THE UNIVERSITY OF MICHIGAN

COLLEGE OF ENGINEERING
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Student Project Reports

INVESTIGATION OF DESIGN MEANS FOR HOME LAUNDRY APPLIANCES

David Kett
Thomas Farrington
Jon D. Tromblee

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ENERGY STORAGE SYSTEM TO COUPLE MOTOR AND PUMP FOR A WASHING MACHINE

David Kett

I. INTRODUCTION

With the advent of a new dc direct-drive motor for use in Whirlpool Washers, certain problems were introduced. One of the design problems concerns a power source that the centrifugal water pump needs for water circulation during wash cycles and pumpout during the spin-dry cycle. The question is whether to buy a separate ac motor to power the centrifugal water pump, or to find a way to use the power available from the dc motor.

This project presents a possible means of using the second alternative. The solution basically involves a transformation of oscillatory energy to constant speed energy, coupled to an energy storage device. This solution assumes that the energy transformation and storage system is capable of accepting a constant speed at the input as well as the oscillatory input. For this condition, the output and input is directly coupled.

II. STATEMENT OF THE PROBLEM

The direct-drive motor, now being used in experimental Whirlpool Washers, drives the bowl of the washer so that the motion of the bowl and the rotor of the motor are the same. The motor oscillates back and forth during wash cycles and turns at 800 rpm during the spin-dry cycle. This motion must somehow be used to power a water pump that develops 0.1 hp at 800 rpm. The normal motion of the motor during agitation is a 180 sweep at 60 agitations per minute. The design problem is to build a power transfer system that converts the above motor motion to the 800 rpm motion needed to drive the pump.

III. APPROACH TO THE SOLUTION OF THE PROBLEM

An energy storage system has certain advantages in this type of problem because the oscillation input represents a surge of motion, momentary stops, and again a surge of motion. An energy storage device could receive this energy and deliver it at a constant rate, thus solving the problem. Springs, flywheels, a lifted weight, and compressed air are just a few of the devices that might be used. A system involving a power spring however, seemed to offer the greatest potential.

The average motion of the motor during agitation is 30 rpm or 15 rpm in each direction. Since the output needed is 800 rpm, a large gear increase is needed. After an investigation of many mechanical gear systems, a planetary gear system was adopted because a large gear increase (or reduction) can be obtained with only a few gears.

IV. EXPLANATION OF THE DESIGN

A. GENERAL OPERATION

A schematic of the system is shown in Figure 8. The system will be best understood if the reader follows this figure while reading the explanation below.

While the washer is in the wash cycle, the input to the system is oscillatory. The system, however, makes use of only one direction of the motion. Looking from the top, clockwise motion is transferred through a one-way clutch (Part 21) into winding two power springs. Counterclockwise motion is not transmitted. The power springs are wound until enough torque is developed to enable the output of the springs to start their motion. A gear is fastened to the bottom of each spring and the output of the springs is transmitted through these gears to a planetary gear system which increases the output by a factor of about 53. When the system reaches steady state, the spring output is about 15 rpm and the system output is about 800 rpm.

When the motor switches to the spin cycle, it turns the input shaft of the system counterclockwise at 800 rpm. Since in the counterclockwise direction, the clutch slips, the spring is not affected by this motion. However, this high speed creates a centripetal acceleration which provides a mechanical linkage to the output shaft. Thus, the output is driven directly at 800 rpm.

B. DESIGN OF POWER SPRINGS

At 800 rpm, the output shaft must develop .1 hp = 55 ft-lb/sec. Since the springs only rotate at about 15 rpm, the torque required at 55 ft-lb/sec of power is

$$Tw = 55 \frac{\text{ft-lb}}{\text{sec}}$$

$$w = 15 \text{ rpm} \times 2\pi \frac{\text{rad}}{\text{rev}} \times \frac{1}{60} = 1.57 \frac{\text{rad}}{\text{rev}}$$

$$T = \frac{55}{1.57} \times 12 = 420 \text{ in-lb}$$

This is the torque that would be needed if there were no mechanical losses in the system. However, mechanical losses invariably exist. Thus the springs should be capable of attaining about 500 in.-lb of torque, each spring having a maximum torque of 250 in.-lb.

The design of such a spring has been well developed in the periodical, Mainspring. The following is a brief survey of the formulae needed for design:

$$T = \frac{\nabla b h^2}{6} \qquad (1)$$

T = torque in in.-lb

Δ = stress on spring (psi)

b = width of spring (in.)

h = thickness (in.)

In order for the spring to deliver the maximum number of turns in a given space, it is necessary for the space occupied by the spring to be one-half the available space, i.e., the drum area minus the arbor area.

$$2Lh = \frac{\pi D^2}{4} - \frac{\pi A^2}{4}$$

or

$$D = \sqrt{2.55\ lh + A^2} \qquad (2)$$

where

L = length of spring (in.)

h = thickness

D = inside diameter of drum

A = diameter of arbor

It has been found experimentally that the arbor size should be from 15 to 25 times the spring stock thickness for best results.

The number of turns, n, can be expressed by the equation

$$n = \frac{\sqrt{2(A^2+D^2)} - (A+D)}{2.55\ h} \qquad (3)$$

With the above three equations and the knowledge of maximum allowable stress levels in spring stock, one can completely design a power spring for almost any application.

C. DESIGN OF THE ONE-WAY CLUTCH

Figure 11 contains an enlarged detailed drawing of the proposed one-way clutch. The clutch is specially fabricated out of two standard identical gears, as shown. The principle of operation involves frictional torque transmitted from the input shaft through four stainless steel balls to the gears (Part 21). This action takes place only when the shaft turns clockwise (as viewed from the top). A counterclockwise motion causes slipage to occur so that the gears remain stationary. The stainless steel balls rest on an inclined surface sloped toward the contact region. This eliminates any slipage that might occur in the drive direction.

D. PLANETARY DRIVE TRAIN DESIGN

The planetary gear speed increaser system involves two sets of planetary gears. These two sets are connected in such a manner that the carrier shafts and the sun gear shafts are the same. The input from the power springs is transmitted to one of the annulus gears (A_2). The other annulus gear (A_1) is fixed or grounded and remains stationary during the operation of the system. The output is the sun gear shaft.

The following is a method for determining the gear increase of such a system. Since the speed ratios are inversely proportional to diameter ratios, the diameter of the gears can be used to determine the speed ratios. The method is as follows:

(1) Write down all the gears involved in the system including the carrier shaft, e.g.,

$$\underline{S_1}\quad \underline{S_2}\quad \underline{A_1}\quad \underline{A_2}\quad \underline{P_1}\quad \underline{P_2}\quad \underline{C}$$

where

S_1 and S_2 are the sun gears

A_1 and A_2 are the annulus gears

P_1 and P_2 are the planet gears

C is the carrier shaft.

(2) Hypothetically consider the whole system to rotate one revolution. Therefore put a +1 under each symbol

$\underline{S_1 = S_2}$ output	$\underline{A_1}$	$\underline{A_2}$ input	$\underline{P_1}$	$\underline{P_2}$	$\underline{C}$
+1	+1	+1	+1	+1	+1

$$\frac{DA_1}{DP_1} \times \frac{DP_1}{DS_1} \quad -1 \quad \frac{-DA_1}{DA_2} \quad \frac{-DA_1}{DP_1} \quad \frac{-DA_1}{DS_1} \quad \frac{DS_2}{DP_2} \quad 0$$

$$1 + \frac{DA_1}{DS_1} \quad 1 - \frac{DA_1}{DA_2} \; .$$

(3) For the rest of the calculation, consider the carrier shaft to be fixed. Thus a zero is placed in the C column.

(4) For this problem, A_1 is fixed and its total motion is zero. Thus a -1 must be placed in this column.

(5) The remainder of the process involves finding the effect of a -1 turn of A_1 on the rest of the gears. The result is shown above.

(6) Add each column. By taking the ratio of the output to the input, the speed ratio can be found.

For the case at hand, the output is the sun shaft and the input is gear A_2. Thus:

$$\text{gain} = \frac{1 + \frac{DA_1}{DS_1}}{1 - \frac{DA_1}{DA_2}}$$

Note: The D with subscript indicates the diameter at that particular gear.

This same procedure can be applied to any palnetary gear system. The actual diameters needed will be determined later in this report. It should be noted, however, that a DA_1/DA_2 ratio close to one will produce a large gain.

V. ANALOG COMPUTER SIMULATION

The system previously described is a dynamic system and as such can be readily simulated on an analog computer. The purpose of this simulation is to examine the operation of the system and to determine the number of turns re-

required in the power springs. The results of this study are displayed in Figures 4, 5, 6, and 7. This study shows that the optimum number of turns is about ten. Higher number of turns cause an extended delay in the output response. Lower number of turns cause very large fluctuations in the output response.

A. EQUATIONS USED IN SIMULATION

The input to the spring can be represented as in Figure 1, assuming that the motor oscillation is sinusoidal. This motion can be simulated by first creating a sine wave with the frequency of 0.5 cps and an amplitude of .5 volt. By rectifying and integrating this signal, the spring input is simulated. Figure 2 displays a representation of the circuit used in this simulation. Amplifiers 1, 2, 3, and 4 and the included potentiometers accomplish the operation.

The spring requirements, as explained earlier, are such that the spring must be capable of delivering at least 500 in.-lb of torque. Figure 13 shows a typical relation between spring torque and the number of turns of the spring. This curve can be approximated by an equation of the form

$$T_S = C(X)^{.5} \qquad (4)$$

where

T_S = torque (in.-lb)

X = number of turns of spring

If we define X_2 as the input displacement to the spring and X_3 as the output displacement of the spring (both expressed in revolutions) then clearly the equation becomes

$$T_S = C(X_2 - X_3)^{.5}$$

Knowing that at X_3-X_2 = maximum, T_S = 500 in.-lb, C can be determined for different total spring turns. This relation was used for total spring turns of 5, 10, 15, and 20. For each case, C is determined.

The output shaft will be driving a centrifugal pump delivering 1/10 hp at 800 rpm. The torque needed for this power value is

$$\frac{1}{10}\text{ hp} = 55\,\frac{\text{ft-lb}}{\text{sec}} = T_o W$$

$$W = 2\pi \times \frac{800}{60} = 83.7\,\frac{\text{rad}}{\text{sec}}$$

$$T_o = \frac{55}{83.7} = .657 \text{ ft-lb} = 7.9 \text{ in.-lb}$$

T_o = torque at output shaft.

It was assumed for practical reasons that it takes 2.0 in.-lb to start the pump.

For most pumps of the design mentioned above, the torque is a function of the square of the speed or rpm. Thus

$$T_o = 2 + C_1 N_1^2 \tag{5}$$

where C_1 can be determined by the condition that at 800 rpm, T_o must equal 7.9 in.-lb

$$C_1 = \frac{5.9}{(800)^2}$$

Between the output shaft and the spring output is a large gear reduction. The spring is designed to operate at 15 rpm output.

Therefore,

$$T_S = \frac{800}{15} T_o \times \frac{1}{.85}$$

and

$$N_2 = \frac{15}{800} N_1$$

where

T_S = spring torque

N_2 = spring output rpm

Note: The factor 1/.85 was included as an efficiency term assuming that mechanical losses occur in the gear reduction.

Substituting the above equations into Eq. (5) gives,

$$T_S = 125 + 1.65 N_2^2 \tag{6}$$

The relation between X_3 and N_2 should be quite obvious

$$X_3 = \int n_2 \, dt \qquad (7)$$

Equations (4), (6), and (7) and the input function X_2 are the basis for the computer program. Figure 3 expresses these equations in algorithm form and gives a block diagram of the mechanical system.

It should be noted that this program actually represents a simulation of the operation of only the spring. It was assumed that the gear system only changed the output by a predictable factor (namely, 800/15). No provision was made for gear backlash or friction which may have a damping effect on the response. The results of this simulation should be analyzed with this in mind.

B. POTENTIOMETER SETTINGS

The values of each of the potentiometers will be explained according to the numbers on Figure 2.

(1) Potentiometers 1 and 2 regulate the frequency of the sine wave. Needed is 5 cps.

$$W = 2\pi F = 2\pi \times .5 = \pi = 3.14$$

Thus the values of both potentiometers is .314.

(2) The product of P_4 and P_3 must equal .005 so that the output of amplifier 4 is the same as Figure 1.

(3) $P_6 = .5$. This along with a gain of 10 across amplifier 5 increases the signal from amplifier 4 by a factor of 5.

(4) Amplifiers 6 and 7 along with the multiplier feedback perform a square root operation. The determination of P_7 is as follows:

Define X ≡ total number of spring turns

$$T_S = C(X_2 - X_3)^{.5} \qquad (8)$$

$$C = \frac{500}{\sqrt{X}}$$

Output of amplifier 7 = $100(X_2-X_3)^{.5}/\sqrt{20}$ if output of amplifier 9 = .1 T_S then

$$.1\, T_S = 10 \times P_7 \times 100 \, \frac{(X_2-X_3)^{.5}}{\sqrt{20}} \qquad (9)$$

Combining (8) and (9) gives

$$X = \frac{20\sqrt{.05}}{\sqrt{.P_7}} \tag{10}$$

$\therefore$

X	P_7
20	.05
15	.089
10	.20
5	.80

(5) P_{15} delays the propagation of the signal from amplifier 9. From the equation

$$T_S = 125 + 1.65 N_2^2$$

it becomes apparent that the value of P_{15} must be .125.

(6) $P_8 = 1/1.65$ to give $.1 N_2^2$ as indicated.

(7) The output of amplifier 14 is $N_2/.316$. In order to get $.2N_2$ out of amplifier 10, P_8 must equal .632.

(8) N_2 is expressed in rpm. X_3 must be expressed in revolutions and the integration is in sec. Therefore:

$$P_{10} = \frac{1}{60 \times .2} = .0834$$

(9) $P_{11} = .5$ for the same reason $P_8 = .5$.

The output as displayed in Figures 4, 5, 6, and 7 were taken from the outputs of amplifiers 9 and 10.

VI. MODELING OF THE PHYSICAL SYSTEM

It was also desirable that a physical system be made to observe the response of the system and to prove (or disprove) the feasibility of such a system.

The Pic Company, a subsidiary of the Benrus Watch Company, produces miniaturized mechanical components built to very rigid tolerances for use in modeling mechanical systems. Due to lack of time, a physical model was not built.

However a physical system was completely designed, using Pic products, to the point where the parts could be ordered and the system built. Figure 8 is a schematic diagram of the entire system. Figures 9-12 are detailed drawings of some of the parts. The following is a description of the individual parts as numbered in Figure 8.

Part 1 (Pic Part Nos.: CW-10-A, AT-1 through 6)

Frame which supports system. There are three such plates that are connected by round spacer posts.

Part 2 (Pic Part No. C1-3)

Precision set screw collars. Four of these parts are needed at various places. They are used as shaft supports.

Part 3 (Pic Part No. AM-5)

Flanged oil-less bearings. Fourteen bearings are needed.

Part 4 (Pic Part No. B2-1)

Laminated brass shim spacers. Four are needed, one for each collar.

Part 5 (Pic Part No. 69-64)

Precision Spur Gears. Two needed to transmit input shaft motion to the power springs. A ratchet must be mechined into the bottom of each gear. This is shown in one of the detailed drawings.

Part 6 (Pic Part No. AM-14)

Minature oil-less bearings. Two needed for Part 21.

Part 7

A retainer that must be used to keep the spring shaft from going in the reverse direction. This part is shown in detail. It is not a standard Pic part and must be fabricated.

Part 8

A centripital clutch device which engages the output shaft directly when the input shaft rotates at 800 rpm. This part is also shown in detail and must be fabricated.

Part 9

Support for the output shaft. This support is connected to the major support system.

Part 10 (Pic Part No. J3-72)

Hubless spur gear. The gears are to be screwed into the bottom of each of the power springs. The spring output is transmitted through these gears.

Parts 11 and 12

Part 11 is a precision internal gear. Two are needed and are connected by a spacer ring, Part 12, which must be made. Part 11 has Pic Part No. CE 3-200. It represents annulus gear A_2.

Part 13

Carrier shafts and assembly for planetary gear system. This must be fabricated as is shown in the detailed drawing.

Part 14 (Pic Part No. Z3-1)

Shaft Retainer Rings. Nine rings are needed to retain different gears.

Part 15 (Pic Part No. J3-83)

Hubless Spur gears. Three gears are needed as planet gears to be mounted on carrier shaft.

Part 16 (Pic Part No. J3-75)

Hubless Spur gears. Three gears are needed as planet gears to be mounted on carrier shaft.

Part 17 (Pic Part No. CE 3-192)

Precision Internal gear. This is annulus gear A_1 which is connected to system support so that it remains stationary.

Part 18

Connecting Link. Part 17 is a bearing surface for annulus gear A_2.

Part 19 (Pic Part No. G9-34)

Precision Spur gear. Upper sun gear which is attached to the output shaft.

Part 20 (Pic Part No. G9-42)

Precision Spur gear. Lower sun gear which is attached to the output shaft.

Part 21 (Pic Part No. G9-64)

One-way clutch. This part is made from two precision spur gears. A detailed drawing is included.

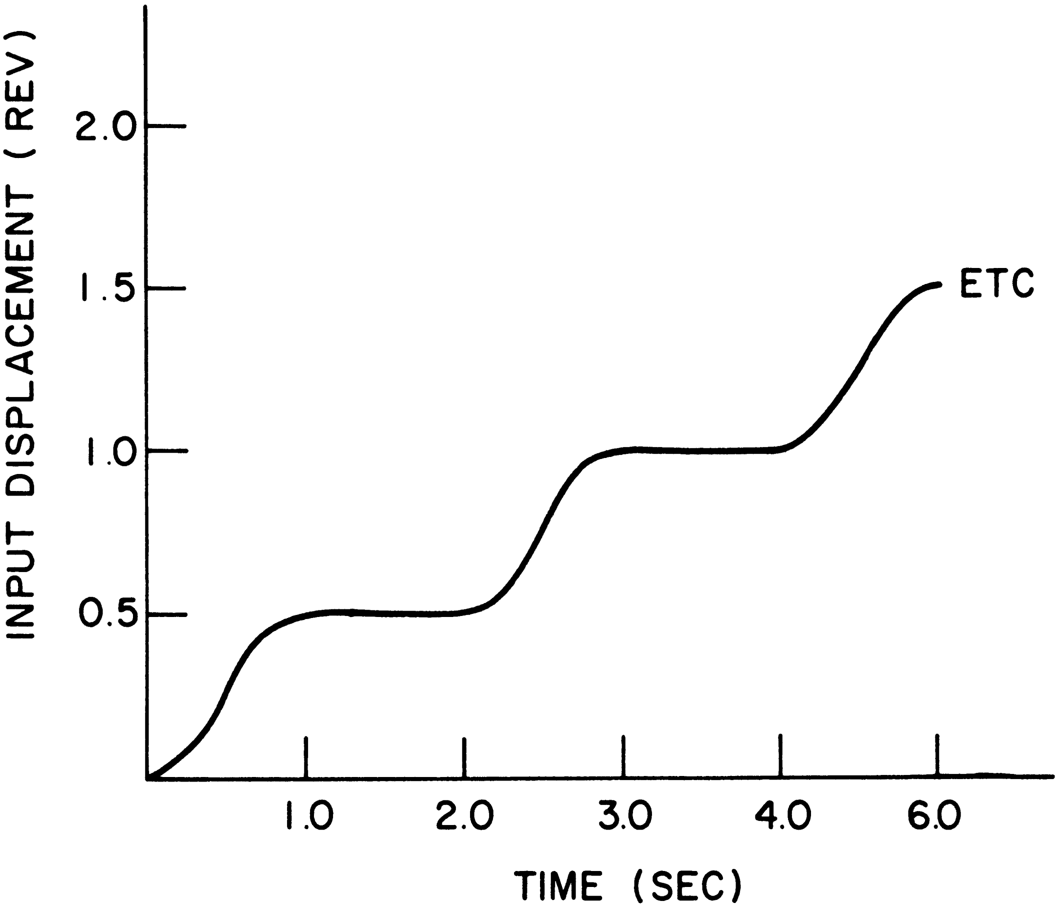

Figure 1. Input function (X_2).

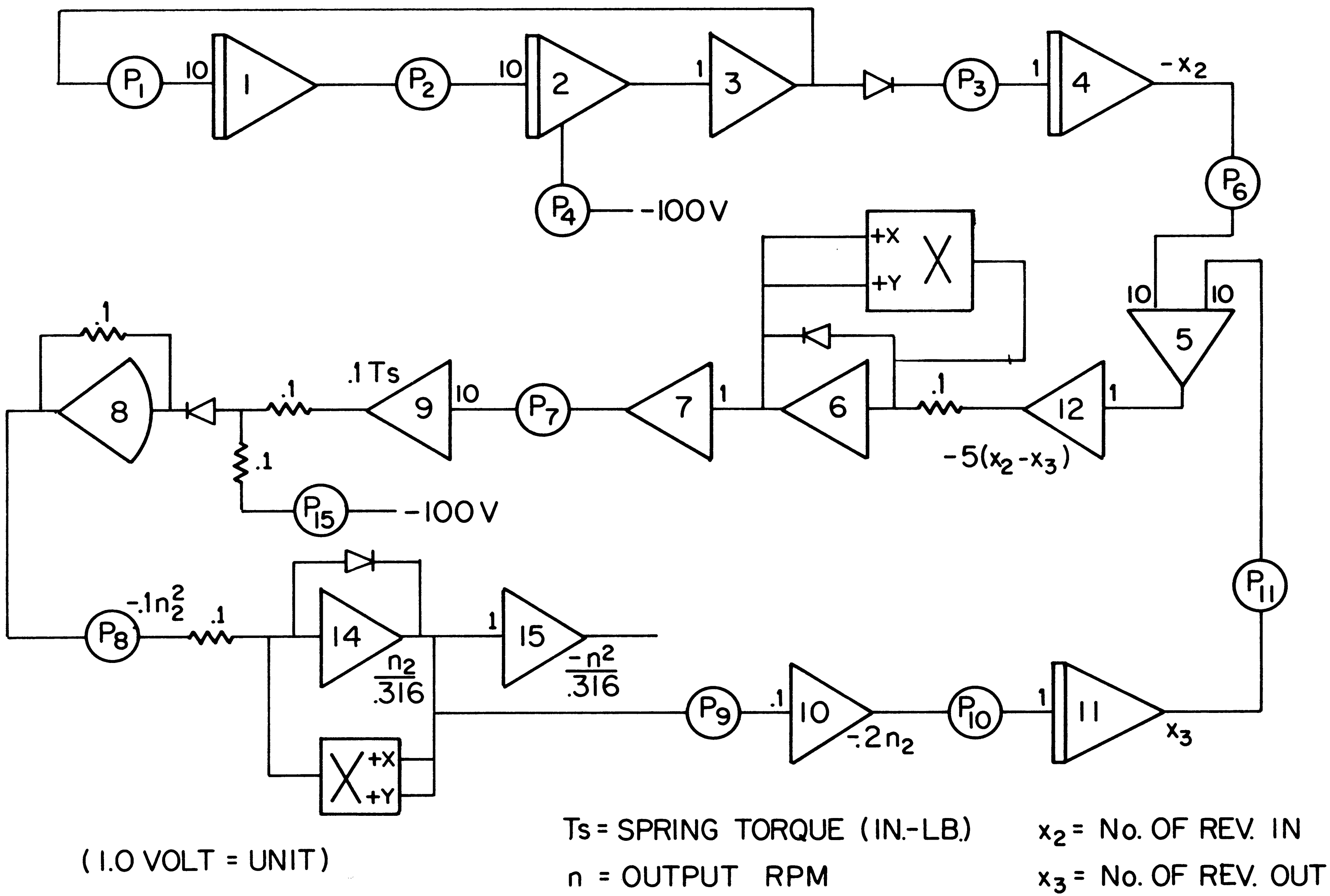

Figure 2. Analog circuit for input and output of spring.

(1) x_2 = INPUT DISPLACEMENT TO SPRING

(2) $T_s = C(x_2 - x_3)^{.5}$, C depends on # of turns of spring

(3) $T_s = 125 + 1.65\, n^2$

(4) $x_3 = \int n \, dt$

WHERE:

x_3 = DISPLACEMENT OF SPRING OUTPUT

Ts = SPRING TORQUE

n = OUTPUT RPM OF SPRING

BLOCK DIAGRAM

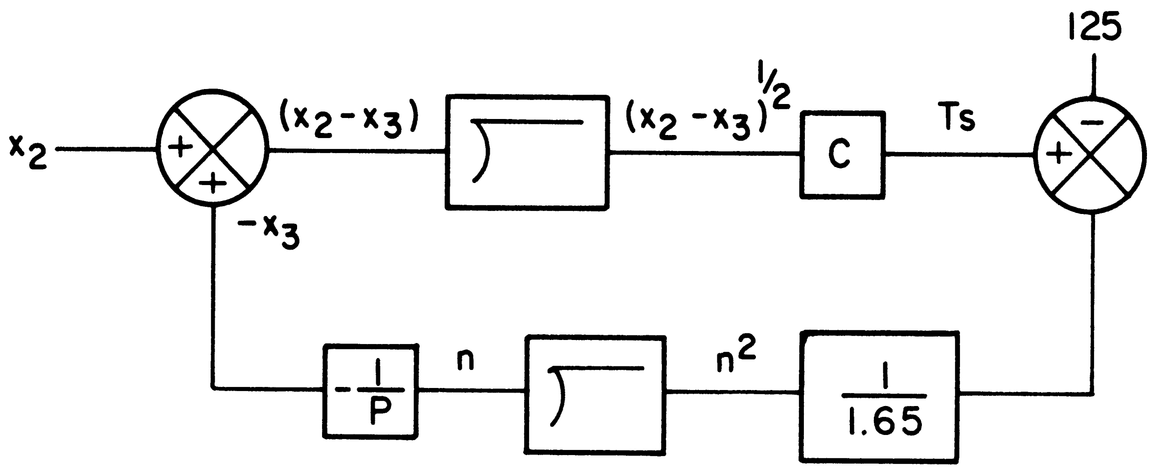

Figure 3. Equations used for analog simulation.

