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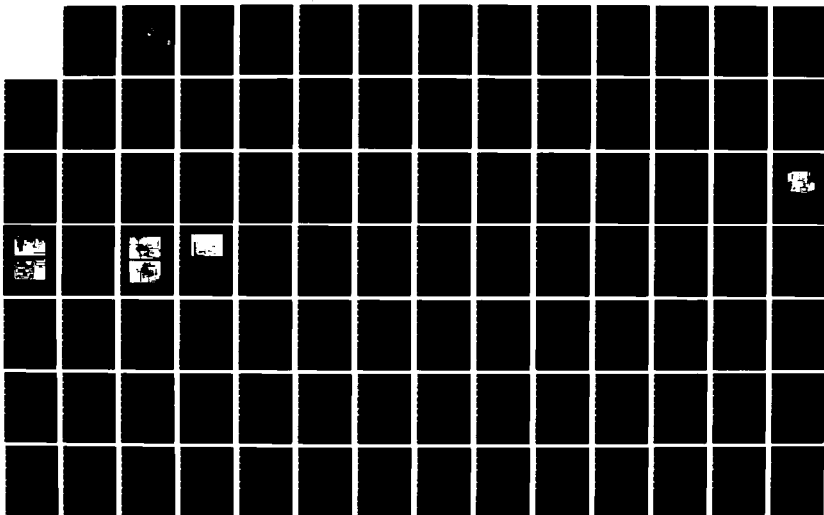
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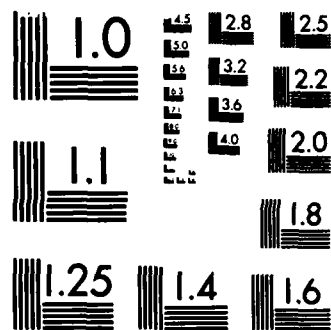
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THESIS

MARINE PROPULSION LOAD EMULATION

by

Philip N. Johnson

June 1985

Thesis Advisor:

David Smith

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Marine Propulsion Load Emulation

by

Philip N. Johnson
Lieutenant Commander, United States Navy
B.S.M.E. Maine Maritime Academy, 1977

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
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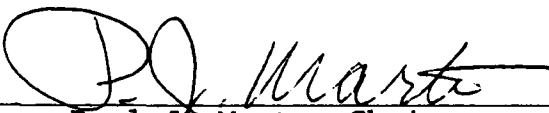
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ABSTRACT

Improved propulsion plant control schemes for gas turbine ships can provide both economic and tactical benefits to the fleet. One way to develop improved propulsion controllers is to use a marine propulsion emulator as an implementation test bed for proposed engine control logic.

This paper describes the development and implementation of a load control system for a marine propulsion emulator which uses a water filled dynamometer and a 160 horsepower gas turbine. Steady state and transient data were collected and analyzed and a dynamic dynamometer model was developed using the Continuous System Modelling Program CSMP III. A proportional plus derivative control system was designed using the nonlinear CSMP model with a cut-and-try design approach. Hardware control elements including valve positioners and microprocessor interfaces were designed and fabricated. The microprocessor-based controller was programmed with the dynamometer control algorithm and the system was tested to verify the emulator design.

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SYMBOLS AND ABBREVIATIONS

ADC	Analog to digital converter
CSMP	Continious System Modelling Program
DAC	Digital to analog converter
DGAIN	Derivative gain
Dl	Digital load signal
Du	Digital unload signal
DYNO	Dynamometer
err	error signal
Id	Dynamometer inertia (ft-lbf-sec ²)
IGAIN	Integral gain
Ipt	Power turbine and reduction gear inertia (ft-lbf-sec ²)
It	Total drivetrain inertia (ft-lbf-sec ²)
Mag	Gas generator air mass flow rate (lbm/min)
Mwl	Dynamometer load water flow rate (lbm/min)
Mwu	Dynamometer unload water flow rate (lbm/min)
Nd	Dynamometer speed (rpm)
Ngg	Gas generator speed (rpm)
Npt	Power turbine speed (rpm)
Pe	Enviromental pressure (in. hg.)
Pd	Dynamometer shell pressure (psi)
PD	Proportional derivative
PGAIN	Proportional gain
Pgg	Gas generator back pressure
PID	Proportional Integral Derivative
Ppt	Power turbine back pressure
rpm	revolutions per minute
Td	Dynamometer torque (ft-lb)
Tpt	Power turbine torque (ft-lb)
Vl	Dynamometer load valve voltage
Vu	Dynamometer unload valve voltage
Ww	Dynamometer water weight (lbm)

%l Dynamometer load valve percent open
%u Dynamometer unload valve percent open

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I. INTRODUCTION

Improvement of marine propulsion control can provide increased ship performance. This performance may be measured in many different forms such as acceleration, fuel economy, and increased engine life.

Generally speaking, there is added complexity with better control systems and their development is often difficult, but several steps can be taken to reduce the risk involved in developing new propulsion systems. As a first step, computer based modelling and simulation have been relied upon heavily to design propulsion control schemes for the newer gas turbine propelled ships [Ref. 1]. These simulations have provided quick feedback at relatively low expense in the testing of control alternatives. This design work has been supplemented with shore-based test facilities to prove out control system implementations using actual equipment. In this way modifications can be more easily applied and tested before costly adaptation into the fleet. These installations can also be well instrumented to detect problems which may otherwise go unnoticed.

In the past few years the Naval Ship Engineering Center, Philadelphia, has been performing dynamic testing of gas turbine control systems through the development of a marine propulsion emulator [Refs. 2,3]. Part of their work has centered around the development of a dynamic control system which is capable of emulating a marine propulsion load. The work discussed in this thesis centers around the design of a similar dynamometer controller and data acquisition system which was developed on a smaller scale.

This thesis discusses all phases of the dynamometer control design cycle, from specifications through hardware selection and implementation.

II. DESIGN SPECIFICATIONS

A. CONTROL SPECIFICATIONS

The controller must provide the ability to control the dynamometer remotely, from outside the engine test cell. It must also provide easy interfacing between computer and operator. The controller must support two modes of control: first, independently selectable dynamometer speed for speeds from 500 to 3000 rpm; and second, marine propulsion emulation scheduled from selectable gas generator speed.

The controller should also allow for future expansion and provide easy modification of the control algorithm. The controller should provide safe and stable operation throughout the dynamometers operating range. The dynamometer controller must interface with an existing gas generator governor.

In order to perform marine propulsion emulation for gas generator transients, the dynamometer control system should be able to assume or shed 85 percent of it's full power dissipation capacity in 15 seconds. This will allow the dynamometer to maintain effective load emulation in the face of gas generator throttle changes. The controller should also have less than 5 percent overshoot to steady state conditions due to speed limitations on the power turbine. Anticipated follow-on work dictates the use of a maximum controller sampling and duty cycle of 100 milliseconds.

Total new equipment purchases must be limited to \$1500 and all new equipment ordered must arrive within 90 days of project inception.

B. DATA COLLECTION AND DISPLAY SPECIFICATIONS

All data displayed under the old facility must be duplicated. The data acquisition system should record all variables normally taken by the students as well as perform an analysis of turbine and dynamometer efficiencies. Transient data should be displayed for a selected group of dynamic variables as shown in Table I.

The data acquisition system should be centered around the HP 85 desk top computer to be consistent with other department needs.

TABLE I

Data acquisition specifications

VARIABLE	UNITS	STEADY	TRANSIENT	COMPUTED
BAROMETRIC PRESSURE	IN. HG.	A		M
CELL PRESSURE FRONT	IN. H ₂ O	A		
CELL PRESSURE REAR	IN. H ₂ O	A		A
CELL PRESSURE AVERAGE	IN. H ₂ O	A		
AIRFLOW BELL PRESSURE RIGHT	IN. H ₂ O	A		
AIRFLOW BELL PRESSURE LEFT	IN. H ₂ O	A		A
AIRFLOW BELL PRESS AVERAGE	IN. H ₂ O	A		
COMPRESSOR DISCH PRESS RIGHT	IN. HG	A		
COMPRESSOR DISCH PRESS LEFT	IN. HG	A		A
COMPRESSOR DISCH PRESS AVE.	IN. HG	A		
NOZZLE BOX PRESSURE	IN. HG	A		
COMPRESSOR INLET TEMP. A	DEG.	A		
COMPRESSOR INLET TEMP. B	DEG.	A		
COMPRESSOR INLET TEMP. C	DEG.	A		
COMPRESSOR INLET TEMP. D	DEG.	A		
COMPRESSOR INLET TEMP. AVE	DEG.	A		A
COMP. DISCH TEMP. RIGHT A	DEG.	A		
COMP. DISCH TEMP. LEFT A	DEG.	A		
COMP. DISCH TEMP. RIGHT B	DEG.	A		
COMP. DISCH TEMP. LEFT B	DEG.	A		
COMP. DISCH TEMP. AVERAGE	DEG.	A		A
FUEL TEMP.	DEG.	A		
TURBINE INLET TEMP. RIGHT A	DEG.	A		
TURBINE INLET TEMP. LEFT A	DEG.	A		
TURBINE INLET TEMP. RIGHT B	DEG.	A		
TURBINE INLET TEMP. LEFT B	DEG.	A		
TURBINE INLET TEMP. AVERAGE	DEG.	A		A
EXHAUST TEMP. RIGHT A	DEG.	A		
EXHAUST TEMP. LEFT A	DEG.	A		
EXHAUST TEMP. RIGHT B	DEG.	A		
EXHAUST TEMP. LEFT B	DEG.	A		
EXHAUST TEMP. AVERAGE	DEG.	A		
UPPER ROTOMETER	NONE	A		M
LOWER ROTOMETER	NONE	A		M
SPECIFIC GRAVITY	NONE	A		M

Table I
Data acquisition specifications (cont'd.)

VARIABLE	UNITS	STEADY	TRANSIENT	COMPUTED
TURBINE SPEED	RPM	A	A	
DYNAMOMETER SPEED	RPM	A	A	
DYNAMOMETER TORQUE	FT-LB	A	A	
FUEL FLOW SENSOR	NONE R.	A		A
COMPRESSOR INLET TEMP.	DEG. R.	A		A
THETA	NONE	A		A
COMPRESSOR INLET PRESSURE	PSIA	A		A
DELTA HEATING VALUE	NONE	A		A
LOWER HEATING VALUE	BTU/LBM	A		A
CORRECTED TORQUE	FT-LB	A		A
CORRECTED COMPRESSOR SPEED	RPM	A		A
CORRECTED DYNAMOMETER SPEED	RPM	A		A
CORRECTED BRAKE HORSE POWER	HP	A		A
CORRECTED MASS FUEL FLOW	LBM/HR	A		A
BRAKE SPECIFIC FUEL CONSUMP.	LBM/HP-HR	A		A
BRAKE THERMAL EFF	PERCENT	A		A
CORRECTED MASS AIR FLOW	LBM/SEC	A		A
AIR FUEL RATIO	NONE	A		A
AVERAGE EXHAUST TEMP. RATIO	DEG R.	A		A
COMPRESSOR PRESSURE RATIO	NONE	A		A
COMPRESSOR DISCHARGE TEMP.	DEG. R.	A		A
AVERAGE TURBINE INLET TEMP.	DEG. R.	A		A
IDEAL COMPRESSOR EFF.	PERCENT	A		A

M = Entered manually
A = Automatically provided

III. APPROACH

A. EVALUATION OF SYSTEM FUNCTION

The preexisting engine test system is shown in Fig 3.1. The system is composed of two components: one for power production, and one for power dissipation.

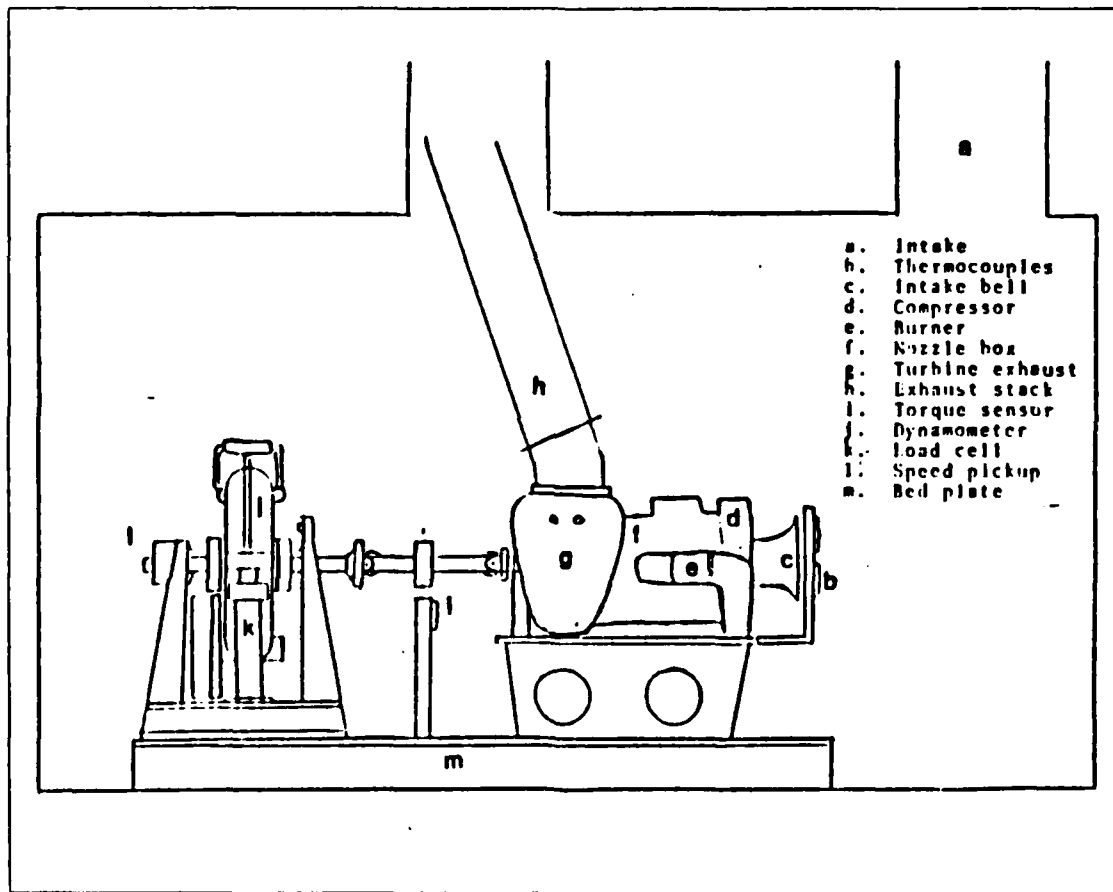


Figure 3.1 Test Cell Configuration.

The power production component consists of a Boeing model 502-2E gas turbine which has two major sections: a gas

generator, and a power output section. The gas generator contains a single-entry centrifugal compressor mechanically coupled to a single-stage axial-flow turbine, two cross-connected can type combustion chambers, and an accessory-drive section. The power output section incorporates a second axial-flow turbine, reduction gears and output shaft, and is driven by the gas generator by a flow of hot gas. The two turbines are not mechanically connected. This arrangement permits the gas generator speed to be controlled independently of the output shaft speed. The resulting engine has output shaft speed variable from 0 to 120 per cent of rated rpm (2900 rpm) for either full or part throttle operation.

The other major component in the test cell is the Claton 17-300 water dynamometer which absorbs the power output from the turbine. It consists of two sub-components: the power absorption unit and the heat exchanger. The power absorption unit may be thought of as a centrifugal water pump with rotor and stator vanes producing a shearing action on the contained water. The torque produced by this shearing action causes the power absorption unit to try to rotate axially on cradle bearings, thus exerting force on a restraining load cell. The quantity of water within the power absorption unit is relatively small, ranging from .5 to 48 lbs. To keep this water at safe operating temperatures the water is circulated by the pumping action of the rotor to the second sub-component, the heat exchanger. An automatic temperature control valve regulates the cooling water to provide near constant water operating temperature.

The unmodified system is dissected into its major functional components in Fig 3.2. The results of a multiport analysis are shown in the Figure which shows how power is exchanged between the various components. Of special interest are those components which interface with the

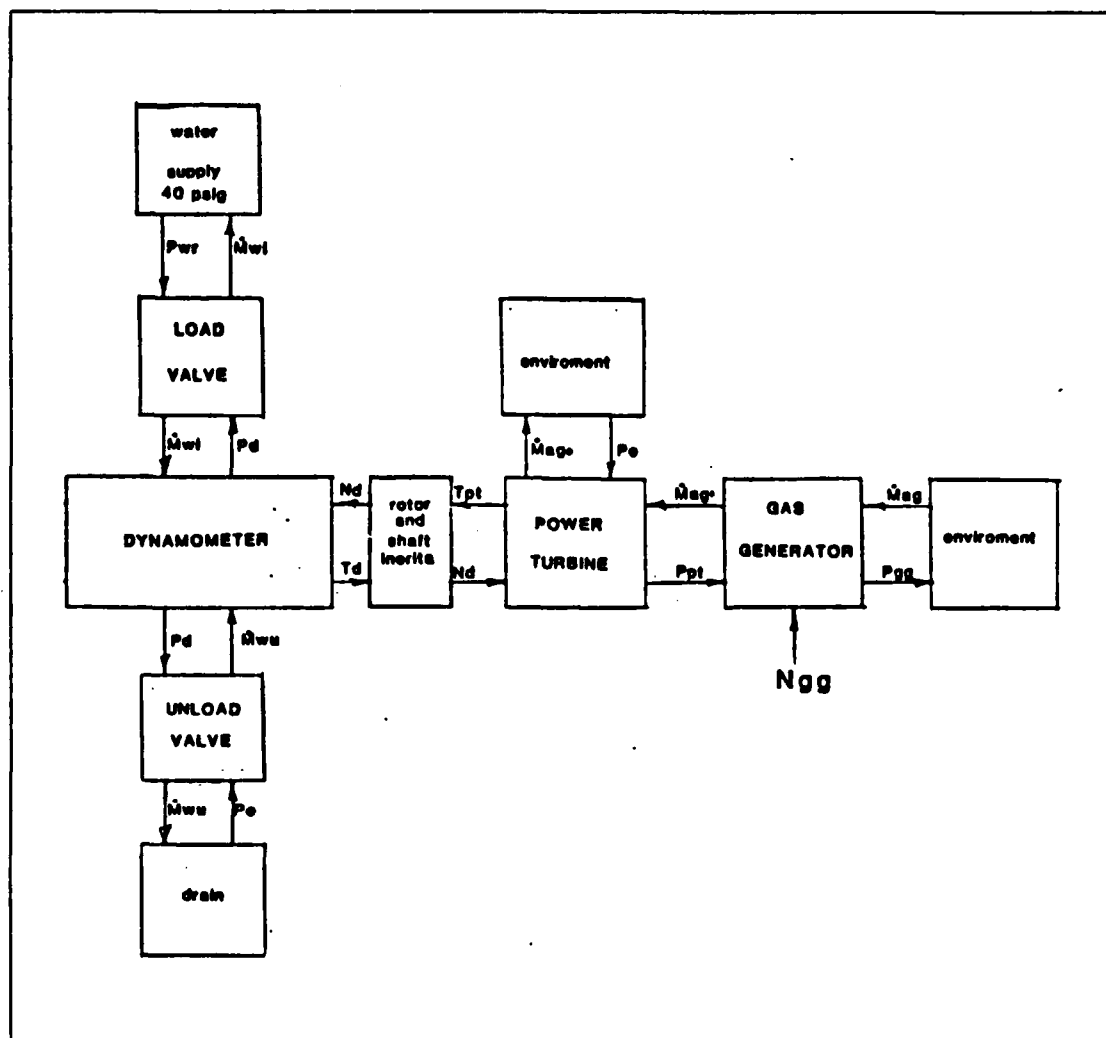


Figure 3.2 Unmodified Hardware Multiport Analysis.

dynamometer. At the onset of this work, two half inch solenoid valves provided for water addition or removal through a network of approximately 80 feet of copper tubing and flex hose. The power turbine and dynamometer were connected by a shaft which acted as a torque summing device, as shown in the Figure. The gas generator was controlled by a speed governor which regulated fuel flow to achieve gas generator speed selections. The load and unload valves were manually controlled to achieve desired shaft speed.

