DRIVE SYSTEM SELECTION
HTLTR OSCILLATORS

D. H. White

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DRIVE SYSTEM SELECTIONHTLTR OSCILLATORS

by

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Engineering Development Department

September, 1966

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HIGH TEMPERATURE LATTICE TEST REACTOR

ABSTRACT

Eleven different systems are evaluated in the selection of a powered drive system to move and control the mechanical oscillator mechanism for the High Temperature Lattice Test Reactor. The mechanism moves and positions reactor components inside the operating reactor. Objects weighing up to 1200 lbs are moved at high speeds and positioned to within tolerances of ± 0.010 in.

An electrohydraulic servo drive system utilizing a servo valve was chosen as the oscillator drive. Characteristics, advantages, disadvantages, and schematic diagrams are presented and discussed.

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DRIVE SYSTEM SELECTION
HTLTR OSCILLATORS

INTRODUCTION

This report describes the choice of a drive system to power the mechanical oscillator mechanism for use on the High Temperature Lattice Test Reactor (HTLTR).

This reactor is to measure reactor parameters at high temperatures (up to 1000°C) and is located at Pacific Northwest Laboratory, Richland, Washington. Certain tests and experiments scheduled to be run in this reactor require that the reactor cells, fuel train, and test samples be positioned, oscillated, and replaced during reactor operation. A mechanical mechanism which meets these requirements was designed and this mechanism when combined with an acceptable drive system is known as the HTLTR Oscillator. The mechanical mechanism converts rotary input torque into linear force and incorporates the proper attachment, cooling, and handling details, leaving many of the major operating and accuracy requirements of this machine to be functions primarily of the drive system. Hence, there was a need for an early selection of the proper drive system with balanced characteristics of accuracy, power, flexibility, and dynamic performance.

SUMMARY

The first step in selecting a drive system was to establish a set of criteria in which position accuracy, variable speed, cost, and reliability were major factors. Next, potential drive systems were tabulated utilizing known concepts, literature searches, and vendor surveys.

The investigation and analysis of these systems resulted in the selection of an electrohydraulic servo drive system (System 1). This is a feedback control system which utilizes the flexibility and accuracy of electronics to control the inherent high power output of a hydraulic system. Major components in the system include a servo valve, hydraulic piston-type motor and pump, and a digital shaft encoder.

The other drive systems investigated were:

2. Viscous-Shear, Slip Clutch Drive System
3. Hydrostatic Drive System
4. Variable Speed D.C. Motor Drive System
5. Hydraulic Stepping Motor Drive System
6. Hydraulic Cylinder - Differential Servo Drive System
7. Mechanical Amplifier Drive System
8. Other Systems

Electrical Clutch - Brake Drive System

Eddy Current Coupled Drive System

Electrical Stepping Motors

Cable Cylinder

It is not implied that this is an all inclusive list of possible drive systems, but because of scheduling and funding limitations only those listed were considered.

CRITERIA

The first step in the drive system selection was to determine all the operational functions required of the drive system and to establish a set of criteria with which to measure the relative merits of the various systems. The basis for these criteria were various documents and letters,^{(1)*} mechanism

*Numbers in parenthesis refer to references in Appendix A, "Sources of Information and Supply."

details, and personal judgment.

The drive system was to power the mechanism, to contain all control functions to and from the computer, and to meet the following criteria:

1. Have a rotary output, both clockwise and counterclockwise, with a controlled, adjustable speed while running and a positive holding force while stopped.
2. Cost under \$10,000 for two similar units.
3. Be a reliable, proven concept. Available "off-the-shelf," or with minimum modifications and development.
4. Position the reciprocal mechanism of both units to within +0.010 in. of an absolute scale. This meets both the initial placement tolerances and the repeatability tolerances required.
5. Move the following loads:
 - a. Light Duty Oscillator, ram - 100 lbs
load - 20 lbs
 - b. Heavy Duty Oscillator, ram - 100 lbs
load 1200 lbs

Using estimated coefficient of friction values of 0.5 for the load and 0.2 for the ram, results in the following output torques:

- a. Light Duty Oscillator 23 ft-lb
- b. Heavy Duty Oscillator 170 ft-lb

6. Have velocity and acceleration characteristics such as to assure the following:

- a. Light Duty Oscillator - Move 108 in. in less than 2-1/2 sec.
- b. Heavy Duty Oscillator - Move 36 in. in less than 5 sec.

