THE PENNSYLVANIA STATE UNIVERSITY
COLLEGE OF ENGINEERING

SYSTEMS AND CONTROLS LABORATORY

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THE SYSTEMS AND CONTROLS LABORATORY

The Systems and Controls Laboratory was started in 1964 to encourage graduate research and development on engineering systems, with emphasis on improving the understanding of basic elements and devices and bridging the gap between theory and practice. The experimental equipment and facilities of this laboratory have been provided for carrying out experimental investigations on devices and systems involving many of the different engineering disciplines. Several graduate assistants are now working on projects and thesis topics of vital interest to engineers concerned with the advanced design and development of new engineering systems.

Much of the work involves mathematical and computer modeling and analysis as well as the experimental investigation of breadboard and prototype systems or devices. A major goal of this work is to accomplish a useful synthesis of analytical and experimental methods in research and development of advanced engineering systems for effective use by engineers and scientists working on future aerospace systems.

Financial support for this program is derived from the University, The National Aeronautics and Space Administration, The United Aircraft Corporation, and the Bendix Corporation. A major share of the projects in the Systems and Controls Laboratory are carried out by graduate assistants and junior staff members under the supervision of Dr. J.L. Shearer, Rockwell Professor of Engineering, and other members of the graduate faculty.

CONTENTS

Signal Noise in Pure Fluid Amplifiers

J.C. Tamulis

Progress toward setting up a theoretical and experimental investigation of noise in pure fluid amplifiers has consisted of literature research and formulation of a program. Some apparatus has already been built and instruments and equipment are being acquired.

Theoretical

A. An extensive bibliography of papers relative to the area of study has been compiled. An ordered review of these papers has been started, beginning with a study of M.J. Lighthill's basic theory on aerodynamic noise (1,2)*. This theory is needed in order to describe fully a control volume basic to the noise problem. In Fig. 1 a blocked load receiver port is represented. What is desired is a description of P(x,y,t) at Section 3. Roughly, the approach is to determine P(x,y,t) at Section 2 as a function of P(x,y,t), V(x,y,t) at 1, and the mixing processes in the volume from 1 to 2. The pressure at 2 is then assumed to be propagated to 3 through a wave guide. Through analysis and measurement, a description of the pressure at 3 is being sought as a function of wave number (k,y) and frequency (f).

* Numbers in parentheses denote references at end

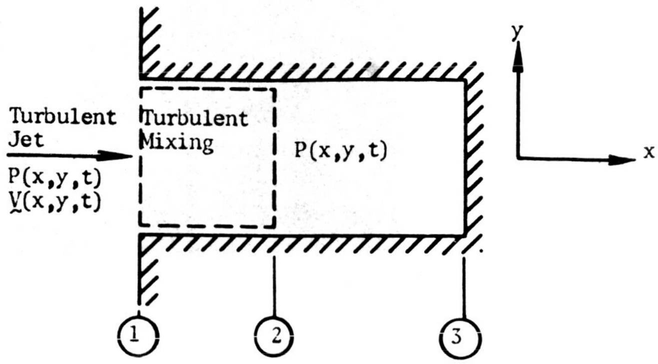


Fig. 1 Basic Control Volume

B. The methods of determining sampling errors in correlation functions obtained from experimental data has been investigated (3,4). The need for such knowledge is based on the fact that the flow fields under consideration are turbulent (random in nature) and that the relationships between signal noise at the receiver ports and noise at the control and/or supply ports will be investigated using correlation techniques. While an analytical expression for the noise function is probably impractical, the correlation functions offer a way of studying the inter action of random processes that is possible to determine experimentally. A knowledge of the accuracy of these experimentally determined functions is of critical importance. Progress in this area is to the stage where accuracy can be determined once the proposed digital data processing system (described below) has been procured.

Experimental

A. Two large-scale fluid amplifiers have been assembled. These amplifiers are large enough to accept existing pressure transducers and anemometer probes. At present one DISA constant temperature anemometer is available. Two additional anemometers and a correlator have been ordered. With this new equipment, spatial correlation functions for the velocity field will be obtainable. One of these amplifiers is shown in Fig. 2.

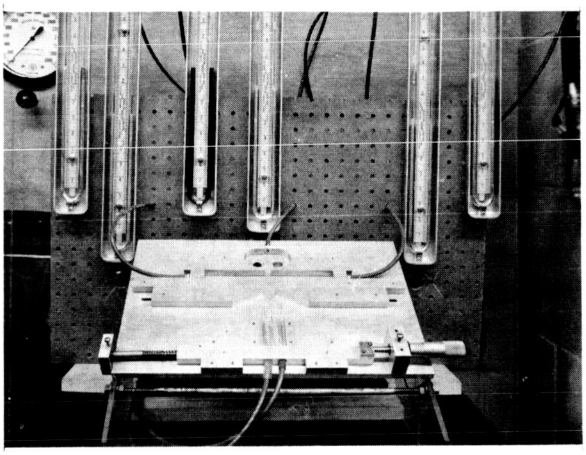


Fig. 2 Large-Scale Fluid Amplifier

B. An air supply compatible with the input impedance of the large-scale fluid amplifiers has been arranged. The supply is a Roots Blower capable of 485 cfm at 3 psig. The excess flow will be spilled.

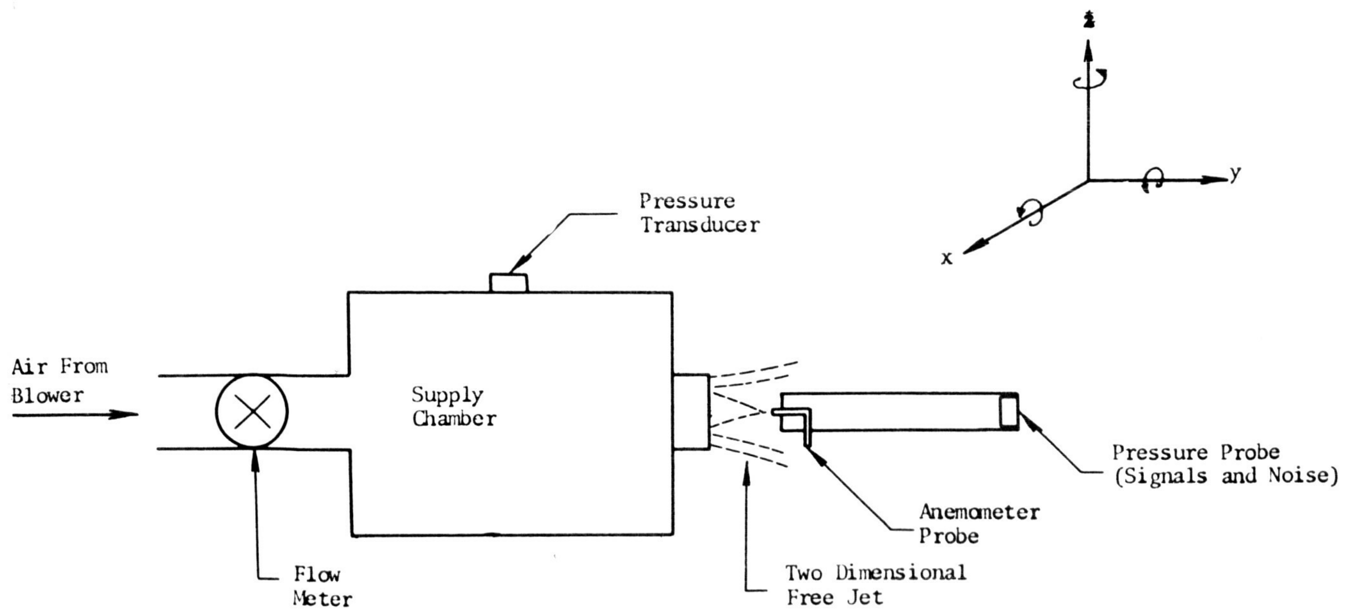


Fig. 3 Free Jet Single Receiver

off or used to run several amplifiers in simultaneous experiments. Since the noise in a working fluid amplifier will be the result of many interacting factors, a simpler test apparatus is being designed that is expected to isolate some of the controlling variables.

The apparatus now being designed is shown in Fig. 3. With this apparatus, the relations for signal noise due to a two-dimensional, free turbulent jet impinging on a blocked-load receiver will be investigated. As the study progresses, the influence of amplifier geometry and control ports can be studied. Results from these experiments will always be interpreted with respect to results obtained from a complete large-scale working fluid amplifier.

C. The digital data processing system mentioned earlier is a general purpose system which is being planned as research equipment for the Department of Mechanical Engineering. A proposal has been submitted to the National Science Foundation for funding the acquisition of this equipment. Fig. 4 represents the proposed data acquisition system. The analog signals will be obtained from an experimental set-up represented by Fig. 5.

