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EMPIRICAL MODELING OF A LYSHOLM HELICAL SCREW EXPANDER

KARL A. BROWN
B.S.M.E. - University of California, Berkeley

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ABSTRACT

Testing of a Lysholm helical screw expander having a rotor diameter of 5.12 inches and an expansion ratio of 5.3 was performed with simulated geothermal fluid at an inlet pressure of 120 psia. The inlet quality, engine speed and pressure ratio were independently varied to produce a three-dimensional data matrix of 104 data points. Statistical curve fitting methods were adapted to produce equations for mass flow rate and power output in terms of the three variables associated with the data matrix. These explicit equations were combined with a steam table subroutine to produce a computer model for prediction of mass flow rate, power, efficiency and exhaust quality at a given set of operating conditions.

The predictive ability observed during validation of the model is adaquate for use in modeling of hybrid geothermal energy conversion systems. Mass flow rates were predicted to within 2.7% of the measured values. For most conditions, predictions of power were within 3.3% of the measured values. Predictions of efficiency reflected the combined errors in prediction of power and mass flow rate. Efficiency predictions were in error by as much as 5% of the measured values. The predictions for exhaust quality were within 0.4% of the measured values.

Isentropic engine efficiencies of over 43% were measured during the tests. An observed peak in efficiency represented an optimum pressure ratio and a trade-off between lower leakage fractions at high speeds and low inlet qualities, increasing frictional losses at high speeds, and increasing fluid and power densities at low qualities.

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Introduction

One of the main problems facing the development of geothermal energy resources is the quality of the fluid at the wellhead. Only a few of the developed geothermal fields, including the geysers in Northern California, have dry steam suitable for direct use in conventional steam turbines[1]. At these unique sites the only treatment required before the steam is directed to the turbines is the removal of solid debris and particulates. At other locations, the low quality of the resource at the wellhead is destructive to bladed steam turbines and requires some sort of system to condition the steam or transfer the energy to another medium.

Techniques used with low quality wells include separation of the liquid phase from the steam, flashing of the steam to a lower pressure and higher quality before separation and the use of a binary cycle with a more thermodynamically suitable working fluid such as isobutane. All of these methods involve a loss of some of the available energy in the steam at the wellhead. Flashing the steam increases its entropy and decreases the available energy. There is also a substantial amount of energy lost in the liquid phase discarded at the separator. In the binary cycle, there is an increase in entropy intrinsically associated with the complexity of such a system. Heat transfer across a non-zero temperature difference and flow losses decrease the available energy. These losses of available energy and the cost involved in these complex conventional systems are the prime motivations for developing engines to handle total two-phase flow from the wellhead.

The most promising of the total flow machines is the Lysholm engine or helical screw expander[Figure 1]. The Lysholm machine consists of a pair of parallel rotors with meshing helical lobes which, with their casing, form compression/expansion chambers. This configuration was patented in Sweden by Alf Lysholm in 1934[2]. The design has found its main use as a heavy duty air compressor for field use. Its attributes in this application

include ruggedness and reliability. The use of Lysholm's design as an engine has been suggested in several applications including use in geothermal energy conversion. In 1973, R. Sprankle patented the use of the Lysholm engine with wells having a quality of less than 15%[3]. Testing of the Lysholm engine has since been conducted by Sprankle, by the Lawrence Livermore Laboratory[4], and currently by the Department of Mechanical Engineering at U.C. Berkeley[5].

In use as an engine, the Lysholm machine accepts pressurized fluid into the chambers as they are formed by the lobes of the meshing rotors. The chambers then expand as they move down the mesh line, extracting energy from the fluid. The fluid is exhausted when the chamber reaches the end of the rotor. The theoretical volumetric expansion ratio is determined by engine geometry. Expansion ratios of up to around 5 are considered practical[6]. The pressure distribution through the engine is dependent on the expansion process. The pressure in the exhaust line, however, is dependent on the characteristics of the other parts of the system. Thus, the pressure in a chamber just before it opens to the exhaust port is not necessarily the same as that in the exhaust line. If the pressure in the chamber is less than the exhaust pressure, power is required to exhaust the fluid. If the chamber pressure is higher than the exhaust pressure, there is an expansion of fluid without work done on the engine. The power output and efficiency are thus heavily dependent on the exhaust pressure imposed on the engine.

The expansion process occuring in each chamber is very efficient. There is, however, a large amount of leakage inherently present in the configuration. The leakage can be divided into two main catagories. There is leakage from chamber to chamber between the rotors and casing and through a "blowhole" formed by the imperfect meshing of the rotors. There is also leakage through the mesh line into the exhausting chambers. The main inefficiency in the engine is due to the isenthalpic expansion during leakage. Because leakage fraction is dependent on engine speed and steam quality, efficiency and power output

are dependent on these parameters as well as on pressure ratio.

The Lysholm engine has been proposed for use in a hybrid system with conventional steam turbines. In this hybrid system, the total flow from the wellhead is expanded in the Lysholm engine, exhausting at some intermediate pressure. The fluid is then directed to a separator where the liquid phase is discarded. The steam phase is then expanded in conventional impulse turbines exhausting at subatmospheric pressures. The turbine exhaust must then be condensed and pumped to atmospheric pressure.

This hybrid system represents a trade-off between the advantages of the total-flow concept and the practical limitations considered when operating with large pressure drops. Because the Lysholm engine efficiency drops off at higher pressure ratios, more than one stage would be required to utilize the pressure drop available at most sites. Much of the fluid flashes to vapor during the Lysholm expansion process, resulting in an increase in volume of the fluid. Since the Lysholm engine is a positive displacement device, increased volumetric flow requires larger engines. The size and cost of larger engines required by the increased volume for each sucessive stage would be prohibitive. A conventional steam turbine is thus being considered for the bottom stage of a hybrid cycle. Such a system is currently being tested and evaluated at U.C. Berkeley.

To best demonstrate the feasibility of a hybrid system, the maximum power from the system for a given resource must be obtained. Optimization of the hybrid system performance requires detailed knowledge of the operating characteristics of all the system components, especially the Lysholm engine. Because the flow through the turbine nozzles is choked, the interstage pressure is dependent on the total mass flow rate through the Lysholm, the Lysholm exhaust quality and the characteristics of the separator. Of course, the system performance is also dependent on the power output and efficiency of the Lysholm engine. Unfortunately, the least well known of the components of the system is the Lysholm engine. It is the purpose of this

investigation to define the Lysholm engine operating characteristics and develop a model which can be used in the prediction of the optimum operating conditions for the hybrid system.

Several types of engine model are possible. A model based on conventional dimensionless parameters, a simulation model considering the physical processes inside the engine and a "black box" empirical model were all considered. To limit the scope of this project to a manageable level, because sufficient knowledge about the engines physical processes and important dimensionless parameters is lacking and because the empirical model is easiest to implement, the empirical model was chosen for initial use in the hybrid system optimization.

To obtain the data required for construction of the model, a program of engine testing in an operating domain including the range expected to be used in the hybrid system had to be undertaken. Data control methods had to be developed to allow the taking of test data with operating conditions controlled to a discrete matrix suitable for the empirical curve fit. A computer program was required to manage the huge task of organizing, storing and reducing data for approximately 100 data points. As a part of this computer program, a steam table subroutine had to be developed for evaluation of thermodynamic properties.

Once a complete set of data is obtained, curve fitting methods in several parameters can be applied to obtain algebraic expressions for system performance in terms of operating conditions. Expressions must be obtained for power, efficiency, mass flow rate and exhaust quality in terms of pressure ratio, inlet quality and engine speed for the maximum obtainable inlet pressure of 120 psia. These expressions must then be converted to the form of a computer code subroutine for implementation in the hybrid model.

The model should be tested for validity by predicting performance under a new set of operating conditions, then testing the engine under these conditions. Points within the current

data matrix can be predicted to test the interpolative action of the model. Points outside the test data matrix can be predicted to determine the extrapolative range of the model.

System Description

The Lysholm engine being tested at U.C. Berkeley is set up to allow power generation in simulation of conditions at a geothermal site[Figure 2]. Conversion to hybrid operation requires no modifications to the system. Instrumentation of the system allows determination of performance parameters and safe operation of the engine.

Saturated steam at approximately 120 psia is drawn from the campus heating system and directed to the Lysholm inlet. The steam supply is also used to heat pressurized industrial water in an eight pass shell and tube type heat exchanger. The flow rates of both water and steam are controlled by pneumatic control valves. Almost any quality of steam can be produced by the mixture of the two flows. The water is injected into the steam just before the engine inlet. An injector allowing the use of variable nozzle size has recently been installed to allow investigation of the effects of different hold-up ratios.

The Lysholm engine is mounted with the rotor axes horizontal and the inlet and exhaust lines pointed downward. Life support systems for the engine include an oil supply system for the journal bearings, an air seal and a water seal. An adjustable load on the engine is provided by an eddy current dynamometer (brake) powered by a DC power supply. The shaft power from the engine is transmitted to the brake by a 3 to 1 reduction Gilmer belt.

The exhaust from the engine is directed through piping to a shell and tube type condenser. The condensed fluid is pumped from the condenser by a centrifugal pump. Subatmospheric exhaust pressures can be maintained by the pump and a steam powered air ejector system used to remove non-condensible gases from the condenser.

When the Lysholm engine is used as the first stage of the hybrid system, its exhaust is directed to a New Zealand type centrifugal separator. The liquid phase is discarded from the

separator. The resulting steam is directed to two impulse turbines which form the second stage of the hybrid system. The exhaust from the turbines is directed to the condenser.

The Lysholm power system is fully instrumented to provide performance data for the Lysholm engine, to monitor the health of the system and to help in controlling the system. The primary instrumentation allows determination of operating conditions and calculation of performance parameters for the engine. A calorimeter on the steam supply line allows determination of the steam enthalpy. A thermocouple on the pressurized, heated water line allows an accurate estimate of the hot water enthalpy. The flow rates of water and steam are determined by measuring the differential pressure across square edged orifice plates with flange taps. A second, interchangeable, orifice plate for the water line was recently designed and manufactured. This plate, having a smaller orifice, allows more precise measurement of the low water flow rates in high quality mixtures. A thermocouple measures the saturation temperature of the steam at the engine inlet to fix the state of the simulated geothermal fluid. A thermocouple in the exhaust line measures the saturation temperatures of the exhausting fluid. This allows determination of the exit state for ideal isentropic expansion in the engine.

The power produced by the engine is measured by the combination of a tachometer and torque sensor. The tachometer consists of a 60-tooth gear on the brake shaft and a magnetic pickup. The torque sensor uses resistance wire strain gages on the shaft with rotary transformers to transmit signals to and from the shaft.

All the primary instrumentation is connected to an Autodata Nine located in the control room[Figure 3]. The thermocouples and power measurement devices send electrical signals to the control room. The flow measurement signals are transmitted pneumatically. The Autodata scans all channels simultaneously, allowing quick accurate measurement and recording of all data when the desired operating conditions are reached. Other data

scanned and recorded by the Autodata during Lysholm engine testing include the oil return temperature, the supply steam line temperature, the condensate flow rate, the cooling water flow rate and the cooling water temperature.

Other remote instrumentation located on the control panel[Figure 3] helps to monitor the system. Pressure gages allow monitoring of the inlet and exhaust conditions independent of the Autodata. Other redundant analog gages indicate flow rate and ratio, engine speed and brake torque. Gages also exist for the monitoring of oil supply pressure, control air supply pressure, air ejector control air pressure and condensate pump pressure.

Calibration of the primary instruments was maintained to insure the accuracy of the data. The installation of a new signal filter before the current series of tests necessitated recalibration of the flow meters. A correlation between weighed mass flow rates and the flow signals was developed for both flow meters. The details of these correlations and the calibration constants are listed in the Appendix. The tachometer and torque meter were both checked to insure that earlier calibrations were still valid and that no corrections to the signals were required. The thermocouples were checked and replaced whenever errant readings were observed.

Testing Limitations and Requirements

Engine testing was required over a wide range of operating conditions, including the conditions expected for use in the hybrid system. The limiting conditions for engine operation are determined by the steam supply, the capacity of the exhaust piping, the capacity of the condensate pump and safety considerations.

The campus heating system maintains a steam pressure of approximately 125 psia. This pressure varies with the heating load on the system and can drop as low as 120 psia. The pressure which can be maintained at the engine inlet varies with and is somewhat lower than the supply pressure. The highest inlet pressure which could be consistently maintained from day to day was 120 psia.

Exhaust pressures were limited by choking of the flow in the exhaust piping. The minimum exhaust pressure varied with mass flow rate and thus with engine speed and inlet quality. This pressure was, in some cases, above atmospheric. In the middle of this series of engine tests, the exhaust piping was modified to provide less flow resistance between the engine and separator during staging. This change increased the tortuousness of the exhaust flow path for the Lysholm engine opertating alone, increasing the minimum exhaust pressure. At the same time, a new technique was initiated to reduce exhaust pressures. By closing the separator drain and letting the exhaust flow in parallel through the separator and separator bypass, the flow area can be increased and exhaust pressures reduced. This technique allows approximately the same minimum exhaust pressure as before the piping change.

Another limitation on the testing range is the capacity of the condensate pump. The maximum total flow rate which can be handled by the pump is about 2.3 lbm/sec. Engine testing at high speeds and low qualities is made difficult by flooding of the condenser when this limit is exceeded.

Because the campus heating steam has a quality of approximately 99% at the main supply valve, inlet conditions are limited to 99% quality at 120 psia. The inlet quality can be varied from pure hot water to approximately 99% quality by adjusting the mixture ratio of water and steam.

The engine speed has been limited to 9000 RPM. This is the highest speed with which the research group at Berkeley has experience. The main concern about going to higher speed has been the reliability of the Gilmer belt. The power/speed specification of the belt is being exceeded at 9000 RPM. Failure of the Gilmer belt would result in runaway of the engine until the steam supply could be shut-off. The engine will be tested at higher speed, but only after a tension test has been performed on the Gilmer belt and after all essential testing has been done on the engine.

The goal of hybrid system testing is to achieve maximum power and maximum efficiency from the system. Maximum power is attained at the highest possible system inlet pressure. Also, it has been noted in previous tests that maximum efficiency for Lysholm engine is attained at the highest inlet pressures. For these reasons and to limit the number of tests required for the model, the series of tests this report is concerned with were all run with the maximum inlet pressure of 120 psia.

The range of testing of the hybrid system is expected to include the full range of inlet qualities to simulate all possible wellhead conditions. Only the operating conditions at higher speeds will be examined, as it has been shown that engine efficiency is higher at higher speeds due to decreased leakage fraction.

Because the flow through the turbine nozzles is choked, the interstage pressures(including the Lysholm exhaust pressure) are dependent only on the mass flow rate through the turbines and the characteristics of the separator. It has been shown that the limited nozzle area in the turbines and the tortuous piping

between the Lysholm and the separator will result in Lysholm exhaust pressures greater than 10 psig. Testing of the engine included exhaust pressures above 10 psig as well as lower pressures to locate efficiency peaks.

Data Matrix

Engine testing was performed at the maximum consistently attainable inlet pressure of 120 psia including speeds from 4500 RPM to 9000 RPM, inlet qualities from 0.17 to 0.99 and over a range of exhaust pressures from the lowest attainable to over 35 psia. The data points obtained formed a three dimensional matrix in speed, inlet quality and exhaust pressure. Obtaining test data in matrix form facilitated correlation of the data and allowed empirical relations for performance parameters to be obtained.

The matrix consists of sets of data points obtained at each of the four speeds: 4500, 6000, 7500 and 9000 RPM. Subsets of data points were taken at each of 5 inlet qualities for each of the four speeds. Water/steam flow ratios of 0/1, 1/1, 2/1, 3/1 and 5/1 corresponding to inlet qualities of 0.99, 0.50, 0.33, 0.25 and 0.17, respectively, were used. Each of these 20 subsets of data points consisted of tests run with identical conditions except for varying exhaust pressure. Thus, a subset of data points with varying exhaust pressure was obtained for each combination of speed and inlet quality.

An attempt was made to obtain the same set of exhaust pressures for each combination of speed and inlet quality. This was only partially successful for two reasons. First, the limitations on exhaust pressure due to the exhaust piping allowed much lower exhaust pressures for some combinations of speed and inlet quality. Second, system control problems made the attaining of high exhaust pressures very difficult. To save time, some data from preliminary hybrid system tests with uncontrolled exhaust pressures was used to fill in the subsets of data points.

The construction of this data matrix was important in correlating the data. It allows the examination of variations in performance parameters with one variable while the others are held constant. The degradation of the data matrix due to inconsistent exhaust pressures limits these correlations to some

extent.

More importantly, a matrix makes model construction by curve fitting in three variables possible. Empirical relationships can be obtained by conventional curve fitting techniques between a given performance parameter and one variable with the other variables held constant. The coefficients from these curve fits can then be plotted against another variable. An expression for the coefficients in terms of this second variable is also obtained by curve fitting. This process can be repeated for other variables to obtain an expression for a given performance parameter in all variables. The degradation of the data matrix reduces the flexibility of this model construction method. Though the method is still workable, the exhaust pressure must be the first variable considered.

Testing Procedures

Testing procedures to improve consistency of data included a prescribed engine warm-up period and checks for residual readings on the flow meters. Data acceptance criteria were followed to insure the quality of the data matrix. In-lab data control calculation methods were developed to aid in evaluating the data. Actual control of the engine was achieved by varying the engine load, the exhaust pressure, the flow rate and the flow ratio.

During start-up, steam is first run through the system to warm the lines (refer to engine start-up check list in the Appendix). At some time during this warm-up, the steam is shut off to insure that the flow meters go to zero. The meters were also checked for zeroes at shutdown. A persistent problem during testing was a residual reading on the steam flow meter upon shutdown. It is thought that non-condensible gases(air) were building up in the low side seal pot. This caused a lowering of the water level in this seal pot during testing, resulting in a level differential of several inches between the seal pots and a residual reading of 5-10% of full scale. Because the flow rate is proportional to the square root of the pressure drop across the orifice plate, this residual translates into only a 1-2% error under normal operating conditions. It was, however, decided to bleed steam from the top of the seal pots to eliminate this problem. Though the bleeding of steam did no doubt eliminate the build-up of non-condensible gases, it caused another problem. The steam bleeding through the low side seal pot was superheated due to the pressure drop across the orifice plate. This superheated steam did not allow normal condensation, causing a fluctuation in the water level in the seal pot. The residual reading from this fluctuation was even more severe than the original residual from the non-condensible gases. The problem was finally solved by the manufacture of water cooling jackets for the lines between the orifice plates and the seal pots. This cooled the bleeding steam to the point where enough would condense to maintain a level in the seal pots.

Other procedures to insure data consistency included an engine warm-up specification and consistent air and water seal operation. The oil return temperature from the engine was always allowed to reach 140 ^{o}F before data was taken. The oil pressure, adjusted by manipuatiion of a bypass valve, was always maintained at approximately 9 psig. An air seal pressure of 40 psig was maintained. The operation of the water seal depended on testing conditions. If only pure steam was used to produce an inlet quality of 99%, the water seal was operated. If any hot water was mixed with the steam to produce other qualities, the water seals were turned off.

Control of operating conditions for the engine was accomplished by varying the load, flow rate, flow ratio, and exhaust pressure. Since flow rate, flow ratio and exhaust pressure are dependent on other operating conditions, automatic control of these variables was required. Load on the engine is independent of other variables, allowing manual control.

Varying the load on the engine was accomplished by varying the current applied to the eddy current brake. Increasing the load increased the inlet pressure while decreasing the speed. If uncontrolled, the exhaust pressure also increased with an increase in load.

The steam flow rate was maintained at a given setpoint by a pneumatic PI-type controller. Increasing the mass flow rate increased both speed and inlet pressure as well as uncontrolled exhaust pressure.

The flow ratio, the ratio between water and steam flow rates, was maintained by a pneumatic PI-type controller working in conjunction with the steam flow controller. Because the controller uses the uncorrected flow rate signals, the controller setpoint corrosponds to different flow ratios at different flow levels. This means that adjustment of the setpoint for changing conditions was necessary.

The domain of testing included water flow rates ranging from less than 0.5 lbm/sec to almost 2 lbm/sec. Precise control of flow is difficult over so wide a range. The valve used, being big enough to handle the large flow rates, cannot precisely control the flow at low flow rates. This resulted not only in a direct decrease in precision of control, but also in a lack of stability of the control system with the usual gain settings. At low flow ratios(1/1), the control gains had to be changed, sacrificing response time to achieve stability. This further increased the difficulty in achieving precise low flow ratios.

Exhaust pressure was controlled with a pneumatic PI-type controller actuating a steam valve on an air ejector. The air ejector removes non-condensible gases from the condenser. Decreasing exhaust pressure increases speed and decreases inlet pressure.

For high exhaust pressures the controller is not usable. Some exhaust pressures higher than those obtained when the air ejector is shut off are required for the model. To obtain these pressures, the exhaust flow must be manually controlled by adjusting the separator bypass valve. This is a slow process which limited the number of data points taken at high exhaust pressure.

Data acceptance criteria were applied to the inlet pressure, the exhaust pressure, the engine speed and the flow ratio or inlet quality. An attempt was made to obtain three consecutive scans for each data point. The data for each scan had to meet the acceptance criteria. The scans were averaged to produce the data used for each point. The practice of obtaining three scans originated in earlier tests when there was much noise in the flow signals. Three scans were then required to average out the noise. Currently, obtaining three scans simply serves to demonstrate that steady state conditions have been reached.

Inlet pressure was monitored as the saturation temperature. The resolution of the temperature measurement was 1 °F. 341 °F,

corresponding to 119.6 psia, was used as the target temperature. The acceptance range for a given scan was plus or minus 1 °F. This corresponded to a pressure range of from 118.0 to 121.2 psia. In practice, control was fairly easy and the data was usually well within these bounds. In particular, the standard deviation of the inlet pressure for the current series of tests was 0.7 psi.

Exhaust pressure was accepted according to the same standards as inlet pressure. This meant an allowable variation in temperature of plus or minus 1 °F. For the pressure range encountered, an error of 1 °F corrosponds to about 0.3 psia.

The acceptance criteria for a single scan of speed was plus or minus 50 RPM. After averaging scans, the result was usually well within this criteria. Table 1 lists the mean of the scan averages, the standard deviation and the high and low scan average for both speed and inlet quality.

The acceptance criterion for inlet quality or enthalpy was less rigorous. The main indicator of inlet quality was an indirect one, the flow ratio. In addition, even the flow ratio had to be calculated using the flow calibration constants. A data control program for flow ratio was developed and assigned as a user mode function on an HP-41C programmable calculator. This program can be found in the Appendix. Since inlet quality and enthalpy are also dependent on the water and steam enthalpies, a data control program for inlet enthalpy was also developed for days when the enthalpies deviated from their usual values. The general data acceptance criterion for a single scan was plus or minus 0.05 in flow ratio. For reasons discussed in the description of the experimental system, the lower water flow rates associated with higher qualities were both harder to control and to measure. In addition, inlet quality and inlet enthalpy are both more heavily dependent on flow ratio for higher inlet qualities. The data control for lower qualities was thus much better than for higher qualities. Of course, for the runs with pure steam having a quality of 99%, control of quality

presented no problem.

Performance Measures and Calculations

Important performance measures for the Lysholm engine include power output and efficiency. Exhaust quality and total mass flow rate are parameters also needed for optimization of the hybrid system. The description of engine performance in dimensionless terms requires the calculation of other parameters including specific size and specific speed. Calculation of most of these parameters requires knowledge of the state and properties of the steam at the inlet and exhaust. In practice, steam table interpolation is required. Manual data reduction involving the many required calculations on the large number of data points taken would have comprised an almost unmanageable task. The manual calculations are both tedious and impossible to perform without error. Steam table interpolation is particularly prone to innaccuracy and error. The organization and storage of raw data is also a difficult task to perform manually. To handle these tasks, FORTRAN computer codes were developed. The codes were run on a PDP-11/60 minicomputer using the RSX-11M operating system. As a part of the codes, a steam table subroutine was constructed and tested. Listings of programs described in this section can be found in the Appendix.

The raw data necessary for all calculations are the uncorrected flow signals, the hot water and calorimeter temperatures, the inlet and exhaust temperatures, and the speed and torque signals. Other information requiring storage for each data point include the run number, the run date, the barometric pressure, a comment code, and a calibration code specifying the appropriate flow calibration constants for determination of the flow rates.

The data values from all scans were averaged for each run, prior to storage in the computer. The raw data entered and stored is in the same form as the output from the Autodata. For the Autodata channel usage and output format, refer to the Appendix.

The computer code utilizes a direct access file for data storage. Each numbered record line in the file holds the raw data for a single point. The data are organized into a three dimensional matrix corresponding to the matrix used for data acquisition. The dimensions of the matrix are flow ratio, speed and exhaust pressure. When accepting data, the computer prompts for a 3-digit code which explicitly specifies the position along each of the three axes of the matrix. The three digits are then converted into a one dimensional array element number corresponding to the number of a record line in the file.

When reduction or output of data is desired, a point or set of points can be obtained using the same 3-digit code. Organization of data in this manner makes the program very versatile in the creation of data files for plotting or analysis. Two versions of the program were used to produce different types of reduced output. One program produces output files with headers and peripheral information such as run number and run date. The other program produces output files in a bare form suitable for use as source files for plotting routines.

Another important feature of the computer codes is a steam table subroutine linked to the main program for determination of steam states. The subroutine is essentially a look-up table with logarithmic interpolation routines for determination of properties inside a skeleton of steam data points. Steam data was obtained from the 1967 ASME Steam Tables[7]. The current range of the table includes the region inside the vapor dome extending out to 70 ^{o}F superheat. The upper and lower pressure limits are 0.95 psia and 153 psia, respectively. The table was originally designed to accept an input state specified by pressure or temperature and one other property including specific volume, enthalpy, entropy or quality. A state specified by pressure and temperature is allowed in the superheated region. The capability for specification of state by specific volume and entropy was later appended for future use in simulation modeling. The state specifying properties are passed as arguments to the

subroutine. The properties used in the state specification are labeled by the use of logical variables passed through a common statement. The output properties are passed as arguments to the main program.

Parameters calculated by the program are inlet and exhaust pressure, pressure ratio, total mass flow rate, inlet enthalpy, inlet quality, power, efficiency, leakage rate, exhaust quality, specific size and specific speed. Other dimensionless parameters potentially useful in modeling include leakage ratio, power specific size, power specific speed and specific power.

The individual mass flow rates are calculated using calibration constants peculiar to the calibration period and orifice plates used for a given run. The total flow rate is simply the sum of the hot water and steam flow rates.

The inlet enthalpy is obtained from the mass flow rates and the enthalpies of the hot water and steam. The hot water enthalpy is estimated using the expression for an idealized incompressible fluid with constant specific heat[8].

$$h_2 - h_1 = c_p(T_2 - T_1) + v(P_2 - P_1)$$

Since the specific volume(v) is very small, the enthalpy can be accurately estimated by taking $T = T_{sat}$. Then:

$$h_w = h_f(T_w) \quad \text{[Steam Table Call]}$$

Where: h_w is the hot water enthalpy

h_f is the enthalpy of a saturated liquid

T_w is the water temperature

The steam enthalpy is measured by flashing the steam to atmospheric pressure in a calorimeter. The state of the throttled steam is defined by the calorimeter temperature and atmospheric pressure. The enthalpy is assumed constant for the throttling process in the calorimeter. The steam enthalpy is then given by:

$$h_s = h_c = h(T_c, P_{atm}) \quad \text{[Steam Table Call]}$$

Where: h_s is the steam enthalpy

h_c is the enthalpy of the throttled steam

T_c is the calorimeter temperature

P_{atm} is the atmospheric pressure.

The inlet enthalpy is then given by:

$$h_1 = (m_w h_w + m_s h_s) / (m_w + m_s)$$

Where: h_1 is the inlet enthalpy

m_w is the hot water flow rate

m_s is the steam flow rate

The inlet state is defined by the inlet enthalpy and temperature. The properties at the inlet are determined using the steam table:

$$x_1 = x(T_1, h_1) \quad \text{[Steam Table Call]}$$

$$s_1 = s(T_1, h_1) \quad \text{[Steam Table Call]}$$

$$v_1 = v(T_1, h_1) \quad \text{[Steam Table Call]}$$

Where: x_1 is the inlet quality

s_1 is the inlet entropy

v_1 is the inlet specific volume

T_1 is the inlet temperature

Inlet pressure, exhaust pressure and pressure ratio are also obtained using the steam table:

$$P_1 = P(T_1) \quad \text{[Steam Table Call]}$$

$$P_2 = P(T_2) \quad \text{[Steam Table Call]}$$

$$\text{PRESSURE RATIO} = P_1 / P_2$$

Where: P_1 is the inlet pressure

P_2 is the exhaust pressure

The leakage rate from the inlet chamber is determined by subtracting the theoretical mass flow rate, assuming no leakage, from the total mass flow rate. The theoretical flow rate is a function of engine geometry, engine speed and inlet density. Gardner-Denver, the manufacturer of the engine, has stated that the intake volume is 13.8 cubic inches per revolution[9]. The theoretical volume flow rate is then given by:

$$V_f = V_i N$$

Where: V_f is the volume flow rate

V_i is the intake volume per revolution

N is the engine speed

The leakage rate from the intake chamber while the inlet is still open is then:

$$m_l = m_t - (V_f / v_1)$$

Where: m_l is the leakage rate

m_t is the total flow rate

Power is calculated as the product of torque and speed. Determination of efficiency, specific speed and specific size require calculation of the isentropic enthalpy drop, which represents the energy available from the steam for a given pressure drop. Insentropic expansion of the steam implies a reversible adiabatic process.

The isentropic enthalpy drop can be determined from the inlet state and the exhaust temperature or pressure. The inlet entropy is used to fix the isentropic exhaust state.

$$h_{2s} = h(T_2, s_1) \quad \text{[Steam Table Call]}$$

$$dh_s = h_1 - h_{2s}$$

Where: h_{2s} is the exhaust enthalpy for isentropic expansion

T_2 is the exhaust temperature

dh_s is the isentropic enthalpy drop

The isentropic efficiency is then given as:

$$\text{EFFICIENCY} = \text{POWER} / (m_t dh_s)$$

The exhaust state can not be determined directly from the available data. The exhaust state can, however, be approximated by assuming an adiabatic engine. The exhaust enthalpy is then determined through conservation of energy:

$$h_2 = h_1 - \text{POWER}$$

The exhaust quality, as well as other exhaust properties, can then be estimated using the steam table:

$$x_2 = x(T_2, h_2) \quad \text{[Steam Table Call]}$$

The specific speed (n_s) and specific size(d_s) are traditionally calculated as:

$$n_s = (N(m_t v_{2s})^{1/2}) / dh_s^{3/4}$$

$$d_s = (D dh_s^{1/4}) / (m_t v_{2s})^{1/2}$$

Where: N is the engine speed [RPM]

m_t is the total mass flow rate [lbm/sec]

v_{2s} is the isentropic exhaust volume [cubic ft/lbm]

dh_s is the isentropic enthalpy drop [ft-lbf/lbm]

D is the rotor diameter [ft]

The specific speed and specific size as calculated have units of $RPM(lbm\text{-}ft/lbf)^{3/4}/sec^{1/2}$ and $(lbf/(lbm\text{-}ft/sec^2))^{1/4}$, respectively. These parameters can be made unitless by multiplication by unit constants:

$$N_s = n_s(7.751E\text{-}3(rad\text{-}min/rev\text{-}sec)(lbf/(lbm\text{-}ft/sec^2)^{3/4})$$

$$D_s = d_s(2.382((lbm\text{-}ft/sec^2)/lbf)^{1/4})$$

There are six dimensional variables currently being considered in dimensional analysis:

Power

Total Mass Flow Rate

Isentropic Enthalpy Drop(Head)

Engine Speed

Engine Size(Rotor Diameter)

Isentropic Exhaust Volume

These six dimensional variables are described by three fundamental dimensions: mass, length and time. The Buckingham-Pi Theorem[10] states that there are then 6 - 3 = 3 independent dimensionless groupings of the dimensional variables. Along with efficiency, the specific speed and specific size comprise one set of 3 independent dimensionless groups.

Other possible dimensionless groups include:

the specific power

$$P^* = (POWER)(v_{2s}) / (N^3D^5)$$

the dimensionless head

$$H^* = dh_s / (N^2D^2)$$

the dimensionless flow

$$F^* = (m_t v_{2s}) / (ND^3)$$

the specific momentum

$$M^* = POWER / (NDm_t dh_s^{1/2})$$

Also, it has been suggested that for power devices, the power specific speed(N_{sp}) and power specific size(D_{sp}) may be

useful[11]:

$$N_{sp} = N((POWER)(v_{2s}))^{1/2} / dh_s^{5/4}$$

$$D_{sp} = dh_s^{1/2} / ((POWER)(v_{2s})(N^2))^{1/5}$$

These additional dimensionless groups are not independent of the first three mentioned. They can be simply derived from proper combinations of any other three independent groups. It should be noted that an entirely new set of dimensionless groups could be obtained by substituting the inlet specific volume for the isentropic exhaust specific volume.

Modeling

Three types of model were considered to describe the performance of the Lysholm engine. A model describing the engine in terms of dimensionless parameters was attempted. It was hoped that a dimensionless model, through similarity, could reduce the number of parameters needed to describe the operating conditions of the engine and fill in the areas of the performance map for which data were not obtained. A simple model with three dimensionless parameters could not, however, consistently describe engine performance. It appears that another, as yet unused, dimensional parameter producing an additional independent dimensionless group is needed to describe engine performance. A fifth dimensionless group, relating to leakage area, would also be required to extend predictions to scaled-up engines. Another type of model is a computer simulation. Computer simulation would attempt to model the physical processes occurring inside the engine. This approach has both the most promise for immediately extending results to scaled-up engines and the most potential for assisting in design improvements. Computer simulation would, however, ba a difficult open-ended project beyond the scope of this investigation. The model finally chosen uses a simple empirical "black box" approach. This approach directly relates the performance parameters of the engine to the operating conditions through purely empirical relations obtained by curve fitting. This approach is easiest to impliment and use. It is, however, of limited use in improving the engine design or extending results to scaled-up engines.

The attempt at dimensionless modeling involved a three dimensional plot of efficiency, specific speed and specific size. The plot included an earlier matrix of data points having constant exhaust pressure and varying inlet pressure. When the current data matrix of points with constant inlet pressure and varying exhaust pressure was added, inconsistencies resulted in the plot. Points with the same combination of specific size and specific speed had very different efficiencies. In particular, higher absolute inlet and exhaust pressures resulted in much

higher efficiencies than similar points with lower pressures. This indicated that there is a fourth independent dimensionless parameter needed to describe engine performance.

The Buckingham-Pi theorem[10] states that another independent dimensionless group requires another dimensional parameter describing the engine operating condition. It is not clear at this time what the additional parameter is.

One possibility is the inlet specific volume. Currently, the isentropic exhaust specific volume and the isentropic enthalpy drop are the only two dimensional parameters used for the dimensionless grouping which describe the state of the working fluid. These two parameters alone are not enough to define explicitly the condition of the steam at any point in the engine. In addition, the choice of the exhaust volume for use in the dimensionless groups may have been an unfortunate one. This choice is based on the traditional approach resulting from the greater importance of the exhaust state in turbine performance[12]. In turbines, the nature of the exhaust velocity triangles is determined by the exhaust specific volume. These velocity triangles directly affect the efficiency of a turbine by defining the magnitude of the leaving losses. In impulse type turbines, the inlet velocity triangle is also determined by the exhaust specific volume. It is not clear, however, that the exhaust state is as important in a positve displacement machine such as the Lysholm engine. In many ways, the Lysholm engine is similar to a reciprocating type engine for which inlet density is very important.

One approach to dimensional analysis might be to add the inlet specific volume, or another inlet property as a dimensional parameter to specify the inlet state. One more independent dimensionless group could then be obtained for correlation of the data.

The discovery of a fourth dimensionless group to characterize the performance of the Lysholm engine would still be inadequate to extend results to other Lysholm engines of

different size. Performance of the engine is heavily dependent on the leakage occuring in the engine. The leakage is dependent on the leakage area defined by the engine clearances. Relative leakage areas will undoubtedly vary in engines of different sizes. A fifth dimensionless parameter involving the engine geometry will be required to predict performance of scaled-up engines. One such possible parameter is described by Balje[12] as the ratio of tip clearance to rotor diameter. The tip clearance would then become an additional dimensional parameter used to describe the operating condition of the engine. Testing of at least one other engine would be required to characterize performance in terms of a dimensionless group involving leakage.

Another problem with dimensional analysis is that determination of the operating condition in terms of specific size and specific speed requires knowedge of one of the dependent dimensional parameters as tested. The exhaust pressure is controlled automatically, as is the flow ratio. Several approaches to the fixing of the other properties are possible. In one approach, the mass flow rate is fixed. The load is then used to control either the inlet pressure or the speed. In this case, the dependent parameters needed to define specific speed and specific size are isentropic enthalpy drop, defined by the inlet pressure, or speed. The method actually used involves adjusting both the mass flow rate and load to fix speed and inlet pressure. In this case, total mass flow rate becomes a dependent variable needed for specific speed and specific size. It is thus difficult to produce a data matrix in specific size and specific speed. Such a data matrix is almost essential in determining relationships between dimensionless groups for a computer model.

The dependent variables needed for the dimensionless parameters also complicate the combination of the Lysholm model with models of other components in optimization of a hybrid system. The solutions are not as explicit, requiring iterations to match components.

A simulation model would be an attempt to explain the engine performance in terms of the physical processes occurring inside the engine. One promising approach would follow the state of the steam in the chambers as they expand down the mesh line. The idealized isentropic expansion of the fluid in the chambers, the leakage from chamber to chamber and into the exhaust, the acceleration of the fluid as it moves into the engine, and the frictional heating of fluid resulting from drag are among the processes to be considered. The rotation of the rotors and expansion of the chambers is simulated incrementally. The effects of all processes involving the fluid in a given chamber at the end of the increment are combined using the principles of conservation of mass and energy to fix the resulting state of the fluid. This becomes the initial condition for the next increment. A stable simulation, starting with some arbitrary set of intermediate chamber conditions, would reach some set of equilibrium intermediate chamber conditions after the model has run for a time. The exhaust state is thus predicted, allowing determination of the exhaust enthalpy and exhaust quality. The enthalpy change and any heat transfer from the engine could be combined to predict the power extracted from the engine. The mass flow rate, predicted from simulation of the inlet, could then be used to predict efficiency.

The simualtion model could predict improvements in performance resulting from design changes affecting the engine processes without further testing. This model could also be applied immediately in simulation of larger engines, predicting the effects of scale.

Unfortunately, a simulation model as described is beyond the scope of this investigation. The constuction of this model would be a very difficult and extensive task. Since not all leakage in the engine is choked flow, closed form solutions for the state in any given chamber are not possible. Another problem with the simulation approach is the difficulty in obtaining information about the physical processes and the geometry inside the engine.

Instrumenting all the internal processes is not possible at this time. This lack of information results in many free parameters in the engine description. Free parameters are those which are uncertain and must be adjusted to fit the performance data. The predictive value of the model is inversely proportional to the number of free parameters.

The model finally chosen uses a simple empirical "black box" approach. This type of model, like the dimensional model, is based on the performance data obtained. Knowledge of the processes involved is used only to determine the form of the relationships between the operating characteristics and performance parameters. The black box appraoch expresses each of the performance parameters in terms of all of the operating conditions through empirical relationships.

The predictive ability of this model in extrapolation is very limited. The ability of the empirical model to predict the effects of scaling is also very poor.

This approach is, however, relatively easy to implement. Also, the resulting model is easy to use in hybrid system optimization.

Test Results

Engine testing to fill the desired data matrix produced 104 data points having an inlet pressure of 120 psia. The data were reduced using the computer codes developed for this project. Trends in mass flow rate, power, efficiency and exhaust quality were examined. As expected, mass flow rate increased with engine speed and decreased with increasing exhaust quality. The flow rate did not vary much with exhaust pressure, dropping off only slightly at high exhaust pressures. Power increased with pressure ratio, approaching an asymptote at very high pressure ratios. The power also increased with speed, leveling off slightly at the highest speed. Efficiency data reflected the combined effects of power and mass flow rate. A peak was observed in efficiency with varying pressure ratio. The efficiency also appeared to approach a peak with increasing engine speed and decreasing inlet quality. The exhaust quality was most heavily dependent on inlet quality and independent of speed. The variation in exhaust quality with pressure ratio was dependent on the inlet quality.

The test results are tabulated in Tables 2a-2e. Included in the listing is a specification of the operating conditions for each data point and the calculated performance parameters. Also listed are the run date, run number, calibration code, barometric pressure and comment code. A listing of the meanings of the comment codes is presented in Table 3.

Mass flow rate for all data points is plotted in Figures 4a-4e. The most important trend apparent in these plots is the lack of variation in mass flow rate with pressure ratio.

Mass flow rate can be divided into two parts, the theoretical flow rate and the leakage rate. Both of these flow rates are determined entirely at the inlet. The theoretical flow rate, the flow rate with no leakage, is dependent entirely on the volume of the inlet chamber upon closing, the speed and the inlet quality. It is independent of exhaust pressure. The leakage rate, the difference between the total flow rate and the

theoretical flow rate, is the amount of fluid leaking out of the inlet while the inlet is open. Since the theoretical flow rate is entirely independent of exhaust pressure, any variation in total flow rate with exhaust pressure is reflected in the leakage flow rate.

The variation in leakage rate with exhaust pressure is thus minimal. The total flow rate and leakage rate are constant at higher pressure ratios. There is only a slight drop off at lower pressure ratios.

There are two leakage mechanisms operating in the engine. One involves leakage from chamber to chamber around the edges of the lobes, or through a blowhole which exists where the rotor seal lines meet at the mesh line. This blowhole forms a zig-zag path from chamber to chamber down the mesh line. The second leakage mechanism is leakage from each chamber into the exhausting chambers through the meshline. The engine geometry is such that across the mesh line from each expanding chamber is a chamber open to the exhaust and having a pressure equal to the exhaust pressure. Since the chamber pressures are over twice the exhaust pressure down much of the mesh line, leakage to the exhaust is often choked flow, and thus independent of exhaust pressure.

Both leakage mechanisms could be operating in the inlet chamber while it is open. The invariance of the leakage flow with exhaust pressure at high pressure ratios suggests that the leakage out of the inlet is all choked flow at lower exhaust pressures. This implies that either the dominant leakage mechanism in the inlet chamber is leakage through the mesh line into the exhaust or that the pressure difference between the inlet and the next chamber is sufficient to result in choked flow. Tests show that pressure in the chambers located near the inlet varies with exhaust pressure, indicating that leakage from chamber to chamber is a significant mechanism. Thus, it appears that the pressure difference between the inlet and next chamber is sufficient to cause choked flow. For steam, a pressure ratio

of approximately 2/1 is required for choked flow. A pressure ratio of 2/1 between the first two chambers would be responsible for a large fraction of the torque and power generated.

