRAHIM, HASHIM, TEW, AND DUDENHOEFER

Ibrahim, et al (1992) developed a numerical model to study oscillating flow with heat transfer between two parallel plates with a sudden change in cross-sectional area. Computations were done over a wide range of Re_{max} , Valensi number, expansion ratio, and fluid displacement. The chosen geometry to study was two flat plates with the flow passing over a step change in area. One of the results that they computed was the instantaneous Nusselt number. They found the instantaneous Nusselt number was much larger, by a factor of 8, at the point of change in the cross-sectional area than elsewhere in the flow.

CHAPTER 2

DESIGN OF APPARATUS

The experimental apparatus was a piston-cylinder arrangement mounted on top of an air compressor, with a DC motor used to drive the compressor. The test piston was attached to the compressor piston causing the test piston to move vertically in the cylinder at the angular velocity of the flywheel. The motion of the test piston caused the working fluid in the cylinder to compress and to expand which forced the pressure, volume, and temperature of the fluid to vary approximately sinusoidally with time. This compression and expansion of the working fluid caused heat transfer within the cylinder. An orifice plate, located near the top center position of the test piston, produced turbulence in the working fluid from flow through the orifice.

The volume of the gas, the gas pressure in the two volume spaces created by the orifice plate, the temperature of the gas flowing through the orifice, and the cylinder wall temperature were directly measured with instrumentation. From these measurements the mean gas temperature and transient wall heat flux could be calculated. Also, heat transfer could be calculated from pressure and volume measurements. In addition, heat transfer could be predicted, from temperature, pressure, and volume measurements, with a complex Nusselt number method. The voltage outputs from the various instruments which took the direct measurements were read by a data acquisition computer. These direct measurements were then manipulated in a data reduction program, at which time the calculated properties were computed.

This chapter will describe the drive apparatus, the test section and all of the related components, the data acquisition setup, and the instrumentation. Experimental parameters which affected the design of the apparatus will also be described in the chapter.

2.1 APPARATUS SETUP FOR DRIVING TEST PISTON

A test matrix was created to cover a wide range of pressure and piston frequency

combinations and is shown in Table 2.1 (this table is located in section 2.5.3). The matrix called for piston frequencies 1.26 to 62.8 rad/sec which corresponded to piston speeds of 12 to 600 RPM. To obtain these speeds the following setup was created: a DC motor (via a belt drive) turned an intermediate shaft which (via a second belt drive) turned a compressor flywheel which drove the test piston. The driving apparatus is illustrated in Figure 2.1.

2.1.1 DRIVING MOTOR

The motor used to drive the test piston was a Westinghouse model SK rated at 5.5 horsepower and 1800 RPM. A 3 phase 240V AC line, varied by a variable autotransformer (Variac), was used to control the voltage to the motor armature. The voltage from the Variac was rectified to DC before being sent to the motor. A 120V AC line, varied by a second variable autotransformer, was used to control the voltage to the motor field. The field voltage was then rectified to DC before being sent to the motor. Figure 2.2 shows the electrical layout of the input voltages to the motor.

The motor speed was controlled by the armature voltage while the field voltage was generally set in one place. If more torque was needed by the motor during the first compression stroke, the field or armature voltage were adjusted. Since motor speed was effected by torque, once the motor was running smoothly the field and armature voltages had to be returned to their initial levels.

2.1.2 INTERMEDIATE SHAFT

The speed of the compressor flywheel was reduced to a level below that of the motor by using two different sheave arrangements placed on an intermediate shaft. One was designed for a speed reduction of 1.6:1, for a maximum piston speed of 1200 RPM (the original testing plan called for speeds this high, but due to problems with o-ring failure the maximum working speed was later decreased to 600 RPM). The second sheave arrangement was designed for a reduction of 9:1, for a maximum speed of 200 RPM. This latter sheave arrangement was used for piston speeds of 30, 60, and 120 RPM, the former for 300 and 600 RPM.

Figure 2.1:
Drive System

Figure 2.2:
Electrical Input to Motor

2.1.3 COMPRESSOR BASE

The piston-cylinder apparatus was mounted on a two stage Curtis air compressor, model C97. The two compressor pistons had a stroke length of 3.75 inches. The test piston was connected to the high pressure compressor piston.

The compressor pistons were originally meant to be lubricated with a splash ring which was attached to the crankshaft. The splash ring would rotate with the crankshaft, pick up oil from the oil pan, and spray the oil to all of the components within the compressor base. The faceplate on the compressor stated that the compressor should not be run at speeds less than 350 RPM, presumably because the splash ring would not adequately lubricate at lower speeds. Since most of the speeds in the test matrix were below this speed, the splash ring was replaced with a pair of spray nozzles which were aimed at the bores of the compressor pistons. Oil was continually pumped to the nozzles by an electrically driven oil pump.

2.2 PISTON-CYLINDER APPARATUS

The test apparatus consisted of the following components:

- the bounce volume,
- the cylinders creating the bounce volume space, the varying volume space, and the fixed volume space,
- the test piston and the connection between it and the compressor piston,
- the orifice plate, and
- the adjustable head.

Figure 2.3 gives a detailed schematic of the test apparatus.

2.2.1 BOUNCE VOLUME

The bounce volume was a large reservoir of gas connected to the cylinder section beneath the area swept by the piston. This large reservoir of gas provided a nearly constant pressure on the back side of the piston. The bounce volume was needed to decrease the load on the motor and to keep the force on the high pressure compressor piston below its rated limits.

The bounce volume was filled to the same fill pressure as the bore (1, 2, 5, 10, and 16.7 atm). The bounce volume was constructed from 12 in. schedule 40 steel pipe and 12 in. butt-welded pipe caps so that the reservoir would be safe at the highest fill pressure. The reservoir was connected to the bounce volume space by a piece of steel tubing, 2.00 in. inside diameter and 18 in. in length. The tube was connected to the cylinder with a flange that was welded to the end of the pipe and then bolted to the side of the cylinder, as shown in Figure 2.4. The flange had a static o-ring, to prevent gas from leaking to the atmosphere, where it was bolted to the cylinder. The minimum volume in the bounce volume space and the bounce volume reservoir, calculated when the piston was at bottom center, was found to be approximately 3990 in³.

The large reservoir of gas provided a nearly constant pressure on the back side of the piston. With the bounce volume, the differential pressure across the piston face was less than it would be if the back side of the piston was exposed to atmosphere. The load on the compressor base and on the motor was decreased because of the smaller differential pressure. With the load on the motor less, the volume signal was more uniform. Without the bounce volume the motor would have had to work hardest at top center; this would cause the test piston to spend a disproportional amount of time at this position as the motor was trying to get the piston "over the hump".

The large bounce volume was also necessary to keep the force on the compressor crankshaft below it's rated limits. Based on the maximum rated pressure for the compressor (250 psi), the force on the high pressure piston (outside diameter of 3.375 in.) was calculated to be 2237 lb_f. This meant that the differential pressure in the testing apparatus (cylinder diameter = 7.50 in.) could not exceed 50.63 psi [(PA)_{hpp} = (PA)_{app}]. The following calculations, at maximum pressure (250 psia) and near adiabatic conditions with helium as the working fluid, were made to make sure that the compressor was not overloaded. A pressure ratio in the bounce volume, calculated from the maximum and minimum volume, was created by the piston motion

$$\left(\frac{V_{\max}}{V_{\min}} \right)^{\frac{\gamma}{2}} = \frac{P_{\max}}{P_{\text{mean}}} \quad 2.1$$

where:

$$V_{\min} = 3990 \text{ in}^3$$

$$V_{\max} = 3990 \text{ in}^3 + \text{volume swept by the piston} = 3990 \text{ in}^3 + 166 \text{ in}^3$$

Equation 2.1 gave a pressure difference, $P_{\max} - P_{\text{mean}}$, due to the pressure ratio in the bounce volume, of 8 psi. There was also a differential pressure across the piston face which was related to volume ratio by equation 2.1. At the conditions of highest volume ratio (1.2) and highest mean pressure (250 psia), equation 2.1 yielded a maximum of 37.7 psi differential pressure across the piston face. When these two pressure forces were combined, the total pressure difference, $P_{\max} - P_{\text{mean}}$, was 46.2 psi nearly 4 psi below the upper limit of 50.63 psi.

2.2.2 CYLINDER SECTIONS

The cylinders were sections of honed steel tubing with an inside diameter of 7.50 in. and a wall thickness of 0.75 in.. The tubing was honed to provide a smooth surface for dynamic and static seals to mount against.

Four cylinders were machined from the honed tubing and are shown in Figure 2.3. The first of these supported the test section, while the other three cylinders created the volume spaces.

- The first cylinder had 2 slots machined through the side wall to allow the experimenter access to the linkage which connected the test piston to the compressor piston, which was also used to adjust the piston-orifice clearance. One of the slots also provided an open area for the vertical travel of a connection between the piston rod and the linear potentiometer. The linkage will be discussed in section 2.2.3.2 and the rigid connection in section 2.4.2.3.

- The second cylinder contained the test piston, the varying volume space, and the bounce volume space. The varying volume space was the volume between the piston face and the orifice plate. Ports were machined into the cylinder wall for a gas thermocouple, a pressure transducer, and a heat flux sensor. A connection to the bounce volume reservoir and a line to pressurize both the bounce volume space and the bounce volume reservoir were located below the area swept by the piston.

- The third and fourth cylinders were used to create the fixed volume space. The lowest volume ratio in the test plan was 1.1. To achieve this volume ratio, with the 3.75 in. stroke length, a cylinder 45.938 in. high was required to create the fixed volume space. Higher volume ratios required less volume and, therefore, shorter cylinders. In order to

Figure 2.3:
Schematic of Test Apparatus

make installation easier, two cylinders, each approximately 24.0 in. high, were machined from the honed tubing. Depending on the test conditions either one or both of the cylinders were needed. The exact volume in the fixed volume space was controlled by the adjustable head (section 2.2.5).

2.2.3 TEST PISTON AND CONNECTING LINKAGE

The test piston was connected to the compressor piston with a linkage which allowed for some misalignment between the two pistons. Since the linkage needed to be adjusted while the piston was in the cylinder, the cylinder containing the linkage had two slots, large enough for access to the linkage, machined into the sides. The slots also provided an open area for the connection between the test piston and the linear potentiometer to run through. Since this cylinder was open to atmosphere it had to be isolated from the other cylinders. Figure 2.4 shows a detailed schematic of the test piston and the linkage.

2.2.3.1 TEST PISTON

The test piston was machined from aluminum and had an outside diameter of 7.497 in. with a wall thickness of 0.500 in. on the piston skirt and 0.270 in. at the piston face. A Parker "T-seal" was placed around the outside diameter of the piston and was located at the base of the piston in order to reduce the effect of frictional heating created by the seal running against the surface of the honed tubing.

The T-seal was similar to a standard o-ring, but was designed specifically for dynamic sealing. As the name implies, the seal was T shaped in cross section. The flange of the T rested against the bottom of the groove and prevented the seal from rolling in the groove. The base of the T sealed to the dynamic surface and acted as a wiper following the dynamic surface. The seal was made out of a self-lubricating nitrile compound which, according to the manufacturer, would decrease the friction between the seal and the cylinder and allow the piston to run at higher speeds without the seal disintegrating. However, the seal failed during initial tests at higher speeds because there was not enough lubrication present. To alleviate this problem, the maximum speed was decreased from 1200 to 600 RPM and a lubricating ring was placed on the bounce volume side of the T-

Figure 2.4:
Test Piston and Linkage

seal. The lubricating ring was made from felt and was saturated with vacuum pump oil before assembly, and was resaturated whenever the apparatus was disassembled. The felt ring was placed on the bounce volume side of seal to keep the oil from contaminating the varying and fixed volume spaces.

The cylinder containing the test piston had to be separated from the cylinder containing the connecting linkage, since this cylinder was exposed to atmosphere while the bounce volume space had to be pressure tight. An aluminum plate was designed to serve this purpose, and was clamped between the two cylinders as shown in Figure 2.4. A static o-ring was placed around the outside diameter of the plate to keep gas from leaking out of the bounce volume space at this junction. The piston rod ran through a hole in the middle of the plate and was then connected to the linkage. A T-seal was placed in the through hole to keep gas in the bounce volume space from leaking around the piston rod. This T-seal also disintegrated during initial testing. Extra lubrication was provided to this seal by coating the piston rod with vacuum pump oil before every experimental run. The oil adhered to the piston rod and lubricated the seal as the rod passed over it.

2.2.3.2 LINKAGE BETWEEN THE TEST AND COMPRESSOR PISTONS

The purpose of the linkage was to transfer the motion of the compressor piston to the test piston. In order to create different clearance volumes, determined by the distance from the piston face at top center to the bottom of the orifice plate, the linkage needed to be adjustable. The linkage also needed to compensate for misalignment between the test piston and the compressor piston. In order to accomplish these tasks the linkage consisted of a yoke (connected to the piston rod of the test piston), a self-aligning rod end, a connecting rod, a second self-aligning rod end, and a second yoke. Figure 2.4 shows this configuration.

The length of the linkage was made adjustable with the threaded connection between the rod ends and the connecting rod. In order to change the clearance volume more or fewer of the threads were engaged, with a minimum possible adjustment of one half the thread pitch of the rod ends, 0.021 in. The clearance volume was set by adjusting the length of the linkage until the desired volume was attained. The depth of the volume was measured from the bottom of the orifice plate to the test piston face with a depth micrometer.

Bearings in the rod ends allowed for motion in the horizontal direction but did not allow for motion in the vertical direction. Some misalignment between the centerline position between the two pistons could be compensated for with the rod ends.

2.2.3.2.1 SLACK IN LINKAGE/ SLACK IN COMPRESSOR BEARINGS

The stroke of the compressor piston was thought to be a constant 3.75 in., independent of pressure and frequency. Initial testing indicated that this was not the case. Different stroke lengths were measured at different pressure/piston frequency combinations. Slack in the connecting linkage and slack in the compressor bearings seemed to effect the stroke length.

2.2.3.2.1.1 SLACK IN BEARINGS

Slack in the various bearings within the compressor base affected the length of the stroke due to an interaction between pressure forces and inertial effects which were due to the acceleration of the piston. At top center inertia tended to keep the piston moving upward. At the same time, the pressure in the varying volume space was greater than that provided by the bounce volume which caused a pressure force to act downward. At bottom center these two effects acted in the opposite directions from top center (see Figure 2.5). The inertial force due to the acceleration of the piston was proportional to $m\omega^2r$ (where $r = L_g/2$, m = mass, and ω = piston frequency).

Figure 2.5:
Pressure Force and Inertial Effects

Nothing could be done to prevent the slack in the compressor bearings from affecting the stroke length, but another cause of slack, the connection between the rod ends and the connecting rods, was addressed.

2.2.3.2.1.2 SLACK IN THE ROD END CONNECTION

In the one space experiments, jam nuts located at the connecting rod - rod end joints were inadvertently left off during the assembly process, and are not shown in Figure 2.4. Without these nuts, the piston free play was increased by the free play in the rod end threads. The jam nuts were added prior to the beginning of the two space experiments. However, the one space experiments were not conducted again since the experimental results matched work by previous experimenters. The slack did create noise at high piston frequencies, which affected heat transfer results.

2.2.3.2.1.3 VOLUME EFFECTED BY SLACK

The slack located in the compressor bearings and in the rod end connection caused the stroke length to change for different pressure/piston frequency combinations. The changing stroke length caused the volume to be different between experimental runs. Stroke length is plotted against RPM in Figures 2.6 and 2.7 for the one and two space experiments. Both plots showed the stroke length decreasing with increasing pressure, with the smallest stroke length occurring when the fill pressure was a maximum and the piston frequency, ω , was a minimum. At high pressures and low speeds the pressure forces dominated causing the decrease in stroke length. In both experimental situations there was a large increase in stroke length between 300 and 600 RPM, with the greatest increase when the fill pressure was a minimum. This indicated that the inertial effects dominated at high speeds and low pressure which caused the stroke length to increase here. As the pressure was increased the inertial effect was proportionally less.

The stroke length in the two space experiments, after the addition of the jam nuts, was smaller than in the one space experiments. Although the stroke length was smaller, the stroke length in the two space experiments showed the same trends with pressure and speed as seen before. This indicated that the slack in the rod end connection only added to the affect of the slack already present in the compressor bearings.

Figure 2.6 Ls vs RPM, One Space
(nominal compressor stroke = 3.75 in.)

Figure 2.7 L_s vs RPM, Two Spaces
(nominal compressor stroke = 3.75 in.)

2.2.3.2.1.4 FLUCTUATION IN THE VOLUME TRACE

The rate of change of volume was calculated from the one and two space raw data on a point by point basis, $(V_n - V_{n-1})/(t_n - t_{n-1})$, in order to see how the jam nuts may have affected the volume - time plot. The cases of 16.7 atm and 30 RPM are shown since the pressure effects seemed to be greatest at this combination. These plots are shown in Figures 2.8 and 2.9. The plots show an initial "hit" in the volume signal at midstroke followed by a "ringing" in the volume. In the two space plot, the "ringing" seemed to be damped out quicker than in the one space plot. In both plots the amplitude of the "hit" seemed to be about the same magnitude, and the frequency was approximately 10x the cycle frequency.

The rate of change of the pressure was also plotted in Figures 2.8 and 2.9. The "hit" was seen in the pressure, but the "ringing" was not. The conclusion was then made that the "hit" was a result of the pressure force and inertial effects switching directions near midstroke. Since the pressure traces did not show the "ringing", the gas in the cylinder must not have experienced any fluctuations. The only way that the "ringing" would be only seen in the volume trace was if the "hit" caused the connection between the piston rod and the potentiometer to vibrate. The plots also showed that the "ringing" was damped out quicker in the two space experiments than in the one space. However, it was unclear why this was so.

2.2.4 ORIFICE PLATE

In the two space experiments an orifice plate was located just above the top center position of the test piston and was designed to produce turbulence due to the inflow of the gas. The orifice plate was machined from aluminum and was clamped between two of the cylinder sections as shown in Figure 2.3. Two different plates were designed to satisfy two values of the ratio Re_{in}/Pe_w (see section 2.5.2.3). The orifice diameters of the two plates were 1.36 and 4.12 in..

The orifice plates separated the test volume into two distinct spaces: a constant volume space (the volume space between the orifice plate and the head) and a varying volume space (the volume space between the piston face and the bottom of the orifice plate). Mass was transferred between the two spaces due to the flow of gas through the

Figure 2.8
 $\Delta V/\Delta t$, $\Delta P/\Delta t$ One Space Results

Figure 2.9
 $\Delta V/\Delta t, \Delta P/\Delta t$ Two Space Results

orifice.

2.2.5 ADJUSTABLE HEAD

A head separated the fixed volume space from the atmosphere. In order to create different volume ratios, the head had to be set at different depths within the fixed volume space.

The head was a disk machined from aluminum with a static o-ring placed around the outside diameter to prevent gas from leaking to atmosphere. A second disk was rigidly attached to the head with a connecting rod. This disk ran over three 7/8-UNF tie rods which ran the length of the apparatus and were used to keep the apparatus in compression. The disk was attached to the tie rods with nuts, one on each side. The head could be positioned at any depth within the bore by threading the nuts up and down the threaded rod. This assembly is shown at the top of Figure 2.3.

One of the components that defined the volume ratio was the depth of the head within the fixed volume space; the stroke length and the clearance volume were the other two components. In order to determine the fixed volume, the distance between the top of the head to the top of the cylinder was measured with a depth micrometer. The fixed volume itself could not be measured directly since it was completely enclosed. The depth of the head had to be adjusted until the required volume was met.

The head contained ports for a differential pressure transducer and a heat flux sensor. It also had a port for a fill line through which the varying and fixed volume spaces could be pressurized. A capillary tube, 0.02 in. inside diameter, was used for this fill line. Such a small tube was used so that the amount of gas trapped inside this volume during the experimental runs would be very small when compared to the volume of gas in the volume spaces. For the case of volume ratio of 1.1, the capillary tube was 12.75 in. long which resulted in a volume of 0.004 in³ in the tube. The minimum volume in the cylinder under this condition was approximately 1675 in³.

2.3 DATA ACQUISITION

The data acquisition system consisted of hardware:

- the data acquisition computer

- the A/D board which sampled the data

and software:

- routines which controlled the sampling
- user created programs which accessed the sampling routines.

The user created programs are presented in Appendix B. The transducers which sent signals to the A/D board will be discussed in section 2.4.

2.3.1 DATA ACQUISITION HARDWARE

The data acquisition computer and the A/D board used to sample the instrumentation output composed the data acquisition hardware. The computer used was an IBM compatible PC with an Intel 80386 microprocessor and clock speed of 25 MHz. A RTI-860 A/D board from Analog Devices was installed in the computer.

2.3.1.1 SAMPLING BOARD

The RTI-860 board sampled the incoming data signals according to the instructions sent to it from the sampling routines. The board had 16 analog input channels and had the ability to sample from individual analog input channels or to perform simultaneous sample and hold.

In simultaneous sample and hold the board could be programmed to sample data simultaneously from groups of four channels. Each channel's reading (from the group) was taken within 50 picoseconds of the others. The sampled data was then held, as a group, until it was converted to digital form. The time between the channel readings was small enough so that it could be assumed that the channels were sampled simultaneously. The board was programmed to use both the simultaneous sampling and individual channel sampling.

The sampling routines provided for a variety of ways for the sampled data to be transferred from board memory. The method chosen involved using a circular double buffer located in the memory of the data acquisition board. Each buffer was 8k. While sampled data was placed in one half of the buffer, existing data was transferred out from the other half. The data was then transferred to a user defined array located in the memory of the data acquisition computer.

The board was configured to receive input in the range of $\pm 5V$.

2.3.2 DATA ACQUISITION SOFTWARE

The data acquisition software consisted of subroutines provided by the board manufacturer to sample the data and computer programs written by the user to access them and process their output. The sampling routines were provided in object code for Microsoft QuickBasic and the user written programs were in that language.

2.3.2.1 SAMPLING ROUTINES

The sampling routines were provided by the board manufacturer and operated on their data acquisition boards. The instrument output could be sampled in a variety of different ways. The different sampling subroutines used in the data acquisition program are declared at the top of the program shown in Appendix B. A description of the use of the routines and the meaning of the parameters passed in the procedure calls can be found in the RTI-850/860 Software Manual (Analog Devices, 1990).

2.3.2.2 DATA ACQUISITION PROGRAMMING

The user written code was created to access the sampling routines and to manipulate the sampled data. The data acquisition program is shown in Appendix B. The data acquisition program determined whether or not steady state had been reached and then sampled all of the transducer output channels for 3 cycles.

Steady state was determined from pressure and volume data. Based on the theoretical piston frequency, calculated from a voltage signal proportional to the armature voltage, the board was programmed to scan the output of one of the pressure transducers for a period of time equivalent to twice the piston period (two revolutions). In this way a complete cycle was assured to have been sampled. After the output voltage was converted to pressure, the trapezoid rule was used to obtain the time averaged pressure over the full cycle. The time from the beginning of the cycle to the end of the cycle was then used to compute the actual piston frequency. After all of the data was manipulated, the process was repeated. When the time averaged pressure and piston frequency

measured from two successive scans of data were within 1 % of each other, steady state was assumed to have been reached.

Once steady state was reached, all of the transducers were scanned (using simultaneous sample and hold) to collect the final data set. Since all of the channels could not be scanned simultaneously (there were seven transducers and the maximum number of channels in a scanned group was four) four instrument channels were simultaneously sampled followed by the other three. Since the phase difference between the pressure and volume was important, the first group of sampled channels contained the pressure and volume instrumentation. The second group of channels contained the instruments which measured the gas temperature and the wall temperature. The time used for this final scan was the time determined from the steady state calculations, as opposed to the time from the theoretical piston frequency. The scan was long enough to ensure that three complete cycles of data were obtained. Three cycles were taken so that any cycle to cycle variation could be determined.

The data from the final scan was read into a file and, at a later time, was reduced to heat transfer data.

2.4 INSTRUMENTATION

The system instrumentation consisted of the transducers, amplifiers, and other analog signal processing components. The instrumentation produced a voltage output which was sampled by the data acquisition hardware. The relationship between the transducers, the amplifier unit, and the data acquisition board is illustrated in Figure 2.10.

2.4.1 AMPLIFIER UNIT

The amplifier unit was a model 2120A strain gage amplifier manufactured by Measurements Group. A power supply unit, model 2110A (also from Measurements Group), provided the excitation voltage to all of the transducers but the linear potentiometer, which had its own dedicated voltage supply. The strain gage amplifier amplified the output signal to the $\pm 5V$ range selected on the A/D board. The frequency response of the amplifier was 15kHz.

The transducer output could be amplified in a range from 2x to 2100x. Each

channel could have a different gain and had a bridge balance and a current balance. By adjusting the bridge balance, the residual imbalance of the bridge transducer could be compensated for. The current balance could be adjusted to eliminate input amplifier offset. The power supply could provide an excitation voltage from 0.266 to 12 volts. Each channel could be excited with a different voltage.

2.4.2 TRANSDUCERS

The following types of transducers were used to measure various quantities within the apparatus:

- two differential pressure transducers, ± 50 psid, measured the pressure in the fixed and varying volume spaces,
- one pressure transducer, 0 - 300 psig, measured the pressure that the differential pressure transducers were referenced to,
- two heat flux sensors to measure transient surface heat flux,
- one linear potentiometer to measure piston displacement,
- one thermocouple to measure the temperature of the gas flowing through the orifice, and
- one voltage divider/ rectifier to measure the motor armature voltage.

2.4.2.1 PRESSURE TRANSDUCERS

To measure the pressure in the volume spaces three pressure transducers were used, two $\pm$ psid differential transducers and one 0 - 300 psig transducer. 50 psid differential pressure transducers were used to measure the change in pressure due to the oscillating pressure flow. Even though the measurement of the total pressure was less accurate than if a single transducer had been used, the use of the differential transducers gave a more accurate measurement of the change in pressure. An accurate measurement of ΔP was needed for the heat flux calculations.

2.4.2.1.1 DIFFERENTIAL PRESSURE TRANSDUCERS

There were two differential pressure transducers in the apparatus to measure the

Figure 2.10:
Amplifier/Transducers

change in pressure; one was located just below the orifice plate to measure the pressure in the varying volume space, while the other was located in the head to measure the pressure in the fixed volume space. The differential transducers chosen were Kulite model XT-140 pressure transducers. The transducers had a differential pressure range of ± 50 psi and the maximum rated pressure for the backing tube was 250 psi. The magnitude of the expected pressure differential across the piston face was ± 37.7 psi (see section 2.2.1), which was calculated at the worst case conditions of: maximum pressure (16.7 atm), near adiabatic conditions, and volume ratio of 1.2.

The pressure sensor was coated with a RTV (silicone rubber) coating to eliminate transient heat transfer effects on the pressure measuring diaphragm. The transducers had a threaded housing and were threaded directly into the mounting surfaces. There was an o-ring at the junction between the transducer and the mounting surface to prevent gas leakage. The transducers were small in size (the tip diameter was 0.093 in.) and flush mounted, leading to a very quick response time. The natural frequency of the transducer diaphragm was 150 kHz.

The differential pressure transducers were calibrated with a dead weight tester over the full range from -50 to 50 psi. Since the transducer would record increasing and decreasing pressure for a single experimental run, the calibration was conducted so as to detect any hysteresis. The differential transducers were calibrated while wired to the amplifier and to the computer in order to reflect any errors associated with the electrical setup. The calibration on both transducers was good to within $\pm 0.5\%$ of the best fit line over the full scale of the transducer.

2.4.2.1.2 BACKING PRESSURE TRANSDUCER

The 0 - 300 psig pressure transducer was used to measure the pressure in the backing tube of the differential pressure transducer. The maximum backing pressure of 250 psia was well within the limits of the transducer.

The pressure transducer used for this purpose was a Teledyne pressure transducer, model 176. The transducer was calibrated with a dead weight tester over the full 0 - 300 psi range. The calibrated pressure was within $\pm 0.3\%$ of the best fit line over the full scale of the transducer. The calibration was conducted with the transducer wired into the amplifier and the computer. In this manner, any errors introduced in the electrical setup

would be reflected in the calibration. Since the backing pressure was nearly a constant, the response time of the pressure transducer was not important.

2.4.2.2 HEAT FLUX SENSOR

Transient surface heat flux was measured from surface temperature measurements. In order to measure the wall surface temperature in the fixed volume space, a heat flux sensor (a resistance temperature detector, model PTF-100-20404 from Medtherm Corporation) was mounted flush to the bottom surface of the end cap. A second sensor, to measure the wall surface temperature in the varying volume space, was to be mounted just below the orifice plate; however, this sensor was damaged during installation and was not replaced.

2.4.2.2.1 TRANSIENT HEAT FLUX

Heat transfer $q''(t)$ is a unique function of wall temperature $T_w(t)$. When $q''(t)$ and $T_w(t)$ are sinusoids as is the case here, this function has a simple form. In order to make use of this form, $T_w(t)$ was Fourier decomposed into a series of sinusoids.

The cylinder wall was then modelled as a semi-infinite solid with surface temperature as a harmonic function of time (Carslaw & Jaeger, 1959). Details of this model are presented in section 3.6.2. This measured surface heat flux was compared to another heat flux measurement and to a predicted value of heat flux.

2.4.2.2.2 INSTRUMENTATION

The RTD was a thin layer of platinum, approximately 0.123 in. x 0.01 in., deposited on a Pyrex™ 7740 substrate. When the thin film thermometer was exposed to a change in temperature, the resistance of the platinum changed. The sensor was installed in a Wheatstone bridge as shown in Figure 2.11. The change in resistance was seen as a change in output voltage from the sensor. The voltage was used to find the measured surface temperature as follows.

For a bridge with a small imbalance:

$$V_{out} = \left[\frac{R_T}{R_T + R_B} - \frac{1}{2} \right] V_{ex}$$

2.2

where:

V_{ex} = excitation voltage,

R_T = resistance of the RTD, and

R_B = resistance of the balancing resistor.

Figure 2.11:
RTD Bridge Network

Equation 2.2 was solved for R_T , the only unknown parameter, then the surface temperature was found by:

$$T = \frac{\frac{R_T}{R_B} - 1}{\alpha} - T_{oc}$$

2.3

where:

$\alpha = 0.0023 \Omega/\Omega^\circ\text{C}$, the temperature coefficient of resistance for the RTD,

T_{oc} = initial wall temperature at bridge balance, and

T = surface temperature in Celsius.

The sensitivity, α , was taken from a ohms vs. temperature calibration provided by Medtherm.

2.4.2.2.3 BALANCING

Since the temperature changes it was designed to measure were small, the heat flux sensor was also very sensitive to changes in ambient temperature. The bridge balance located on the strain gage amplifier was not sensitive enough to balance out these changes. Because the instrument imbalance could be too great for the amplifier bridge balance to compensate for, a second balancing arrangement was created as a coarse adjustment to the fine balance provided by the amplifier. The schematic of the sensor's bridge, Figure 2.11, shows the coarse balance, R_B . The coarse balance consisted of a 109.6Ω resistor being placed in parallel with a variable 0 to 1000Ω resistor and provided one of the four arms to the wheatstone bridge. The fine balance was usually used to balance out any small fluctuations in room temperature that could affect the transducer between experimental runs.

2.4.2.2.4 SELF-HEATING

Self-heating is a problem with heat flux sensors which affects the accuracy of the transducer. Self-heating occurs when the power delivered to the transducer creates a significant amount of internally generated heat flux. This internally generated heat flux results in the surface of the transducer being hotter than the surrounding wall.

Originally, the power delivered to the bridge was to be such that the internal heat flux generated would be less than 1% of the expected heat flux. The excitation voltage needed to satisfy this requirement, 0.7 V, created too much self-heating in the heat flux sensor. Following conversations with the manufacturer, it was found that 1 mA or less of current should be delivered to the bridge in order to eliminate the effects of the transducer self-heating. However, less than 1 mA could not be delivered to the bridge due to the power supply used. The smallest excitation voltage possible was 0.266 V, which would result in 1.3 mA being delivered to the bridge. It was felt that 1.3 mA was close enough to 1 mA; therefore, the excitation voltage was then reduced to 0.266 V. Once this change

was made, no more problems from self-heating were evident.

2.4.2.3 LINEAR POTENTIOMETER

A linear potentiometer was used to measure the piston position which was, in turn, used to measure the volume of the gas in the varying volume space. The potentiometer used was a Novotechnik model TS100A502 with a wiper stroke length of 3.94 in., while the stroke of the test piston was 3.75 in.. The potentiometer was mounted to the side of the apparatus and an arm rigidly connected the actuating shaft to the piston rod, as shown in Figures 2.4 and 2.12. A rigid connection to the piston rod was necessary to eliminate any error in the volume measurement due to the connection between the transducer and the test piston.

Results of initial testing indicated that the stroke length of the test piston was not constant and that the linear potentiometer was giving inaccurate volume measurements. The inaccuracy was due to a spring located at the connection between the potentiometer actuating shaft and the connecting arm, and was corrected before the single space testing was started. This will be further explained in section 2.4.2.3.2.

2.4.2.3.1 CALIBRATION

The linear potentiometer was calibrated with a depth micrometer. The calibration was conducted with the transducer wired into the data acquisition board and the computer in order to reflect any errors caused by the electrical setup. To calibrate the potentiometer, the compressor flywheel was rotated in small increments and the depth of the piston was measured with a depth micrometer at each of these points. The voltage output, at these points, from the linear potentiometer was sampled by the data acquisition system. The potentiometer calibration was accurate to within $\pm 0.2\%$ of the best fit line over the full stroke length.

2.4.2.3.2 SLACK IN POTENTIOMETER LINKAGE

One of the features of the potentiometer was that it had a self-aligning coupling to compensate for some misalignment in the horizontal direction. The alignment was created

with a ball joint and a spring as shown in Figure 2.12. Initial testing showed a much larger stroke length than was expected. The actuating shaft was actually hitting the potentiometer housing during the stroke. The inaccuracy in the volume measurement was determined to be caused by the spring. It was found that the spring was not stiff enough to accelerate the moving parts of the potentiometer at the required rate. At top center, the spring was too weak to keep the ball joint against the actuating shaft. It was not clear if the spring was overloaded during initial testing and was damaged, or if the spring had not been stiff enough to begin with.

The problem was corrected by increasing the stiffness of the spring by placing a spacer between the bottom of the spring and the surface it rested on. This spacer compressed the spring, increasing the stiffness. With the spacer in place, the actuating shaft and linkage moved as one unit at all test speeds.

Figure 2.12:
Slack In Linear Potentiometer

2.4.2.3.3 VOLUME MEASUREMENTS

The slack in the compressor bearings affected the actual volume change. The true volume was measured except for the "ringing", as discussed in section 2.2.3.2.1. This "ringing" resulted in inaccurate volume measurements in both the one and two space experiments.

2.4.2.4 GAS THERMOCOUPLE

A thermocouple was used to measure the temperature of the gas flowing through the orifice. The thermocouple used was a Type E thermocouple, model 320, manufactured by the Paul Beckman Company. The thermocouple was located just below the orifice plate and was threaded into the cylinder wall. An o-ring was located at the junction of the threaded connection and the cylinder to prevent gas leakage.

The length of the thermocouple probe was 3.75 in., to approximately place the probe tip in the center of the bore. The Chromel-Constantan wires making up the thermocouple were 0.0007 in. thick and terminated in a fused Micro-Disk junction 0.001 in. thick. The probe containing the wires was insulated and was 0.003 in. in diameter. Due to the small wires, very fast response was expected. The small diameter of the probe meant that the probe would create minimal disturbance to the flow of gas through the orifice.