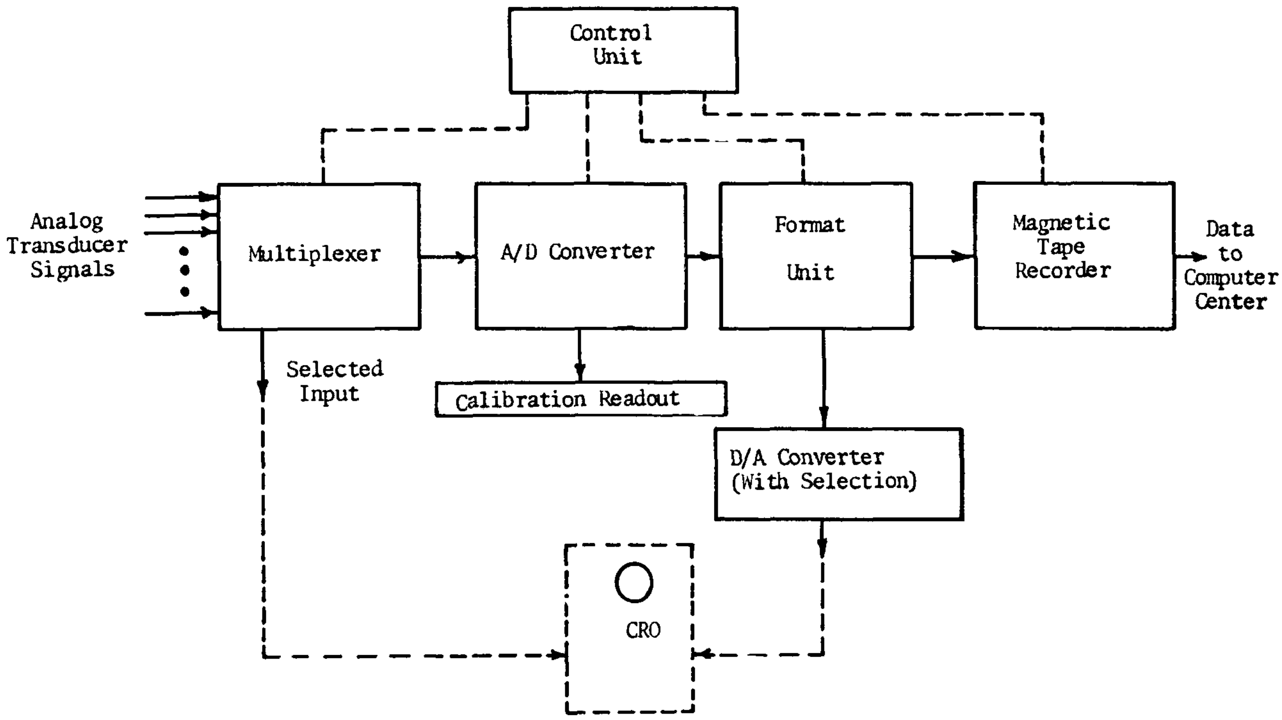


Fig. 4 Block Diagram of Proposed Data Acquisition System

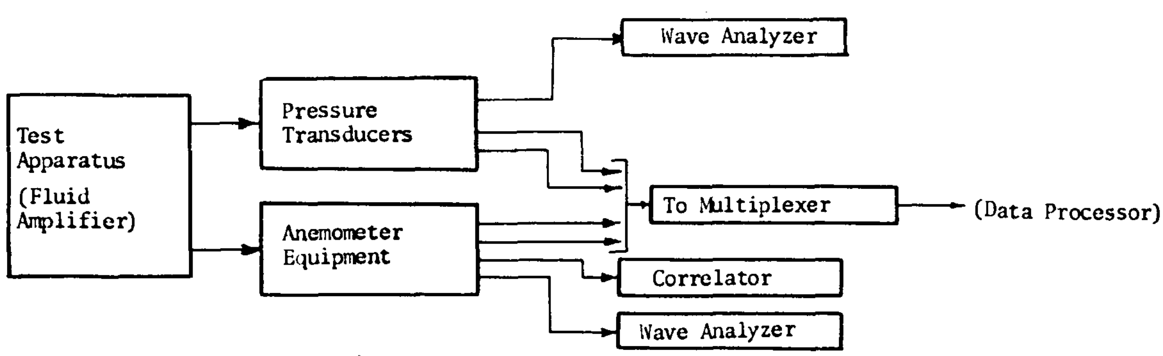


Fig. 5 Experimental System

D. A deep water-table has been built in order to help in the study of fluid amplifiers. While the table is of doubtful help in determining the structure of signal noise, it is expected to be of use in the final stages of investigation. One of the hoped-for results of signal noise study is to find ways of controlling noise by adjusting the overall geometry of the device. When this is done, other amplifier characteristics will be changed. The water-table will then be used to observe the changes introduced in the overall flow field. Fig. 6 depicts the water-table which is 4' x 8' in size, with a glass bottom and appropriate lighting equipment to expedite photographic recording.

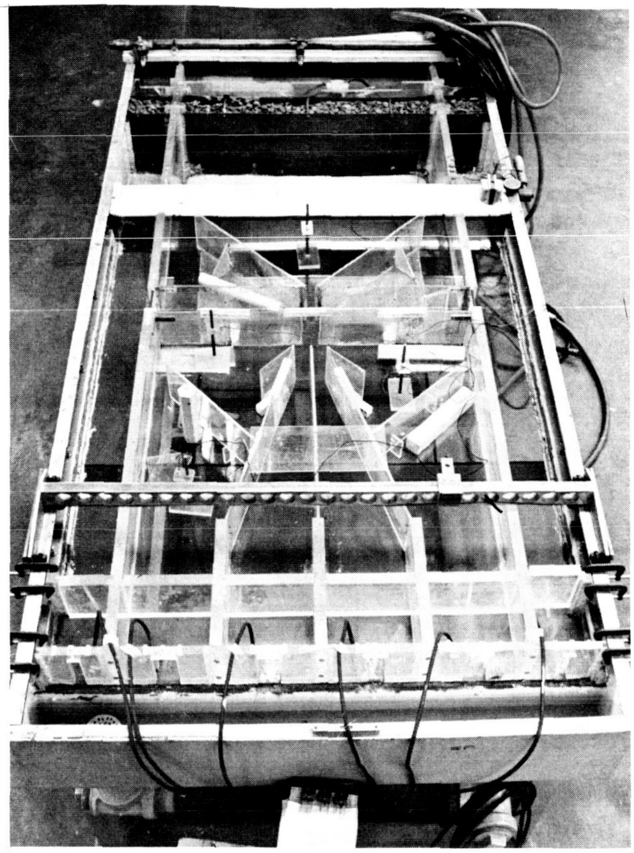


Fig. 6 Photograph of Water Table

Pneumatic Signal Function Generator

J.L. Shearer

In connection with current and anticipated work in the Systems and Controls Laboratory on fluid control systems, there will be a consistent need for generating pneumatic pressure signals such as sine waves, square waves, and randomly varying signals for use in experimental investigations. In order to meet this need, a Master's thesis by Mr. F.J. Elia (5) was recently completed which describes analytical and experimental work on a closed loop electropneumatic function generator intended to produce pneumatic pressure signals from input voltage signals generated by an electronic function generator.

A schematic block diagram of the function generator which was developed in breadboard form is shown in Fig. 7. The power amplifier was a Kintel Model 111BF wideband D-C amplifier and the summing, differentiating, and other amplifying operations were accomplished through the use of Philbrick P65A operational amplifiers. Dynisco TCPT 25-50 pressure transducers were used to obtain electrical feedback signals representing the load pressures. A specially modified flapper-nozzle pilot stage of a Moog servovalve was used as the electropneumatic transducer.

This system is unique in that it is capable of developing balanced push-pull signals to drive dual input devices, and it is capable of generating pneumatic signals at considerably higher frequencies than could be achieved with previously developed systems. When the pneumatic capacitance in the load is limited to a volume of 0.1 in.^{3}, it is capable

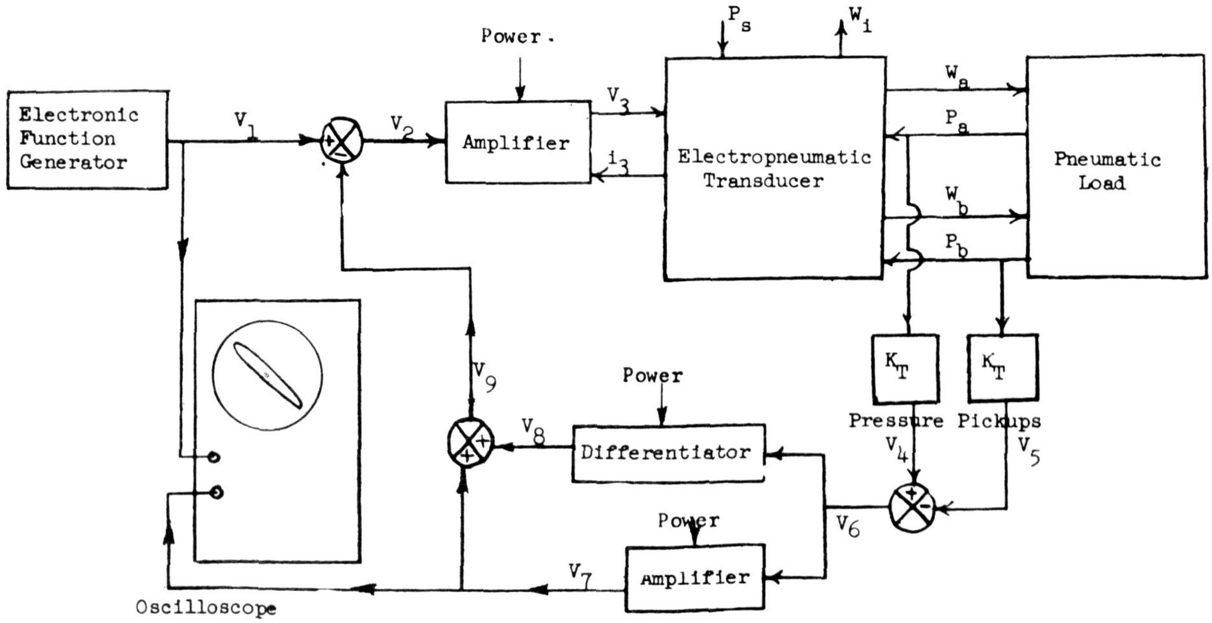
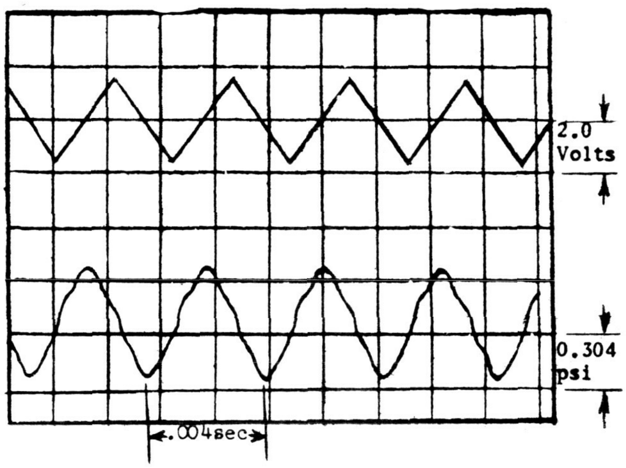


Fig. 7 Schematic Block Diagram of Pneumatic Signal Function Generator

of generating single sinusoudal pressure signals with amplitudes of ±2.5 psi at all frequencies up to 500 cps, and it can generate square waves with amplitudes of ±2.5 psi at all frequencies up to 100 cps. The oscilloscope trace shown in Fig. 8. shows the pneumatic output signal for a triangular wave form electric input signal at 250 cps.

