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TITLE Alternate Valve Studies			
ABSTRACT This report is the summary of an investigation conducted for KAPL on alternate valving techniques. The valve handles liquid metal and is installed for emergency use.			
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CONCLUSIONS The following requirements were investigated: 1. Alternate valve closure mechanisms. 2. Positive seal, other than bellows, between the internal valve mechanism and the external driving force. 3. Mechanical over-ride mechanisms. These requirements can be fulfilled through the application of one or more ideas presented in the proposed mechanisms. In view of the frozen sodium seal and the magnetic drive, it is recommended that these be investigated more fully.			

For list of contents—drawings, photos, etc. and for distribution see next page (FN-610-2).

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CONTENTS PAGE

II

CONTENTS OF REPORT

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LIST OF ILLUSTRATIONS

<u>Figure</u>	<u>Title</u>
1.	Composite Sheet
2.	Force-Displacement Curve, 8" Valve
3.	Air Convection Freezing Curve
4.	Directly Operated Valve
5.	Slotted Rotor Valve
6.	Hydraulic Valve
7.	Piston Locking Device, Rotor Operated
8.	Piston Locking Device, Solenoid Operated
9.	Hydraulic Toggle Valve
10.	Magnetic Disc Toggle Valve
11.	Toggle Force-Displacement Curve
12.	Emergency Shaft Seal
13.	Mechanical Over-ride, Rotary Motion

~~IV~~

TABLE OF CONTENTS

	<u>Page</u>
I INTRODUCTION	1
II PLAN	1
III WORK	2
A. Search for Means	3
B. Evaluation of Means	3
C. Selection of Best Means	10
D. Analysis of Force-Displacement Relationships	11
E. Proposals	12
IV CONCLUSIONS	17
V APPENDIX	19
A. Valve Stem Size (Prop. #1)	19
B. Stem Column Size	20
C. Air Convection Freezing	20
D. Breaking Sodium Seal	22
E. Slotted Rotor Size (Prop. #2)	23
F. Hydraulic Piston Size (Prop. #3)	25
G. Closing Time Hydraulic Valve	25
H. Toggle Force-Displacement Curve (Prop. #4)	25
J. Toggle Drive-Disc Size	26
K. Toggle Linkage Friction Study	26

I INTRODUCTION

The Knolls Atomic Power Laboratory has need for a valve to stop the flow of liquid metal. Leakage cannot be tolerated either around the valve seat or to the outside. A globe type valve has been found satisfactory for shutting off flow, provided sufficient force is applied to the valve disc. However, a conventional valve is subject to leakage around the valve stem. For this application, conventional valve stem packings are totally unsatisfactory.

The problem, therefore, was to propose one or more means of accomplishing the following on an 8" globe type valve:

1. Provide displacement of the valve disc equivalent to 4" lift.
2. Apply 20,000 to 30,000 lbs. seating force.
3. Accomplish parts 1 and 2 through a positive seal other than bellows.

The solutions to this problem were to be carried only through the investigation stage. The proposed schemes must offer critical physical dimensions of the key component parts. An estimate of magnitude of forces, motion, and power required must also accompany each proposal.

II PLAN

The study was broken into five separate phases. These phases were as follows:

- A. Search for Means: Search for as many means as conceivable for obtaining force, displacement and seal.
- B. Evaluation of Means: Evaluate the schemes which appear on the composite sheet in order to determine their relative merit.

- C. Selection of Best Means: Select two or three schemes from the list which bear investigation and detail analysis. A globe-type valve seating was selected as the type for further study because it was felt that only in this type of valve could certain leakage requirements be met. Consequently, the study thereafter concerned itself principally with means of actuating and seating a globe-type valve. Tests had indicated that large forces were required to obtain the necessary minimum leakage rates. As a design number, 25,000 lbs. was selected for use in the study.
- D. Analysis of Force-Displacement Relationships: Determine how the forces on the valve disc vary with respect to the displacement.