Assuming constant acceleration for half of the distance and constant deceleration for the other half results in the following values:

	<u>Max. Speed</u>	<u>Acceleration</u>
a. Light Duty Oscillator	350 rpm (108 in/sec)*	36 rad/sec ² (108 in/sec ²)*
b. Heavy Duty Oscillator	77 rpm (24 in/sec)*	6 rad/sec ² (18 in/sec ²)*

*Linear equivalent based on a 6 in. effective cable drum diameter within the mechanism.

Since this computational technique results in the minimum acceleration and a changing velocity which is maximum at the midpoint (a situation difficult to control), most drive systems analyzed had characteristics of higher acceleration and lower maximum velocities.

7. Have a maximum travel of:
 - a. Light Duty Oscillator

9.1 rev.
(14 ft-2 7/8 in)*
 - b. Heavy Duty Oscillator

7.75 rev.
(12 ft-2 7/8 in)*
8. Have a control system compatible with the HTLTR computer. Position feedback signals (and possibly velocity signals) could be sent to the computer and position controls signals could be received from it.
9. Must fit within the available space, conform to the mechanism details, and have a total weight compatible with frame and carriage.
10. Have a high degree of interchangeability of components between the two units.
11. Connect to an inertia (motion transfer mechanism minus the ram) of 0.428 lb/sec²/in.

DRIVE SYSTEMS

The purpose of this investigation was to choose the best drive system for our use. The one chosen was the electrohydraulic servo drive system, Number 1, below. Systems based on a number of different concepts were considered and those appearing to have any promise at all were evaluated further. The systems we considered are discussed below in order of relative merit for our particular application.

1. ELECTROHYDRAULIC SERVO DRIVE SYSTEM

This is the system we chose to drive the oscillator. The system was evaluated with minimum delay and with maximum assurance of performance because of experience gained in a large number of past industrial applications. There are several different types of commercially available electrohydraulic servo systems. The one felt to be most responsive to our need and also most economical, utilizes a servo valve.

*Linear equivalent based on a 6 in. effective cable drum diameter within the mechanism.

Principle and Description

The electrohydraulic servo system is a combination of the high torque and high speed characteristics associated with hydraulic power systems and the response, accuracy, and flexibility characteristics associated with electronic control systems. The key to this system is a servo valve which uses electronic signals to control the flow of fluid to the motor.

Figure 1 is a schematic block diagram of this system and a simplified operational description follows. First, a flow of fluid at high pressure is provided by a conventional hydraulic power supply unit. Second, closed loop, electronic feedback signals of oscillator position are sent to the digital computer which compares the desired position with the actual position and generates an error signal for control. Third, both the control signal and the pressurized fluid go to the servo valve, the heart of the system, where the electronic control signal modulates the flow of fluid to the hydraulic drive motor. The servo valve is close coupled to the motor through a manifold plate which has motor protection devices and circuitry and minimizes fluid compression effects for faster feedback response.

Advantages and Disadvantages

The major advantages of this type of system are relative low cost, availability and satisfactory past experience, extremely fast response, small size and weight attached to oscillator, the ability to vary speed and torque independently, by changing the flow and pressure, respectively; high positional accuracy, and long life. The disadvantages are minimal but include high noise levels and a number of auxiliary components which must be located at specific locations on and near the oscillator.