An engineering prototype of this system is now being built for general purpose testing and research on fluid control systems in the Systems and Controls Laboratory.



Upper Trace	V volts
Lower Trace	P_a lb/in^2

Fig.8 Generation of a Triangular Waveform at 250 cps.

Fluid Stepping Motors

With the development of digital and hybrid digital-analog control systems of various kinds, a need has developed also for output actuators which deliver discrete steps of output motion in accordance with discrete input signals. In some systems, electric stepping motors have been used with good success. These motors are mostly very small and are able to deliver only small amounts of power. At higher power levels it appears that fluid stepping motors will be required. Also, newly developed all-fluid control systems may require very low power level fluid stepping motors as output actuators.

Therefore, work is now underway in the Systems and Controls Laboratory on both hydraulic and pneumatic stepping motors. Different approaches are being used for these two devices, due largely to the disparity of power levels involved. The hydraulic stepper is being investigated for higher power level applications, and the pneumatic stepper is of primary interest for lower power level applications.

In both cases, units are being developed which will respond to coded pulse-type signals traveling down a single transmission line in such a wav that incremental outputs can be commanded in either direction of the output shaft.

A Pneumatic Stepping Motor

R.P. Martin

One of the steppers under development is a pneumatic, rotary-output device designated the PSM.

Desirable features of a Pneumatic Stepping Motor were determined during early phases of the study. A Pneumatic Stepping Motor should, as a minimum:

1. Move in the proper direction
2. Move the same number of degrees each time
3. Stay in the proper position after each move
4. Move every time a signal is provided

Many types and combinations of mechanisms, actuators, and locking or detent devices were considered and studied. From the many concepts synthesized, one which seemed to hold particular promise was selected.

Several different configurations for implementing the basic concept were studied by means of preliminary design layouts, calculations, and evaluation based on engineering judgment. After selection of a basic reversible double ratchet-pawl configuration, the detail design of a prototype Pneumatic Stepping Motor was begun. During this phase of the design, many modifications were incorporated to make the prototype easier to fabricate, assemble, test, and modify. The parts for the prototype have been fabricated and initial modifications have been made. The unit is shown in Fig. 9.

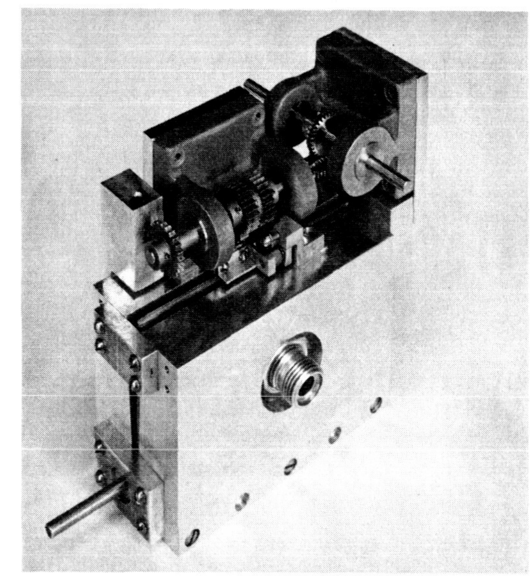


Fig. 9 Photograph of Pneumatic Stepping Motor

One unique feature of this device is the use of flexible bags instead of bellows for accomplishing the conversion of pressure pulses to movement of the pawl carrier.

When the basic design effort for the prototype was completed and while parts were being made, the initial plans for testing were formulated. It became evident that a pneumatic pulse generator would be essential for many phases of testing. Several methods of making a pulse generator were considered. Selection of one method was made, and design and fabrication of the basic pulse generator has been completed.

The development test phase has just been initiated. The test program will have as its primary goals the determination of the problem areas and their solution, capabilities, operating characteristics and some indication of the future potential of the Pneumatic Stepping Motor. The details of the test program will be governed by time and money limitations but will be guided primarily by the performance of the prototype unit influenced by the development of test facilities and equipment. It is hoped that a pneumatic function generator, based on a Master's thesis by F.J. Elia (5) can be adjusted to supply input signals.

Initial testing at reduced pressure levels and moderate speeds has begun. The pulse generator has operated satisfactorily although much work remains if optimum performance is desired. The performance of the prototype unit has been promising, but the testing is in the very early stages.

This work, which is part of a Master's thesis in Mechanical Engineering, is being supported by NASA Grant NGR 39-009-023.

A Hydraulic Stepping Motor

D.L. Ruppert

Another project has involved a computer-aided investigation of a unique mechanism which has been proposed for use in a hydraulic stepper designated the HSM.

The input pressure signal consists of pressure pulses coded for forward and reverse motion as shown in Fig. 10.

The basic mechanism chosen for this study is shown schematically in Fig. 11. A piston or bellows, depending on whether or not a hermetically sealed fluid system is desired, is used to obtain a reciprocating linear movement from the input hydraulic pressure signal. This is used to drive a cam in the ±Y direction, where X and Y are coordinate axes fixed in space as shown on the figure. The cam is supported by frictionless roller guides and is free to move in the ±Y direction only. Referring to Fig. 12, it is attached to a damper and a spring, which is loaded in compression to balance the piston (bellows) force resulting from the steady state pressure component, P_s. Therefore, only pressure excursions from this steady state value would tend to change the position of the piston (bellows) from its steady state position.

The purpose of the cam groove is to produce a net increment of linear or rotary output movement from the reciprocating motion of the cam. Using a shape of the cam profile, $Y_r = F_o(X_r)$, which generally resembles that of a single cycle of a sine wave or similar doublet of pulses, and assuming that the cam follower pin is free to move only in the ±X direction, the cam follower pin will move between incremental positions of progressively increasing or

