OPEN LOOP PNEUMATIC CONTROL
OF A LYSHOLM ENGINE OR TURBINE EXHAUST PRESSURE

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Submitted in partial satisfaction of the requirements for
the degree of

MASTER OF SCIENCE
in
MECHANICAL ENGINEERING
in the
GRADUATE DIVISION
of the
UNIVERSITY OF CALIFORNIA, BERKELEY

July 17, 1981

AT03-80ET27252

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Abstract

A Lysholm engine, or helical screw expander, is currently being evaluated at the University of California, Berkeley for staging with a conventional turbine in geothermal energy conversion. A pneumatic closed loop, proportional-integral control system was implemented to control the Lysholm engine's exhaust pressure for performance testing of the engine at constant inlet/outlet pressure ratios. The control system will also be used to control the exhaust pressure of the conventional turbine during future testing of the staged Lysholm-turbine system. Analytical modeling of the control system was performed and successful tuning was achieved by applying Ziegler-Nichol's tuning method. Stable control and quick response, of approximately 1 minute, was demonstrated for load and set point changes in desired exhaust pressures.

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I. Introduction

I.1 Description of the Lysholm Engine

Research is currently being conducted at the University of California, Berkeley (UCB) to test the performance characteristics of a Lysholm engine for geothermal energy conversion. The Lysholm engine or helical screw expander, was first patented in Sweden in 1934 by Alf Lysholm (Ref. 1). Since then, its primary use has been as a gas compressor, which is known for its reliability in severe environments. Use of the helical screw expander as an engine was noted by H.R. Nilsson in 1952 (Ref. 2) and Lysholm himself in an ASME paper in 1966 (Ref. 3). In 1973, R. Sprankle of the Hydrothermal Power Company, patented the use of the Lysholm engine for limited geothermal application (Ref. 4). Sprankle converted a Gardner-Denver 5 in. diameter, rotor air compressor to an engine. In 1974, he field tested the engine's geothermal power capability at the Cerro Prieto field in Mexico and at the East Mesa field in the Imperial Valley, California. Further testing of the Lysholm engine, with improved data gathering methods, was undertaken at the Lawrence Livermore National Laboratory in 1975 (Ref. 5) and 1977 (Ref. 6). The engine, a modified Gardner-Denver SC-71 air compressor, was then brought to UCB in 1979 for more extensive testing. The current objectives of the test program at UCB are: 1) Further performance testing of the Lysholm engine at slightly higher speeds, larger inlet/outlet pressure ratios and more accurately controlled steam qualities. 2) Performance evaluation of a hybrid, Lysholm-conventional steam turbine, geothermal power system.

The Lyshom engine's ability to accept two-phase flow of a water-steam mixture make it ideal for operation in geothermal energy conversion. Mechanically, the engine consists of two parallel intermeshing rotor shafts supported at either end by bearings which are fitted with floating carbon ring seals (Fig.1). The two rotors, a four-lobed male rotor and a six-lobed female rotor, are connected by timing gears and have no direct contact with each other. The engine takes in a fixed volume of fluid, obtains work as the fluid expands against its two rotors, and releases the fluid at a lower pressure. These cycle events overlap such that intake, expansion and exhaust are essentially continuous.

In field operation, the geothermal resource exists at a given pressure and quality. With minor throttling losses, the fluid enters the Lysholm at a slightly different pressure and quality. Closure at the inlet port isolates the charge from both inlet and exhaust conditions. The charge is then expanded to 5.3 times its original volume, with this expansion ratio fixed by the geometric configuration of the intake port. The inlet/outlet pressure ratios will differ somewhat from the volume ratios, because some fraction of the liquid phase flashed to vapor, as saturation temperature decreases with pressure. This expansion process can be modeled as a polytropic process with a variable exponent dependent on steam quality and other system operating parameters.

I.2 Operation of Test Equipment

The Lysholm engine and supporting equipment are shown in Fig. 2 with a schematic available in Fig. 3. Lubricating oil, seal water, and seal air are supplied to the engine, which is equipped with rotameters and pressure gauges to measure these quantities. Thermocouples measure engine inlet, exhaust, and oil temperatures, as well as the temperatures in other parts of the system.

A Foxboro fluid control system is used to vary qualities of water and steam to simulate a geothermal resource. Campus heating steam, at 340-346°F and 124-130 psia, is obtained from the UCB physical operations plant. The liquid phase is obtained by pressurizing building industrial water to about 160 psia and heating it to 336-342°F in an eight-pass, Thermxchanger 1208, heat exchanger. The fluid control system controls both the total flow rate and the quality by individually controlling the steam and water flows. Slightly upstream of the engine's intake port, the two phases are mixed, using a nozzle to accelerate the water to match the velocity of the steam. Since some water flashes to steam as the two phases are mixed, the quality changes. The process is rapid enough however, to be considered isenthalpic.

The Lysholm engine is connected to the brake shaft by a 3 to 1 reduction gear belt. Shaft torque is measured by a Lebow torque sensor, model 1605-2k, mounted on the brake shaft. A magnetic pickup and 60-tooth gear measure rotational speed. Brake and load control on the engine are provided by adjusting the power supply to a Telma CA135 eddy current dynamometer. Neither the Lebow torque sensor, magnetic pickup or eddy current brake have any wearing devices in contact with the engine's brake shaft.

Power for the brake is provided by a 50 amp., 20 volt Hewlett Packard, LVR series 6261B, power supply.

Most of the auxiliary equipment required to run the engine; such as the main condenser, are located in the sunken pump room, which is one level below the main operating equipment. Fig. 4 shows a sketch of the physical layout of the test and auxiliary equipment. During normal operation, the Lysholm engine exhausts to the main condenser, a Westinghouse two-pass, down-flow, shell and tube model. The exhaust is then condensed and removed by a condensate pump to maintain a slightly subatmospheric pressure in the engine's exhaust line.

Most of the engine's operating characteristics are recorded with a 40 channel, Autodata Nine printer. The Autodata records and prints engine rpm, engine torque, and steam and water flow rates. In addition, it prints the temperatures of several thermocouples located throughout the test and auxiliary equipment system. These temperatures are used to monitor the system during runs, to fix inlet and exhaust pressures and calculate enthalpies and efficiencies. The Autodata also records pressures, obtained from pressure transducers located throughout the system. The pressure signals must first be conditioned by carrier demodulators before entering the Autodata. Data were consistently recorded for the following conditions: inlet pressures of 75, 90, 105, and 120 psia, engine speeds of 4500, 6000, 7500 and 9000 rpm and water/steam ratios of 0/1, 2/1, 3/1 and 5/1 (Ref. 7).

When a desired data condition is reached, the engine's performance is allowed to stabilize and three data scans are taken of about 15 seconds each. These data values are then averaged

for data reduction. Fig. 5 shows a photograph of the Lysholm engine control panel and Autodata Nine printer.

Two modes of control are available to adjust the performance of the engine to reach a desired operating point. The first, with a positive effect on the engine's inlet pressure and speed, is flow rate adjustment. By setting the Foxboro flow controller into automatic mode, a fixed water-steam ratio will be maintained automatically as steam flow rate is adjusted. As flow rate is increased, both the inlet pressure and speed of the engine will increase. The other control mode, with an inverse effect on engine speed and, to a lesser extent, a positive effect on inlet pressure,^{is} the the adjustment of current to the power supply of the eddy current brake. As current is increased to the eddy current brake, its load on the engine's brake shaft increases, which serves to slow the engine's speed down. With constant mass flow rate maintained by the Foxboro controller, the inlet pressure to the engine increases slightly. Since all three variables; flow rate, inlet pressure and engine speed are dynamically coupled to each other, iterative adjustments of both the flow rate and supplied brake current are necessary to attain a desired operating point. Engine performance is stable in response to set point changes of either mode, with engine speed exhibiting the longest response time.

I.3 Need for Control of Exhaust Pressure

Performance testing of the Lysholm engine was conducted over a range of qualities (17-100% steam), at fixed inlet pressures (75-120 psia), and at various speeds (4500-9000rpm). The results of these tests are plotted as Willan's lines, plots of mass flow rate verses power output for fixed inlet/outlet pressure ratios and inlet enthalpy. Normally a Willian's line is essentially straight for a turbine, but at the highest inlet pressures and largest flow rates a pronounced hook was observed for the Lysholm data. This is caused by an increase in back pressure in an inadequately sized exhaust line and is a characteristic of our test system and not the Lysholm.

As shown in Fig. 3 the Lysholm exhausts to the main condenser which maintains an atmospheric or slightly sub-atmospheric pressure in the exhaust line. If the pressure in the Lysholm exhaust port, just as it is uncovered, is higher than that of the exhaust line the charge will immediately rush into the exhaust line. This would best be described as an isenthalpic or throttling process, producing little useful shaft work. On the other hand, if the chamber pressure is lower than the exhaust line pressure, shaft work is actually consumed to forceably exit the charge. Therefore, as back pressure is increased in the line, engine efficiency will decrease. The inlet/outlet pressure ratio is very sensitive to changes in the exhaust pressure, as exhaust pressure becomes a rather small divisor of the ratio when compared with inlet pressure.