Details

Systems of this type are available commercially.⁽²⁾ Additional details and performance characteristics of one brand⁽³⁾, are as follows:

The hydraulic power package has a 10 gpm output at 2000 psi, a 20 gallon reservoir, and a variable displacement, pressure compensated pump directly connected to a 3 phase, 440 v, 10 Hp, electric motor. The servo valve is a two-stage type with an internal linkage of 2.5:1, an adjustable dither from 60 to 400 cps, and a maximum dead band of 0.2% open loop. A constant displacement, piston type hydraulic motor is used. This motor has a variable

FIGURE 1
Schematic Block Diagram - Electrohydraulic Servo System

speed of 0 to 3600 rpm, clockwise and counterclockwise, and develops 15 lb-in. of torque per 100 psi pressure while displacing 1.0 cu. in. of fluid per revolution.

The manifold plate connecting the servo valve to the motor is such that the volume of oil under compression is less than 2 cu. in. This plate also incorporates internal cross-relief valves and a cross-line variable orifice which are protection devices assuring proper motor performance. An electronic operational amplifier is required between the computer and the servo valve. This amplifier is solid state with a high gain, linear output and has controls for gain, balance, and dither. A commercially available⁽⁴⁾, digital, shaft encoder generates electronic feedback signals corresponding to the linear position of the oscillating mechanism. A small cable on a spring loaded drum couples the encoder to the oscillating mechanism and converts linear input of the mechanism to rotary output for the encoder. A pulse is produced every revolution as well as a pulse for every 0.0063 in. of linear travel, with separate outputs for forward and reverse. Very flexible control circuitry for the PMACS computer* has been designed giving options of an up-down counter control with minimum computer time or a direct control mode whereby almost any complex movement pattern may be programmed for direct computer control.

2. VISCOUS SHEAR, SLIP CLUTCH DRIVE SYSTEM

This system utilizes clutches which work against each other. The basic simplicity of such an arrangement is appealing and this concept was evaluated in some depth before the final drive system was chosen.

Principle and Description

The principle of operation is based on control of clutch slip. Slip is defined as a difference in speeds of the clutch plates. The amount of energy transmitted by a clutch is inversely related to the slip. For example, when the output clutch plate is turning at half the speed of the input, half of the input energy is being transferred and half is dissipated as heat. If the amount of slip can be controlled, the amount of energy, the speed, and the position can be controlled. This system utilizes counter rotating, opposed clutches with controllable slip characteristics to obtain the desired output motions and accuracies.

*PMACS stands for Programmed Measurement and Control System, the main control and measurement system for the HTLTR complex. The heart of this system is a digital computer.

The outputs of these two clutches are connected to a common output shaft through gears such that the input rotation tends to drive one direction through one clutch and the opposite direction through the other clutch. Figure 2 is a schematic block diagram of this system. Any type of clutch may be arranged in this manner but for routine operation with one or both clutches slipping, the clutch must dissipate heat at a high rate and the wear on the clutch faces must be low.

One type of clutch which satisfies these requirements uses the viscous shear characteristics of a thin film of oil between specially shaped clutch faces.⁽⁵⁾

This clutch uses air pressure to adjust the clutch plate spacing, hence, an electrical to air transducer is required between the control electronics and the power package. While the clutch packs are standard, the gears, timing belts, and housing are unique for our application.

A clutch pack consists of alternate steel and bronze discs. The special groove pattern on the sintered bronze discs maintains a thin, but positive oil film. Torque is transferred between discs by the viscous shear of the oil. The amount of torque transmitted is a function of clamping pressure on the disc stack. Clamping pressure is supplied by an internal air piston, hence, change of air pressure modulates the transmitted torque. The positive oil flow from the clutch hub also removes the heat generated as a result of the speed difference between input and output shafts.

Advantages and Disadvantages

The advantages of this system includes its simplicity and the minimum amount of associated equipment that is required. The use of standard clutch packs and a price well within our budget were also considered advantageous.