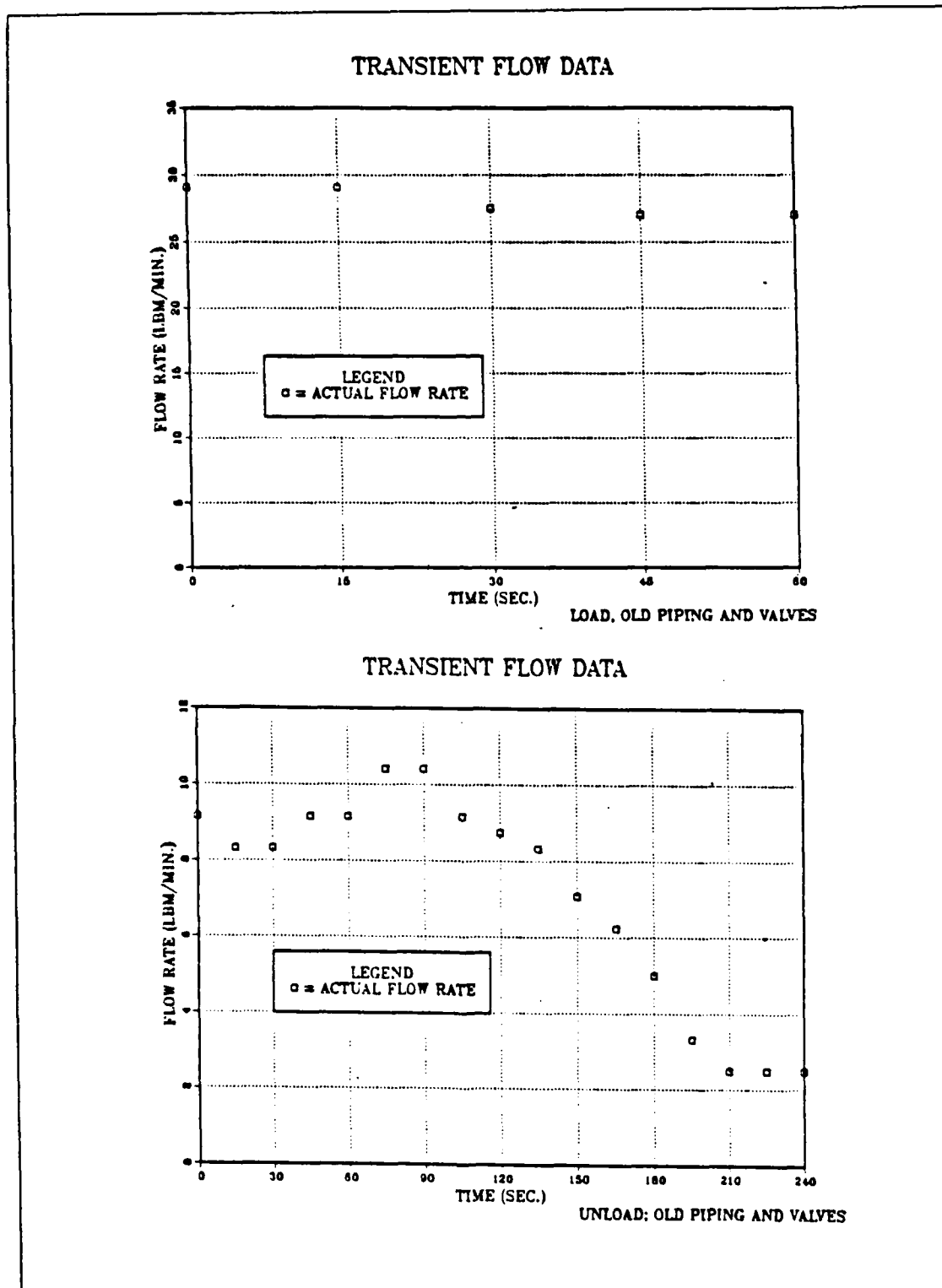


Figure 3.3 Solenoid Valve Flow Rates.

The evaluation of the solenoid valves operation is shown in Fig 3.3. The Figure shows that the dynamometer loading flow rate remained nearly constant through the load transient and could to be modelled as constant. However, unload flow rate varied widely. This droop in the unload flowrate was due to the large change in the valve up stream pressure (P_d) which was observed to vary between 0 and 10 psig, depending on dynamometer speed and loading. It was thought that the creation of a constant pressure chamber in the dynamometer by the addition of a regulated air source would improve this droop in flow rate so that it too could be assumed as constant. Furthermore, in order to decrease the load transient time to meet specifications, the flow needed to be increased. This required increasing the pipe diameters and shortening the piping run. Also, proportional water flow control was required to meter the flow rate and hence control the dynamic transition of load states.

B. PLANT HARDWARE MODIFICATIONS

Figure 3.4 shows a multiport analysis of the modified system. Here, the supply and return water lines were shortened and increased to one inch in diameter. An air pressure regulator was added to the dynamometer shell which provided a four psig head pressure to the dynamometer unit during operation. These changes provided the capability necessary to remove the 15 lbs. of water in 15 seconds as called forth in the specifications for power dissipation. Proportional valves were also designed and added to moderate flow rates to and from the dynamometer. A control implementation of speed regulation was chosen over torque regulation because of the necessity to develop a speed regulator for lab requirements and because the signals were less noisy. Servovalve positioning was to be accomplished by valves

which were designed to attain their desired position independently of microprocessor control, i.e. , position based solely on an analog reference voltage, with internal regulation. This was to speed the control duty cycle.

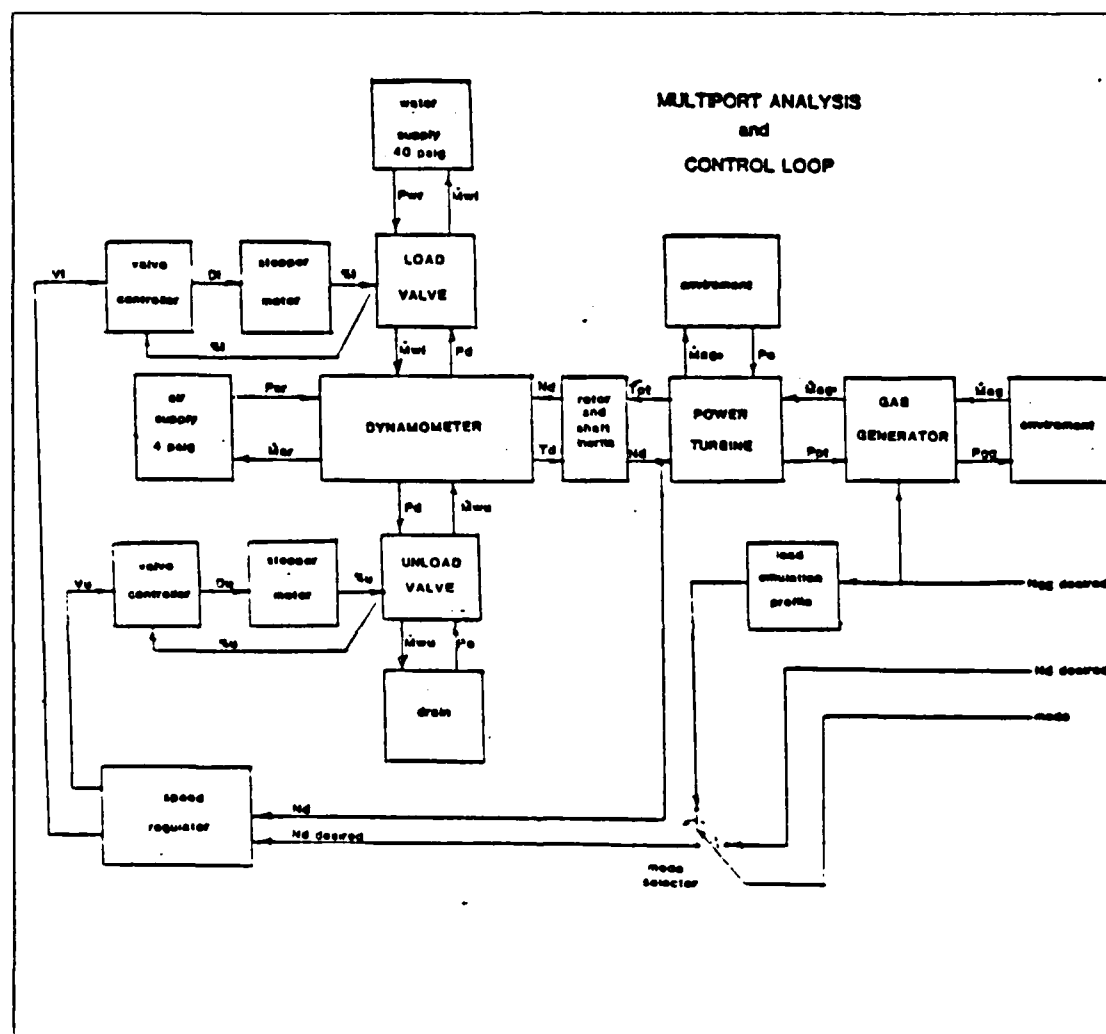


Figure 3.4 Modified Multiport Analysis.

C. PLANT MODELLING

In order to simulate the plant dynamics, the equations describing component behavior had to be developed. These equations described the torque absorbed by the dynamometer as a function of water weight and rotational speed, as well as the torque developed by the power turbine as a function of gas generator speed and power turbine speed. The dynamic effects were lumped into an inertia resistance of the drive train, gears, rotors and water volume in the dynamometer.

A least-squares technique was employed to develop a best fit equation for each of the respective components with the data collected.

The turbine rotor inertia was obtained from the turbine technical reference manual [Ref. 4]. Steady state turbine performance was collected from previous lab work performed by students enrolled in ME 3241 at the Naval Postgraduate School. Additional data was also collected to verify performance at lower operating speeds. Dynamometer rotor inertia was obtained directly from the manufacturer, as was steady state performance data.

Strip chart recorders were used to record transient data for time constant analyses and simulation comparisons.

D. OPEN LOOP SIMULATION

The open loop plant and valve dynamics were programmed into the Continuous System Modelling Program CSMP III to validate their accuracy by direct comparison to strip chart recordings.

E. CONTROL AND DATA PROCESSING HARDWARE SELECTION

The basic concept of propulsion emulation is shown in Figure 3.5. Here, a load emulation curve (A-B-C) is shown

DYNAMOMETER PROPULSION EMULATION

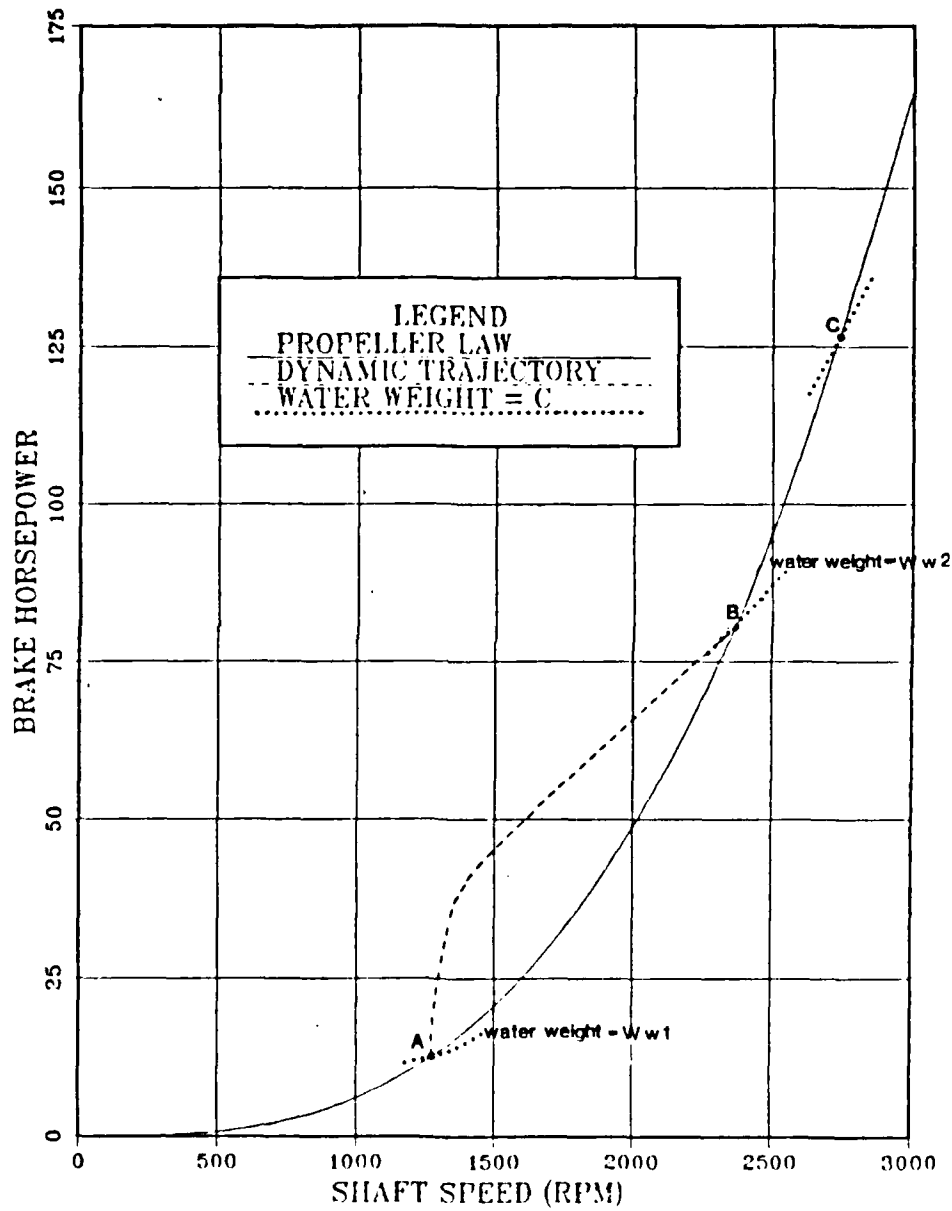


Figure 3.5 Propulsion Plant Emulation.

for a proposed emulator. The propulsion emulator sensed dynamometer and gas generator speed, and computed the torque generated by the power turbine, (point A). It then established a desired speed based on gas generator state along the propeller curve, (point B). For the purposes of this work, a cubic equation was used for the propeller law which passed through 165 horsepower at 3000 rpm and the origin. Having established a desired speed, the speed regulator manipulated the valves in order to transit to the new setpoint (dashed line A to B). Since the controller works by adding or removing water from the dynamometer, in this example the dynamometer went from water weight $Ww1$ to $Ww2$ as shown in the Figure for the transition of A to B.

Due to the need for flexibility in this and subsequent work, a microprocessor based approach was selected to implement the control algorithm. Reduction of the data from the operator panel and student instrument panel was also implemented in order to provide easier and more comprehensible results for laboratory work. However, since computer data acquisition transducers are particularly expensive, the use of dedicated transducers in the turbine test cell was found to be cost prohibitive. Therefore, a data acquisition system was implemented which provides the flexibility to be disconnected for use on other experiments and is simple enough to be easily setup at any location.

F. CONTROLLER DESIGN

The simulation tool CSMP III was used to design the controller (regulator) in a cut-and-try fashion.

The following guidelines were used in the selection of proportional, derivative, and integral controller gains. All gains were initially set to zero. The proportional gain was increased until an oscillatory condition existed, then

reduced slightly until a smooth transition to steady-state was achieved. The derivative gain was selected next to meet overshoot and transit time requirements. The integral gain was to be selected last to provide reset action in a reasonable period.

G. CONTROLLER PROGRAMMING

Several computer programs were required to implement computer aided aquisition and control of the dynamometer. The computer first sampled all input data sensors, then computed the correct control commands and finally output the commands. It was decided to perform this procedure on a fixed interval for simplicity.

The data acquisition system provided the necessary prompts and error checking for the user. The output listings included direct readings as well as a limited reduction of the data which was dissplayed in accordance with the specifications.

In this approach, it was decided to test Tiny Basic as a candidate control development language. If the computer-interpretted code for controller processing took longer than the 100 milliseconds as required by specification, then the controller would need to be programed in machine language since an assembler or compiler was not available. However, the Tiny Basic program testing revealed cycle times of 400 milliseconds. This was four times too slow for the anticipated requirements, therefore, in order to increase execution speed, control routines were programed in machine language. The task of converting a higher level language to machine language is best accomplished through the use of a compiler. A secondary approach is to use an assembler to develop the machine routines. The least efficient and most time comsuming method is to hand compile the routines and

program them directly into the microprocessor. However, because of budgetary and lead time restrictions the last and least desirable method was utilized. Each Basic line was individually broken down into machine language and sequentially programmed into memory. Because there was no assembler or compiler, monitoring detection and correction of errors was difficult.

IV. RESULTS

A. PLANT HARDWARE SETPOINTS

The dynamometer's steady state performance could be changed substantially by the adjustment of an internal flow control valve integral to the heat exchanger (see Figure 4.1 and 4.2). The dynamometer technical reference manual

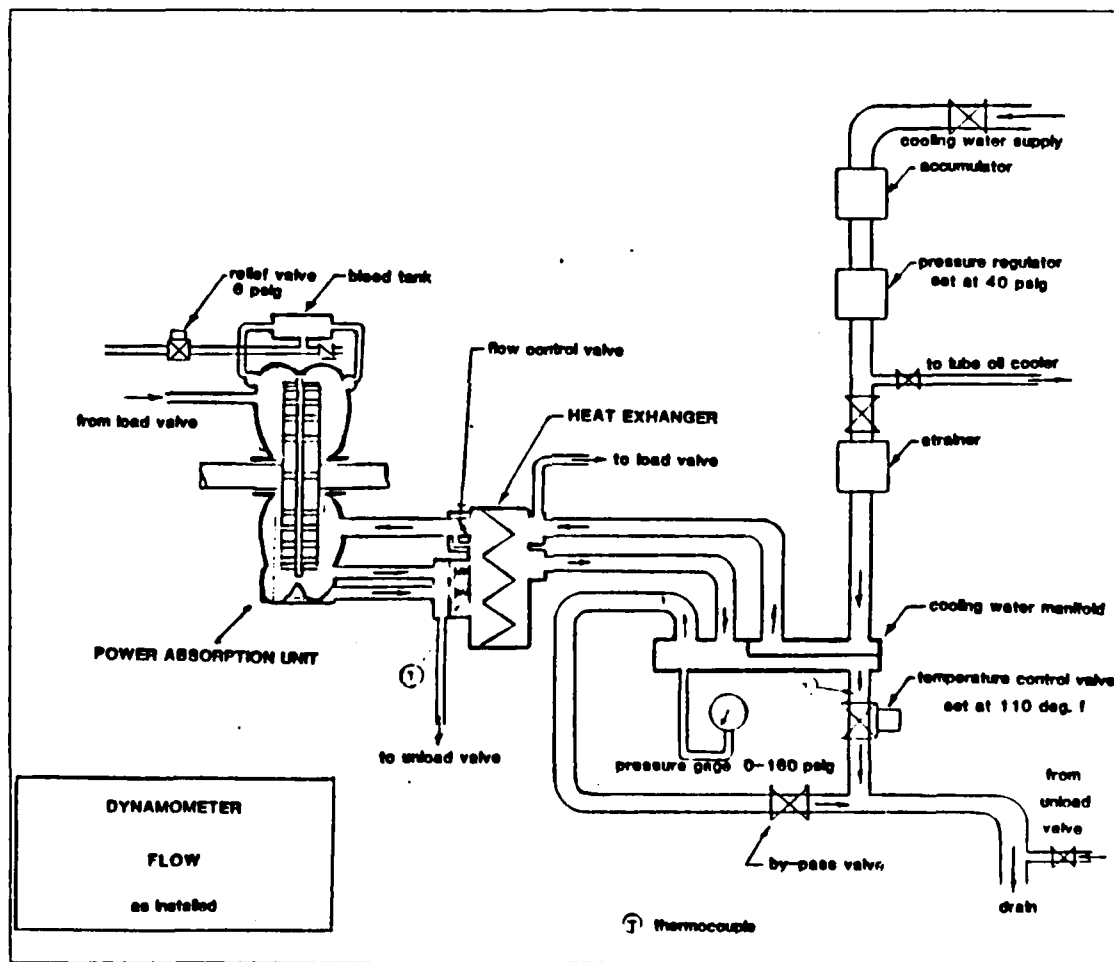


Figure 4.1 Unmodified Dynamometer Flow.

Figure 1 is a graph titled "DYNAMOMETER CHARACTERISTIC CURVE". The vertical axis is labeled "HORSEPOWER (HP)" and ranges from 0 to 350 in increments of 25. The horizontal axis is labeled "DYNAMOMETER SPEED (RPM)" and ranges from 0 to 4400 in increments of 400. The graph shows a curve that rises steeply from the origin, reaching a peak horsepower of approximately 300 HP at around 1000 RPM, and then gradually declines to about 100 HP at 4400 RPM. The area under the curve is divided into two regions: the upper region is labeled "operating with flow control valve open" and is filled with diagonal hatching; the lower region is labeled "operating range with flow control valve closed" and is filled with cross-hatching. The boundary between the two regions is a solid line that starts at approximately 100 HP at 400 RPM and remains relatively flat until about 2000 RPM, after which it begins to decline. Within the "operating range with flow control valve closed" region, there are several data points plotted as small circles, and a dashed line is drawn through them, showing a peak of about 100 HP at 1000 RPM and then a decline.

[Ref. 5] indicated that this valve is used to control stability at light loading. It is important to note that the dynamometer power absorption unit is capable of absorbing up to 750 horsepower when provided with two heat exchangers but, as currently configured, a limit of 300 horsepower is mandated. The flow control valve must be in the open position while operating above 150 horsepower in order to provide adequate cooling, but this provides instability at light loads. Thus, as a compromise, the flow

control valve was set at 25 percent open and left there for all subsequent work.

Dynamometer temperature was also monitored during runs and ranged between 90 and 120 deg f. when regulated by a constant temperature regulator set at 110 deg f. It should be noted that dynamometer water weights between .5 lbm. and 48 lbm. define the limits of operation and that lab usage often requires use of this full range.

B. PLANT MODELLING

Inertia data was collected by subjecting the dynamometer and power turbine to a near constant torque as provided by operator and gas generator throttle position, and measuring the speed signal with a strip chart recorder. The acceleration was estimated from the speed trace and the inertia was calculated from equation 4.1 below. The sum of manufacturers inertias of 1.8 lb-ft-sec² for the turbine and 14.81 lb-ft-sec² for the dynamometer agrees closely with 16.7 lb-ft-sec² experienced during acceleration runs. The water volume inertia seemed to have little effect as evidenced by accelerations performed at various degrees of loading. The negligible difference between the manufacturers and experimental inertias was attributed to additional inertias of shaft and universals which couple the dynamometer and power turbine.

$$I_t = T_d / N_d \quad (\text{eqn 4.1})$$

The relationship between dynamometer water weight and dynamometer torque and speed was also needed as part of this study. This data was collected by measuring torques and speeds at various gas generator speeds with a constant water volume. The water weight was measured at the conclusion of

each experiment by draining the dynamometer. A total of six runs with 28 data points each were obtained and are shown in Figure 4.3 The dynamometer manufacturer supplied additional information which agrees with data collected.