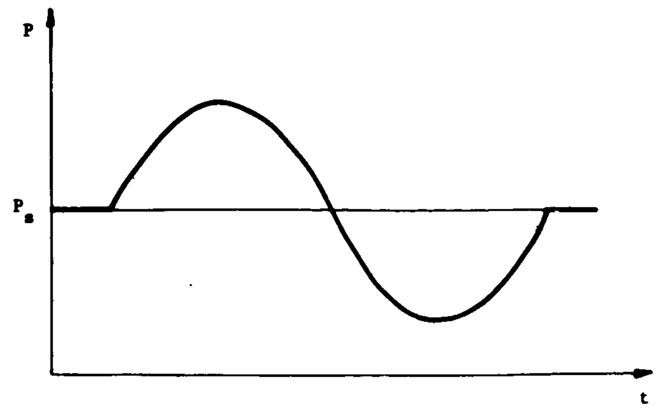


Fig. 10a Forward Motion Pressure Signal

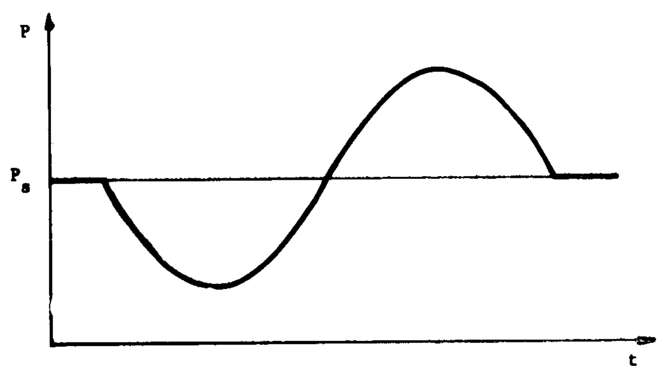


Fig. 10b Reverse Motion Pressure Signal

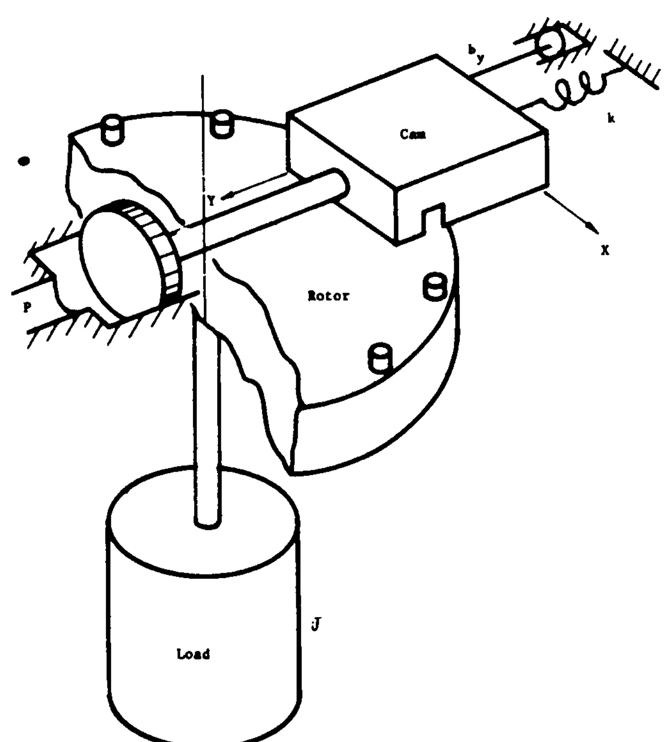


Fig. 11 Stepping Motor Schematic Diagram

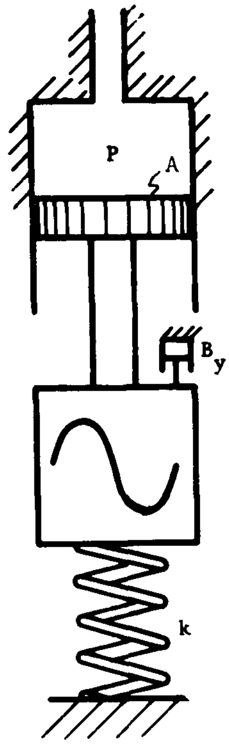


Fig. 12 Cam Assembly

decreasing values of X, depending on the sign of the input pressure signal. The combined momentum of the cam follower, gear train (if used), and the load must be great enough to carry the follower past the points on the cam profile where $X_r = \pm Y_{rmax}$. Coordinate axes X_r and Y_r to depict relative motion between the cam and the pin are fixed on the reciprocating cam as shown on Fig.13.

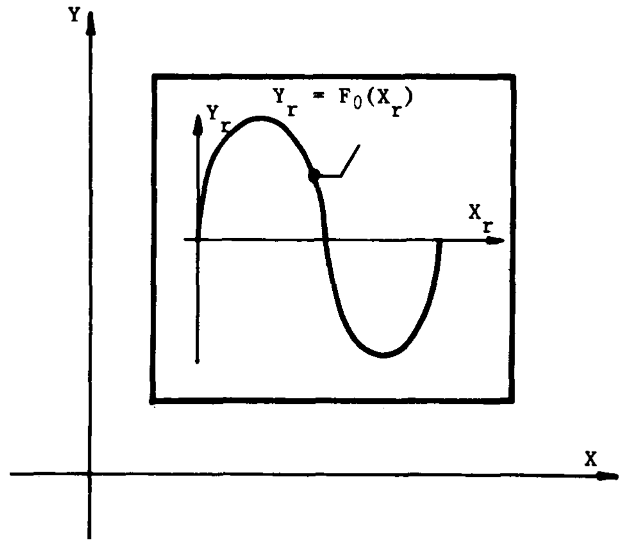


Fig. 13 Cam Groove Profile

The cam profile may be chosen so that the slope in the region corresponding to the steady-state position of the stepper is very large. This would permit high starting torque for loads with coulomb friction and/or large inertia and high resistance to load effects in the steady-state position. If the cam slope were infinite at its steady-state position, complete locking of the load would be attained at the incremental steady-state output positions.

This investigation was concerned with a hydraulic stepping motor having a maximum time-averaged power output of 2 horsepower and a time-varying driving force, G = PA, whose period is 0.01 seconds per cycle. The output is rotary with an incremental

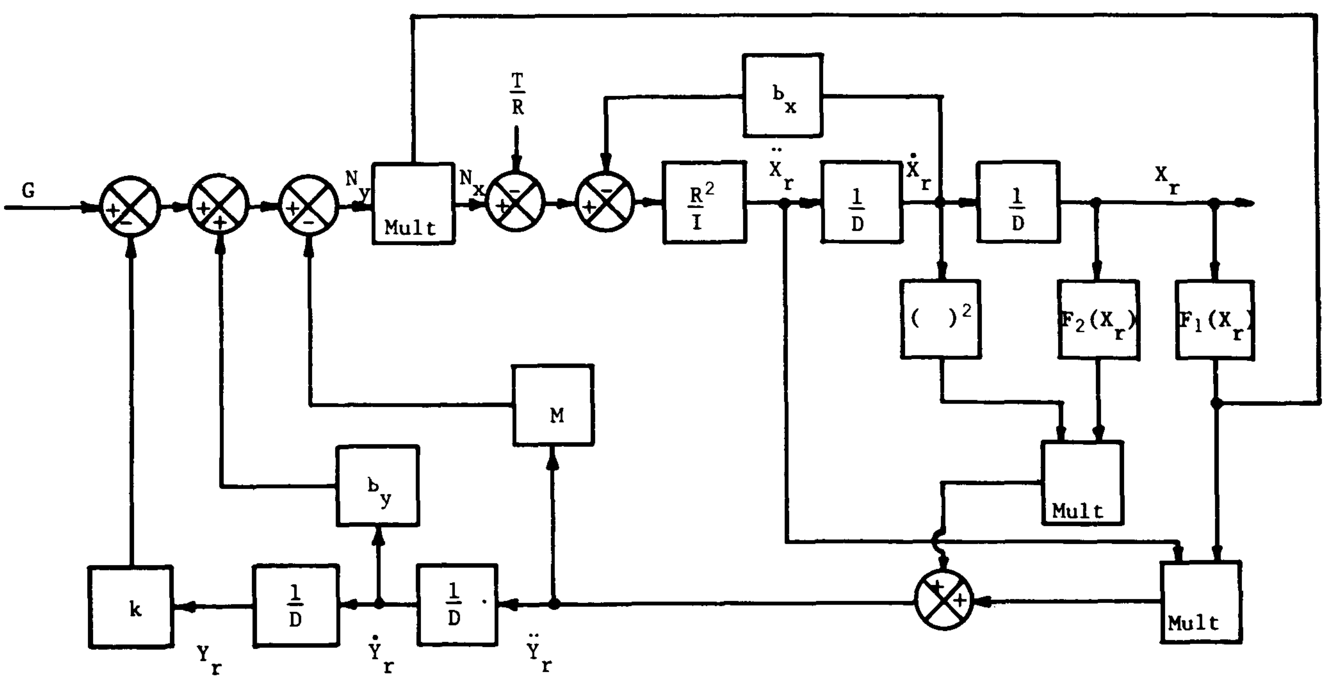


Fig. 14 Analog Computer Block Diagram of HSM

step angle of 2 degrees. In order to accommodate the geometry of cam and follower hardware while keeping the overall size of the unit at a minimum, a gear train with a 10:1 ratio was used between rotor and load. This has a desirable secondary result of reducing the effect of load inertia changes by the square of this ratio, in this case, by a factor of 100. For the purposes of this particular investigation, the cam profile, $Y_r = F_o(X_r)$, consisted of parabolas fitted together piecewise to form the complete cycle shown in Fig. 13.

The prime objective of this investigation has been to determine quantitatively the nature of the transient response of the proposed stepping motor and to determine whether or not such a device would be feasible for use in a practical application. The feasibility of the device was determined using such qualitative criteria as tolerance of the system to changes in important parameters and system behavior with different initial conditions.

The transient performance of the hydraulic stepping motor was simulated by both electronic analog and digital computer solutions of the system differential equations of motion. Fig. 14 is the block diagram for the analog computer solution. The Applied Dynamics Computer in the Dynamic Simulation Laboratory is shown in Fig. 15 as it was set up for this investigation. The flow diagram for the digital computer solution is shown in Fig. 16. Using the same input data in both analog and digital computers for a given solution enabled the digital computer to verify the solution obtained from the analog computer.

The operational tolerances of the simulated system were then observed by changing the magnitudes of the following system parameters, in which variations are anticipated in practical applications: driving force amplitude, A; load torque, T; moment-of-inertia, J; spring constant, k; and viscous damping constant, b_y. In addition, the driving force wave form was changed from a sine wave to a square wave.

Finally, appropriate system initial conditions for the continuing pair of driving force pulses in a series of repeated signals were used in a transient response simulation to determine whether the response for the second pair would differ appreciably from the response to the first pair of pulses.

Both the analog and the digital computer solutions provided a reasonably accurate indication of the nature of the system transient response by virtue of their close agreement. Typical results of these solutions are shown in Fig. 17. The X curve is the desired output. The few deviations that existed in the two solutions were due mainly to analog computer errors which could be isolated or due to errors resulting from limited dynamic response of the X-Y Recorder.

The system proved to be quite tolerant to changes in the magnitudes of important parameters. It exhibited considerable flexibility in accommodating a wide range of load moment-of-inertia and load torque values, suggesting a possible application to spring loads. Further evidence of system flexibility was seen in its ability to operate with an impressed square wave driving force as well as a sine wave, suggesting that intermediate wave forms would also be satisfactory.

The system transient response for the second driving force signal in a series of repeated signals was approximately the same as that for the first signal. It was therefore assumed that further repeated signals would produce a system transient response similar to that produced by the first signal.