The average leakage flow rate for a given speed and inlet quality is plotted versus speed in Figure 5a. There is a sharp increase in leakage flow rate as the quality decreases to zero. This can be attributed to the higher density of the leaking fluid at low qualities. For lower inlet qualities, the leakage rate decreases at higher speeds. This may be due to the need to accelerate the liquid up to the speeds at which the chambers travel down the mesh line and at which the liquid eventually travels around the casing. Characteristic velocities for the fluid and lobes are presented in Figure 6. At the inlet, the liquid can not leak through the blowhole or around the edges of the rotors until it has been accelerated up to the speed of the rotors. This theory is supported by results from earlier tests showing a slight decrease in efficiency when a smaller water nozzle, producing higher water velocities, was used. The effect of speed decreases to nil at saturated inlet conditions. As the steam velocity at the inlet is already on the order of magnitude of the rotor velocities, vapor leakage is independent of engine speed.

Though the leakage rate increases with decreasing inlet quality, the theoretical flow rate also increases with decreasing inlet quality. The leakage fraction actually decreases with decreasing inlet quality as well as with increasing speed. Figure 5b is a plot of average leakage fraction versus inlet quality. These trends will be important in the later discussion of efficiency trends.

Power versus pressure ratio is plotted in Figures 7a-7d. While the power outputs for different speeds fall into distinct groups, there appears to be no significant variation in power with inlet quality. This implies that the pressure distribution on the rotors inside the engine is independent of inlet quality. The pressure distribution is dependent on the overall nature of

the expansion process.

The Lysholm engine expansion process can be modeled by the combination of two processes. The fluid which remains inside a given chamber as it expands down the mesh line can be considered to be undergoing an isentropic expansion. It is this process which results in work being done by the fluid on the rotors and the power output from the engine. The second process is the leakage rate from the chambers into the neighboring chambers or into the exhaust. This process can be considered to be a pure isenthalpic throttling process contributing nothing to the power output of the engine.

The pressure variation during an isentropic expansion can be described by the polytropic equation[8]:

$$Pv^n = \text{constant}$$

For an ideal gas undergoing an isentropic process, n is the ratio of specific heats. For an isentropic process on steam inside the vapor dome, the polytropic exponent has no such intrinsic meaning. This exponent is, however, still usefull in describing the expansion process. K will be used to represent the polytropic exponent for a general isentropic process.

It would be helpful in this analysis to be able to examine the value of K during an expansion process characteristic of the Lysholm engine for varying inlet qualities. The steam table subroutine developed for use in data reduction was used for this purpose. An isentropic expansion from 120 psia to 14.7 psia was considered over a range of inlet qualities from 0 to 1. The value of K for this process was determined by the equation

$$K = \ln(P_1/P_2) \ / \ \ln(v_2/v_1)$$

Since K does vary slightly with pressure, this represents only an effective K for the whole process. This is a good approximation to the value of K for any point in the process. A plot of effective K versus inlet quality is presented in Figure 8. There is only a modest increase in K with inlet quality for the range of inlet qualities considered in the tests(0.17 - 1.00). This implies that for the same volume change applied by the chambers

during expansion, there is a slightly greater pressure drop during expansion for higher inlet qualities.

Leakage would tend to decrease the pressure drop during expansion by equalizing the chamber pressures. The leakage fraction is the pertinent paramenter to consider because it represents the amount of leakage relative to the amount of fluid in a chamber. The leakage fraction and thus the pressure equalization is greater for higher inlet qualities. The small variations in K and in leakage appear to negate each other producing a power output effectively independent of inlet quality.

Although power appears to be independent of inlet quality, utilization of the total flow from the wellhead is still desirable. Examination of mass flow rate data[Table 2] reveals that for a given speed, increasing inlet quality decreases the required mass flow rate of steam. Thus, for a given speed and pressure ratio, decreasing inlet quality reduces the amount of steam required to produce the same amount of power. In addition, the expansion of the water phase in the engine makes more steam available for use by the second stage of a hybrid system.

The power increase with speed is due to the increased mass flow rate. This trend starts to level off by 9000 RPM with a peak probably occurring at higher speeds. This leveling off is due to the losses associated with drag, required acceleration of the fluid and bearing friction.

The power increases with pressure ratio from an output of zero at some mimimum pressure ratio required to turn the engine with no load. The power appears to reach a maximum value at very high pressure ratios. This asymptotic behaivior can be explained by examining the effects of exhaust pressure on leakage and on the power required to expel fluid from the engine.

Ideal isentropic expansion would be independent of the back pressure. The pressure in a chamber just before it opens to the exhaust would not be affected by the exhaust pressure. When this

last chamber is opened, the pressure must equalize to the exhaust pressure. High back pressures result in fluid rushing back into the exhausting chamber, creating an adverse pressure differential across the last set of lobes. This adverse pressure differential decreases net torque and power. As the back pressure is decreased, the adverse pressure differential decreases resulting in more net power. As exhaust pressure decreases below final chamber pressure, a favorable pressure difference across the exhausting rotor lobe continues to increase power output. When leakage is considered, the pressures are equalized somewhat but the same general effects result in the observed increase in power with pressure ratio.

At higher pressure ratios the leakage throughout the engine becomes choked flow, freezing the pressure distribution inside the engine. Further reduction of exhaust pressure can only increase the pressure differential across the final lobe to increase power. This effect becomes minimal at higher pressure ratios as the pressure difference increases more slowly for increasing pressure ratio. The power thus approaches an asymptote at very high pressure ratios.

Engine efficiency is plotted versus pressure ratio in Figures 9a-9d. The efficiency results reflect the individual effects of the variations in flow rate and power. The propagation of errors in measurement of power and mass flow rate can also be seen in the scatter of the efficiency data. For any given speed and inlet quality, the efficiency increases with pressure ratio from the minimum pressure ratio required to turn the engine with no load. At low speeds and high inlet qualities, the rising part of the curve is not covered by the data taken. It is obvious, however, that the efficiency of a real engine must approach zero as the pressure ratio goes to zero because of frictional losses. As mass flow rate is relatively independent of exhaust pressure, variation of efficiency with pressure ratio reflects the effects of variations in power. The rising part of the efficiency curve reflects the reduction in back pressure with increasing pressure ratio. After reaching a peak, the efficiency

drops off at higher pressure ratios. Though the declining part of the curve is out of the testing range for low inlet qualities and high speeds, the efficiency is leveling off even for these conditions. The decrease in efficiency is due to the asymptotic behaivior of the power at high pressure ratios. The leveling off of power with increasing pressure ratios is accompanied by a continued increase in isentropic enthalpy drop, resulting in decreased efficiency.

Variations in efficiency with inlet quality are due entirely to variations in leakage fraction and isentropic enthalpy drop. As the leakage fraction decreases with decreasing inlet quality, the efficiency increases. This effect seems to be in effect down to very low inlet qualities where the efficiency begins to level off. The reason for the leveling off can be explained by the sharply increasing fluid density at low inlet qualities. At high qualities, the increase in fluid density with decreasing quality is accompanied by a roughly equivalent decrease in isentopic enthalpy drop[Figure 10a]. This results in a relatively constant amount of power available to the engine. At low qualities, a sharp increase in fluid density and mass flow rate is accompanied by only a modest decrease in isentropic enthalpy drop. The power available per volume of fluid at the inlet rises sharply at low inlet qualities with no corresponding increase in power output[Figure 10b]. This results in decreasing efficiency at very low exhaust qualities.

An increase in efficiency with engine speed is observed at all inlet qualities. This can be attributed to the decreased leakage fraction at high speeds. A leveling off of the efficiency at speeds approaching 9000 RPM is caused by the increased frictional losses and the increased power required to accelerate the fluid through the engine.

Plots of exhaust quality versus pressure ratio are presented in Figure 11. Exhaust quality does not vary much from the inlet quality and is thus most dependent on inlet quality. There is very little variation in exhaust quality with speed. The exhaust

quality increases with pressure ratio at lower inlet qualities while decreasing at an inlet quality of 0.99. These trends reflect the change in quality during an isentropic expansion as seen in a T-S diagram[Figure 12]. For low qualities, the quality increases during an isentropic expansion. For higher qualities, the same isentopic expansion results in a decrease in quality.

Curve Fitting

An empirical engine model was created by applying statistical curve fitting techniques to the test data. Methodology was developed for using such traditional curve fitting methods to obtain expressions in three or more variables. The equation forms considered were limited to transformations of the linear equation $y = a + bx$ and the parabolic equation. Explicit expressions for power and mass flow rate in terms of pressure ratio, inlet quality and engine speed at an inlet pressure of 120 psia were obtained. The model was completed by combining the predictions for mass flow rate and power with a steam table to produce predictions for efficiency and exhaust quality.

There is an extremely large number of equation forms which can be used to fit experimental data. In order to limit the time involved in the modeling process, the types of equation considered for use in modeling were limited. An attempt was made to employ only transformations of the linear equation $y = a + bx$[13, 14]. These include:

the expotential equation,

$$y = ae^{bx}$$

the logarithmic equation,

$$y = a + b \ln(x)$$

the power equation,

$$y = ax^b$$

the hyperbola forced through a point(x_1, y_1),

$$y = ((x - x_1) / (a + bx)) + y_1$$

and the parabolic equation forced through a point(x_1, y_1),

$$y = (y_1 - ax_1) + (a - bx_1)x + bx^2$$

The regression coefficients for these equations were determined using the method of least squares. To reduce the chance of error and the time involved in this extensive task, a programable calculator was employed. A statistics software pack in the form of a ROM was used with an HP-41C. A peripheral printer was used to record and check the data entries.

Because of the degradation of the data matrix in the variation of exhaust pressure, pressure ratio had to be the first variable considered in the curve fitting process. Since different values of exhaust pressure were obtained for each combination of speed and inlet quality, there are not sufficient groups of data with consistent pressure ratio to allow other variables to be considered first.

Mass flow rate reaches an asymptote at high pressure ratio for any combination of speed and inlet quality. In addition, the mass flow rate has a theortical value associated with no leakage flow at a pressure ratio of one. A hyperbola forced through this point can thus be used to describe the variation in mass flow rate with pressure ratio. The important quality of the hyperbola in this application is that it approaches an asymptote in y for large values of x. The transformed equation:

$$(PR - 1) / (m - m_t) = a + b(PR)$$

was used to obtain equations of the form:

$$m = ((PR - 1) / (a + b(PR)) + m_t$$

Where: PR is the pressure ratio

m is the mass flow rate(lbm/sec)

Here, m_t is the theoretical mass flow rate given by

$$m_t = (V_i N) / (v_f + v_{fg} x_1)$$

Where: V_i is the theoretical intake volume(ft^3-min/rev-sec)

N is the engine speed(RPM)

v_f is the specific volume of saturated liquid(120 psia)

v_{fg} is the specific volume difference across the vapor dome(120 psia)

x_1 is the inlet quality

The other term in the equation represents the leakage rate. The leakage rate goes to zero as the pressure ratio decreases to one and reaches an asymptote of 1/b as the pressure ratio becomes large.

A transformation involving ln(PR) instead of PR-1 was attempted because it appeared to have more physical meaning. This approach did not produce a good correlation and was

abandoned

The coefficients a and b could now be related to the speed and inlet quality by additional levels of curve fitting. In general, it was found that the quality of the curve fit improved if the variables with which there is the most variation in the dependent quantity are considered first. Once the constraint to use pressure ratio first had been followed, this criteria was used to select the order of the fits. Since the leakage flow rate varies more with quality than with speed, the coefficients a and b were next related to the inlet quality for a given speed.

The coefficient b was found to vary linearly with inlet quality:

$$b = a_b + b_b x_1$$

This is in agreement with the theory that the leakage rate varies with the density of the leaking fluid. The reciprocal of the leakage flow asymptote (1/b) varies linearly with the inlet quality while the specific volume or reciprocal of inlet density also varies linearly with inlet quality. The coefficient a also varies linearly with inlet quality:

$$a = a_a + b_a x_1$$

Finally, the coefficients a_a, b_a, a_b and b_b were related to speed by the equations:

$$a_a = a_{aa}\exp(b_{aa}N)$$
$$b_a = a_{ba} + b_{ba}N$$
$$a_b = a_{ab}\exp(b_{ab}N)$$
$$b_b = a_{bb} + b_{bb}N$$

Power also appears to reach an asymptote with increasing pressure ratio. In addition, power must pass through zero at some minimum pressure ratio required to turn the engine with no load. A hyperbolic equation forced through this point could then be used to describe the variation in power with pressure ratio. The point where the curve crosses the pressure ratio axis was originally estimated using the x intercepts obtained from an attempt to fit the power data to a logarithmic equation. The logarithmic equation did not level off fast enough in pressure

ratio and was rejected. The x-intercepts were later adjusted to improve the fit. The power variation with pressure ratio was thus described by equations of the form:

$$\text{Power} = (PR - PR_1) / (a + b(PR))$$

Where: PR_1 is the minimum pressure ratio required to turn the engine at a given speed

Since there appears to be no significant variation of power with inlet quality, only one more step was required in the curve fitting process for power. The coefficients a, b and PR_1 were related to the speed by the equations:

$$a = a_a + b_a N$$

$$b = a_b + b_b N + c_b N^2$$

$$PR_1 = \exp(b_c N)$$

The parabolic variation in b with speed confirms that the power reaches a peak as the speed increases. The reciprocal of asymptotic power(1/b) bottoms out at the peak in power with speed.

The correlation of the final version of the model was substantially improved by a subtle variation on the technique described. After coefficients associated with the first level of curve fitting were obtained, the coefficient a was related to the other variables as usual. Now, however, values of a were generated by the curve fit for a given speed and inlet quality. The original data were then used to force new curves through these values of a, obtaining new values of the coefficient b. Expressions for b in terms of the other variables were then obtained as before. This method could be used at all but the last level of the curve fit.

The forcing technique reduced the effect of errant sets of coefficients, a and b, having self-negating but significant variations from the rest of the coefficients. These sets of coeeficients may have best fit the particular set of data but wreaked havoc on subsequent curve fits. Values for the coefficients used in the final version of the model can be found in Table 4.

Attempts were made to obtain an explicit relationship for efficiency in terms of pressure ratio, inlet quality and speed. Since efficiency peaked with pressure ratio, a polynomial of at least second order would be required. Parabolic and cubic equations were tried with little success. Higher order equations produced good fits for a given set of data but went wild outside the limits of the data and resulted in poor correlations at the next levels of the curve fit. A satisfactory model for efficiency was finally obtained by combining the predictions for mass flow rate and power with a steam table.

Exhaust quality was easy to model directly. However, it is possible to model the exhaust quality using the same approach as with efficiency. This approach was taken to reduce the complexity of the model.

The engine model was converted into the form of a Fortran-IV subroutine. This subroutine must be linked with the steam table developed for use in data reduction. Coefficients for the model are stored as block data. The input variables for the model as well as the predicted performance parameters are passed to and from the subroutine as arguments. A listing of the subroutine is presented in the Appendix.

This subroutine was used to predict engine performance over a range of pressure ratios and for speeds and inlet qualities corresponding to the target values used in engine testing. These predictions were graphed for comparison with data points using an X-Y plotter, driven by the GDA plotting routine. The plotter was implemented by the PDP-11/60 minicomputer used for data reduction. These plots are presented as Figures 13-16.

Model Validation

The Lysholm engine model was validated by predicting performance under new operating conditions and testing the engine under these conditions. The operating conditions selected for prediction and testing included varied and untested inlet qualities and speeds, with both subatmospheric and above atmospheric exhaust pressures. As a part of the validation, the model was used to predict the operating conditions for maximum efficiency. The engine was then tested under these conditions. The model performed adequately, predicting performance with expected accuracy. Mass flow rate, power and exhaust quality were all predicted very well by the model. The efficiency predictions included the combined errors in prediction of mass flow rate and power. Efficiency predictions were somewhat disappointing, but not critical to the major intended use of the model in hybrid system optimization.

The first day of model validation involved operating points having an inlet quality of 0.20. The model was used to predict the pressure ratio for maximum efficiency at this inlet quality and speeds of 7500 and 9000 RPM. Predictions were also made for an inlet quality of 0.20, a pressure ratio of 6.0 and speeds of 4500, 6000 and 7500 RPM. The engine was tested under these conditions.

The first day of validation also included runs under conditions predicted to have the maximum engine efficiency. The engine model was employed by a searching routine to find the conditions resulting in the maximum efficiency. This routine is listed in the Appendix. Operating conditions for a maximum efficiency of 43.3% were predicted to be:

INLET QUALITY = 0.24

SPEED = 8400 RPM

PRESSURE RATIO = 6.0

The engine was tested under these conditions.

The second day of testing included operating conditions chosen to thoroughly exercise the model by covering a range of

qualities, speeds and pressure ratios. Inlet qualities ranging from 0.18 to 0.99 were included with speeds of from 5000 to 9000 RPM and exhaust pressures of from 12 to 23 psia. The models interpolative ability was tested by choosing operating conditions between data matrix conditions. For some points, all three operating conditions were off matrix. Unfortunately, the extrapolative abilities of the model could not be easily tested. Inlet qualities of much less than 0.17 are difficult to use because of limitations on the total flow rate imposed by the condensate pump. The full range of attainable pressure ratios has already been covered by the data matrix. Only engine speed can be increased outside the data matrix range. Testing at higher speeds was not performed at this time to avoid endangering the engine.

Data acceptance criteria used for validation testing were the same as those used for the data matrix. Actually, only the inlet pressure was really critical in validation as the model was based on a single inlet pressure of 120 psia.

The results of the model validation are presented in Table 5. The model performed very well in predicting the performance of the Lysholm engine. Some parameters were predicted with better accuracy than others. The predictions for exhaust quality were excellent. Measured exhaust qualities were all within 0.4% of the predicted values. Predictions for mass flow rate were also quite good. Measured mass flow rates were all within 2.7% of the predicted values. The mean of the percentage errors in prediction of mass flow rate was 0.4% with a standard deviation of 1.6%.

Predictions of power were generally good with the exception of two points having an inlet quality of 99%. The power predictions for these two data points were 2kW higher than the measured values. All other predictions of power were within 1.1kW and 3.3% of the tested performance. The mean and standard deviation of the errors are 0.8% and 2.8%, respectively. When the points having an inlet quality of 0.99 are not considered,

the mean and standard deviation are 0.01% and 2.0%, respectively. The large errors in prediction of power at inlet qualities of 99% may indicate that power is not absolutely independent of inlet quality as assumed by the model.

Predictions of efficiency suffered from the combination of errors in prediction of power and mass flow rate. Some predictions for efficiency differed from the measured values by as much as 2.1% absolute and 5.2% of the tested value. In general, the efficiency predictions were mediocre with a mean and standard deviation of errors expessed as percentage of tested value of 0.8% and 2.9%, respectively.

The main use of the model will be in the optimization af the hybrid system. This optimization only requires prediction of the mass flow rate, exhaust quality and power. The efficiency of the Lysholm engine alone is not required. The lack of accuracy in efficiency prediction is thus not critical and should not hamper the application of the model.

The attempt to locate the efficiency peak using the model was only partially successful. Testing of the engine at the predicted peak conditions showed an efficiency of over 42.1%. This is somewhat lower than the 43.3% predicted. Other operating conditions in the same region, tested during the model validation and during the completion of the data matrix, produced efficiencies of over 43%. These variations are within the range of obseved experimental error and repeatability. For an analysis of the experimental errors, refer to the Appendix.

The combination of the flatness of the efficiency peak and the experimental error make it very difficult to locate the exact optimum operating point. The model appears to be able to locate only the region of highest efficiency. Because of the flatness and broadness of the peak, this is all that is required of or can be expécted from the model.

Conclusions and Recommendations

The empirical Lysholm engine model described in this report is adequate for use in modeling of hybrid geothermal energy conversion systems. Test results obtained for the construction of the model indicate that the engine has promise for geothermal application. The major loss mechanism, fluid leakage, has been analyzed and quantitatively evaluated. Further investigation, including the development of more sophisticated models could predict the effectiveness of design changes in arresting leakage. Significant improvements in engine efficiency should be possible with such improvements in design and with scaling-up of the engine.

Modeling of hybrid systems including the Lysholm engine requires prediction of engine power, mass flow rate and exhaust quality for a given pressure ratio, inlet quality, speed and inlet pressure. These important parameters are predicted accurately for an inlet pressure of 120 psia by an engine model derived from current test data. The model is currently implemented in the form of a FORTRAN-IV subroutine. This subroutine can be linked to models of a separator and turbines to form a computer model of the hybrid system currently under test at U.C. Berkeley.

The models for mass flow rate and power are explicit equations in terms of the engine operating conditions. Though the exhaust quality is currently derived from the mass flow rate, power and tabulated steam data; a similar explicit expression for exhaust quality is easily obtainable. These explicit equations could be used to form an analytical system model as an alternative to the computer simulation.

The Lysholm engine model can predict efficiency with only modest accuracy. The model was only able to locate the general range of operating conditions for maximum efficiency. Isentropic efficiencies of over 43% were observed in this region. The broad peak in efficiency represents an optimum pressure ratio and a trade-off between the minimization of leakage fraction at high

speeds and low inlet qualities, sharply increasing fluid and power densities at low inlet qualities and increased minor losses such as friction and drag associated with high speeds.

Since leakage fractions on the order of 50% are observed under typical operating conditions, the most productive route for increasing engine performance involves design changes to reduce leakage. Reducing the rotor tip clearances or simply scaling up the engine to reduce relative leakage area are two promising approaches. Further investigation into the nature of the leakage flows is needed to determine the most productive design changes.

More sophisticated models can quantitatively predict increases in performance involved with design changes. Traditional models based on similarity laws can immediately extend prediction to geometricly similar engines. The application of such traditional models involving dimensionless parameters is more difficult for positive displacement machines such as the Lysholm engine. The need to define the state of the working fluid throughout the engine increases the number of dimensionless parameters required. Testing of other engines may be necessary to extend the predictive ability of dimensionless models to scaled-up engines or to engines with design changes.

A simulation model has the potential to immediately predict the effects of design changes and scaling up of the engine. New engine designs could be tested cheaply and quickly using a simulation model. Such a model could significantly shorten the time required to develop practical Lysholm expanders for geothermal application. Because of this large potential payoff, the development of a simulation model should be pursued concurrently with efforts at traditional modeling involving similarity laws.

TABLE 1 - DATA EVALUATION (Based on data after scan averaging)

DATA CONTROL TARGET	MAXIMUM	MINIMUM	MEAN	STANDARD DEVIATION
INLET PRESSURE = 120 PSIA	121.2 PSIA	118.0 PSIA	119.8 PSIA	0.7 PSIA
SPEED = 4500 RPM	4545 RPM	4456 RPM	4507 RPM	22 RPM
SPEED = 6000 RPM	6040 RPM	5962 RPM	6010 RPM	23 RPM
SPEED = 7500 RPM	7539 RPM	7457 RPM	7501 RPM	26 RPM
SPEED = 9000 RPM	9035 RPM	8966 RPM	8997 RPM	20 RPM
INLET QUALITY = 0.990	0.994	0.990	0.992	0.001
INLET QUALITY = 0.500	0.506	0.486	0.495	0.005
INLET QUALITY = 0.333	0.337	0.330	0.332	0.003
INLET QUALITY = 0.250	0.252	0.247	0.249	0.001
INLET QUALITY = 0.167	0.169	0.164	0.167	0.002
FLOW RATIO = 0 (Water/Steam)	0	0	0	0
FLOW RATIO = 1	1.04	0.95	1.00	0.02
FLOW RATIO = 2	2.02	1.96	2.00	0.02
FLOW RATIO = 3	3.05	2.97	3.00	0.02
FLOW RATIO = 5	5.05	4.96	4.99	0.04

TABLE 2a - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
324	6/30/81	16	119.6	9.1	13.08	4508.	0.461	1183.6	27.6	24.3	0.00	2.52	2.03	0.300	0.992	0.992
342	7/ 7/81	16	119.6	10.0	12.01	4482.	0.452	1182.7	29.0	24.4	0.00	2.44	2.12	0.292	0.991	0.988
340	7/ 7/81	16	119.6	11.1	10.81	4475.	0.450	1182.7	29.5	23.8	0.00	2.38	2.21	0.289	0.991	0.987
335	7/ 2/81	16	119.6	13.0	9.17	4519.	0.460	1183.6	29.5	22.9	0.00	2.37	2.31	0.298	0.992	0.988
331	7/ 1/81	16	119.6	14.7	8.13	4512.	0.460	1182.7	28.8	21.2	0.00	2.33	2.41	0.298	0.991	0.988
5009	6/ 2/81	1	121.2	21.2	5.72	4533.	0.455	1183.2	28.9	17.9	0.00	2.23	2.73	0.290	0.992	0.988
5001	4/20/81	1	121.2	32.5	3.72	4533.	0.449	1183.6	28.0	13.2	0.00	2.23	3.12	0.284	0.992	0.989
325	6/30/81	16	119.6	10.0	12.01	6021.	0.514	1183.2	31.7	30.3	0.00	3.49	1.99	0.298	0.992	0.984
326	6/30/81	16	119.6	11.1	10.81	6035.	0.517	1183.2	32.8	30.4	0.00	3.45	2.06	0.301	0.992	0.982
336	7/ 2/81	16	119.6	13.0	9.17	6019.	0.510	1183.6	33.2	28.5	0.00	3.32	2.19	0.294	0.992	0.982
332	7/ 1/81	16	119.6	14.7	8.13	6011.	0.511	1182.2	32.7	26.8	0.00	3.27	2.28	0.295	0.991	0.982
5010	6/ 2/81	1	120.4	22.0	5.48	6038.	0.513	1183.2	32.0	21.9	0.00	3.16	2.60	0.295	0.992	0.984
5015	8/28/81	1	120.4	24.3	4.95	5965.	0.508	1184.9	32.2	20.6	0.00	3.10	2.69	0.293	0.994	0.986
5002	4/20/81	1	119.6	35.4	3.38	6003.	0.517	1182.7	28.6	14.4	0.00	3.22	2.96	0.301	0.991	0.988

TABLE 2b - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
353	7/17/81	1	119.6	11.1	10.81	7511.	0.560	1184.2	34.2	34.4	0.00	4.46	1.98	0.290	0.993	0.980
337	7/ 2/81	16	119.6	12.8	9.36	7521.	0.558	1183.2	35.2	33.4	0.00	4.36	2.08	0.288	0.992	0.978
333	7/ 1/81	16	119.6	14.7	8.13	7499.	0.561	1181.7	35.1	31.5	0.00	4.27	2.18	0.291	0.990	0.977
483	12/ 4/81	1	119.6	15.3	7.82	7466.	0.560	1184.6	36.7	32.5	0.00	4.22	2.21	0.293	0.994	0.978
5031	10/ 6/81	1	121.2	24.1	5.03	7492.	0.554	1184.2	34.9	24.6	0.00	4.05	2.58	0.282	0.993	0.981
5017	8/28/81	1	119.6	26.4	4.54	7524.	0.555	1184.2	33.9	22.6	0.00	4.10	2.63	0.286	0.993	0.983
5011	8/21/81	1	118.0	34.2	3.45	7538.	0.544	1184.2	31.3	16.9	0.00	4.16	2.85	0.277	0.993	0.987
5004	4/20/81	1	119.6	34.8	3.43	7501.	0.559	1182.7	27.8	15.4	0.00	4.18	2.83	0.290	0.991	0.989
5003	4/20/81	1	121.2	37.9	3.20	7461.	0.564	1182.7	26.7	14.1	0.00	4.18	2.89	0.293	0.991	0.990
328	6/30/81	16	119.6	11.1	10.81	8992.	0.606	1182.7	36.9	40.0	0.00	5.56	1.90	0.283	0.991	0.974
338	7/ 2/81	16	119.6	13.0	9.17	9016.	0.612	1182.2	36.7	37.8	0.00	5.45	2.00	0.288	0.991	0.975
334	7/ 1/81	16	119.6	14.7	8.13	9021.	0.606	1181.7	36.7	35.6	0.00	5.34	2.10	0.282	0.990	0.975
264	12/ 0/80	9	118.0	15.0	7.87	9033.	0.614	1183.2	35.4	34.4	0.00	5.40	2.09	0.294	0.992	0.979
5032	10/ 6/81	1	119.6	25.4	4.70	9017.	0.601	1183.7	35.2	25.9	0.00	5.11	2.51	0.278	0.992	0.981
5018	8/28/81	1	118.0	28.8	4.10	8988.	0.592	1184.2	33.7	22.4	0.00	5.10	2.61	0.274	0.993	0.984
5005	4/20/81	1	119.6	37.9	3.16	9014.	0.604	1182.7	23.1	13.0	0.00	5.28	2.78	0.281	0.991	0.993

TABLE 2c - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
407	9/11/81	12	119.6	10.8	11.04	4515.	0.785	746.6	31.7	24.4	1.00	3.84	1.87	0.462	0.495	0.563
352	7/15/81	1	121.2	12.8	9.49	4545.	0.790	744.4	32.6	23.6	1.02	3.76	1.99	0.458	0.491	0.557
359	7/17/81	1	119.6	14.4	8.30	4525.	0.789	743.6	33.6	22.9	1.02	3.69	2.07	0.463	0.491	0.553
400	9/11/81	13	119.6	15.3	7.82	4519.	0.785	743.8	33.4	22.1	1.01	3.65	2.12	0.459	0.491	0.552
430	10/ 2/81	1	120.7	24.8	4.86	4521.	0.781	744.6	32.9	16.9	1.01	3.50	2.51	0.452	0.492	0.542
406	9/11/81	12	119.6	11.1	10.81	5977.	0.866	747.1	36.4	30.5	1.00	5.31	1.79	0.439	0.495	0.559
357	7/17/81	1	119.6	13.0	9.17	6025.	0.849	753.4	37.4	29.2	0.97	5.14	1.92	0.424	0.502	0.561
358	7/17/81	1	119.6	14.4	8.30	5989.	0.864	744.9	38.0	28.4	1.02	5.11	1.98	0.434	0.493	0.551
399	9/11/81	13	119.6	15.3	7.82	5997.	0.880	740.9	37.4	27.5	1.03	5.14	2.01	0.445	0.488	0.546
410	9/11/81	1	119.6	24.2	4.94	5962.	0.864	738.8	38.6	21.9	1.04	4.90	2.36	0.430	0.486	0.533
355	7/17/81	15	118.0	13.0	9.05	7457.	0.930	751.8	39.9	33.8	0.98	6.70	1.83	0.410	0.501	0.558
404	9/11/81	12	119.6	14.7	8.13	7522.	0.947	749.6	40.8	33.5	0.99	6.68	1.90	0.412	0.498	0.552
298	4/14/81	1	121.2	15.3	7.92	7497.	0.950	749.5	39.9	32.5	0.99	6.61	1.93	0.409	0.497	0.552
431	10/ 2/81	1	120.1	24.5	4.90	7539.	0.948	747.6	39.1	24.6	1.00	6.43	2.26	0.407	0.496	0.541
5021	9/14/81	1	119.6	25.4	4.70	7491.	0.935	742.8	37.6	22.6	1.02	6.38	2.30	0.394	0.490	0.537
5025	9/14/81	1	121.2	27.8	4.36	7478.	0.941	746.1	39.2	22.7	1.00	6.33	2.36	0.397	0.493	0.537
405	9/11/81	12	119.6	13.0	9.17	8981.	1.027	749.1	41.4	38.8	0.99	8.45	1.75	0.388	0.498	0.554
402	9/11/81	12	119.6	14.7	8.13	9000.	1.028	749.5	41.8	37.3	0.99	8.33	1.82	0.388	0.498	0.551
299	4/14/81	1	119.6	15.3	7.82	9011.	1.032	756.6	41.2	36.7	0.95	8.28	1.84	0.401	0.506	0.558
432	10/ 2/81	1	119.6	24.8	4.82	9035.	1.029	746.7	39.4	26.6	1.01	8.05	2.18	0.382	0.495	0.540
5030	10/ 6/81	1	119.6	26.8	4.46	8990.	1.014	744.3	35.9	22.7	1.02	7.96	2.25	0.367	0.492	0.538
5023	9/14/81	1	120.4	28.3	4.25	8993.	1.028	746.4	37.0	23.0	1.00	7.99	2.27	0.379	0.494	0.538
5026	9/14/81	1	119.6	29.8	4.01	8986.	1.028	744.3	36.6	21.9	1.01	8.04	2.31	0.381	0.492	0.535

TABLE 2d - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
382	9/ 2/81	12	119.6	13.2	9.04	4532.	1.069	606.0	33.2	23.2	1.98	4.83	1.82	0.592	0.334	0.422
380	9/ 2/81	12	121.2	14.7	8.24	4509.	1.073	602.6	32.5	21.6	2.02	4.74	1.90	0.585	0.330	0.416
389	9/ 2/81	1	118.8	15.1	7.84	4518.	1.065	603.9	34.8	22.6	2.01	4.74	1.92	0.589	0.333	0.415
461	11/20/81	1	120.1	20.8	5.78	4472.	1.062	602.8	34.3	18.9	2.01	4.53	2.16	0.583	0.331	0.404
423	9/22/81	1	119.6	24.4	4.90	4505.	1.056	604.5	35.9	17.9	2.00	4.52	2.28	0.579	0.333	0.400
484	12/ 4/81	1	120.1	30.2	3.98	4491.	1.040	604.5	36.9	15.7	1.99	4.47	2.46	0.562	0.332	0.392
411	9/18/81	12	119.6	13.0	9.23	5983.	1.201	603.3	35.4	27.8	2.00	6.80	1.71	0.564	0.331	0.418
364	8/21/81	127	119.6	14.7	8.13	6024.	1.170	604.4	38.5	27.9	2.00	6.63	1.81	0.531	0.333	0.414
388	9/ 2/81	1	119.6	15.3	7.82	6033.	1.186	604.1	39.4	28.4	2.01	6.65	1.83	0.546	0.332	0.412
462	11/20/81	1	120.1	20.4	5.89	6024.	1.183	603.1	38.7	24.0	2.00	6.44	2.03	0.539	0.331	0.403
424	9/22/81	1	118.5	24.7	4.80	6007.	1.166	603.2	39.9	21.6	2.01	6.37	2.18	0.534	0.332	0.397
482	12/ 4/81	1	120.1	29.8	4.03	6040.	1.157	606.1	39.2	18.8	1.97	6.33	2.32	0.517	0.334	0.394
381	9/ 2/81	12	120.1	14.7	8.17	7494.	1.303	603.9	38.7	31.3	2.01	8.69	1.72	0.504	0.332	0.413
368	8/26/81	17	119.6	15.6	7.67	7523.	1.286	607.8	41.3	32.3	1.96	8.58	1.77	0.498	0.337	0.414
464	11/20/81	1	119.6	20.0	5.97	7522.	1.309	603.2	42.1	29.1	2.01	8.49	1.92	0.509	0.331	0.402
425	9/22/81	1	119.6	24.2	4.94	7496.	1.283	603.6	41.2	25.0	2.00	8.30	2.06	0.487	0.332	0.397
5012	8/21/81	1	119.6	33.1	3.61	7519.	1.242	596.8	38.1	17.7	2.06	8.32	2.33	0.424	0.324	0.381
387	9/ 2/81	1	119.6	15.5	7.71	9000.	1.415	605.7	41.8	35.8	1.99	10.80	1.68	0.466	0.334	0.412
463	11/20/81	1	119.6	20.3	5.90	9015.	1.419	604.6	43.1	32.1	1.99	10.57	1.85	0.464	0.333	0.402
426	9/22/81	1	119.6	24.5	4.88	9017.	1.400	603.8	39.9	26.2	2.00	10.42	1.99	0.442	0.332	0.397
5013	8/21/81	1	119.6	36.0	3.32	8966.	1.345	606.9	29.2	14.1	1.96	10.32	2.28	0.403	0.335	0.391

TABLE 2e - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
415	9/18/81	14	118.0	15.3	7.72	4478.	1.330	531.4	35.2	22.7	3.01	5.66	1.79	0.711	0.250	0.344
457	11/16/81	1	119.6	20.1	5.94	4502.	1.301	531.4	36.2	19.8	3.00	5.46	2.00	0.668	0.249	0.334
456	11/16/81	1	119.6	24.5	4.88	4513.	1.293	531.6	35.7	17.2	3.00	5.41	2.15	0.659	0.250	0.327
480	12/ 4/81	1	119.6	30.4	3.94	4506.	1.279	531.9	37.4	15.4	2.99	5.38	2.32	0.647	0.250	0.319
414	9/18/81	14	119.6	15.1	7.90	6018.	1.505	529.3	38.9	28.4	3.05	8.09	1.68	0.650	0.247	0.340
458	11/16/81	1	119.6	20.8	5.75	6028.	1.464	531.4	40.2	24.2	3.00	7.73	1.91	0.617	0.250	0.331
429	9/22/81	1	119.6	24.2	4.94	6024.	1.457	532.2	41.3	22.6	2.99	7.66	2.02	0.613	0.250	0.326
481	12/ 4/81	1	119.6	29.8	4.01	5985.	1.440	531.2	40.0	18.7	3.00	7.58	2.18	0.598	0.249	0.318
376	8/31/81	1	119.6	16.9	7.09	7532.	1.606	533.3	39.7	29.7	2.99	10.27	1.69	0.556	0.252	0.340
459	11/16/81	1	119.6	20.4	5.86	7462.	1.630	530.0	42.5	28.8	3.02	10.13	1.80	0.574	0.248	0.329
427	9/22/81	1	119.6	24.7	4.85	7460.	1.601	531.3	41.4	24.6	3.00	9.95	1.94	0.552	0.249	0.324
5019	9/ 4/81	1	119.6	28.6	4.17	7504.	1.596	531.7	38.3	20.4	2.99	9.99	2.04	0.542	0.250	0.320
383	9/ 2/81	1	119.6	17.5	6.82	8977.	1.758	531.7	41.2	32.9	3.02	12.79	1.64	0.498	0.250	0.337
460	11/16/81	1	121.2	20.8	5.83	8968.	1.765	532.2	42.5	31.2	2.98	12.55	1.74	0.488	0.249	0.331
428	9/22/81	1	119.6	25.1	4.76	8985.	1.770	529.6	39.7	25.6	3.01	12.62	1.85	0.497	0.247	0.323
5020	9/ 4/81	1	120.1	31.4	3.82	8981.	1.742	532.8	34.6	19.0	2.97	12.50	2.01	0.480	0.251	0.319

TABLE 2f - LYSHOLM ENGINE TEST RESULTS

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/SEC]	INLET ENTHL [BTU/SEC]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/SEC]	INLET QUAL	EXIT QUAL
416	9/18/81	1	118.5	16.9	7.03	4495.	1.814	457.9	36.5	22.7	5.03	7.28	1.66	0.884	0.166	0.268
466	11/25/81	1	119.6	20.4	5.86	4499.	1.785	456.6	36.3	19.9	5.04	7.11	1.80	0.834	0.164	0.259
469	12/ 3/81	1	120.4	23.6	5.09	4541.	1.773	457.6	37.7	18.7	5.05	7.08	1.90	0.811	0.165	0.254
476	12/ 3/81	1	119.6	29.8	4.01	4456.	1.734	458.7	40.1	16.5	5.04	6.88	2.09	0.805	0.167	0.244
475	12/ 3/81	1	119.6	18.8	6.36	5987.	1.998	458.4	41.7	27.0	5.01	10.03	1.65	0.747	0.166	0.262
467	11/25/81	1	121.2	21.4	5.65	6000.	2.006	457.6	40.7	24.5	5.01	9.96	1.73	0.722	0.164	0.257
470	12/ 3/81	1	119.6	25.4	4.70	6036.	1.968	459.6	40.4	21.4	5.00	9.89	1.85	0.717	0.168	0.252
477	12/ 4/81	1	120.7	28.3	4.27	6028.	1.963	459.2	38.7	18.9	4.98	9.84	1.93	0.694	0.167	0.248
474	12/ 3/81	1	119.6	20.8	5.75	7492.	2.200	459.2	43.4	29.3	4.98	13.06	1.63	0.643	0.167	0.259
471	12/ 3/81	1	120.1	25.1	4.78	7481.	2.166	459.8	41.0	24.1	4.98	12.84	1.76	0.608	0.168	0.253
478	12/ 4/81	1	120.1	28.8	4.17	7536.	2.174	459.4	37.9	20.3	4.96	12.97	1.84	0.601	0.167	0.247
473	12/ 3/81	1	119.6	22.4	5.34	8993.	2.359	459.8	42.0	29.0	4.96	16.17	1.61	0.497	0.168	0.257
472	12/ 3/81	1	119.6	25.4	4.70	8971.	2.352	460.9	39.7	25.3	4.91	16.03	1.69	0.509	0.169	0.254
479	12/ 4/81	1	120.1	30.2	3.98	8983.	2.365	459.4	34.5	19.3	4.96	16.16	1.79	0.490	0.167	0.246

TABLE 3 - DATA TABLE CODES

Comment Codes

0 - 0.250 in. Diameter(Small) Water Nozzle Used

1 - Karl Brown's Data (1981-2)

2 - Repeated Run, Seal Pot Error

3 - Repeated Run, Off-Matrix

4 - Repeated Run, Inconsistent Data

5 - Repeated Run, Repeatability Check

6 - Seal Pot Error Corrected For

7 - Water Temperature Estimated

8 - Superheated Inlet Conditions

9 - Ralph Berger's Data (1980)

Calibration Codes

1 - Pre-Signal Filter(before 3/80)

2 - Large Water Orifice Plate(after 3/80)

3 - Small Water Orifice Plate(after 3/80)

Note: Run numbers in the 5000's are staging runs.

TABLE 4 - COEFFICIENTS FOR LYSHOLM ENGINE MODEL

i) Mass Flow Rate

a_{aa} = -5.403E-2
b_{aa} = 3.638E-4

a_{ba} = -4.031
b_{ba} = 3.375E-4

a_{ab} = 2.694E-1
b_{ab} = 1.968E-4

a_{bb} = 3.685
b_{bb} = -2.050E-4

V_1 = 1.331E-4 ft^3-min/rev-sec
v_f = 1.789E-2 ft^3/lbm (@ 120 psia)
v_{fg} = 3.710 ft^3/lbm (@ 120 psia)

ii) Power

a_a = 6.093E-2
b_a = -9.934E-6

a_b = 6.651E-2
b_b = -9.590E-6
c_b = 5.113E-10

b_c = 1.144E-4

Note: Coefficients are expressed in FORTRAN scientific notation. The mantissa precedes the E and the exponent follows it.

