· ALTERING THE VOLUMETRIC EXPANSION RATIO OF
A LYSHOLM HELICAL SCREW EXPANDER

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ABSTRACT

This is an analysis of the effects of the volumetric expansion ratio on the operation of a Lysholm helical screw expander. Extensive testing of the University of California Lysholm engine with a 5.3 volumetric expansion ratio was performed by Karl Brown¹ and Ralph Berger². In this experiment, the expansion ratio has been reduced from 5.3 to 4.0. Tests were performed at a variety of speeds and qualities for a 5.0 pressure ratio and at a variety of speeds and pressure ratios for 99 percent quality.

It was predicted that decreasing the volumetric expansion ratio would decrease the leakage fraction and thereby increase efficiency. This occurred as predicted. Isentropic efficiency increased 20 percent for 50 percent quality steam and 16 percent for 33 percent quality steam, all for an inlet pressure of 120 psia and a speed of 9000 rpm. A maximum efficiency of 47.2 percent was reached at 33 percent quality, 9000 rpm and a pressure ratio of 5.0. It was noted that the leakage rate did not appear to be a function of expansion ratio, as had been expected. Using this fact the previous empirical models of the engine were extended to include volumetric expansion ratio as a variable parameter.

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INTRODUCTION

The Lysholm helical screw expander is a positive displacement engine consisting of two parallel intermeshing helical rotors enclosed in a casing with an intake port at one end and an exhaust port at the other (Figures 1-3). It is a durable device, capable of producing power from geothermal steam sources using two phase flow. Fluids of qualities from 100 percent to pure water may be utilized. The geometry of its ports determine the duration of the intake and exhaust cycles. As the rotors turn past the intake port, a volume of steam is let in, determined by the geometry of the port and of the rotors. Further rotation brings a new cavity in line with the intake and isolates the initial fluid. A chevron shaped cavity, formed by the intersecting rotors and the end plate, expands and moves down the rotor length as rotation continues. It is during this expansion process that work is extracted from the steam. Once the chamber has arrived at the end of the rotor length the cavity is opened, as the rotors unmesh, and exhaust occurs. At this point, the steam may either be discarded or conditioned for use in a second stage process, usually a turbine.

The Lysholm configuration was invented by Alf Lysholm and patented in 1934. It was first used as a compressor, and still is today. The possibility of applications as an engine were first noted in 1952 by H.R. Nilsson and in 1966 by Alf Lysholm. The first suggestion of using the Lysholm as an engine with two phase fluids came in 1973 when R. Sprankle patented the engine for use with low quality geothermal steam. Tests were performed by Sprankle in conjunction with Jet Propulsion Laboratory, by Lawrence Livermore National Laboratory and currently at the Uni-

versity of California.

Previous experiments have revealed that, while the expansion is efficient, a great deal of energy is lost due to leakage. High leakage rates are an inherent difficulty with the design. The sealing of the expansion chamber is defined by the meshline between the two rotors, the circumference of the rotors against the case and by the high pressure ends of the rotors against the inlet plate. This complicated seal line results in a large area over which leakage may pass. The leakage fraction typically has been fifty percent of the total flow.

There are several ways that the leakage fraction may be decreased. For instance, clearances may be decreased. Increasing engine speed is another way to reduce the leakage fraction. Problems are encountered with either one of these approaches. In the current configurations, clearances are already close enough that the rotors have locked together during cooling several times. Increasing the speed is not a good solution because, at 9000 rpm, we are already exceeding the recommended operation speed of the flexible belt between the engine and the brake by 2500 rpm. Failure of this belt could be catastrophic for the engine. Also, at higher speed, dynamic losses increase, associated with accelerating the water.

Another way to decrease leakage flow is to increase the flow rate. This can be accomplished by increasing the number of degrees of rotation through which the inlet port is open, thereby increasing the volume of intake and decreasing the volumetric expansion ratio. This was the

method decided on for this experiment. It is of special interest since, while a great deal of experimentation has been performed with different ratios of inlet to outlet pressures, none has been performed with different volumetric expansion ratios for the same geothermal machine.

The University of California Lysholm engine is a modified Gardener Denver air engine. The two helical rotors are 6.519 inches long and 5.12 inches in diameter. They consist of one female rotor with six lobes and one male rotor with four lobes (Figures 2,3). The four lobe rotor rotates one and a half times for each revolution of the six lobe rotor. The shaft of the high speed rotor is the engine output shaft. The rotors are made of stainless steel coated with teflon; two timing gears hold them from touching. The timing gears, rotors and two oil slingers are the only moving parts in the engine.

Rotation of the rotors and the inlet port geometry define all stages of the expansion process. For the 4.0 expansion ratio port, intake occurs from the time the rotors first begin to unmesh until a rotation of 89 degrees, for the male rotor (133 degrees for the female rotor), has occurred. After this angle, the inlet port becomes completely blocked by the subsequent lobes, and isolated expansion occurs for the next 178 degrees of rotation for the male rotor. At this point, the expansion chamber has moved down the rotor length to the exhaust end of the case and exhaust begins occurring when the chamber unmeshes. Free exhaust occurs for 43 degrees, and forced exhaust occurs for 234 degrees, as the lobes remesh on the back side.

Since the above cycle is occurring simultaneously in all the chambers of the rotors, the inlet and exhaust flows are fairly constant. At all times there is at least one inlet port open, and, for slightly less than half the time, two are open.

One of the difficulties with the Berkeley engine is

that water from the working fluid travels up the rotor shafts and finds its way into the oil supplied for bearing lubrication. An arrangement of successive areas of water pressure, air pressure and oil pressure has been designed to prevent this occurrence (Figure 4). Guages are fitted to the engine to monitor the flow of seal oil, water and air.

Other guages fitted to the engine for fluid monitoring and control purposes are for seal pressures, oil inlet and outlet temperatures, fluid intake temperature and exhaust temperature. Information from these sources coupled with that from the other test equipment are sufficient to completely specify operating conditions.

The working fluid for the engine was a mixture of hot water and steam, as a synthetic geothermal steam. Steam at any desired quality between ninety-nine percent and zero could be produced at pressures ranging from 120 psia to subatmospheric. Campus heating steam, supplied at 125 psia and 99 percent quality, was mixed with water heated by running steam through a heat exchanger. Before mixing, the mass flow rate of each fluid was measured by orifice plates. From this, quality was computed. Pressure was controlled by throttling the steam and water lines upstream from the orifice plates.

Flow rates were measured by square edge orifice plates using flange type pressure taps. Foxboro 13A differential pressure cells were used to convert the pressure drop to pressure signals between 3 - 15 psig. The pressure signals were sent to Foxboro M40G square root extractors. The square roots of the signals were multiplied by appropriate constants and sent as flow-rate signals to an Autodata Nine and to a Foxboro 130M controller for steam and 130F Controller for water.

The engine's output power ran through a 3 to 1 reduction, via a flexible belt, to a jack shaft and an eddy current brake. Torque and speed were taken from the jack shaft by a Lebow torque sensor Model 1605-2k which incorporated resistance wire strain gauges for torque measurement and a magnetic pickup for signal transmission. The signals were sent to displays on the control panel and to the Autodata Nine data logger.

The Brake was a Telma CA 135 eddy current dynamometer.

Its resistance was controlled by input power from a Hewlett Packard LVR 6261B power supply mounted on the control panel.