The steady state turbine data was obtained from runs performed by students enrolled in ME 3241. In addition, data was also taken at compressor speeds below 27000 rpm in order to completely map the power turbine performance. During this work, the gas generator was assumed to have a perfect regulator. Thus, once Ngg was selected it was assumed to remain constant during all dynamometer transients. This assumption was validated through data recordings of Ngg. While the shape of the curves for power turbine torque output agree qualitatively with those from the manufacturer, our results are somewhat less in magnitude. This was attributed to the thirty years of turbine usage since installation at the Naval Postgraduate School. The equation is plotted in Figure 4.4.

A least-squares technique was employed to develop equations which represent the data obtained. For the dynamometer, the torque absorbed was assumed to be a function of water weight and input speed (rpm) and is shown in equation 4.2. The power turbine steady state output torque was assumed to be a function of power turbine output speed and gas generator speed as shown in equation 4.3.

$$T_d = -20 + ((0.00046 * (W_w / 16.6))^{*1.3} + 4.0E-6) * (N_d^{*2}) \quad (\text{eqn 4.2})$$

$$T_{pt} = (-725.76 + (0.0363642 * N_{gg})) + (0.05267138 - (4.454586E-6 * N_{gg})) * N_d \quad (\text{eqn 4.3})$$

These two equations were the result of a trial and error procedure of investigating various unsuccessful function forms for dynamometer and power turbine torque. The unsuccessful candidates are shown in equations 4.4 to 4.8.

DYNO TORQUE/SPEED CURVE

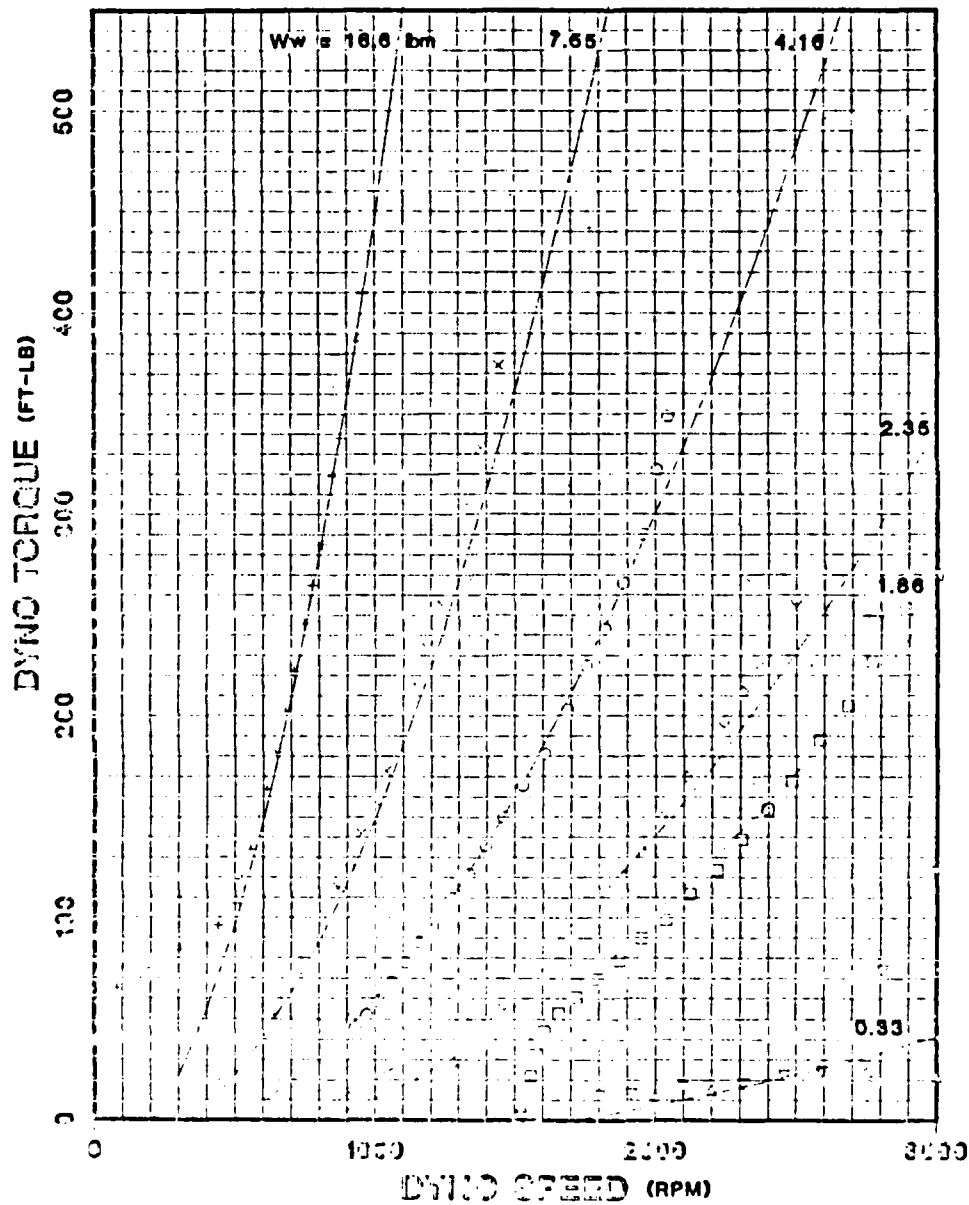


Figure 4.3 Dynamometer Capacity Curves.

TURBINE TORQUE/SPEED CURVE

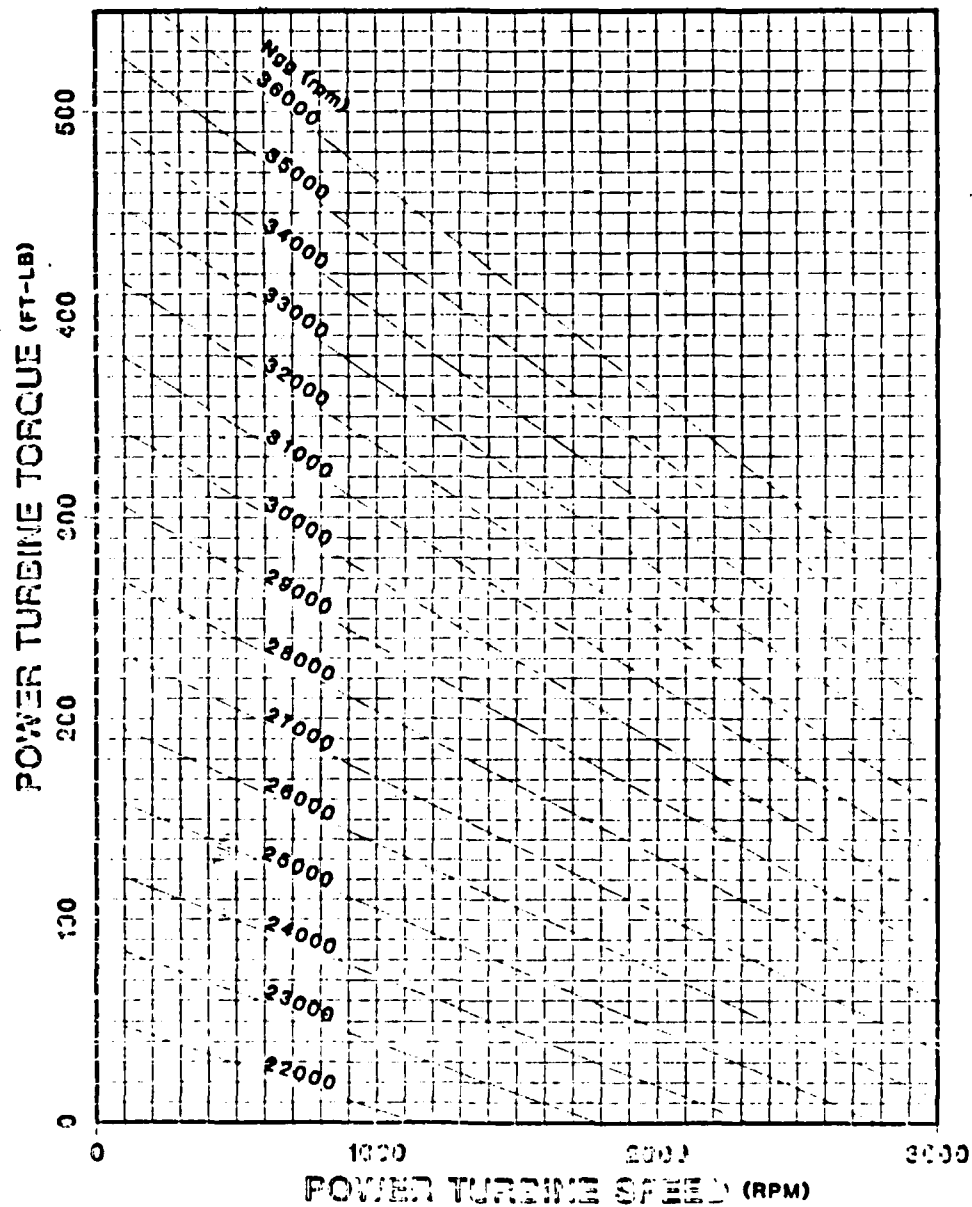


Figure 4.4 Turbine Torque Speed Performance.

$$T_d = ((.112659205 + 3.1511985E-2 * \dot{w}_w) * N_d) - 205 \quad (\text{eqn 4.4})$$

$$T_d = -30 + (((0.00046 * ((0.10 + (\dot{w}_w / 18.5))) ** 1.5)) * (N_d ** 2)) \quad (\text{eqn 4.5})$$

$$T_d = (((1.25293E-7) * \dot{w}_w ** 3) - ((1.35816E-6) * \dot{w}_w ** 2) + ((5.61E-6 * \dot{w}_w) * (-6.176477E-6)) * (N_d) ** (2.64718 - (3.603287E-2) * \dot{w}_w))) \quad (\text{eqn 4.6})$$

$$T_{pt} = (N_{gg} - 18666.0) / 33.3 - 0.0816 * N_d \quad (\text{eqn 4.7})$$

$$T_{pt} = (-412.97 + (0.029 * N_{gg})) + (0.052671 - (4.4586E-6 * N_{gg})) * N_d \quad (\text{eqn 4.8})$$

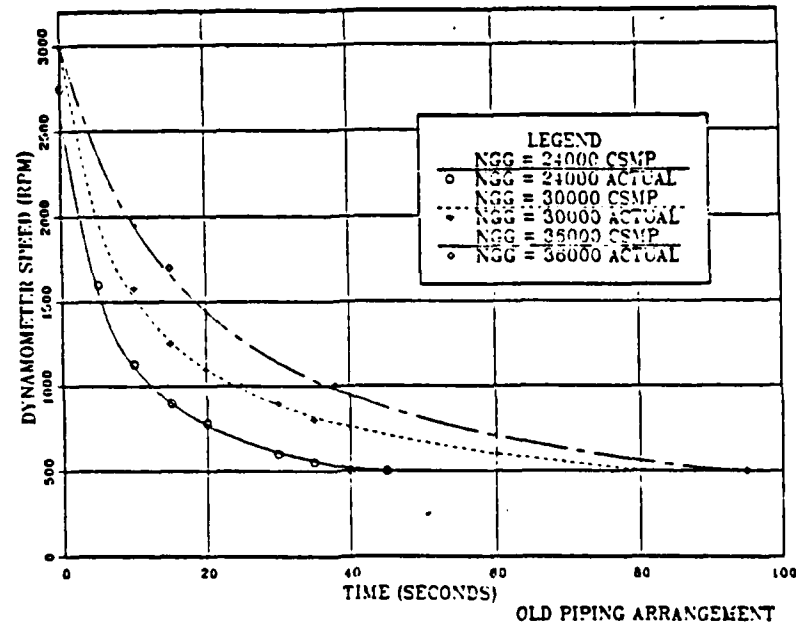
The steady-state turbine and dynamometer data was used as a model for both the static and dynamic states of performance of those units. All inertias were lumped into the rotor model and the acceleration was based on equation 4.9.

$$N_d = (1/I_t) * (T_{pt} - T_d) \quad (\text{eqn 4.9})$$

C. OPEN LOOP SIMULATION

The unmodified plant was simulated to validate the dynamometer-inertia-turbine model. The equations developed from dynamic and steady state modelling were used to simulate the system dynamics via CSMP III. The model was subjected to similar transients as those recorded in the data collection phase, which is shown in Figure 4.5. The simulation for dynamometer loading Figure 4.5 (top) shows good agreement over the normal operating ranges of the equipment. However, unloading results were quite different, Figure 4.5 (bottom). This difference was due to the poor model of unload valve fluid dynamics. Actually, when the dynamometer was unloaded, initial water flow rates were high, therefore the actual dynamometer unloaded faster than the simulation. As the real dynamometer approached 3000 rpm

CSMP MODEL VALIDATION



CSMP MODEL VALIDATION

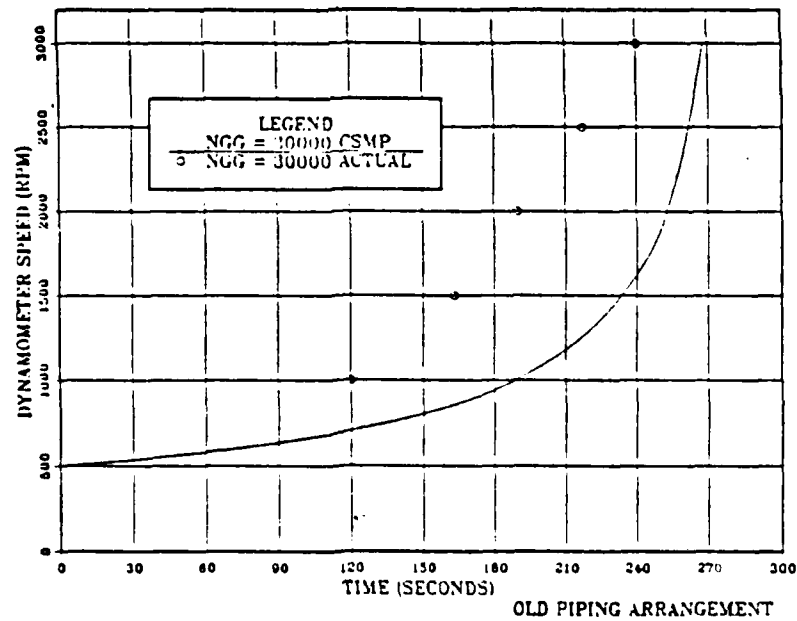


Figure 4.5 Open Loop Simulation.

the flow rate was less than the model and it approached slower than the simulation. However, since unload flow was substantially less than the load flow rate and the dynamometer was to be modified with a constant pressure regulator, it was decided that the assumption of constant pressure drop across the valves could be maintained and control development could proceed.

D. CONTROL AND DATA PROCESSING HARDWARE IMPLEMENTATION

Governed by the need to provide an 85 percent load change in fifteen seconds, and knowing the volume of water necessary to provide changes in operating points, the valve and pipes were sized as discussed earlier (section III.B). The pipe and valve sizes were chosen to provide proper loading capacity to prevent overspeed of drive train if subjected to maximum acceleration of the gas generator. The load and unload sizes were chosen to be the same for simplicity.

In order to keep overall pipe size small and to provide a more constant pressure differential across the unload valve, an air pressure regulator was installed to maintain dynamometer shell pressure at a constant 4 psig (Figure 4.6). The piping modification also required an increase in water supply and removal capability. For the load system, water was tapped from the cooling supply header. Since the header and supply were designed for 750 horsepower operation, their capacity was sufficient to handle the needed load water. Additional water was supplied through the ports normally used to empty an additional heat exchanger. Because of the increase in filling capacity, an additional 1 inch relief valve was installed whose relieving pressure was approximately 8 psig.

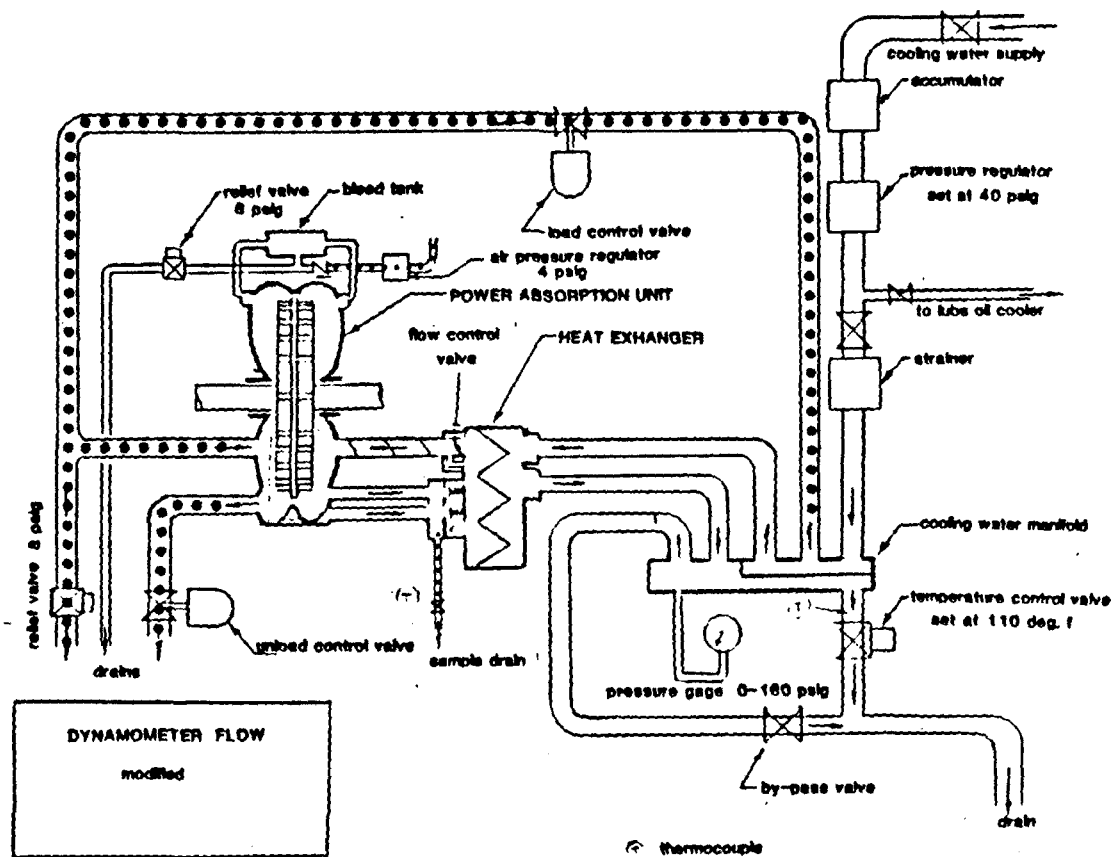


Figure 4.6 Modified Dynamometer Flow.

For the unload system, the water was extracted from the unused heat exchanger supply ports. This location is virtually the same as in the old configuration, however, it offers a much greater capacity. The unload water was then dumped into the same drain as the heat exchanger cooling water. Flexible hoses were used to pipe to the dynamometer to avoid torque shunts from the dynamometer shell to the test cell foundation.

Globe valves were selected to control the various flow rates required during loading and unloading operations. The required time constants of the valves were estimated to be approximately 3 seconds. These valves were designed to position independently of microprocessor control (i.e., position based solely on an analog voltage and internal regulation). A review of currently available electrically operated analog positioning valves revealed that none were available at a reasonable cost.

The design of the motor-actuated analog-voltage controlled valves started with the use of standard 1 inch brass globe valves. The number of revolutions as well as torque required to open and close the valve under pressure was determined. Motor selection was based on a torque requirement of 600 inch lbs. and a speed of approximately 40 rpm. A gear head motor was desirable, however, a slow synchronous motor was selected due to cost considerations. The use of a synchronous 72 rpm motor provided an interesting control problem in itself and is discussed in appendix E. The use of slow sync motors provided constant opening and closing rate. These motors had good torque characteristics and provided a time constant of 2 seconds.

Flow characteristics of the valves were measured using the on site water supply regulator for the load valve and a constant upstream pressure for unload valve. The result of transient and capacity testing is shown in Figure 4.7. The load and unload valves had similar characteristics under these conditions.

Microprocessor control was selected to provide the ease of modification needed for future work, and twelve bit input and output analog resolution was selected to provide smooth operation. Because analog output was required for valves and throttle positioning, a digital-to-analog interface board had to be designed. The details of this interface design are included as Appendix F.