These units were quite heavy (150 lbs and 300 lbs, respectively), fairly large, and the whole unit would have to mount on the oscillator mechanism frame. These three items were the major disadvantages. Serious problems with reliability were caused by this unique arrangement.

FIGURE 2
Schematic Block Diagram - Viscous Shear, Slip Clutch Drive System

3. HYDROSTATIC DRIVE SYSTEM

Another system which has many of the desirable characteristics of a hydraulic system is the hydrostatic drive system. These systems are in extensive use and are widely available.⁽⁶⁾

Principle and Description

Rotational direction and speed is a function of the flow direction and volume of the supply fluid. The hydrostatic drive is based on this principle. The changing flow from a variable volume hydraulic pump runs a hydraulic motor.

Figure 3 is a schematic block diagram of this system. The hydraulic pump and motor are directly connected. While a number of different arrangements exist (such as both pump and motor variable, or motor only variable), the concept that was evaluated was that the pump, only, was varied. The pump plungers are attached to a movable wobble plate and movement of this plate controls the pump output and hence the motor rotation. Figure 4 is a schematic illustration of the wobble plate-pump-motor connections and operating interdependency.

It is standard industrial practice to attach a hydraulic cylinder to the wobble plate arm and to then control this cylinder with a hand valve, a solenoid, a torque motor, servo valve, cams, or a servo motor. These control methods were judged too inaccurate for our requirements.*

The wobble plate plunger has an excursion of 2-3/4 in. and requires from 100 to 200 lb. of force to move. After evaluating a number of ways to control this wobble plate, a stepping motor driving through a torque amplifier (for example, a ball screw, gear reducer, or mechanical amplifier, see Section 7) was chosen to move the wobble plate plunger.

In addition to the aforementioned items, the system would require other standard hydraulic components such as a reservoir, filters, relief and check valves, and an electric motor.

* It should be noted that an electrohydraulic servo valve will position this cylinder accurately, but since it is capable of performing the whole job alone (see Section 1), it is uneconomical to use this valve as pilot control in this case.

FIGURE 3
Schematic Block Diagram - Hydrostatic Drive

FIGURE 4
Wobble Plate Schematic

Advantages and Disadvantages

The major advantages of the hydrostatic drive system is high torque, smooth, fast response, and costs just below our budget allotment. The main disadvantages are associated with control and accuracy. The most accurate control actuator in standard practice is a servo valve*, which will give rotational accuracies of about 5°. Our tolerances required that our control positions be 15 times more accurate than this. It is felt that the stepping motor - torque amplifier control system would be satisfactory, but it would require complete development and proof testing of rather expensive components.

The fluid between the pump and the motor is part of the control system and any volumetric change, either from fluid compression or hose expansion, will increase the time required for the total system to respond to the controls. The maximum system response requires close motor-pump coupling with both pump and motor on the oscillator mechanism and connected with rigid tubing. On the other hand, the best mounting arrangement in our application is to mount the pump on the carriage with hose connections to the motor. The need to choose between these two alternatives results in a serious loss of design and operational flexibility and becomes a disadvantage.

4. VARIABLE SPEED D.C. MOTOR

The major method of transferring energy throughout industry is by electricity and the electrical motor has become the standard prime mover. Because of the choice of power characteristics and the control flexibility available, a variable speed D.C. motor system was evaluated. Motors of this type are commonly available from any number of suppliers and manufacturers. (7)

Principle and Description

The speed of a D.C. electrical motor can be varied by regulating current in the armature, or field, or both. For our purposes the third arrangement, the compound wound motor, was chosen. The silicon controlled rectifier (SCR) was chosen as the best control system over motor-generator sets and tube type rectifiers because of cost, reliability, and control advantages. Variable speed D.C. motors do not operate smoothly near zero rpm. The recommended ratio of high speed to low speed varies from 15:1 to 20:1 depending on manufacture

*This servo valve should not be confused with the electrohydraulic servo valve described in Section 1.