The Autodate Nine was set up to record all the necessary parameters. These were steam inlet temperature and flow rate, calorimeter temperature, exhaust temperature, speed and torque. Three printouts of each of these variables were made at each data point to ensure that steady state had been reached. Average values of each parameter were recorded and entered into computer programs for data reduction.

The previous experimentation, done with 5.3 volumetric expansion ratio inlet port, utilized data reduction programs written by Karl Brown¹ for use on the PDP 11/60 mini computer. The 4.0 volumetric expansion ratio data were reduced by two slightly modified versions of those programs.

Raw data for both the 5.3 and 4.0 port are stored in separate three dimensional matrices. Each data point is indexed by "i", "j" and "k" locations where these letters correspond to codes for quality, speed and pressure ratio. The values of these codes are given in Table 1.

To locate and reduce a data point, the three code numbers are entered and it is located in the direct access data file. This file is written in machine language, hence it must be read by the program. The raw data are reduced according to one of the reduction programs. No reduced data are stored.

The two reduction programs included in this text yield output of different natures. One is for human consumption and prints with the data in columns with headings and peripheral information such as run numbers date and calibration codes. The other is a source file for the graphics device.

It is not possible to reduce 5.3 volumetric expansion data with either of these programs. Reduction of 5.3 data will result in errant flow rates and leakage rates since a 4.0 expansion ratio is assumed by the program.

Test Equipment Limitations

One of the most serious limitations is the 3 to 1 speed reduction from the engine to the brake. This reduction is necessitated by the speed limitations of the brake and is accomplished with a two inch wide toothed belt running over a 4 inch and a 12 inch pulley. At our highest operating speed, 9000 rpm, we are exceeding the manufacture's recommendations for the operation of the belt by 2500 rpm. At this speed the radial accelerations is 4600 g's. In the interest of safety and desire to protect the machine we decided not to operate above 9000 rpm.

The exhaust system also limited performance. Tortuous piping restricts the minimum exhaust pressure available. This minimum value is a function of mass flow rate and quality. For 9000 rpm and 25 percent quality, our highest mass flow rate run, 25 psia was the lowest attainable exhaust pressure.

On the inlet side, the maximum pressure was limited by the steam pressure consistently available in the campus

line. At the beginning of testing this was about 117 psia, later the available pressure rose to 125 psia. All tests in the data matrix are taken at 120 psia.

As tests were performed, especially those with low quality steam, water found its way into the oil. It took about one hour at 50 percent quality to fill the oil tank with water, thus requiring that the system be shut down. This shortcoming of the system required that tests be taken quickly and allowed only a short period of time for conditions to stabilize at each data point. This has contributed to the scatter of data points taken on this machine. An examination of those points that appear in error often revealed that they were taken on days with a high productivity and hence less time per point.

The engine start-up procedure proposed by Ralph Berger² was followed. Steam was directed through the engine bypass and then shut off to check that the flow meters went to zero. The engine was run slowly on pure steam until the oil temperature exceeded 140°F.

With the engine warm, data points were taken at speeds of 4500, 5000, 7500 and 9000 rpm and qualities of 25, 33, 50 and 99 percent for a pressure ratio of 5 to 1. This created a data matrix of sixteen points. Data were also taken at a range of pressure ratios and speeds for 99 percent quality.

Initially, a data point was taken as an average of three scans in which all the controlled parameters were within acceptable limits. Later, it was determined that it was necessary for these scans to be consecutive or nearly consecutive to produce satisfactory points.

All runs were made with an inlet pressure of 120 psia. Since the intake was always two phase, temperature was chosen as a measure of correct inlet pressure. Temperature was more accurately measured than pressure by our set up. The same procedure was followed with the exhaust pressure.

Inlet pressure was controlled by throttling inlet steam and water and automatic valves and Foxboro controllers. The desired exhaust pressure was attained by a trial and error setting of the separator bypass valve and held steady by the automatically controlled air ejector which raised and lowered the pressure in the condenser. Achieving the proper exhaust pressure was not easy, and it took time to reach a steady state.

The flow ratio was controled to some degree by the Foxboro automatic controller designed for that purpose. In theory, a ratio could be set and then, while the flow rate changed, the correct ratio would be maintained. However, since the controller received uncorrected flow rate signals straight from the square root extractor, the set point had to constantly be changed. A calculator program was written to correlate the Autodata readings for water and steam flow to produce the desired ratio (Figure 5).

With these four parameters controled, data scans were taken of steam, water, steam calorimeter and exhaust temperatures as well as water and steam flow rates, torque and speed.

A great deal of care was taken in verifying the accuracy of the points taken. For flow rate measures, large volumes of fluid were measured over long periods of time to be certain that transient effects in the system were insignificant. For both steam and water, between 300 and 400 pounds of fluid were run through the orifice meters into weigh tanks for each point on the calibration curve. Data points were also analyzed carefully for engine runs. Points were plotted in many different fashions; any points which appeared inconsistent were run again to verify repeatability.

In previous experiments with the Livermore engine, it has been determined that leakage through the rotors accounted for about fifty percent of the total flow, making leakage the number one loss. One of the highest priorities in the development of the Lysholm engine is the reduction of leakage losses. The choice of a smaller volumetric expansion ratio was an attempt in this direction.

A smaller expansion ratio decreases the leakage fraction, since a smaller volume ratio means a larger inlet volume and hence a greater mass flow while the leakage remains relatively constant. Reducing the volumetric expansion ratio from the previous 5.3 to 4.0 yields an increase intake volume and intake mass of $(5.3 - 4.0)/4.0$ or thirty-three percent. Meanwhile the leakage rate was expected to stay near constant since, at the pressure ratios at which the machine was to be operated, the leakage paths were calculated to be fully choked.

Flow rate for a given expansion ratio was predicted by determining an ideal flow rate for that ratio, and then adding it to the leakage rate observed in 5.3 expansion ratio runs at the same inlet pressure. Ideal or theoretical flow rate was calculated by multiplying the density of the fluid and the speed. Also, in this manner efficiency can be predicted.

Since,

$$\eta_{VR} = \frac{\dot{M}_{VR} (\Delta h_{theory})}{(\dot{M}_{VR} + \dot{M}_L) \Delta h}$$

Then by dividing by a previously observed efficiency and leakage, a theoretical efficiency can be calculated for any volumetric expansion ratio.

η_{vr} = Efficiency at an arbitrary volumetric expansion ratio

$\dot{M}_{vr}$ = Theoretical flow at expansion ratio VR

$\dot{M}_L$ = Leakage

$$\frac{\eta_{vr}}{\eta_{5.3}} = \frac{\dot{M}_{vr}}{\dot{M}_{vr} + \dot{M}_L} \cdot \frac{\Delta h_{vr \text{ THEORY}}}{\Delta h_{vr}} \cdot \frac{\Delta h_{5.3}}{\Delta h_{5.3 \text{ THEORY}}} \cdot \frac{\dot{M}_{5.3} + \dot{M}_L}{\dot{M}_{5.3}}$$

$$\eta_{vr} \cong \eta_{5.3} \left(\frac{\dot{M}_L}{\dot{M}_{5.3} + 1} \right) / \left(\frac{\dot{M}_L}{\dot{M}_{vr} + 1} \right)$$

Where $\eta_{5.3}$ is taken at a similar pressure ratio.
Similarity is described by:

$$PR_{5.3} = PR_{vr} \cdot \frac{5.3}{VR}$$

It should be noted that, while all the above calculations were made using data from the 5.3 expansion ratio, any expansion ratio could be used. 5.3 was chosen only because it was available at the time these calculations were made.