During operation the cold junction was not maintained at 32 °F. A cold junction compensator was purchased from Hades Corp and was used in place of an ice bath. The compensator employed a self-compensating electrical bridge network. The network had a temperature sensitive resistance element as one of the legs and was thermally integrated with the cold junction. The change in resistance of the thermal element canceled out the error due to the cold junction being above 0°C. Standard thermocouple tables and curve fits with a 0°C reference could then be used.

An accepted mV vs. temperature curve fit (NIST Monograph 125) was used to fit the Type E thermocouple data over the expected temperature range. While wired to the cold junction compensator, to the amplifier unit, and to the data acquisition computer, the thermocouple bridge was balanced with the probe tip immersed in an ice bath. In this way, the reference temperature of the thermocouple matched that of the curve fit used.

2.4.2.5 VOLTAGE SIGNAL FROM MOTOR

A voltage proportional to the motor armature voltage was sent to the data acquisition system. After the 3 phase AC voltage was varied by the variable autotransformer, the voltage was reduced by two 10k resistors as shown in Figure 2.2. The voltage was then rectified to produce a DC signal, which was read by the data acquisition computer. When the variable autotransformer did not provide any reduction to the armature voltage, 3.66 V DC was produced.

The reduced armature voltage was calibrated versus piston frequency. The calibration was conducted with the linear potentiometer, the complete piston drive, and the computer in place. A separate calibration was done for both sheave arrangements. The calibration of armature voltage against RPM was conducted while the piston face was exposed to atmosphere, i.e. no gas was being compressed. During operation, the actual piston frequency would be less than the theoretical values due to the load on the motor. The calibration was not required to be exact because the data acquisition program calculated the exact piston frequency for each run after steady state conditions were reached.

2.4.2.6 NOISE

Noise had to be kept to a minimum. Many of the output signals were differentiated, and any noise in the original signal was amplified in the differentiated signal.

To reduce the amount of noise in the signals, all of the cabling connecting the transducers to the data acquisition unit was shielded and twisted. The shielding was grounded to the pin connectors at the amplifier box. These two precautions eliminated the effects of the electromagnetic fields created by sources carrying AC current (lighting, the input voltage to the motor).

2.5 EXPERIMENTAL PARAMETERS

Several experimental parameters were used during the design of the apparatus. Most of them fall into two categories; those that defined the engine geometry and those which related the geometry to thermal and fluid properties. The bore/stroke ratio, volume

ratio, and piston-head clearance ratio defined the engine geometry. The oscillating Peclet number and the inlet Reynolds number related the geometry to the thermal and fluid properties. Another experimental parameter was the specific heat ratio, which depended on gas properties alone.

2.5.1 ENGINE GEOMETRY

The desired bore/stroke ratio, volume ratio, and piston-head clearance ratio affect the physical make-up of a Stirling machine. The designer must work within reasonable limits for these parameters when designing the apparatus. All of the parameters are linked, directly or indirectly, to the stroke length (L_s) of the driving piston.

2.5.1.1 BORE/STROKE RATIO

The bore/stroke ratio is the ratio of the diameter of the bore to the length of the stroke of the driving piston. Typical values in free piston machines range from 2 to 7 (Kornhauser, 1991).

In the test apparatus the stroke length of the compressor piston was fixed at 3.75 in.. With this large stroke length, even the lower bound on the range of bore/stroke ratio required a bore of 7.5 in. in diameter. Creating larger bore/stroke ratios would have been difficult due to the required diameter of the cylinders making up the bore. A bore/stroke ratio of 2.0 was therefore used for all experiments.

2.5.1.2 VOLUME RATIO

The volume ratio is the ratio of maximum to minimum volume in the apparatus' cycle. If $V_{\max}$ is the volume of gas at piston bottom center and $V_{\min}$ is the volume of gas at piston top center, then the volume ratio is defined as:

$$R_v = \frac{V_{\max}}{V_{\min}} \quad 2.4$$

From geometry

$$V_{\max} - V_{\min} = \frac{\pi (Bore)^2 L_g}{4} \quad 2.5$$

Equations 2.4 and 2.5 can be manipulated to obtain:

$$V_{\min} = \frac{\pi (Bore)^2 L_g}{4 (R_v - 1)} \quad 2.6$$

A similar expression can be derived for $V_{\max}$.

For free piston machines the pressure ratio, $P_{\max}/P_{\min}$, rather than volume ratio has usually been defined. This ratio usually lies between 1.1 and 1.3 (Kornhauser, 1991).

Since an engine is bounded by the adiabatic and isothermal conditions, a range for the volume ratio can be defined. The adiabatic and isothermal relations are as follows:

$$\left(\frac{V_{\max}}{V_{\min}} \right)^\gamma = \frac{P_{\max}}{P_{\min}} \quad 2.7$$

and

$$\frac{V_{\max}}{V_{\min}} = \frac{P_{\max}}{P_{\min}} \quad 2.8$$

The following range on the volume ratio was calculated from the isothermal and adiabatic relationships:

for monatomic gasses, $\gamma = 1.67$, : $1.07 < R_{v(ad)} < 1.21$

$$1.1 < R_{v(is)} < 1.3$$

for diatomic gasses, $\gamma = 1.4$, : $1.06 < R_{v(ad)} < 1.17$

$$1.1 < R_{v(is)} < 1.3$$

To cover all possible ranges, effective bounds on R_v were 1.05 to 1.3.

The testing apparatus was designed to operate at $R_v \geq 1.1$. It would be difficult to operate at lower ratios due to the required height of the fixed volume section. Lower volume ratios create smaller pressure amplitudes as well. At a volume ratio of 1.05 and a P_{mean} of 1 atm, the magnitude of the pressure signal would only be +/- 0.6 psi, while at R_v

= 1.1 the amplitude would be ± 1.1 psi. Such a small signal, at $R_v = 1.05$, could result in a small signal to noise ratio in the differentiated signal.

2.5.1.3 PISTON-HEAD CLEARANCE RATIO

The piston-head clearance ratio is the ratio of the distance between the piston face at top center and the head to the stroke length, and for free piston machines this ratio is typically between 0.1 to 0.2 (Kornhauser, 1991). In this apparatus, the orifice plate was the head. With the stroke length fixed at 3.75 in., the range of the clearance ratio was between 0.375 and 0.75 in.. The test apparatus was designed to cover this entire range by using the adjustable linkage discussed in section 2.2.3.

2.5.2 THERMAL FLUID CHARACTERISTICS

The oscillating Peclet number, Pe_w , and the inlet Reynolds number, Re_{in} , are both non-dimensional parameters used in the study of Stirling engines. The oscillating Peclet number relates heat transfer to work. The Reynolds number is usually a measure of whether the fluid is in a turbulent or laminar state. The specific heat ratio, γ , is a defining gas property.

2.5.2.1 SPECIFIC HEAT RATIO

The specific heat ratio, γ , is the ratio c_p/c_v . For monatomic gases $\gamma = 1.67$ and for diatomic gases $\gamma = 1.4$. Most Stirling engines use either monatomic or diatomic gases for the working fluid. Helium and argon, both monatomic gases, were chosen as working fluids for the apparatus. Kornhauser and Smith (1993) found no appreciable differences in the heat transfer results between various monatomic and diatomic gases. Nothing in the apparatus design would have had to be changed if a diatomic gas was chosen as the working fluid.

2.5.2.2 OSCILLATING PECLET NUMBER

The oscillating Peclet number, Pe_w , represents the ratio of the rate of work to the

rate of heat transfer to the wall. The oscillating Peclet number was defined as:

$$Pe_{\omega} = \frac{\rho_o C_p \omega D_h^2}{4k} \quad 2.9$$

where:

D_h = hydraulic diameter calculated from the total surface area and total volume calculated at midstroke ($4 V_{mid}/A_{smid}$),

ω = piston frequency (rad/sec),

ρ_o = density calculated from mean properties,

C_p = specific heat at constant pressure, and

k = thermal conductivity of the gas.

For a given apparatus and gas, the oscillating Peclet number was primarily a function of ω and P_o , the mean pressure, due to the facts that:

1) the hydraulic diameter was set by the apparatus geometry and was constant for any run at given volume and clearance ratios,

2) T_o was nearly constant for all operating conditions, so the density varied only with pressure, and

3) C_p and k are nearly constant and depend on the working fluid.

A test matrix, using helium as the working fluid, was constructed to experimentally test over a wide range of Pe_{ω} by varying the fill pressure and piston frequency, and is shown in Table 2.1. The desired fill pressures and piston frequencies used to create the test matrix were:

Fill pressure : 1, 2, 5, 10, 16.7 atm

ω : 1.26, 3.14, 6.28, 12.57, 31.41, 62.83 rad/s

: 12, 30, 60, 120, 300, 600 RPM

The theoretical range for the oscillating Peclet number was between 38 and 31,550. If argon was used as the working fluid, the values for Pe_{ω} in Table 2.1 need to be multiplied by 8.4.

The experiments were not expected to occur at the theoretical values of the oscillating Peclet number in the test matrix, since the RPM calibration was inexact (as described in section 2.4.2.5) and the cyclic mean pressure, P_o , varied from the fill pressure.

Table 2.1: Test Matrix
 Pe_ω as a Function of P_o and ω

P_o (atm)	1	2	5	10	16.7
ω (rad/sec)					
1.26	38	76	189	378	631
3.14	95	190	473	945	1578
6.28	190	380	945	1890	3155
12.57	380	760	1890	3780	6310
31.41	950	1900	4725	9450	15775
62.83	1900	3800	9450	18900	31550

2.5.2.3 INLET REYNOLDS NUMBER

The inlet Reynolds number, Re_{in} , was used as a parameter to design the orifice plate. The inlet Reynolds number described the flow at the inlet of the orifice. Re_{in} had the inlet velocity, instead of the free stream velocity, as the characteristic velocity and was defined as:

$$Re_{in} = \frac{\rho_o V_{inmax} D_h}{\mu} \quad 2.10$$

where:

D_h = the hydraulic diameter defined as 4 times cylinder volume/ total surface area at midstroke,

V_{inmax} = the maximum velocity of the gas through the orifice and occurs at midstroke, and

ρ_o is the mean density.

The maximum inlet velocity was a function of orifice geometry and was given by:

$$V_{in_{max}} = V_{piston_{max}} \left(\frac{A_{cs}}{A_{in}} \right) \quad 2.11$$

where:

A_{cs} is the cylinder cross sectional and

A_{in} is the cross sectional area of the orifice.

Kornhauser (1992), using approximations based on free pistons, found that the ratio of inlet Reynolds number to oscillating Peclet number was proportional to $3 \cdot A_{cs}/A_{in}$. He also found that Re_{in}/Pe_w was between 10 and 100 for various Stirling machines. Two orifice plates were constructed to obtain $Re_{in}/Pe_w = 10$ and 100. The design of the orifice plates was based on Re_{in}/Pe_w being calculated by the hydraulic diameter of the varying volume space alone. In the two space experiments D_h was calculated for the total volume space in the cylinder, and, therefore, the design value of Re_{in}/Pe_w was higher than the value obtained experimentally.

Table 2.2 shows a listing of Re_{in}/Pe_w for various Stirling machines. Figure 2.13 compares the theoretical range of the test matrix, Table 2.1, to these machines. Using both helium and argon as working fluids and both of the orifice plates, the test matrix would cover most of the Stirling machines.

2.6 EXPERIMENTAL PROCEDURE

A test matrix, Table 2.1, was developed to test at a wide range of oscillating Peclet numbers. Two orifice plates were made to test at two values of Re_{in}/Pe_w . The following experimental procedure was developed to accurately test at the above conditions.

If the apparatus was ever disassembled (to change orifice plates, to change the clearance ratio, to change the volume ratio, etc.), it was then evacuated to 99.9% of a full vacuum after reassembly. By evacuating the apparatus to 99.9%, there would hardly be any air in the cylinder to mix with the working fluid during pressurization. Therefore, the gas in the cylinder was considered a pure substance and the gas properties (μ , k , c_p) were precisely known.

The apparatus was leak tested after reassembly. The leak testing was usually done at 35 psig and the rate of pressure drop was limited to 0.02 psi/min. If the rate was larger

Table 2.2
 Re_{in}/Pe_w for Stirling Machines

Machine	Compression Space		Expansion Space		Symbol		Notes
	Re_{in}/Pe_w	Pe_w	Re_{in}/Pe_w	Pe_w	C	E	
RE1000	20.7	6840	56.4	2070	A	a	3,1
SPRE	12.1	30800	27.1	5510	B	b	3,1
SDSE	21.7	3120	24.6	832	C	c	3,1
MOD-1	14.5	52500	79.1	6960	D	d	4,1
STC250PT	122.1	2370	99.6	2510	E	e	3,1
GPU3	33.3	7080	58.4	1275	F	f	4,1
ARMY	1.3	551000	5.6	90700	G	g	3,1
STC300	49.1	2410	33.5	14920	H	h	5,1
OXFORD	922.1	2	67.2	538		i	5,1
FORD 4215	24.8	108700	73.2	15490	J	j	4,2
GM 4L23	47.0	28400	53.7	4320	K	k	4,2

Notes:

1. Calculated from data presented in D. Gedeon memo 12/1/89
2. Calculated from data presented in NACA CR-182108 (Simon and Seume)
3. Free-Piston Engine
4. Kinematic Engine
5. Cryocooler

Figure 2.13:
Theoretical Range of Re_{in}/Pe_w

than this the source of the leak had to be located and repaired. Once the apparatus was pressure tight, the following testing procedure was followed:

- 1) The compressor flywheel was pinned in place at the geometric mean volume.
- 2) While the apparatus was at atmospheric pressure, the pressure transducers and heat flux sensor were adjusted to correct for instrument drift.
- 3) At the start of the day, atmospheric pressure was recorded from a barometer located in the test area.
- 4) The bounce volume space, varying volume space, and fixed volume space were filled to the pressure in the test matrix (Table 2.1). The apparatus was filled in incremental steps to maintain the 50 psi differential pressure for the Kulite pressure transducers. The bounce volume and the backing tube of the Kulite transducers filled faster than the fixed and varying volume space.
- 5) The data acquisition program was then started.
- 6) The computer sampled the initial pressure, volume, and gas temperature in the apparatus. The mass in the cylinder was calculated from these initial values.
- 7) The armature voltage was set to obtain the desired piston frequency to satisfy the test matrix, Table 2.1.
- 8) The pin at the compressor flywheel was then pulled.
- 9) A switch was thrown which allowed the armature voltage to flow to the motor. The motor would then turn over and the compressor piston would begin on the compression stroke.
- 10) Next, the computer took bursts of data to check for steady state. The results of each burst were compared to the results of the previous burst. If the time-averaged pressure and piston frequency were within 1% of the last cycle taken then steady state was assumed to have been reached.
- 11) The computer then took a burst of data from the pressure transducers, the linear potentiometer, the gas thermocouple, and the heat flux sensor. The length of the burst was 4 times the piston period to ensure that 3 complete cycles of data were taken. Approximately 400 measurements were taken by each transducer during each cycle.
- 12) After the experimental run was complete the apparatus had to sit for 20 to 30 minutes to allow the dynamic seals to cool down.
- 13) If no more testing was to be done for a given day, the pressure in the apparatus was vented to a level below 50 psi; therefore, if a leak were to occur before the start of the

next testing period, the differential pressure transducers would never be subjected to more than 50 psi. The pressure was slowly vented to a container of vacuum oil; the vent was line below the level of the oil to keep air from entering the system.

14) The raw data were then analyzed to obtain heat transfer results.

CHAPTER 3

ONE SPACE RESULTS

The test apparatus was initially run without the orifice plate; thus, there was only one working space. By analyzing the results of these experimental runs, the performance of the apparatus was evaluated. The data was compared to previous experimental results obtained by Kornhauser (1989) and Kornhauser and Smith (1993) and also to a model developed by Lee (1983). If the data compared favorably to these previous works, then experimentation would proceed with the orifice plate installed.

Kornhauser conducted experiments with a similar apparatus. The same measurements that were made in his study were measured in this experiment. These included:

- a measure of the magnitude of the oscillating pressure, P^*
- a measure of the phase difference between the pressure and the volume, P_{lead}
- the cyclic lost work, $\oint PdV$
- a first law analysis to measure heat flux, and
- a prediction of heat flux and of the phase difference between heat transfer and the gas to wall temperature difference ($T - T_w$).

Lee developed a model to predict heat transfer and cyclic lost work. In his model, the magnitude of the cyclic lost work was a function of volume ratio. It allowed formulation of a non-dimensional cyclic lost work which was independent of volume ratio. The model also predicted the magnitude of heat transfer and the phase difference between heat transfer and gas temperature as a function of oscillating Peclet number.

Lee's model was based on volume ratios approaching 1.0. The tests conducted by Kornhauser and Smith, and Kornhauser were done at volume ratios of 2, 4, and 8. The results at $R_v = 2$ matched the model, but those at 4 and 8 did not. Since the model was based on $R_v \approx 1.0$, they assumed that the model would be good for the range $1 \leq R_v \leq 2$.

The data from this test apparatus will hopefully verify the data taken by Kornhauser

and Kornhauser and Smith, while extending their data sets to higher oscillating Peclet numbers and lower volume ratios. This data can then be used to test their assumption concerning applicable volume ratios.

3.1 TEST CONDITIONS

The one space experiments were run under the following testing conditions:

- helium was used as the working fluid, $\gamma = 1.67$
- no orifice plate was installed
- volume ratio = 1.1
- bore/stroke = 2.0
- pressure and piston frequency were varied according to Table 2.1.

Due to limitations on the motor torque, the following pressure/piston frequency combinations were not attainable: 31.4 rad/sec at 16.7 atm and all of the 1.26 rad/sec points. The motor could not compress the gas at these points.

3.2 PRESSURE MAGNITUDE

The pressure magnitude, P^* , gave an indication of the magnitude of the measured pressure in comparison to the corresponding adiabatic and isothermal pressure. This quantity was first proposed by Chafe (1988), and then used by Kornhauser (1989). P^* was given by:

$$P^* = \frac{P_1 - P_{1_{is}}}{P_{1_{ad}} - P_{1_{is}}} \quad 3.1$$

The pressures used in equation 3.1 were the magnitudes of the first harmonic for the different waves, as found by Fourier analysis. The calculations for the adiabatic and isothermal waves were as follows:

$$P_{ad_n} = P_o \left(\frac{V_o}{V_n} \right)^\gamma \quad 3.2$$

and

$$P_{is_n} = P_o \left(\frac{V_o}{V_n} \right) \quad 3.3$$

where P_o and V_o were the initial pressure and volume of the gas.

When P^* equalled 1.0, the magnitude of the measured pressure would be the same as that of the adiabatic pressure. Here, the heat transfer would be zero and the change in energy would be equal to work. If P^* were 0.0, the magnitude of the measured pressure would be the same as that of the isothermal pressure. At this condition, the change in energy would be zero and the heat transfer would be equal to work.

3.2.1 CURRENT EXPERIMENTAL RESULTS

The pressure magnitude was found to be a function of oscillating Peclet number, with P^* increasing with increasing Pe_w , as shown in Figure 3.1. Most of the values of pressure magnitude were above 0.90; therefore, the amount of the cyclic gas to wall heat transfer was low relative to the work for most of the experimental runs.

3.2.2 PREVIOUS RESULTS

Kornhauser (1989) measured the pressure magnitude at different engine geometries and with different working fluids. The oscillating Peclet number brought together all of the bore/stroke data; but, data at different volume ratios and different working fluids were not brought together. The runs made with the various working fluids showed that the monatomic gases (helium and argon) correlated well with Pe_w and the diatomic gases (nitrogen and hydrogen) correlated well with Pe_w ; however, the diatomic gases showed slightly higher pressure magnitudes than the monatomic gases for the same oscillating Peclet number. This agreed with an expression for pressure magnitude developed from Lee's model. He also observed that, for a common oscillating Peclet number, the lower the volume ratio the higher the pressure magnitude.

The data was found to be a function of the oscillating Peclet number at all of the engine geometries and with all of the working fluids. At low Pe_w the pressure magnitude

Figure 3.1: One Space P^* Results
 $R_v = 1.1$, $D/L_s = 2.0$, He

was near 0.0. As Pe_w was increased, the pressure magnitude increased towards 1.0.

3.2.3 COMPARISON TO PREVIOUS RESULTS

The current experimental data was plotted with Kornhauser's measurements at different volume ratios with helium as the working fluid as shown in Figure 3.2. The experimental data at $R_v = 1.1$ was very similar to Kornhauser's data at $R_v = 2$; although, the data at $R_v = 1.1$ showed a slightly higher magnitude. The two sets of data showed that there was not much difference in the volume ratio at low ratios, less than 2, but a great deal of difference at higher ratios. Kornhauser and Smith's assumption was then confirmed, and Lee's model was valid for the range $1.0 \leq R_v \leq 2.0$

3.3 PRESSURE PHASE LEAD

The pressure phase lead, Φ_p , was defined as the amount of phase shift between the measured pressure and the corresponding adiabatic or isothermal pressure. The phase was calculated from the first harmonics of these pressures. Since the adiabatic and isothermal waves were exactly 180° out of phase with the measured volume signal, the pressure phase lead also represented the amount of phase shift between the pressure and volume signals. The phase of a signal was defined as:

$$phase = \tan^{-1} \frac{\alpha}{\beta} \quad 3.4$$

where:

α = the cosine component of the first harmonic and

β = the sine component of the first harmonic.

The phase lead of the pressure signal was:

$$\Phi_p = phase_{signal} - phase_{ad} = phase_{signal} - phase_{is} \quad 3.5$$

Figure 3.2
 Current and Previous P^* Results

3.3.1 CURRENT PHASE LEAD RESULTS

The pressure phase lead results were found to be a function of the oscillating Peclet number and are plotted in Figure 3.3. The pressure led the volume at all pressure/piston frequency combinations. P_{lead} decreased in magnitude, proportionally to $Pe_\omega^{-1/2}$, as the oscillating Peclet number was increased, which indicated that the pressure and volume were closer in phase to each other. The lead decreased from a maximum of 4.4° , at $Pe_\omega = 115$, to a minimum of 0.28° at a Pe_ω of 46,000.

3.3.2 PREVIOUS PHASE LEAD RESULTS

Kornhauser (1989) measured pressure phase lead at different engine geometries and working fluids. The data at different bore/stroke ratios were brought together with the oscillating Peclet number, but different volume ratios and working fluids were not. He found that, for the same oscillating Peclet number, the diatomic gases (hydrogen and nitrogen) showed lower amounts of lead than the monatomic gases (helium and argon). Also, higher volume ratios (4 and 8) showed a larger amount of pressure lead than volume ratio of 2. The affect of γ was predicted by an extension of Lee's model, but the R_v effect was not.

For all experimental conditions, the pressure phase lead was small at low oscillating Peclet numbers. As Pe_ω was increased, the amount of lead increased proportionally to Pe_ω . At Pe_ω of approximately 30, the lead started to decrease with increasing Pe_ω . This decrease was found to be proportional to $Pe_\omega^{-1/2}$. In a range of intermediate Peclet number, approximately 10 to 100, a breakdown in the correlation was observed. In this breakdown region low speed - high pressure points had higher amounts of lead than high speed - low pressure points. This breakdown in the correlation was seen in all of Kornhauser's plots correlating the different engine geometries and working fluids.

3.3.3 COMPARISON TO PREVIOUS RESULTS

The current experimental data was plotted with Kornhauser's pressure phase lead data at different volume ratios with helium as the working fluid, as shown in Figure 3.4. The current experimental data, at volume ratio of 1.1, matches extremely well with

Figure 3.3:
One Space Pressure Phase Lead Results
 $R_v = 1.1$, $D/L_s = 2.0$, He

Figure 3.4:
Current and Previous Pressure Phase Lead Results

Kornhauser's data at volume ratio of 2 and was lower in magnitude than at R_v equal to 4 and 8. No data points were taken in the intermediate range of oscillating Peclet numbers where the breakdown in the correlation occurred in Kornhauser's data.

3.4 NON-DIMENSIONAL LOSS

Loss is a useful parameter in the study of gas springs. The cyclic lost work is the difference between the work put into the spring and the work extracted from it, and is calculated from $-\oint PdV$ around the cycle. Losses will be presented in a non-dimensional form shown theoretically by Lee (1983) and experimentally by Kornhauser and Smith (1993).

3.4.1 TYPES OF LOSS

The overall loss in Stirling machines is attributed to different areas. Chen, et al. (1984) described four important loss mechanisms: adiabatic cylinder effects, transient heat transfer loss in semi-adiabatic cylinders, pressure drop, and gas leakage. Each of these mechanisms reduced the efficiency due to the level of irreversibility introduced.

Losses due to adiabatic cylinder effects would be created when a gas at one temperature flows between spaces and comes into contact with a gas at a different temperature. This source of loss was not present in the one space experiments; however, this loss will occur in the experiments with the orifice plate. In the one-space apparatus there was no source of pressure drop, and gas leakage from the various volume spaces was eliminated as much as possible. Therefore, the largest contribution to cyclic lost work in the one space apparatus was the loss associated with cyclic gas to wall heat transfer. This type of heat transfer resulted in an increase in entropy because the gas released heat at high temperatures, and then regained the heat later in the cycle, when the gas was at a lower temperature. This was an irreversible thermodynamic process.

3.4.2 PRESSURE-VOLUME PLOTS

The lost work of a gas spring can be graphically visualized by plotting pressure against volume around the cycle. If the cycle was adiabatic this plot would result in a

Figure 3.5: Pressure vs Volume
Minimum Pe_ω

Figure 3.6: Pressure vs Volume
Maximum Pe_ω

curve; however, because the cycle was not truly adiabatic the plot resulted in an enclosed loop. The area enclosed by the loop defined the lost work. The inefficiency of the cycle was the ratio of the area enclosed by this loop to the area of the plot beneath the loop. The smaller this ratio, the more efficient the cycle was. Figures 3.5 and 3.6 show the pressure plotted versus volume at maximum and minimum Pe_w . The amount of lost work was less at the maximum Peclet number, while the efficiency was greater.

3.4.3 METHOD OF CALCULATION

A non-dimensional loss was used experimentally by Kornhauser and Smith (1993) to calculate the magnitude of loss:

$$LOSS_{nd} = \frac{-\oint P dV}{P_o V_o \left(\frac{P_a}{P_o} \right)^2 \frac{\gamma - 1}{\gamma}} \quad 3.6$$

where:

P_o = the mean pressure over the cycle,

V_o = the mean volume, and

P_a/P_o = a measure of the pressure variation and was defined as

$$\frac{P_a}{P_o} = \frac{P_{max} - P_{min}}{P_{max} + P_{min}} \quad 3.7$$

This non-dimensional loss correlated their experimental data well at different geometries and working fluids.

3.4.4 CURRENT LOSS RESULTS

The non-dimensional loss correlated well with oscillating Peclet number and is plotted in Figure 3.7. The magnitude of the loss decreased with increasing oscillating Peclet number and this decrease was proportional to $Pe_w^{-1/2}$. The non-dimensional loss decreased from 0.404 at $Pe_w = 115$ to 0.024 at $Pe_w = 46,000$. This decrease in the

Figure 3.7: One Space Loss Results
 $R_v = 1.1$, $D/L_s = 2.0$, He

magnitude of the loss indicated a decrease in the amount of cyclic gas to wall heat transfer.

3.4.5 PREVIOUS LOSS RESULTS

Kornhauser and Smith (1993) measured loss with different engine geometries and with different working fluids. The non-dimensional loss, equation 3.6, brought together the data from different bore/stroke ratios, but not the data at different volume ratios and for different working fluids (different γ). For points on common Pe_w , the amount of loss at higher volume ratios was larger than at lower volume ratios. The non-dimensional loss brought together the two lighter gases (helium and hydrogen) while the two heavier gases (nitrogen and argon) showed a lot of scatter. Lee predicted that the non-dimensional loss would not be a function of either R_v or molecular weight.

The previous data showed that points of low oscillating Peclet number, < 1 , corresponded to low loss. Loss increased with increasing oscillating Peclet number, and the increase was found to be proportional to Pe_w . At an oscillating Peclet number of approximately 30, the magnitude of the loss started to decrease with increasing Pe_w . This decrease was found to be proportional to $Pe_w^{-1/2}$. At intermediate values of the oscillating Peclet number, between 10 and 100, there was a breakdown in the correlation. Here, points of low speed - high pressure had higher loss than points of high speed - low pressure. This breakdown occurred under all engine geometries and with all working fluids.

3.4.6 COMPARISON TO PREVIOUS RESULTS

The current data, at volume ratio of 1.1, was plotted with Kornhauser and Smith's data for the different volume ratios with helium as the working fluid, as is shown in Figure 3.8. The current data matched extremely well versus the previous data at $R_v = 2$. This confirmed their assumption; that the magnitude of loss at volume ratios below 2 would be essentially the same as 2. Kornhauser and Smith's data set was also extended to higher values of oscillating Peclet number and to lower volume ratios.

3.5 TEMPERATURE MEASUREMENTS

The temperature of the gas in the test section was found in two ways. The mass

Figure 3.8
Current and Previous Loss Results

mean temperature was calculated with the ideal gas law and pressure and volume measurements:

$$T_n = \frac{P_n V_n}{m_o R} \quad 3.8$$

where:

m_o = the initial mass in the system, calculated from the initial pressure, volume, and temperature of the gas in the system, and

n = the n^{th} point in the cycle.

The temperature was also measured with a thermocouple approximately located in the centerline of the varying volume space. The gas thermocouple was used to see if the temperature of the gas flowing through the orifice was equal to the mass mean temperature; here no orifice was present.

3.5.1 COMPARISON BETWEEN TEMPERATURE MEASUREMENTS

The gas temperatures found with the two different techniques were not equal. The gas thermocouple lagged the mass mean temperature and the thermocouple measurement also exhibited a dip when the piston was at top center. The mass mean temperature was larger in magnitude and in amplitude than the gas thermocouple measurement. These differences between the two temperatures are shown in Figure 3.9, which shows the measured temperatures at 6.28 rad/sec and 1 atm. The lag of the thermocouple signal, the dip at top center, and the difference in the magnitude and amplitude between the temperatures are all shown in the plot.

3.5.1.1 PHASE LAG IN THE GAS THERMOCOUPLE

The phase difference between the two temperatures was calculated from the distance, in degrees of the cycle, between the signals at maximum temperature. The thermocouple lagged the mass mean temperature at all pressure/piston frequency combinations, by as much as 30°. For points of common oscillating Peclet number, there was a greater amount of lag in the points at higher piston frequencies. This lag was

Figure 3.9: Temperature Measurements

thought to be due to the response time of the thermocouple. The thermocouple did not seem to be quick enough to measure the instantaneous gas temperature at higher piston frequencies.

3.5.1.2 DIP IN GAS THERMOCOUPLE MEASUREMENT

The temperature measured from the thermocouple experienced a dip when the piston was near top center; however, this dip was not seen in the mass mean temperature. This dip was seen at low pressures and low to intermediate piston frequencies.

The dip was probably caused by a boundary layer attached to the piston face. The temperature measured by the thermocouple would drop if the probe tip was inside this boundary layer, since the temperature inside a boundary layer would be lower than that of the free stream. The piston face was modelled as an oscillating flat plate (Schlichting, 1979), and the thickness of the resulting velocity boundary layer was given by:

$$\delta = \sqrt{\frac{\nu}{n}} \quad 3.9$$

where:

n = frequency of the oscillation and

ν = kinematic viscosity.

After the kinematic viscosity was replaced by μ/ρ and after the density and frequency were replaced with measured quantities (ω , P , T), the following form was obtained:

$$\delta = \sqrt{\frac{\mu R T_o 2\pi}{\omega P_o}} \quad 3.10$$

where:

μ = dynamic viscosity,

R = the gas constant,

T_o = mean temperature of the gas, and

P_o = mean pressure of the gas.

This analysis showed that the boundary layer would decrease with increasing pressure and

piston frequencies. For a Prandtl number of 0.72 (the Prandtl number for Helium is 0.70), the thermal boundary layer is larger than the velocity boundary layer (Walz, 1969) and,

$$\frac{\delta}{\delta_T} = 0.85 \quad 3.11$$

The thickness of the thermal boundary layer was calculated from the model for different pressure/piston frequency points. Since the thermocouple was approximately 0.25 in. above the piston face at top center, a thermal boundary layer thicker than this would affect the thermocouple measurement. The predicted thermal boundary layer thicknesses ranged from 1.14 in. at low Pe_w to 0.06 in. at high Pe_w . The model accurately predicted which points were effected by the thermal boundary layer. However, the model also predicted a boundary layer thick enough to affect the thermocouple reading at pressure/piston frequency combinations that did not suffer a dip in the measurement. The predicted boundary layer thickness at those points, which did not match the experimental results, was 0.34 in. or 0.09 in. above the actual thermocouple placement.

3.5.1.3 AMPLITUDE AND MAGNITUDE DIFFERENCE BETWEEN THE TEMPERATURES

The amplitude and magnitude of the mass mean temperature was larger than that of the temperature measured from the gas thermocouple.

The amplitude of the mass mean temperature was between 30 and 33 R for all pressure/piston frequency combinations. For a given pressure, the amplitude of the thermocouple measurement was similar to that of the mass mean temperature at low piston frequency. However, the amplitude of the thermocouple measurement then decreased as the frequency was increased (for the same pressure). Therefore, at a given pressure the amplitude difference between the mass mean temperature and the thermocouple measurement increased with increasing piston frequency. One possibility for this difference in amplitude was heat radiating from the thermocouple to the piston face. These radiative affects would be more pronounced at low piston frequencies because, at these frequencies, the piston face was at near top center positions for a greater amount of time. Another possibility for the drop in the thermocouple measurement with increasing piston frequency was the response time of the thermocouple.

The magnitudes of the two temperatures were also different. The temperature measured by the pressure and volume data had a larger magnitude, for all points, than the thermocouple measurement. This larger magnitude caused the mean temperature calculated from the mass mean temperature to be larger than the thermocouple. This difference in magnitude was probably due to the leakage in the piston-cylinder. The mass used in the calculation of the mass mean temperature was calculated using the ideal gas law with the initial pressure, volume, and temperature of the gas in the apparatus. During the running of the experiment, the amount of leakage was small for one cycle, but the cumulative leakage would have been larger. By the time steady state was reached, the mass in the system was probably different than the initial mass in the system. Using the ideal gas law, a 1% change in the mass of the system would result in a difference in mass mean temperature between 5 and 6 R (since the temperature swing was between 500 and 600 R).

3.5.2 MODIFIED MASS MEAN TEMPERATURE

The temperature difference ($T - T_{\text{wall}}$) had to be known in order to predict the heat flux. Since the gas thermocouple measurement lagged the pressure and volume, the mass mean temperature was to be used in the prediction. However, the mass mean temperature had a magnitude offset from the temperature measured from the gas thermocouple. Therefore, the temperature measurement and the calculation of temperature had different mean values. It was assumed that the mean of the mass mean temperature was offset due to mass leakage in the system which caused the initial mass used in equation 3.8 to be different than the mass which was actually in the system when the calculation was made.

In order to account for this magnitude offset, a modified mass mean temperature was calculated based on the assumption that the mass mean temperature should have had the same mean value as the temperature measured by the gas thermocouple. The modified mass mean temperature was calculated by:

$$T'_{m_n} = T_{m_n} + \left(\frac{\sum T_{T/C}}{\#pts} - \frac{\sum T_m}{\#pts} \right) \quad 3.12$$

where the subscript n was the nth temperature point.