E. Proposals:

- #1 - Directly Operated Valve
- #2 - Slotted Rotor Valve
- #3 - Hydraulic Valve
- #4 - Magnetic Disc Toggle Valve
- #5 - Mechanical Over-rides

III WORK

The various phases outlined in the previous section are discussed in this section of the report. The details of these phases are expanded to show the reasoning leading to the conclusions.

A. Search for Means

The search for means began with several group meetings. These meetings consisted of a group of people who were interested in contributing to the problem. The objective of these sessions was to gather ideas, unprejudiced by judicious thinking, which could be directed toward solutions to this problem.

The results of the group meetings were classified and tabulated on a sheet as shown in Fig. 1. The first column lists various methods of valving techniques for stopping flow. The remaining columns segregate the various methods into means of obtaining force and displacement. The various categories are air, gas, heat, electrical, hydraulic, impact, mechanical, and pressure. All of these means preclude the bridging of a positive barrier in the transfer to energy from without the valve body to within.

B. Evaluation of Means

This discussion gives a brief description of the various items given in Fig. 1. The reasons why each scheme was either retained or rejected are also given. The discussion is arranged in the order in which the columns appear on Fig. 1.

1. Means of Closure

Most of the means listed are familiar to the reader. The more obscure ones are the sleeve, grid and slide valves. The sleeve valve is one similar to a stopcock used in laboratory glass equipment. The grid valve consists of a series of strips of material to cover parallel slits and is used in steam or pneumatic applications. The slide valve is the one used on steam locomotive cylinders.

Tests at KAPL prove that extremely high unit pressures between sealing surfaces are necessary to stop the flow of liquid metal. This factor alone eliminates all the means of closure except globe, gate, and freezing. The necessary dimensional tolerances of a gate valve are very difficult to make and to maintain under these conditions. For this reason, the globe valve is more suitable than the gate valve. However, one must not overlook the fact that any of the valves listed could be used to impede the flow in order to freeze the liquid at the valve or at some convenient

point in the pipe. However, this particular approach requires an enormous rate of heat transfer to either stop or release flow. The transfer of such quantities of thermal energy not only requires large auxiliary equipment but introduces an undesirable time lag. Furthermore, the ultimate user of the valve would like to be assured that when the procedure of valve closure has been rendered, the liquid metal will absolutely not flow.

2. Air

Frequently, air is used in the drive and control mechanisms for valving operations. Air is also used for seal purposes by introducing it, under regulated pressure, to oppose the liquid pressure and its subsequent flow. The use of air was rejected for this application because of its effect on the piped liquid. The liquid metal oxidizes very readily. This oxide is an undesirable material in the system.

3. Gas

Gas is used in valving systems just the same as air. An inert gas such as nitrogen or argon could be considered for this purpose since it does not react with sodium. However, other factors prohibit the use of a pneumatic system. These are as follows:

- a. The gas must be mechanically acted upon to actuate the valve through some positive seal. The seal introduces complications and is a possible source of contamination.
- b. Relatively large displacements are necessary to actuate valving mechanisms. This is not easy to accomplish through a positive seal.
- c. Very likely, a supply of the gas must be kept on hand for replenishment. This imposes an undesirable problem upon the user.
- d. The gas is apt to be caught in the liquid stream, under an accidental condition, and could be detrimental to the operation of the system.

4. Heat

Since the valve is to control a system carrying liquid metal at temperatures of around 850° F, the utilization of this heat energy seems to be an economic means of valve operation.

The natural phenomena of thermal expansion of metals and gases offers the most direct method of converting heat energy to mechanical energy. This initial change in volume of the substance being thermally expanded could then be utilized to actuate a linkage, gear train or cylinder.