The back pressure builds up in the exhaust line due to the presence of noncondensable gases which cannot be condensed or

removed by the main condenser. These noncondensable gases are introduced into the engine's operating fluid and into the exhaust line, by the building water supply, which is used to simulate different qualities of steam. At the highest mass flow rates and lowest qualities, the main condenser is no longer able to sufficiently remove the increasing amount of noncondensable gases. This causes a back pressure to build up in the exhaust line which drives the engine's exhaust pressure up, thereby decreasing the inlet/outlet pressure ratio across the engine.

The air ejector, consisting of one single-stage jet air ejector mounted on an after condenser, serves to further remove noncondensable gases from the main condenser, which decreases the back pressure build-up in the exhaust line.

The air ejector, shown in Fig. 6, consists of a steam chest (1), a nozzle(2), a mixing chamber(4), and a diffuser(6). Steam for a 126 psia building supply line enters the ejector steam chest(1) and acquires a high velocity as it passes through the nozzle(2). This high-velocity steam jet entrains air and noncondensable gases from the main condenser into the mixing chamber. This mixture is then discharged into the diffuser where nearly all the kinetic energy of the mixture is converted into internal energy. The stream of steam and air enters the after condenser(10) at atmospheric pressure and low velocity. In the after condenser, the noncondensable gases are vented to the atmosphere while the ejector motive steam and any steam or condensable vapors brought from the main condenser, are condensed and drain out as condensate.

The ejector is sensitive to variations in steam pressure. A 3/4" hand valve on the ejector steam inlet allows manual throttling control of the steam pressure, determining the ejector's effectiveness in removing non-condensable gases from the main condenser.

The ejector will serve two main functions in obtaining Lysholm engine test data. The first purpose will be to maintain constant inlet/outlet pressure ratios for data correlation in drawing Willan's lines at high mass flow rates and low qualities. The second function will be to give control over inlet/outlet pressure ratios so engine efficiency and power output can be investigated at various expansion ratios. These data will enable us to find the optimum expansion ratio for maximum efficiency and then, for maximum power output. The ratios will not necessarily be the same. Preliminary data indicate that the optimum engine efficiency occurs at a pressure ratio between 8 and 9. The discrepancy between this optimum pressure ratio and the volumetric expansion ratio of 5.3, will be related to the amount of leakage through the engine and the polytropic exponent relating pressure to volume.

Anticipated pressure ratios to be examined for the Lysholm engine require control of exhaust pressure between 7 and 15 psia. Manual testing of the air ejector fully open was performed. A maximum sub-atmospheric pressure of 8 psia was obtained for operating conditions of 100% steam, 120 psia inlet pressure and 6000 rpm. As inlet quality decreases and mass flow rate increases for higher engine speeds, the load of non-condensable gases in the exhaust line will increase and less of a sub-atmospheric pressure will be attained. The mass flow rate capacity of the condensate pump is another limiting constraint.

Control of the turbine exhaust pressure is desired for future staging operation of the combined Lysholm engine-turbine system. A schematic of the staged system is available in Fig. 7. The Lysholm engine will serve as the high-pressure stage of the system. The Lysholm exhaust will enter a separator to remove the liquid phase of the fluid mixture. The resulting, 99- 100% pure, saturated steam will then enter both or one of the Westinghouse, impulse turbines to complete the second, low-pressure stage of the system. The turbine's exhaust will then enter the main condenser where it will be condensed back into the liquid phase. Anticipated operating conditions at the Lysholm inlet will be low quality steam mixtures at high (120 psia) inlet pressure. Non-condensable gases will be introduced into the turbine exhaust and therefore, will require use of the air ejector to eliminate the build up of back pressure in the turbine exhaust line.

Two modes exist for overall staging control of the Lysholm-turbine system. One mode is to just utilize pure flow control, with the turbines running below 3600 rpm. This mode would be controlled by the Foxboro flow controllers. The other mode, would utilize governor control at the turbines, in addition, to flow control at the Lysholm. Controlling the turbine exhaust pressure would add another dimension to these two modes by allowing the interstage pressure to be varied between the Lysholm and the turbines. It must be stressed, however, that exhaust-pressure control can only be exerted for conditions when the flow is not choked in the exhaust line. Due to an inadequate amount of nozzles on the turbines, flow becomes quite easily choked and only a small range of slightly sub-atmospheric controllable pressures exist. Choking also occurs for flows

Lysholm exhaust line. Once the automatic, exhaust-pressure control system is set up, the practical limits of control, before choking, will have to be established.

To achieve maximum power output of the staged system; however, the air ejector will just be manually fully open, with the turbine exhaust line choked, and no exhaust pressure control desired.

Automatic, exhaust pressure control is desirable because it involves removing air and other non-condensable gases from the exhaust and does not interfere with the Foxboro flow controller. Lysholm engine speed and consequently inlet pressure will be affected once exhaust pressure is changed; however, automatic control of exhaust pressure should free the operator from constant adjustments to the air ejector and allow him to concentrate on controlling only the mass flow rate and eddy current brake to achieve his desired operating points.

I.4 Problems with Manual Exhaust Control

The hand valve located in the ejector steam inlet line did not prove adequate for manual exhaust control during testing operations. Since, the valve was located in the sunken pump room beside the ejector, another operator was required to constantly adjust the valve in response to commands from the control room operator, who was monitoring the exhaust temperature on the Autodata Nine. Note: exhaust pressure is found from saturated steam exhaust temperature. This procedure proved to be very tedious and time consuming. In addition, hand adjustments on the valve were rather coarse for fine adjustments in exhaust pressure.

Remote, exhaust control was achieved at the control panel by placing a pneumatic, Jamesbury Model IMO-32 1/2" ball, control valve in the ejector steam inlet line. The pneumatic valve was placed in series and downstream of the manual hand valve as shown in Fig. 8. The control valve is rated at 3-15 psig air-to-close. The operator adjusts a small, air regulator to control the air supply to the valve. When remote, operator control is not desired, the Jamesbury control valve will remain open and the hand valve can be used for control.

Monitoring the exhaust temperature on the Autodata, the operator can now adjust the air regulator's pressure to the control valve until the desired exhaust pressure is reached. Effectively, the operator is exerting closed loop control on the exhaust pressure. Exhaust control was still difficult, as the control valve proved to be very sensitive, in a narrow band, to the control air supply. The desired exhaust pressure was often overshoot in frustration to the apparently long time response of

the control valve-ejector system. Thus, another operator was required to adjust the air regulator while the main operator monitored the engine speed and operating performance. This was still a tedious process as engine speed, exhaust pressure and inlet pressure are all coupled to each other. At this point, the operating characteristics of the Jamesbury valve and the possibility of automatic, closed loop control were investigated.

The valve's movement was characterized by measuring the relative position of a pointer on the valve face (Fig. 8) as a function of control air. Fig. 9 shows a graph of the Jamesbury valve movement as a function of control air. The test data is available in Table 1. The graph shows fairly linear movement of indicated position as a function of control air. However, this movement only occurs for ^acontrol air supply of 7.6 to 12.6 psig. This gives a control band of only 5 psi as compared to the manufacturer's 3-15 psig rating. This controllability band becomes even smaller when you ^{have} a hysteresis of almost 0.3 psi. The limited control air band explains why the valve appears to be so sensitive and is difficult for the operator to control.

Since engine running time is limited, due to water leakage into the oil system, it is essential that some sort of automatic control system be implemented to provide quick and stable control of the exhaust pressure. The controller would be required to automatically maintain a desired exhaust pressure while engine speed, flowrate and inlet pressure were varied to obtain data points.

Proportional-integral (PI) control is therefore required to maintain the desired set point exhaust pressure in response to changing loads, without the offset introduced by proportional control operating alone.

II. OPEN LOOP SYSTEM

II.1 Description of the Open Loop System

Before designing the closed loop control system, the static and dynamic characteristics of the open loop system must be measured and analyzed. The open loop system exhibits inherently stable behavior. That is, for a set point change in control air to the pneumatic control valve, the exhaust pressure will stabilize at some new value. This stability assures that our system is controllable.

Fig. 10 shows a block diagram of the open loop control system. With the small air regulator on the Lysholm control panel, the operator is able to adjust air to the pneumatic valve. The resulting valve movement controls the ejector steam chest pressure which determines the air ejector's effectiveness in removing non-condensable gases from the main condenser. As non-condensable gases are removed from the main condenser and the Lysholm or turbine exhaust line, the backpressure exerted on the respective exhaust port is decreased, allowing the exiting fluid to expand to a lower pressure.