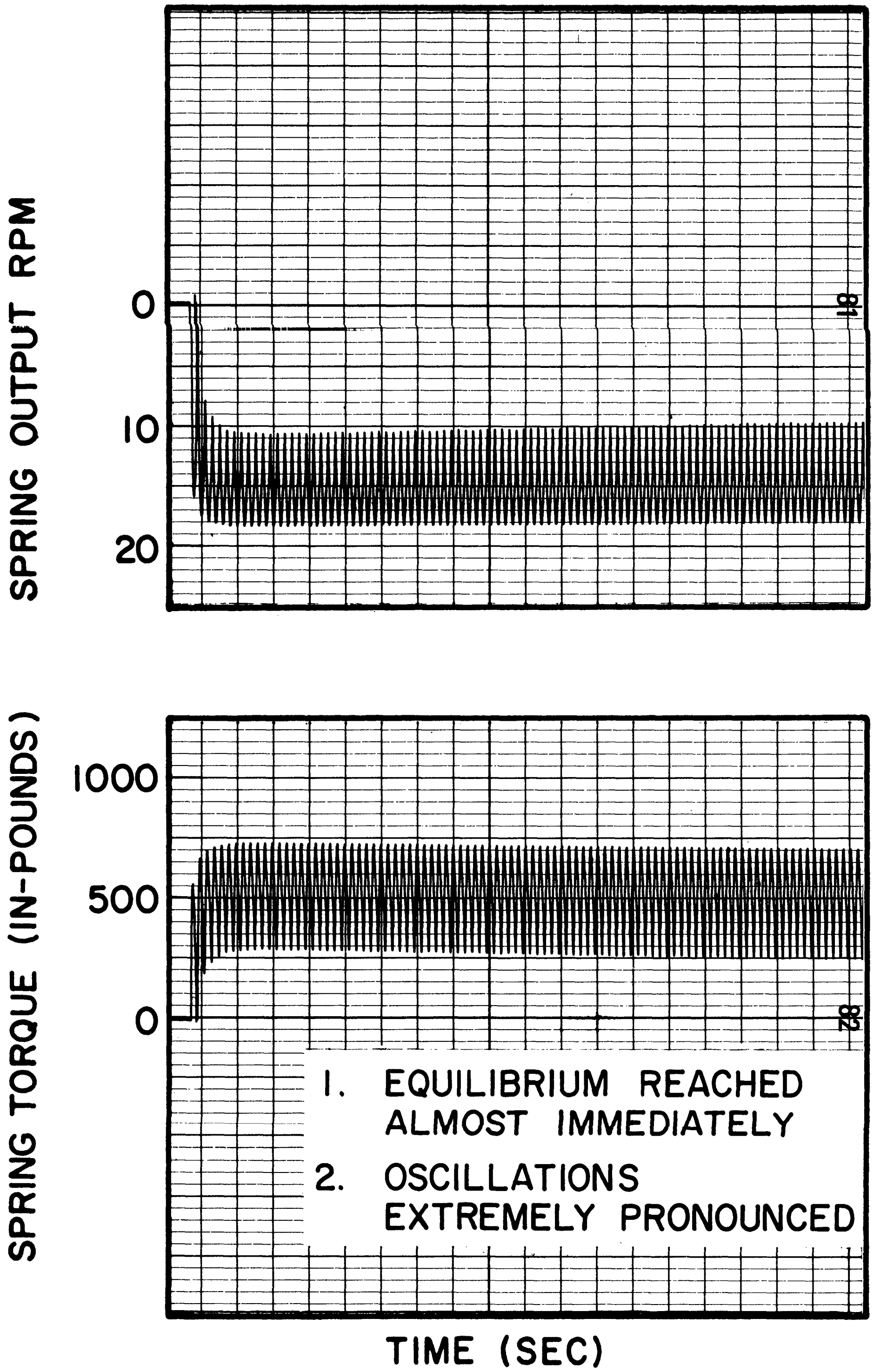

Figure 4. Power spring torque and output rpm for spring with five turns.

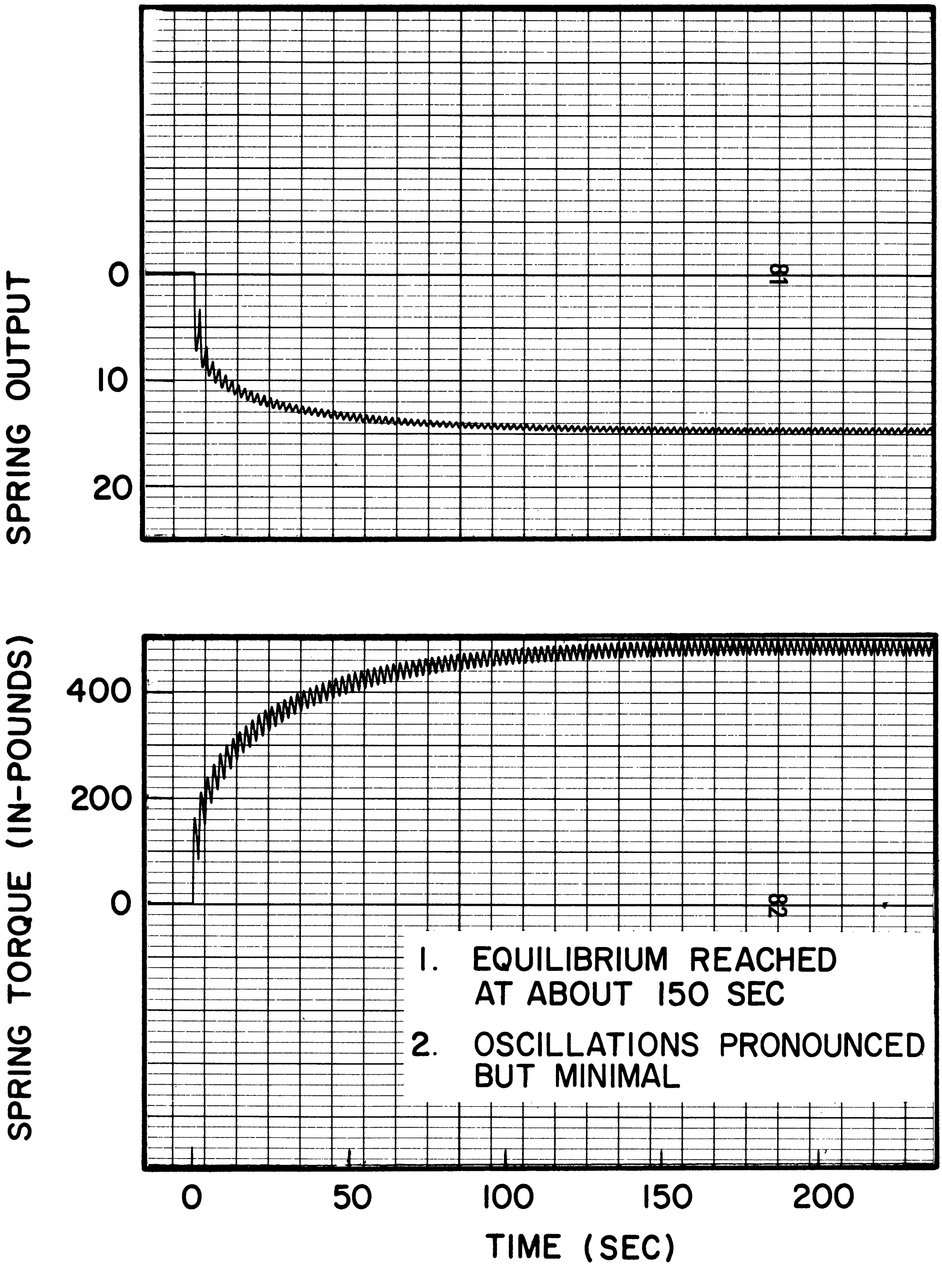

Figure 5. Power spring torque and output rpm for spring with ten turns.

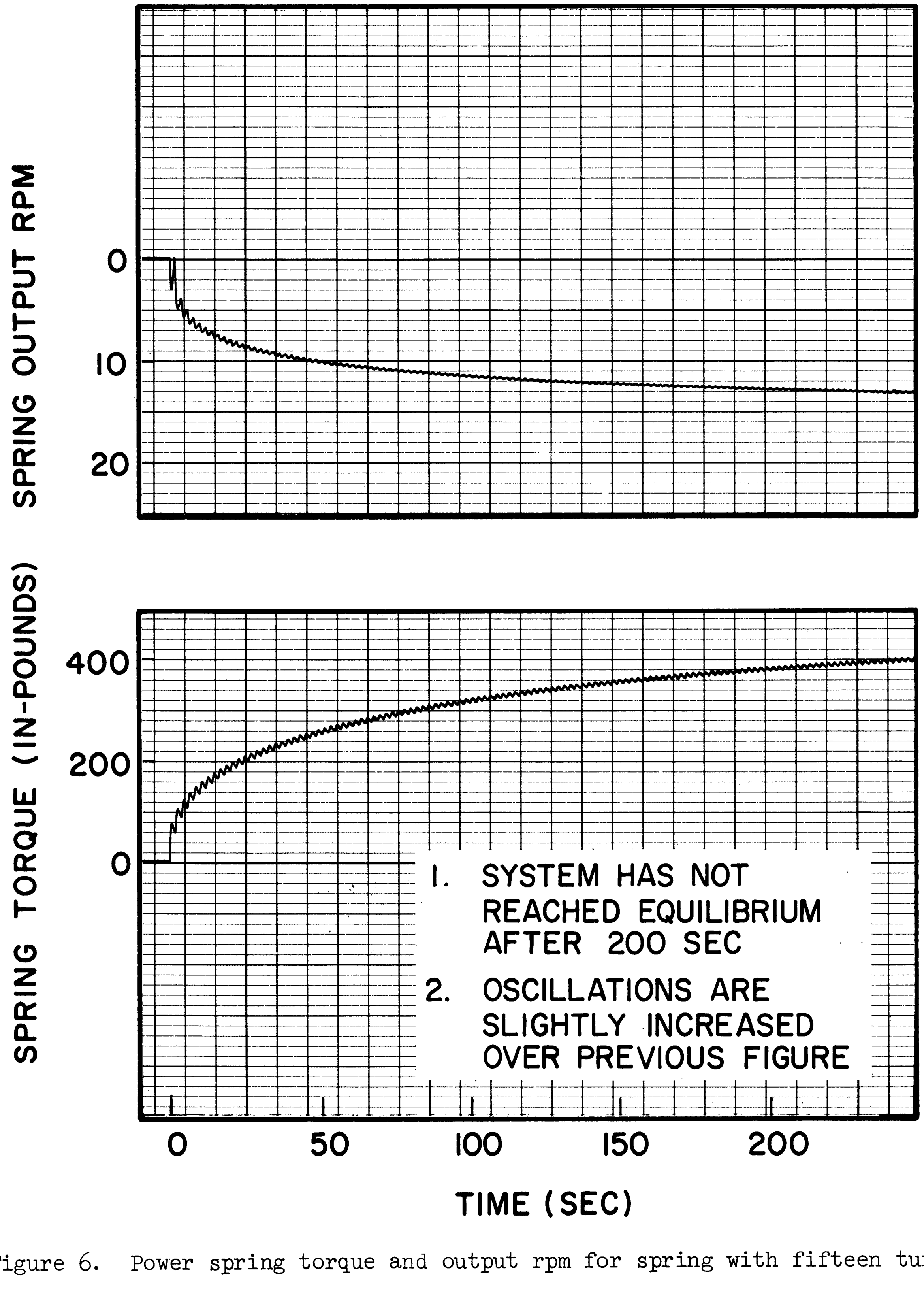

Figure 6. Power spring torque and output rpm for spring with fifteen turns.

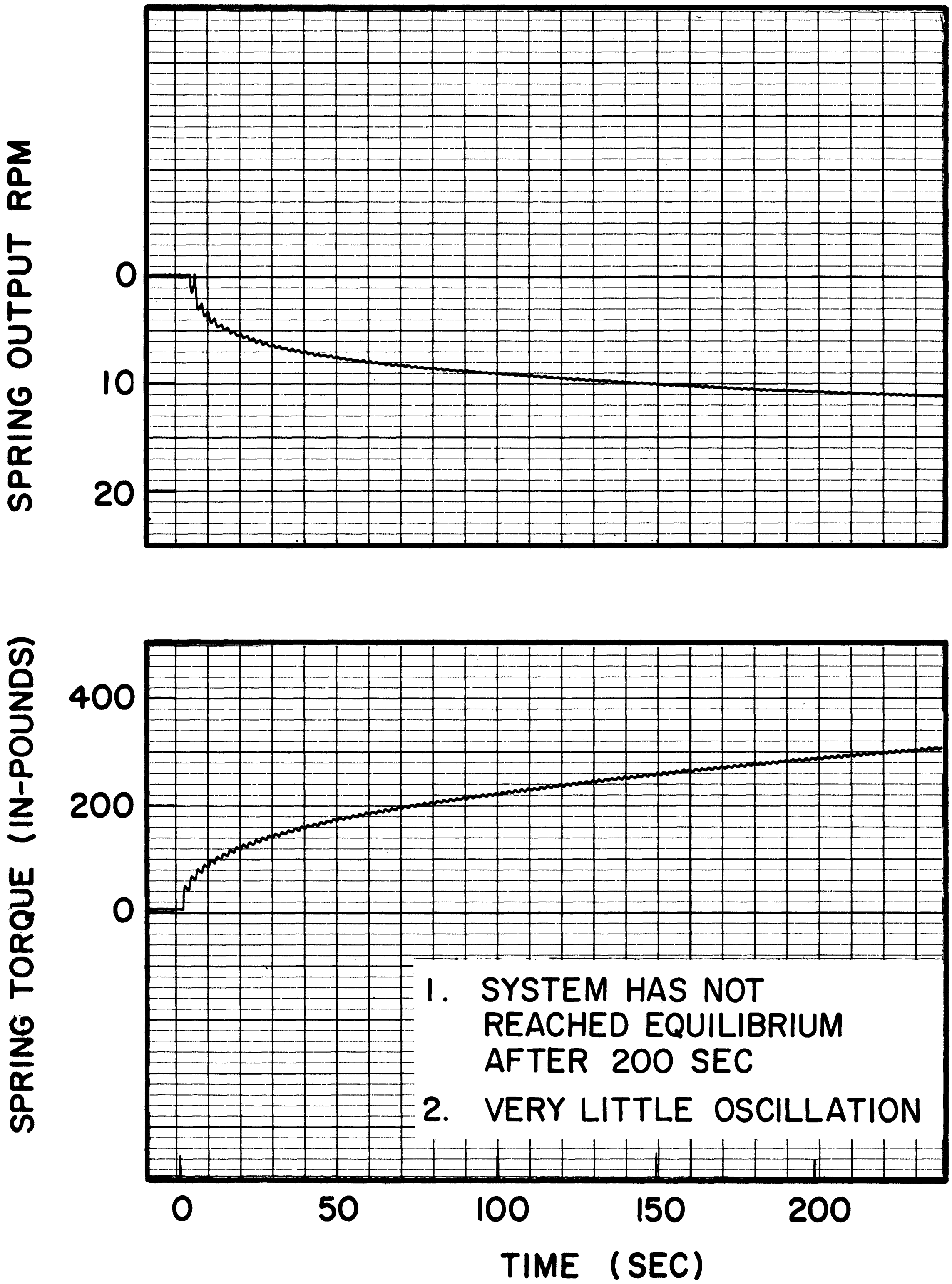

Figure 7. Power spring torque and output rpm for spring with twenty turns.

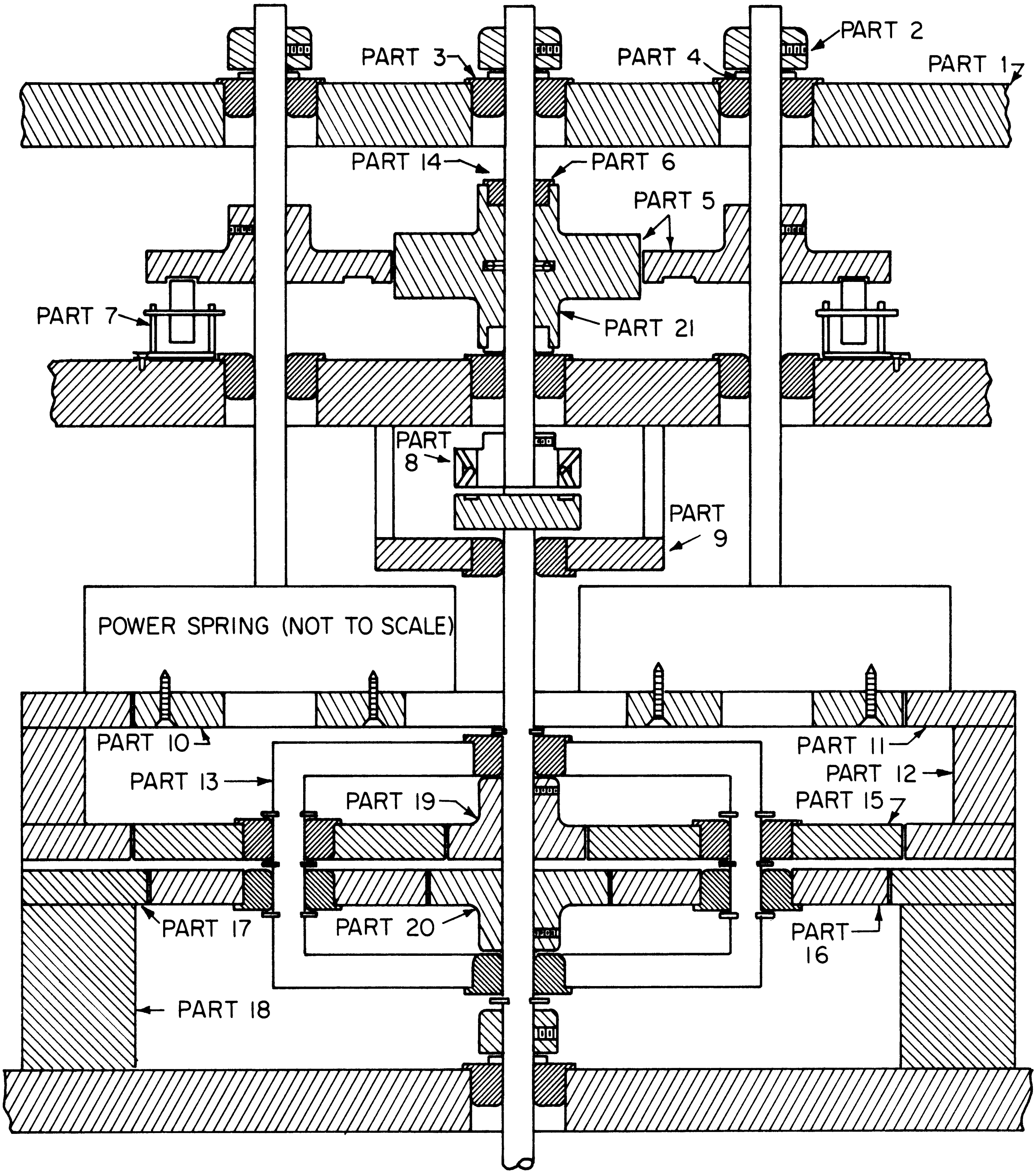

Figure 8. Schematic of system.

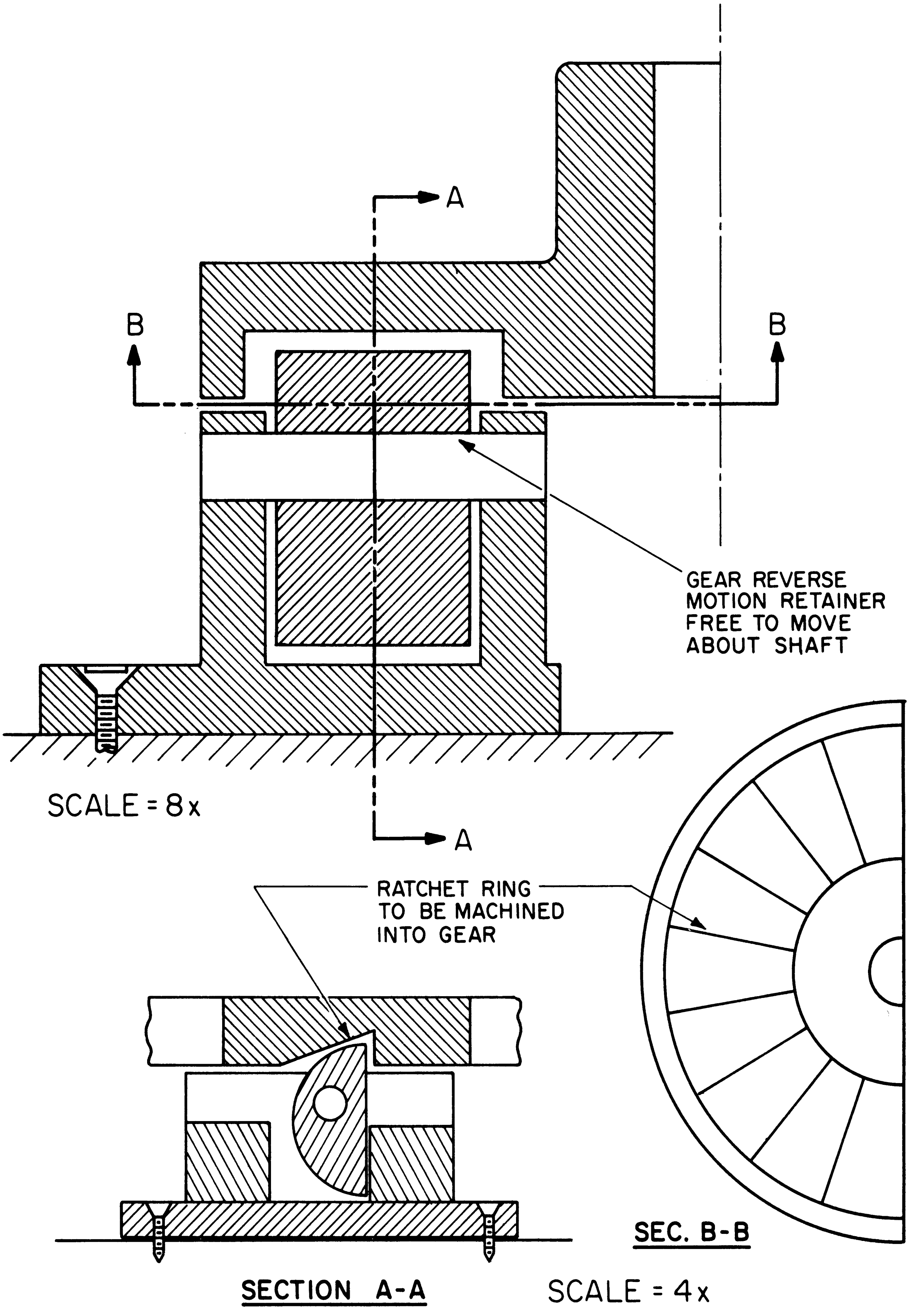

Figure 9. Detail of Parts 5 and 7.

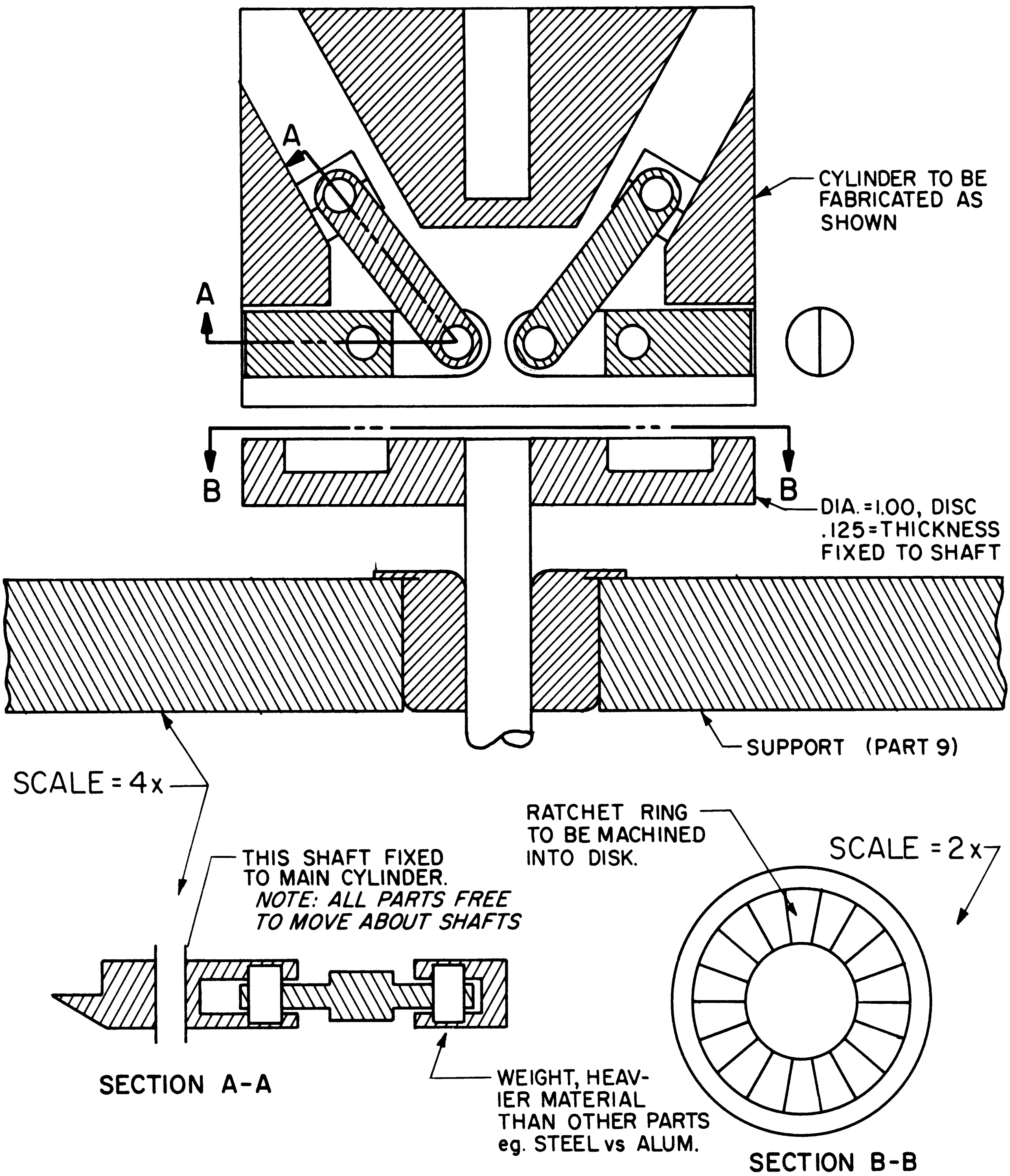

Figure 10. Detail of Part 8.