Design considerations for this device indicate that the cam follower is most likely the weakest link in the mechanism. In order to permit minimum rotor diameter, cam profile amplitude, gear ratio,

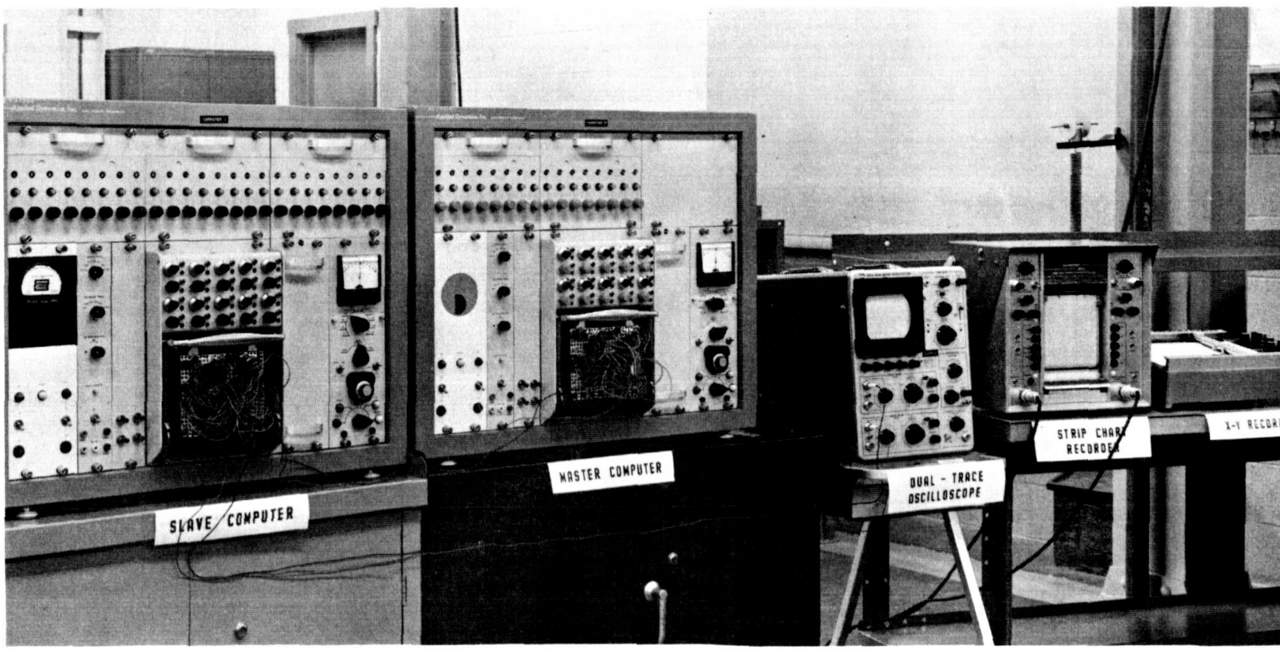


Fig. 15 Photograph of Analog Computer Set-up for HSM

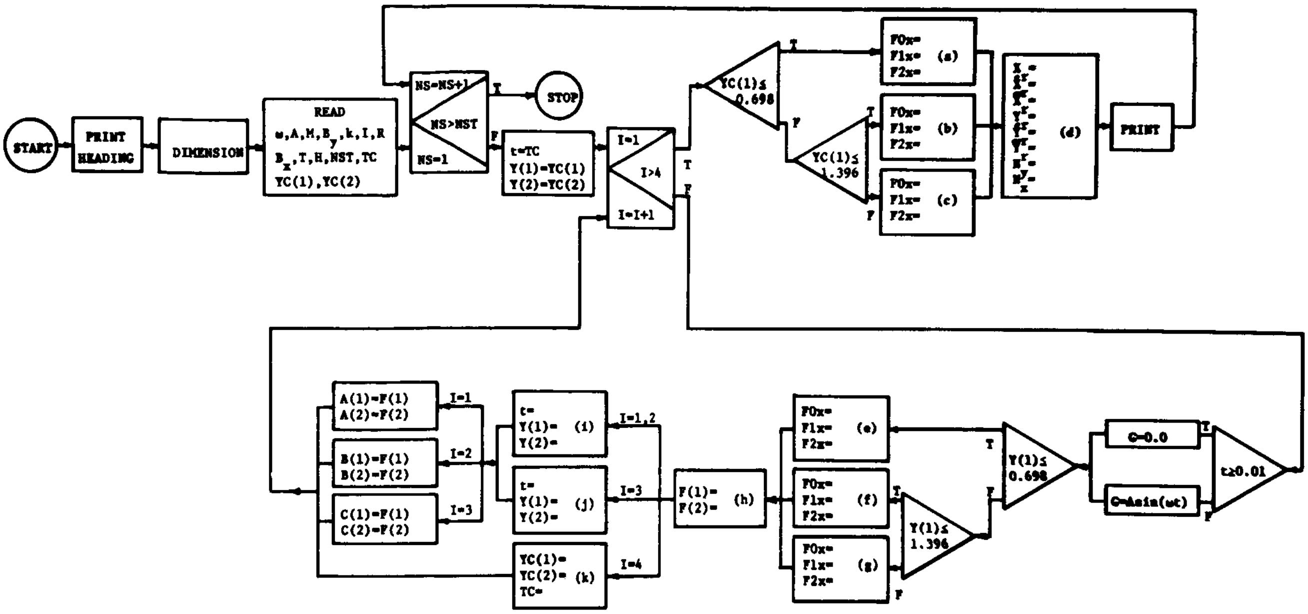


Fig. 16 Digital Computer Flow Diagram for HSM

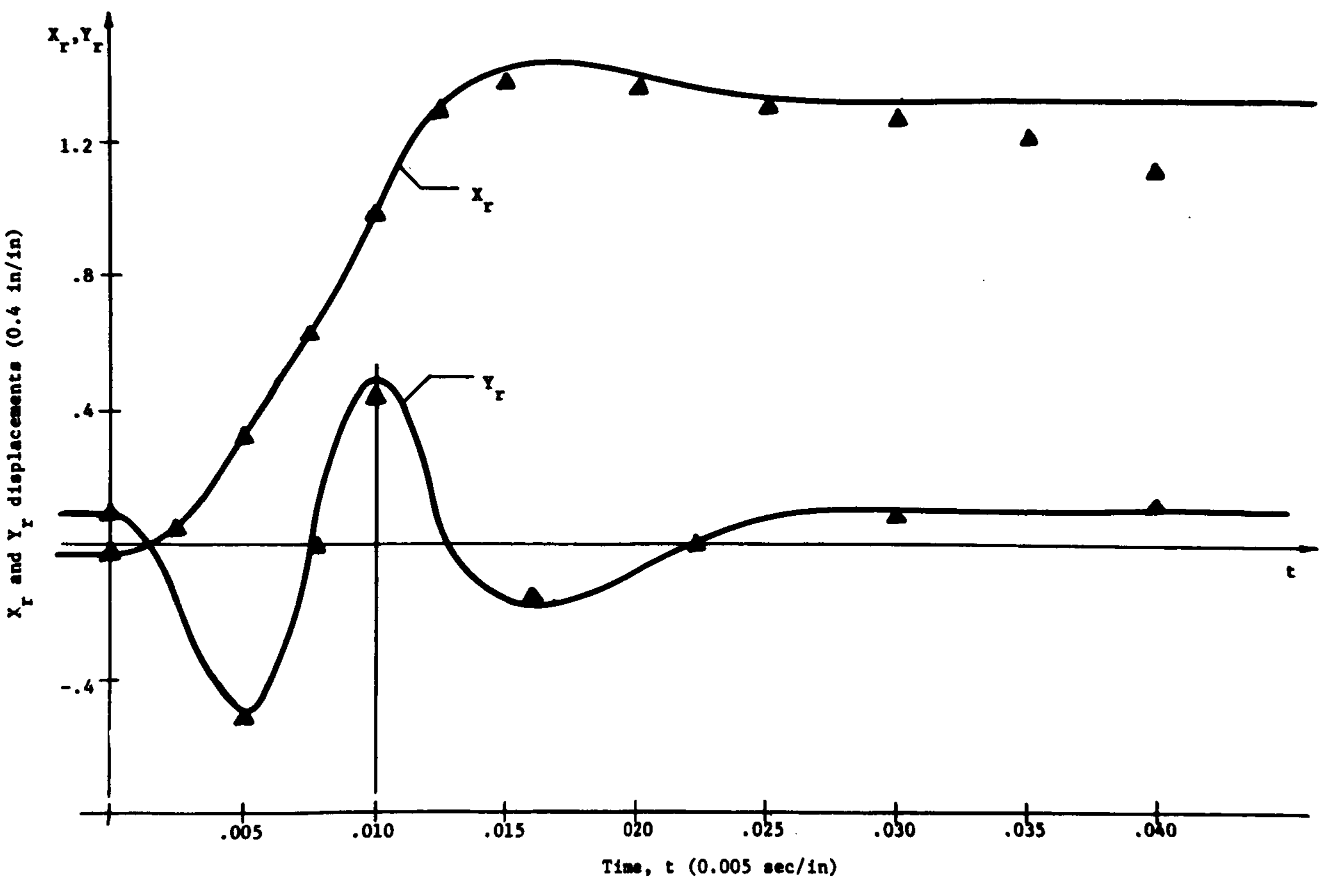


Fig. 17 Transient Response Results, X_r and Y_r

and follower spacing, keeping the pin diameter at a minimum is a necessity. The follower, however, should be able to withstand the dynamic loads for the desired power transmission. It is believed, however, that strength requirements for the follower can be met by judicious choice of system geometry and follower material and design.

The configuration of the mechanism shown on Fig.11 was selected in order that the follower reaction force at the rotor bearing would be radial. In an alternate configuration, the cam profile could be mounted in a plane perpendicular to that of the rotor with the followers mounted radially. This arrangement, however, would involve a very difficult cam and follower interface problem and an undesirable bearing load configuration.

Referring again to Fig.11, it would be quite possible to use a continuous cam profile fixed to the rotor and drive a pin in the Y direction with the piston or bellows. This would enable the oscillating mass to be reduced considerably. The configuration shown in Fig.11, however, requires less machining, because only one cam profile cycle is used.

It is assumed that in constructing this stepping motor a cam follower employing frictionless bearings may be used. Lubrication difficulties are therefore anticipated if the stepper is to be operated in a hard vacuum. Specifically, these would affect the rotor bearings, the cam guides, and the follower. Under these conditions, hermetically sealing the entire system would be highly desirable.