For the purpose of control theory modeling, the pneumatic control valve, air ejector, ejector steam supply, after condenser, main condenser and the exhaust line of either machine, will all be considered the "plant" of our open loop system. For a given control air input to the plant, a corresponding exhaust pressure will be the output. This input-output relationship of a system, specified in the Laplace domain, is called the transfer function of the system.

II.2 Modeling of the Open Loop System

The open loop control system is a rather complex, non-linear system with many coupled variables and effects. For analytical purposes, the system has to be modeled as a linear system. This is not too much of a simplification, as the transfer function can be piece-wise linearized around specific operating points and only the dominant time responses of the system will be analyzed.

A model can be made of the open loop system to study the internal behavior of the system elements. The reason for this internal analysis is to gain greater understanding of the plant dynamics and to identify which element exerts the dominant effect on the exhaust pressure. The modeling method used will be the "black box" approach of the transfer function method. So, rather than treat the whole plant as a black box initially, the plant will be subdivided into its four major elements: air regulator, control valve, the ejector steam chest pressure and the combined ejector, main condenser and exhaust line. A transfer function can be written from the control air input to the output of each element (Fig11).

A linear first order transfer function is written in the form: $G(s) = K/(Ts+1)$, where K is the static gain of the system and T is the system's time response, in units of time. The " s " indicates that the function is transformed into the Laplace domain. The analytical approach taken will be somewhat qualitative and consist of hypothesizing the order of each element's transfer function. Experimental testing of each element's gain and time

response to setpoint changes will provide a more accurate description of the transfer function.

The first system element to be analyzed is the air regulator, used to adjust the control air signal to the pneumatic valve. Assuming ideal behavior for the regulator, a direct correspondence should exist between the manual pressure set by the operator and the regulator's output pressure. This implies that the regulator changes instantaneously in response to input, and set point changes. This is a valid assumption since output pressure has very fast response, fractions of a second, to the throttling action introduced by a set point change. This time delay is small compared to other responses in the system and can therefore be neglected.

The other assumption implied by the ideal behavior is that there is no transportation lag (time delay) required for the new output signal change to reach the control valve. This was not a valid assumption initially, when the air regulator was mounted on the Lysholm control panel. As seen in the physical layout of the Lysholm engine and auxiliary equipment (Fig 12), the control valve is installed a considerable distance from the regulator. Test data verified that a small time lag occurred (≈ 1 sec.). Also, the step response of the regulator decayed into a ramp input by the time the signal change reached the control valve. This occurs because the air regulator is unable to bring the entire control line instantaneously up to the new pressure value.

This nonideal behavior was eliminated by placing the air regulator in the sunken pump room, directly in front of the valve's control air inlet. All testing of control system response was then conducted with the regulator in this position. During closed-loop operation, the controller's output

will also be placed directly in front of the control valve's control air input, therefore no transportation lag will exist for closed loop operation either. The air regulator can then be modeled for ideal behavior with a transfer function, $G_{ar} = 1$.

The air regulator is then a static element in the control system with no time dependency or delay.

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The control valve movement

will no longer be considered independently and will be buried in the transfer function of the ejector steam chest pressure to control air. The transfer function of the ejector steam chest pressure is therefore; $G_s(s) = K_s/(T_1s+1)$.

From a physical standpoint, operation of the open loop control system is as follows: as control air is increased to the pneumatic valve, the valve closes, throttling the steam supply pressure to the ejector chest. The reduced pressure in the ejector chest decreases the effectiveness of the ejector in removing non-condensable gases from the main condenser, allowing a back pressure to build up in the exhaust line which drives the Lysholm or turbine exhaust pressure up.

Since it is feasible that a time response will occur between the ejector and the resulting Lysholm or turbine exhaust pressure another time constant, T_2 , will be introduced to the system. The exhaust pressure will also have some gain, K_e , with respect to the ejector steam chest pressure, ^{The resulting transfer function} will be $G_e(s) = K_e/(T_2s+1)$.

The overall transfer function of the open loop system, the plant, will simply be the product of $G_s(s)$ and $G_e(s)$ and is equal to $G_p(s) = \frac{K_p}{(T_1s+1)(T_2s+1)}$, where $K_p = K_s K_e$. In conclusion, the plant is modeled as a linear second-order system with a transfer function;

$$G_p(s) = \frac{K_p}{(T_1s+1)(T_2s+1)}$$

II.3 Static and Dynamic Testing of the Open Loop System

Testing of the open loop system was performed in two phases. The first was a static test to determine the relationship between the control air and each one of the variables shown in Fig. 13. The test procedure was to introduce setpoint changes in the control air input, allowing the entire system to stabilize, before measuring each of the variables. The static relationships determined are the gains for each transfer function. The second phase was to dynamically test the time response of each element by continuously monitoring the instantaneous response of each variable to a set point change of control air. Assuming a first order, linear system, the time it takes each variable to reach 63% of its final value, is the time constant, T , of the element's transfer function.

Fig. 13 shows a diagram of the test set-up and how each variable was measured. For a given change in control air, the following variables were measured: 1) control air at the valve inlet, 2) control valve position, 3) air ejector steam chest pressure, 4) turbine exhaust pressure. The effectiveness of the system was tested on the turbine exhaust assuming that the turbine could simulate a non-condensable load for itself and the Lysholm engine on the main condenser and the ejector. The Lysholm engine was not used, as its running time is limited, due to the build up of water in the oil tank. It should be noted that the major purpose in testing the open loop characteristics of the system is to demonstrate and design the control system. Once the closed loop control system is implemented, the controller can be specifically tuned for the Lysholm and the turbine.

Static Testing

The first static test performed consisted of measuring the control valve position, ejector steam chest pressure, and turbine exhaust pressure as a function of control air. Note: For this test, the air regulator for control air was still mounted on the control panel. However, the transportation lag introduced has no effect on the static test results. During the test, the turbine speed was kept at 3600 rpm so that the turbine's speed-control, governor would automatically throttle the turbine inlet chest pressure to maintain constant speed at 3600 rpm. The governor, speed-control mode was used, as this mode responds quickly to changes in load or operating pressures. A fixed load of 40% was applied to the turbine.

The data from the test are available in the Appendix A. The following conclusions can be drawn from the test data:

- 1) Valve movement agrees within $\pm 5\%$ with the earlier test performed to characterize the control valve movement. The valve moved between 7.6 and 12.2 psig, resulting in a 4.6 psi control band. Hysteresis was 0.5 psi in the worst case.
- 2) The air ejector steam chest pressure, plotted in Fig. 14, only responds to a 3 psi control air range of 8.75- 11.75 psig. This indicates the effective closing movement of the valve. Although the steam pressure response is not linear initially, a pronounced linear region is observed. A gain of -100 psia steam chest pressure per psig control air is measured. Accounting for hysteresis of approximately 0.5 psig, there actually exists only a 1 psi band where the ejector steam chest pressure dramatically responds.

- 3) The response of the turbine exhaust pressure to control air is very nonlinear. The turbine control response band is, however, almost identical to that of the ejector steam chest pressure. This verifies that the ejector steam chest pressure is the dominant controlling variable of exhaust pressure. No linear gain can be approximated as the turbine exhaust decreases from 20 to 3 psia, while the control valve closes. The turbine appears to exert a very nonlinear influence into the system, which the closed loop controller will have to adjust for, in order to hold the turbine exhaust pressure constant for load and mass flow rate changes.

The above statement necessitates that the controller used in the closed loop system have integrating action, in addition to proportional control. The purpose of the integral action is to eliminate offset introduced by the proportional control in response to load changes at a given set point.

During closed loop modeling, the Lysholm engine and the turbine will actually be modeled as part of the plant. They act to introduce disturbances independently of the controller, which directly effect the exhaust pressure. Since the transfer functions of either the Lysholm or turbine cannot be modeled linearly, it will suffice to say that they introduce nonlinear, disturbances in the control system. One of the performance criteria of the closed loop control system will be to evaluate how well the controller compensates for these dynamic effects. In practical terms, this means that the controller will have to maintain constant exhaust pressure as the engine or turbine's speed, flow rate, or inlet pressure vary.

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To easily obtain more extensive data in the active control air band (approximately 8.5- 12 psig) a static test was performed on the main condenser. Both main inlet valves to the condenser were closed, and a small air intake valve was open to atmosphere to simulate a load of non-condensable gases. The data obtained from the test are available in the Appendix A, with the results as follows:

- 1) The graph of ejector steam chest pressure verses control air, (Fig. 15), agrees strongly with that obtained from the earlier turbine exhaust test (Fig. 14). This is to be expected as the Lysholm/turbine disturbances are introduced downstream of the steam chest pressure. A 3.2 psi control band of 8.6- 11.8 psig is observed, with hysteresis of 1 psig. A minimum change of 0.6 psi is required to overcome hysteresis and get the ejector pressure to reverse. The gain, identical to the previous result, is -100.
- 2) The graph of condenser pressure verses control air (Fig. 16), exhibits linear behavior with a 2 psi control band from 9.5- 11.5 psig. This linearity proves that the open loop system can be correctly modeled linearly with nonlinear disturbances introduced by the turbine or Lysholm. The condenser pressure has a gain of 10 psia exhaust pressure per psig control air. Hysteresis of almost 1 psi occurred, passed on from the ejector steam response. The condenser exhaust ranges from 15- 3 psia with an actual control band of 1 psi, considering hysteresis.