Figures 6 and 7 give plots of the above theoretical efficiencies versus speed for a number of different qualities and two expansion ratios. Figure 8 plots theoretical efficiency versus pressure ratio for five different volumetric ratios.

For the 4.0 volume ratio port, a maximum efficiency was calculated of 46.4 percent at 33 percent quality and 9000 rpms.

There are four paths along which leakage through the engine may occur. At the mesh line, leakage may pass between the rotors to the exhaust. Along the seal line between the rotor and the case, leakage may pass to the next chamber. Leakage also passes through the gap formed at the intersection of the rotors and the case (figure 2). These last two paths also both lead to the next chamber.

Leakage occurs continuously through all four of these paths throughout the cycle. However, the method of calculating leakage only takes into account that leakage which occurs while the inlet port is open. Leakage flow rate was calculated as the total flow rate minus the theoretical flow rate. The actual flow rate of leakage from the expansion chambers is about three times this value. Leakage and loss, however, are two different things; only the leakage through the mesh line goes to the exhaust. Over 70 percent of the total leakage goes to the chamber of the next lower pressure and does work there.

The area of the leakage paths is a function of rotation angle, since the circumferential seal line changes length and since for some ranges of angle more than one inlet port is open. Figures 9 and 10 give leakage area as a function of angle.

The areas of the leakage paths were calculated and measured. The area of the case-rotor intersection gap was measured by extruding a piece of clay through the space between the rotors and measuring its dimensions. The mesh

line clearance was measured by the insertion of feeler gauges and wires of known diameter. The circumferential seal line area was calculated from measurements of rotor diameter and casing bore and rotor wrap angle. The leakage area around the front of the rotors was taken as the product of the nominal high pressure end clearance and the seal length.

Because of the small clearances being measured, high accuracy was difficult to obtain, but accuracy of the relative total leakage areas between 5.3 and 4.0 was expected to be high. Accuracy of the circumferential seal line area was estimated to be 10 percent. The accuracy of the other three areas was 20 percent. The total leakage area is within 17 percent.

The areas are as follows:

	<u>4.0 Port</u>	<u>5.3 Port</u>
Leakage around the front face	.012 in ²	.012 in ²
Average circumferential seal line	.0297	.019
Average mesh line	.0324	.0269
Case-rotor intersection	.0361	.0361
Average total area	.110	.094
Ratio of leakage areas	1.17	

As a comparison, the leakage area may be calculated using Grashoff's relation. This was done by Ralph Berger² for the 5.3 inlet port.

His results were:

<u>Pressure</u>	<u>Area</u>
75 psia	.146 in ²
90	.160
105	.159
102	.174

These results are larger than the areas measured. Water in the steam accounts for some of the discrepancy. Grashoff's relation applies strictly to pure steam; data used for the above calculation were at an inlet quality of 99 percent. Water in the fluid was probably thrown to the circumference by centrifugal force. Water that accumulated along the seal line would be forced through by the differential pressure; since there is no choking for water flow, mass flow could exceed that possible for pure steam.

For accurate analysis, it is desirable to know the actual volumetric expansion ratio as precisely as possible. From measurements made while the engine was disassembled, the areas of the male and female lobes were found to be 1.626 in^2 and 1.303 in^2 respectively. The increases in intake angles were also measured for the new port. By multiplying these by the appropriate function of the wrap angle, the inlet volume was found to increase 4.75 in^3 over the 5.3 in^3 inlet volume. Adding this number to the 5.3 in^3 volume yielded 18.76 in^3 . The exhaust volume was found by multiplying the lobe areas by the rotor length. In this manner the volumetric expansion was found to be 4.07.

The above technique was dependent on the accuracy of measuring the wrap angle, the lobe areas and the accuracy of the given 5.3 in^3 inlet port volume. An alternate measurement was made by packing the inlet volume with clay and then carefully removing the clay and measuring its volume. This method, which substituted the inaccuracy of the 5.3 in^3 volume specification for the inaccuracy of the clay volume, yielded a ratio of 3.94. The nominal value of 4.0 was chosen. It is considered to be correct within ± 2 percent.

The test results showed good consistency of data, especially at high inlet qualities. The Willans' lines were very nearly straight for ninety-nine and fifty percent qualities (Figure 11). For the twenty-five percent quality a curving up of the Willans' line at high speeds was noticed. This was consistent with previous observations and attributed to dynamic losses associated with accelerating the water.

Figure 13 shows that for high quality runs we have yet to reach the speed where efficiency begins to turn down. It is therefore realistic to expect higher efficiencies at higher speeds. Using the similarity method of the previous section, a maximum efficiency for 10,500 rpms is predicted to be fifty percent at fifty percent quality and at a pressure ratio of 4.66.

One of the most striking features of the 4.0 expansion ratio data is that the leakage rate is the same as for the 5.3 port. One of the characteristics of the larger inlet port is an increase in area through which leakage passes. The major portion of the increased area is due to an increased length of the circumferential seal line. It has been presumed that the leakage flow rate is a function of Grashoff's relation, since a pressure differential of greater than the 0.57 required for choking is present. Thus an increase in leakage would be expected in direct proportion to the increased leakage area.

Figure 15 shows leakage rates for various qualities for both ports. They are almost identical.

There are five logical explanations for this phenomenon. Leakage area is constant; Grashoff's relation doesn't apply; circumferential leakage is insignificant; the machine was altered during changing of the ports; or some change in external parameters has occurred. Closer analysis has revealed no solid justification for any of these reasons.

The length of the circumferential seal line increases with an increased inlet opening angle. This accounts for a major part of the difference in leakage area between the ports. The leakage area is 1.17 times as great for the 4.0 inlet port. There is no evidence that choking is occurring at any other point.

To check whether Grashoff's relation applied, the assumption was made that all the leakage flow went into the second chamber and remained there. Even in this unrealistic situation, the pressure in the second chamber would only be 75 psia; yielding a pressure ratio between the first and the second chambers of 0.63. Although this is not enough for choking, the mass flow rate would be very near that of choked flow, since the mass flow rate versus pressure curve approaches choking parabolically.

The third explanation, that the circumferential seal line is insignificant, seems unlikely since the circumference accounts for about a third of the leakage area. There are however, three factors tending to seal the circumference; these are dynamic pressure created by the rotor tip velocity, differential thermal expansion between the rotor and the casing and water sealing. Analysis reveals these to all be insignificant. Although the rotor lobes move at a velocity

in excess of 200 ft./sec., their dynamic contribution to flow resistance is only about one psi. Thermal expansion is small, also. If the temperature difference between the rotor and the casing were 100°F greater with the 4.0 port than with the 5.3 port, the total leakage area would only be reduced by a twelfth. This unlikely extreme situation would still not be sufficient to account for the observed phenomenon.

The last case, water sealing, is unlikely since at 99 percent quality the volume of water is one twenty-thousandth of that of steam.

While it is possible that the machine was changed during reassembly, the only likely change would be a slight alteration of the spacing between the rotors when the timing was reset, however the possible area change would be very small in relation to the total, about 0.012 in² maximum.

