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DESIGN, CONSTRUCTION AND EVALUATION
OF A SIMULATED GEOTHERMAL
FLOW SYSTEM

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By

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July 28, 1980

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ABSTRACT

A system was designed and built to simulate the flow from a geothermal well. The simulated flow will be used to power a Lysholm engine, the performance of which will then be evaluated for different simulated geothermal flows.

This report covers two main subjects: 1) the design, construction and evaluation of the behavior of the system that simulates the geothermal flow; included in that topic is a discussion of the probable behavior of the Lysholm engine when it is put into operation, and 2) the investigation of the use of dynamic modeling techniques to determine whether they can provide a suitable means for predicting the behavior of the system.

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1. INTRODUCTION

Due to the depletion of available oil reserves, the countries of the world are being forced to develop alternative sources of energy such as geothermal energy. Some experts estimate the geothermal energy reserves in the United States to be 1,000,000 mega-watt-centuries of thermal energy.¹ With such a large potential of usable energy available, many ways are being developed to utilize it. In the search for ways to convert the thermal energy to usable power, various energy conversion systems are being investigated. One of the ways being studied to try to increase the efficiency of geothermal energy conversion systems is through the use of a two-phase expander known as the Lysholm engine. Comparing a hybrid system using a Lysholm engine and one conventional engine with a comparable two-stage double-flash system, it is found that the system using the lysholm would be nearly ten percent more efficient.² Because use of the Lysholm engine has not been fully investigated, a project is being carried on at the University of California at Berkeley to further test the Lysholm engine. The tests will be run in order to obtain more accurate knowledge of the engine's performance characteristics. In order to run the engine performance tests it was necessary to simulate the geothermal fluid flow. Along with the simulation of flow, a dynamic model of the total system was created to investigate the effectiveness of dynamic modeling for predicting the behavior of multiple variable systems.

The Lysholm engine being tested has a maximum power output of 50 KW. The engine is also known as a helical screw expander.

which operates by taking in a volume of fluid and expanding the fluid against two side-by-side meshing helical screws. The engine is a positive displacement type with the energy conversion coming from the fluid doing work on the screws.^{2,9} (See reference 9 for a description of the Lysholm engine operation.) One of the major advantages of using this type of engine in a geothermal plant is its ability to expand low quality fluid and produce usable power while operating at high efficiencies.

Previous tests have been conducted on the Lysholm engine at the Lawrence Livermore Laboratory in Livermore, California, and the results of their tests showed that the efficiency of the engine increased as the speed increased.⁸ However, the test set-up at Lawrence Livermore Laboratory limited the speed of the engine because it limited the amount of fluid and quality of the fluid obtainable. In the Berkeley project it was necessary to design a system which would deliver the flow conditions that were not obtained at Livermore and which would simulate the geothermal wellhead conditions.

2. FLUID ANALYSIS AND SIMULATION

The type of fluid output from geothermal wells varies with each well.³ For this project the flow was designed to resemble the flow from wells usually found in the Imperial Valley in California, a flow which is a two-phase mixture of liquid and vapor that fluctuates as to its quality, temperature, and pressure. The wellhead flow conditions range from fluid with quality up to 40 percent, temperature between 350 - 450 degrees fahrenheit and pressure up to 400 psia.²

A preliminary analysis was made on the type of fluid flow regime which would be expected in the two-phase mixture. The primary factors in determining the type of flow were 1) the volume fractions of steam and water and 2) the velocity of each fraction. In this case with a flow of 20 percent quality the volume fraction of steam versus water was about 99 percent steam, the flow velocities of the water and steam being estimated by using the volume fraction and the known mass flow rates of the steam and water. (See Appendix I) Therefore with these two pieces of information, it was determined that the flow would be in the annular mist regime.^{4,5}

Annular mist flow is flow that has a film of water on the walls of the pipe which has a thickness that depends on the velocity of the fluid, type of pipe, and the velocity fraction. The rest of the water is contained within the central portion as entrained water droplets in the steam, the size of the droplets being determined by the velocity of fluid and the volume fraction. To prevent hold-up of the steam by the water, or visa-versa, which would cause excessive pressure losses, reduction in mass flow rate, and thermodynamic and flow regime instability, it was necessary to inject the water into the steam at a velocity close to the velocity of the steam. The system constructed combined the water and steam so that the flow regime would be close to annular mist flow as it entered the turbine.

3. FLUID AND CONTROL SYSTEM DESCRIPTION

To construct the required fluid system which was set-up in Hesse Hall at the University of California at Berkeley, a

significant amount of time was spent acquiring and locating the necessary equipment. The system was designed to use the existing steam and water supplies in Hesse Hall.

The maximum pressure of the steam supply is 110 psig. The steam supply pressure is the limiting factor in determining the range of pressures that could be used to run the Lysholm, with knowledge of this limitation, all the rest of the system was designed around the steam supply. The cold water supply is at a pressure of 80 psig.

It was necessary to combine hot water with the steam so that thermal equilibrium between the steam and water could be reached before entering the turbine. In order to provide the water at the required pressure and temperature a ten horsepower pump and a heat exchanger were purchased. The steam from the building supply line was used to heat the water in the heat exchanger and also to mix with the hot water to create the two-phase mixture. Figure 2 shows the position of the Lysholm and the Westinghouse turbines in Hesse Hall. Figure 3 is an overall system drawing. Figure 4 is a drawing of the steam and hot water mixing section. Figure 5 shows actual pictures of the system components.

To control the flows of the water and the steam a control system was purchased. (See figure 1). The control system consists of two air actuated control valves, and two orifices with square root converters to obtain signals for the flow rates from the difference in pressure across the orifices. The system also contained one flow controller and one flow ratio controller, both with proportional and integral control. The

signals from the square root converters also went to air-to-volt converters. The voltage signal went to a signal monitor where the value was read by the operator.

The amount of flow of the steam was set by the flow controller. The signal from the square root converter on the steam line went to the ratio controller which then multiplied this signal by the ratio selected, and this signal was then applied to the water control valve. Thus, the quality could be adjusted by changing the value of the ratio.

The total mass flow was determined by setting the flow controller, then setting the ratio controller. The states of the fluid at important points in the system were obtained by inserting thermocouple probes in the steam and water lines. The values for the thermocouples, mass flow rates and pressures were sent to a recorder that gave a continuous readout of the various states.

4. SYSTEM CALIBRATION

After the construction of the system was completed, and before the actual tests were run, an extensive calibration check was made. Each of the pneumatic instruments was checked carefully in order to determine that the instrument performed in the proper manner. It was found that each one of the instruments tested required calibration. Because the calibration is so important in order to obtain accurate knowledge of the operating conditions it was decided that periodic calibration checks should be made to prevent inaccuracies in measurement.

After the instrumentation was checked, actual flows were

run through the steam and water lines and the flow rates were measured using a weigh tank. The measured flow was compared to the indicated flow to check the accuracy. The results obtained were satisfactory, the difference between the indicated and measured flow never being more than four percent.

From the testing, it was found to be very important to keep the lines from the orifice to the differential transmitters full of water, because any steam in the lines caused large oscillations in the controller. To ascertain there was no steam in the lines, and to insure proper level of each signal tube, seal pots were installed in each tube line from the steam orifice.

5. TESTS OF CONTROL VALVE RESPONSE AND RESULTS

After the calibration of the various control elements was completed the system was tested. The elements first tested were both the steam and water control valve systems, and various tests were performed to find the best gains for the controller.

After steady flow was reached in the line being tested, a step change in the flow rate was introduced to the system, this step change being generated by changing the set point on the controller. The response of the flow was obtained by taking the voltage output from the air-to-volts converter, which changed the pressure output signal from the differential pressure cell and square root converter, to a voltage thus representing a flow rate. This flow rate voltage was monitored by a strip chart recorder. Sample results are shown in Appendix II.