In order to satisfy both the accuracy and the speed criteria, two parallel drive paths per system were required. Figure 5 is a block schematic diagram of this arrangement. It should be noted that the two drive paths, the high speed and the low speed, are controlled independently and their mechanical output is summed mechanically at the differential transmission. See Appendix B for a short discussion of mechanical differentials.

Advantages and Disadvantages

The major advantage is that all components are standard and easily obtained. There are a number of disadvantages, however. The double control system would be fairly complex and costly. Preliminary calculations indicate that the gear reducers for the low speed path could run as high as 400:1. This implies very precise gears if such a large reduction is to meet our backlash requirements. And even with this gear reduction, the motor would have to be stopped with an accuracy of 1/10 of a revolution to meet the requirements -- a possible, but not necessarily simple operation. The final disadvantage was that because of the number of components required, the costs would exceed our budget allotment.

5. HYDRAULIC STEPPING AMPLIFIER

During the evaluation of drive systems, a new concept was brought to our attention. This drive system, known as a hydraulic stepping amplifier, attempts to get discreet stepped rotary output from a hydraulic motor. The discreet steps would provide the ability to directly control the inherent high power of hydraulics. The concept has been assigned to an engineer-manufacturer by the inventor and a patent is pending.⁽⁸⁾ Demonstration units have been built, but no system was in use at the time of the evaluation.

Principle and Description

This proposed system is based on the amplifier principle -- power with control signal characteristics is the output from a device whose inputs are control and power. In this case, the power is produced by a standard hydraulic package and the signal is produced by an electrical stepping motor. The stepping motor was picked because of its compatibility with the digital computer.

FIGURE 5
Schematic Block Diagram - Variable Speed, D.C. Motor

The number of cylinders in the hydraulic motor determines the number of steps required for one revolution. The practical limit of 15 cylinders in our case means that one motor system was incapable of satisfying both the speed and accuracy requirements. The dual system block schematic that was chosen for evaluation is shown as Figure 6.

The "big step" path would be used for high speeds and approximate position while the "small step" path would give the positional accuracy. The differential would sum these outputs mechanically.

The transmitter portion of an amplifier pair resembles a rotary valve with the transmitter output feeding on individual lines to the respective hydraulic motor cylinder. An input of X increments at the transmitter results in an exact output of X increments of motor rotation.

Advantages and Disadvantages

Originally, it was felt that this arrangement would produce discreet units of motion with a high power output. However, the accuracy and high speed requirements were not compatible with the relatively large stepping increments, thus requiring a dual drive system. This doubling of equipment which complicates the system and lowers reliability was the major disadvantage. Other disadvantages include the lack of operational or experimental experience, the need to run separate hydraulic lines to each cylinder, and the cost estimate exceeded the budget allocations.

6. HYDRAULIC CYLINDER - DIFFERENTIAL SERVO SYSTEM

After it became obvious that the Hydraulic Stepping Amplifier had more disadvantages than advantages, the vendor who had proposed that system proposed a new system which can best be described as a hydraulic cylinder - differential servo system. Their proposal was to engineer, develop, build, and test this drive for us. This concept is probably not too practical for our application, but it was one of the first concepts considered which combined hydraulic power and a servo feedback control. Its major contribution in this study is that it was the first step down a course of investigation which resulted in the choice of an electrohydraulic servo system.

FIGURE 6
Schematic Block Diagram - Hydraulic Stepping Amplifier

Principle and Description

The key component in this system is the cam and valve which is driven through a differential. Figure 7, a schematic diagram of this system, shows this cam and how it is connected to the other components. One input to the differential is connected directly to the drive shaft and the other input is coupled to a stepping motor. The cam drives a special valve which resembles a servo valve in both construction and action. A null point is established at the sloping face of the cam by the valve which balances the two opposed hydraulic cylinders. As the control moves the cam from this null point, the resulting cylinder pressure imbalance drives the cam, and mechanism, to a new null position.