The modified mass mean temperature was very similar in magnitude and amplitude to the thermocouple measurement, but did not have the instrument lag or the dip associated with it. The modified mass mean temperature was essentially a rescaled version of the mass mean temperature. This temperature was used in the calculation of the heat transfer coefficients (Nu_c), equations 3.23 and 3.24, and in the prediction of the heat flux, equation 3.22.

3.6 HEAT FLUX

In-cylinder heat flux, the rate of heat transfer per unit area, was measured in two ways. The first method used the pressure and volume measurements and the first law of thermodynamics, while the second method used the surface temperature measurements provided by the heat flux sensor.

In-cylinder heat flux was predicted as well as measured. A complex Nusselt number was calculated from the pressure, temperature, and volume measurements and then was used to predict the heat flux. The complex Nusselt number method was a circular procedure since the data to be predicted by the method, measured heat flux, was used in the calculation of the complex Nusselt number. However, the method had demonstrated the ability to predict the phase difference between the heat flux and temperature difference which occurs under conditions of oscillating pressure and oscillating flow.

The measured and predicted heat flux were compared against one another, while the complex Nusselt numbers were compared against the Nusselt numbers calculated by Kornhauser (1989). Lee (1983) evaluated a complex Nusselt number in his model of heat flux in oscillating pressure. The experimentally calculated phase shift between heat flux and temperature was also compared with Lee's model.

3.6.1 HEAT FLUX MEASURED FROM PRESSURE AND VOLUME

Heat flux was calculated with pressure and volume data. This calculation used the first law of thermodynamics as well as the ideal gas law.

For a closed thermodynamic system, the first law states that the heat transfer is a function of the change in internal energy and of work. With heat and work defined as positive out of the system,

$$\delta Q = -dU - PdV \quad 3.13$$

For an ideal gas with constant specific heat, the internal energy was given by:

$$U - U_{ref} = mc_v(T - T_{ref}) \quad 3.14$$

The ideal gas law was then used to convert the temperature in equation 3.14 to pressure and volume. After differentiating the internal energy, and using the relationship $c_v/R = 1/(\gamma-1)$ the following form was obtained:

$$\delta Q = -\left(\frac{\gamma}{\gamma-1}\right) PdV - \left(\frac{1}{\gamma-1}\right) VdP \quad 3.15$$

Heat flux was the rate of heat transfer per unit area, so

$$\dot{q}''_n = \frac{1}{A} \frac{\delta Q}{dt} = -\frac{V_n}{A_n} \left(\frac{1}{\gamma-1}\right) \left(\frac{dP}{dt}\right)_n - \frac{P_n}{A_n} \left(\frac{\gamma}{\gamma-1}\right) \left(\frac{dV}{dt}\right)_n \quad 3.16$$

where:

A = total surface area and

the subscript n indicates the n th point, as heat flux was calculated on a point by point basis within the cycle.

The heat flux in equation 3.16 was a spatially averaged heat flux.

The pressure was measured by the differential pressure transducer located in the head, while the volume was measured with the linear potentiometer. The time derivatives were calculated using Fourier transforms (see Appendix A).

3.6.2 HEAT FLUX MEASURED FROM THE HEAT FLUX SENSOR

Heat flux was also measured from the wall surface temperature measured by the heat flux sensor in the head. The measured wall temperature is shown in Figure 3.10 for the conditions of 5 atm and 12.6 rad/sec.

Figure 3.10:
Measured Wall Temperature

The cylinder wall was modeled as a 1-D, semi-infinite solid, with thermal properties independent of temperature, which was subjected to a time dependant sinusoidal surface temperature variation of the form

$$T = T_o + T_c \cos(\omega t) + T_s \sin(\omega t) \quad 3.17$$

The heat flux, after initial transient disturbances had died away, was given by Carslaw & Jaeger (1959) as:

$$\dot{q}'' = \beta_q \sqrt{\frac{\omega}{2}} [(T_s + T_c) \cos(\omega t) + (T_s - T_c) \sin(\omega t)] \quad 3.18$$

where $\beta_q = (\rho c_p k)^{1/2}$ was a measure of the thermal properties of the wall. For the PyrexTM 7740 substrate, which the thin film resistor was mounted on, $\beta_q = 0.0743$ BTU/ft²Rs^{1/2}.

The surface temperature measured from the heat flux sensor was decomposed by Fourier decomposition into sine and cosine components and was then recombined to give:

$$\begin{aligned} \dot{q}''_n = \beta_q \sqrt{\frac{\omega}{2}} \sum_{j=1}^{IT} \sigma_j [j^{1/2} (\beta_j + \alpha_j) \cos(\frac{\pi j t_n}{t_{tot}}) \\ + j^{1/2} (\beta_j - \alpha_j) \sin(\frac{\pi j t_n}{t_{tot}})] \end{aligned} \quad 3.19$$

where (inside the summation):

α = cosine component of Fourier decomposition,

β = sine component of Fourier decomposition,

σ = Lanczos smoothing factor, and

IT = the number of Fourier coefficients used.

For more information on the method of Fourier decomposition see Appendix A.

The quantity β_q for PyrexTM 7740 was different than that for steel (the cylinder surface). For the Pyrex substrate $\beta_q = .0743$ BTU/ft²Rs^{1/2}, while for steel it was 0.61 BTU/ft²Rs^{1/2}. This difference in the thermal properties between the steel and the substrate

caused an inaccuracy in the wall temperature measurement because of a temperature gradient introduced where the two dissimilar substances were joined. However, since the heat flux measured from this transducer was to be a secondary measurement, this inaccuracy was not very important. The one space heat flux results showed that the heat flux measured from the heat flux sensor was extremely similar to other methods used to evaluate heat flux, so the inaccuracy in the measurement seemed minor.

3.6.3 PREDICTION OF HEAT FLUX USING A COMPLEX NUSSELT NUMBER METHOD

An effort was made to predict the heat flux using a complex Nusselt number, Nu_c , which was calculated from temperature and pressure/volume data. The complex Nusselt number and the mass mean temperature were then used to predict heat flux.

Various studies, summarized in sections 1.6.2.2 and 1.6.2.3, have shown that, under oscillating conditions, the heat flux was out of phase with temperature difference. The complex Nusselt number method incorporates this phase difference, while ordinary convective heat transfer models do not.

3.6.3.1 PREDICTING HEAT FLUX WITH Nu_c

Lee (1983) developed a model which calculated heat flux in oscillating pressure and oscillating flows. In the model, a complex Nusselt number, which resulted from the use of a complex temperature difference to solve the heat conduction problem, was used to evaluate the heat flux. The complex Nusselt number was obtained by dividing complex heat flux by complex temperature difference.

Newton's law of convection can be written in terms of complex temperature, complex Nusselt number, and complex heat transfer:

$$\dot{q}'' = \frac{k}{Dh} Nu_c (T_c - T_w) \quad 3.20$$

Based on a sinusoidal temperature variation, Kornhauser (1989) calculated the heat transfer to be

$$\dot{q}'' = \frac{k}{D_h} [Nu_r (T - T_w) + \frac{Nu_i}{\omega} \frac{d\Delta T}{dt}] \quad 3.21$$

where:

T_w = wall temperature,

$\Delta T = T - T_w$,

T = the modified mass mean temperature (given in equation 3.12),

k = thermal conductivity of the gas,

D_h = hydraulic diameter,

Nu_r = the real portion of the complex Nusselt number, and

Nu_i = the imaginary portion of the complex Nusselt number.

In the derivation, the variation in wall temperature was assumed to be negligible in comparison to the variation in the gas temperature.

The heat flux in equation 3.21 consists of one part proportional to the temperature difference plus one part proportional to the rate of change of the temperature difference.

In order to predict heat flux with this method the complex Nusselt number needed to be calculated.

3.6.3.2 CALCULATING THE COMPLEX NUSSELT NUMBER

The heat transfer expression which used the complex Nusselt number, equation 3.21, has two coefficients: Nu_r and Nu_i . Therefore, at least two $q'' - \Delta T - d\Delta T/dt$ were needed to calculate the complex Nusselt number. Previous works had suggested that the complex Nusselt number was a function of Pe_w . Since instantaneous Pe_w varies throughout the cycle (because density varies throughout), Kornhauser (1989) calculated a mean Nu_c over the cycle which would have no dependence on Pe_w and still be a best-fit to all of the data. The cyclic mean complex Nusselt number was calculated to give an optimum least squares fit to all $q'' - \Delta T - d\Delta T/dt$ data points. This fit gave:

$$Nu_r = \left(\frac{D_h}{k} \right) \frac{\sum (\dot{q}''_n \Delta T_n) \sum \Delta T_n^2 - \sum (\dot{q}''_n \Delta T'_n) \sum \Delta T_n \Delta T'_n}{\sum \Delta T_n^2 \sum \Delta T'^2_n - (\sum \Delta T_n \Delta T'_n)^2} \quad 3.22$$

and

$$Nu_i = \left(\frac{\omega D_h}{k} \right) \frac{\sum \dot{q}''_n \Delta T'_n \sum \Delta T_n^2 - \sum \dot{q}''_n \Delta T_n \sum \Delta T_n \Delta T'_n}{\sum \Delta T_n^2 \sum \Delta T'^2_n - (\sum \Delta T_n \Delta T'_n)^2} \quad 3.23$$

In equations 3.22 and 3.23:

$$\sum = \sum_{n=0}^{360} \text{ and } \Delta T' = \frac{d\Delta T}{dt}$$

The wall temperature used in the calculation of the real and imaginary portions of the Nusselt number was the averaged wall temperature. Previous studies have shown that the amplitude of the wall temperature was very small in comparison to the amplitude of the gas temperature; therefore, the wall temperature was assumed to be a constant. In the one-space data, the amplitude of the wall temperature was a maximum of 3.4% of the gas temperature; this was measured at the maximum mean pressure.

3.7 HEAT FLUX RESULTS

The results of the two different methods of measuring heat flux will be compared against one another, while the predicted heat flux will be compared against the heat flux measured from the pressure and volume data. The results of the complex Nusselt number will be compared against Kornhauser's results and also to Lee's model.

3.7.1 MEASURED HEAT FLUX RESULTS

The two methods of measuring the heat flux were compared at piston frequencies of 6.28 and 62.8 rad/sec at low, intermediate, and high mean pressures. These plots are shown in Figures 3.11 to 3.16. A simple error analysis, Appendix E, showed that the heat flux measurements were accurate to +/- 5%.

3.7.1.1 COMPARISON BETWEEN THE MEASUREMENT METHODS

The heat flux measured from both methods was found to be a function of oscillating Peclet number. The amplitude of the heat flux increased as the oscillating Peclet number was increased. The amplitude of the heat flux was similar for common Pe_ω calculated at different pressure/piston frequency combinations.

The heat flux measured from the heat flux sensor was similar in magnitude and amplitude to the heat flux measured from the pressure and volume data. However, the large oscillations, which occurred at high piston frequencies, in the heat flux measured from the pressure and volume data made comparisons difficult at these points.

Both of the measurements showed that the maximum heat transfer occurred prior to the top center position, while the pressure, volume and temperature were maximum at top center. This was due to the fact that, under conditions of compression and expansion, the heat transfer can lead the temperature by as much as 45°. This phenomena will be discussed with the results of the complex Nusselt number in section 3.7.2.2.

3.7.1.2 NOISE IN THE HEAT FLUX

The heat flux measured from the pressure and volume data showed a high frequency noise component, while the heat flux measured from the sensor did not. The amplitude of this noise increased as ω was increased. The frequency of the noise was approximately 10x the cycle frequency for all pressure/piston frequency points.

This noise was probably the result of the "hit" and "ringing" measured by the linear potentiometer at midstroke (see section 2.2.2.2.1.4). This was supported by the following observations:

1) The heat flux sensor did not show the noise, and if the gas were undergoing some sort of fluctuation the sensor would have measured it.

2) The noise was present in the high and low speed runs which used different sheave arrangements. This indicated that the noise was not related to the drive apparatus.

3) The affects of the noise could be manipulated numerically by changing the amount of coefficients used to decompose the volume measurement. In this process 20 Fourier coefficients were used to decompose the original pressure and volume signals. In Fourier decomposition, the fewer the amount of coefficients used the more noise that was

eliminated. However, if too few coefficients were kept then part of the original signal could have been lost. With this in mind, the pressure and volume data were differentiated in two ways. First all 20 coefficients were kept on the pressure and only 10 with the volume, then 20 were kept on the volume and 10 on the pressure. When the amount of pressure coefficients was reduced hardly any change in the heat flux was seen. The amplitude of the noise was similar to the original and the frequency dropped from 10x to 9x. However, a greater change was observed when the number of volume coefficients was reduced. The amplitude of the noise dropped by a factor of 3 and the frequency dropped to 7x. This testing was done on the 16.7 atm, 62.8 rad/sec experimental run since the amplitude of the noise was greatest at this condition.

4) When the number of Fourier coefficients were doubled (to 40) on the pressure and volume, the peak magnitude of the noise increased slightly and intermediate humps of noise were noticed between the original spikes of noise.

5) The noise in the heat flux was unchanged when a least squares method was used to calculate dV/dt and dP/dt . This indicated that the technique used to smooth the data and to calculate these derivatives was not the source of the problem.

After the above observations were made it was felt that the noise in this heat flux measurement was related to the "hit" and "ringing" which affected the volume measurement (heat flux was a function of dV/dt). However, this was an assumption since nothing definitive could be proven.

3.7.2 RESULTS OF PREDICTED HEAT FLUX/NU_c

The heat flux was predicted using the complex Nusselt number method. The predicted heat flux was compared to the heat flux measured from pressure and volume data to determine the accuracy of the prediction. The magnitude of the complex Nusselt number and the phase difference between the heat flux and the temperature were both compared to Kornhauser's data (1989). The phase difference was also compared to Lee's model (1983).

**Figure 3.11: Measured Heat Flux
Low Pressure, Low Piston Frequency**

Figure 3.12: Measured Heat Flux
Low Pressure, High Piston Frequency

**Figure 3.13: Measured Heat Flux
Medium Pressure, Low Frequency**

**Figure 3.14: Measured Heat Flux
Medium Pressure, High Frequency**

Figure 3.15: Measured Heat Flux
High Pressure, Low Piston Frequency

**Figure 3.16: Measured Heat Flux
High Pressure, High Piston Frequency**

3.7.2.1 COMPARISON OF PREDICTED AND MEASURED HEAT FLUX

The method used to predict the heat flux was accurate at all pressure/piston frequency combinations. The magnitude and the phase of the predicted heat flux accurately reflected the heat flux measured from the pressure and volume data. The measured and predicted heat fluxes were plotted for the conditions of low pressure - low piston frequency and for intermediate pressure - high piston frequency. These plots are presented in Figures 3.17 and 3.18 and are typical of all experimental conditions. The predicted heat flux resulted in a smoother version of the measured heat flux, regardless of the amount of noise in the volume measurement.

3.7.2.2 COMPLEX NUSSELT NUMBER RESULTS

The complex Nusselt number was composed of a real part and an imaginary part. These two components were used to find the magnitude and the phase of the complex Nusselt number. The magnitude was given by

$$Nu_{c_{mg}} = \sqrt{Nu_r^2 + Nu_i^2} \quad 3.24$$

while the phase was given by

$$phase = \tan^{-1} \frac{Nu_i}{Nu_r} \quad 3.25$$

The magnitude and the phase difference were compared to Kornhauser's results, and then the phase difference was compared to Lee's model.

3.7.2.2.1 CURRENT RESULTS

The real and imaginary parts of Nu_c and its magnitude were both found to be functions of the oscillating Peclet number, and are shown in Figures 3.19 and 3.20. Both the magnitude of the complex Nusselt number and its real and imaginary parts increased with increasing Pe_w . The phase difference, calculated from Nu_r and Nu_i , varied from 39° to

**Figure 3.17: Predicted Heat Flux
Low Pressure, Low Piston Frequency**

**Figure 3.18: Predicted Heat Flux
Medium Pressure, High Frequency**

Figure 3.19: Complex Nusselt Number

Figure 3.20: Magnitude of Nu_c
 $R_v = 1.1$, $D/L_s = 2.0$, Helium

45° for most of the pressure/piston frequency points. There was an increase in the phase difference for the runs made at maximum piston frequency, $\omega = 62.8$ rad/sec. It was felt that this increase was due to the noise in the volume measurement, which affected the heat flux. This noise seemed to affect the calculation of the phase difference more than the calculation of the magnitude.

A plot of the predicted heat flux versus the modified mass mean temperature is shown in Figure 3.21. This plot shows the heat flux leading the temperature by 39°. The plot was made for $\omega = 12.6$ rad/sec at 5 atm and was typical of the other pressure/piston frequency combinations. Since there was no difference between the phase of the predicted heat flux and the heat flux measured from pressure and volume data, the predicted heat flux was plotted since it was smoother.

3.7.2.2.2 PREVIOUS RESULTS

Kornhauser (1989) calculated Nu_c for various engine geometries and for different working fluids. He found that the results from different bore/stroke ratios and with different working fluids correlated well with the oscillating Peclet number, but that results at different volume ratios did not. The correlation with volume ratio appeared to fail at high Pe_ω where Nu_r and Nu_i were smaller at lower volume ratios.

All of Kornhauser's Nu_c data also showed the following trend with the oscillating Peclet number. At low Pe_ω , < 100 , the real part of the complex Nusselt number was larger than the imaginary part, while at high Pe_ω the real part and the imaginary part were nearly the same. This indicated that at low Pe_ω the heat transfer occurred soon after the temperature difference was introduced, while at high Pe_ω the heat flux led the temperature difference by about 45°. All of the correlations showed a breakdown in the range of $10 < Pe_\omega < 100$ where, for an oscillating Peclet number, high pressure - low speed runs showed a larger real part and smaller imaginary part than low pressure - high speed runs. This was the same breakdown region that was seen in the correlations of pressure magnitude, pressure lead, and non-dimensional loss.

Lee used his model of heat transfer to predict the phase difference between heat flux and temperature difference. His model predicted that for $Pe_\omega < 0.2$ the heat flux would occur, without delay, as soon as the temperature difference was introduced, while at $Pe_\omega > 200$ the heat flux would lead the temperature by 45°. This was experimentally

Figure 3.21:
Heat Flux vs Temperature
Phase Difference = 39°
Medium Pressure, Medium Frequency

verified by Kornhauser's work.

3.7.2.2.3 COMPARISON TO PREVIOUS RESULTS

The current experimental data was plotted against Kornhauser's data for various volume ratios with helium as the working fluid. The real and imaginary parts are compared in Figure 3.21, while Figure 3.22 shows the magnitude of the complex Nusselt number.

The current experimental data, $R_v = 1.1$, matched that of Kornhauser's results made at $R_v = 2.0$, while higher volume ratios ($R_v = 4$ and 8) showed larger real parts of the complex Nusselt number. The magnitude of the complex Nusselt number matched that of the previous data at $R_v = 2.0$ and was smaller than the data at $R_v = 4$ and 8 .

The calculated phase difference between the heat flux and temperature difference matched Lee's model. The model predicted that, for the range of Pe_ω tested, the phase difference would be approximately 45° . The experimental results indicated phase differences between 39° and 45° .

3.8 ONE SPACE CONCLUSIONS

The following conclusions were made from the single space experimentation.

- 1) The current experimental data for volume ratio of 1.1 matched that of Kornhauser's data taken at $R_v = 2.0$. Pressure magnitude, pressure lead, non-dimensional loss, and complex Nusselt number data were comparable between the two volume ratios.
- 2) The current experimental data verified a result from Lee's model, that the heat transfer would lead the temperature difference by about 45° at high oscillating Peclet number.
- 3) The data also verified an assumption made about Lee's model. It had been assumed that results in the range of $1 \leq R_v \leq 2$ would match those of $R_v = 2$.
- 4) The complex Nusselt number method was capable of predicting the phase-shifted heat flux.
- 5) Although the good agreements with previous results does not ensure accuracy, this fact gave confidence that the apparatus was performing as designed.
- 6) Kornhauser's data set was extended to lower volume ratios and higher oscillating Peclet numbers.

Figure 3.22:
 Current and Previous Nu_c and Nu_i Results

Figure 3.22: Magnitude of Nu_c
 Current and Previous Results

7) There existed noise in the measured heat flux data and the calculation of phase shift between heat flux and temperature difference. It was thought, but was not proven, that this noise had its source in the volume measurement.

CHAPTER 4

TWO SPACE RESULTS

The addition of an orifice plate to the one space test apparatus created the two space apparatus. This orifice plate was placed slightly above piston top center and created two distinct volume spaces with mass flow between them. These two spaces will be referred to as the "varying volume space" (the volume bounded by the piston and the plate) and the "fixed volume space" (the volume bounded by the plate and the head piece). During the compression stroke mass flowed from the varying volume space to the fixed volume space; the opposite was true during the expansion stroke.

Turbulence was expected to be generated due to the mass flow through the orifice. It was thought that this turbulence would cause an increase in in-cylinder heat transfer and cyclic lost work. The pressure drop across the orifice plate also created an additional source of loss.

In the future, a computer model/simulation, developed by Finkbeiner and Kornhauser (1993), will be used to calculate heat fluxes in the separate volume spaces. At the present time this simulation is not complete.

The two space experiments were run at the same pressures and piston frequencies as the one space experiments. However, the two space experiments were run at a volume ratio of 1.2, while the one space experiments were run at 1.1. Since the one space results ($R_v = 1.1$) matched Lee's model and Kornhauser's data set (at $R_v = 2.0$), the only difference between the results of the two sets of data was expected to be due to the presence of the orifice plate. The pressure magnitude, pressure phase lead, and non-dimensional loss results from the two space apparatus will be presented in comparison to the one space results. Averaged in-cylinder heat flux will also be measured and calculated in a manner similar to that used for the single space experiments.

4.1 TEST CONDITIONS

The two space experiments were run under the following test conditions:

- helium was used as the working fluid, $\gamma = 1.67$
- orifice diameter = 1.36 in.
 - $Re_{in}/Pe_{\omega} = 56$ (D_h based on the total cylinder space)
- clearance gap height = 0.41 in.
 - clearance gap/ $L_g = 0.11$
- volume ratio = 1.2
- bore/stroke = 2.0
- the fill pressure and piston frequency were varied according to Table 2.1.

Due to the lack of adequate motor torque, the following pressure/piston frequency points were not attainable: $\omega = 31.4$ and 62.8 rad/sec at 10 and 16.7 atm, and all of the points at $\omega = 1.26$ rad/sec.

4.1.1 ACTUAL Re_{in}/Pe_{ω} TO DESIGN

The orifice plate that was used in the two space experiments was designed for a ratio of Re_{in}/Pe_{ω} of 100. This design was based on approximations by Kornhauser (1992). However, after the two space testing was completed, the ratio of inlet Reynolds number to oscillating Peclet number was calculated and found to be 56 instead of 100. The experimental Re_{in}/Pe_{ω} was plotted against the Stirling machines (from Table 2.2, pg. 67) as shown in Figure 4.1.

The difference between the design and the experimental ratio was found to be the definition of the hydraulic diameter (Re_{in}/Pe_{ω} was proportional to $1/D_h$). In his approximation, Kornhauser calculated Re_{in}/Pe_{ω} with a hydraulic diameter based solely on a varying volume space. In the experimental apparatus a hydraulic diameter could be calculated for the fixed volume space, the varying volume space, or the total volume (both spaces added together). These three different hydraulic diameters resulted in the following Re_{in}/Pe_{ω} : 52, 109, 56. The ratio calculated based on the varying volume space, 109, was very similar to the result which was expected from the approximation.

Due to the mass flow between the two spaces, the mass was not known in either of the volume spaces at any instant in time; therefore, the heat transfer could not be

Figure 4.1: Actual Re_{in}/Pe_w
 "V" Denotes Experimental Points
 Table 2.2 (pg. 67) Lists the Machines Plotted

measured in either of the two volume spaces. To solve this problem, an averaged in-cylinder heat transfer was measured based on the total mass and total surface area in the system. The oscillating Peclet number was then calculated based on the total surface area and the total volume, from which $Re_{in}/Pe_{\omega} = 56$ was obtained.

4.2 PRESSURE DROP ACROSS THE ORIFICE

The pressure drop across the orifice was measured by subtracting the two pressure signals. One of the pressure transducers was located in the fixed volume space (approximately 18 inches above the orifice plate), while the other was placed in the varying volume space (approximately 0.19 inches below the orifice plate). The pressure difference was then smoothed with Fourier analysis, and the maximum pressure drop was calculated as the maximum difference of this smoothed measurement. The measured pressure drop was then compared to a model which was based on a one-dimensional, steady, incompressible flow.

4.2.1 MEASURED PRESSURE DROP

The measured pressure drop was plotted against piston frequency in Figure 4.2. The measured pressure drop increased with increasing piston frequency with the maximum drop approximately 2 psi and the minimum approximately 0.02 psi. At high ω (≥ 31.4 rad/sec) the pressure drop was proportional to ω^2 , while at low ω (≤ 6.28 rad/sec) the pressure drop was nearly constant for a given pressure.

4.2.2 THEORETICAL PRESSURE DROP

A steady, incompressible, 1-D flow analysis (White, 1986) yielded

$$V_{\max} = C \left(\frac{\Delta P_{\max}}{\rho} \right)^{1/2} \quad 4.1$$

For our apparatus,

Figure 4.2:
Pressure Drop vs Omega

$$V_{\max} \approx \omega \frac{L_s}{2} \left(\frac{A_{cyl}}{A_{orif}} \right)^2 \quad 4.2$$

and the density at midstroke , where $V = V_{\max}$, was approximately

$$\rho_o = \frac{P_o}{RT_o} \quad 4.3$$

where:

ρ_o , P_o , and T_o were cyclic mean values.

Equations 4.3 and 4.2 were substituted into equation 4.1, and then the expression was solved for ΔP . Since the mean temperature was nearly constant for all experimental runs, the following expression was obtained for the theoretical pressure drop

$$\Delta P_{\max} = C\omega^2 P_o \quad 4.4$$

The pressure drop across the orifice was expected to follow this proportionality.

4.2.3 COMPARISON OF MEASURED PRESSURE DROP TO MODEL

The measured pressure drop was non-dimensionalized by dividing it by the cyclic mean pressure. This non-dimensional pressure drop was then plotted against piston frequency and is shown in Figure 4.3. At high piston frequencies (≥ 31.4 rad/sec), the points of $\Delta P/P_o$ were brought together for all of the pressures; however, the data at lower frequencies ($\omega \leq 6.28$ rad/sec) were not brought together. Also, the 1 and 2 atmosphere data, at these lower values of ω , were separated from the higher pressure data.

The proportionality predicted by the model, equation 4.4, was also plotted in Figure 4.3. The model predicted the pressure drop accurately at high piston frequencies, where $\Delta P/P_o$ brought all of the data together. However, at lower piston frequencies the model did not match the data, here the $\Delta P/P_o$ points were larger than the model predicted. There were a couple of possible reasons why the measured pressure drop deviated from the model at low speeds. There could have been a high level of inaccuracy in the differential

Figure 4.3:
Pressure Drop Across Orifice vs Model

pressure transducers at low pressures since the pressure change here was very small. This may explain the low pressure measurements being offset from the ω^2 proportionality at low piston frequencies, but does not explain the deviation at high mean pressure. Another possible reason would be viscous losses through the orifice, which were not included in the derivation of the theoretical pressure drop. At the lower speeds the viscous losses might have been of the same magnitude as the theoretical pressure drop loss in equation 4.4.

4.3 PRESSURE MAGNITUDE

The pressure magnitude, P^* , was calculated in a manner similar to that used for the one space results. With the two spaces, two separate values of P^* were calculated, one was calculated from the pressure measured in the varying volume space and the other from pressure in the fixed volume space.

4.3.1 CALCULATION METHOD

Two calculations of pressure magnitude were made, one based on the pressure measured in the fixed volume space and the other based on the pressure in the varying volume space. In both measurements P^* was given by:

$$P_{(f,v)}^* = \frac{P_{(f,v)} - P_{iso}}{P_{ad} - P_{iso}} \quad 4.5$$

Even though two values of P^* were measured, the adiabatic and isothermal pressures in both measurements were based on the total volume ($V_{var} + V_{fix}$) in the test apparatus. Since the total volume was used for the two calculations of pressure magnitude, both results were computed as if the orifice was not installed (i.e. as if the two space apparatus was a one space apparatus with the pressure in the apparatus equal to either the pressure in the fixed or in the varying volume space). In this way the two sets of experiments were compared on an equal basis, and the difference between the two sets of data was due to the effects of the orifice plate.

4.3.2 COMPARISON TO ONE SPACE RESULTS

The two sets of pressure magnitude results are shown in Figure 4.4. The two space results were lower in magnitude, or further from adiabatic conditions, for the same value of Pe_ω than the one space results. This indicated that the ratio of heat transfer to work was greater in the two space experiments, and seemed to be caused by an increase in in-cylinder heat transfer.

4.3.3 TWO SPACE RESULTS

The two space pressure magnitude data was not brought together with the oscillating Peclet number as is shown in Figure 4.4. The pressure magnitude calculated in the varying volume space was consistently higher than that calculated in the fixed volume space. The difference between the two was plotted in Figure 4.5. It was shown that the difference between the two pressure magnitudes was fairly constant, 0.0038 to 0.0055, at low piston frequencies (≤ 12.6 rad/sec), while, for most cases, the difference increased at higher piston frequencies (≥ 31.4 rad/sec). The increase in the difference between the two P^* calculations occurred at the same piston frequencies where the theoretical pressure drop matched the measured pressure drop. Therefore, it was felt that the difference between the two space P^* calculations at high ω was a result of the pressure drop across the orifice.

A volume averaged pressure magnitude was also calculated. Since the fixed volume was approximately 8x the varying volume, the volume averaged P^* results were very close to the fixed volume data. However, there was still a lot of scatter in the volume averaged P^* data. It was not clear whether the scatter was representative of a real phenomenon or if it was related to experimental noise. In Figure 4.4 the range on the y-axis was very small, which tends to magnify scatter.

4.4 PRESSURE PHASE LEAD

The pressure phase lead was calculated in a manner similar to that described in section 3.3. The pressure phase lead was calculated for the fixed volume space and for the varying volume space, with the adiabatic and isothermal pressures calculated from the total

Figure 4.4: Two and One Space P^* Results

Figure 4.5:
Two Space Interspatial P^*

volume (see section 4.3.1). The phase difference between the two volume spaces was also calculated.

4.4.1 COMPARISON TO ONE SPACE RESULTS

The two space pressure lead results were compared to the one space pressure phase lead results as shown in Figure 4.6. The two space P_{lead} results showed the pressure leading the volume by a greater amount than in the one space experiments.

4.4.2 TWO SPACE P_{lead} RESULTS

The two space pressure phase lead data was shown in Figure 4.6. Not all of the data was brought together with the oscillating Peclet number. The fixed volume data, denoted by 1, was brought together with Pe_{ω} ; all of the fixed space P_{lead} results decreased with increasing Pe_{ω} . The varying volume P_{lead} data, denoted by 2, did not correlate with the oscillating Peclet number. Most of the varying volume data points decreased with increasing Pe_{ω} ; however, six points did not. These points corresponded to the experimental runs made at 31.4 and 62.8 rad/sec.

The difference in pressure phase lead between the two volume spaces, the interspatial P_{lead} , was then plotted against piston frequency as shown in Figure 4.7. A line proportional to ω^2 , the pressure drop proportionality calculated in section 4.2, was also plotted on this graph. At high piston frequencies the difference in the phase lead data between the two volume spaces was found to be proportional to ω^2 . These piston frequencies corresponded to the points of pressure phase lead data in the varying volume space which were separated from the rest of the data. At low piston frequencies the interspatial pressure phase lead measurements were not brought together with ω . Since these same trends at high and low piston frequency were first established by the pressure drop measurements, the interspatial pressure phase lead data at high piston frequencies seemed to be related to the pressure drop across the orifice.

If a volume averaged pressure phase lead were calculated, these points would be very similar to the fixed volume space (since the fixed volume was approximately 8x the varying volume). This volume averaged data correlated well with Pe_{ω} , but the values were higher than the one space data.

Figure 4.6:
Two and One Space Pressure Phase Lead Results

Figure 4.7:
Two Space Interspatial P_{lead}

4.5 NON-DIMENSIONAL LOSS

The loss was non-dimensionalized with the same expression used in the one space results (see equation 3.6). In the two space experiments there existed additional sources of loss which were not present in the one space apparatus. One of these additional sources of loss was due to the pressure drop across the orifice. A second non-dimensional loss expression was developed to study the effects of this loss.

4.5.1 TYPES OF LOSS

There were two additional sources of loss within the two space apparatus which were not present in the one space apparatus; these two sources of loss were created by the orifice plate. The first source of loss was due to the mass flow through the orifice. The mass flow across the pressure drop was an irreversible thermodynamic process and, thus, generated a loss. The other additional form of loss was due to the mixing of gases at two different temperatures. During the compression stroke, the working fluid flowed from the varying volume space to the fixed volume space. The temperature of the gas which entered the fixed volume space was at a different temperature than the gas already within that volume space, also leading to an irreversibility. The same was true during the expansion stroke when gas flowed from the fixed volume space to the varying volume space.

As in the one space experiments, the loss due to gas leakage was virtually eliminated, and there was still a loss associated with cyclic wall to gas heat transfer.

4.5.2 NON-DIMENSIONAL LOSS CALCULATION

The non-dimensional loss was calculated in a manner similar to that presented in section 3.4. However, the two volume spaces had to be accounted for. With the two volume spaces, the loss equation was represented by:

$$LOSS_{nd} = \frac{-\left(\oint_1 P dV + \oint_2 P dV\right)}{P_o V_o \left(\frac{P_*}{P_o}\right)^{\frac{\gamma-1}{\gamma}}} \quad 4.6$$

where:

1 = the fixed volume space,

2 = the varying volume space,

P_a/P_o = the approximate pressure variation given as $(P_{\max} - P_{\min})/(P_{\max} + P_{\min})$

the values for $P_{\max}$ and $P_{\min}$ were measured in the varying volume space,

the rest of the parameters were the same as in equation 3.6

Since the volume contained by the fixed volume space was constant, the first cyclic integral was zero.

4.5.3 COMPARISON TO ONE SPACE RESULTS

The one space and the two space non-dimensional loss results are shown in Figure 4.8. In most of the two space data, the non-dimensional loss decreased with increasing Pe_w , as did the one space results. However, the two space loss results were greater in magnitude, for a common Pe_w , than the one space results.

Lee's model (1983) predicted that the amount of cyclic lost work at volume ratios less than 2.0 would be very similar to the amount at a volume ratio of 2.0. This was verified with the one space experiments. Therefore, without the orifice plate, the two space results ($R_v = 1.2$) should have matched the one space results ($R_v = 1.1$). Since the results did not match, the difference had to be due to the presence of the orifice plate. The increased loss in the two space experiments was due to increased heat transfer, the pressure drop across the orifice, and to the mixing of the gas between the two volume spaces.

4.5.4 TWO SPACE LOSS RESULTS

In the two space results the non-dimensional loss was not brought together with the oscillating Peclet number (see Figure 4.8). Most of the points decreased with increasing Pe_w ; however, 6 points showed an increase in loss with increasing Pe_w . These points corresponded to all of the runs made with piston frequencies greater than or equal to 31.4 rad/sec, and were the same points that experienced an increase in the pressure phase lead (see Figure 4.6). Since the increase in the pressure phase lead was linked to the pressure drop across the orifice, it was also felt that this increase in loss was also due to

Figure 4.8:
Two and One Space Loss Results

the pressure drop. Therefore, a second loss expression was developed which accounted for the pressure drop loss.