It is now apparent why manual control was so difficult. A 5 psi control band for valve movement deteriorates to a 3 psi band for ejector steam chest pressure response and to a 2 psi band for final exhaust pressure control. These bands are effectively reduced further, when accounting for hysteresis of approximately 0.5 to 1.0 psi.

Dynamic Testing

The procedure to dynamically test the time response of the open loop system consisted of introducing step changes in control air while continuously monitoring, on the Autodata, control air pressure, ejector steam chest pressure, and turbine exhaust pressure as functions of elapsed time. Figures 17-20 show graphs of 4 different step responses with the data available in the Appendix B. Rigorous analysis was attempted to determine the time response of the turbine exhaust pressure. From the open loop model, we expect to see two dominant time responses corresponding to a second order system. It is apparent from the inflection point observed in each exhaust response curve, that at least two time constants exist, with one responding several seconds after the first.

Time constants derived from each of the curves yield values ranging from approximately 1 to 20 seconds. This variance is explained by the highly nonlinear response of the turbine, which seems to dominate the system response. The approximate nature of these values is stressed, as the Autodata, in the continuous mode printed out data every 2 seconds. It appeared, however, to be actually less than two seconds with the period fixed by an internal clock. A data output rate every 2 seconds is not quick enough for a system with a response time in the same order of magnitude. A strip chart should be used next time. Another source of error results from the turbine exhaust pressure calculated from the saturation pressure of the exhaust temperature. The time response of the sensing thermocouple is greater than the pressure response time.

The ejector steam chest pressure responds very rapidly and consistently, with a time response on the order of a second or less. Control air step inputs, introduced with the regulator placed at the control valve inlet, appear to take approximately 2-3 seconds to reach their new levels. Apparently, this is an error resulting from the 2 second recording time rate of the Autodata.

Overall, the step response curves give a general indication of the system response. Initially, with remote operator control response times on the order of 30 seconds were anticipated. In conclusion, static and dynamic testing of the open loop system yields a second order transfer function, $G_p(s) = \frac{10}{(T_1s+1)(T_2s+1)}$, with T_1 , T_2 apparently in the range of only a few seconds. The exhaust controllability band corresponds to a 2 psi control air band from 9.5-11.5 psig.

III. CONTROLLER DESIGN

III.1 Design of Closed Loop Control System

The basic requirement of the closed loop control system is to provide quick and stable control of the Lysholm or turbine exhaust pressure, at different set point pressures, in response to mass flow rate and load disturbances. Available hardware, from a former LLNL control panel, consisted of a Foxboro(13 series) pneumatic Differential Pressure Transmitter (abbreviated as D/P), a pneumatic Moore (model 50) Nullmatic PI Controller, and numerous pressure regulators and gauges.

The model series 50, Nullmatic Controller is a two-mode controller which operates on the pneumatic "Null" balance principle. It controls the process variable, exhaust pressure, by regulating its output of control air to the pneumatic valve, as it tries to match a feedback of the exhaust with the set point reference. The difference between the set point reference and the measured, feedback of the exhaust pressure, is the error seen by the controller. The controller's air output signal is then proportional to the error and the integral of the error, taken over the integral, or reset time. The controller features an adjustable proportional band of 2- 200%, inversely corresponding to a gain of 50-.5. Reset action is available at 0.1- 50 minutes per reset, corresponding to integral time constants of 6- 3000 seconds. The nullmatic controller operates entirely on a force-balance principle, with all forces acting on diaphragms along the long axis of the instrument (Fig. 21).

The Foxboro D/P will serve two main functions in the control system: 1) To provide a suitable (3-15 psig) output signal, characteristic of the exhaust pressure, as input to the controller's process variable. 2) To apply gain to the differential exhaust pressure measured, which will increase the controller's sensitivity to fluctuations in exhaust pressure.

The Foxboro, 13 series D/P transmitter is rated to measure differential pressures from 0-250" H₂O (9psi) across its high and low ports. It produces a proportional, full-scale, signal of 12 psi above an offset pressure. This offset pressure, the output when the two ports are equally pressurized, is normally taken to be 3 psig. This establishes the conventional, D/P output range of 3-15 psig. The D/P sensitivity gain can be varied by adjusting the differential span, with reference to the fixed output range of 12 psi.

Although the model 13 Foxboro D/P is rated at 250" H₂O (9 psi) maximum span, the span was extended to 333" H₂O (12 psi) by turning the range wheel up as far as it would go. The D/P cell operates on a force-balance principle, with the span or range wheel acting as the pivot in determining the gain between the output and input. A 12 psi differential span gives a minimum possible gain of 1. Anticipated pressures for Lysholm and turbine exhaust control are 4-18 psi and 3-15 psi, respectively. These ranges do not consider the effects of choking or constraining the limited capacity of the condensate pump. It is impossible for the D/P to measure across the entire 14 psi exhaust band of the Lysholm without two different references. One

reference, atmospheric, is required to measure differential vacuum pressures, while the other reference, a regulated air supply, will measure above-atmospheric exhaust pressures. Maximum flexibility of the D/P is achieved by installing two, 3 way Hoke $\frac{1}{4}$ " ball valves on the inlets to the high and low ports of the D/P. Either the Lysholm or turbine exhaust could be addressed at the low port while, the high port would reference atmosphere or, a 0-20 psig Fairchild (model #15) air regulator.

Consideration must now be given to optimally adjust the differential span of the D/P and determine the controller's set point reference as a function of the desired exhaust pressure. Some physical factors must be considered first. Both the Lysholm and turbine exhaust pressures are measured from taps in their exhaust lines directly below their exit ports. These pressure lines then travel approximately six vertical feet down, into the sunken pump room, before they enter the D/P transmitter. These signal lines will fill with water as the steam condenses from vapor to liquid, as a result of heat transfer across the copper tubing. This vertical water column contributes a static pressure head on the D/P port. Anticipating a value of approximately 2.25 psi from calculations made, the static pressure heads of the Lysholm and turbine D/P lines were measured. This was accomplished by comparing the saturation pressure of the exhaust temperature with the pressure recorded by the D/P. The static head, the difference between these two pressures, was found to be 2.15 and 2.35 psi $\pm$ 0.25 for the turbine and Lysholm, respectively.

Although this static head introduces an offset in the D/P

reading, it can be used to an advantage in measuring greater vacuum pressures on the exhausts. For example, a maximum span of 12 psid across the D/P corresponds to the lowest measurable pressure of 14.14 psig vacuum (atm. -12-2.15 psi) at the turbine exhaust. The turbine exhaust pressure is actually 2.15 psi less than its measured value at the D/P inlet.

Although it's beneficial for the controller to have high sensitivity to exhaust pressure fluctuations (corresponding to high D/P gains or, low span settings) there is a tradeoff between sensitivity and lowest measurable vacuum exhaust pressure. This results from the inability to put a sub-atmospheric reference pressure into the high port of the D/P. Initially a span of 12 psi was chosen to give maximum measurable vacuum exhausts for the Lysholm and turbine. This gain sensitivity of only 1 proved inadequate for later stable control operation of the turbine. Therefore, the maximum measurable exhaust pressure was sacrificed a little and a gain of 2 was chosen. This corresponds to a span of 166.67" H₂O (6 psi). With the static head contribution, this span setting allows pressures of 8.15 psig vacuum (6.35 psia) for the turbine and 8.30 psig vacuum (6.50 psia) for the Lysholm. For lower vacuum pressures, the air ejector will have to be fully open ^{IN MANUAL CONTROL} he ejector can pull a 4 psia exhaust on the turbine and approximately ^{AM} 8 psia exhaust on the Lysholm at pure steam, 120 psia inlet pressure and 6000rpm. For increased flowrates (i.e. higher speeds) and lower steam qualities, the exhaust pressure will be greater due to the increased load of non-condensable gases in the exhaust line. So, in actuality, only the turbine suffered a slight loss (4-6.35 psia) in maximum vacuum controlled exhaust pressure.

As seen in Fig. 21, the PI controller has four inputs and one output. One input is the 20 psig supply air which it modulates out to the control valve at any value necessary to match the process variable (P_{dp}) to the operator inputted set point (SP). The setpoint is introduced by the operator from a small air regulator and attached 0-30 psig gage on the control panel. The controller also has the option for internal feedback (manual bypass) or external feedback. External feedback facilitates smooth transistion from manual control to automatic by feeding manual control air to the controller reset reference chamber as a referenced output, while the transistion occurs. External feedback would necessitate an additional manual control air regulator and switching valve. Manual bypass is sufficient as long as care is taken to smoothly transfer to automatic control.