TABLE 5a - MODEL VALIDATION, TEST RESULTS (PREDICTED/MEASURED)

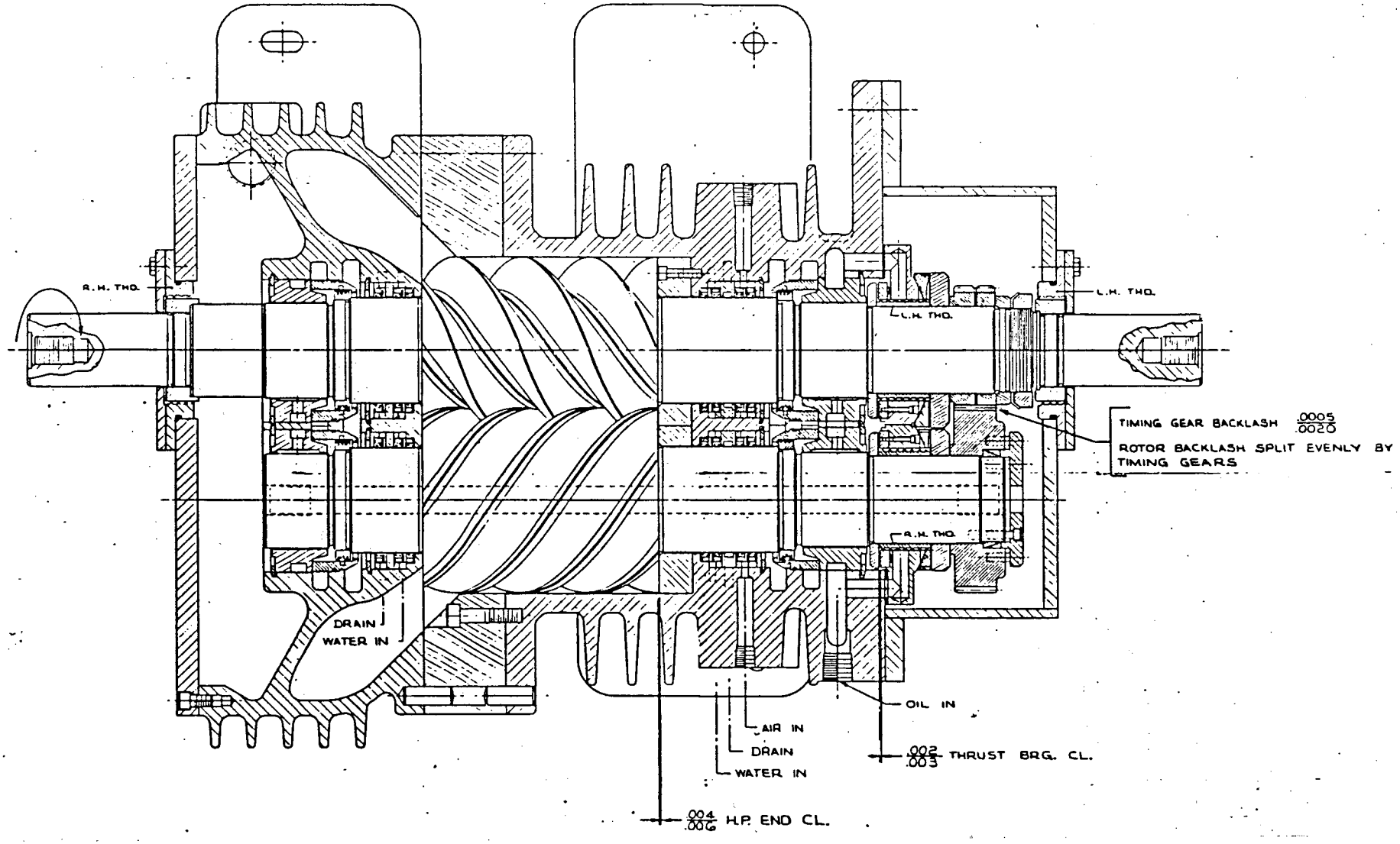
Day 1 (2/24/82) Measured Inlet Pressures = 120 ± 0.5 psia

OPERATING CONDITIONS INLET QUALITY ENGINE SPEED PRESSURE RATIO	FLOW RATE LBM/SEC	POWER kW	EFFI-CIENCY PERCENT	EXHAUST QUALITY
0.238, 8443 RPM, 5.97 (Predicted Optimum)	1.755 / 1.803	30.9 / 30.7	43.3 / 42.0	0.321 / 0.321
0.237, 8431 RPM, 5.92 (Predicted Optimum)	1.759 / 1.808	30.7 / 30.7	43.3 / 42.1	0.320 / 0.321
0.200, 4499 RPM, 5.92	1.556 / 1.556	19.7 / 19.8	35.6 / 35.9	0.291 / 0.291
0.200, 6043 RPM, 5.97	1.734 / 1.759	24.3 / 24.4	39.3 / 38.9	0.290 / 0.290
0.199, 7539 RPM, 5.86	1.910 / 1.951	28.5 / 27.7	42.5 / 40.4	0.287 / 0.288
0.200, 7504 RPM, 5.27	1.894 / 1.925	26.6 / 27.2	42.6 / 43.0	0.284 / 0.284
0.198, 9004 RPM, 5.64	2.090 / 2.116	30.4 / 29.4	42.5 / 40.6	0.285 / 0.286

TABLE 5b - MODEL VALIDATION, TEST RESULTS (PREDICTED/MEASURED)

Day 2 (3/5/82) Measured Inlet Pressures = 120 ± 0.5 psia

OPERATING CONDITIONS INLET QUALITY, ENGINE SPEED, PRESSURE RATIO	FLOW RATE LBM/SEC	POWER kW	EFFICIENCY PERCENT	EXHAUST QUALITY
0.398, 8045 RPM, 8.81	1.171 / 1.173	35.2 / 34.1	40.5 / 39.1	0.470 / 0.471
0.334, 4995 RPM, 9.75	1.127 / 1.110	25.2 / 25.3	33.1 / 33.8	0.424 / 0.423
0.333, 5015 RPM, 8.18	1.127 / 1.109	24.0 / 24.4	34.2 / 35.4	0.417 / 0.416
0.334, 6997 RPM, 8.64	1.268 / 1.261	31.7 / 31.9	39.1 / 39.5	0.416 / 0.416
0.285, 8034 RPM, 7.67	1.503 / 1.518	33.6 / 33.5	42.0 / 41.5	0.370 / 0.370
0.182, 7015 RPM, 6.82	1.984 / 2.002	29.1 / 30.1	40.8 / 41.8	0.279 / 0.279
0.332, 8011 RPM, 8.13	1.339 / 1.346	34.3 / 34.7	41.4 / 41.7	0.412 / 0.412
0.994, 8003 RPM, 8.12	0.575 / 0.563	34.2 / 32.0	37.1 / 35.4	0.977 / 0.980
0.994, 8011 RPM, 10.65	0.578 / 0.565	36.9 / 35.1	35.7 / 34.8	0.979 / 0.981



GEOTHERMAL EXPANDER FOR STEAM-WATER MIXTURE

FIGURE 1 - CROSS SECTION OF LYSHOLM ENGINE

GARDNER-DENVER CO.
SC-71
SECTIONAL VIEW
RESEARCH

FIGURE 1b - LYSHOLM ENGINE DATA (Reference 9)

Theoretical Inlet Volume	0.00798 ft^3/rev
Expansion Ratio	5.3/1
Hydrostatic Pressure Tests to	500 psig
Maximum Inlet Pressure	250 psia
Maximum Inlet Temperature	500 ^oF
Rotor Diameter	5.120 in.
Rotor Length	6.519 in.
Maximum Recommended Male Rotor Shaft Speed	9000 RPM
Minimum Exhaust Pressure	4 psia

FIGURE 2 - LYSHOLM ENGINE/HYBRID SYSTEM SCHEMATIC

-- Indicates Thermocouple in line

(Includes Primary Lysholm Instrumentation)

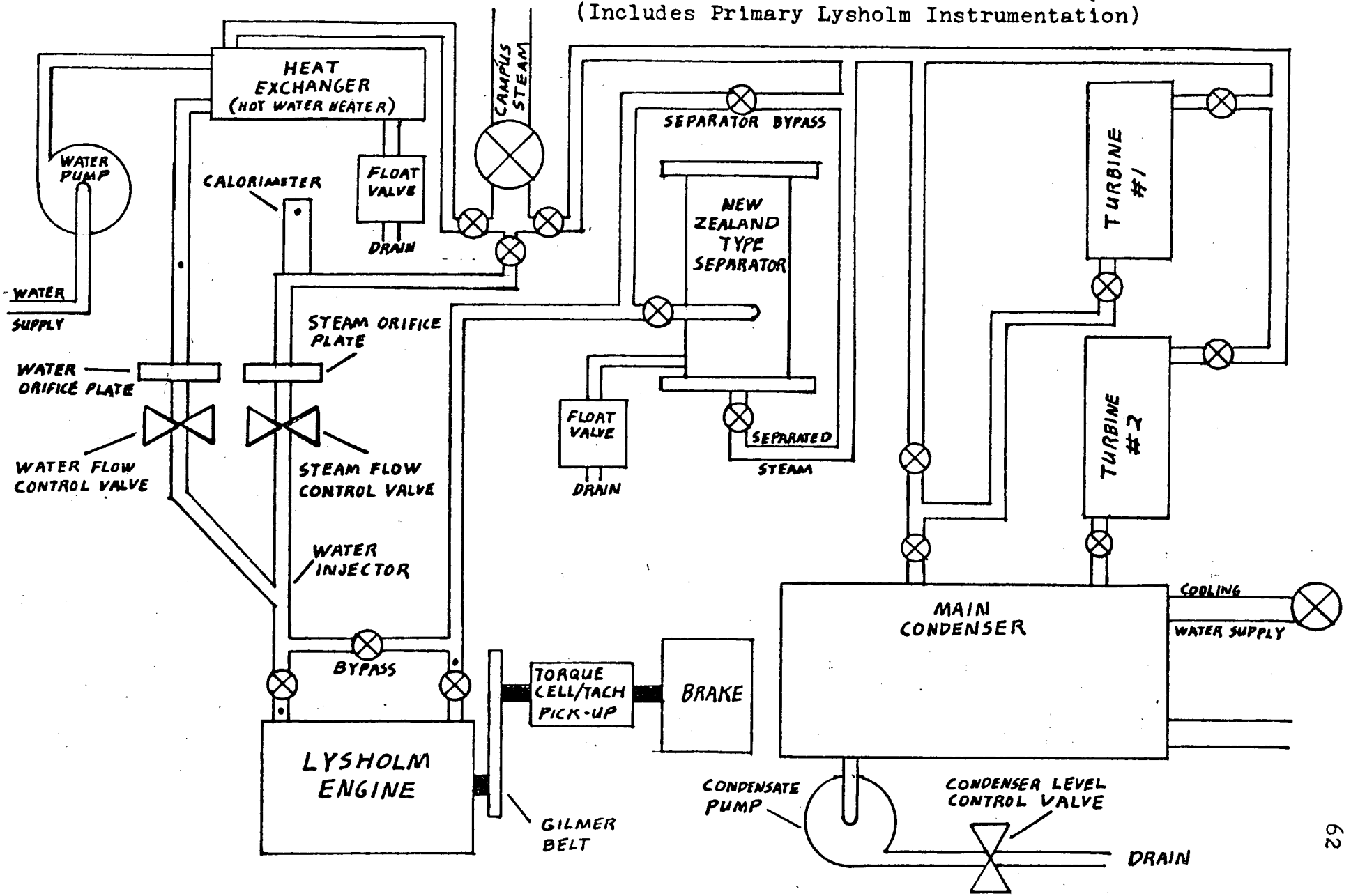
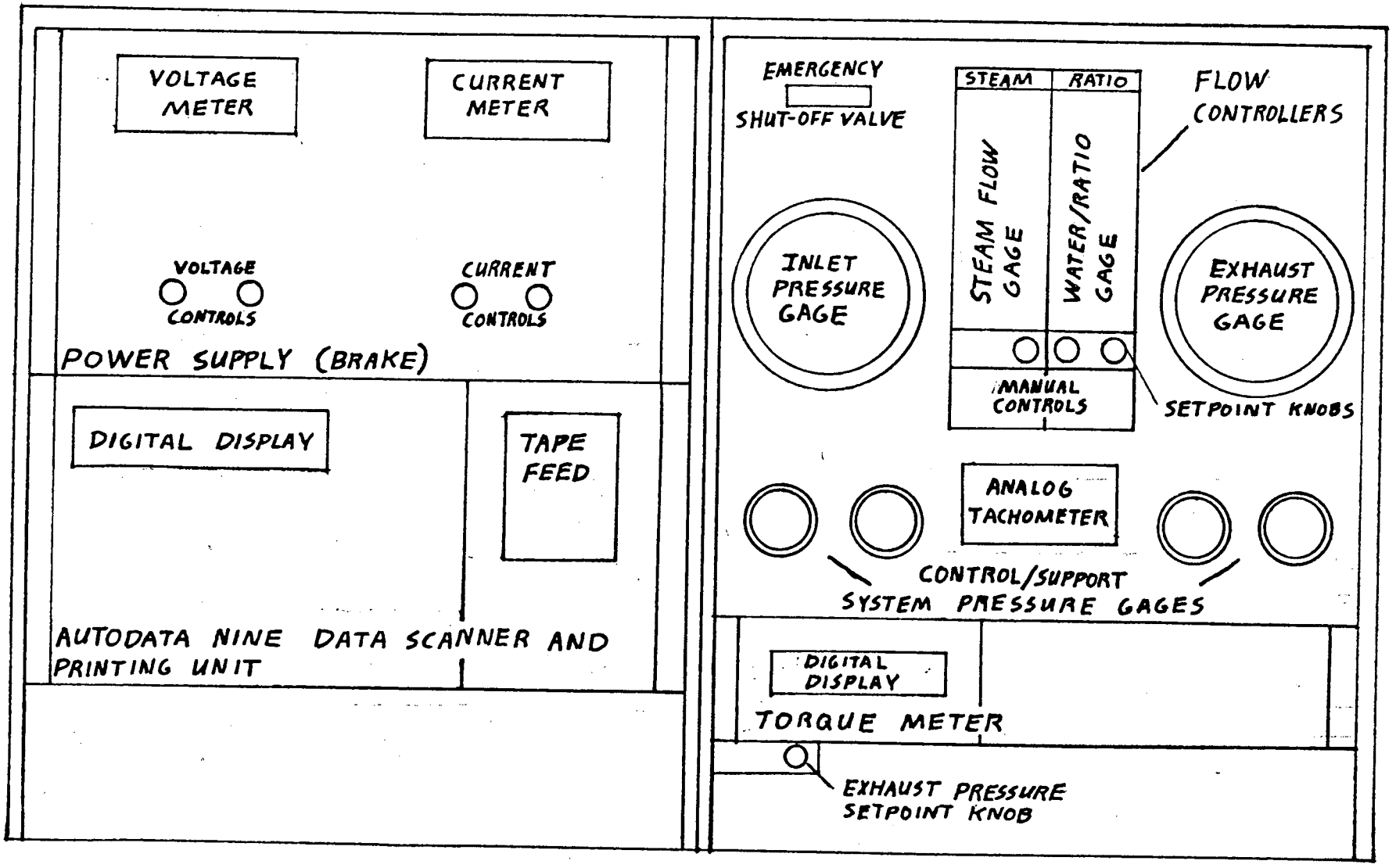


FIGURE 3 - LYSHOLM ENGINE CONTROL PANEL LAYOUT



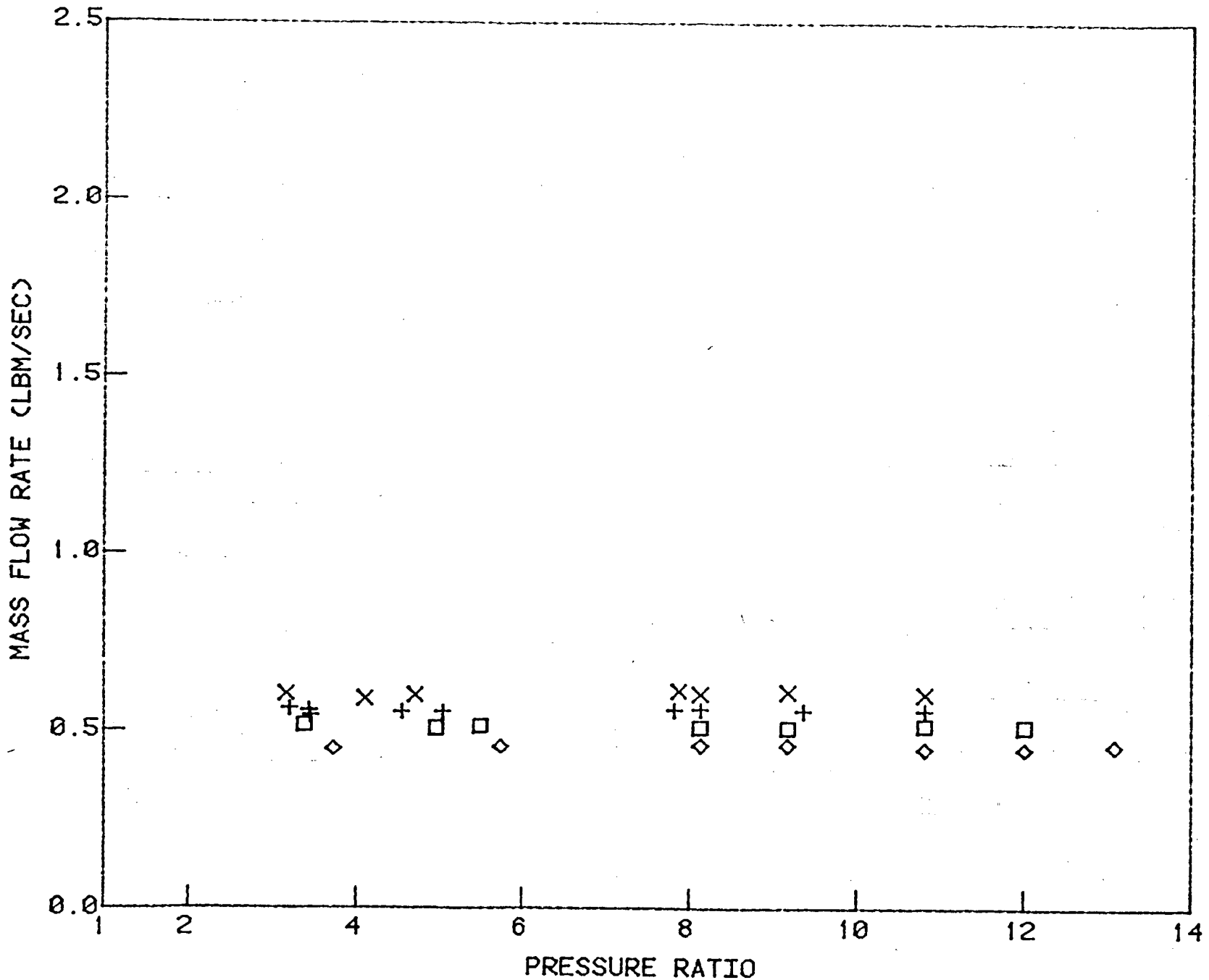


FIGURE 4a - LYSHOLM ENGINE PERFORMANCE

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.99

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

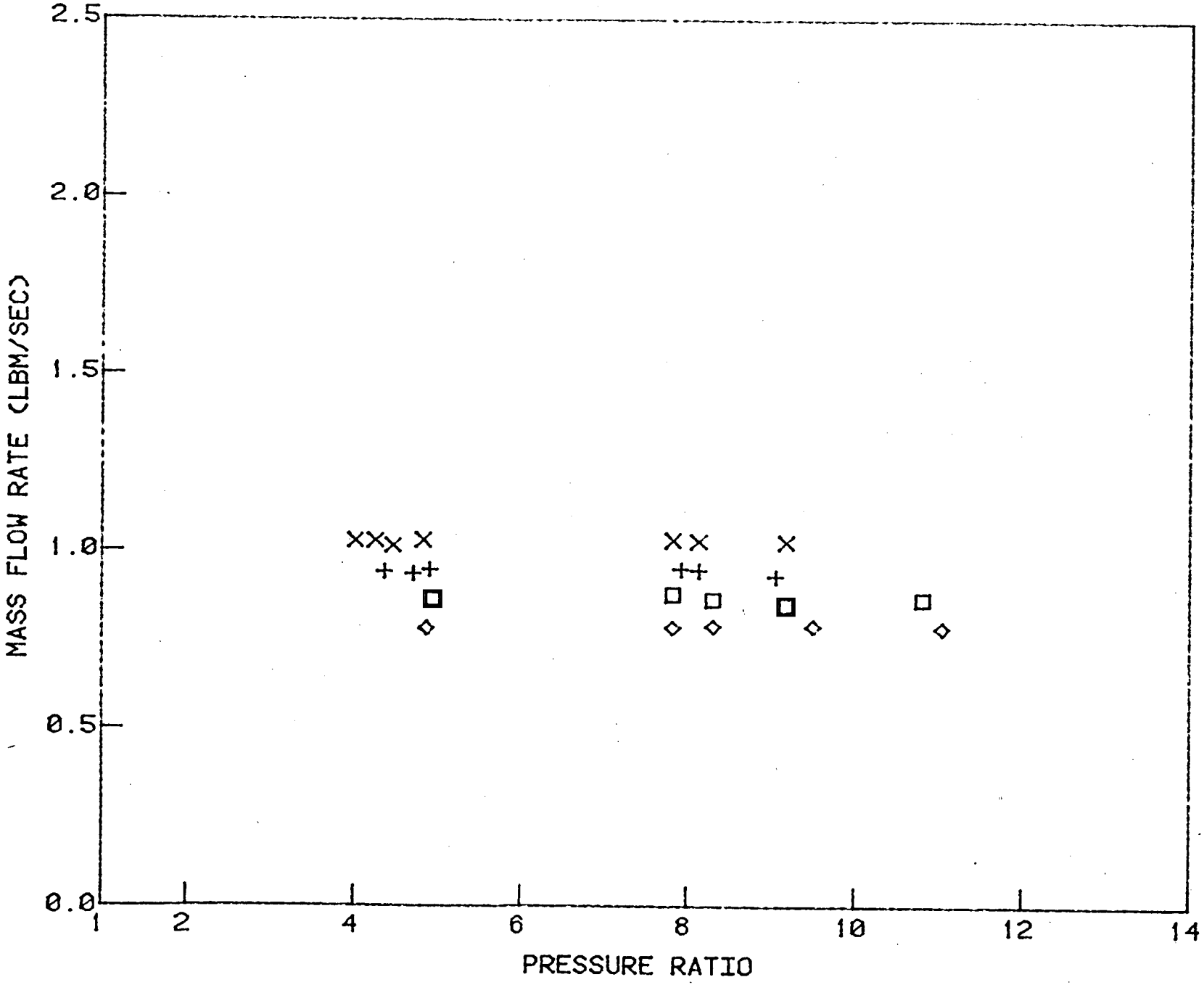


FIGURE 4b - LYSHOLM ENGINE PERFORMANCE

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.50

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

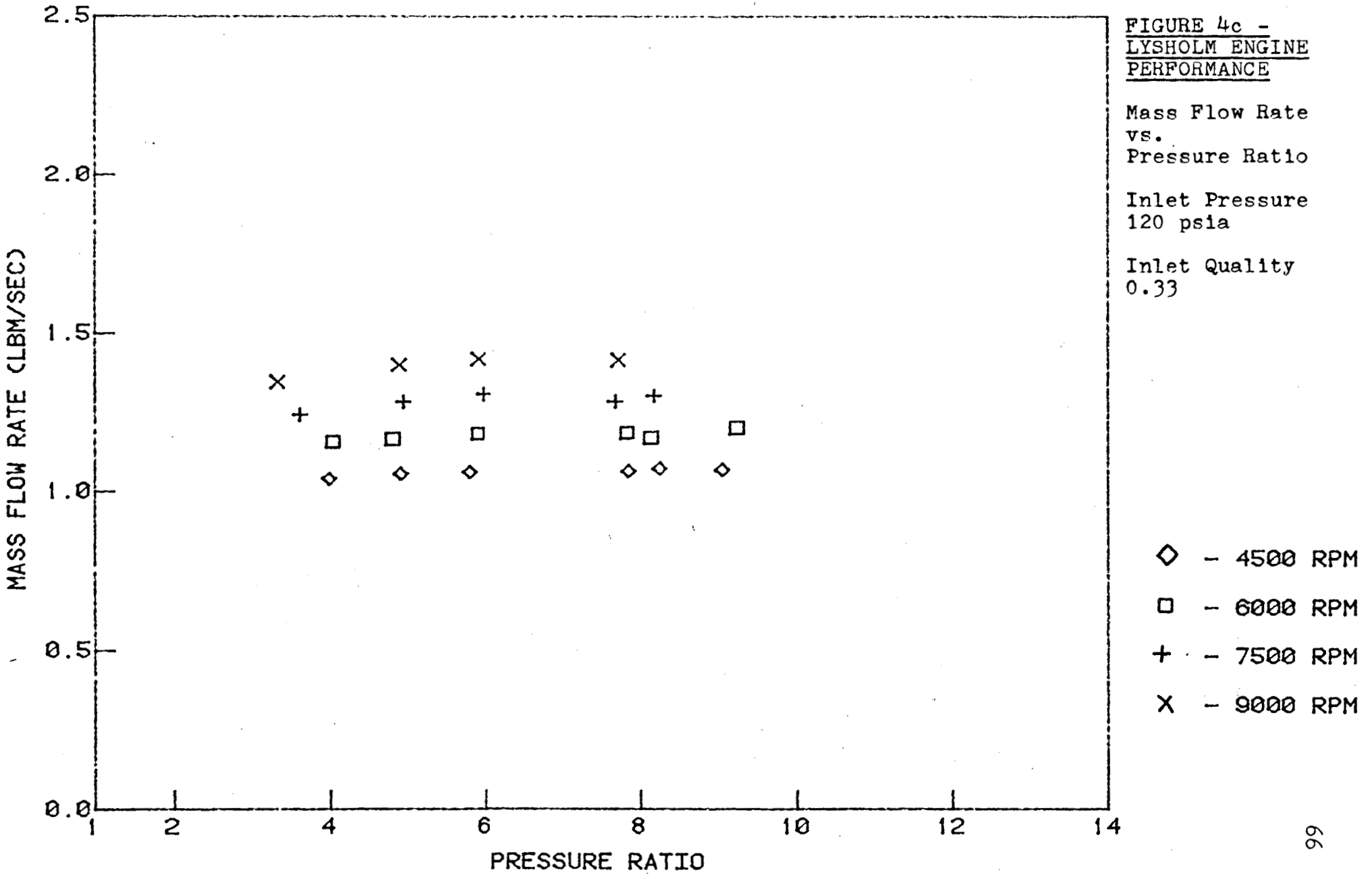


FIGURE 4c - LYSHOLM ENGINE PERFORMANCE

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.33

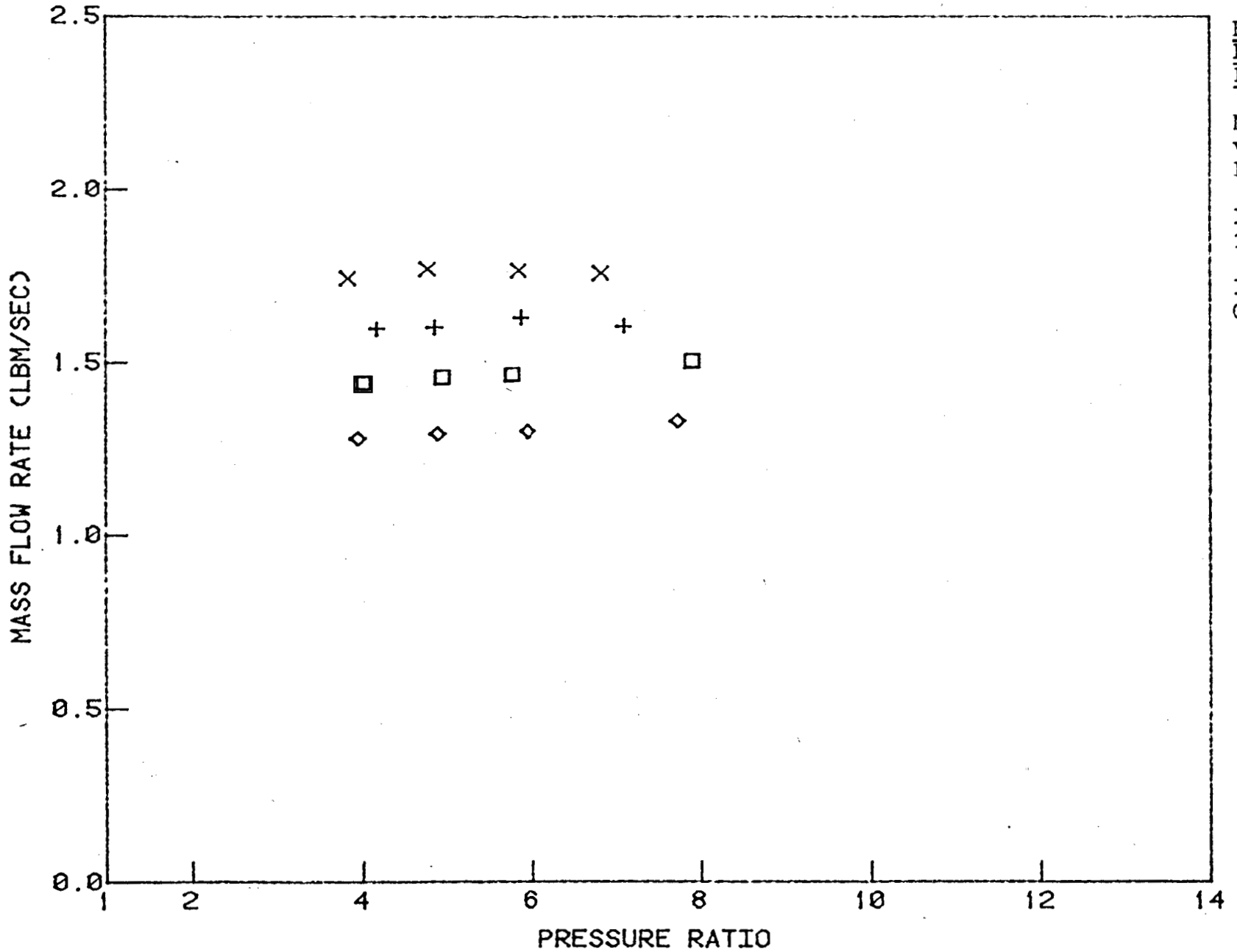


FIGURE 4d - LYSHOLM ENGINE PERFORMANCE

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.25

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

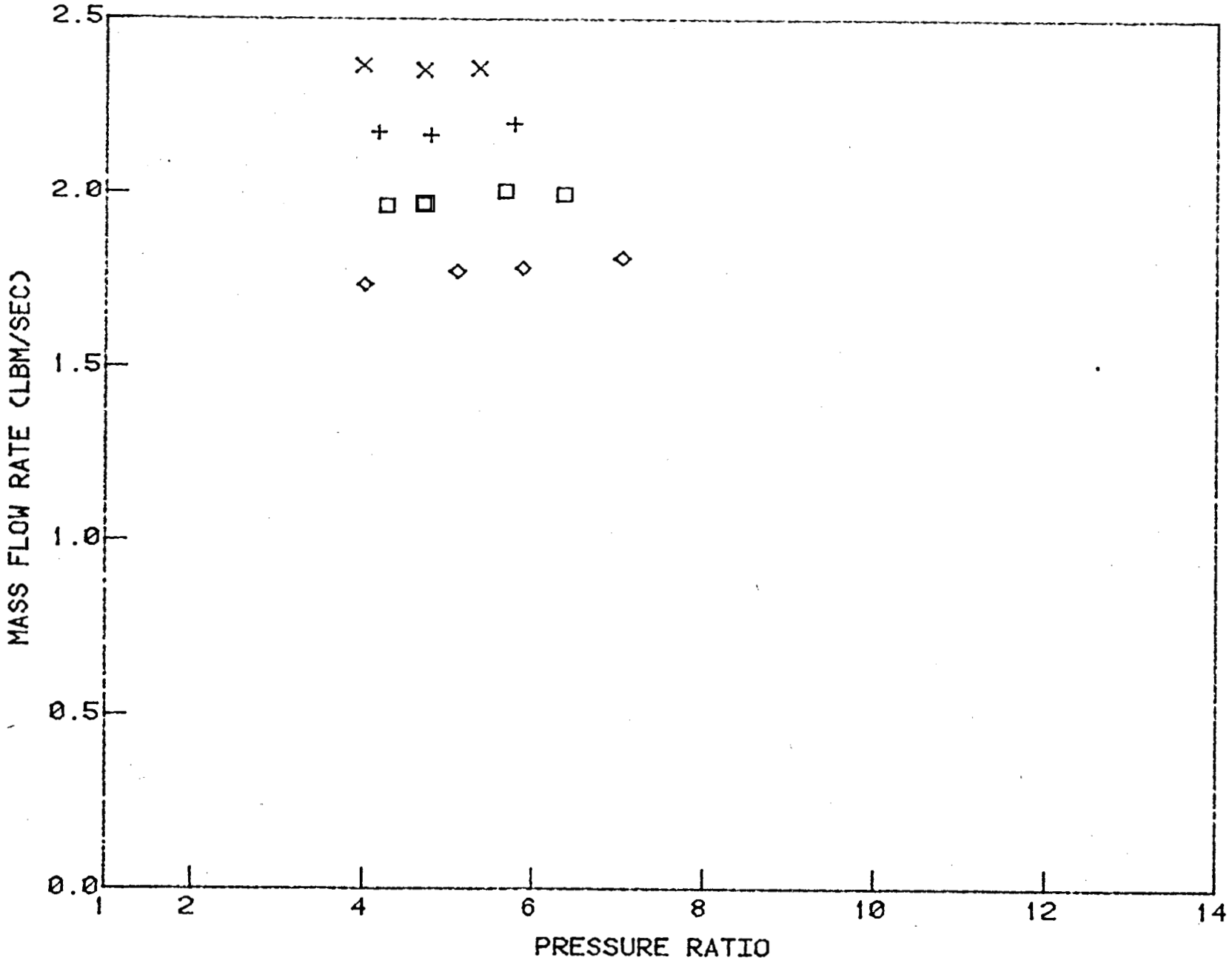


FIGURE 4e - LYSHOLM ENGINE PERFORMANCE

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.17

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

× - 9000 RPM

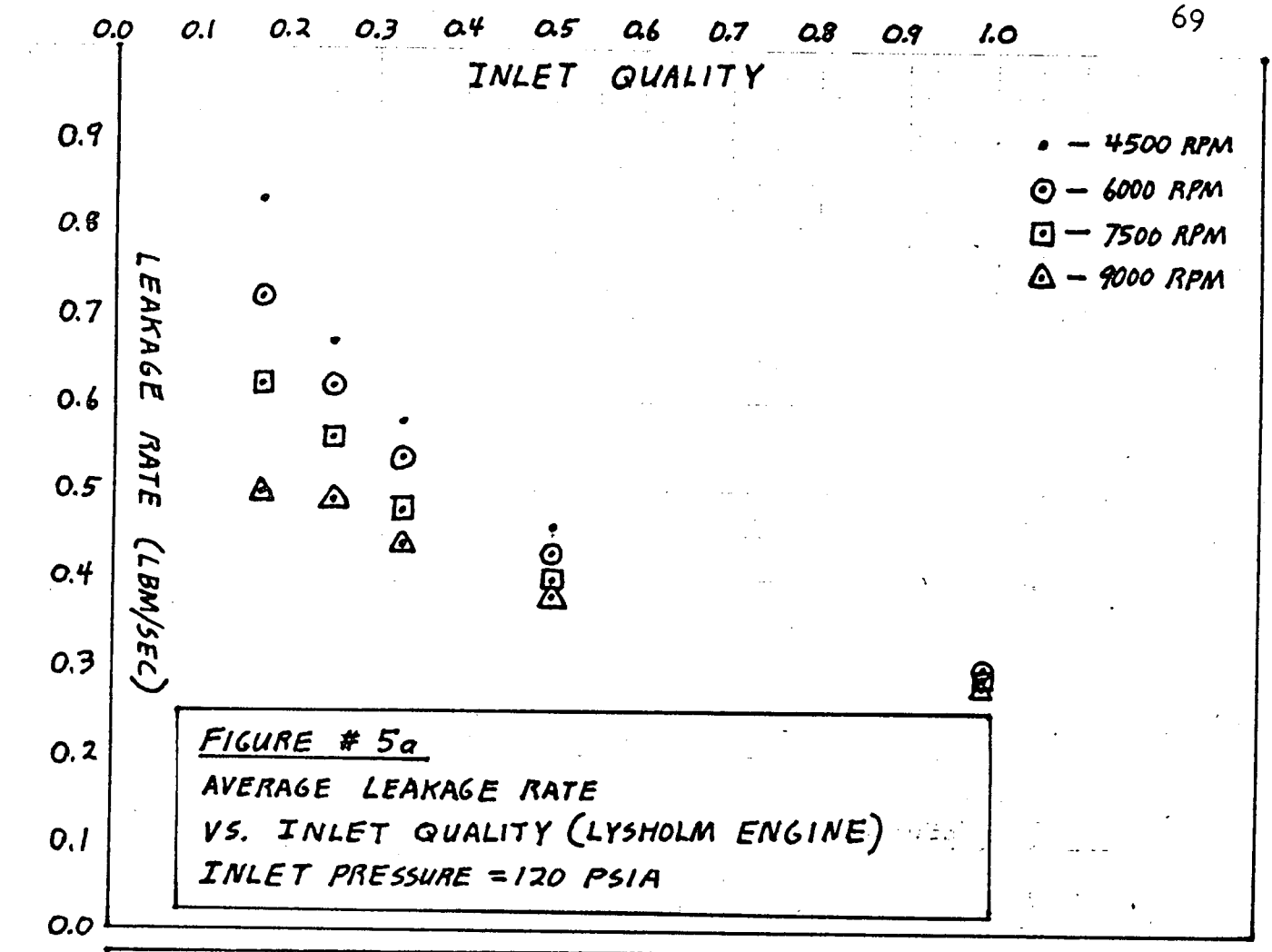


FIGURE # 5a
AVERAGE LEAKAGE RATE
VS. INLET QUALITY (LYSHOLM ENGINE)
INLET PRESSURE = 120 PSIA

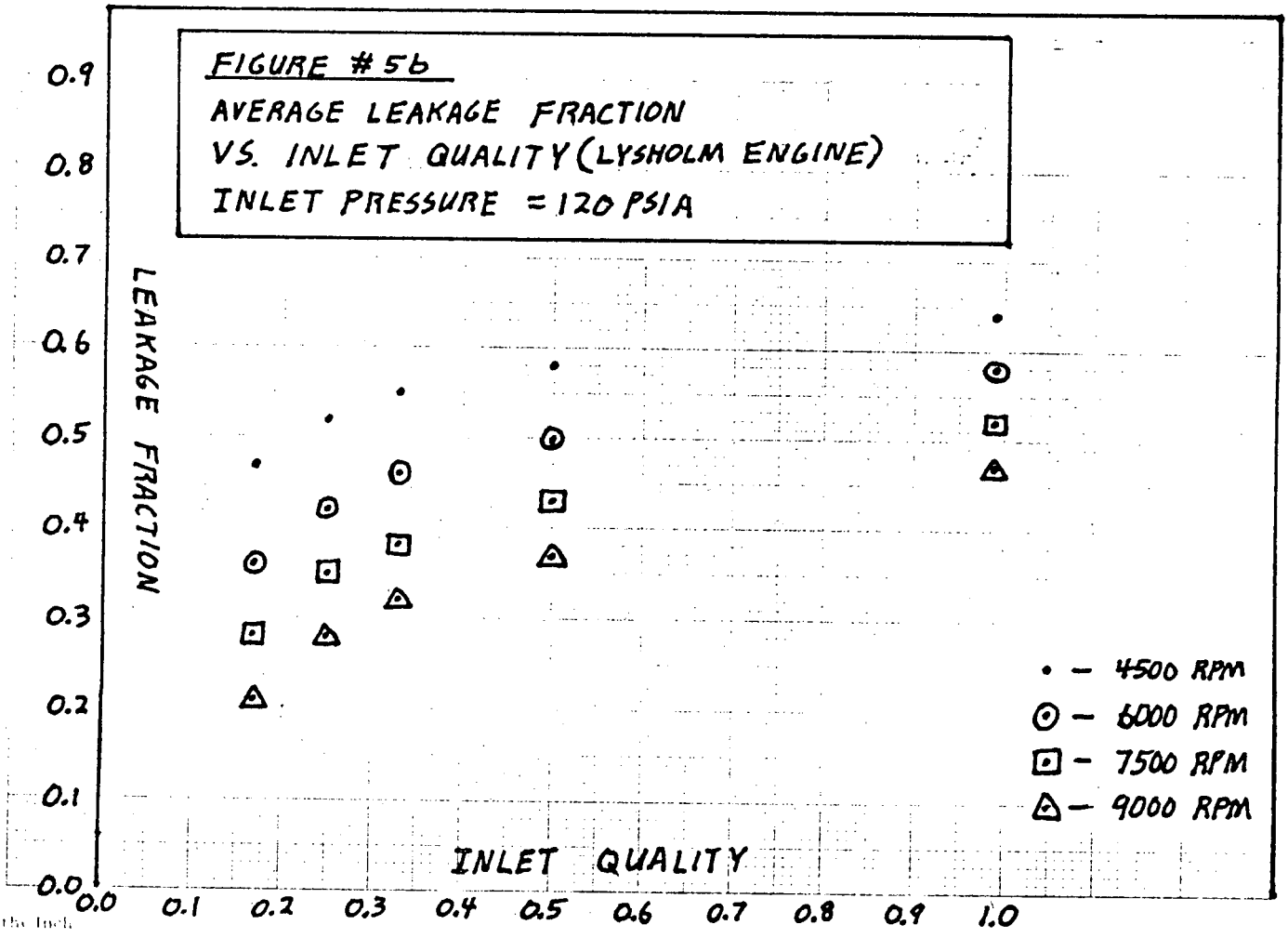


FIGURE # 5b
AVERAGE LEAKAGE FRACTION
VS. INLET QUALITY (LYSHOLM ENGINE)
INLET PRESSURE = 120 PSIA

FIGURE #6 – CHARACTERISTIC VELOCITIES (LYSHOLM ENGINE)

ρ_l = 56 LBM/FT³ (SATURATED LIQUID @ 120 PSIA)

ρ_v = 0.27 LBM/FT³ (SATURATED VAPOR @ 120 PSIA)