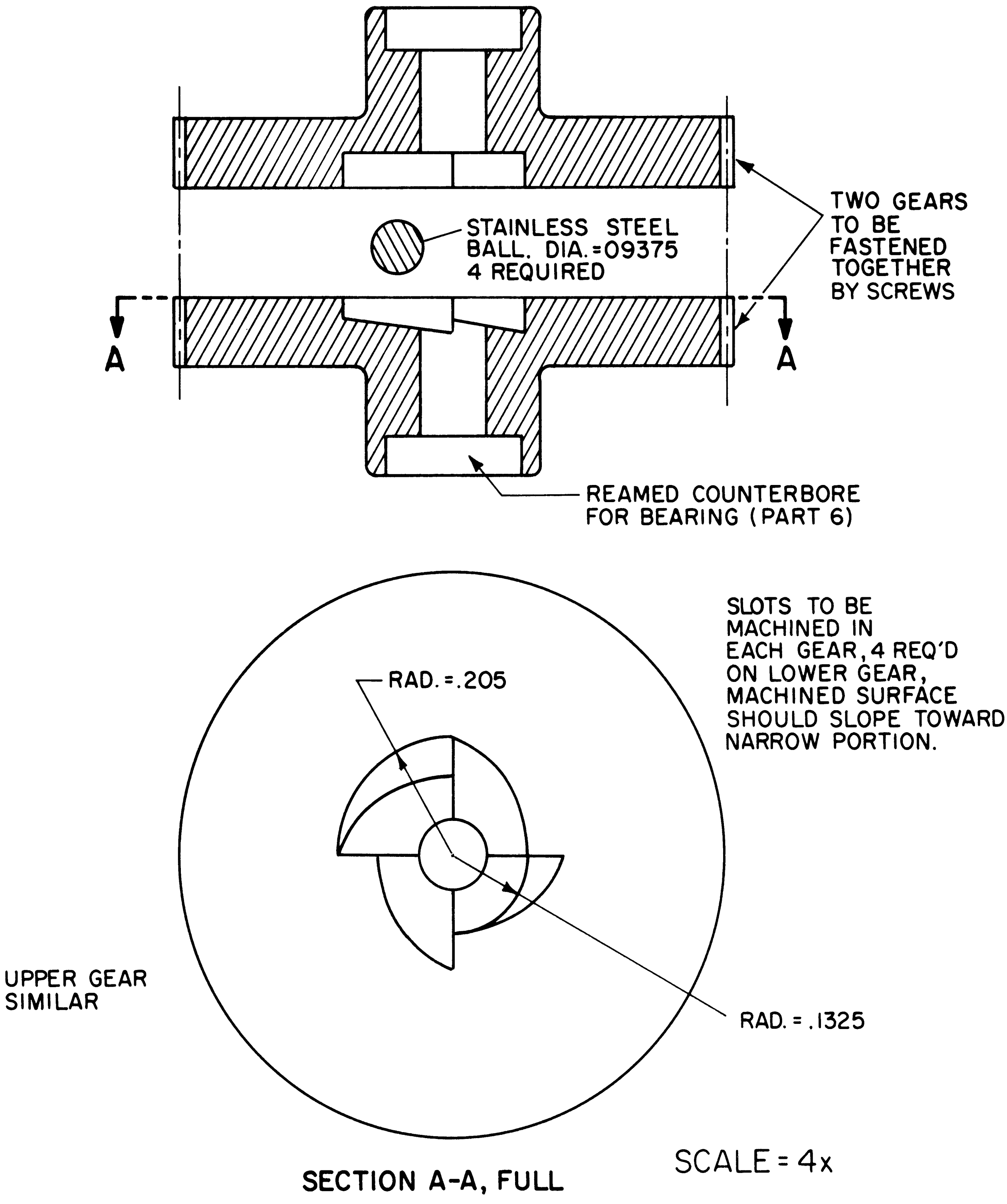

Figure 11. Detail of Part 21.

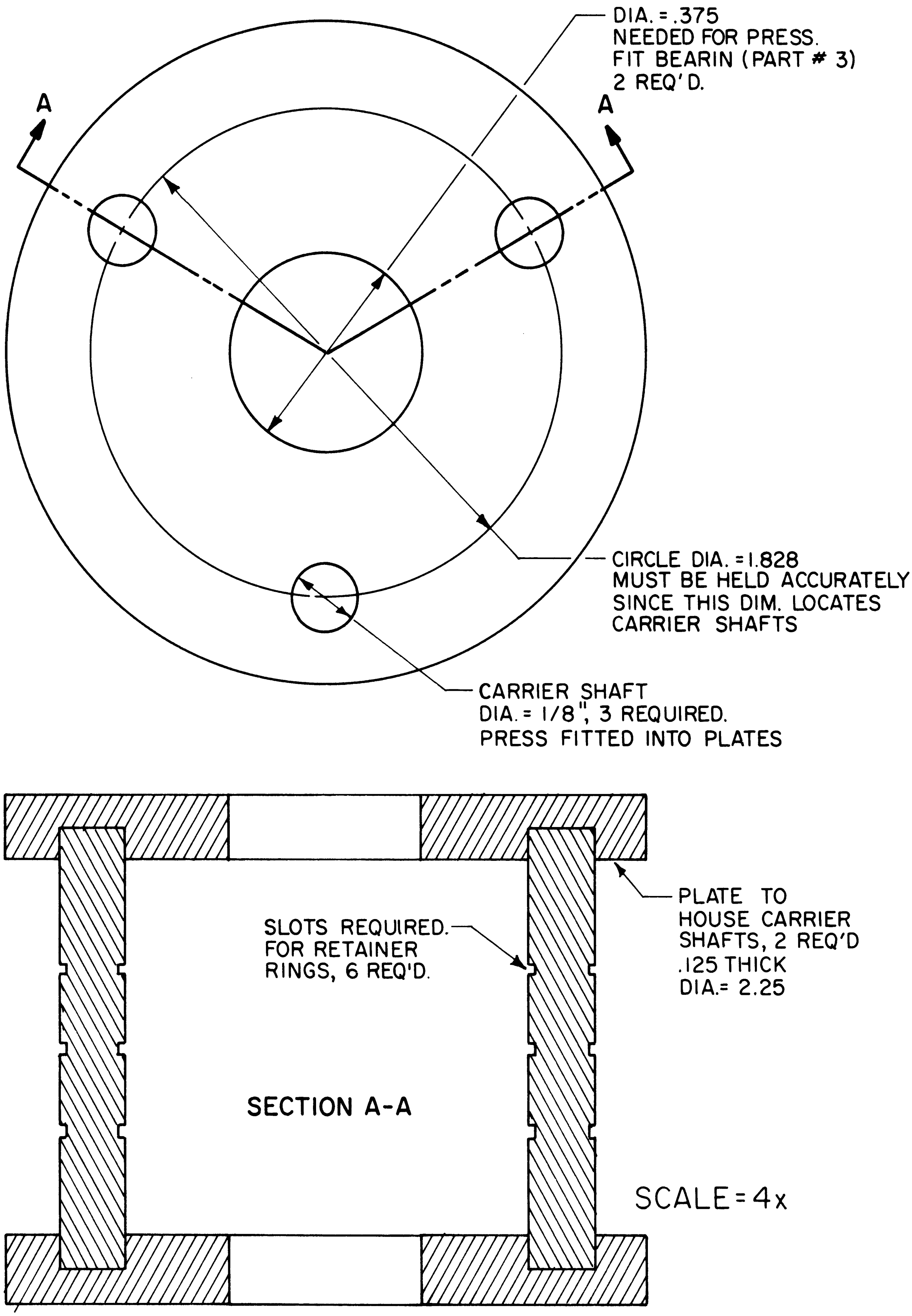

Figure 12. Detail of Part 13.

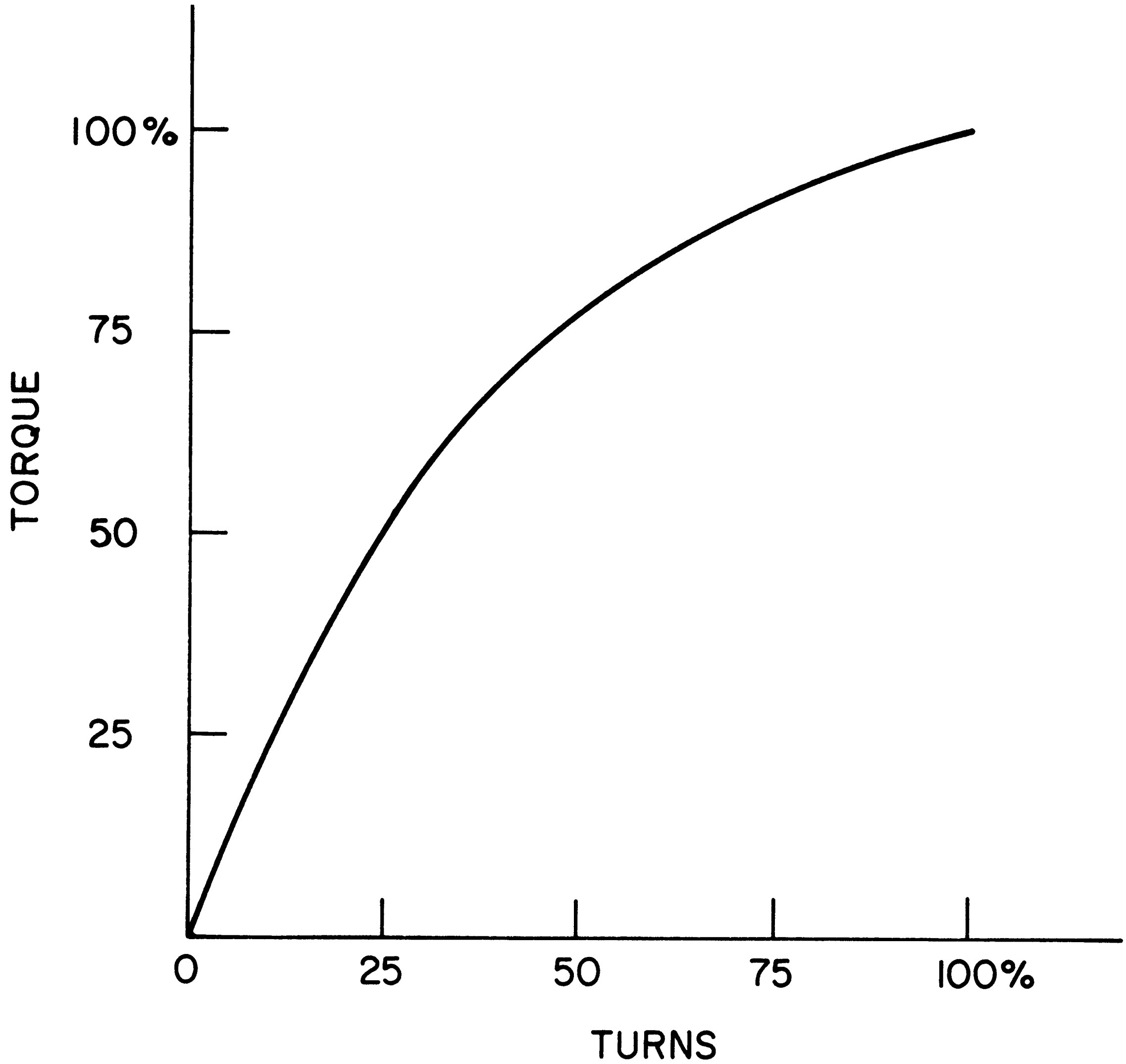

Figure 13. Typical torque curve for power springs.

HEAT AND MASS TRANSFER IN THE DRYING OF TEXTILES

Thomas Farrington

I. INTRODUCTION

"Drying" is a very general term used in industry today. It describes a process that might now be more properly classified as an art than a science. The knowledge of the process of heat and mass transfer involved in the removal of a liquid from within a body has not entirely made the transition from theory to practice. Today's approach is usually either to approximate data obtained from a system by purely empirical equations, or to greatly simplify theoretical equations to fit such data. In either case, the working relationships are dependent to an unknown degree on the system parameters. The intent of this report is to discuss what has been done in the field of heat and mass transfer and to try to bridge the theory-practice gap in a manner useful to the Whirlpool Corporation.

The scope of this report has necessarily been narrowed because of time limitations and the background of its author. It is meant to be used as an aid to more comprehensive evaluation of the ideas it points to and as a guide to the literature presently available on the subject.

II. THEORETICAL APPROACH TO HEAT AND MASS TRANSFER IN POROUS BODIES

There are two rather distinct fields of study in the literature on heat and mass transfer in moist bodies. The first of these is research and development on textile materials, mainly wools, with emphasis on a particular phenomenon the author wishes to describe. The second is research and development done on a more general scale, but primarily aimed at increasing knowledge in the area of building and structural materials. The first section of my report is devoted to a description of the more general studies, while the next section is devoted to the more specific textile studies.

A general theoretical approach to drying and combined heat and mass transfer in general is an extremely complicated undertaking. My research indicates that very little has been published on the subject by American or European authors, with the exception of the Russians, who appear to have done a considerable amount of work. Luikov[6] and Mikhalov and Lykov[9] have done theoretical analyses which are the most comprehensive I could find. Though both are concerned with structural materials rather than textiles, their books cover the development of their theories along with broader applications. Their work is so extremely complicated and sophisticated, however, that I can only outline what they have done, and leave the reader to delve into the original for a more complete understanding.

In the drying of a textile containing water, the concern is with removing as much water as possible in the least amount of time, without damaging the textile material. This requires a knowledge of just how the water and the cloth are coupled so that we can formulate a way to separate them. The most comprehensive descriptions of this coupling that I found were given by Luikov[6]. He classified the bonds of the water and textile or water and any dispersed media into four possible categories according to their magnitudes and nature. The first of these is chemically combined water. This is water bound in the form of hydroxyl ions and in the form of molecular compounds of the crystal hydrate type. The second is absorptionally bound water or the water that forms a monomolecular layer on the internal and external surface of a capillary-porous body. Third, there is capillary-bound water which is contained in the capillaries of the body and bound by a free minisci. The fourth bond group is that of osmotically bound or entropy-combined water. These bonding groups appear to have been ordered more for mathematical simplicity than physical differences in some cases, but they do cover all the types of moisture retention I encountered in my research; no other author's explanation did this.

Luikov's and Mikhalov and Lykov's approaches were both to derive the general system of combined heat and mass transfer equations that fit porous bodies and then to define the parameters within these equations, using the relationsips they felt best described the bonding of water and textile. Once they accomplished this, they attempted to apply their equations by assuming simplifying conditions that they felt fit the practical situations they were concerned with. By this procedure they obtained results that were in agreement with experimental data. For anyone capable of following the mathematics and the assumptions based on the mathematics, Luikov's and Mikhalov and Lykov's works are excellent references.

III. HEAT AND MASS TRANSFER IN TEXTILES

The second field of investigation I encountered was concerned primarily with the drying of textiles. Each author tended to strive for a solution to one particular phenomenon, as opposed to the general solutions discussed in the previous section. Their approaches and theories seem to be the most promising for immediate further work.

The most refined approach, and the one I think best fits Whirlpool's needs was introduced by Cassie[2]. Cassie's theory and development began with a control volume of differential length dx in a bed of fibres conditioned at some definite temperature and water vapor concentration. At time zero, Cassie assumed a flow of incident air through a unit cross-section normal to the x-direction, which contained a different temperature and water vapor concentration than the original bed, as illustrated in Figure 1.

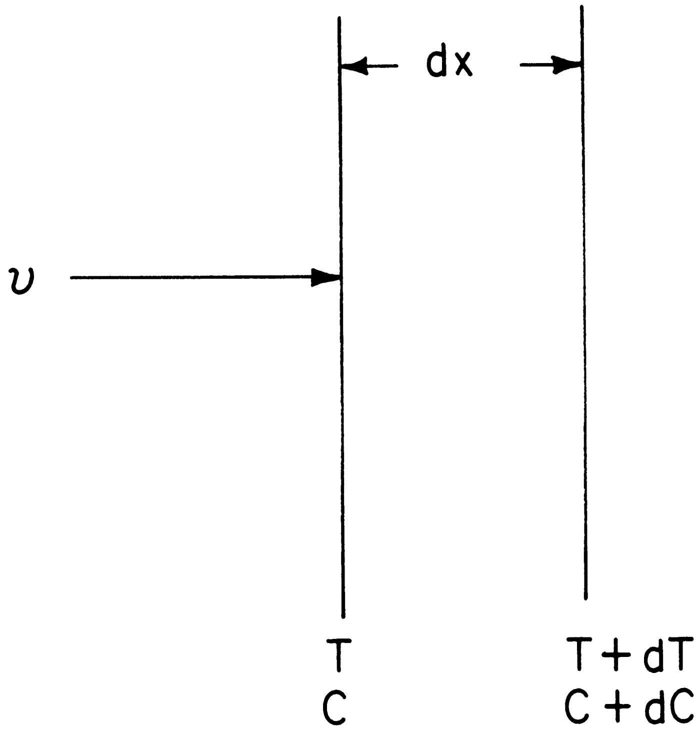

Figure 1

The equations he used were:

$$S \frac{\partial T}{\partial t} = - \rho_a c_a v \frac{\partial T}{\partial x} + q\rho \frac{\partial M}{\partial t} \quad \text{heat balance}$$

$$\frac{\partial C}{\partial t} = - v \frac{\partial C}{\partial x} - \rho \frac{\partial M}{\partial t} \quad \text{water vapor balance}$$

where S = heat required to raise the temperature of 1 cc of the air-textile mixture through 1°C (assuming no mass of water vapor change)

ρ_a = density of air

c_a = specific heat of air at constant pressure

q = heat of absorption of lag of water by the textile

ρ = the mass per cc of the textile

M = fractional regain of the fibres

C = water vapor concentration

T = temperature

v = velocity of incident air flow

t = time

x = direction of incident air flow.

Cassie then made the assumption that the fractional regain of the fibres varied linearly with temperature and water vapor concentration so that

$$\frac{\partial M}{\partial t} = \sigma \frac{\partial C}{\partial t} - \omega \frac{\partial T}{\partial t}$$

where σ and ω are the constants of linearity. Substituting this equation into the heat and mass balance equation gave

$$V \frac{\partial T}{\partial x} + \frac{\partial T}{\partial t} - \alpha \frac{\partial C}{\partial t} = 0$$

$$W \frac{\partial C}{\partial x} + \frac{\partial C}{\partial t} - \beta \frac{\partial T}{\partial t} = 0$$

where

$$V = \frac{v}{(S + q\rho\omega/\rho_a c_a} \qquad \alpha = \frac{q\rho\sigma}{(S + q\rho\omega)}$$

$$W = \frac{v}{(1 + \rho\sigma)} \qquad \beta = \frac{\rho\omega}{(1 + \rho\sigma)}$$

Cassie solved these equations by the method of normal functions. Multiplying the two equations above by r/V and z/W respectively and adding the two resulting equations yielded

$$u \frac{\partial f}{\partial x} + \frac{\partial f}{\partial t} = 0 \qquad (1)$$

where

$$f = (rT + zC) \qquad (2)$$

and u is given by

$$(\frac{1}{u} - \frac{1}{V})(\frac{1}{u} - \frac{1}{W}) = \frac{\alpha\beta}{VW} \qquad (3)$$

From Eq. (3), Cassie deduced that there are two values of u and from Eqs. (1) and (2), that there are two functions f(x,t) of the form

$$f(x,t) = g(x - ut)$$

This, he concluded, means that there are two linear combinations of C and T which are propagated unchange in form through the textile with the two velocities given by Eq. (3). By denoting the two functions given by Eq. (2) as

$$f_1 = (r_1 T + z_1 C)$$

$$f_2 = (r_2 T + z_2 C)$$

and solving by normal functions, he obtained

$$T = \frac{\frac{1}{r_1} f_1 - \frac{W}{\beta}(\frac{1}{V} - \frac{1}{u_1})\frac{1}{z_2} f_2}{1 - \frac{VW}{\alpha\beta}(\frac{1}{V} - \frac{1}{u_1})(\frac{1}{W} - \frac{1}{u_2})}$$

$$C = \frac{\frac{V}{\alpha}(\frac{1}{W} - \frac{1}{u_2})\frac{1}{r_1} f_1 - \frac{1}{z_2} f_2}{\frac{VW}{\alpha\beta}(\frac{1}{V} - \frac{1}{u_1})(\frac{1}{W} - \frac{1}{u_2}) - 1}$$

For the case of a textile conditioned to a fixed temperature and water vapor concentration and the air flowing through the textile is suddenly increased in temperature and water vapor concentration by $\Delta_o T$ and $\Delta_o C$, Cassie obtained the following equations for subsequent values of (x,t) for the increase ΔT and ΔC:

$$\Delta T = \Delta_o T \frac{(\frac{1}{V} - \frac{1}{u_2})f_1 - (\frac{1}{V} - \frac{1}{u_1})f_2}{(\frac{1}{u_1} - \frac{1}{u_2})} + \Delta_o C \frac{\frac{\alpha}{V}}{(\frac{1}{u_1} - \frac{1}{u_2})} (f_2 - f_1) \qquad (4)$$

$$\Delta C = \Delta_o T \frac{\frac{\beta}{W}}{(\frac{1}{u_1} - \frac{1}{u_2})} (f_2 - f_1) + \Delta_o C \frac{(\frac{1}{V} - \frac{1}{u_2})f_2 - (\frac{1}{V} - \frac{1}{u_1})f_1}{(\frac{1}{u_1} - \frac{1}{u_2})} \qquad (5)$$

Cassie interpreted the physical meaning of these equations by the following reasoning. In order to observe what a change in temperature or water vapor concentration separately would cause, all that needs to be done is to set either $\Delta_o C$ or $\Delta_o T$ equal to zero in the two equations above. Cassie's investigation was concerned with the propagation of the temperature change through the textile, so he discussed the case of $\Delta_o C$ being equal to zero, but because of the similarities and interrelationship between the equations, his ideas are also applicable to the moisture concentration change. He next assumed the case where $S = \rho_a c_a$ so that the textile could be regarded merely as a water absorbing medium without influencing heat exchange. He stated that this assumption introduced little error except when numerical calculations involving the second velocity of propagation were used, and that it greatly simplified the equations. Assuming $\Delta_o C = 0$ allowed Cassie to come up with the two equations:

$$\omega = \sigma \frac{dC}{dT}$$

$$\frac{1}{C}\frac{dC}{dT} = \frac{Q}{RT^2} - \frac{1}{T}$$

The first equation is Cassie's assumption of linear variation of M with T and C, with dM set equal to zero. The second is Korchhoff's relation for a condition of constant regain. σ is obtainable from known graphs of M plotted against C for a constant temperature; Q is the heat of absorption per molecule of water vapor, and R is the gas constant per molecule. This gives ω equal to $5 \times 10^{-3}/°C$ when σ is 10^4 cc/g which is the order of magnitude for most textiles and atmospheres common in Britain. From these calculations, Cassie further assumed ω/σ equal to $\rho_a c_a/q$ which he felt was valid for normal water vapor concentrations. From this he obtained

$$V = W = \frac{v}{(1 + \rho\sigma)}$$

$$\frac{\alpha\beta}{VW} = \frac{\rho^2\sigma^2}{v^2}$$

Substituting these values into Eq. (3) and noting $\rho\sigma$ was of the order of 2×10^3, he obtained:

$$\frac{1}{u_1} = \frac{2}{V} = \frac{2}{W} = \frac{2\rho\sigma}{v}$$

$$\frac{1}{u_2} = \frac{1}{v} \ll \frac{1}{u_1}$$

Finally, substituting these relationships into Eq. (4) with $\Delta_o C = 0$ gave

$$\Delta T = 1/2\ \Delta_o T(f_1 + f_2) \qquad (6)$$

From this reasoning, physical interpretation of the case of air at a constant vapor concentration flowing through a textile with changed initial temperature was easy for Cassie. First, the temperature change within the textile is propagated in two stages; one half of the total change at the same speed as the air, the other half at roughly 2.5×10^{-4} times this speed. If the textile had zero specific heat and no hydroscopic property, the total change would be propagated at the speed of the air. Since the fibres do have hydroscopic properties, the vapor concentration of the air must also remain in equilibrium with the textile. A difference in temperature between the fibres and the air, therefore, forces an equilibrium of temperature that effects the moisture equilibrium. If the temperature of the air is increased, the fibres also increase in temperature, and the water vapor concentration of the air must increase at the expense of the textile's water. If there was no heat of desorption the propagation would be unchanged, but since there is, heat must be given up by the air to maintain temperature equilibrium. This forces the air passing through the fibres of the textile to be cooled even though the specific heat of the fibres is zero. The cooling continues until the water vapor concentration is that required for equilibrium with the textile at the temperature it assumes, and at its original regain.

Equation (6) shows that this temperature is midway between the initial and final temperatures, and since

$$dC = (\omega/\sigma)dT$$

the water vapor concentration in the air simultaneously increases by ω/σ times the temperature increase. The new conditions advance just as rapidly as the atmosphere flows through the fibres, because the atmosphere and fibres can produce the change spontaneously when $\omega/\sigma = \rho_a c_a/q$.

At this point, Cassie noted that since the temperature acquired by the air and textile is determined largely by the properties of the air, the textile can be regarded merely as a system for maintaining constant relative humidity in the atmosphere when temperature changes occur. This extreme simplification is based on the fact that the heat of absorption of water vapor by the fibres is the only factor associated with the textile, and this is roughly equal to the latent heat of evaporation of water. The amount of heat required to maintain a constant relative humidity with a change of temperature by evaporating water is roughly equal to the heat obtained by cooling dry air the same amount. It is this fact that gives a temperature equal to the mean of the initial and final temperature and is unaffected by the textile.

The regain of the textile has not changed appreciably at this stage, but it is continually losing water to the atmosphere, so that after a much longer period the regain is changed sufficiently to be conditioned to equilibrium with the incident air temperature and water vapor concentrations. The assumptions made in deducing the mathematical equations imply that there is a sharp front between the regions of unchanged and changed regain; thus the slow component represents propagation of change of regain, while the fast one represents propagation of temperature and water vapor changes without change of regain.

The same processes are followed if concentration is increased while temperature remains constant; one front passes rapidly through the textile and across it roughly one-half of the total concentration change occurs the slower front follows bringing the second half of the concentration change and the total regain change. The faster front also brings with it an increase in temperature which disappears as the slower front proceeds; this corresponds to the first term in Eq. (5).

Experiments by Cassie and Baxter[3] supported Cassie's theory, but pointed up two deviations. The first was that with actual numerical calculations, rather than the two fronts equally sharing the temperature and moisture changes, the first or fast front contained the majority of the temperature change and the second or slow front the majority of the moisture change. Second was that the fronts, and in particular the slow front, are not usually sharp. The deviation of the amount of change attributed to each front was easily explainable by the fact that the theory was based on approximations, but the broadening of the fronts was not so easily explained. Cassie and Baxter attributed the broadening to the fact that the air flow through their experimental system did not constitute a uniform, single-velocity flow throughout the cloth. Daniels[4] examined the effect of temperature and water vapor diffusion at the front and concluded that it could account for the broadening to the extent observed by Cassie and Baxter.