The constant leakage rate however, while unexplained, simplifies extending the previous empirical model of the engine to include expansion ratio. As noted in previous experiments by Karl Brown², power, at high pressure ratios, was independent of quality and flow rate was independent of pressure ratio. With these three observations and the data collected for constant quality at different pressure ratios along with the data for constant pressure ratio for different quantities, the flow rate and output power for any set of operating conditions may be predicted and the efficiency may be calculated.

In this manner the empirical model of the Lysholm¹ was

extended to include volume ratio. The efficiency, power output, flow rate and exhaust conditions can be predicted for any quality, speed and pressure ratio. It has been presumed that this model, derived from 4.0 and 5.3 volumetric expansion ratio data, may be used fairly accurately at volumetric expansion ratios between 5.3 and 4.0. Care should be taken in extrapolating with the model, since it is based on only two different volume ratios.

By decreasing the volumetric expansion ratio from 5.3 to 4.0, the leakage fraction was decreased by twelve percent and the efficiency was increased by about nine percent. This is very nearly what was predicted using the empirical model¹ and the similarity relationship outlined in the theory section of this paper.

The maximum efficiency for the 4.0 port occurs at a pressure ratio of 4.7. This yields desirable interstage pressures if the Lysholm is to be run at maximum efficiency as the first stage of a staged operation with the University's two Westinghouse turbines.

Analysis of the data yielded the surprising result that the leakage rates for the 5.3 port and the 4.0 port were identical. Also evident was that higher efficiencies may be expected at higher speeds. For a speed of 10,500 rpm, a maximum efficiency of 50 percent was predicted to occur at 50 percent quality and a pressure ratio of 4.7. Still higher efficiencies will probably occur at higher speeds.

For future operation of the engine, it would be desirable to install a gear box to eliminate the speed restriction of 9000 rpms imposed by the flexible belt. Further improvements would be a microprocessor for engine control and a vacuum line to the separator drain to permit higher mass flow rates.

Casing inlet (illuminated at back)

Exhaust

FIGURE 2a. LYSHOLM ENGINE GEOMETRY

4.0 (top) and 5.3 volumetric expansion ratio inlet ports.

4.0 port shown with rotors

FIGURE 2b. lysholm engine geometry

Rotors

Rotors

FIGURE 2c.

MESH LINE LEAKAGE PATH

Paper triangle shows location of the ROTOR-CASING INTERSECTION LEAKAGE PATH

FIGURE 2d.

FIGURE 3. END VIEW OF 4.0 INLET PORT AND ROTORS

FULL SCALE

FIGURE 4. Seal fluids Drawing by Dr. David Pankow

FIGURE 4. Seal fluids Drawing by Dr. David Pankow

Figure 5

CALCULATOR PROGRAM FOR ADJUSTING
FLOW RATIO CONTROLLER

For data taken with 1.375 in. diameter square edged orifice
plate for steam and 0.8 in. diameter square edge orifice
plate for water. (Calibration constant number four.)

Channel 10 = steam flow

Channel 11 = water flow

using the calibration curves, Figures 16 and 17.

$$\frac{[(\text{ch. } 10)(0.1) - 0.461] (\text{flow ratio}) - .0756}{0.875} = \text{Ch. } 11$$

Program for HP21C =

LBL 1

.1

X

RCL 1

-

RCL 3

-

RCL 4

÷

RTN

To run:

STO 1 = 0.0461

STO 2 = Flow Ratio

STO 3 = 0.0756

STO 4 = 0.875

Enter Ch. 10

Check: 6.5787 ==> .749

@ Flow Ratio = 1

FIGURE 6.

FIGURE 7.

FIGURE 9.

FIGURE 10.

FIGURE 11. WILLANS' LINES

FIGURE 12. WILLANS' LINES

FIGURE 13.

FIGURE 14.

FIGURE 15.

EFFICIENCY VS. PRESSURE RATIO

FIGURE 18.

PRESSURE RATIO

EFFICIENCY VS. PRESSURE RATIO

FIGURE 19.

CODES FOR DATA MATRIX POINT LOCATION

i = Flow ratio, water to steam

0 = 99% quality
1 = 50% quality
2 = 33% quality
3 = 25% quality
4 = 16.7% quality

j = Speed

0 = 4500 RPM
1 = 6000
2 = 7500
3 = 9000

k = Pressure Ratio

10 = 9.20
11 = 8.13
12 = 5.0
13 = 4.1
14 = 3.2

Data locations are not exact, data in a given matrix position may vary by 10% from its assigned value. There are some points at other locations in the matrix.

Specification of a zero in the "k" location will yield an error.

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS [PSIA]	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/ SEC]	INLET ENTHL [BTU/ LBM]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/ SEC]	INLET QUAL	EXIT QUAL
510	4/ 6/83	10	120.1	23.8	5.05	4492.	0.522	1186.1	32.4	21.6	0.00	2.37	2.64	0.309	0.995	0.987
508	4/ 6/83	10	119.6	23.6	5.06	6004.	0.581	1186.1	37.9	28.1	0.00	3.35	2.49	0.297	0.995	0.980
504	4/ 1/83	10	119.6	23.6	5.06	7515.	0.637	1184.6	39.8	32.4	0.00	4.39	2.38	0.281	0.994	0.976
505	4/ 1/83	10	120.1	23.9	5.02	8998.	0.704	1184.1	42.4	37.9	0.00	5.51	2.27	0.275	0.993	0.972
512	4/ 8/83	10	121.2	23.5	5.15	4499.	0.922	740.0	35.3	21.9	1.03	3.79	2.27	0.482	0.486	0.537
513	4/ 8/83	10	121.2	24.1	5.03	6007.	1.046	732.7	41.2	28.2	1.08	5.41	2.16	0.449	0.478	0.525
514	4/ 8/83	10	118.5	23.6	5.01	7500.	1.127	733.5	44.0	32.4	1.07	7.09	2.06	0.401	0.480	0.525
515	4/ 8/83	10	119.6	23.9	5.00	9018.	1.232	741.1	47.2	38.6	1.03	8.83	1.97	0.366	0.488	0.530
516	4/11/83	10	120.5	23.6	5.10	4503.	1.257	603.4	35.7	21.6	2.00	4.92	2.07	0.617	0.331	0.400
517	4/11/83	10	120.5	23.5	5.13	6000.	1.393	605.0	40.7	27.6	1.98	6.89	1.96	0.545	0.333	0.399
518	4/11/83	10	121.2	24.0	5.06	7493.	1.596	591.7	44.3	32.7	2.11	9.30	1.86	0.480	0.317	0.384
519	4/11/83	10	121.2	23.8	5.10	9000.	1.661	606.5	46.4	37.4	1.96	11.24	1.81	0.387	0.334	0.398
538	4/22/83	10	119.6	23.8	5.03	4513.	1.529	534.7	38.1	22.3	2.95	5.86	1.95	0.700	0.253	0.330
521	4/11/83	10	120.5	24.0	5.03	5999.	1.651	539.7	44.8	28.9	2.85	8.02	1.88	0.563	0.258	0.332
522	4/11/83	10	120.1	23.6	5.08	7501.	1.903	532.1	47.0	34.2	2.98	10.89	1.75	0.503	0.250	0.325
539	4/22/83	10	119.6	23.8	5.03	9009.	2.127	531.6	44.2	35.7	2.99	13.85	1.66	0.450	0.250	0.325

TABLE 2.