The shape of the response curves varied considerably as the

gains of the controller were changed. For a large proportional gain a large overshoot was obtained, which agrees with control theory, as the error is multiplied by the proportional gain. The integral portion is used to avoid offset error by the controller. The signal from the integral side increases with time, as long as the error is positive and decreases when the error is negative, thus causing cycling around the steady state point. Too high of a reset rate produces a large amount of cycling and a long settling time whereas too low of a reset rate will produce only a small amount of cycling but a long time before offset will be removed. Overall, the control valve response compared very well with the response which theory predicted.^{10,11}

The response curves showed areas of nonideal response. Most of the nonideal response was caused by the limitation of the pneumatics to handle rapid changes in flow. A large change in flow caused an excess demand on the supply air. Therefore the air supply could not fulfill the requirements of the controllers, and the output signals reflected these fluctuations in supply air. When the demand for air was low the response showed little signs of nonideal areas. The supply air could not meet the demand because the supply for both valves, differential pressure transmitters and controllers was through a single quarter inch stainless steel thin-walled tube and one regulator. Therefore in order to remove the problem two supply tubes will be used; one supply tube to supply the two differential pressure (d/P) transmitters and the other tube to supply the control panel. Each d/P transmitter has a regulator and there will be two regulators in the control panel.

Another part of the system response checked was to determine how the steam and water flow controllers worked together. To determine the response, both steam and water sides were put into operation.

The ratio and the flow of steam were arbitrarily selected. When both flows were steady, a step change in steam flow was generated by making a step change in the set point, which caused both the steam and water control valves to respond.

The response of the valves was monitored again by a strip chart recorder, which recorded the flow rate of both water and steam. (Results of tests are found in Appendix III). The steam and water curves showed nearly identical response curves excepting that the water curve was approximately 180 degrees out of phase with the steam curve. For all the tests the settling times were approximately the same and the time constant for the logarithmic trend of the response was about three times longer for the water than the steam.

6. TESTS OF STEAM AND TURBINE SYSTEM AND RESULTS

To predict how the steam system will respond in actual operation, steam was used to run the existing Westinghouse steam turbine. At the time the test was run, the necessary support equipment for the Lysholm engine was not installed and therefore the Lysholm could not be run. Running the steam and the Westinghouse turbine together would give a good indication of the dynamic response of the system when both the Lysholm and the simulated geothermal flow are run. From the tests, the amount of time required for steady state operation

of the turbine could be obtained, and a better knowledge of the behavior of the total system would be obtained.

To run the tests, the steam flow rate and the turbine speed were monitored. The system could be run by running water and steam together, and then separating the steam out, or by running saturated steam directly to the turbine. In order to get a measured mass flow rate with which to check the controller reading, only steam was run to the turbine with no separation. The speed of the turbine was monitored by recording the voltage output from the RPM gauge of the turbine. The flow rate of the steam and the turbine speed were recorded by the voltage recording section of an Autodata 9, and it was also recorded by a strip chart recorder.

In order to observe how the turbine responds, the turbine had to be operated at a speed below which the governor engages, which is 3600 RPM. The tests consisted of running the turbine at a constant speed with a given flow rate of steam, then giving a step increase in the steam flow rate. The speed of the turbine was recorded until it had reached a steady value.

The step input was made with the turbine in the loaded and unloaded modes and for step changes of flow rate of .025 lb/sec (5%) and .05 lb/sec (10%) of steam. Data and plots from the tests are provided in Appendix IV.

7. DISCUSSION OF TEST RESULTS

The two-phase system responded very rapidly with a delay of no more than ten seconds and the output was very stable. Taking run number seven as a typical run for the response of

the steam flow system, the standard deviation was .0013 lb/sec at a steam flow rate of .227 lb/sec. (See Appendix 4). The system in its present arrangement is capable of providing steam and water mixtures from 8 to 100 percent quality and total mass flow rates of 6.06 lb/sec at 8 percent to .5 lb/sec at 100 percent. These flows can be altered by changing the orifices to give the desirable flow ranges.

The inlet pressure was a maximum pressure of 110 psig. All operations of changing the flows can be performed at a control panel which can be placed a safe distance away from the operating equipment. Changes in steam flow rate are made by moving the set point; changes in water flow rate are made by changing the ratio. The steam and water two-phase supply system performed very well the output being stable and showing no tendency to be otherwise.

The response of the turbine to changes in mass flow rate was nearly logarithmic. Fitting data curves from runs one and two to a logarithmic response gives a correlation coefficient of about 0.95. For step changes of 5 percent, which corresponds to .025 lb/sec mass flow rate, the time constant was nearly the same for each test, for both loaded and unloaded conditions and at different initial speeds. The time to reach 63.2 percent of the final value was about 150 seconds for each of the tests.

The difference found in the time constants was caused by several factors. 1) At times the turbine may have not quite been stable when the step change was initiated resulting in a

faster speed response. 2) The turbine may not have arrived at a steady final speed yet when the test was stopped thus causing a low time constant to be obtained. 3) There were also small differences in the exact size of the step which would result in differences in speed response.

Test number six gave results which were significantly different from the other tests. It gave a time constant of approximately 230 seconds, where the expected time constant was 150 seconds. Looking at the plot from the run gives no indication as to why the results were so different. Since there are enough data at the small step change to show this as being significantly inconsistent it will not be considered as valid, and it will be assumed that something in the test procedure was different for that run.

For the step change of 10 percent or .05 lb/sec of steam only two tests were run. The results were reasonably close, giving an average time constant of approximately 220 seconds. The tests were restricted to only these small changes in flow rate because any changes larger than these would cause the turbine's governor to be activated thus eliminating the free response of the system. The turbine response was not affected by the existence of the load, thus confirming that the load does not change with speed, but, does in fact stay constant.

From the testing of the turbine to step changes in flow, it is obvious that the largest factor affecting the response time of the system is the turbine itself. The time constant of the turbine was approximately fifteen times that of the

settling time of the steam supply response. The supply of steam was stable at least three minutes before the turbine reached 60 percent of its final value. The factor causing the slow response of the turbine was its inertia and the inertia of the coupled generator. The inertia of the turbine prevented the fast reaction of the system to step changes in flow. Because of the slow response, and the presence of the governor there is little danger of the turbine overspeeding without warning. From the tests, it was observed that in order to reach a stable speed, at least sixteen minutes must be allowed. Also, when step changes larger than .05 lb/sec are used the turbine will rotate at the governor set speed of 3600 RPM.

8. PROJECTION OF LYSHOLM BEHAVIOR

Projecting the response time to the Lysholm system, it would be reasonable to assume that the response time for the Lysholm would be much faster than the Westinghouse turbine. The faster response would be due to the smaller inertia of the rotating elements in the Lysholm. The Lysholm's response should again be logarithmic as is the Westinghouse turbine, but with a time constant smaller than the turbine. The inertia of the turbine was estimated to be about 2 slug ft² and that of the generator to also be 2 slug ft². Making a rough calculation for the Lysholm, the inertia of the rotors is about .02 slug ft², which is one hundred times less than the Westinghouse turbine. However, the Lysholm is also connected to a eddy current dynamometer which has an estimated inertia of around 2 slug ft² so overall the Lysholm engine could

have an inertia of half of the Westinghouse turbine. The time constant of the Lysholm system would therefore be half that of the Westinghouse. Since the response of the steam-water supply system is fast and the flow rate is steady there should be no problem of the flow affecting the response of the turbine or fluctuations in flow affecting the response. The Lysholm engine should accelerate smoothly to a steady speed and then stay at that speed.