McMahon and Downes[8] next pursued the work done by Cassie. They carried out experiments with a much higher air velocity than Cassie and Baxter had used. They also observed that the slow front was very broad. However, they had carefully constructed their test system to eliminate any nonuniformities of air flow and made calculations based on Daniel's theory of the effect of diffusion, which did not explain the front they observed. They therefore extended the theory.

From the considerations of the alternative explanations for the broadness of the fronts, McMahon and Downes decided that the assumption made by Cassie, that the fibres come instantaneously into equilibrium with the temperature and moisture content of the air around them, may not always be justified, particularly at higher values of air flow. They felt that the assumption of instantaneous temperature equilibrium with the surrounding air was justifiable, but that the measurements made by others in the field indicated that the assumption of instantaneous moisture equilibrium was not.

To take into account the finite rate of approach to moisture equilibrium between a fibre and its surrounding air, they assumed

$$\frac{\partial M}{\partial t} = \omega(M' - M)$$

where M = instantaneous regain

M' = equilibrium value of regain corresponding to the temperature, T, and humidity of the surrounding air

ω = rate constant.

They noted that this relationship was only an approximation but they felt that the increase in accuracy was significant. Using Cassie's moisture balance equation, they substituted their equation and obtained

$$M = M' + \frac{1}{\rho\omega} (\frac{\partial C}{\partial t} + v \frac{\partial C}{\partial x})$$

Eliminating M between the moisture balance equation

$$\frac{1}{\omega} \frac{\partial}{\partial t}(\frac{\partial C}{\partial t} + v \frac{\partial C}{\partial x}) + \frac{\partial C}{\partial t} + v \frac{\partial C}{\partial x} + \rho \frac{\partial M'}{\partial t} = 0 \tag{7}$$

The quantity M' is a function of C and the temperature T, as Cassie had assumed.

McMahon and Downes recognized that an analytical solution of Eq. (7) would present serious difficulties. They therefore attempted to find a solution similar to Cassie's. They assumed Cassie's wave solution form of fronts of constant velocity propagated through the textile mass at constant velocities. They found that solutions of this type indicated that C and T are connected by a single-valued expression. With T and C so related, M' could be expressed as a function of C alone so that

$$\frac{\partial M'}{\partial t} = \frac{\partial C}{\partial t} \frac{dM'}{dC}$$

The assumption of a wave solution gave

$$C = C(x - ut)$$

so that

$$\frac{\partial C}{\partial t} = - u \frac{\partial C}{\partial x} \quad , \text{ etc.,}$$

and u is the velocity of propagation.

Substituting these relations into Eq. (7) gave

$$\frac{\partial^2 C}{\partial x^2} + (\frac{\omega}{v - u} \frac{dM'}{dC} - \frac{\omega}{u})\frac{\partial C}{\partial x} = 0$$

By assuming the mass of fibres to be of infinite thickness with C_α, M'_α denoting the conditions at $x = \infty$ and C_β, M'_β denoting the conditions at $x = -\infty$, they further simplified Eq. (7). The amount of water picked up by the air per second in unit cross section of the mass normal to the x-axis can be expressed as

$$(v - u)(C_\alpha - C_\beta)$$

and also as

$$\rho u(M'_\alpha - M'_\beta)$$

Equating these two expressions results in

$$u = \frac{v - u}{\rho} \frac{C_\alpha - C_\beta}{M'_\alpha - M'_\beta}$$

Substituting this further simplifies Eq. (7) to

$$\frac{\partial^2 C}{\partial x^2} + \frac{\rho\omega}{v - u}\left(\frac{dM'}{dC} - \frac{M'_\alpha - M'_\beta}{C_\alpha - C_\beta}\right)\frac{\partial C}{\partial x} = 0$$

and since

$$\frac{\partial}{\partial C}\left(\frac{\partial C}{\partial x}\right) = \left(\frac{\partial^2 C}{\partial x^2}\right)\Big/\frac{\partial C}{\partial x}$$

Then

$$\frac{\partial}{\partial c}\left(\frac{\partial C}{\partial x}\right) = \frac{\rho\omega}{v - u}\left(\frac{dM'}{dC} - \frac{M'_\alpha - M'_\beta}{C_\alpha - C_\beta}\right)$$

Integrating and evaluating by the boundary conditions that

$$\frac{\partial C}{\partial x} = 0 \text{ at } C = C_\beta \text{ and } M' = M'_\beta$$

finally gives

$$\frac{\partial C}{\partial x} = \frac{\rho\omega}{v - u}\left\{\frac{M'_\alpha - M'_\beta}{C_\alpha - C_\beta}(C - C_\beta) - (M' - M'_\beta)\right\} \tag{8}$$

Equation (8) was derived from the assumption of a unique relationship between C and T. McMahon and Downes showed this relationship by taking Cassie's equations of heat and moisture balance and eliminating $\partial M/\partial t$ between them to give

$$\frac{\partial C}{\partial t} + v\,\frac{\partial C}{\partial x} + \frac{S}{q}\,\frac{\partial T}{\partial t} + \frac{\rho_a c_a v}{q}\,\frac{\partial T}{\partial x} = 0$$

Substituting their wave form assumptions of the temperature and moisture change propagations led to

$$(v - u)\frac{\partial C}{\partial x} + \frac{S}{q}\left(\frac{\rho_a c_a v}{S} - u\right)\frac{\partial T}{\partial x} = 0$$

or

$$C + \frac{S}{q} \; \frac{(\rho_a c_a v/S)-u}{v-u} \; T = \text{constant}$$

which is the single valued relationship they sought.

Since the above equation is not in a particularly convenient form, McMahon and Downes made the assumption

$$\frac{\rho_a c_a}{S} \cong 1$$

that Cassie made, but thought that it was not justifiable under usual conditions. Instead, they thought that assuming

$$u \ll v$$

which is easily satisfied, or

$$u \ll \frac{\rho_a c_a}{S} v$$

which is usually justifiable for the slow front, gave the same results of

$$C + \frac{\rho_a c_a}{q} T = \text{constant} \tag{9}$$

which was a much more convenient form.

Using the most general assumption given above, Eq. (8) gave them

$$\frac{\partial C}{\partial x} = \frac{\rho\omega}{v}\left\{\frac{M'_\alpha - M'_\beta}{C_\alpha - C_\beta}(C - C_\beta) - (M' - M'_\beta)\right\} \tag{10}$$

which can be evaluated by means of Eq. (9) and experimentally determined isotherms.

McMahon and Downes noted that the term within the brackets in Eq. (10) is the difference between the ordinate of a given C on a line drawn between (C_β, M'_β) and (C_α, M'_α) and the ordinate of the same C value on a C-M' curve through the same two points. This means that if the C-M' curve is concave upwards, a stable wave solution exists only for the desorption case. Absorption change is not propagated as a stable wave of constant velocity. Conversely, the stable wave solution for a concave downward C-M' curve exists only for a sorption change, but such a curve occurs only at very low values of regain for textile fibres. McMahon and Downes ended their evaluation by giving a method of approximately solving Eq. (10) analytically and finding the width of the front and the time for the front to pass a point from this approximate solution.

Nordon[10] developed a model of the previous systems after reassessing some of the assumptions that had been made. His model was formed with the method of finite differences in mind in order to give numerical solutions for transient states as well as steady-state. Nordon began with the same type of bed as the others. He assumed that the equilibrium between the concentration of water in the fibres and vapor in the air is a function of relative humidity only and invariant with temperature.

$$\text{Isotherm } Y = f_1\left(\frac{M_b}{d}\right)$$

where

$$\text{Relative humidity } Y = \frac{P_a}{P_o(T_a)}$$

and

$$\text{Saturation vapor pressure } P_o = Ae^{-B/TA}$$

where

M_b = concentration of water absorbed by the fibres
d = packing density of dry fibres in the bed
P_a = vapor pressure of water in the air
T_a = air temperature
A,B = constants

Further, he assumed that absorption or desorption of unit mass of water by the fibres liberates or absorbs a quantity of heat greater than the latent heat of water vapor at that temperature. This quantity is a differential heat and a function of regain only, independent of temperature

$$\lambda = f_2\left(\frac{M_b}{d}\right)$$

Nordon assumed that the rate at which the fibres change their water content was proportional to the difference between the vapor pressure of water in air at the chosen point and time and the equilibrium vapor pressure which the water absorbed by the fibres exerts at the same point. Neglecting hysteresis, he expressed the equilibrium vapor pressure of the fibres (P_b) by

$$P_b(X,t) = f_1[\frac{M_b(X,t)}{d}]\ P_o[T_b(X,t)]$$

He believed that this assumption was a plausible simplification of sorption kinetics and explained that he used it because it is formally equivalent to the situation where the mass transfer rate is controlled by a film resistance similar to heat transfer.

Nordon then wrote the governing equations for a unit section of the bed as:

Conservation of Mass

$$v\frac{\partial M_a}{\partial x}\ \frac{\partial M_b}{\partial t} = 0$$

Conservation of Heat

$$\zeta v C_a \frac{\partial T_a}{\partial x} + \lambda v \frac{\partial M_a}{\partial x} + Cd\frac{\partial T_b}{\partial t} = 0$$

Mass Transfer

$$v\frac{\partial M_a}{\partial x} = k\ (P_b - P_a)$$

Heat transfer

$$\zeta v C_a \frac{\partial T_a}{\partial x} = h(T_b - T_a)$$

where

- v = volume rate of air flow per unit gross bed area at right angles to air flow
- M_a = concentration of water vapor in the air
- ζ = density of air
- C = specific heat of the fibres
- T_b = temperature of the fibres

k = rate constant for mass transfer, and
h = heat transfer coefficient

These equations are based on the assumptions that v, C_a, C are independent of temperature;

ζ is constant for the range of temperature and pressure encountered,

the product Cd remains constant although it is strictly a function of regain,

the mass of vapor held in the inter-fibre space in the bed at any given time is negligible,

the total heat of the air-vapor mixture held in the inter-fibre space in the bed at any given time is negligible.

Nordon noted that the last two assumptions are applicable only at low temperatures when the saturation vapor pressure of water is a small fraction of the total pressure. The former assumption neglects the changing water content in the inter-fibre space compared to that of the fibres and allowed him to neglect an additional term in the conservation of mass equation. The latter assumption neglects the heat content of the inter-fibre air-water vapor mixture, as compared to the third term in the conservation of heat equation. This avoided an additional term in that equation.

Nordon made two further assumptions before he did this calculations. The first of these was that n in the heat transfer equation was infinitely large. This forced $T_b = T_a$ in order to maintain $\partial T_a/\partial x$ finite and forced the temperature front to be propagated through the bed, unchanged in shape, at a constant velocity equal to the air stream velocity. Nordon recognized that this was not strictly correct, but he was interested in the second front that carried the moisture change, which was not affected by this assumption. Similarly, the second assumption he made was that the contribution of a change in the sensible heat of the fibres in the heat balance equation could be neglected. This is represented by the term $Cd\ (\partial T_b/\partial t)$ in the equation and only introduces error when $\partial T_b/\partial t$ is large, which again involves the temperature front only. This final assumption allowed him to rewrite the heat balance equation as

$$\zeta C_a \frac{\partial T_a}{\partial x} + \lambda \frac{\partial M_a}{\partial x} = 0$$

Nordon applied the finite difference form of solution in the following manner;

Mass transfer

$$M_a(x + 1,t) = M_a(x,t) + \frac{\Delta x}{v} F(x,t)$$

where

$$F(x,t) = k[P_b(x,t) - P_a(x,t)]$$

Conservation of Mass

$$v[M_a(x + 1,t) - M_a(x,t)]\Delta t + [M_b(x,t + 1) - M_b(x,t)]\Delta x = 0$$

or

$$M_b(x,t + 1) = M_b(x,t) - \Delta t F(x,t)$$

Conservation of Heat

$$T_a(x + 1,t) = T_a(x,t) - \frac{\lambda \Delta x}{\zeta v C_b} F(x,t)$$

In order to evaluate $P_a(x + 1,t)$, he assumed that the water vapor in the air obeys the perfect gas law and that changes in air temperature are sufficiently small to permit the pressure to be evaluated at a constant temperature, i.e., the temperature of the air stream entering the bed of fibres. This produced

$$P_a(x,t) = \frac{RT}{M} M_a(x,t)$$

where T is always $T_a(o,t)$. It then followed that

$$P_a(x + 1,t) = P_a(x,t) + \frac{RT}{Mv} \Delta x F(x,t)$$

which together with the other equations and the boundary conditions

$$M_b(x,o)$$

$$T_a(o,t)$$

$$M_a(o,t)$$

allowed Nordon to make his calculations.

Nordon carried out calculations for a bed of wool and gave the numerical data he used in an appendix of Ref. 3. He developed eleven predictions from this model and stated that a study of the actual behavior of a wool bed was in progress at the time of his paper, but had not been completed.

The final method of approach that I feel should be discussed here is the one taken by Bell and Grosberg[1]. Bell and Grosberg were concerned only with the falling rate period of drying, not the total process of drying. It is in this falling rate period that the characteristics of the textile and textile fibres begin to play a part that can no longer be neglected. The way in which Bell and Grosberg handled these characteristics prompted me to include a discussion of their theories.

Bell and Grosberg noted that other researchers had shown that the water surface within a textile retreats inward in the drying of cylinders made from the material. The balance between mass and heat transfer across the dry materials then produces a constant temperature within—the pseudo-wet-bulb temperature. Nissan, Kaye, and Bell gave the following equation for this condition;

$$k(t_a - t_w) = 2.886 \times 10^{-4}(\varepsilon D_o)\lambda_w\left(\frac{P_w}{T_w} - \frac{P_a}{T_a}\right)$$

where

k = thermal conductivity coefficient,
t_a = temperature of the air stream,
t_w = temperature of the water surface (pseudo-wet-bulb temperature),
ε = porosity or fraction of the volume occupied by air,
D_o = coefficient of vapor diffusion,
λ_w = latent heat of evaporation at the temperature of the water surface,
P_w = vapor pressure at the water surface,
P_a = partial pressure in the air stream,
T_w = absolute temperature at the water surface, and
T_a = absolute temperature of the air stream.

Experimental results indicated that this equation was not accurate for cloth. By reevaluating the diffusion of the water vapor through the dry porous material, Bell and Grosberg came up with the equation

$$k(t_a - t_w) = 2.886 \times 10^{-4}\ \alpha D_o\lambda_w\left(\frac{P_w}{T_w} - \frac{P_a}{T_a}\right)$$

where

k = permeability coefficient, and
α = function of ε

and showed graphs for attaining α. They did not take into consideration the boundary layer at the surface of the cloth because, they stated, for layers of cloth over 0.03 in. in thickness, the error is of the order of 5% or less in t_w. This equation, therefore, gave the temperature of the core which they gave data to support.

Bell and Grosberg then stated their theory and reasoning of how the water within the core was dryed up. Others usually attribute the falling rate to a breakdown in the capillary flow network that feeds the surface and therefore forces the water surface to recede inward. Bell and Grosberg pointed out that experimental evidence indicates that the water content of the water core continues to decrease after the water surface begins to retreat inward, which a capillary breakdown would not account for. They postulated that the water movement was viscous controlled after the critical moisture content was reached to begin the falling rate and they showed calculations and experimental evidence that the moisture content within the core agreed closely with this theory. The equation they deduced was

$$W - G \propto 1 - \left(\frac{r}{r_1}\right)^{2^{1/4}}$$

where

W = moisture content at any radius r of a cylinder of textile,
G = fractional moisture content of fixed water in the structure, and
r_1 = radius of the water surface.

This argument did not give satisfactory agreement with the rate of movement of the water surface. However, Bell and Grosberg reasoned that this could be because the structure contains both fixed and movable water and that the fixed water would tend to block and cut off the movement of the movable water and therefore slow the retreat of the water surface. They stated that this interaction would be complex to analyze so they assumed

$$F = (\bar{w}-G)/\bar{w}$$

where

$\bar{w}$ = average fractional moisture content, on a dry basis, of the core at any point, and
F = proportion of the total water demanded by the conditions of drying at the water surface which is supplied by water flow to that surface,

which they showed agreed well with data. It also pointed up that F is not unity at the critical moisture content.

Using this equation for F, Bell and Grosberg were able to derive an equation for the position of the core in a textile bobbin;

$$G\left(\frac{1}{W_1} - \frac{1}{W_c}\right) = \frac{1}{\pi\rho L}\left[\frac{1}{(r_1^2 - r_o^2)} - \frac{1}{(R^2 - r_o^2)}\right]$$

where

W_1 = total weight of water in the bobbin,
W_c = critical moisture content,
ρ = density of bobbin material,
L = length of the bobbin,
r_o = radius of insulating material on which the textile is wound, and
R = radius of the bobbin.

In this equation W_1 is the total water content of the core minus the absorbed water and G can be obtained from the intercept of AW_1-W plot since

$$W_1 = (\text{constant})(\bar{w}-G)$$

They also showed data to support these two equations.

Finally, Bell and Grosberg summed up their article by deriving a formula to predict the drying rate curve. This equation is

$$\lambda_w \frac{dW}{d\theta} = \frac{k(t_a - t_w)\pi L(R + r_1)}{R-r_1 + \frac{k}{n}\frac{R + r_1}{zr}}$$

Since t_w was obtained at the beginning of their article and r_1 just previously, n can be calculated from the constant rate period of drying. From this equation they plotted a drying rate curve which fit data well, except when thin sections caused the drying process to proceed to the point of removing the absorbed water.

IV. CONCLUSION

The previous sections of this report have been concerned with the work that has been done in relation to combined heat and mass transfer, as it can be applied to the drying of textile materials. This section is intended as a guide to what I feel should be further research in this area.

The methods of Nordon and Bell and Grosberg are, in my estimation, the best choices for further development. They proved that their theories are plausible and can give results. Each was concerned with a different type of drying process, but the two combined cover the process of drying in a practical manner from initiation to the falling rate period. Both combine a practical approach to the problem with a concern for the phywical and mathematical processes that govern it. Finally, both have documented the decisions they made and the manner in which they developed their conclusions and both exhibit them in working formulas.

For further work I would pursue a combined method of calculation using Nordon's approach to simulate drying from initiation until the critical moisture content is reached, and Bell and Grosberg's approach from critical moisture until the final desired moisture content of the cloth is reached. This would require a coordinate transformation of one of the two theories, but by using the computer as a means of doing the calculations, this should not impose any extreme complications. Once this is to a stage of completion and the necessary parameters for numerical calculations can all be recognized, an experimental procedure would have to be set up to verify the relationships derived and the accuracy of the computational methods used. I think that the most promising type of experimental technique would be one similar to that used by Bell and Grosberg.

I think that the immediate goal of further development should be a working, practical equation. For this reason, I suggest that the experimental evaluation be made in an apparatus as similar as possible to a commercial home laundry dryer. The principal error within the suggested procedure seems to lie in the critical moisture content determination and in the determination of equivalent physical parameters of the system to be used in the equations. If the preliminary work is done from a system as physically similar to a dryer as possible, the evaluation of these parameters to give a working equation can be made empirically and the equations used while an exact analytical solution is being derived. Immediate practical application with a sound basis of theory is the factor that influenced which approaches I concentrated on in compiling this report. There are certain to be pitfalls and problems in refining the theories given here, but I believe that none will be insurmountable.

V. REFERENCES

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APPENDICES

These Appendices point to further research into the practical application of heat and mass transfer theory to the drying of textiles. The information contained within each is as follows:

APPENDIX A

This is a collection of those references that I was unable to obtain, but whose titles suggest that they could be of value either for their technical knowledge of experimental data. For ease of use they are arranged in a bibliographical form.

APPENDIX B

This is a collection of those references which I was able to investigate, but did not directly make use of while compiling this report. I do feel that they could be of value for a deeper evaluation of the subject. These are also presented in a bibliographical form.

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DESIGN AND DEVELOPMENT OF A

FLUIDIC-PNEUMATIC CONTROL SYSTEM FOR AN AUTOMATIC WASHER

Jon D. Tromblee

I. INTRODUCTION

Since 1959, when the modern growth of fluidic technology began, fluidic systems have been investigated for possible applications in a variety of fields; including industrial production control, medicine, and national defense. The systems that have been successfully employed have at least one of the following characteristics; the ability to sense specific environmental conditions or the ability to produce either digital or proportional control signals. Thus, an interesting question for home appliance manufacturers arises. Can fluidics be successfully applied to the automatic controls found in consumer-oriented products in which sensing and control are vital operations, e.g., automatic washing machines, clothes dryers, and dishwashers? The answer must first include a consideration of feasibility; then cost, reliability, producibility, and consumer appeal.

The Department of Mechanical Engineering at The University of Michigan started a project in September 1968 to design and develop a fluidic-pneumatic control system for an automatic washing machine. The purpose of this paper is to report on that project, to draw certain conclusions from the results obtained to date, and to try to determine whether a washing machine can be controlled automatically during a particular washing cycle by an integrated system of fluidic elements acting through appropriate pneumatic interfaces. An affirmative answer may not fully establish feasibility, because of the physical limitations of the experimental apparatus, but it will at least point to the desirability of further research in this area.

The report itself is technically oriented; it describes in detail an experimental project in product development. Its approach is largely qualitative rather than quantitative. No attempt has been made to describe the system behavior mathematically, although several data curves are presented that characterize component operation.

I wish to express my appreciation to Professor Robert B. Keller of the Department of Mechanical Engineering at The University of Michigan for valuable assistance in project guidance and practical advice on design considerations. I am also indebted to Mr. Sheldon Roll and his staff at The University of Michigan for their careful work and cooperation in manufacturing unusual hardware for the experimental apparatus.

II. QUALITATIVE DISCUSSION OF FLUIDIC DEVICES EMPLOYED

Three types of fluidic devices have been considered for use in the proposed control system: bistable fluid amplifiers, monostable fluid amplifiers, and proximity sensors. A brief discussion of their operation will be given before considering specific applications of these elements in the integrated control system.

A. BISTABLE FLUID AMPLIFIER

A fluid amplifier, in general, is a device without solid moving parts that is used to control a primary fluid flow by secondary fluid control signals. The term "amplifier" is derived from the fact that the supply pressures and flow rates of the secondary fluid control signals are usually significantly less than those associated with the primary flow. The term "bistable" indicates that the fluid amplifier has two stable states. This means that the amplifier, which has two output channels, is in a stable state whenever all the flow leaves the amplifier by one of its output channels. The exit flow channel containing the primary flow is determined by the application of a control signal at one of two control ports which lie on opposite sides of the primary flow in the plane containing the exit flow channels.

Refer to the schematic representation of the Bistable Fluid Amplifier given in Appendix A and consider the following example. Initially, primary fluid flow enters the amplifier at S and leaves arbitrarily at O1 with no signals applied at either C1 or C2. Later a secondary fluid signal is applied at C2. If the signal has the proper pressure and flow rate, all the primary flow will switch to the other channel exiting at O2 and will remain there even after the fluid control signal at C2 ceases. Conversely, if a fluid signal with proper pressure and flow rate is now applied at C1, the primary flow will switch from O2 to O1 and remain at O1 even after the control signal at C1 ceases. Any recurrence of the control signal at C1 while the flow is out O1 will have no appreciable effect on the output flow. The same statement holds when the output flow is at O2 and the signal recurs at C2. When equal control signals are simultaneously applied at both C1 and C2, ideally the output flow should not switch. However, in practice the output is considered indeterminate under such conditions. When the primary flow is initially supplied at S, the output is also indeterminate unless a known control signal is applied at either C1 or C2.

The stability of the fluid flow in either of the two stable states is the result of a phenomenon known as the Coanda effect. Because of the pressure differences existing across the flow, the fluid stream attaches itself to the channel wall leading to one of the two exits. The flow remains there until its equilibrium is disturbed by the presence of a secondary fluid signal at the appropriate control port. This control port inserts the secondary fluid flow so that the pressure increases next to the wall to which the primary flow is attached. This forces the fluid stream to the opposite channel wall where it reattaches itself and seeks the other amplifier exit. In order to accomplish this, the secondary fluid flow must have a supply pressure large enough to sustain a flow rate through the control port to overcome entrainment of the secondary flow by the primary flow and increase the pressure between the stream and the wall.

The bistable amplifier described in Appendix A is the same model used in the control system under consideration. The curves accompanying the description show the operating characteristics of the amplifier as determined by the manufacturer. The information provided by the curves proves useful when more than one amplifier must be arranged interdependently. This situation occurs when two amplifiers are cascaded or when one amplifier output supplies control signals for as many as four other amplifiers.

An interesting feature of this model, which operates with gaseous fluids, is that the output channels are provided with atmospheric vents. The vents prevent flow into the wrong output channel even though the correct output channel is maintained in a zero flow status. Fluid flow from one or both of the output channels can be effectively blocked without affecting the ability of the amplifier to switch normally. This is important in instances where maximum pressure recovery and output channel isolation are vital.

Another important feature of this amplifier is that two opposing control signals can be stacked so that the amplifier obeys the stronger control signal applied at any one time. By using this technique, a bistable amplifier can be controlled normally or it may be converted to monostable operation. This is done by utilizing the weaker control signal as a continuous bias which automatically resets the amplifier whenever the stronger control signal is absent.

B. MONOSTABLE FLUID AMPLIFIER

A monostable fluid amplifier, like other fluid amplifiers, controls a primary fluid flow by secondary fluid control signals without the use of solid moving parts. The term "monostable" conveys the idea that this type of amplifier has but one stable state. However, monostable amplifiers have two output channels and the capability of switching the primary flow from one output channel to the other just as bistable amplifiers do. The difference is that a monostable fluid amplifier will maintain the entire primary flow in its "excited" output channel only as long as a control signal is applied at one of its control ports. When the control signal is removed, the entire primary flow switches to its "stable" or "relaxed" output channel. The monostable amplifier may have more than one control port, but all of these are positioned on the same side of the primary flow. When a control signal is applied at any of the control ports, the primary flow is diverted to the "excited" output channel.

Refer to the schematic representation Monostable Fluid Amplifier given in Appendix A and consider the following example. A primary fluid flow enters the amplifier at S and exits at O1 when no control signal is present at either of the two control ports C1 and C2. When a control signal of sufficient pressure and flow rate is applied at either one or both of the control ports, C1 and C2, the primary flow switches completely to the output channel exiting at O2. When both control signals are absent, the primary flow switches back to the output channel exiting at O1. The flow at O1 represents the "stable" output and the flow at O2 represents the "excited" output. There is no appreciable change in the output at O2 if more than one control signal is supplied to the amplifier.

