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CONCEPTUAL STUDY OF EQUIPMENT AND HARDWARE TO PROVIDE RANDOM MOTION OF VEHICLE SIMULATORS

~~FOR~~

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INTRODUCTION

THE WORK COVERED BY THIS REPORT CONSISTS OF A CONCEPTUAL STUDY OF BOTH CONVENTIONAL AND ENERGY STORAGE HYDRAULIC CONTROL SYSTEMS FOR A VIBRATION GENERATOR. THREE TYPES OF SYSTEMS ARE STUDIED IN DETAIL AND COMPARED AS TO COST, PERFORMANCE, ACCURACY, PROBABILITY OF SUCCESS, AND EASE OF CHANGING FREQUENCY AND AMPLITUDE.

TWO TYPES OF ENERGY STORAGE SYSTEMS WERE ANALYZED. BOTH SYSTEMS USE RESONANCE PHENOMENA AS A MEANS OF ENERGY STORAGE. ONE SYSTEM USES MECHANICAL SPRINGS FOR ENERGY STORAGE, AND THE SECOND SYSTEM UTILIZES FLUID COMPRESSION AS A MEANS OF ENERGY STORAGE. THE THIRD SYSTEM STUDIED IS A CONVENTIONAL SYSTEM WHICH DOES NOT UTILIZE ENERGY STORAGE BUT PROVIDES ALL THE POWER REQUIRED TO DRIVE THE LOAD.

A COMPARISON OF ALL THREE SYSTEMS IS SHOWN IN SECTION IV.

SECTION I

SPRING STORAGE SYSTEM

System Description

THE BASIC SYSTEM CONSISTS OF A VARIABLE DELIVERY PISTON PUMP WHICH POWERS A CLOSED LOOP SERVO VALVE-CYLINDER THAT CONTROLS THE POWER INPUT TO THE LOAD. THE POWER REGENERATION IS DERIVED FROM THE MASS VELOCITY MOTION BEING STORED IN HELICAL SPRINGS WHICH ARE CONNECTED TO THE LOAD MASS. THUS, WHEN THE MASS IS IN MOTION, ALL OF ITS ENERGY IS TRANSFERRED TO THE SPRINGS. WHEN THE MASS HAS GIVEN UP ITS ENERGY, THE SPRINGS NOW HAVE "STORED" ENERGY, AND THIS IS USED TO RE-ACCELERATE THE MASS. THUS, WE HAVE A REGENERATIVE SYSTEM WHICH WOULD REMAIN IN OSCILLATION IF NO MECHANICAL LOSSES WERE PRESENT AND WHOSE RATE OF OSCILLATION WOULD BE DETERMINED BY THE MASS AND THE SPRING RATE. SINCE LOSSES ARE PRESENT, ALL THE SPRING MASS SYSTEM REQUIRES FROM AN EXTERNAL SOURCE IS AN AMOUNT OF ENERGY EQUAL TO THESE MECHANICAL LOSSES. THE HYDRAULIC COMPONENTS WERE SIZED SO THAT THIS AMOUNT OF POWER COULD READILY BE APPLIED TO THE LOAD.

THE SELECTED COMPONENTS INCLUDE 2 POWER UNITS (ONE FOR EACH AXIS). A PRESSURE COMPENSATED PUMP IS UTILIZED WHICH PROVIDES ONLY THE OUTPUT FLOW THAT IS REQUIRED BY THE SYSTEM, THUS DECREASING THE POWER REQUIRED FOR DUTY CYCLES OF LESS THAN MAXIMUM. IT IS ASSUMED THAT ONLY ONE ARM WILL BE OPERATED AT ONE TIME SO THAT ONLY 2 POWER UNITS ARE REQUIRED. THESE PROVIDE THE HYDRAULIC FLOW REQUIRED BY THE HIGH FLOW SERVO VALVES. IN ORDER TO USE A SINGLE ENDED CYLINDER WITH THE DESIRED CLEVIS TYPE MOUNT, A 3-WAY SERVO VALVE WAS SELECTED IN ORDER TO PROVIDE MINIMUM FLOW FROM THE PUMP AT MAXIMUM DUTY CONDITIONS.

SYSTEM OPERATION

THE CONTROL SYSTEM FOR THE SPRING MASS LOAD UTILIZES A FREQUENCY GENERATOR AS THE COMMAND DEVICE, A LINEAR AMPLIFIER WHICH COMPARES THE COMMAND SIGNAL WITH THE LOAD POSITIONAL SIGNAL, AMPLIFIES THE ERROR SIGNAL BETWEEN THE COMMAND AND FEEDBACK AND POSITIONS THE SERVO VALVE SPOOL ACCORDINGLY. THUS, THE FLOW TO THE LOAD CYLINDER AND POWER TO THE LOAD IS CONTROLLED. THE SYSTEM WILL EITHER ADD OR SUBTRACT POWER FROM THE LOAD TO LIMIT THE AMPLITUDE OF THE LOAD POSITION TO ITS PROPER VALUE.

THE SPRING-MASS SYSTEM WILL DETERMINE ITS OWN RESONANT FREQUENCY UNLESS FORCED TO A DIFFERENT FREQUENCY. AT THIS RESONANT FREQUENCY POINT, THE POWER INPUT WILL BE AT A MINIMUM. IF THE COMMAND SIGNAL IS AT OTHER THAN THE SPRING MASS RESONANT FREQUENCY, THE POWER REQUIRED BY THE SYSTEM WILL INCREASE, AND THE SYSTEM WILL OSCILLATE AT THE COMMAND FREQUENCY. THUS, THE SYSTEM CAN BE FORCED TO OSCILLATE AT A FREQUENCY OTHER THAN THE NATURAL FREQUENCY, BUT ONLY WITHIN THE LIMITS OF POWER AVAILABILITY FROM THE POWER UNIT.

CHANGES IN THE FREQUENCY OF LOAD OSCILLATION CAN BE OBTAINED BY CHANGING THE NUMBER OF SPRINGS IN THE SPRING MASS ASSEMBLY. THE #8 Y AXIS HAS BEEN SIZED TO USE 9 DOUBLE SPRINGS ON EACH SIDE OF THE MASS. EACH DOUBLE SPRING CONSISTS OF 2 OUTER SPRINGS IN SERIES WITH EACH OTHER AND 2 INNER SPRINGS IN SERIES WITH EACH OTHER BUT IN PARALLEL WITH THE OUTER SPRINGS. THUS, THERE ARE 19 POSSIBLE NATURAL RESONANT FREQUENCIES AVAILABLE FROM .84 TO .2 CPS - THE .2 CPS POINT BEING OBTAINED WITH NO SPRINGS IN THE SYSTEM. HERE THE HYDRAULIC OIL IN THE SYSTEM STORES ENERGY THE SAME AS THE MECHANICAL SPRINGS DID PREVIOUSLY.

ADVANTAGES AND DISADVANTAGES OF THE SPRING STORAGE SYSTEM

THE RELATIVE IMPORTANCE OF THE FOLLOWING ADVANTAGES AND DISADVANTAGES CAN READILY BE DETERMINED BY THE CUSTOMER, SO NO ATTEMPT IS MADE HERE TO EVALUATE THEIR IMPORTANCE.

ADVANTAGES:

1. MECHANICAL SPRINGS ARE HIGHLY EFFICIENT.
2. MECHANICAL SPRINGS PRODUCE NATURAL SELF-CENTERING OF THE LOAD.
3. MECHANICAL SPRINGS ARE VERY LINEAR IN SPRING RATE AND HENCE WILL TEND TO PRODUCE A GOOD SINE WAVE OF POSITION.
4. BY CHANGING SPRINGS TO CHANGE THE FREQUENCY OF OSCILLATION, THE SYSTEM ALWAYS OPERATES AT ITS OPTIMUM EFFICIENCY, THAT IS, ITS NATURAL RESONANT FREQUENCY.
5. THE SPRING RATE CHANGE WITH TEMPERATURE IS EXTREMELY SMALL IN COMPARISON TO GAS TYPE ENERGY STORAGE SYSTEMS.

DISADVANTAGES:

1. MECHANICAL SPRINGS ARE AWKWARD TO HANDLE.
2. THE FREQUENCY VARIATION IS OBTAINED BY MECHANICAL STEP CHANGES INSTEAD OF ELECTRICALLY CHANGING THE COMMAND FREQUENCY ALONE.
3. MECHANICAL SPRINGS REQUIRE CONSIDERABLE SPACE FOR LARGE HORSEPOWER STORAGE.
4. MECHANICAL SPRINGS HAVE ABOUT A ±10% TOLERANCE ON SPRING RATE, WHICH MEANS APPROXIMATELY ±5% ON THE NATURAL FREQUENCY OF OSCILLATION.
5. CONTROL OF ANY TYPE OF RESONANT CIRCUIT IS DIFFICULT.

Specifications of Spring Storage System

A. Load frequency: ±5% of set frequency.

B. Frequency adjustment: By adding or removing springs.

C. Amplitude tolerance: ±5% of set amplitude.

D. Amplitude adjustment: Electrical.

E. Phase adjustment between axis: Electrical.

F. Command signal: Both axes from same source.

G. Power amplification: 580% at spring mass natural resonance.

H. Time to reach maximum amplitude: Approximately .5 min.

J. Operating fluid temperature: 125° ±25°F.

K. Spring space requirement (3Y axis) 116" L x 157.5" W x 17.5"H. Progressively less for other axis by approximately horsepower storage requirement.

L. Viscous friction: Assumed to be 5% of maximum force at maximum velocity.

SYSTEM COMPONENTS REQUIRED

1 POWER UNIT CONSISTING OF:

- 1 PVA120 PUMP WHICH INCLUDES:
  - 1 PVA120-RS-10 PUMP
  - 1 CAP-SCR7-10-S-- PRESSURE COMPENSATOR CONTROL
  - 1 V210-6-1C-12-LH-S214 AUXILIARY PUMP
  - 1 249708 AUXILIARY PUMP MOUNTING KIT
  - 2 FL2-120-28 BWL-10 FLANGE
- 1 PVA120 PUMP WHICH INCLUDES:
  - 1 PVA120-RS-10 PUMP
  - CAP-SCR7-10-S-- PRESSURE COMPENSATOR CONTROL
  - 2 FL2-120-28-BWL-10 FLANGE
- 1 CT-06-B-10 RELIEF VALVE
- 2 CT-10-B-10 RELIEF VALVE
- 2 CF-16-H RELIEF VALVES
- 6 CGVMS-06-10
- 2 ELECTRIC MOTOR AND STARTER
  - 3 ø 440 VAC, 150 HP -
  - 1200 RPM OPEN DRIPPROOF
- FILTERS
- 2 50V8SA-1C-10 VANE PUMP
- 2 CCW-6-10 OIL COOLER
- 1 CCRV-1-06-10 TEMP. CONTROL VALVE
- 2 CCST-1-06-10 WATER STRAINERS
- 1 CCBW-1-10 BULB WELL
- 1 OIL TANK
- 2 CHECK VALVES
- 16 DTPG2-04-10 GLOBE VALVE
- 16 2 INCH GLOBE VALVE - 5000 PSI

Cylinder Sizes

Axis	Cylinder (Inches) Bore	Stroke	Servo Valve	Springs Volume (Cu. Ft)
2Y	1.5	12	Medium flow	3.3
2X	1.5	12	Medium flow	1.9
3Y	1.5	15	Medium flow	5.2
3X	1.5	15	Medium flow	2.9
4Y	1.5	18	Medium flow	10
4X	1.5	18	Medium flow	8
5Y	1.5	32	Medium flow	35
5X	1.5	32	Medium flow	24
6Y	2	36	High flow	64
6X	2	36	High flow	40
7Y	2	48	High flow	120
7X	2	48	High flow	75
8Y	2	60	High flow	184
8X	2	60	High flow	123
9Y	2	64	High flow	112
9X	2	64	High flow	60.5

ELECTRICAL COMPONENTS REQUIRED PER AXIS: COMMON TO ALL 16 AXES

1 FEEDBACK MECHANISM

1 AMP

1 POWER SUPPLY (ELCOR)

1 SIGNAL GENERATOR

1 PREAMPLIFIER AND POWER SUPPLY

MISCELLANEOUS ELECTRONIC EQUIPMENT & ENCLOSURE

ENGINEERING TASKS

- SERVO VALVE
  - ENGINEERING
  - TEST
- EVAL. SPEC. CIRCUITS
  - TEST
- 16 SPECIFIC SYSTEM ANALYSES
- OVERALL SYSTEM ANALYSIS
- SECTION OF COMPONENTS
- SYSTEM SCHEMATIC
  - ELECTRICAL & HYDRAULIC
- CUSTOMER COORDINATION
- VENDOR COORDINATION
- PARTS LISTS
- POWER UNIT TEST
- INSTRUCTION MANUALS
- SPEC. TEST MATERIALS
- MFG. LIAISON
- SPEC. CYL.
- PRECONTRACTUAL
- TRIPS

SECTION II

FLUID STORAGE SYSTEM

INTRODUCTION

THE ENERGY STORAGE METHOD CONSIDERED IN THIS PORTION OF THE STUDY IS THE FLUID SPRING. ENERGY IS STORED BY COMPRESSION OF THE HYDRAULIC FLUID AND IS RELEASED DURING DECOMPRESSION TO DRIVE THE LOAD.

System Description

The drive system consists of a hydraulic cylinder actuator attached to the load. The cylinder actuator is operated by a servo controlled pump which generates the flow and pressure required to position the load.

With this system, the hydraulic fluid serves as the energy storage medium. The hydraulic system is tuned to resonance at the maximum drive frequency of the load. Cylinder actuators have been sized so that hydraulic resonance occurs near this frequency on all drives.

Two PVA120 servopumps operating in parallel are used as the controlling device on each axis. This permits the operation of any one station. The actuators on the remaining stations are isolated from the servopump by globe valves.

System Operation

The system operates basically as a spring-mass resonant system with the servopump output acting as the forcing function of the system. The hydraulic fluid serves as the spring in this case, with the hydraulic system tuned to the maximum driving frequency of the load. This provides maximum power conservation at the frequency where maximum load power is required.

The hydraulic fluid absorbs energy from the load during deceleration by compression of the fluid. During acceleration, energy is returned to the load as the fluid decompresses.

At resonance, the servopump, which acts as the forcing function, must make up the system losses. A viscous drag coefficient for the load was assumed which would provide a viscous drag force at maximum velocity equal to 5% of the maximum accelerating force. At resonance the pump must make up the losses due to this damping force plus system leakages. Away from resonance, energy conservation is no longer optimum, so that at intermediate frequencies the pump must supply a greater amount of energy to the system than at resonance. Some additional power could be saved by retuning the hydraulic system for these intermediate frequencies. Since the hydraulic resonance is inversely proportional to the square root of the oil volume under compression, it is theoretically possible to retune the system by adding to the oil volume. The hydraulic system may also be tuned by changing the actuator area since resonant frequency is proportional to the cylinder area. This method would require the use of multiple cylinders. Because either a large volume of oil or multiple actuators would be required to obtain a reasonable power saving at intermediate frequencies, the proposed system is tuned only to the maximum drive frequency and additional power is added at intermediate frequencies.

SYSTEM OPERATION (CONTINUED)

THE PUMP OUTPUT IS CONTROLLED BY A POSITIONAL SIGNAL FROM THE LOAD. THIS POSITIONAL SIGNAL IS MIXED WITH THE OUTPUT OF A SIGNAL GENERATOR WHICH ACTS AS THE REFERENCE SOURCE FOR FREQUENCY AND AMPLITUDE. SINCE A LIGHTLY DAMPED SERVO SYSTEM IS DIFFICULT TO CONTROL AT ITS RESONANT POINT, A SECONDARY FEEDBACK OF ACCELERATION IS PROPOSED TO PERMIT MORE IDEAL CLOSED LOOP SYSTEM CHARACTERISTICS. FINAL SELECTION OF FEEDBACK COMPONENTS WHICH WOULD BE NEEDED FOR A SATISFACTORILY OPERATING SERVO SYSTEM WOULD REQUIRE A MORE DETAILED SERVO ANALYSIS AND WOULD HAVE TO INCLUDE LOAD RESONANCE, LOAD DAMPING AND FRICTION, PLUS OTHER NONLINEAR LOAD AND ACTUATOR CHARACTERISTICS.

SPECIFICATIONS OF FLUID STORAGE SYSTEM

A. LOAD FREQUENCY: ±5% OF SET FREQUENCY.

B. FREQUENCY ADJUSTMENT: BY ELECTRICALLY CHANGING THE FORCING FREQUENCY FROM 0.2 TO 0.84 CPS. HYDRAULIC SYSTEM IS TUNED TO RESONANT AT 0.84 CPS.

C. AMPLITUDE TOLERANCE: ±5%.

D. AMPLITUDE ADJUSTMENT: ELECTRIC ADJUSTMENT OF INPUT SIGNAL.

E. PHASE ADJUSTMENT BETWEEN AXES: ELECTRICAL ADJUSTMENT OF PHASE BETWEEN THE INPUT SIGNALS.

F. COMMAND SIGNAL: BOTH AXES FROM SAME SIGNAL SOURCE WITH PROVISION FOR PHASE ADJUSTMENT BETWEEN THE TWO SIGNALS.

G. POWER AMPLIFICATION: ABOUT 10 TIMES AT RESONANCE (0.84 CPS) TO UNITY AT 0.2 CPS.

H. TIME TO REACH MAXIMUM AMPLITUDE: ABOUT .5 MINUTES.

J. OPERATING FLUID TEMPERATURE: 125° ±25°F.

K. SPACE REQUIREMENTS: APPROXIMATELY 13.5 INCHES X 13.5 INCHES X 73 INCHES FOR 7 INCH CYLINDER ON AXES 8 Y WITH TRUNNION MOUNT, PLUS ADDITIONAL 60 INCH SPACE REQUIRED LENGTHWISE TO ALLOW EXTENSION OF DOUBLE ROD. PROPORTIONALLY SMALLER SPACE REQUIRED ON OTHER AXES.

L. VISCOUS FRICTION: ASSUMED TO BE OF MAXIMUM FORCE AT MAXIMUM VELOCITY.

ADVANTAGES AND DISADVANTAGES OF THE FLUID STORAGE SYSTEM

ADVANTAGES:

1. POWER CONSERVATION IS OBTAINED BY UTILIZING THE COMPRESSIBILITY CHARACTERISTIC OF THE HYDRAULIC FLUID.
2. SERVOPUMP CONTROL ELIMINATES POWER LOSS DUE TO PRESSURE DROP THROUGH METERING VALVES (SERVO VALVES).
3. NO EXTERNAL SPRINGS OR PRECHARGED ACCUMULATORS ARE REQUIRED.
4. EFFECTIVE CHANGE IN SPRING RATE WITH PRESSURE IS MUCH LESS WITH HYDRAULIC FLUID AS COMPARED TO A GAS ACCUMULATOR. A NONLINEAR SPRING RATE CAN RESULT IN HARMONICS IN THE OUTPUT AND ADDITIONAL POWER LOSSES.
5. A SERVOPUMP HAS INHERENTLY MORE LINEAR OPERATING CHARACTERISTICS THAN A STANDARD SERVO VALVE SYSTEM (EXCEPT WHEN A CONSTANT GAIN SERVO VALVE IS USED).

DISADVANTAGES:

1. A LARGE VOLUME OF LFUID IS REQUIRED TO OBTAIN RESONANCE AT THE OPERATING FREQUENCY.
2. THERE IS A SPRING RATE CHANGE WITH PRESSURE AND TEMPERATURE. THIS WOULD CAUSE THE RESONANT FREQUENCY TO CHANGE BY ABOUT 10% FOR 0-5000 PSI OPERATION AND ABOUT 10% FOR A ±25° TEMPERATURE CHANGE. (THIS IS ALSO TRUE OF AN AIR ACCUMULATOR TO A MUCH GREATER EXTENT.)

ADVANTAGES AND DISADVANTAGES OF THE FLUID STORAGE SYSTEM

DISADVANTAGES (CONTINUED)

3. SERVO CONTROL IS CRITICAL SINCE THE CLOSED LOOP CONTAINS A VERY LOW FREQUENCY LIGHTLY DAMPED SECOND ORDER TRANSFER FUNCTION.

4. ADJUSTMENT OF SYSTEM RESONANCE TO LOWER FREQUENCIES FOR GREATER POWER CONSERVATION OVER THE FULL FREQUENCY RANGE WOULD REQUIRE:

   A. A LARGE INCREASE IN OIL VOLUME.

   B. USE OF MULTIPLE CYLINDERS.

5. A DOUBLE ENDED CYLINDER MUST BE USED WITH SERVOPUMP CONTROL.

COMPONENTS - AXIS DRIVE UNIT - 2 SETS REQUIRED

(E) SERVO PUMP - PVA120 - ASSEMBLY — 1 REQ.
TO INCLUDE:

1 PVA120-RD-10 PUMP

1 CAP-DR-10 E.H. CONTROL

2 FL2-120-288WL-10 FLANGES

(F) SERVO PUMP - PVA120 - ASSEMBLY — 1 REQ.
TO INCLUDE:

1 PVA120-LD-10 PUMP

1 CAP-DR-10 E.H. CONTROL

2 FL2-120-288WL-10 FLANGES

1 249710 AUX. PUMP MTG. KIT

1 V4230-26-5-1C-10-LH-S214

(G) ELECTRIC MOTOR AND MOTOR STARTER — 1 REQ.
3 Ø, 440 VAC, 200 HP, 1800 RPM
OPEN, DRIPPROOF

(H) RELIEF VALVE - CF-16-H — 2 REQ.
FLANGES FL-16-PS-20 — 6 REQ.