Except for the special valve, cam, and mountings, most components are standard, including the conventional hydraulic power supply system.

Advantages and Disadvantages

The advantage of this system is that accurate positions are possible with standard, non-precision components. This tends to be true for most servo systems. The use of cylinders also eliminated the need for limit switches, which is a minor advantage.

The disadvantages are numerous. First, the arrangement of opposed, compound looped cable drive connected to the cylinders is awkward and requires more space than was available. Second the most critical part of the system, the valve and cam, would require complete development. Third, the required double stepping motor inputs are expensive and complicated. Next, the entire mechanical feedback system is complex and unique, resulting in low reliability. Then, to obtain the desired speeds and forces, large hydraulic pipe and cylinders are required. Finally, the cost exceeded our budget by a large factor.

7. MECHANICAL POWER AMPLIFIER

A unique device which is entirely mechanical in nature and based on the principle of amplification was brought to our attention. The concept was appealing but closer evaluation showed that it was incapable of meeting our requirements. These units are available as standard products.⁽⁹⁾ Inquiries

FIGURE 7
Schematic Diagram - Hydraulic Cylinder - Differential Servo System

were made concerning larger units and greater power generation by cascading the output of one amplifier into another. The vendors reply was that larger units are planned, but unavailable until long after our deadline, and that cascaded systems in the past had been only partially successful.

Principle and Description

The principle of operation is similar to a powered capstan drive. As illustrated in Figure 8, Capstan Drive Elements, a control input on one end of the rope is amplified to produce a power input on the other end.

The Mechanical Power Amplifier (MPA) replaces the rope coils by a spiraled steel band and obtains clockwise and counterclockwise rotation with two, side-by-side drums mounted as shown in Figure 9, Simplified Elements. The drums are rotated continuously by a standard electric motor. The bands are tightened on either one drum or the other by the input device. Inputs can be manual, a stepping motor, servo receivers, or servo motors -- in fact, most anything which has control characteristics but inherent low power.

The speed out is equal to the speed in, the phase is synchronized, and the torque out is 80 times the torque in. The response of the amplifier itself has been measured at 0.005 sec. and present units deliver 0 to 400 rpm clockwise or counterclockwise rotation.

Advantages and Disadvantages

The advantages of this system is the simplicity of the concept and mechanism, assuming that an input device with the required control characteristics is available. There are two major disadvantages. First, our requirement for a dual stepping motor input with a large step path for speed and a small step path for accuracy. Second, the amplification available in the present units is not large enough to handle the oscillator so two would have to be cascaded, the output of the first feeding the input of the second. The manufacturer discouraged this type of an arrangement.

8. OTHER SYSTEMS

In addition to the above-mentioned systems, which were analyzed at some length, a number of other concepts were considered but rejected because of significant disadvantages. A few of these are discussed below.

FIGURE 8
Capistan Drive Elements

FIGURE 9
Simplified Elements of a Mechanical Power Amplifier

Electrical Clutch-Brake System

A system similar to the viscous shear clutch, number 2 above, but utilizing standard, electrically powered, dry clutches and brakes was investigated. Because of our requirements of wide speed adjustment and variable positioning, a system of this type would have to run a majority of the time with the clutch slipping. We could find no clutch of this type capable of adequately dispersing the heat generated with this kind of operation.

Eddy Current Coupling

A coupling which transmits energy across an air gap by utilizing a magnetic field formed from induced eddy currents is commercially available from a number of suppliers. An arrangement somewhat similar to the viscous shear clutch system but utilizing the eddy current couplings was considered. This concept had two disadvantages. First, the heat problem similar to dry clutches as discussed above. Second, a problem of how to obtain control. Original contacts with makers of these devices were discouraging on both counts, however, later literature was obtained which suggests that this concept probably had more merit than we originally believed, but certainly still not competitive with the chosen drive system.