The explanation of the monostable behavior is similar to that of the bistable behavior. The only difference is that monostable devices are designed so that once the fluid stream has been detached from the "stable" output channel wall, it is prevented from re-attaching to the "excited" output channel wall. When the control flow which caused the detachment is removed, the primary flow again establishes itself in the "stable" output channel. The control flow necessary to switch a monostable amplifier must meet requirements similar to those discussed for bistable switching.

The monostable amplifier described in Appendix A is the same model used in the experimental control system under investigation. The curves accompanying the description are provided by the manufacturer. The close similarity between the operating curves for bistable and monostable amplifiers allows integration of both types of amplifiers in multiply connected fluidic circuits.

This particular monostable amplifier is equipped with atmospheric vents in each output channel. Their function is identical to that described above for the bistable model.

An important modification was made to one monostable amplifier for this project, which effectively added a third control port. The added control port, however, does not regulate switching by the normal means of introducing a positive secondary control flow. In the modified version, a complete switch in output is obtained by occluding the new control port. The modified amplifier behavior is completely analogous to regular monostable amplifier behavior. The occlusion control signal has the same priority as the standard control signals and follows the same pattern of behavior.

The two control ports of the standard model monostable amplifier provide an effective means of isolating signals from separate sources, even though they are applied at essentially the same control point. There is no appreciable crossover between the control ports so that when a signal is applied to one, the other experiences no important change in pressure.

C. PROXIMITY SENSOR

A proximity sensor is any device capable of detecting and reacting to the distance between the device and some solid or liquid surface. The reaction is converted to a usable output signal capable of controlling other performing elements in a system. In a system where many of the performing elements are fluid amplifiers, the appropriate output signal for a proximity sensor would be a pressure signal. One device that satisfies the definition of proximity sensor and whose output is a pressure signal is the cone jet sensor. Specific information on the model used in the control system being studied is located in Appendix A.

Investigation of the cone jet sensor performance revealed that the output signal from this device is capable of reliably controlling the fluid amplifiers only if the distance between the sensor head and some solid surface is held to values less than 0.010 in. This statement holds when both sensor and amplifier are operated at the same supply pressure. This seems to contradict the data obtained from the sensor operating curve, which indicates that even at distances greater than 0.010 in., sufficient pressure should be supplied by the sensor to control this type of fluid amplifier operating at the same supply pressure. The contradiction is resolved by considering the following facts: (1) the sensor pressure output was recorded for zero flow rate; (2) the fluid amplifiers require a control signal that must meet a minimum flow rate supplied at a switching pressure that is dictated by the amplifiers' characteristics for a particular supply pressure. The sensor output must be increased until it can satisfy the requirements for switching posed by the amplifiers. The sensor output in this case is increased sufficiently by decreasing the distance from sensor head to surface to less than 0.010 in.

III. DESIGN CONSIDERATIONS GIVEN TO EXPERIMENTAL APPARATUS

The subsystems used in the experimental control system can be classified as follows: (1) fluid-mechanical timer, (2) logic circuitry, (3) interface devices. Before each of these are discussed in detail, the restraints imposed on the system design will be discussed. Following considerations of the subsystems, the layout of the experimental system will be discussed.

A. DESIGN RESTRAINTS

The machine chosen for the project was a 1967, three-speed, Whirlpool Imperial Mark XII. The washing cycle selected for duplication by the new control system included both the Super and Normal cycles. The ability to preselect different standard motor speeds for agitation and spin, and different water temperatures for wash and rinse was desired. The selections were to be made by means comparable to those used on standard machines. Infinite water level adjustability and resetability were also to be included in the design.

The system was to be operated by pressurized air supplied from a shop source. The decision to use an external air supply was made to expedite testing procedures. No attempt was made to make the air supply self-contained. Self-contained here means that the air supply and control system could be mounted somewhere within the machine. However, included in the testing procedure was the effort to operate the resultant control system on as low a supply pressure as practicable. In this way the system would approach as closely as possible one which could be completely self-contained, except for an electrical power supply. Latest tests showed that the system could be satisfactorily operated at 3.0 psig, consuming approximately 5.0 cfm at that pressure. Tests also indicated that the system could be operated satisfactorily, except for the spin actuator, at pressure down to 1.5 psig. Below this pressure the agitation does not operate.

Air was chosen as the operating medium for several reasons. First, the supply of air is abundant, even though the means of pressurizing air are limited. Second, low-pressure air can be transported and utilized within the system without major concern for damage by leaking or sudden rupture. Third, a variety of existing fluidic devices can be operated with air. Finally, the effects of gravity and machine motion are negligible when air is used.

The system was designed to have no automatic control over the air supply. Air was to be supplied through a manually controlled pressure regulator. By using the pressure regulator, prescribed variations in start-up, running, and shut-down supply pressures could be achieved. Except for the pneumatic interfaces, the major portion of the control system was to be bread-boarded, so that modifications could be made as quickly and easily as possible.

B. FLUID-MECHANICAL TIMER

Two possible designs were considered for developing a timer capable of being integrated with fluidic and pneumatic devices. One method would utilize existing timer hardware suitably modified to regulate pressure rather than electrical signals. The alternative approach would rely entirely on fluid amplification to obtain properly timed pressure pulses throughout the cycle.

Despite the decided advantage of no moving parts in a totally fluidic timer, the decision was made to modify existing hardware. The choice was based on the estimation that developing the fluidic timer would require about the same amount of time as the remainder of the control system. It was estimated that a fluid-mechanical timer could be designed and built in significantly less time.

The timer finally used, shown in Figure 1B,* consists of the mechanism of the standard timer with the electrical circuit breakers and associated cams removed. A round aluminum shaft, 1/2 in. in diameter, fits over and is securely

*The letter following the number indicates the appendix in which the figure appears.

attached to the original cam shaft. The new shaft extends out of the mechanism through an enlarged hole in the front plate, which acts as a bearing. The length of the shaft external to the mechanism is approximately 6 in. Along this length, six Plexiglas cams are mounted. Each cam is 1/4 in. thick and 4 in. in diameter.

The purpose of the cams is to control the output of six cone jet sensors mounted around the circumference of the cams; each cam controls one sensor. The sensors are mounted to a plastic cylinder so that the air streams are directed radially against the cylindrical surface of the cams. The faces of the sensor heads are held at less than 0.010 in. from the cam surface for reasons explained in the discussion of proximity sensors. The plastic cylinder supporting the sensors is securely attached to a base plate rigidly monnted on the front plate of the timer mechanism. In some places the sensor heads rub the cylindrical surface of the cams. To facilitate proper timer motion under these torque requirements, the cam surfaces were lubricated with petroleum jelly. Also, a stiffer spring was inserted in the timer mechanism.

In operation the sensors remain stationary and the cams move as dictated by the timer mechanism powered by an electric motor. Each 2 min the cams rotate 5.625°, the motion being accomplished in a matter of seconds so that the cams remain stationary for the major portion of each 2-min interval. Because of this motion, the sensors can be operated like monostable devices with the cam shapes determining when "control signals" are given to each sensor. A vacant position on any cam yields zero pressure output from the corresponding sensor; an occupied position results in a positive pressure signal capable of switching some fluid amplifier. Sketches of the cams appear in Figure 4C.

C. LOGIC CIRCUITRY

The fluidic logic circuit is the portion of the control system that receives timed control signals from the timer and combines these with certain information collected from the machine itself. Once the combination is made, the circuit arranges the output of its fluidic elements to produce the appropriate output signals for proper machine control. A specific description of the logic circuit operation is presented in Section IV. A photograph of the circuit appears in Figure 2B while a schematic representation appears in Figure 1C. The schematic representation is the source for the reference numbers given to various amplifiers in the following discussion.

1. Water Level Detection

Only one item of information from the machine is required by the logic circuit; water level in the tub. This is determined by amplifier 12, a modified monostable amplifier with an occlusion control port. The control port is connected by a flexible air line to a tube which can be raised or lowered inside a standpipe adjacent to the tub. The standpipe is connected to the tub so that changes in the tub water level lead to changes in the standpipe water level. When no significant change occurs in the tub level, the two levels become equal. When the water level is high enough in the standpipe to cover the end of the adjustable tube, amplifier 12 is switched to its "excited" output. This output can be used effectively to inform other elements in the circuit that a particular water level in the tub has been reached.

A needle valve is connected to the air line leading to the occlusion control port. The valve allows flow into the line from the atmosphere after the tube end of the air line has been occluded by the water's surface. When the tube is occluded, the pressure inside begins to drop, causing the amplifier to switch. However, as the pressure drops, the water begins to rise in the tube and may enter the air line and amplifier unless the pressure is controlled. The purpose of the needle valve is to maintain the air line pressure after occlusion so that the amplifier can be switched without an excessive rise of water within the adjustable tube.

A photograph of the water level detection apparatus appears in Figure 3B.

2. Water Temperature Selection and Flow Control

Any of nine temperature combinations for wash and rinse may be selected by properly setting the four temperature-selection valves connected to the outputs of amplifier 1. In the experimental setup, the valves only simulate what could be accomplished by perhaps presetting a rotary valve; but the resulting performance in either case is identical. The output of amplifier 1, fixed by the timer position, determines whether the machine is in a wash or rinse phase of the cycle. The two valves in line with each output then determine which downstream amplifiers receive control signals.

Flow control is actually influenced by the operation of many amplifiers. However, the final commands governing water flow come from the outputs of amplifiers 6 and 7, which regulate hot and cold water, respectively. Amplifiers 6 and 7 are separated from the other elements in the logic circuit by amplifiers 2, 3, 4, and 5, whose outputs depend on the temperature-selection valves and the output of amplifiers 1 and 8. Any instructions concerning water flow must pass through this temperature-selection bridge before the instructions can be carried out.

Amplifiers 2, 3, 4, and 5 are bistable amplifiers whose control signals are stacked to provide an override capability for the "stop flow" signal coming from the "excited" output of amplifier 8. The "stable" output of amplifier 8, which exits to the atmosphere, represents the "start flow" signal. It is transmitted across the bridge to the proper valve or valves by the appropriate switching of amplifiers 2, 3, 4, or 5.

3. Transmission Control

Transmission control in this instance is defined as regulation of the functions carried out by the machine transmission and motor. Amplifiers 15, 16, and 17 are directly responsible for transmission control according to this definition. Amplifier 15 places the machine in agitation at a preselected speed only if both the proper tub water level and timer signal exist simultaneously. Amplifier 16, which controls pumpout, is governed by signals from the timer only. During pumpout, the machine motor always runs at high speed. Two sensors control amplifier 16 as a result of early attempts to design a system that would put the machine in total pumpout at certain times and in partial pumpout near the end of the Super wash. However, the control system does not yet have the partial pumpout feature, so both sensors are used together for total pumpout at the proper times. Amplifier 17 places the machine in spin at a preselected speed on command from the timer only.

4. Reset Subcircuit

Because an arbitrary status for some of the bistable amplifiers cannot be tolerated for proper machine control, a means of fixing their outputs is necessary at certain times during the cycle. Fixing these outputs is accomplished by automatically generating a reset pulse within the logic circuit whenever the logic circuit senses certain conditions. The reset pulse is guided to the control ports of the selected bistable amplifiers, whose outputs are then determined.

The pulse, whose duration is approximately 2 sec, is produced immediately after any switch of amplifier 12 to its "stable" output. A portion of the signal from amplifier 12 causes amplifier 9 to switch immediately, supplying power fluid to amplifier 10. Amplifier 10 produces a relatively weak signal in its "stable" channel until the delayed portion of the signal from amplifier 12 causes a switch in output. The switching of amplifier 10 is delayed by two needle valve resistors and a volume capacitor. The weak pulse from amplifier 10 is then amplifier in 11 before being distributed to the logic circuit. The specific times for pulse generation are noted in the discussion on sequence of events in Section IV.

The reset pulse was originally controlled by cams A and B in the timer. This accounts for their rather complicated shapes. The pulsing method, however, was modified to simplify the logic circuit and allow for simplification in the timer. Despite the modification of the circuit, cams A and B still operate satisfactorily. The proposed simplification, which at the time of this report has not yet been made, would eliminate cam A, cam B, and the bistable amplifier in the #1 position.

5. Spin-Spray Subcircuit

To facilitate periodic water flow during the spin portions of the machine cycle, a fluidic multivibrator is included in the logic circuit. It functions continuously during the cycle, but its output reaches the flow-control switches only during spin. The water flow during spin occurs at the preselected rinse temperature. The entire spin-spray subcircuit consists of amplifiers 18, 19, 20, and 21. The control signals for amplifier 21 are stacked to provide monostable operation during spin and bistable operation at all other times.

D. INTERFACE DEVICES

Interface devices are the components in the control system which convert the pressure signals produced by the logic circuit into mechanical and electrical signals readily usable by the machine. Three types of interface devices are used: pressure switches, linear actuators, and air-piloted water valves.

1. Pressure Switches

Four pressure switches are used to convert air pressure inputs to electrical outputs. Their arrangement in relation to other electrical circuitry is shown schematically in Figure 2C. Three of these pressure switches are used to operate the main motor in agitation, pumpout, and spin; the fourth is used to oper- the timer motor. Figure 4B is a photograph of all four devices mounted on the experimental apparatus. The photograph shows that the pressure switches are identical to those used in water level control on standard machines.

2. Linear Actuators

The two electrical solenoids which control the position of the transmission cam bars on a standard machine have been replaced by rolling diaphragm linear actuators on the experimental apparatus. The agitation and pumpout actuator is shown in Figure 5B. Two views of the spin actuator are presented in Figures 6B and 7B. Figure 8B shows the position of both actuators in relation to the transmission and main motor.

Each actuator, model S-4-F-BP-UM produced by Bellofram Corporation, was initially equipped with an internal coil spring to facilitate actuator return. In order to reduce the operating pressure, this spring was removed and a gravity return was employed for both actuators. Attached to each actuator shaft is a lifting arm over which the spin or agitation wig-wag activating pin slides. The shape of each lifting arm, visible in Figures 5B and 7B, is adapted to the orientation of the cam bar relative to the actuator's mounting. When the actuator is pressurized, the actuator shaft moves up, forcing the activating pin into the upper slot on the cam bar. The cam bar then moves just as in standard operation. When pressure is released from the actuator, the activating pin is allowed to drop into the lower slot, and the cam bar moves to its original position.

3. Air-Piloted Water Valves

The pressure signals from the logic circuit controlling water flow are transmitted to two diaphragm operated water valves, one each for hot and cold water. Shown in Figure 9B, the valves are connected directly to the water supply. The water hoses connected to the valves lead to the tub. Each valve, model F210-C3 produced by Asco, may be operated with air pressures ranging between 0.5 and 5.0 psig and water pressures between 0 and 125 psig. During system tests, air was supplied at 1.0 psig and water pressure at zero flow was approximately 80 psig.

E. EXPERIMENTAL LAYOUT OF TOTAL SYSTEM

Some physical features of the total system have already been described in the discussion of the various subsystems and design restraints. The following discussion will place these in proper physical perspective, while giving a brief overall description of the interconnected experimental layout. Figures 10B and 11B show the total apparatus ready for testing. These photographs will give considerable aid in understanding the discussion.

The entire washer mechanism usually contained within the machine cabinet is mounted in an open wood frame by means of the standard suspension system. The base plate is elevated nearly 2 ft above its normal height in production models. The open mounting and elevated base plate allow easy access to the motor, transmission, and tub.

The drain pipe, located to the right of the test stand, has been lengthened to prevent accidental siphoning of water from the machine. The air-piloted water valves are connected to the water supply by the drain pipe.

The logic circuit and timer, mounted on pegboard, are located in front of the test stand. The elements of the circuit are interconnected by 1/8-in. I.D. Tygon tubing and supplied from a central manifold. The manifold is connected to the pressure regulator located to the right of the test stand. A pressure gage, also connected to the manifold, provides the manifold air pressure. The gage reading is only an estimate of the supply pressure for each element because of the length and arrangement of some of the connecting tubing. A manometer is available to indicate the flow of a particular amplifier or sensor output. Another means of determining output and control flows is to use a piece of tubing approximately 2-1/2 ft long as a listening device, since the air flow within the amplifiers produces distinctly audible sounds.

The water level detection apparatus is mounted on the upper left side near the back of the test stand. Tygon tubing is also used here to supply air and transmit pressure signals. A metal rod provides support for the adjustable tube extending into the standpipe.

The pressure switches controlling the electrical circuit are mounted on the front of the test stand on the upper left corner. Electrical power, supplied from an outlet to the right of the test stand, is distributed to the pressure switches, main motor, and timer motor by a wiring harness. The linear actuators, mounted on the transmission case, and the pressure switches and water valves are all connected to the logic circuit by Tygon tubing.

IV. INVESTIGATION OF INTEGRATED SYSTEM BEHAVIOR

A qualitative description of the activities of the individual elements in the system, as they progress through the washing cycle, is presented here to calrify more specifically how machine functions are controlled. The activities of individual elements are explained in the section on fluidic devices, while the role of individual elements in establishing machine control is explained in the sections on the timer, circuitry, and interfaces.

Following this description of activities, the deviations in machine behavior under experimental control from machine behavior under standard control will be discussed. Knowledge of the activities of the various individual elements provides a means of understanding these deviations and the modifications needed to remove them.

A. SEQUENCE OF EVENTS IN CYCLE

The timer is assumed to be set initially for the beginning of the Super part of the wash cycle. Relevant comments about startup in other timer positions will be made throughout the discussion. For simplicity of description, warm water is assumed for both wash and rinse water temperatures. Numbers refer to amplifiers and letters to timer signals, as shown in Figure 1C. The term "vented mode" describes an output flowing directly to the atmosphere. Amplifier outputs and timer signals remain as last stated until a specific change is noted.

Air is supplied simultaneously by the manually operated regulator through the manifold to all the fluidic devices in the timer and logic circuit. Initially A is the only signal produced by the timer. The output of #1 is automatically determined by A, which places the machine in a wash phase.

Since there is no water in the tub initially, #12 remains in its stable mode and supplies control flow to #9 and #10. The offset switching of #9 and #10 causes a pulse from #11. This pulse resets #2, #3, #4, #5, #13, and #21. The reset pulse from #11 is supplied to #2, #3, #4, and #5 through #8 and to #21 through #20. After the reset pulse passes, #3 and #5 remain in their vented modes while #2 and #4 are switched to their fill modes by #1. Both piloted water valves open when #2 and #4 switch #6 and #7, respectively. The tub begins to fill.

When the water reaches the desired level in the standpipe adjacent to the tub, the opening of the dip tube control line to #12 is covered. Because of the capacitive effect of the tube, several seconds pass before #12 switches. This allows the water level to rise above the end of the tube so that the opening will not be accidentally uncovered during agitation. If the cycle is stopped and then restarted with the proper water level already reached at this timer position, #12 will remain in its stable mode for several seconds, causing a reset pulse and a start-fill signal. However, as soon as the dip tube air pressure is reduced sufficiently, #12 switches and the water flow stops.

When #12 switches, control flow is supplied to #13 and #21. The output of #21, originally in a vented mode, is now switched to #8. The output of #8 overrides the output of #1 and switches #2 and #4 to their vented modes. This causes #6 and #7 to relax and the water valves close. Tub fill stops. Simultaneously #13 switches from its vented mode to supply power to #14. Initially C is not present so #15 is switched to the agitate mode. The timer pressure switch operates as soon as #12 switches. The agitation pressure switch and actuator operate as soon as #15 switches to the agitate mode. Agitation begins and continues for 6 min.

At any time during the cycle, the water level may be reset by simply moving the dip tube so that its opening is above the existing water level in the standpipe. This feature also allows the dip tube to act as high- and low-level water indicator for the machine.

After 6 min, the timer moves into its fourth station and produces signal C. At this point A and C are the only timer signals being transmitted to the circuit by the timer. C switches #14 to a vented mode causing #15 to relax. The timer continues to operate but the machine is now removed from agitation. This is the soak portion of the cycle; the machine is idle except for the running timer motor. If the cycle were to start at this timer position, the tub would fill and then begin the 2-min soak. If the machine were already full of water, it would attempt to fill as before, shut off, and then begin the 2-min soak.

When the timer stations itself again, D is introduced to #16. The switched output of #16 operates the pumpout pressure switch and supplies control flow to #12. The tub begins to empty. To prevent a reset pulse and resulting fill signal when the dip tube control line is opened, #12 remains in its "excited" mode. Pumpout continues for 2 min. Startup in this timer position does not result in a reset pulse since all the questionable outputs are determined by the switched output of #12. Regardless of water level, the machine is placed in pumpout.

As the timer again stations itself, C and D cease. Pumpout stops as #16 relaxes. This allows #12 to relax and results, as before, in a pulse output from #11. The pulse is distributed to the circuit exactly as before. The tub fills to the proper level and the machine agitates as before. Exactly the same sequence of events is followed by the individual elements. Agitation continues for 14 min.

As the timer advances into the next station, C and F are introduced simultaneously to #14 and #16, respectively. This stops agitation and begins pumpout as described above. The sequence is exactly the same, except C and F occur simultaneously. F in this case replaces D. Both F and D are used for reasons discussed in the section describing the timer. Pumpout continues here for 2 min.

The next timer advance produces E which switches #17 to the spin mode. At the same time F ceases. When #17 switches, #12 is maintained in a swtiched mode, as in the case of pumpout, to prevent a reset pulse. The spin pressure switch and actuator are also operated. By this time B has replaced A in controlling #1, thus placing the system in the rinse phase. The machine begins its spin operation. Spin continues for 4 min.

If the cycle begins in this timer position, no reset pulse is produced and the machine is placed directly in spin. Unfortunately, spin can occur regardless of tub water level. This will be discussed in the section on deviations.

When #17 switches, the control signal to #19 is removed and the periodic pulsed output from #18 is transmitted through #19 to #20. Here the periodic pulsing is amplified before going on to #21. The output of #20 overrides the signal from #12 so that the output of #21 becomes periodic like that of #18. The result is that #8 and #1 alternately switch #3 and #5 back and forth, so that the water valves open and close with a frequency established by #18. This gives the rinse temperature spin-spary required during spin.

After 4 min, E and C cease. Now B is the only signal being produced by the timer. The absence of E allows #17 and #12 to relax and results in a reset pulse. The machine is removed from spin. Fill and agitation occur as before, except with rinse water temperature. Rinse agitation lasts for 2 min. The sequence then goes into pumpout and spin exactly as before, except that final spin lasts for 6 min.

The comments on startup for various timer positions during the wash phase apply also to comparable timer positions during the rinse phase.

B. DEVIATIONS FROM STANDARD OPERATION

In operating the entire system through approximately 15 complete trials, the following deviations were noted.

The cycle does not include a partial tub pumpout following the soak period in the Super part of the cycle. The pumpout empties the tub completely before going into the Normal part of the cycle.

When the cycle progresses from agitation to pumpout, there is a period of several seconds when both the agitation and pumpout pressure switches are closed. If agitation speed is chosen as anything other than high speed, two different motor windings are energized. This produces an audible hum until the agitation pressure switch drops out, thus relegating control to the pumpout pressure switch. The effect is caused by the presence of the agitation actuator in parallel with the agitation pressure switch. When #15 relaxes, the actuator does not allow the agitation pressure switch to reset immediately. However, #16 swtiches at the same time #15 relaxes and operates the pumpout pressure switch before the agitation line pressure has dropped sufficiently to open the agitation pressure switch.

For simplicity in the apparatus, a dispensing subsystem was not included in the experimental layout.

Spin-spray frequency is higher than in standard operation, but total flow duration is approximately the same.

Water level adjustability on the experimental apparatus is comparable to that on the standard machine. However, resetability is somewhat limited by the fact that the water in the standpipe rises slightly above the dip tube opening. The reset level must be above the water level in the standpipe. This fact reduces the range for reset selection.

The machine is not prevented from spinning when excessive amountsof water are present in the tub. This fact makes it poor practice to advance the timer into spin manually from other timer positions or to start the cycle in spin when the tub contains water.

Proposed modifications to correct all of these deviations will be investigated in May and June of 1969.

V. CONCLUSIONS

This report has attempted to answer the following question. Can a washing machine be controlled automatically during a particular washing cycle by an integrated system of fluidic elements acting through appropriate pneumatic interfaces? After considering the results presented in the report, the answer must be in the affirmative, with certain qualifications because of the physical limitations of the experimental apparatus by which the answer was obtained.

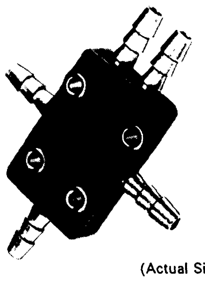
Control of the washing machine used in the experiment was not fully automatic. Air was supplied manually to the timer and logic circuit. It would have been a relatively simple matter to control the external shop air supply by electrically operated valves. However, this solution would not satisfy the practical requirement of being self-contained for home use. Although tests indicate that the supply pressure and flow rate can be significantly reduced after some modification, further investigation is needed to ascertain positively the possibility of operating the system from a machine-mounted pressurized air source.

Machine operation under the experimental control system did not exactly duplicate machine operation under the standard control system. The deviations were discussed in an earlier section. Further investigation is required to guarantee applicability of the modifications proposed to correct these deviations.

APPENDIX A

CHARACTERISTIC DATA ON FLUIDIC DEVICES EMPLOYED

FILE CATALOGUE: **Fluidic Components**
SECTION: **Application Data**
TYPE: **1000BO1**


(Actual Size)

BISTABLE

FLUID AMPLIFIER

DESCRIPTION

The Aviation Electric Type 1000BO1 Bistable Fluid Amplifier is a two input wall attachment digital control element that operates with air and most other commonly available gases. The unique method of venting (Patent pending) eliminates the impedance matching and rarefaction switching problems usually associated with other types of fluid amplifiers, and permits the straightforward design of fluidic circuitry using standard off-the-shelf components.

OPERATING CHARACTERISTICS

Function: Two Input Bistable Flip-Flop

Operating Medium: Gaseous Fluids

Operating Principle: Wall Attachment with vented vortex

Temperature Range: −140°F to +270°F

	MAXIMUM	NOMINAL	MINIMUM
Input Pressure	15 psig	2.5 psig	1.0 psig
Power Consumption		1.3 watts (see Fig. 1)	
Pressure Recovery (Blocked)		45%	
Flow Recovery (Open)		125%	
Frequency Response	800 cps		
Response Time		0.0004 sec.	
Switching Flow	0.050 scfm	0.030 scfm	0.015 scfm
Switching Pressure	0.85 psig	0.35 psig	0.15 psig

Loading Capacity: Eiement is stable with zero inactive leg flow from fully opened to fully blocked loading conditions on the active leg (see Fig. 2)

Note: Detailed operating characteristics shown on reverse side of this data sheet.