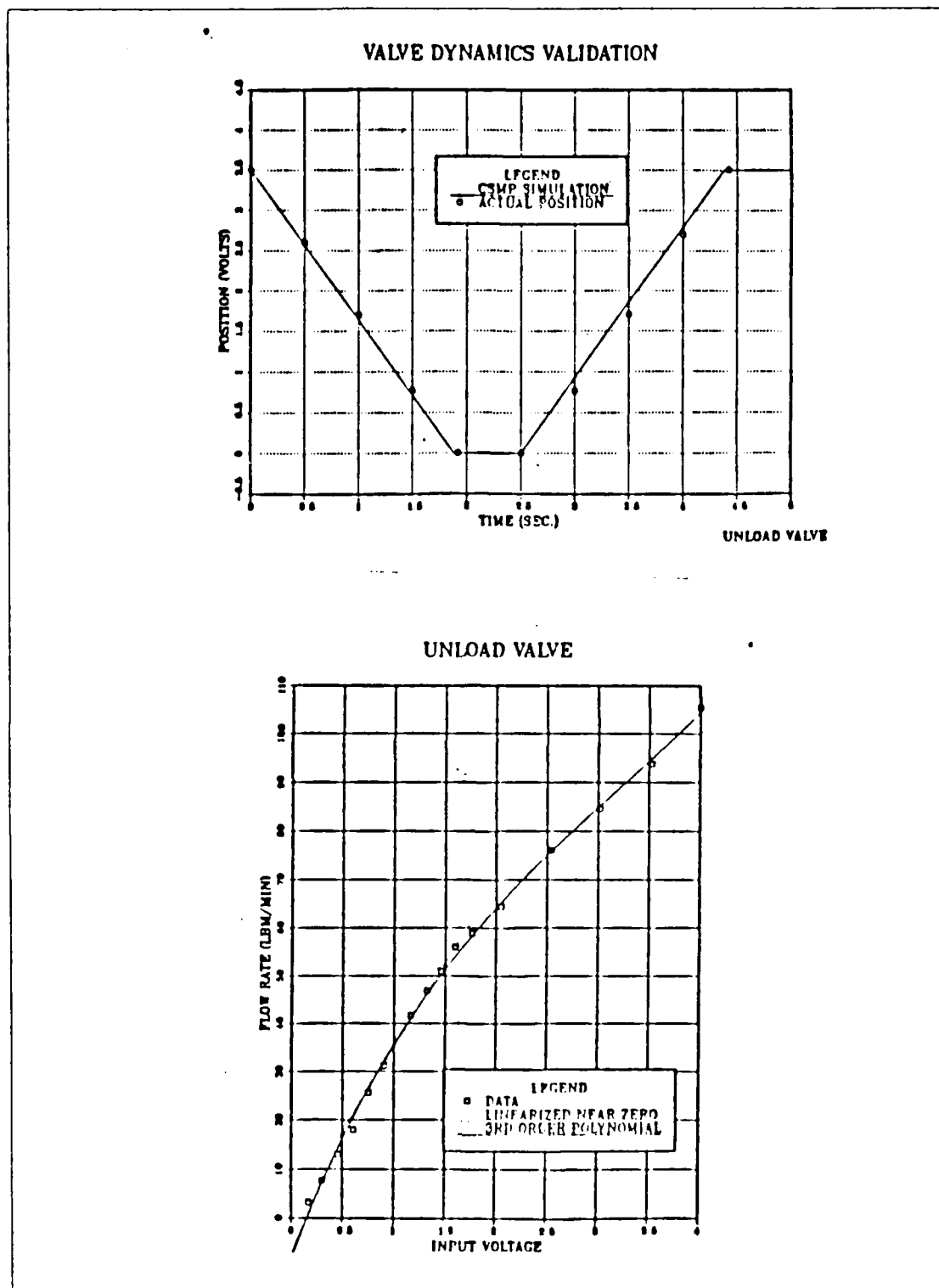


Figure 4.7 Load Valve Characteristics.

System modifications are shown in the following series of photographs. The first photograph Figure 4.8 shows the turbine test cell. On the left the side is the observation window, in the foreground is the gas turbine, and behind it the dynamometer.

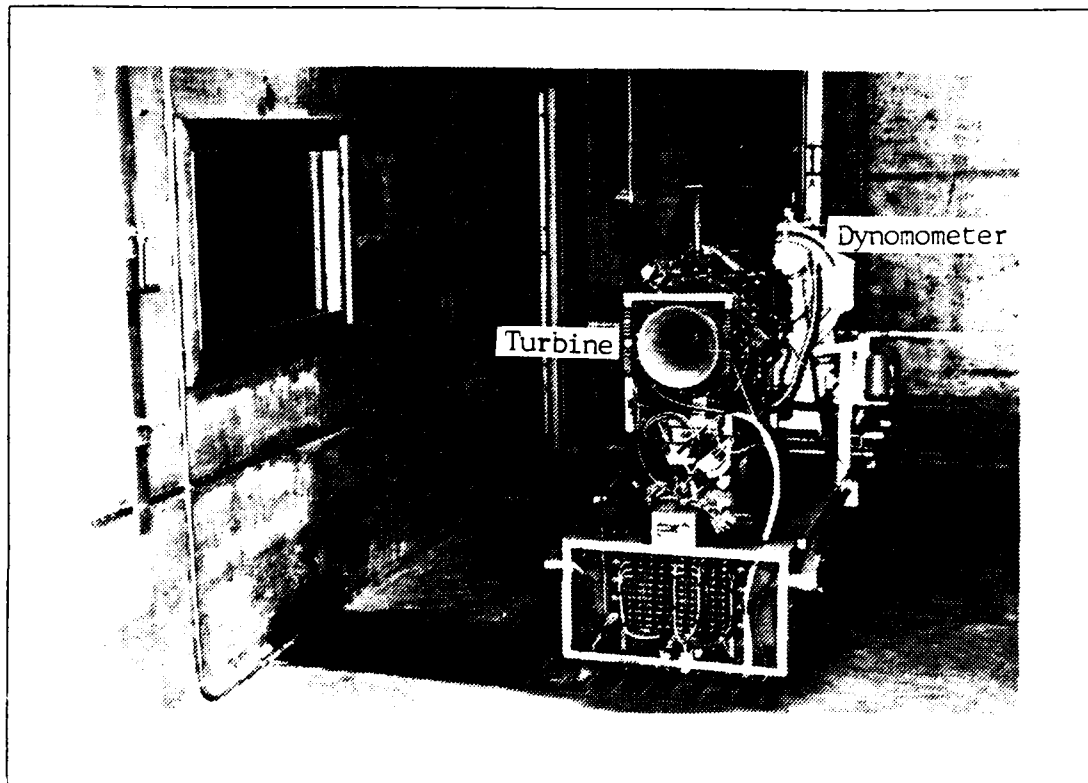


Figure 4.8 Photograph of Turbine Test Cell.

Figure 4.9 (top) shows the observation area. Here the test cell is located behind the wall at the left. The operators station is located against that wall and faces the observation window. The student instrument pannel is located against the far wall, and on the right is the computer data aquisition system. The photograph in Figure 4.9 (bottom) provides a closeup view of the data acquisition

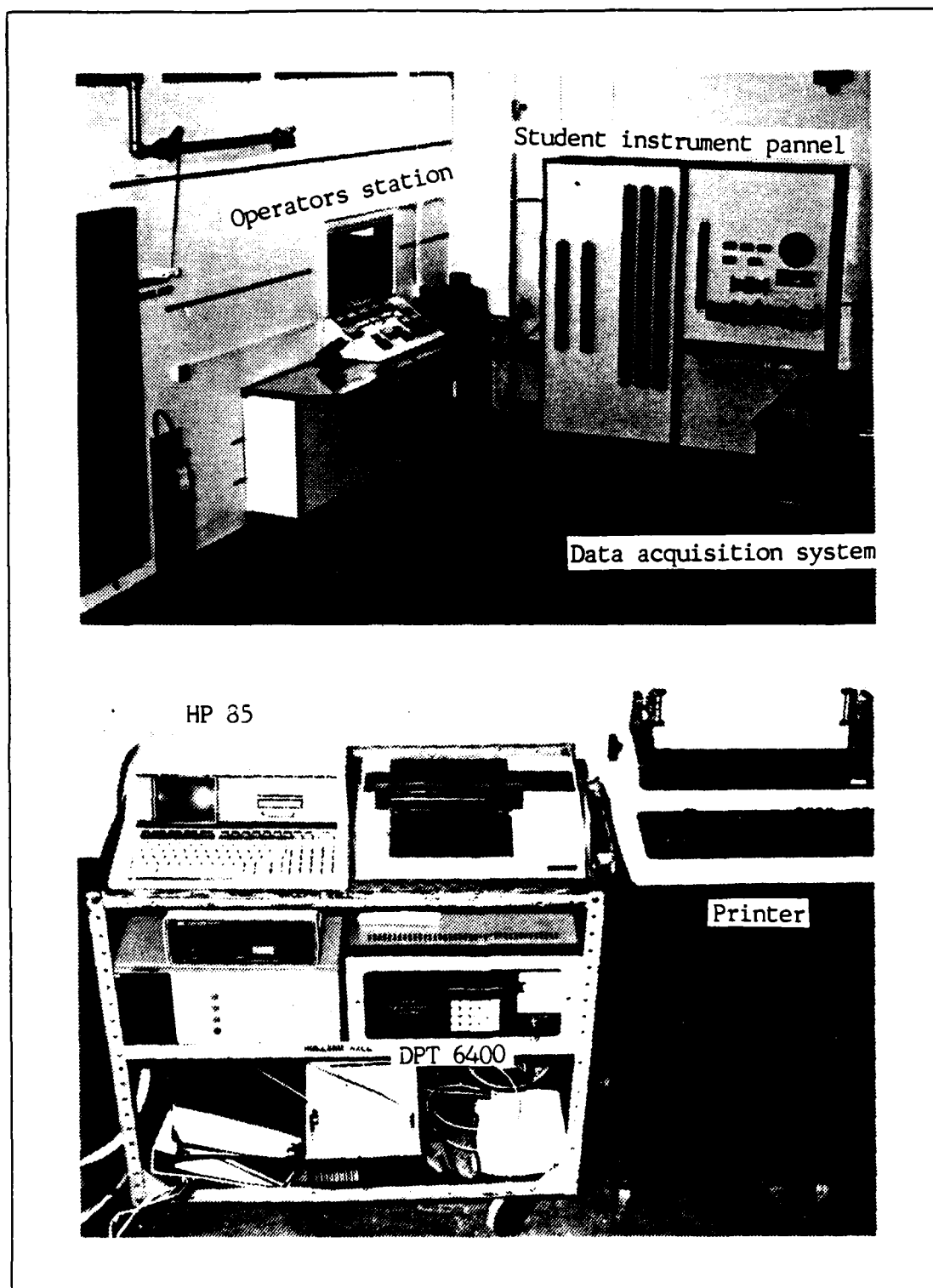


Figure 4.9 Observation Area and Acquisition System.

system. The HP 85 which controls the acquisition equipment, is located on the top left side of the photograph. Below the computer is the disk drive which provides storage capability. Below the disk drive is the HP 9741 used to convert speeds and temperatures into digital information for the HP 85. To the right of the HP9741 is a DPT 6400 which senses pressures for the HP 85. A printer and plotter are located to the far right.

Figure 4.10 (left) is a rear view of the dynamometer. In this photograph, the water supply system is in the lower left corner. Just above and to the right of the inlet lines is the air pressure regulator which maintains constant shell pressure. A shell pressure gage which is affixed to the top of the power absorption unit for the operators viewing. Attached to the side of the power absorption unit is the load cell. Just below and to the left is the drain valve used to collect water volume samples. Behind the drain valve is the heat exchanger whose discharge piping can be seen exiting near the water supply lines.

Figure 4.10 (right) is a front view of the dynamometer. On the left is the rear section of the power turbine. To the right is the shafting and dynamic torque sensor and the dynamometer power absorption unit. Below the absorption unit is the valve positioners, valves and piping.

Figure 4.11 shows a closeup view of the one inch globe valves and stepper motors. Here the details of how the motor is mounted to the frame are shown, as well as the three to one gear set that drives a ten turn potentiometer for position sensing. A spline arrangement was designed to act just above the valve wheel which permits axial valve travel.

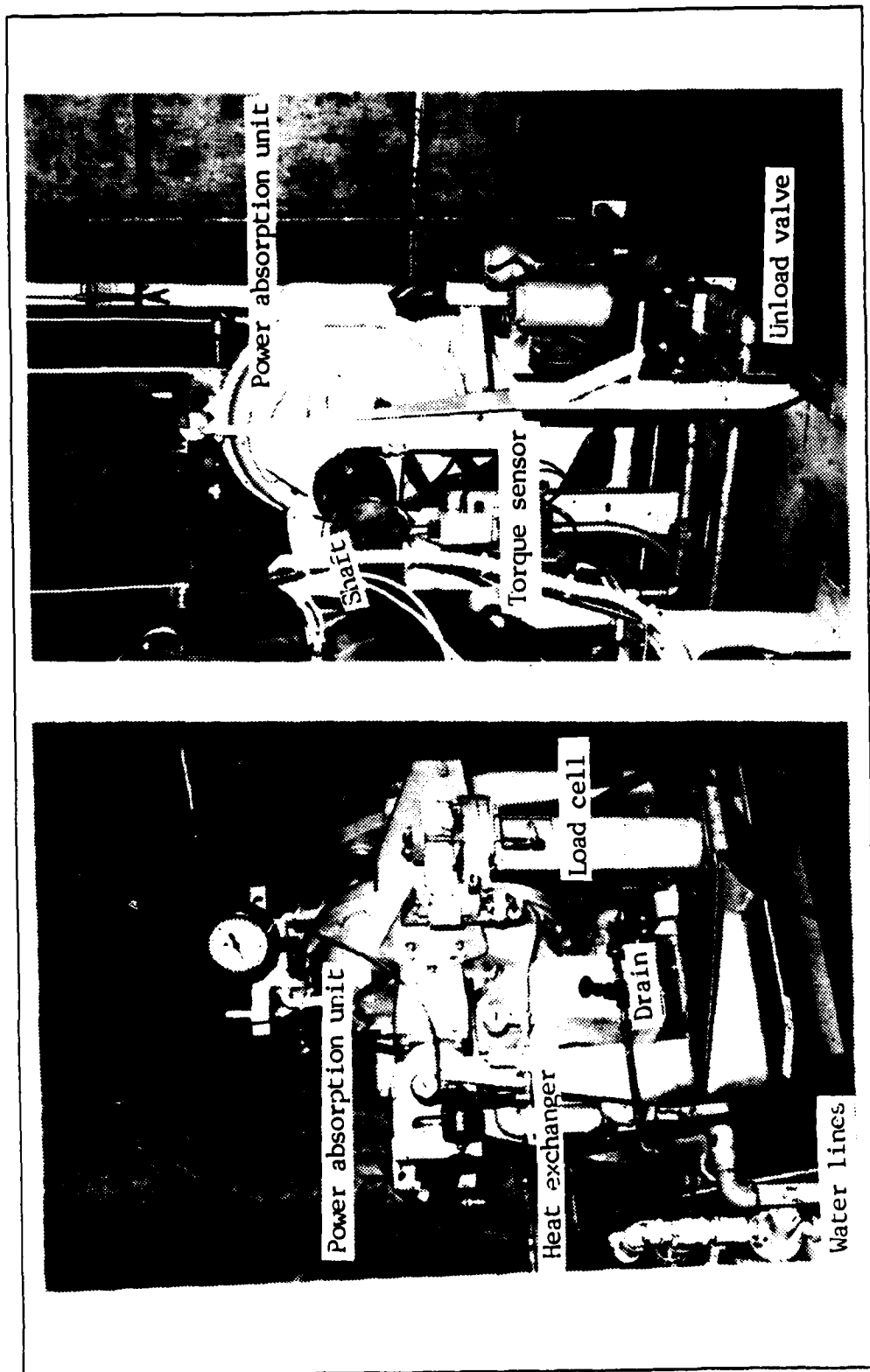


Figure 4.10 Front and Rear Views of Dynamometer.

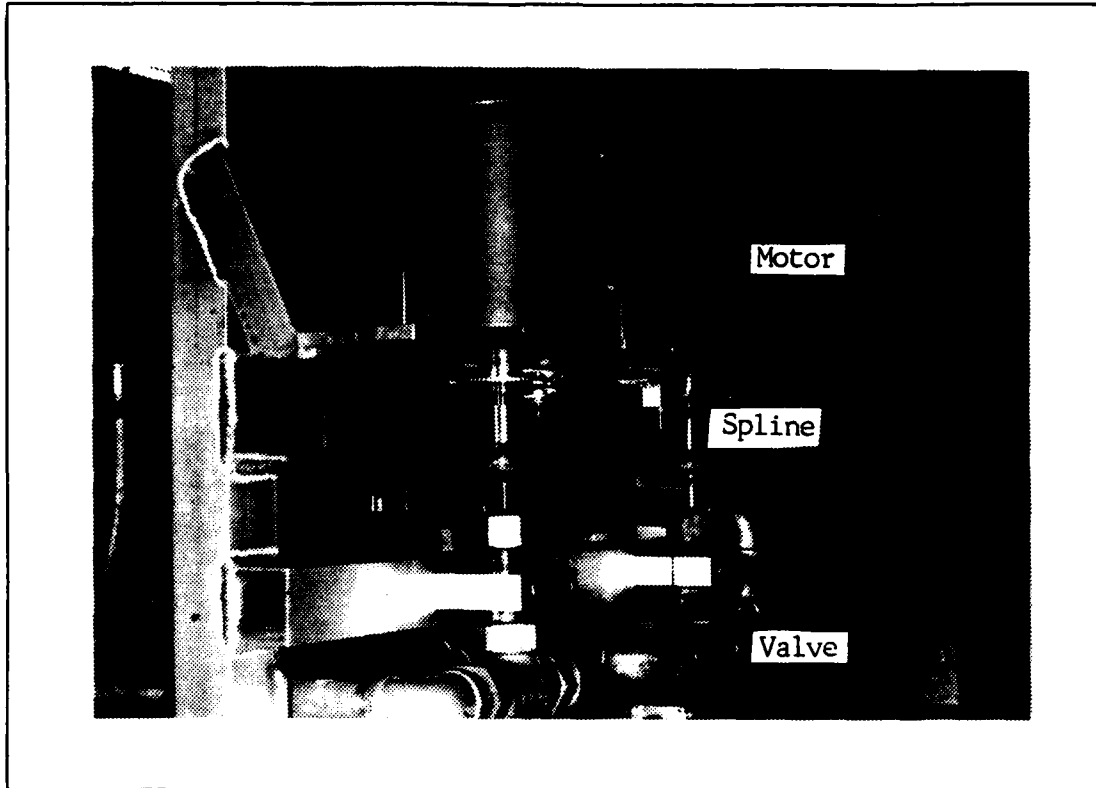


Figure 4.11 Closeup View of Valves and Positioner.

E. CONTROLLER DESIGN

The overall system simulation is shown in Figure 4.12. The Figure shows the three major system components. The block for plant open loop dynamics contains all the transient and steady state properties of the plant. The valve model block contains the transfer functions necessary to convert the controller output voltages into flow rate to the dynamometer, including the dynamics of valve positioning. The controller block simulates a digital type controller with fixed sample intervals and adjustable proportional, integral and derivative gains.

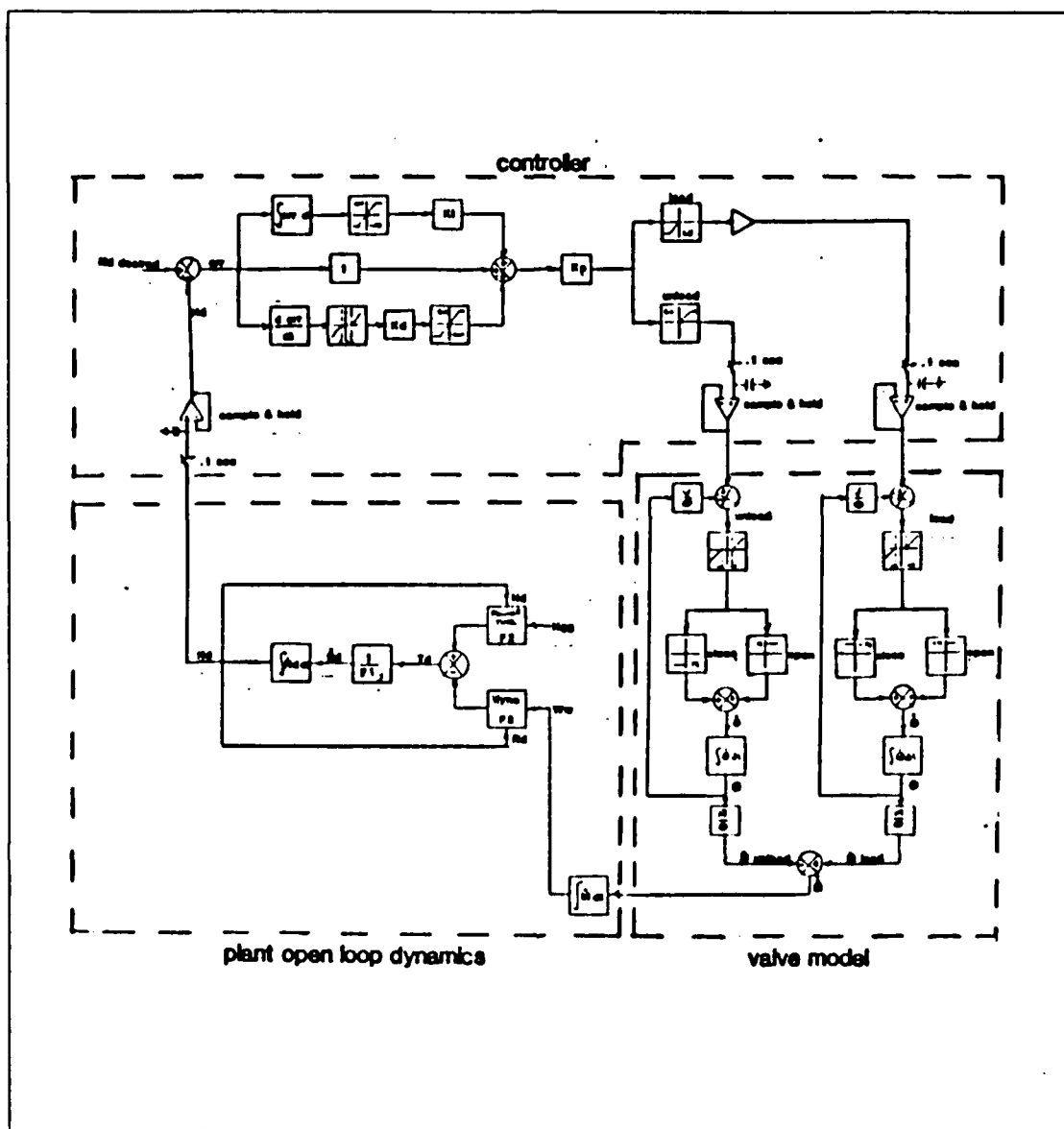


Figure 4.12 Block Diagram of System Simulation.