Unlike the Westinghouse turbine there is no governor on the Lysholm, so there is no way to control the speed except by the dynamometer or the fluid supply. There is, therefore, a danger of the Lysholm running too fast. To avoid this from happening the operation of the Lysholm should be done with great care. There should be no step changes larger than 5 percent without corresponding changes in load, and there should be someone monitoring the speed of the engine at all times.

9. SYSTEM MODELING

Whenever there is a control system placed in a larger system there is always some uncertainty as to how the system will perform. Before the system is put in operation it is hard to know which control gains would give the quickest response and still give a stable system. For the fluid-turbine system being tested the number of different dynamic elements is small, so that adjusting the control gains would be fairly simple. If the system contained many more stages of separators, turbines, condensers, reheaters, etc., the job of optimizing the control settings would be very tedious.

A model was made of the steam and turbine system to get a better idea of how the system will behave when subject to step changes in inputs or to disturbances in supply pressures. If the modeling was successful then it could be expanded to be used to predict the behavior of a plant on the scale of PG and E power plants. This method could provide accurate means of predicting the behavior of the elements of the power plant, thus reducing the necessary start-up time.

10. BOND GRAPH MODELING

To make the model of the system, the method of bond graphing was used. To use the method of bond graphs to model dynamic systems, the system must be broken into separate elements. The places where the elements can be joined are places where power can flow between them; the places where power can flow in and out of an element are called ports. On each of the ports there are associated variables which are called power variables either effort or flow. The product of an effort variable and a flow variable is power. (example: force (flow) times velocity (effort) gives power.) Different elements have been designated to represent certain physical elements. (example: "I" elements store energy and can be used to represent springs or inductors.) The modeling elements are used to describe how the power of the overall system is affected by the interconnection of all the physical elements. One disadvantage of using bond graphs is that it is very difficult to model nonlinear systems. Where the constitutive relation in this system was nonlinear, the range of values of the variable was determined, and then the constitutive

relation was approximated as being linear over this range.

Therefore, the model was only good for a certain operating range. (For a more detailed description of the use of bond graphs see references 6,7.)

After the model is completed it can be analyzed to give a set of linear state equations for the system. The state variables are the variables which represent the energy storage level of the energy storage elements. (example: the state variable for an "I" element would be the flow variable.) From the state equations the system response is found.

The first step in modeling the system was to develop a bond graph model for each control valve. The input to each valve was a pressure from the controller; the valve is an air actuated, spring opposed, equal percentage, globe type control valve. The elastic elements of the valve were accounted for in the energy storing "C" element. The mass of the moving valve parts made up the inertia of the control valve and was included in the "I" element.

The values for the resistance of the bearings and bushings and the values for effective inertia and spring constant were determined using reasonable approximations. The system was then tested for several different inputs.

Using a program called PARASOL nearly any set of linear differential equations can be analyzed. Placing the differential equations for this system in the computer and using PARASOL to solve them provides curves showing the response of the valve. The tests showed the predicted response of the valve for the

given step inputs. See Appendix V for the bond graph and analysis; see Appendix VI for the results.

After the control valves were analyzed, a similar analysis of the steam supply and turbine system was performed. This system (see in Appendix V) included only the steam supply line and the turbine itself. In analyzing the system, it was easy to develop a large number of state variables. The number of state variables was reduced by lumping as many of the energy storing elements together as possible.

When reducing the number of state variables, there are always innaccuracies. There is a trade-off between accuracy and the amount of work necessary to obtain the results from the analysis. The model analyzed provides sufficient accuracy for the system being analyzed.

The energy storage elements include the inertia of the moving fluid, compressibility of the steam and inertia of the rotating elements. Some minor effects which were not included, were the torsional flexure of the shafts in the turbine and the dynamometer.

This bond graph is an adequate model for a nearly non-compressible fluid system, but for compressible thermofluid flow it is inadequate. The steam is compressible and as the pressure drops the volumetric flow rate increases, the energy is transferred from thermal to kinetic. As the bond graph now stands it does not show these effects. To try to model these effects a pseudo bond graph can be used.¹² It is called pseudo because the product of the effort and the flow variables is no

longer a pure power term but a pseudo power. An example of a possible model is shown in figure 5.3 of Appendix V. To avoid the tremendously more complicated pseudo bond graph model an incompressible model was utilized using some simplifying assumptions.

To remove the problem of representing the energy mode change of the steam from thermal to kinetic in the bond graph, the change in mode is calculated ahead of time. Knowing the actual amount of thermal energy and kinetic energy of the steam, the response of the system can be approximated. The system is modeled with the source conditions being that of the inlet to the turbine and with the compressibility and inertial affects of the fluid neglected. Therefore, the bond graph now consists of the input to the turbine and the turbine losses and the turbine and generator inertia. See figure 5.4 in Appendix 5.

This system is a extreme simplification which was made necessary because of the compressiblity and thermal properties of the steam. Even though it is a large simplification using the state equation and the known inlet condition the response of the turbine can be predicted. See Appendix V.

In order to get a total model for the system, the control valve, flow controller and turbine had to be connected. The model for the system is shown in figure 5.5 of Appendix V. This shows the model for the incompressible fluid flow model. The assumption of incompressible fluid flow is made again because of the difficulty in modeling compressible thermo-fluid systems. Figure 5.5 shows the control valve feedback

loop. The feedback loop consists of a part where there is no power flow. This part is represented by a block diagram rather than a bond graph. Because of the feedback loop, the state equations are nonlinear, but, in order to solve the state equations, the equations must be linearized about some operating point. The state variables are now deviations from the operating point.

To make a model for the system that would be operating with the Lysholm engine requires even more complicated modeling. Figure 5.6 shows the modeling of the hot water and steam piping in the incompressible system. The model shows the feedback loops for both control valves and how the ratio controller interacts to set the value of the water flow based on the steam flow. Again, to make a more accurate model of how the steam and hot water mix, psuedo bond graphs must be used.

11. RESULTS OF MODELING

Using the bond graph model the valve response and the turbine response were simulated. The purpose of the simulation was to determine what response the model predicts and to compare that with the real response. As indicated, the values for the parameters involved were estimated from observing the physical parts. The valve was first tested to determine how it responded to a given step input in pressure. The response of the valve with the estimated parameters was a very rapid response with a time constant less than one second.

The computer simulation of the turbine with the estimated value for the inertia and the resistance resulted in a response

similar to actual tests. The bond graph predicts a logarithmic response which is what was obtained in actual tests. The value of the resistance was obtained by predicting the losses due to leakage, windage, friction, blade efficiency and leaving losses. The simulation predicted a time constant of approximately 300 seconds for a 5 percent step change in steam flow. (See Appendix V).

From the two small computer simulations of the valve and the turbine, several responses were predicted by the simulation and verified by the actual tests. The simulation predicted a very fast response of the control valve. The real tests showed that there was no noticeable time lag in the valves response which agrees with the simulation. The simulation predicted a logarithmic response of the turbine speed and the actual tests again agreed. Both the computer simulation and the actual tests on the turbine gave similar time constants. The simulation also showed that the dominant factor affecting the response of the system (the dominant pole), is the inertia of the turbine. More time could be spent on the modeling process and a model for the whole system could be made which includes the thermodynamic changes and feedback loops, but the work done gives the necessary information.