APPLICATIONS

This device has been designed for use in digital and pulse processing fluidic circuits which may be built up in the same way as electronic circuits using solid state flip-flops. The use of fluidics represents a cost saving and reliability improvement over conventional electronics and is particularly advantageous in applications where temperature, radiation, shock, vibration or explosion hazards rule out the choice of electronic circuitry.

SPECIFICATIONS

Dimensions: See drawing on reverse side of sheet

Material: Polycarbonate Thermoplastic

Fittings: Tri-serrated brass suitable for 1/8" I.D. flexible tubing

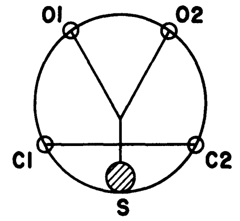
Compatibility: Compatible with most common gases, water, oils, acids and alcohols.

Engineering services are available to review application feasibility.

Please contact the Sales Director for further information.

AVIATION ELECTRIC LIMITED

THE Bendix CORPORATION Canadian Affiliate



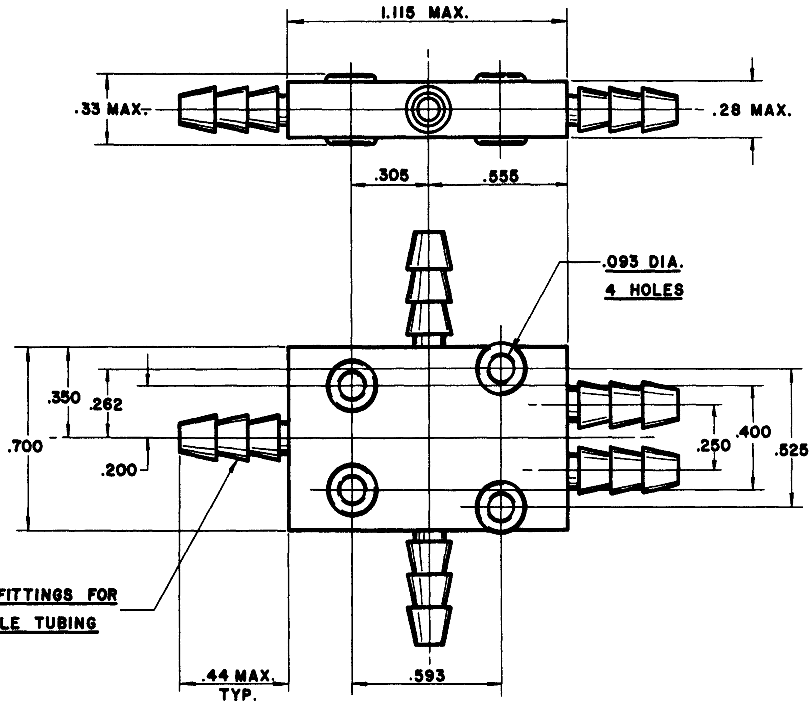
S = SUPPLY

C1 = LEFT CONTROL

C2 = RIGHT CONTROL

O1 = LEFT OUTPUT

O2 = RIGHT OUTPUT



OUTPUT AND LOADLINE CHARACTERISTICS

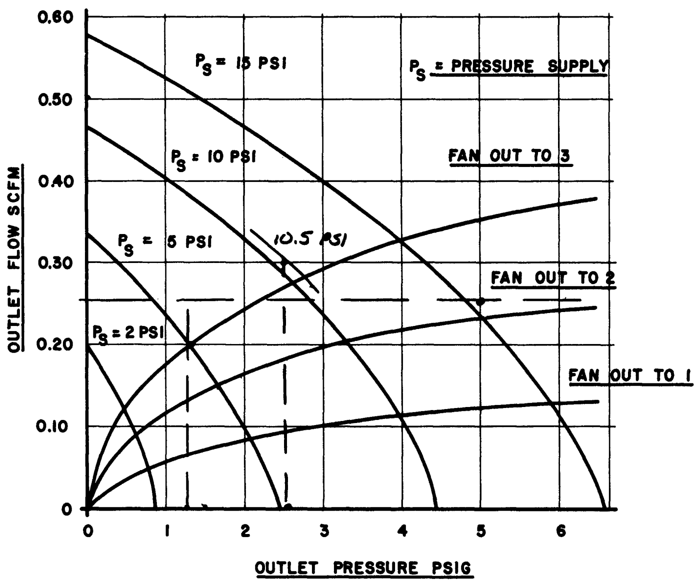


FIGURE 2

SUPPLY PRESSURE AND FLOW

CHARACTERISTICS

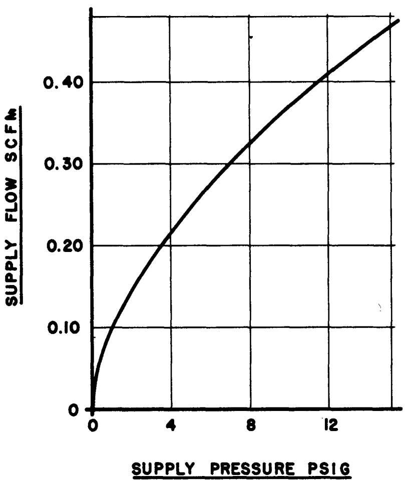
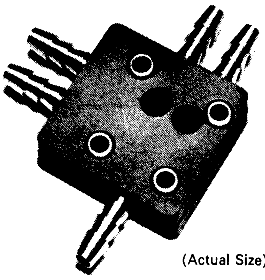


FIGURE 1

H3-21 (REV. 8/66) PRINTED IN CANADA

FILE CATALOGUE: Fluidic Components
SECTION: Application Data
TYPE: 1100MO01


(Actual Size)

MONOSTABLE

FLUID AMPLIFIER

DESCRIPTION

The Aviation Electric type 1100MO1 Monostable Fluid Amplifier is a two-input wall attachment digital control element that operates with air and most other commonly used gases. The unit provides the "or/nor" function required in digital control circuitry. The unique method of venting (patent pending) eliminates the impedance matching and rarefaction switching problems usually associated with no-moving-parts fluid amplifiers and permits the straightforward design of pneumatic circuitry using standard off-the-shelf components.

OPERATING CHARACTERISTICS

Function: Two-input Monostable Flip-Flop

Operating Medium: Gaseous Fluids

Operating Principle: Wall Attachment

Temperature Range: −140°F to +270°F.

	MAXIMUM	NOMINAL	MINIMUM
Input Pressure	15 psig	2.5 psig	1.0 psig (See Fig. 1)
Power Consumption		1.1 watts	
Pressure Recovery (blocked)		42%	
Flow Recovery (unblocked)		125%	
Frequency Response		800 cps	
Response Time		0.0004 sec.	
Switching Pressure		0.35 psi max.	

Loading Capacity: Element is stable with zero inactive leg flow from fully opened to fully blocked conditions on the active leg. (See Fig. 2)

Note: Detailed operating characteristics given on the reverse side of this data sheet.

APPLICATIONS

This device has been designed for use in digital and pulse processing fluidic circuits which may be built up in the same way as electronic circuits using solid-state "or/nor" flip-flops. The use of fluidics represents a cost saving and reliability improvement over conventional electronics, and is particularly advantageous in applications where temperature, radiation, shock, vibration, or explosion hazards rule out the choice of electronic/electric circuitry.

SPECIFICATIONS

Dimensions: See drawing on reverse side of sheet

Material: Polycarbonate Thermoplastic

Fittings: Tri-serrated brass suitable for ⅛" I.D. flexible tubing.

Compatibility: Compatible with most common gases, water, oils, acids and alcohols.

Engineering services are available to review applications feasibility.

Please contact the Sales Director for further information.

AVIATION ELECTRIC LIMITED

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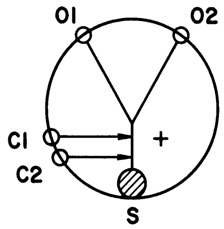
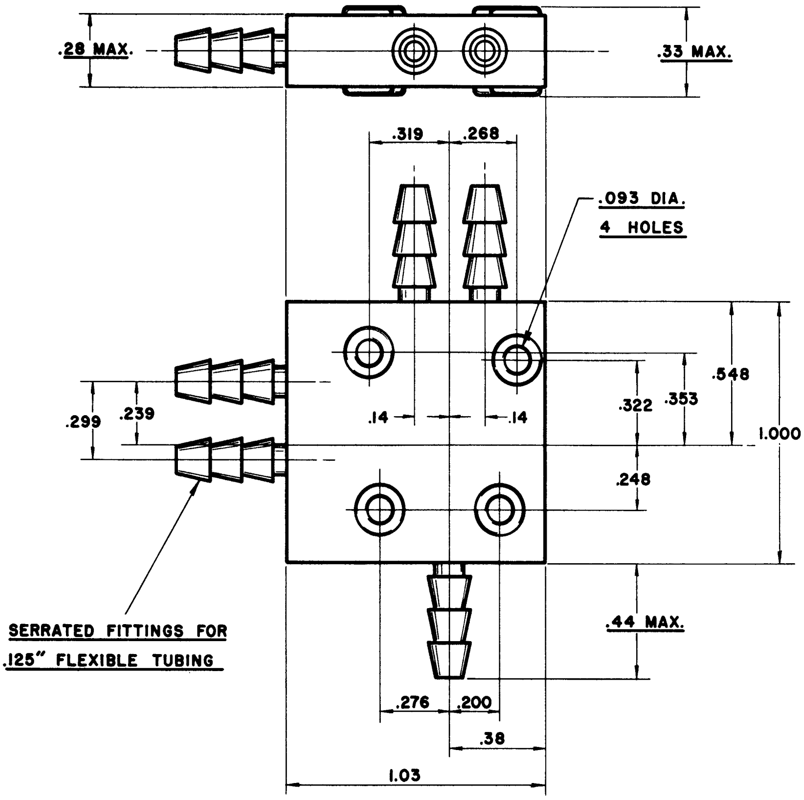


S= SUPPLY

C1, C2 = INPUTS

$O1 = \overline{C1} \cdot \overline{C2}$ = OUTPUT

O2 = C1 + C2 = OUTPUT



OUTPUT AND LOADLINE CHARACTERISTICS

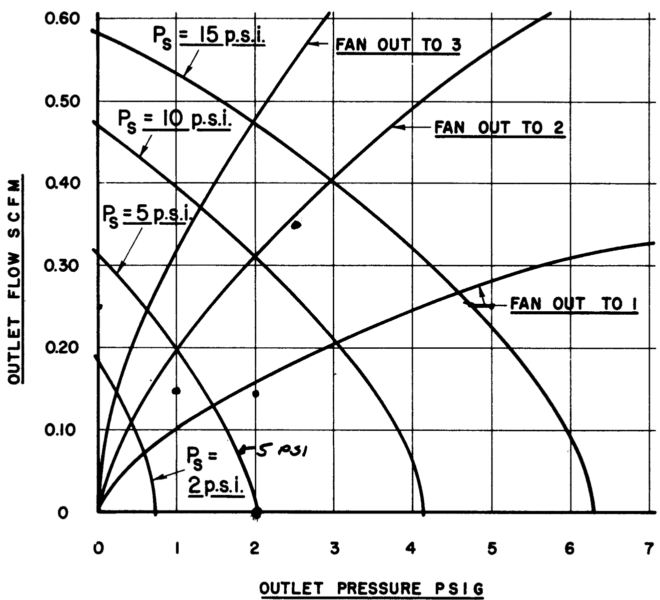


FIGURE 2

SUPPLY PRESSURE

AND

FLOW CHARACTERISTICS

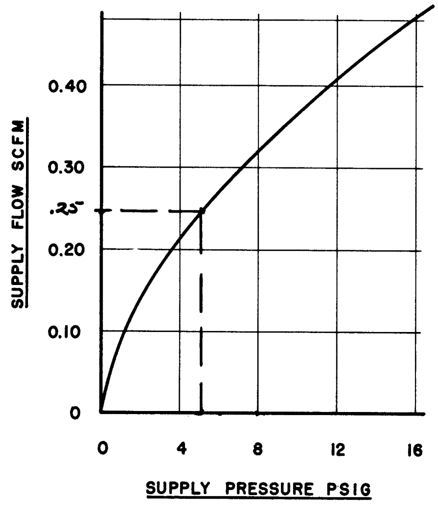


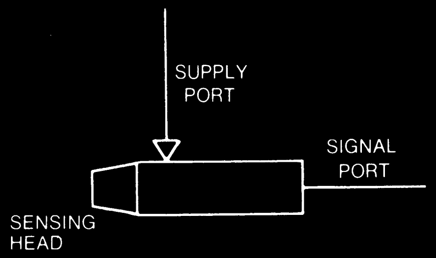
FIGURE 1

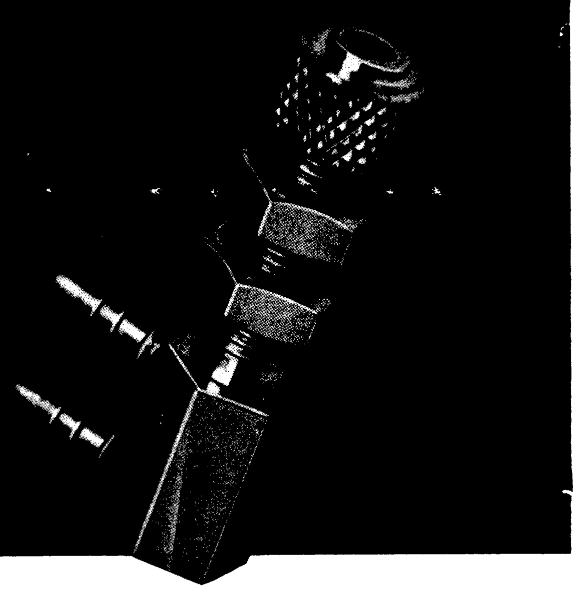
H3-22 (REV. 8/66) PRINTED IN CANADA

FLUIDONICS®

CONE-JET SENSOR

Part No. 300218





GENERAL DESCRIPTION: The Fluidonics Cone-Jet Sensor is a true fluidic element which permits sensing the presence of objects without physical contact. It allows as much as ten times the sensing gap provided by an ordinary back pressure sensor with far less flow consumption. Operation is based on the increase of pressure within a converging conic flow pattern when resistance to that flow is created by presence of an object or an opposing jet stream. It is operable on all gases. Operating characteristics are compatible with all Fluidonics Fluidi-Stak devices, Sensicon units, the Momentum Amplifier, and some interface devices.

OPERATIONAL CONSIDERATIONS: In its simplest form, the Cone-Jet Sensor is connected as in Figure 1. Both signal and supply ports have barbed connectors for ¼″ O.D. plastic tubing. The output impedance is low and flow recovery high enough to control other Fluidonics fluidic devices operating at the same supply pressure as the Cone-Jet at sensing gaps up to ⅛ inch. The converging conic flow pattern of high velocity gas allows precision in sensing edges, steps, grooves, narrow (small as 1/16 inch), or small diameter objects, and even cloth or screen mesh. The efficiency of its pressure and flow recovery makes practical sensing gaps as great as 0.2 inch. If the device is connected as in Figure 2, a biasing flow into the signal port can significantly increase the sensing gap obtainable. However, flow consumption is also increased considerably.

Flow to the signal port of the device is from within the conic flow so that contamination from the surroundings cannot reach a control port through the sensor. This protective capability in addition to its sensitivity makes the Cone-Jet an excellent receiver in an interruptible jet sensing mode as shown in Figure 3. Note that while nearly immune to external contamination, the narrow passages of the Cone-Jet demand dry air filtered to at least 40 microns for maximum reliability.

The recovery pressure of the Cone-Jet Sensor is so nearly directly related to supply pressure that a dimensionless curve of pressure recovery versus sensing gap can be made (Figure 5).

MATERIALS OF CONSTRUCTION:

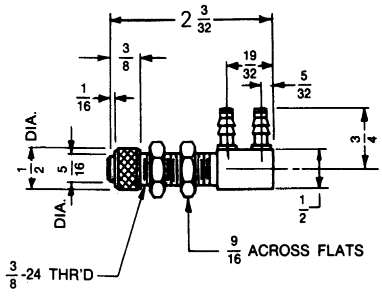
Aluminum. Other materials available on special order.

PERFORMANCE:

Supply pressure range:	2-35 psig
Supply impedance:	See Figure 4
Pressure Recovery:	See Figure 5

TYPICAL APPLICATIONS:

Position
Limit Sensing
Limit Air Gaging
Proximity detections



DIMENSIONAL DRAWING

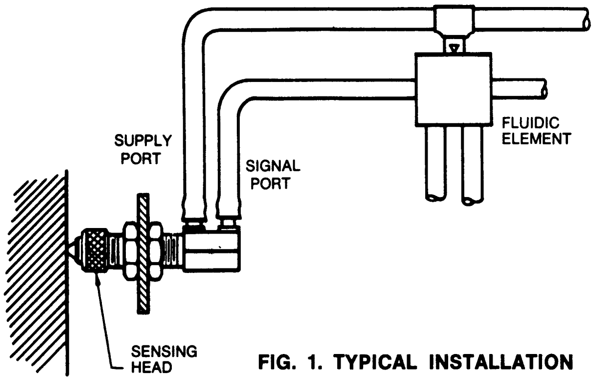


FIG. 1. TYPICAL INSTALLATION

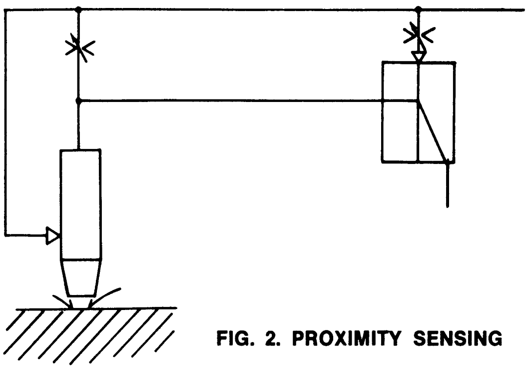


FIG. 2. PROXIMITY SENSING

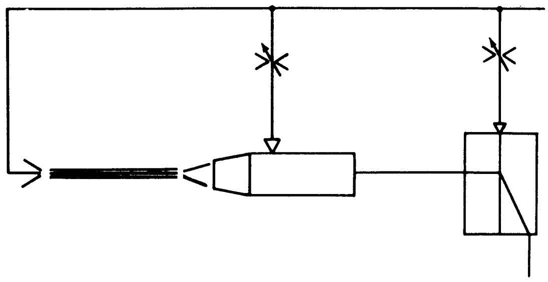


FIG. 3. INTERRUPTABLE JET SENSING

TYPICAL PERFORMANCE CURVES

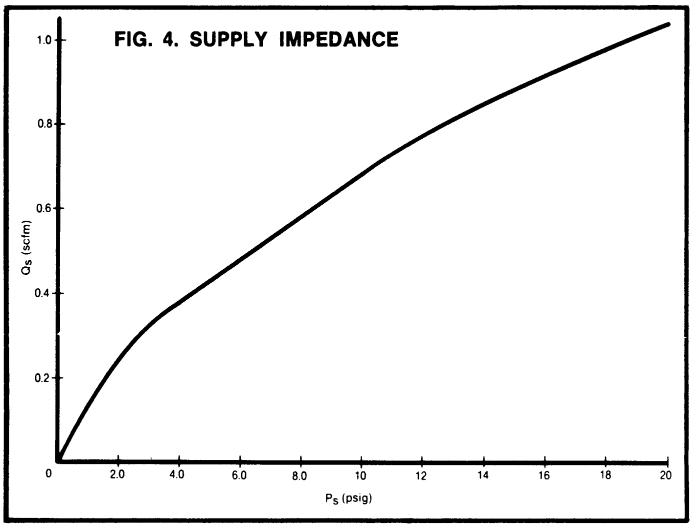


FIG. 4. SUPPLY IMPEDANCE

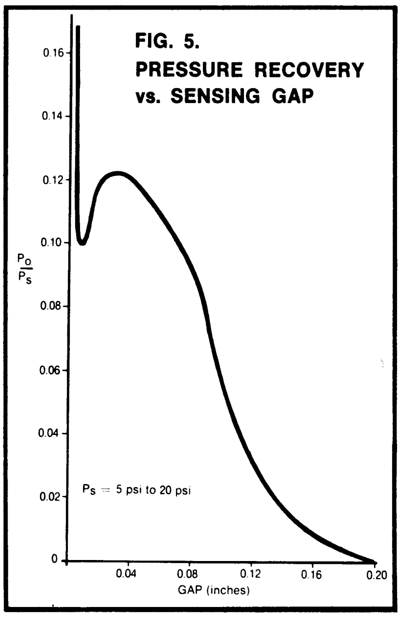


FIG. 5. PRESSURE RECOVERY vs. SENSING GAP

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FORM NO. 6030 PRINTED IN U.S.A.

APPENDIX B

PHOTOGRAPHS OF EXPERIMENTAL APPARATUS

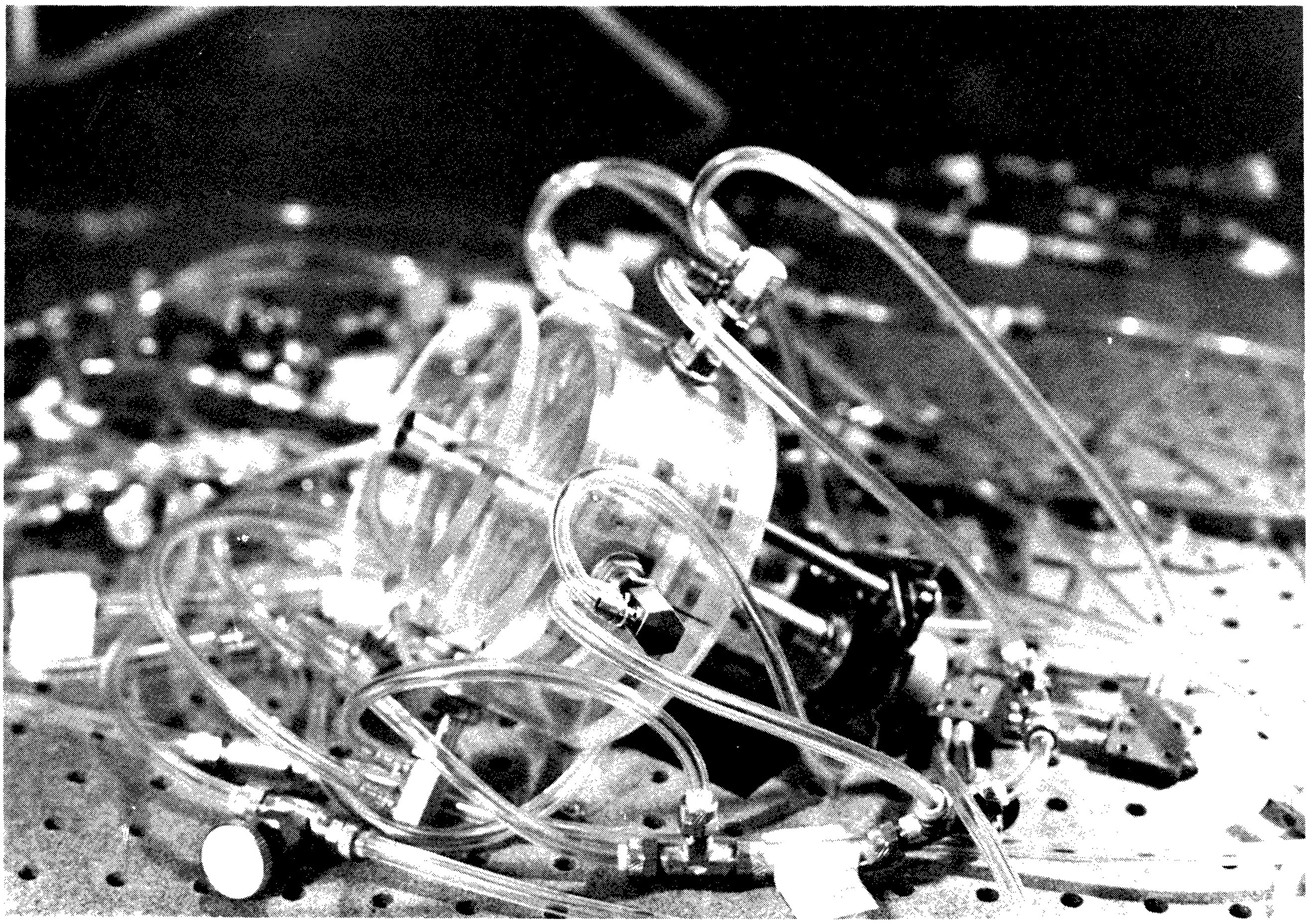


Figure 1B. Fluid-mechanical timer.

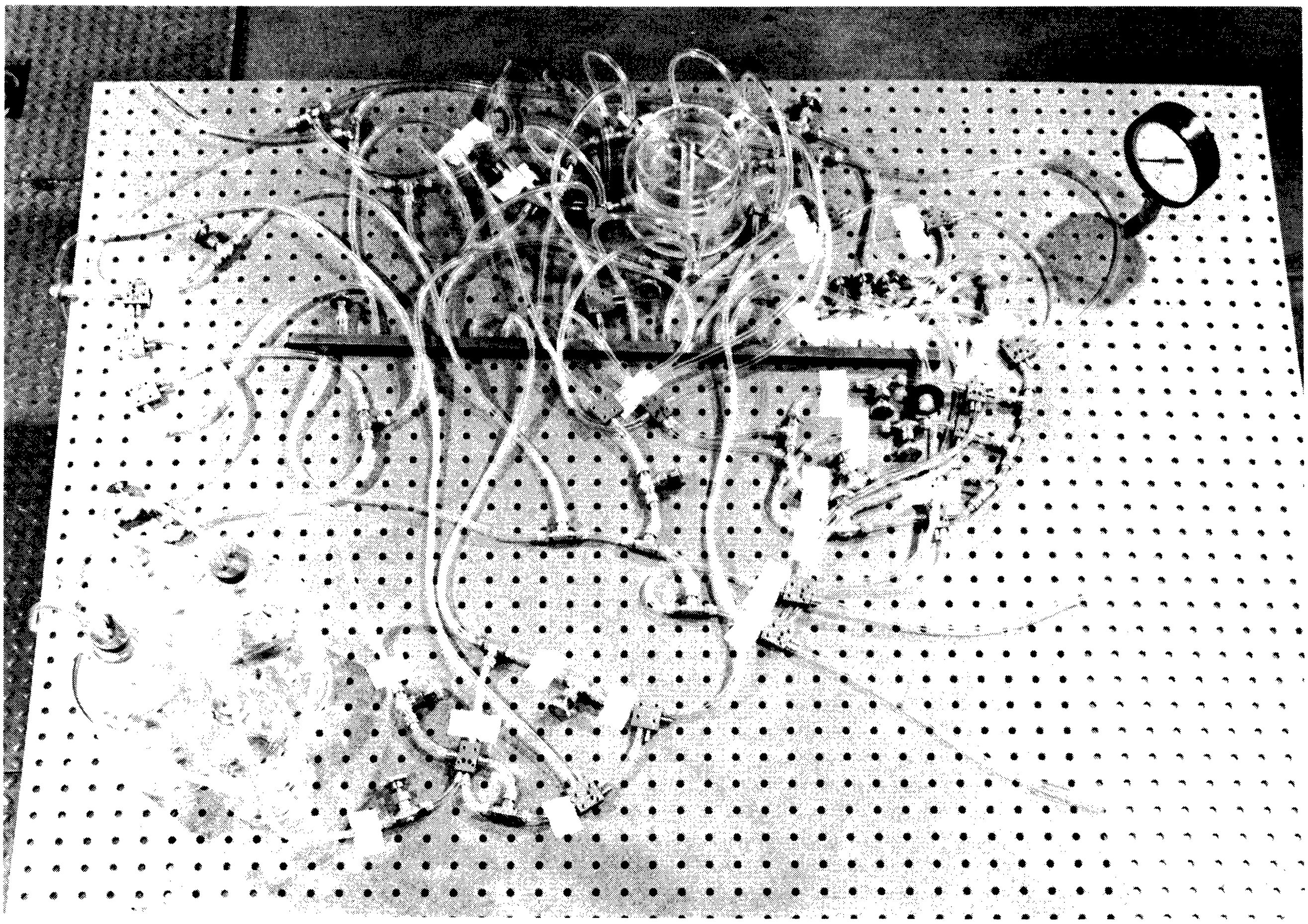


Figure 2B. Logic circuit.