4.5.4.1 CALCULATION OF LOSS DUE TO PRESSURE DROP

The pressure drop loss was calculated from an entropy balance across the orifice. In the calculation the density in the varying volume space was assumed to be constant, even though it was not; therefore, the loss due to the pressure drop was approximate.

The difference in entropy across the orifice was given by:

$$s_{out} - s_{in} = c_p \ln \frac{T_{out}}{T_{in}} - R \ln \frac{P_{out}}{P_{in}} \quad 4.7$$

Since flow through the orifice was an adiabatic throttling process, it could be shown that $h_{in} = h_{out}$ and, since helium was an ideal gas, $h = h(T)$. Thus, since $h_{in} = h_{out}$ and $h = h(T)$, it followed that $T_{in} = T_{out}$.

Therefore,

$$s_{out} - s_{in} = -R \ln \frac{P_{out}}{P_{in}} = R \ln \frac{P_{in}}{P_{out}} \quad 4.8$$

The entropy generated each cycle was,

$$\text{entropy transport} = s\dot{m} = S_{gen} = \oint R \ln \frac{P_{in}}{P_{out}} dm_{in} \quad 4.9$$

For flow from 1 → 2

$$S_{gen} = \int_{1/2cy} R \ln \frac{P_1}{P_2} dm_2 \quad 4.10$$

and for flow from 2 → 1

$$S_{gen} = \int_{1/2cy} R \ln \frac{P_2}{P_1} dm_1 \quad 4.11$$

By combining equations 4.10 and 4.11 and using the relationships $dm_2 = -dm_1$ and $\ln(P_1/P_2) = -\ln(P_2/P_1)$, the following expression was obtained:

$$S_{gen} = \oint R \ln \frac{P_2}{P_1} dm_1 \quad 4.12$$

The loss due to the pressure drop was:

$$LOSS = T_w S_{gen} = RT_w \oint \ln \frac{P_2}{P_1} dm_1 \quad 4.13$$

dm_1 was approximated by assuming that the density was constant and that, initially in the system, $T_{gas} = T_w$. Therefore, the mass was given by:

$$m_1 = m_{tot} \left(\frac{V_1}{V_{tot}} \right) = \frac{P_{init} V_{init}}{RT_w} \left[\frac{V_1}{V_1 + V_2} \right] \quad 4.14$$

and

$$dm_1 = \frac{P_{init} V_{init}}{RT_w} \left[\frac{V_2}{(V_1 + V_2)^2} dV_1 \right] \quad 4.15$$

By substituting equation 4.15 into 4.13 and combining terms, the loss due to the pressure drop was approximated as:

$$LOSS_{\Delta P} = V_2 V_{init} P_{init} \int \frac{\ln \frac{P_2}{P_1}}{(V_1 + V_2)^2} dV_1 \quad 4.16$$

The expression for non-dimensional loss which accounted for the pressure drop was then given by combining equations 4.16 and 4.6,

$$(LOSS - LOSS_{\Delta P})_{nd} = \frac{-\int_1 P dV - V_2 P_o V_o \int \frac{\ln \frac{P_2}{P_1}}{(V_1 + V_2)^2} dV_1}{P_o V_o \left(\frac{P_s}{P_o} \right)^2 \frac{\gamma - 1}{\gamma}} \quad 4.17$$

where:

- 1 : conditions in the varying volume space,
- 2 : conditions in the fixed volume space, and
- o : initial conditions.

4.5.4.2 APPROXIMATE PRESSURE DROP LOSS RESULTS

The approximate pressure drop loss was subtracted from the cyclic lost work (equation 4.17) and the result then plotted against Pe_w as shown in Figure 4.9. With the subtraction of the approximate loss due to the pressure drop across the orifice, all of the two space loss results were brought together with Pe_w . The loss now decreased with increasing Pe_w for all points. However, the two space loss results still remained greater in magnitude than the one space results.

While the loss decreased proportionally to $Pe_w^{-1/2}$ in the one space results, the decrease in the two space results was more gradual and was found to be proportional to $Pe_w^{-0.29}$. This meant that as the oscillating Peclet number was increased, the difference between the two sets of loss data also increased. Since the loss due to the pressure drop was accounted for, the increased loss in the two space results had to be due to the mixing loss and to increased cyclic gas to wall heat transfer.

Figure 4.9: Two Space Loss Results
Subtracting Pressure drop Loss

4.6 TEMPERATURE MEASUREMENTS

The temperature of the gas flowing through the orifice was measured with a gas thermocouple placed just below the orifice plate and approximately on the centerline of the orifice. The gas thermocouple was used to see how well the temperature of the gas which flowed through the orifice matched that of the mixed mean temperature. Since mass flowed continually from one volume space to the other, the exact temperature in either space was not known. The mixed mean temperature was developed to measure the average temperature in the apparatus as a whole. The mixed mean temperature was also used in the prediction of the heat flux using the complex Nusselt number method.

4.6.1 DERIVATION OF MIXED MEAN TEMPERATURE

Since mass was continually being transferred from one space to the other, the exact temperature could not be measured in either of the volume spaces. The mixed mean temperature was developed as a means of measuring the average temperature in the apparatus as a whole. Crucial to this calculation was the fact that, while the mass in either volume space was not known, the overall mass in the system was known (barring any leaks in the apparatus). The mixed mean temperature was formulated as the mass averaged temperature in each space. The derivation of the mixed mean temperature was as follows.

The mixed mean temperature was defined as:

$$T_m = \frac{T_1 m_1 + T_2 m_2}{m_1 + m_2} \quad 4.18$$

with

$$T_1 = \frac{P_1 V_1}{m_1 R} \quad \text{and} \quad T_2 = \frac{P_2 V_2}{m_2 R} \quad 4.19$$

The expressions in equation 4.19 were substituted into equation 4.18 which yielded:

$$T_m = \frac{P_1 V_1 + P_2 V_2}{mR} \quad 4.20$$

where:

- m : total mass in the test apparatus measured from the initial pressure, temperature, and volume within the apparatus,
- 1 : conditions within the varying volume space, and
- 2 : conditions within the fixed volume space.

4.6.2 COMPARISON OF THE TEMPERATURES

The measured temperature of the gas flowing through the orifice was compared to the mixed mean temperature, and they were found to be unequal. The gas thermocouple measurement lagged the mixed mean temperature and experienced a dip at top center. The amplitudes of the two temperatures also seemed to be different.

In Figures 4.10 and 4.11, the two temperatures were compared at intermediate pressure (5 atm). Figure 4.10 is a typical plot of the temperatures at low to intermediate piston frequencies, while Figure 4.11 is a typical plot of higher piston frequencies.

4.6.2.1 DIFFERENCE IN MAGNITUDE

As in the one space temperature measurements, the magnitude of the gas thermocouple measurement was different than the calculated temperature. However, unlike the one space experiments, no correction for the magnitude offset was introduced. In the one space experiments it was felt that the gas thermocouple measurement was an accurate approximation of the gas temperature in the entire cylinder. The magnitude offset between the measurement and calculation was then eliminated by assuming that the mean temperatures from both of these temperatures were equal. However, in the two space experiments this offset was not eliminated since it was felt that the gas thermocouple measurement was only representative of the gas flowing through the orifice, and not of the whole cylinder. The mixed mean temperature was an attempt to calculate the averaged gas temperature in the whole cylinder. There did not seem to be a way to correct for the

**Figure 4.10: Temperature Measurements
Low to Intermediate Piston Frequency**

**Figure 4.11: Temperature Measurements
High Piston Frequencies**

magnitude offset. This lack of correction probably affected the heat flux prediction in an adverse manner.

4.6.2.2 LAG OF THERMOCOUPLE MEASUREMENT

The thermocouple measurement lagged the mixed mean temperature at all pressure/piston frequency combinations. The amount of lag increased with increasing piston frequency, which indicated that the lag was due to the response time of the thermocouple. In the two plots, 4.10 and 4.11, a greater amount of lag can be seen in the plot with the higher piston frequency.

4.6.2.3 DIP IN THERMOCOUPLE MEASUREMENT

The gas thermocouple measurement experienced a drop when the piston was near top center for all of the pressure/piston frequency combinations, except for 62.8 rad/sec at 5 atm. The magnitude of the drop decreased with increasing pressure and increasing piston frequency. In Figures 4.10 and 4.11 the drop was less severe at the higher piston frequency (both plots were at similar mean pressures).

The one space gas thermocouple measurements also dipped at top center. However, in the one space experiments only a few pressure/piston frequency points were affected, while nearly all of the combinations were affected in the two space experiments. In section 3.5 a model was presented, for the one space experiments, of a boundary layer which grew on the piston face and which affected the temperature measurement when the piston was near top center, thus causing the dip in the temperature measurement. This model did not adequately predict which pressure/piston frequency points were affected in the two space experiments.

Even though the model presented in section 3.5 did not nearly predict which two space points would experience the drop in magnitude, the predicted trend that measurements at lower Pe_w would have a larger drop at top center was still true. There seemed to be two possible reasons for the increased dip in the two space temperature measurements. There could have been additional boundary layer effects from a boundary layer growing on the face of the orifice plate, as the gas thermocouple was approximately 0.19 in. below the orifice plate and approximately 0.22 in. above the piston face at top

center. Also, as the gas flowed through the orifice the boundary layer probably travelled through the orifice as well. Since the thermocouple was approximately placed on the centerline of the orifice, the boundary layer flowing through the orifice would pass over the thermocouple probe and lower the measured temperature. Both of these possibilities would occur at top center, where the dip was seen in the gas thermocouple measurement.

4.6.2.4 DIFFERENCE IN AMPLITUDE

Due to the drop in magnitude experienced by the thermocouple measurement at top center, it was difficult to make comparisons between the amplitudes of the mixed mean temperature and of the thermocouple measurement. It seemed that the amplitude of the mixed mean temperature was smaller than that of the thermocouple measurement at low piston frequencies (≤ 12.6 rad/sec), while at high piston frequencies (≥ 31.4 rad/sec) the mixed mean temperature had a larger amplitude. This was due to the amplitude of the mixed mean temperature increasing with piston frequency, while the amplitude of the thermocouple measurement decreased with increasing piston frequency. This was similar to the one space experiments, in which the thermocouple measurement decreased in amplitude with increasing piston frequency.

As in the one space experiments, heat would radiate from the thermocouple to the piston face which would lower the amplitude of the thermocouple measurement. This seemed to be more important at lower piston frequencies. At higher piston frequencies, the slow response time of the thermocouple may have resulted in a decreased amplitude measured from the thermocouple.

4.7 HEAT FLUX

The heat flux was measured with pressure and volume data and with the heat flux sensor located in the head. The method for measuring the heat flux with pressure and volume data was slightly altered from the method presented in section 3.6.1 for the one space apparatus. The method for measuring the heat flux from the heat flux sensor remained unchanged, from what was presented in section 3.6.2.

The heat flux was also predicted with the complex Nusselt number method as described in section 3.6.3. The only change in this method was the use of the mixed mean

temperature in the two space prediction, instead of the mass mean temperature.

The two space heat flux results were compared against the two space prediction of heat flux. Also, the real and imaginary parts of the complex Nusselt number, its phase and magnitude were calculated for the two space experiments. The magnitude and the phase were compared against two models of oscillating flow with inflow-produced turbulence.

The two space heat flux measured from pressure and volume data was compared against the one space measurement at points of common pressure/piston frequency. The heat flux measured from the heat flux sensor was also compared between the one and two space experiments. Nu_r , Nu_i , Nu_{mag} , and Nu_{phase} were also compared between the one and two space experiments.

A simple error analysis, located in Appendix E, indicated that the error in the heat flux measurement was approximately $\pm 5\%$.

4.7.1 MEASURED HEAT FLUX RESULTS

The heat flux was measured with two independent measurements: heat flux measured from pressure and volume data, and heat flux measured from the heat flux sensor. The two measurement methods were compared for low and high piston frequencies at low and intermediate mean pressures. Since high piston frequencies were not attainable at high mean pressures, the high mean pressure results were plotted at intermediate piston frequency. These plots are presented in Figures 4.12 through 4.16. The two space heat flux measurements from pressure and volume data were compared against the heat flux measured in the one space experiments for the same pressure/piston frequency combinations, as shown in Figures 4.17 and 4.18. The heat flux comparisons were presented for low pressure - low piston frequency and for high pressure - intermediate piston frequency combinations. At these same pressure/piston frequency combinations, the heat flux measurements from the heat flux sensor were compared, as shown in Figures 4.19 and 4.20.

4.7.1.1 TWO SPACE HEAT FLUX MEASURED FROM PRESSURE AND VOLUME DATA

The heat flux measured from the pressure and volume data essentially followed the same calculation technique as was presented in section 3.6.1. However, the heat flux from

Figure 4.12:
Two Space Heat Flux Results
Low Pressure - Low Piston Frequency

Figure 4.13:
Two Space Heat Flux Results
Low Pressure - High Piston Frequency

Figure 4.14
Two Space Heat Flux Results
Medium Pressure - Low Frequency

Figure 4.15:
Two Space Heat Flux Results
Medium Pressure - High Frequency

Figure 4.16:
Two Space Heat Flux Results
High Pressure - Medium Frequency

Figure 4.17:
 Comparison of One/Two Space Heat Flux
 Measured from Pressure/Volume
 Low Pressure - Low Piston Frequency

Figure 4.18:
 Comparison of One/Two Space Heat Flux
 Measured from Pressure/Volume
 High Pressure - Medium Frequency

Figure 4.19:
 Comparison of Heat Flux Sensor
 One/Two Space Measurements
 Low Pressure - Low Piston Frequency

Figure 4.20:
 Comparison of Heat Flux Sensor
 One/Two Space Measurements
 High Pressure - Medium Frequency

both volume spaces had to be accounted for. Once again, the first law of thermodynamics was used to measure the heat flux,

$$\delta Q = -dU - (P_1 dV_1 + P_2 dV_2) \quad 4.21$$

with

$$U - U_{ref} = mc_v(T - T_{ref}) \quad 4.22$$

and, from equation 4.20, the mixed mean temperature was defined as

$$T = \frac{P_1 V_1 + P_2 V_2}{mR} \quad 4.23$$

where:

1 : conditions in the varying volume space, and

2 : conditions in the fixed volume space.

The temperature from equation 4.23 was substituted into the expression for the internal energy and then differentiated to get:

$$dU = \frac{c_v}{R} P_1 dV_1 + \frac{c_v}{R} V_1 dP_1 + \frac{c_v}{R} P_2 dV_2 + \frac{c_v}{R} V_2 dP_2 \quad 4.24$$

Since the fixed volume section was at a constant volume, the dV_2 terms in equations 4.24 and 4.21 were zero. Equation 4.24 was then substituted into 4.21 and, after terms were combined and using the relation $(c_v/R) = 1/(\gamma - 1)$, the following expression was obtained for the two space heat transfer:

$$\delta Q = -\left(\frac{\gamma}{\gamma - 1}\right) P_1 dV_1 - \left(\frac{1}{\gamma - 1}\right) V_1 dP_1 - \left(\frac{1}{\gamma - 1}\right) V_2 dP_2 \quad 4.25$$

Then, after differentiating over time, the following expression was obtained for the heat flux. The heat flux in equation 4.26 was a space averaged in-cylinder heat flux.

$$\dot{q}'' = \frac{1}{A} \frac{\delta Q}{dt} = -\left(\frac{\gamma}{\gamma-1}\right) \frac{P_1}{A} \frac{dV_1}{dt} - \left(\frac{1}{\gamma-1}\right) \left[\frac{V_1}{A} \frac{dP_1}{dt} + \frac{V_2}{A} \frac{dP_2}{dt} \right] \quad 4.26$$

The heat flux measurement made from the heat flux sensor data remained unchanged from the method described in section 3.6.2. The method used to predict the heat flux using the complex Nusselt number was the same as was presented in section 3.6.3 except for the use of the mixed mean temperature, equation 4.20, instead of the mass mean temperature.

4.7.1.2 COMPARISON OF ONE AND TWO SPACE MEASURED HEAT FLUX

The two space heat flux measured from pressure and volume data and the heat flux measured from the heat flux sensor were compared to the one space heat flux measured at the same pressure/piston frequency points (see Figures 4.17 and 4.18, 4.19 and 4.20). In the plots, the one space results were multiplied by the ratio of the adiabatic temperature differences to eliminate any differences between the one and two space results which could have been attributed to this. The adiabatic temperature difference was given as:

$$T - T_w = T_w (R_v^{\gamma-1} - 1) \quad 4.27$$

After assuming that the wall temperature was the same in both the one and two space experiments, the following one space heat flux multiplication factor was obtained:

$$\dot{q}_1'' \frac{(T - T_w)_{R_{v2}}}{(T - T_w)_{R_{v1}}} = \dot{q}_1'' \frac{(1.2^{\frac{0.67}{2}} - 1)}{(1.1^{\frac{0.67}{2}} - 1)} = \dot{q}_1'' 1.973 \quad 4.28$$

The differences in amplitude, magnitude, and phase between the one and two space measured heat flux results were compared. The noise which affected the one space heat flux results was also studied in the two space results.

4.7.1.2.1 COMPARISONS OF MAGNITUDE, AMPLITUDE, AND PHASE

The one space heat flux measurements were multiplied by the ratio of the adiabatic temperature differences to eliminate any differences between the two data sets which resulted from different engine geometries (see Figures 4.17 and 4.18, 4.19 and 4.20). With the adjustment mentioned above, the amplitude of the two space heat flux calculated from pressure and volume data was approximately 2x that of the amplitude of the one space measurement at the high Pe_w case, while at low Pe_w the one space heat flux was very similar in magnitude to the two space. This indicated that the amount of in-cylinder heat transfer was larger in the two space experiments. Also, the one and two space heat flux measurements appeared to be in phase with each other.

The heat flux measured from the heat flux sensor in the two space apparatus was also larger in magnitude than in the one space apparatus. This indicated that the local heat flux, as well as the spatially averaged heat flux, was greater in magnitude in the two space apparatus presumably due to the turbulence generated by the flow through the orifice. It can also be seen that the heat flux measured by the sensor in the two space apparatus lags that of the one space. It was assumed that the induced turbulence caused lag in the measurement of the two space local heat flux.

4.7.1.2.2 EFFECTS OF NOISE

In the one space heat flux measurements made from pressure and volume data, noise affected the measurement. The frequency of this noise was approximately 10x the cycle frequency and, at high piston frequencies, the amplitude of the noise was much larger than that of the measured heat flux. The one and two space measurements can be compared in Figures 4.17 and 4.18 and, at higher piston frequencies, by comparing Figure 3.12 to 4.13 and 3.14 to 4.15.

The amplitude of the noise seemed to be similar in the two sets of heat flux measurements. However, due to the larger amplitude of the two space heat flux measurement, the affect of the noise was not as noticeable in the two space heat flux measurements made at high piston frequencies. However, it seemed that the noise in the one space results had more of a sinusoidal quality to it than in the two space measurements.

4.7.1.3 COMPARISON OF THE TWO SPACE HEAT FLUX MEASUREMENTS

In the two space experiments, the heat flux was measured with pressure and volume data and with the heat flux sensor as shown in Figures 4.12 to 4.16. These results were compared on the basis of phase and amplitude.

4.7.1.3.1 DIFFERENCES IN PHASE

The heat flux measured from the pressure and volume data led the heat flux measured by the heat flux sensor at all pressure/piston frequency combinations. It was shown previously that the two space heat flux measured from pressure and volume data was in phase with the one space heat flux measurements (see Figures 4.17 and 4.18), which indicated that the response of the pressure measurement did not change in the two space experiments. The two space local heat flux measurement also lagged the one space measurement (see Figures 4.19 and 4.20), even though the heat flux was measured by the same sensor in the one and two space experiments. After this phase difference was discovered, the pressure measured in the fixed volume section was compared to the measured cylinder wall temperature. Since both of these transducers were located in the head, they both should have received the information at the same time. However, it was found that the pressure measurement did, in fact, lead the temperature measurement. In the one space results, the measurements from the pressure transducer and the heat flux sensor were in phase with each other.

Since nothing about the heat flux sensor was changed between the one and two space experimental runs, the lag in the measurement of the heat flux sensor in the two space experiments must be related to the inflow-produced turbulence. Since the heat flux measured from the pressure and volume data was a spatially averaged measurement and the heat flux sensor measured a local heat flux, the phase lag between the two might indicate that the effects of turbulence were uneven over the space.

4.7.1.3.2 DIFFERENCES IN AMPLITUDE

The amplitudes between the two heat flux measurements were also different. At low piston frequencies (≤ 12.6 rad/sec), the heat flux measured from pressure and volume

data was lower than the heat flux measured from the heat flux sensor. At higher piston frequencies (≥ 31.4 rad/sec) the heat flux measured from the pressure and volume data was equal to or slightly greater than the amplitude of the heat flux measured by the sensor. These higher piston frequency points also corresponded to where the pressure drop matched the theoretical proportionality and where the uncorrected non-dimensional loss increased, instead of decreased, with increasing Pe_w . The difference in amplitude could also be the result of comparing a spatially averaged heat flux (pressure/volume calculation) to a local heat flux measurement (the heat flux sensor).

4.7.2 RESULTS OF THE PREDICTED HEAT FLUX/ Nu_c

The heat flux predicted from the complex Nusselt number method was compared to the heat flux measured from the pressure and volume data. The comparisons were made on the basis of the differences in the phase and amplitude between the two heat flux results.

The complex Nusselt number was also calculated, and Nu_r , Nu_i , and the magnitude and the phase of Nu_c were compared to the one space results. The magnitude and the phase of the complex Nusselt number were also compared to the theoretical models of Kornhauser (1993) and Gedeon (1989). Kornhauser and Gedeon both modelled the in-cylinder heat transfer with inflow produced turbulence.

4.7.2.1 COMPARISON OF PREDICTED AND MEASURED HEAT FLUX

In the one space results and in Kornhauser's previous data (1989), the heat flux predicted by the complex Nusselt number smoothed the measured heat flux and was in phase with the measured heat flux. However, the two space predicted heat flux did not follow these prior trends. For most experimental conditions, the predicted heat flux led the measured heat flux and the amplitude of the predicted heat flux was smaller than the measured heat flux. However, the predicted heat flux still resulted in a signal in which the noise was eliminated. The predicted and measured heat fluxes are shown in Figures 4.12 to 4.16. The differences in phase and amplitude between the two methods could be due to the limitations of the complex Nusselt number method and due to the use of the mixed mean temperature.

4.7.2.1.1 DIFFERENCES DUE TO METHOD

In the complex Nusselt number method, a mean Nu_c was calculated over the entire cycle and was then considered to be the Nu_c for all points within the cycle. This was a very good approximation for the one space experiments; however, due to the flow through the orifice, this approximation may not have been valid in the two space experiments where the heat flux in both of the volume spaces was different. It was conceivable that the heat and mass transfer was such that the complex Nusselt number could be different during the compression and expansion strokes. Also, previous work for oscillating flow with a sudden change in cross section (Ibrahim, et al, 1992) indicated that the instantaneous Nusselt number was much larger, compared to other points in the flow, at the point where the cross section changed. Therefore, it was also conceivable that, due to the interaction through the orifice, the Nu_c could even vary from point to point within the cycle and a mean complex Nusselt number would not adequately reflect the heat transfer within both spaces.

4.7.2.1.2 DIFFERENCES DUE TO MIXED MEAN TEMPERATURE

The mixed mean temperature used in the prediction of the heat flux was not corrected for a possible magnitude offset (see section 4.6.2.1). A difference in magnitude would account for the difference in amplitude between the predicted and measured heat flux. The mixed mean temperature also led both of the pressure measurements and the volume measurement. Both of these factors could have attributed to the difference in phase and in magnitude between the predicted and measured heat flux.

4.7.2.2 COMPLEX NUSSELT NUMBER RESULTS

The real and imaginary parts of the complex Nusselt number were calculated for the prediction of the heat flux. The Nu_r and Nu_i calculated in the two space experiments were compared to the same results from the one space experiments. The magnitude and the phase of the complex Nusselt number were also compared against the one space results and against two theoretical models.

4.7.2.2.1 THEORETICAL MODELS

Gedeon (1989) and Kornhauser (1992) both developed models of in-cylinder heat transfer with turbulence due to inflow. Kornhauser's model extended the work of Pfriem (1943), in which the cylinder space was separated into a laminar boundary layer and a fully turbulent core region, by calculating the laminar boundary layer thickness, δ , as a function of the inflowing gas velocity. Gedeon's model extended the work of Lee (1983), in which in-cylinder heat transfer was modelled for laminar flow conditions, by replacing the molecular conductivity term with a turbulent conductivity term, which Gedeon assumed to increase linearly with distance from the wall. The models of Gedeon and Kornhauser both solved for the magnitude and the phase of the complex Nusselt number.

For the same value of Re_{in}/Pe_w , the models of Gedeon and Kornhauser predicted different amounts for the phase and magnitude of the complex Nusselt number. Gedeon's model predicted higher magnitude of the Nu_c than Kornhauser's model, but also predicted a lower phase angle. However, the trends in both of the parameters were the same between the two turbulence models; the magnitude of the Nu_c increased with increasing Pe_w and the phase of the Nu_c decreased with increasing Pe_w , with the phase being nearly zero at high Pe_w (i.e. at highly turbulent conditions). The models indicated that, since the phase approached zero, there would be no phase difference between heat flux and temperature for turbulent conditions. This was different than what had been experimentally demonstrated for laminar conditions, where the heat flux led the temperature by 45° at high Pe_w .

The experimental Nu_c magnitude and phase data were also compared to a numerical model which fit the experimental data to Kornhauser's δ modification of Pfriem's model. The model accepted an a and a b to satisfy the form

$$\frac{\delta}{D_h} = a Re_{in}^b \quad 4.29$$

δ/D_h was then solved by using the minimum of equation 4.29 or 0.25 to solve for the complex Nusselt number in the form:

$$Nu_c = z \frac{[1 - e^{-2\frac{\delta}{D_h}x} + z(\frac{1}{4} - \frac{\delta}{D_h})(1 + e^{-2\frac{\delta}{D_h}x})]}{[(1 - e^{-2\frac{\delta}{D_h}x})^2 + z(\frac{1}{4} - \frac{\delta}{D_h})(1 - e^{-2\frac{\delta}{D_h}x})]} \quad 4.30$$

where $z = (1 + i) (2 Pe_\omega)^{1/2}$.

From the above expression Nu_r and Nu_i were calculated, from which the magnitude and the phase of the complex Nusselt number were found.

4.7.2.2.2 MAGNITUDE OF Nu_c

The magnitude of the complex Nusselt number was calculated by

$$Nu_{c_{mag}} = \sqrt{Nu_r^2 + Nu_i^2} \quad 4.31$$

The magnitude of the Nu_c was compared to the one space experimental results and to the theoretical models.

4.7.2.2.2.1 COMPARISON TO ONE SPACE RESULTS

The real and imaginary parts of the complex Nusselt number calculated from the two space data were both larger than the results from the one space data, as shown in Figure 4.21. The magnitude of the complex Nusselt number was plotted in Figure 4.22 and shows that the magnitude of the Nu_c was larger for the two space results than for the one space results, which indicated a higher amount of in-cylinder heat transfer in the two space experiments. In both sets of data the magnitude of the Nu_c increased linearly with increasing Pe_ω .

4.7.2.2.2.2 COMPARISON TO MODELS

The magnitude of the complex Nusselt number was compared against the models of Kornhauser/Pfriem. Gedeon's expressions were complex (with the use of Bessel functions)

Figure 4.21: Nu_r and Nu_i
 Comparison Between One and Two Space Experiments

Figure 4.22:
 Two and One Space Nu_c Magnitude

Figure 4.23: NUC Magnitude
Comparison of Data to Models

and the process to evaluate his expressions for a particular Re_{in}/Pe_w would have been too time consuming. Therefore, the experimental data was compared to the predictions of the model of Kornhauser/Pfriem at $Re_{in}/Pe_w = 56$, shown in Figure 4.23. The experimental data was shifted to the right of the Kornhauser/Pfriem expression (i.e. for the same Pe_w the experimental data was at a lower magnitude), but the slopes seemed to be in good agreement. Since Kornhauser's Nu_c magnitude expression was lower in magnitude than Gedeon's expression, Gedeon's would have been even further from the experimental data.

The two space experimental data was then fit, based on Pfriem's expressions, to an exponential fit of the form $Nu_c = aRe_{in}^b$ as shown in Figure 4.21. It was found that the two space data was best fit with a line corresponding to:

$$\frac{D_h}{\delta} = 0.033Re_{in}^{0.74} \quad 4.32$$

This expression compared favorably to Kornhauser's expression (1992), for highly turbulent conditions (large Re_{in}) which was in the form:

$$\frac{D_h}{\delta} = 0.061Re_{in}^{0.73} \quad 4.33$$

The slopes of the two equations were nearly identical (0.74 and 0.73) and the leading constants (0.033 and 0.061) were well within an order of magnitude of each other. This indicated that Kornhauser's model compared favorably to the experimental data.

4.7.2.2.3 PHASE OF Nu_c

The phase of the complex Nusselt number was calculated by:

$$Nu_{c_{phase}} = \tan^{-1} \left(\frac{Nu_i}{Nu_r} \right) \quad 4.34$$

Equation 4.31 shows that as Nu_i becomes small in comparison to Nu_r (or Nu_r becomes large in comparison to Nu_i) the phase would approach zero. Also, as the real and imaginary parts approach each other, the phase approaches 45° . Both of the turbulence models predicted

that Nu_i would approach zero and, therefore, the phase of the Nu_c would approach zero. As with the two space magnitude data, the two space phase data was compared to the theoretical models and to the one space experimental results.

4.7.2.2.3.1 COMPARISON TO ONE SPACE RESULTS

The real and imaginary portions of the complex Nusselt number, calculated for both the one and two space experimental results, show that the two values were very similar in magnitude, as seen in Figure 4.21. This indicated that in both experiments the heat flux led the temperature difference by nearly 45° . However, in the two space experiments, due to the expected turbulence, it was thought that the phase would be much lower.

It was previously determined that the noise in the volume signal affected the heat flux data. It seemed probable that this noise affected the values of Nu_r and Nu_i which would result in incorrect values for the phase; however, it was not known that if the difference between the expected phase (near 0°) and the calculated phase (near 45°) was due entirely to the noise problem.

4.7.2.2.3.2 COMPARISON TO MODELS

The phase of the complex Nusselt number calculated from the experimental data was much larger than the models of Kornhauser and Gedeon predicted. The experimental phase data also did not seem to decrease with Pe_w ; rather, the phase was nearly constant for all values of Pe_w . The experimental phase data, taken at $Re_{in}/Pe_w = 56$, was graphically compared to the Kornhauser/Pfriem model, $Re_{in}/Pe_w = 56$, as shown in Figure 4.24. The experimentally determined phase was plotted with the expression that correlated the magnitude of the complex Nusselt number (equation 4.32). As can be seen, the experimental phase data was much larger than what was predicted by Kornhauser's expression or by the experimentally determined expression. Since Kornhauser's model predicted larger phase angles than Gedeon's model did, Gedeon's model was even further from the experimental data.

The leading constant in the expression which fit the magnitude data, currently 0.061, could be changed to achieve a better fit to the phase data; however, due to the fact that the phase data was affected by the noise in the heat flux measurement this seemed to

Figure 4.24: NU_c Phase
Comparison of Data to Models

be a pointless exercise.

4.8 TWO SPACE CONCLUSIONS

At this time nothing definitive can be concluded from the results of the two space apparatus. In order to make definitive statements more tests need to be conducted over a wider range of experimental conditions (Re_{in}/Pe_w , R_v , clearance volume). The conclusions presented here for the two space apparatus are based solely on the one orifice plate used ($D_o = 1.36$ in.) and the one volume ratio ($R_v = 1.2$).

1) The pressure magnitude, pressure phase lead, corrected non-dimensional loss, and measured heat flux results all indicated that there was more in-cylinder heat transfer in the two space experiments than in the one space experiments. This seemed to be due to the turbulence generated from the mass flow through the orifice.

2) At high piston frequencies, ≥ 31.4 rad/sec, the pressure phase lead and uncorrected non-dimensional loss were definitely effected by the pressure drop across the orifice. The pressure magnitude results showed some effects of the pressure drop as well.

3) The heat flux calculated from pressure and volume data was done by lumping both of the volume spaces together and solving for a spatially averaged heat flux. This method may have given heat flux results which were not representative of the heat flux in either volume space. A numerical model (Finkbeiner and Kornhauser, 1993), not experimental data, was originally intended to calculate the heat flux in both of the volume sections. However, the model was still being developed at this time and could not be used to calculate the heat fluxes.

4.) There existed a difference in the magnitude and phase between the heat flux measured from pressure and volume data and the heat flux measured from the heat flux sensor. The phase difference was due to the measurement from the heat flux sensor lagging the measurement of the pressure and volume, presumably due to the effect of the inflow produced turbulence. These differences were not seen in the one space experiments.

5) The complex Nusselt number method did not predict the heat flux in the two space experiments as well as it did in the one space. It was unclear whether the difference between the two space predicted and measured heat flux was due to the complex Nusselt number method itself, or due to the use of the mixed mean temperature.

6) The magnitude of the complex Nusselt number matched well with the Kornhauser/Pfriem model and was lower than what Gedeon predicted, while the experimental phase data did not match either model. An exponential expression, which was similar to Kornhauser's expression for highly turbulent flow, was fit to the magnitude data; this expression did not model the phase data at all. The phase seemed to be affected by the noise in the heat flux measurement to a greater degree than the magnitude was.

CHAPTER 5

DIRECTION OF FUTURE WORK

The two space experiments, made with a single orifice at a single engine geometry, were the initial tests in our experimental study of in-cylinder heat transfer with inflow-produced turbulence. From these results, it was apparent that some improvements could be made to the apparatus to improve future results and that more data needed to be taken to fully understand the results.

5.1 RECOMMENDATIONS

The testing apparatus and the results may be improved with the following recommendations.

1) Try to reduce or eliminate the noise in the volume measurement which seemed to affect the measured and predicted heat flux. This would improve the heat flux measurements and the calculation of the real and imaginary parts of the complex Nusselt number. Currently, this noise greatly affects the phase angle determined by the complex Nusselt number.

2) Place a heat flux sensor in the varying volume section so that surface transient heat flux can be compared between the fixed and varying volume spaces. This will help future researchers to better understand the heat transfer in both spaces.

3) Place a gas thermocouple in the fixed volume space (there is currently a gas thermocouple in the varying volume space). With the second thermocouple the mixed mean temperature could be compared to the approximate gas temperature in both volume spaces. If the mixed mean temperature does not predict either measurement well, then a different method for measuring an average temperature should be used. The mixed mean temperature, equation 4.20, should be close to the temperature measured in the fixed volume section, since in the calculation the fixed volume was nearly 8x the varying volume.

4) Continue to develop the numerical model for calculating the heat flux in both of

the volume spaces. The spatially averaged heat flux presented here could then be compared to the results from the model.

5.2 DIRECTIONS FOR FUTURE WORK

There needs to be much more testing completed before any conclusions can be made about in-cylinder heat transfer with inflow and outflow.

The current experiments suggested that, after a certain piston frequency, the pressure drop across the orifice plate greatly effected the hysteresis loss and, therefore, the efficiency of the machine. However, it is not realistic to assume that for every orifice plate (resulting in a certain Re_{in}/Pe_w) or for every clearance ratio (the distance between the piston and the orifice plate at top center) that this pressure drop effect will be seen at the same piston frequency.