12. DISCUSSION OF MODELING

In using bond graphs to model a real system, it is evident that there are many problems that must be faced. One of the major problems faced in modeling engineering systems is the fact that almost all systems are nonlinear. Therefore, the

bond graph model is only usable for a range of responses around the equilibrium point. The most significant problem in using bond graphs is in their application for compressible thermo-fluid systems. Bond graphs in their pure form are entirely inadequate for modeling these systems. To model the system, pseudo bond graphs must be used which present an unreasonable amount of work for the size and sophistication of this system which was analyzed. For the purpose of this report it is evident that it is not reasonable to go into a lot of detail in modeling the system when by careful engineering judgment, it is obvious as to what the response of the system will be. However, what the bond graphs have shown is that by using them one can get an approximation of what are the state variables and what are the state equations. Then the engineer can visualize what are the important variables of the system and how these variables relate to each other. From the additional insights into the system the engineer can make a reasonable estimate of the behavior of the system.

13. CONCLUSIONS

The following conclusions were reached as a result of the project.

1. The two-phase fluid mixing system provides a flow similar to the flow from a geothermal well.
2. The system provides a stable flow of a given quality over a wide range of mass flow rates.
3. Response of the fluid system to changes in set point is very fast, and shows no tendency to be unstable.
4. The dominant factor in the speed of response of the

turbine is the inertia of the turbine and generator.

5. Projected speed of response of the Lysholm engine is about half that of the turbine. The speed response curve should follow a logarithmic path for a given step input.

6. Bond graphs provide a means of determining the important state variables. However, bond graphs appear to be too complicated for use in simple compressible thermofluid systems.

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FIGURES

1. Control System Schematic
2. Hesse Hall Layout
3. Overall System
4. Steam and Water Mixing
5. Pictures of Equipment

Figure 1

Control System Schematic

Figure 2
 HESSE HALL - ROOM 120 - STEAM SUPPLY PIPING
 (MECHANICS BUILDING) (1/4" = 1')

Figure 3
Overall System

THERMOCOUPLE PROBE LOCATIONS

- 1) INLET STEAM CALORIMETER
- 2) SEPARATOR CALORIMETER
- 3) SEPARATOR STEAM
- 4) LYSHOLM EXHAUST
- 5) LYSHOLM INLET
- 6) HOT WATER
- 7) STEAM INPUT

NOTES:

- 1 PRESSURE REGULATING VALVE 1" IT SHOULD BE PLACED SO THAT IT IS EASILY ADJUSTED
- 2 THERMOCOUPLE LOCATIONS ARE DESIGNATED BY NUMBERS INSIDE TRIANGLES Δ .
- 3 ALL SYMBOLS $\bigcirc$ P ARE BOURDON TYPE PRESSURE GAUGES.

SCALE 1" = 1'

COLD WATER
HOT WATER
STEAM
CONDENSATE
AIR

Figure 4

Steam and Water Mixing

Figure 5

Westinghouse Steam Turbine

Lysholm Engine

Separator and Steam Piping

Steam and Water Control Valves

Steam Control Valve and Orifice-foreground
Differential Pressure Transmitters-behind

Pump and Heat Exchanger

Lysholm System

Control Panel
Flow Controllers - Center
Input Pressure - Left Center
Output Pressure - Right Center
RPM Meter - Lower Center

APPENDIX I

FLOW REGIME CALCULATIONS

The flow regime is determined by using a flow regime map. Two different maps are used to predict the flow regime for the given operating conditions.

The first flow regime map is figure 1.1 from reference 4 page 8. This flow regime map plots the modified gas phase mass flux versus the modified gas phase and liquid phase mass flux ratio.

Parameters

- G_L - mass flux liquid
- G_G - mass flux gas
- ρ_G - density of gas
- ρ_L - density of liquid
- ρ_A - density of air at atmospheric pressure
- ρ_w - density of water at atmospheric pressure
- σ_w - surface tension of water at atmospheric pressure
- σ_L - surface tension of liquid
- μ_L - viscosity of liquid
- μ_w - viscosity of water at atmospheric pressure

$$\textcircled{1} \lambda_B = \left[\left(\frac{\rho_G}{\rho_A} \right) \left(\frac{\rho_L}{\rho_w} \right) \right]^{\frac{1}{2}}$$

$$\textcircled{2} \psi_B = \left(\frac{\sigma_w}{\sigma_L} \right) \left[\frac{\mu_L}{\mu_w} \left(\frac{\rho_w}{\rho_L} \right)^2 \right]^{\frac{1}{3}}$$

Value of Parameters

Operating Pressure = 110 psig

$$\rho_A = 0.00237 \text{ slug/ft}^3$$

$$\rho_G = 0.00767 \text{ slug/ft}^3$$

$$\rho_w = 1.94 \text{ slug/ft}^3$$

$$\rho_L = 1.74 \text{ slug/ft}^3$$

$$\mu_w = 271.4 \times 10^{-7} \frac{\text{lb} \cdot \text{sec}}{\text{ft}^2}$$

$$\sigma_w = 73.5 \frac{\text{dyne}}{\text{cm}}$$

$$\mu_L = 34.3 \times 10^{-7} \frac{\text{lb} \cdot \text{sec}}{\text{ft}^2}$$

$$\sigma_L = 58 \frac{\text{dyne}}{\text{cm}}$$

Substituting into equation 1

$$\lambda_B = \left[\left(\frac{0.00767}{0.00237} \right) \left(\frac{1.74}{1.94} \right) \right]^{0.5} = 1.705$$

Substituting into equation 2

$$\psi_B = \left(\frac{73.5}{58} \right) \left[\frac{34.3}{271.4} \left(\frac{1.94}{1.74} \right)^2 \right]^{\frac{1}{3}} = 0.684$$

Input area of pipe

2 inch schedule 40 - 2.067 I.D.
area = .0233 sq. ft.

Choose flow rate

2 lb/sec 20% quality
.4 lb/sec steam, 1.6 lb/sec water

$$G_G = 61,800 \frac{\text{lb}}{\text{hr ft}^2} \quad G_L = 2.47 \times 10^5 \frac{\text{lb}}{\text{hr ft}^2}$$

$$\frac{G_G}{\lambda_B} = \frac{61,800}{1.705} = 36,400$$

$$\frac{G_L \lambda_B \psi_B}{G_G} = \frac{2.47 \times 10^5 (1.7) (.684)}{36,400} = 7.897$$

Using these values in figure 1.1 locates a point within the annular mist region.

To double check this prediction the second flow regime map is used. This map is shown in figure 1.2.

Parameters

- Q_L - volumetric flow rate of liquid
- Q_G - volumetric flow rate of gas
- d_o - equivalent diameter
- A - cross-sectional area
- g - acceleration due to gravity

$$Q_L = 1.63 \text{ ft}^3/\text{sec}$$

110 psig

$$Q_G = .0285 \text{ ft}^3/\text{sec}$$

$$A = .0233 \text{ ft}^2$$

$$d_o = .1717 \text{ ft}$$

$$g = 32.2 \text{ ft}/\text{sec}^2$$

$$\left(\frac{Q_L + Q_G}{A} \right)^2 / g d_o = \left(\frac{1.63 + .0285}{.0233} \right)^2 / 32.2 \times .1717 = 944$$

$$\frac{Q_G}{Q_L + Q_G} = \frac{.0285}{1.63 + .0285} = .983$$

These values also give a point within the annular mist region of the map. Thus, both methods predict annular mist flow in the pipe.

Figure 1.3 shows another flow regime map for water nad air mixture. Table 1.1 shows how the size and number of droplets and film thickness changes for the different types of flows.

Parameters

V_{SL} - superficial velocity of liquid
 V_{SG} - superficial velocity of gas

$$V_{SL} = \frac{Q_L}{A} = \frac{1.63 \text{ ft}^3/\text{sec}}{.0233 \text{ ft}^2} = 70.0 \text{ ft}/\text{sec}$$

$$V_{SG} = \frac{Q_G}{A} = \frac{.0285 \text{ ft}^3/\text{sec}}{.0233 \text{ ft}^2} = 1.22 \text{ ft}/\text{sec}$$

FIG. 2.3. Flow pattern diagram for horizontal flow. (After Baker, 1954, as modified by Scott, 1963.)