III.2 Modeling of the Closed Loop System

Fig. 23 shows a control theory block diagram of the closed loop control system. The D/P is shown in a position to measure a subatmospheric exhaust pressure. As the exhaust pressure decreases from the desired control value, the differential output of the D/P will increase with respect to the controller's set point input. The relationship between the set point value and the desired exhaust pressure is calculated in Appendix C. To bring the exhaust pressure back^{up}, steam pressure in the ejector must be decreased. This requires that the control valve close further. Since the control valve requires air-to-close, the PI controller output should also increase. Similiary, a decrease in the process variable, P_{dp} , requires a decrease in the controller's output to bring P_{dp} up to the set point reference. This relationship of controller output to the process variable input, requires that the controller operate with positive feedback. The Moore controller action plate should be set to "increase input/increase output." Note: The^{OVERALL} control system operates with negative feedback, as indicated in Fig. 23, because the D/P introduces a negative gain into the system when measuring the exhaust pressure.

Fig. 24 shows a control theory block diagram of a typical feedback control system. Note: that the plant has two transfer functions: $G_p(s)$, in response to setpoint changes of the reference input, $R(s)$ and another transfer function; $N(s)$, in response to load or mass flow rate disturbances, $V(s)$. During open loop testing of the system we were able to characterize a linear model for $G_s(s)$, however; $N(s)$ introduced non-linearities into the system^{that} was not modeled.

Fig. 25 shows the control block diagram of the physical

elements of the closed loop system. The block diagram is identical to that of Fig. 24 and the chart available in Fig. 26 can be used to relate the system's physical variables to the control theory variables.

Although open loop test results were rather qualitative, due to the limitations of the available test equipment, enough information exists to characterize the general closed loop response. The root locus method will be used to obtain qualitative information concerning the stability and performance of the system. The root locus method consists of locating the closed loop poles in the S-plane of a real-imaginary coordinate system and graphing the root locus migration as a function of a system parameter. The system parameter studied in this controller design is the effect of gain on the stability and response of the system.

As shown in Fig. 11 the individual transfer functions of the control system are as follows:

$$\text{Plant: } G_p(s) = \frac{10}{(T_1s+1)(T_2s+1)}$$

where T_1, T_2 are in order of a few sec.

$$\text{PI Controller: } G_c(s) = K_c(1 + 1/T_1s)$$

where $.5 \leq K_c \leq 50$
 $6 \leq T_1 \leq 3000$
 sec.

$$\text{Feedback: } H(s) = \frac{2}{(2s+1)}$$

The transfer function, $H(s)$ of the feedback element, the Foxboro D/P, was obtained as follows: The gain of the D/P was set at 2, so $K = 2$. Foxboro literature gives the 3 decibel, corner frequency of the D/P response as $w = 1$ hertz. This corresponds to a time constant of 1 second. To conservatively

allow for a longer time response, a time constant of 2 seconds will be taken. Therefore the transfer function of the feedback element is $H(s) = \frac{2}{(2s+1)}$.

The open loop transfer function of the system is;

$$G(s) = G_p(s) G_c(s) H(s)$$

$$G(s) = \frac{K(s + 1/T_i)}{s(s+p_1)(s+p_2)(s+.5)}$$

$$\text{where } K = \frac{10 K_c}{p_1 p_2}, \quad p_1 = 1/T_1, \quad p_2 = 1/T_2$$

The open loop system has one zero at $-1/T_i$ and four poles at 0, $-.5$, $-p_1$, $-p_2$. Results of the open loop dynamic tests indicate that the step input response was heavily damped with no oscillations. This indicates that the two dominant poles of the plant lie very close to, or on, the negative real axis. This observation results from the fact that complex roots would tend to produce greater frequencies of oscillation as they move away from the real axis. Therefore, we know that we have all open loop real poles along the negative real axis, assuring stable system behavior.

Although the exact values of the plant time constants are unknown, it is really not important, as the stability of the control system has been demonstrated. The major purpose of modeling the system was to prove that closed loop control will be stable and can therefore be attempted.

The closed loop transfer function of the system is $G(s) + 1 = 0$. The roots of this equation are the closed loop poles. Before any further analysis is performed, the control system will be installed and tuned. Information from the tuning process should be helpful in further describing the location of the closed loop poles.

III.3 Implementation and Tuning of the Closed Loop System

The controller system was hooked up and installed as explained in Appendix D. The Foxboro D/P and the Moore PI controller were then calibrated and zeroed. This procedure is also described in the Appendix.

The next step was to separately tune the controller for ~~STABILITY~~ with the Lysholm engine and the turbine. The tuning procedure used was the Ziegler-Nichols, Ultimate Sensitivity, Tuning Method. It is an empirical approach to finding the optimum adjustments of the controller parameters for a given system (Ref. 8). The basic approach followed was to turn the integral section of the PI controller off and then keep turning the proportional gain of the controller up until the system went unstable, in response to a small step change in the set point input. Ziegler-Nichols then recommends that the optimum controller adjustments for PI action are:

$$K_c = .45 K_u$$

$$T_i = .83 P_u \quad \text{where } K_u \text{ is defined as the stability limit gain.}$$

P_u is the resulting period of cycling that occurs at this gain.

From the view point of the root locus; $w = 2\pi/P_u$, corresponds to the location where the root locus crosses the imaginary axis. Note: as the controller gain is increased, the system's response to introduced set point and load changes will improve; however, more cycling will occur in response to cyclic, transient inputs such as noise. As the integral time is decreased, system response will improve but, again more cycling and oscillations will occur as the system becomes more lightly damped. The Ziegler-Nichols tuning method should give a good approximation of the optimum values of K_c and T_i .

The lysholm engine was tuned first. By constantly turning the gain up and introducing small step inputs in the set point, system instability was noted at a proportional band of 60%. This corresponds to a gain of 1.67. The induced period of oscillation was measured by a strip chart recording of the D/P output pressure, as read from the Autodata. The period was found to be 33 seconds. Applying the Ziegler-Nichols formula, the proportional band was set to 132% (gain = .76) and the reset time was set to 27.4 seconds, or approximately .5 minutes per repeat. The system was unstable at this point, probably due to hysteresis of the control valve. The proportional band was turned further up to 200% (gain = .5) resulting in stable performance. To compensate for the slower time response at this gain, the reset time was set down to .35 minutes (21 seconds) per repeat. The controller operated quite well at this point, stabilizing in less than 1 minute to induced set point changes. Controller performance was then tested for load changes of flow rate and inlet pressure. Again, the system responded and stabilized within about 1 minute for small load changes, and approximately a minute and a half for large load changes.

The turbine can be run in either a flow control or governor, speed control mode. In the governor control mode the speed of the turbine is maintained at 3600 rpm allowing an automatic governor speed control on the turbine to throttle the inlet steam pressure to maintain this speed. This gives the turbine rather stable operation and quick response to load changes. Flow control involves running the turbines at speeds below 3600 rpm so the governor is not engaged. Loads on the electric generator are then manually adjusted in response to changes in mass flow rate and inlet pressure. The

controller must be tuned separately for each mode of control. Only the governor control mode has been tuned so far.

Running the engine at 3600 rpm, the Ziegler-Nichols tuning method was applied. The system went unstable at a proportional band of 35%, with a period of oscillation of 9 seconds. Applying the Ziegler-Nichols formula, the proportional band was then turned to 80% (gain = 1.25), with a reset time of 7.5 seconds (approximately .12 minutes) per repeat. The system was stable here, but the exhaust pressure tended to fluctuate a little. These fluctuations were decreased by turning the gain down, or the proportional band up to 200% (gain = .5). The controller was then finely tuned in response to load changes of 40-80% on the generator. The reset time was finally adjusted to 115 minutes (9 seconds) per repeat. The controlled exhaust pressure stabilized within 1.5 minutes for step changes in the set point, and within 0.5 minutes for step changes in the generator load.

III.4 Correlation of Model to the Tuning Parameters

An attempt was made to correlate the results of the Ziegler-Nichols tuning method with the qualitative, closed loop model to obtain a greater understanding of the shape of the root locus plot. During Ziegler-Nichols tuning the reset time was turned all the way up to its maximum value of 50 minutes, effectively shutting the integral action of the PI controller off. The open loop transfer function now becomes $G(s) = \frac{K}{(s+P_1)(s+P_2)(s+.5)}$

where $K = \frac{10K_c}{P_1 P_2}$. The root locus for the system now

corresponds to 3 asymptotes with direction angles of 60, 120 and 180 degrees for the center of gravity (cg) on the real axis (Ref. 8). The center of gravity is simply the sum of all the poles divided by the number of poles or, $cg = (P_1 + P_2 + .5)/3$.