(J) CHECK VALVES DF10P1-24-5-20 — 4 REQ.
FLANGES FL-20-PS-20 — 8 REQ.

(K) RELIEF VALVE CT-10-B-10 — 2 REQ.

(L) OIL COOLER OCW-6-10 — 1 REQ.
TEMP. CONTROL VALVE OCRV-1-06-10 — 1 REQ.
WATER STRAINER OCST-1-06-10 — 1 REQ.
BULB WELL OCBW-1-10 — 1 REQ.

(M) CHECK VALVE C3-825-S3 — 1 REQ.

(N) CHECK VALVE C2-825 — 1 REQ.

(P) PUMP INLET FILTER, 505-238-M-3-P1-41 — 1 REQ.

(R) RESERVOIR, AND RELATED EQUIPMENT, 60 GAL — 1 REQ.

(AA)	SERVO AMPLIFIER AND POWER SUPPLY EAPS-A-11	2 REQ.
*(BB)	FUNCTION GENERATOR - GENERAL RADIO 1305 AR	1 REQ.
*(CC)	POWER SUPPLY FOR YOKE POTENTIOMETER ELCOR AT100-20	2 REQ.
*(DD)	PREAMPLIFIER AND POWER SUPPLY MICRO GEE PRODUCTS MODEL 505AS	1 REQ.

MISCELLANEOUS ELECTRONIC EQUIPMENT

(*PREAMPLIFIER AND POWER SUPPLY, Y AXIS ONLY, 1 REQ.)

COMPONENTS - ACTUATOR UNITS

(A) CYLINDER, DOUBLE ROD,

2X 5 IN. CYL, 3.5 IN ROD, 13 IN. STROKE
2Y 5 IN. CYL, 3.5 IN ROD, 13 IN. STROKE

3X 5 IN CYL, 3.5 IN ROD, 15 IN. STROKE
3Y 5 IN CYL, 3.5 IN ROD, 15 IN. STROKE

4X 5 IN CYL., 2.5 IN ROD, 18 IN STROKE
4Y 6 IN CYL., 4.0 IN ROD, 18 IN STROKE

5X 5 IN CYL, 2.5 IN ROD, 30 IN STROKE
5Y 6 IN CYL, 4.0 IN ROD, 30 IN STROKE

6X 5 IN CYL, 2.5 IN ROD, 35 IN STROKE
6Y 7 IN CYL, 5.0 IN ROD, 35 IN STROKE

7X 5 IN CYL, 2.5 IN ROD, 47 IN STROKE
7Y 7 IN CYL, 5.0 IN ROD, 47 IN STROKE

8X 5 IN CYL, 2.5 IN ROD, 60 IN STROKE
8Y 7 IN CYL, 5.0 IN ROD, 60 IN STROKE

9X 5 IN CYL, 3.5 IN ROD, 63 IN STROKE
9Y 5 IN CYL, 2.5 IN ROD, 63 IN STROKE

(B) GLOBE VALVE, 5000 PSI

1.5 INCH	4 REQ.
2.0 INCH	2 REQ.
2.5 INCH	6 REQ.
3.0 INCH	4 REQ.
4.0 INCH	12 REQ.
5.0 INCH	4 REQ.

(C) POTENTIOMETER - LOAD POSITION 16 REQ.

(D) ACCELEROMETER 16 REQ.

FEEDBACK POTENTIOMETER GEARING 16 REQ.

CONVENTIONAL

System Description

THE BASIC SYSTEM CONSISTS OF FIXED DISPLACEMENT VANE PUMPS WHICH ACT AS THE HYDRAULIC POWER SOURCE. THE POWER INTO THE LOAD IS CONTROLLED BY A CLOSED LOOP POSITIONAL SYSTEM USING A SERVO VALVE AND CYLINDER. A POTENTIOMETER ON THE CYLINDER IS USED TO FEED BACK LOAD POSITION. THE COMMAND FOR LOAD POSITION IS PROVIDED BY A FUNCTION GENERATOR WHICH HAS TWO OUTPUTS ADJUSTABLE IN PHASE RELATIVE TO ONE ANOTHER. SPECIAL HIGH FLOW SERVO VALVES ARE USED ON THE HIGHER FLOW AXES.

FOUR BASIC POWER UNITS ARE PROVIDED. TWO HAVE A HIGH FLOW OUTPUT AND ARE USED TO SUPPLY THE X AND Y AXES OF STATIONS 5 THROUGH 9. TWO HAVE A LOWER OUTPUT AND ARE USED TO SUPPLY STATIONS 2 THROUGH 4.

TO OBTAIN A MINIMUM AVERAGE POWER INPUT AND PROVIDE GREATER CONTROL SYSTEM STABILITY, A PRESSURE DEMAND TYPE OF SERVO VALVE SUPPLY IS USED. WITH THIS SYSTEM, A CONSTANT PRESSURE DROP IS MAINTAINED ACROSS THE SERVO VALVE. THIS DROP IS SET AT THE MINIMUM PRESSURE AT WHICH THE VALVE WILL PASS THE PEAK FLOW. SERVO VALVE SUPPLY PRESSURE IS THUS MAINTAINED AT LOAD PRESSURE PLUS THE FIXED VALVE DROP. WITH THIS SYSTEM, THE SERVO VALVE OPERATES AT CONSTANT GAIN AND PROVIDES ESSENTIALLY CONSTANT HYDRAULIC DAMPING FOR BETTER CONTROL SYSTEM CHARACTERISTICS AND CONSEQUENTLY THE HYDRAULIC SYSTEM CONTRIBUTES LESS TO THE HARMONIC CONTENT OF THE OUTPUT. SINCE THE SERVO VALVE SUPPLY PRESSURE IS ADJUSTED BY LOAD DEMAND, ONLY ONE DRIVE CAN BE OPERATED BY A SINGLE HYDRAULIC SUPPLY. WITH THE PROPOSED SYSTEM, ONE HIGH FLOW STATION (5-9) AND ONE LOW FLOW STATION (2-4) CAN BE OPERATED AT THE SAME TIME. WITH ADDITIONAL VALVING AND PIPING TO PROVIDE A FLEXIBLE

SELECTION OF SUPPLY PUMPS FOR EACH DRIVE, IT WOULD BE POSSIBLE TO DRIVE MORE THAN TWO DRIVES AT ONE TIME, PROVIDING THAT THE TOTAL FLOW CAPACITY WERE WITHIN THE CAPABILITY OF THE POWER UNITS.

SYSTEM OPERATION

THE CONTROL SYSTEM USES A FREQUENCY GENERATOR AS THE COMMAND DEVICE. THIS COMMAND IS COMPARED TO THE SIGNAL FROM A FEEDBACK POTENTIOMETER. THE RESULTANT ERROR SIGNAL IS AMPLIFIED AND POSITIONS THE SERVO VALVE SPOOL TO CONTROL THE FLOW INTO THE LOAD CYLINDER. THE LOAD IS THUS POSITIONED IN RESPONSE TO THE COMMAND VOLTAGE FROM THE GENERATOR. THE GENERATOR HAS A SECONDARY OUTPUT WHICH IS ADJUSTABLE IN PHASE RELATIVE TO THE PRIMARY OUTPUT. THIS IS USED TO COMMAND THE SECOND AXIS.

THE POWER UNITS WHICH SUPPLY FLOW FOR THE SERVO VALVES CONSIST OF A NUMBER OF HIGH FLOW, HIGH PRESSURE VANE PUMPS. THE NUMBER OF PUMPS WHICH WOULD BE TURNED ON TO SUPPLY A PARTICULAR DRIVE WOULD BE DETERMINED BY THE PEAK FLOW REQUIREMENTS OF THE DRIVE BEING CONTROLLED. CHECK VALVES IN THE OUTPUT LINES OF EACH PUMP PREVENT REVERSE FLOW THROUGH THE PUMP WHEN THE PARTICULAR PUMP IS NOT IN OPERATION.

SYSTEM PRESSURE IS SET BY THE PRESSURE DEMAND VALVE WHICH SENSES THE MAXIMUM PRESSURE IN THE LINES TO THE LOAD AND ADJUSTS THE SUPPLY PRESSURE A FIXED AMOUNT ABOVE THE LOAD LINES. THIS MAINTAINS A CONSTANT PRESSURE DROP ACROSS THE SERVO VALVE INDEPENDENT OF THE FLOW THROUGH IT. THERE ARE THREE MAIN ADVANTAGES TO USING A PRESSURE DEMAND SYSTEM.

1. IT PERMITS A REDUCTION IN AVERAGE INPUT POWER WITH FIXED DISPLACEMENT SUPPLY PUMPS SINCE SUPPLY PRESSURE IS NOT FIXED AT A HIGH LEVEL.
2. THE AVERAGE PRESSURE SEEN BY THE SUPPLY PUMPS IS LOWER GIVING LONGER PUMP LIFE.

SYSTEM OPERATION (CONTINUED)

3. SINCE A CONSTANT PRESSURE DROP IS MAINTAINED ACROSS THE SERVO VALVE, IT HAS A CONSTANT GAIN OVER VARYING FLOW AND PRESSURE CONDITIONS AND HYDRAULIC DAMPING IS ESSENTIALLY CONSTANT PROVIDING BETTER CONDITIONS FOR SERVO CONTROL AND CONTRIBUTING TO A MORE IDEAL SINE WAVE OUTPUT.

ADVANTAGES AND DISADVANTAGES OF THE CONVENTIONAL HYDRAULIC SYSTEM

ADVANTAGES:

1. BETTER CONTROL CONDITIONS ARE OBTAINED WITH A CONVENTIONAL SYSTEM. SINCE THE SYSTEMS WHICH INCORPORATE ENERGY STORAGE DEPEND ON OPERATING AT OR NEAR THE SYSTEM RESONANT POINT, THEY HAVE THE PROBLEMS ASSOCIATED WITH THE CONTROL OF A RESONANT CIRCUIT. THE HIGHER INHERENT RESONANT FREQUENCY AND THE CONSTANT GAIN CHARACTERISTICS OF THE PRESSURE DEMAND SYSTEM USED MAKE THE CONVENTIONAL SYSTEM MORE EASILY CONTROLLED.
2. BECAUSE OF THE BETTER CONTROL CHARACTERISTICS OF THE CONVENTIONAL SYSTEM, THE ACCURACY OF AMPLITUDE CONTROL WILL BE BETTER THAN THE RESONANT SYSTEMS.
3. HARMONIC CONTENT OF THE OUTPUT WILL LIKELY BE LOWER BECAUSE OF THE MORE LINEAR CHARACTERISTICS OF THE CONVENTIONAL SYSTEM.
4. AMPLITUDE AND FREQUENCY ARE EASILY ADJUSTED BY CHANGING THE INPUT COMMAND.

DISADVANTAGES:

1. SINCE THERE IS NO RETURN OF POWER TO THE SYSTEM DURING DECELERATION OF THE LOAD, A LARGE AMOUNT OF INPUT POWER IS REQUIRED.

DISADVANTAGES (CONTINUED)

2. RELATIVELY COMPLEX PIPING AND VALVING IS REQUIRED IF MORE THAN TWO DRIVES ARE TO BE RUN AT ONE TIME.

COMPONENTS - POWER SUPPLY

1. POWER SUPPLY, HIGH FLOW CONSISTING OF: 1 REQ.

2 RESERVOIRS, SPECIAL 500 GAL, WITH ACCESSORIES

B 2 OIL COOLERS AND ACCESSORIES 1400 GPM CAPACITY, 400 HP OF HEAT AT MAXIMUM CAPACITY

C 2 FILTERS, RETURN LINE, 1400 GPM CAPACITY 25 MICRON

D PUMP MODULES CONSISTING OF:

D 1 ELECTRIC MOTOR, DOUBLE ENDED 400 HP, 1000 RPM, 3 Ø, 440 VAC

2 VANE PUMPS 50V109A-1010 (ONE LEFT HAND AND ONE RIGHT HAND)

F 2 FILTERS, 150 GPM, 10 MICRON, 3000 PSI

G 2 CHECK VALVES, DF10P1-24-5 WITH TWO FL-24-PC FLANGES

1 PUMP MODULE CONSISTING OF:

H 1 ELECTRIC MOTOR, 200 HP, 1000 RPM, 3Ø, 440 VAC,

J 1 VANE PUMP 50V109A-1010

K 1 FILTER, 150 GPM, 10 MICRON, 3000 PSI

L 1 CHECK VALVE, DF10P1-24-5 WITH TWO FL-24-PC FLANGES

2 PUMP MODULES CONSISTING OF:

M 1 ELECTRIC MOTOR, 10 HP, 1000 RPM, 3Ø, 440 VAC,

N 1 PUMP, PRESSURE COMPENSATED PV820-FPL-10-C

1 FILTER, 24 GPM, 10 MICRON EUROLATOR F-424, 3000 PSI

1 RELIEF VALVE, DF-06-C

1 FILTER PUMP INLET, 200-149-4-3-F1-41

MISCELLANEOUS PARTS:

10 INLET STRAINERS, F10-12-10 (5 IN EACH RESERVOIR)

POWER SUPPLY (CONTINUED)

- 10 CHECK VALVES, OIL COOLER BYPASS, [illegible]-24-5-63 WITH TWO FL-24 FL FLANGES
- [illegible] RELIEF VALVES, [illegible]-24-[illegible]-10 EACH WITH THREE FL-24-[illegible] FLANGES

11. POWER SUPPLY, LOW FLOW CONSISTING OF: 1 REQ.

- 2 RESERVOIRS, 60 GALLONS, WITH ACCESSORIES
- 2 OIL COOLERS, [illegible]-6-10 AND ACCESSORIES
- 2 FILTERS, RETURN LINE, [illegible]-202

1 PUMP MODULE CONSISTING OF:

- 1 ELECTRIC MOTOR, DOUBLE END 75 HP, 1800 RPM, 3Ø, 440 [illegible]
- 1 VANE PUMP 25V14-[illegible]
- 1 VANE PUMP 25V 21-[illegible]10L
- 3 FILTERS, 10 MICRON, 3000 PSI PUROLATOR PF424
- 1 CHECK VALVE, 02-915
- 1 CHECK VALVE, 02-920

1 PUMP MODULE CONSISTING OF:

- 1 ELECTRIC MOTOR, DOUBLE ENDED 125 HP, 1800 RPM, 3Ø, 440 VAC
- 1 VANE PUMP, 35V 24-[illegible]10
- 1 VANE PUMP, 35V 3[illegible]-[illegible]10L
- 4 FILTERS, 10 MICRON, 3000 PSI PUROLATOR PF424
- 2 CHECK VALVES, 02-925

2 PUMP MODULES CONSISTING OF:

- 1 ELECTRIC MOTOR, 5 HP, 1800 RPM, 3Ø, 440 VAC,
- 1 PUMP PRESSURE COMPENSATED PV810-[illegible]-10-C
- 1 FILTER, 12 GPM, 10 MICRON PUROLATOR FF412, 3000 PSI

COMPONENTS - POWER SUPPLY (CONTINUED)

AN 1 RELIEF VALVE, CT-06-C

BU 1 FILTER PUMP INLET 10S-149-M-3-P1-50

MISCELLANEOUS PARTS:

AP 2 INLET STRAINERS CF3-12-10

AQ 1 CHECK VALVE, OIL COOLER, BYPASS, C2-825

AR 1 CHECK VALVE, OIL COOLER, BYPASS, C2-830

AS 2 RELIEF VALVES, DT-10-F

ACTUATOR COMPONENTS

CYLINDERS, DOUBLE ROD, FOR 3000 PSI AND HIGH SPEED OPERATION, 1 EACH OF THE FOLLOWING:

AT	2X	2 IN CYL, 1 IN ROD, 13 IN STROKE	26 IN/SEC
AU	2Y	2.5 IN CYL, 1.375 IN ROD, 13 IN STROKE	
AV	3X	2 IN CYL, 1 IN ROD, 15 IN STROKE	
AW	3Y	3.25 IN CYL, 2 IN ROD, 15 IN STROKE	33 IN/SEC
AX	4X	3.25 IN CYL, 2 IN ROD, 18 IN STROKE	
AY	4Y	4 IN CYL, 2.5 IN ROD, 18 IN STROKE	41 IN/SEC
AZ	5X	4 IN CYL, 2 IN ROD, 30 IN STROKE	
BA	5Y	5 IN CYL, 2.5 IN ROD, 30 IN STROKE	73 IN/SEC
BB	6X	5 IN CYL, 2.5 IN ROD, 35 IN STROKE	
BC	6Y	6 IN CYL, 2.5 IN ROD, 35 IN STROKE	87 IN/SEC
BD	7X	6 IN CYL, 3.5 IN ROD, 47 IN STROKE	
BE	7Y	7 IN CYL, 3 IN ROD, 47 IN STROKE	118 IN/SEC
BF	8X	7 IN CYL, 4 IN ROD, 60 IN STROKE	
BG	8Y	8 IN CYL, 3.5 IN ROD, 60 IN STROKE	152 IN/SEC
BH	9X	7 IN CYL, 5 IN ROD, 63 IN STROKE	
BJ	9Y	6 IN CYL, 2.5 IN ROD, 63 IN STROKE	159 IN/SEC

BK 3 SERVO VALVES, SA4-06-**-220-10 TWO STAGE

BL 5 SERVO VALVES, SPECIAL, 3 STAGE WITH SA4-03 FIRST AND SECOND STAGE AND 2 INCH DIAMETER 3RD STAGE

BM 8 SERVO VALVES, SPECIAL, 3 STAGE WITH SA4-03 FIRST AND SECOND STAGE AND 4 INCH DIAMETER 3RD STAGE

BN 6 PRESSURE DEMAND CONTROL VALVES CAPACITY TO 70 GPM

BP 3 PRESSURE DEMAND CONTROL VALVES CAPACITY TO 340 GPM

BQ 7 PRESSURE DEMAND CONTROL VALVES CAPACITY TO 1400 GPM

BR 16 SHUTTLE VALVES, SPECIAL

BS 16 FILTERS, 12 GPM, 10 MICRON PUROLATOR PR412, 3000 PSI

BT 16 ACCUMULATORS, AI-75-10

ACTUATOR COMPONENTS (CONTINUED)

GLOBE VALVES, 3000 PSI, AS FOLLOWS:

BU	4	1.5 INCH
BV	2	2.0 INCH
BW	6	2.5 INCH
BX	4	3.0 INCH
BY	12	4.0 INCH
BZ	4	5.0 INCH

ELECTRICAL COMPONENTS

4 SERVO AMPLIFIERS, EAPS-A-11

2 FUNCTION GENERATORS, GENERAL RADIO 1305AR

16 POTENTIOMETERS, FEEDBACK, LOAD POSITION

16 GEARING, FEEDBACK POTENTIOMETER

16 ACCELEROMETERS

2 PREAMPLIFIER AND POWER SUPPLY MICRO GEE PRODUCTS MODEL 50546

MISCELLANEOUS ELECTRICAL EQUIPMENT

MOTOR STARTERS AC COMBINATION MAGNETIC STARTERS LINE VOLTAGE TYPE, CIRCUIT BREAKER TYPE, BIMETALLIC OVERLOAD RELAYS, NEMA TYPE 1, 440 VAC

3 NEMA SIZE 6 FOR 400 HP MOTOR

1 NEMA SIZE 5 FOR 200 HP MOTOR

1 NEMA SIZE 5 FOR 125 HP MOTOR

1 NEMA SIZE 4 FOR 75 HP MOTOR

1 NEMA SIZE 1 FOR 10 HP MOTOR

1 NEMA SIZE 0 FOR 5 HP MOTOR

SPECIFICATIONS OF THE CONVENTIONAL SYSTEM

- LOAD FREQUENCY: ±5% OF SET FREQUENCY.
- FREQUENCY ADJUSTMENT: BY ELECTRICALLY CHANGING THE COMMAND FREQUENCY FROM 0.2 TO 2.84 CPS.
- AMPLITUDE TOLERANCE: ±5%.
- AMPLITUDE ADJUSTMENT: ELECTRIC ADJUSTMENT OF INPUT SIGNAL.
- PHASE ADJUSTMENT BETWEEN AXES: ELECTRICAL ADJUSTMENT OF PHASE BETWEEN THE INPUT SIGNALS.
- COMMAND SIGNAL: BOTH AXES FROM SAME SIGNAL SOURCE WITH PROVISION FOR PHASE ADJUSTMENT BETWEEN THE TWO SIGNALS.
- POWER AMPLIFICATION: ABOUT 0.8 AT MAXIMUM POWER WITH 500 PSI VALVE DROP.
- TIME TO REACH MAXIMUM AMPLITUDE: ABOUT ONE CYCLE.
- OPERATING FLUID TEMPERATURE: 125° ±25°F.
- SPACE REQUIREMENTS: APPROXIMATELY 30 INCHES X 14.5 INCHES X 75 INCHES FOR CYLINDER AND VALVES ON AXES X Y WITH TRUNNION MOUNT, PLUS ADDITIONAL 60 INCH SPACE REQUIRED LENGTHWISE TO ALLOW EXTENSION OF DOUBLE ROD. PROPORTIONALLY SMALLER SPACE REQUIRED ON OTHER AXES.
- VISCOUS FRICTION: ASSUMED TO BE OF MAXIMUM FORCE AT MAXIMUM VELOCITY.