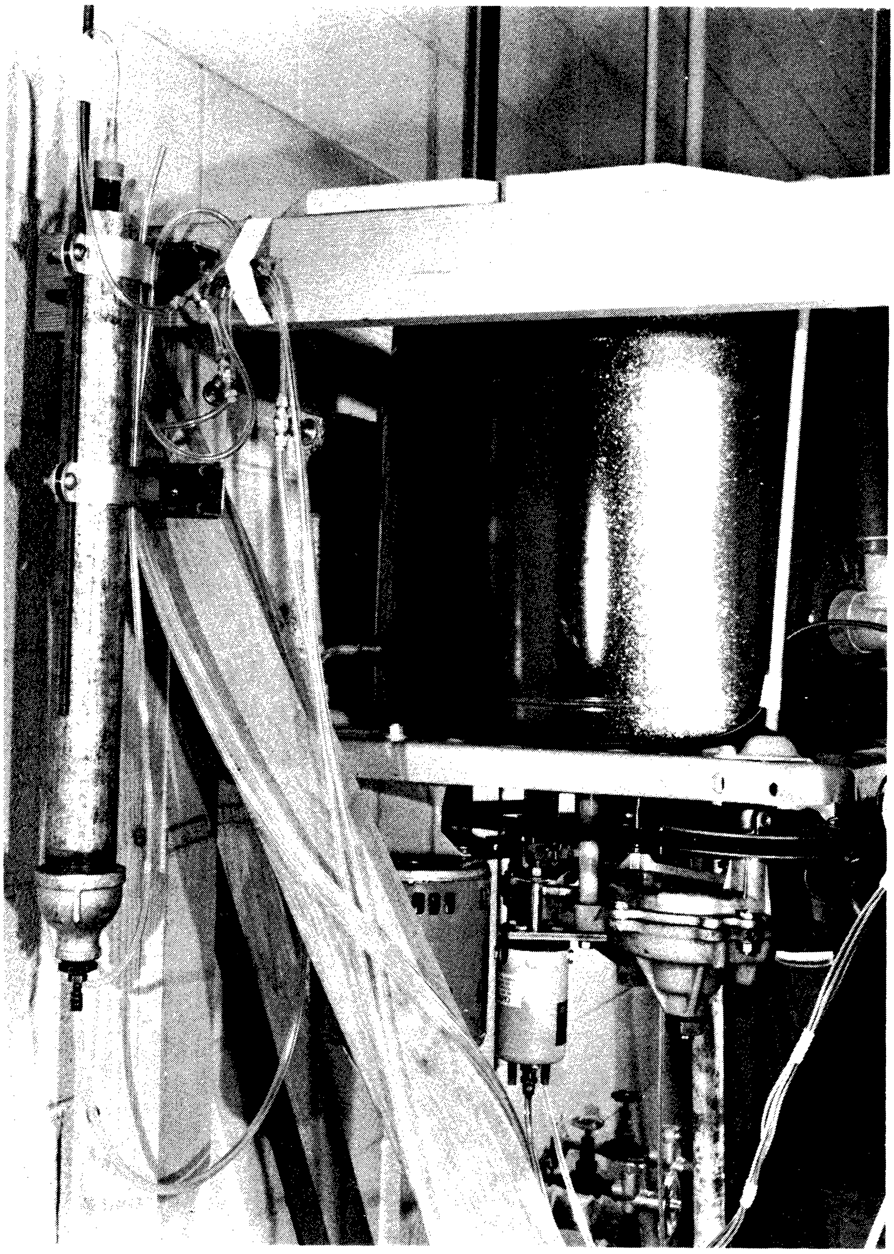


Figure 3B. Water level detector apparatus.

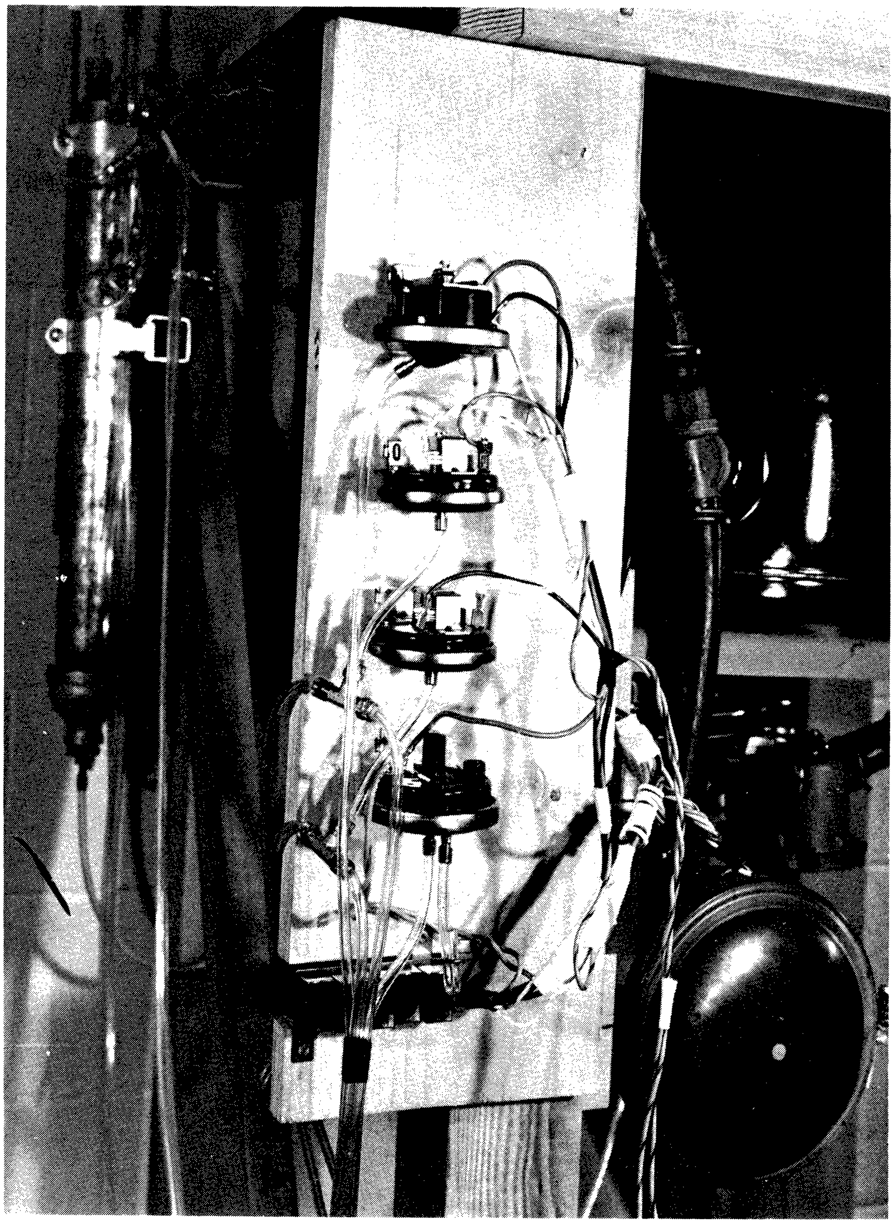


Figure 4B. Pressure switches.

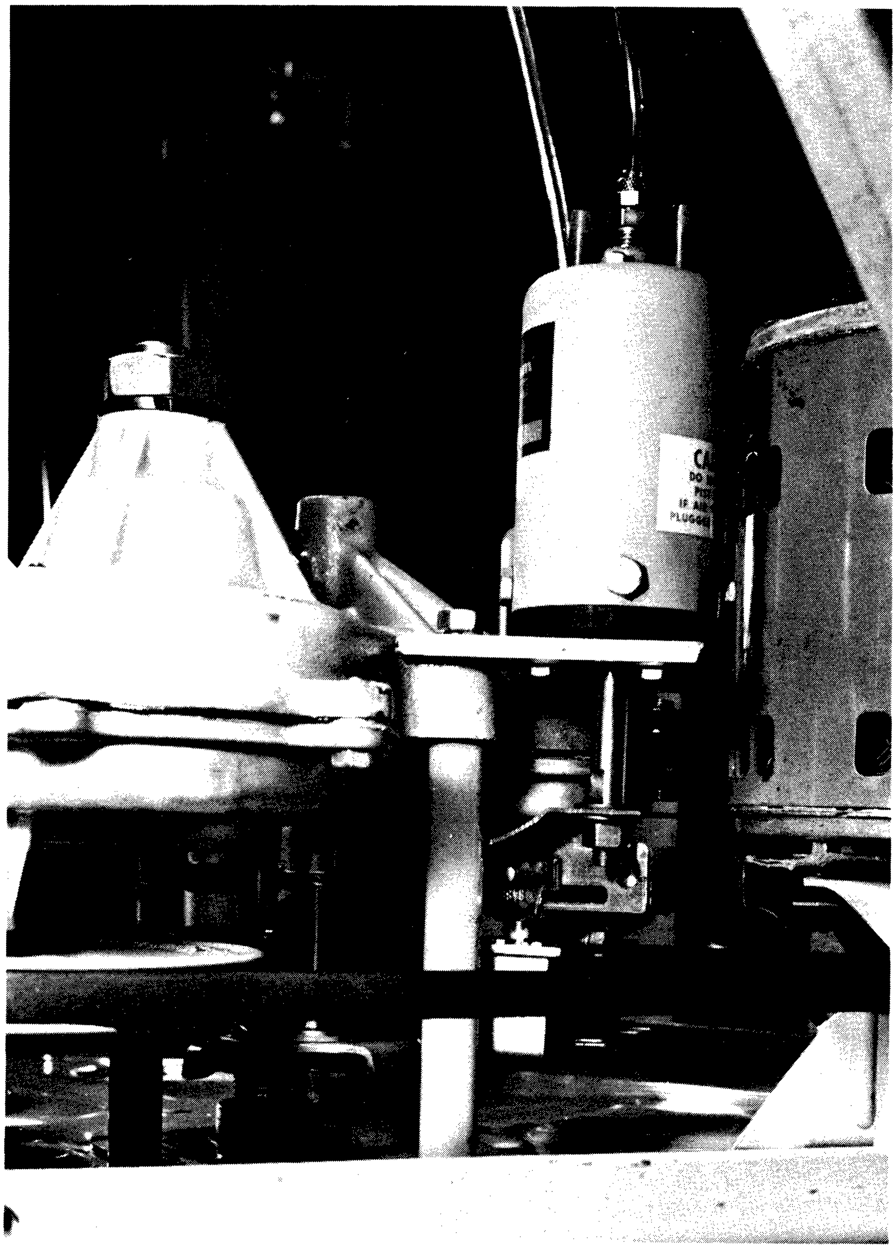


Figure 5B. Agitation linear actuator—mounting and lifting arm.

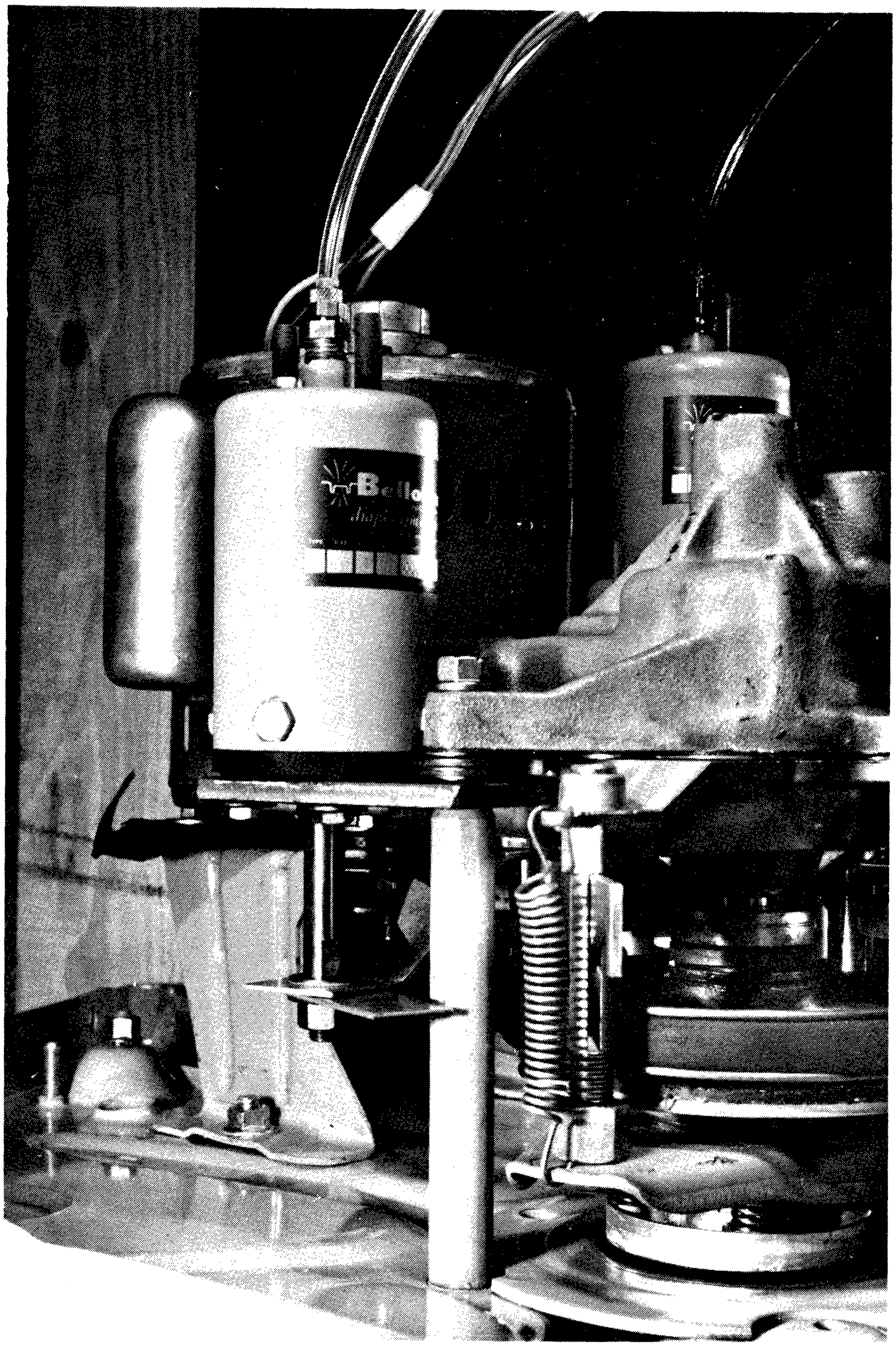


Figure 6B. Spin linear actuator—mounting.

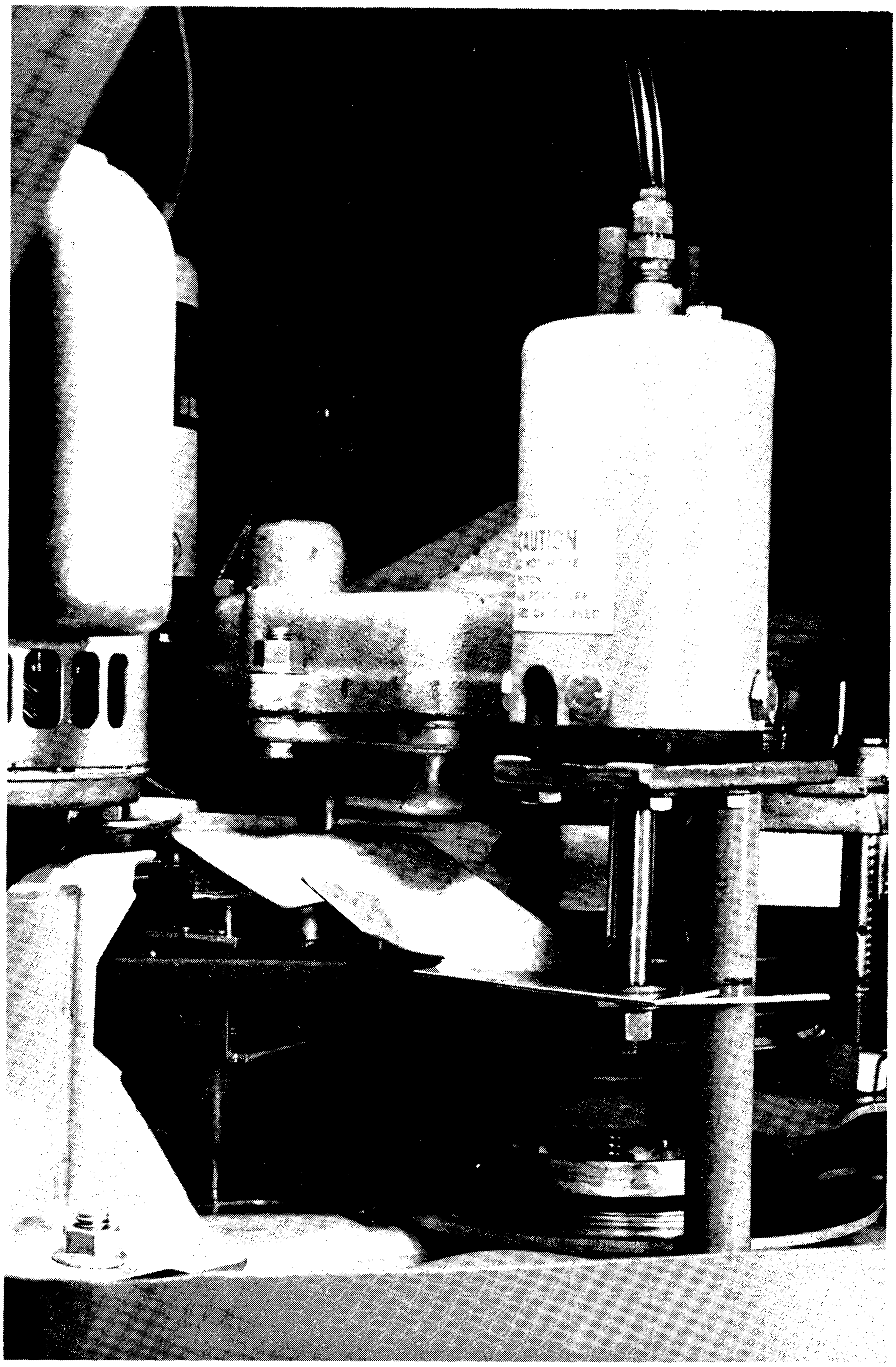


Figure 7B. Spin linear actuator—lifting arm.

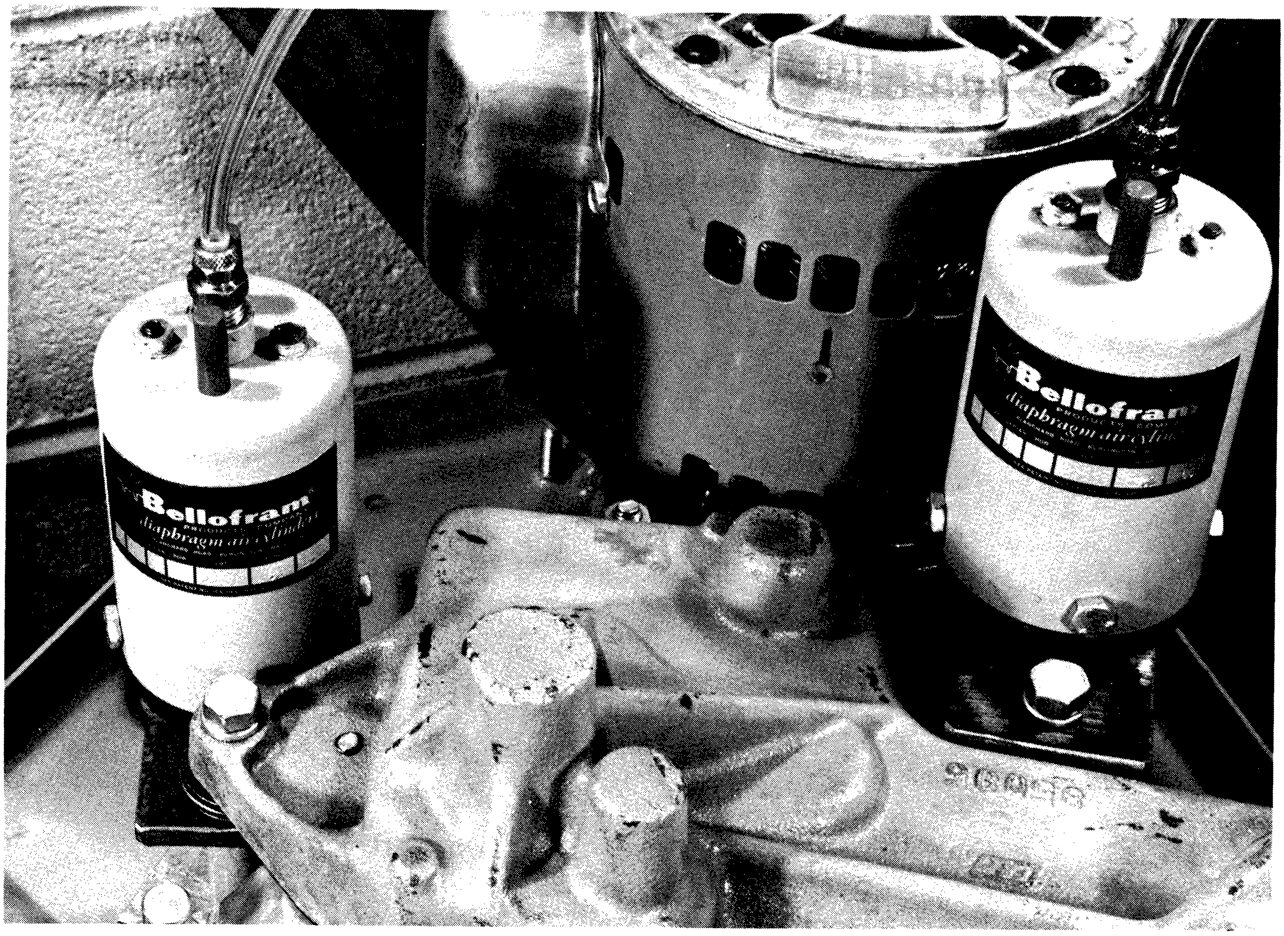


Figure 8B. Relative position of actuators.

Figure 9B. Air-piloted water valves.

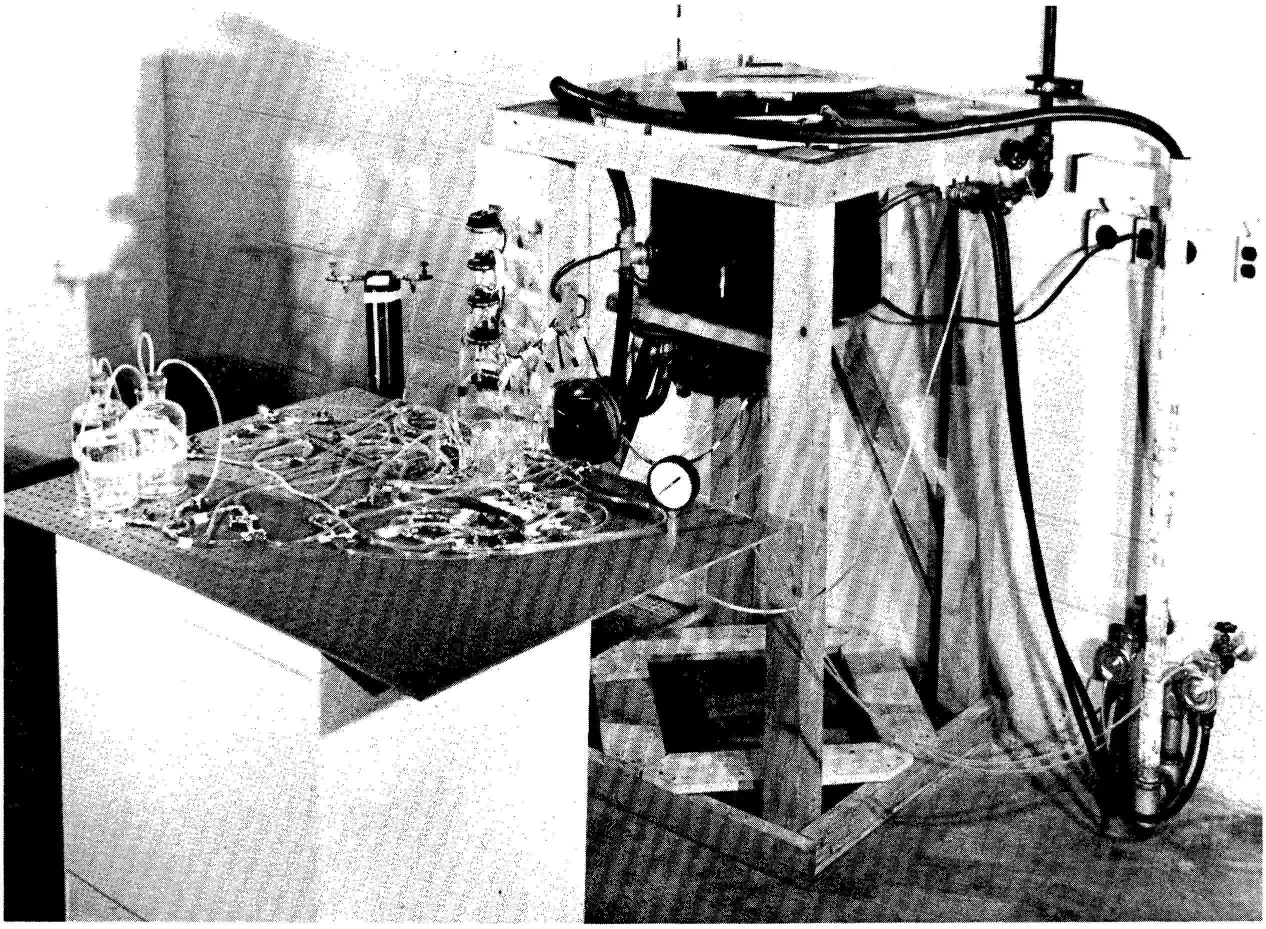


Figure 10B. Overall layout—right perspective.