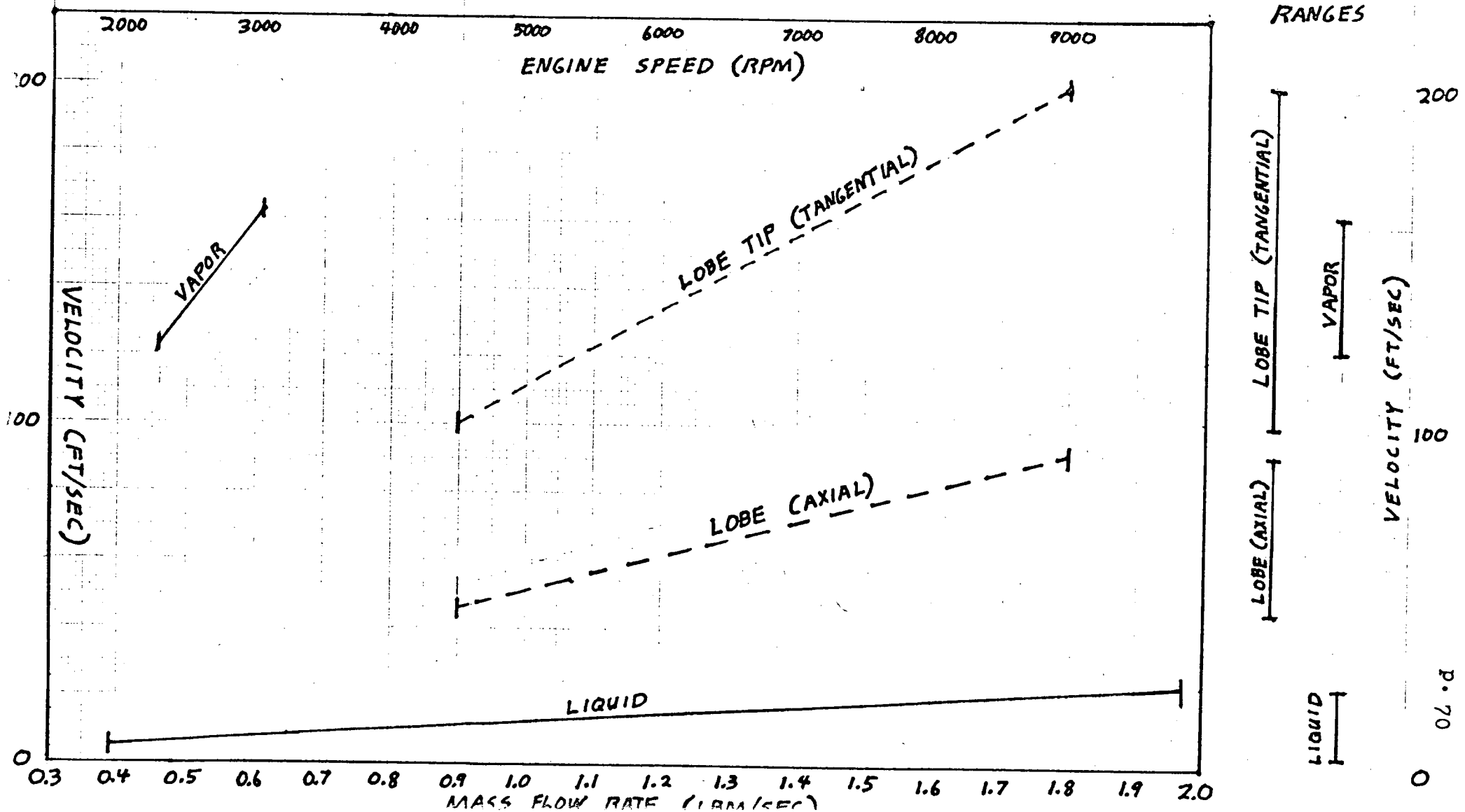
D_l = 0.5 in (WATER NOZZLE)

$A_v \approx$ 2 in (LYSHOLM INLET PORT AREA)

MALE ROTOR DIAMETER = 5.12 in

ROTOR LENGTH = 6.52 in

WRAP ANGLE = 320°



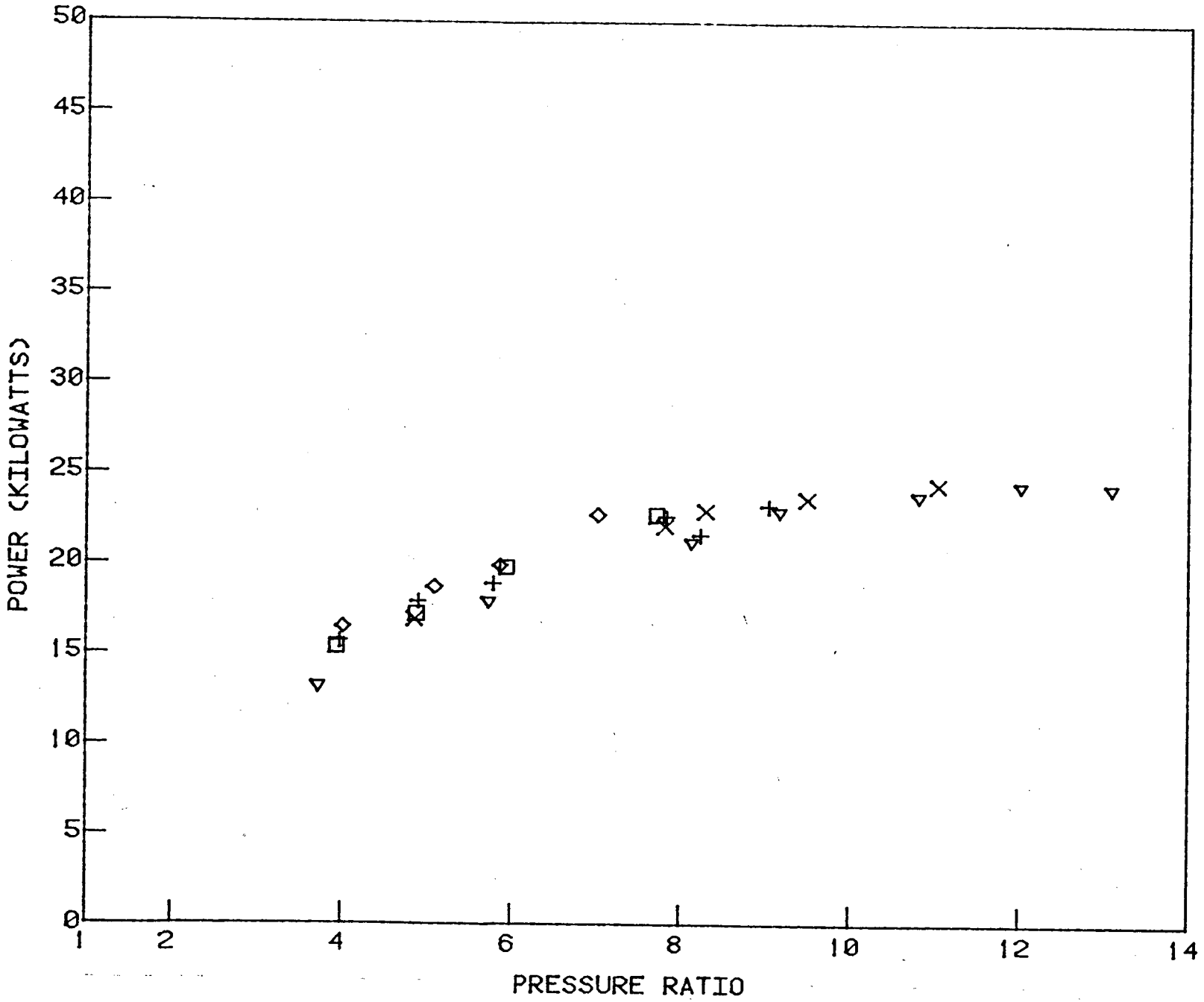


FIGURE 7a - LYSHOLM ENGINE PERFORMANCE

Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
4500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

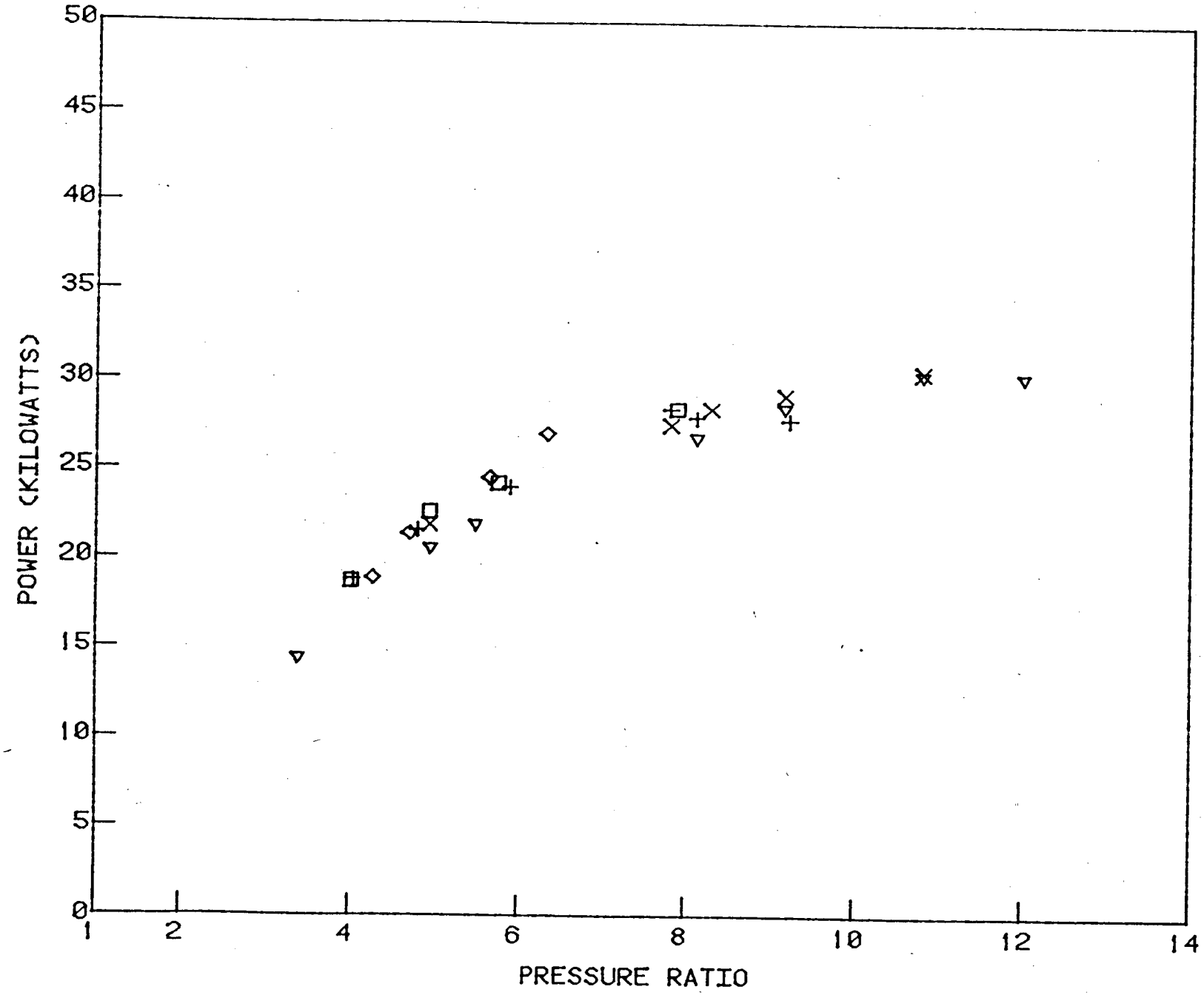


FIGURE 7b - LYSHOLM ENGINE PERFORMANCE

Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
6000 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

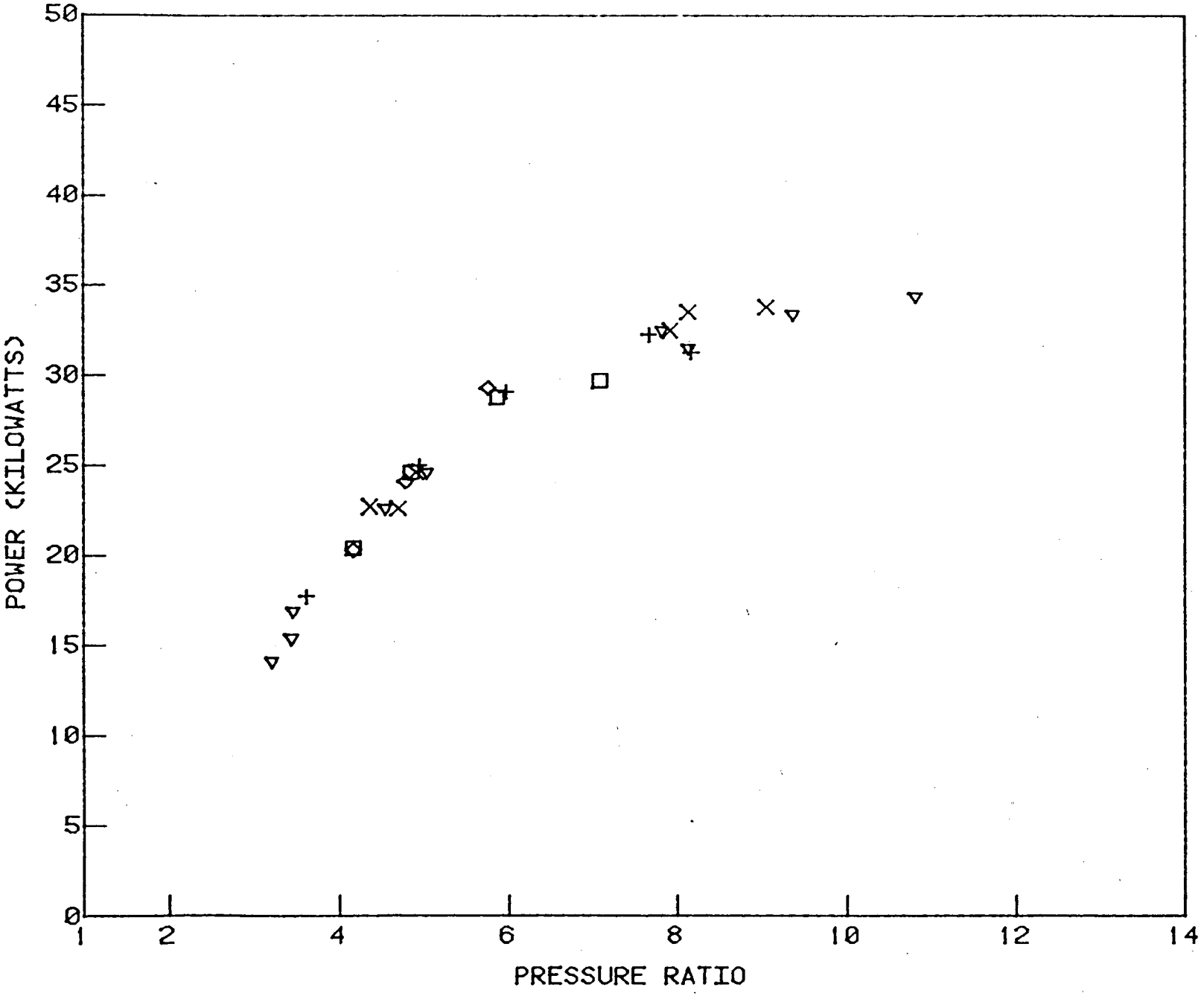


FIGURE 7c - LYSHOLM ENGINE PERFORMANCE

Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
7500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

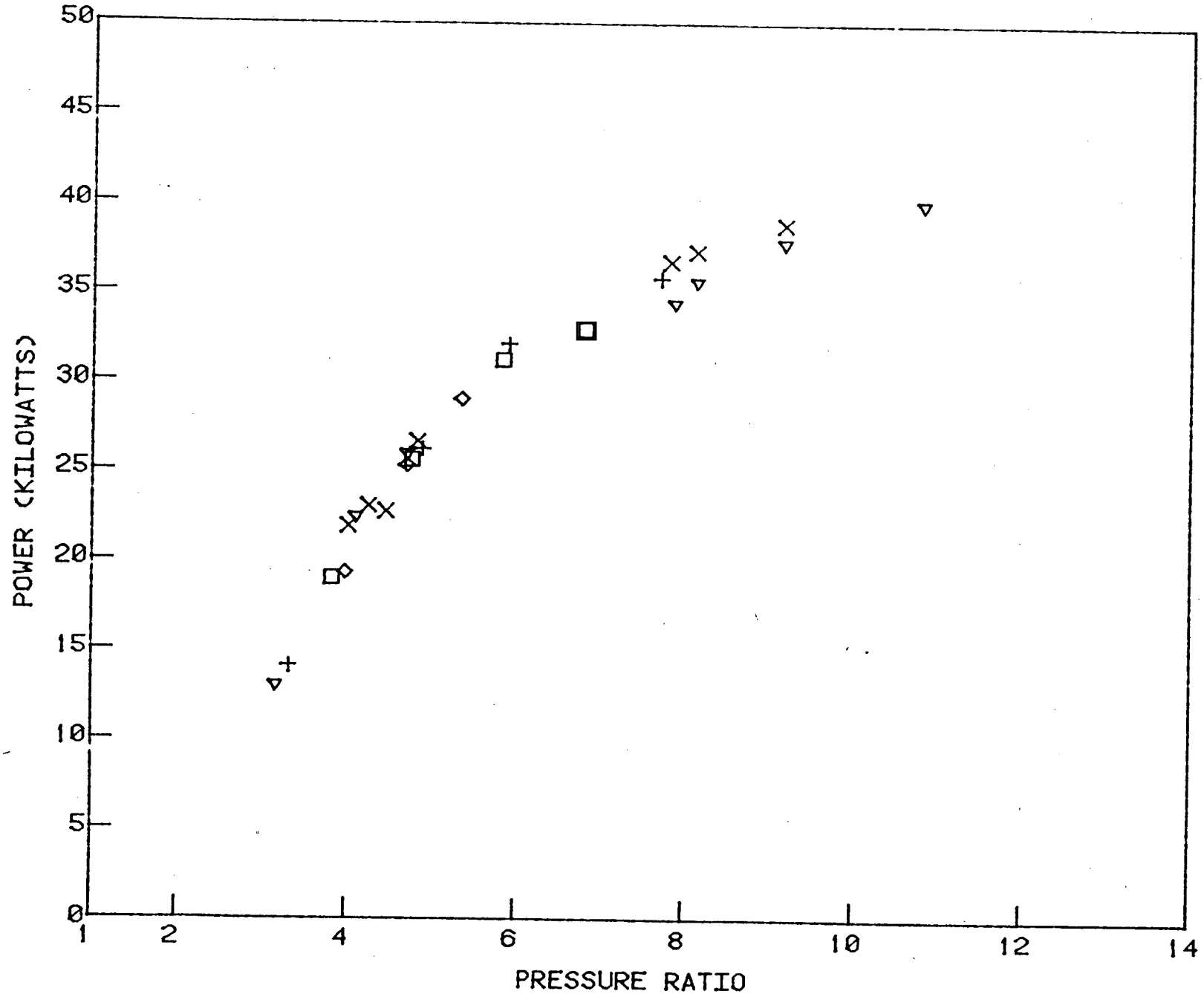


FIGURE 7d - LYSHOLM ENGINE PERFORMANCE

Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
9000 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

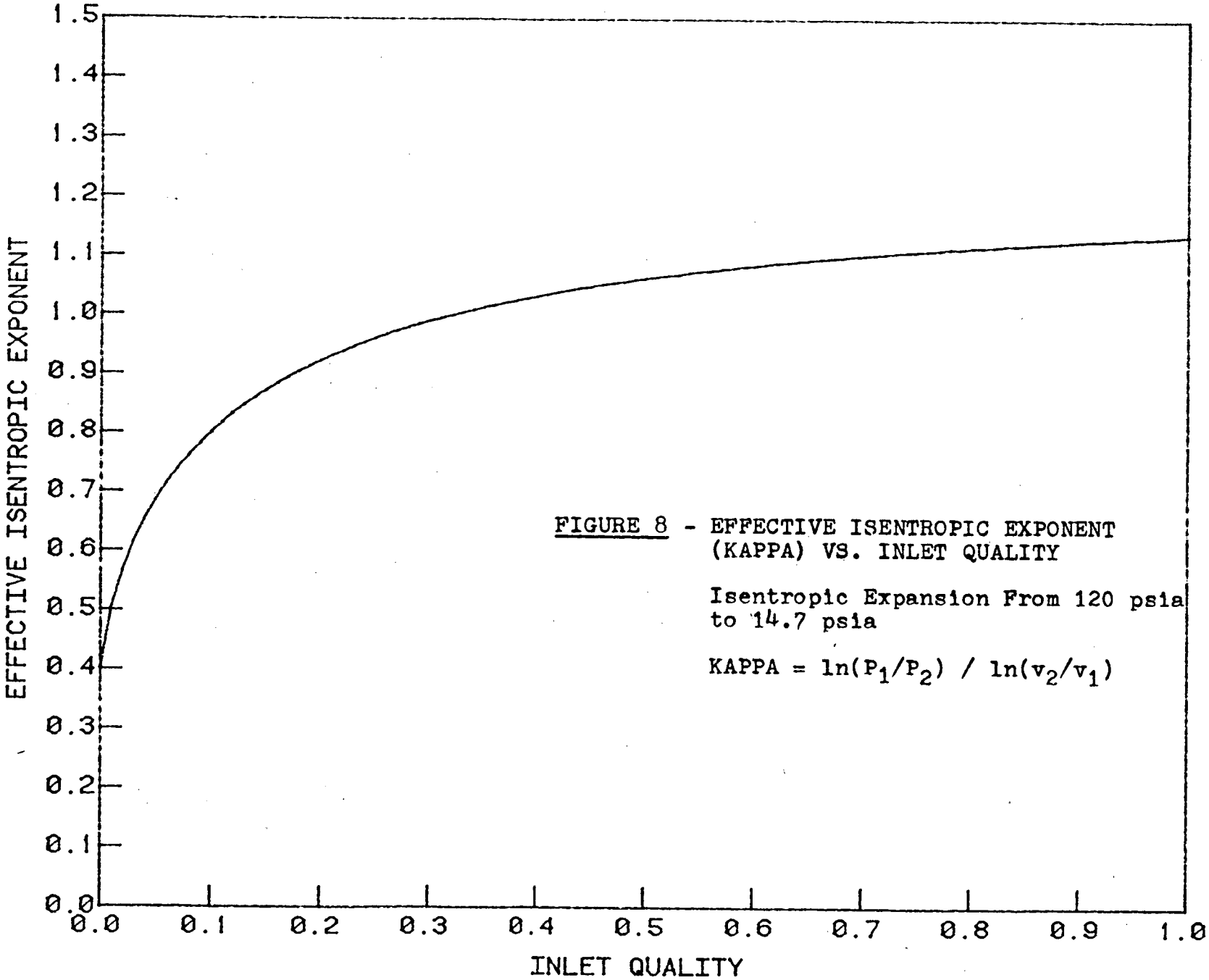


FIGURE 8 - EFFECTIVE ISENTROPIC EXPONENT (KAPPA) VS. INLET QUALITY

Isentropic Expansion From 120 psia to 14.7 psia

KAPPA = $\ln(P_1/P_2) / \ln(v_2/v_1)$

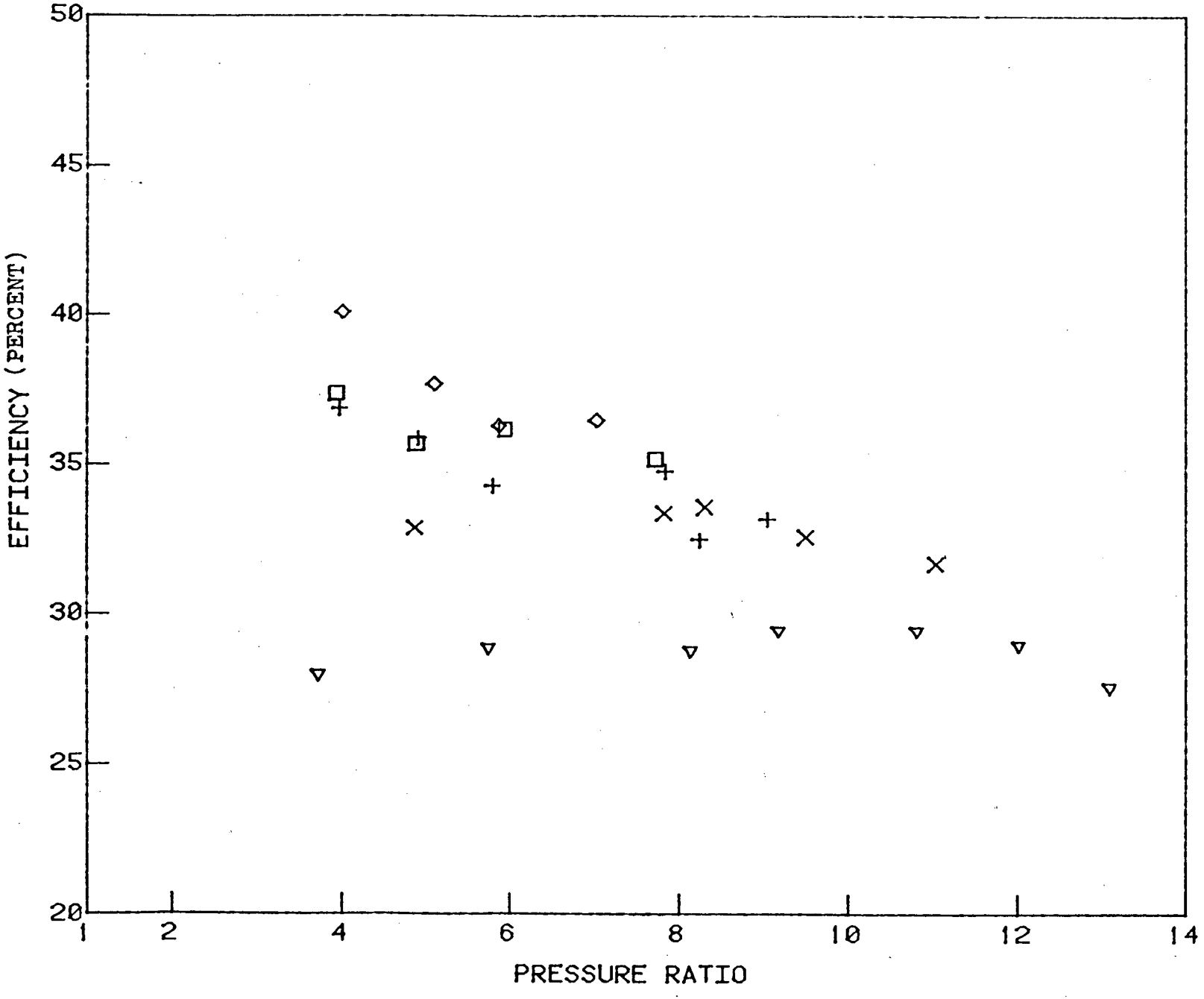


FIGURE 9a - LYSHOLM ENGINE PERFORMANCE

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 4500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

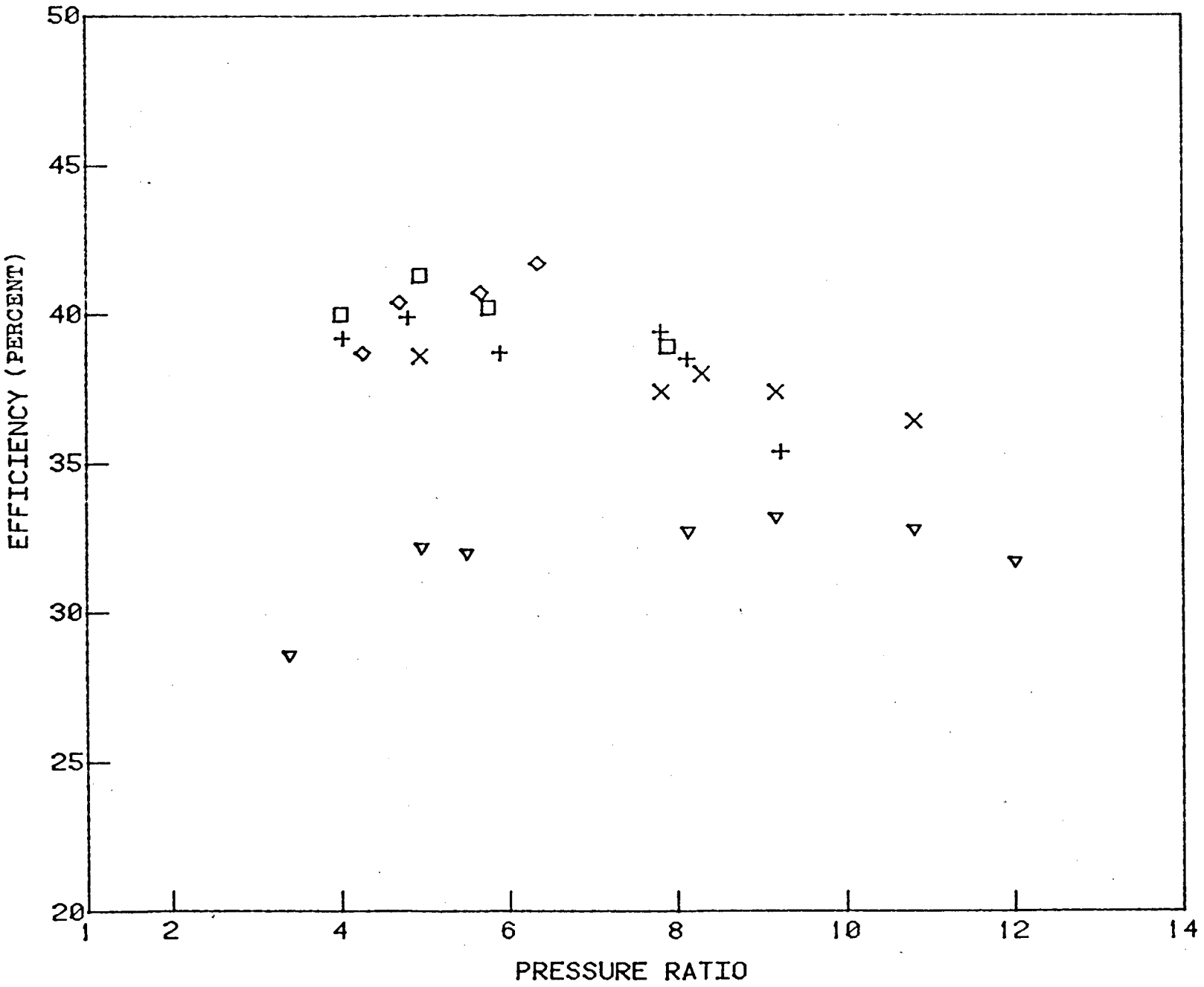


FIGURE 9b - LYSHOLM ENGINE PERFORMANCE

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 6000 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

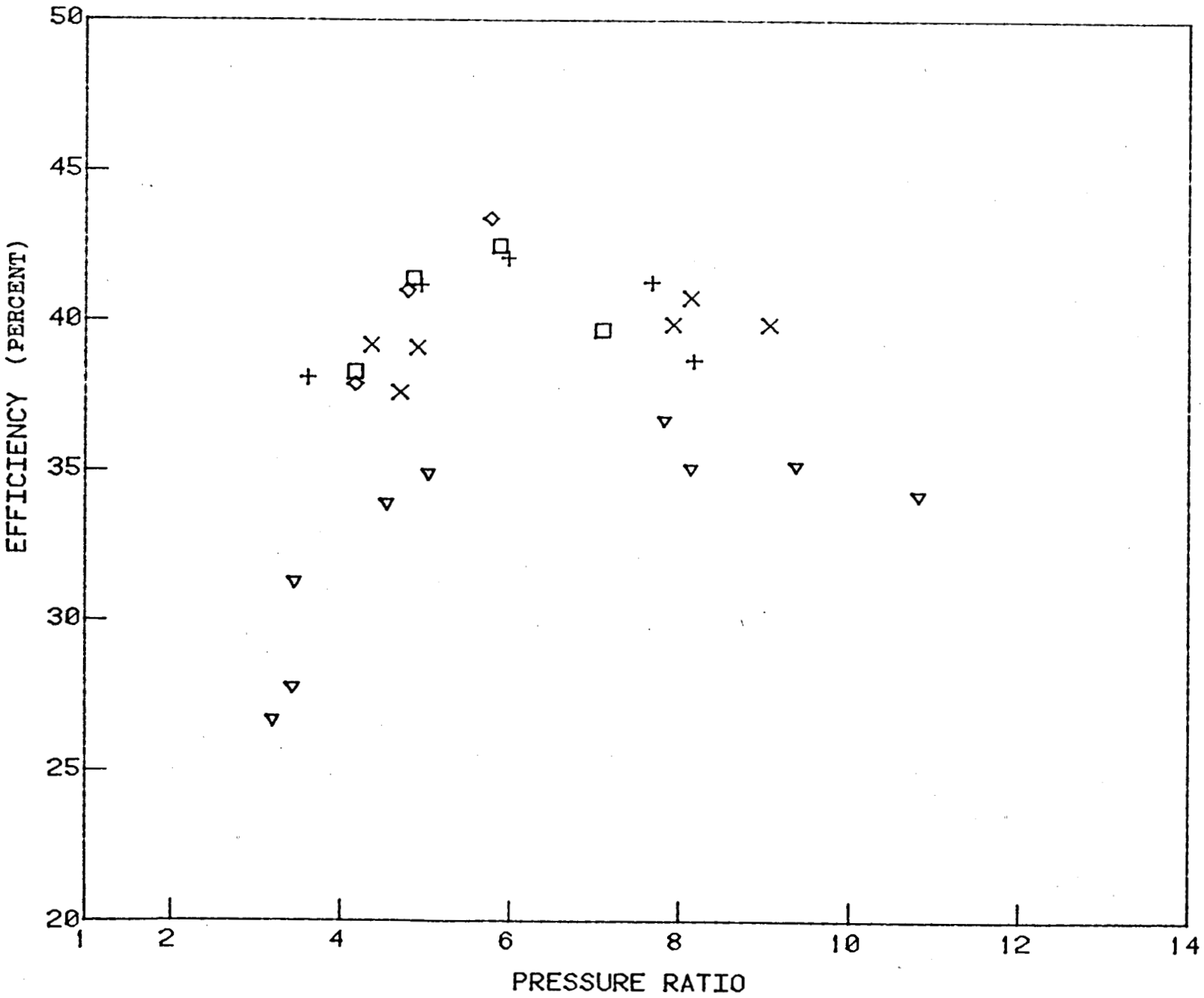


FIGURE 9c - LYSHOLM ENGINE PERFORMANCE

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 7500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

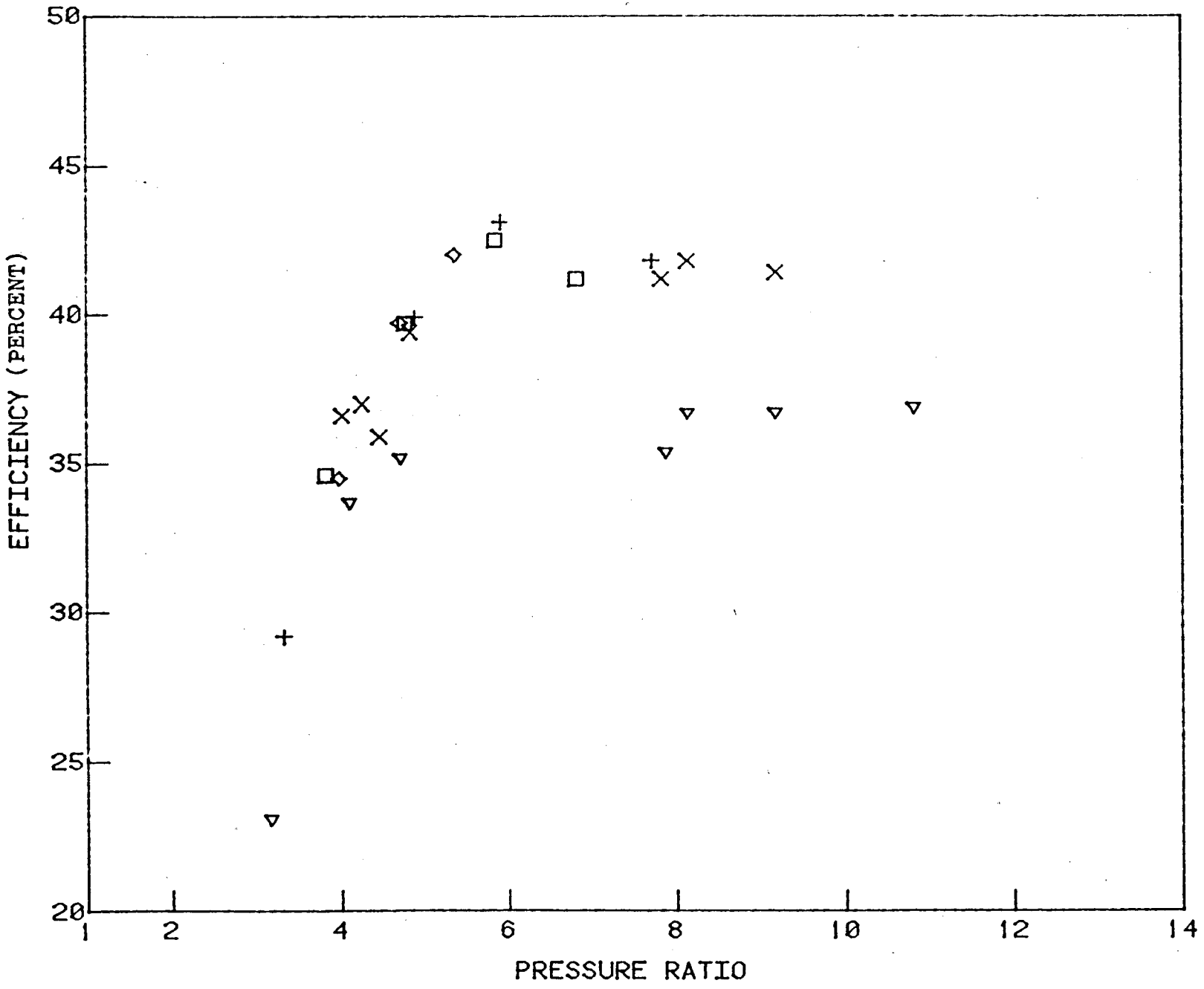


FIGURE 9d - LYSHOLM ENGINE PERFORMANCE

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 9000 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