Since the valve lift must be on the order of 4 inches, the power source, ie., the expanded material, must have an expansion of sufficient magnitude to accomplish this lift. Of the four sources proposed (liquid, metal, bi-metal and gas), bi-metal and gas more closely approach this requirement. The expansion of metals and liquids are of such small magnitude so as to be unusable in this application. However, by use of a bi-metal strip or thermally expanded gases, the problem may be met. Examining these two methods, the expansion of gases is rejected because of:

- a. possibility of this gas (though inert) to enter the system and cause trouble through accumulation over long period of time.
- b. the large size piston needed and high temperatures (over 850°) to be attained in order to gain enough force to close the valve.
- c. the additional power to heat this gas as well as its inherent time lag.

The bi-metal strip has these drawbacks:

1. Its exact rate of movement and its magnitude are dependent upon the heat available and control of this heat.

2. If the valve is to remain closed for any period of time, there is danger of excessive creep taking place in the bi-metal due to its highly stressed condition at elevated temperatures and thus causing valve leakage.
3. In emergencies, the valve may have to be closed in a very short period of time (say 10 seconds). A longer period would probably be required to cause this bi-metal strip to close the valve. This time factor alone is the prime difficulty which must be overcome in a thermally operated valve. Since this valve is of the emergency type, time is of the essence. Any thermal system will of necessity require a certain amount of time before its expansion will accomplish a valve closure. Another method of applying thermal energy would be through sodium "steam". However, certain difficulties of this proposal are at once apparent. Sodium vaporizes at a temperature of 1621° F. Such temperatures would require materials with properties as yet barely attained by present day metallurgists and ceramicists. There is also the problem of adding enough energy to the liquid sodium to vaporize it. Therefore, thermally operated valves are rejected as unsuitable for this particular application.

5. Electrical

Operating the valve electrically seems to offer a way of circumventing the seal problem. Electricity can be applied in many ways, but perhaps the most applicable to this problem is to transfer electrical energy through the seal by magnetic means.

The first method considered was a "canned rotor" motor. The rotor of an induction motor would be encased within the valve body and its field rotated mechanically about it. The major drawback to this method is that the magnetic field must travel through the "can" in order to reach the rotor, resulting in very low efficiencies. The problem of temperature and high speed effects upon the bearings must also be considered. If the motor requirements could be reduced in some way, the "canned rotor" motor has good possibilities.

A solenoid presents another method of valve actuation. By using two sets of windings, a "push-pull" solenoid could be effected to operate the valve. In view of the high seating forces required, however, the windings would require special consideration. Since the current would only be applied for a short time, the coil size may be reduced by eliminating the extra space required for cooling. In any case, solenoids offer excellent possibilities as component parts. They may be employed as locking devices or to actuate small valves in hydraulic applications.

A form of magnetic clutch offers a way of overcoming the low efficiencies in "canned" devices. By surrounding the can with magnets, pull could be exerted on a series of discs which would in turn operate the valve upon displacement of the outside magnets (see Fig. 10). Permanent magnets could be used to provide the magnetic field. The disc material must be such that it retains ferromagnetic properties at elevated temperatures. Therefore, a valve actuator can be effected which could operate through mechanical force-reducers such as screws and linkages.

The same force which causes a current carrying conductor to move in a magnetic field may be applied to the liquid metal in the form of an electro-magnetic pump. Since no bearings or moving parts are required, the pressure generated can be applied to some suitable valve drive such as a piston or turbine (see Fig. 9).

Our problem is one of converting electrical energy to mechanical energy. This very thing is accomplished in magneto-strictive transducers. However, for our particular application, this method is unsuitable because:

- a. the magnitude of motion is so small as to be almost negligible (40 to 60 parts per million).
- b. at elevated temperatures, the magneto-strictive properties of metal greatly diminish. Piezoelectric transducers also convert electrical energy into mechanical energy. The piezoelectric effect is much more pronounced than the magneto-strictive; however, the resultant motion is too small to be of any practical value. Furthermore, the amount of energy which can be converted in this manner is almost negligible.

6. Hydraulic

Sodium in the liquid state is similar to water and as such would be suitable as a hydraulic fluid. By using sodium, danger of system contamination is averted.