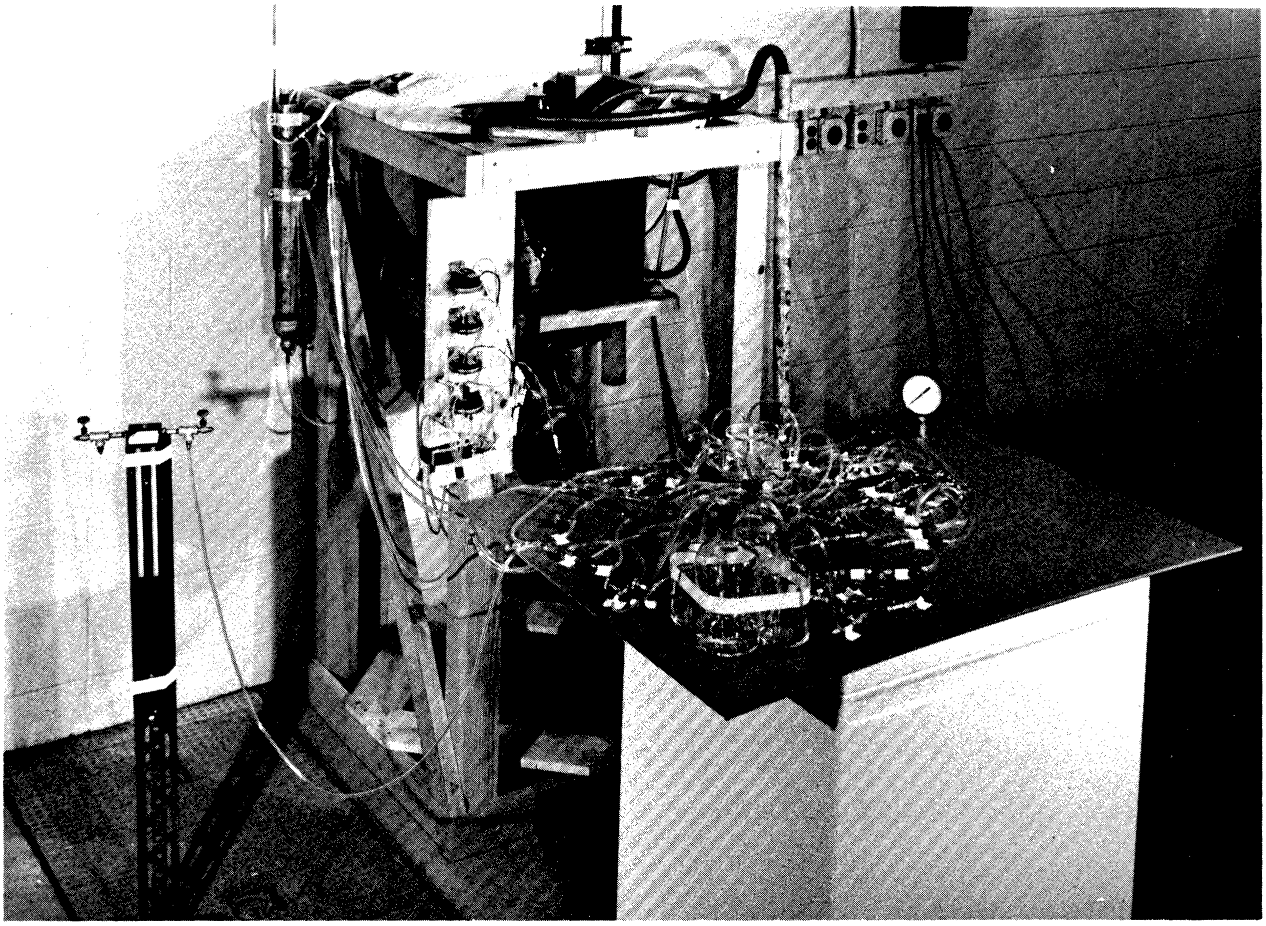


Figure 11B. Overall layout--left perspective.

APPENDIX C

SKETCHES OF SYSTEM LOGIC

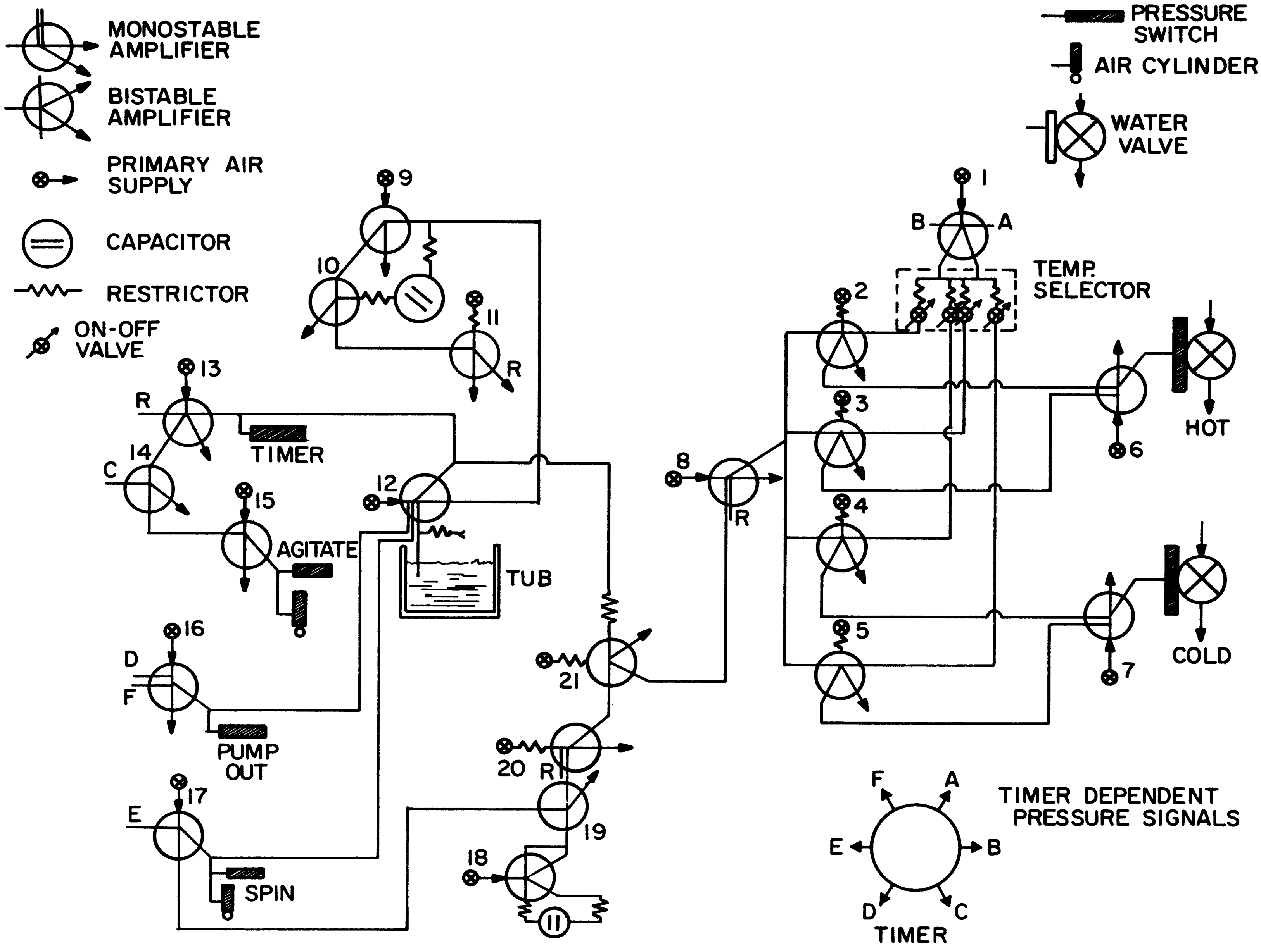


Figure 1C. Schematic diagram of logic circuit.

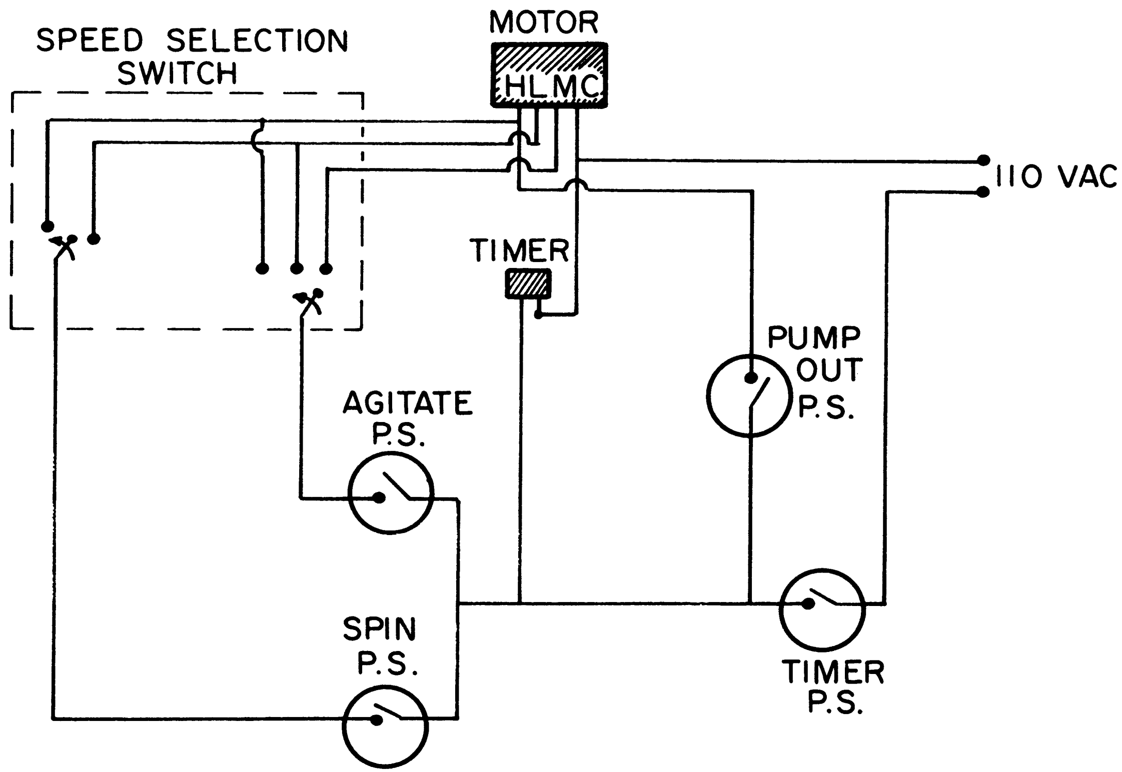


Figure 2C. Schematic diagram of electrical circuit.

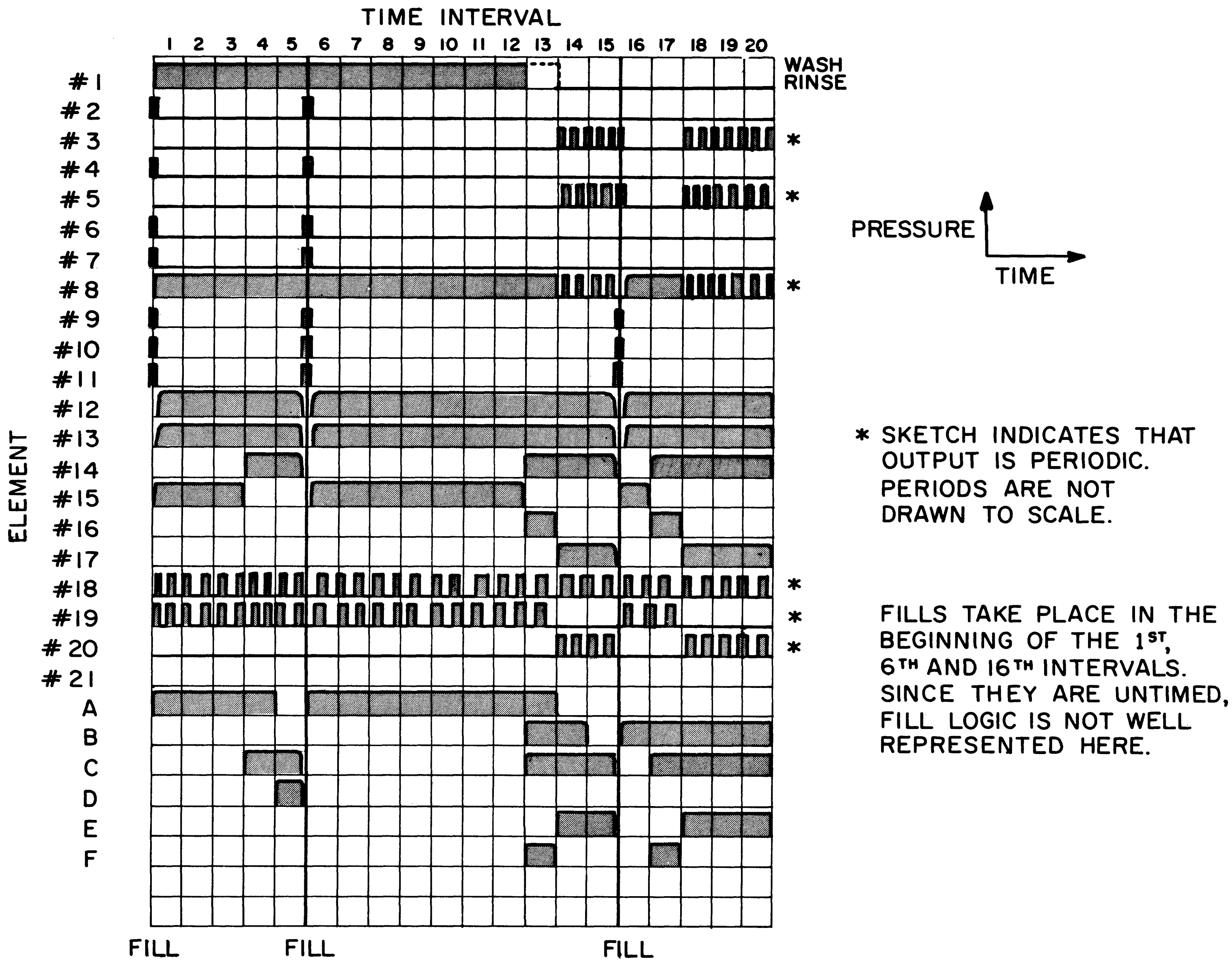


Figure 3C. Logic patterns for elements of logic circuit.

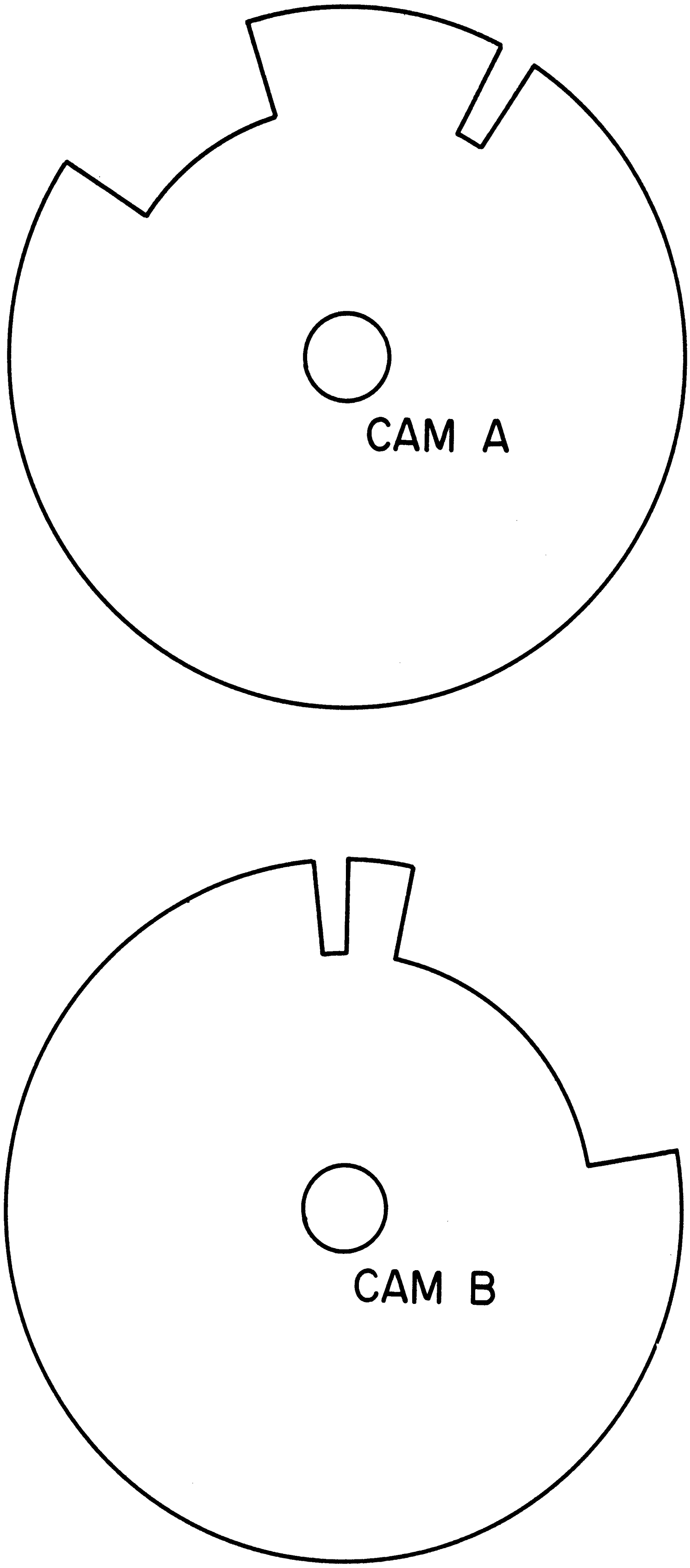
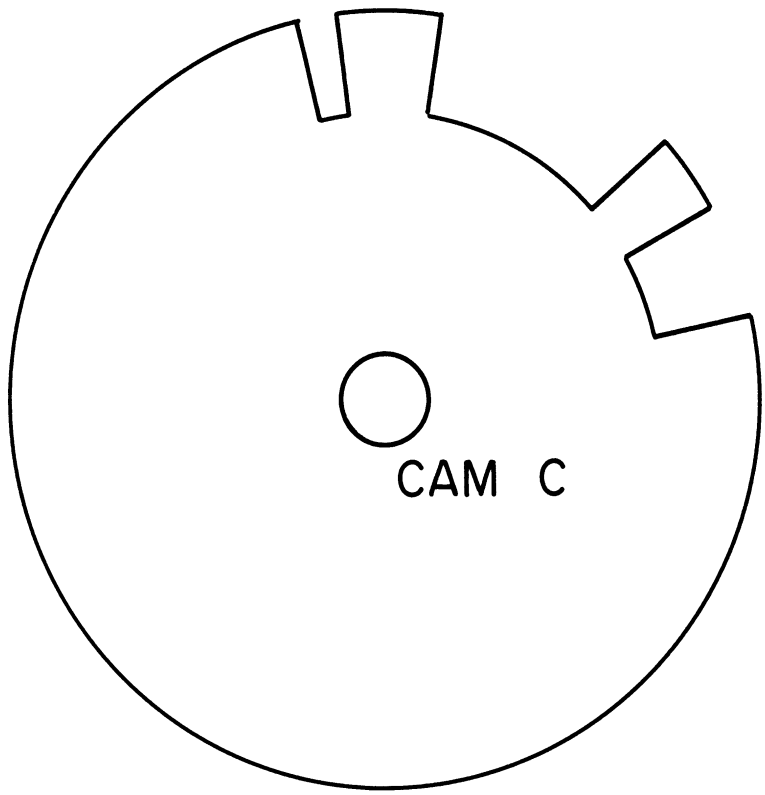


Figure 4C. Cam diagrams.



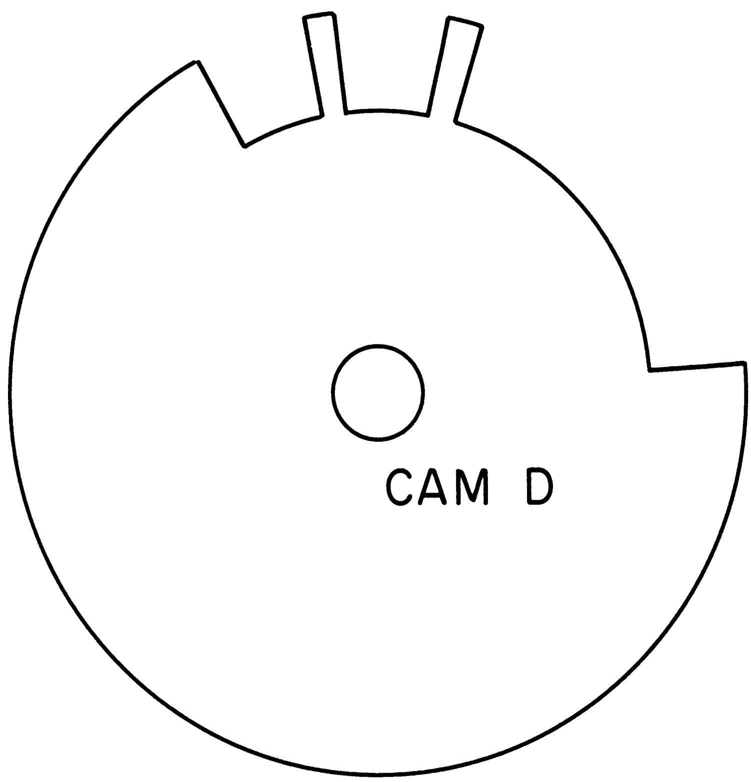
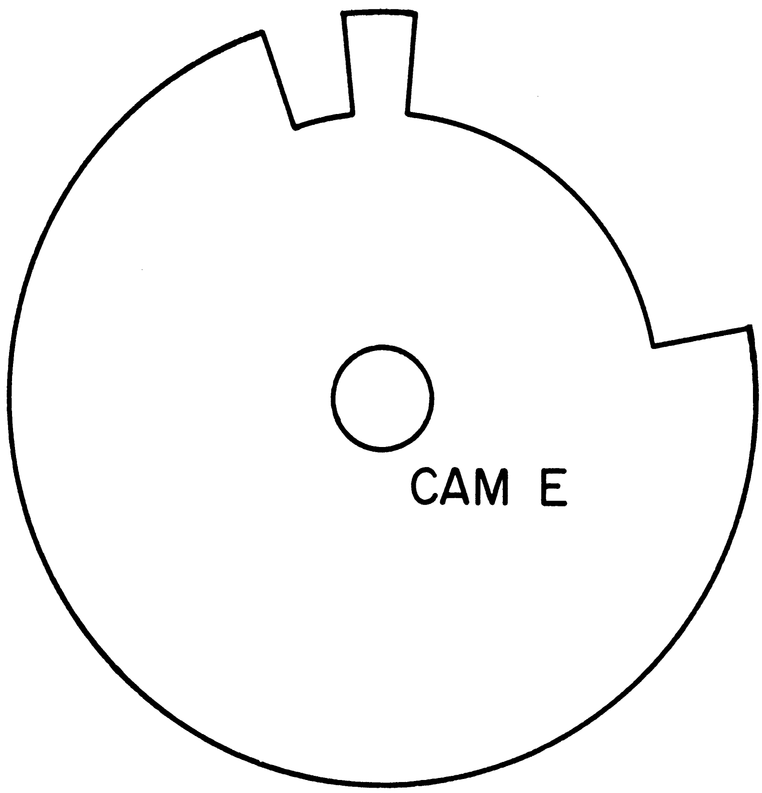


Figure 4C. Continued.



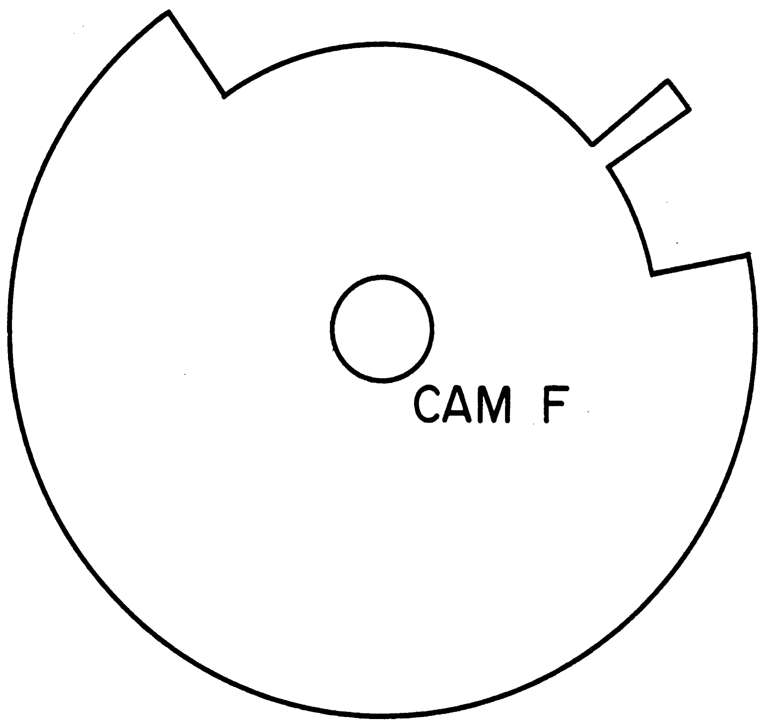


Figure 4C. Concluded.