The use of a microprocessor controller introduced additional complexity to both the system model and the control algorithm design. Because the sampled signals were digitized, their derivative became very noisy at either very high or very low rates of signal change. Furthermore,

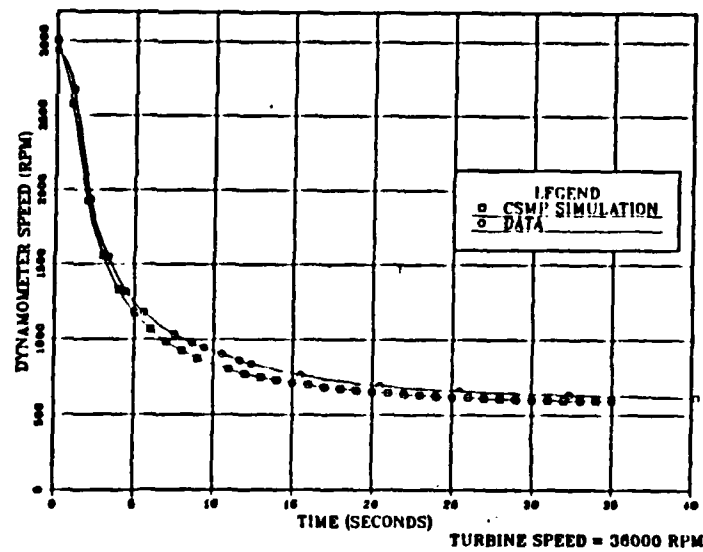
because this noise was most noticeable at or near setpoint where it has little value in the control solution, it was filtered out by elimination of all derivative values below and above thresholds. The integrated error signal was observed to saturate the controller when the system was subjected to step changes in demand. In order to compensate for this effect, maximum integral limits were established as shown in Figure 4.12.

The valves were tested in order to obtain a transfer function which related input voltage signal to output flow-rate. The valve dynamics analysis assumed that the synchronous motors would transit at a constant speed. This assumption was based on the fact that the torque of the motors was relatively high compared to the inertia of the valve and rotor. Verification of valve dynamics with actual transient data proved this assumption to be accurate enough for this study (Figure 4.7).

As mentioned above the predicted valve performance was based upon the assumption of constant shell pressure of 4 psig. The error of this assumption is shown in Figure 4.13 where the closed-loop simulation and actual data are overlaid. Note that loading transients, where pressure drops across the valve are nearly constant, produce good agreement with the simulation, Figure 4.13 (top). Unloading at low speeds and high torques, where shell pressures were higher than modelled, lead to speed increases which are faster than simulated (Figure 4.13, bottom).

The most arduous control scenario observed was used as the test case for controller development. This transient occurred at constant gas generator speed. It started at 500 rpm with the dynamometer fully loaded and required the rapid emptying of the dynamometer to attain 3000 rpm, where speed regulation was the most sensitive to water volume changes (similar to Figure 4.13 bottom). Later, after gains were

TRANSIENT RESPONSE



TRANSIENT RESPONSE

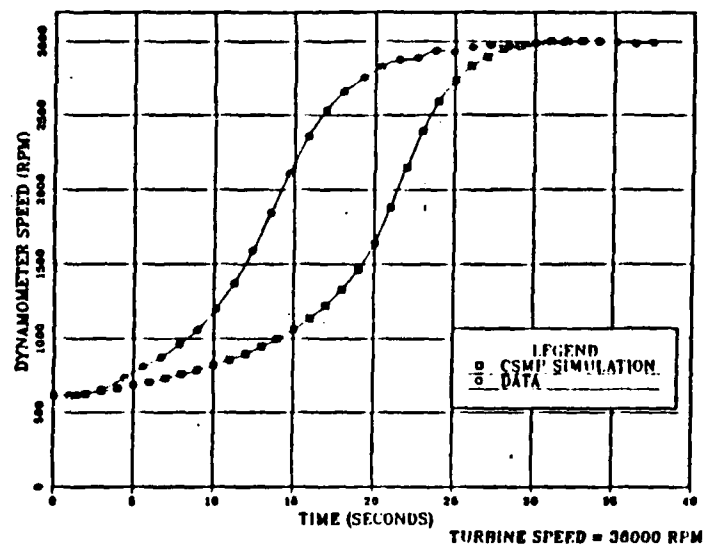


Figure 4.13 Close Loop Simulation.

selected, other transients were investigated to verify control over the full range of operation.

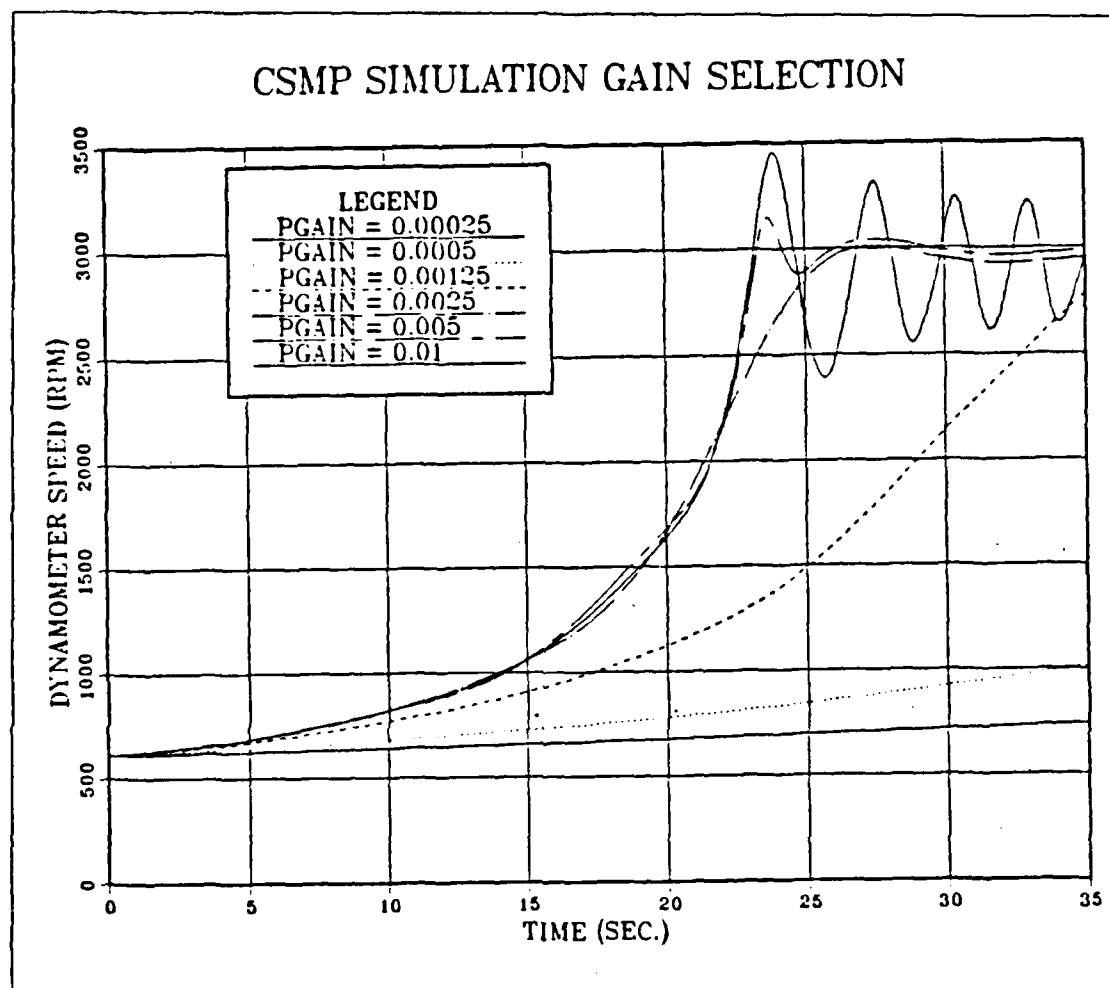


Figure 4.14 Proportional Gain Selection.

The selection of gains followed the guidelines discussed earlier. The proportional gains selection is shown in Figure 4.14. Note that for the first half of the transient, several of the the control solutions with different gains are overlayed on each other. This indicates saturation of the unload valve. Also shown is the effect that high

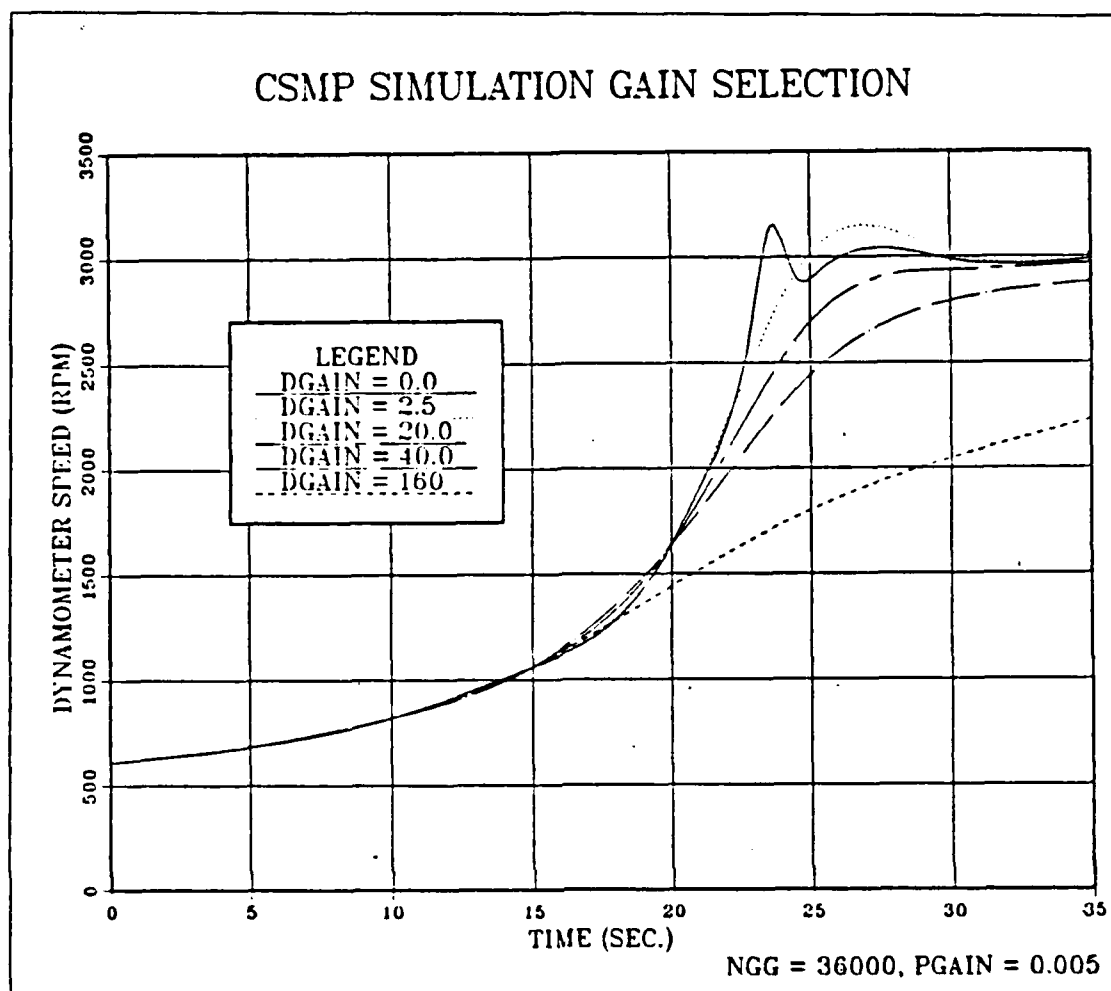


Figure 4.15 Derivative Gain Selection.

proportional gains produced oscillations around the desired speed, whereas low gains failed to meet transient time requirements called for in the specifications. The selection of a gain of .005 was between these two limits and provided some margin for system and controller degradation.

The derivative gain was selected next, as shown in Figure 4.15 A derivative gain of 20.0 was selected to meet over shoot and transient requirements, again selecting a gain which best provided margins for safety.

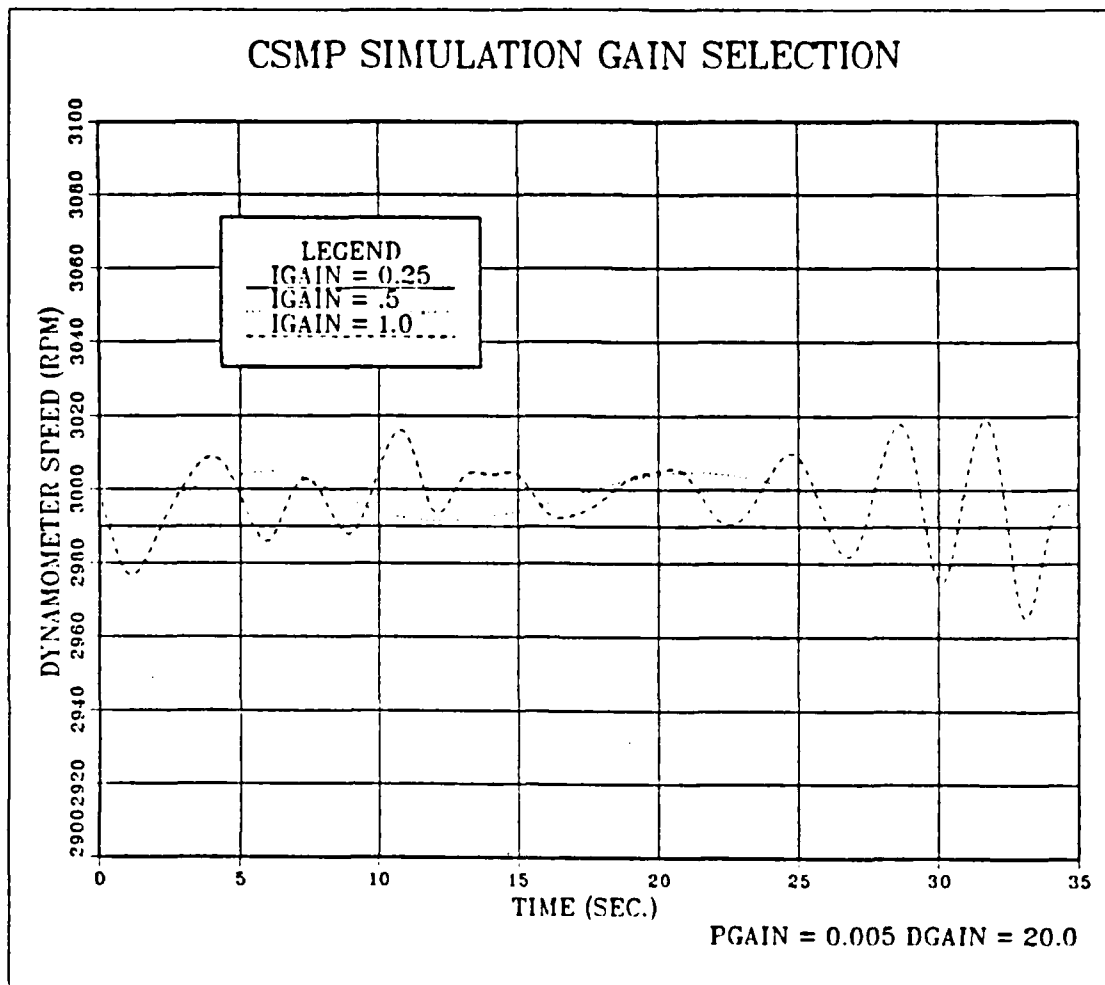


Figure 4.16 Integral Gain Selection.

Lastly, the integral gain was evaluated. An integral gain of .5 was initially selected to provide reset action in a ressonable period. In addition, the integrator output was limited to ± 100 to avoid saturation and balance the net controller output. Closer examination of the system behavior around set point showed that the integral action was generating a limit cycle when valve backlash occurred (Figure 4.16). However, the P-D controller provided a 1 percent error in speed regulation without the added

complexity of a PID controller, so the integral gain was set to zero.

F. CONTROLLER PROGRAMMING

Having selected proportional, integral, and derivative gains using CSMP III, efforts then centered on conversion of the control algorithm to a real time computer control program. The program was first written and tested in Tiny Basic. Cycle times of 400 milliseconds were experienced and it was decided that machine language was necessary to reduce cycle times to less than 100 milliseconds. Programming in machine language without the use of an assembler proved to be a formidable task. However, programming was facilitated by the development of macros for often-used control tasks. The program was structured so that control constants, such as gains and flags, were set in Basic to allow for easy modification. The control solution algorithm for real time control was performed in machine language for speed of execution. It is important to note that while gains were selected for a 100 millisecond cycle time, the inputs are sensed, control solution calculated, and output prepared in approximately 3 milliseconds. This provides ample time for later expansion into complex control algorithms. The overall improvement in performance between Basic and machine language speed of execution was observed to be approximately two orders of magnitude.

Both machine language and Basic suffer the same mathematical constraints of the 8073 microprocessor associated with sixteen bit fixed point arithmetic. This type of arithmetic requires a control program to continually check to determine if an overflow condition exists. Prechecks in addition and multiplication must be used to prevent the overflow condition from occurring.

V. CONCLUSIONS

Control development using a nonlinear simulation was successful in the development of a marine propulsion emulator and speed regulator for a dynamometer. A control algorithm was developed using a CSMP III simulation which accurately depicted the performance of the system. The algorithm was translated directly to the microprocessor controller and successfully run as written. The good agreement between simulated and actual closed loop response thus validates the accuracy of both the control simulation and the approach to emulation. This also provides a plant model for future gas turbine controller development.

Machine language was required in order to realize real time solutions for the control algorithm. This was true even though system time constants were large and the algorithm was of modest size. It was determined that the equivalent control algorithm written in Tiny Basic ran 100 times slower than its machine language counter part.

Linearization of system performance for nonlinear systems can sometimes nullify the important characteristics which dominate the control problem. This was found in the area of valve dynamics modelling of the designed valves which transit at a constant rate. The limit cycle behavior predicted by a nonlinear model of valve dynamics would not have been observed with a linear valve model.

Numerous equations were developed to represent the steady state performance of both the dynamometer and power turbine. While they all represent the data to varying degrees, each had it's own complexity. Consequently, the selection of an appropriate equation was a tradeoff between knowledge of how the physical system should relate to inputs and what equation best fits the data.

Dynamometer pressures were assumed to be constant with the addition of the constant pressure regulator. The pressurization of the dynamometer was thought to improve the unload capacity and ease the modelling problem by providing a constant pressure drop across the valves. While pressurization helped to double the unload capacity, it did not fully correct for varying discharge pressures.

The use of slow synchronous stepper motors and an analog digital controller proved to be a successful, low cost, simple, and effective means for controlling valve position. The valves were capable of regulating the flowrates from 0 to 200 lbm/min. It should be noted that while the motors provided good positioning of the 1 inch globe valves, they did contain some backlash. This effect was due to 3 degrees of resolution, which corresponded to approximately one half lbm/min near full closure.

VI. RECOMMENDATIONS

The data acquisition system could be improved by reprogramming it to take multiple readings in order to determine fluctuations in data and error. Currently, fuel flow data is entered manually. The installation of a turbine type flow meter could negate this requirement. The data acquisition system could also be interfaced with the turbine and dynamometer controllers to permit totally automatic data acquisition for a preprogrammed test schedule.

To reduce the burden of manually converting control algorithms into fast machine language code, an assembler should be obtained for the microprocessor.

The dynamometer unload valve performance was in poor agreement with the model. A detailed investigation of dynamometer discharge pressures and incorporation of this affect on the valve transfer function would substantially improve the simulations accuracy.

The current gas turbine control system was developed on short notice and has substantial room for improvement. A detailed study into the gas generator dynamics, as well as the development of an optimum controller, should be undertaken. Additional control loops which incorporate the turbine, dynamometer and gas generator could also be developed, thus providing a control system which is more in line with the bridge throttle lever concept where shaft speed is selected and the controller regulates to attain that speed. The emulation of sea state disturbances to be injected into proposed control systems could be performed in order to evaluate optimum designs. A start sequencer controller which provides automatic start up and operation of the turbine should be developed. This system could also monitor

vital parameters such as oil pressure, turbine inlet temperature, dynamometer and turbine speeds and take corrective action should they exceed normal operating limits.