In any hydraulic system a differential pressure must be generated to actuate a piston, bellows, or other mechanism. The pressure drop in the sodium loop is not available so we must use auxiliary means. Both positive displacement, axial flow, and centrifugal-type pumps were considered. All mechanical pumps require a driving motor which presents a potential seal problem. Metal parts operating in sliding contact in liquid sodium show unfavorable galling tendencies. This wear on mechanical pumps would greatly decrease their usable life. In view of these difficulties, an electro-magnetic pump seems to offer the solution, as it would require no seals and no moving parts. Therefore, a valving system utilizing hydraulic principles with an electro-magnetic pump as the pressure source offers one method of valve sealing and operation (see Fig. 9).

Torque converters were considered as a means of achieving a high-speed low-effort output for raising and lowering the valve and a low-speed high-effort output for seating and unseating the valve. However, this means of energy conversion was rejected because of its complexity. We felt that a much simpler means of achieving the same end could be found.

7. Impact

Since the liquid in the pipelines is in motion, the thought arose that perhaps the inertia or impact force of this liquid could be utilized. However, it is immediately obvious that the liquid is not always in motion and so would not be a reliable energy source.

Other means than this could be used to harness impact forces of bodies in motion. One of these means would be to store up energy in a flywheel and use this to operate the valve. Another would be to transfer motion and energy by jarring a metal diaphragm which, in some way, is linked to the valve disc. In general, noisy operating systems, jarring or hammering devices do not seem advisable. Furthermore, a large diameter, relatively thin diaphragm exposed to the internal pressure seemed an invitation to failure.

High velocity parts, such as flywheels, must operate in the liquid metal with consequent high windage losses, making their use prohibitive. The flywheel, if operated externally, would still need a seal to enter the valve proper and, as such, offers no outstanding contribution to the problem.

8. Mechanical

The field of "mechanical" means for valve closure embraces a wide variety of methods. Examples of "mechanical" means are inclined planes, linkages, springs, etc.

As stated before, the main problem is one of sealing a shaft which will transmit the desired force-displacement characteristics to the valve disc. The most direct means, of course, is through a simple screw thread, and providing a seal by freezing any sodium that endeavors to leak out. This method appears to be the most feasible approach to a solution (see Fig. 4).

The use of springs is governed by the ability of the material to withstand high temperatures over long periods of time without excessive creep. This means low stresses in any spring which is used. Leaf springs lend themselves most readily to this application, so perhaps they may be employed as a component of some valving arrangement.

Turbines, running on the piped fluid under pressure, offer a means of actuating the valve disc, but due to the torque requirements, would be too large and complex as a main power source. Possibly they could be used as auxiliaries, but in any case, bearing and material problems must be faced.

Flexible sections such as diaphragms of thin metal offer means of transmitting force to the valve disc and, at the same time, provide positive seal. In order to transfer practical magnitudes of motion through a diaphragm, the diameter must be quite large. Large diameters and thin sections exposed to the internal pressure of the liquid metal are undesirable.

Force multiplying linkages show promise as a means of reducing the initial applied force required for valve operation. Force is gained at the sacrifice of linkage displacement. Such linkages should be kept as simple as possible to eliminate potential failure points.

9. Pressure

In order for fluid to flow, there must be a pressure differential. If there is no pressure differential, all flow will cease. Using this principle, the valve problem may be approached by conceiving means of eliminating the pressure drop and consequently stopping flow.

The pressure differential could be eliminated by generating an opposing head via one of the methods listed in Fig. 1. The major problem which immediately arises is the difficulty of balancing the two heads so as to provide no flow in either direction. A self-balancing system could be employed, similar to mercury in a pressure manometer or some sort of siphon arrangement, but such systems would be too space consuming. The most unfavorable characteristic of such valving methods is the lack of positive flow shut-off.

Therefore, these pressure methods will be eliminated since they are not of sufficient value to contribute substantially to the problem.