Data also needs to be taken with different orifice plates, at different volume ratios, and at different clearance gaps so that our understanding of in-cylinder heat transfer with generated turbulence can be better understood. Hopefully with this data, engine designers can design better engines by minimizing the affects of in-cylinder heat transfer.

The complex Nusselt number method used to predict the heat flux needs to be reevaluated for use with turbulence. This method was based on no turbulence and did not do well to predict the phase shifted heat flux with the introduction of turbulence. It might be necessary to look into calculations of the instantaneous Nusselt number to see how it varies over the cycle.

APPENDIX A

FOURIER ANALYSIS

Fourier analysis is a mathematical procedure used to eliminate high frequency noise from a measurement which is a periodic function. In Fourier analysis, the periodic function is decomposed into a number of sine and cosine components, called Fourier coefficients. The Fourier coefficients are then recombined and the result is a measurement which is a smoother form of the original measurement. The number of Fourier coefficients can range from 1 to the maximum number of data points making up the function. The lower the number of coefficients, the more high frequency noise that is eliminated from the measurement; however, a low number of coefficients may also mean that too much of the original non-noise portion of the measurement may also be lost. If the full number of coefficients are used, the Fourier analysis will result in a signal which is an exact duplicate of the original signal.

In the data reduction program the Fourier analysis was based on chapter 7 of Wylie (1960). There were 361 equally spaced data points in each cycle. The pressure, volume, gas temperature, and surface temperature data were decomposed with 20 Fourier coefficients. Fourier analysis was also used to obtain the time derivatives of those measurements. Lanczos smoothing (Hamming, 1983) was used during the recombination of the Fourier coefficients to eliminate the effects of Gibb's phenomenon.

A.1 FOURIER ANALYSIS

A periodic function can be broken into sine and cosine components. For a function of interval $2p$, the function can be expanded by

$$f(t) = \frac{1}{2}a_0 + a_1\cos\left(\frac{\pi t}{p}\right) + a_2\cos\left(\frac{2\pi t}{p}\right) + \dots + a_n\cos\left(\frac{n\pi t}{p}\right)$$

$$+ b_1 \sin\left(\frac{\pi t}{p}\right) + b_2 \sin\left(\frac{2\pi t}{p}\right) + \dots + b_n \sin\left(\frac{n\pi t}{p}\right) \quad \text{A.1}$$

where:

$2p$ = the period of the function, here $p = [t(360) - t(0)]/2$ and

n = the n^{th} Fourier coefficient, here $0 \leq n \leq 20$.

The sine and cosine coefficients (a and b) were calculated from

$$a_n = \int_a^{d+2p} f(t) \cos\left(\frac{n\pi t}{p}\right) dt$$

$$b_n = \int_a^{d+2p} f(t) \sin\left(\frac{n\pi t}{p}\right) dt \quad \text{A.2}$$

In the data reduction program, the integrals calculating the Fourier coefficients were solved with the trapezoid rule as follows:

$$a_n = \frac{\Delta t}{p} \left[\frac{1}{2} f(0) + \sum_{i=1}^{359} f(i) \cos\left(\frac{n\pi t(i)}{p}\right) + \frac{1}{2} f(360) \cos\left(\frac{n\pi t(360)}{p}\right) \right]$$

$$b_n = \frac{\Delta t}{p} \left[\sum_{i=1}^{359} f(i) \sin\left(\frac{n\pi t(i)}{p}\right) + \frac{1}{2} f(360) \sin\left(\frac{n\pi t(360)}{p}\right) \right] \quad \text{A.3}$$

where Δt was the time step between points.

Since the data points were equally spaced, Δt was a constant for each cycle. The Fourier coefficients (a_n and b_n) were then recombined in the form of equation A.1,

$$f(t)_i = \frac{a_0}{2} + \sum_{n=1}^N [a_n \cos\left(\frac{n\pi t(i)}{p}\right) + b_n \sin\left(\frac{n\pi t(i)}{p}\right)] \quad \text{A.4}$$

where $0 \leq i \leq 360$.

The Fourier analysis was also used to calculate the time derivatives of the measurements. To calculate the derivatives, equation A.4 was differentiated with respect

to time,

$$f'(t)_1 = \sum_{n=1}^{\infty} \left[\frac{n\pi}{p} \left\{ b_n \cos\left(\frac{n\pi t(i)}{p}\right) - a_n \sin\left(\frac{n\pi t(i)}{p}\right) \right\} \right] \quad \text{A.5}$$

Since $\pi/p = (2\pi)/(\text{time of the cycle}) = \omega$, equation A.5 was rewritten as:

$$f'(t)_1 = \omega \sum_{n=1}^{\infty} n \left[b_n \cos\left(\frac{n\pi t(i)}{p}\right) - a_n \sin\left(\frac{n\pi t(i)}{p}\right) \right] \quad \text{A.6}$$

A.2 GIBB'S PHENOMENON AND LANCZOS SMOOTHING

In order to do Fourier analysis, the series is always truncated. Unfortunately, truncating the series always causes oscillations in the Fourier smoothed series, these oscillations are known as Gibb's phenomenon. Lanczos found that these oscillations had the period of either the first term neglected or the last term kept. He found that these oscillations were eliminated by multiplying the Fourier coefficients with a factor of:

$$\sigma(IT, n) = \sin \frac{\left(\frac{\pi n}{IT}\right)}{\left(\frac{\pi n}{IT}\right)} \quad \text{A.7}$$

At the truncated point, $k = IT$, the Lanczos smoothing factor was zero and the effect was that the term of frequency IT had a multiplier of zero. After adding the Lanczos smoothing factor, equations A.4 and A.6 became:

$$f(t)_1 = \frac{a_0}{2} \sum_{n=1}^{\infty} \sigma(IT, n) \left[a_n \cos\left(\frac{n\pi t(i)}{p}\right) + b_n \sin\left(\frac{n\pi t(i)}{p}\right) \right] \quad \text{A.8}$$

and

$$f'(t)_1 = \omega \sum_{n=1}^{\infty} \sigma(IT, n) n \left[b_n \cos\left(\frac{n\pi t(i)}{p}\right) - a_n \sin\left(\frac{n\pi t(i)}{p}\right) \right] \quad \text{A.9}$$

APPENDIX B

DATA ANALYSIS PROGRAM

The data acquisition programming was essentially the same for the one and two space experiments. The only difference between the two programs were the volume calculations, due to the presence of the orifice plate in the two space experiments and to the different volume ratios between the experiments. All of the subroutines listed at the beginning of the program are routines provided by the board manufacturer, Analog Devices. All of the parameters passed to these routines are explained fully in the RTI-860 software manual (Analog Devices, 1990).

Since both data acquisition programs were essentially the same only the two space program was listed.

```
REM DATA ACQUISITION PROGRAM
REM TWO SPACE EXPERIMENTS
```

```
REM VOLUME CALCULATIONS MUST BE CHANGED WITHIN THE PROGRAM
FOR EVERY
REM NEW VOLUME RATIO
REM VOLUME CALCULATED IS FOR THE VARYING VOLUME SPACE
```

```
REM INITIALIZE THE PROCEDURE CALLS
```

```
DECLARE SUB INITIALIZE CDECL (SEG ERSTAT%)
DECLARE SUB AITYPE800 CDECL (BYVAL BRD%, BYVAL SCHAN%, BYVAL
NCHAN%, BYVAL MODTYPE%, SEG ERSTAT%)
DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%, BYVAL
PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG AVERAGE!, SEG
ERSTAT%)
DECLARE SUB CHECK CDECL (BYVAL LCHAN%, SEG ERSTAT%)
DECLARE SUB XCSCAN CDECL (BYVAL LCHAN%, SEG RARRAY!, SEG
ERSTAT%)
DECLARE SUB CSCAN860 CDECL (BYVAL LCHAN%, BYVAL BRD%, SEG
CHANARR%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, BYVAL
```

```

SIMUL%, SEG ERSTAT%)
REM DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%,
BYVAL PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
REM DECLARE SUB CHECK CDECL (BYVAL LCHAN%, SEG ERSTAT%)
REM DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG SCNDAVG!, SEG
ERSTAT%)
DECLARE SUB CSCAN860 CDECL (BYVAL LCHAN%, BYVAL BRD%, SEG
FINARR%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, BYVAL
SIMUL%, SEG ERSTAT%)
DECLARE SUB XCSCAN CDECL (BYVAL LCHAN%, SEG DATARRAY!, SEG
ERSTAT%)
REM DECLARE SUB CHECK CDECL (BYVAL LCHAN%, SEG ERSTAT%)
DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%, BYVAL
PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG TNOTV!, SEG
ERSTAT%)
REM DECLARE SUB CHECK CDECL (BYVAL LCHAN%, SEG ERSTAT%)
DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%, BYVAL
PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG VOLUMEINIT!, SEG
ERSTAT%)
REM DECLARE SUB CHECK CDECL (BYVAL LCHAN%, SEG ERSTAT%)
DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%, BYVAL
PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG POVOLT!, SEG
ERSTAT%)
DECLARE SUB CLCHAN CDECL (BYVAL LCHAN%, SEG ERSTAT%)
DECLARE SUB CAVG860 CDECL (BYVAL LCHAN%, BYVAL BRD%, BYVAL
PCHAN%, BYVAL RANGE%, BYVAL MULT%, BYVAL RCOUNT%, SEG
ERSTAT%)
DECLARE SUB XCAVG CDECL (BYVAL LCHAN%, SEG PBVOLT!, SEG
ERSTAT%)

REM MAKE ARRAYS DYNAMIC SO THAT THE SPACE IN MEMORY IS NOT
ALLOCATED
REM UNTIL THE ARRAY IS USED
'$DYNAMIC
DIM DATARRAY!(12800)
DIM CHANARR%(4)
DIM RARRAY!(2400)
DIM P5ARRAY!(1600)
DIM P6ARRAY!(1600)
DIM P7ARRAY!(1600)
DIM VOLARRAY!(1600)
DIM TEMPARRAY!(1600)

```

```

DIM HF9ARRAY!(1600)
DIM HF10ARRAY!(1600)
DIM SSV!(800)
DIM SSP!(800)
DIM SSBP!(800)
DIM FINARR%(3)
DEFINT A-Z
CLS

REM TRANSDUCER CHANNEL ON THE DATA ACQUISITION BOARD
REM AMPLIFIER 5 - BOARD CHANNEL 1  PCHAN 0  REFERENCE
PRESSURE TRANSDUCER
REM AMPLIFIER 7 - BOARD CHANNEL 2  PCHAN 1  PISTON PRESSURE
TRANSDUCER
REM AMPLIFIER 6 - BOARD CHANNEL 4  PCHAN 2  VOLUME PRESSURE
TRANSDUCER
REM          - BOARD CHANNEL 5  PCHAN 3  VOLUME
POTENTIOMETER
REM AMPLIFIER 8 - BOARD CHANNEL 7  PCHAN 4  THERMOCOUPLE
REM AMPLIFIER 9 - BOARD CHANNEL 8  PCHAN 5  PISTON HEAT FLUX
S/N 64
REM AMPLIFIER 10 - BOARD CHANNEL 10  PCHAN 6  VOLUME HEAT
FLUX S/N 11

REM INITIALLY CLEAR ARRAY CONTAINING FINAL BURST OF DATA
FOR N% = 0 TO 12800
    DATARRAY!(N%) = 0
NEXT N%

CALL INITIALIZE(ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "INITIALIZATION ERROR"; ERSTAT%: STOP
END IF

REM SET THE BOARD FOR +/- 5 VOLTS
BRD% = 1
SCHAN% = 0
NCHAN% = 15
MODTYPE% = 5
CALL AITYPE800(BRD%, SCHAN%, NCHAN%, MODTYPE%, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN AITYPE800"; ERSTAT%: STOP
END IF

REM CLEAR ALL CHANNELS
LCHAN% = -1
CALL CLCHAN(LCHAN%, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN CLEARING CHANNELS ="; ERSTAT%: STOP

```


END IF

REM READY AVERAGING ROUTINE FOR RPM VOLTAGE SIGNAL
REM MOTOR VOLTAGE SIGNAL PLUGGED INTO BOARD CHANNEL 20

LCHAN% = 1

PCHAN% = 13

RANGE% = 10000

MULT% = 150

RCOUNT% = 20

CALL CAVG860(LCHAN%, BRD%, PCHAN%, RANGE%, MULT%, RCOUNT%,
ERSTAT%)

IF ERSTAT% <> 0 THEN

PRINT "ERROR IN SETTING UP AVERAGING ROUTINE FOR RPM
VOLTAGE = "; ERSTAT%: STOP

END IF

REM SET UP ROUTINE TO RECORD INITIAL TEMPERATURE

LCHAN% = 100

PCHAN% = 4

RANGE% = 10000

MULT% = 150

RCOUNT% = 20

CALL CAVG860(LCHAN%, BRD%, PCHAN%, RANGE%, MULT%, RCOUNT%,
ERSTAT%)

IF ERSTAT% <> 0 THEN

PRINT "ERROR IN SETTING UP THERMOCOUPLE FOR TNOT =
"; ERSTAT%: STOP

END IF

REM SET UP ROUTINE TO FIND INITIAL VOLUME

LCHAN% = 200

PCHAN% = 3

RANGE% = 10000

MULT% = 75

RCOUNT% = 20

CALL CAVG860(LCHAN%, BRD%, PCHAN%, RANGE%, MULT%, RCOUNT%,
ERSTAT%)

IF ERSTAT% <> 0 THEN

PRINT "ERROR IN ROUTINE SETTING UP INITIAL VOLUME
="; ERSTAT%: STOP

END IF

REM SET UP AVERAGE TO TAKE AVERAGE BALANCE PRESSURE FOR P0!

LCHAN% = 75

PCHAN% = 0

RANGE% = 10000

MULT% = 30

RCOUNT% = 20

CALL CAVG860(LCHAN%, BRD%, PCHAN%, RANGE%, MULT%, RCOUNT%,

```

ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN SETTING UP ROUTINE FOR INITIAL PBAL
="; ERSTAT%: STOP
END IF

REM SET UP ROUTINE TO TAKE AVERAGE CYLINDER PRESSURE FOR P0!
LCHAN% = 50
PCHAN% = 1
RANGE% = 10000
MULT% = 30
RCOUNT% = 20
CALL CAVG860(LCHAN%, BRD%, PCHAN%, RANGE%, MULT%, RCOUNT%,
ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN SETTING UP ROUTINE FOR P0 =";
ERSTAT%: STOP
END IF
REM END OF ROUTINE SETUP

REM START OF MAIN PROGRAM
PRINT "PROGRAM BEGINS"
INPUT "ENTER ATMOSPHERIC PRESSURE (in Hg)"; PATM!
REM CONVERT ATMOSPHERIC PRESSURE TO PSIA
PATM! = PATM! * .491

REM FIND INITIAL PRESSURE FROM TRANSDUCER LOCATED IN THE
HEAD
LCHAN% = 50
CALL XCAVG(LCHAN%, POVOLT!, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN TAKING P0 DATA = "; ERSTAT%: STOP
END IF
DO WHILE ERSTAT% = 0
    CALL CHECK(LCHAN%, ERSTAT%)
LOOP
    IF ERSTAT% <> 117 THEN
        PRINT "ERROR IN P0 CONVERSION ="; ERSTAT%:
STOP
    END IF
REM CONVERT V TO PSI
PDIFFT! = 10.67297# * POVOLT!

REM HAVE TO ADD PDIFF TO PBAL
LCHAN% = 75
CALL XCAVG(LCHAN%, PBVOLT!, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN TAKING INITIAL PBAL DATA = ";
ERSTAT%: STOP

```

```

END IF
DO WHILE ERSTAT% = 0
    CALL CHECK(LCHAN%, ERSTAT%)
LOOP
    IF ERSTAT% <> 117 THEN
        PRINT "ERROR IN INITIAL PBAL CONVERSION =";
    ERSTAT%: STOP
    END IF
    REM CONVERT V TO PSI
    PBINIT! = (60.7867786# * PBVOLT! + .263678879# * PBVOLT! ^
    2) + PATM!
    REM CALCULATE INITIAL PRESSURE IN THE CYLINDER SPACE
    PNOT! = PBINIT! + PDIFFT!

    INPUT "ENTER VOLUME RATIO"; RV!
    REM GAP = DISTANCE FROM PISTON AT TDC TO O-PLATE
    INPUT "ENTER GAP DISTANCE"; GAP!

    REM FIND INITIAL TEMPERATURE
    LCHAN% = 100
    CALL XCAVG(LCHAN%, TNOTV!, ERSTAT%)
    IF ERSTAT% <> 0 THEN
        PRINT "ERROR IN FINDING AVERAGE TO VOLTAGE =";
    ERSTAT%: STOP
    END IF
    DO WHILE ERSTAT% = 0
        CALL CHECK(LCHAN%, ERSTAT%)
    LOOP
    IF ERSTAT% <> 117 THEN
        PRINT "ERROR IN CONVERTING TNOT DATA ="; ERSTAT%:
    STOP
    END IF
    REM CONVERT V TO T (F)
    GAIN! = 1608
    MV! = (TNOTV! / GAIN!) * 1000
    TOF! = 32.0137637# + 30.62481861# * (MV!) - .359504665# *
    (MV!) ^ 2

    REM FIND INITIAL VOLUME
    LCHAN% = 200
    CALL XCAVG(LCHAN%, VOLUMEINIT!, ERSTAT%)
    IF ERSTAT% <> 0 THEN
        PRINT "ERROR IN FINDING VOLTAGE FOR INITIAL VOLUME
    ="; ERSTAT%: STOP
    END IF
    DO WHILE ERSTAT% = 0
        CALL CHECK(LCHAN%, ERSTAT%)
    LOOP
    IF ERSTAT% <> 117 THEN

```

```

        PRINT "ERROR IN CONVERTING INITIAL VOLUME DATA =";
ERSTAT%: STOP
END IF
REM 800.098 IS COSTANT VOLUME ABOVE THE ORIFICE PLATE
REM 0.1836 IS 1/2 OF THE SMALL ORIFICE THRU HOLE
INITIALVOLUME! = ((2.29243# - .4027# * VOLUMEINIT!) *
((7.502 ^ 2) / 4 * 3.1416)) + 800.098 + .1836

REM WITH VOLUME,PRESSURE,AND TEMPERATURE CALCULATE INITIAL
MASS IN CYLINDER
REM R = 386.3 FTLbf/LbmR    12 IS A CONVERSION FACTOR    MASS
IN Lbm
TOR! = TOF! + 459.67
MASS! = PNOT! * INITIALVOLUME! / (TOR! * 386.3 * 12)

REM INPUT DESIRED RPM TO FIND OUT WHAT VOLTAGE TO SET THE
VARIAC TO
INPUT "ENTER DESIRED RPM'S FOR THIS RUN"; RPMWANT!
INPUT "ENTER BELT SETTING - L = LOW    H = HIGH"; BS$
IF BS$ = "L" THEN
    IF RPMWANT! > 38! THEN
        VSET! = -1.066038578# + .029925708# * RPMWANT! -
.000019803# * RPMWANT! ^ 2
    END IF
    IF RPMWANT! < 38! THEN
        VSET! = .002753403# * RPMWANT!
    END IF
ELSE
    VSET! = -.747871634# + .006342897# * RPMWANT! -
.000002132# * RPMWANT! ^ 2
END IF
PRINT "TO OBTAIN DESIRED RPM'S SET VARIAC TO "; VSET!

INPUT "HIT KEY TO CONTINUE "; WAIT$

REM READ THE VOLTAGE THAT THE VARIAC WAS SET TO AND WHAT THE
APPROXIMATE RPM
REM SHOULD BE
REM OBTAIN AVERAGE VOLTAGE COMING OUT OF BOX FOR MOTOR RPM
LCHAN% = 1
AVERAGE! = 0!
CALL XCAVG(LCHAN%, AVERAGE!, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN OBTAINING RPM AVERAGE = "; ERSTAT%:
STOP
END IF
DO WHILE ERSTAT% = 0
    CALL CHECK(LCHAN%, ERSTAT%)
LOOP

```

```

IF ERSTAT% <> 117 THEN
    PRINT "ERROR IN CONVERTING DATA ="; ERSTAT%: STOP
END IF
REM CONVERT MOTOR VOLTAGE TO RPM
PRINT "AVERAGE VOLTAGE FROM VARIAC EQUALS "; AVERAGE!
IF BS$ = "L" THEN
    IF RPMWANT! > 38! THEN
        RPM! = 35.45541 + 39.75636 * AVERAGE! - 2.50241 *
AVERAGE! ^ 2 + .663283 * AVERAGE! ^ 3
    END IF
    IF RPMWANT! < 38! THEN
        RPM! = AVERAGE! / .002753403#
    END IF
ELSE
    RPM! = 121.9876 + 187.506 * AVERAGE! + 27.19912 *
AVERAGE! ^ 2 - 70.7828 * AVERAGE! ^ 3 + 47.18236 * AVERAGE!
^ 4 - 8.38255 * AVERAGE! ^ 5
END IF
PRINT "RPM OF RUN SHOULD BE "; RPM!

REM TAKE VOLTAGE DELTA ON RTD TO TAKE CARE OF ANY INITIAL
NON EQUILIBRIUM
INPUT "ENTER VOLTAGE DIFFERENCE ON RTD TOP TO SUBTRACT FROM
ALL DATA POINTS AS AN OFFSET"; RTDV!

INPUT "TAKE PIN OUT, START MOTOR HIT KEY TO CONTINUE"; WAIT$
REM MOTOR TURNS OVER

REM SECTION OF PROGRAM TO CHECK FOR STEADY STATE
REM TAKE SCAN OF 800 POINTS, AT LEAST 2 CYCLES
REM ON THE BASIS THAT 400 POINTS PER CYCLE - ALWAYS GET FULL
CYCLE
REM COMPARE GROUPINGS OF 400 POINTS
SS% = 0
REM VREF! IS THE VOLTAGE CORRESPONDING TO THE GEOMETRIC MEAN
VOLUME
VREF! = -.218

REM SET UP CHANNEL 5,7, AND VOLUME POT S.STATE DATA HERE
SINCE RPMS HAVE BEEN FOUND
LCHAN% = 300
CHANARR%(0) = 3
CHANARR%(1) = 2
CHANARR%(2) = 3
CHANARR%(3) = 0
REM 3 CHANNELS BEING TAKEN PCHAN 0,2, AND 3
SIMUL% = 0
RANGE% = 1
RCOUNT% = 2400

```

```

RCOUNT! = RCOUNT&
RANGE! = RANGE%
REM FIGURE OUT TIME FOR 3 CYCLES
REM TIME = (MULT) (RANGE) (RCOUNT)
TMIN! = 1 / RPM!
REM TIME FOR 3 CYCLES
TTOTMIN! = 3 * TMIN!
REM TIME IN MICROSECONDS
TTOTMSEC! = TTOTMIN! * 60 * 1000000!
MULT! = TTOTMSEC! / (RCOUNT! * RANGE!)
MULT% = MULT! \ 1
REM TAKE 800 POINTS TO ENSURE 1 COMPLETE CYCLES
CALL CSCAN860(LCHAN%, BRD%, CHANARR%(0), RANGE%, MULT%,
RCOUNT&, SIMUL%, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN COLLECTING P AND V FOR S.STATE = ";
ERSTAT%: STOP
END IF

REM LOOPING TO DETERMINE WHETHER STEADY STATE WAS REACHED
PAVGCYC1! = 0!
PAVGCYC2! = 0!
TCYC1! = 0!
TCYC2! = 0!
WHILE SS% = 0

REM ZERO OUT ARRAY CONTAINING STEADY STATE DATA
Q% = 0
FOR Q% = 0 TO 2400
    RARRAY!(Q%) = 0
NEXT Q%
Q% = 0
FOR Q% = 0 TO 800
    SSP!(Q%) = 0
    SSV!(Q%) = 0
    SSBP!(Q%) = 0
NEXT Q%

REM NOW TAKE AN SCAN OF THE NEXT 800 POINTS FROM P AND V
(2400 TOTAL)
CALL XCSCAN(LCHAN%, RARRAY!(0), ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN COLLECTING STEADY STATE DATA =";
ERSTAT%: STOP
END IF
DO WHILE ERSTAT% = 0
    CALL CHECK(LCHAN%, ERSTAT%)
LOOP
IF ERSTAT% <> 117 THEN

```

```

        PRINT "ERROR IN CONVERTING STEADY STATE DATA =";
ERSTAT%: STOP
END IF

```

```

REM SEPARATE DATA FROM RARRAY INTO THREE SEPARATE ARRAYS
REM DIFFERENTIAL PRESSURE, VOLUME, BALANCE PRESSURE
REM LEAVE IN VOLTAGE VALUES SINCE CAN STILL WORK IN THOSE
UNITS

```

```
P% = 0
```

```
V% = 0
```

```
BP% = 0
```

```
FOR R% = 0 TO 2400
```

```
    IF R% MOD 3 = 0 THEN
```

```
        SSP!(P%) = RARRAY!(R%)
```

```
        P% = P% + 1
```

```
    END IF
```

```
    IF R% MOD 3 = 1 THEN
```

```
        SSV!(V%) = RARRAY!(R%)
```

```
        V% = V% + 1
```

```
    END IF
```

```
    IF R% MOD 3 = 2 THEN
```

```
        SSBP!(BP%) = RARRAY!(R%)
```

```
        BP% = BP% + 1
```

```
    END IF
```

```
NEXT R%
```

```
PRINT "HAVE SEPARATED STEADY STATE DATA INTO ARRAYS"
```

```

REM SCAN OF DATA COMPLETED NOW FIND AVERAGE PRESSURE AND
TIME PER CYCLE

```

```

REM TWO CYCLES OF DATA HOPEFULLY FOUND ANALYZE PRESSURE THEN
VOLUME

```

```
REM FIND INITIAL PRESSURE AND TIME POINTS BASED ON VREF
```

```
REM PRESSURE DATA - DATA CONTAINED IN SSP! ARRAY
```

```
REM ASSUME P AND V DATA TAKEN VIRTUALLY SIMULTANEOUSLY
```

```
REM MUST ACCESS VOLUME DATA TO SEE WHERE CYCLE STARTS
```

```
REM FIND THE TWO POINTS THAT VREF LIES BETWEEN
```

```

REM MUST SKIP OVER THE VREF THAT LIES IN THE MIDDLE OF THE
CYCLE

```

```

REM THEREFORE FIND VREF WHEN POINT v IS BELOW VREF AND V+1
IS ABOVE

```

```
V! = 0
```

```
VVALUE! = SSV!(V!)
```

```
IF VVALUE! > VREF! THEN
```

```
    WHILE VVALUE! > VREF!
```

```
        V! = V! + 1
```

```
        VVALUE! = SSV!(V!)
```

```
    WEND
```

```
    WHILE VVALUE! < VREF!
```

```

        V! = V! + 1
        VVALUE! = SSV!(V!)
    WEND
END IF
IF VVALUE! < VREF! THEN
    WHILE VVALUE! < VREF!
        V! = V! + 1
        VVALUE! = SSV!(V!)
    WEND
END IF
PRINT "FOUND FIRST CROSSOVER POINT FOR STEADY STATE"

REM LINEARLY INTERPOLATE TO SEE WHERE VREF LIES IN TIME
REM CURVE FIT STRAIGHT LINE -  $V = A0 + A1(T)$ 
REM 0 REFERS TO POINT BELOW REFERENCE, 1 IS POINT ABOVE
REFERENCE
FIRST! = V! - 1
SECOND! = V!
V0! = SSV!(FIRST!)
V1! = SSV!(SECOND!)
T0! = RANGE! * MULT! * (FIRST!) * 3
T1! = RANGE! * MULT! * (SECOND!) * 3
A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - A1! * T0!
REM TIME FOR TFIRST! TO OCCUR
TFIRST! = (VREF! - A0!) / A1!

REM FIND CROSSOVER POINT FOR END OF FIRST CYCLE BEGINNING OF
SECOND CYCLE
VVALUE! = SSV!(V!)
WHILE VVALUE! > VREF!
    V! = V! + 1
    VVALUE! = SSV!(V!)
WEND
WHILE VVALUE! <= VREF!
    V! = V! + 1
    VVALUE! = SSV!(V!)
WEND
PRINT "FOUND SECOND CROSSOVER POINT FOR STEADY STATE"
REM NOW V! SHOULD BE ABOVE VREF!
REM CURVE FIT STRAIGHT LINE -  $V = A0 + A1(T)$ 
REM 0 REFERS TO POINT BELOW REFERENCE, 1 IS POINT ABOVE
REFERENCE
THIRD! = V! - 1
FOURTH! = V!
V0! = SSV!(THIRD!)
V1! = SSV!(FOURTH!)
T0! = RANGE! * MULT! * (THIRD!) * 3
T1! = RANGE! * MULT! * (FOURTH!) * 3

```



```

A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - A1! * T0!
REM TIME FOR VREF! TO OCCUR
TSECOND! = (VREF! - A0!) / A1!

PRINT "FOUND ALL CROSSOVER POINTS"
REM HAVE ALL OF THE POINTS IN THE ARRAY WHERE THE CROSSOVERS
OCCUR
REM ALSO HAVE ALL OF THE TIMES AT CROSSOVERS
REM WORK WITH BACKING PRESSURE SO AS TO HAVE ACCURATE
DIFFERENTIAL PRESSURE VALUES
REM WORK WITH RANGE FOUND BY VOLUME INFORMATION
REM FIND THE AVERAGE BACKING PRESSURE
PBTOT! = 0
FOR Q! = FIRST! TO FOURTH!
    VOLT! = SSBP!(Q!)
    PB! = (60.7867786# * VOLT! + .263678879# * VOLT ^ 2)
+ PATM!
    PBTOT! = PBTOT! + PB!
NEXT Q!
PBAVG! = PBTOT! / (FOURTH! - FIRST! + 1)

REM CALCULATE CYLINDER PRESSURE, ADD DIFFERENTIAL TO AVERAGE
BACKING
FOR Q! = FIRST! TO FOURTH!
    VOLT! = SSP!(Q!)
    PCYLDIF! = 10.44478218# * VOLT!
    PCYL! = PCYLDIF! + PBAVG!
    SSP!(Q!) = PCYL!
NEXT Q!

REM DO LINEAR CURVE FIT TO FIND PRESSURE CROSSOVER POINTS
P0! = SSP!(FIRST!)
P1! = SSP!(SECOND!)
T0! = FIRST! * MULT! * RANGE! * 3
T1! = SECOND! * MULT! * RANGE! * 3
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
PFIRST! = (A1! * TFIRST!) + A0!
P0! = SSP!(THIRD!)
P1! = SSP!(FOURTH!)
T0! = THIRD! * MULT! * RANGE! * 3
T1! = FOURTH! * MULT! * RANGE! * 3
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
PSECOND! = (A1! * TSECOND!) + A0!

REM ALL PRESSURE CROSSOVER POINTS FOUND
REM FIRST CYCLE

```

```

REM USE TRAPEZOIDAL RULE FOR FINDING AVERAGE PRESSURE
REM TRIA REFERS TO THE TRIANGULAR SECTIONS NOT COVERED BY
RULE
REM TRIANGLES LIE ON TOP OF RECTANGLES
P2! = SSP!(SECOND!)
T2! = SECOND! * MULT! * RANGE! * 3
P3! = SSP!(THIRD!)
T3! = THIRD! * RANGE! * MULT! * 3
TRIA1! = .5 * (P2! - PFIRST!) * (T2! - TFIRST!)
AREA1! = TRIA1! + (PFIRST! * (T2! - TFIRST!))
TRIA2! = .5 * (PSECOND! - P3!) * (TSECOND! - T3!)
AREA2! = TRIA2! * (P3! * (TSECOND! - T3!))
REM MIDPOINTS REFERS TO POINTS 2 THRU N-1 IN A TRAP. RULE
MIDPOINTS! = 0
FOR Q! = (SECOND! + 1) TO (THIRD! - 1)
    MIDPOINTS! = MIDPOINTS! + (SSP!(Q!) * MULT! * RANGE!
    * 3)
NEXT Q!
TCYC1! = TSECOND! - TFIRST!
PAVGCYC1! = (AREA1! + AREA2! + MIDPOINTS! + (.5 * MULT! * 3
* RANGE! * (SSP!(SECOND!) + SSP!(THIRD!)))) / TCYC1!
PRINT "FIRST SS PRESSURE FOUND"

```

```

REM BOTH STEADY STATE PAVG AND TIME FOUND SEE IF CLOSE
ENOUGH TO STOP
REM SET LIMIT ON DIFFERENCES TO FIND STEADY STATE
REM PALLOW! IN TERMS OF VOLTAGE
PALLOW! = .02
TALLOW! = .02
PDIFF! = ABS(PAVGCYC2! - PAVGCYC1!)
TDIFF! = ABS(TCYC2! - TCYC1!)
IF (PDIFF! / PAVGCYC1! <= PALLOW!) THEN
    IF (TDIFF! / TCYC1! <= TALLOW!) THEN
REM IF THIS OCCURS SS% SHOULD BE SET TO KICK OUT OF LOOP
        SS% = 1
        PRINT "STEADY STATE REACHED"
        PRINT PAVGCYC1!, PAVGCYC2!
        PRINT TCYC1!, TCYC2!
    ELSE
        SS% = 0
        PRINT "STEADY STATE NOT REACHED"
        PAVGCYC2! = PAVGCYC1!
        TCYC2! = TCYC1!
    END IF
ELSE
    SS% = 0
    PRINT "STEADY STATE NOT REACHED"
    PAVGCYC2! = PAVGCYC1!