Figure 1.1

FIG. 2.5. Flow pattern diagram suggested by Griffith and Wallis (1961) for vertical upwards flow.

Figure 1.2

Figure 8.1. Flow patterns for air-water mixtures flowing in a 1.025-inch pipe based on observations of Govier, Radford, and Dunn, 1957, and calculations.

Figure 1.3

TABLE 2.1
Flow Pattern Details for Air-Water Mixture Flowing in
1.025 Inch I.D. Vertical Pipe (See Figure 8.1)

Sketch Identi- fication	Flow Pattern	Continuous Phase	Superficial Velocity		Volume Fraction Air, E_g	Holdup Ratio, H	Slip Velocity, S ft/sec
			Water, V_{sz} ft/sec	Air, V_{sc} ft/sec			
A	bubble	water	0.1	0.1	0.47	1.15	0.03
B	bubble	water	0.1	0.5	0.67	2.50	0.45
C	slug	water	0.1	1.0	0.70	4.30	1.10
D	slug	water	0.1	3.5	0.75	11.6	4.26
E	froth	neither	0.1	10.0	0.77	29.0	12.5
F	annular mist	air	0.1	50.0	0.96	20.5	49.5
G	annular mist	air	0.1	100.0	0.988	12.5	93.1
H	bubble	water	1.0	0.1	0.09	1.05	0.05
I	bubble	water	1.0	0.5	0.294	1.20	0.28
J	slug	water	1.0	1.0	0.43	1.35	0.61
K	slug	water	1.0	3.5	0.62	2.15	3.02
L	froth	neither	1.0	10.0	0.69	4.40	11.1
M	annular mist	air	1.0	50.0	0.877	7.0	48.9
N	annular mist	air	1.0	100.0	0.93	7.0	91.7
O	bubble	water	10.0	0.1	0.01	1.01	0.10
P	bubble	water	10.0	0.5	0.04	1.1	1.04
Q	bubble	water	10.0	1.0	0.08	1.13	1.41
R	slug	water	10.0	5.0	0.27	1.35	4.80

Approximate Only

Sketch Identi- fication	Flow Pattern	Diameter of Bubbles or Droplets, d_p , in.	Number of Bubbles or Droplets, n_p^a	Average Film Thickness, δ_{aw} , in.	Length of Liquid Slug, L_s , in.	Length of Taylor Bubble, L_b , in.
A	bubble	0.25	540	—	—	—
B	bubble	0.25	780	—	—	—
C	slug	0.25	350	—	8	18
D	slug	0.2-0.25	150	—	8	32
E	froth	0.2-0.5	30	0.025	—	—
F	annular mist	—	—	0.08	—	—
G	annular mist	0.027	2100	0.0025	—	—
H	bubble	0.25	100	—	—	—
I	bubble	0.25	350	—	—	—
J	slug	0.25	400	—	3	0.6
K	slug	0.2-0.25	70	—	8	16
L	froth	0.2-0.5	80	0.035	—	—
M	annular mist	—	—	0.015	—	—
N	annular mist	0.05	1300	0.015	—	—
O	bubble	0.25	12	—	—	—
P	bubble	0.25	50	—	—	—
Q	bubble	0.25	95	—	—	—
R	slug	0.25	45	—	8	3.7

^a Per foot of 1-inch pipe; in the slug flow pattern it refers to the small bubbles in the liquid slug only.

Table 1.1

APPENDIX II

Control Valve Response Data and Curves

WATER CONTROL VALVE RESPONSE

RUN	% OVERSHOOT	SETTLING TIME (sec)	STEP SIZE	PROPORTIONAL GAIN %	INTEGRAL GAIN SEC/REPEAT	NOTES OSCILLATIONS
1	49	5.5	44.4mV	75	.037	Small
2	80	5.2	25.6	75	.037	Small
3	34	4.3	53.6	75	.037	Small
4	15.4	9.3	103.6	75	.037	Large
5	49	11.4	52.4	75	.075	Small
6	110	7.8	50.8	75	.02	small
7	0	12.9	52.8	150	.037	Very small
8	0	18.8	50.4	150	.075	Very small
9	159	6.25	50.8	50	.037	large

Settling Time : No peaks over 10% of final value.

Step size : $100 \text{ mV} = 10\% = .556 \text{ 1/sec}$

Valve Response Curves

APPENDIX III

Parallel Flow Response Data and Curves

WATER & STEAM VALVE RESPONSE

RUN	STEP CHANGE STEAM	SETTLING TIME WATER	SETTLING TIME STEAM	TIME CONSTANT WATER	TIME CONSTANT STEAM	RATIO
1	+ 10%	14.7 sec	12.2 sec	5 sec	1.8 sec	1
2	- 10%	17.5	14.4	4.3	1.5	1
3	+ 5%	9.6	8.5	2.5	1.8	.7
4	- 10%	11.8	10.2	3.6	.9	.7

Step Change : 10% = .05 lb/sec - Steam

YORK

No. 9280-0258

RECORDING CHARTS

GRAPHIC CONTROLS CORPORATION BUFFALO, N.Y.

APPENDIX IV

Steam and Turbine Test Results

STEAM AND TURBINE RESPONSES

RUN	STEAM CHEST	CONTROLLER READING	OUTPUT READING	FLOW AUTODATA	RPM AUTODATA	LOAD
1	BEFORE ~3.5 psig	~ 25 %	~ 7	2.539 Volt	3.694 Volt	0
	AFTER ~ 6 psig	~ 35 %	~ 12	3.605 Volt	5.53 Volt	
2	~ 6 psig	~ 35	12	3.607	5.73	0
	~ 8 psig	~ 40	14	4.111	8.425	
3	~ 9	~ 20	5	2.019	4.25	0
		~ 35	OVERSPEED			
4	~ 7	~ 40	12	4.048	5.033	10 %
	~ 8.5	~ 45	15	4.612	6.233	
5	4.5	25	3.5	2.56	3.70	0
	6.0	35	7	3.504	4.815	
6	6.0	35	7	3.504	4.827	0
	8.0	40	13	4.042	6.76	
7	8.0	40	11	4.022	5.001	10 %
	9.5	45	14	4.53	6.012	
8	21.5	70	30	7.075	5.75	80 %
	24	75	36	7.618	7.9	

STEP CHANGE SIZE	RUN	TIME CONSTANT	LOAD	C
5%	2	170sec	No	.94
5	4	120sec	Yes	.89
5	6	235sec	No	.903
5	7	140sec	Yes	.942
5	8	150sec	Yes	
10	1	250sec	No	.95
10	5	190sec	No	
15	3	OVERSPEED		

C - correlation coefficient of data fitted to a logarithmic response curve.