Considering the above discussion, it was concluded that the proposed hydraulic stepping motor is feasible for use in a practical application such as a digital-controlled actuator for use in remote control and/or automatic control systems.

This investigation is described in greater detail in a Master's Thesis entitled: "Computer Simulation of a Hydraulic Stepping Motor", being written by Mr. David L. Ruppert for the Department of Mechanical Engineering. It was supported in part by NASA Grant NGR 39-009-023, and by a grant from The American Machine and Foundry Co.

Hysteretic Non-Linearities in Closed Loop Control Systems

R.S. Raizada

Hysteresis type of backlash and dead zone in a closed loop control system causes instability at small signal levels which results in limit cycle oscillations. This instability was studied analytically and with phase plane techniques. For an input-output relationship for a hysteresis element shown in Fig. 18, the transfer function for a typical system shown in Fig. 19 can be written as:

$$\frac{\theta_o}{\theta_i} = \frac{KG(s)}{1 + KG(s)}\left[1 - \left(\frac{H_n}{\theta_i}\right)\left(\frac{\dot{E}}{|\dot{E}|}\right)\right] \qquad (1)$$

It can be shown that the transfer function (1) can be reduced to:

$$\frac{\theta_o}{\theta_i} = \frac{KG(s)}{1 + KG(s)} \qquad (2)$$

by introducing a switching logic network, N(s), either in series or parallel of the type,

$$N(s) = F \cdot \frac{\dot{E}}{|\dot{E}|} \qquad (3)$$

and the system can be restored to its former degree of stability. The problems in synthesis of N(s) were studied for series and parallel correction techniques using logic theory and it was shown that, in general, a series type of correction, Fig. 20, would result in a simpler arrangement.

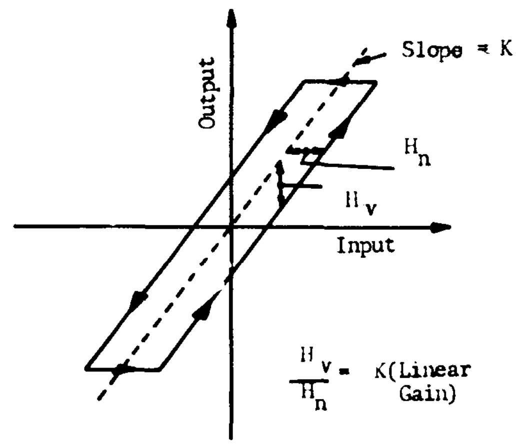


Fig. 18 Hysteresis Characteristic

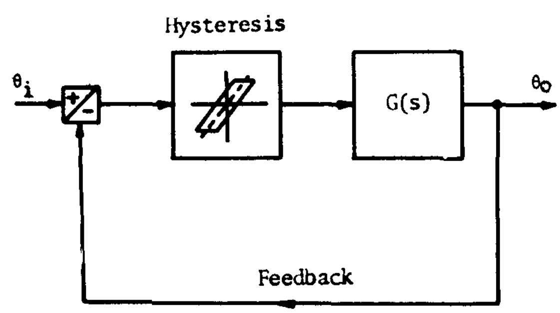


Fig. 19 Typical Control System with Hysteresis

The type of switching logic necessary to synthesize the correction network, N(s), has been shown in Fig. 21 schematically. A second order system with a variable hysteresis in the error path was simulated on an analog computer and it was found that the switching logic of the type shown in Fig. 21 restores the system stability. The effects of over-correction and under-correction were studied and results from

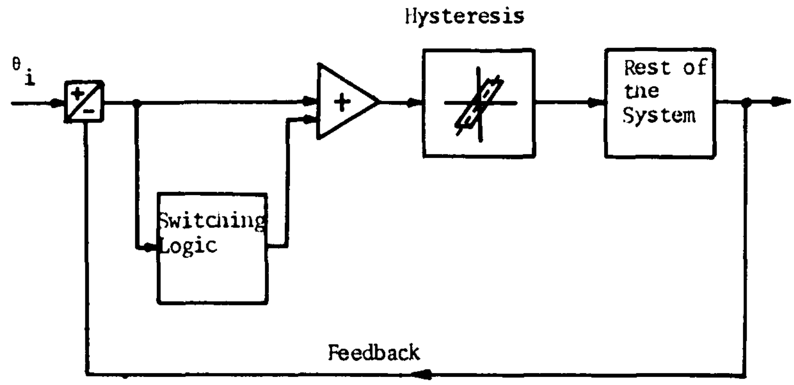


Fig. 20 Series Correction of Instability Due to Hysteresis in a Closed Loop System by Means of a Switching Logic Network

analog simulation were found to agree with the analysis.

For further details on this project, reference may be made to No. 6 in the List of References.

This work was supported by NASA Grant No. NGR 39-009-023.

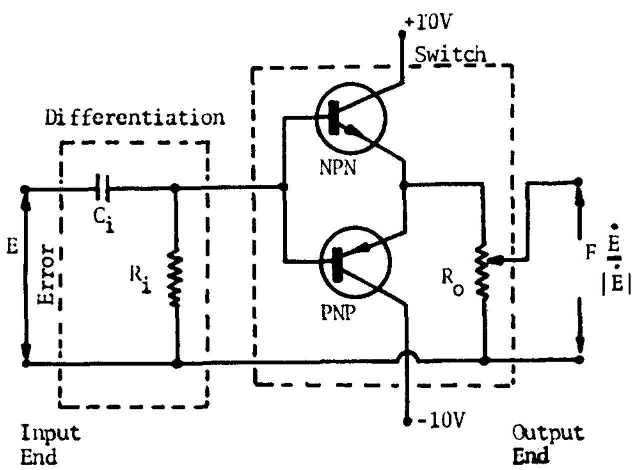


Fig..21 Schematic Electronic Circuit for Correction of Hysteresis Type of Non-Linearity by Means of a Switching Logic Network

Dynamics of Fluid Transmission Lines

R.S. Raizada

Dynamics of loaded fluid transmission lines was studied by using a simplified linear model. Special attention was paid to the problem of transmission of pulse signals in a fluid line using the Laplace transform technique in conjunction with Mellin's inversion theorem. A model is, however, also being developed for the Matrix transfer function approach. It can be shown that the Matrix transfer function can be reduced to:

$$\frac{P}{Q}\bigg|_{\text{Input}} = \frac{\text{Cosh}\,\Gamma + A\,\text{Sinh}\,\Gamma}{\text{Cosh}\,\Gamma + B\,\text{Sinh}\,\Gamma} \qquad (1)$$

$$\text{where } A = \frac{Z_c}{Z_t}$$

$$\text{and } B = \frac{Z_t}{Z_c}$$

Z_c being the characteristic impedance for the line, and Z_t the terminal impedance at the load end.

For a general case (1) can be rewritten as,

$$\frac{P}{Q}\bigg|_{\text{Input}} = \left[\frac{f_1(s) + \text{Tanh}\,\Gamma(s)}{f_2(s) + \text{Tanh}\,\Gamma(s)}\right] f_3(s)$$

$$= \frac{F_1(s)}{F_2(s)} \qquad (2)$$

$$= F(s)$$

In order to evaluate (2), it is necessary to have some form of an expansion in a power series or otherwise. Expanding $F_1(s)$ and $F_2(s)$ each in a power series results in inclusion of false zeros or poles, when the series are truncated after a finite number of terms, predicting instability whereas instability may not occur in an actual system. This difficulty can be avoided by a Taylor series expansion of $F(s) = F_1(s)/F_2(s)$, taking into account $F_1(s)$ and $F_2(s)$ simultaneously. Such an expansion is being studied at present for its validity, especially where convergence may be a problem.

Experimental High Performance Electric Servomotor With Minimum Weight and Size

M.C. Justice

Recently published analytical studies (7,8) have revealed the possibility of designing electric servomotors with greatly improved performance. A thesis project is under way toward providing a working model shell-type D-C motor for experimental verification of analytical predictions that imposed performance requirements of aerospace applications can be met.

A literature search for further information (9,10,11,12) disclosed that no experimental results have been published on shell-type servomotors and that analytical results are limited. However, a visit with engineers of the General Electric Co. in Erie, Pennsylvania, revealed that some experimental work had been done on a shell-type servomotor for industrial use.

The results of their limited investigations were encouraging and reinforced our belief that we should move ahead with research on a high-performance, low weight and small size device. Design studies have proceeded with their recommendations in mind.

The analytical work referred to is for a shell-type motor, shown in Fig. 22, consisting of an armature in the form of a hollow shell which revolves around a stationary ferromagnetic core. Field flux through the poles is provided by permanent magnets designed to produce a required field flux density at the air gaps. The most important performance parameter is the torque-squared-to-inertia ratio, or power rate (7,8), and is given in terms of heat transfer as well as electromagnetic phenomena in order to minimize the mass and inertia of the motor.