RUN#	DATE	COMMENT	INLET PRESS [PSIA]	EXH PRESS	PRESS RATIO	SPEED [RPM]	FLOW RATE [LBM/ SEC]	INLET ENTHL [BTU/ LBM]	EFF	POWER [KW]	FLOW RATIO	SPEC SPEED	SPEC SIZE	LEAK RATE [LBM/ SEC]	INLET QUAL	EXIT QUAL
532	4/13/83	10	121.2	13.2	9.17	4518.	0.552	1182.7	30.7	28.5	0.00	2.58	2.12	0.334	0.991	0.985
510	4/ 6/83	10	120.1	23.8	5.05	4492.	0.522	1186.1	32.4	21.6	0.00	2.37	2.64	0.309	0.995	0.987
533	4/13/83	10	119.6	13.0	9.17	5997.	0.602	1182.4	35.1	35.5	0.00	3.60	2.02	0.316	0.991	0.978
543	5/ 2/83	10	119.6	14.4	8.30	6010.	0.590	1183.0	35.1	33.5	0.00	3.52	2.11	0.304	0.992	0.978
508	4/ 6/83	10	119.6	23.6	5.06	6004.	0.581	1186.1	37.9	28.1	0.00	3.35	2.49	0.297	0.995	0.980
544	5/ 2/83	10	119.6	29.2	4.10	6003.	0.573	1183.7	37.7	24.3	0.00	3.33	2.67	0.288	0.992	0.979
545	5/ 2/83	10	119.1	37.5	3.18	6000.	0.575	1183.5	36.6	19.6	0.00	3.43	2.84	0.290	0.992	0.981
546	5/ 2/83	10	119.6	14.4	8.30	7503.	0.641	1183.5	39.1	40.6	0.00	4.58	2.02	0.285	0.992	0.973
504	4/ 1/83	10	119.6	23.6	5.06	7515.	0.637	1184.6	39.8	32.4	0.00	4.39	2.38	0.281	0.994	0.976
547	5/ 2/83	10	119.6	29.2	4.10	7520.	0.641	1183.3	40.8	29.4	0.00	4.42	2.52	0.284	0.992	0.975
548	5/ 2/83	10	119.6	36.6	3.26	7501.	0.627	1183.3	38.4	23.0	0.00	4.45	2.71	0.270	0.992	0.979
535	4/13/83	10	120.1	13.0	9.21	9011.	0.703	1182.4	39.1	46.3	0.00	5.83	1.87	0.272	0.991	0.971
549	5/ 2/83	10	119.6	14.7	8.13	9021.	0.693	1183.3	41.5	46.1	0.00	5.71	1.96	0.264	0.992	0.969
505	4/ 1/83	10	120.1	23.9	5.02	8998.	0.704	1184.1	42.4	37.9	0.00	5.51	2.27	0.275	0.993	0.972
550	5/ 2/83	10	119.6	28.8	4.15	8996.	0.697	1182.7	42.4	33.4	0.00	5.51	2.41	0.269	0.991	0.972
551	5/ 2/83	10	119.6	37.3	3.21	9001.	0.683	1182.8	39.3	25.2	0.00	5.59	2.61	0.255	0.991	0.978

TABLE 3.

LYSHOLM ENGINE DATA

Theoretical Inlet Volume	.00798 ft ³ /rev for 5.3 ratio port
	.0106 ft ³ /rev for 4.0 ratio port
Maximum Inlet Pressure	250 psia
Maximum Inlet Temperature	500°F
Maximum Recommended Male Rotor Shaft Speed	9000 rpm
Minimum Exhaust Pressure	4 psia
Rotor Length	6.519 in
Rotor Diameter	5.118 in
Casing Bore	5.128 in
Center Line Distance Between Rotors	4.150 in

	MALE	FEMALE
Pitch Diameter	3.320 in	4.980 in
Minor Diameter	3.210	3.190
Lobe Wrap Angle	321°	214°
Seal Region Shaft Dia.	2.900	2.900
Bearing Shaft Dia.	2.700	2.700
End Clearance	.004-.006	.004-.006
Seal Line	15.75 in	11.57 in

Timing 4.0 Inlet Port:

Intake Angle	133°	89°
Expansion	278°	185°
Free Exhaust	45°	30°
Forced Exhaust	405°	270°

Timing 5.3 Inlet Port:

Intake Angle	111°	74°
Expansion	300°	200°
Free Exhaust	45°	30°
Forced Exhaust	405°	270°

CALIBRATION AND COMMENT CODES FOR DATA REDUCTION
PROGRAMS

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)
- 10 - Michael Dunbar's Data (1983)

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)
- 4 - Large Water Orifice Plate, 0.8 in. dia.; Large Steam Orifice plate, 1.375 in. dia. (1983)

- 1) Brown, K.; Empirical Modeling of a Lysholm Helical Screw Expander; Masters project; June 1982.
- 2) Berger, R.E.; Performance Characteristics of the Lysholm Engine; Masters project; 1981.
- 3) Fox, R.W. and McDonald, A.T.; Introduction to Fluid Mechanics; second edition; John Wiley and Sons; 1978.
- 4) Lysholm, A.; The Fundamentals of a New Screw Engine; ASME Paper Number 66-GT-86; 1966.
- 5) McKay, R. and Sprankle, R.; "Helical Rotary Screw Expander Power System"; Proc. Conf. Res. Devel. Geothermal Resources, Pasadena, 1974; (Jet Propulsion Laboratory, Pasadena); 1974.
- 6) Reynolds, W. and Perkins, H.; Engineering Thermodynamics; McGraw Hill, Inc.; 1977.
- 7) Steidel, R.; Pankow, D. and Berger, R.; "Performance Characteristics of the Lysholm Engine as Tested for Geothermal Power Applications"; Proceedings of the 16th Intersociety Energy Conversion Conference; ASME Paper No. 819594, 1981, P. 1334-1340.
- 8) Steidel, R.; Weiss, H. and Flower, J.; Performance Characteristics of the Lysholm Engine As Tested For Geothermal Power Applications in the Imperial Valley; Lawrence Livermore Laboratory Report UCRL - 80151, 1977.
- 9) Weiss, H.; Steidel, R.; and Lundberg, A.; Performance Test of a Lysholm Engine; Lawrence Livermore Laboratory Report UCRL - 51861, 1975.