SECTION IV COMPARISON OF THREE SYSTEMS

Summary of Conventional and Energy Storage Systems

THREE HYDRAULIC DRIVE SYSTEMS ARE ANALYSED IN THIS REPORT: (1) CONVENTIONAL SERVO VALVE-CYLINDER POSITIONING SYSTEM WITH PRESSURE DEMAND, (2) ENERGY STORAGE SYSTEM USING MECHANICAL SPRINGS WITH SERVO VALVE-CYLINDER CONTROL, AND (3) ENERGY STORAGE SYSTEM USING FLUID COMPRESSIBILITY WITH SERVO PUMP-CYLINDER CONTROL. THE SYSTEMS WHICH USE ENERGY STORAGE ABSORB ENERGY DURING DECELERATION OF THE LOAD AND RETURN ENERGY TO THE LOAD DURING ACCELERATION.

IN ADDITION, A FOURTH SYSTEM USING AIR-OIL ACCUMULATORS FOR ENERGY STORAGE WAS INVESTIGATED. THIS SYSTEM WAS RULED OUT BECAUSE OF THE EXCESSIVE SPRING RATE CHANGE OF THE SYSTEM OVER THE OPERATING CYCLE. THIS EFFECTIVE SPRING RATE CHANGE WOULD NULLIFY MUCH OF THE ANTICIPATED POWER SAVING AND ADD TO THE HARMONIC CONTENT OF THE OUTPUT.

THE CONVENTIONAL SERVO VALVE-CYLINDER POSITIONING SYSTEM USES A SERVO VALVE TO POSITION THE LOAD CYLINDER. A THIRD STAGE IS USED ON THE SERVO VALVE TO MEET THE FLOW DEMAND ON AXES WHERE THE FLOW REQUIREMENT IS HIGH. TWO DIFFERENT SIZE THIRD STAGES ARE PROPOSED TO MEET THE RANGE OF FLOW DEMANDS OF THE VARIOUS DRIVES. SYSTEM FLOW IS SUPPLIED BY A NUMBER OF FIXED DISPLACEMENT VANE PUMPS. THE NUMBER OF PUMPS IN USE IS DETERMINED BY THE FLOW REQUIREMENTS OF THE DRIVES BEING OPERATED.

THE MECHANICAL SPRING SYSTEM USES MECHANICAL SPRINGS WITH THE SPRING RATE SET SO THAT THE SPRING MASS RESONANT FREQUENCY OF THE SYSTEM OCCURS AT THE DESIRED OSCILLATION FREQUENCY. THE SERVO VALVE CYLINDER POSITIONAL CONTROL IN THIS CASE ACTS AS THE FORCING FUNCTION

COMPARISON CHART FOR THREE SYSTEMS

SYSTEM	CONVENTIONAL	MECHANICAL SPRING STORAGE	FLUID STORAGE
COST, TOTAL	HIGHEST	MEDIUM	LOWEST
PERFORMANCE	GOOD	FAIR	FAIR
EASE OF CHANGING FREQUENCY	GOOD	POOR	FAIR
EASE OF CHANGING AMPLITUDE	GOOD	GOOD	GOOD
FREQUENCY ACCURACY	GOOD	GOOD	GOOD
AMPLITUDE ACCURACY	GOOD	FAIR	FAIR
HARMONIC CONTENT	LOW	MEDIUM	MEDIUM
TIME TO COMPLETE	12 MONTHS	13 MONTHS	11 MONTHS
PROBABILITY OF SUCCESS	GOOD	FAIR	FAIR

Cost Comparison

Costs shown here are approximate for comparison purposes only. Costs for pipe, fittings, and accessory hardware is not included. A summary of relative costs for the three systems is outlined here:

A. Conventional System

Components	$300,000
Design engineering	$153,000
Set up and field engineering	$181,000
Total	$634,000

B. Spring Storage System

Components	$ 90,000
Design engineering	$178,000
Set up and field engineering	$208,000
Total	$476,000

C. Fluid Storage System

Components	$125,000
Design engineering	$156,000
Set up and field engineering	$156,000
Total	$437,000

FOR THE SYSTEM AND IS REQUIRED TO MAKE UP THE MECHANICAL LOSSES OF THE SYSTEM. THE INPUT POWER REQUIREMENT OF THE SYSTEM WILL INCREASE WHEN IT IS NOT OSCILLATING AT ITS NATURAL FREQUENCY. FOR THIS REASON, THE NATURAL FREQUENCY OF THE SYSTEM IS MATCHED TO THE DESIRED OSCILLATING FREQUENCY OF THE LOAD BY CHANGING THE NUMBER OF SPRINGS USED.

THE FLUID STORAGE SYSTEM IS ANALOGOUS TO THE MECHANICAL SPRING SYSTEM, EXCEPT THAT ENERGY IS STORED IN THE COMPRESSED FLUID INSTEAD OF IN THE MECHANICAL SPRINGS. WITH THIS SYSTEM, THE HYDRAULIC NATURAL FREQUENCY IS SET AT 0.84 CPS WHERE THE MAXIMUM SYSTEM POWER REQUIREMENT OCCURS. THIS PROVIDES MAXIMUM POWER GAIN AT THE MAXIMUM POWER POINT. IT WAS NOT CONVENIENT IN THIS CASE TO RETUNE THE SYSTEM FOR LOWER OSCILLATING FREQUENCIES BECAUSE OF THE EXCESSIVE OIL VOLUMES OR MULTIPLE CYLINDERS WHICH WOULD BE REQUIRED. THEREFORE, AT LOWER FREQUENCIES WHERE POWER REQUIREMENTS ARE NOT AS HIGH, OPTIMUM POWER GAIN IS NOT UTILIZED.

Performance Comparison

1. Conventional System

All three systems are essentially positional control systems. The conventional system will have the most absolute control of system performance. Its resonant frequency will be well above the frequency at which the system is being oscillated resulting in much better control system characteristics than the two energy storage systems. Also, the full power required by the load is available from the control system. As a result, the conventional system can be expected to have the minimum transient time on startup and have the best control of oscillation amplitude. It will also be less effected by changes in system parameters due to temperature changes and changes in friction, etc.

2. Spring Storage System

The spring storage system depends on obtaining a system resonant frequency close to the frequency of oscillation of the load in order to provide power amplification. The control system must therefore provide control of a system which is oscillating at its natural frequency. The servo problems associated with this type of control tend to limit the system sensitivity which can be obtained. As a result, the control of the oscillation amplitude will be more effected by drifts and parameter changes than the conventional system. Also, because of the limited power available from the control, several cycles will be required on initial startup to bring the system up to peak amplitude. The conservation of energy then is obtained at the expense of reduced accuracy of control and system responsiveness.

3. Fluid Storage System

The fluid storage system like the spring storage system depends

ON OPERATION NEAR SYSTEM RESONANCE TO OBTAIN POWER AMPLIFICATION. IN THIS CASE, THE COMPRESSIBILITY OF THE HYDRAULIC FLUID IS USED INSTEAD OF A SPRING. CONTROL PROBLEMS RESULTING FROM OPERATION OF THE SYSTEM AT ITS RESONANT FREQUENCY APPLY TO THE FLUID STORAGE SYSTEM AS WELL AS THE SPRING STORAGE SYSTEM. WITH THE FLUID STORAGE SYSTEM PROPOSED, THE RESONANT FREQUENCY IS TUNED ONLY FOR THE MAXIMUM OPERATING FREQUENCY AND THEREFORE PROVIDES LESS POWER CONSERVATION AT INTERMEDIATE FREQUENCIES THAN THE SPRING STORAGE SYSTEM. LIKE THE SPRING STORAGE SYSTEM, ACCURACY OF AMPLITUDE CONTROL AND SYSTEM RESPONSIVENESS IS LESS THAN FOR THE CONVENTIONAL SYSTEM.

Frequency Stability and Relative Ease of Changing Frequency

1. Conventional System

With the conventional system, it is only necessary to change the frequency setting of the function generator command signal to change system frequency. Since the system follows the function generator signal, the accuracy of frequency control is the same as the signal generator (±3%). Harmonic content of the signal generator is about ±3%. To minimize the total harmonic content in the output, a pressure demand system is used with the servo valve. This provides more linear operating characteristics to the servo valve and tends to reduce any harmonics which might be generated in the load motion due to the servo valve.

2. Spring Storage System

Since a spring mass system in steady state oscillation oscillates at the frequency of its forcing function, accuracy of frequency control will again depend on the accuracy of the function generator.

In addition to changing the frequency of the command voltage, to change the frequency of oscillation of this system requires that the natural frequency of the system be changed. This is done by removing or adding springs, so that the system natural frequency equals the desired oscillating frequency. The ease of frequency adjustment then depends on the ease with which the mechanical springs can be removed or replaced. Harmonic content in the output depends on nonlinearities in the system such as servo valve nonlinearity and the change of spring rate with deflection and temperature.

3. Fluid Storage System

The accuracy of steady state control of frequency, like the spring storage system, depends on the accuracy of the function generator. This

FREQUENCY STABILITY AND RELATIVE EASE OF CHANGING FREQUENCY

FLUID STORAGE SYSTEM (CONTINUED)

SYSTEM SACRIFICES SOME POWER AMPLIFICATION BY NOT RE-ADJUSTING THE SYSTEM NATURAL FREQUENCY TO LOWER THAN 0.84 CPS. FREQUENCY CHANGES ARE THEN MADE ONLY BY ADJUSTMENT OF THE COMMAND VOLTAGE FROM THE FUNCTION GENERATOR. ACCELERATION FEEDBACK IS PROPOSED TO FLATTEN THE CLOSED LOOP FREQUENCY RESPONSE CURVE AND MINIMIZE THE NEED FOR AMPLITUDE RE-ADJUSTMENT WHEN FREQUENCY IS CHANGED. HARMONIC CONTENT IN THE OUTPUT WILL BE ESSENTIALLY A FUNCTION OF SYSTEM NONLINEARITIES. A SERVOPUMP CONTROL IS PROPOSED FOR THIS SYSTEM RATHER THAN A SERVOVALVE CONTROL. THIS WILL PROVIDE MORE LINEAR OPERATION TO MINIMIZE HARMONICS.

Structural Frequency Requirements

As a general rule, it is desirable to have the lowest system resonance at least three times the maximum desired closed loop frequency response of significant magnitude, and the second lowest system resonance at least three times the lowest system resonance.

On the conventional system, the expected hydraulic resonant frequency is computed to be in the neighborhood of 9-10 cps. This would, according to the rule of thumb require a minimum structural frequency of at least 3 cps.

On the two energy storage systems, the lowest system resonance is at the frequency of oscillation for the mechanical spring system and at 0.84 cps for the fluid storage system. Since this operating mode could be expected to present control problems due to operation at system resonance, it may be well in this case to have the structural resonance well over 3 cps.

SCHEMATIC DIAGRAMS

SCHEMATIC DIAGRAMS FOR THE THREE SYSTEMS ARE SHOWN ON PAGES 40 TO 41.* PARTS LISTS ARE INCLUDED AT THE END OF THE SECTIONS DESCRIBING THE RESPECTIVE SYSTEMS. LINE SIZES ARE NOT SHOWN ON THE DRAWINGS. THE CRITERION FOR MINIMUM LINE SIZE WILL BE 15 FT/SEC FLUID VELOCITY FOR PRESSURE LINES AND 3 FT/SEC. FOR INTAKE LINES. LINE SIZES WILL BE MATCHED TO PORT SIZES ON THE COMPONENTS WHERE PRACTICAL. THE MINIMUM INTERNAL DIAMETER OF THE PIPING WILL BE $d = 0.165\sqrt{GPM}$ FOR PRESSURE LINES AND $d = 0.370\sqrt{GPM}$ FOR INTAKE LINES. LINE SIZES CORRESPONDING TO THE MAIN FLOW ON EACH AXIS ARE TABULATED IN THE **APPENDIX SECTION OF EACH SYSTEM.**

*CONVENTIONAL SYSTEM DRAWING 12433-K IS INCLUDED.

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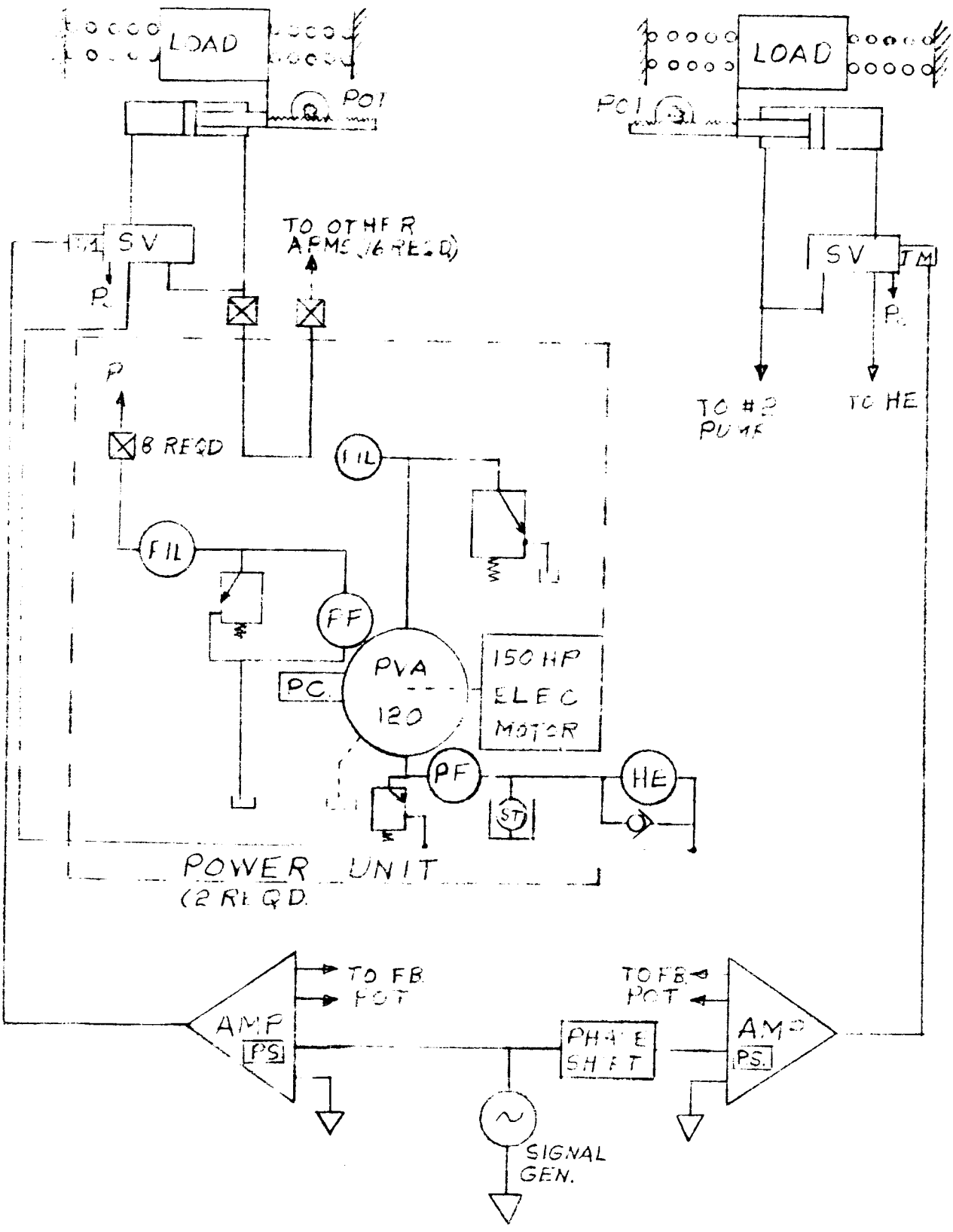
D.L.R	ENGINEERING CALCULATION FORM	PROGRAM & PROJECT NO.
CHECKED BY	SUBJECT: RANDOM MOTION SIMULATOR	8-4801-056-132
		DATE 3-4-63



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7-60

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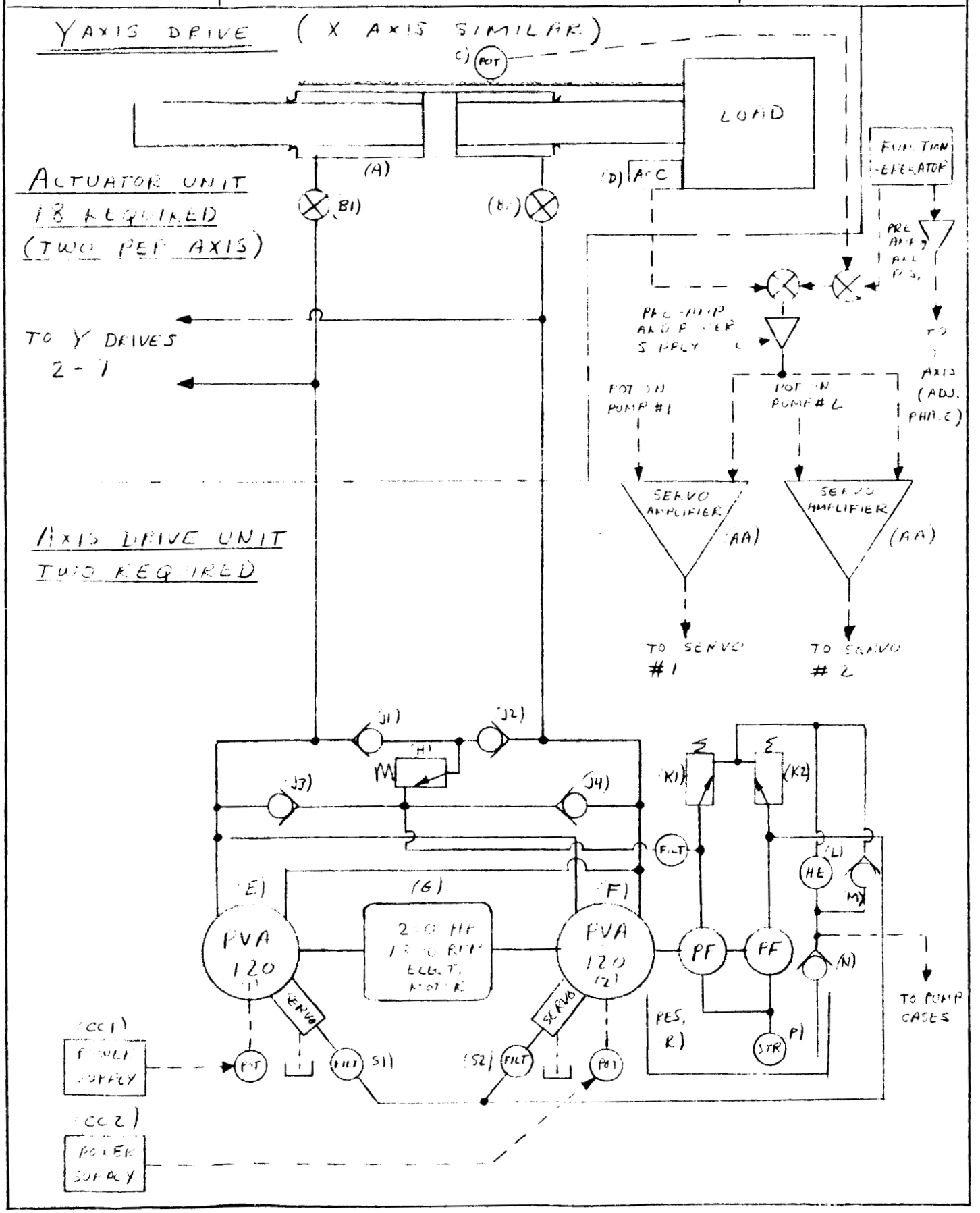
G. McCandless	ENGINEERING CALCULATION FORM	PROGRAM & PROJECT NO.
CHECKED BY	SUBJECT: NASA - Random Motion Simulator	DATE 2-27-63



APPENDIX TO SECTION I

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DLR	ENGINEERING CALCULATION FORM	
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		DATE

Power Required:

1. Loss in Spring due to hysteresis
2. Loss due to Viscous friction

1. Energy Loss in Spring. (.2% Max Hysteresis)

$$\mathcal{E} = \tfrac{1}{2} K x^2 (.002)$$

$$= .001 \times 2942 \times (28.7)^2$$

$$= 2420 \text{ lb-in}$$

$f = .84$ cps $\quad$ $\tfrac{1}{4}$ cycle $= \frac{1}{4 \times .84} = .298$ sec

$$\text{Loss} = \frac{2420 \text{ lb-in}}{.298 \text{ sec}} = 8130 \text{ lb-in/sec}$$

2. Viscous friction - Assumed to be 5% of Max. Force required at Max Lead Velocity.

$$\mathcal{E} = .05 \times 90{,}900 \text{ lb} \times 152.5 \frac{\text{in}}{\text{sec}}$$

$$= 6.93 \times 10^5 \text{ in-lb/sec. (Peak)}$$

Total Loss: $6.93 \times 10^5 + 8 \times 10^3 = 7.01 \times 10^5$ lb-in/sec

$$HP_{MAX} = \frac{7.01 \times 10^5 \frac{\text{in lb}}{\text{sec}}}{6.6 \times 10^3 \frac{\text{in lb}}{\text{sec} \cdot \text{HP}}} = 106 \text{ HP}$$

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		DATE

Cylinder Size: (2:1 Differential Area)

$$HP = \frac{P \times GPM}{1714}$$ Assume $P = 4250$ PSI

$$GPM = \frac{1714\ HP}{P} = \frac{1714 \times 106}{4250} = 42.7\ GPM$$

$$= 42.7 \frac{GAL}{MIN} \times 231 \frac{cu.in}{GAL} \times \frac{1\ MIN}{60\ sec} = 165 \frac{cu.in}{sec}$$

This flow required when $V = 152.5$ in/sec

$$\therefore \text{Area } A = \frac{165 \frac{cu.in}{sec}}{152.5\ in/sec} = 1.08\ in^2$$

Smallest Area in excess of $1.08\ in^2 = 1.66\ in^2$
2 in dia. Cylinder. Head End $= 3.14\ in^2$

$\therefore$ Max flow $= 65.7$ GPM from Pump

$$P = \frac{1714\ HP}{GPM} = \frac{1714 \times 106}{65.7} = 2760\ PSI$$

However, A servo valve drop of 750 PSI on the Head end will be maintained.