Turbine Speed vs. Time

Turbine Speed vs. Time

$$\dot{m}_0 = 2.539 \text{ V} \quad \omega_0 \approx 3.694$$

$$\dot{m}_f = 3.605 \text{ V} \quad \omega_{ss} \approx 5.53$$

$$\Delta \dot{m} = 1.066 \text{ V} \quad \Delta \omega = 1.837$$

$$\tau \approx 255 \text{ sec}$$

$$A_0 = 1.847$$

RUN # 1

Time (sec)	Steam Flow Volts	Steam Flow lb/sec	Turbine Speed Volts	ΔV
0	2.539	$\dot{m}_{\text{measured}}$	3.694	0
64	3.663	$= .131$	3.755	
80	3.585	$\dot{m}_{\text{calc}} = .127$	3.896	.2016
94	3.554		3.978	
110	3.604	$\dot{m}_{\text{calc}} = .180$	4.009	.315
125	3.622	$\dot{m}_{\text{meas}} = .180$	4.104	
140	3.621		4.263	.569
155	3.612		4.362	
170	3.589		4.397	.703
185	3.610		4.501	
200	3.605		4.531	.837
215	3.627		4.636	
230	3.620		4.686	
245	3.613		4.798	1.104
260	3.584		4.829	
290	3.608		4.946	1.252
320	3.608		5.011	
350	3.591		5.060	1.366
380	3.596		5.155	
410	3.603		5.207	1.513
440	3.625		5.247	
470	3.639		5.295	1.601
500	3.611		5.356	
530	3.621		5.351	1.657
560	3.596		5.393	
590	3.627		5.439	
620	3.545		5.446	1.752
680	3.591	53	5.470	
740	3.597		5.509	
800	3.624		5.545	1.851
860	3.590		5.538	

RUN #2

$\dot{m}_0 = 3.607V$ $W_0 \approx 5.73V$
 $\dot{m}_f = 4.111V$ $W_{ss} = 8.425V$
 $\Delta \dot{m} = .504V$ $\Delta W = 2.695V$

$\tau \approx 170 \text{ sec}$

Time (sec)	Steam Flow Volts	Steam flow lb/sec	Turbine Speed Volts	ΔV	
0	3.607	$\dot{m}_{meas} = .180$	5.730		
5	4.135	$\dot{m}_{calc} = .180$	5.729		
10	4.108		5.782		
15	4.112	$\dot{m}_{meas} = .202$	5.789		
30	4.090	$\dot{m}_{calc} = .206$	5.883	.153	
45	4.084		6.040		
60	4.112		6.147	.417	
75	4.151		6.294		
90	4.125		6.449	.719	
105	4.148		6.597		
120	4.118		6.732	1.002	
135	4.125		6.877		
150	4.083		7.005	1.275	
165	4.099		7.127		
180	4.126		7.236	1.506	
195	4.111		7.468		
210	4.059		7.535	1.806	
225	4.098		7.589		
255	4.094		7.679	1.949	
285	4.097		7.756		
315	4.107		7.842	2.112	
345	4.111		7.897		
375	4.104		7.977		
405	4.09		8.026	2.296	
435	4.097		8.065		
465	4.109		8.111		
495	4.147		8.136		
525	4.11		8.176	2.446	
555	4.138		8.211		
585	4.095		8.261		
615	4.088		8.284		
645	4.138		8.32		
675	4.122		8.352		
705	4.12		8.38		
735	4.107		8.414		
765	4.129		8.434		

1065 4.116 8.438
 1125 4.145 8.416
 1185 4.095 8.421

$\bar{V} = 8.425$

RUN #3

Time (sec)	Steam Flow volts	Steam Flow 1/sec	Turbine Speed volts	Turbine Speed RPM
0	2.019		4.25	
30	3.636		4.397	
60	3.535		4.759	
90	3.57		5.24	
120	3.57		5.84	
150	3.58		6.603	
180	3.616		7.46	
210	3.553		8.31	
240	3.608		8.89	
270	3.561		9.272	
Governor Activated				

RUN #4

$$\begin{aligned} m_0 &= 4.048V \\ w_f &= 4.612V \\ \Delta m &= .564V \end{aligned}$$

$$\begin{aligned} w_0 &= 5.033V \\ w_f &= 6.233V \\ \Delta W &= 1.2V \end{aligned}$$

$$\tau \approx 120 \text{ sec}$$

Time (sec)	Steam Flow Volts	Steam Flow lb/sec	Turbine Speed Volts	Turbine speed RPM	ΔV
0	4.048		5.033		0
30	4.634		5.183		.15
60	4.625		5.429		.396
90	4.632		5.659		.626
120	4.617		5.803		.77
150	4.608		5.90		.867
180	4.607		5.982		.949
210	4.620		6.068		
240	4.589		6.119		1.086
270	4.589		6.136		
300	4.624		6.148		
330	4.648		6.19		1.157
360	4.59		6.225		
420	4.598		6.231		
450	4.625		6.23		
480	4.638		6.233		
520	4.603		6.251		1.2
550	4.638		6.271		
580	4.598		6.249		
610	4.608		6.221		
640	4.591		6.236		
670	4.562		6.225		

$$\dot{m}_0 = 2.56 \text{ V}$$

$$\dot{m}_f = 3.504 \text{ V}$$

$$\omega_0 = 3.70 \text{ V}$$

$$\omega_f = 4.815 \text{ V}$$

RUN #5

$$\Delta \dot{m} = .944 \text{ V}$$

$$\Delta \omega = 1.115 \text{ V}$$

$$\tau = 191 \text{ sec}$$

Time (sec)	Steam Flow volts	Turbine Speed volts	ΔV
0	2.56	3.70	
30	3.52	3.709	
60	3.479	3.918	.218
90	3.496	4.046	.346
120	3.501	4.24	.54
150	3.495	4.269	.569
180	3.53	4.354	.654
210	3.51	4.48	.78
240	3.525	4.554	.854
270	3.47	4.538	
300	3.496	4.567	.867
330	3.509	4.595	
360	3.504	4.609	.909
390	3.485	4.636	
420	3.529	4.666	.966
450	3.503	4.656	
480	3.522	4.687	.987
510	3.445	4.663	
540	3.525	4.693	.993
570	3.517	4.719	
600	3.498	4.756	1.056
630	3.501	4.741	
660	3.509	4.750	
690	3.507	4.743	
720	3.506	4.770	
750	3.526	4.787	
780	3.503	4.823	1.115
810	3.522	4.835	
840	3.516	4.783	
870	3.537	4.805	
900 +		4.827	

RUN #6

$$\begin{aligned} m_i &= 3.504 \text{ V} \\ m_f &= 4.042 \text{ V} \\ \Delta m &= .54 \text{ V} \end{aligned}$$

$$\begin{aligned} w_o &= 4.827 \text{ V} \\ w_f &= 6.76 \text{ V} \\ \Delta w &= 1.933 \text{ V} \end{aligned}$$

$$\tau = 235 \text{ sec.}$$

Time (sec)	Steam Flow Volts	Turbine Speed Volts	ΔV		
0	3.504	4.827	0		
30	4.059	4.955			
60	4.035	5.179	.352		
90	4.029	5.382	.555		
120	4.034	5.531	.704		
150	4.054	5.695	.868		
180	4.069	5.838	1.011		
210	4.049	5.958			
240	4.019	6.068	1.241		
270	4.054	6.131			
300	4.016	6.183	1.356		
330	4.054	6.273			
360	4.080	6.325	1.498		
390	4.014	6.344			
420	4.048	6.372	1.545		
450	4.032	6.407			
480	4.034	6.435	1.608		
510	4.043	6.465			
540	4.032	6.498			
570	4.050	6.505	1.678	900	3.989 6.676
600	4.066	6.512	930	4.062	6.706
630	4.066	6.547	960	4.082	6.709
660	4.048	6.534	1.707	990	4.070 6.692
690	4.050	6.543	1020	4.046	6.709
720	4.028	6.587	1050	4.085	6.739
750	4.030	6.561	1080	4.035	6.719
780	4.047	6.605	1120		6.758
810	4.031	6.619			6.740
840	4.036	6.650	1.823		6.761
870	4.040	6.662			6.757
					6.769