APPENDIX

CALCULATION OF MAXIMUM POSSIBLE SECOND
CHAMBER PRESSURE

52

Assume all leakage passes into the second chamber and
stays there

$$\text{Volume of steam @ 120 psia} = 13.8 \text{ in}^3 \frac{5.3}{4.0} = 6.9 \text{ in}^3$$

(theoretical + leakage)

$$= 25.2 \text{ in}^3$$

Area of second port

at closing of first port rotation angle of second
port is 222° and 148° for the male and female rotors
respectively.

$$\frac{222^\circ - 111.3^\circ}{321^\circ} \text{ inlet angle } (6.519 \text{ in})(1.626 \text{ in}^2) = 3.66 \text{ in}^3$$

wrap angle

$$\frac{148^\circ - 74^\circ}{214^\circ} (6.519 \text{ in})(1.303 \text{ in}^2) = 2.94 \text{ in}^3$$

$$\text{Total volume} = 4(3.66 + 2.94 \text{ in}^3) + 13.8 = 40.17 \text{ in}^3$$

$$\text{Pressure} = \frac{25.2 \text{ in}^3}{40.17 \text{ in}^3} 120 \text{ PSIA} = 75.3 \text{ psia}$$

CALCULATION OF MINIMUM STATIC PRESSURE RATIO

$$P_o = P_i + \frac{1}{2} \rho V^2$$

$$P_o = 75.28 + \frac{1}{2} \frac{\overset{\text{MAX. TIP VEL.}}{(201 \text{ ft/sec})} (.27 \text{ lb./ft.}^3)}{(32.2 \text{ ft./s.}^2) (144 \text{ IN}^2/\text{ft.}^2)}$$

$$= 76.46 \text{ psia.}$$

$$\text{MINIMUM RATIO} = \frac{76.46}{120.0} = .637$$

CALCULATE PRESSURE DROP ACROSS INLET PORT

$$\frac{V^2}{2Jg} = \Delta h_s$$

ASSUME: ISOTROPIC

$$V = \sqrt{2Jg \Delta h_s}$$

$$\Delta h_s = \left(\frac{200 \text{ ft/s.} - 30 \text{ ft/s}}{224} \right) = .576 \text{ BTU}$$

From h - s plot .57 BTU corresponds to .46 psi pressure drop

REQUIRED CHANGE IN PRESSURE DROP ACROSS INLET PORT TO
ACCOUNT FOR CONSTANT LEAKAGE

Total flow @ 9000 rpm 99% quality = .704 lb/s

actual leakage = .275 lb/s

expected leakage = .275 lb/s (ratio of leakage areas) = .275 (1.17)
= .322 lb/s

$$P_{\text{inlet}} = \frac{.704 - \text{Leakage}}{\text{Theoretical M}} (120 \text{ psia})$$

$$\text{Leakage} = .322 \text{ lb/s} \frac{P_{\text{inlet}}}{120 \text{ psia}}$$

hence,

$$P_{\text{inlet}} = 120 \cdot \frac{[.704 - (.322 \text{ lb/s}) (P_{\text{inlet}}) / 120 \text{ psia}]}{.429 \text{ lb/s}}$$

$$P_{\text{inlet}} = 112.5 \text{ psia}$$

a drop of 7.5 psia
across the inlet

CALCULATION OF VOLUMETRIC EXPANSION RATIO

Inlet volume measured by packing clay into chamber = 19.77 in^3

Inlet volume calculated by increase in volume over 5.3
port volume.

$$\text{Vol. 5.3} = 13.8 \text{ in}^3$$

$$\begin{aligned} \text{inlet angle 5.3} &= 111.3^\circ \text{ male} \\ &= 74^\circ \text{ female} \\ &= 132^\circ \end{aligned}$$

$$\begin{aligned} \text{inlet angle 4.0} &= 132^\circ \text{ male} \\ &= 88^\circ \text{ female} \end{aligned}$$

Area of section between rotor lobes was measured by cutting
out the shape from graph paper and weighing it.

$$\begin{aligned} \text{Perpendicular to rotor axis area} &= 1.626 \text{ in}^2 \text{ male} \\ &= 1.303 \text{ in}^2 \text{ female} \end{aligned}$$

$$\text{Change in area} = \frac{(\text{Change in inlet angle})}{(\text{Total wrap angle})}$$

$$\bullet (\text{Perpendicular Area}) (\text{Rotor length})$$

$$\begin{aligned}\text{change in area, male rotor} &= \frac{132^\circ - 111.3^\circ}{321^\circ} (6.52)(1.626) 55 \\ &= .6835 \text{ in}^3\end{aligned}$$

$$\begin{aligned}\text{change in area, female rotor} &= \frac{88^\circ - 74^\circ}{214^\circ} (6.519 \text{ in})(1.303 \text{ in}^2) \\ &= 0.5557 \text{ in}^3\end{aligned}$$

$$\begin{aligned}\text{total inlet volume} &= 4(.5557 + .6835 \text{ in}^3) + 13.8 \text{ in}^3 \\ &= 18.76 \text{ in}^3\end{aligned}$$

$$\begin{aligned}\text{exhaust volume} &= 4(\text{Area})(\text{Length}) \\ &= 4(1.626 + 1.303)(6.519) \\ &= 76.38 \text{ in}^2\end{aligned}$$

$$\text{Volume ratio} = \frac{76.38}{18.76} = 4.07$$

Or

$$\begin{aligned}\text{inlet volume} &= \frac{\text{exhaust}}{(\text{old vol. ratio})} + \text{vol. change} \\ &= 76.38 / 5.3 + 4.956 = 19.36\end{aligned}$$

$$\text{Volume Ratio} = \frac{76.38}{19.36} = 3.94$$

The two methods of calculating volumetric expansion ratio yeild 4.07 and 3.94. Take 4.0 as Volumetric expansion ratio.

CALCULATION OF MINIMUM OBTAINABLE EXHAUST PRESSURE

Employing a smaller volumetric expansion ratio results in an increased mass flow rate. The following calculation estimates the minimum obtainable exhaust pressure for a 3.5 and a 4.0 volumetric expansion ratio when operating at 9000 rpm and 33 percent quality.

The ratios of the friction factors within the exhaust piping were calculated for different mass flow rates and qualities. It was found that the pressure drop for 37% quality fluid (the exhaust quality expected from 33% quality inlet) would be two times greater than that for 25% quality fluid and 1.31 times greater than that for 33.7% quality fluid.

The pressure drops for 25 and 33.7% quality fluid were known. Assuming a minimum condenser pressure of 8 psia, a minimum exhaust pressure of 26.9 psia was calculated for a 3.5 expansion ratio and 20.4 psia for a 4.0 expansion ratio.

Exhaust piping pressure drop calculation:

Vol. RATIO	MASS FLOW RATE	EXHAUST CONDITIONS	
3.0	2.17 lb/s	PR = 5.0	PR = 6.0
3.5	1.93 lb/s	$\nu = 6.47 \text{ ft}^2/\text{lb}$	$\nu = 7.44 \text{ ft}^2/\text{lb}$
4.0	1.75 lb/s	$x = .368$	$x = .37$
		$T = 236.9^\circ\text{F}$	$T = 228^\circ\text{F}$

$$\mu = .4772 \times 10^{-5} + 1.492 \times 10^{-8} T$$

$$\text{Flow rate of Liquid} = (1 - x) \dot{M}$$

$$\text{Flow rate of vapor} = (x) \dot{M}$$

$$Re_{\text{vapor}} = \frac{(\text{Vel}_{\text{vapor}}) (D_{\text{Pipe}})}{\nu \mu (12)} = \frac{2 \dot{M} x}{\pi r \mu}$$

$$\text{Roughness} = .0018/\text{ft}$$

$$\text{PRESSURE DROP FOR VAPOR} = \frac{(\text{FRICTION FACTOR}) (\text{FLUID VEL})^2}{(\text{PIPE DIA}) 2 g_c \nu}$$