This reflects as $750 \times \frac{3.14}{1.66} = 1420$ PSI

$\therefore$ Max Pressure $= 2760 + 1420 = 4180$ PSI.

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$D_{\text{DAMPING factor}} = \frac{90,900 \times .05}{152.5} = 29.8 \ \frac{lb\text{-}sec}{IN}$

$\text{MASS} = \frac{43,500 \ lb}{386 \ in/sec^2} = 112.6 \ \frac{lb\text{-}sec^2}{IN}$

Spring Effect due to oil compression:

With All spring removed, we want $f_N = .2$ cps

or $\omega_n = .2 \times 6.28 = 1.256$ rad/sec

$\omega_n = \sqrt{\frac{K_H}{M}}$

$\therefore K_H = M\omega_n^2 = 112.6 \times 1.58 = 178 \ lb/IN$

$K_H = \frac{A^2}{VE}$ or $V = \frac{A^2}{K_H E} = \frac{3.14^2}{178 \times 5 \times 10^6}$

$V = 1.11 \times 10^4$ cu. in

for .84 cps = 5.27 rad/sec

$K_T = K_{spring} + K_H = M\omega_n^2$

$= 112.6 \times 27.7 = 3120 \ lb/in$

$K_S = 3120 - 178 = 2942 \ lb/in$

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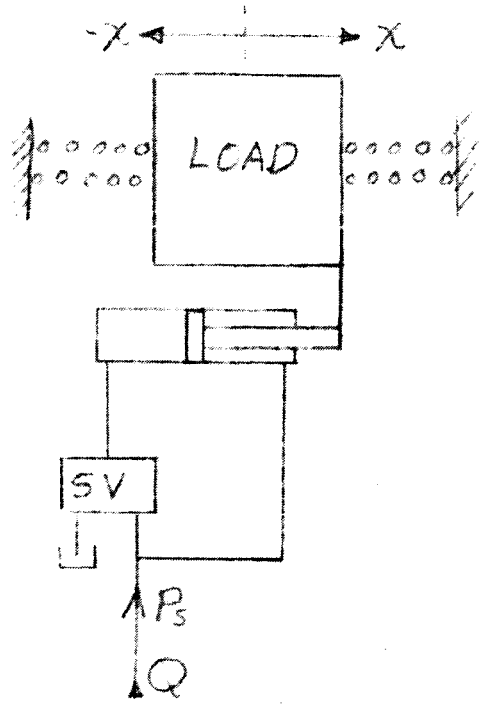
		DATE



Transfer function Derivation:

$$P = \left(\frac{Ms^2 + Ds + K}{A}\right) x(s)$$

$$Q = PL + Asx + VE\frac{dP}{dt}$$

$$\frac{dP}{dt} = s\left[\frac{Ms^2 + Ds + K}{A}\right] x(s)$$

$$Q = x \cdot s\left[\frac{VEM}{A}s^3 + \left(\frac{VED}{A} + \frac{ML}{A}\right)s^2 + \left(\frac{LD}{A} + A + \frac{VEK}{A}\right)s + \frac{KL}{A}\right]$$

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7-60

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		DATE

Transfer Function - Load Position/Flow IN

$$\frac{x}{Q} = \frac{1}{\frac{VEM}{A} S^3 + \frac{VEL + LM}{A} S^2 + \left(A + \frac{VD + KL}{A}\right) S + \frac{KL}{A}}$$

Constants Used: (8 Y AXIS)

$V = 1.11 \times 10^{4}$ cu. in.
$E = 5 \times 10^{-6}$ in^2/lb
$M = 1.126 \times 10^{2}$ lb-sec^2/in
$D = 29.8$ lb-sec/in
$A = 3.14$ in^2
$L = 1.74 \times 10^{-3}$ cu. in/sec-PSI
$K = 2942$ lb/in

$$\frac{x}{Q} = \frac{1}{1.99 S^3 + .589 S^2 + 55.16 S + 1.63}$$

$$= \frac{1}{(S + .0296)(1.99 S^2 + .530 S + 55.14)}$$

$$= \frac{\frac{1}{55.14}}{(S + .0296)\left(\frac{1.99}{55.14} S^2 + \frac{.530}{55.14} S + 1\right)}$$

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CHECKED BY	SUBJECT	PROGRAM & PROJECT NO.
		DATE

$$\omega_n = \sqrt{\frac{55.14}{1.99}} = \sqrt{27.7} = 5.27 \frac{rad}{sec}$$

$$f_n = \frac{5.27}{6.28} = .84 \text{ cps}$$

for $K = 0$

$$\frac{X}{Q} = \frac{1}{1.99S^3 + .589S^2 + 3.16}$$

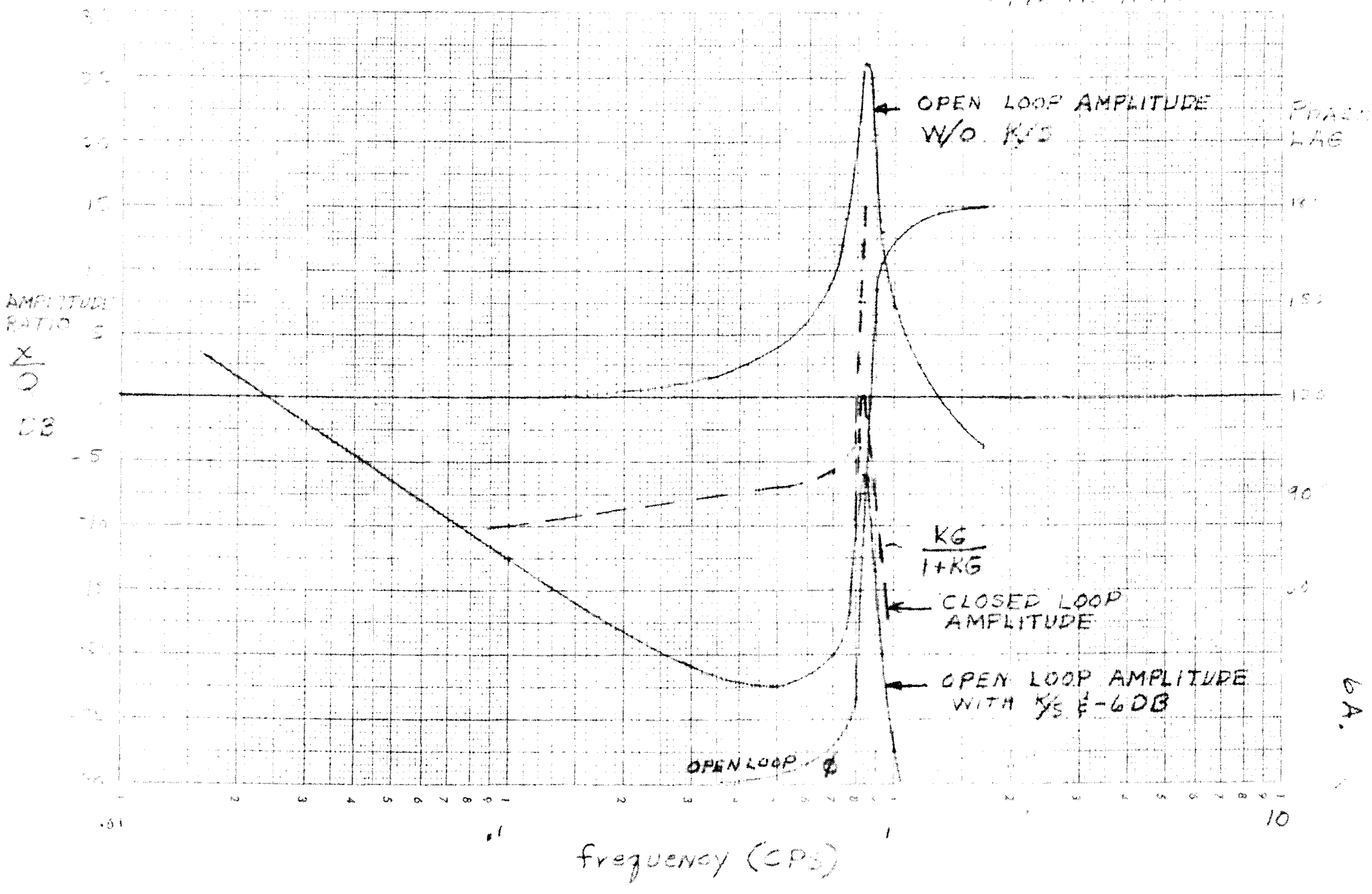
$$\omega_n = \sqrt{\frac{3.16}{1.99}} = \sqrt{1.585} = 1.265 \text{ rad/sec}$$

$$f_n = .2 \text{ cps}$$

Frequencies obtain by removing sets of Springs

No. of Spring Columns	Ks	ω_n	f_N (cps)
9	2942	5.27	.84
8	2616	4.98	.792
7	2287	4.68	.745
6	1962	4.35	.693
5	1635	4.10	.638
4	1308	3.64	.580
3	981	3.21	.511
2	654	2.72	.433
1	327	2.12	.337
0	0	1.26	.20

Max Possible deviation from Listed $f_n = \pm 4.8\%$


OPEN LOOP AMPLITUDE
W/O K/S
PHASE LAG
AMPLITUDE RATIO
X/O
DB
KG / 1+KG
CLOSED LOOP AMPLITUDE
OPEN LOOP AMPLITUDE
WITH K/S ≠ -6DB
OPEN LOOP φ
.01
.1
1
10
frequency (CPS)

6A.

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Axis	MAX HP_{IN}	MAX HP_{LOAD}	MAX $HP_{DRIVE\ RAL}$	MAX Motor HP (90% GA. Eff.)
8Y	163	1050	106	181
8X	123	650	66	137
7Y	109	635	65	121
7X	85	378	41	95
6Y	72	388	40	80
6X	54	210	22	60
9Y	119	585	59.5	132
9X	92	318	33.	103
3Y		22.5		
3X		15.		
2Y		17.5		
2X		10		
4Y		55	6[illegible]	
4X		40	3[illegible]0	
5Y	46	183	19.3	51
5X	41	128	13.8	46

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$$HP_{\substack{DRIVE\\LOAD}} = \frac{F_{MAX} \times .05 \times V_{MAX}}{6600} + 1_{HYST.\ SPRING}$$

$$HP_{LOAD} = \frac{.5\ F_M\ V_{MAX}}{6600}$$

$$HP_{IN} = HP_{DRIVE\ LOAD} + \frac{750\ Q_{MAX}}{1714}$$

Data on Page 7 from these formulae.

APPENDIX TO SECTION II

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BASIC ENERGY STORAGE METHODS

1. SPRINGS

   A. MECHANICAL - COIL SPRING, TORSION BAR

   B. GAS - ACCUMULATOR

   C. HYDRAULIC - OIL VOLUME UNDER COMPRESSION

2. INERTIA - LINKAGE AND COUNTERBALANCE WEIGHT

3. GRAVITY - PENDULUM

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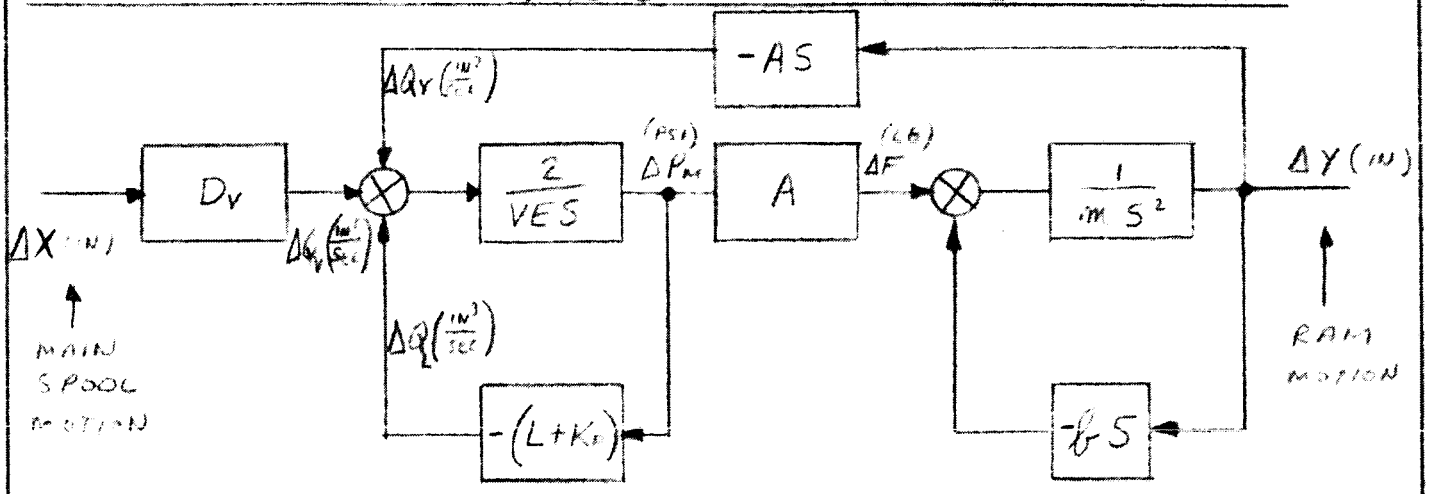
1. BASIC SERVO VALVE TO CYLINDER SYSTEM



D_V = VALVE FLOW GAIN (IN^3/SEC-IN)

V = VOLUME OF OIL UNDER COMPRESSION (IN^3)

E = COMPRESSIBILITY OF FLUID ($\frac{IN^2}{LB}$)

ΔX = INCREMENTAL CHANGE IN VALVE MAIN SPOOL POSITION (IN)

ΔQ_V = INCREMENTAL CHANGE IN VALVE FLOW ($\frac{IN^3}{SEC}$)

ΔQ_Y = INCREMENTAL CHANGE IN RAM FLOW ($\frac{IN^3}{SEC}$)

ΔQ_L = INCREMENTAL CHANGE IN EQUIVALENT LEAKAGE FLOW ($\frac{IN^3}{SEC}$)

L = RAM LEAKAGE COEFFICIENT ($\frac{IN^3}{SEC\text{-}PSI}$)

K_P = SERVO VALVE FLOW - PRESSURE GRADIENT ($\frac{IN^3}{SEC\ PSI}$)

A = RAM AREA (IN^2)

m = MASS LOAD ON RAM ($\frac{LB\text{-}SEC^2}{IN}$)

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b = LOAD AND RAM VISCOUS DAMPING COEFICIENT (LB-SEC/IN)

ΔY = INCREMENTAL CHANGE IN RAM MOTION (IN)

SERVO VALVE TO RAM TRANSFER FUNCTION

$$\frac{\Delta Y}{\Delta X} = \frac{D_V/A}{S\left(\frac{1}{\omega_{VR}^2}S^2 + \frac{2\zeta_{VR}}{\omega_{VR}}S + 1\right)}$$

$$\omega_{VR} = \sqrt{\frac{2(L+K_P)b + 2A^2}{VEM}}$$

VALVE TO RAM NATURAL FREQUENCY (1/SEC)

$$\zeta_{VR} = \frac{(L+K_P)M + \frac{1}{2}VEb}{2\sqrt{\frac{1}{2}VEM\left\{(L+K_P)b + A^2\right\}}}$$

VALVE TO RAM DAMPING FACTOR

NEGLECTING b

$$\omega_{VR} = \frac{\sqrt{2}\,A}{\sqrt{VEM}}$$

$$\zeta_{VR} = \frac{(L+K_P)M}{2\sqrt{\frac{1}{2}VEM}\;A}$$

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FOR MAXIMUM HORSEPOWER DRIVE #8

BY

1. WEIGHT - 43,500 LB, MASS 112.7 $\frac{LB\ SEC^2}{IN}$
2. AMPLITUDE OF OSCILLATION - ± 28.7 IN
3. FREQUENCY OF OSCILLATION - 0.2 - 0.84 cps (1.26 - 5.28 $\frac{RAD}{SEC}$)
4. PEAK VELOCITY - 152.5 $\frac{IN}{SEC}$
5. PEAK ACCELERATION - 800 $\frac{IN}{SEC^2}$ (COMPUTED)
6. MAXIMUM FORCE - 90,700 LB
7. CYLINDER AREA FOR MAXIMUM FORCE AT 3000 PSI

$$\frac{90,900\ LB}{3,000\ LB/IN^2} = 30.3\ IN^2$$

BX

1. WEIGHT - 26,900 LB, MASS 69.7 $\frac{LB\ SEC^2}{IN}$
2. AMPLITUDE OF OSCILLATION - ± 28.7 IN
3. FREQUENCY OF OSCILLATION - 0.2 - 0.84 cps (1.26 - 5.28 $\frac{RAD}{SEC}$)
4. PEAK VELOCITY - 152.5 $\frac{IN}{SEC}$
5. PEAK ACCELERATION - 800 $\frac{IN}{SEC^2}$ (COMPUTED
6. MAXIMUM FORCE - 56,200 LB
7. CYLINDER AREA FOR MAXIMUM FORCE AT 3000 PSI

$$\frac{56,200\ LB}{3,000\ LB/IN^2} = 18.7\ IN^2$$

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ON ZY USE 8 INCH DIA CYL., 4.5 INCH DIA ROD
ANNULUS AREA - 34.361 IN^2

ON ZX USE 7 INCH DIA CYL, 5 INCH DIA ROD
ANNULUS AREA - 18.850 IN^2

MAXIMUM ΔP

ZY 90,900 LB / 34.361 IN^2 = 2650 PSI

ZX 56,200 LB / 18.850 IN^2 = 2990 PSI

MAXIMUM FLOW

ZY $152.5 \frac{IN}{SEC} \times 34.361 IN^2 = 5240 \frac{IN^3}{SEC}$ (1362 GPM)

ZX $152.5 \frac{IN}{SEC} \times 18.850 IN^2 = 2880 \frac{IN^3}{SEC}$ (747 GPM)

SERVO VALVE TO RAM NATURAL FREQUENCY
NEGLECTING b

$$\omega_{VR} = \sqrt{\frac{2A^2}{VEM}}$$

	ZY	ZX
A	34.361 IN^2	18.850 IN^2
V	1010 IN^3	603 IN^3
E	$5.6 \times 10^{-6} \frac{IN^2}{LB}$	$5.6 \times 10^{-6} \frac{IN^2}{LB}$
M	$112.7 \frac{LB\ SEC^2}{IN}$	$69.7 \frac{LB\ SEC^2}{IN}$
ω_{VR}	$64.4 \frac{1}{SEC}$	$53.1 \frac{1}{SEC}$
f_{VR}	10.26 cps	9.24 cps

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INSTANTANEOUS DAMPING FACTOR

$$K_P = \frac{Q}{2(P_S - P_L)}$$

$L = 0.0004 \frac{IN^3}{SEC\text{-}PSI}$ 7Y, $0.0003 \frac{IN^3}{SEC\text{-}PSI}$ 8X

MINIMUM FOR TYPICAL VALVES $K_P = 0.012 \frac{IN^3}{SEC\text{-}PSI}$ 8Y $0.008 \frac{IN^3}{SEC\text{-}PSI}$ 8X