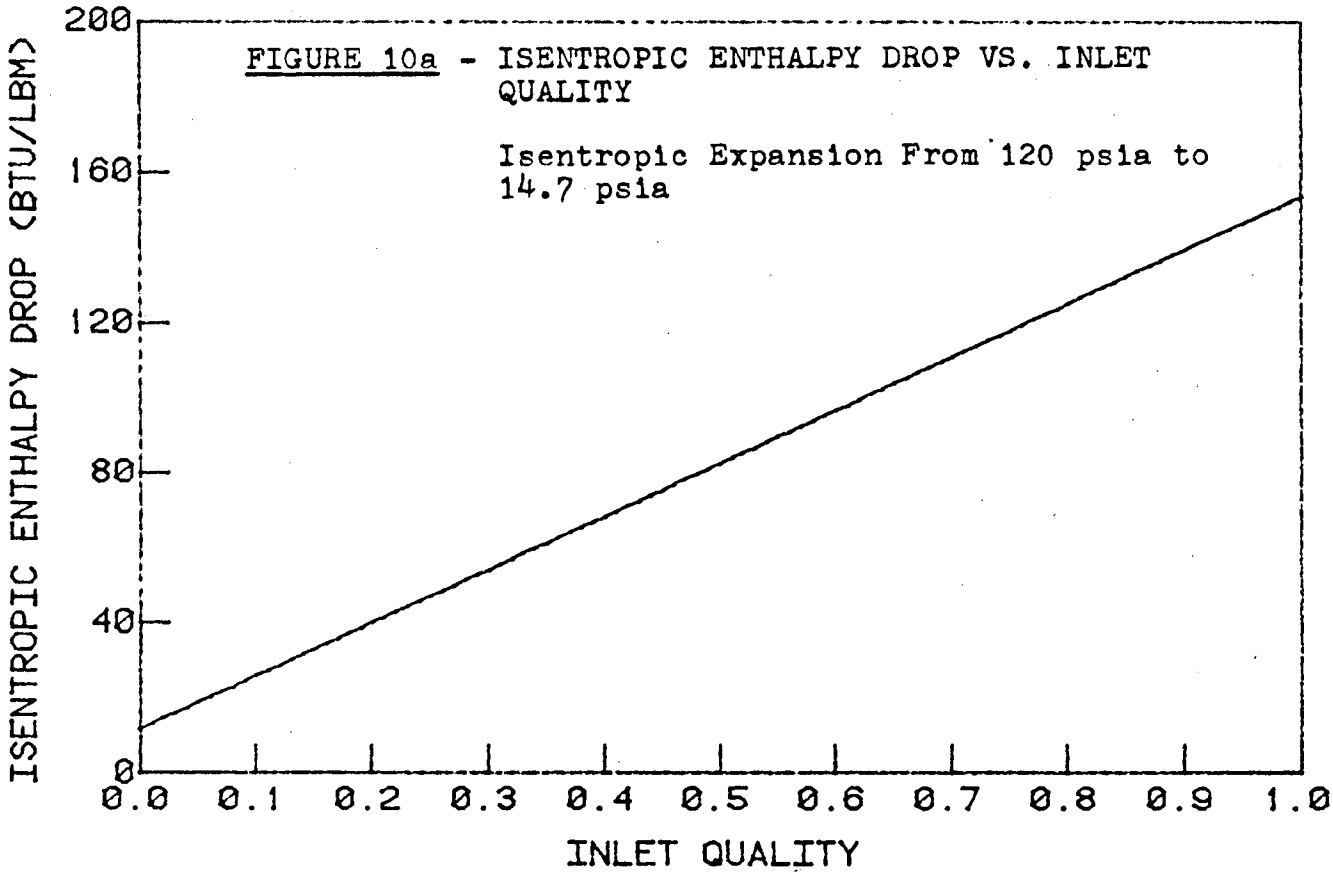


FIGURE 10a - ISENTROPIC ENTHALPY DROP VS. INLET QUALITY

Isentropic Expansion From 120 psia to 14.7 psia

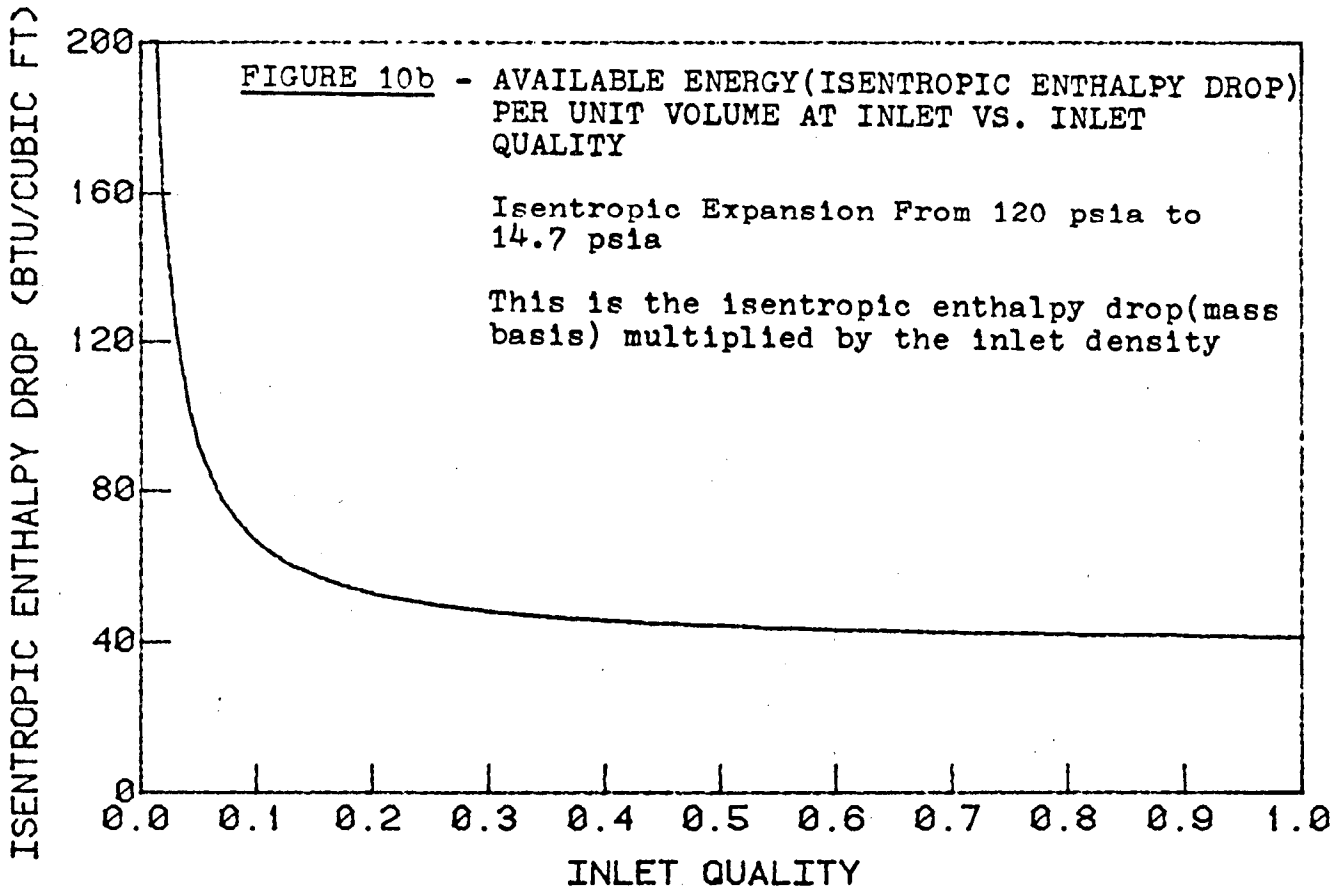


FIGURE 10b - AVAILABLE ENERGY(ISENTROPIC ENTHALPY DROP) PER UNIT VOLUME AT INLET VS. INLET QUALITY

Isentropic Expansion From 120 psia to 14.7 psia

This is the isentropic enthalpy drop(mass basis) multiplied by the inlet density

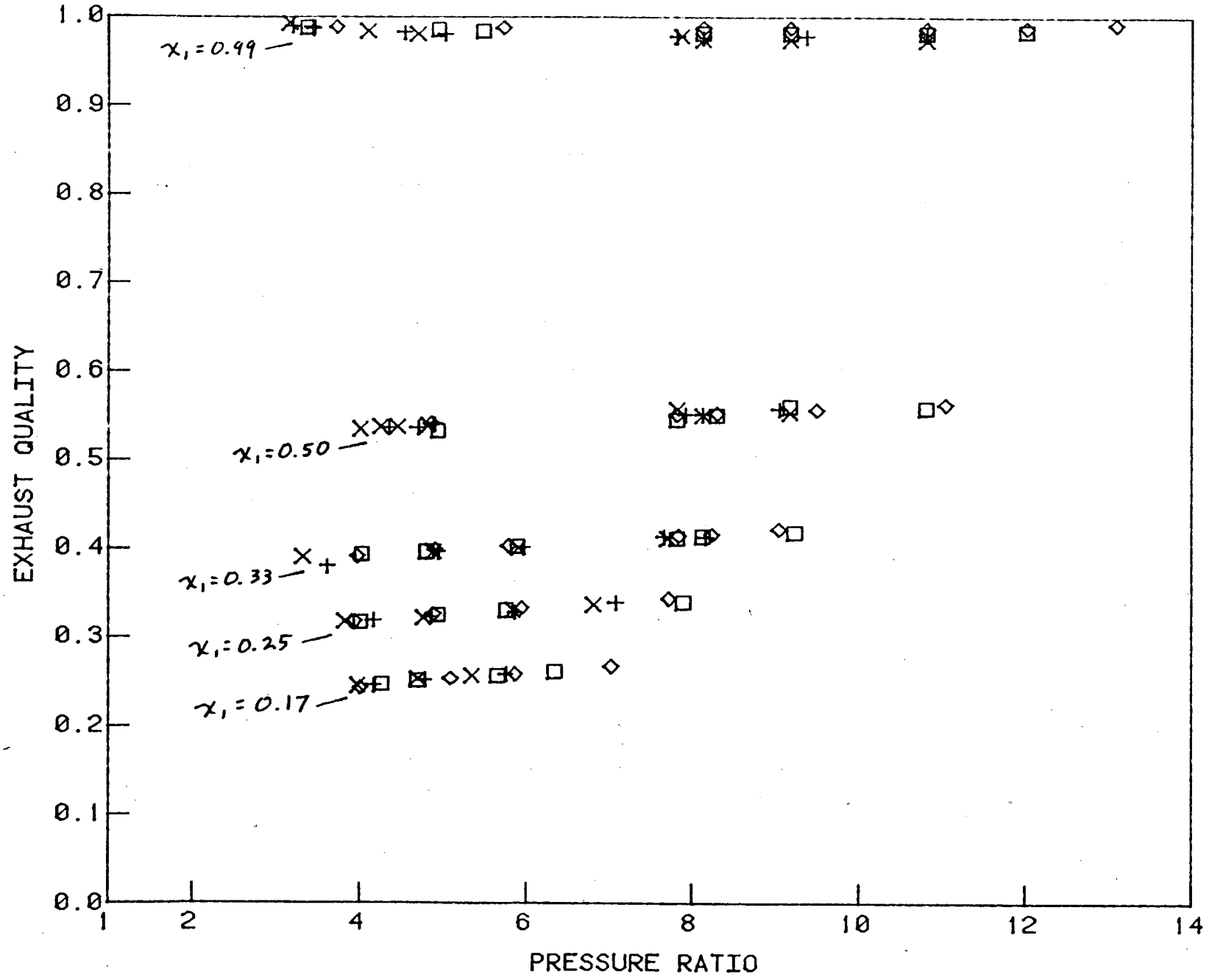


FIGURE 11 - LYSHOLM ENGINE PERFORMANCE

Exhaust Quality vs. Pressure Ratio

Inlet Pressure 120 psia

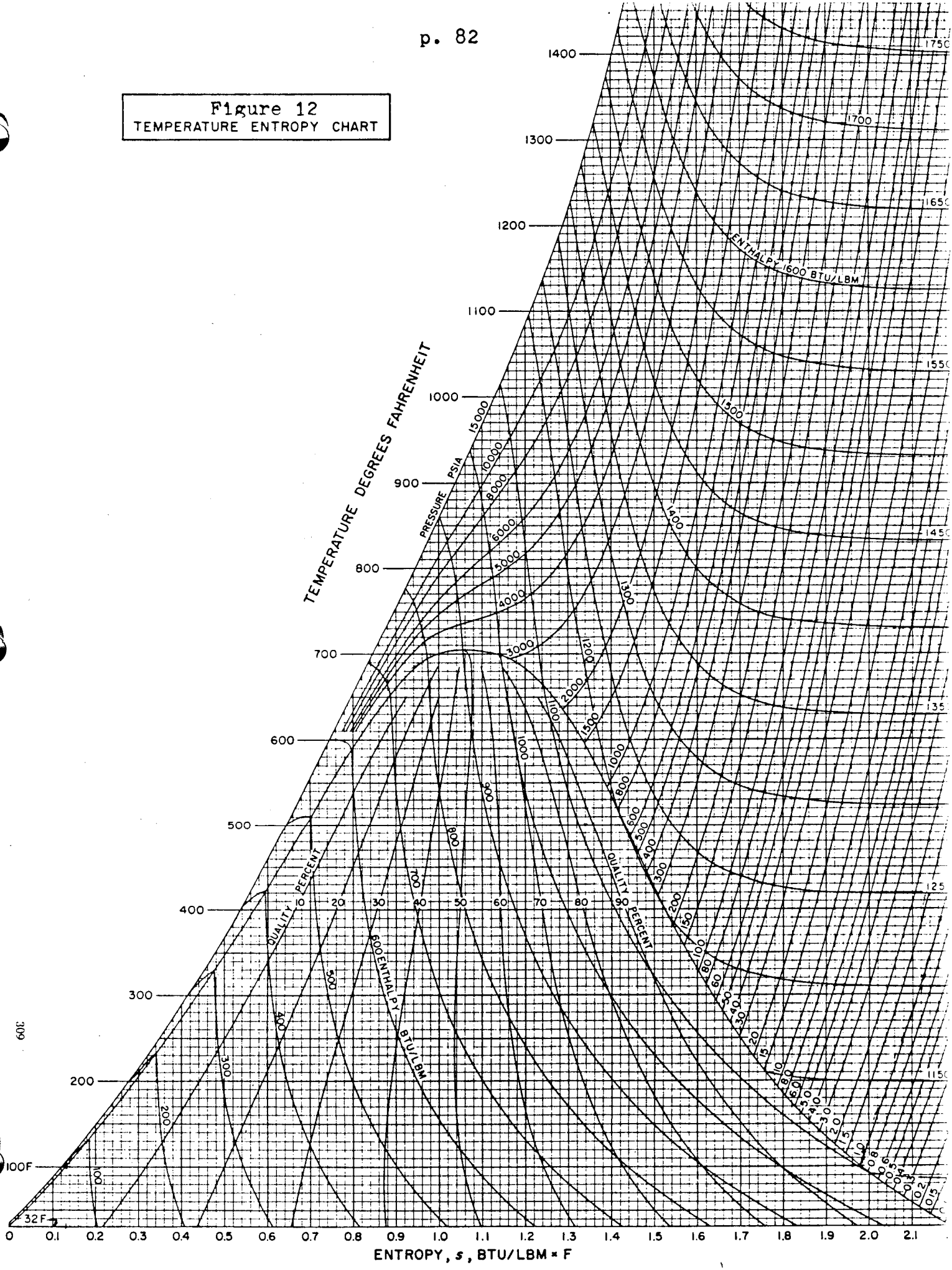
◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

Figure 12
TEMPERATURE ENTROPY CHART



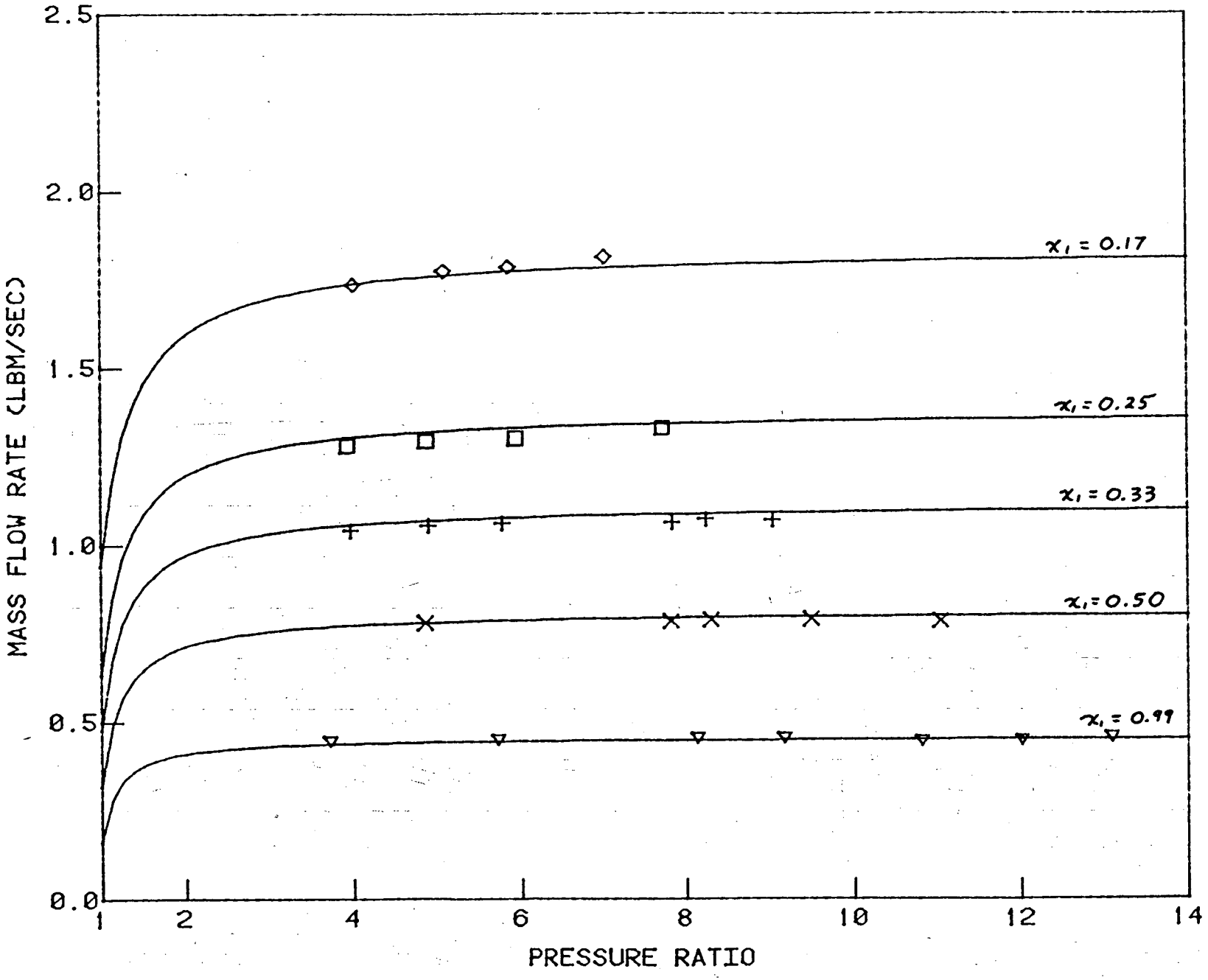


FIGURE 13a - LYSHOLM ENGINE MODEL

Mass Flow Rate
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
4500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

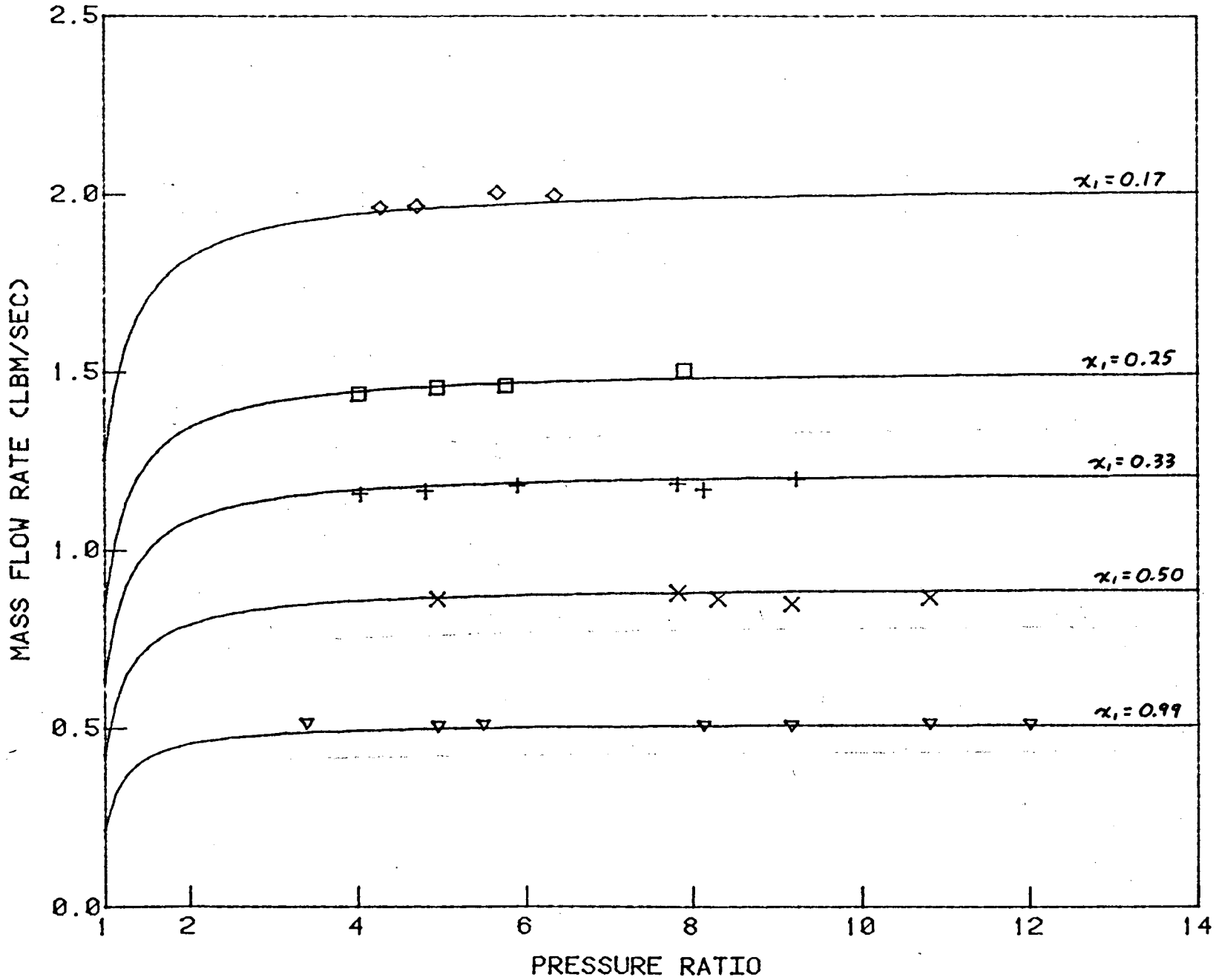


FIGURE 13b - LYSHOLM ENGINE MODEL

Mass Flow Rate vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 6000 RPM

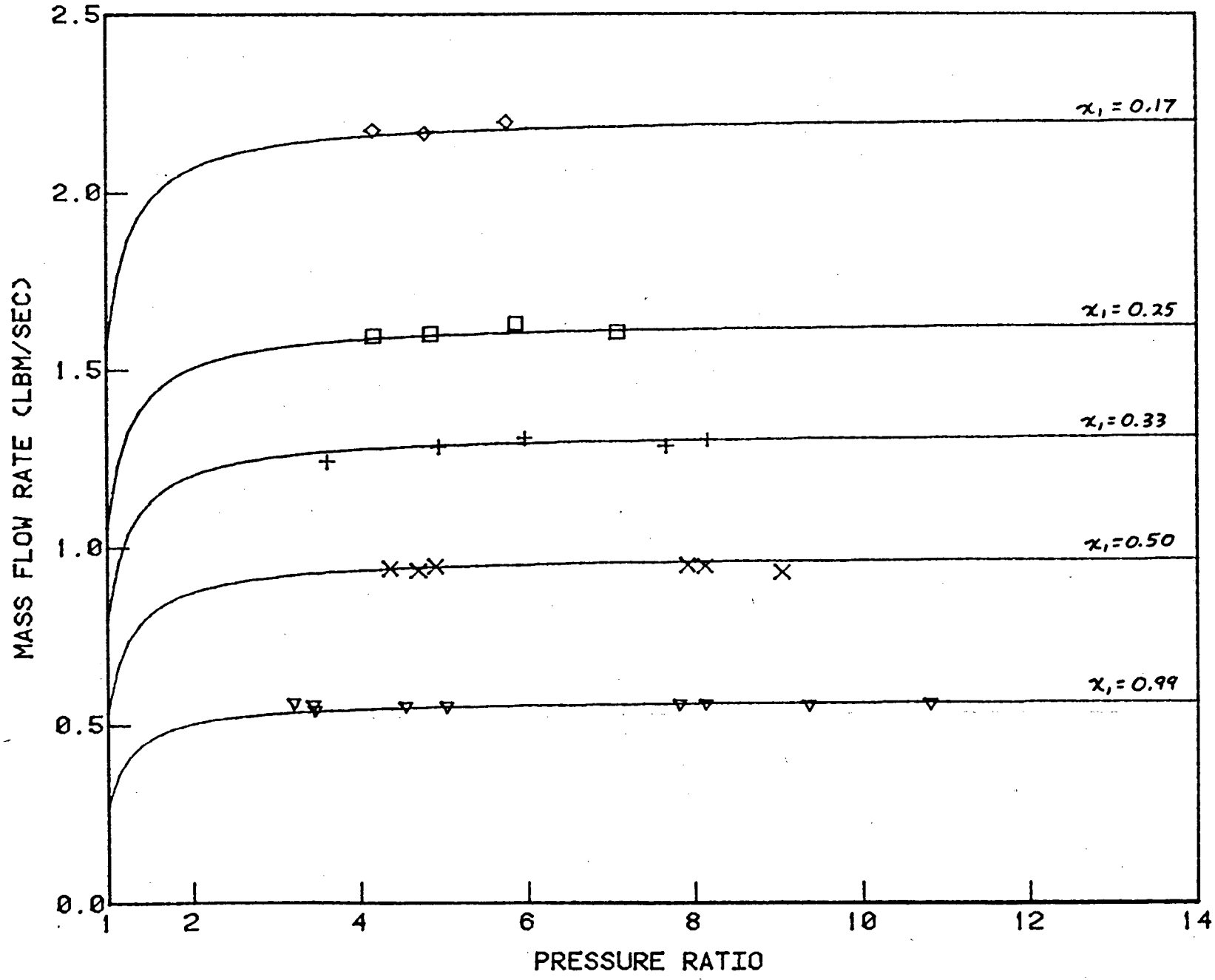
▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17