RUN #7

$m_o = 4.022 \text{ V}$
 $m_f = 4.53 \text{ V}$
 $\Delta m = .508 \text{ V}$

$w_o = 5.001 \text{ V}$
 $w_f = 6.012 \text{ V}$
 $\Delta w = 1.002 \text{ V}$

$\tau \approx 140 \text{ sec}$

Time (sec)	Steam Flow Volts	Turbine speed Volts	ΔV
0	$\bar{v} = 4.022$	$\bar{v} = 5.001$	
30	4.56	5.107	
60	4.542	5.272	.271
90	4.558	5.454	.453
120	4.565	5.542	.541
150	4.457	5.678	
180	4.524	5.732	.731
210	4.566	5.795	.794
240	4.526	5.825	.824
270	4.532	5.852	
300	4.527	5.886	.885
330	4.495	5.925	
360	4.542	5.903	.902
390	4.562	5.923	
420	4.542	5.98	.979
450	4.541	5.946	
480	4.532	5.989	.989
510	4.510	5.978	
540	4.557	5.983	.982
570	4.546	5.988	
600	4.551	6.013	1.012
630	4.523	5.999	
660	4.543	6.008	1.007
690	4.555	6.023	
720	4.537	6.011	1.01
750	4.513	6.018	
	$\bar{v} = 4.534 \text{ V}$	6.036	1.035
	$\delta = .025 \text{ V}$	6.023	
		6.006	1.005
		6.00	
		6.025	
		6.002	

RUN #8

$$m_f = 7.618 V$$

$$\Delta m = .543 V$$

$$u_f \approx 7.9 V$$

$$\Delta W \approx 2.25 V$$

$$\tau \approx 150 \text{ sec}$$

Time (sec)	Steam Flow Volts	Turbine Speed Volts	ΔV
0	$\bar{V} = 7.075$	$\bar{V} = 5.75 V$	
30	7.628	6.039	.289
60	7.590	6.351	.605
90	7.635	6.652	.902
120	7.635	6.915	1.165
150	7.640	7.114	1.364
180	7.646	7.246	1.496
210	7.627	7.34	1.59
240	7.595	7.44	
270	7.635	7.504	1.754
300	7.591	7.535	
330	7.626	7.595	1.845
360	7.616	7.634	
390	7.603	7.673	1.923
420	7.612	7.695 _{.028}	
450	7.619	7.723 _{.029}	
480	7.604	7.761 _{.009}	2.011
510	7.632	7.770 _{.063}	
540	7.596	7.773 _{.020}	
570	7.617	7.793 _{.002}	2.043
600	7.615	7.795 _{.125}	
630	7.626	7.826 _{.026}	
660	7.645	7.846 _{.016}	2.096
690	7.615	7.862	
		Projected $\approx 7.9 V$	
		60	

APPENDIX V

Figure Bond Graphs and State Equations

- 5.1 Valve Bond Graph
- 5.2 Steam Supply and Turbine Bond Graph
- 5.3 Steam Supply and Turbine Pseudo Bond Graph
- 5.4 Simplified Steam and Turbine Bond Graph
- 5.5 Steam Supply and Turbine Total System Bond Graph
- 5.6 Hot Water, Steam and Turbine Total System Bond Graph

Figure 5.1
VALVE BOND GRAPH

State Equations:

$$\dot{e}_y = \frac{1}{C_y} \left(s_{e,r} - \frac{e_y}{R_3} - f_7 \right)$$

$$\dot{f}_7 = \frac{e_y}{I_7}$$

$$\dot{x}_1 = e_y$$

$$I = 1/K$$

$$C = M$$

K - spring constant

M - mass of moving parts

R - resistance to motion

GY - gyrator: converts pressure to a force

Figure 5.2
STEAM SUPPLY AND TURBINE BOND GRAPH
(incompressible hydraulic system)

State Equations:

$$\dot{e}_y = \frac{Se_1}{R_2 C_y} - \frac{e_y}{R_2 C_y} - \frac{f_6}{C_y}$$

$$\dot{f}_6 = \frac{e_y}{I_6} - \frac{f_6 R_7}{I_6} - \frac{A}{I_6 R_{10}} (f_6 A - e_{12})$$

$$\dot{e}_{12} = \frac{f_6 A}{R_{10} C_{12}} - \frac{e_{12}}{R_{10} C_{12}}$$

R_2 - valve resistance

C_y - compressibility

I_6 - fluid inertia

R_7 - piping losses

R_{10} - turbine losses

C_{12} - rotary inertia

STEAM SUPPLY AND TURBINE PSEUDO BOND GRAPH
(adiabatic compressible thermalfuid system)

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Figure 5.4
SIMPLIFIED STEAM TURBINE BOND GRAPH

State Equation

$$\dot{e}_5 = \frac{-e_5}{R_3 C_5} + \frac{S_F A}{R_3 C_5}$$

R - turbine losses

C - inertia of turbine

S_f - flow source

GY - gyrator: converts fluid power to mechanical power.

Figure 5.5

STEAM SUPPLY AND TURBINE TOTAL SYSTEM BOND GRAPH
(incompressible hydraulic system)

State Equations: (nonlinear)

$$\dot{e}_4 = -\frac{e_4}{C_1 X_1 C_4} - \frac{f_6}{C_4} + \frac{Se_1}{C_1 X_1 C_4}$$

$$\dot{f}_6 = \frac{e_4}{I_6} - \frac{f_6 R_7}{I_6} - \frac{A}{I_6 R_{10}} (f_6 A - e_{12})$$

$$\dot{e}_{12} = \frac{f_6 A}{R_{10} C_{12}} - \frac{e_{12}}{R_{10} C_{12}}$$

$$\dot{X}_{C1} = P_{set1} - \frac{(Se_1 - e_4) G_1}{C_1 X_1}$$

$$\dot{e}_{21} = \frac{1}{C_{21}} (Se_{19} C_{R2} - \frac{e_{21}}{R_{23}} - f_{22}), \quad Se_{19} = \frac{K_{c1} X_{C1}}{T_1} + K_{c1} \dot{X}_{C1}$$

$$\dot{f}_{22} = \frac{e_{21}}{I_{22}}$$

$$\dot{X}_1 = e_{21}$$

Figure 5.6

Hot Water, Steam and Turbine Total System Bond Graph
(incompressible hydraulic system)

APPENDIX VI

Simulation Results

1. Valve Simulation
2. Turbine Simulation

Valve Simulation (from valve bond graph)

Valve Parameters:

effective area of diaphragm

$$(3^2 - .75^2)\pi = 28.3 \text{ in}^2 = .196 \text{ ft}^2$$

spring constant,

total deflection .75 in. at
15 psi.

force = K*deflection

$$\frac{15 \text{ lb/in}^2 \times 28.3 \text{ in}^2}{.75 \text{ in}} = K = 566 \frac{\text{lb}}{\text{in}} = 6790 \frac{\text{lb}}{\text{ft}}$$

S_e - pressure input to valve

M - mass of moving parts
diaphragm and shaft approximately 11b.

M = .031 slug

1/R - resistance due to bushings proportional to the
velocity. Teflon bushings are used so the
resistance is small.

Value of Parameters for Simulation

LR = .196

$S_e = 140 \text{ lb/ft}^2$

M = .031 slugs

K = 6790 lb/ft

R = 1 & 10 ft/lb sec

Results

Test #1

R = 1

settling time = .4 sec

Test #2

R = 10

settling time = 4 sec

The valves response is very rapid. The amount of resistance has
a significant affect on the settling time the most probable value
for R is approximately 1 ft/lb sec which gives almost no
oscillation and a settling time less than .5 seconds.