```

```

        TCYC2! = TCYC1!
END IF
WEND

REM NO LONGER NEED ARRAYS
ERASE SSP!
ERASE RARRAY!
ERASE SSV!
ERASE SSBP!

REM TAKE 1440 POINTS - LONGER THAN 3 CYCLES - OF DATA OF ALL
8 CHANNELS
REM TAKE FINAL BURST OF DATA
REM SET UP XCSCAN
LCHAN% = 400
FINARR%(0) = 2
RANGE% = 1
RCOUNT& = 12800
SIMUL% = 1
RCOUNT! = RCOUNT&
RANGE! = RANGE%
FINARR%(1) = 0
FINARR%(2) = 1
MULT! = (TCYC2! / 800)
MULT% = MULT! \ 1
CALL CSCAN860(LCHAN%, BRD%, FINARR%(0), RANGE%, MULT%,
RCOUNT&, SIMUL%, ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN SETTING UP FINAL SCANNING PROCEDURE
="; ERSTAT%: STOP
END IF

CALL XCSCAN(LCHAN%, DATARRAY!(0), ERSTAT%)
IF ERSTAT% <> 0 THEN
    PRINT "ERROR IN READING MULTIPLE DATAS ="; ERSTAT%:
STOP
END IF
DO WHILE ERSTAT% = 0
    CALL CHECK(LCHAN%, ERSTAT%)
LOOP
IF ERSTAT% <> 117 THEN
    PRINT "ERROR IN CONVERTING RAW COUNTS IN FINAL DATA
CHECK ="; ERSTAT%: STOP
END IF

INPUT "FINAL SCAN TAKEN, CAN TURN OFF MOTOR"; WAIT$

REM READING DATA FROM DATARRAY INTO SEPARATE ARRAYS
PRINT "OPENING FILES TO WRITE TO"

```

```

INPUT "ENTER PATH AND NAME OF GENERAL INFORMATION FILE
?????.GEN"; FILNAME$
OPEN FILNAME$ FOR OUTPUT AS #4
INPUT "ENTER PATH AND NAME OF FIRST CYCLE DATA FILE
???1.DAT"; FILENAME$
OPEN FILENAME$ FOR OUTPUT AS #1
INPUT "ENTER PATH AND NAME OF SECOND CYCLE DATA FILE
???2.DAT"; FILENAME$
OPEN FILENAME$ FOR OUTPUT AS #2
INPUT "ENTER PATH AND NAME OF THIRD CYCLE DATA FILE
???3.DAT"; FILENAME$
OPEN FILENAME$ FOR OUTPUT AS #3

```

```

REM CONVERT TRANSDUCER VOLTAGE TO FINAL MEASUREMENT AND
PLACE IN APPROPRIATE ARRAY

```

```

A% = 0
B% = 0
C% = 0
D% = 0
E% = 0
F% = 0
G% = 0
FOR N% = 0 TO 12800
    IF N% MOD 8 = 0 THEN
        VOLT! = DATARRAY!(N%)
        P! = (60.7867786# * VOLT! + .263678879# *
(VOLT! ^ 2)) + PATM!
        P5ARRAY!(A%) = P!
        A% = A% + 1
    END IF

```

```

REM DIFFERENTIAL PRESSURE TRANSDUCERS DO NOT HAVE THE
AVERAGED BACKING PRESSURE ADDED
REM TO THE MEASUREMENT, THIS WAS DONE IN THE DATA REDUCTION
PROGRAM

```

```

    IF N% MOD 8 = 1 THEN
        VOLT! = DATARRAY!(N%)
        P! = 10.67297# * (VOLT!)
        P! = P! + P5ARRAY!(B%)
        P6ARRAY!(B%) = P!
        B% = B% + 1
    END IF
    IF N% MOD 8 = 2 THEN
        VOLT! = DATARRAY!(N%)
        P! = 10.44478218# * VOLT!
        P! = P! + P5ARRAY!(C%)
        P7ARRAY!(C%) = P!
        C% = C% + 1
    END IF
    IF N% MOD 8 = 3 THEN

```

```

REM THIS IS ONLY THE VOLUME IN THE VARYING SPACE 0.1836
REPRESENTS 1/2 OF THE SMALL ORIFICE OPENING
      VOLT! = DATARRAY!(N%)
      DEPTH! = 2.29243# - (.40279# * VOLT!)
      VOLUME! = ((3.1416 * 7.502 ^ 2 * DEPTH!) /
4) + .1836
      VOLARRAY!(D%) = VOLUME!
      D% = D% + 1
    END IF
    IF N% MOD 8 = 4 THEN
      VOLT! = DATARRAY!(N%)
      GAIN! = 1608
      MV! = (VOLT! / GAIN!) * 1000
      TEMP! = 32.0137637# + 30.62481861# * MV! -
.359504665# * (MV!) ^ 2
      TEMPARRAY!(E%) = TEMP!
      E% = E% + 1
    END IF
    IF N% MOD 8 = 5 THEN
      REM ALPHA IN CELSIUS, CHANGE TEMP FROM F TO
C AND BACK
      VOLT! = DATARRAY!(N%)
      RB! = 113.68
      ALPHA! = .0023
      VEEX! = 1.21
      TOC! = (TOF! - 32) * (5 / 9)
      RT! = (RB! - (RB! * (VOLT! / VEEX! + .5))) /
(VOLT! / VEEX! + .5)
      T! = (((RT! / RB!) - 1) / ALPHA!) + TOC!
      T! = (9 / 5) * T! + 32
      HF9ARRAY!(F%) = T!
      F% = F% + 1
    END IF
    IF N% MOD 8 = 6 THEN
      VOLT! = DATARRAY!(N%)
      RB! = 98.93
      ALPHA! = .0023
      VEEX! = .266
      GAIN! = 2100
      V! = (VOLT! - RTDV!) / GAIN!
      TOC! = (TOF! - 32) * 5 / 9
      RT! = (RB! * (V! / VEEX! + .5)) / (.5 - V! /
VEEX!)
      T! = (((RT! / RB!) - 1) / ALPHA!) + TOC!
      T! = (9 / 5) * T! + 32
      HF10ARRAY!(G%) = T!
      G% = G% + 1
    END IF
  NEXT N%

```

```
REM NO LONGER NEED DATARRAY
ERASE DATARRAY!
```

```
REM ALL DATA NOW IN INDIVIDUAL ARRAYS
REM FIND OUT WHERE POINT IS FOR THE FIRST COMPLETE CYCLE
FROM THE VOLUME ARRAY
```

```
VOLREF! = ((2.29243 - (.40279 * -.218)) * 3.141592654# *  
7.502 ^ 2 / 4) + .1836
```

```
V! = 0
```

```
VVALUE! = VOLARRAY!(V!)
```

```
IF VVALUE! > VREF! THEN
```

```
    WHILE VVALUE! > VOLREF!
```

```
        V! = V! + 1
```

```
        VVALUE! = VOLARRAY!(V!)
```

```
    WEND
```

```
    WHILE VVALUE! < VOLREF!
```

```
        V! = V! + 1
```

```
        VVALUE! = VOLARRAY!(V!)
```

```
    WEND
```

```
ELSE
```

```
    WHILE VVALUE! < VOLREF!
```

```
        V! = V! + 1
```

```
        VVALUE! = VOLARRAY!(V!)
```

```
    WEND
```

```
END IF
```

```
REM ASSUME ALL CHANNELS TAKEN SIMULTANEOUSLY
```

```
FIRST! = V!
```

```
REM FIND LAST POINT FOR FIRST CYCLE, FIRST POINT FOR SECOND  
CYCLE
```

```
VVALUE! = VOLARRAY!(V!)
```

```
WHILE VVALUE! > VOLREF!
```

```
    V! = V! + 1
```

```
    VVALUE! = VOLARRAY!(V!)
```

```
WEND
```

```
WHILE VVALUE! < VOLREF!
```

```
    V! = V! + 1
```

```
    VVALUE! = VOLARRAY!(V!)
```

```
WEND
```

```
SECOND! = V! - 1
```

```
THIRD! = V!
```

```
REM FIND END OF SECOND CYCLE AND BEGINNING OF THIRD CYCLE
```

```
VVALUE! = VOLARRAY!(V!)
```

```
WHILE VVALUE! > VOLREF!
```

```
    V! = V! + 1
```

```
    VVALUE! = VOLARRAY!(V!)
```

```
WEND
```

```
WHILE VVALUE! < VOLREF!
```

```

        V! = V! + 1
        VVALUE! = VOLARRAY!(V!)
WEND
FOURTH! = V! - 1
FIFTH! = V!

REM FIND END OF THIRD CYCLE AND BEGINNING OF FOURTH
VVALUE! = VOLARRAY!(V!)
WHILE VVALUE! > VOLREF!
    V! = V! + 1
    VVALUE! = VOLARRAY!(V!)
WEND
WHILE VVALUE! < VOLREF!
    V! = V! + 1
    VVALUE! = VOLARRAY!(V!)
WEND
SIXTH! = V! - 1
SEVENTH! = V!

REM NOW PUT ALL POINTS INCLUDING TIME FOR EACH CYCLE INTO
ITS OWN DATA FILE
REM CAN RECOMBINE COMMON CYCLES LATER
REM ALSO AT THIS POINT CALCULATE AVERAGE PRESSURE FOR
CHANNEL 5, 6, AND 7
REM ASSUME ALL CHANNEL DATA START AND END AT THE SAME POINT
IN TIME

REM FIND TIME FROM VOLUME DATA
V0! = VOLARRAY!(FIRST! - 1)
V1! = VOLARRAY!(FIRST!)
T0! = MULT! * (FIRST! - 1) * RANGE! * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - (A1! * T0!)
TFIRST! = (VOLREF! - A0!) / A1!
V0! = VOLARRAY!(SECOND!)
V1! = VOLARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - (A1! * T0!)
TSECOND! = (VOLREF! - A0!) / A1!
V0! = VOLARRAY!(FOURTH!)
V1! = VOLARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - (A1! * T0!)
TTTHIRD! = (VOLREF! - A0!) / A1!

```

```

V0! = VOLARRAY!(SIXTH!)
V1! = VOLARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (V1! - V0!) / (T1! - T0!)
A0! = V0! - (A1! * T0!)
TFOURTH! = (VOLREF! - A0!) / A1!

REM FIND TIME FOR EACH OF THE 3 CYCLES
TCYC1! = TSECOND! - TFIRST!
TCYC2! = TTHIRD! - TSECOND!
TCYC3! = TFOURTH! - TTHIRD!

REM FIND ACTUAL RPM'S OF CYCLES
RPM1! = 1 / (TCYC1!) * 60 * 1000000!
RPM2! = 1 / TCYC2! * 60 * 1000000!
RPM3! = 1 / TCYC3! * 60 * 1000000!
RPM AVG! = (RPM1! + RPM2! + RPM3!) / 3

REM FIND AVERAGE BACKING PRESSURE
REM ASSUME CONSTANT ENOUGH TO USE A STRAIGHT AVERAGING
REM NEED TO FIND CROSSOVER POINTS
P0! = P5ARRAY!(FIRST! - 1)
P1! = P5ARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P5FIRST! = A1! * TFIRST! + A0!
P0! = P5ARRAY!(SECOND!)
P1! = P5ARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P5SECOND! = A1! * TSECOND! + A0!
P0! = P5ARRAY!(FOURTH!)
P1! = P5ARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P5THIRD! = A1! * TTHIRD! + A0!
P0! = P5ARRAY!(SIXTH!)
P1! = P5ARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)

```



```

P5FOURTH! = A1! * TFOURTH! + A0!

REM FIND AVERAGE BACKING PRESSURE FOR THE FIRST CYCLE
P5TOT! = 0
FOR Q! = FIRST! TO SECOND!
    P5TOT! = P5ARRAY!(Q!) + P5TOT!
NEXT Q!
P5CYC1! = (P5TOT! + P5FIRST! + P5SECOND!) / (SECOND! -
FIRST! + 1)

REM FIND AVERAGE BACKING PRESSURE FOR THE SECOND CYCLE
P5TOT! = 0
FOR Q! = THIRD! TO FOURTH!
    P5TOT! = P5TOT! + P5ARRAY!(Q!)
NEXT Q!
P5CYC2! = (P5TOT! + P5SECOND! + P5THIRD!) / (FOURTH! -
THIRD! + 1)

REM FIND THE AVERAGE BACKING PRESSURE FOR THE THIRD CYCLE
P5TOT! = 0
FOR Q! = FIFTH! TO SIXTH!
    P5TOT! = P5TOT! + P5ARRAY!(Q!)
NEXT Q!
P5CYC3! = (P5TOT! + P5THIRD! + P5FOURTH!) / (SIXTH! - FIFTH!
+ 1)

REM WORK ON P6ARRAY (DIFFERENTIAL PRESSURE), USE TRAPEZOIDAL
RULE
REM FIND CROSSOVER POINTS
P0! = P6ARRAY!(FIRST! - 1)
P1! = P6ARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P6FIRST! = A1! * TFIRST! + A0!
P0! = P6ARRAY!(SECOND!)
P1! = P6ARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P6SECOND! = A1! * TSECOND! + A0!
P0! = P6ARRAY!(FOURTH!)
P1! = P6ARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)

```

```

P6THIRD! = A1! * TTHIRD! + A0!
P0! = P6ARRAY!(SIXTH!)
P1! = P6ARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P6FOURTH! = A1! * TFOURTH! + A0!

REM CALCULATE PRESSURE (P7) CROSSOVER POINTS
P0! = P7ARRAY!(FIRST! - 1)
P1! = P7ARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P7FIRST! = A1! * TFIRST! + A0!
P0! = P7ARRAY!(SECOND!)
P1! = P7ARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P7SECOND! = A1! * TSECOND! + A0!
P0! = P7ARRAY!(FOURTH!)
P1! = P7ARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P7THIRD! = A1! * TTHIRD! + A0!
P0! = P7ARRAY!(SIXTH!)
P1! = P7ARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (P1! - P0!) / (T1! - T0!)
A0! = P0! - (A1! * T0!)
P7FOURTH! = A1! * TFOURTH! + A0!

REM WORK ON TRAPEZOIDAL RULE FOR P6 AND P7
REM AREAS REFER TO AREAS NOT COVERED BY TRAP. RULE
REM THERE ARE 6 SUCH AREAS
REM AREA61 REFERS TO CHANNEL 6, FIRST TRIANGULAR AREA +
RECTANGULAR AREA
REM T? = TIME OF THE CROSSOVER POINT
T1! = MULT! * RANGE! * FIRST! * 2
T2! = MULT! * RANGE! * SECOND! * 2
T3! = MULT! * RANGE! * THIRD! * 2
T4! = MULT! * RANGE! * FOURTH! * 2

```

```

T5! = MULT! * RANGE! * FIFTH! * 2
T6! = MULT! * RANGE! * SIXTH! * 2
AREA61! = (.5 * (T1! - TFIRST!) * (P6ARRAY!(FIRST!) -
P6FIRST!)) + (P6FIRST! * (T1! - TFIRST!))
AREA71! = (.5 * (T1! - TFIRST!) * (P7ARRAY!(FIRST!) -
P7FIRST!)) + (P7FIRST! * (T1! - TFIRST!))
AREA62! = (.5 * (TSECOND! - T2!) * (P6SECOND! -
P6ARRAY!(SECOND!)) + (P6ARRAY!(SECOND!) * (TSECOND! - T2!))
AREA72! = (.5 * (TSECOND! - T2!) * (P7SECOND! -
P7ARRAY!(SECOND!)) + (P7ARRAY!(SECOND!) * (TSECOND! - T2!))
AREA63! = (.5 * (T3! - TSECOND!) * (P6ARRAY!(THIRD!) -
P6SECOND!)) + (P6SECOND! * (T3! - TSECOND!))
AREA73! = (.5 * (T3! - TSECOND!) * (P7ARRAY!(THIRD!) -
P7SECOND!)) + (P7SECOND! * (T3! - TSECOND!))
AREA64! = (.5 * (TTHIRD! - T4!) * (P6THIRD! -
P6ARRAY!(FOURTH!)) + (P6ARRAY!(FOURTH!) * (TTHIRD! - T4!))
AREA74! = (.5 * (TTHIRD! - T4!) * (P7THIRD! -
P7ARRAY!(FOURTH!)) + (P7ARRAY!(FOURTH!) * (TTHIRD! - T4!))
AREA65! = (.5 * (T5! - TTHIRD!) * (P6ARRAY!(FIFTH!) -
P6THIRD!)) + (P6THIRD! * (T5! - TTHIRD!))
AREA75! = (.5 * (T5! - TTHIRD!) * (P7ARRAY!(FIFTH!) -
P7THIRD!)) + (P7THIRD! * (T5! - TTHIRD!))
AREA66! = (.5 * (TFOURTH! - T6!) * (P6FOURTH! -
P6ARRAY!(SIXTH!)) + (P6ARRAY!(SIXTH!) * (TFOURTH! - T6!))
AREA76! = (.5 * (TFOURTH! - T6!) * (P7FOURTH! -
P7ARRAY!(SIXTH!)) + (P6ARRAY!(SIXTH!) * (TFOURTH! - T6!))

```

REM ENDPOINTS REFER TO THE FIRST AND NTH POINTS IN TRAP.
 RULE

```

REM THERE IS ONE SET OF ENDPOINTS FOR EACH CYCLE
ENDPTS61! = .5 * (P6ARRAY!(SECOND!) + P6ARRAY!(FIRST!)) *
MULT! * RANGE! * 2
ENDPTS62! = .5 * (P6ARRAY!(FOURTH!) + P6ARRAY!(THIRD!)) *
MULT! * RANGE! * 2
ENDPTS63! = .5 * (P6ARRAY!(SIXTH!) + P6ARRAY!(FIFTH!)) *
MULT! * RANGE! * 2
ENDPTS71! = .5 * (P7ARRAY!(SECOND!) + P7ARRAY!(FIRST!)) *
MULT! * RANGE! * 2
ENDPTS72! = .5 * (P7ARRAY!(FOURTH!) + P7ARRAY!(THIRD!)) *
MULT! * RANGE! * 2
ENDPTS73! = .5 * (P7ARRAY!(SIXTH!) + P7ARRAY!(FIFTH!)) *
MULT! * RANGE! * 2

```

REM MIDPOINTS ARE THE POINTS 2 TO N-1

```

MIDPTS61! = 0
MIDPTS71! = 0
FOR COUNT! = (FIRST! + 1) TO (SECOND! - 1)
  MIDPTS61! = MIDPTS61! + (P6ARRAY!(COUNT!) * MULT! *
  RANGE! * 2)

```

```

        MIDPTS71! = MIDPTS71! + (P7ARRAY!(COUNT!) * MULT! *
RANGE! * 2)
NEXT COUNT!
MIDPTS62! = 0
MIDPTS72! = 0
FOR COUNT! = (THIRD! + 1) TO (FOURTH! - 1)
        MIDPTS62! = MIDPTS62! + (P6ARRAY!(COUNT!) * MULT! *
RANGE! * 2)
        MIDPTS72! = MIDPTS72! + (P7ARRAY!(COUNT!) * MULT! *
RANGE! * 2)
NEXT COUNT!
MIDPTS63! = 0
MIDPTS73! = 0
FOR COUNT! = (FIFTH! + 1) TO (SIXTH! - 1)
        MIDPTS63! = MIDPTS63! + (P6ARRAY!(COUNT!) * MULT! *
RANGE! * 2)
        MIDPTS73! = MIDPTS73! + (P7ARRAY!(COUNT!) * MULT! *
RANGE! * 2)
NEXT COUNT!

```

REM FIND AVERAGE PRESSURES BASED ON THE ABOVE RESULTS

```

CYCP16! = AREA61! + AREA62! + (ENDPTS61! + MIDPTS61!)
CYCP26! = AREA63! + AREA64! + (ENDPTS62! + MIDPTS62!)
CYCP36! = AREA65! + AREA66! + (ENDPTS63! + MIDPTS63!)
CYCP17! = AREA71! + AREA72! + (ENDPTS71! + MIDPTS71!)
CYCP27! = AREA73! + AREA74! + (ENDPTS72! + MIDPTS72!)
CYCP37! = AREA75! + AREA76! + (ENDPTS73! + MIDPTS73!)

```

```

PAVG16! = CYCP16! / (TSECOND! - TFIRST!)
PAVG26! = CYCP26! / (TTHIRD! - TSECOND!)
PAVG36! = CYCP36! / (TFOURTH! - TTHIRD!)
PAVG17! = CYCP17! / (TSECOND! - TFIRST!)
PAVG27! = CYCP27! / (TTHIRD! - TSECOND!)
PAVG37! = CYCP37! / (TFOURTH! - TTHIRD!)

```

REM WORK ON TEMPARRAY, HF9ARRAY, HF10ARRAY FOR CROSSOVER POINTS

```

TEMP0! = TEMPARRAY!(FIRST! - 1)
TEMP1! = TEMPARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (TEMP1! - TEMP0!) / (T1! - T0!)
A0! = TEMP0! - (A1! * T0!)
TEMPFIRST! = A1! * TFIRST! + A0!
TEMP0! = TEMPARRAY!(SECOND!)
TEMP1! = TEMPARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (TEMP1! - TEMP0!) / (T1! - T0!)

```

```

A0! = TEMP0! - (A1! * T0!)
TEMPSECOND! = A1! * TSECOND! + A0!
TEMP0! = TEMPARRAY!(FOURTH!)
TEMP1! = TEMPARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (TEMP1! - TEMP0!) / (T1! - T0!)
A0! = TEMP0! - (A1! * T0!)
TEMPTHIRD! = A1! * TTHIRD! + A0!
TEMP0! = TEMPARRAY!(SIXTH!)
TEMP1! = TEMPARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (TEMP1! - TEMP0!) / (T1! - T0!)
A0! = TEMP0! - (A1! * T0!)
TEMPFOURTH! = A1! * TFOURTH! + A0!

HF0! = HF9ARRAY!(FIRST! - 1)
HF1! = HF9ARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2
T1! = MULT! * RANGE! * FIRST! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF9FIRST! = A1! * TFIRST! + A0!
HF0! = HF9ARRAY!(SECOND!)
HF1! = HF9ARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF9SECOND! = A1! * TSECOND! + A0!
HF0! = HF9ARRAY!(FOURTH!)
HF1! = HF9ARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF9THIRD! = A1! * TTHIRD! + A0!
HF0! = HF9ARRAY!(SIXTH!)
HF1! = HF9ARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF9FOURTH! = A1! * TFOURTH! + A0!

HF0! = HF10ARRAY!(FIRST! - 1)
HF1! = HF10ARRAY!(FIRST!)
T0! = MULT! * RANGE! * (FIRST! - 1) * 2

```

```

T1! = MULT! * RANGE! * FIRST! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF10FIRST! = A1! * TFIRST! + A0!
HF0! = HF10ARRAY!(SECOND!)
HF1! = HF10ARRAY!(THIRD!)
T0! = MULT! * RANGE! * SECOND! * 2
T1! = MULT! * RANGE! * THIRD! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF10SECOND! = A1! * TSECOND! + A0!
HF0! = HF10ARRAY!(FOURTH!)
HF1! = HF10ARRAY!(FIFTH!)
T0! = MULT! * RANGE! * FOURTH! * 2
T1! = MULT! * RANGE! * FIFTH! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF10THIRD! = A1! * TTHIRD! + A0!
HF0! = HF10ARRAY!(SIXTH!)
HF1! = HF10ARRAY!(SEVENTH!)
T0! = MULT! * RANGE! * SIXTH! * 2
T1! = MULT! * RANGE! * SEVENTH! * 2
A1! = (HF1! - HF0!) / (T1! - T0!)
A0! = HF0! - (A1! * T0!)
HF10FOURTH! = A1! * TFOURTH! + A0!

REM WRITE AVERAGES AND TIME TO SCREEN AND FILES
REM WRITE ALL DATA TO FILES

PRINT FIRST!
PRINT THIRD!
PRINT FIFTH!
PRINT SEVENTH!
INPUT "HIT ANY KEY TO CONTINUE"; ANS$

WRITE #4, "RUN", FILNAME$
WRITE #4, "TAMB (F)", TOF!
PBALAVG! = (P5CYC1! + P5CYC2! + P5CYC3!) / 3
PTOPAVG! = (PAVG36! + PAVG26! + PAVG16!) / 3
PBOTAVG! = (PAVG17! + PAVG27! + PAVG37!) / 3
TIMEAVG! = (TCYC3! + TCYC2! + TCYC1!) / 3
WRITE #4, "PBAL (lbf/in^2)", PBALAVG!
WRITE #4, "PTOP (lbf/in^2)", PTOPAVG!
WRITE #4, "PBOT (lbf/in^2)", PBOTAVG!
WRITE #4, "TIME", TIMEAVG!
WRITE #4, "CLEARANCE (in)", GAP!
WRITE #4, "Rv", RV!
WRITE #4, "OMEGA", (RPMAVG! / 60) * 2 * 3.1416
WRITE #4, "Pnot (lbf/in^2)", PNOT!

```

```

WRITE #4, "Vnot (in^3)", INITIALVOLUME!
WRITE #4, "MASS", MASS!
WRITE #4, "ORIF.DIAM", 1.3675
ONE$ = "PBAL"
TWO$ = "PTOP"
THREE$ = "PBOT"
FOUR$ = "TEMP"
FIVE$ = "RTDBOT"
SIX$ = "RTDTOP"
SEVEN$ = "VVAR"
NINE$ = "VCONST"
EIGHT$ = "TIME"
WRITE #4, "THIS IS THE FORMAT OF THE DATA"
WRITE #4, ONE$, TWO$, THREE$, FOUR$, FIVE$, SIX$, SEVEN$,
NINE$, EIGHT$
WRITE #1, P5FIRST!, P6FIRST!, P7FIRST!, TEMPFIRST!,
HF9FIRST!, HF10FIRST!, VOLREF!, 800.098, TFIRST!
FOR X! = FIRST! TO SECOND!
    TIME! = MULT! * RANGE! * X! * 2
    WRITE #1, P5ARRAY!(X!), P6ARRAY!(X!), P7ARRAY!(X!),
TEMPARRAY!(X!), HF9ARRAY!(X!), HF10ARRAY!(X!),
VOLARRAY!(X!), 800.098, TIME!
NEXT X!
WRITE #1, P5SECOND!, P6SECOND!, P7SECOND!, TEMPSECOND!,
HF9SECOND!, HF10SECOND!, VOLREF!, 800.098, TSECOND!
WRITE #2, P5SECOND!, P6SECOND!, P7SECOND!, TEMPSECOND!,
HF9SECOND!, HF10SECOND!, VOLREF!, 800.098, TSECOND!
FOR X! = THIRD! TO FOURTH!
    TIME! = MULT! * RANGE! * X! * 2
    WRITE #2, P5ARRAY!(X!), P6ARRAY!(X!), P7ARRAY!(X!),
TEMPARRAY!(X!), HF9ARRAY!(X!), HF10ARRAY!(X!),
VOLARRAY!(X!), 800.098, TIME!
NEXT X!
WRITE #2, P5THIRD!, P6THIRD!, P7THIRD!, TEMPTHIRD!,
HF9THIRD!, HF10THIRD!, VOLREF!, 800.098, TTHIRD!
WRITE #3, P5THIRD!, P6THIRD!, P7THIRD!, TEMPTHIRD!,
HF9THIRD!, HF10THIRD!, VOLREF!, 800.098, TTHIRD!
FOR X! = FIFTH! TO SIXTH!
    TIME! = MULT! * RANGE! * X! * 2
    WRITE #3, P5ARRAY!(X!), P6ARRAY!(X!), P7ARRAY!(X!),
TEMPARRAY!(X!), HF9ARRAY!(X!), HF10ARRAY!(X!),
VOLARRAY!(X!), 800.098, TIME!
NEXT X!
WRITE #3, P5FOURTH!, P6FOURTH!, P7FOURTH!, TEMPFOURTH!,
HF9FOURTH!, HF10FOURTH!, VOLREF!, 800.098, TFOURTH!

INPUT "END OF PROGRAM REACHED, HIT KEY TO CONTINUE"; ANS$
REM NO LONGER NEED ANY OF THE ARRAYS
ERASE P5ARRAY!

```

```
ERASE P6ARRAY!  
ERASE P7ARRAY!  
ERASE TEMPARRAY!  
ERASE VOLARRAY!  
ERASE HF9ARRAY!  
ERASE HF10ARRAY!  
END
```


APPENDIX C

DATA REDUCTION PROGRAMS

The data reduction took the raw data from the data acquisition program and calculated all of the heat transfer related measurements presented in chapters 3 and 4. The raw data was smoothed with Fourier analysis prior to any of the calculations.

The data reduction program for the one and two space experiments were presented in the appendix since there were some significant differences between the two, most notably in the measurement of heat flux from pressure and volume data. There were also additions to the two space data reduction program which were not in the one space program.

C.1 ONE SPACE DATA REDUCTION PROGRAM

REM SINGLE SPACE DATA REDUCTION PROGRAM

```
REM DECLARE SUBROUTINES
DECLARE SUB EQSPC (DELTA!, X!(), Y!(), MAX%)
DECLARE SUB FOCOEFF (X!(), A!(), B!(), P!, DELTA!, Y!(),
IT%)
DECLARE SUB FORECOMB (X!(), A!(), B!(), P!, Y!(), IT%, Z!())
DECLARE SUB MINMAX (Y!(), MAX%, MIN%)
DECLARE SUB MEAN (Y!(), X!(), DELTA!, MN!)
DECLARE SUB FODER (A!(), B!(), IT%, Z!(), RPMHZ!, PI!, Y!(),
P!)
```

CLS

```
INPUT "ENTER FILE NAME AND PATH FOR FILE TO BE LOOKED AT";
FILENAME$
OPEN FILENAME$ FOR INPUT AS #1
```

```
REM INPUT HOW MANY DATA POINTS WERE IN THE CYCLE
INPUT "ENTER NUMBER OF POINTS TAKEN PER CHANNEL ", X%
```

```
REM DECLARE ARRAYS DYNAMIC SO THAT THE MEMORY WILL ONLY BE
ALLOCATED WHEN
```

```

REM THE ARRAY IS CALLED
'$DYNAMIC
DIM PBAL! (X%)
DIM PTOP! (X%)
DIM PBOT! (X%)
DIM TEMP! (X%)
DIM RTDBOT! (X%)
DIM RTDTOP! (X%)
DIM VOL! (X%)
DIM TIME! (X%)
DIM PTOP2! (360)
DIM PBOT2! (360)
DIM TIME2! (360)
DIM PBAL2! (360)
DIM TEMP2! (360)
DIM RTD2TOP! (360)
DIM VOL2! (360)
DIM A! (100)
DIM B! (100)
DIM PTOPAD! (360)
DIM PTOPIISO! (360)
DIM PTOP2RAW! (360)
DIM PTOPD! (360)
DIM VOLD! (360)
DIM TEMPD! (360)
DIM ASUR! (360)
DIM RTDDTOP! (360)
DIM RTDA! (100)
DIM RTDB! (100)
DIM HFRTD! (360)
DIM TBAND! (360)
DIM T! (360)
DIM AQ! (100)
DIM BQ! (100)
DIM QF! (360)
DIM QFPRIME! (360)
DIM PBOT2RAW! (360)
DIM PBOTD! (360)
DIM TEQSTOP! (360)
DIM TEQSBOT! (360)
DIM TEQS2TOP! (360)
DIM TEQS2BOT! (360)
DIM HFTOP! (360)
DIM HFBOT! (360)
DIM DELTMPTOP! (360)
DIM DELTMPBOT! (360)
DIM DEL2TMPTOP! (360)
DIM DEL2TMPBOT! (360)
DIM DDELTMPPTOP! (360)

```

```

DIM DDELTMPBOT!(360)
DIM HFNUTOP!(360)
DIM HFNUBOT!(360)
DIM DP!(360)
DIM DV!(360)

```

```

PI! = 3.141592654#

```

```

REM THE FOLLOWING INPUTS COME FROM THE .GEN FILE CREATED IN
THE DATA ACQUISITION PROGRAM
INPUT "ENTER VOLUME RATIO"; RV!
INPUT "ENTER RPM OF RUN"; RPM!
REM CONVERT RPM TO HZ
RPMHZ! = RPM! / 60
REM CONVERT TO OMEGA (1/S)
OMEGA! = RPMHZ! * 2 * PI!

```

```

REM CYLINDER ID IN INCHES
CYLD! = 7.502
REM STROKE LENGTH IN INCHES
LS! = 3.75
REM GAS PROPERTIES
INPUT "ENTER GAS USED (HE, AR)"; GAS$
IF GAS$ = "HE" THEN
    GAMMA! = 1.667
    REM R IN FT LBF/LBM R
    R! = 383.3
    REM CP! IN BTU/LBM F
    CP! = 1.24

```

```

END IF
INPUT "ENTER MASS (LBM)"; MASS!
INPUT "ENTER PNOT (PSIA)"; PNOT!
INPUT "ENTER VNOT (IN^3)"; VNOT!

```

```

REM BRING IN ALL INFORMATION FROM SPREADSHEET FILE INTO
SEPARATE ARRAYS
Q% = 0
DO UNTIL EOF(1)
    INPUT #1, PBAL!(Q%), PTOP!(Q%), PBOT!(Q%),
    TEMP!(Q%), RTDBOT!(Q%), RTDTOP!(Q%), VOL!(Q%), TIME!(Q%)
    Q% = Q% + 1
LOOP
CLOSE #1
MAX% = X% - 1

```

```

REM ALL DATA READ TO ARRAYS
REM CALL ROUTINE FOR EVENLY SPACED POINTS (FOR FOURIER

```

```

TRANSFORMS)
NOPTS! = 360
DELTA! = (TIME!(MAX%) - TIME!(0)) / NOPTS!
CALL EQSPC(DELTA!, PTOP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, PBAL!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, PBOT!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, TEMP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, RTDTOP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, VOL!(), TIME!(), MAX%)
REM ALL DATA POINTS HAVE BEEN CHANGED, TIME REMAINS
UNCHANGED
NEWTIME! = TIME!(0) + DELTA!
I% = 1
TIME2!(0) = TIME!(0)
DO WHILE I% <= 359
    TIME2!(I%) = NEWTIME!
    I% = I% + 1
    NEWTIME! = TIME!(0) + DELTA! * I%
LOOP
TIME2!(360) = TIME!(MAX%)

REM IN THE EQUAL SPACING SUBROUTINE THE 400 DATA POINTS WERE
CHANGED TO 361
REM EQUALLY SPACED POINTS; HOWEVER, THE ORIGINAL ARRAY NOW
CONTAINS THE 361
REM DATA POINTS PLUS THE ORIGINAL DATA POINTS FROM 361 TO
EOF
REM WANT TO TAKE THE 361 POINTS AND JUST PLACE THEM IN A NEW
ARRAY
I% = 0
DO WHILE I% <= (360)
    PTOP2!(I%) = PTOP!(I%)
    PBAL2!(I%) = PBAL!(I%)
    PBOT2!(I%) = PBOT!(I%)
REM TEMP IN RANKINE
    TEMP2!(I%) = TEMP!(I%) + 459.67
    RTD2TOP!(I%) = RTDTOP!(I%) + 459.67
    VOL2!(I%) = VOL!(I%)
    I% = I% + 1
LOOP

REM SUBTRACT OUT THE BACKING PRESSURE FROM THE PRESSURE
MEASUREMENTS, SO THAT THE
REM AVERAGE BACKING PRESSURE CAN BE ADDED TO ALL OF THE
PRESSURE POINTS
FOR Q% = 0 TO 360
    PTOP2RAW!(Q%) = PTOP2!(Q%) - PBAL2!(Q%)
    PBOT2RAW!(Q%) = PBOT2!(Q%) - PBAL2!(Q%)
NEXT Q%

```

```

REM FIND AVERAGE BALANCE PRESSURE
SUM% = 0
FOR Q% = 0 TO 360
    SUM! = SUM! + PBAL2!(Q%)
NEXT Q%
PBAVG! = SUM! / 361
FOR Q% = 0 TO 360
    PTOP2!(Q%) = PTOP2RAW!(Q%) + PBAVG!
    PBOT2!(Q%) = PBOT2RAW!(Q%) + PBAVG!
NEXT Q%

REM ERASE ARRAYS THAT ARE NO LONGER NEEDED TO FREE UP MEMORY
ERASE PTOP2RAW!
ERASE PBOT2RAW!
ERASE PTOP!
ERASE TIME!
ERASE PBAL!
ERASE TEMP!
ERASE RTDTOP!
ERASE VOL!
ERASE PBOT!

REM SEND EQUALIZED SPACE DATA TO SUBROUTINE TO CALCULATE
FOURIER COEFF.
REM FOLLOW THAT WITH ROUTINE TO RECOMBINE FOURIER
COEFFICIENTS TO SMOOTH DATA

P! = (TIME2!(360) - TIME2!(0)) / 2

REM USE 20 COEFFICIENTS IN FOURIER ANALYSIS
IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOP2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, PTOP2!(), IT%,
PTOPD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PBOT2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, PBOT2!(), IT%,
PBOTD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, TEMP2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, TEMP2!(), IT%,
TEMPD!())

IT% = 20

```

```

CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, RTD2TOP!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, RTD2TOP!(), IT%,
RTDDTOP!())
REM CHANGE NAMES OF ALPHA AND BETA FOR SURFACE HF CALC.
FOR Q% = 0 TO IT%
    RTDA!(Q%) = A!(Q%)
    RTDB!(Q%) = B!(Q%)
NEXT Q%

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, VOL2!(), IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, VOL2!(), IT%,
VOLD!())

REM START TO CALCULATE NECESSARY FUNCTIONS

REM CALCULATE MINIMUM AND MAXIMUM PRESSURE FROM BOTH
PRESSURE TRANSDUCERS
MIN% = 0
MAX% = 0
CALL MINMAX(PTOP2!(), MAX%, MIN%)
PTOP2MAX! = PTOP2!(MAX%)
PTOP2MIN! = PTOP2!(MIN%)

MIN% = 0
MAX% = 0
CALL MINMAX(PBOT2!(), MAX%, MIN%)
PBOT2MAX! = PBOT2!(MAX%)
PBOT2MIN! = PBOT2!(MIN%)

REM FIND MAXIMUM AND MINIMUM VOLUME
MIN% = 0
MAX% = 0
CALL MINMAX(VOL2!(), MAX%, MIN%)
VOLMAX! = VOL2!(MAX%)
VOLMIN! = VOL2!(MIN%)
PRINT "MINIMUM VOLUME IS "; VOLMIN!
PRINT "MAXIMUM VOLUME IS "; VOLMAX!

REM CALCULATE PMEAN (PAVG) FOR BOTH PRESSURE TRANSDUCER
MEASUREMENTS
MN! = 0
CALL MEAN(PTOP2!(), TIME2!(), DELTA!, MN!)
PTOPMEAN! = MN!
PRINT "PTOP(MEAN) ="; PTOPMEAN!

MN! = 0

```

```

CALL MEAN(PBOT2!(), TIME2!(), DELTA!, MN!)
PBOTMEAN! = MN!
PRINT "PBOT(MEAN) ="; PBOTMEAN!