CALCULATION OF PRESSURE DROPS

EXHAUST QUALITY = .37 VAPOR

$$Re_{\text{vapor}} = 3.45 \times 10^5$$

$$\text{Relative roughness} = 4.5 \times 10^{-4}$$

FRICTION FACTOR CAN BE
LOOKED UP FOR THESE CONDITIONS

$$FF = .0180$$

PRESSURE DROP DUE TO
VAPOR

$$P.D._v = 8.92 \times 10^{-3} \text{ psi/ft}$$

EXHAUST QUALITY = .255

$$Re_{\text{vapor}} = 2.08 \times 10^5$$

$$\text{Relative Roughness} = 4.5 \times 10^{-4}$$

$$FF = .0185$$

$$P.D._v = 3.09 \times 10^{-3} \text{ psi/ft}$$

LIQUID

$$Re_{\text{liquid}} = 2.7 \times 10^4$$

$$F = .025$$

The pressure drop can be shown to be:

$$PD_L = 2.74 \times 10^{-3} \text{ psi/ft}$$

$$Re_{\text{liquid}} = 3.4 \times 10^4$$

$$F = .024$$

$$PD_L = 1.17 \times 10^{-3} \text{ psi/ft}$$

Total Two Phase Pressure Drop

$$\Phi v^2 = 3.29$$

$$\Phi v^2 = 3.36$$

Total pressure drop

can be shown to

be 2.94×10^{-2} psi/ft

for .37 quality fluid — and —

for .255 quality fluid

1.48×10^{-2} psi/ft

The ratio of the pressure drop = $2.94/1.48 \approx 2$

For a volumetric expansion ratio of 3.5 the exhaust quality is expected to be .37 and the mass flow rate would be 1.93 lb/s. The exhaust piping pressure drop can be approximated from existing data.

$$\text{RUN \# 416} - \dot{M} = 1.84 \text{ lb/s} \quad P_{\text{exhaust}} = 16.9 \text{ psia} \\ x = .255$$

$$P_{\text{exhaust } 3.5} = 16.9 - 8 \text{ psia} \left(\frac{1.93}{1.814} \right)^2 + 8 \text{ psia}$$

$$= 26.9 \text{ psia} \quad \text{MIN. Pressure exhaust @ } \begin{matrix} 33\% \text{ quality} \\ 9000 \text{ rpm} \\ 3.5 \text{ V.R.} \end{matrix}$$

SIMILARLY FOR EXHAUST AT:

$$x = .337 \quad T = 220^\circ \text{F}$$

$$P = 17.5 \text{ psia}$$

$$\text{Velocity Vapor} = 160 \text{ ft/sec}$$

$$\text{Velocity Liquid} = 0.224 \text{ ft/sec}$$

$$Re_{\text{vapor}} = 2.85 \times 10^5$$

$$Re_{\text{liquid}} = 2.46 \times 10^4$$

$$f_{\text{vapor}} = .018 \quad (\text{from Moody Diagram})$$

$$f_{\text{liquid}} = .026$$

$$\text{Pressure drop(vapor)} = 6.46 \times 10^{-3} \text{ PSI/FT}$$

$$\text{Pressure drop(Liq.)} = 2.52 \times 10^{-3} \text{ PSI/FT}$$

Pressure drop for Mixture:

as before

$$\left[4.065 \left(\frac{\Delta P_{\text{liq}}}{\Delta P_{\text{vapor}}} \right)^{1/2} (6692) + 1.22 \right]^2 \Delta P_{\text{vapor}} = 2.24 \times 10^{-2} \text{ PSI/FT}$$

RATIO OF PRESSURE DROPS FOR .37 and .337
EXHAUST QUALITY:

$$\frac{\Delta P_{.37}}{\Delta P_{.337}} = 2.94 / 2.24 = 1.31$$

FOR AN EXPANSION RATIO OF 4.0 EXHAUST
QUALITY AND MASS FLOW RATE ARE EXPECTED
TO BE:
 $x = .37$ and $\dot{m} = 1.75 \text{ lb/s}$

FROM RUN # 303 $P_{\text{exhaust}} = 17.5 \text{ PSIA}$ $\dot{m} = 1.758 \text{ lb/s}$
 $x = .337$

$$P_{4.0} = (17.5 - 8) (1.75 / 1.758) 1.31 + 8 \text{ psia}$$

= 20.4 psia MIN. exhaust pressure at 33% quality
inlet
9000 rpm
VR = 4.0

LEAKAGE AREAS

60

BLOW HOLE :

MESH LINE :

TOTAL AREA = $.0218 \text{ in}^2$

LEAKAGE AROUND FRONT FACE :

61

$$2.4 \text{ in} \times .005 \text{ in} = .012 \text{ in}^2$$

CIRCUMFERENTIAL SEAL LINE :

$$\text{Seal line length} = (\text{Total lobe length}) \frac{(\text{Inlet opening angle})}{(\text{Wrap angle})}$$

$$\text{Male Rotor length} = 15.62 \text{ in} \frac{(\text{Inlet angle})}{(318^\circ)}$$

$$\text{Female Rotor length} = 11.48 \text{ in} \frac{(\text{Inlet angle})}{(214^\circ)}$$

$$\text{AREA} = (\text{Male} + \text{Female rotor length}) \times .005 \text{ in}$$

TOTAL LEAKAGE AREA FOR 4.0 PORT :

$$\text{Total leakage} = .0361 + .012 + .0218(N) + .005(\text{Male} + \text{Female Length})$$

ANGLE	0°	10°	20°	30°	40°	47.5°	50°	60°	70°	80°	90°
MALE SEAL LENGTH	1.28	1.60	1.93	2.25	2.53	2.83	2.91	3.24	3.57	3.90	4.22
FEMALE SEAL LENGTH	.77	1.31	1.85	2.39	2.93	3.34	3.48	4.02	4.56	5.10	5.64
NO. PORTS OPEN	1	1	1	1	1	1 (2)	2	2	2	2	2
TOTAL AREA [in ²]	.0802	.0845	.0889	.0931	.0975	.101 (.123)	.124	.128	.132	.137	.141

$$\text{Ave Total area} = .1102 \text{ in}^2$$

TOTAL LEAKAGE AREA FOR 5.3 PORT :

ANGLE	0°	6.8°	7°	20°		40°		60°	68.7°	80°	90°
MALE SEAL LENGTH	0	0	.01	.71		1.80		2.88	3.35	3.96	4.50
FEMALE SEAL LENGTH	0	0	.68	1.11		1.76		2.42	2.70	3.07	3.40
NO. PORTS OPEN	1	1	1	1		1		1	1 (2)	2	2
TOTAL AREA [IN ²]	.070	.070	.074	.079		.083	.0	.096	.10 (.122)	.127	.131

Ave. Circumferential seal area = .0189 in²

Ave. Mesh line area = .0269 in²

Ave. Total area = .09406 in²

FIGURE 20.

□ 4.0 INLET PORT
◇ 5.3 INLET PORT

FIGURE 21.

□ 4.0 INLET PORT
◇ 5.3 INLET PORT

FIGURE 23.

□ 4.0 INLET PORT
◇ 5.3 INLET PORT

FIGURE 22.

□ 4.0 INLET PORT
◇ 5.3 INLET PORT

FIGURE 24.

× 4.0 INLET PORT
+ 5.3 INLET PORT

FIGURE 25.

× 4.0 INLET PORT
+ 5.3 INLET PORT

FIGURE 26.

X 4.0 INLET PORT
+ 5.3 INLET PORT

FIGURE 27.

◇ THEORETICAL FOR 4.0 PORT
X ACTUAL FOR 4.0 PORT

FIGURE 28.

◇ THEORETICAL FOR 4.0 PORT
× ACTUAL FOR 4.0 PORT

FIGURE 29.