MAXIMUM $= \frac{Q}{2P_V}$

$$K_P = \frac{5240 \frac{IN^3}{SEC}}{3000 PSI} = 1.747 \frac{IN^3}{SEC\text{-}PSI} \quad 8Y$$

$$K_P = \frac{2880 \frac{IN^3}{SEC}}{3000 \frac{IN^3}{PSI}} = 0.960 \frac{IN^3}{SEC\text{-}PSI} \quad 8X$$

$$\xi_{VR} = \frac{(L + K_P) M \omega_{VR}}{2 A^2}$$ NEGLECTING b

	8Y	8X
$(L + K_P)_{MIN}$	$0.012 \frac{IN^3}{SEC\text{-}PSI}$	$0.008 \frac{IN^3}{SEC\text{-}PSI}$
$(L + K_P)_{MAX}$	$1.748 \frac{IN^3}{SEC\text{-}PSI}$	$0.960 \frac{IN^3}{SEC\text{-}PSI}$
M	$112.7 \frac{LB\ SEC^2}{IN}$	$69.7 \frac{LB\ SEC^2}{IN}$
A	$34.361\ IN^2$	$18.850\ IN^2$
ω_{VR}	$64.4 \frac{1}{SEC}$	$58.1 \frac{1}{SEC}$
ξ_{VR}	$3.06(L + K_P)$	$5.70(L + K_P)$
$\xi_{VR\,MIN}$	0.0368	0.0244
$\xi_{VR\,MAX}$	5.35	5.47

BRUTE FORCE SYSTEM. ξ_{VR}(MIN) WILL LIKELY BE HIGHER IN THE ACTUAL SYSTEM BECAUSE OF DAMPING DUE TO LOAD FRICTION

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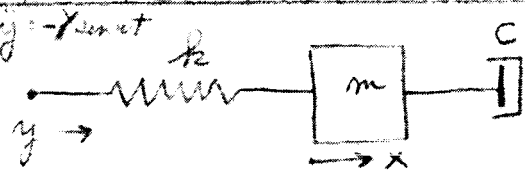
RESONANCE DRIVING TO REDUCE POWER INPUT

$y = Y \sin \omega t, \; \dot{y} = Y\omega \cos \omega t$



$$m\ddot{x} + C\dot{x} + k(x - y) = 0$$

$$Ky = m\ddot{x} + C\dot{x} + kx = KY \sin \omega t$$

STEADY STATE SOLUTION

$$x = X \sin(\omega t - \alpha), \quad \dot{x} = X\omega \cos(\omega t - \alpha), \quad \ddot{x} = -X\omega^2 \sin(\omega t - \alpha)$$

$$X = \frac{KY}{\sqrt{(k - m\omega^2)^2 + C^2\omega^2}}$$

$$\alpha = \tan^{-1} \frac{C\omega}{k - m\omega^2}$$

$$\frac{X}{Y} = \frac{K}{\sqrt{(k - m\omega^2)^2 + C^2\omega^2}}$$

ASSUME RESONANT FREQUENCY = 0.84 cps

$\omega_N = 5.28 \frac{1}{sec}$, $m = 112.7 \frac{lb\ sec^2}{in}$ FOR

$$\omega_N = \sqrt{\frac{k}{m}}$$

$$k = m\omega_N^2 = 112.7 \frac{lb\ sec^2}{in} \times \left(5.28 \frac{1}{sec}\right)^2 = 3140 \frac{lb}{in}$$

$C = 2\zeta m\omega_N$ FOR $\zeta = 0.1$

$$C = 0.2 \times 112.7 \frac{lb\ sec^2}{in} \times 5.28 \frac{1}{sec}$$

$$C = 119.0 \frac{lb\text{-}sec}{in}$$

$$\alpha = \tan^{-1} \frac{119\omega}{3140 - 112.7\omega^2}$$

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$$\frac{X}{Y} = \frac{3140}{\sqrt{(3140 - 112.7\,\omega^2)^2 + 14.17\,\omega^2}}$$

f (cps)	ω (1/sec)	$112.7\omega^2$	$14.70\omega^2$	Denom	X/Y
0.84	5.28	3140	375,000	629	5.00
0.70	4.40	2180	274,000	2005	3.70
0.50	3.14	1111	139,700	2005	1.567
0.30	1.89	403	50,700	2737	1.147
0.20	1.26	179	22,500	2353	1.175

$$\text{Horsepower} = F\dot{X} \quad (\text{output})$$
$$= F\dot{y} \quad (\text{input})$$

$$F = k(y - x) = k(Y \sin \omega t - X \sin(\omega t - \alpha))$$

$$\dot{X} = X\omega \cos(\omega t - \alpha)$$

$$\dot{y} = Y\omega \cos \omega t$$

ω (1/sec)	117ω	$3140 - 112.7\omega^2$	$\tan\alpha$	α	Motion Equation
5.28	628	0	∞	90.0°	$X = 5Y \sin(5.28t + 1.57)$
4.40	524	960	0.547	28.7°	$X = 3.70Y \sin(4.40t + 0.501)$
3.14	374	2029	0.1842	10.5°	$X = 1.567Y \sin(3.14t + 0.1832)$
1.81	225	2737	0.0822	4.7°	$X = 1.147Y \sin(1.81t + 0.0820)$
1.26	150	2761	0.0507	2.9°	$X = 1.175Y \sin(1.26t + 0.0506)$

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$$F = k\left(Y \sin \omega t - X \sin \omega t \cos \alpha + X \cos \omega t \sin \alpha\right)$$

$$\dot{x} = X\omega\left(\cos \omega t \cos \alpha + \sin \omega t \sin \alpha\right)$$

$$\dot{y} = Y\omega \cos \omega t$$

AT RESONANCE $\alpha = 90°$, $\cos \alpha = 0$, $\sin \alpha = 1$

$$F = k\left(Y \sin \omega t + X \cos \omega t\right)$$

$$\dot{x} = X\omega \sin \omega t$$

$$\dot{y} = Y\omega \cos \omega t$$

<u>POWER INTO LOAD</u>

$$P_L = F\dot{x} = kX\omega\left(Y \sin^2 \omega t + X \cos \omega t \sin \omega t\right)$$

AT PEAK POWER

$$\frac{\partial P}{\partial t} = 0 = kX\omega\left(2Y\omega \sin \omega t \cos \omega t + X\omega \cos^2 \omega t - X\omega \sin^2 \omega t\right)$$

$$0 = kX\omega^2\left(Y \sin 2\omega t + X \cos 2\omega t\right)$$

OR $$\tan 2\omega t = -\frac{X}{Y}$$

FOR $\zeta = 0.1$, $X/Y = 5$,

$2\omega t = 101.3°$, $\omega t = 50.7°$

$\sin 50.7° = 0.7738$, $\sin^2 50.7 = 0.5980$

$\cos 50.7° = 0.6334$

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$$P_L(MAX) = k\omega X^2\left(\frac{Y}{X}\sin^2\omega t + \cos\omega t \tan\omega t\right)$$

$$P_L(MAX) = k\omega X^2\left(0.2 \times 0.5980 + 0.6334 \times 0.7738\right)$$

$$P_L(MAX) = k\omega X^2\left(0.1196 + 0.490\right)$$

$$P_L(MAX) = k\omega X^2\left(0.6096\right)$$

$$P_L(MAX) = 3140 \times 5.28 \times (28.7)^2 \times (0.6096) \ \frac{IN\text{-}LB}{SEC}$$

$$P_L(MAX) = 8{,}320{,}000 \ \frac{IN\text{-}LB}{SEC} \Rightarrow 1{,}260 \ HP$$

<u>POWER INPUT FROM ACTUATOR</u>
(NEGLECTING FRICTION LOSSES)

$$P_A = F\dot{y} = k Y\omega\left(Y \sin\omega t \cos\omega t + X\cos^2\omega t\right)$$

AT PEAK POWER

$$\frac{\partial P}{\partial t} = 0 = k Y\omega\left(Y\omega\cos^2\omega t - Y\omega\sin^2\omega t - 2X\omega\cos\omega t\sin\omega t\right)$$

$$0 = k Y\omega^2\left(Y\cos 2\omega t - X\sin 2\omega t\right)$$

$$\tan 2\omega t = Y/X$$

FOR $\zeta = 0.1$, $Y/X = 0.2$

$2\omega t = 11.5$ DEG, $\omega t = 5.75$ DEG

$\sin 5.75° = 0.1001$

$\cos 5.75° = 0.9950$, $\cos^2 5.75° = 0.990$

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$$P_A(MAX) = k_2 Y^2 \omega \left(\sin \omega t \cos \omega t + \frac{X}{Y} \cos^2 \omega t \right)$$

$$P_A(MAX) = k_2 Y^2 \omega \left(0.1001 \times 0.995 + 5 \times 0.990 \right.$$

$$P_A(MAX) = k_2 Y^2 \omega \left(0.0996 + 4.950 \right)$$

$$P_A(MAX) = k_2 Y^2 \omega \left(5.050 \right)$$

$$P_A(MAX) = 3140 \times (5.74)^2 \times 5.28 \times 5.05 \ \frac{IN\text{-}LB}{SEC}$$

$$P_A(MAX) = 2{,}770{,}000 \ \frac{IN\text{-}LB}{SEC} \Rightarrow 417 \ HP$$

POWER INTO LOAD FOR $\zeta = 0.05$

FOR $\zeta = 0.05$, $X/Y = 10$

$2\omega t = 95.7°$, $\omega t = 47.7°$

$\sin 47.7° = 0.7420$ $\quad \sin^2 47.7° = 0.551$

$\cos 47.7° = 0.6704$

$$P_L(MAX) = k_2 \omega X^2 \left(\frac{Y}{X} \sin^2 \omega t + \cos \omega t \sin \omega t \right)$$

$$P_L(MAX) = k_2 \omega X^2 \left(0.1 \times 0.551 + 0.6704 \times 0.7420 \right)$$

$$P_L(MAX) = k_2 \omega X^2 \left(0.0551 + 0.498 \right)$$

$$P_L(MAX) = k_2 \omega X^2 \left(0.553 \right)$$

$$P_L(MAX) = 1260 \ HP \left(\frac{0.553}{0.6076} \right) = 1145 \ HP$$

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POWER INPUT FROM ACTUATOR FOR $\zeta = 0.05$

FOR $\zeta = 0.05$, $Y/X = 0.1$

$2\omega t = 5.75°$, $\omega t = 2.88°$

$\sin 2.88° = 0.0505$

$\cos 2.88° = 0.999$, $\cos^2 2.88° = 0.998$

$$P_A(MAX) = k Y^2 \omega \left(\sin \omega t \cos \omega t + \frac{X}{Y} \cos^2 \omega t\right)$$

$$P_A(MAX) = k Y^2 \omega \left(0.0505 \times 0.999 + 10 \times 0.998\right)$$

$$P_A(MAX) = k Y^2 \omega \left(0.050 + 9.98\right)$$

$$P_A(MAX) = k Y^2 \omega \left(10.03\right)$$

$$P_A(MAX) = 3140 (2.87)^2 \times 5.28 \times 10.03 \ \frac{IN\text{-}LB}{SEC}$$

$$P_A(MAX) = 1,313,000 \ \frac{IN\text{-}LB}{SEC} \Rightarrow 208 \ HP$$

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RESONANT SYSTEM WITH MECHANICAL SPRINGS

$C = 117.0 \frac{LB\ SEC}{IN} \times \frac{0.05}{0.1} = 59.5 \frac{LB\ SEC}{IN}$ (SEE P. 7)

FOR DAMPING FACTOR OF 0.05

SPRING RATE REQUIRED 3140 $\frac{LB}{IN}$ FOR 1 SPRING

1570 $\frac{LB}{IN}$ FOR 2 PARALLEL

785 $\frac{LB}{IN}$ FOR 4 PARALLEL

STROKE ± 28.7 IN

DESIGN SPRINGS ASSUMING 1 SPRING AND 4 PARALLEL, REF. "HELICAL COMPRESSION SPRINGS" MACHINE DESIGN, 12-6-62

$P = 90,100$ LB MAXIMUM WORKING LOAD

$\delta = 28.7$ IN DEFLECTION FOR MAXIMUM WORKING LOAD

$G = 11.5 \times 10^6$ PSI TORSIONAL MODULUS OF ELASTICITY

$N_{ia} = 5$ NUMBER OF INACTIVE COILS

$n = 0.22$ EXPONENT FOR CARBON SPRING STEEL WIRE

$A = 35,000$ PSI STRESS FACTOR SEVERE DUTY FOR CARBON STEEL

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DESIGN CONSTANT

$$X = \frac{0.212\ \delta G}{n_{ia}\ P^{0.438}\ A^{0.562}}$$

1 SPRING

$$X = \frac{0.212 \times 28.7 \times 11.5 \times 10^6}{n_{ia}\ (90,100)^{0.438}\ (35,000)^{0.562}}$$

$$X = \frac{70 \times 10^6}{n_{ia} \times 150 \times 400}$$

$$X = \frac{1167}{n_{ia}}$$

4 PARALLEL

$$X = \frac{0.212 \times 28.7 \times 11.5 \times 10^6}{n_{ia}\ (22,725)^{0.438}\ (35,000)^{0.562}}$$

$$X = \frac{70 \times 10^6}{n_{ia} \times 81 \times 400}$$

$$X = \frac{2165}{n_{ia}}$$

LET $n_{ia} = 5$

$X = 233$

$C = 4.8$ MIN. WT.
21.2 MIN LGT.
2.95 MIN VOL

SPRING INDEX

$X = 433$

5.2 MIN WT.
23.6 MIN LGT
3.55 MIN VOL.

$$d = \frac{2.55\ KPC}{F}$$

WIRE DIA (IN)

K = WAHL FACTOR
= 1.21

$$d = \frac{2.55 \times 1.21 \times 90,100 \times C}{35,000}$$

$d = 2.02\ C$

$$d = \frac{2.55 \times 1.21 \times 22,725 \times C}{35,000}$$

$d = 2.01\ C$

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$n_a = \frac{8Gd}{8PC^3}$ NUMBER OF ACTIVE COILS

$D = Cd$ MEAN DIA (IN.)

$L = (n_a + n_{ia})d$ LENGTH OF SPRING (IN)

1 SPRING	LET $C = 3$	4 PARALLEL
$d = 24.06$ IN		$d = 6.01$ IN
$D = 72.18$ IN		$d = 18.03$ IN
$n_a = \frac{28.7 \times 11.5 \times 10^6 \times 24.06}{8 \times 40,100 \times 27}$		$n_a = \frac{28.7 \times 11.5 \times 10^6 \times 6.01}{8 \times 22,775 \times 27}$
$n_a = 408$		$n_a = 404$
$L = 9980$ IN (831 FT)		$L = 2460$ IN (205 FT)

IMPRACTICAL

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RESONANT SYSTEM WITH FLUID SPRINGS

$$f = \frac{\sqrt{2}\,A}{\sqrt{VE}} \qquad A = 34.4 \text{ IN}^2, \quad E = 5\times10^{-6}\ \frac{\text{IN}^2}{\text{LB}}$$

$$k = 314 \ \frac{\text{LB}}{\text{IN}}$$

$$V = \frac{2\,A^2}{k\,E}$$

$$V = \frac{2\,(34.4)^2 \text{ IN}^4}{1400\ \frac{\text{LB}}{\text{IN}} \times 5\times10^{-6}\ \frac{\text{IN}^2}{\text{LB}}}$$

$$V = 100{,}700 \text{ IN}^3 \quad (3 \text{ FT DIA} \times 12.34 \text{ FT LONG})$$

VOLUME REQUIRED ON EACH SIDE

THE ADIABATIC TANGENT BULK MODULUS $\left(\frac{1}{E}\right)$ CHANGES FOR MIL-H-5606 FROM 0 - 3000 PSI AT 100° F IS

$$\frac{270{,}000 - 225{,}000}{225{,}000} \times 100\% = 20\%$$

THIS REPRESENTS A CHANGE IN RESONANT FREQUENCY OF 9.6%

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RESONANT SYSTEM WITH AIR SPRINGS (ACCUMULATOR)

$$VE = \frac{2A^2}{K}$$

ACCUMULATORS ACT AS PARALLEL SPRINGS

$$VE = \frac{2 \times (34.4)^2 \text{ IN}^4}{3140 \frac{\text{LB}}{\text{IN}}} = 0.753 \frac{\text{IN}^3}{\text{PSI}} \quad \left(0.377 \frac{\text{IN}^3}{\text{PSI}} \text{ EACH SIDE}\right)$$

FOR ADIABATIC EXPANSION AND COMPRESSION

$$P_1 V_1^{1.4} = P_2 V_2^{1.4}$$

$$\Delta V = 2 \times 28.7 \text{ IN} \times 34.4 \text{ IN}^2$$

$$\Delta V = 1986 \text{ IN}^3$$

$$\Delta P = \frac{1986 \text{ IN}^3}{0.377 \frac{\text{IN}^3}{\text{PSI}}} = 5280 \text{ PSI} \quad \pm 2640 \text{ PSI}$$

ASSUME $P_1 = 360$ PSI , $P_2 = 3000$ PSI

$$\left(\frac{V_1}{V_2}\right)^{1.4} = \frac{P_2}{P_1} = 8.33$$

$$\frac{V_1}{V_2} = 4.55 \quad , \quad V_1 - V_2 = 1986 \text{ IN}^3$$

$$4.55\, V_2 - V_2 = 1986 \text{ IN}^3$$

$$3.55\, V_2 = 1986 \text{ IN}^3$$

$$V_2 = 559 \text{ IN}^3 \text{ AT } 3000 \text{ PSI}$$

$$V_1 = 2540 \text{ IN}^3 \text{ AT } 360 \text{ PSI}$$

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INITIAL VOLUME OF COMPRESSED GAS - EACH ACCUMULATOR

$$V_0 = (2540 + 559)/2 \ IN^3 = 1550 \ IN^3$$

V_A (IN^3)	V_B (IN^3)	P_A (PSI)	P_B (PSI)	ΔP (PSI)	CHANGE IN ΔP (PSI)	$\frac{\Delta V}{\Delta P}$ ($\frac{IN^3}{PSI}$)
1550	1550	718	718	0		
					265	0.753
1750	1350	607	872	265		
					308	0.649
1950	1150	521	1094	573		
					374	0.507
2150	950	456	1423	967		
					619	0.323
2350	750	402	1938	1586		
					1118	0.1788
2550	550	358	3062	2704		

$$P_{A(0)} = P_1 \left(\frac{V_1}{V_{A(0)}}\right)^{1.4} = 360 \left(\frac{2540}{V_{A(0)}}\right)^{1.4}$$

SPRING RATE IS VERY NONLINEAR

TO OBTAIN A REASONABLY LINEAR SPRING RATE WOULD REQUIRE OPERATION WITH A SMALL ΔP AS COMPARED TO OPERATING PRESSURE.