The planned experimental working model is shown schematically in Fig. 23. A detailed design, which is now under way, involves considerations of permanent magnet design, armature configuration, heat transfer, instrumentation, and experimental testing of the device. Some of the possibilities for improving heat transfer characteristics are spray mist or immersion cooling, forced air convection, choice of materials with high heat conductivity, provision of heat sinks, and good bearing design. A relatively simple armature design will be employed because of limited fabrication facilities, but it will be designed for good electrical efficiency and low inertia. Once the model is built, dynamic response characteristics and thermal transients will be investigated experimentally. Use of the digital computer is expected to

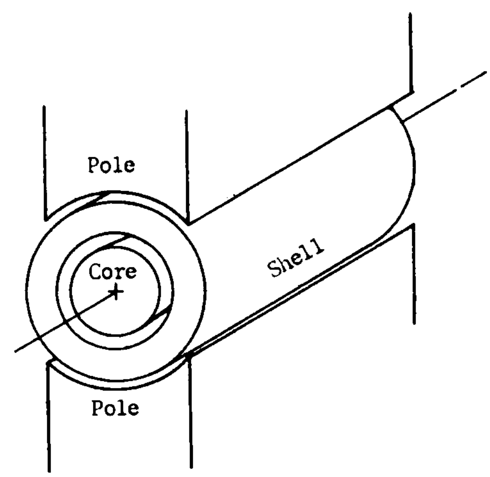


Fig. 22 Shell-Type D-C Motor

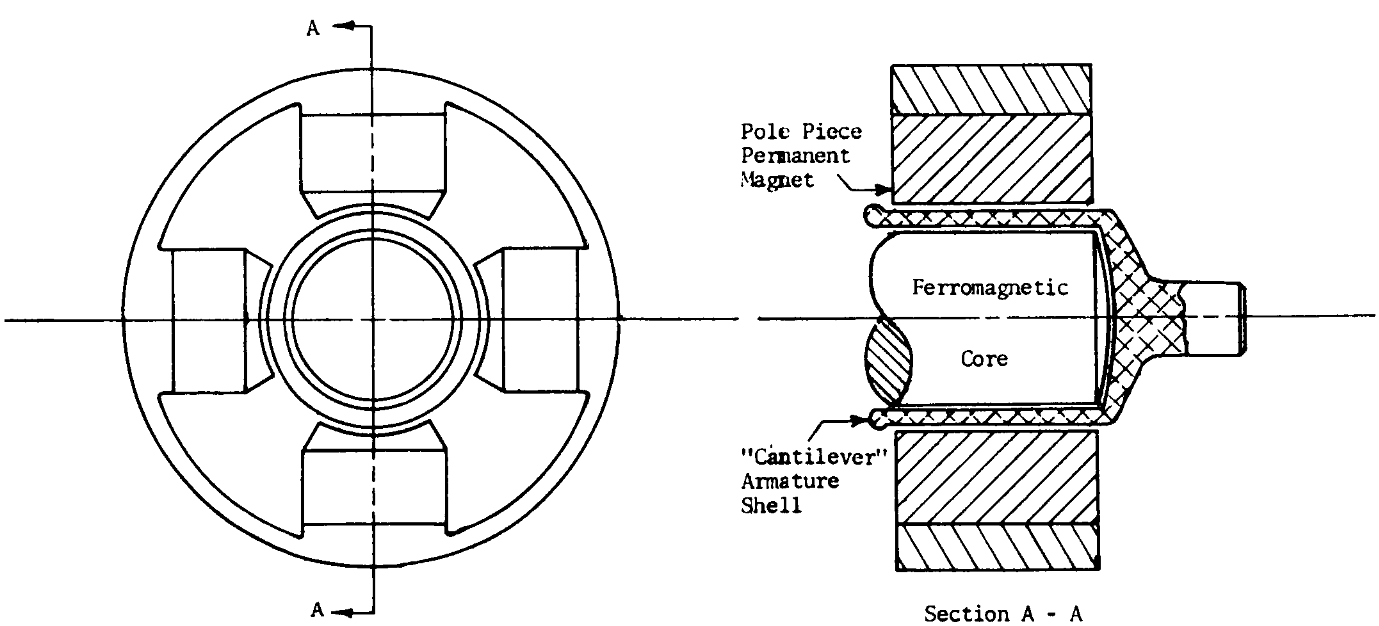


Fig. 23 Schematic Diagram of the Planned Experimental D-C Motor

facilitate the evaluation of design trade-offs and help to evolve a rational approach to the design of devices of this kind.

Due to a frequent need among the Systems and Controls group, a digital computer program was developed for finding roots of polynomials. A literature survey revealed many available methods(13,14), none of which are truly general or foolproof, and an initial decision was made to use the Lin-Bairstow method. The disadvantage of this method is that an approximate set of roots (a quadratic) must be fed into the program and convergence to a true set of roots occurs only if the approximation is good enough. Future work is contemplated on a more efficient program(15), however, the present one is usually successful and has been applied to a number of problems including computation for root locus tabulations.

This work is being supported by NASA Grant No. NGR 39-009-023.

Fluid Amplifiers and Stability

A.K. Stiffler

The major emphasis of this work is on non-linear stability in fluid amplifiers. As of now, solutions have not been outlined nor has the specific problem been well-defined. An effort has been made toward these ends by a program of familiarization with analytical tools and the experimental investigation of the fluid amplifiers themselves.

Analytical

We are interested in the fluid amplifier as a component in an engineering system. The first fundamental concept upon which system analysis is based is the necessity to describe the state of each element in the system. A modern approach to this problem involves the use of state space techniques. Two areas of its development are stability (Lyapunov) and optimal control.

An extensive review has been made of the literature on Lyapunov's direct method of non-linear stability analysis (16,17,20). Most of the main theorems have been explored in depth. In particular, applications of Lyapunov techniques to distributed parameter systems (21,22) were examined. Theoretical work here is very limited. In addition, there is a need for extended theory in the use of Lyapunov's general energy concept for transient behavior prediction and for the evaluation of ultimate design criteria.

A review of the current state of optimal control theory has been carried out with emphasis on the use of classical methods of variational calculus. The use of variational methods is hampered by computational difficulties and by its failure to handle saturation constraints. Of special interest is the close relationship of dynamic programming, widely attributed to Bellman (28) and the maximum principle of Pontriagin (27) which now makes it possible to solve a wider range of optimum control problems.

Experimental

A laboratory program has been started to obtain a better understanding of fluid amplifiers. This program has taken two directions.

A. Low pressure (0.2 to 14.7 psia) studies are being made on the use of a fluid amplifier as a pressure sensor. This includes, in particular, the effect of a mean control port pressure below the sink pressure on the far side of the knife edge. Some amplifier designs, fabricated to our specifications by Corning Glass Works, Bradford, Pa., proved to be unsatisfactory because of input impedance problems and the lack of a center dump between the receiver ports. An amplifier was fabricated in our own laboratory to correct these difficulties so that experimental work now can be carried forward on determining operating characteristics and preferred methods of measuring them. A photograph of this device is shown in Fig.24. A drawing of the profile employed in this design is shown in Fig.25.

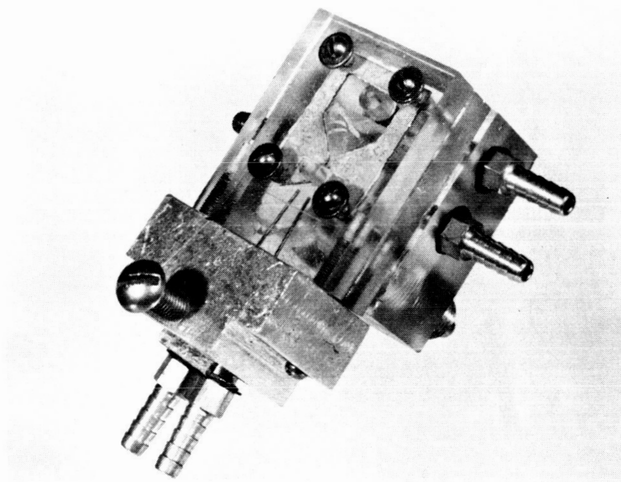


Fig. 24 Photograph of Experimental Fluid Amplifier

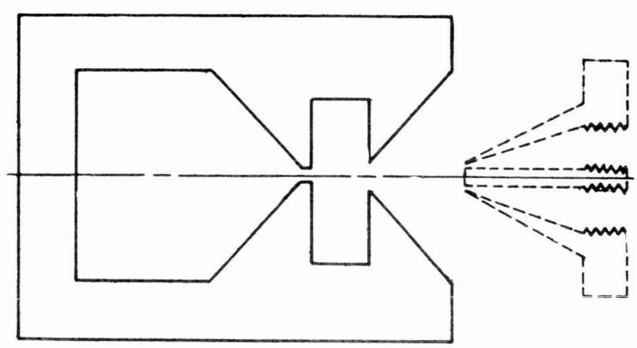


Fig. 25 Control Section of Fluid Amplifier

B. Some of the Corning devices, as well as our own, have been set up to collect data on static input and output impedance characteristics. This includes instrumentation and fabrication of sets of calibrated high flow fluid resistors. Eventually, dynamic studies will follow with the aid of a pneumatic function generator which is being developed along the lines of the breadboard system described in a recently completed Master's thesis by F.J. Elia, (5) . This work on fluid amplifier pressure sensors is sponsored by the Hamilton Standard Division of United Aircraft Corporation, Hartford, Connecticut.

Time-Delay Characteristics in Systems Employing Vortex Devices

H.H. Herritt

The purpose of the project is to investigate the effects of dynamic response characteristics of vortex devices on overall control system performance.

At the beginning of the report period, this project was just getting under way with a literature study of vortex amplifiers and other fluid devices to determine the pure-time delay characteristic of each. The majority of the effort required to complete the study has been expended this period. The project will be completed during the next period.

Work began with a literature survey of fluid devices to determine the amount of pure time-delay associated with these devices. No general time-delay information could be found and no information was available on the time-delay of vortex devices in the literature. However, preliminary information was obtained in a letter from Bowles Engineering Co.(33) which gave a range of values of time-delay for each fluid device in a proposed torpedo control system.

An investigation was made of various means of simulating pure time-delay for use with the analog computer. Based on the delay values given by Bowles, computer circuits composed of operational amplifiers (34,35) were found to be adequate for the range of interest of time-delay and frequency.