Mass Flow Rate
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
7500 RPM

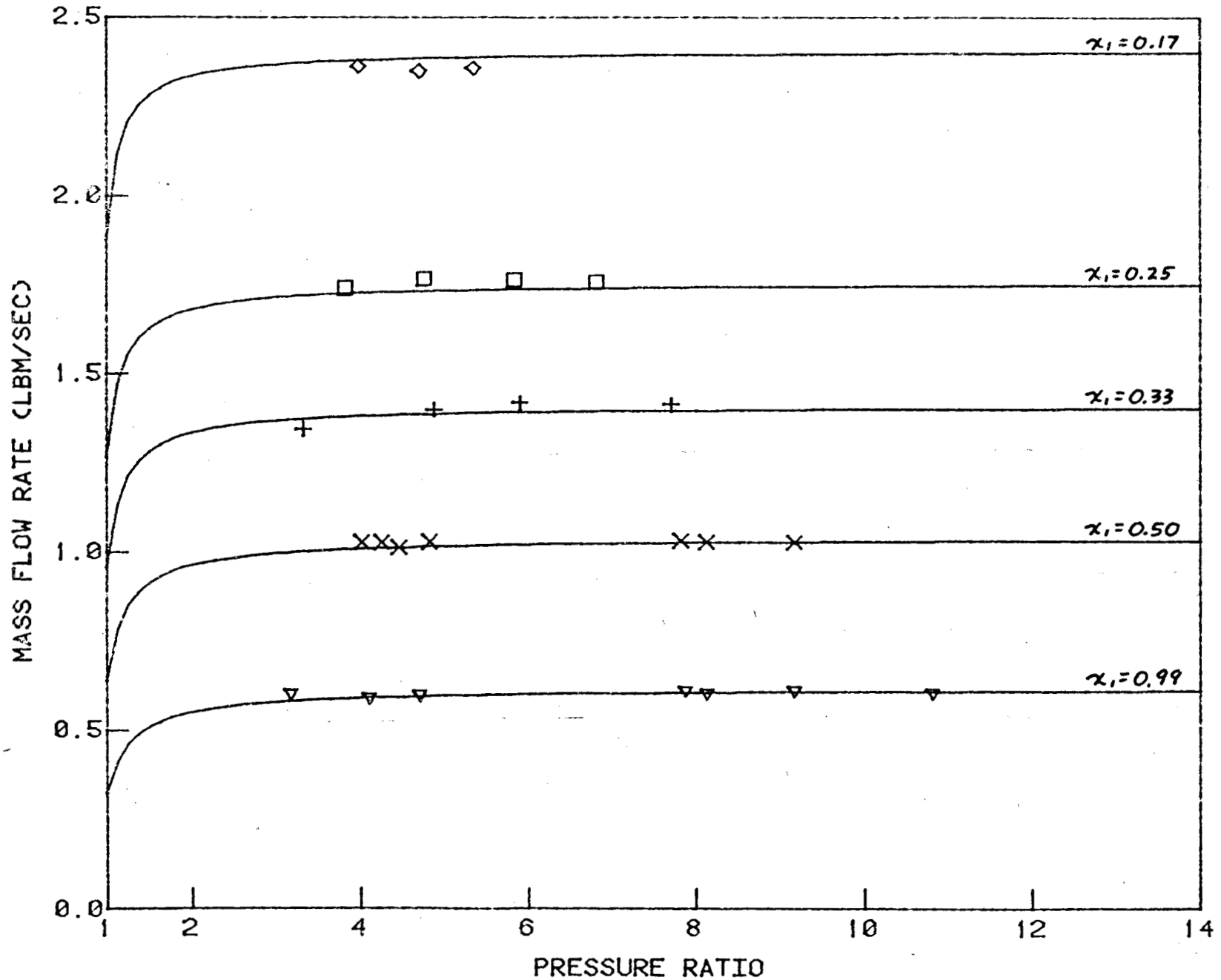
▽ – x1 = 0.99

X – x1 = 0.50

+ – x1 = 0.33

□ – x1 = 0.25

◇ – x1 = 0.17



Mass Flow Rate
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
9000RPM

▽ – x1 = 0.99

X – x1 = 0.50

+ – x1 = 0.33

□ – x1 = 0.25

◇ – x1 = 0.17

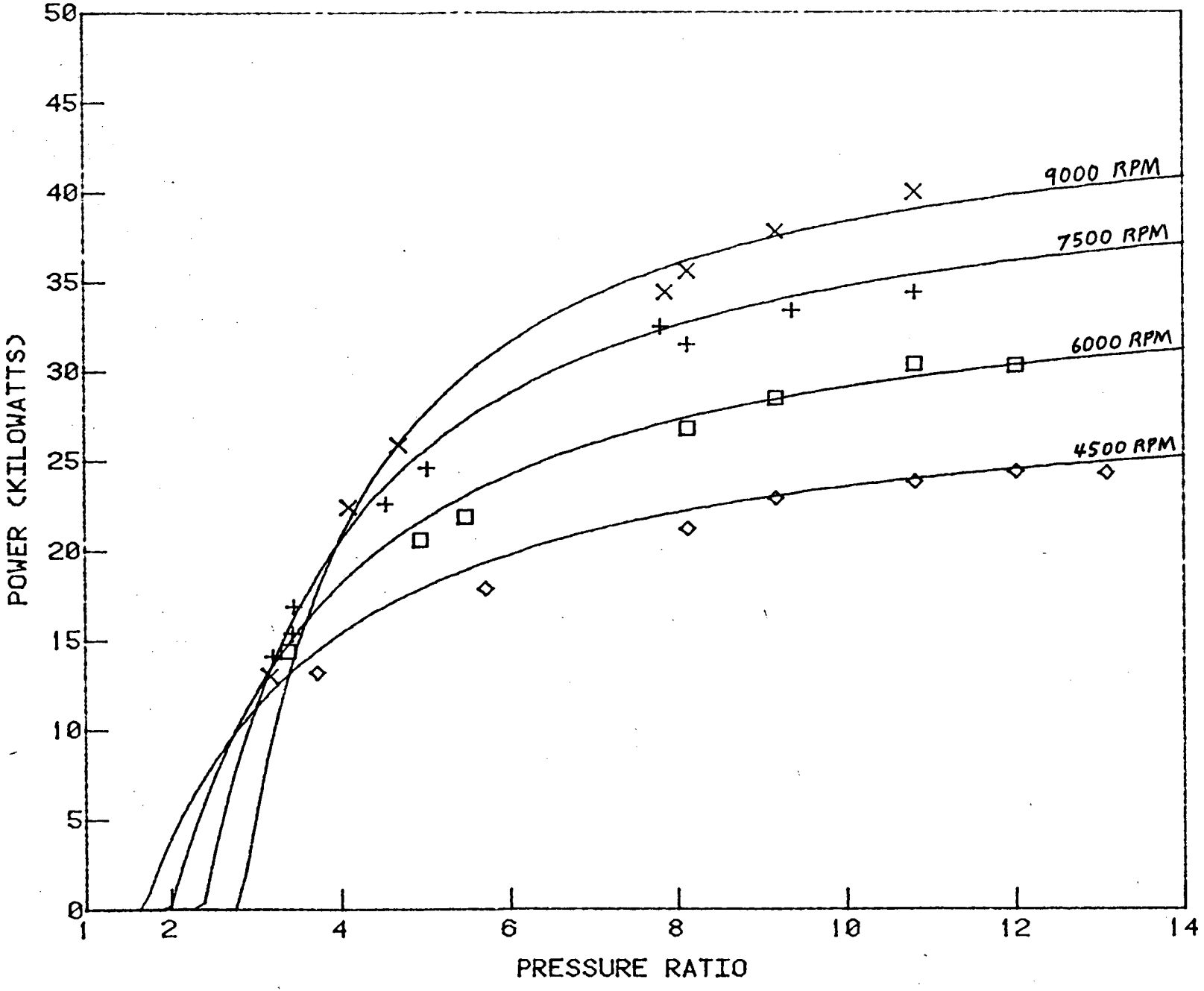


FIGURE 14a - LYSHOLM ENGINE MODEL

Power vs. Pressure Ratio

Inlet Pressure 120 psia

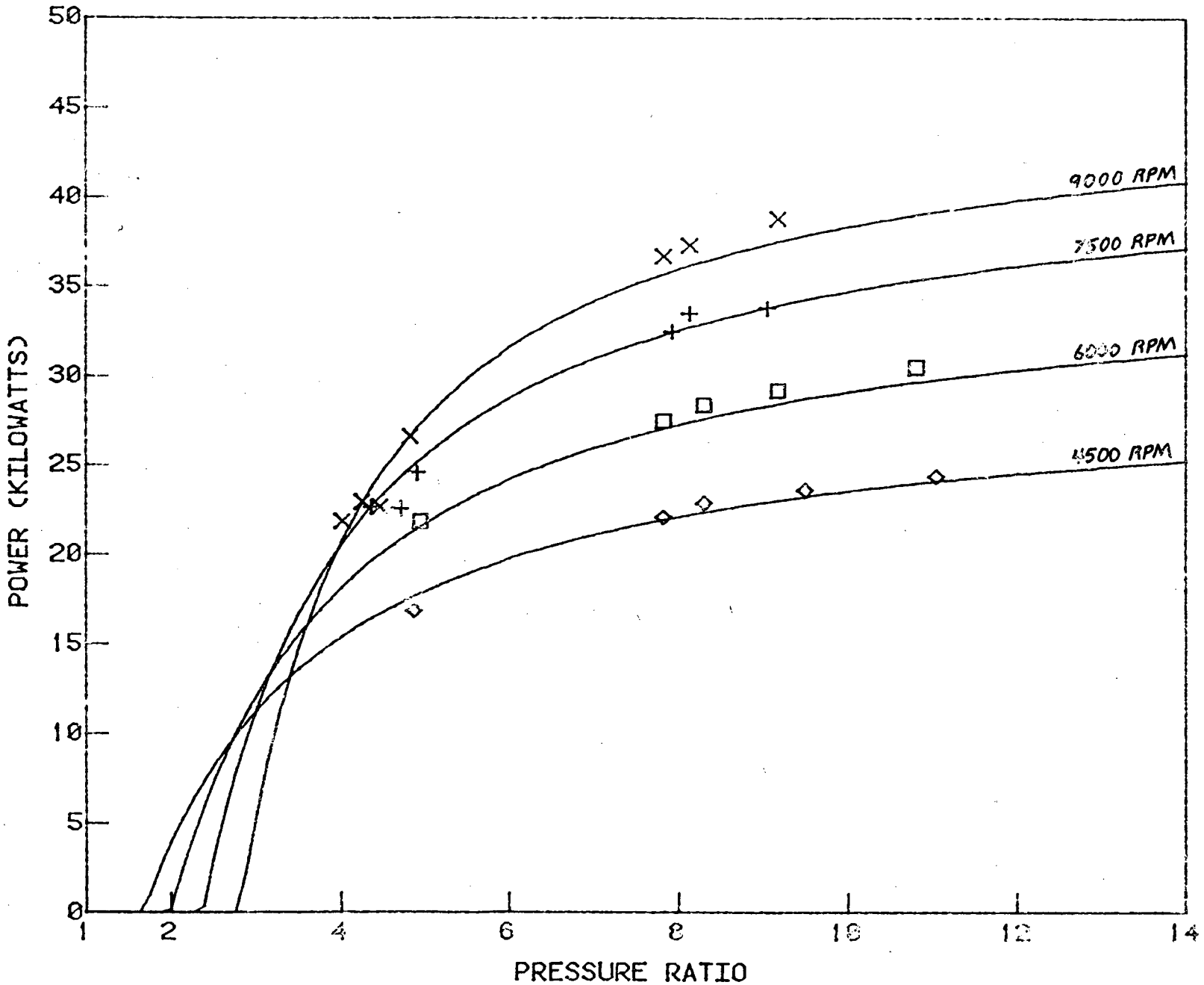
Inlet Quality 0.99

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

× - 9000 RPM



Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

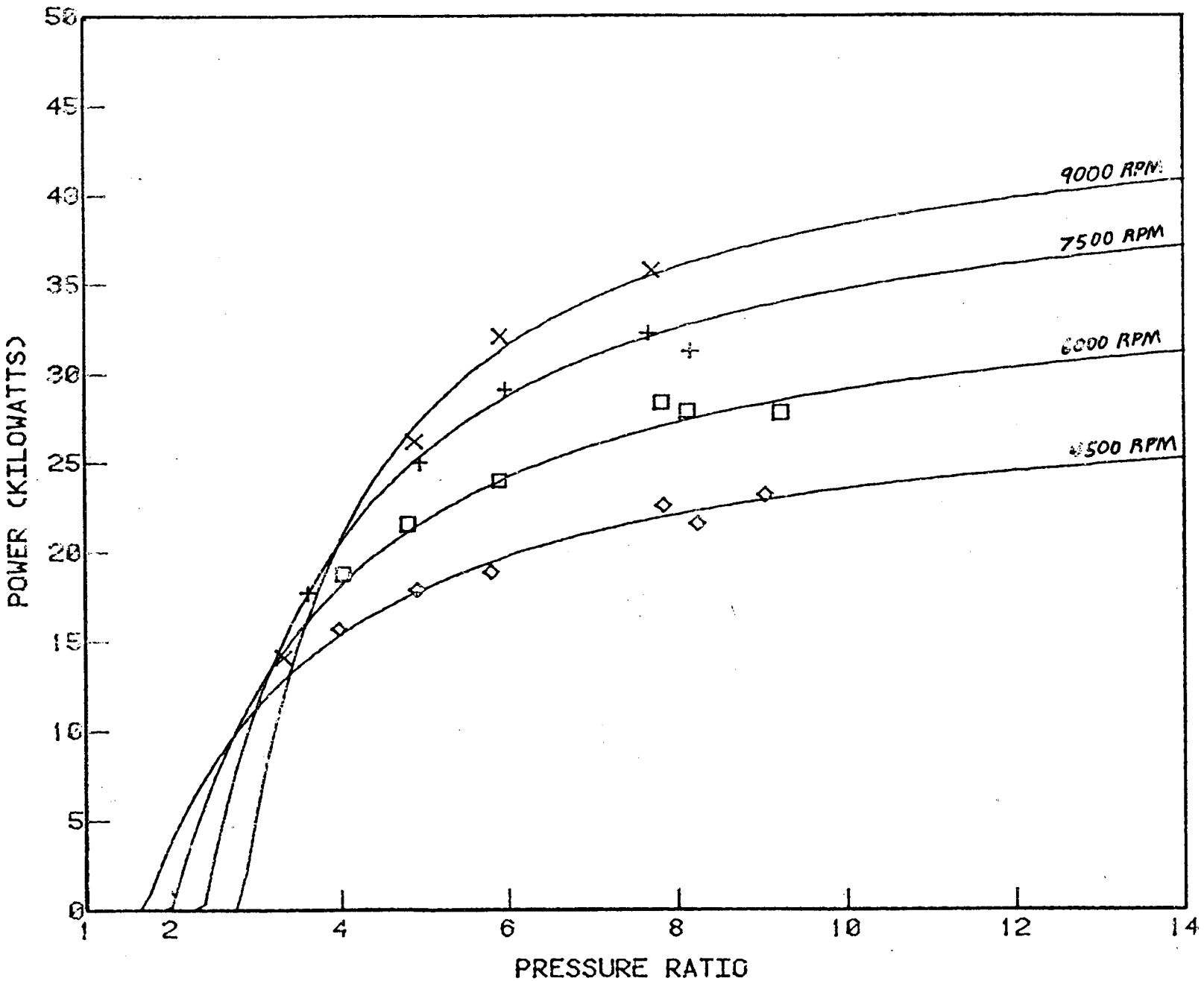
Inlet Quality
0.50

◇ – 4500 RPM

□ – 6000 RPM

+ – 7500 RPM

X – 9000 RPM



Power
vs.
Pressure Ratio

Inlet Pressure
120 psia

Inlet Quality
0.33

◇ – 4500 RPM

□ – 6000 RPM

+ – 7500 RPM

X – 9000 RPM

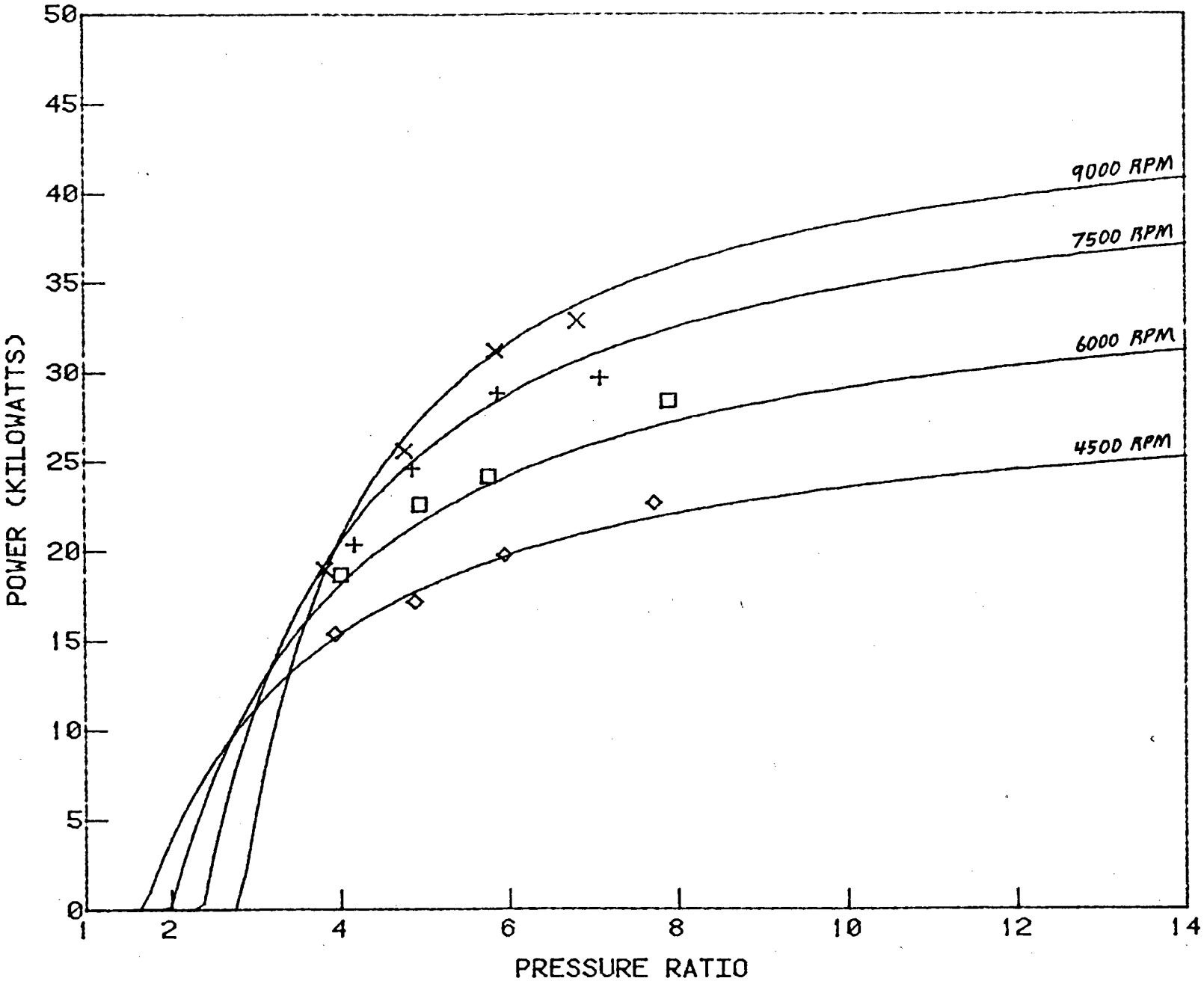


FIGURE 14d - LYSHOLM ENGINE MODEL

Power vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.25

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

× - 9000 RPM

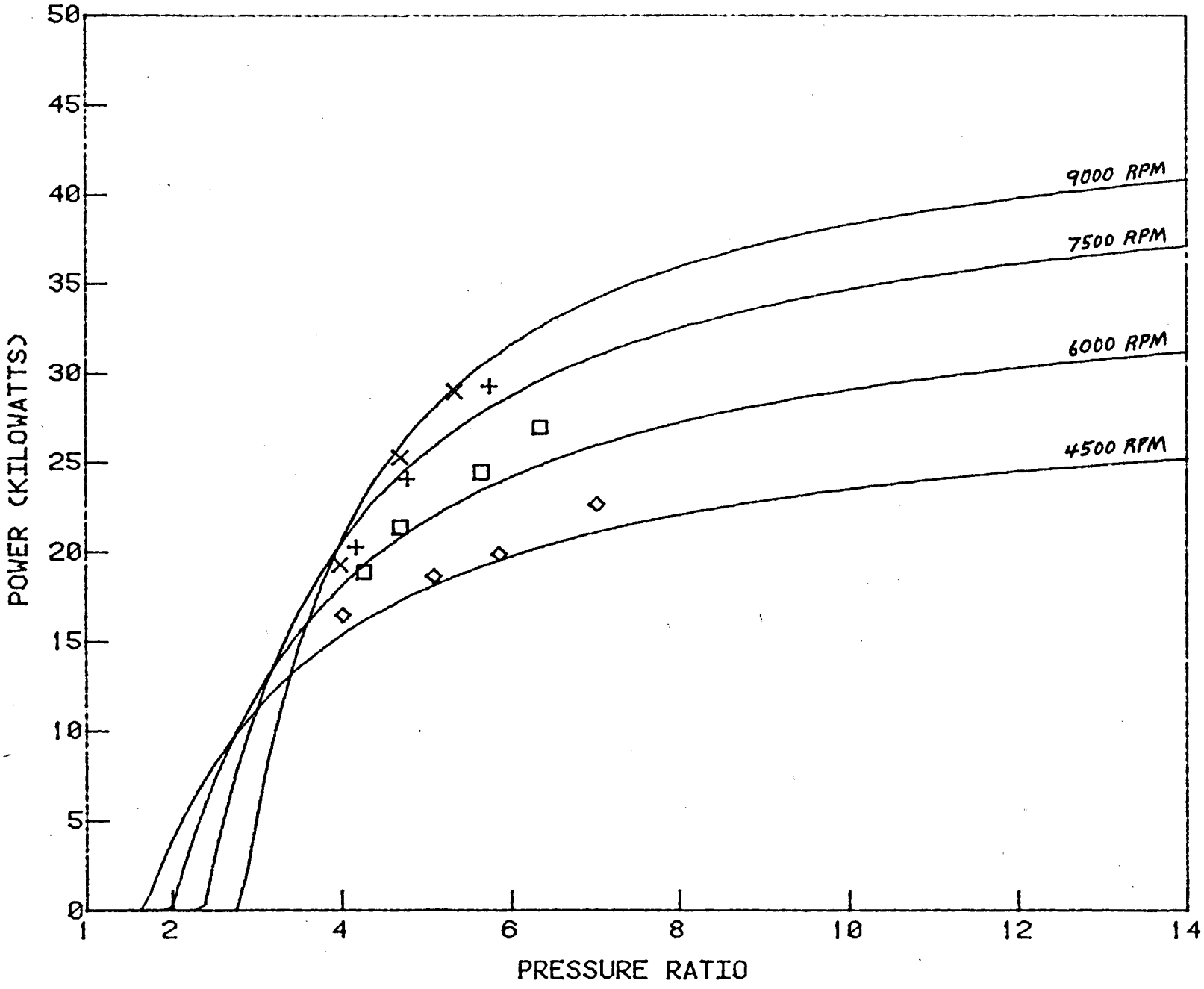


FIGURE 14e - LYSHOLM ENGINE MODEL

Power vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.17

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

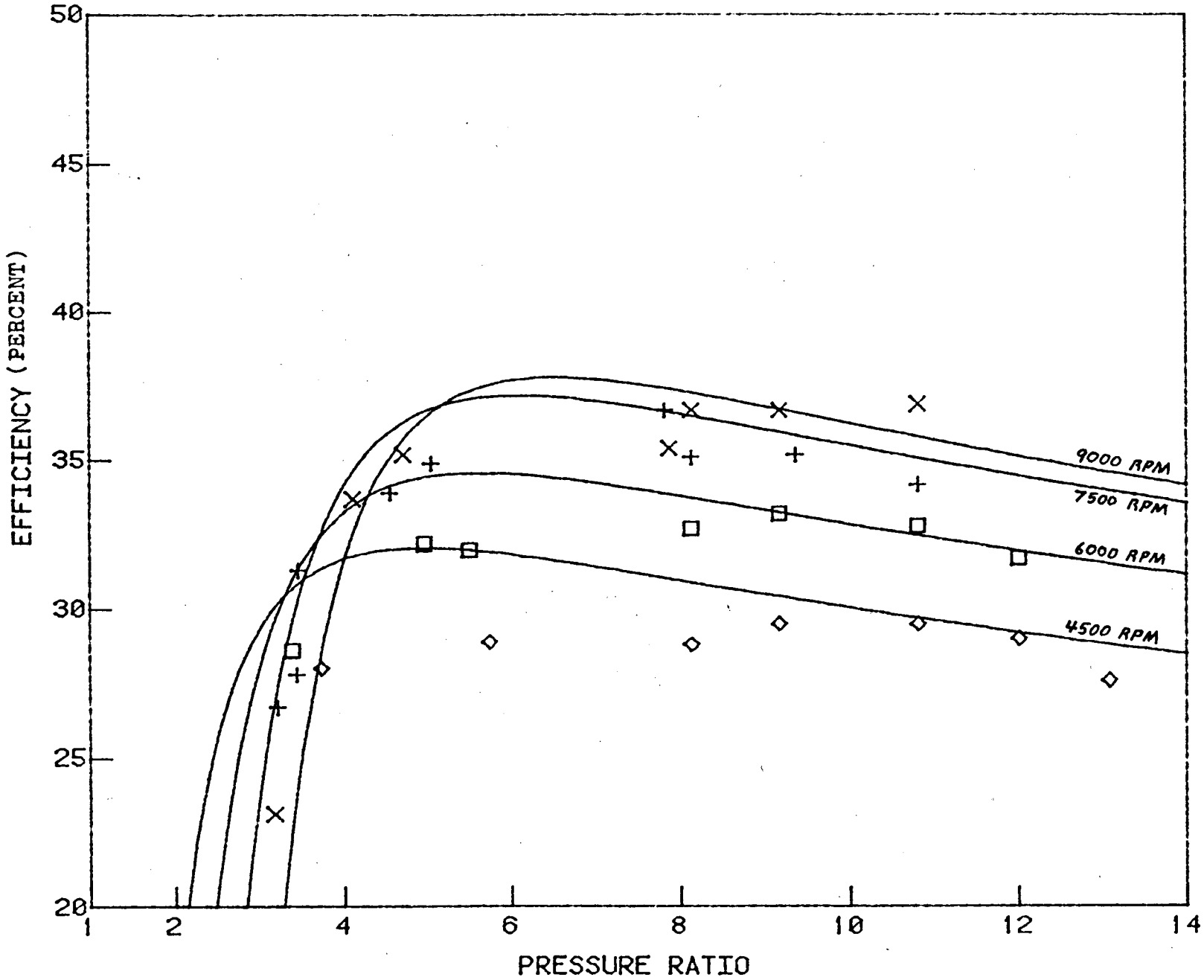


FIGURE 15a - LYSHOLM ENGINE MODEL

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.99

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

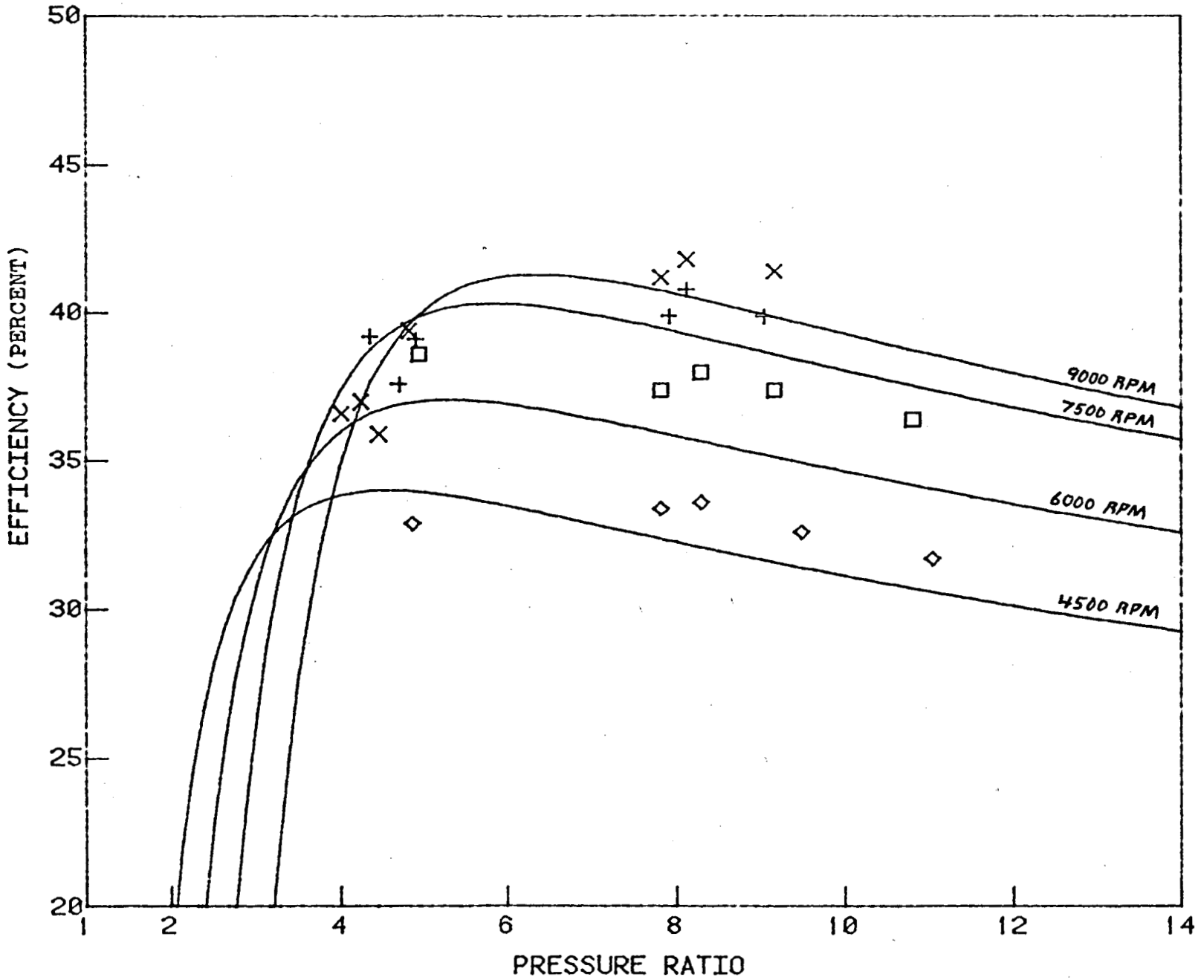


FIGURE 15b - LYSHOLM ENGINE MODEL

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.50

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

× - 9000 RPM

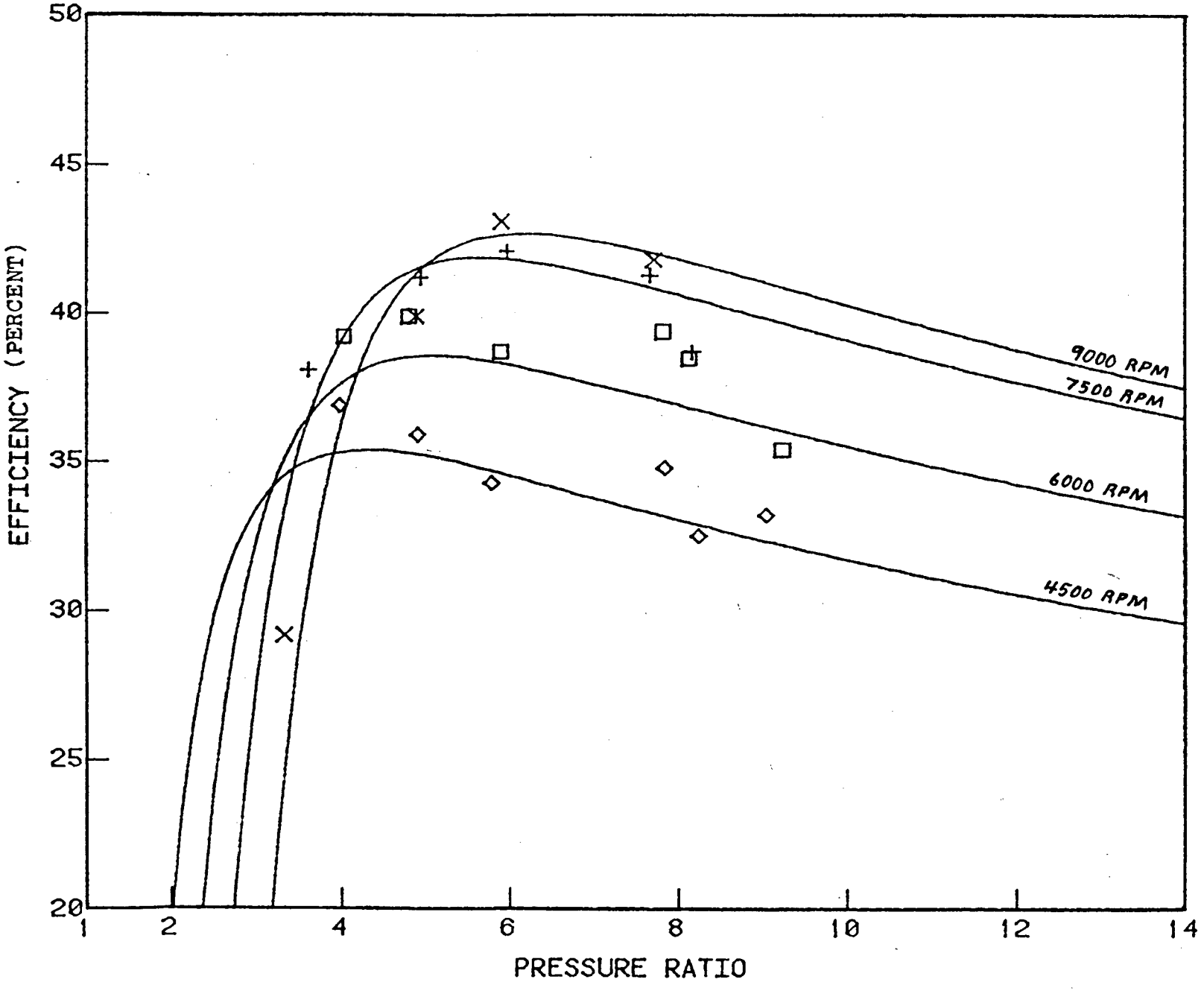


FIGURE 15c - LYSHOLM ENGINE MODEL

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.33

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

× - 9000 RPM

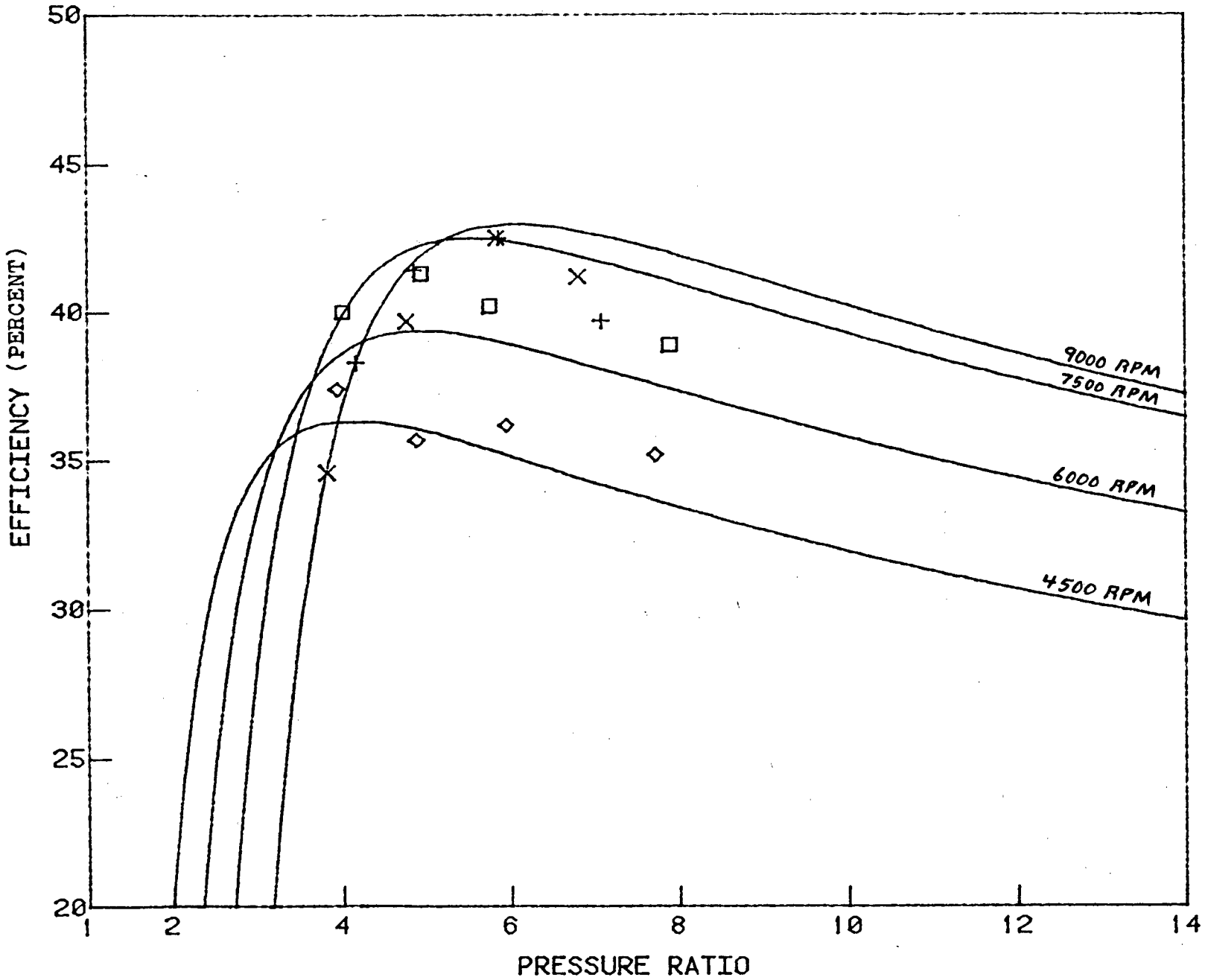


FIGURE 15d - LYSHOLM ENGINE MODEL

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.25

◇ - 4500 RPM

□ - 6000 RPM

+ - 7500 RPM

X - 9000 RPM

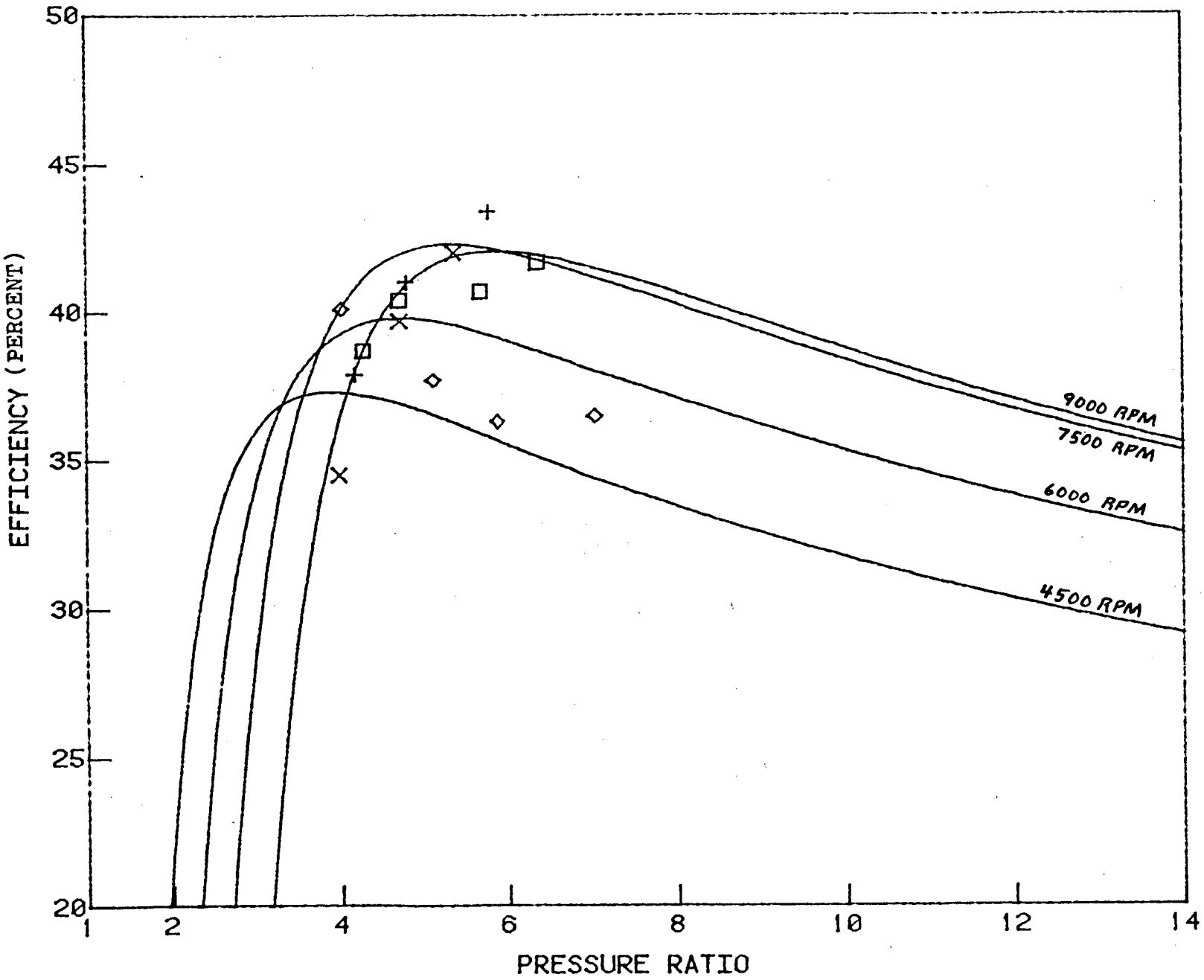


FIGURE 15e - LYSHOLM ENGINE MODEL

Efficiency vs. Pressure Ratio

Inlet Pressure 120 psia

Inlet Quality 0.17

◇ – 4500 RPM

□ – 6000 RPM

+ – 7500 RPM

X – 9000 RPM

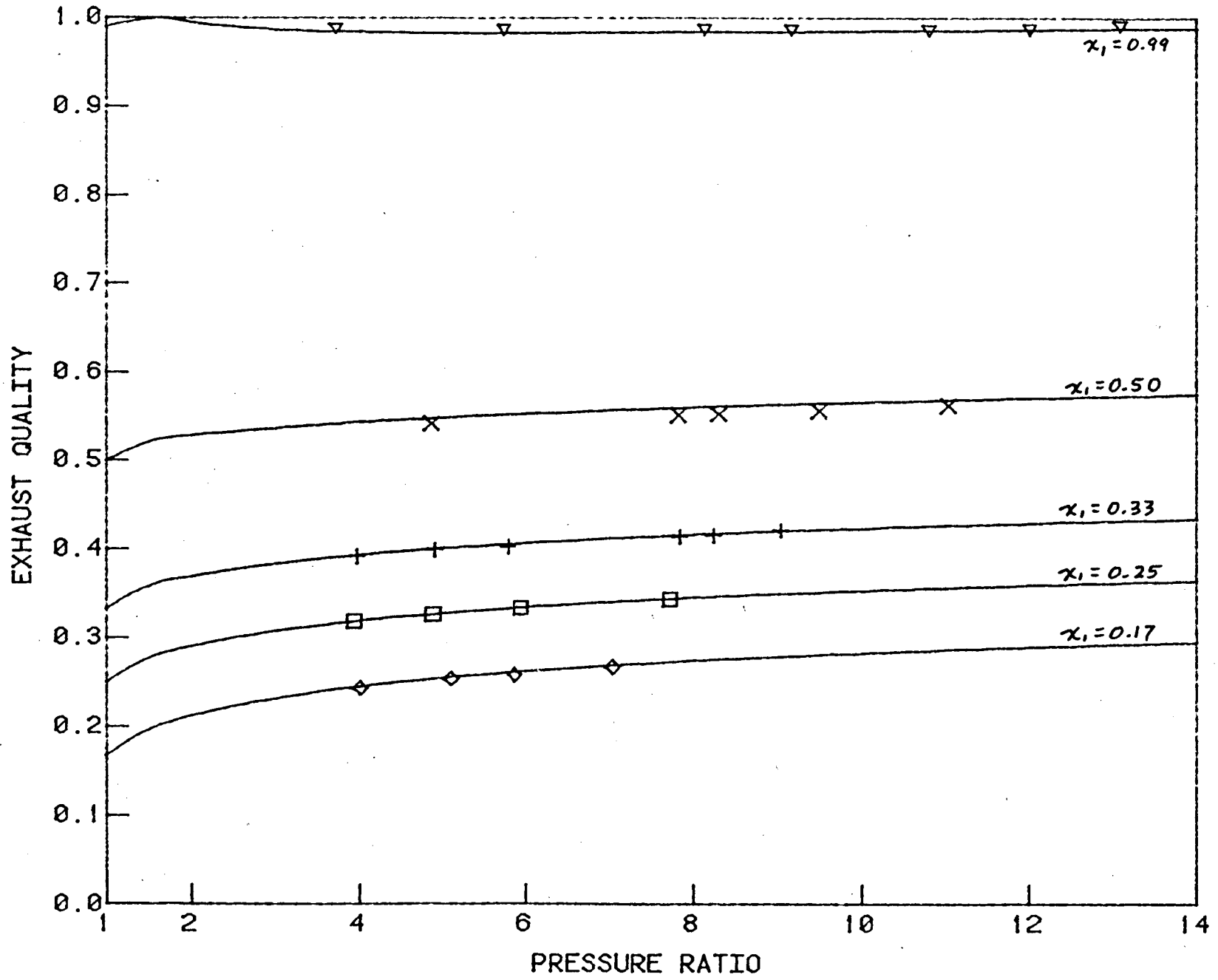


FIGURE 16a - LYSHOLM ENGINE MODEL

Exhaust Quality vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 4500 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

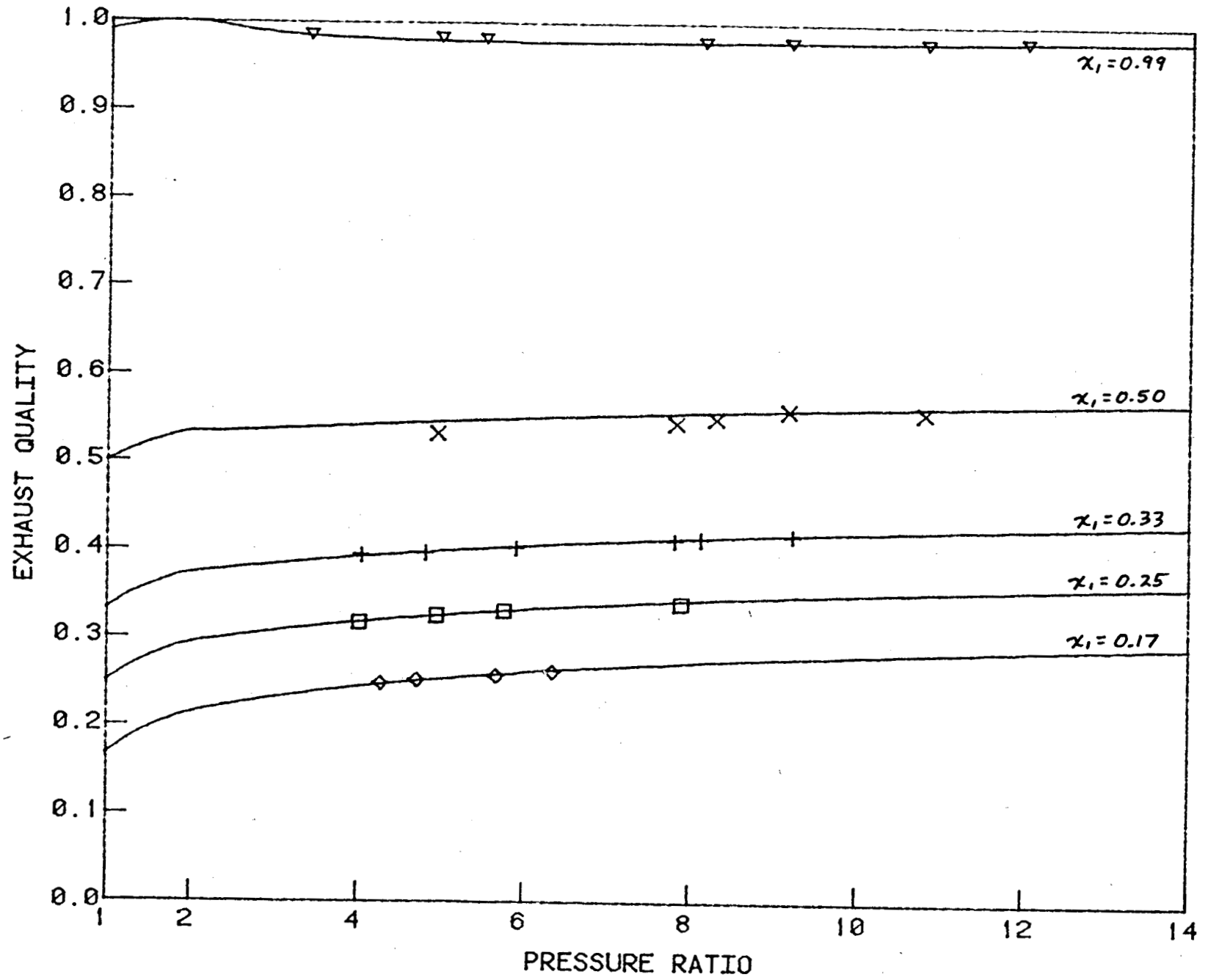


FIGURE 16b - LYSHOLM ENGINE MODEL

Exhaust Quality vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 6000 RPM

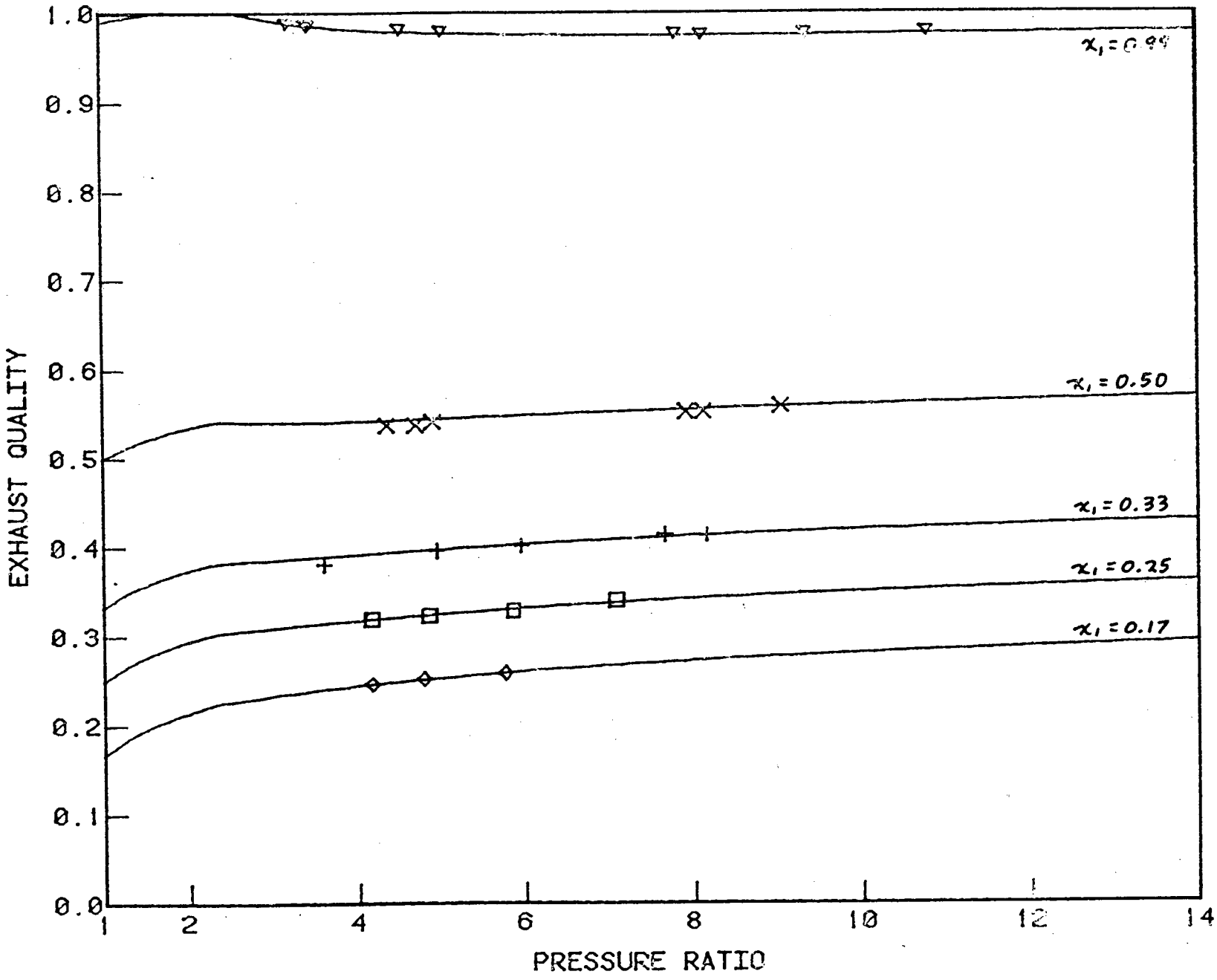
▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17



Exhaust Quality
vs.
Pressure Ratio

Inlet Pressure
120 psia

Engine Speed
7500 RPM

▽ – x1 = 0.99

X – x1 = 0.50

+ – x1 = 0.33

□ – x1 = 0.25

◇ – x1 = 0.17

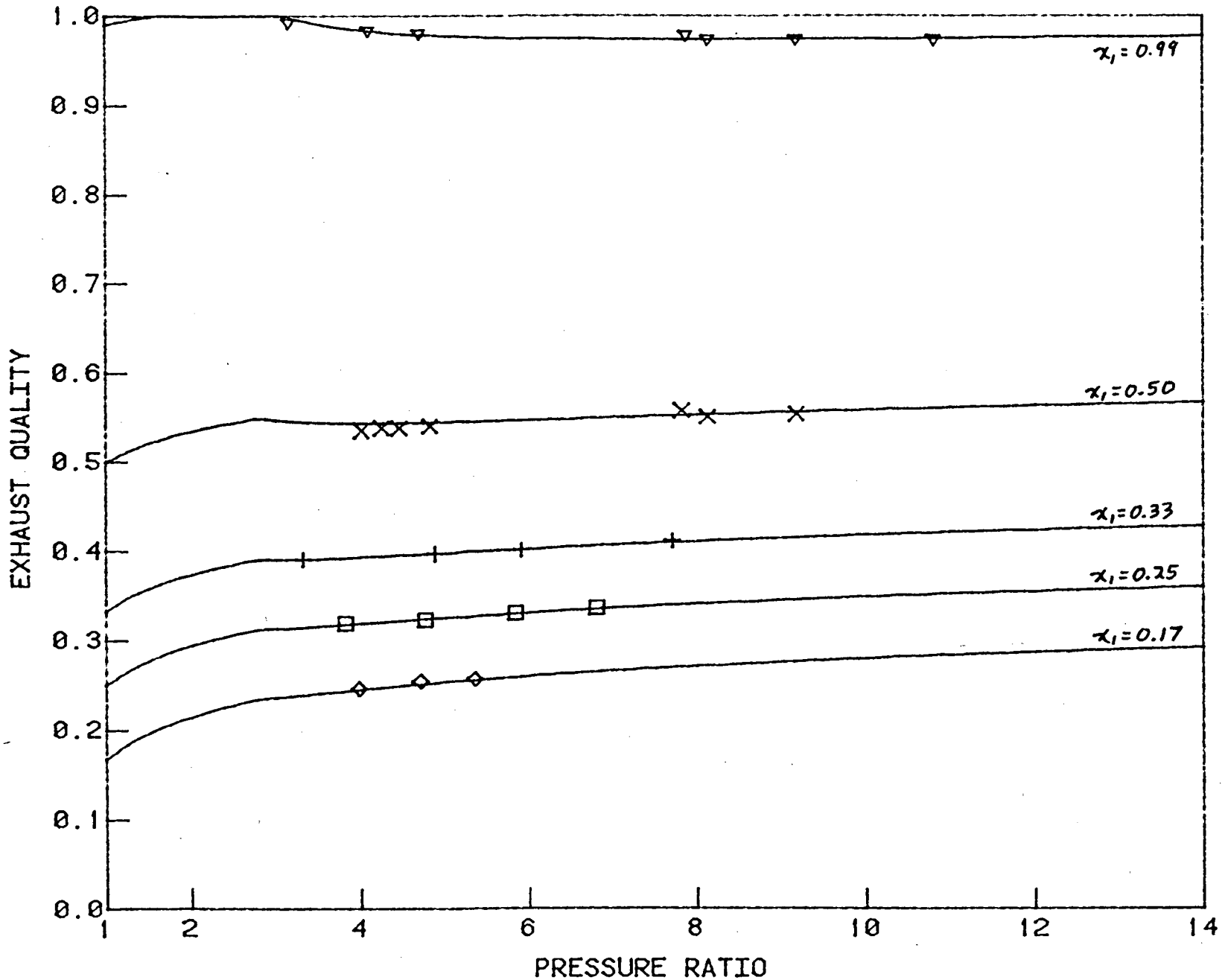


FIGURE 16d - LYSHOLM ENGINE MODEL

Exhaust Quality vs. Pressure Ratio

Inlet Pressure 120 psia

Engine Speed 9000 RPM

▽ - x1 = 0.99

X - x1 = 0.50

+ - x1 = 0.33

□ - x1 = 0.25

◇ - x1 = 0.17

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APPENDIX

to

EMPIRICAL MODELING OF A LYSHOLM HELICAL SCREW EXPANDER

Karl A. Brown
B.S.M.E. - University of California, Berkeley

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LYSHOLM ENGINE STARTING PROCEDURE

Turn On Controls

1) Plug in power

2) Turn on control air supply

3) Bleed control air regulators

4) Turn on Autudata

5) Check control settings:
   a) set flow controls to manual
   b) set flow controls to zero
   c) set vacuum control to zero vac.
   d) set torque filter to minimum value
   e) set brake controls to zero

6) Turn on brake power

Check engine condition

7) Disconnect jacking motor

8) Check oil system:
   a) check tank level
   b) check leakage return cans
   c) open oiltank vent
   d) open leakage return line
   e) open oil bypass

9) Turn on main air supply

10) open seal air supply valve

11) Bleed dp regulators

12) Close water seal valve

13) Check seal drain valves(oil return closed, drain open)

14) Open main water valve

15) Close engine ports;
   a) inlet and exhaust closed
   b) bypass open

16) Check steam path:
   a) open separator valves <u>or</u> separator bypass
   b) close turbine inlet valves
   c) close #2 turbine exhaust port
   d) open steam supply bypass to condenser
   e) open #1 turbine exhaust port

17) Turn on air ejector:
   a) open line to condenser
   b) open cooling water valve ¼ turn
   c) crack steam supply valve
   d) bleed ejector control line
   e) open ejector control valve to build pressure
   f) open stem supply valve

18) Clear condensate path:
   a) turn on condensate control air
   b) open two valves on condensate line
   c) close regulator bypass
   d) open orifice bypass
   e) open weigh tank drain and/or bypass

Turn On Steam System

19) Open steam supply:
   a) open main steam valve
   b) open steam line drain
   c) open calorimiter 3/4 turn
   d) crack engine steam supply valve to warm lines

   e) close steam line drain
   f) open engine steam supply valve

20) Turn on steam dp:
   a) open one side
   b) close bypass
   c) open other side

21) Turn on condensor cooling water:
   a) open open weigh tank drain
   b) open main water valve
   c) set control valve

22) Turn on condensate pump(check lubrication)

23) Start steam flow at 25% to warm lines

24) Stop flow; if flow does not read zero, bleed dps and seal pots)

Engine Start

25) Open engine ports:
   a) open exhaust port
   b) close bypass
   c) open inlet port

26) Set seal air pressure(40 psig)

27) Set seal water flow(dry steam operation)

28) Turn on oil pump

29) Set oil pressure (10 psig)

30) Set steam flow to 25%, turn over engine, load if necessary to keep speed below 3000 rpm.