Valve Position vs Time

Valve Bond Graph Simulation PARASOL Equations

```
#F7= E4 K *
#E4= F7 $CHS M / E4 R / M / - SE1 LR * M / +
K= 6790.
M= 0.3000E-01
R= 1.000
SE1= 140.0
LR= 0.1960
#X1= E4
END
```

$$R = 1.0$$

0 to .1 sec.

```
>>RUN
DT:.0001
TMAX:.1
ODE MODE
```

```
0.000E-01
6.635E-03
4.017E-03
1.878E-03
6.935E-03
2.028E-03
4.265E-03
5.448E-03
1.984E-03
5.584E-03
3.734E-03
3.143E-03
5.494E-03
2.872E-03
4.365E-03
4.601E-03
3.024E-03
4.919E-03
3.737E-03
3.703E-03
4.749E-03 .1
```

0 to .25 sec.

```
>>RUN
DT:.0005
TMAX:.25
ODE MODE
```

```
0.000E-01
2.029E-03
3.735E-03 .85
4.602E-03
4.749E-03 .1
4.523E-03
4.227E-03 .15
4.023E-03
3.941E-03 .2
3.947E-03
3.988E-03 .25
```

0 to .5 sec.

```
>>RUN
DT:.001
TMAX:.5
ODE MODE
```

```
0.000E-01
3.752E-03
4.751E-03 .15
4.218E-03
3.938E-03 .2
3.990E-03
4.052E-03 .35
4.053E-03
4.041E-03 .4
4.039E-03
4.041E-03 .5
```

R = 10.0

0 to 1 sec.

```
>>IN
R=10
END
>>RUN
DT:.001
TMAX:1
ODE MODE
```

0.000E-01
3.261E-03 .05
7.130E-03 .1
5.909E-03
2.179E-03 .2
1.741E-03
4.703E-03 .3
6.233E-03
4.345E-03 .4
2.319E-03
3.110E-03 .5
5.123E-03
5.248E-03 .6
3.606E-03
2.856E-03 .7
3.942E-03
4.998E-03 .8
4.500E-03
3.418E-03 .9
3.411E-03
4.318E-03 1

0 to 2 sec.

```
>>RUN
DT:.001
TMAX:2
ODE MODE
```

0.000E-01
7.130E-03
2.179E-03
4.703E-03
4.345E-03
3.110E-03
5.248E-03
2.856E-03
4.998E-03
3.418E-03
4.318E-03 1
4.059E-03
3.819E-03 1.2
4.368E-03
3.699E-03 1.4
4.333E-03
3.838E-03 1.6
4.146E-03
4.025E-03 1.8
3.992E-03
4.128E-03 2

0 to 4 sec.

```
>>RUN
DT: /\. /\. .001
TMAX:4
ODE MODE
```

0.000E-01
2.179E-03
4.345E-03
5.248E-03
4.998E-03
4.318E-03
3.819E-03
3.699E-03
3.838E-03
4.025E-03
4.128E-03
4.129E-03
4.078E-03
4.032E-03
4.014E-03
4.021E-03
4.036E-03
4.047E-03
4.049E-03
4.045E-03
4.041E-03

Turbine Simulation (from turbine bond graph figure 5.4)

Input Conditions: (run #1)

$m = .18 \text{ lb/sec}$ saturated steam
pressure in steam chest = 5 - 7 psig

Exhaust Conditions:

$m = .18 \text{ lb/sec}$
pressure = 14.7 psia

inlet enthalpy $h_1 = 1157.1 \text{ btu/lb}_m$ at 21 psia

exhaust enthalpy assuming isentropic expansion

$h_{2s} = 1131.6 \text{ btu/lb}_m$ at 14.7 psia

The change in enthalpy for an isentropic expansion from 1 to 2s is:

$h_{12s} = 25.5 \text{ btu/lb}_m$

For the ideal expansion the available power output is:

3568. ft lb/sec or 4.78 KW

The losses of the turbine are lumped together into one R element.

The losses are estimated to be approximately 3 KW.

Therefore the net power output of the turbine is 1.78 KW.

The steady state speed was recorded as 3100 RPM or 330 rad/sec.

The preceding information is now used to determine the value for R.

$$S_f = .18 \text{ lb/sec} * 19.4 \text{ ft}^3/\text{lb} = 3.5 \text{ ft}^3/\text{sec}$$

$$C = 4 \text{ slug ft}^2$$

$$e_5 = 330 \text{ rad/sec}$$

$$P_t = \text{total power } 3568 \text{ ft lb/sec (4.78 KW)}$$

Equations from the bond graph figure 5.4.

$$1) f_2 = f_3 = f_5 = \tau$$

$$2) e_2 = e_3 + e_5$$

$$3) S_f * A = e_2$$

$$4) e_3 = f_4 * R$$

The differential equation is: $\dot{e}_5 = \frac{-e_5}{RC} + \frac{S_f A}{RC}$

to find the transfer function I used $G(s) = C(sI - A)^{-1}B$

$$(sI - A) = (s + \frac{1}{RC}) \quad (sI - A)^{-1} = \frac{1}{s + \frac{1}{RC}}$$

$$C = \frac{A}{RC}$$

$$G(s) = \frac{A/RC}{s + \frac{1}{RC}} = \frac{e_5}{S_f}$$

step input $S_f = \frac{F}{s}$

$$e_5(s) = \frac{\frac{FA}{RC}}{s(s + \frac{1}{RC})} = \frac{K_1}{s} + \frac{K_2}{s + \frac{1}{RC}}$$

$$K_1 = FA$$

$$K_2 = -FA$$

taking the inverse Laplace gives:

$$e_5(t) = FA(1 - e^{-t/RC})$$

The steady state value is $F * A$.

$F * A$ must equal 330 rad/sec, therefore $F = 3.5$ and $A = 94.3$

R can now be obtained knowing that the losses due to the torque equal 3 KW.

$$\text{losses} = 2238 \text{ ft lb/sec} = \tau^2 R$$

$$\tau_o = \frac{S_f A}{R}$$

$$\left(\frac{S_f A}{R}\right)^2 = \frac{(330)^2}{R} = 2238$$

$$R = 49 \text{ 1/sec ft lb}$$

The values of the parameters are:

$$S_f = 3.5 \text{ ft}^3/\text{sec}$$

$$A = 94.3$$

$$R = 49 \text{ l/sec ft lb}$$

$$C = 4 \text{ slug ft}^2$$

The value for R is only usable for this mass flow rate and speed because the resistance changes as the velocity changes.

Since this is a first order system the time constant is the product of R and C.

$$R * C = 4 * 49 = 196 \text{ seconds}$$

$$\text{measured time constant \#1} = 250 \text{ seconds}$$

Steam Turbine Bond Graph Simulation

PARASOL Equations

```

RUN $PARASOLTX
>>RFILE
FILE NAME HELP1.PAR
>>IN
>>IN
A=94.3
R3=49
END
>>LI
#E5= SF1 A * R3 / C5 / E5 R3 / C5 / -
SF1= 3.500
A= 94.30
R3= 49.00
C5= 4.000
Y= E5
END

```

0 to 300 sec.

```

>>RUN
DT:1
TMAX:300
ODE MODE

```

```

0.000E-01
2.432E+01
4.684E+01
6.771E+01
8.704E+01
1.049E+02
1.215E+02
1.369E+02
1.511E+02
1.643E+02
1.765E+02
1.878E+02
1.983E+02
2.080E+02
2.170E+02
2.253E+02
2.330E+02
2.402E+02
2.468E+02
2.529E+02
2.586E+02
>>STOP

```

0 to 800-sec.

```

>>PRINT
VARIABLES: Y
INTERVAL: 40
>>RUN
DT:1
TMAX:800
ODE MODE

```

```

0.000E-01
6.093E+01
1.106E+02-80
1.511E+02
1.842E+02-160
2.111E+02
2.330E+02-240
2.510E+02
2.656E+02-320
2.775E+02
2.872E+02-400
2.951E+02
3.015E+02-480
3.068E+02
3.111E+02-560
3.146E+02
3.174E+02-640
3.198E+02
3.217E+02-720
3.232E+02
3.245E+02-800

```

Turbine Speed vs. Time