◇ THEORETICAL FOR 4.0 PORT
X ACTUAL FOR 4.0 PORT

FIGURE 30.

X 4.0 INLET PORT
+ 5.3 INLET PORT

POWER VS. LBM./KW-HR., 4.0 VOL. RATIO

FIGURE 31.

POWER VS. LBM./KW.-HR., 5.3 VOL. RATIO

FIGURE 32.

FIGURE 31.

FIGURE 33

FILE LYST2.FTN

CONTENTS: MAIN PROGRAM LYST6
 (REQUIRES SUBROUTINE STMTBL)
 (REQUIRES BLOCK DATA CDATA)
 (REQUIRES DATA FILE LYST)

BLOCK DATA CDATA

THIS PROGRAM IS A MODIFICATION OF A PROGRAM WRITTEN
 BY KARL BROWN. IT MANIPULATES AND REDUCES DATA WITH
 A VOLUMETRIC EXPANSION RATIO OF 4.0 FROM THE FILE
 LYST.DAT
 THE OUTPUT FROM DATA REDUCTION IS IN A FORMAT SUITABLE
 FOR PLOTTING.
 OUTPUT FROM THIS FILE MAY BE FOUND IN FILES NAMED
 LOUT.DAT

PROGRAM MANIPULATES AND REDUCES LYSHOLM DATA
 COMMON /LOGIN/ TL, PL, HL, SL, VL, XL, SPRHT
 COMMON /CDATA/ SC(4), BC(4), SST(4), BST(4), SLW(4), BW(4)
 LOGICAL TL, PL, HL, SL, VL, XL, SPRHT
 LOGICAL CONT
 DIMENSION NRUN(585), MONTH(585), MDAY(585), MYEAR(585),
 1 NSCAN(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 = 'LOUT.DAT', TYPE = 'NEW',
 1 CARRIAGECONTROL = 'LIST', ACCESS = 'SEQUENTIAL')
 C PROMPT FOR TASK INPUT
 5 TYPE 10
 10 FORMAT (1H, 'CHOOSE OPTION:', 120, '1 - STORE DATA')
 TYPE 20
 20 FORMAT (1H, 120, '2 - RETRIEVE DATA')
 TYPE 30
 30 FORMAT (1H, 120, '3 - REDUCE DATA AND OUTPUT RESULTS')
 TYPE 35
 35 FORMAT (1H, 120, '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:', 130, '2 - OUTPUT FILE
 1 (EXITS PROGRAM)')

```

TYPE 215
215  FORMAT (1H, 130, '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    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 600
      TYPE *, 'THE MATRIX POSITION SPECIFIED ALREADY CONTAINS DATA.'
      TYPE *, 'WRITING IN NEW DATA WILL ERASE THE OLD DATA'
      TYPE *, 'CONTINUE?(Y OR N)'
      ACCEPT *, CONT
      IF(.NOT. CONT) GO TO 5
600  CONTINUE
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 MATRIX ELEMENTS
      TYPE *, 'ENTER I1ST, I1ST, J1ST, J1ST, K1ST, K1ST'
      ACCEPT *, I1ST, I1ST, J1ST, J1ST, K1ST, K1ST
      GO TO (5, 1050, 1020), IOPT
1020 CONTINUE
1050 CONTINUE
      DO 1955 I = I1ST, I1ST
      DO 1954 J = J1ST, J1ST
      DO 1953 K = K1ST, K1ST
      NREC = 95 * I + 19 * J + K
C    READ DATA FROM FILE
      READ (1, NREC) NRUN(NREC), MONTH(NREC), MDAY(NREC),

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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

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      TL = .TRUE.
      SL = .TRUE.
      CALL STMTBLCT3, 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      IMPLICIT 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 * 5.3 / (4.0 * V2)
      VS = 4.0 * V2
      VL = .TRUE.
      SL = .TRUE.
      CALL STMTBLCT5, 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 STMTBLCT3, P3, H3, S3, V3, X3, SH3)
C      WRITE RESULTS TO OUTPUT FILE
      WRITE (IOUT, 1920),
1          P2, P3S, PRAT, SPEED,
2          TFLOW, H2, EFF, POWER, FRAT, SS1, SD1, RLEAK, CH,
3          X2, X3
1920      FORMAT (1H, E5.1,
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, F4.2, 2X, F5.3, 2X, F5.3)
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
C

BLOCK DATA

BLOCK DATA

C

CALIBRATION CONSTANTS FOR FLOW METERS

COMMON /CADATA/ SC(4), BC(4), SST(4), BST(4), SLW(4), BW(4)

DATA SC/ 0.8664, 0.8664, 0.8664, 0.8664/

DATA BC/ 0.0707, 0.0707, 0.0707, 0.0707/

DATA SST/ 1.0, 1.0214, 1.0214, 1.0/

DATA BST/ 0.0, -0.0059, -0.0059, 0.0461/

DATA SLW/ 1.0, 0.8538, 0.1861, 0.8538/

DATA BW/ 0.0, 0.0756, -0.0122, 0.0756/

END

```

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
IMPLICIT 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 * 5.3 / ( 4.0 * V2 )
VS = 4.0 * 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,

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2 QUAL QUAL')
WRITE (IOUT, 1915)
1915 FORMAT (1H, 125, ' LPSIAB) CRPM3 LBM/ LBTU/
1 LKWD LBM/')
WRITE (IOUT, 1917)
1917 FORMAT (1H, 125, ' SECT LBM)
1 SECT')
WRITE (IOUT, *) ' '
1050 CONTINUE
DO 1955 I = I1ST, I1ST
DO 1954 J = J1ST, J1ST
DO 1953 K = K1ST, K1ST
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, 14, 2X, 12, '/', 12, '/', 12, 2X, 11, 2X, 11,
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 LYSBOLM 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.

```

```

215  FORMAT (1H, '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    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 600
      TYPE *, 'THE MATRIX POSITION SPECIFIED ALREADY CONTAINS DATA.'
      TYPE *, 'WRITING IN NEW DATA WILL ERASE THE OLD DATA'
      TYPE *, 'CONTINUE?(Y OR F)'
      ACCEPT *, CONT
      IF (.NOT. CONT) GO TO 5
600  CONTINUE
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 MATRIX ELEMENTS
      TYPE *, 'ENTER I1ST, I1ST, J1ST, J1ST, K1ST, K1ST'
      ACCEPT *, I1ST, I1ST, J1ST, J1ST, K1ST, K1ST
      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, '125, 'PRESS, PRESS RATIO, RATE
1      ENTHL, RATIO, SPEED, SIZE, RATE

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i      2      IFLOW, H2, EFF, POWER, FRAT, SSI, SDI, 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      BLOCK DATA
C
      BLOCK DATA
C      CALIBRATION CONSTANTS FOR FLOW METERS
      COMMON /CDATA/ SC(4), BC(4), SST(4), BSI(4), SLW(4), BW(4)
      DATA SC/ 0.8664, 0.8664, 0.8664, 0.8664/
      DATA BC/ 0.0707, 0.0707, 0.0707, 0.0707/
      DATA SST/ 1.0, 1.0214, 1.0214, 1.0/
      DATA BSI/ 0.0, -0.0059, -0.0059, 0.0461/
      DATA SLW/ 1.0, 0.8538, 0.1861, 0.8538/
      DATA BW/ 0.0, 0.0756, -0.0122, 0.0756/
      END

```