31) Turn on water system:
   a) turn on water pump
   b) open trap on heat exchanger line
   c) open heat exchanger steam valve
   d) close trap on heat exchanger line

32) Open water dp(see step 20 for procedure)

33) Bleed water dp if flow does not read zero

34) Start a very low water flow

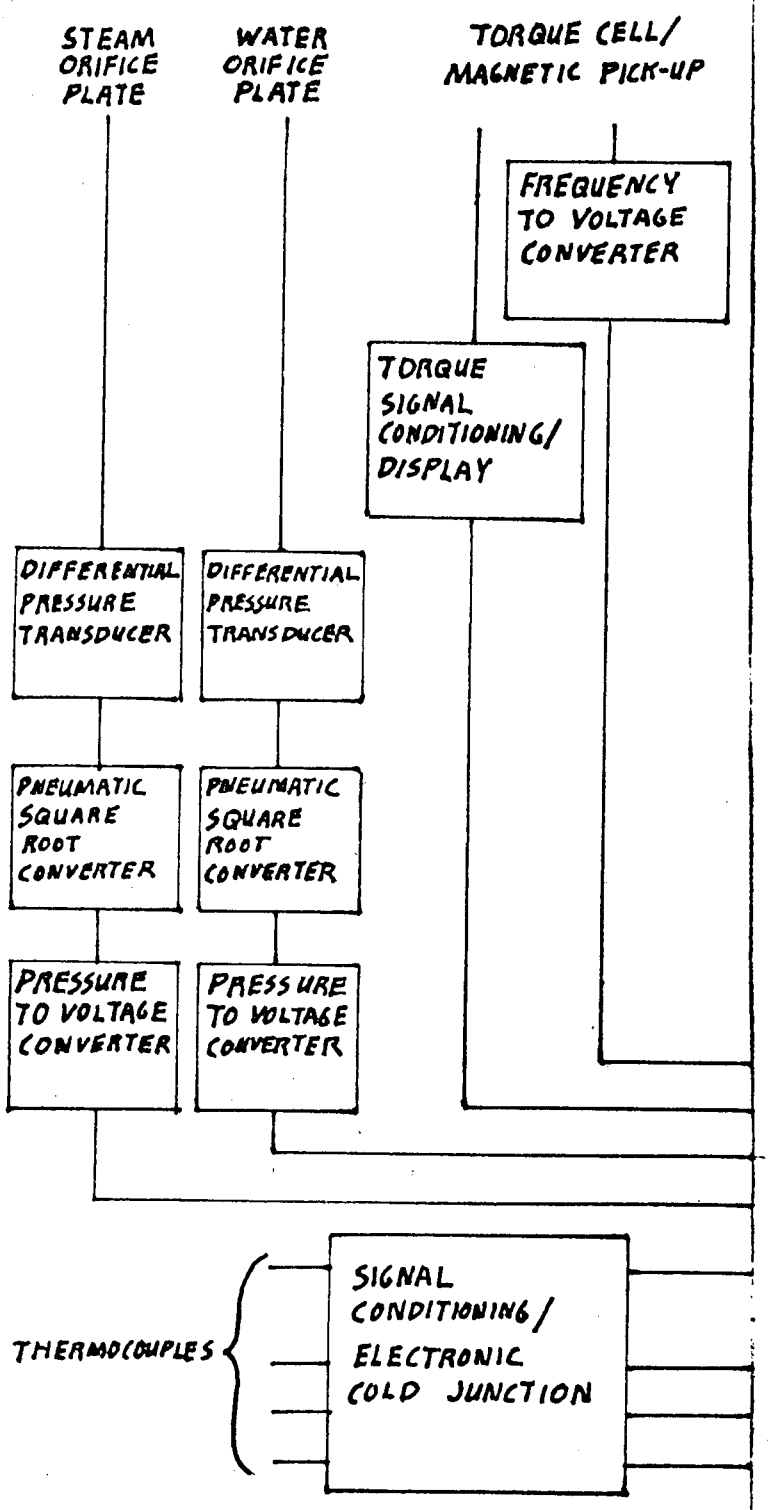
35) Allow water temperature to reach 336°F

36) Allow oil temperature to reach 140°F

37) Record in log - date, engine run #s

38) Bring engine to operating conditions

AUTO DATA CHANNEL ASSIGNMENTS AND PRIMARY INSTRUMENTATION SCHEMATIC

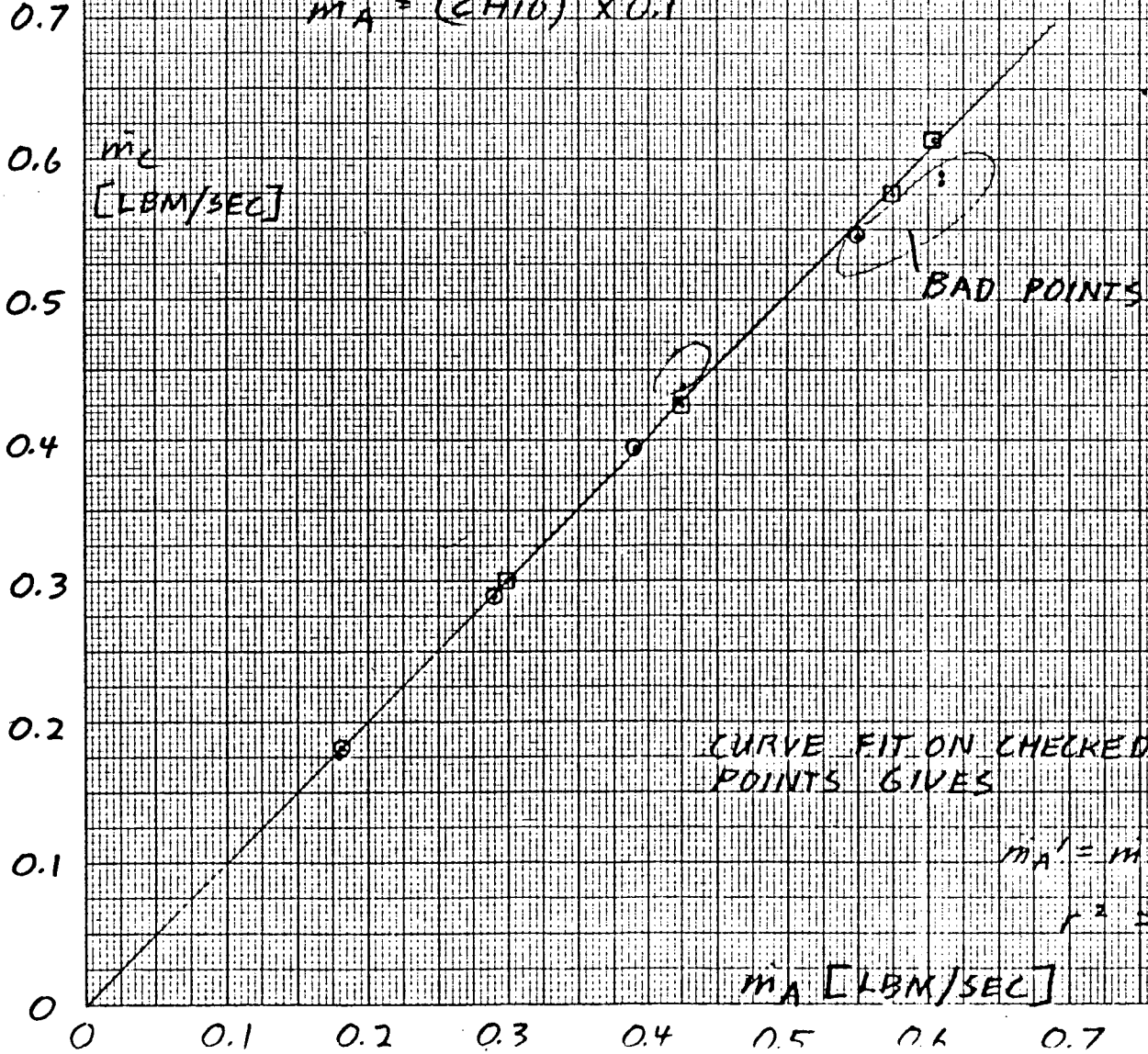


	WESTINGHOUSE TURBINES	
039		
038	#2 RPM	1v = 1K rpm
037	#2 Field	1v = 1 amp
036	#1 Field	1v = 1 amp
035	#2 Power	1v = 10 kw
034	#1 Power	1v = 10 kw
033	Cooling	1pct = 10 lb/sec
032	Condensate	1pct = .8 lb/sec
031	#2 Chest	psi = .2 psi
030	#1 Chest	psi
029	Condensate	°F
028	Cooling Out	°F
027	Cooling In	°F
026	#2 Exhaust	°F
025	#2 Chest	°F
024	#2 Line	°F
023	#1 Exhaust	°F
022	#1 Chest	°F
021	#1 Line	°F
020	Calorimeter	°F
	LYSHOLM ENGINE	
019		
018		
017	Seal Air	psi
016	Oil Supply	psi
015	Front Seal	psi
014	Rear Seal	psi
013	RPM	1v = 10K rpm
012	Torque	1v = 1K in-lb
011	Water Flow	1v = 1 lb/sec
010	Steam Flow	1v = .1 lb/sec
009	Calorimeter	°F
008	Steam Supply	°F
007	Water Supply	°F
006	Exhaust	°F
005	Inlet	°F
004	Separator	°F
003	Oil Return	°F
002		
001		
000		

CALIBRATION CHECK, STEAM FLOW METER (3/25/81, 6/10/81, 7/15/81, 7/16/81)

MEASURED FLOW ($\dot{m}_c$) VS ADJUSTED AUTODATA READING

$\dot{m}_A = (\overline{CH10}) \times 0.1$



	$\dot{m}_A$ [LBM/SEC]	$\dot{m}_c$ [LBM/SEC]
3/25/81	0.4220	0.428 ✓
	0.1798	0.177 ✓
(•)	~~0.6096~~	~~0.583~~
	0.3051	0.302 ✓
	~~0.4263~~	~~0.437~~
	~~0.6101~~	~~0.540~~
6/10/81	0.1816	0.1822 ✓
	0.2889	0.2878 ✓
(⊙)	0.3893	0.3960 ✓
	~~0.5508~~	~~0.5460~~
7/15/81	0.6033	0.6148 ✓
(□)		
7/16/81	0.5770	0.5774 ✓
	0.5157	0.5208 ✓
(□)	0.4230	0.4246 ✓
	0.2987	0.2985 ✓

CURVE FIT ON CHECKED POINTS GIVES

$\dot{m}_A' = \dot{m}_c = 1.0214\dot{m}_A - 0.0059$

$r^2 = 0.9995 \quad \sigma_m = 0.0069 \quad \sigma_b = 0.0028$

5

10 Millimeters to the Centimeter

12-168

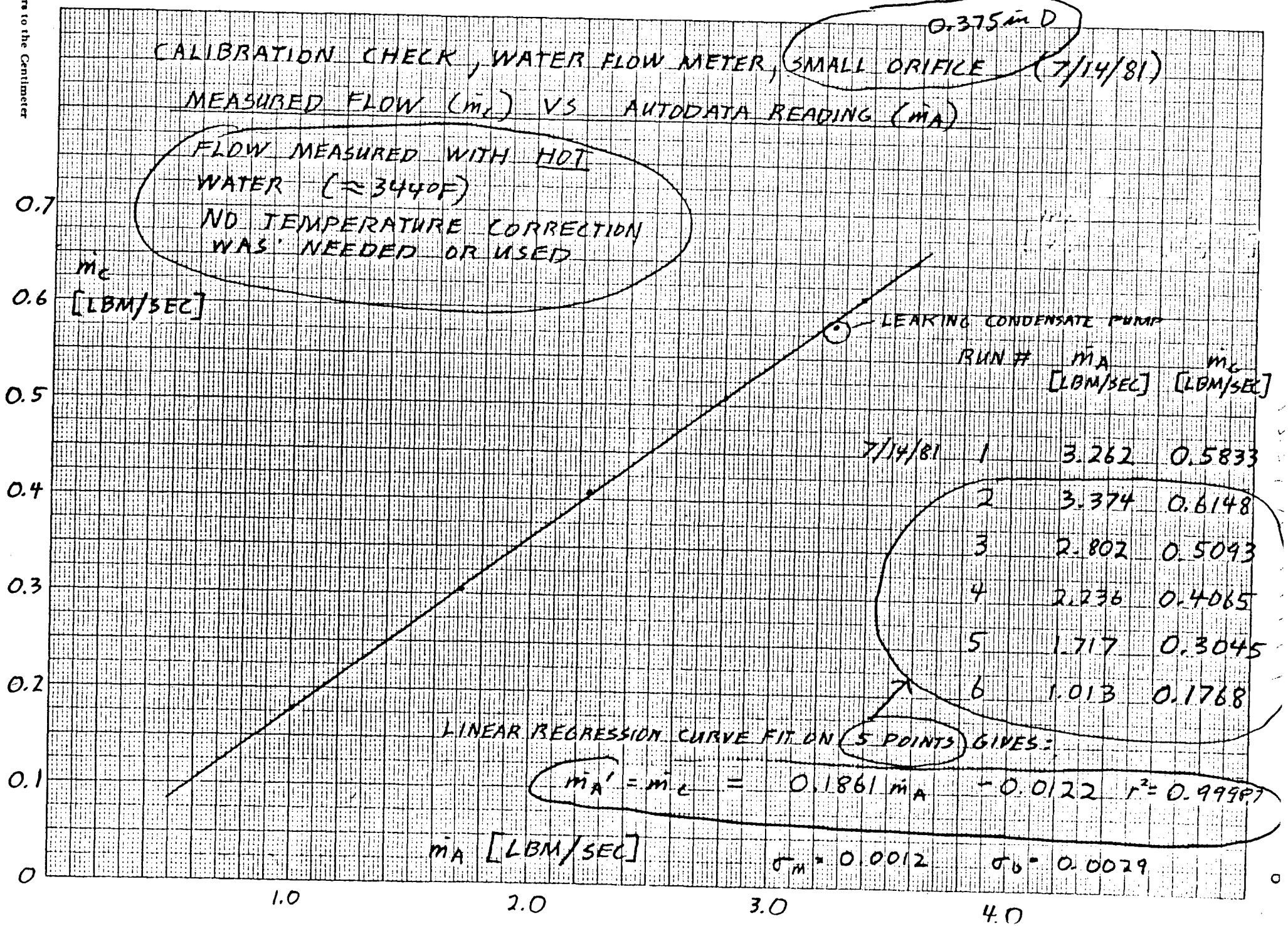
10 Millimeters to the Centimeter

12-188

CALIBRATION CHECK, WATER FLOW METER, SMALL ORIFICE (0.375 in D) (7/14/81)

MEASURED FLOW ($\dot{m}_c$) VS AUTODATA READING ($\dot{m}_A$)

FLOW MEASURED WITH HOT WATER (≈340°F)
NO TEMPERATURE CORRECTION WAS NEEDED OR USED



	RUN #	$\dot{m}_A$ [LBM/SEC]	$\dot{m}_c$ [LBM/SEC]
7/14/81	1	3.262	0.5833
	2	3.374	0.6148
	3	2.802	0.5093
	4	2.236	0.4065
	5	1.717	0.3045
	6	1.013	0.1768

LINEAR REGRESSION CURVE FIT ON 5 POINTS GIVES:

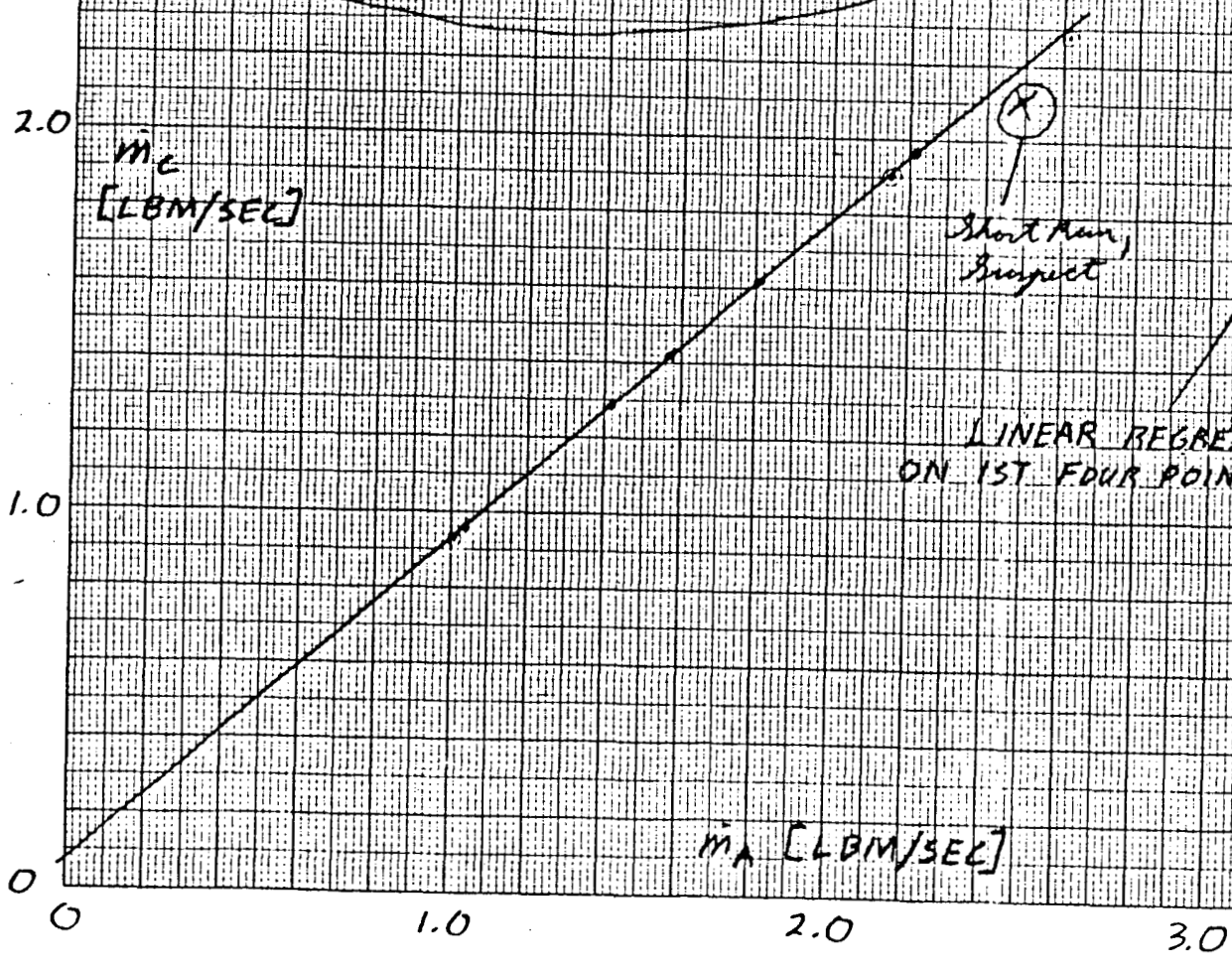
$\dot{m}_A' = \dot{m}_c = 0.1861\,\dot{m}_A - 0.0122 \quad r^2 = 0.99987$

$\sigma_m = 0.0012 \quad \sigma_b = 0.0029$

CALIBRATION CHECK, WATER FLOW METER (6/12/81)

MEASURED FLOW ($\dot{m}_C$) VS ADJUSTED AUTODATA READING ($\dot{m}_A$)

FLOW MEASURED WITH HOT WATER (≈ 340°F)
NO TEMPERATURE CORRECTION WAS NEEDED OR USED



	$\dot{m}_A$ [LBM/SEC]	$\dot{m}_C$ [LBM/SEC]
6/12/81	1.036	0.957
	1.421	1.294
	1.809	1.619
	2.220	1.970
	2.493	2.105

LINEAR REGRESSION ON 1ST FOUR POINTS

$\dot{m}_A = \dot{m}_C = 0.8538\dot{m}_A + 0.0756$

$r = 0.99993$

$\sigma_m = 0.0035$ $\sigma_b = 0.0059$

10 Millimeters to the Centimeter
12-188

ERROR ANALYSIS (For a typical point - Inlet Pressure = 120 psia, Inlet Quality = 0.33, Speed = 7500 RPM, Pressure Ratio = 8.0)

Note: All errors (σ) based on a confidence level of 95%

$\partial Q/\partial x$ - Determined by variation of parameters using the data reduction computer code		Q			
		Flow Rate lbm/sec	Power kW	Efficiency %	Exhaust Quality
x	Channel 5 Inlet Temperature - °F σ = 1 °F Nominal Value - 341.	0	0	0.23	0
	Channel 6 Exhaust Temperature - °F σ = 1 °F Nominal Value - 213.	0	0	0.33	0.0007
	Channel 7 Water Temperature - °F σ = 1 °F Nominal Value - 342.	0	0	0.08	0.0007
	Channel 9 Calorimeter Temperature - °F σ = 1 °F Nominal Value - 282.	0	0	0.02	0.0002
	Channel 10 Steam Flow Rate - Volts σ = 0.08 V Nominal Value - 4.30	0.10	0	8.0	0.05
	Channel 11 Water Flow Rate - Volts σ = 0.01 V Nominal Value - 0.930	0.90	0	10.0	0.20
	Channel 12 Torque - Volts σ = 0.003 V Nominal Value - 0.340	0	90	110	0
	Channel 13 Speed - Volts Nominal σ = 0.004 V Value - 0.75	0	40	50	0
	$\sigma = \sqrt{\sum_i ((\partial Q/\partial x_i)^2 (\sigma_i)^2)}$	0.012	0.31	0.86	0.0046
	σ /Q (Percent)	0.9	1.0	2.2	1.1
	Nominal Q	1.303	30.9	38.8	0.412

Characteristic Velocity Calculations

① Liquid Velocity

$$V_\ell = \frac{\text{MASS FLOW RATE}}{(\text{DENSITY})(\text{FLOW AREA})} = \frac{\dot{m}}{\rho \quad \pi(\text{NOZZLE RADIUS})^2}$$

$$V_\ell = \frac{\dot{m}}{(56\ \text{lbm/ft}^3)(\pi)(0.25\ \text{in})^2}\left(12\ \text{in/ft}\right)^2$$

$$V_\ell = \underline{\dot{m}\ (13.1\ \text{ft/lbm})} \qquad \rho = 56\ \text{lbm/ft}^3 \left(\begin{array}{c}\text{Saturated}\\ \text{Liquid}\\ @\ 120\ \text{psia}\end{array}\right)$$

② Steam (Vapor) Velocity

$$V_v = \frac{\dot{m}}{\rho\ (\text{LYSHOLM INLET PORT AREA})}$$

$$V_v = \frac{\dot{m}}{(0.27\ \text{lbm/ft}^3)(2\ \text{in}^2)}\left(12\ \text{in/ft}\right)^2$$

$$V_v = \underline{\dot{m}\ (270\ \text{ft/lbm})} \qquad \rho = 0.27\ \text{lbm/ft}^3 \left(\begin{array}{c}\text{Saturated}\\ \text{Vapor}\\ @\ 120\ \text{psia}\end{array}\right)$$

③ Lobe Tip Velocity (Tangential), Male Rotor

$$V_T = (\text{ROTATIONAL SPEED})(\text{ROTOR CIRCUMFERENCE})$$

$$V_T = N\left(2\pi\,(\text{ROTOR RADIUS})\right)$$

$$V_T = N\left(2\pi\,(0.2134\ \text{ft})\right)\left(1\ \text{min}/60\ \text{sec}\right)$$

$$V_T = \underline{N\left(2.23\times10^{-2}\ \text{ft-min/rev-sec}\right)}$$

④ Lobe Velocity (Axial), Male Rotor

$$V_A = (\text{ROTATIONAL SPEED})(\text{ROTOR LENGTH})/(\text{WRAP FRACTION})$$

$$V_A = \left[N\ (6.52\ \text{in})/(320°/360°)\right]\left(1\ \text{ft}/12\ \text{in}\right)\left(1\ \text{min}/60\ \text{sec}\right)$$

$$V_A = \underline{N\left(1.02\times10^{-2}\ \text{ft-min/rev-sec}\right)}$$

Wrap Angle = 320°

Correction For Residual Reading In Steam Flow Rate

Dates of Runs with Error

Date	Residual
6/30/81	?
7/1/81	?
7/2/81	?
7/7/81	~15%
7/9/81	~20%

Correction

Testing of the system indicates that the error due to vapor in the seal pots does not take long (~10 min) to develop and probably soon reaches an equilibrium value close to the residual reading at shutdown.

Thus the same correction should be applied to all points on a given day.

For simplicity, a correction corresponding to a 15% error will be applied to runs on all dates

$$\dot{m}_A(\text{Corrected}) = \left(\dot{m}_{A,100\%}\right)\left[\sqrt{-(R)^2 + \left(\dot{m}_A/\dot{m}_{A,100\%}\right)^2}\right]$$

R = Fractional Residual = <u>0.15</u>

$\dot{m}_{A,100\%}$ = Full-Scale Autodata Reading

= 6.03

DATA CONTROL PROGRAMS
LYSHOLM ENGINE TESTING
HP - 41CV

```
R54=  0.102
R55= -0.006
R56=  0.854
R57=  0.076
R58=  0.186
R59= -0.012
R60=  0.475
R61=  1050.050
R62=  1.045
R63= -44.000
```

```
PRP "FRATIO"

01♦LBL "FRATIO"
   02 FS? 03
   03 GTO 01
   04 RCL 56
   05 *
   06 RCL 57
   07 +
   08 GTO 02

   09♦LBL 01
   10 RCL 58
   11 *
   12 RCL 59
   13 +

   14♦LBL 02
   15 X<>Y
   16 RCL 54
   17 *
   18 RCL 55
   19 +
   20 /
   21 END
```

```
PRP "HCNTRL"

01♦LBL "HCNTRL"
   02 STO 05
   03 RDN
   04 STO 04
   05 RDN
   06 STO 03
   07 RDN
   08 STO 01
   09 RCL 05
   10 FS? 03
   11 GTO 01
   12 RCL 56
   13 *
   14 RCL 57
   15 +
   16 GTO 02

   17♦LBL 01
   18 RCL 58
   19 *
   20 RCL 59
   21 +

   22♦LBL 02
   23 STO 06
   24 RCL 04
   25 RCL 54
   26 *
   27 RCL 55
   28 +
   29 STO 02
   30 RCL 03
   31 RCL 60
   32 *
   33 RCL 61
   34 +
   35 RCL 02
   36 *
   37 RCL 01
   38 RCL 62
   39 *
   40 RCL 63
   41 +
   42 RCL 00
   43 *
   44 +
   45 RCL 00
   46 RCL 02
   47 +
   48 /
   49 END
```

Flow Ratio Control (FRATIO) HP-41CV

ASN -81, (FR) USER, (x = 0?)

01	LBL^T FRATIO	R54	0.10214	(SST(2,3))
02	FS? 03	R55	-0.0059	(BST(2,3))
03	GTO 01	R56	0.8538	(SW(2))
04	RCL 56	R57	0.0756	(BW(2))
05	*	R58	0.1861	(SW(3))
06	RCL 57	R59	-0.0122	(SW(3))
07	+			
08	GTO 02			
09	LBL 01			
10	RCL 58			
11	X			
12	RCL 59			
13	+			
14	LBL 02			
15	x ≤ y			
16	RCL 54			
17	X			
18	RCL 55			
19	+			
20	/			
21	END			

Running Procedure

CH 10 ↑
CH 11 (FR) (USER)
or XEQ FRATIO

Display Flow Ratio

Flags

03 Set → 0.375 Water Orifice (3)
Clear → 0.800 Water Orifice (2)

Test Examples

SF 03	5.194	↑		
	2.016	FR	→	0.6919 ✓
CF 03	3.096	↑		
	1.621	FR	→	4.7035 ✓

Inlet Enthalpy Control (HCNTRL), HP-41CV

ASN -61, (HC) USER, ($x \leq y$?)

01	LBL^T HCNTRL	46	RCL 08
02	STO 11	47	+
03	R↓	48	/
04	STO 10	49	END
05	↓		
06	STO 09		
07	↓		
08	STO 07		
09	RCL 11		
10	FS ? 00		
11	GTO 01		
12	RCL 56		
13	*		
14	RCL 57		
15	+		
16	GTO 02		
17	LBL 01		
18	RCL 58		
19	*		
20	RCL 59		
21	+		
22	LBL 02		
23	STO 06		
24	RCL 10		
25	RCL 54		
26	*		
27	RCL 55		
28	+		
29	STO 08		
30	RCL 09		
31	RCL 60		
32	*		
33	RCL 13		
34	+		
35	RCL 61		
36	*		
37	RCL 07		
38	RCL 62		
39	*		
40	RCL 63		
41	+		
42	RCL 06		
43	*		
44	+		
45	RCL 06		

R54	0.10214	(SST (2,3))
R55	-0.0059	(BST (2,3))
R56	0.8538	(SW (2))
R57	0.0756	(BW (2))
R58	0.1861	(SW (3))
R59	-0.0122	(BW (3))
R06	$\dot{m}_W$	[LBM/SEC]
R07	CH 07	
R08	$\dot{m}_{ST}$	[LBM/SEC]
R09	CH 07	
R10	CH 10	
R11	CH 11	
R60	0.475	[BTU/LBM°R]
R61	1050.05	[BTU/LBM]
R62	1.045	[BTU/LBM°R]
R63	-44.	[BTU/LBM]

Running Procedure

CH 07 ↑
CH 09 ↑
CH 10 ↑
CH 11 HC (USER)
or XEQ HCNTRL

Display H_2 [BTU/LBM]

Flags

03 Set → 0.375 Water Orifice (3)
Clear → 0.800 Water Orifice (2)

Test Examples (HCNTRL)

SF 03	340.0	↑		
	282.0	↑		
	4.266	↑		
	2.295	HC	→	755.3621 ✓
CF 03	324.0	↑		
	285.0	↑		
	3.477	↑		
	1.111	HC	→	521.1102 ✓