The location where the asymptote crosses the imaginary axis (w) corresponds to $w = 2/P_u$, where P_u is the period of oscillation in seconds, measured during the Ziegler-Nichols tuning stability limit gain. For the Lysholm engine and the turbine P_u was measured to be 33 and 9 seconds, respectively. This corresponds to $w = .19$ and $.70$ rad/sec.

Drawing the asymptote at 60 from the real axis to intersect with $.7j$ on the imaginary axis gives a cg of $-.4$ for the turbine. The root locus is sketched in Fig. 30.

Fig. 30 . Root Locus for Turbine Control during Tuning.

Solving back for $P_1 + P_2$ in the calculation of the center of gravity implies that $P_1 + P_2$ must equal $-.7$, corresponding to a total sum of time constants equal to 1.4 seconds. This seems a little too fast for the observed system. Typical values of P_1 and P_2 were taken and the Routh Horowitz stability limit was applied to give the gain and the root on the imaginary axis where the asymptote crosses. The calculated gain an order of magnitude less than that found from the Ziegler-Nichols tuning process and the root was 5 times higher than that observed. Correlation was even worse for the Lysholm engine tuning results.

Other attempts were made to use the Ziegler-Nichols' RL method on the step response data obtained during open loop testing of the turbine exhaust (Appendix A). Again, Again, there was no correlation with the ultimate sensitivity tuning data. So, in the final analysis only a qualitative model of the closed loop system can be achieved. It should be noted that this is more than adequate to meet the objectives of the project as it correctly identifies the stability of the control system and displays the relative migration of the closed loop poles as the controller gain changes.

The root locus for both the Lysholm engine and the turbine exhaust control systems is shown in Fig. 31. The location of the closed loop zero must be somewhere to the right of $s = .167$ on the real axis corresponding to a zero $= 1/T_i$ where T_i has the range of $6 - 3000$ seconds. The recommended Ziegler-Nichols controller tuning parameters and the optimal parameter settings found should yield roots that lie somewhere along the root locus.

Fig. 31 . Characteristic Closed Loop Root Locus

IV. CONCLUSIONS AND RECOMMENDATIONS

IV.1 Performance Evaluation

In terms of forfilling the objectives of the project, the closed-loop, pneumatic control system was a success. The control system exhibited stable control of both the Lysholm engine and turbine exhaust pressures at different set point pressures and under various load disturbances. The PI controller's parameters were set using the Ziegler-Nichols tuning method, with fine tuning accomplished in response to load disturbances. The optimal controller parameters were found to be: proportional band = 200% and reset time = .40 and .15 minutes/reset for the Lysholm engine and the turbine, respectively. Results of the tuning procedure, as well as performance response of the exhaust pressures to load inputs, are shown in Appendix E.

At optimal controller settings, the turbine exhaust pressure took approximately 45 seconds to stabilize in response to either a set point change in exhaust pressure, or a 50% - 70% change in applied generator load. At greater load changes, the turbine exhaust took approximately 90 seconds to stabilize. The Autodata recordings of the exhaust pressure were stable to one degree Fahrenheit, indicating a pressure fluctuation of less than $\pm .25$ psi. However, since the temperature measurement of the thermocouple has a longer response time than the pressure fluctuations, the pressure calculated from the recorded temperature is actually the average of the instantaneous pressure values. From an overall system view while taking data, the averaging effect of the thermocouples is beneficial as it more accurately describes the pressure during the longer time interval that it takes the operating values of the system to respond.

At optimal controller settings, the Lysholm exhaust pressure took approximately 63 seconds to stabilize to the mass flow rate changes corresponding to speed changes of 4500 to 6000 rpm, and 72 seconds to respond to a change of 6000 to 7500 rpm. For a load change of 6000 to 9000 rpm, where power to the eddy current brake was decreased with no corresponding increase of mass flow rate, the Lysholm engine took approximately 2.5 minutes to stabilize. For set point changes in exhaust pressure, the Lysholm took approximately 1.5 minutes to stabilize.

On the whole, the Lysholm engine has a slightly longer response times than the turbine, with approximately the same amount of damping. This can be attributed to the longer integral time setting on the PI controller for Lysholm engine exhaust pressure control. The turbine's governor speed-control also aided in the turbine's relatively quick response time. To substantiate this hypothesis, the controller should be tuned for the turbine running in the flow control mode. Measured response time should then be longer than that of the turbine in the governor-control mode and the Lysholm engine, due to the high rotational inertia of the turbine.

In either case, response times of 1 to 2 minutes are less than the dynamic response of the engine and the turbine and are therefore sufficient to control exhaust pressure during data gathering operations.

An operating procedure for switching from manual to automatic control of exhaust pressure is given in Table 2, with the shut-down procedure given in Table 3. The only real consideration stressed is to gradually switch into automatic control and to monitor the pressure in the condenser to that the

condenser does not flash, causing the condensate pump to cavitate.

IV. 2 Recommendations

One way to improve system stability and response time, by allowing higher gains and lower integral time setting for the PI controller, would be to attempt to reduce hysteresis in the movement of the control valve. Hysteresis could be reduced by eliminating some of the friction caused by tight valve packing. The valve should be taken apart and the ball sanded down a little, to reduce friction between it and the packing as it rotates. After this is performed, the new movement of the valve verses control air should be tested and then the control system retuned for operation with the Lysholm engine and the turbine exhaust pressures again. Results should show reduced hysteresis as well as a wider control air band for valve movement. This will translate to better control, with higher gains and lower integral times, as the valve will now be more finely controlled by the PI controller's output signal.

With reduced hysteresis, the analytical model of the control loop system should be much more easily and perhaps, quantitatively, correlated to the Ziegler-Nichols tuning parameters. In the developed model of this project, hysteresis was not accounted for. Hysteresis, in addition to the nonlinear response of the turbine or Lysholm to load changes, were the major reasons for the quantitative discrepancies between the model and tuning.

Another task to be performed is to tune the controller for the turbine operating in the flow control mode. The results of this process should be compared to those of the Lysholm and the turbine operating with governor control to gain a greater understanding of the dynamics of the turbine. The last thing to be explored is the extent of staging control possible during

operation of the Lysholm-turbine energy conversion process.

FIGURES:

PLAN SECTION VIEW

SIDE SECTION VIEW

Fig. 1 . The Lysholm Engine

Fig. 2 . Photograph of the Lysholm Engine and Operating Equipment

Fig. 3 . The Lysholm Engine Operating System

Control lines -----
Flow pipes ————

Fig. 5 LYSHOLM ENGINE CONTROL PANEL AND AUTODATA NINE

Fig. 6 . Air Ejector and After-Condenser

Fig. 7 . The Lysholm - Turbine Staged Operating System

Fig. 8. Jamesbury Control Valve installed in steam supply line to air ejector

Fig. 8'. Expanded View of Valve-Position Face Plate.

Fig. 9 . Movement of Jamesbury Model IMO-32, $\frac{1}{2}$ " Ball, Control Valve

Fig. 10 . Block Diagram of Open Loop Control System

Fig.10'. Photograph of Pneumatic Control Valve and Air Ejector System

$$G_{ar}(s) = 1$$

$$G_v(s) = K_v / (T_1 s + 1)$$

$$G_s(s) = K_s / (T_1 s + 1)$$

$$G_e(s) = K_e / (T_2 s + 1)$$

$$G_p(s) = \frac{K_p}{(T_1 s + 1)(T_2 s + 1)}, \quad K_p = K_s K_e$$

Fig. 11. Block Diagram of Transfer Functions of Open Loop System

Fig. 12. Physical Layout of Lysholm Engine and Turbine Test and Auxiliary Equipment

Measured:

- 1 - Control Air at Valve inlet:
Autodata C.N. 31 using Validyne Pressure Transducer
uncertainty = ± 0.2 psi
- 2 - Control Valve Position:
recorded position of marker (% closed) on valve face plate,
uncertainty = $\pm 5\%$ closed
- 3 - Ejector Steam Chest Pressure:
Autodata C.N. 30 using Validyne Pressure Transducer
uncertainty = ± 0.5 psi
- 4 - Turbine Exhaust Pressure:
Autodata C.N. 26 temperature converted to saturation temperature
uncertainty = ± 0.75 psi , turbulence fluctuated temp. readings

Fig. 13 . Test Set-Up to Measure Open-Loop Transfer Functions

Fig. 14 . Ejector Steam Chest Pressure verses Control Air to Valve
-Controlling Turbine Exhaust with Air Regulator at Control Panel

Fig. 15 . Ejector Steam Chest Pressure verses Control Air to Valve

Fig. 16 . Condenser Pressure verses Control Air to Valve

Fig. 17. Step Response #1 of Turbine-Controlled Exhaust

- Set Point change of 11.2 - 10.2 psig

Fig. 18 . Step Response # 2 of Turbine-Controlled Exhaust

Set Point change of 10.0 - 9.4 psig

Fig. 19 . Step Response # 3 of Turbine-Controlled Exhaust

Set Point Change of 8.8 - 12.2 psig

Fig. 20. Step Response #4 of Turbine-Controlled Exhaust

- set point change of 9.6 - 17.8 psig

Fig. 21. **NULLMATIC CONTROLLER MODEL 50**

PRESSURES

SUPPLY		RESET
SETPOINT		INTERMEDIATE
MEASURED-VARIABLE		PILOT-NOZZLE
RESET-REFERENCE		CONTROLLER OUTPUT (B)

Fig. 21. Block Diagram of Exhaust Controller System

Fig. 23. Control Block Diagram of Closed Loop System

Fig. 24. Control Theory Block Diagram of Typical Feedback Control System.