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COMPUTATION OF EFFECTIVE SPRING RATE

$$k_2 = \frac{2A^2}{VE} \quad \text{HYDRAULIC SPRING RATE}$$

$$P_A V_A^{1.4} = P_B V_B^{1.4} = P_0 V_0^{1.4}$$

$$P_A = P_0 \left(\frac{V_0}{V_A}\right)^{1.4} = P_0 \left(\frac{V_0}{V_0 + AY}\right)^{1.4} = P_0 \left(1 + \frac{1}{V_0} AY\right)^{-1.4}$$

$$P_B = P_0 \left(\frac{V_0}{V_B}\right)^{1.4} = P_0 \left(\frac{V_0}{V_0 - AY}\right)^{1.4} = P_0 \left(1 - \frac{1}{V_0} AY\right)^{-1.4}$$

$$k_2 = A^2 \left(\frac{2}{VE}\right) = A^2 \frac{\partial (P_A - P_B)}{\partial \Delta V} = A^2 \frac{\partial (P_A - P_B)}{\partial AY}$$

$$P_A - P_B = P_0 \left[\left(1 + \frac{1}{V_0} AY\right)^{-1.4} - \left(1 + \frac{1}{V_0} AY\right)^{-1.4}\right]$$

$$k_2 = A^2 \frac{\partial}{\partial AY} (P_A - P_B) = \frac{-1.4 A^2 P_0}{V_0} \left[\left(1 + \frac{1}{V_0} AY\right)^{-2.4} + \left(1 - \frac{1}{V_0} AY\right)^{-2.4}\right]$$

$$k_2 = \frac{-1.4 (34.4)^2 \times 718}{1550} \frac{LB}{IN} \left[\left(1 + \frac{34.4}{1550} Y\right)^{-2.4} + \left(1 - \frac{34.4}{1550} Y\right)^{-2.4}\right]$$

$$k_2 = -768 \frac{LB}{IN} \left[(1 + 0.0222 Y)^{-2.4} + (1 - 0.0222 Y)^{-2.4}\right]$$

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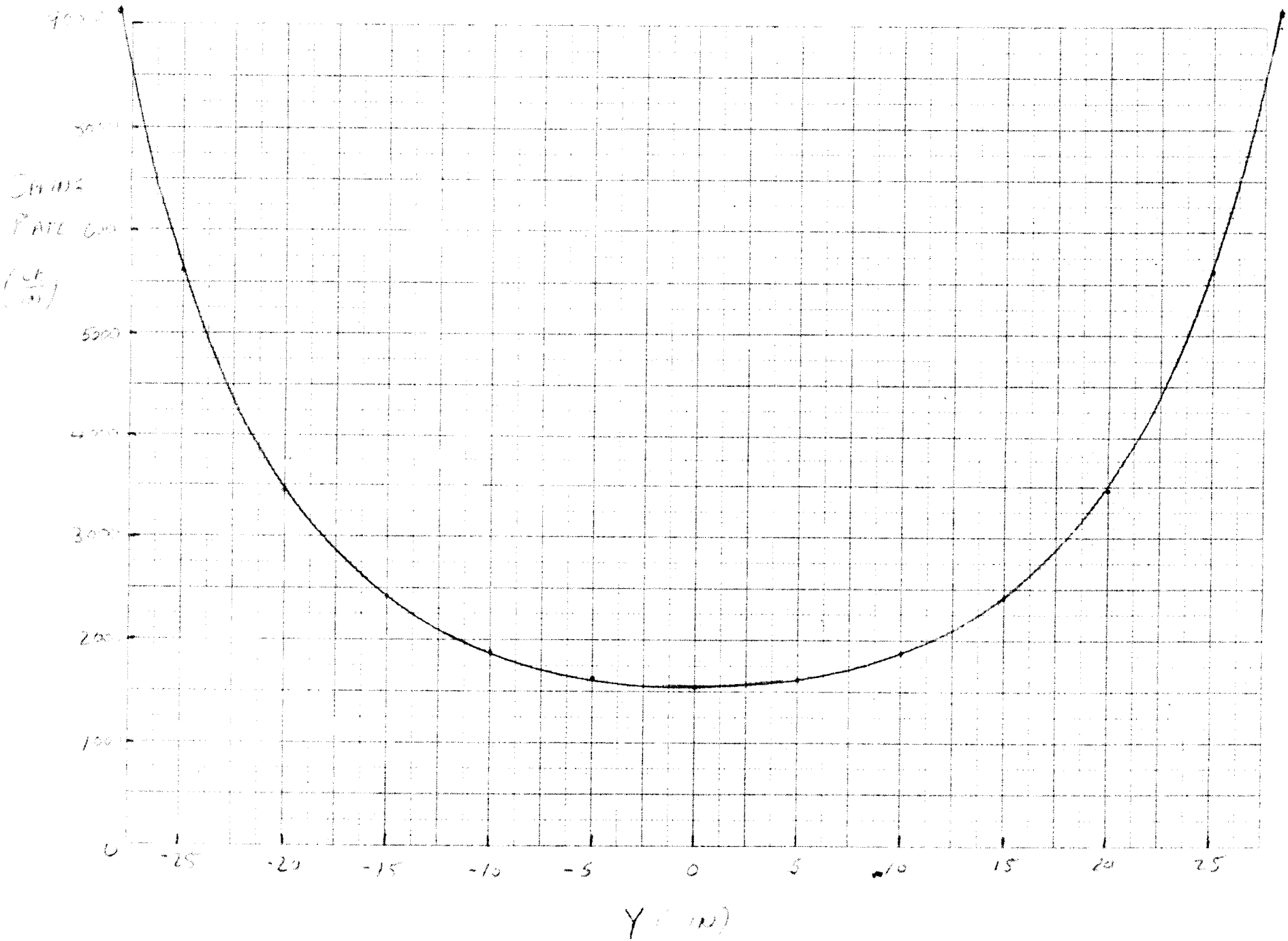
Y	0.0222X	$(1+0.0222Y)^{-2.4}$	$(1-0.0222Y)^{-2.4}$	k
0	0.000	1.000	1.000	1535
±5	0.111	0.777	1.326	1615
±10	0.222	0.618	1.817	1870
±15	0.333	0.502	2.64	2410
±20	0.444	0.418	4.38	3455
±25	0.555	0.347	6.17	5615
±28.7	0.637	0.306	11.60	9150

UPCP-RADIUM MOTION SIMULATOR
EFFECTIVE SPRING RATE ACCUMULATOR SYSTEM



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RESONANT SYSTEM WITH FLUID SPRINGS - SERVO VALVE DRIVES

AXIS 8Y

AT SYSTEM RESONANCE FOR 0.1 DAMPING FACTOR VALVE MUST SUPPLY 1/5 PEAK FLOW AT FULL PRESSURE

$$Q_V = \frac{1}{5} \times 1362 \text{ GPM} = 272 \text{ GPM}$$

$$P_L = 2650 \text{ PSI}, \quad P_V = 500 \text{ PSI}, \quad P_S = 3150 \text{ PSI}$$

DAMPING FACTOR

$$\zeta_{VR} = \frac{(L + K_P) M \omega_{VR}}{2A^2} = 0.251 (L + K_P) \quad \begin{cases} A = 34.4 \text{ IN}^2 \\ M = 112.7 \frac{\text{LB-SEC}^2}{\text{IN}} \\ \omega_{VR} = 5.28 \frac{1}{\text{SEC}} \end{cases}$$

$$K_{P(MAX)} = \frac{1048 \frac{\text{IN}^3}{\text{SEC}}}{2500 \text{ PSI}} = 1.048 \frac{\text{IN}^3}{\text{SEC PSI}}$$

$$K_{P(MIN)} = 0.0028 \frac{\text{IN}^3}{\text{SEC PSI}}$$

FOR DAMPING = 0.1

$$(L + K_P) = \frac{0.1}{0.251} \frac{\text{IN}^3}{\text{SEC PSI}} = 0.398 \frac{\text{IN}^3}{\text{SEC-PSI}}$$

$K_{P(MIN)}$ CAN BE INCREASED TO POSSIBLY 0.1 BY USING A GROSS LINE ORIFICE

$K_P(MAX)$ WILL RESULT IN LARGE INHERENT SYSTEM DAMPING AND INCREASED POWER LOSS.

SOME DAMPING WILL BE PROVIDED BY MECHANICAL SYSTEM

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RESONANT SYSTEM WITH FLUID SPRINGS - SERVO PUMP DRIVES

$Q_{PUMP} = 272$ GPM PUMP SUPPLIES $\frac{1}{3}$ PEAK FLOW AT PEAK PRESSURE FOR 0.1 DAMPING FACTOR

$P_L = 2650$ PSI

$$\omega_{PR} = \sqrt{\frac{2A^2}{VEM}} = 5.26 \frac{1}{sec}$$

FOR $A = 34.4 \text{ IN}^2$

$E = 5 \times 10^6 \frac{\text{IN}^2}{\text{LB}}$

$V = 150{,}700 \text{ IN}^3$

$M = 112.7 \frac{\text{LB sec}^2}{\text{IN}}$

$$\zeta_{PR} = \frac{(L_c + L_p) M \omega_{PR}}{2A^2}$$

$\zeta_{PR} = 0.251 (L_c + L_p)$ $L_p = 0.008 \frac{\text{IN}^3}{\text{sec PSI}}$ (NOMINAL)

$\zeta_{PR} = 0.00226$ $L_c = 0.001 \frac{\text{IN}^3}{\text{sec PSI}}$

SOME ADDITIONAL DAMPING COULD BE PROVIDED BY CROSS LINE ORIFICE

LOAD WILL ALSO ADD DAMPING

ASSUMING 5% OF LOAD FORCE IS VISCOUS DAMPING

$$b = \frac{4545 \text{ LB}}{152.5 \text{ IN/sec}} = 27.8 \frac{\text{LB-sec}}{\text{IN}}$$

ADDITIONAL DAMPING PROVIDED BY LOAD

$$\zeta_{PR}(\text{LOAD}) \approx \frac{\sqrt{VE}\, b}{2A\sqrt{2M}}$$

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$$\zeta_{PR}(\text{LOAD}) = \frac{\sqrt{0.1507 \times 10^{6} \times 5 \times 10^{-6}} \times 27.8}{2 \times 34.4 \sqrt{2 \times 112.7}} = 0.0250$$

WITH ACCELERATION FEEDBACK AND LOAD DAMPING
REFERENCE P. 37 AND P 38.

ω ($\frac{1}{sec}$)	HP	PUMP FLOW GPM
1.257	26.8	292
2.51	87.4	480
3.77	215.0	474
5.02	137.0	134
5.28	103.8	65.2

$$Q = \frac{4820}{2650} \times Q\,(P.37)$$

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SERVO PUMP DRIVE - 5000 PSI SYSTEM

CYLINDER AREA FOR MAXIMUM FORCE AT 5000 PSI

$$\frac{90,900 \text{ LB}}{5000 \text{ LB/IN}^2} = 18.18 \text{ IN}^2$$

USE 7 INCH DIA CYL. WITH 5 INCH DIA ROD

AREA = 18.850

$$\Delta P = \frac{10,900}{18.85} = 4820 \text{ PSI}$$

MAX FLOW $= 18.85 \times 152.5 \frac{\text{IN}^3}{\text{SEC}} = 2330 \frac{\text{IN}^3}{\text{SEC}}$ (747 GPM)

ASSUME PUMP MUST SUPPLY $\frac{1}{5}$ FLOW

PUMP FLOW = 149 GPM

$$V = \frac{2A^2}{E M \omega_{PR}^2}$$

$$V = \frac{2 \times (18.18)^2}{5 \times 10^{-6} \times 112.7 \times (5.28)^2}$$

$V = 42,000 \text{ IN}^3$

$A = 18.18 \text{ IN}^2$

$E = 5.6 \times 10^{-6} \frac{\text{IN}^2}{\text{LB}}$

$M = 112.7 \frac{\text{LB-SEC}^2}{\text{IN}}$

$\omega_{VR} = 5.28 \frac{1}{\text{SEC}}$

$L_c = 0.001 \frac{\text{IN}^3}{\text{SEC PSI}}$

$L_P = 0.004 \frac{\text{IN}^3}{\text{SEC PSI}}$

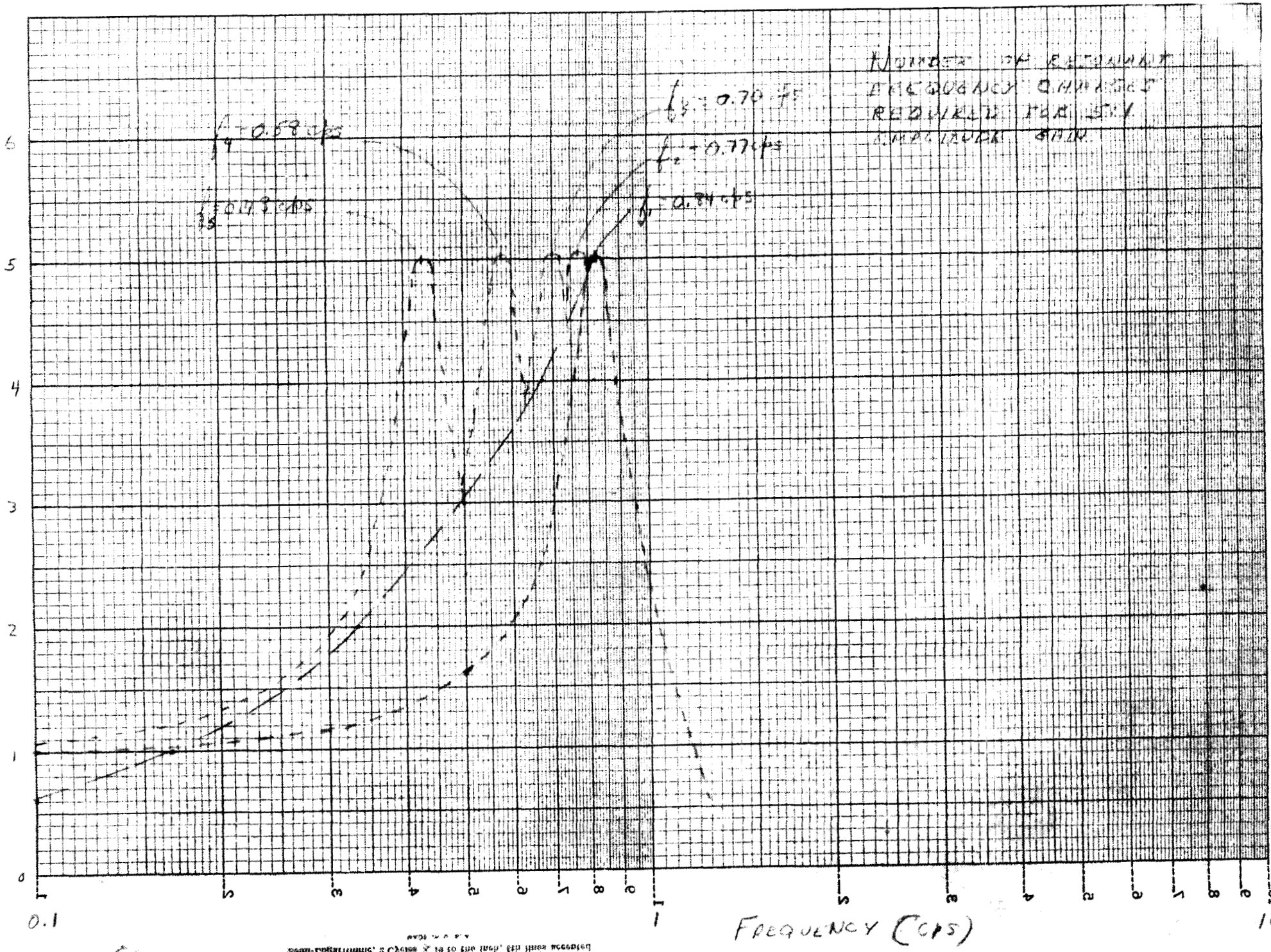
$$\zeta_{PR} = \frac{(L_c + L_P) M \omega_{VR}}{2A^2}$$

$\zeta_{PR} = (L_c + L_P)\, 0.903$

$\zeta_{PR} = 0.0045$

$$\zeta_{PR}\,(\text{CHK}) = \frac{\sqrt{0.042 \times 10^6 \times 5 \times 10^{-6} \times 27.6}}{2 \times (18.85) \times \sqrt{2 \times 112.7}} = 0.0216$$

(SEE P. 20)



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Semi-Logarithmic 3 Cycles X 10 to the inch, 6 to the inch Y

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PUMP - CYLINDER EQUATIONS

$$Q_{P_1} = A\dot{y} + L(P_A - P_B) + VE\dot{P}_A$$

$$Q_{P_2} = A\dot{y} + L(P_A - P_B) - VE\dot{P}_B$$

$$Q_P = A\dot{y} + L(P_A - P_B) + \frac{VE}{2}(\dot{P}_A - \dot{P}_B)$$

$$P_A - P_B = \frac{M}{A}\ddot{y}$$

$$\dot{P}_A - \dot{P}_B = \frac{M}{A}\dddot{y}$$

$$Q_P = A\dot{y} + \frac{LM}{A}\ddot{y} + \frac{VEM}{2A}\dddot{y}$$

ASSUME $Q_P = Q$ = const

STEADY TIME SOLUTION

LET $y = -Y\cos(\omega t - \alpha)$

$\dot{y} = Y\omega\sin(\omega t - \alpha)$

$\ddot{y} = Y\omega^2\cos(\omega t - \alpha)$

$\dddot{y} = -Y\omega^3\sin(\omega t - \alpha)$

BY ANALOGY TO EQUATIONS ON P. 7

$X \Rightarrow Y\omega$, $1 \Rightarrow \dot{y}$, $m \Rightarrow \frac{VEM}{2A}$

$b \Rightarrow \frac{LM}{A}$, $k \Rightarrow A$

TERM ON LEFT IS REPLACED BY TERM ON RIGHT

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$$Y_\omega = \frac{Q}{\sqrt{\left(A - \frac{VEM}{2A}\omega^2\right)^2 + \frac{L^2M^2}{A^2}\omega^2}}$$

$$\alpha = \tan^{-1} \frac{(LM/A)\,\omega}{A - \frac{VEM}{2A}\omega^2}$$

$A = 18.18 \text{ IN}^2$

$E = 5.6 \times 10^{-6} \frac{\text{LB}}{\text{IN}^2}$

$M = 112.1 \frac{\text{LB-SEC}^2}{\text{IN}}$

$V = 42{,}000 \text{ IN}^3$

$YA = 522 \text{ IN}^3$

$$\omega_N = \frac{\sqrt{2A}}{\sqrt{VEM}} \qquad \omega_N^2 = \frac{2A^2}{VEM}$$

$\omega_N = 5.28 \frac{1}{\text{SEC}}$

FOR $\zeta = 0.1$

$$L = 2A\sqrt{\frac{2VE}{M}} = 0.1173 \frac{\text{IN}^3}{\text{SEC-PSI}}$$

$\omega\left(\frac{1}{\text{SEC}}\right)$	ω/ω_n	AY_ω/Q	Q (GPM)	α	$Y_\omega A\left(\frac{\text{IN}^3}{\text{SEC}}\right)$	$Q\left(\frac{\text{IN}^3}{\text{SEC}}\right)$
1.257	0.238	1.08	157.5	2.5°	656	607
2.51	0.477	1.52	238	8°	1310	794
3.77	0.714	2.02	255	17°	1180	781
5.02	0.952	4.8	141.7	63°	2620	546
5.28	1.000	5.0	143.4	90°	2760	554

$$\frac{A\,Y_\omega}{Q} = \frac{1}{\sqrt{\left(1 - \frac{VEM}{2A^2}\omega^2\right)^2 + \frac{L^2M^2}{A^4}\omega^2}}$$

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ACCELERATION PRESSURE

$$P = \frac{M\ddot{y}}{A} = \frac{MY\omega^2}{A} \cos(\omega t - \alpha)$$

ω $(\frac{1}{sec})$	$\frac{MY\omega^2}{A}$ (P.I)	α	$90-\alpha$	PEAK POWER CORRECTION	POWER $\frac{in\ lb}{sec}$	HP
1.257	281	2.5°	87.5°	0.510	0.898×10^5	13.2
2.51	1122	8°	82°	0.565	6.30×10^5	96
3.77	2530	17°	73°	0.640	15.7×10^5	241
5.02	4480	63°	27°	0.940	23.0×10^5	349
5.28	4970	90°	0°	1.000	27.6×10^5	420

$$\Delta P_{PUMP} = \frac{MY\omega^2}{A} \sin(\omega t + 90 - \alpha)$$

$$Q_{PUMP} = Q \sin \omega t$$

$$POWER = \frac{MY\omega^2 Q}{A} \sin \omega t \sin(\omega t + 90 - \alpha)$$

CROSS LINE LEAKAGE = $4170 \times 0.1173 \frac{in}{sec} = 5.46 \frac{in^3}{sec}$
(MAXIMUM AT $\omega = 5.28 \frac{1}{sec}$)
(INCLUDED IN Q_{PUMP}) = 115.2 GPM

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WITH VISCOUS LOAD DAMPING

$$Q_P = A\dot{y} + L(P_A - P_B) + \frac{VE}{2}(\dot{P}_A - \dot{P}_B)$$

$$P_A - P_B = \frac{M}{A}\ddot{y} + \frac{b}{A}\dot{y}$$

$$\dot{P}_A - \dot{P}_B = \frac{M}{A}\dddot{y} + \frac{b}{A}\ddot{y}$$

$$Q_P = \left(A + \frac{Lb}{A}\right)\dot{y} + \left(\frac{LM}{A} + \frac{VEb}{2A}\right)\ddot{y} + \frac{VEM}{2A}\dddot{y}$$

$$A \gg \frac{Lb}{A}$$

$$Q_P = A\dot{y} + \left(\frac{LM}{A} + \frac{VEb}{2A}\right)\ddot{y} + \frac{VEM}{2A}\dddot{y}$$

b = LOAD VISCOUS DAMPING COEFFICIENT $\left(\frac{LB\text{-}SEC}{IN}\right)$

PART OF SYSTEM DAMPING WILL BE PROVIDED BY VISCOUS DAMPING OF LOAD.

DECREASED FLOW REQUIREMENT DUE TO SMALLER CROSS LINE DAMPING ORIFICE WILL BE OFFSET BY INCREASED CYLINDER SIZE REQUIRED TO COUNTERBALANCE ADDITIONAL FORCE TO OVERCOME VISCOUS FRICTION OF THE LOAD.