Work is now being done on an analog computer simulation of a control system using fluid devices as rate and course sensors. Gains and other parameters will be adjusted to provide the response expected of an operational system. Step response and frequency response will be recorded for values of pure time-delay as given by Bowles Engineering, and stability limits of the parameters will be predicted.

A digital computer program is being investigated which will be used to check the results given by the analog computer.

During the next period, a trip is planned to Bowles Engineering Co. to investigate details of the fluid system. Any necessary changes or additions will be incorporated into the computer study. The details and results of this study will be written up in a report to give the fluid control system preliminary design requirements and its performance during the coming period.

Some of this information will be used in a Master's thesis which will deal with techniques for improving the response of systems with dead times. This work is being done under Exploratory and Foundational support from Penn State's Ordnance Research Laboratory.

Steam-To-Air Energy Converter

R.T. Harper

The many potential advantages in the use of high pressure air in place of superheated steam in fluid power control systems prompted this program in the development of a steam-to-air energy converter suitable for use in an underwater weapon guidance system. This program was begun during the period covered by this report. The design proposal specified a 15 hp. unit capable of delivering 3000 psi air using 500 psi 1000°F supply steam. Specific design parameters to be kept to minimum values include size, weight, and operating noise.

A preliminary literature search uncovered no suitable existing hardware or design. Project advisors further limited the scope of the project to an investigation of the application of the free piston concept to the problem. An extensive search of the literature in this area was conducted. Little of the previous work done on free piston engines could be applied.

After surveying various free piston design configurations, the most promising of these configurations was selected for intensive investigation. In Fig. 26 steam enters the center chamber at a constant supply pressure and forces the two pistons apart. The outward motion of the pistons compresses the air

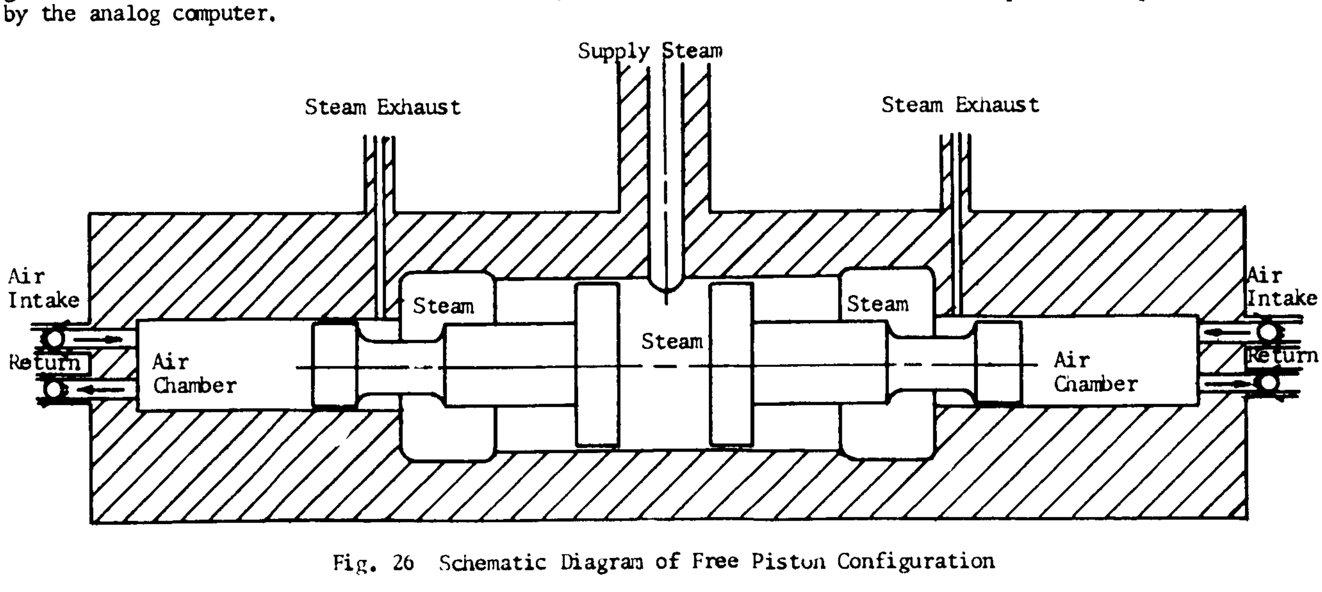


Fig. 26 Schematic Diagram of Free Piston Configuration

in the air chambers and forces it out check valves. On nearing the outer ends of the cylinder, channels in the cylinder walls allow steam flow past the pistons. The forces on the pistons are now unbalanced in the opposite direction and the pistons slow to a stop and reverse direction of motion. The inward motion of the pistons draws additional air into the air chamber and expands the steam in the bounce chamber. On nearing the center of the cylinder, the piston allows the expanded steam to exhaust, again reversing the direction of the unbalanced forces and completing the cycle.

The study of this design configuration will include necessary calculations to optimize the various parameters, instrumentation of a single free piston laboratory model to provide experimental data for confirmation and/or refinement of the theory, and analysis of problems relating to response to load disturbances and synchronization of opposed pistons.

A design for a single piston laboratory model was selected to provide a maximum of useful experimental data, Fig. 27. Maximum steam pressure available in the laboratory is 250 psia, one half that specified in the design proposal. Air line pressures have been reduced to maintain the original ratios of pressures. It is anticipated that the frequency of piston motion in this unit will be about 25 cps.

Machine shop and assembly drawings for all parts have been completed and sent out for competative bids. Necessary instrumentation and standard fittings have been ordered.

An electrical analog model has been set up with assistance from the ORL Computer Laboratory personnel, and the program has been wired and debugged.

During the next report period the analog solution will be studied in detail. In addition, the laboratory model will be set up and operated. Much of this work will constitute a Master's thesis for the Department of Mechanical Engineering. This work is being done under Exploratory and Foundational support from Penn State's Ordnance Research Laboratory.

For further information about the projects mentioned in this report or the activities of the Systems and Controls Laboratory, inquiries should be addressed to: Director, Systems and Controls Laboratory, 213 Mechanical Engineering Building, University Park, Pennsylvania, 16802.

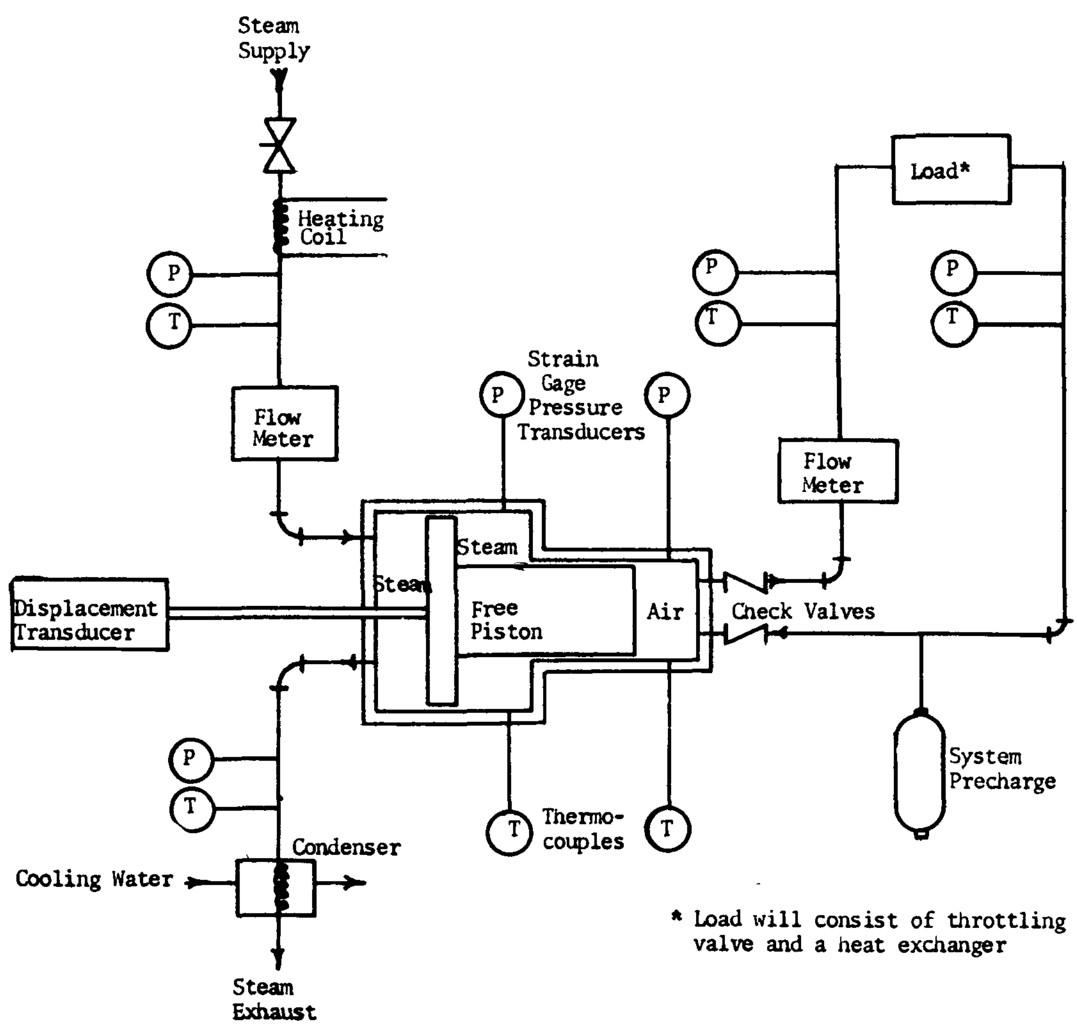


Fig. 27 Schematic of Laboratory Apparatus

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