Fig.25. Block Diagram of Exhaust-Pressure Control System

VARIABLES	CONTROL DEFINITION	PHYSICAL ELEMENT	UNITS	EQUATION
R(s)	Desired (reference) variable	Set point(SP) introduced by operator	0- 20 psig	SP= 2(H-L)
C(s)	Controlled, process variable	Lysholm or turbine Exhaust pressure (P _{ex})	4- 18 psia	
G _c (s)	Controller transfer function		G _c (s)= K _c (1 + 1/T _i s)	
B(s)	Feedback Signal	(P _{dp}) D/P measured output of exhaust	4.3- 16.3 psig	P _{dp} = 2(H-L)
E(s)	Error Signal	Difference between set. point and D/P output	0- 15 psia	E(s)=R(s)-B(s)
M(s)	Manipulated variable	Controller's output air signal	3- 15 psig	M(s)=G _c (s)E(s)
G _p (s)	Plant transfer function		G _p (s)= $\frac{K_p}{(T_1s+1)(T_2s+1)}$	
V(s)	Plant disturbances	Load & mass flow rate changes in plant		
N(s)	Plant transfer function to V(s)	Transfer function of Lysholm or turbine	Too nonlinear to model!	

Fig.26 . Comparison of Physical Variables of Exhaust Control System with Control Theory Variables

TABLES:

Table. 1 . Movement of Jamesbury Control Valve
- Data plotted in Fig. 9 .

Control Air		Jamesbury Valve Position ⁴ (% closed)
Validyne ² Transducer (psig)	Air ³ Regulator (psig)	
0 ↓	0 ↓	0 ↓
7.6	8.5	0
8.8	9.5	17
9.2	10.2	33
10.2	11.0	50
11.0	11.7	67
11.6	12.4	83
12.8	13.5	100
12.0	13.0	100
11.2	12.0	83
10.6	11.5	67
10.0	10.5	50
9.4	10.2	33
8.6	9.5	17
7.6	8.5	0

Notes:

1. Valve tested by supplying control air from an air regulator placed directly in front of valve's inlet.

2. Control Air read from a pressure tap between air regulator and valve inlet. Measured with Validyne DP-7 Pressure Transducer, uncertainty ± 0.2 psi. *

3. Control Air read from 0- 20 psig pressure gage on Fairchild Air Regulator Model #10, uncertainty ± 0.5 psi.

4. Valve Position read directly from Face Plate as % closed, uncertainty $\pm 5\%$ closed.

* - Not only is the control air read from the Validyne Pressure more accurate than the reading from the Air Regulator's pressure gage, it also represents the valve's control air signal better as it is measured downstream of the air regulator.

Table 2:

Start-Up Procedure for Automatic Exhaust Control System

1. With control system off, manually turn on air ejector to warm up lines. Once warmed up, shut hand valve on ejector steam supply line.
2. Set air regulator on control panel to a low set point pressure less than 4.3 psig. This should cause the controller to hold the exhaust pressure at atmospheric or, the pressure the high side of the D/P is referenced to.
3. Turn on supply air to controller panel in the sunken pump room.
4. While you are allowing steam to warm up the Lysholm engine or turbine lines you can turn the exhaust control system into effect. Slowly open the hand valve on the steam supply line watching to see the pneumatic control valve gradually closing. This occurs because there is very little load on the air ejector system so the controller should cause the valve to close.
5. You are now in the automatic control mode and you can gradually move the set point pressure where you want it using the equations:
$$SP = 2(H - P_{ex}) \text{ for the turbine.}$$
$$SP = 2(H - P_{ex}) - .3 \text{ for the Lysholm.}$$

Be careful to monitor the control panel gauge for the condenser so you do not flash it and cavitate the condensate pump by trying to pull a sub-atmospheric exhaust pressure too rapidly.

6. Monitor the exhaust pressure on the Autodata and fine tune your set point until you achieve your desired value.
Note: initially when running the Lysholm on an inlet pressure on less than 90 psia you will have little control as the volumetric expansion ratio already exhausts at sub-atmospheric pressure.

Table 3:

Shut-Down Procedure for Automatic Exhaust Control System

1. Close hand valve on steam line to air ejector.
2. Turn off supply air to controller panel in the sunken pump room.

APPENDIX A:

Test Data From Open Loop Static Test On Turbine Exhaust

Test Data From Open Loop Static Test On Condenser Pressure

Open Loop Static Test Data for Turbine

Air Supply (psig)	Turbine Exhaust		Ejector Steam (psia)
	Gauge (psia)	Sat. Temp. (psia)	
5.8 ± .2	2.4 ± 1	3.7 ± .3	126 ± 1
6.2		3.2	126
8.6	2.2	3.1	126
8.8	1.9	3.6	125
9.6	4.7	3.3	113
9.6	4.9	8.0	112
10.2	7.9	10.6	93
10.2	10.9	12.2	72
10.4	5.4	11.5	109
10.8	2.4	8.5	100
11.0	4.7	5.8	86
11.2	12.5	12.2	56
11.4	13.2	14.1	31
11.6	14.2	15.3	0
12.0	13.4	15.1	34
12.2	15.1	15.3	0
13.2	14.4	15.3	0

Open Loop Static Test Data for Condenser

- Condenser opened to atmosphere with small air valve partially open so full steam 3 psia pressure.

Control Air (psig)	Ejector Steam Chest Pressure (psia)	Condenser Pressure (psia)	Control Valve Position (% closed)
8.0 ± .2	126 ± 1	3 ± .5	0 ± 5
9.8	123	3	40
10.4	119		
10.6	112		
10.8	98	9	60
11.0	92		
11.0	84	10.5	60
11.2	66	12.2	75
11.4	55	13.2	75
11.4	42	14.6	75
11.6	23	15.0	90
11.8	0	15.0	
10.6	17	14.9	75
10.6	32	14.6	60
10.4	36	14.2	60
10.4	45	13	60
10.2	61	11.7	55
10.2	75	10.5	55
10.0	87	8.5	50
9.8	101	4.9	44
9.6	110	3.2	44
9.4	117	3.2	38
9.0	121	3.2	38
8.8	122	3.2	25
8.6	124	3.0	0

- Air valve on condenser fully open to simulate new flow rate.

Control Air (psig)	Ejector Steam Chest Pressure (psia)	Condenser Pressure (psia)	Control Valve Position (% closed)
6.4 ± .2	126 ± 1	3 ± .5	0 ± 5
6.6	126	3.2	0
7.8	125	3.2	0
8.4	126	3.2	88
9.6	124	3.2	15
9.8	122	3.2	33
10.0	121	3.2	38
10.4	117	3.2	50
10.4	114	3.2	50
10.8	101	7.4	57
10.2	102	7.4	57
9.8	108	5.5	42
10.2	107	5.5	50
10.4	107	5.2	57
10.8	103	6.4	63
10.8	97	9.0	67
11.0	89	9.9	69
11.1	75	11.4	71
11.3	67	12.2	75
11.5	52	13.4	79
11.5	37	14.5	100
11.6	24	15.0	100
12.0	0	15.0	100

APPENDIX B:

Data For Open Loop Step Response Tests # 1-4

Open Loop Step Response Data # 1

Elapsed Time	Control Air	Ejector Steam	Turbine Exhaust
(sec.)	Pressure (psig)	Pressure (psia)	Pressure (psia)
0	11.2 [±] .2	55 [±] 1	12.25 [±] .3
2	10.4	55	12.25
3	10.2	55	12.00
5		58	12.25
7		60	12.00
8		62	12.00
10		63	12.25
12		64	12.25
14-15		64	11.50
17		65	11.50
19-20		65	11.27
22-26		66	11.27
27-29		66	11.03
32		67	11.03
		Oscillates 67-68 psia	

Open Loop Step Response Data #2

Elapsed Time (sec.)	Control Air Pressure (psig)	Ejecctor Steam Pressure (psia)	Turbine Exhaust Pressure (psia)
0	10.0±.2	60±1	11.03±.3
1	9.6	101	10.8
3	9.6	106	10.14
5	9.6	108	9.31
6	9.4	109	8.36
8	9.4	109	7.66
10	↓	109	7.01
11		109	6.55
13		110	6.26
15		↓	5.98
17			5.84
18			5.58
20		111	5.08
22		110	4.62
23		110	4.30
25		↓	4.09
27			3.99
29-30	↓	111	3.90
31-34		Oscillates 110-111 psi	3.80