Stepping Motors

There were reports of the existence of large stepping motors capable of driving these mechanisms directly. An effort was made to see if such a system existed. While power type stepping motors exist -- we could find none to meet our requirements. It was further determined that because of the combination of speed and accuracy, even a control type stepper with a usable combination of steps per revolution and stepping rate was unavailable.

Cable Cylinder

The oscillator is a reciprocating mechanism which has to fit a very tight space. A cable cylinder is available⁽¹⁰⁾ which produces a linear motion with a working length only slightly less than the total available space. Because of these characteristics, a cable cylinder drive appeared

to be a reasonable drive system candidate.

Figure 10, Schematic Cable Cylinder, illustrates the arrangement which allows usable travel over most of the mounting length. The rods of a standard double-ended cylinder were replaced by a nylon coated cable. The nylon cable covering permits a seal to be effected at the cylinder-cable interface and hence is very necessary. However, our anticipated temperatures are higher than nylon can stand. No acceptable solution for this major disadvantage could be found.

Other problems were fluid compression effects in system response and significant space limitations in the mechanism design.

REMARKS

This report discusses those concepts considered as possible solutions for a specific problem. No claims are made that all possible concepts were considered or even that all aspects were investigated for those concepts that were considered. The type and depth of analysis possible was significantly influenced by tight financial and time schedules.

The sources of supply and information listing in the Appendix are considered minimal only. In most cases there are numerous other suppliers.

It is hoped that this report might prove to be of some value to others with similar drive requirements by providing a partial list of ideas for initial consideration.

FIGURE 10
Schematic - Cable Cylinder

APPENDIX ASOURCES OF SUPPLY AND INFORMATION

- (1) a. Informal oscillator criteria, R. J. Hoch, 4/1/64 and revised 4/10/64;
b. HWS-7560, Procurement Specification for Oscillator Drive System -
Building 318; c. Comment letter about oscillators to AEC from
D. L. Ballard, 4/25/65; and d. verbal instructions by AEC to Equipment
Development personnel.
- (2) a. Vickers, Inc., Troy, Mich.; b. Racine-Vickers Armstrong, Inc., Racine,
Wis.; and c. Moog Servocontrols, Inc., East Aurora, New York.
- (3) Vickers, Inc.
- (4) a. Dynamic Research Corporation, Stoneham, Mass.; b. Wayne-George Corp.,
Newton, Mass.; c. W&L. E. Gurley, Troy, New York; and d. Norden,
Norwalk, New York.
- (5) Force Controls, Inc., New Castle, Ind.
- (6) New York Air Brake Co., Watertown, New York; b. Vickers, Inc., Troy,
Mich.; c. Racine Hydraulics, Racine, Wis.; and d. Denison Engineering,
Columbus, Ohio.
- (7) a. Reliance Electric, Cleveland, Ohio; b. General Electric, Schenectady,
New York; c. Westinghouse, Pittsburgh, Pa; and d. Louis Allis,
Milwaukee, Wis.
- (8) Wilkins and Associates, Inc., Tacoma, Wash.
- (9) Seneca Falls Machine Corp., Seneca Falls, New York
- (10) Tol-O-Matic, Inc., Minneapolis, Minn.

APPENDIX B
MECHANICAL DIFFERENTIALS

A mechanical differential is shown in Figure A. Such an arrangement of bevel gears acts as a mechanical summation device, among other things.

There are two inputs and one output. An investigation of this gear arrangement shows the following: (assuming that input speeds are constant and equal.)

- a. Input shafts rotating in the same direction causes the output shaft to rotate at the input speed.
- b. Input shafts rotating in opposite directions result in a stationary output shaft.

In the schematic block diagrams of this report, a schematic symbol for this differential, as illustrated in Figure B, is used for simplicity and symmetry.

FIGURE A
Mechanical Differential

FIGURE B
Schematic - Mech. Differential

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