```

```

REM FIND MEAN VOLUME
MN! = 0
CALL MEAN(VOL2!(), TIME2!(), DELTA!, MN!)
VOLMEAN! = MN!

```

```

REM CALCULATE SURFACE AREA FOR EACH TIME POSITION
FOR Q% = 0 TO 360
    ASUR!(Q%) = 4 * VOL2!(Q%) / CYLD! + PI! * CYLD! ^ 2
/ 2
NEXT Q%

```

```

REM FIND MEAN SURFACE AREA FOR HYDRAULIC DIAMETER
MN! = 0
CALL MEAN(ASUR!(), TIME2!(), DELTA!, MN!)
ASURFMN! = MN!

```

```

REM CALCULATE MEAN TEMPERATURE FROM GAS T/C
MN! = 0
CALL MEAN(TEMP2!(), TIME2!(), DELTA!, MN!)
TCMN! = MN!

```

```

REM CALCULATE MEAN MASS
MASSMN! = PTOPMEAN! * VOLMEAN! / (12 * TCMN! * R!)
MASSMN! = PBOTMEAN! * VOLMEAN! / (12 * TCMN! * R!)

```

```

REM CALCULATE MASS MEAN TEMPERATURE FROM IDEAL GAS LAW
REM MASS MEAN TEMPERATURE WAS CALCULATED FOR BOTH PRESSURE
MEASUREMENTS
REM TO SEE IF THERE WERE ANY DIFFERENCES
FOR Q% = 0 TO 360
    TEQSTOP!(Q%) = PTOP2!(Q%) * VOL2!(Q%) / (MASS! * R!
* 12)
    TEQSBOT!(Q%) = PBOT2!(Q%) * VOL2!(Q%) / (MASS! * R!
* 12)
NEXT Q%

```

```

REM CALCULATE MODIFIED MASS MEAN TEMPERATURE
REM MODIFIED MASS MEAN TEMPERATURE WAS CALCULATED FOR BOTH
PRESSURE MEASUREMENTS
REM CALCULATE TEMP FROM EQ OF STATE, COMPARE TO GAS T/C
SUMTC! = 0
SUMIGLTOP! = 0
SUMIGLBOT! = 0

```

```

FOR Q% = 0 TO 360
    SUMTC! = SUMTC! + TEMP2!(Q%)
    SUMIGLTOP! = SUMIGLTOP! + TEQSTOP!(Q%)
    SUMIGLBOT! = SUMIGLBOT! + TEQSBOT!(Q%)
NEXT Q%

FOR Q% = 0 TO 360
    TEQS2TOP!(Q%) = TEQSTOP!(Q%) + (SUMTC! / 361 -
SUMIGLTOP! / 361)
    TEQS2BOT!(Q%) = TEQSBOT!(Q%) + (SUMTC! / 361 -
SUMIGLBOT! / 361)
NEXT Q%

REM CALCULATE MEAN MODIFIED MASS MEAN TEMPERATURE, IN R
MN! = 0
CALL MEAN(TEQS2TOP!(), TIME2!(), DELTA!, MN!)
TEMPMNRTOP! = MN!
MN! = 0
CALL MEAN(TEQS2BOT!(), TIME2!(), DELTA!, MN!)
TEMPMNRBOT! = MN!

REM EXPRESS TEMPMEAN IN KELVINS FOR K! CALCULATION OF
THERMAL CONDUCTIVITY
TEMPMNKTOP! = TEMPMNRTOP! / 1.8
TEMPMNKBOT! = TEMPMNRBOT! / 1.8

REM CALCULATE THERMAL CONDUCTIVITY IN BTU/S FT F BASED ON
TEMPMEAN
REM CALCULATION COMES FROM MILLER, SHAW, YOUNG "PROPERTIES
OF GASES & LIQUIDS"
REM THERMAL CONDUCTIVITY CALCULATED FROM BOTH OF THE
MODIFIED MASS MEAN TEMPS
KTOP! = (.5778 / 3600) * (.03722 + .0003896 * TEMPMNKTOP! -
7.45E-08 * TEMPMNKTOP! ^ 2 + 1.29E-11 * TEMPMNKTOP! ^ 3)
KBOT! = (.5778 / 3600) * (.03722 + .0003896 * TEMPMNKBOT! -
7.45E-08 * TEMPMNKBOT! ^ 2 + 1.29E-11 * TEMPMNKBOT! ^ 3)

REM CALCULATE DENSITY FROM MEAN PROPERTIES
REM TWO DENSITIES CALCULATED BASED ON BOTH PRESSURES AND
TEMPERATURES
RHOTOP! = PTOPMEAN! / (R! * TEMPMNRTOP!)
RHOBOT! = PBOTMEAN! / (R! * TEMPMNRBOT!)

REM CALCULATE HYDRAULIC DIAMETER (IN) FOR PEW
DH! = (4 * VOLMEAN! / ASURFMN!)

REM CALCULATE Pew, ONCE AGAIN CALCULATE TWO SEPARATE VALUES
PEW2TOP! = (PTOPMEAN! * CP! * OMEGA! * DH! ^ 2) / (KTOP! *

```



```

R! * 4 * TEMPMNRTOP!)
PEW2BOT! = (PBOTMEAN! * CP! * OMEGA! * DH! ^ 2) / (KBOT! *
R! * 4 * TEMPMNRBOT!)
PRINT "PECLET IN TOP SECTION F(DH,OMEGA) = "; PEW2TOP!
PRINT "PECLET IN BOTTOM SECTION F(DH,OMEGA) = "; PEW2BOT!
PRINT "OMEGA"; OMEGA!

REM CALCULATE LOSS (INTEGRAL OF PdV)
LOSSTOP! = 0
LOSSBOT! = 0
FOR I% = 1 TO 360
    LOSSTOP! = LOSSTOP! + (((PTOP2!(I%) + PTOP2!(I% -
1)) / 2) * (VOL2!(I%) - VOL2!(I% - 1)))
    LOSSBOT! = LOSSBOT! + (((PBOT2!(I%) + PBOT2!(I% -
1)) / 2) * (VOL2!(I%) - VOL2!(I% - 1)))
NEXT I%

REM CALC. Pa/Po FROM APPROXIMATION 3.16 IN KORN.
DISSERTATION
PVARTOP! = (PTOP2MAX! - PTOP2MIN!) / (PTOP2MAX! + PTOP2MIN!)
PVARBOT! = (PBOT2MAX! - PBOT2MIN!) / (PBOT2MAX! + PBOT2MIN!)

REM CALCULATE LOSSnd EQ. 3.25 KORN. DISSERTATION
LOSSNDTOP! = LOSSTOP! / (PTOPMEAN! * VOLMEAN! * (PVARTOP!) ^
2 * ((GAMMA! - 1) / GAMMA!))
LOSSNDBOT! = LOSSBOT! / (PBOTMEAN! * VOLMEAN! * (PVARBOT!) ^
2 * ((GAMMA! - 1) / GAMMA!))
PRINT "LOSSnd BASED ON PTOP(EQ. 3.16) = "; LOSSNDTOP!
PRINT "LOSSnd BASED ON PBOT(EQ. 3.16) = "; LOSSNDBOT!

REM CALCULATE ADIABATIC PRESSURE FOR p* AND Plead
FOR N% = 0 TO 360
    PTOPAD!(N%) = PNOT! * (VNOT! / VOL2!(N%)) ^ GAMMA!
NEXT N%

REM CALCULATE ISOTHERMAL PRESSURE
FOR N% = 0 TO 360
    PTOPIISO!(N%) = PNOT! * VNOT! / VOL2!(N%)
NEXT N%

REM CALCULATE MINIMUM AND MAXIMUM ADIABATIC AND ISOTHERMAL
PRESSURES
MIN% = 0
MAX% = 0
CALL MINMAX(PTOPAD!(), MAX%, MIN%)
PTOPADMAX! = PTOPAD!(MAX%)
PTOPADMIN! = PTOPAD!(MIN%)

MIN% = 0

```

```

MAX% = 0
CALL MINMAX(PTOPISO!(), MAX%, MIN%)
PTOPISOMAX! = PTOPIISO!(MAX%)
PTOPISOMIN! = PTOPIISO!(MIN%)

REM FIND MAGNITUDE AND PHASE SHIFT
REM CALCULATE FROM FIRST HARMONIC OF PRESSURE WAVES
IT% = 1
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOPAD!(),
IT%)
ALPHAAD! = A!(1)
BETAAD! = B!(1)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOPIISO!(),
IT%)
ALPHAISO! = A!(1)
BETAISO! = B!(1)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOPI2!(),
IT%)
ALPHATOP! = A!(1)
BETATOP! = B!(1)
MAGAD! = (ALPHAAD! ^ 2 + BETAAD! ^ 2) ^ (1 / 2)
MAGISO! = (ALPHAISO! ^ 2 + BETAISO! ^ 2) ^ (1 / 2)
MAGTOP! = (ALPHATOP! ^ 2 + BETATOP! ^ 2) ^ (1 / 2)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PBOT2!(),
IT%)
ALPHABOT! = A!(1)
BETABOT! = B!(1)
MAGBOT! = (ALPHABOT! ^ 2 + BETABOT! ^ 2) ^ (1 / 2)

PSTARTOP! = (MAGTOP! - MAGISO!) / (MAGAD! - MAGISO!)
PRINT "P* BASED ON PTOPI(PRESSURE RATIO) = "; PSTARTOP!
PSTARBOT! = (MAGBOT! - MAGISO!) / (MAGAD! - MAGISO!)
PRINT "P* BASED ON PBOT(PRESSURE RATIO) = "; PSTARBOT!

ANG! = 360 / (2 * PI!)
PHASETOP! = ATN(ALPHATOP! / BETATOP!) * ANG!
PHASEBOT! = ATN(ALPHABOT! / BETABOT!) * ANG!
PHASEAD! = ATN(ALPHAAD! / BETAAD!) * ANG!
PHASEISO! = ATN(ALPHAISO! / BETAISO!) * ANG!
LEAD1 = PHASETOP! - PHASEAD!
LEAD2! = PHASETOP! - PHASEISO!
PRINT "PHASE ANGLE BASED ON PTOPI(BASED ON AD.) = "; LEAD1!
PRINT "PHASE ANGLE BASED ON PTOPI(BASED ON ISO.) = "; LEAD2!
LEAD1 = PHASEBOT! - PHASEAD!
LEAD2! = PHASEBOT! - PHASEISO!
PRINT "PHASE ANGLE BASED ON PBOT(BASED ON AD.) = "; LEAD1!
PRINT "PHASE ANGLE BASED ON PBOT(BASED ON ISO.) = "; LEAD2!

```

```

REM CALCULATE HEAT TRANSFER RESULTS
REM HF BASED ON DP/Dt, DV/Dt

REM MEASURE HEAT FLUX FROM PRESSURE AND VOLUME DATA
REM HF PER UNIT AREA BASED ON PTOP
FOR Q% = 0 TO 360
    ONE! = -VOL2!(Q%) * (1 / (GAMMA! - 1)) * PTOPD!(Q%)
    TWO! = -PTOP2!(Q%) * (GAMMA! / (GAMMA! - 1)) *
VOLD!(Q%)
    HFTOP!(Q%) = ONE! + TWO!
REM CHANGE UNITS FROM LBF/(IN*MICROSEC) TO BTU/(H*FT^2)
    HFTOP!(Q%) = HFTOP!(Q%) / ASUR!(Q%) * 12 * 1000000!
    * 3600 / 778.16
NEXT Q%

REM HF PER UNIT AREA BASED ON PBOT
FOR Q% = 0 TO 360
    ONE! = -VOL2!(Q%) * (1 / (GAMMA! - 1)) * PBOTD!(Q%)
    TWO! = -PBOT2!(Q%) * (GAMMA! / (GAMMA! - 1)) *
VOLD!(Q%)
    HFBOT!(Q%) = ONE! + TWO!
REM CHANGE UNITS FROM LBF/(IN*MICROSEC) TO BTU/(H*FT^2)
    HFBOT!(Q%) = HFBOT!(Q%) / ASUR!(Q%) * 12 * 1000000!
    * 3600 / 778.16
NEXT Q%

REM BEGIN CALCULATIONS FOR COMPLEX NUSSELT NUMBER METHOD TO
PREDICT HEAT FLUX

REM CALCULATE DELTA T (T/C - RTD)
REM AVERAGE THE WALL TEMPERATURE
SUM! = 0
FOR Q% = 0 TO 360
    SUM! = SUM! + RTD2TOP!(Q%)
NEXT Q%
RTDTOPAVG! = SUM! / 361

REM OBTAIN TEMPERATURE DIFFERENCE
FOR Q% = 0 TO 360
    DELTMPTOP!(Q%) = TEQS2TOP!(Q%) - RTDTOPAVG!
    DELTMPBOT!(Q%) = TEQS2BOT!(Q%) - RTDTOPAVG!
NEXT Q%

FOR Q% = 0 TO 360
    DEL2TMPTOP!(Q%) = DELTMPTOP!(Q%)
    DEL2TMPBOT!(Q%) = DELTMPBOT!(Q%)
NEXT Q%
REM WANT TO MAINTAIN ORIGINAL DELTA, NOT TO ONLY HAVE IT
SMOOTHED

```

REM FIND D(DELTMP)/Dt FOR BOTH TEMPERATURE MEASUREMENTS

IT% = 20

CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!,
DEL2TMPTOP!(), IT%)

CALL FORECOMB(TIME2!(), A!(), B!(), P!, DEL2TMPTOP!(), IT%,
DDELTMPOT!())

IT% = 20

CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!,
DEL2TMPBOT!(), IT%)

CALL FORECOMB(TIME2!(), A!(), B!(), P!, DEL2TMPBOT!(), IT%,
DDELTMPBOT!())

REM CALCULATE SUMS FOR CALCULATIONS OF NUR, NUI FOR TOP
SECTION

HFDELT! = 0

DDELOTSQR! = 0

HFDDELT! = 0

DELTDDELT! = 0

DELTSQR! = 0

FOR Q% = 0 TO 360

HFDELT! = HFDELT! + HFTOP!(Q%) * DELTMPTOP!(Q%)

DDELOTSQR! = DDELOTSQR! + DDELTMPOT!(Q%) ^ 2

HFDDELT! = HFDDELT! + HFTOP!(Q%) * DDELTMPOT!(Q%)

DELTDDELT! = DELTDDELT! + DELTMPTOP!(Q%) *

DDELTMPOT!(Q%)

DELTSQR! = DELTSQR! + DELTMPTOP!(Q%) ^ 2

NEXT Q%

REM CALCULATE REAL AND IMAGINARY NUSSELT #

RSUM! = ((HFDELT! * DDELOTSQR!) - (HFDDELT! * DELTDDELT!)) /
((DELTSQR! * DDELOTSQR!) - (DELTDDELT!) ^ 2)

ISUM! = ((HFDDELT! * DELTSQR!) - (HFDELT! * DELTDDELT!)) /
((DELTSQR! * DDELOTSQR!) - DELTDDELT! ^ 2)

NURTOP! = (DH! * RSUM!) / (KTOP! * 3600 * 12)

NUITOP! = (OMEGA! * DH! * ISUM!) / (KTOP! * 1000000! * 3600
* 12)

PRINT "REAL PORTION OF NUSSELT # FOR TOP SECTION ="; NURTOP!

PRINT "IMAGINARY PORTION OF NUSSELT # FOR TOP SECTION =";

NUITOP!

REM CALCULATE HF BASED ON NUSSELT #'S

FOR Q% = 0 TO 360

HFNUTOP!(Q%) = ((NURTOP! * KTOP! * 3600 * 12 *
DELTMPTOP!(Q%)) / DH!) + ((NUITOP! * KTOP! * 12 * 1000000! *
3600 * DDELTMPOT!(Q%)) / (OMEGA! * DH!))

NEXT Q%

```

REM REPEAT NUSSELT CALCULATIONS FOR BOTTOM SECTION
HFDELT! = 0
DDELOTSQR! = 0
HFDDELT! = 0
DELTDDDELT! = 0
DELTSQR! = 0

FOR Q% = 0 TO 360
    HFDELT! = HFDELT! + HFBOT!(Q%) * DELTMPBOT!(Q%)
    DDELOTSQR! = DDELOTSQR! + DDELTMPBOT!(Q%) ^ 2
    HFDDELT! = HFDDELT! + HFBOT!(Q%) * DDELTMPBOT!(Q%)
    DELTDDDELT! = DELTDDDELT! + DELTMPBOT!(Q%) *
DDELTMPBOT!(Q%)
    DELTSQR! = DELTSQR! + DELTMPBOT!(Q%) ^ 2
NEXT Q%

REM CALCULATE REAL AND IMAGINARY NUSSELT #
RSUM! = ((HFDELT! * DDELOTSQR!) - (HFDDELT! * DELTDDDELT!)) /
((DELTSQR! * DDELOTSQR!) - (DELTDDDELT!) ^ 2)
ISUM! = ((HFDDELT! * DELTSQR!) - (HFDELT! * DELTDDDELT!)) /
((DELTSQR! * DDELOTSQR!) - DELTDDDELT! ^ 2)
NURBOT! = (DH! * RSUM!) / (KBOT! * 3600 * 12)
NUIBOT! = (OMEGA! * DH! * ISUM!) / (KBOT! * 1000000! * 3600
* 12)
PRINT "REAL PORTION OF NUSSELT # FOR BOTTOM SECTION =";
NURBOT!
PRINT "IMAGINARY PORTION OF NUSSELT # FOR BOTTOM SECTION =";
NUIBOT!

REM CALCULATE HF BASED ON NUSSELT #'S
FOR Q% = 0 TO 360
    HFNUBOT!(Q%) = ((NURBOT! * KBOT! * 3600 * 12 *
DELTMPBOT!(Q%)) / DH!) + ((NUIBOT! * KBOT! * 12 * 1000000! *
3600 * DDELTMPBOT!(Q%)) / (OMEGA! * DH!))
NEXT Q%

REM CALCULATE SURFACE HEAT FLUX FROM RTD'S
IT! = 20
FOR Q% = 0 TO 360
    HFRTD!(Q%) = 0
NEXT Q%
FOR Q! = 0 TO IT!
    AQ!(Q!) = (RTDB!(Q!) + RTDA!(Q!)) * Q! ^ .5
    BQ!(Q!) = (RTDB!(Q!) - RTDA!(Q!)) * Q! ^ .5
NEXT Q!
IT% = 20
CALL FORECOMB(TIME2!(), AQ!(), BQ!(), P!, QF!(), IT%,
QFPRIME!())
FOR Q% = 0 TO 360

```

```

        HFRTD!(Q%) = QF!(Q%) * .0743 * 3600 * (OMEGA! / 2) ^
.5
NEXT Q%

REM WRITE ALL DATA TO FILE
INPUT "ENTER PATH AND NAME FOR FILE CONTAINING ANALYZED
DATA"; FILE2NM$
OPEN FILE2NM$ FOR OUTPUT AS #5
A$ = "PTOP"
N$ = "PBOT"
B$ = "PTAD"
C$ = "PTISO"
D$ = "VOL"
E$ = "TGAS"
K$ = "RTDTOP"
L$ = "TEQSTOP"
O$ = "TEQSBOT"
M$ = "TEQS2TOP"
P$ = "TEQS2BOT"
F$ = "DELTTOP"
Q$ = "DELTBOT"
G$ = "HFTOP"
R$ = "HFBOT"
H$ = "HFNUTOP"
S$ = "HFNUBOT"
I$ = "HFRTD"
J$ = "TIME"
WRITE #5, A$, N$, B$, C$, D$, E$, K$, L$, O$, M$, P$, F$,
Q$, G$, R$, H$, S$, I$, J$
FOR Q% = 0 TO 360
    WRITE #5, PTOP2!(Q%), PBOT2!(Q%), PTOPAD!(Q%),
PTOPISO!(Q%), VOL2!(Q%), TEMP2!(Q%), RTD2TOP!(Q%),
TEQSTOP!(Q%), TEQSBOT!(Q%), TEQS2TOP!(Q%), TEQS2BOT!(Q%),
DELTMTOP!(Q%), DELTMPBOT!(Q%), HFTOP!(Q%), HFBOT!(Q%),
HFNUTOP!(Q%), HFNUBOT!(Q%),
HFRTD!(Q%), TIME2!(Q%)
NEXT Q%

REM ERASE ARRAYS TO UNCLUTTER MEMORY
ERASE PTOP2!
ERASE PTOPAD!
ERASE PTOPIISO!
ERASE VOL2!
ERASE TEMP2!
ERASE RTD2TOP!
ERASE TEQSTOP!
ERASE TEQSBOT!
ERASE TEQS2TOP!
ERASE TEQS2BOT!

```

```

ERASE DELTMPTOP!
ERASE DELTMPBOT!
ERASE HFTOP!
ERASE HFBOT!
ERASE HFNUTOP!
ERASE HFNUBOT!
ERASE HFRTD!
ERASE TIME2!
ERASE RTDA!
ERASE RTDB!
ERASE DEL2TMPTOP!
ERASE DEL2TMPBOT!
ERASE A!
ERASE B!
ERASE PTPD!
ERASE PBOTD!
ERASE VOLD!
ERASE TEMPD!
ERASE RTDDTOP!
ERASE ASUR!
ERASE DDELTMPPTOP!
ERASE DDELTMPBOT!
ERASE PBOT2RAW!
ERASE PBOT2RAW!
CLOSE #5
END

```

```

REM $STATIC
SUB EQSPC (DELTA!, Y!(), X!(), MAX%)
REM TAKE THE ORIGINAL AMOUNT OF DATA POINTS AND CREATE MAX%
EQUALLY SPACED POINTS
REM LINEARLY INTERPOLATE TO FIND THE EQUALLY SPACED POINTS
REM  $Y! = M * X! + B!$ 
I% = 1
J% = I%
NEWTIME! = X!(0) + DELTA!
DO WHILE I% <= 359
REM FIND WHERE NEWTIME! IS LOCATED
DO WHILE NEWTIME! > X!(J%)
J% = J% + 1
LOOP
REM LINEAR FIT TO FIND NEW Y!() VALUE
Y1! = Y!(J%)
Y0! = Y!(J% - 1)
X1! = X!(J%)
X0! = X!(J% - 1)
M! = (Y1! - Y0!) / (X1! - X0!)
B! = Y0! - (M! * X0!)
Y!(I%) = M! * NEWTIME! + B!

```

```

        NEWTIME! = X!(0) + DELTA! * I%
        I% = I% + 1
LOOP
Y!(360) = Y!(MAX%)
END SUB

SUB FOCOEFF (X!(), A!(), B!(), P!, DELTA!, Y!(), IT%)
REM PROGRAM TO FIND THE FOURIER COEFFICIENTS
REM A = ALPHA, B = BETA
FOR Q% = 0 TO IT%
        A!(Q%) = 0
        B!(Q%) = 0
NEXT Q%
J% = 0
PI! = 3.141592654#
A!(J%) = 0
B!(J%) = 0
DO WHILE J% <= IT%
        FOR R% = 1 TO 359
                A!(J%) = A!(J%) + (Y!(R%) * COS(J% * PI! *
X!(R%) / P!))
                B!(J%) = B!(J%) + (Y!(R%) * SIN(J% * PI! *
X!(R%) / P!))
        NEXT R%
        A!(J%) = A!(J%) + .5 * ((Y!(0) * COS(J% * PI! *
X!(0) / P!)) + (Y!(360) * COS(J% * PI! * X!(360) / P!)))
        A!(J%) = (DELTA! / P!) * A!(J%)
        B!(J%) = B!(J%) + .5 * ((Y!(0) * SIN(J% * PI! *
X!(0) / P!)) + (Y!(360) * SIN(J% * PI! * X!(360) / P!)))
        B!(J%) = (DELTA! / P!) * B!(J%)
        J% = J% + 1
LOOP
END SUB

SUB FODER (A!(), B!(), IT%, Z!(), RPMHZ!, PI!, Y!(), P!)
REM ROUTINE JUST TO CALCULATE THE DERIVATIVE OF THE FOURIER
EXPANSION
FOR Q% = 0 TO 360
        Z!(Q%) = 0
NEXT Q%

IT! = IT%

FOR I! = 0 TO 360
        FOR J! = 1 TO IT!
                LANC! = (SIN(PI! * J! / IT!)) / (PI! * J! /
IT!)
                Z!(I!) = Z!(I!) + LANC! * ((B!(J!) * (PI! *
J! / P!) * COS(PI! * Y!(I!) * J! / P!)) - (A!(J!) * (PI! *

```



```

J! / P!) * SIN(PI! * Y!(I!) * J! / P!)))
    NEXT J!
NEXT I!

END SUB

SUB FORECOMB (X!(), A!(), B!(), P!, Y!(), IT%, Z!())
REM ROUTINE TO RECOMBINE FOURIER COEFFICIENTS FOR ORIGINAL
SIGNAL
REM AND FOR THE DERIVATIVE
REM LANZOS SMOOTHING TERM = LANC
REM X! = TIME, Y! = P,V,TEMP, Z! = DERIVATIVE OF Y! WRT
TIME
PI! = 3.141592654#
FOR Q% = 0 TO 360
    Z!(Q%) = 0
NEXT Q%

IT! = IT%
FOR I% = 0 TO 360
    SUM! = 0
    FOR N! = 1 TO IT!
        SUM! = SUM! + A!(N!) * COS(PI! * N! * X!(I%
/ P!) + B!(N!) * SIN(PI! * N! * X!(I% / P!)
        LANC! = (SIN(PI! * N! / IT!)) / (PI! * N! /
IT!)
        Z!(I%) = Z!(I%) + LANC! * ((B!(N!) * (PI! *
N! / P!) * COS(PI! * X!(I%) * N! / P!)) - (A!(N!) * (PI! *
N! / P!) * SIN(PI! * X!(I%) * N! / P!)))
    NEXT N!
    Y!(I%) = A!(0) / 2 + SUM!
NEXT I%

END SUB

SUB MEAN (Y!(), X!(), DELTA!, MN!)
REM ROUTINE TO CALCULATE THE MEAN VALUE OF THE PASSED ARRAY

FOR I% = 0 TO 360
    MN! = MN! + Y!(I%)
NEXT I%
MN! = MN! / 361
END SUB

SUB MINMAX (Y!(), MAX%, MIN%)
REM ROUTINE TO CALCULATE THE MINIMUM AND MAXIMUM POINT IN
THE PASSED ARRAY

```

```

FOR I% = 1 TO 360
    IF Y!(I%) > Y!(MAX%) THEN
        MAX% = I%
    END IF
    IF Y!(I%) < Y!(MIN%) THEN
        MIN% = I%
    END IF
NEXT I%
END SUB

```

C.2 TWO SPACE DATA REDUCTION PROGRAM

REM TWO SPACE DATA REDUCTION PROGRAM

```

REM DECLARE SUBROUTINES
DECLARE SUB EQSPC (DELTA!, X!(), Y!(), MAX%)
DECLARE SUB FOCOEFF (X!(), A!(), B!(), P!, DELTA!, Y!(),
IT%)
DECLARE SUB FORECOMB (X!(), A!(), B!(), P!, Y!(), IT%, Z!())
DECLARE SUB MINMAX (Y!(), MAX%, MIN%)
DECLARE SUB MEAN (Y!(), X!(), DELTA!, MN!)
DECLARE SUB FODER (A!(), B!(), IT%, Z!(), RPMHZ!, PI!, Y!(),
P!)

```

CLS

```

INPUT "ENTER FILE NAME AND PATH FOR FILE TO BE LOOKED AT";
FILENAME$
OPEN FILENAME$ FOR INPUT AS #1

```

```

REM ENTER THE NUMBER OD DATA POINTS WITHIN THE CYCLE
INPUT "ENTER NUMBER OF POINTS TAKEN PER CHANNEL ", X%

```

```

REM DECLARE ARRAYS DYNAMIC SO THAT SPACE IN MEMORY IS NOT
ALLOCATED UNTIL THE
REM ARRAY IS DEFINED
'$DYNAMIC
DIM PBAL!(X%)
DIM PTOP!(X%)
DIM PBOT!(X%)
DIM TEMP!(X%)
DIM RTDBOT!(X%)
DIM RTDTOP!(X%)
DIM VOL!(X%)

```

```

DIM TIME! (X%)
DIM PTOP2! (360)
DIM PBOT2! (360)
DIM TIME2! (360)
DIM PBAL2! (360)
DIM TEMP2! (360)
DIM RTD2TOP! (360)
DIM VOL2! (360)
DIM A! (100)
DIM B! (100)
DIM PAD! (360)
DIM PISO! (360)
DIM PTOP2RAW! (360)
DIM PTOPD! (360)
DIM VOLD! (360)
DIM TEMPD! (360)
DIM ASUR! (360)
DIM RTDDTOP! (360)
DIM RTDA! (100)
DIM RTDB! (100)
DIM HFRTD! (360)
DIM T! (360)
DIM AQ! (100)
DIM BQ! (100)
DIM QF! (360)
DIM QFPRIME! (360)
DIM PBOT2RAW! (360)
DIM PBOTD! (360)
DIM VOLT! (X%)
DIM VOLC! (X%)
DIM VOLTD! (360)
DIM VOL2D! (360)
DIM HFTPV! (360)
DIM HFT! (360)
DIM TEQS! (360)
DIM DDELTMP! (360)
DIM DELTMP! (360)
DIM DEL2TMP! (360)
DIM PDIFF! (360)
DIM PDIFFD! (360)
DIM DTEQS! (360)
DIM HFNU! (360)

```

```

PI! = 3.141592654#

```

```

REM THE VALUES FOR THE INPUTTED PARAMETERS ARE FOUND IN THE
.GEN FILE
REM WHICH WAS CREATED IN THE DATA ACQUISITION PROGRAM
INPUT "ENTER VOLUME RATIO"; RV!

```

```

INPUT "ENTER OMEGA OF RUN"; OMEGA!

REM CYLINDER ID IN INCHES
CYLD! = 7.502
REM STROKE LENGTH IN INCHES
LS! = 3.75
REM GAS PROPERTIES
INPUT "ENTER GAS USED (HE, AR)"; GAS$
IF GAS$ = "HE" THEN
    GAMMA! = 1.667
    REM R IN FT LBF/LBM R
    R! = 383.3
    REM CP! IN BTU/LBM F
    CP! = 1.24
END IF
INPUT "ENTER MASS (LBM)"; MASS!
INPUT "ENTER PNOT (PSIA)"; PNOT!
INPUT "ENTER VNOT (IN^3)"; VNOT!

REM BRING IN ALL INFORMATION FROM SPREADSHEET FILE INTO
SEPARATE ARRAYS
Q% = 0
DO UNTIL EOF(1)
    INPUT #1, PBAL!(Q%), PTOP!(Q%), PBOT!(Q%),
    TEMP!(Q%), RTDBOT!(Q%), RTDTOP!(Q%), VOL!(Q%), VOLC!(Q%),
    TIME!(Q%)
    Q% = Q% + 1
LOOP
CLOSE #1
MAX% = X% - 1

REM ALL DATA READ TO ARRAYS
REM CALL ROUTINE FOR EVENLY SPACED POINTS FOR FOURIER
ANALYSIS
NOPTS! = 360
DELTA! = (TIME!(MAX%) - TIME!(0)) / NOPTS!
CALL EQSPC(DELTA!, RTDTOP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, PTOP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, PBAL!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, PBOT!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, TEMP!(), TIME!(), MAX%)
CALL EQSPC(DELTA!, VOL!(), TIME!(), MAX%)
REM ALL DATA POINTS HAVE BEEN CHANGED, TIME REMAINS
UNCHANGED
NEWTIME! = TIME!(0) + DELTA!
I% = 1
TIME2!(0) = TIME!(0)
DO WHILE I% <= 359
    TIME2!(I%) = NEWTIME!