Open Loop Step Response Data #3

Elapsed Time (sec.)	Control Air Pressure (psig)	Ejector Steam Pressure (psia)	Turbine Exhaust Pressure (psia)
0	8.8+ .2	124+1	4.00+ .3
2	11.0	110	4.00
3	11.6	49	4.00
5	11.8	23	4.00
7	12.0	0	6.25
9	12.2		9.72
10	↓	↓	13.00
12			14.66
14	↓	↓	14.96

Open Loop Step Response Data #4

Elapsed Time (sec.)	Control Air Pressure (psig)	Ejector Steam Pressure (psia)	Turbine Exhaust Pressure (psia)
0	9.6±.2	112.5±1	3.5±.3
1	12.8	23.5	4.0
2	15.8	0	6.8
4	17.2		10.7
6	17.6		13.9
7	17.8		15.0
9	17.8		
11	17.8		
13	18.0		

APPENDIX C:

Calculation of The Set Point Pressure

The Foxboro D/P also requires a 20 psig supply air input which it modulates out (P_{dp}) as a function of the differential pressure (ΔP) across its high and low ports. Fig. 22 shows a block diagram of the D/P with all the inputs and outputs labeled. Normally with a 12 psi output span the zero offset (P_{zero}) is

Fig. 22 Block Diagram of Foxboro D/P

set to 3 psi when the high port (H) equals the low port (L). This equilibrium offset can be set to anything less than around 7 psig, however, P_{zero} was set to a value that would compensate for the water head offset (P_h) to simplify calculation of the set point. This calculation is as follows:

$$P = H - L$$

$$P = H - (P_{ex} + P_h) \text{ where } P_{ex} \text{ is the exhaust pressure in psia of the lysholm or turbine}$$

$$P_{dp} = 2 * P + P_{zero}$$

we want the set point to match our process variable, P_{dp} , so:

$$SP = 2 * P + P_{zero}$$

$$SP = 2 * (H - P_{ex} - P_h) + P_{zero}$$

setting $P_{zero} = 2 * P_h$ eliminates the offset.

Therefore,

$$\text{Eq.1} \quad \text{SP} = 2 * (\text{H} - \text{P}_{\text{ex}})$$

when $\text{P}_{\text{zero}} = 4.3$ psig for the turbine, $= 5.7$ psig for the lysholm

The D/P output, P_{dp} , will then range between 4.3-16.3 psig for the turbine and between 5.7-17.7 psig for the lysholm,

Eq. 1 is valid for high port (H) references of atmosphere ($\text{H} = 14.5$ psia) or regulated air references measured in psia.

The following is a sample calculation of the setpoint. Suppose the operator wished to control a 10 psia exhaust on the lysholm. Referencing the high port of the D/P to atmosphere, his setpoint would simply be : $\text{SP} = 2 * (14.5 - 10 \text{ psia})$ or, 9 psig when P_{zero} has been calibrated to 5.7 psig. As a long hand check of this,

$$\text{P} = \text{H} - \text{L} = 14.5 - 10 - 2.85 \text{ psia} = 1.65 \text{ psia}$$

$$\text{P}_{\text{dp}} = 2 * \text{P} + \text{P}_{\text{zero}} = 2 * 1.65 + 5.7 \text{ psig} = 9 \text{ psig} \checkmark$$

APPENDIX D:

Implementation and Calibration of the Closed Loop Control
System

The closed loop control system was installed as shown in the photograph of Fig. 27. The controller panel shown was mounted on the wall adjacent to the control valve to minimize any transportation lag between the controller's output and the valve. 95 psig supply air for the two air regulators was obtained from a building compressed air line, located at the base of the main condenser. One regulator supplies 20 psig supply air to the D/P and PI controller while the other regulator is used to provide a regulated reference pressure to the high side of the D/P.

Once all the hardware was set up, the first step was to calibrate the D/P. All pressure gauges used in the system were calibrated with a dead weight tester to assure their validity. The span on the D/P was set as close as possible to 166.7" H₂O on the range scale. Fig. 28 shows the test set-up used to more finely adjust the range on the D/P and set P_{zero}.

The low side of the D/P was opened to atmosphere, while the high side was referenced to the air regulator so any psig pressure inputted from the air regulator resulted in the same magnitude but, psi differential (psid) pressure across the D/P. The output of the D/P was then fed into the Validyne pressure transducer for Autodata C.N. 31. Similarly, the low side of the transducer was opened to atmosphere, so the D/P's psig output could be read as an absolute value. The validyne pressure transducer for Autodata C.N. 31 has a 32 psid range diaphragm in it. The transducer has been previously calibrated for the turbine #2 steam chest. The calibration was confirmed by first checking the zero and then applying regulated air supply directly into the high side of the transducer. The output of the

Fig. 27. Photographs of Exhaust Pressure Control Panel

Fig. 28. Test Set-Up to Calibrate the Foxboro D/P

transducer is a millivolt signal which is then amplified and conditioned by a Validyne Carrier Demodulator (CD-18) to output 0- 10 volts to Autodata C.N. 31. The output of C.N. 31 can be read either as a pressure (1 psi output calibrated to .2 psi) or, as a voltage (1 volt equal to 3 psi). Reading the voltage is more accurate as it can be read to four decimal places, with an uncertainty of ± 0.02 psi. The pressure output can only be read to whole numbers, providing an uncertainty of $\pm .25$ psi. Note: the regulated air supply to the high side of the D/P is only accurate to $\pm .25$ psi, so either Autodata readout is suitable.

Venting both sides of the validyne transducer, the zero on the CD-18 was readjusted slightly to read $\pm .2$ millivolts. Venting both sides of the D/P, the zero on the D/P was adjusted to read 4.3 psi (1.500 volts or 21-22 psi on Autodata CN 31). Applying a regulated 6 psig pressure to the high side of the D/P, the span and zero were iteratively readjusted as they effect each other, until the desired values were reached. The D/P has been calibrated to have a P_{zero} of 4.3 psi (for the turbine static pressure head) and a span of 6 psi differential pressure across its inlet. This corresponds to a gain of 2.

The next step was to calibrate the zero on the PI controller following the procedure outlined by the Moore Company product literature. The set-up for this is shown in Fig. 29. The controller's set point input was connected to the output and also to a pressure gauge. The process input was connected to an air regulator supply and to a gauge. The controller action reversing plate was set to increase input/increase output. The bypass feedback plate which could be left in either position, was set to manual bypass. The reset needle valve was "opened" to 0.1 minute per repeat and the proportional band was opened to 30%. The air regulator input to the process was then set to

Fig. 29. Zero Calibration of PI Controller.

approximately 10% of its span, or 2 psig. The controller's zero screw was adjusted until $P_1 = P_2$. The regulator was then set to 90% of its span, or 18 psig. If $P_1 \neq P_2$, then the controller's tracking screw was adjusted with a 3-fold overcorrection. The air regulator was then set back to 10% and the procedure was repeated until $P_1 = P_2$ for either regulator setting. Intermediate operation was also verified. The PI controller was found to be fairly well zeroed to begin with.

APPENDIX E:

Tuning Performance Graphs of Lysholm Engine and Turbine
Exhaust Pressures

Step Response of Turbine Exhaust Pressure at Proportional-Control
Stability Limit Gain

Proportional Band = 35 %

Gain = 2.9

T = 2.0 seconds

Damped Step Response of Turbine Exhaust Pressure

Proportional Band = 200 %

Gain = 0.5

Integral time = 0.2 minutes (12 seconds)

Performance of Turbine Exhaust Pressure Control at Optimum Settings

Proportional Band = 200 %

Gain = 0.15

Integral Time = 0.15 minute (9 seconds)

Damped Step Response of Lysholm Exhaust Pressure during Ziegler-
Nichols Tuning

Proportional Band = 70 %, no Integral action

Gain = 1.4

Settling Time = 24 seconds

Step Response of Lysholm Exhaust Pressure at Proportional-Control
Stability Limit Gain

Proportional Band = 60 %

Gain = 1.7

T = 33 seconds

Settling Time:
= 72 seconds

6000 to 7500 rpm

Settling time:
= 63 seconds

OPTIMAL SETTINGS

Proportional Band= 200%

Integral time = .4 min

← 0.6 psi →

4500 to 6000 rpm

Response of Lysholm Exhaust Pressure during Data runs:

100% steam, 120 psia inlet, 6000 rpm
7500 rpm

Response of Lysholm Exhaust Pressure to Load Change of 6000 - 9000 rpm
- accomplished by decreasing power to Eddy Brake

Proportional Band = 200 %

Integral time = .4 minutes (24 seconds)

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