42-381 50 SHEETS 5 SQUARE
42-382 100 SHEETS 5 SQUARE
42-389 200 SHEETS 5 SQUARE
NATIONAL

```
C
C       SUBROUTINE
C
        SUBROUTINE MOD(QUAL1, SPEED, PRATIO, FLOW, EFF, QUAL2,
        1        POWER, PMIN)
C       VERSION #7, 2/16/82
C       SUBROUTINE USES EMPIRICAL RELATIONS TO DETERMINE PERFORMANCE
C       OF LYSHOLM ENGINE FOR A GIVEN SET OF OPERATING CONDITIONS
        COMMON /COEFF/ F(11), P(6), PM(2)
        COMMON /LOGIN/ TL, PL, HL, SL, VL, XL, SPRHT
        LOGICAL TL, PL, HL, SL, VL, XL, SPRHT
C       DETERMINE MASS FLOW RATE
        AA = F(1) * EXP(F(2) * SPEED)
        BA = F(3) + F(4) * SPEED
        AB = F(5) * EXP(F(6) * SPEED)
        BB = F(7) + F(8) * SPEED
        A = AA + BA * QUAL1
        B = AB + BB * QUAL1
        C = (F(9) * SPEED) / (F(10) + F(11) * QUAL1)
        FLOW = (PRATIO - 1.0)/(A + B * PRATIO) + C
C       DETERMINE POWER
        A = P(1) + P(2) * SPEED
        B = P(3) + P(4) * SPEED + P(5) * SPEED**2.0
        C = EXP(P(6) * SPEED)
        POWER = (PRATIO - C) / (A + B * PRATIO)
C       LIMIT POWER TO REALISTIC VALUES FOR PLOTTING
        IF (POWER .LT. 0.0) POWER = 0.0
        IF (POWER .GT. 75.0) POWER = 0.0
C       DETERMINE INLET STATE
        P1 = 120.0
        PL = .TRUE.
        XL = .TRUE.
        CALL STMTBL(T1, P1, H1, S1, V1, QUAL1, SH1)
        P2 = P1 / PRATIO
C       AVOID DIVISION BY ZERO AT PRATIO = 1
        IF (PRATIO .GT. 1.0) GO TO 100
        EFF = 0.0
        GO TO 200
100     CONTINUE
C       DETERMINE ISENTROPIC EXHAUST STATE
        PL = .TRUE.
        SL = .TRUE.
        CALL STMTBL(T2S, P2, H2S, S1, V2S, X2S, SH2S)
C       CALCULATE EFFICIENCY
        DELTAH = H1 - H2S
        EFF = 100 * POWER * 0.9480 / (FLOW * DELTAH)
200     CONTINUE
C       DETERMINE ADIABATIC EXHAUST STATE(ASSUMING NO LOSSES)
        H2 = H1 - POWER * 0.9480 / FLOW
C       DETERMINE EXIT STATE
        PL = .TRUE.
        HL = .TRUE.
        CALL STMTBL(T2, P2, H2, S2, V2, QUAL2, SH2)
C       TO PRESERVE CONTINUITY OF EXHAUST QUALITY PLOT, SET QUAL2 TO
C       1.0 IF EXHAUST IS SUPERHEATED
```

```
      IF (SPRHT) QUAL2 = 1.0
C     DETERMINE MINIMUM ATTAINABLE EXHAUST PRESSURE FOR THE MASS
C     FLOW RATE
      PMIN = PM(1) * EXP(PM(2) * FLOW)
      RETURN
      END
C
C     BLOCK DATA
C
      BLOCK DATA
C     COEFFICIENTS FOR EMPIRICAL LYSHOLM ENGINE MODEL
      COMMON /COEFF/ F(11), P(6), PM(2)
      DATA F/ -5.403E-2, 3.638E-4, -4.031, 3.375E-4,
     1        2.694E-1, 1.968E-4, 3.685, -2.050E-4,
     2        1.331E-4, 1.789E-2, 3.710/
      DATA P/ 6.093E-2, -9.934E-6,
     1        6.651E-2, -9.590E-6, 5.113E-10,
     2        1.144E-4/
      DATA PM/ 7.500, 4.500E-1/
      END
```

```
C
C       SUBROUTINE
C
        SUBROUTINE STMTBL(T, P, H, S, V, X, SH)
C       CALCULATES THE PROPERTIES OF STEAM AT A GIVEN STATE
        COMMON /LOGIN/ TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
        COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
        1       VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
        2       SGT(27), SSUPT(27), TSUPT(27)
        LOGICAL TABLES, STATES, SPRHTS, TLS, PLS, HLS, SLS, VLS, XLS
C       INITIALIZE SUPERHEAT LOGICAL VARIABLE
        SPRHTS = .FALSE.
C       SET PIVOT PARAMETER ARRAY SIZE
        N =27
1       CONTINUE
C       DETERMINE PIVOT PARAMETER
        IF ( PLS ) GO TO 20
        IF ( TLS ) GO TO 30
        IF ( VLS ) GO TO 2000
        STATES = .FALSE.
10      CONTINUE
C       IF STATE HAS NOT BEEN PROPERLY SPECIFIED, TYPE 'STATE NOT-
C       PROPERLY SPECIFIED' AND STOP EXECUTION
        TYPE 16
16      FORMAT (1H, 'STATE NOT PROPERLY SPECIFIED')
        STOP
20      CONTINUE
        CALL BOUND(P, PSAT, TABLES, I, N)
C       CHECK THAT THE STATE IS IN THE TABLE RANGE
        IF ( TABLES ) GO TO 25
22      CONTINUE
C       IF OUT OF TABLE RANGE, TYPE 'OUT OF STEAM TABLE RANGE' AND
C       STOP EXECUTION
        TYPE 23
23      FORMAT (1H, 'OUT OF STEAM TABLE RANGE')
        STOP
25      CONTINUE
        FRAC = YLOG(P, PSAT(I), PSAT(I+1))
C       CHECK FOR IMPLICIT SPECIFICATION OF SUPERHHEAT CONDITIONS(T AND
        IF ( TLS ) SPRHTS = .TRUE.
        IF ( TLS ) GO TO 40
        GO TO 35
30      CONTINUE
        CALL BOUND(T, TSAT, TABLES, I, N)
C       CHECK THAT THE STATE IS IN THE TABLE RANGE AND EXIT SUBROUTINE I
        IF ( TABLES ) GO TO 32
        GO TO 22
32      CONTINUE
        FRAC = YLOG(T, TSAT(I), TSAT(I+1))
35      CONTINUE
40      CONTINUE
C       DETERMINE SATURATED VAPOR PROPERTIES FOR DECISION ON SUPERHEAT
        HG = ZLOG(FRAC, HGT(I), HGT(I+1))
        SG = ZLOG(FRAC, SGT(I), SGT(I+1))
        VG = ZLOG(FRAC, VGT(I), VGT(I+1))
```

```
C       GO TO SUPERHEAT SECTION IF IMPLICITLY SPECIFIED
        IF ( SPRHTS ) GO TO 500
C       DETERMINE SATURATED LIQUID PROPERTIES
        HF = ZLOG(FRAC, HFT(I), HFT(I+1))
        SF = ZLOG(FRAC, SFT(I), SFT(I+1))
        VF = ZLOG(FRAC, VFT(I), VFT(I+1))
C       CALCULATE QUALITY FROM SECOND INPUT VARIABLE
        IF ( XLS ) GO TO 90
        IF ( HLS ) GO TO 100
        IF ( SLS ) GO TO 110
        IF ( VLS ) GO TO 120
        GO TO 10
90      CONTINUE
        IF ( HLS .OR. VLS .OR. SLS) GO TO 10
        GO TO 200
100     CONTINUE
        IF ( SLS .OR. VLS ) GO TO 10
C       CHECK FOR SUPERHEAT CONDITIONS
        IF ( H .GT. HG ) GO TO 490
        X = YLIN(H, HF, HG)
        GO TO 200
110     CONTINUE
        IF ( VLS ) GO TO 10
        IF ( S .GT. SG) GO TO 490
        X = YLIN(S, SF, SG)
        GO TO 200
120     CONTINUE
        IF ( V .GT. VG ) GO TO 490
        X = YLIN(V, VF, VG)
200     CONTINUE
C       CHECK FOR NEGATIVE QUALITY INDICATING SUBCOOLED CONDITIONS
        IF ( X .LE. -0.00005 ) GO TO 220
        GO TO 230
220     CONTINUE
        TYPE 225
225     FORMAT (1H, 'NEGATIVE QUALITY INDICATES SUBCOOLED CONDITIONS-OUT
        1 OF TABLE RANGE')
230     CONTINUE
        IF (X .LE. 1.005) GO TO 240
        GO TO 10
240     CONTINUE
C       CALCULATE UNKNOWN PROPERTIES
        IF ( PLS ) GO TO 250
        P = ZLOG(FRAC, PSAT(I), PSAT(I+1))
250     CONTINUE
        IF ( TLS ) GO TO 260
        T = ZLOG(FRAC, TSAT(I), TSAT(I+1))
260     CONTINUE
        IF ( HLS ) GO TO 270
        H = ZLIN(X, HF, HG)
270     CONTINUE
        IF ( SLS ) GO TO 280
        S = ZLIN(X, SF, SG)
280     CONTINUE
        IF ( VLS ) GO TO 290
```

```
      V = ZLIN(X, VF, VG)
290   CONTINUE
      GO TO 730
490   CONTINUE
      IF ( PLS ) GO TO 495
      GO TO 800
495   CONTINUE
      SPRHTS = .TRUE.
500   CONTINUE
C     DETERMINE PROPERTIES AT 70 DEGREES SUPERHEAT
      TSUP = ZLOG(FRAC, TSUPT(I), TSUPT(I+1))
      HSUP = ZLOG(FRAC, HSUPT(I), HSUPT(I+1))
      SSUP = ZLOG(FRAC, SSUPT(I), SSUPT(I+1))
      VSUP = ZLOG(FRAC, VSUPT(I), VSUPT(I+1))
      TSATG = ZLOG(FRAC, TSAT(I), TSAT(I+1))
      IF ( TLS ) GO TO 600
      IF ( HLS ) GO TO 610
      IF ( SLS ) GO TO 620
      IF ( VLS ) GO TO 630
      GO TO 10
C     CALCULATE SUPERHEAT FRACTION FROM SECOND INPUT VARIABLE
600   CONTINUE
      IF ( HLS .OR. SLS .OR. VLS ) GO TO 10
      SHF = YLIN(T, TSATG, TSUP)
      GO TO 650
610   CONTINUE
      IF ( SLS .OR. VLS ) GO TO 10
      SHF = YLIN(H, HG, HSUP)
      GO TO 650
620   CONTINUE
      IF ( VLS ) GO TO 10
      SHF = YLIN(S, SG, SSUP)
      GO TO 650
630   CONTINUE
      SHF = YLIN(V, VG, VSUP)
650   CONTINUE
      IF (SHF .LE. -0.00005) GO TO 660
      GO TO 670
660   CONTINUE
      TYPE 665
665   FORMAT (1H, 'NEGATIVE SUPERHEAT INDICATES P AND T ILLEGALY
      1 SPECIFIED SUBCOOLED CONDITIONS')
670   CONTINUE
C     CALCULATE UNKNOWN PROPERTIES
      SH = SHF * (TSUPT(1) - TSAT(I))
      IF ( TLS ) GO TO 700
      T = ZLIN(SHF, TSATG, TSUP)
700   CONTINUE
      IF ( HLS ) GO TO 710
      H = ZLIN(SHF, HG, HSUP)
710   CONTINUE
      IF ( SLS ) GO TO 720
      S = ZLIN(SHF, SG, SSUP)
720   CONTINUE
      IF ( VLS ) GO TO 730
```

```
      V = ZLIN(SHF, VG, VSUP)
730   CONTINUE
      GO TO 950
800   CONTINUE
      IF (HLS) GO TO 810
      IF (SLS) GO TO 820
      IF (VLS) GO TO 830
      GO TO 10
810   CONTINUE
      CALL TSPR(T, H, FRAC, HGT, HSUPT, SHF1, SHF2, M)
      GO TO 900
820   CONTINUE
      CALL TSPR(T, S, FRAC, SGT, SSUPT, SHF1, SHF2, M)
      GO TO 900
830   CONTINUE
      CALL TSPR(T, V, FRAC, VGT, VSUPT, SHF1, SHF2, M)
900   CONTINUE
      IF (HLS) GO TO 910
      CALL DINT(H, FRAC, SHF1, SHF2, HGT, HSUPT, M)
910   CONTINUE
      IF (SLS) GO TO 920
      CALL DINT(S, FRAC, SHF1, SHF2, SGT, SSUPT, M)
920   CONTINUE
      IF (VLS) GO TO 930
      CALL DINT(V, FRAC, SHF1, SHF2, VGT, VSUPT, M)
930   CONTINUE
      P = ZLOG(FRAC, PSAT(M), PSAT(M+1))
      GO TO 950
2000  CONTINUE
      CALL QUAL(V,S,P)
      GO TO 1
950   CONTINUE
C     INITIALIZE LOGICAL INPUT SPECIFIERS FOR SUBSEQUENT CALLS
      TLS = .FALSE.
      PLS = .FALSE.
      HLS = .FALSE.
      SLS = .FALSE.
      VLS = .FALSE.
      XLS = .FALSE.
1000  RETURN
      END
C
C     BLOCK DATA
C
      BLOCK DATA
C     STEAM DATA, ENGLISH UNITS (REFERENCE 1967 ASME STEAM TABLES)
      COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
     1        VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
     2        SGT(27), SSUPT(27), TSUPT(27)
      DATA TSAT/ 100.0, 110.0, 120.0, 130.0, 140.0, 150.0,
     1        160.0, 170.0, 180.0, 190.0, 200.0, 210.0, 220.0,
     2        230.0, 240.0, 250.0, 260.0, 270.0, 280.0, 290.0, 300.0,
     3        310.0, 320.0, 330.0, 340.0, 350.0, 360.0/
      DATA PSAT/ 0.94924, 1.2750, 1.6927, 2.2230, 2.8892, 3.7184,
     1        4.7414, 5.9926, 7.5110, 9.340, 11.526, 14.123,
```

```
2       17.186, 20.779, 24.968, 29.825, 35.427, 41.856, 49.200,
3       57.550, 67.005, 77.667, 89.643, 103.045, 117.992,
4       134.604, 153.010/
DATA VFT/ 0.01613, 0.01617, 0.01620, 0.01625, 0.01629, 0.01634,
1       0.01640, 0.01645, 0.01651, 0.01657, 0.01664, 0.01671,
2       0.01678, 0.01685, 0.01693, 0.01701, 0.01709, 0.01718,
3       0.01726, 0.01736, 0.01745, 0.01755, 0.01766, 0.01776,
4       0.01787, 0.01799, 0.01811/
DATA VGT/ 350.4, 265.39, 203.26, 157.33, 123.00, 97.07,
1       77.29, 62.06, 50.225, 40.957, 33.639, 27.816, 23.148,
2       19.381, 16.321, 13.819, 11.762, 10.060, 8.6439, 7.4603,
3       6.4658, 5.6260, 4.9138, 4.3069, 3.7878, 3.3418, 2.9573/
DATA HFT/ 67.999, 77.98, 87.97, 97.96, 107.95, 117.95,
1       127.96, 137.97, 148.00, 158.04, 168.09, 178.15,
2       188.23, 198.33, 208.45, 218.59, 228.76, 238.95,
3       249.17, 259.4, 269.7, 280.0, 290.4, 300.8, 311.3,
4       321.8, 332.3/
DATA HGT/ 1105.1, 1109.3, 1113.6, 1117.8, 1122.0, 1126.1,
1       1130.2, 1134.2, 1138.2, 1142.1, 1146.0, 1149.7,
2       1153.4, 1157.1, 1160.6, 1164.0, 1167.4, 1170.6, 1173.8,
3       1176.8, 1179.7, 1182.5, 1185.2, 1187.7, 1190.1, 1192.3,
4       1194.4/
DATA SFT/ 0.1295, 0.1472, 0.1646, 0.1817, 0.1985, 0.2150,
1       0.2313, 0.2473, 0.2631, 0.2787, 0.2940, 0.3091,
2       0.3241, 0.3388, 0.3533, 0.3677, 0.3819, 0.3960,
3       0.4098, 0.4236, 0.4372, 0.4506, 0.4640, 0.4772,
4       0.4902, 0.5032, 0.5161/
DATA SGT/ 1.9825, 1.9577, 1.9339, 1.9112, 1.8895, 1.8686,
1       1.8487, 1.8295, 1.8111, 1.7934, 1.7764, 1.7600,
2       1.7442, 1.7290, 1.7142, 1.7000, 1.6862, 1.6729, 1.6599,
3       1.6473, 1.6351, 1.6232, 1.6116, 1.6003, 1.5892, 1.5784,
4       1.5678/
DATA VSUPT/ 394.6, 298.36, 228.15, 176.32, 137.64, 108.48,
1       86.27, 69.19, 55.94, 45.57, 37.405, 30.902, 25.702,
2       21.507, 18.104, 15.323, 13.039, 11.151, 9.581, 8.271,
3       7.170, 6.241, 5.454, 4.783, 4.210, 3.7174, 3.2932/
DATA HSUPT/ 1136.6, 1141.0, 1145.3, 1149.7, 1154.0, 1158.2,
1       1162.5, 1166.8, 1170.9, 1175.1, 1179.2, 1183.3, 1187.3,
2       1191.2, 1195.1, 1198.9, 1202.7, 1206.4, 1210.0, 1213.6,
3       1217.0, 1220.4, 1223.7, 1226.9, 1229.9, 1232.9, 1235.8/
DATA SSUPT/ 2.0356, 2.0101, 1.9857, 1.9623, 1.9399, 1.9816,
1       1.8982, 1.8785, 1.8597, 1.8416, 1.8243, 1.8076, 1.7916,
2       1.7761, 1.7612, 1.7469, 1.7331, 1.7197, 1.7067, 1.6942,
3       1.6821, 1.6703, 1.6589, 1.6478, 1.6370, 1.6265, 1.6163/
DATA TSUPT/ 170.0, 180.0, 190.0, 200.0, 210.0, 220.0,
1       230.0, 240.0, 250.0, 260.0, 270.0, 280.0, 290.0,
2       300.0, 310.0, 320.0, 330.0, 340.0, 350.0, 360.0,
3       370.0, 380.0, 390.0, 400.0, 410.0, 420.0, 430.0/
END
C
C       SUBROUTINE
C
        SUBROUTINE TSPR(TT, Q, FRACT, G, SUP, SHFT1, SHFT2, J)
        COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
1       VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
```

```
     2        SGT(27), SSUPT(27), TSUPT(27)
      DIMENSION G(27), SUP(27), SHFT(27), QSH(27)
      NT = 27
      DO 100 J = 1, NT
      SHFT(J) = (TT - TSAT(J)) / (TSUPT(J) - TSAT(J))
      QSH(J) = ZLIN(SHFT(J), G(J), SUP(J))
100   CONTINUE
      CALL RBOUND (Q, QSH, TABLES, J, NT)
      FRACT = YLOG(Q, QSH(J), QSH(J+1))
      SHFT1 = SHFT(J)
      SHFT2 = SHFT(J+1)
      RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE DINT(QD, FRACD, SHFD1, SHFD2, GD, SUPD, KD)
      DIMENSION GD(27), SUPD(27)
      QD1 = ZLIN(SHFD1, GD(KD), SUPD(KD))
      QD2 = ZLIN(SHFD2, GD(KD+1), SUPD(KD+1))
      QD = ZLOG(FRACD, QD1, QD2)
      RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE BOUND(B, BT, BTABLE, IB, NB)
      DIMENSION BT(NB)
      LOGICAL BTABLE
      NBM = NB - 1
      DO 20 IB = 1, NBM
      IF ( B .GE. BT(IB)) GO TO 10
      GO TO 30
10    IF ( B .LE. BT(IB + 1)) GO TO 40
20    CONTINUE
30    BTABLE = .FALSE.
      GO TO 50
40    BTABLE = .TRUE.
50    RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE RBOUND(B, BT, BTABLE, IB, NB)
      DIMENSION BT(NB)
      LOGICAL BTABLE
      NBM = NB - 1
      DO 20 IB = 1, NBM
      IF (B .LE. BT(IB)) GO TO 10
      GO TO 30
10    IF (B .GE. BT(IB + 1)) GO TO 40
20    CONTINUE
30    BTABLE = .FALSE.
      GO TO 50
40    BTABLE = .TRUE.
```

```
50      RETURN
        END
C
C       FUNCTION
C
        FUNCTION ZLIN(A, B1, B2)
        ZLIN = B1 + A * (B2 - B1)
        RETURN
        END
C
C       FUNCTION
C
        FUNCTION ZLOG(C, D1, D2)
        ZLOG = EXP(ALOG(D1) + C * (ALOG(D2) - ALOG(D1)))
        RETURN
        END
C
C       FUNCTION
C
        FUNCTION YLIN(E, E1, E2)
        YLIN = (E - E1)/(E2 -E1)
        RETURN
        END
C
C       FUNCTION
C
        FUNCTION YLOG(F, F1, F2)
        YLOG = (ALOG(F) - ALOG(F1))/(ALOG(F2)- ALOG(F1))
        RETURN
        END
C
C       SUBROUTINE
C
        SUBROUTINE QUAL(V,S,P)
        COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
        1        VSUFT(27), HFT(27), HGT(27), HSUFT(27), SFT(27),
        2        SGT(27), SSUFT(27), TSUFT(27)
        COMMON /LOGIN/ TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
        DIMENSION Q(27), VM(27)
        LOGICAL TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
        NT = 27
        DO 100 J=1,NT
        Q(J) = YLIN(S, SFT(J), SGT(J))
100     CONTINUE
        DO 200 J=1,NT
        VM(J) = ZLIN(Q(J),VFT(J),VGT(J))
200     CONTINUE
        CALL BOUND(V,VM,TABLES,J,NT)
        Y = YLOG(V,VM(J),VM(J+1))
        P = ZLOG(Y,PSAT(J),PSAT(J+1))
        VLS = .FALSE.
        PLS = .TRUE.
        RETURN
        END
```

```
C       CURVE.FTN
C       PROGRAM COMPUTES DATA FOR EVALUATION OF MOD
        DIMENSION QUAL1(6), SPEED(4)
        DATA QUAL1 / 0.167, 0.250, 0.333, 0.500, 0.990, 0.200/
        DATA SPEED /4500.0, 6000.0, 7500.0, 9000.0 /
        DO 300 I = 1,6
        DO 200 J = 1,4
C       OPEN OUTPUT FILE
        OPEN (UNIT = 1, NAME = 'CURVE.DAT', TYPE = 'NEW',
        1       CARRIAGE CONTROL = 'LIST', ACCESS = 'SEQUENTIAL')
        PRATIO = 1.0
        DO 100 K = 1,113
        CALL MOD(QUAL1(I), SPEED(J), PRATIO, FLOW,
        1       EFF, QUAL2, POWER, PMIN)
C       WRITE RESULTS TO DATA FILE
        WRITE (1,50) QUAL1(I), SPEED(J), PRATIO, FLOW,
        1       POWER, EFF, QUAL2, PMIN
50      FORMAT(1H ,F5.3, T9, F5.0, T18, F5.2, T27, F7.5, T36,
        1       F8.3, T45, F8.3, T54, F7.5, T63, F6.3)
        PRATIO = PRATIO + 1.0 / 8.0
100     CONTINUE
        CLOSE (UNIT = 1)
200     CONTINUE
300     CONTINUE
        STOP
        END
```

```
C       OPT.FTN
C       PROGRAM SEARCHES FOR OPERATING POINT FOR MAXIMUM EFFICIENCY
C       IN ENGINE MODEL
        EFFM2 = 0.0
        EFFM1 = 0.0
        SPEED = 6000
        DO 300 I = 1,25
        QUAL1 = 0.0
        DO 200 J = 1,51
        PRAT = 3.0
        DO 100 K = 1,21
        CALL MOD(QUAL1, SPEED, PRAT, FLOW, EFF, QUAL2, POWER, PMIN)
        IF(EFF .LE. EFFM1) GO TO 10
        EFFM1 = EFF
        SPEED1 = SPEED
        QUAL11 = QUAL1
        PRAT1 = PRAT
        FLOW1 = FLOW
        QUAL21 = QUAL2
        POWER1 = POWER
        PMIN1 = PMIN
10      CONTINUE
        PE = 120.0 / PRAT
        IF ((EFF .LE. EFFM2) .OR. (PE .LT. PMIN) .OR. (FLOW .GT. 2.4))
        1       GO TO 20
        EFFM2 = EFF
        SPEED2 = SPEED
        QUAL12 = QUAL1
        PRAT2 = PRAT
        FLOW2 = FLOW
        QUAL22 = QUAL2
        POWER2 = POWER
        PMIN2 = PMIN
20      CONTINUE
        PRAT = PRAT + 0.50
100     CONTINUE
        QUAL1 = QUAL1 + 0.02
200     CONTINUE
        SPEED = SPEED + 200.0
300     CONTINUE
        TYPE 400, EFFM1, EFFM2
400     FORMAT(1H ,'MAXIMUM EFFICIENCY', T30, F4.1, T40, F4.1)
        TYPE 500, QUAL11, QUAL12
500     FORMAT(1H ,'INLET QUALITY', T30, F5.3, T40, F5.3)
        TYPE 600, SPEED1, SPEED2
600     FORMAT(1H ,'SPEED', T30, F6.0, T40, F6.0)
        TYPE 700, PRAT1, PRAT2
700     FORMAT(1H ,'PRESSURE RATIO' T30, F5.2, T40, F5.2)
        TYPE 800, POWER1, POWER2
800     FORMAT(1H ,'POWER', T30, F4.1, T40, F4.1)
        TYPE 900, FLOW1, FLOW2
900     FORMAT(1H ,'MASS FLOW RATE', T30, F5.3, T40, F5.3)
        TYPE 1000, QUAL21, QUAL22
1000    FORMAT(1H ,'EXHAUST QUALITY', T30, F5.3, T40, F5.3)
        TYPE 1100, PMIN1, PMIN2
```

```
1100    FORMAT(1H ,'MINIMUM EXHAUST PRESSURE', T30,
        1        F4.1, T40, F4.1)
        STOP
        END
```

```
C       POINT.FIN
C       PROGRAM PREDICTS PERFORMANCE FOR SELECTED POINTS
C       USING MOD
C       OPEN OUTPUT FILE
        OPEN (UNIT = 1, NAME = 'CURVE.DAT', TYPE = 'NEW',
        1       CARRIAGE CONTROL = 'LIST', ACCESS = 'SEQUENTIAL')
        WRITE(1,10)
10      FORMAT(1H ,'QUAL1', T9, 'SPEED', T18, 'PRATIO', T27, 'FLOW',
        1       T36, 'POWER', T45, 'EFF', T54, 'QUAL2', T63, 'PMIN')
20      CONTINUE
        TYPE *, ' ENTER INLET QUALITY, SPEED, PRESSURE RATIO'
        ACCEPT *, QUAL1, SPEED, PRATIO
        IF (SPEED .EQ. 0.0) GO TO 100
        CALL MOD(QUAL1, SPEED, PRATIO, FLOW,
        1       EFF, QUAL2, POWER, PMIN)
C       WRITE RESULTS TO DATA FILE
        WRITE(1,40)
40      FORMAT(1H ,'  ')
        WRITE (1,50) QUAL1, SPEED, PRATIO, FLOW,
        1       POWER, EFF, QUAL2, PMIN
50      FORMAT(1H ,F5.3, T9, F5.0, T18, F5.2, T27, F5.3, T36,
        1       F4.1, T45, F4.1, T54, F5.3, T63, F4.1)
        GO TO 20
100     CONTINUE
        CLOSE (UNIT = 1)
        STOP
        END
```

```
C       KAPPA1.FTN
C       PROGRAM COMPUTES EFFECTIVE KAPPA, ISENTROPIC ENTHALPY DROP AND S
C       DENSITY FOR EXPNSION FROM
C       120 PSIA TO ATMOSPHERIC PRESSURE FOR STARTING QUALITIES
C       OF 0.0 TO 1.0
C       OUTPUT FILE IS SUITABLE FOR USE AS A SOURCE FILE FOR GDA
        COMMON /LOGIN/ TL, PL, HL, SL, VL, XL, SPRHT
        REAL KAPPA
        LOGICAL TL, PL, HL, SL, VL, XL, SPRHT
C       OPEN RESULT FILE
        OPEN (UNIT =1, NAME = 'KAPPA.DAT', TYPE = 'NEW',
        1       CARRIAGECONTROL = 'LIST', ACCESS = 'SEQUENTIAL')
        P1 = 120.0
        P2 = 14.7
        QUAL1 = 0.0
        DO 100 I = 1,101
C       DETERMINE STARTING STATE
        PL = .TRUE.
        XL = .TRUE.
        CALL STMTBL(T1, P1, H1, S1, V1, QUAL1, SH1)
C       DETERMINE EXHAUST STATE
        PL = .TRUE.
        SL = .TRUE.
        CALL STMTBL(T2, P2, H2, S1, V2, QUAL2, SH2)
C       COMPUTE EFFECTIVE KAPPA(P * V**KAPPA = CONSTANT)
        KAPPA = (ALOG(P1 /P2)) / (ALOG(V2 / V1))
        DELTAH = H1 - H2
        DEN1 = 1.0 / V1
C       WRITE TO OUTPUT FILE
        WRITE (1,50) QUAL1, KAPPA, DELTAH, DEN1
50      FORMAT (1H , F4.2, T10, F8.6, T20, F7.3, T30, F9.4)
        QUAL1 = QUAL1 + 0.01
100     CONTINUE
        CLOSE (UNIT = 1)
        STOP
        END
```

```
C       LYST1.FTN
C       PROGRAM MANIPULATES AND REDUCES LYSHOLM DATA
        COMMON /LOGIN/ TL, PL, HL, SL, VL, XL, SPRHT
        COMMON /CDATA/ SC(3), BC(3), SST(3), BST(3), SLW(3), BW(3)
        LOGICAL TL, PL, HL, SL, VL, XL, SPRHT
        DIMENSION NRUN(585), MONTH(585), MDAY(585), MYEAR(585),
        1       MSCAN(585), MCAL(585), MCOM(585), BP(585),
        2       CH5(585), CH6(585), CH7(585), CH9(585), CH10(585),
        3       CH11(585), CH12(585), CH13(585)
C       OPEN DATA STORAGE FILE
        OPEN ( UNIT = 1, NAME = 'LYST.DAT', TYPE = 'OLD',
        1       ACCESS = 'DIRECT', RECORDSIZE = 15)
C       OPEN OUTPUT FILE
        OPEN (UNIT = 2, NAME = 'LTOUT.DAT', TYPE = 'NEW',
        1       CARRIAGECONTROL = 'LIST', ACCESS = 'SEQUENTIAL')
C       PROMPT FOR TASK INPUT
5       TYPE 10
10      FORMAT (1H, 'CHOOSE OPTION:', T20, '1 - STORE DATA' )
        TYPE 20
20      FORMAT (1H, T20, '2 - RETRIEVE DATA' )
        TYPE 30
30      FORMAT (1H, T20, '3 - REDUCE DATA AND OUTPUT RESULTS')
        TYPE 35
35      FORMAT (1H, T20, '4 - EXIT PROGRAM')
        ACCEPT *, IOPT
        GO TO (45, 45, 45, 2000), IOPT
        GO TO 5
45      CONTINUE
        GO TO (500, 200, 200, 2000), IOPT
200     CONTINUE
C       PROMPT FOR OUTPUT METHOD
        TYPE 210
210     FORMAT (1H, 'CHOOSE OUTPUT METHOD:', T30, '2 - OUTPUT FILE
        1 (EXITS PROGRAM)')
        TYPE 215
215     FORMAT (1H, T30, '5 - TERMINAL')
        ACCEPT *, IOUT
        GO TO (200, 230, 200, 200, 230), IOUT
        GO TO 200
230     CONTINUE
        GO TO 1000
500     CONTINUE
C       PROMPT FOR MATRIX POSITION
        TYPE *, 'ENTER I, J, K'
        ACCEPT *, I, J, K
        NREC = 95 * I + 19 * J + K
C       PROMPT FOR DATA
        TYPE *, 'ENTER NRUN, MONTH, DAY, YEAR, SCANS, CAL, BP, COM'
        ACCEPT *, NRUN(NREC), MONTH(NREC), MDAY(NREC), MYEAR(NREC),
        1       MSCAN(NREC), MCAL(NREC), BP(NREC), MCOM(NREC)
        TYPE *, 'ENTER CH5, CH6, CH7, CH9'
        ACCEPT *, CH5(NREC), CH6(NREC), CH7(NREC), CH9(NREC)
        TYPE *, 'ENTER CH10, CH11, CH12, CH13'
        ACCEPT *, CH10(NREC), CH11(NREC), CH12(NREC),CH13(NREC)
800     CONTINUE
```

```
C       WRITE DATA TO FILE
        WRITE (1,' NREC) NRUN(NREC), MONTH(NREC), MDAY(NREC),
        1       MYEAR(NREC), MSCAN(NREC), MCAL(NREC), BP(NREC),
        2       MCOM(NREC), CH5(NREC), CH6(NREC), CH7(NREC), CH9(NREC),
        3       CH10(NREC), CH11(NREC), CH12(NREC), CH13(NREC)
        GO TO 5
1000    CONTINUE
C       PROMPT FOR MATIX ELEMENTS
        TYPE *, 'ENTER I1ST, ILST, J1ST, JLST, K1ST, KLST'
        ACCEPT *, I1ST, ILST, J1ST, JLST, K1ST, KLST
        GO TO (5, 1050, 1020), IOPT
1020    CONTINUE
C       WRITE HEADING TO RESULTS TABLE
        WRITE (IOUT, 1900)
1900    FORMAT (1H , 'RUN#    DATE   COMMENT INLET  EXH   PRESS  SPEED
        1  FLOW   INLET   EFF   POWER FLOW  SPEC   SPEC  LEAK
        2 INLET  EXIT')
        WRITE (IOUT, 1910)
1910    FORMAT (1H , T25, 'PRESS  PRESS RATIO          RATE
        1   ENTHL                RATIO SPEED  SIZE  RATE
        2  QUAL   QUAL')
        WRITE (IOUT, 1915)
1915    FORMAT (1H , T25, '   [PSIA]            [RPM]  [LBM/  LBTU/
        1       [KW]                        [LBM/')
        WRITE (IOUT, 1917)
1917    FORMAT (1H , T25, '                                SEC]   SEC]
        1                                  SEC]')
        WRITE (IOUT, *) '   '
1050    CONTINUE
        DO 1955 I = I1ST,ILST
        DO 1954 J = J1ST,JLST
        DO 1953 K = K1ST,KLST
        NREC = 95 * I + 19 * J + K
C       READ DATA FROM FILE
        READ (1 ' NREC) NRUN(NREC), MONTH(NREC), MDAY(NREC),
        1       MYEAR(NREC), MSCAN(NREC), MCAL(NREC), BP(NREC),
        2       MCOM(NREC), CH5(NREC), CH6(NREC), CH7(NREC), CH9(NREC),
        3       CH10(NREC), CH11(NREC), CH12(NREC), CH13(NREC)
        IF (NRUN(NREC) .EQ. 0) GO TO 1950
        GO TO (5, 1160, 1500), IOPT
1160    CONTINUE
C       WRITE DATA TO OUTPUT FILE
        WRITE (IOUT, 1200) NRUN(NREC), MONTH(NREC), MDAY(NREC),
        1       MYEAR(NREC), MSCAN(NREC), MCAL(NREC), BP(NREC),
        2       MCOM(NREC), CH5(NREC), CH6(NREC), CH7(NREC), CH9(NREC),
        3       CH10(NREC), CH11(NREC), CH12(NREC), CH13(NREC)
1200    FORMAT (1H , I4, 2X, I2, '/', I2, '/', I2, 2X, I1, 2X, I1,
        1       2X, F5.2, 2X, I10, 2X, F5.1, 2X, F5.1, 2X, F5.1,
        2       2X, F5.1, 2X, F5.3, 2X, F5.3, 2X, F6.4, 2X, F6.4)
        GO TO 1950
1500    CONTINUE
C       REDUCE LYSHOLM DATA
C       TRANSLATE AUTODATA READINGS TO DATA
        T2 = CH5(NREC)
        T3 = CH6(NREC)
```

```
      TW = CH7(NREC)
      TCAL = CH9(NREC)
      MC = MCAL(NREC)
      STFLOW = 0.1 * SST(MC) * CH10(NREC) + BST(MC)
      IF (CH11(NREC) .LE. 0.0) GO TO 1600
      WFLOW = SLW(MC) * CH11(NREC) + BW(MC)
      GO TO 1700
1600  WFLOW = 0.0
1700  CONTINUE
      TORQ = 1000 * CH12(NREC)
      SPEED = 10000 * CH13(NREC)
C     DETERMINE TOTAL FLOW RATE
      TFLOW = STFLOW + WFLOW
C     DETERMINE STEAM ENTHALPY
      PCAL = 0.491 * BP(NREC)
      TL = .TRUE.
      PL = .TRUE.
      CALL STMTBL(TCAL, PCAL, HCAL, SCAL, VCAL, XCAL, SHCAL)
C     DETERMINE WATER ENTHALPY ASSUMING SATURATED CONDITIONS
      XW = 0.0
      TL = .TRUE.
      XL = .TRUE.
      CALL STMTBL(TW, PW, HW, SW, VW, XW, SHW)
C     CALCULATE INLET ENTHALPY
      H2 = (HCAL * STFLOW + HW * WFLOW) / TFLOW
C     DETERMINE INLET PROPERTIES
      TL = .TRUE.
      HL = .TRUE.
      CALL STMTBL(T2, P2, H2, S2, V2, X2, SH2)
C     FIX ISENTROPIC EXHAUST STATE
      S3S = S2
C     DETERMINE ISENTROPIC EXHAUST PROPERTIES
      TL = .TRUE.
      SL = .TRUE.
      CALL STMTBL(T3, P3S, H3S, S3S, V3S, X3S, SH3S)
C     CALCULATE ISENTROPIC ENTHALPY DROP
      DELTAH = H2 -H3S
C     CALCULATE MEASURED POWER
      DPOWER = TORQ * SPEED * 1.1832E-5
C     CORRECT FOR BELT LOSSES AND PARASITIC LOADS
      POWER = DPOWER + SPEED * 0.00012 - 0.155
C     CALCULATE EFFICIENCY(PERCENT)
      EFF = 100 * POWER * 0.9480 / ( TFLOW * DELTAH )
C     CALCULATE SPECIFIC SPEED
      SS1 = ( TFLOW * V3S ) ** 0.5 * SPEED / ( DELTAH * 778 ) ** 0.75
C     CALCULATE SPECIFIC SIZE
      D = 0.4267
      SD1 = D * ( DELTAH * 778 ) ** 0.25 / ( TFLOW * V3S ) ** 0.5
C     CALCULATE PRESSURE RATIO
C     IMPLICT ASSUMPTION: ALL EXHAUST STATES ARE INSIDE THE VAPOR DOME
      PRAT = P2 / P3S
C     CALCULATE FLOW RATIO
      FRAT = WFLOW / STFLOW
C     CALCULATE LEAKAGE FLOW RATE
      RLEAK = TFLOW - SPEED * 1.331E-4 / V2
```

```
      VS = 5.3*V2
      VL = .TRUE.
      SL = .TRUE.
      CALL STMTBL(TS,PS,HS,S2,VS,XS,SHS)
      CH = PS/P3S
C     CALCULATE EXHAUST ENTHALPY (ASSUMING NO LOSSES)
      H3 = H2 - POWER *0.9480/TFLOW
C     DETERMINE EXIT STATE
      TL = .TRUE.
      HL = .TRUE.
      CALL STMTBL(T3, P3, H3, S3, V3, X3, SH3)
C     WRITE RESULTS TO OUTPUT FILE
      WRITE (IOUT, 1920), NRUN(NREC), MONTH(NREC), MDAY(NREC),
     1        MYEAR(NREC), MCOM(NREC), P2, P3S, PRAT, SPEED,
     2        TFLOW, H2, EFF, POWER, FRAT, SS1, SD1, RLEAK,
     3        X2, X3
1920  FORMAT (1H , I4, 2X, I2, '/', I2, '/', I2, 2X, I5, 2X, F5.1,
     1        2X, F4.1, 2X, F5.2, 2X, F5.0, 2X, F5.3, 2X, F6.1, 2X,
     2        F4.1, 2X, F4.1, 2X, F4.2, 2X, F5.2, 2X, F4.2, 2X, F5.3,
     3        2X, F5.3, 2X, F5.3)
      WRITE (IOUT, *) '    '
1950  CONTINUE
1953  CONTINUE
1954  CONTINUE
1955  CONTINUE
      GO TO (5, 2000, 5, 5, 5), IOUT
      GO TO 5
2000  CONTINUE
      CLOSE (UNIT = 1)
      CLOSE (UNIT = 2)
      STOP
      END
C
C     SUBROUTINE
C
      SUBROUTINE STMTBL(T, P, H, S, V, X, SH)
C     CALCULATES THE PROPERTIES OF STEAM AT A GIVEN STATE
      COMMON /LOGIN/ TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
      COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
     1        VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
     2        SGT(27), SSUPT(27), TSUPT(27)
      LOGICAL TABLES, STATES, SPRHTS, TLS, PLS, HLS, SLS, VLS, XLS
C     SET PIVOT PARAMETER ARRAY SIZE
      N =27
1     CONTINUE
C     DETERMINE PIVOT PARAMETER
      IF ( PLS ) GO TO 20
      IF ( TLS ) GO TO 30
      IF ( VLS ) GO TO 2000
      STATES = .FALSE.
10    CONTINUE
C     IF STATE HAS NOT BEEN PROPERLY SPECIFIED, TYPE 'STATE NOT-
C     PROPERLY SPECIFIED' AND STOP EXECUTION
      TYPE 16
16    FORMAT (1H, 'STATE NOT PROPERLY SPECIFIED')
```

```
      STOP
20    CONTINUE
      CALL BOUND(P, PSAT, TABLES, I, N)
C     CHECK THAT THE STATE IS IN THE TABLE RANGE
      IF ( TABLES ) GO TO 25
22    CONTINUE
C     IF OUT OF TABLE RANGE, TYPE "OUT OF STEAM TABLE RANGE" AND
C     STOP EXECUTION
      TYPE 23
23    FORMAT (1H, 'OUT OF STEAM TABLE RANGE')
      STOP
25    CONTINUE
      FRAC = YLOG(P, PSAT(I), PSAT(I+1))
C     CHECK FOR IMPLICIT SPECIFICATION OF SUPERHHEAT CONDITIONS(T AND
      IF ( TLS ) SPRHTS = .TRUE.
      IF ( TLS ) GO TO 40
      GO TO 35
30    CONTINUE
      CALL BOUND(T, TSAT, TABLES, I, N)
C     CHECK THAT THE STATE IS IN THE TABLE RANGE AND EXIT SUBROUTINE I
      IF ( TABLES ) GO TO 32
      GO TO 22
32    CONTINUE
      FRAC = YLOG(T, TSAT(I), TSAT(I+1))
35    CONTINUE
40    CONTINUE
C     DETERMINE SATURATED VAPOR PROPERTIES FOR DECISION ON SUPERHEAT
      HG = ZLOG(FRAC, HGT(I), HGT(I+1))
      SG = ZLOG(FRAC, SGT(I), SGT(I+1))
      VG = ZLOG(FRAC, VGT(I), VGT(I+1))
C     GO TO SUPERHEAT SECTION IF IMPLICITLY SPECIFIED
      IF ( SPRHTS ) GO TO 500
C     DETERMINE SATURATED LIQUID PROPERTIES
      HF = ZLOG(FRAC, HFT(I), HFT(I+1))
      SF = ZLOG(FRAC, SFT(I), SFT(I+1))
      VF = ZLOG(FRAC, VFT(I), VFT(I+1))
C     CALCULATE QUALITY FROM SECOND INPUT VARIABLE
      IF ( XLS ) GO TO 90
      IF ( HLS ) GO TO 100
      IF ( SLS ) GO TO 110
      IF ( VLS ) GO TO 120
      GO TO 10
90    CONTINUE
      IF ( HLS .OR. VLS .OR. SLS) GO TO 10
      GO TO 200
100   CONTINUE
      IF ( SLS .OR. VLS ) GO TO 10
C     CHECK FOR SUPERHEAT CONDITIONS
      IF ( H .GT. HG ) GO TO 490
      X = YLIN(H, HF, HG)
      GO TO 200
110   CONTINUE
      IF ( VLS ) GO TO 10
      IF ( S .GT. SG) GO TO 490
      X = YLIN(S, SF, SG)
```

```
      GO TO 200
120   CONTINUE
      IF ( V .GT. VG ) GO TO 490
      X = YLIN(V, VF, VG)
200   CONTINUE
C     CHECK FOR NEGATIVE QUALITY INDICATING SUBCOOLED CONDITIONS
      IF ( X .LE. -0.00005 ) GO TO 220
      GO TO 230
220   CONTINUE
      TYPE 225
225   FORMAT (1H, 'NEGATIVE QUALITY INDICATES SUBCOOLED CONDITIONS-OU
      1 OF TABLE RANGE')
230   CONTINUE
      IF (X .LE. 1.005) GO TO 240
      GO TO 10
240   CONTINUE
C     CALCULATE UNKNOWN PROPERTIES
      IF ( PLS ) GO TO 250
      P = ZLOG(FRAC, PSAT(I), PSAT(I+1))
250   CONTINUE
      IF ( TLS ) GO TO 260
      T = ZLOG(FRAC, TSAT(I), TSAT(I+1))
260   CONTINUE
      IF ( HLS ) GO TO 270
      H = ZLIN(X, HF, HG)
270   CONTINUE
      IF ( SLS ) GO TO 280
      S = ZLIN(X, SF, SG)
280   CONTINUE
      IF ( VLS ) GO TO 290
      V = ZLIN(X, VF, VG)
290   CONTINUE
      GO TO 730
490   CONTINUE
      IF ( PLS ) GO TO 495
      GO TO 800
495   CONTINUE
      SPRHTS = .TRUE.
500   CONTINUE
C     DETERMINE PROPERTIES AT 70 DEGREES SUPERHEAT
      TSUP = ZLOG(FRAC, TSUPT(I), TSUPT(I+1))
      HSUP = ZLOG(FRAC, HSUPT(I), HSUPT(I+1))
      SSUP = ZLOG(FRAC, SSUPT(I), SSUPT(I+1))
      VSUP = ZLOG(FRAC, VSUPT(I), VSUPT(I+1))
      TSATG = ZLOG(FRAC, TSAT(I), TSAT(I+1))
      IF ( TLS ) GO TO 600
      IF ( HLS ) GO TO 610
      IF ( SLS ) GO TO 620
      IF ( VLS ) GO TO 630
      GO TO 10
C     CALCULATE SUPERHEAT FRACTION FROM SECOND INPUT VARIABLE
600   CONTINUE
      IF ( HLS .OR. SLS .OR. VLS ) GO TO 10
      SHF = YLIN(T, TSATG, TSUP)
      GO TO 650
```

```
610     CONTINUE
        IF ( SLS .OR. VLS ) GO TO 10
        SHF = YLIN(H, HG, HSUP)
        GO TO 650
620     CONTINUE
        IF ( VLS ) GO TO 10
        SHF = YLIN(S, SG, SSUP)
        GO TO 650
630     CONTINUE
        SHF = YLIN(V, VG, VSUP)
650     CONTINUE
        IF (SHF .LE. -0.00005) GO TO 660
        GO TO 670
660     CONTINUE
        TYPE 665
665     FORMAT (1H, 'NEGATIVE SUPERHEAT INDICATES P AND T ILLEGALY
        1 SPECIFIED SUBCOOLED CONDITIONS')
670     CONTINUE
C       CALCULATE UNKNOWN PROPERTIES
        SH = SHF * (TSUPT(I) - TSAT(I))
        IF ( TLS ) GO TO 700
        T = ZLIN(SHF, TSATG, TSUP)
700     CONTINUE
        IF ( HLS ) GO TO 710
        H = ZLIN(SHF, HG, HSUP)
710     CONTINUE
        IF ( SLS ) GO TO 720
        S = ZLIN(SHF, SG, SSUP)
720     CONTINUE
        IF ( VLS ) GO TO 730
        V = ZLIN(SHF, VG, VSUP)
730     CONTINUE
        GO TO 950
800     CONTINUE
        IF (HLS) GO TO 810
        IF (SLS) GO TO 820
        IF (VLS) GO TO 830
        GO TO 10
810     CONTINUE
        CALL TSPR(T, H, FRAC, HGT, HSUPT, SHF1, SHF2, M)
        GO TO 900
820     CONTINUE
        CALL TSPR(T, S, FRAC, SGT, SSUPT, SHF1, SHF2, M)
        GO TO 900
830     CONTINUE
        CALL TSPR(T, V, FRAC, VGT, VSUPT, SHF1, SHF2, M)
900     CONTINUE
        IF (HLS) GO TO 910
        CALL DINT(H, FRAC, SHF1, SHF2, HGT, HSUPT, M)
910     CONTINUE
        IF (SLS) GO TO 920
        CALL DINT(S, FRAC, SHF1, SHF2, SGT, SSUPT, M)
920     CONTINUE
        IF (VLS) GO TO 930
        CALL DINT(V, FRAC, SHF1, SHF2, VGT, VSUPT, M)
```

```
930     CONTINUE
        P = ZLOG(FRAC, PSAT(M), PSAT(M+1))
        GO TO 950
2000    CONTINUE
        CALL QUAL(V,S,P)
        GO TO 1
950     CONTINUE
C       INITIALIZE LOGICAL INPUT SPECIFIERS FOR SUBSEQUENT CALLS
        TLS = .FALSE.
        PLS = .FALSE.
        HLS = .FALSE.
        SLS = .FALSE.
        VLS = .FALSE.
        XLS = .FALSE.
        SPRHTS = .FALSE.
1000    RETURN
        END
C
C       BLOCK DATA
C
        BLOCK DATA
C       STEAM DATA, ENGLISH UNITS (REFERENCE 1967 ASME STEAM TABLES)
        COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
        1       VSUFT(27), HFT(27), HGT(27), HSUFT(27), SFT(27),
        2       SGT(27), SSUFT(27), TSUFT(27)
        DATA TSAT/ 100.0, 110.0, 120.0, 130.0, 140.0, 150.0,
        1       160.0, 170.0, 180.0, 190.0, 200.0, 210.0, 220.0,
        2       230.0, 240.0, 250.0, 260.0, 270.0, 280.0, 290.0, 300.0,
        3       310.0, 320.0, 330.0, 340.0, 350.0, 360.0/
        DATA PSAT/ 0.94924, 1.2750, 1.6927, 2.2230, 2.8892, 3.7184,
        1       4.7414, 5.9926, 7.5110, 9.340, 11.526, 14.123,
        2       17.186, 20.779, 24.968, 29.825, 35.427, 41.856, 49.200,
        3       57.550, 67.005, 77.667, 89.643, 103.045, 117.992,
        4       134.604, 153.010/
        DATA VFT/ 0.01613, 0.01617, 0.01620, 0.01625, 0.01629, 0.01634,
        1       0.01640, 0.01645, 0.01651, 0.01657, 0.01664, 0.01671,
        2       0.01678, 0.01685, 0.01693, 0.01701, 0.01709, 0.01718,
        3       0.01726, 0.01736, 0.01745, 0.01755, 0.01766, 0.01776,
        4       0.01787, 0.01799, 0.01811/
        DATA VGT/ 350.4, 265.39, 203.26, 157.33, 123.00, 97.07,
        1       77.29, 62.06, 50.225, 40.957, 33.639, 27.816, 23.148,
        2       19.381, 16.321, 13.819, 11.762, 10.060, 8.6439, 7.4603,
        3       6.4658, 5.6260, 4.9138, 4.3069, 3.7878, 3.3418, 2.9573/
        DATA HFT/ 67.999, 77.98, 87.97, 97.96, 107.95, 117.95,
        1       127.96, 137.97, 148.00, 158.04, 168.09, 178.15,
        2       188.23, 198.33, 208.45, 218.59, 228.76, 238.95,
        3       249.17, 259.4, 269.7, 280.0, 290.4, 300.8, 311.3,
        4       321.8, 332.3/
        DATA HGT/ 1105.1, 1109.3, 1113.6, 1117.8, 1122.0, 1126.1,
        1       1130.2, 1134.2, 1138.2, 1142.1, 1146.0, 1149.7,
        2       1153.4, 1157.1, 1160.6, 1164.0, 1167.4, 1170.6, 1173.8,
        3       1176.8, 1179.7, 1182.5, 1185.2, 1187.7, 1190.1, 1192.3,
        4       1194.4/
        DATA SFT/ 0.1295, 0.1472, 0.1646, 0.1817, 0.1985, 0.2150,
        1       0.2313, 0.2473, 0.2631, 0.2787, 0.2940, 0.3091,
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     2        0.3241, 0.3388, 0.3533, 0.3677, 0.3819, 0.3960,
     3        0.4098, 0.4236, 0.4372, 0.4506, 0.4640, 0.4772,
     4        0.4902, 0.5032, 0.5161/
     DATA SGT/ 1.9825, 1.9577, 1.9339, 1.9112, 1.8895, 1.8686,
     1        1.8487, 1.8295, 1.8111, 1.7934, 1.7764, 1.7600,
     2        1.7442, 1.7290, 1.7142, 1.7000, 1.6862, 1.6729, 1.6599,
     3        1.6473, 1.6351, 1.6232, 1.6116, 1.6003, 1.5892, 1.5784,
     4        1.5678/
     DATA VSUPT/ 394.6, 298.36, 228.15, 176.32, 137.64, 108.48,
     1        86.27, 69.19, 55.94, 45.57, 37.405, 30.902, 25.702,
     2        21.507, 18.104, 15.323, 13.039, 11.151, 9.581, 8.271,
     3        7.170, 6.241, 5.454, 4.783, 4.210, 3.7174, 3.2932/
     DATA HSUPT/ 1136.6, 1141.0, 1145.3, 1149.7, 1154.0, 1158.2,
     1        1162.5, 1166.8, 1170.9, 1175.1, 1179.2, 1183.3, 1187.3,
     2        1191.2, 1195.1, 1198.9, 1202.7, 1206.4, 1210.0, 1213.6,
     3        1217.0, 1220.4, 1223.7, 1226.9, 1229.9, 1232.9, 1235.8/
     DATA SSUPT/ 2.0356, 2.0101, 1.9857, 1.9623, 1.9399, 1.9816,
     1        1.8982, 1.8785, 1.8597, 1.8416, 1.8243, 1.8076, 1.7916,
     2        1.7761, 1.7612, 1.7469, 1.7331, 1.7197, 1.7067, 1.6942,
     3        1.6821, 1.6703, 1.6589, 1.6478, 1.6370, 1.6265, 1.6163/
     DATA TSUPT/ 170.0, 180.0, 190.0, 200.0, 210.0, 220.0,
     1        230.0, 240.0, 250.0, 260.0, 270.0, 280.0, 290.0,
     2        300.0, 310.0, 320.0, 330.0, 340.0, 350.0, 360.0,
     3        370.0, 380.0, 390.0, 400.0, 410.0, 420.0, 430.0/
     END
C
C    BLOCK DATA
C
     BLOCK DATA
C    CALIBRATION CONSTANTS FOR FLOW METERS
     COMMON /CDATA/ SC(3), BC(3), SST(3), BST(3), SLW(3), BW(3)
     DATA SC/ 0.8664, 0.8664, 0.8664/
     DATA BC/ 0.0707, 0.0707, 0.0707/
     DATA SST/ 1.0, 1.0214, 1.0214/
     DATA BST/ 0.0, -0.0059, -0.0059/
     DATA SLW/ 1.0, 0.8538, 0.1861/
     DATA BW/ 0.0, 0.0756, -0.0122/
     END
C
C    SUBROUTINE
C
     SUBROUTINE TSPR(TT, Q, FRACT, G, SUP, SHFT1, SHFT2, J)
     COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
     1        VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
     2        SGT(27), SSUPT(27), TSUPT(27)
     DIMENSION G(27), SUP(27), SHFT(27), QSH(27)
     NT = 27
     DO 100 J = 1, NT
     SHFT(J) = (TT - TSAT(J)) / (TSUPT(J) - TSAT(J))
     QSH(J) = ZLIN(SHFT(J), G(J), SUP(J))
100  CONTINUE
     CALL RBOUND (Q, QSH, TABLES, J, NT)
     FRACT = YLOG(Q, QSH(J), QSH(J+1))
     SHFT1 = SHFT(J)
     SHFT2 = SHFT(J+1)
```

```
      RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE DINT(QD, FRACD, SHFD1, SHFD2, GD, SUPD, KD)
      DIMENSION GD(27), SUPD(27)
      QD1 = ZLIN(SHFD1, GD(KD), SUPD(KD))
      QD2 = ZLIN(SHFD2, GD(KD+1), SUPD(KD+1))
      QD = ZLOG(FRACD, QD1, QD2)
      RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE BOUND(B, BT, BTABLE, IB, NB)
      DIMENSION BT(NB)
      LOGICAL BTABLE
      NBM = NB - 1
      DO 20 IB = 1, NBM
      IF ( B .GE. BT(IB)) GO TO 10
      GO TO 30
10    IF ( B .LE. BT(IB + 1)) GO TO 40
20    CONTINUE
30    BTABLE = .FALSE.
      GO TO 50
40    BTABLE = .TRUE.
50    RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE RBOUND(B, BT, BTABLE, IB, NB)
      DIMENSION BT(NB)
      LOGICAL BTABLE
      NBM = NB - 1
      DO 20 IB = 1, NBM
      IF (B .LE. BT(IB)) GO TO 10
      GO TO 30
10    IF (B .GE. BT(IB + 1)) GO TO 40
20    CONTINUE
30    BTABLE = .FALSE.
      GO TO 50
40    BTABLE = .TRUE.
50    RETURN
      END
C
C     FUNCTION
C
      FUNCTION ZLIN(A, B1, B2)
      ZLIN = B1 + A * (B2 - B1)
      RETURN
      END
C
C     FUNCTION
```

```
C
      FUNCTION ZLOG(C, D1, D2)
      ZLOG = EXP(ALOG(D1) + C * (ALOG(D2) - ALOG(D1)))
      RETURN
      END
C
C     FUNCTION
C
      FUNCTION YLIN(E, E1, E2)
      YLIN = (E - E1)/(E2 -E1)
      RETURN
      END
C
C     FUNCTION
C
      FUNCTION YLOG(F, F1, F2)
      YLOG = (ALOG(F) - ALOG(F1))/(ALOG(F2)- ALOG(F1))
      RETURN
      END
C
C     SUBROUTINE
C
      SUBROUTINE QUAL(V,S,P)
      COMMON /STDATA/ TSAT(27), PSAT(27), VFT(27), VGT(27),
     1        VSUPT(27), HFT(27), HGT(27), HSUPT(27), SFT(27),
     2        SGT(27), SSUPT(27), TSUPT(27)
      COMMON /LOGIN/ TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
      DIMENSION Q(27), VM(27)
      LOGICAL TLS, PLS, HLS, SLS, VLS, XLS, SPRHTS
      NT = 27
      DO 100 J=1,NT
      Q(J) = YLIN(S, SFT(J), SGT(J))
100   CONTINUE
      DO 200 J=1,NT
      VM(J) = ZLIN(Q(J),VFT(J),VGT(J))
200   CONTINUE
      CALL BOUND(V,VM,TABLES,J,NT)
      Y = YLOG(V,VM(J),VM(J+1))
      P = ZLOG(Y,PSAT(J),PSAT(J+1))
      VLS = .FALSE.
      PLS = .TRUE.
      RETURN
      END
```