BY ANALOGY TO EQUATIONS ON p. 24

$$Q_P = Q \sin \omega t$$

$$y = -Y \cos(\omega t - \alpha)$$

$$\dot{y} = Y\omega \sin(\omega t - \alpha)$$

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$$Y_\omega = \frac{Q}{\sqrt{\left(A - \frac{VEM}{2A}\omega^2\right)^2 + \left(\frac{LM}{A} + \frac{VEb}{2A}\right)^2 \omega^2}}$$

$$\frac{AY_\omega}{Q} = \frac{1}{\sqrt{\left(1 - \frac{VEM}{2A^2}\omega^2\right) + \left(\frac{LM}{A^2} + \frac{VEb}{2A^2}\right)^2 \omega^2}}$$

$$\omega_N = \frac{\sqrt{2}A}{\sqrt{VEM}}, \quad \omega_N^2 = \frac{2A^2}{VEM}$$

$A = 18.18 \text{ IN}^2$

$E = 5.6 \times 10^{-6} \frac{\text{IN}^2}{\text{LB}}$

$\omega_N = 5.28 \frac{1}{\text{SEC}}$

$M = 112.7 \frac{\text{LB-SEC}^2}{\text{IN}}$

$V = 42{,}000 \text{ IN}^2$

$$\zeta = \left[\frac{LM}{2A^2} + \frac{VEb}{4A^2}\right]\omega_N$$

$YA = 522 \text{ IN}^3$

ASSUME $b = \frac{4545}{152.5} \frac{\text{LB-SEC}}{\text{IN}} = 21.8 \frac{\text{LB-SEC}}{\text{IN}}$

$$\zeta_b = \frac{42{,}000 \times 5.6 \times 10^{-6} \times 21.8 \times 5.28}{4\,(18.18)^2} = 0.068$$

$\zeta_q = 0.0720$

$$L = \frac{\zeta_L\, 2A^2}{\omega_N\, M} = \frac{0.072 \times 2 \times (18.18)^2}{5.28 \times 112.7} \frac{\text{IN}^3}{\text{SEC PSI}} = 0.080 \frac{\text{IN}^3}{\text{SEC PSI}}$$

MAXIMUM LEAKAGE $= 4170 \text{ PSI} \times 0.08 \frac{\text{IN}^3}{\text{SEC PSI}} = 318 \frac{\text{IN}^3}{\text{SEC}}$
$= 103.3 \text{ GPM}$

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CLOSED LOOP RESPONSE

$$\frac{Y}{Q_P} = \frac{K}{S\left(\frac{1}{\omega^2}S^2 + \frac{2\zeta}{\omega}S + 1\right)}$$

$$K = \frac{1}{A}$$

$$\omega_N = \frac{\sqrt{2}\,A}{\sqrt{VE\,M}} = 5.28 \frac{1}{SEC}$$

$$\zeta = \left[\frac{LM}{2A^2} + \frac{VE\,b}{4A^2}\right]\omega_N = 0.1$$

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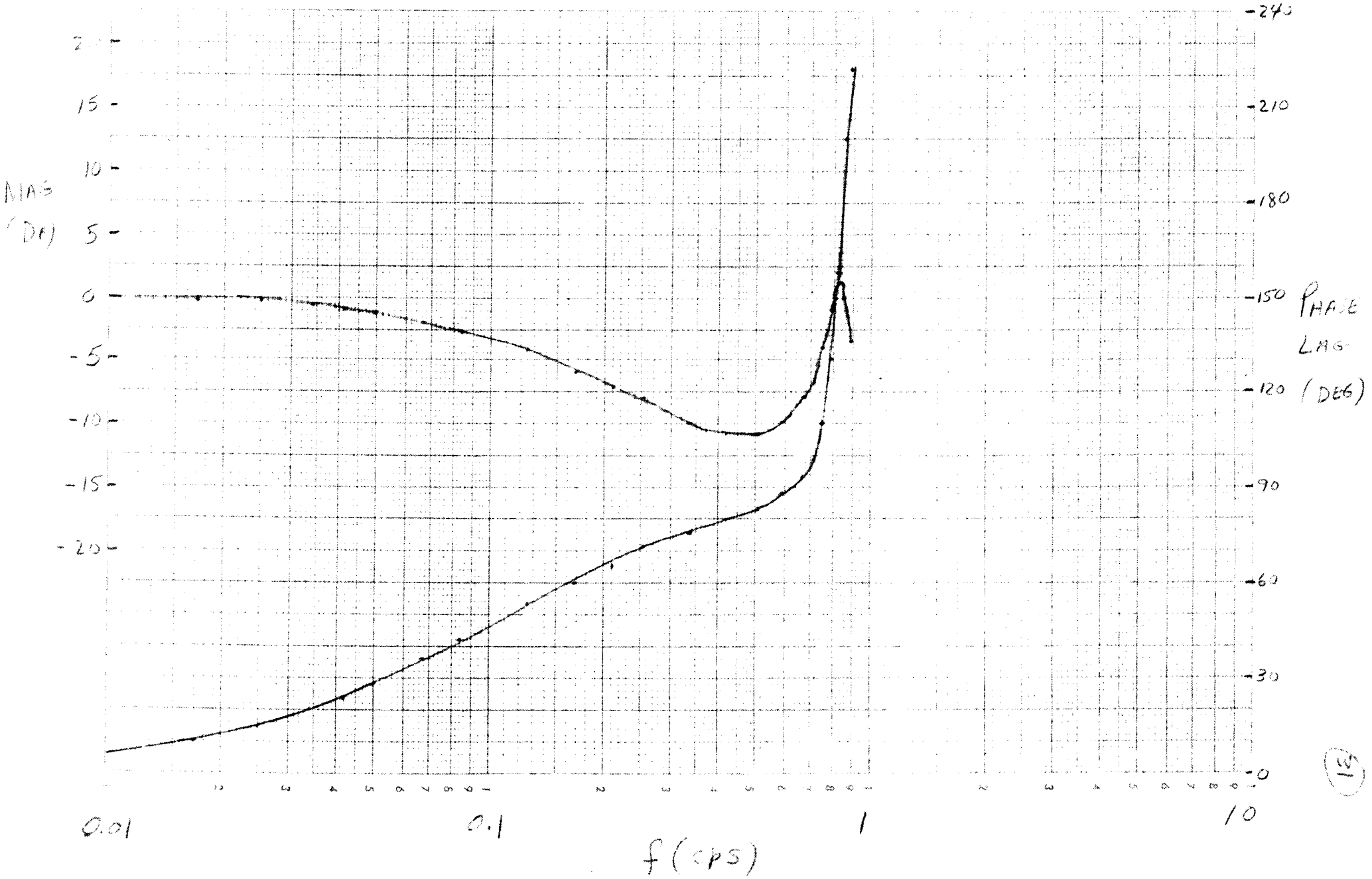
CLOSED LOOP RESPONSE S-MM



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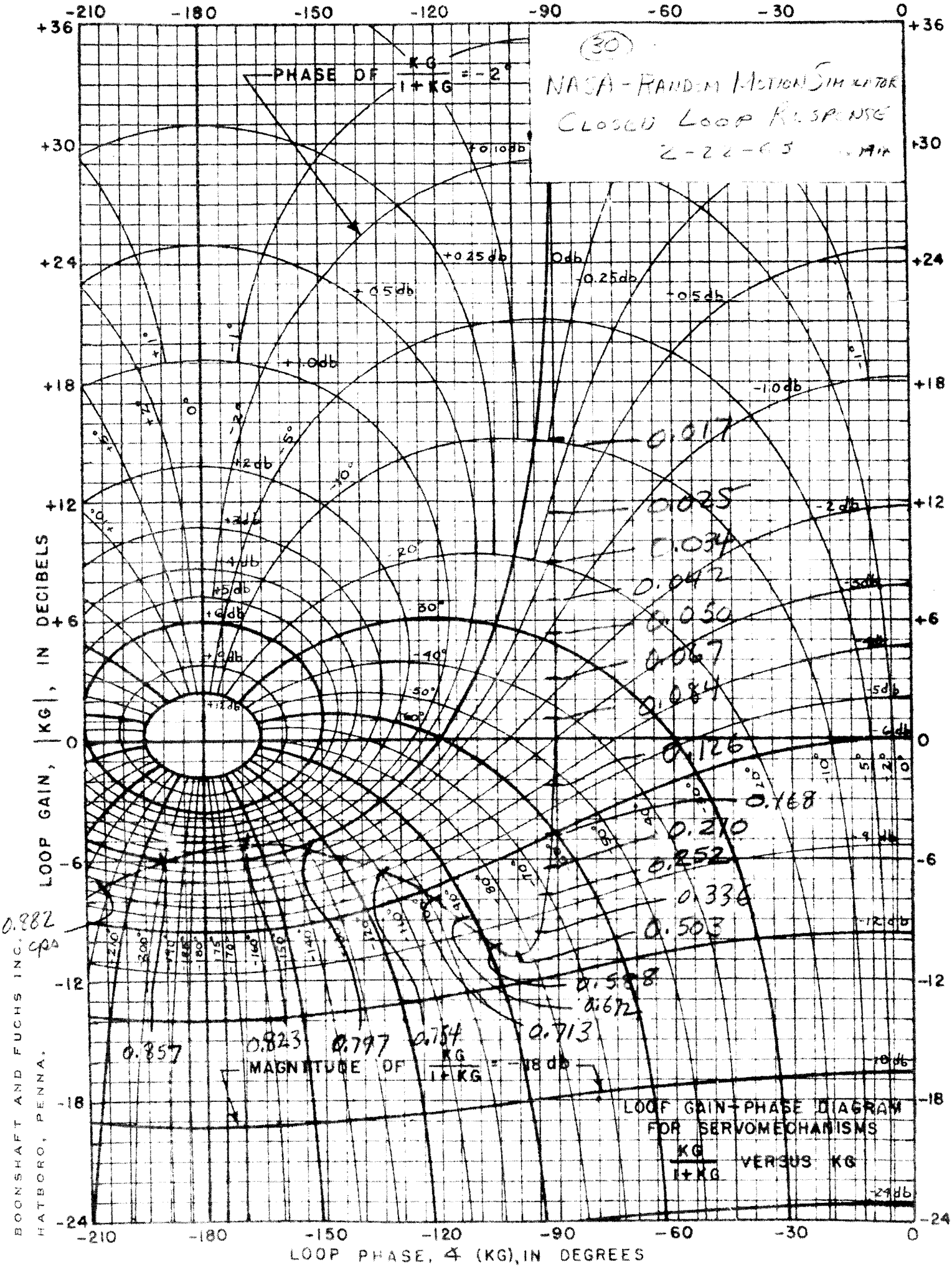
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NASA - RANDOM MOTION SIMULATOR
CLOSED LOOP RESPONSE
2-24-65
PHASE OF KG/(1+KG) = -2°
MAGNITUDE OF KG/(1+KG) = -18 db
LOOP GAIN+PHASE DIAGRAM
FOR SERVOMECHANISMS
KG/(1+KG) VERSUS KG
LOOP GAIN, |KG|, IN DECIBELS
LOOP PHASE, ∠ (KG), IN DEGREES
0.882 cps
0.857
0.823
0.797
0.754
0.713
0.672
0.588
0.503
0.336
0.252
0.210
0.168
0.126
0.084
0.067
0.050
0.042
0.034
0.025
0.017
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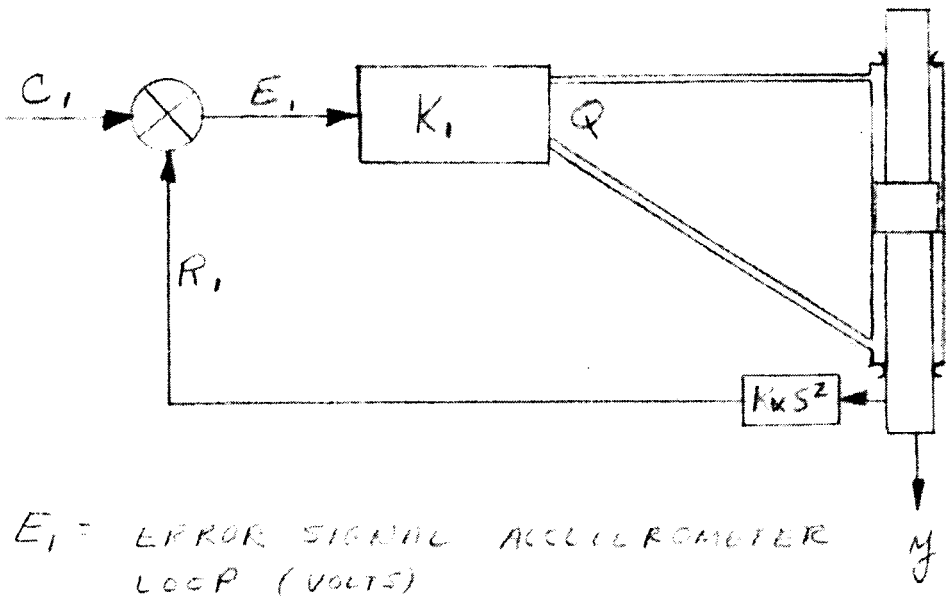
ACCELERATION FEEDBACK

TO CONSERVE POWER PROVIDE SYSTEM DAMPING WITH ACCELERATION FEEDBACK INSTEAD OF LEAKAGE FLOW



E_1 = ERROR SIGNAL ACCELEROMETER LOOP (VOLTS)

K_1 = ERROR SIGNAL TO PUMP OUTPUT GAIN $\left(\frac{IN^3}{SEC}/VOLT\right)$

$$Q_P = A\dot{y} + \frac{LM}{A}\ddot{y} + \frac{VLM}{2A}\dddot{y} = K_1 E_1$$

IN OPERATIONAL FORM

$$\frac{y}{Q_P} = \frac{1/A}{S\left(1 + \frac{LM}{A^2}S + \frac{VLM}{2A^2}S^2\right)}$$

$$\frac{y}{E_1} = \frac{K_1/A}{S\left(1 + \frac{LM}{A^2}S + \frac{VLM}{2A^2}S^2\right)}$$

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CLOSED LOOP

$$\frac{C_1}{R_1} = \frac{1}{\frac{1}{\ } + H} = \frac{1}{\frac{S\left(1 + \frac{LM}{A^2}S + \frac{VEM}{2A^2}S^2\right)}{K_1/A} + K_x S^2}$$

$$\frac{C_1}{R_1} = \frac{K_1/A}{S\left(1 + \left\{\frac{LM}{A^2} + \frac{K_1 K_x}{A}\right\} S + \frac{VEM}{2A^2}S^2\right)}$$

$$\omega_N = \frac{\sqrt{2}\,A}{\sqrt{VEM}} = 5.28 \frac{1}{SEC}$$

$A = 18.18\ IN^2$

$E = 5.6 \times 10^{-6} \frac{IN^2}{LB}$

$$\zeta = \zeta_1 + \zeta_2 = \frac{LM}{2A^2}\omega_N + \frac{K_1 K_x}{2A}\omega_N$$

$M = 112.7 \frac{LB\text{-}SEC^2}{IN}$

$$\zeta_1 = \frac{0.005 \times 112.7 \times 5.28}{2\ (18.18)^2}$$

$L = 0.005 \frac{IN^3}{SEC\text{-}PSI}$

$$\zeta_1 = 0.0045$$

FOR GOOD CLOSED LOOP PERFORMANCE LET $\zeta = 0.5$

$$\zeta_2 \approx 0.5 = \frac{K_1 K_x \omega_N}{2A}$$

$$K_1 K_x = \frac{2 \times 18.18 \times 0.5}{5.28}\ IN^2\text{-}SEC = 3.44\ IN^2\text{-}SEC$$

FROM PUMP FLOW - LOAD MOTION TRANSFER FUNCTION

$$Q_p = G \sin \omega t \quad , \quad y = -Y \cos(\omega t - \alpha)$$

$$\dot{y} = Y\omega \sin(\omega t - \alpha) \quad , \quad \ddot{y} = Y\omega^2 \cos(\omega t - \alpha)$$

$$\dddot{y} = Y\omega^3 \sin(\omega t - \alpha)$$

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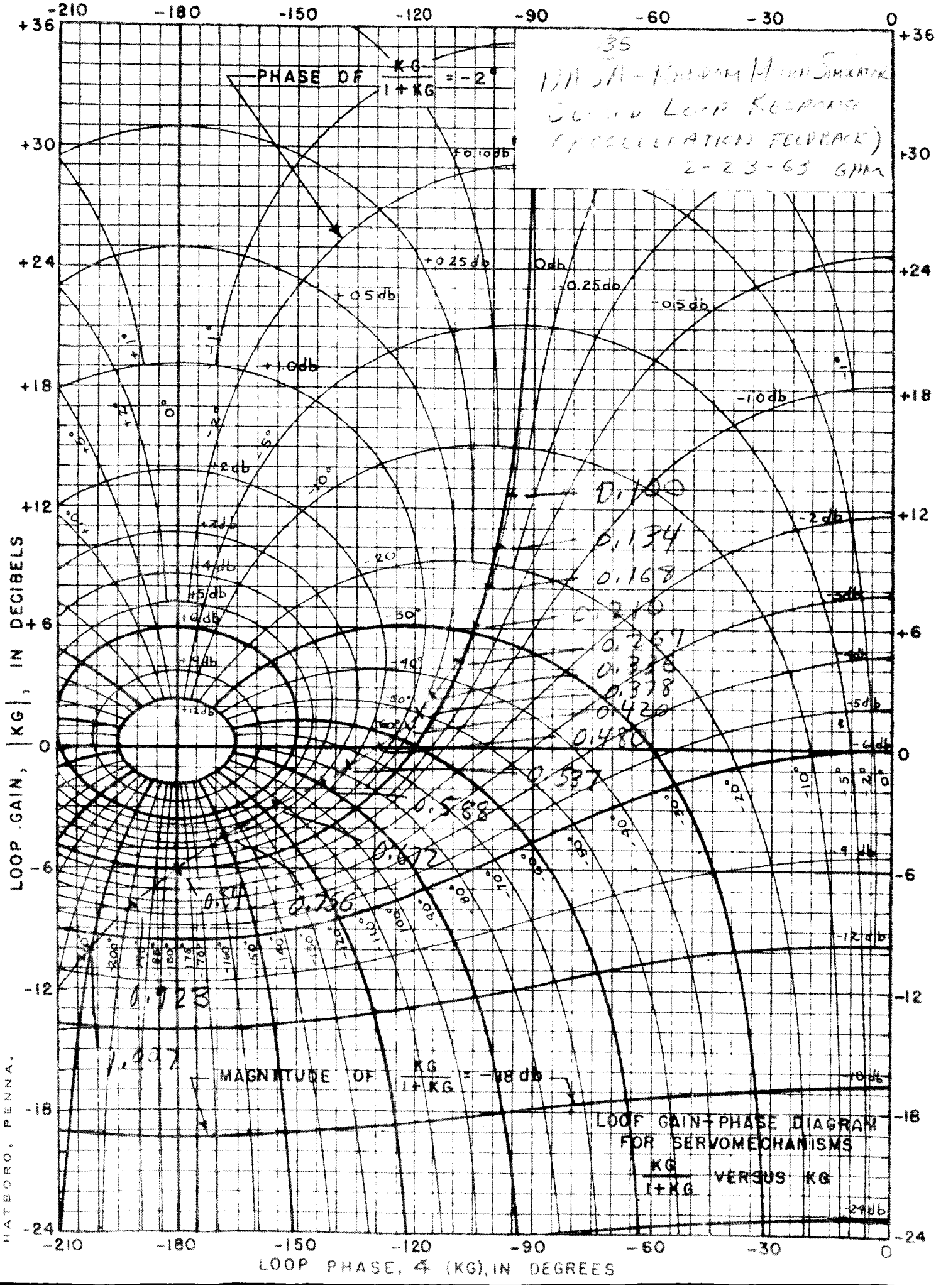
$$\frac{AY\omega}{Q} = \frac{1}{\sqrt{\left(1 - \frac{VEM}{2A^2}\omega^2\right)^2 + \frac{L^2M^2}{A^4}\omega^2}}$$

$\omega\left(\frac{1}{sec}\right)$	ω/ω_n	$AY\omega/Q$	Q (GPM)	α	$Y\omega A\left(\frac{in^3}{sec}\right)$	$Q\left(\frac{in^3}{sec}\right)$
1.257	0.238	1.05	162	0°	656	624
2.51	0.477	1.30	262	0°	1310	1007
3.77	0.714	2.0	257	1°	1780	990
5.02	0.952	10.2	67.5	4°	2620	260
5.28	1.000	124	5.78	90°	2760	22.3

$$Q_{PUMP} = Q \sin \omega t, \quad \Delta P_{PUMP} = \frac{MY\omega^2}{A} \sin(\omega t + 90 - \alpha)$$

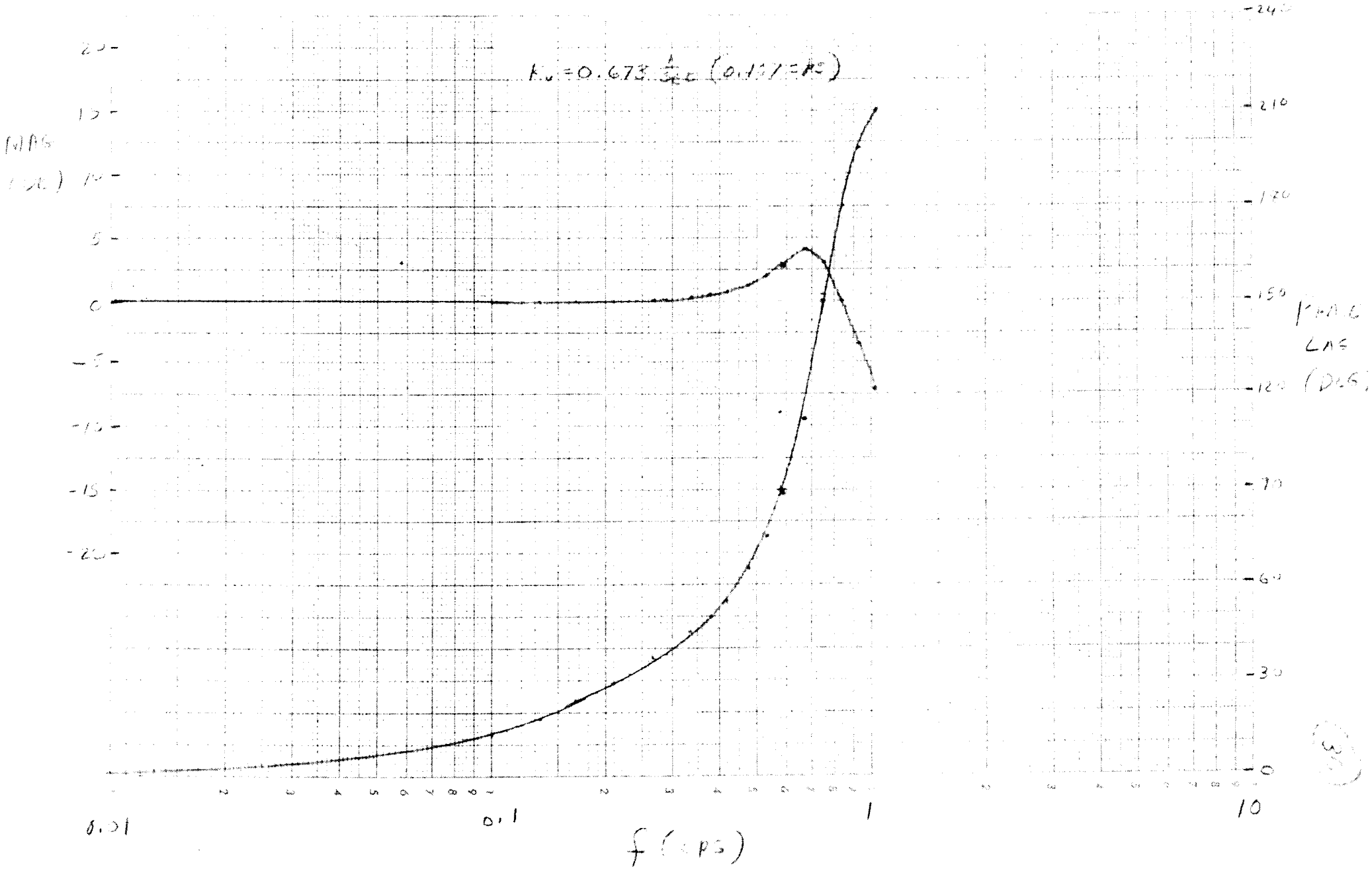
$$\text{Power} = \frac{MY\omega^2 Q}{A} \sin \omega t \sin(\omega t + 90 - \alpha)$$

$\omega\left(\frac{1}{sec}\right)$	$\frac{MY\omega^2}{A}$ (psi)	α	$90-\alpha$	PEAK POWER CORRECTION	PEAK POWER $\frac{in\text{-}lb}{sec}$	HP
1.257	281	0°	90°	0.500	0.877×10^5	13.3
2.51	1122	0°	90°	0.500	5.65×10^5	85.7
3.77	2530	1°	89°	0.500	12.52×10^5	190
5.02	4480	4°	86°	0.545	6.36×10^5	96.3
5.28	4710	90°	0	1.000	1.11×10^5	16.9


35
Open Loop Response
(Acceleration Feedback)
2-23-63 GHM
PHASE OF KG / 1+KG = -2°
MAGNITUDE OF KG / 1+KG = -18 db
LOOP GAIN+PHASE DIAGRAM
FOR SERVOMECHANISMS
KG / 1+KG VERSUS KG
LOOP GAIN, |KG|, IN DECIBELS
LOOP PHASE, ∠ (KG), IN DEGREES
BOONSHAFT AND FUCHS INC.
HATBORO, PENNA.