APPENDIX A
CSMP SIMULATION PROGRAM

```
// JOHNSON JOB (2844,1431), 'LAB2', CLASS=C
// *MAIN ORG=NPGVM1.2844P
// *FORMAT PR, DDNAME=PLOT.SYSVECTR, DEST=LOCAL
// EXEC CSMPXV
// X.PLOTPARM DD *
// &PLOT SCALE=7 &END
// X.SYSIN DD *
*
*   PHILIP N JOHNSON
*
INITIAL
*   VALVE PARAMETERS
*   UNLOAD VALVE
      PARAM TU=0.0
      PARAM VPU=0.0
      CONSTANT KCU=1.54
*   LOAD VALVE
      PARAM TL=0.0
      PARAM VPL=0.0
      CONSTANT KCL=1.4366666
      CONSTANT MV=1.2
*   DYNAMIC PARAMETERS
      CONSTANT WINIT=1.85
      CONSTANT NDINIT=3000.0
      CONSTANT NDFIN=3000.0
      PARAM PGAIN=.00500
      CONSTANT TDGAIN=20.0
      CONSTANT TIGAIN=.25
*   CONSTANT TIGAIN=0.0
*   CONSTANT TDGAIN=0.0
      INCON NG=36000
*
DYNAMIC
*
*   NOSORT
*   STEP SPEED DESIRED AT 5 SECONDS
*   FROM NDINIT TO NDFIN
      XXX = STEP (0.0)
      SSD=(NDFIN*XXX)+(ND*(1-XXX))
*
*   UNLOAD VALVE ACTUATOR
*
      UE=VLU-VPU
      EVU=DEADSP(-0.03,0.03,UE)
      UCV=FCNSW(EVU,0.0,0.0,MV)
      UOV=FCNSW(EVU,MV,0.0,0.0)
      TUDOT=UCV-UOV
      TU=INTGRL(0.0,TUDOT)
      VPU=KCU*TU
*   THIRD ORDER MASS FLOW RATE EQUATION
      UMDTT=((1.163393487*(VPU)**3)-(10.94193298*(VPU)**2)...
      + (53.25267468*VPU)-7.88662148))/60
      UMDOT=LIMIT(0.0,200.0,UMDTT)
*   FIRST ORDER MASS FLOW RATE EQUATION
*   UMDOT=35.07 * VPU/60
*
*   LOAD VALVE ACTUATOR
*
      LE=VLL-VPL
```

```

EVL=DEADSP(-0.03,0.03,LE)
LCV=FCNSW(EVL,0.0,0.0,MV)
LOV=FCNSW(EVL,MV,0.0,0.0)
TLDOT=LCV-LOV
TL=INTGRL(0.0,TLDOT)
VPL=KCL*TL
*
THIRD ORDER MASS FLOW RATE EQUATION
LMDTT=((4.67291038*(VPL)**3)-(41.60427991*(VPL)**2)...
+ (146.27120228*VPL)-12.07589)/60
LMDOT=LIMIT(0.0,200.0,LMDTT)
*
FIRST ORDER MASS FLOW RATE EQUATION
LMDOT= 100*VPL/60
*
INTEG DYNO WATER VOLUME
*
W2=LMDOT-UMDOT
W1=INTGRL(WINIT,W2)
W=LIMIT(0.0,48.0,W1)
*
CALC DYNAMICS
*
INERTIA * CONSTANT
F1=(.6738)*(.10471976)
*
POWER TURBINE TORQUE
F2=(-725.76+(0.0363642*(NG)))+(0.05267138...
- (4.454586E-6*(NG)))*(ND)
*
DYNAMOMETER TORQUE
FIRST ORDER EQUATION
F3=((112659205+3.1511985E-2*(W))*(ND))-205
*
SECOND ORDER EQUATION
F3=-20+(0.00046*(W/16.6)**1.3)+4.00E-6*(ND**2)
NDDOT=(1/F1)*(F2-F3)
ND=INTGRL(NDINIT,NDDOT)
BHP = ND*F3*(2.0*3.1415/33000.)
*
CONTROL SECTION
*
ANALOG TO DIGITAL CONVERSION 100 MILLISECOND CYCLE TIME
*
SST=IMPULS(0.0,0.1)
SAM=ZHOLD(SST,ND)
IF(SST.EQ.0.0) GO TO 1
*
ONLY PERFORM CONTROL AT CYCLE INTERVAL
*
ERR CAN BE FROM 0 TO 3000
ERR=SSD-SAM
*
DNSERR = DERIV(0.0,ERR)
INSERR = INTGRL(0.0,ERR)
INSERR = INSERR + (ERR/10)
*
DNDERR = DNSERR/10.0
INDERR = INSERR/10.0
INDERR = LIMIT(-100.0,100.0,INSERR)
INSERR = INDERR
V1=PGAIN*ERR
V2=PGAIN*(TIGAIN*INDERR)
V3A=PGAIN*(TDGAIN*DNDERR)
V3 = LIMIT(-10.0,10.0,V3A)
*
V3 LIMITED FOR DISPLAY ONLY NO EFFECT ON CONTROL
V=V1+V2+V3
VELX=LIMIT(-3.5,0.0,V)
VEL=-VELX
VEU=LIMIT(0.0,4.0,V)
1
CONTINUE
*
VALVE VOLTAGE LIMITS
*
DIGITAL TO ANALOG CONVERSION
VLL = ZHOLD (SST,VEL)
VLU = ZHOLD (SST,VEU)
SORT
END OF LOOP
TIMER FINTIM = 40.0, OUTDEL = 0.20, PRDEL = 0.20

```

```

TERMINAL
* PRINTER PLOT
* PRTPLT V,V1,V2,V3
* LABEL TIME VS. OUTPUT P I D
* PRTPLT ERR,INDERR,DNDERR
* LABEL TIME VS. CALC PID
* PRTPLT VLL,VPL,LMDOT,W
* LABEL TIME VS. LOADVALVE
* PRTPLT VLU,VPU,UMDOT,W
* LABEL TIME VS. UNLOADVALVE
* PRTPLT W,ND,BHP
* LABEL TIME VS WATER SPEED POWER
* PRTPLT NG,ND,NDDOT,W,F1,F2,F3
* LABEL TIME VS. SIGNALS
* VERSITEC PLOT
* OUTPUT TIME , ND
* PAGE XYPLOT
* LABEL TIME VS. DYNO SPEED
* OUTPUT TIME , W
* PAGE XYPLOT
* LABEL TIME VS. WATER WT.
* OUTPUT TIME , V,V1,V2,V3
* PAGE XYPLOT
* LABEL TIME VS. ERRORS
END
STOP
ENDJOB
/*

```

APPENDIX B
BASIC CONTROL PROGRAM

VARIABLES USED:

A	ACTUAL TURBINE SPEED
B	DESIRED TURBINE SPEED
C	TURBINE CONTROL GAIN
D	DYNO DERIVATIVE CONTROL GAIN
E	DYNO SPEED ERROR (N-M)
F	DYNO DERIVATIVE SIGNAL
G	DYNO INTEGRAL SIGNAL
H	MARINE SIMULATION FLAG (1=ON,0=OFF)
I	DYNO INTEGRAL GAIN
J	DYNO INTEGRAL
K	DYNO MAXIMUM INTEGRAL LIMIT +/-
L	TEMP STORAGE
M	DYNO ACTUAL SPEED
N	DYNO DESIRED SPEED
O	DYNO LAST CYCLE SPEED FOR DERIVATIVE
P	DYNO PROPORTIONAL GAIN
Q	TEMP STORAGE
R	DYNO MAXIMUM DERIVATIVE LIMIT
S	DYNO MINIMUM EFFECTIVE DERIVATIVE LIMIT
T	NOT USED
U	DYNO FLOW SIGNAL INVERSE (-V)
V	DYNO FLOW VALVE SIGNAL
W	TURBINE THROTTLE OUTPUT
X	DYNO VARIABLE GAIN CONTROLLER FLAG (1=ON,0=OFF)
Y	TURBINE TEMP STORAGE
Z	TURBINE INTEGRATED THROTTLE VALUE

BASIC CONTROL PROGRAM

```

10 P=2
20 I=0
30 D=20:R=4000:S=4
40 K=100
50 C=100:X=0
60 PRINT ENTER TURBINE SPEED / 10 (1700-3650) ,
70 INPUT B
80 IF B<1700 THEN PRINT TO LOW:GOTO 60
90 IF B>3650 THEN PRINT TO HIGH:GOTO 60
100 MARINE PROPULSION SIMULATION ? (YES=1,NO=0)
110 INPUT H
120 IF H=1 THEN GOTO 170
130 PRINT ENTER DYNAMOMETER SPEED (500-3000),
140 INPUT N
150 IF N<500 THEN PRINT TO LOW:GOTO 130
160 IF N>3000 THEN PRINT TO HIGH:GOTO 130

```

```
170 PRINT
180 PRINT CONTROL SETPOINTS
190 PRINT TURBINE SPEED =
200 PRINT DYNAMOMETER SPEED = ,B
210 B=B*2-3228
220 LINK #1400
230 GOTO 60
```

APPENDIX C
8073 ASSEMBLY LANGUAGE PROGRAM

ADD	INSTRUCTION	LABEL	MNEMONIC	COMMENT
1450	C4 08	TRIG A	LDA 08	07+01 =CHANNEL 1
1452	27 10 0A		LD P3 0A10	ADC CHANNEL ADDRESS
1455	CB 00		ST A, P3+00	
1457	C4 FF	DELAY A	LD A, FF	CYCLE 256 TIMES
1459	00		NOP	
145A	00		NOP	
145B	00		NOP	
145C	00		NOP	
145D	00		NOP	
145E	00		NOP	
145F	00		NOP	
1460	FC 01		SUB A, 01	SUBTRACT 1 FORM A
1462	00		NOP	
1463	00		NOP	
1464	00		NOP	
1465	00		NOP	
1466	00		NOP	
1467	00		NOP	
1468	00		NOP	
1469	7C EE		BNZ #EE	IF A<>0 GO BACK 17 STEPS
146B	00		NOP	
146C	00		NOP	
146D	00		NOP	
146E	00		NOP	
146F	27 10 0A	READ A	LD P3#1000	BASIC ADDRESS TABLE
1472	26 10 0A		LD P2, 0A10	ADC READ ADDRESS
1475	82 01		LD EA, P2+01	LOAD EA P2, 0A11, 2
1477	01		XCH E WITH A	CONDITION INPUT
1478	0C		SR EA	
1479	0C		SR EA	
147A	0C		SR EA	
147B	8B 18		ST EA, P3+18	STORE1018, 9 'M'
147E	00		NOP	
147F	00		NOP	
1480	C4 09	TRIG B	LD A 09	07+02 =CHANNEL 2
1482	27 10 0A		LD P3 0A10	ADC CHANNEL ADDRESS
1485	CB 00		ST A, P3+00	
1487	C4 FF	DELAY B	LD AFF	CYCLE 256 TIMES
1489	00		NOP	

148A 00	00		NOP		
148B 00	00		NOP		
148C 00	00		NOP		
148D 00	00		NOP		
148E 00	00		NOP		
1490 FC	01		SUB A, #01	SUBTRACT 1 FORM A	
1492 00	00		NOP		
1493 00	00		NOP		
1494 00	00		NOP		
1495 00	00		NOP		
1496 00	00		NOP		
1497 00	00		NOP		
1498 00	00		NOP		
1499 7C	EE		BNZ 20	IF A<>0 GO BACK 17 STEPS	
149B 00	00		NOP		
149C 00	00		NOP		
149D 00	00		NOP		
149E 00	00		NOP		
149F 27	00 10	READ B	LD P3 1000		
14A2 26	10 0A		LD P2 0A10	BASIC ADDRESS TABLE	
14A5 82	01		LD EA, P2+01	ADC READ ADDRESS	
14A7 01			KCH E WITH A	LOAD EA P0A11,2	
14A8 0C			SR EA	CONDITION INPUT	
14A9 0C			SR EA		
14AA 0C			SR EA		
14AB 0C			SR EA		
14AC 8B	00		ST EA, P3+00	STORE 1000, 1A	
14AE 00			NOP		
14AF 00			NOP		
14B0 84	00 00	IF H=1	LD EA0000	IF H=1 SIMULATE DYNO SPEED	
14B3 BB	0E		ADD EA P3+0E	ADD H	
14B5 6C	0D		BZ 0D	BRANCH IF ZERO +12	
14B7 B4	6C 07		LD EA076C	LOAD EA WITH 1900 DEC	
14BA B3	00		ADD EA, P3+00	ADD EA WITH 1000, 1 A	
14BC A4	02 00		LD T 0002		
14BF 0D			DIV		
14C0 8B	1A		ST EA, P3+1A	STORE EA 1001A, B N	
14C2 00			NOP		
14C3 00			NOP		
14C4 84	00 00	IF X=1	LD EA#0000	IF X=1 TURN ON VARIABLE GAIN	
14C7 BB	2E		ADDEA P3+2E	ADD X	
14C9 6C	10		BZ 10	BRANCH IF ZERO + 16	
14CB 83	18 02		LD EA, P3+00	LOAD EA WITH 1000 A	
14CD A4	90 02		LD T, #0290	LOAD T WITH 2000 DEC	
14D0 0D			DIV		
14D1 8B	20		ST EA, P3+20	STORE 1020, 1 Q TEMP	
14D3 84	05 00		LD EA0005		

14D6 BB 20		SUB EA, P3+20	SUBTRACT 1020, 1 Q
14D8 8B 1E		ST EA, P3+1E	STORE 101E, F P
14DA 00 02	IFB-A>C	NOP	
14DB 83 02		LD EA, P3+02	LOAD EA 1002, 3 B
14DD BB 00		SUB EA, P3+00	SUBTRACT 1000, 1 A
14DF 8B 30		ST EA, P3+30	STORE EA 1030, 1 Y
14E1 83 04		LD EA, P3+04	LOAD EA 1004, 5 C
14E3 09 30		LD T WITH EA	
14E4 BB 30		SUB EA, P3+30	SUBTRACT 1030, 1 Y
14E6 01 03		XCH A WITH E	LOOK AT HIGH BYTE FOR SIGN
14E7 64 03		BP 03	BRANCH IF POSITIVE +3
14E9 0B 30		LD EA WITH T	
14EA 8B 30		ST EA, P3+30	STORE EA 1030, 1 Y
14EC 00 30		NOP	
14ED 83 30	IFB-A<-C	LD EA, P3+30	LOAD EA 1030, 1 Y
14EF B3 04		ADD EA, P3+04	ADD EA 1004, 5 C
14F1 01 07		XCH A WITH E	LOOK AT HIGH BYTE FOR SIGN
14F2 64 00		BP 07	BRANCH IF POSITIVE
14F4 84 00		LD EA 0000	
14F7 BB 04		SUB EA, P3+04	SUBTRACT 1004, 5 C
14F9 8B 30	Z+Y	ST EA, P3+30	STORE EA 1030, 1 Y
14FB 83 32		LOAD EA, P3+32	LOAD EA 1032, 3 Z
14FD B3 30		ADD EA, P3+30	ADD EA 1030, 1 Y
14FF 8B 32		ST EA, P3+32	STORE EA 1032, 3 Z
1501 84 FF	4F IFZ>20480	LD EA, 4FFF	LOAD EA WITH 4095 DEC
1504 09 32		LD T WITH EA	
1505 BB 32		SUB EA, P3+32	SUBTRACT 1032, 3 Z
1507 01 03		XCH A WITH E	LOOK AT HIGH BYTE FOR SIGN
1508 64 03		BP 03	BRANCH IF POSITIVE +03
150A 0B 32		LD EA WITH T	
150B 8B 32		ST EA, P3+32	STORE EA 1032, 3 Z
150D 84 00		LD EA, 0000	LOAD EA WITH 0
1510 09 32	IF Z<0	LD T WITH EA	
1511 83 32		LD EA, P3+32	LOAD 1032, 3 Z
1513 01 03		XCH A WITH E	
1514 64 03		BP 03	BRANCH IF POSITIVE +3
1516 0B 32		LD EA WITH T	
1517 8B 32		ST EA, P3+32	STORE 0 1032, 3 Z
1519 83 32	W=Z/5	LD EA, P3+32	LOAD EA 1032, 3 Z
151B A4 05		LD T 0005	LOADT WITH 5
151E 0D 2C		DIV	
151F 8B 2C		ST EA, P3+2C	STORE EA 102C, D W
1538 0B 12		LD EA WITH T	
1539 8A 12		ST EA, P2+ 12	STORE 1012, 3 J
153B 00 12		NOP	
153C 82 14	J<-K, J=-K	LD EA, P2+ 12	LOAD 1012, 3 J
153E B2 01		ADD EA, P2+ 14	ADD 1014, 5 K
1540 01 01		XCH A WITH E	LOOK AT HIGH BYTE FOR SIGN

1640	74	0F	00	BRA	EA 0000	BRANCH UNCONDITIONALY + OF
1642	84	00	00	LD	EA WITH 0	LOAD EA WITH 0
1645	8B	02		ST	EA, P3+02	STORE EA F002, 3 CLOSE
1647	82	2A		LD	EA, P2+2A	LOAD EA WITH P102A V
1649	0C			SR	EA	ADJUST VARIABLE FOR OUTPUT
164A	0C			SR	EA	
164B	0C			SR	EA	
164C	0C			SR	EA	
164D	8B	00		ST	EA, P3+00	STORE EA F000, 1
164F	74	00		BRA	EA 0000	BRANCH UNCONDITIONALY +13
1651	84	00		LD	EA 0000	
1654	8B	00		ST	EA, P3+00	STORE EA 000, 1 CLOSE
1656	0C	28		LD	EA, P2+28	LOAD E 1028, 9 U
1658	0C			SR	EA	ADJUST VARIABLE FOR DAC
1659	0C			SR	EA	
165A	0C			SR	EA	
165B	0C			SR	EA	
165C	8B	02		ST	EA, P3+02	STORE EA F002, 3
165E	00			NOP		
165F	00			NOP		
1660	00			NOP		
1661	00			NOP		
1662	00			NOP		
1694	82	2C	THROTTLE	LD	EA, P2+2C	LOAD EA 102C, D W
1696	0C			SR	EA	CONDITION OUTPUT
1697	0C			SR	EA	
1698	0C			SR	EA	
1699	0C			SR	EA	
169A	8B	04		ST	EA, P3+04	STORE AT DAC C
169C	00			NOP		
169D	00			NOP		
169E	00			NOP		
169F	00			NOP		
16A0	8B	06	TRIG DAC	ST	EA, P3+06	OUTPUT DAC VOLTAGES
16A2	27	00	0A BUTTON	LD	P3, 0A00	
16A5	C3	01		LD	A, P3+00	CHECK HIGH BIT
16A7	D4	80		AND	A WITH 80	BRANCH IF NOT ZERO
16A9	7C	11		BNZ	11	
16AB	26	00	F0	LD	P2 F000	
16AE	84	00	00	LD	EA 0	CLOSE VALVE
16B1	8A	00		ST	EA, P2+00	CLOSE VALVE
16B3	8A	02		ST	EA, P2+02	EXECUTE
16B5	8A	06		ST	EA, P2+06	RETURN TO BASIC
16B7	5C			RIN		
16B8	00			NOP		
16B9	00			NOP		
16BA	27	00	CLOCK	LD	P3, 0A00	TURN ON LED
16BD	C4	FF		LD	A FF	

16BF	CB	00	ST A, P3+00	READ CLOCK
16C1	C3	00	LD A, P3+00	CHECK HIGH BIT
16C3	D4	80	AND A WITH 80	BRANCH IF NOT ZERO -5
16C5	7C	FA	BNZ FA	TURN OFF LED
16C7	CB	00	ST A, P3+00	
16C9	30	CA 02	LD EA WITH PC	RELATIVE PROGRAM START
16CD	BC		SUB EA 02CA	
16D0	4C		XCH PC WITH EA	

MEMORY DUMP

1100	30	10	P=2	20	I=0	30
1110	30	40	K=50	50	X=0	40
1120	30	60	C=100	70	PR	80
1130	30	80	PR ENTER			90
1140	30	100	TURBINE SPEED	/		110
1150	30	120	(17000-3650)			130
1160	30	140	INPUT B.100			150
1170	30	160	IF B<17000 THEN PR			170
1180	30	180	TO LOW:GO 10.110			190
1190	30	200	IF B>36500 THEN PR			210
11A0	30	220	TO HIGH:GO 10.1			230
11B0	30	240	PR.130 PR			250
11C0	30	260	PROPULSION EMULA			270
11D0	30	280	TION ? (1=YES 0=			290
11E0	30	300	NO).140 INPUT			310
11F0	30	320	H.130 IF H=1 THE			330
1200	30	340	N.G0210.160 PR			350
1210	30	360	170 PR ENTER			370
1220	30	380	DYNAMOMETER SPEE			390
1230	30	400	D (500-3000).1			410
1240	30	420	80 INPUT N.190 I			430
1250	30	440	F N<500 THEN PR			450
1260	30	460	TO LOW:GO 160. TH			470
1270	30	480	200 IF N>3000			490
1280	30	500	EN PR TO HIGH:			510
1290	30	520	GO 160.210 PR			530
12A0	30	540	.220 PR CONTRO			550
12B0	30	560	L SETPOINTS.230			570
12C0	30	580	PR.240 PR			590
12D0	30	600	TURBINE = B,			610
12E0	30	620	X 10.250 PR			630
12F0	30	640	.260 PR DYNAMOM			650
1300	30	660	ETER = N.265 B			670
1310	30	680	=B*2-3228.270 LI			690