```

```

        I% = I% + 1
        NEWTIME! = TIME!(0) + DELTA! * I%
LOOP
TIME2!(360) = TIME!(MAX%)

REM IN THE ROUTINE WHICH CREATES THE EQUALLY SPACED POINTS,
THE EQUALLY SPACED POINTS ARE PLACED WITHIN
REM THE FIRST 361 SPACES IN THE ARRAY, THE FINAL POINTS IN
THE ARRAY
REM ARE LEFT FROM THE ORIGINAL DATA SET. THEREFORE, TAKE
THE FIRST 361 POINTS
REM AND PLACE THEM IN A SECOND ARRAY, THESE ARE THE EQUALLY
SPACED POINTS
I% = 0
DO WHILE I% <= (360)
    PTOP2!(I%) = PTOP!(I%)
    PBAL2!(I%) = PBAL!(I%)
    PBOT2!(I%) = PBOT!(I%)
REM TEMP IN RANKINE
    TEMP2!(I%) = TEMP!(I%) + 459.67
    RTD2TOP!(I%) = RTD2TOP!(I%) + 459.67
    VOL2!(I%) = VOL!(I%)
    VOLT!(I%) = VOL2!(I%) + VOLC!(I%)
    I% = I% + 1
LOOP

REM SUBTRACT OUT THE BALANCE PRESSURE FROM THE PRESSURE
MEASUREMENT
REM THEN ADD BACK THE AVERAGED BALANCE PRESSURE
FOR Q% = 0 TO 360
    PTOP2RAW!(Q%) = PTOP2!(Q%) - PBAL2!(Q%)
    PBOT2RAW!(Q%) = PBOT2!(Q%) - PBAL2!(Q%)
NEXT Q%

REM FIND AVERAGE BALANCE PRESSURE
SUM! = 0
FOR Q% = 0 TO 360
    SUM! = SUM! + PBAL2!(Q%)
NEXT Q%
PBAVG! = SUM! / 361
FOR Q% = 0 TO 360
    PTOP2!(Q%) = PTOP2RAW!(Q%) + PBAVG!
    PBOT2!(Q%) = PBOT2RAW!(Q%) + PBAVG!
NEXT Q%

REM SEND EQUALIZED SPACE DATA TO SUBROUTINE TO CALCULATE
FOURIER COEFF.
REM FOLLOW THAT WITH ROUTINE TO RECOMBINE FOURIER
COEFFICIENTS TO SMOOTH DATA

```

P! = (TIME2!(360) - TIME2!(0)) / 2

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOP2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, PTOP2!(), IT%,
PTOPD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PBOT2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, PBOT2!(), IT%,
PBOTD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, TEMP2!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, TEMP2!(), IT%,
TEMPD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, RTD2TOP!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, RTD2TOP!(), IT%,
RTDDTOP!())
REM CHANGE NAMES OF ALPHA AND BETA FOR SURFACE HF CALC.
FOR Q% = 0 TO IT%
 RTDA!(Q%) = A!(Q%)
 RTDB!(Q%) = B!(Q%)
NEXT Q%

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, VOLT!(), IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, VOLT!(), IT%,
VOLTD!())

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, VOL2!(), IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, VOL2!(), IT%,
VOLD!())

REM START TO CALCULATE NECESSARY FUNCTIONS

REM CALCULATE MINIMUM AND MAXIMUM PRESSURE

MIN% = 0
MAX% = 0
CALL MINMAX(PTOP2!(), MAX%, MIN%)
PTOP2MAX! = PTOP2!(MAX%)
PTOP2MIN! = PTOP2!(MIN%)

```

MIN% = 0
MAX% = 0
CALL MINMAX(PBOT2!(), MAX%, MIN%)
PBOT2MAX! = PBOT2!(MAX%)
PBOT2MIN! = PBOT2!(MIN%)

REM FIND MAXIMUM AND MINIMUM VOLUME IN TOTAL CYLINDER
MIN% = 0
MAX% = 0
CALL MINMAX(VOLT!(), MAX%, MIN%)
VOLMAX! = VOLT!(MAX%)
VOLMIN! = VOLT!(MIN%)
PRINT "MINIMUM VOLUME IS "; VOLMIN!
PRINT "MAXIMUM VOLUME IS "; VOLMAX!

REM CALCULATE PMEAN (PAVG)
MN! = 0
CALL MEAN(PTOP2!(), TIME2!(), DELTA!, MN!)
PTOPMEAN! = MN!
PRINT "PTOP(MEAN) ="; PTOPMEAN!

MN! = 0
CALL MEAN(PBOT2!(), TIME2!(), DELTA!, MN!)
PBOTMEAN! = MN!
PRINT "PBOT(MEAN) ="; PBOTMEAN!

REM FIND MEAN VOLUME OF TOTAL CYLINDER SPACE
MN! = 0
CALL MEAN(VOLT!(), TIME2!(), DELTA!, MN!)
VOLTMEAN! = MN!

REM FIND MEAN VOLUME OF VARYING CYLINDER SPACE
MN! = 0
CALL MEAN(VOL2!(), TIME2!(), DELTA!, MN!)
VOLMEAN! = MN!

REM CALCULATE SURFACE AREA FOR EACH TIME POSITION
FOR Q% = 0 TO 360
    ASUR!(Q%) = 601.452555# + .533191149# * VOL2!(Q%)
NEXT Q%

REM FIND MEAN SURFACE AREA FOR HYDRAULIC DIAMETER
MN! = 0
CALL MEAN(ASUR!(), TIME2!(), DELTA!, MN!)
ASURFMN! = MN!

REM CALCULATE MEAN TEMPERATURE FROM GAS T/C
MN! = 0

```

```

CALL MEAN(TEMP2!(), TIME2!(), DELTA!, MN!)
TCMN! = MN!

REM CALCULATE MIXED MEAN TEMPERATURE
FOR Q% = 0 TO 360
    TEQS!(Q%) = (PBOT2!(Q%) * VOL2!(Q%) + PTOP2!(Q%) *
VOLC!(Q%)) / (MASS! * R! * 12)
NEXT Q%

REM CALCULATE MEAN TEMPERATURE, IN R
MN! = 0
CALL MEAN(TEQS!(), TIME2!(), DELTA!, MN!)
TEMPMNR! = MN!
PRINT "MEAN MASS MEAN TEMPERATURE ="; TEMPMNR!

REM EXPRESS TEMPMEAN IN KELVINS FOR K! CALCULATION
TEMPMNK! = TEMPMNR! / 1.8

REM CALCULATE THERMAL CONDUCTIVITY IN BTU/S FT F BASED ON
TEMPMN
K! = (.5778 / 3600) * (.03722 + .0003896 * TEMPMNK! -
7.45E-08 * TEMPMNK! ^ 2 + 1.29E-11 * TEMPMNK! ^ 3)

REM CALCULATE AVERAGE DENSITY BASED ON AVERAGED PRESSURE AND
TMIX
RHO! = ((PTOPMEAN! + PBOTMEAN!) / 2) / (R! * TEMPMNR!)

REM CALCULATE HYDRAULIC DIAMETER (IN) FOR PEW
DH! = (4 * VOLTMEAN! / ASURFMN!)
PEW2! = (RHO! * CP! * OMEGA! * DH! ^ 2) / (K! * 4)
PRINT "PECLET F(DH,OMEGA) = "; PEW2!
PRINT "OMEGA"; OMEGA!

REM CALCULATE PRESSURE DROP ACROSS THE ORIFICE
FOR Q% = 0 TO 360
    PDIFF!(Q%) = PTOP2!(Q%) - PBOT2!(Q%)
NEXT Q%

IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PDIFF!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, PDIFF!(), IT%,
PDIFFD!())

REM CALCULATE MINIMUM AND MAXIMUM DELTA
MIN% = 0
MAX% = 0
CALL MINMAX(PDIFF!(), MAX%, MIN%)

```



```

PDIFFMAX! = PDIFF!(MAX%)
PDIFFMIN! = PDIFF!(MIN%)
PDROP! = PDIFFMAX! - PDIFFMIN!
PRINT "MAXIMUM PRESSURE DROP ="; PDROP!

```

```

REM CALCULATE LOSS (INTEGRAL OF PdV)
REM SINCE TOP SPACE IS CONSTANT VOLUME INTEGRAL IS ZERO
LOSSBOT! = 0
FOR I% = 1 TO 360
    LOSSBOT! = LOSSBOT! + (((PBOT2!(I%) + PBOT2!(I% - 1)) / 2) * (VOL2!(I%) - VOL2!(I% - 1)))
NEXT I%

```

```

REM CALCULATE PRESSURE DROP LOSS
LOSSPD! = 0
FOR Q% = 1 TO 360
    A! = (PTOP2!(Q%) + PTOP2!(Q% - 1)) / 2
    B! = (PBOT2!(Q%) + PBOT2!(Q% - 1)) / 2
    LOSSPD! = LOSSPD! + ((LOG(A! / B!)) / (VOLT!(Q%) ^ 2)) * (VOL2!(Q%) - VOL2!(Q% - 1))
NEXT Q%
LOSSPD! = LOSSPD! * PNOT! * VNOT! * VOLC!(1)
PRINT "LOSS NORMAL "; LOSSBOT!
PRINT "LOSSPD "; LOSSPD!

```

```

REM CALC. Pa/Po FROM APPROXIMATION 3.16 IN KORN. DISSERTATION
PVARBOT! = (PBOT2MAX! - PBOT2MIN!) / (PBOT2MAX! + PBOT2MIN!)

```

```

REM CALCULATE LOSSnd EQ. 3.25 KORN. DISSERTATION
LOSSNDBOT! = (-LOSSBOT!) / (PBOTMEAN! * VOLTMEAN! * (PVARBOT!) ^ 2 * ((GAMMA! - 1) / GAMMA!))
LOSSNDPD! = (-LOSSBOT! - LOSSPD!) / (PBOTMEAN! * VOLTMEAN! * (PVARBOT!) ^ 2 * ((GAMMA! - 1) / GAMMA!))

```

```

PRINT "LOSSnd BASED ON PBOT(EQ. 3.16) = "; LOSSNDBOT!
PRINT "LOSS TO INCLUDE PRESSURE DROP ="; LOSSNDPD!

```

```

REM BEGIN CALCULATIONS FOR p* AND Plead
REM CALCULATE ADIABATIC PRESSURE
FOR N% = 0 TO 360
    PAD!(N%) = PNOT! * (VNOT! / VOLT!(N%)) ^ GAMMA!
NEXT N%

```

```

REM CALCULATE ISOTHERMAL PRESSURE
FOR N% = 0 TO 360
    PISO!(N%) = PNOT! * VNOT! / VOLT!(N%)
NEXT N%

```

```

MIN% = 0
MAX% = 0
CALL MINMAX(PAD!(), MAX%, MIN%)
PADMAX! = PAD!(MAX%)
PADMIN! = PAD!(MIN%)

MIN% = 0
MAX% = 0
CALL MINMAX(PISO!(), MAX%, MIN%)
PISOMAX! = PISO!(MAX%)
PISOMIN! = PISO!(MIN%)

REM FIND MAGNITUDE AND PHASE SHIFT
IT% = 1
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PAD!(), IT%)
ALPHAAD! = A!(1)
BETAAD! = B!(1)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PISO!(), IT%)
ALPHAISO! = A!(1)
BETAISO! = B!(1)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PTOP2!(), IT%)
ALPHATOP! = A!(1)
BETATOP! = B!(1)
MAGAD! = (ALPHAAD! ^ 2 + BETAAD! ^ 2) ^ (1 / 2)
MAGISO! = (ALPHAISO! ^ 2 + BETAISO! ^ 2) ^ (1 / 2)
MAGTOP! = (ALPHATOP! ^ 2 + BETATOP! ^ 2) ^ (1 / 2)
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, PBOT2!(), IT%)
ALPHABOT! = A!(1)
BETABOT! = B!(1)
MAGBOT! = (ALPHABOT! ^ 2 + BETABOT! ^ 2) ^ (1 / 2)

PSTARTOP! = (MAGTOP! - MAGISO!) / (MAGAD! - MAGISO!)
PRINT "P* BASED ON PTOP(PRESSURE RATIO) = "; PSTARTOP!
PSTARBOT! = (MAGBOT! - MAGISO!) / (MAGAD! - MAGISO!)
PRINT "P* BASED ON PBOT(PRESSURE RATIO) = "; PSTARBOT!

ANG! = 360 / (2 * PI!)
PHASETOP! = ATN(ALPHATOP! / BETATOP!) * ANG!
PHASEBOT! = ATN(ALPHABOT! / BETABOT!) * ANG!
PHASEAD! = ATN(ALPHAAD! / BETAAD!) * ANG!
PHASEISO! = ATN(ALPHAISO! / BETAISO!) * ANG!
LEAD1 = PHASETOP! - PHASEAD!
LEAD2! = PHASETOP! - PHASEISO!
PRINT "PHASE ANGLE BASED ON PTOP(BASED ON AD.) = "; LEAD1!
PRINT "PHASE ANGLE BASED ON PTOP(BASED ON ISO.) = "; LEAD2!
LEAD1 = PHASEBOT! - PHASEAD!
LEAD2! = PHASEBOT! - PHASEISO!

```

```

PRINT "PHASE ANGLE BASED ON PBOT(BASED ON AD.) = "; LEAD1!
PRINT "PHASE ANGLE BASED ON PBOT(BASED ON ISO.) = "; LEAD2!

REM CALCULATE HEAT FLUX
REM HEAT FLUX MEASURED FROM P AND V
REM HEAT FLUX MEASURED FROM TMIX
REM HEAT FLUX MEASURED FROM HEAT FLUX SENSOR
REM PREDICTED HEAT FLUX WITH COMPLEX NUSSELT NUMBER METHOD

REM HF BASED ON T mass mean BROKEN INTO P AND V
FOR Q% = 0 TO 360
    ONE! = -PBOT2!(Q%) * (GAMMA / (GAMMA! - 1)) *
VOLD!(Q%)
    TWO! = -VOL2!(Q%) * (1 / (GAMMA! - 1)) * PBOTD!(Q%)
    THREE! = -VOLC!(Q%) * (1 / (GAMMA! - 1)) *
PTOPD!(Q%)
    HFTPV!(Q%) = ONE! + TWO! + THREE!
REM CHANGE UNITS FROM LBF/(IN*MICROSEC) TO BTU/(H*FT^2)
    HFTPV!(Q%) = HFTPV!(Q%) / ASUR!(Q%) * 12 * 1000000!
    * 3600 / 778.16
NEXT Q%

REM CALCULATE HEAT FLUX BASED ON T mass mean
REM NEED DT FOR THIS CALCULATION
IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, TEQS!(), IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, TEQS!(), IT%,
DTEQS!())

FOR Q% = 0 TO 360
    ONE! = -MASS! / GAMMA! * CP! * DTEQS!(Q%)
    TWO! = -PBOT2!(Q%) * VOLD!(Q%)
    HFT!(Q%) = (ONE! * 144 * 1000000! * 3600) + (TWO! *
12 * 1000000! * 3600 / 778.16)
    HFT!(Q%) = HFT!(Q%) / ASUR!(Q%)
NEXT Q%

REM CALCULATE TEMPERATURE DIFFERENCE
REM AVERAGE THE WALL TEMPERATURE
SUM! = 0
FOR Q% = 0 TO 360
    SUM! = SUM! + RTD2TOP!(Q%)
NEXT Q%
RTDTOPAVG! = SUM! / 361

REM TEMPERATURE DIFFERENCE IS TMIX - AVG WALL TEMP
FOR Q% = 0 TO 360
    DELTMP!(Q%) = TEQS!(Q%) - RTDTOPAVG!

```

```

NEXT Q%

FOR Q% = 0 TO 360
    DEL2TMP!(Q%) = DELTMP!(Q%)
NEXT Q%

REM WANT TO MAINTAIN ORIGINAL DELTA, NOT TO ONLY HAVE IT
SMOOTHED
REM FIND D(DELTMP)/Dt
IT% = 20
CALL FOCOEFF(TIME2!(), A!(), B!(), P!, DELTA!, DEL2TMP!(),
IT%)
CALL FORECOMB(TIME2!(), A!(), B!(), P!, DEL2TMP!(), IT%,
DDELTMP!())

REM CALCULATE SUMS FOR CALCULATIONS OF NUR, NUI
REM USE THE HEAT FLUX MEASURED FROM PRESSURE AND VOLUME
HFDELT! = 0
DDELTSQL! = 0
HFDDELT! = 0
DELTDDELT! = 0
DELTSQR! = 0
FOR Q% = 0 TO 360
    HFDELT! = HFDELT! + HFTPV!(Q%) * DELTMP!(Q%)
    DDELTSQL! = DDELTSQL! + DDELTMP!(Q%) ^ 2
    HFDDELT! = HFDDELT! + HFTPV!(Q%) * DDELTMP!(Q%)
    DELTDDELT! = DELTDDELT! + DELTMP!(Q%) * DDELTMP!(Q%)
    DELTSQR! = DELTSQR! + DELTMP!(Q%) ^ 2
NEXT Q%

REM CALCULATE REAL AND IMAGINARY NUSSELT #
RSM! = ((HFDELT! * DDELTSQL!) - (HFDDELT! * DELTDDELT!)) /
((DELTSQR! * DDELTSQL!) - (DELTDDELT!) ^ 2)
ISM! = ((HFDDELT! * DELTSQR!) - (HFDELT! * DELTDDELT!)) /
((DELTSQR! * DDELTSQL!) - DELTDDELT! ^ 2)
NUR! = (DH! * RSM!) / (K! * 3600 * 12)
NUI! = (OMEGA! * DH! * ISM!) / (K! * 1000000! * 3600 * 12)
PRINT "REAL PORTION OF NUSSELT # =" ; NUR!
PRINT "IMAGINARY PORTION OF NUSSELT # =" ; NUI!

REM CALCULATE HF BASED ON NUSSELT #'S
FOR Q% = 0 TO 360
    HFNU!(Q%) = ((NUR! * K! * 3600 * 12 * DELTMP!(Q%)) /
DH!) + ((NUI! * K! * 12 * 1000000! * 3600 * DDELTMP!(Q%)) /
(OMEGA! * DH!))
NEXT Q%

REM CALCULATE SURFACE FEAT FLUX FROM RTD'S

```

```

IT! = 20
FOR Q% = 0 TO 360
    HFRTD!(Q%) = 0
NEXT Q%
FOR Q! = 0 TO IT!
    AQ!(Q!) = (RTDB!(Q!) + RTDA!(Q!)) * Q! ^ .5
    BQ!(Q!) = (RTDB!(Q!) - RTDA!(Q!)) * Q! ^ .5
NEXT Q!
IT% = 20
CALL FORECOMB(TIME2!(), AQ!(), BQ!(), P!, QF!(), IT%,
QFPRIME!())
FOR Q% = 0 TO 360
    HFRTD!(Q%) = QF!(Q%) * .0743 * 3600 * (OMEGA! / 2) ^
.5
NEXT Q%

REM WRITE ALL DATA TO FILE
INPUT "ENTER PATH AND NAME FOR FILE CONTAINING ANALYZED
DATA"; FILE2NM$
OPEN FILE2NM$ FOR OUTPUT AS #5
A$ = "PTOP"
N$ = "PBOT"
B$ = "PTAD"
C$ = "PTISO"
D$ = "VOLV"
F$ = "VOLC"
G$ = "VOLT"
E$ = "TGAS"
K$ = "RTDTOP"
L$ = "TMIX"
J$ = "TIME"
M$ = "HFPV"
Q$ = "HFTMIX"
O$ = "HFNU"
P$ = "HFRTD"
R$ = "PDIFF"

WRITE #5, A$, N$, B$, C$, D$, F$, G$, E$, K$, L$, J$, M$,
Q$, O$, P$, R$
FOR Q% = 0 TO 360
    WRITE #5, PTOP2!(Q%), PBOT2!(Q%), PAD!(Q%),
PISO!(Q%), VOL2!(Q%), VOLC!(Q%), VOLT!(Q%), TEMP2!(Q%),
RTD2TOP!(Q%), TEQS!(Q%), TIME2!(Q%), HFTPV!(Q%), HFT!(Q%),
HFNU!(Q%), HFRTD!(Q%), PDIFF!(Q%)
NEXT Q%

REM ERASE ALL ARRAYS TO FREE UP MEMORY
ERASE PTOP!
ERASE PAD!

```

```

ERASE PISO!
ERASE VOL2!
ERASE TEMP2!
ERASE RTD2TOP!
ERASE HFRTD!
ERASE TIME2!
ERASE RTDA!
ERASE RTDB!
ERASE A!
ERASE B!
ERASE PTPD!
ERASE PBOTD!
ERASE VOLD!
ERASE TEMPD!
ERASE RTDDTOP!
ERASE ASUR!
ERASE PBOT2RAW!
ERASE PBOT2RAW!
CLOSE #5
END

REM $STATIC
SUB EQSPC (DELTA!, Y!(), X!(), MAX%)
REM Y! = M*X! + B!
I% = 1
J% = I%
NEWTIME! = X!(0) + DELTA!
DO WHILE I% <= 359
REM FIND WHERE NEWTIME! IS LOCATED
    DO WHILE NEWTIME! > X!(J%)
        J% = J% + 1
    LOOP
REM LINEAR FIT TO FIND NEW Y!() VALUE
    Y1! = Y!(J%)
    Y0! = Y!(J% - 1)
    X1! = X!(J%)
    X0! = X!(J% - 1)
    M! = (Y1! - Y0!) / (X1! - X0!)
    B! = Y0! - (M! * X0!)
    Y!(I%) = M! * NEWTIME! + B!
    NEWTIME! = X!(0) + DELTA! * I%
    I% = I% + 1
LOOP
Y!(360) = Y!(MAX%)
END SUB

SUB FOCOEFF (X!(), A!(), B!(), P!, DELTA!, Y!(), IT%)
FOR Q% = 0 TO IT%
    A!(Q%) = 0

```

```

        B!(Q%) = 0
NEXT Q%
J% = 0
PI! = 3.141592654#
A!(J%) = 0
B!(J%) = 0
DO WHILE J% <= IT%
    FOR R% = 1 TO 359
        A!(J%) = A!(J%) + (Y!(R%) * COS(J% * PI! *
X!(R%) / P!))
        B!(J%) = B!(J%) + (Y!(R%) * SIN(J% * PI! *
X!(R%) / P!))
    NEXT R%
    A!(J%) = A!(J%) + .5 * ((Y!(0) * COS(J% * PI! *
X!(0) / P!)) + (Y!(360) * COS(J% * PI! * X!(360) / P!)))
    A!(J%) = (DELTA! / P!) * A!(J%)
    B!(J%) = B!(J%) + .5 * ((Y!(0) * SIN(J% * PI! *
X!(0) / P!)) + (Y!(360) * SIN(J% * PI! * X!(360) / P!)))
    B!(J%) = (DELTA! / P!) * B!(J%)
    J% = J% + 1
LOOP
END SUB

SUB FODER (A!(), B!(), IT%, Z!(), RPMHZ!, PI!, Y!(), P!)
FOR Q% = 0 TO 360
    Z!(Q%) = 0
NEXT Q%

IT! = IT%
J! = 0
FOR I! = 0 TO 360
    FOR J! = 1 TO IT!
        LANC! = (SIN(PI! * J! / IT!)) / (PI! * J! /
IT!)
        Z!(I!) = Z!(I!) + LANC! * ((B!(J!) * (PI! *
J! / P!) * COS(PI! * Y!(I!) * J! / P!)) - (A!(J!) * (PI! *
J! / P!) * SIN(PI! * Y!(I!) * J! / P!)))
    NEXT J!
NEXT I!

END SUB

SUB FORECOMB (X!(), A!(), B!(), P!, Y!(), IT%, Z!())
REM X! = TIME, Y! = P,V,TEMP, Z! = DERIVATIVE OF Y! WRT
TIME
PI! = 3.141592654#
FOR Q% = 0 TO 360
    Z!(Q%) = 0
NEXT Q%

```

```

IT! = IT%
FOR I% = 0 TO 360
    SUM! = 0
    FOR N! = 1 TO IT!
        SUM! = SUM! + A!(N!) * COS(PI! * N! * X!(I%)
/ P!) + B!(N!) * SIN(PI! * N! * X!(I%) / P!)
        LANC! = (SIN(PI! * N! / IT!)) / (PI! * N! /
IT!)
        Z!(I%) = Z!(I%) + LANC! * ((B!(N!) * (PI! *
N! / P!) * COS(PI! * X!(I%) * N! / P!)) - (A!(N!) * (PI! *
N! / P!) * SIN(PI! * X!(I%) * N! / P!)))
    NEXT N!
    Y!(I%) = A!(0) / 2 + SUM!
NEXT I%

END SUB

SUB MEAN (Y!(), X!(), DELTA!, MN!)
FOR I% = 0 TO 360
    MN! = MN! + Y!(I%)
NEXT I%
MN! = MN! / 361
END SUB

SUB MINMAX (Y!(), MAX%, MIN%)
FOR I% = 1 TO 360
    IF Y!(I%) > Y!(MAX%) THEN
        MAX% = I%
    END IF
    IF Y!(I%) < Y!(MIN%) THEN
        MIN% = I%
    END IF
NEXT I%
END SUB

```


APPENDIX D

FINAL DATA SETS

D.1 ONE SPACE FINAL RESULTS

Pew	Omega rad/sec	P _{mean} psia	p*	Plead degrees	LOSSnd
115.4352	2.76	13.74	0.804171	4.405014	0.403974
270.4128	6.47	13.78	0.896638	3.1292	0.268691
530.1744	12.66	13.8	0.9042	2.2508	0.196987
1292.666	30.84	13.87	0.9547	1.5605	0.140575
2754.773	65.56	13.89	0.9497	1.144	0.100658
289.1295	2.8	34.06	0.8842	2.9683	0.263256
693.0841	6.47	35.37	0.9139	2.01791	0.176615
1321.998	12.49	34.95	0.9522	1.5443	0.126581
3408.689	30.45	37.05	0.9565	0.9441	0.082671
7013.426	62.92	37.04	0.9431	0.7693	0.071359
682.4129	3.29	68.53	0.935	2.0153	0.175314
1383.171	6.51	70.16	0.9464	1.4929	0.126498
2674.535	12.64	69.88	0.9609	1.1104	0.092289
7003.225	31.09	74.82	0.9764	0.6535	0.056134
13452	59.64	74.72	0.9654	0.5229	0.04512
1348.905	3.03	146.9	0.9505	1.4825	0.132042
2891.931	6.51	146.67	0.968	1.0707	0.090557
5559.96	12.5	146.82	0.9777	0.8232	0.069329
13624.22	29.91	149.49	0.9812	0.504	0.042491
29188.94	64.05	149.51	0.9618	0.3446	0.031576
2337.214	3.25	237.99	0.9498	1.1713	0.1074
4677.339	6.49	237.68	0.9635	0.8703	0.074842
8950.395	12.45	237.21	0.9765	0.641	0.054204
46098.41	62.63	242.33	0.9508	0.2873	0.024353

Pew	Nu(r)	Nu(i)	Numag	Phase degrees
115.4352	18.87046	15.16644	24.20982	38.7893
270.4128	27.2868	23.16374	35.79285	40.3279
530.1744	38.63025	36.02166	52.81909	42.99871
1292.666	62.28192	59.85096	86.37809	43.85972
2754.773	96.65846	141.6125	171.4554	55.68428
289.1295	27.48314	22.64612	35.61137	39.48854
693.0841	43.12139	40.60786	59.23219	43.28051
1321.998	55.9877	46.89394	73.03194	39.94872
3408.689	94.32844	128.6615	159.5357	53.75292
7013.426	171.2764	292.7975	339.2138	59.67384
682.4129	39.13013	26.09679	47.03413	33.70024
1383.171	58.16803	46.89104	74.71472	38.87336
2674.535	81.12073	66.35855	104.8047	39.28392
7003.225	126.77	119.9314	174.5112	43.41216
13452	197.1048	251.1979	319.2972	51.88016
1348.905	55.58764	39.35189	68.10695	35.29564
2891.931	83.36453	55.84651	100.3418	33.81842
5559.96	123.3796	78.20086	146.075	32.36747
13624.22	184.4441	154.4462	240.5686	39.94148
29188.94	299.8415	542.2306	619.6119	61.05838
2337.214	75.5376	66.64056	100.7318	41.41929
4677.339	110.5402	98.6431	148.154	41.74486
8950.395	156.244	156.1047	220.8639	44.97445
46098.41	370.1814	1169.346	1226.542	72.43371

D.2 TWO SPACE FINAL RESULTS

PEw	Pmean psia	Omega rad/sec	Delta P max psia	P* FIX	P* VAR
90.41309	13.84685	3.330919	0.020752	0.7632	0.7663
174.064	13.85748	6.432955	0.027093	0.8296	0.8349
336.8172	13.86635	12.44956	0.029314	0.8876	0.8923
814.7897	13.94613	30.01327	0.089838	0.9238	0.9323
1736.188	13.98256	63.9809	0.386856	0.9261	0.9502
214.9534	33.60924	3.284847	0.035052	0.8063	0.8121
409.3796	33.60522	6.276324	0.033907	0.8563	0.8608
803.0883	33.60025	12.28919	0.047496	0.8959	0.9006
1937.869	33.67795	29.75324	0.212661	0.9148	0.9221
4228.909	33.78011	64.93181	0.986931	0.9186	0.9301
473.4475	73.64415	3.300437	0.047832	0.861	0.8648
919.1058	73.70319	6.403563	0.049639	0.8923	0.8962
1789.183	73.74493	12.48541	0.113585	0.9143	0.9191
4416.643	76.92129	29.89387	0.488417	0.939	0.94
9307.856	76.9613	62.54888	2.081486	0.9405	0.9511
876.1673	147.5092	3.068118	0.105914	0.8826	0.8879
1812.121	147.5302	6.355197	0.122374	0.905	0.9103
3544.999	147.2578	12.43517	0.24353	0.9287	0.9325
1349.491	244.8127	2.859611	0.187533	0.8743	0.8802
2968.304	245.0366	6.278666	0.203104	0.8966	0.9021
5890.824	244.8971	12.47497	0.339038	0.914	0.9178

PEw	Plead FIX degrees	Plead VAR degrees	LOSSnd	loss corr for del P
90.41309	5.5923	5.641206	0.529967	0.527776
174.064	4.757	4.83942	0.443343	0.435673
336.8172	3.598003	3.763523	0.335738	0.323197
814.7897	3.1609	3.971645	0.348969	0.288805
1736.188	2.353958	6.46085	0.574925	0.253821
214.9534	4.428291	4.457382	0.411334	0.408897
409.3796	3.72818	3.767334	0.339209	0.336166
803.0883	3.076	3.2293	0.283944	0.272172
1937.869	2.399914	3.298023	0.289697	0.219077
4228.909	1.728722	6.063232	0.538367	0.200312
473.4475	3.538348	3.559945	0.329081	0.32692
919.1058	2.944633	3.002689	0.267538	0.262377
1789.183	2.469852	2.632847	0.229863	0.216367
4416.643	1.880207	2.787155	0.243914	0.172611
9307.856	1.176414	5.210182	0.460682	0.145651
876.1673	2.977509	2.995884	0.288805	0.286625
1812.121	2.459221	2.515167	0.225638	0.220528
3544.999	2.001111	2.161141	0.188046	0.175056
1349.491	2.541058	2.561142	0.265787	0.263183
2968.304	2.111343	2.160675	0.198395	0.193564
5890.824	1.692032	1.831921	0.16015	0.149046

PEw	NUr	NUi	NUmag	Phase degrees	Rein/Pew
90.41309	13.33072	13.90403	19.26214	46.20594	55.92592
174.064	20.27937	19.76133	28.31542	44.25876	56.14554
336.8172	26.53971	29.91406	39.99009	48.42061	56.189
814.7897	60.76107	48.17506	77.54189	38.40951	56.31848
1736.188	105.8116	98.79302	144.7624	43.03535	56.48976
214.9534	32.11929	25.10022	40.76358	38.00651	56.30655
409.3796	45.90628	37.82949	59.48493	39.49054	56.48268
803.0883	69.23869	57.03765	89.70669	39.48111	56.3679
1937.869	138.1608	113.0078	178.4914	39.28124	56.68719
4228.909	284.8287	275.0907	395.9827	44.00362	56.86167
473.4475	52.64988	38.17188	65.03155	35.94262	56.28171
919.1058	82.4519	59.50358	101.6808	35.81709	56.29527
1789.183	130.8733	94.07994	161.1796	35.71095	56.41695
4416.643	253.7387	174.0408	307.6906	34.44645	57.07762
9307.856	455.029	434.3748	629.073	43.66969	56.69852
876.1673	80.24416	54.16308	96.81304	34.0185	56.6283
1812.121	135.174	92.08484	163.5592	34.26394	56.72208
3544.999	211.0397	147.0073	257.1943	34.86053	56.62958
1349.491	106.0713	88.55264	138.1763	39.8565	56.87224
2968.304	192.2864	171.6198	257.7352	41.74961	56.82237
5890.824	301.0086	296.4761	422.4976	44.56536	56.85603

APPENDIX E

ERROR ANALYSIS

A simple analysis was conducted to attempt to quantify the amount of error introduced into the heat flux measurement from possible instrument errors. This analysis was done by introducing the error into the raw data sets, and then evaluating those erroneous sets in the data reduction program. A statistical uncertainty analysis, such as Kline and McClintock (1953), on the heat flux would have been very complicated mathematically.

E.1 CASE ANALYSIS

There were four independent cases studied in the analysis. The first two were simulated gain error on the differential pressure transducers. In these cases the differential pressure was changed by a gain error of 0.5%. In case 1, the pressure was changed on the differential transducer located in the fixed volume section, for case 2 the varying volume pressure measurement was changed.

The latter two cases simulated offset error in the differential pressure transducers and in the volume transducer. In these cases an offset amount was added to all pressure or volume points in the cycle. In case 3, 0.33 in³ was added to all of the volume points. This represented the maximum percent deviation between the calibration and the best fit line over the full scale of the instrument. In case 4, 0.05 psi was added to all of the points of both differential pressure transducers. This seemed to be the maximum error due to bridge imbalance which could go uncorrected by the experimenter.

The above cases were run at a low pressure-low piston frequency experimental run (13.85 psia, 3.33 rad/sec) and a high pressure-intermediate piston frequency experimental run (244.9 psia, 12.47 rad/sec) for the two space experiments. The two space experiments were chosen since the total volume was lower; therefore, this would represent a worst case error on the volume since the total cylinder volume was less than in the one space experiments. Since the magnitude of the volume was consistent for all of the

experimental runs, it was felt that the affect of the error would be seen in the pressure measurements, which ranged from 1 atm to 16.7 atm. The two experimental runs chosen were made at maximum and minimum pressure.

E.2 ERROR RESULTS

The results of the error analysis are shown in Tables E.1 and E.2. The data showed that the introduced error, from cases 1 - 4, did not significantly affect the calculations of P^* , P_{load} , or $Loss_{nd}$. In fact, the maximum error in any of those results was $\pm 0.4\%$. The major affect of the introduced error was seen in the calculation of Nu_r , Nu_i , and the phase and magnitude of Nu_c . For the low pressure analysis, Table E.1, a maximum error of 8.6% was seen in the phase, while for the high pressure analysis the maximum error in the phase was 4.5%. In both pressure cases, the introduced error had more of an affect on the phase than on the magnitude.

The heat flux was also calculated from the pressure and volume data with the introduced error. The percent error in the heat flux associated with the four error cases is shown in Figures E.1 and E.2. The heat flux calculated with the simulated error were phase shifted from the baseline case. This caused a large amount of percent error at the points where the baseline case was very low in magnitude and, due to the phase shift, the simulated heat flux had a slightly different magnitude. This occurred twice during the cycle. In both cases the error in the heat flux seemed to be $\pm 5\%$. In the case of the high pressure - intermediate piston frequency run the error was larger. This was due to noise in the baseline case; due to the phase shift in the simulated cases, the percent error at this case was amplified.

Table E.1: Results of Error Analysis
Low Pressure

	baseline	case 1	case 2	case 3	case 4
p^* fix	.7614	.7623	.7614	.7635	.7614
p^* var	.7645	.7645	.7652	.7666	.7645
P_{lead} fix (deg)	5.592	5.590	5.592	5.592	5.592
P_{lead} var (deg)	5.641	5.641	5.638	5.641	5.641
$Loss_{nd}$	.5278	.5279	.5277	.5276	.5296
Nu_r	13.33	13.17	13.33	13.18	12.12
Nu_i	13.90	13.66	13.96	13.83	14.51
Nu_{mag}	19.26	18.97	19.30	19.10	18.91
Nu_{phase} (deg)	46.2	46.0	46.3	46.4	50.1

Table E.2: Results of Error Analysis
High Pressure

	baseline	case 1	case 2	case 3	case 4
P^* fix	.914	.9176	.914	.9163	.914
P^* var	.9178	.9178	.9214	.9201	.9178
P_{lead} fix (deg)	1.692	1.691	1.692	1.692	1.692
P_{lead} var (deg)	1.836	1.836	1.835	1.836	1.836
$Loss_{nd}$	.1490	.1493	.1488	.1490	.1491
Nu_r	301.02	299.61	301.50	300.89	301.60
Nu_i	296.39	275.73	298.95	292.60	300.48
Nu_{mag}	422.45	407.18	424.59	419.70	425.74
Nu_{phase} (deg)	44.6	42.6	44.8	44.2	44.9

Figure E.1: Error Results
Low Mean Pressure

**Figure E.2: Error Results
High Mean Pressure**

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