Mr. A. - Kinden [illegible]
Closed-Loop Response
[illegible] Acceleration Feedback

2-26-65
[illegible]



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WITH ACCELERATION FEEDBACK AND LOAD DAMPING
REF P. 33-34 AND P. 27-28

$$\frac{Q_1}{K_1} = \frac{K_1/A}{S\left(1 + \left\{\frac{LM}{A^2} + \frac{VE\beta}{2A^2} + \frac{K_1 K_\alpha}{A}\right\} S + \frac{VEM}{2A^2} S^2\right)}$$

$$\omega_N = \frac{\sqrt{2}A}{\sqrt{VEM}} = 5.28 \ \frac{1}{sec}$$

$$\zeta_1 = \frac{LM}{2A^2}\omega_N + \frac{VE\beta}{4A^2}$$

$$\zeta_1 = 0.0045 + 0.028 \qquad \text{(see p. 28)}$$

$$\zeta_1 = 0.0325$$

$$Q_P = A\dot{y} + \left(\frac{LM}{A} + \frac{VE\beta}{2A}\right)\ddot{y} + \frac{VEM}{2A}\dddot{y}$$

$$Q_{PUMP} = Q \sin \omega t, \quad L_{PUMP} = \frac{MY\omega^2}{A} \sin(\omega t + 90 - \alpha)$$

$$\text{POWER} = \frac{MY\omega^2 Q}{A} \sin \omega t \sin(\omega t + 90 - \alpha)$$

$\omega \left(\frac{1}{sec}\right)$	ω/ω_N	$AY\omega/Q$	Q (GPM)	α	$Y\omega A \left(\frac{in^3}{sec}\right)$	$Q \left(\frac{in^3}{sec}\right)$
1.251	0.238	1.04	164	1°	656	631
2.51	0.477	1.27	264	2°	1310	1015
3.77	0.714	1.77	261	5°	1760	1004
5.02	0.952	9.23	73.7	26°	2620	284
5.28	1.000	20.0	35.7	90°	2760	138

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ω (rad/sec)	$\frac{MY\omega^2}{A}$ (psi)	α	$90-\alpha$	PEAK POWER CORRECTION	PEAK IN-LB/SEC	POWER HP
1.257	281	1°	89°	0.510	$1.77 \cdot 10^5$	26.6
2.51	1122	2°	88°	0.518	5.70×10^5	89.4
3.77	2530	5°	85°	.538	14.17×10^5	215
5.02	4480	26°	64	0.720	9.17×10^5	139.0
5.26	4170	90°	0°	1.000	6.86×10^5	103.8

ADDITIONAL POWER WILL BE REQUIRED

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POWER CONSIDERING LOAD VISCOUS DRAG

$$P = \frac{MY\omega^2}{A}\cos(\omega t - \alpha) + \frac{bY\omega}{A}\sin(\omega t - \alpha)$$

$$P_p = Q \sin \omega t$$

$$P = C \sin(\omega t - \alpha + \beta)$$

$$C = \sqrt{\left(\frac{MY\omega^2}{A}\right)^2 + \left(\frac{bY\omega}{A}\right)^2}$$

$$\tan\beta = \frac{MY\omega^2 A}{A\, bY\omega} = \frac{M\omega}{b} = 3.735\,\omega$$

ω	$\left(\frac{MY\omega^2}{A}\right)^2$	$\left(\frac{bY\omega}{A}\right)^2$	C^2	C	$\tan\beta$	β°	α°	$\beta-\alpha^\circ$
1.257	0.0770×10^6	0.00351×10^6	0.114×10^6	871	4.76	78.1	1	77.1
2.51	1.258×10^6	0.01317×10^6	1.27×10^6	1128	9.5	84.0	2	82.0
3.77	6.46×10^6	0.0315×10^6	6.48×10^6	2540	14.3	86.0	5	81.0
5.02	20.06×10^6	0.0557×10^6	20.14×10^6	4470	17.0	87.0	26	61.0
5.28	24.7×10^6	0.0625×10^6	24.76×10^6	4970	20.0	87.1	90	-2.9

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Q ($\frac{in^3}{sec}$)	C (psi)	CQ ($\frac{in\text{-}lb}{sec}$)	Power Correction	Power $\frac{in\text{-}lb}{sec}$	HP	ω
631	891	5.62×10^5	0.61	3.43×10^5	51.8	1.257
1015	1128	11.46×10^5	0.57	6.54×10^5	99.1	2.51
1004	2540	25.5×10^5	0.58	14.78×10^5	224	3.77
284	4470	12.76×10^5	0.745	9.50×10^5	144	5.02
138	4970	6.85×10^5	1.00	6.85×10^5	104	5.28

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$K_1 = 2 K_{PUMP}$ PVA 120 pumps (2)

$$K_{PUMP} = \frac{30 \frac{REV}{SEC} \times 18.71 \frac{IN^3}{REV}}{2.16 \text{ IN}} \times \frac{2.16 \text{ IN}}{50 \text{ VOLTS}} = 14.22 \frac{IN^3}{SEC\ VOLT}$$

$K_K K_1 = 3.44 \text{ IN}^2\text{-SEC}$

$$K_K = \frac{3.44 \text{ IN}^2\text{-SEC}}{28.44 \text{ IN}^3/\text{SEC-VOLT}} = 0.1211 \frac{\text{VOLT-SEC}^2}{\text{IN}}$$

Accelerometer output (maximum) = 17 volts
at 800 in/sec² (preamp req.)
(2.08 G)

$K_v = \frac{K_{PO} K_1}{A}$, $K_{PO} = \frac{A K_v}{K_1}$

$$K_{PO} = \frac{18.18 \text{ IN}^2 \times 0.673 \frac{1}{SEC}}{28.44 \frac{IN^3}{SEC\text{-}VOLT}} = 0.430 \frac{VOLTS}{IN}$$

Without acceleration feedback

$K_v \approx 2\pi \times 0.096 \frac{1}{SEC} = 0.603 \frac{1}{SEC}$

$$K_{PO} = \frac{18.18 \times 0.603}{28.44} \frac{VOLTS}{IN} = 0.385 \frac{VOLTS}{IN}$$

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USE OF DUAL CYLINDERS TO PROVIDE RESONANT FREQUENCY CHANGE -

	(1)	(2)	(3)
OBTAIN RESONANCE AT	0.28 cps	0.56 cps	0.84 cps
CYLINDER AREA THEO.	6.06 IN^2	12.12 IN^2	18.18 IN^2
CYLINDER AREA (STD. CYL.)	4.90 IN^2	15.71 IN^2	20.63 IN^2

$$\omega_{N_3} = \frac{\sqrt{2}\,A}{\sqrt{VEM}} = 5.28 \frac{1}{SEC} (0.84\,cps) \qquad A = 20.63\ IN^2$$

$$E = 5.6 \times 10^{-6} \frac{IN^2}{LB}$$

$$V = \frac{2A^2}{EM\omega_{N_3}^2} = \frac{2(20.63)^2}{(5.6\times10^{-6})(112.7)(5.28)^2} \qquad M = 112.7 \frac{LB\text{-}SEC^2}{IN}$$

$$V = 48{,}400\ IN^3 \qquad L = 0.005 \frac{IN^3}{SEC\cdot PSI}$$

$$b = 27.8 \frac{LB\text{-}SEC}{IN}$$

$$\omega_{N_2} = \omega_{N_3} \times \frac{15.71}{20.63} = 4.02 \frac{1}{SEC} (0.64\,cps)$$

$$\omega_{N_1} = \omega_{N_3} \times \frac{4.90}{20.63} = 1.26 \frac{1}{SEC} (0.20\,cps)$$

$$\zeta_3 = \left[\frac{LM}{2A^2} + \frac{VEb}{4A^2}\right]\omega_N$$

$$\zeta_3 = \left[\frac{0.005\times112.7}{2(20.63)^2} + \frac{48{,}400\times5.6\times10^{-6}\times27.8}{4(20.63)^2}\right]5.28 = 0.0035 + 0.025$$

$$\zeta_3 = 0.028$$

$$\zeta_2 = \zeta_3\left(\frac{20.63}{15.71}\right)^2 \times \frac{15.71}{20.63} = 0.037$$

$$\zeta_1 = \zeta_3\left(\frac{20.63}{4.90}\right)^2 \times \frac{4.90}{20.63} = 0.113$$

$$Y = 28.7\ IN$$

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f (rps)	ω (rad/sec)	~~ω/ω_n~~	YωA (in³/sec)	YωA/Q	Q (in³/sec)	Q (GPM)	α
0.84	5.26	1.000	3130	17.8	176	45.7	90°
0.80	5.03	0.953	2980	9.0	331	86.0	30°
0.75	4.71	0.892	2790	4.8	582	151.0	13°
0.70	4.41	0.833	2610	3.5	746	194	9°
0.75	4.71	1.172	2130	2.7	788	205	166°
0.70	4.41	1.093	1988	4.7	423	110	158°
0.60	3.77	0.938	1700	7.5	227	58.8	30°
0.50	3.14	0.782	1417	2.7	524	136.2	8°
0.40	2.52	0.625	1137	1.7	669	173.8	3°
0.30	1.89	0.468	852	1.3	656	170.2	2°
0.20	1.26	0.197	567	1.1	516	134	1°

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$\omega\ (\frac{1}{sec})$	$\frac{MY\omega^2}{A}$ (psi)	α	$90-\alpha$	Peak Power Correction	Peak in-lb/sec	Power HP
5.26	4370	90°	0°	1.00	7.69×10^5	116.5
5.03	3970	30°	60°	0.76	9.98×10^5	151.2
4.71	3480	13°	77°	0.62	12.57×10^5	170.5
4.41	3040	9°	81°	0.58	13.15×10^5	198.2
4.71	4570	166°	-76°	0.62	22.3×10^5	338.0
4.41	4010	158°	-68°	0.67	11.72×10^5	177.5
3.77	2930	30°	60°	0.76	5.06×10^5	76.7
3.14	2030	8°	82°	0.57	6.07×10^5	91.9
2.56	1310	3°	87°	0.53	$4.6\ \ 10^5$	70.4
1.81	735	2°	88°	0.52	2.51×10^5	38.0
1.26	327	1°	89°	0.51	0.858×10^5	13.0

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SUMMARY OF CONTROL SYSTEM USING HYDRAULIC FLUID FOR ENERGY STORAGE

CYLINDER - 7" DIA, 5" ROD CYLINDER, ANNULUS AREA - 18.18 IN², DOUBLE ENDED

MAX ΔP - 4820 PSI

MAX PUMP FLOW - 261 GPM

SERVO PUMPS - TWO PVA-120, 1800 RPM INPUT

PEAK HP - 224 FROM PUMP

OIL VOLUME REQUIRED/SIDE - 42,000 IN²

MAXIMUM LOAD FLOW - 7.18 GPM

LOAD PEAK PRESSURE-FLOW PRODUCT - $133.0 \times 10^5 \frac{\text{IN-LB}}{\text{SEC}}$ (2020 HP)

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SYSTEMS FOR ALL AXES

AXIS	MASS $\left(\frac{LB\text{-}SEC^2}{IN}\right)$	FORCE (LB)	CYL. AREA (IN^2)	CYL SIZE	PEAK PRESSURE (PSI)	PEAK SPEED $\left(\frac{IN}{SEC}\right)$
2X	31.6	4,400	10.01	5 IN HD	439	25.9
2Y	58.3	8,100	14.74	5 IN HD	550	25.9
3X	31.6	5,600	10.01	5 IN HD	559	32.8
3Y	58.3	10,400	14.74	5 IN HD	716	32.8
4X	60.2	12,800	14.74	5 IN ID	867	40.2
4Y	86.8	18,400	15.08	6 IN HD	1220	40.2
5X	60.2	23,000	14.74	5 IN ID	1560	72.4
5Y	86.8	33,200	15.08	6 IN HD	2200	72.4
6X	61.7	32,000	14.74	5 IN ID	2170	86.7
6Y	112.7	51,800	18.85	7 IN HD	2750	86.7
7X	61.7	43,600	14.74	5 IN ID	2970	117.9
7Y	112.7	70,500	18.85	7 IN HD	3740	117.9
8X	69.9	56,200	14.74	5 IN ID	3810	151.8
8Y	112.7	90,900	18.85	7 IN HD	4820	151.8
9X	31.6	26,400	10.01	5 IN HD	2640	158.7
9Y	58.3	48,500	14.74	5 IN ID	3300	158.7

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Axis	± Stroke	Peak IN^3/SEC	Flow GPM
2X	4.7	259	67.3
2Y	4.9	382	97.2
3X	6.2	328	85.2
3Y	6.2	483	125.3
4X	7.6	592	153.7
4Y	7.6	605	157.2
5X	13.7	1068	278
5Y	13.7	1092	284
6X	16.4	1280	333
6Y	16.4	1634	424
7X	22.4	1740	452
7Y	22.4	2220	577
8X	28.7	2240	582
8Y	28.7	2860	743
9X	30.0	1570	412
9Y	30.0	2340	607

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DETERMINATION OF QUARTER WAVE LENGTH FREQUENCY FROM VELOCITY OF SOUND

$$f = c/4\ell$$

$$C = \sqrt{B/\rho}$$

$$B = 177{,}000 \ \frac{LB}{IN^2}$$

$$\rho = 54 \frac{LB}{FT^3} \times \frac{FT^3}{1728 \ IN^3} \times \frac{1 \ SEC^2}{386 \ IN}$$

$$\rho = 8.11 \times 10^{-5} \ \frac{LB\text{-}SEC^2}{IN^4}$$

$$C = 0.4697 \times 10^5 \ \frac{IN}{SEC}$$

$$f = \frac{11{,}740}{\ell} \ \frac{IN}{SEC}$$

f = QUARTER WAVE LENGTH FREQUENCY (CPS)

C = VELOCITY OF SOUND IN OIL (IN/SEC)

ℓ = LINE LENGTH (IN)

B = BULK MODULUS OF OIL (LB/IN^2)

ρ = MASS DENSITY OF OIL $\left(\frac{LB\text{-}SEC^2}{IN^4}\right)$

ASSUME 530 FT MAXIMUM PUMP - CYLINDER DISTANCE (6360 IN)

$$f = 1.85 \text{ cps}$$

APPENDIX TO SECTION III

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BRUTE FORCE POSITIONING SYSTEM

AXIS	(± STROKE-IN) ↓ MASS $\left(\frac{LB\text{-}SEC^2}{IN}\right)$	PEAK FORCE (LB)	PEAK SPEED $\left(\frac{IN}{SEC}\right)$	REQUIRED AREA AT 2500 PSI (IN^2)	CYLINDER SIZE (ROD ↓)	CYLINDER AREA (IN^2)	PEAK FLOW (GPM)
2X (±4.9)	31.6	4400	25.9	1.76	2"-STD-1"	2.357	15.85
2Y (±4.9)	58.3	8100	25.9	3.24	2½"-INT-1¾"	3.424	23.1
3X (±6.2)	31.6	5600	32.8	2.24	2"-STD-1"	2.357	20.1
3Y (±6.2)	58.3	10,400	32.8	4.16	3¼"-HVY-2"	5.154	43.9
4X (±7.6)	60.2	12,800	40.2	5.12	3¼"-HVY-2"	5.154	44.0
4Y (±7.6)	86.8	18,400	40.2	7.36	4"-HVY-2½"	7.666	65.3
5X (±13.7)	60.2	23,000	72.4	9.20	4"-INT-2"	9.424	177.2
5Y (±13.7)	86.8	33,200	72.4	13.28	5"-INT-2½"	14.735	278
6X (±16.4)	67.7	32,000	86.7	12.80	5"-INT-2½"	14.735	332
6Y (±16.4)	112.7	51,800	86.7	20.8	6"-STD-2½"	23.234	526
7X (±22.4)	67.7	43,600	117.7	17.42	6"-INT-3½"	18.653	571
7Y (±22.4)	112.7	70,500	117.7	28.2	7"-STD-3"	31.416	962
* 8X (±28.7)	67.7	56,200	151.8	22.5	6"-STD-2½"	23.374	912
8Y (±28.7)	112.7	90,100	151.8	36.1	8"-STD-3½"	40.644	1377
9X (±30.0)	31.6	26,400	158.7	10.57	5"-HVY-2½"	16.014	413
* 9Y (±30.0)	58.3	48,600	158.7	19.43	6"-STD-2½"	23.374	963

$$\text{PEAK FLOW (GPM)} = \frac{\text{CYLINDER AREA } (IN^2) \times \text{PEAK SPEED } \left(\frac{IN}{SEC}\right)}{3.85}$$

* REVISED CYLINDER SELECTION

8X	7" INT-4"-25.313	1013
9X	7"-HVY-5"-18.850	778

230
7-60

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AXIS	AVERAGE FLOW (GPM)	PEAK PRESSURE (PSI)
2X	10.07	1866
2Y	14.72	2370
3X	12.80	2380
3Y	28.0	2020
4X	28.1	2470
4Y	41.7	2400
5X	112.8	2440
5Y	177.0	2250
6X	212	2170
6Y	335	2230
7X	364 *	2340 *
7Y	612	2250
8X	582	2410
8Y	890	2240
9X	277	2640
9Y	647	2080

AVERAGE FLOW = 0.637 PEAK FLOW

* REVISED CYLINDER SELECTION

8X	646	2180
9X	769	1420

230
7-60

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SERVO VALVES FOR BRUTE FORCE SYSTEM

PRESENT HIGH FLOW SERVO VALVE PASSES 50 GPM AT 250 PSI VALVE DROP, 2.5 cps AT 45° WITH 300 PSI CONTROL PRESSURE

USING 1000 PSI CONTROL PRESSURE FREQUENCY WILL BE

$$2.5\ cps \times \sqrt{\frac{1000}{300}} \approx 4.5\ cps$$

FLOW AT (PLUS LINE AND PASSAGE DROP)

500 PSI VALVE DROP ≈ 70.7 GPM

3000 PSI VALVE DROP ≈ 173 GPM

MAXIMUM LOAD FLOW PER VALVE ≈

173 GPM × 0.8 = 138 GPM

USE 3" DIRECTIONAL VALVE AS 3RD STAGE FOR SA4

BORE DIA 4", FULL ANNULAR PORTING,

$$Q = \frac{100 \times 12.56\ in \times y \times \sqrt{P_v/2}}{3.85}\ GPM = 231\, y\sqrt{P_v}$$