13340	0	NK#1400	280	GO	1
13350	0	0.290	STOP	L	
13360	0	INK#1500	280	GO	
13370	0	70.290	STOP	27	
13380	0	LINK#1500	280		
13390	0	GO	70.290	STOP	28
13400	0	GO	70.290	STOP	
13410	0				
13420	0				
13430	0				
13440	0				
13450	0				
13460	0				
13470	0				
13480	0				
13490	0				
13500	0				
13510	0				
13520	0				
13530	0				
13540	0				
13550	0				
13560	0				
13570	0				
13580	0				
13590	0				
13600	0				
13610	0				
13620	0				
13630	0				
13640	0				
13650	0				
13660	0				
13670	0				
13680	0				
13690	0				
13700	0				
13710	0				
13720	0				
13730	0				
13740	0				
13750	0				
13760	0				
13770	0				
13780	0				
13790	0				
13800	0				
13810	0				
13820	0				
13830	0				
13840	0				
13850	0				
13860	0				
13870	0				
13880	0				
13890	0				
13900	0				
13910	0				
13920	0				
13930	0				
13940	0				
13950	0				
13960	0				
13970	0				
13980	0				
13990	0				
14000	0				
14010	0				
14020	0				
14030	0				
14040	0				
14050	0				
14060	0				
14070	0				
14080	0				
14090	0				
14100	0				
14110	0				
14120	0				
14130	0				
14140	0				
14150	0				
14160	0				
14170	0				
14180	0				
14190	0				
14200	0				
14210	0				
14220	0				
14230	0				
14240	0				
14250	0				
14260	0				
14270	0				
14280	0				
14290	0				
14300	0				
14310	0				
14320	0				
14330	0				
14340	0				
14350	0				
14360	0				
14370	0				
14380	0				
14390	0				
14400	0				
14410	0				
14420	0				
14430	0				
14440	0				
14450	0				
14460	0				
14470	0				
14480	0				
14490	0				
14500	0				
14510	0				
14520	0				
14530	0				
14540	0				
14550	0				
14560	0				
14570	0				
14580	0				
14590	0				
14600	0				
14610	0				
14620	0				
14630	0				
14640	0				
14650	0				
14660	0				
14670	0				
14680	0				
14690	0				
14700	0				
14710	0				
14720	0				
14730	0				
14740	0				
14750	0				
14760	0				
14770	0				
14780	0				
14790	0				
14800	0				
14810	0				
14820	0				
14830	0				
14840	0				
14850	0				
14860	0				
14870	0				
14880	0				
14890	0				
14900	0				
14910	0				
14920	0				
14930	0				
14940	0				
14950	0				
14960	0				
14970	0				
14980	0				
14990	0				
15000	0				
15010	0				
15020	0				
15030	0				
15040	0				
15050	0				
15060	0				
15070	0				
15080	0				
15090	0				
15100	0				
15110	0				
15120	0				
15130	0				
15140	0				
15150	0				
15160	0				
15170	0				
15180	0				
15190	0				
15200	0				
15210	0				
15220	0				
15230	0				
15240	0				
15250	0				
15260	0				
15270	0				
15280	0				
15290	0				
15300	0				
15310	0				
15320	0				
15330	0				
15340	0				
15350	0				
15360	0				
15370	0				
15380	0				
15390	0				
15400	0				
15410	0				
15420	0				
15430	0				
15440	0				
15450	0				
15460	0				
15470	0				
15480	0				
15490	0				
15500	0				
15510	0				
15520	0				
15530	0				
15540	0				
15550	0				
15560	0				
15570	0				
15580	0				
15590	0				
15600	0				
15610	0				
15620	0				

APPENDIX D
GAGE FACE GENERATION PROGRAM

```

LARGE GAGE FACE PROGRAM
*****
C * LT PHILIP N. JOHNSON U.S.N.
C * 17 JANUARY 1985
C * NAVAL POSTGRADUATE SCHOOL, MONTEREY, CA
C * *****
C CALL DISPLA GRAPHICS PACKAGE
C REAL MIN,MAX,DIV,SDIV,A1,AR2,AR3,P1X,P1Y,P2X,P2Y,L1,L2,C1,C2
C REAL CAP,AR4,P3X,P3Y,XL1,XL2,F1
C CALL TEK618
C CALL COMPRS
C ***** START LEVEL 1 WORK *****
C CALL NOBRDR
C CALL SWISSM
C CALL BASALF({'STANDARD'})
C CALL MIXALF({'L/CSTD'})
C CALL HWROT({'MOVIE'})
C CALL HWSAL({'SCREEN'})
C CALL PAGE(8.5,11.0)
C W=5.75
C CALL AREA2D(W,9.25)
C ***** START LEVEL 2 WORK *****
C CALL FRAME
C CALL SHDCHR(45.,1,.005,1)
C SCA = 1.0
C CALL HEIGHT(SCA/5.5)
C XL1=XMESS('OIL TEMPERATURES',15)
C CALL MESSAG('OIL TEMPERATURES',15,(W/2.0)-(XL1/2),
C *8.7*SCA)
C XL2=XMESS('DEG. F',6)
C CALL MESSAG('DEG. F',6,(W/2.0)-(XL2/2),7.2*SCA)
C CALL HEIGHT(SCA/7.0)
C CALC NUMBER WIDTH
C XL3=XMESS('12345',4)
C XL3=XL3/2.0
C ***** START LEVEL 3 WORK *****
C MIN = 0.0
C MAX = 300.0

```

```

DIV= 50.0
SDIV=5.0
L1=((MAX-MIN)/DIV)
L2=((MAX-MIN)/SDIV)
C1=38.0
C2=100.0
VL= 5
DO 100 I=1,L1+1
CAP=DIV*(I-1)
A1=(90+(C1/2.0))-(C1/L1*(I-1))
AR2=A1/57.3
PIY=((8.0*SCA*COS(AR2))+(W/2.0))
PIY=((8.0*SCA*SIN(AR2))
AR3=((90+(C2/2.0))+180.0)-(C2/L1*(I-1))/57.3
P2X=((VL*SCA*COS(AR3))+PIY)
P2Y=((VL*SCA*SIN(AR3))+PIY)
AR4=AR3+3.14159
P3X=((25*SCA*COS(AR4))+PIY)
P3Y=PIY
P3Y=((25*SCA*SIN(AR4))+PIY)
P3Y=((25*SCA)+PIY
ICAP=FIX(CAP)
XL3=XINT(ICAP)
XL3=XL3/2.0
CALL INTNO (ICAP,P3X-XL3,P3Y)
CALL VECTOR (PIY,PIY,P2X,P2Y,1100)
CONTINUE
VL= 25
DO 200 I=1,L2+1
A1=(90+(C1/2.0))-(C1/L2*(I-1))
AR2=A1/57.3
PIY=((8.0*SCA*COS(AR2))+(W/2.0))
PIY=((8.0*SCA*SIN(AR2))
AR3=((90+(C2/2.0))+180.0)-(C2/L2*(I-1))/57.3
P2X=((VL*SCA*COS(AR3))+PIY)
P2Y=((VL*SCA*SIN(AR3))+PIY)
CALL VECTOR (PIY,PIY,P2X,P2Y,1100)
CONTINUE
CALL ENDPL(0)
CALL DONEPL
STOP
END
C *****

```

SMALL GAGEFACE PROGRAM

C *****
C *


```

DO 100 I=1, L1+1
CAP=DIV*(I-1)
AL=(90+(C1/2.0))-(C1/L1*(I-1))
AR2=A1/57.3
PIY=((YNEED*SCA*COS(AR2))+(W/2.0))
PIY=((YNEED*SCA)
AR3=((90+(C2/2.0))+180.0)-(C2/L1*(I-1))/57.3
P2X=((VL*SCA*COS(AR3))+PIY)
P2Y=((VL*SCA*SIN(AR3))+PIY)
AR4=AR3+3.14159
P3X=((25*SCA*COS(AR4))+PIY)
P3Y=((250*SCA*SIN(AR4))+PIY)
ICAP=FIX(CAP)
CALL INTNO (ICAP, P3X-XL3, P3Y)
CALL VECTOR (PIX, PIY, P2X, P2Y, 1100)
CONTINUE
VL=065
DO 200 I=1, L2+1
AL=(90+(C1/2.0))-(C1/L2*(I-1))
AR2=A1/57.3
PIY=((YNEED*SCA*COS(AR2))+(W/2.0))
PIY=((YNEED*SCA*SIN(AR2))
PIY=((YNEED*SCA)
AR3=((90+(C2/2.0))+180.0)-(C2/L2*(I-1))/57.3
P2X=((VL*SCA*COS(AR3))+PIY)
P2Y=((VL*SCA*SIN(AR3))+PIY)
CALL VECTOR (PIX, PIY, P2X, P2Y, 1100)
CONTINUE
CALL ENDPL(0)
CALL DONEPL
STOP
END

```

100

C

200

APPENDIX E
DATA ACQUISITION PROGRAM

```

HP 85 PROGRAM
READ 5
10JP(17),T(10),K(10),N(10),S(10)
20 PRINTER IS 10,120
30 CLEAR
40 DISP ENTER DATE;;
50 INPUT K1$
60 DISP ENTER TIME;;
70 INPUT K2$
80 DISP ENTER BAROMETRIC PRESS (IN.HG.);
90 INPUT P0
100 DISP
110 DISP ENTER FUEL SPECIFIC GRAVITY ;
120 INPUT S1
125 CLEAR
130 DISP
140 DISP ENTER UPPER ROTOMETER READING;
150 INPUT R2
160 DISP
170 DISP ENTER LOWER ROTOMETER READING;
180 INPUT R1
190 SETTIME 0,0
200 IMAGE K,3D.2D,K
210 IMAGE K,4D.D,K
220 IMAGE K,1D.4D,K
230 IMAGE K,5D.,K
240 ABORTIO 7
250 SET TIMEOUT 7:1000
260 OUTPUT 705 SC0 1-9 ,CHR$(13)
270 FOR I=1 TO 9
280 ENTER 705 USING %,K ; B$
31)
300 NEXT I
310 ABORTIO 7
320 SET TIMEOUT 7:1000
330 OUTPUT 705 SC0 10-16 ,CHR$(13)
340 FOR J=1 TO 7
350 ENTER 705 USING %,K ; C$
41)
370 NEXT J

```

```

380 ABORTIO 7
390 REM SUBROUTINE FOR TYPE T
400 FOR I=0 TO 9
410 GOSUB 450
420 T(I)=T0
430 NEXT I
440 GOTO 590
450 REM
460 OUTPUT 709 ;AC,I
470 M0=1008609
480 M1=25727.94369
490 M2=-767345.8295
500 M3=78025595.81
510 M4=-924786589
520 M5=697688000000
530 M6=-2.66192E13
540 M7=3.94078E14
550 ENTER 709 ; V
560 T0=M0+V*(M1+V*(M2+V*(M3+V*(M4+V*(M5+V*(M6+V*(M7))))))
570 T0=9/5*T0+32
580 RETURN
590 REM INPUT TYPE K
600 FOR I=10 TO 19
610 GOSUB 650
620 K(I-9)=K0
630 NEXT I
640 GOTO 800
650 REM
660 L0=226584602
670 L1=24152.109
680 L2=67233.4248
690 L3=2210340.682
700 L4=-860963914.9
710 L5=48350600000
720 L6=-1.18452E12
730 L7=1.3869E13
740 L8=-6.33708E13
750 OUTPUT 709 ;AC,I
760 ENTER 709 ; V
770 K0=L0+V*(L1+V*(L2+V*(L3+V*(L4+V*(L5+V*(L6+V*(L7+V*(L8))))))
780 K0=9/5*K0+32
790 RETURN
800 REM SPEEDS ETC.
810 FOR I=20 TO 28
820 OUTPUT 709 ;AC I
830 ENTER 709 ; N(I-19)
840 NEXT I
850 REM S(1)=DYNO SPEED

```


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1820	REM	R1	=	LOWER ROTOMETER READING
1830	REM	R2	=	UPPER ROTOMETER READING
1840	REM	S1	=	SPECIFIC GRAVITY OF FUEL
1850	REM	P0	=	ATM. PRESSURE
1860	REM			
1870	REM			CALCULATED VALUES
1880	REM			
1890	REM	T0	=	AVERAGE INLET TEMP
1900	REM	D0	=	THETA CORRECTION FACTOR
1910	REM	D1	=	DELTA CORRECTION FACTOR
1920	REM	P9	=	CORRECTED TURBINE INLET PRESSURE
1930	REM	A1	=	API AT 60 DEG
1940	REM	S2	=	CORRECTED SPECIFIC GRAVITY TO 60 DEG F.
1950	REM	Q2	=	LOWER HEATING VALUE
1960	REM	Q3	=	CORRECTED TORQUE
1970	REM	N3	=	CORRECTED COMPRESSOR SPEED
1980	REM	N4	=	CORRECTED DYNAMOMETER SPEED
1990	REM	B1	=	CORRECTED BRAKE HORSE POWER
2000	REM	M1	=	MASS FLOW RATE LOWER FLOAT
2010	REM	M2	=	MASS FLOW RATE UPPER FLOAT
2020	REM	C1	=	ROTOMETER CONVERSION FACTOR
2030	REM	M3	=	CORRECTED MASS FLOW RATE UPPER FLOAT
2040	REM	M4	=	CORRECTED MASS FLOW RATE LOWER FLOAT
2050	REM	M5	=	CORRECTED MASS FLOWRATE
2060	REM	B2	=	BRAKE SPECIFIC FULE CONSUMPTION
2070	REM	B3	=	BRAKE THERMAL EFFICIENCY
2080	REM	E1	=	AVERAGE INLET BELL THROAT PRESS
2090	REM	E2	=	INLET BELL THROAT PRESS
2100	REM	D2	=	DELTA PRESSURE IN THROAT
2110	REM	M6	=	MASS FLOW RATE OF AIR
2120	REM	M7	=	CORRECTED MASS FLOW RATE OF AIR
2130	REM	A2	=	AIR FUEL RATIO
2140	REM	K9	=	AVE EXHAUST TEMP
2150	REM	O1	=	AVE COMPRESSOR DISCHARGE PRESS
2160	REM	O2	=	PRESSURE RATIO
2170	REM	O3	=	IDEAL THERMAL EFFICIENCY
2180	REM	O4	=	AVE COMPRESSOR DISCHARGE TEMP
2190	REM	O5	=	AVE TURBINE INLET TEMP
2200	REM	T1	=	84.8
2210	REM	T2	=	88
2220	REM	T3	=	91.7
2230	REM	T4	=	80.4
2240	REM	P0	=	29.7326
2250	REM	P1	=	1.1546
2260	REM	S1	=	.8546
2270	REM	T9	=	71
2280	REM	P2	=	1
2290	REM	Q1	=	129.7

```

2300 N1=27000
2310 N2=1500
2320 R1=99
2330 R2=204
2340 P3=10.6
2350 P4=10.6
2360 P5=26.5
2370 P6=26.5
2380 K5=524
2390 K6=508
2400 K7=530
2410 K8=502
2420 K1=1200
2430 K2=1220
2440 K3=1190
2450 K4=1200
2460 T5=234.4
2470 T6=227.6
2480 T7=232
2490 T8=234
2500 REM CALCULATE CORRECTION FACTORS
2510 T0=(T(0)+T(1)+T(2)+T(3))*25+460
2520 D0=T0/520
2530 P9=P0*(14.696/29.92)+(P(15)+P(14))/2*(14.696/406.92)
2540 D1=P9/14.696
2550 REM CALCULATE LOWER HEATING VALUE
2560 S2=S1+(T(8)-60)/3600
2570 A1=141.5/S2-131.5
2580 Q2=16380+60*A1
2590 REM CALCULATE CORRECTED TORQUE
2600 Q3=S(4)/D1
2610 REM CALCULATE CORRECTED COMPRESSOR SPEED
2620 N3=S(2)/SOR(D0)
2630 REM CALCULATE CORRECTED DYNAMOMETER SPEED
2640 N4=S(1)/SOR(D0)
2650 REM CALCULATE CORRECTED BRAKE HORSEPOWER
2660 B1=2*3.141592*Q1*S(1)/(3000*D1*SQR(D0))
2670 REM CALCULATE MASS FLOW RATE OF FUEL
2680 M2=607954*R2-4.627907
2690 M1=1.17442*R1-8.556818
2700 C1=SQR((7.82-S1)*S1)/SOR((7.82-.8637)*.8637)
2710 M3=C1*M1
2720 M4=C1*M2
2730 M5=(M3+M4)/2/(D1*SQR(D0))
2740 REM CALCULATE BRAKE SPECIFIC FUEL CONSUMPTION
2750 B2=M5/B1
2760 REM CALCULATE BRAKE THERMAL EFFICIENCY
2770 B3=2545/(B2*Q2)

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2780 REM CALCULATE MASS FLOW RATE OF AIR
2790 E1 = -((P(16)+P(11))/2)
2800 E2 = P0*(14.696/29.92)-E1*(14.696/406.92)
2810 D2 = P9-E2
2820
2830 M6 = 8.02*.98*P9*21.8956/SQR(53.34*T0)*SQR(D2/P9-1.5/1.4*(D2/P9) 2)
2840 M7 = M6*SQR(D0)/D1
2850 REM CALCULATE AIR/FUEL RATIO
2860 A1 = M7*3600/M5
2870 REM CALCULATE EXHAUST GAS TEMP
2880 K9 = (K(5)+K(6)+K(7)+K(8))/4+460
2890 REM CALCULATE PRESSURE RATIO
2900 O1 = P9*(P(5)+P(2))/2*14.696/29.92
2910 O2 = O1/P9
2920 REM CALCULATE COMPRESSOR DISCHARGE TEMP
2930 O4 = .25*(T(4)+T(5)+T(6)+T(7))+460
2940 REM CALCULATE TURBINE INLET TEMP
2950 O5 = (K(1)+K(2)+K(3)+K(4))/4+460
2960 REM CALCULATE IDEAL THERMAL EFFICIENCY
2970 O3 = 1-1/O2
2980 PRINT ANALYSIS
2990 PRINT
3000 PRINT USING 210 ; COMPRESSOR INLET TEMP. = ,T0, DEG. R.
3010 PRINT THETA = ,D0
3020 PRINT USING 200 ; COMPRESSOR INLET PRESSURE = ,P9, PSIA
3030 PRINT DELTA = ,D1
3040 PRINT USING 230 ; LOWER HEATING VALUE = ,Q2, BTU/LBM
3050 PRINT USING 230 ; CORRECTED TORQUE = ,Q3, FT-LB
3060 PRINT USING 230 ; CORRECTED COMPRESSOR SPEED = ,N3, RPM
3070 PRINT USING 230 ; CORRECTED DYNAMOMETER SPEED = ,N4, RPM
3080 PRINT USING 230 ; CORRECTED BRAKE HORSE POWER = ,B1, HP
3090 PRINT USING 200 ; CORRECTED MASS FUEL FLOW = ,M5, LBM/HR
3100 PRINT USING 210 ; BRAKE SPECIFIC FUEL CONSUMPTION = ,B2, LBM/HP-HR
3110 PRINT USING 200 ; BRAKE THERMAL EFF. = ,B3*100, PERCENT
3120 PRINT USING 200 ; CORRECTED MASS AIR FLOW = ,M7, LBM/SEC
3130 PRINT AIR FUEL RATIO = ,A1
3140 PRINT USING 210 ; AVERAGE EXHAUST TEMP. = ,K9, DEG R.
3150 PRINT COMPRESSOR PRESSURE RATIO = ,O2
3160 PRINT USING 210 ; COMPRESSOR DISCHARGE TEMP. = ,O4, DEG. R.
3170 PRINT USING 210 ; AVERAGE TURBINE INLET TEMP. = ,O5, DEG. R.
3180 PRINT USING 200 ; IDEAL COMPRESSOR EFF. = ,O3*100, PERCENT
3190 GOTO 125

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SAMPLE OUTPUT

TURBINE ANALYSIS PROGRAM

DATE: 6 JUNE 1985

TIME: 1300

PRESSURES

BAROMETRIC PRESSURE = 30.13 IN.HG.
CELL PRESSURE FRONT = -.04 IN.H2O
CELL PRESSURE REAR = -.07 IN.H2O
CELL PRESSURE AVERAGE = -.06 IN.H2O
AIRFLOW BELL PRESSURE RIGHT = -9.62 IN.H2O
AIRFLOW BELL PRESSURE LEFT = -9.53 IN.H2O
AIRFLOW BELL PRESSURE AVERAGE = -9.58 IN.H2O
COMPRESSOR DISCHARGE PRESSURE RIGHT = 22.82 IN.HG
COMPRESSOR DISCHARGE PRESSURE LEFT = 22.66 IN.HG
COMPRESSOR DISCHARGE PRESSURE AVERAGE = 22.74 IN.HG
NOZZLE BOX PRESSURE = 22.41 IN.HG

TEMPERATURE

COMPRESSOR INLET TEMP. A = 68.9 DEG. F
COMPRESSOR INLET TEMP. B = 70.8 DEG. F
COMPRESSOR INLET TEMP. C = 71.6 DEG. F
COMPRESSOR INLET TEMP. D = 72.4 DEG. F
COMPRESSOR INLET TEMP. AVERAGE = 70.9 DEG. F
COMPRESSOR DISCHARGE TEMP. RIGHT A = 201.6 DEG. F
COMPRESSOR DISCHARGE TEMP. LEFT A = 203.3 DEG. F
COMPRESSOR DISCHARGE TEMP. RIGHT B = 196.7 DEG. F
COMPRESSOR DISCHARGE TEMP. LEFT B = 202.8 DEG. F
COMPRESSOR DISCHARGE TEMP. AVERAGE = 201.1 DEG. F
FUEL TEMP = 68.1 DEG. F
TURBINE INLET TEMP. RIGHT A = 1181.4 DEG. F
TURBINE INLET TEMP. LEFT A = 1341.9 DEG. F
TURBINE INLET TEMP. RIGHT B = 1200.7 DEG. F
TURBINE INLET TEMP. LEFT B = 1345.7 DEG. F
TURBINE INLET TEMP. AVERAGE = 1267.4 DEG. F
EXHAUST TEMP. RIGHT A = 902.6 DEG. F
EXHAUST TEMP. LEFT A = 947.0 DEG. F
EXHAUST TEMP. RIGHT B = 893.7 DEG. F
EXHAUST TEMP. LEFT B = 930.0 DEG. F
EXHAUST TEMP. AVERAGE = 918.3 DEG. F

SPEEDS ETC.

UPPER ROTOMETER = 185.0
LOWER ROTOMETER = 90.0
SPECIFIC GRAVITY = .8650
TURBINE SPEED = 24881. RPM
DYNAMOMETER SPEED = 985. RPM
DYNAMOMETER TORQUE = 138. FT-LB
FUEL FLOW SENSOR = 106. UPPER ROTOMETER

ANALYSIS

COMPRESSOR INLET TEMP. = 530.9 DEG. R.
THETA = 1.02097892304
COMPRESSOR INLET PRESSURE = 14.60 PSIA
DELTA = .993601469311
LOWER HEATING VALUE = 18398. BTU/LBM
CORRECTED TORQUE = 139. FT-LB
CORRECTED COMPRESSOR SPEED = 24624. RPM
CORRECTED DYNAMOMETER SPEED = 975. RPM
CORRECTED BRAKE HORSE POWER = 24. HP
CORRECTED MASS FUEL FLOW = 112.58 LBM/HR
BRAKE SPECIFIC FUEL CONSUMPTION = 4.6 LBM/HP-HR
BRAKE THERMAL EFF. = 2.98 PERCENT
CORRECTED MASS AIR FLOW = 2.30 LBM/SEC
AIR FUEL RATIO = 73.5663700269
AVERAGE EXHAUST TEMP. = 1378.3 DEG R.
COMPRESSOR PRESSURE RATIO = 1.76492110916
COMPRESSOR DISCHARGE TEMP. = 661.1 DEG. R.
AVERAGE TURBINE INLET TEMP. = 1727.4 DEG. R.
IDEAL COMPRESSOR EFF. = 14.98 PERCENT

APPENDIX F

DIGITAL TO ANALOG INTERFACE

The digital to analog interface accomplishes the conversion of digital signals to analog voltages, decodes bus addresses, and provides computer and actuator signal isolation. The design requirements dictated the need for a design having a minimum of three independent analog output ports. The precision required varied from the throttle output port, which needs 12 bit or 4096 count accuracy, to the valves which only required 8 bit or 256 count accuracy. For simplicity the interface card was designed for 12 bit accuracy on all ports.

The interface uses a 74LS139 to decode addresses F000 to F008 from the bus [Ref. 6]. Three resident DAC1230's are connected directly to the data bus and receive information in two successive 8 bit transfers. Address F006 is used to trigger all output ports and execute a transfer of digital information to the DAC. The output is buffered by a MC3403 operational amplifier which isolates the DAC's from the environment. A gain adjustment controls the voltage output of an LM337 which is used as a reference voltage for all DAC's. This gain pot permits the adjustment of the range of voltage corresponding to a 0 to 4095 change in input, and provides approximately 1 to 10 volt full scale output. A zero adjustment provides zeroing capability to the DAC's output and may be adjusted from -0.1 volts to 2 volts.

Figure F.1 shows the general layout of the card. The adjustment pots and test connections were placed at the bottom edge of the card to permit adjustment while in operation. Figure F.2 show the wiring of one of three DAC's. Wiring of other two DAC's are similar.

Figure F.2 Digital to Analog Wiring Diagram

APPENDIX G

VALVE POSITIONER CONTROLLER

The valve position controller utilizes a feedback potentiometer, operational amplifier, and solid state relays to apply voltage to the motor windings phased in such a way as to move the motor in the desired direction.

The desired valve position has two possible inputs depending on the position of the selector switch (figure G.1)

If the selector switch is in the computer position, the input voltage is provided by the DAC's of the microprocessor interface board. If the selector switch is in the manual position, the desired valve position is provided by an adjustable voltage bridge. This bridge is a variable resistor of 20k ohm resistance connected across +5 volts and ground. It's output therefore may be varied between 0 and +5 volts. From the selector switch the desired valve position in the form of a voltage between 0 and 5 volts which is compared with actual valve position. Actual valve position is generated in the same manner as manual desired valve position except the potentiometer is connected to the valve via a gearing arrangement. This position feedback system measures valve position in the form of a voltage, 0 volts corresponding to fully closed and +5 volts to fully open. This voltage is also fed to a voltage follower circuit which is used to drive a meter which mimics valve position.

Both the desired position from the selector switch and the actual position from the valves are fed into an operational amplifier which has a resistor network to provide windowing in order to prevent valve oscillation. The output of the operational amplifiers drive transistors which in turn drive

a light emitting diode and a solid state relay which close to provide 120 volt alternating current to a resistor and capacitor network. Thus the valve is turned in the correct direction [Ref. 8].

APPENDIX H

POWER SUPPLY

The power supply used for the operators panel is a converted 28 volt power supply. Printed circuit board traces were cut and regulators installed to provide ± 15 and ± 18 volts. An additional transformer was installed to provide +5 volts. Each supply is capable of delivering 1 amp of current at the voltage specified. Component locations are shown in figure H.1 and the circuit schematic is shown in figure H.2

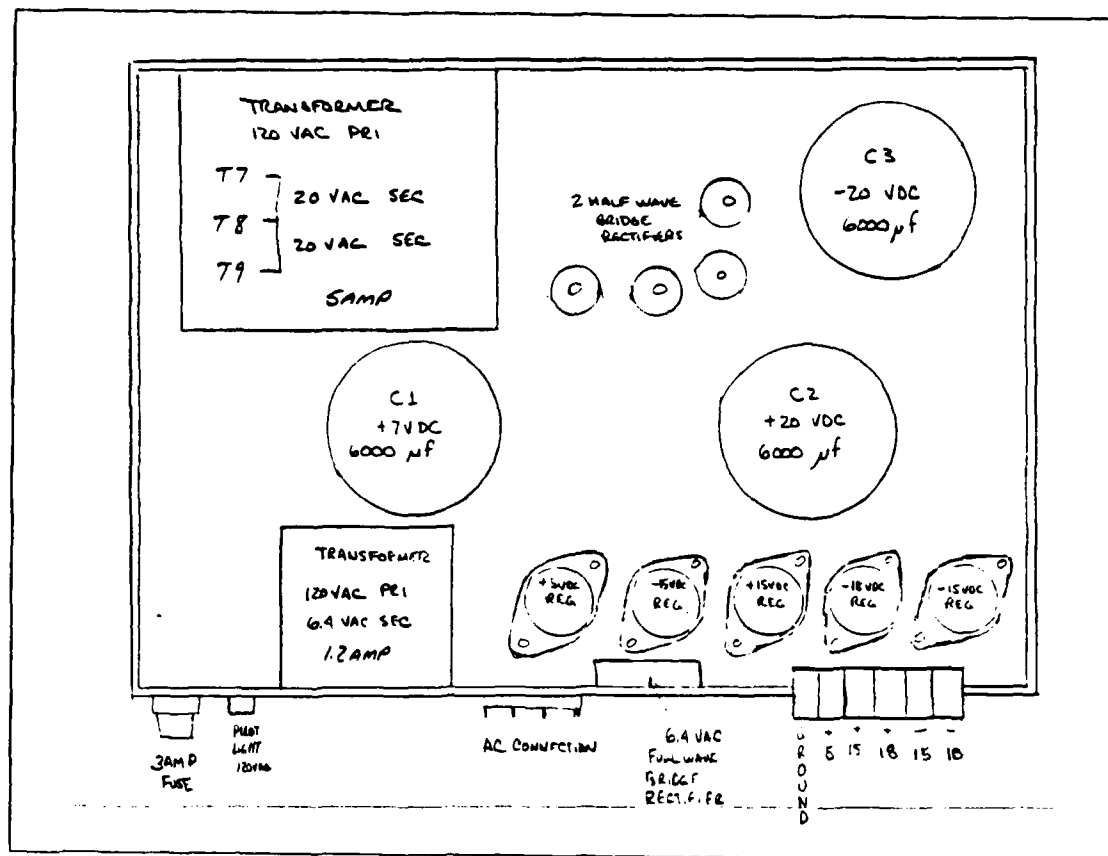


Figure H.1 Power supply layout.

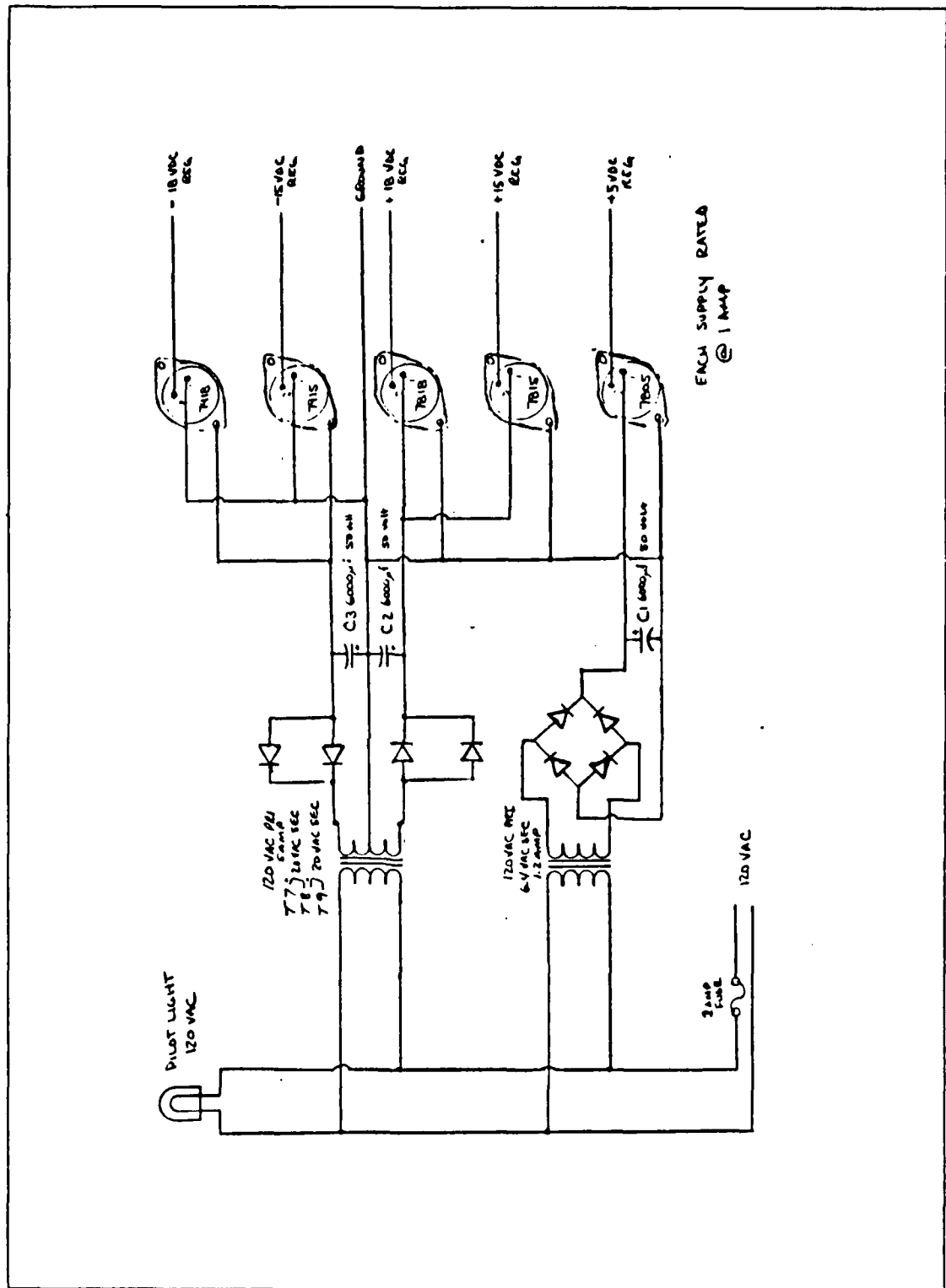


Figure H.2 Power Schematic Layout

APPENDIX I
TURBINE OPERATING PROCEDURES

TURBINE START CHECKLIST

1. Align and start low pressure air compressor. (a minimum of 50 psig is required for turbine operation.)
2. Check in-ground fuel storage tank for adequate fuel. (a minimum of 50 gallons is required to prevent loss of suction on transfer pump.) Note: The sounding rod is stored behind the student gage panel.
3. Close power breaker for fuel transfer pump, turn on fuel transfer switch located on operators panel, and push start button on controller box. Verify that fuel transfer pump maintains a normal level in head tank.
4. Turn on power to the student instrument panel (plug inside).
5. Turn on control computer and terminal.
6. Turn on computer acquisition equipment, and load acquisition program.
7. Insure 120 vac and 24 vdc fuel trips, glow coil, igniter and starter switches are in the off position.
8. Turn 24 vdc power on. (note: damper position indicators should illuminate.)
9. Turn dynamometer flow valve control knobs to the full counter clockwise position.
10. Place both dynamometer and gas turbine controllers in manual control.
11. Turn 120 vac power and blower on. (note that instruments should respond to their current sensed values and valve motors should be at rest.)
12. Turn test cell lighting on using wall switch adjacent to door.

13. Check module for loose debris, check turbine for correct oil level and check both turbine and dynamometer for any loose sensing or control connections.
14. Open the following valves located behind the dynamometer. a) Lube oil cooler water supply valve. b) Fuel oil supply valve. c) Air supply valve. d) Dynamometer cooling water supply valve.
15. Adjust dynamometer shell pressure to 4 psig.
16. Verify dynamometer load cell pressure regulator is set at 37 psig.
17. Open inlet and exhaust dampers, and insure front and front-side doors are shut tightly.
18. Exit and shut rear cell door.
19. Don protective hearing gear.
20. Review operating parameters for turbine and dynamometer. a) Dynamometer temp should not exceed 140 deg. f. b) Dynamometer speed should not exceed 3300 rpm. c) Dynamometer should not be run empty. d) Dynamometer shell pressure should not exceed 6 psig. e) Dynamometer speeds below 960 rpm may not be obtainable at high compressor speeds. f) Turbine oil temp should not exceed 220 deg.f. g) Turbine speed should not exceed 36500 rpm. h) Turbine speed must be maintained above 17000 rpm. i) turbine exhaust temp must not exceed 1350 deg f. j) Rapid accelerations/deccelerations should be avoided. k) Minimum oil pressure is 40 psig. l) Maximum starter operation is 30 seconds.
21. Remember there are no safety trips on this equipment and the operator is responsible for keeping the equipment within safe operating limits.
22. Turbine start procedure. a) Verify throttle control by cycling actuator then set throttle at 1.5. (this corresponds to 20000 rpm idle) b) Cycle dynamometer valves to verify control then oper load valve to position 5 for 5 seconds, then close. c) Turn on glow coil,

igniters and 120 vac fuel solenoid and depress the master start button for 5 seconds, then switch on starter. d) When turbine speed reaches 5000 rpm turn on 24 vdc fuel solenoid, turbine speed should continue to rise, at 15000 rpm release master start button and turn off starter, glow coil, and igniters, turbine speed of 20000 rpm should be attained within 5 seconds of DC fuel valve opening. e) Should turbine fail to start or is slow in acceleration to 20000 rpm (greater than 5 seconds), close fuel trips and turn off igniters and glow coils but continue to motor turbine for 15 seconds. Secure engine and determine cause of failure to start before attempting to start again. (Note: a minimum of 15 minutes is required to allow starter to cool between starts.) f) When turbine reaches idle verify operating parameters are within limits, especially. 1) Dynamometer speed. 2) Oil pressure. 3) Alternator current. 4) Fuel pressure. g) View cell form operating window and check for abnormalities.

23. Enter turbine speed and dynamometer selection on control terminal (note flashing internal led indicates correct computer operation.)
24. Switch dynamometer control from manual to computer on operators console and observe that speed attains set point.
25. Manually bring turbine speed to slightly above the selected set point and observe computer throttle position decreases from full throttle, when manual and computer signals are matched, switch throttle to computer control.
26. To select new operating speeds simply depress the white reset button on the microprocessor controller. (Note: while new values are being entered, dynamometer valves are closed and throttle remains fixed.)

2/2

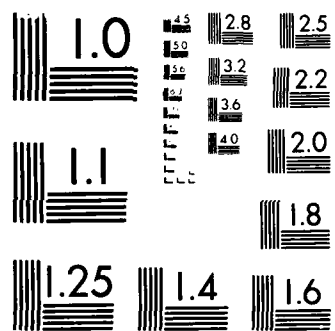
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MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963-A

27. Should the operator exit the program abnormally the turbine should be taken back into manual control.
28. Instabilities in dynamometer operation may exist at high operating speeds and low loads, should this occur adjustment of the flow control valve located near the dynamometer heat exchanger may be necessary. This flow control valve must be opened when operating dynamometer above 150 horsepower.

TURBINE SHUTDOWN CHECKLIST

1. Bring turbine to 20000 rpm idle for 1 minute.
2. Place both turbine and dynamometer in manual control.
3. Unload dynamometer completely.
4. Secure both 24 vdc and 120 vac fuel trips.
5. Enter test cell and close the following valves.
a) Lube oil cooler water supply valve. b) Fuel oil supply valve. c) Air supply valve. d) Dynamometer cooling water supply valve.
6. Close supply and exhaust dampers.
7. Exit and close rear cell door.
8. Turn off power to data acquisition equipment.
9. Turn off (unplug) student instrument panel.
10. Open fuel transfer pump breaker.
11. Turn off computer controller and terminal.

12. Turn off all power to operators panel except blower power.
13. Secure blower 20 minutes after shutdown.
14. Cover equipment and extinguish lighting.

APPENDIX J
STUDENT PANEL INSTRUMENTS

GAS GENERATOR SPEED NEWPORT (DIGITAL COUNTER) MODEL 6130A 0 TO 99,999 RPM RESOLUTION +/- 1 RPM INTERGRATED OVER 10 SECONDS SERIAL # 9020267 - 25

DYNAMOMETER SPEED NEWPORT (DIGITAL COUNTER) MODEL 6130A 0 TO 9999 RPM RESOLUTION +/- 1 RPM INTERGRATED OVER 1 SECOND SERIAL # 9041670 - 25

TYPE "K" DIGITAL PYROMETER NEWPORT MODEL 267-B KF-1 -225 TO +2500 DEG. F RESOLUTION +/- 1 DEG. F ERROR +/- 2.5 DEG. F 8 CHANNELS SERIAL # 9440010

TYPE "T" DIGITAL PYROMETER NEWPORT MODEL 267-B TF-2 -150.0 TO +750 DEG. F RESOLUTION +/- 0.1 DEG. F ERROR +/- 0.56 DEG. F 8 CHANNELS SERIAL # 3021263 - 25

FUEL TEMP. PYROMETER NEWPORT MODEL 267-B TF-2 -150.0 TO +750 DEG. F RESOLUTION +/- 0.1 DEG. F ERROR +/- 0.56 DEG. F SERIAL # 2207514 - 25

AIR FLOW NOZZLE PRESSURE MERIAM INSTRUMENT CO. MODEL 30EB25 RANGE 0 TO 30 IN. WATER, 0.1 IN. WATER DIVISIONS LEFT SIDE SERIAL # W67993 RIGHT SIDE SERIAL # W69521

COMPRESSOR DISCHARGE AND NOZZLE BOX PRESSURE MERIAM INSTRUMENT CO. MODEL A-324 RANGE 0 TO 60 IN. HG., 0.1 IN. HG. DIVISIONS COMPRESSOR DISCHARGE LEFT SERIAL # 44F552 COMPRESSOR DISCHARGE RIGHT SERIAL # 44F548 NOZZLE BOX PRESSURE SERIAL # 44F559

FUEL FLOW ROTAMETER BROOKS ROTAMETER CO. TUBE TAPER 9M-600 LI TWIN FLOAT RANGE 0 TO 600 DIVISIONS BY 2

CELL PRESSURE FRONT AND BACK MERIAM INSTRUMENT MODEL 40GD10WM RANGE 0 TO 1 IN. WATER, 0.01 IN. DIVISIONS

DYNAMIC TORQUE / SPEED ACUREX CORPORATION MODEL 1206D SPEED RANGE (X10) 0-4000 RPM, +/- 10 RPM TORQUE RANGE 0 TO

700 FT-LB, +/- 1 FT-LB HORSEPOWER (NOT USED) SERIAL #
1-278

STATIC TORQUE WALLACE AND TIERNAN MODEL FA-145 RANGE 0 TO
1000 FT-LB, 0-350 BY 1 FT-LB DIVISIONS 350-1000 BY 2
FT-LB DIVISIONS SERIAL #225B

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