C. Selection of Best Means

Evaluation of the means of valve actuation shown in Fig. 1 yields four practical methods. The proposals are listed below in order of preference:

1. A mechanically driven valve sealed by a frozen sodium annulus surrounding the valve stem (see Fig. 4).
2. A slow speed canned rotor motor of the synchronous type using permanent magnets mechanically rotated about the rotor to provide torque (see Fig. 5).

3. A hydraulically actuated valve utilizing a linkage mechanism for a locking device (see Fig. 6).
4. A series of ferro-magnetic discs operated by permanent magnets to drive a valve actuating mechanism (see Fig. 10).

D. Analysis of Force-Displacement Relationships

In one of the meetings with KAPL engineers, it was agreed that, inasmuch as a globe-type valve has been found satisfactory, effort should be directed toward the mechanism effecting closure. The principal reason for the choice of this type of valve lies in the need for an extremely high unit pressure between the sealing surfaces. As a figure of magnitude, a force of 25,000 lbs on the valve disc was necessary to stop the flow through an 8 inch valve.

One of the first analyses made was that of determining the force-displacement relationship of the globe valve. This relationship was plotted as a curve and is shown in Fig. 2. This curve is based upon assumed values of adverse conditions. The flow was assumed to be 3000 gpm and the static head 60 psi.

At full open position, the only force on the valve disc is that exerted by the moving metal stream. This force is the velocity pressure. Computations show that this is 35 lbs.

As the valve is closed, flow will slowly be restricted. For the first 95% of travel, it will be almost negligible, the only reaction being a slight increase in velocity pressure. During the last 5% of travel, the velocity pressure of the main stream will decrease rapidly with consequent increase of static pressure. The static pressure will reach a maximum on the impending seating of the valve disc. With a static pressure differential of 60 psi the force now exerted on the disc is 300 lbs. The extremely high force of 25,000 lbs comes after the valve seats. However, this very high force is exerted through a negligible distance. From this, one outstanding conclusion arises: The most efficient operating mechanism is a constant power system which can put out a low-effort fast-travel for 95% of displacement and high-effort slow-travel for the remaining 5% of displacement.

E. Proposals

Knowing the full requirements of the valve closure, they may be applied to the preliminary design of all five recommended proposals.

1. Proposal #1 - Directly Operated Valve

This is a mechanically driven valve sealed by a frozen sodium annulus surrounding the valve stem (see Fig. 4).

Freezing the sodium seems to be the most direct approach to the problem of sealing or leak prevention. By reducing the temperature of a thin annulus of sodium surrounding the valve stem to 208°F or less, the sodium will solidify, adhering tightly to all surfaces it contacts. This provides a positive seal, similar to a soldered joint.

Obviously, the prime consideration is to determine whether or not this sodium film can be frozen by reasonably simple means.

However, before this can be done, an idea of the size of the various valve parts must be obtained to get a clearer view of the heat transfer problem.

Computations show a stem size of 1.875" diameter to be satisfactory (see Appendix A). This stem is supported by a surrounding column. The column diameter is 2.75" (see Appendix B).

Knowing the valve stem size and its surrounding column size, the area required for freezing the sodium film can be computed. Calculations show that natural air convection freezing of the sodium film is quite feasible (see Appendix C and Fig. 3).

As an added safety factor, or, if preferred, a water cooled jacket may be employed. This type of cooling may be done by either circulating water through a heat exchanger with a pump or simply by dumping water around the valve stem from a reservoir to cool it almost instantly. This water cooling jacket would be much smaller than that required for air cooling. As mentioned before, this water jacket would make an excellent safety device. Mounted above the air cooled portion of the stem (see Fig. 4), it could readily be flooded if emergency conditions taxed the cooling capacity of the air cooled portion of the stem.

Freezing an annulus of sodium between the stem and the column introduces a problem. This is the problem of the torque required to break the seal in order to operate the valve. Calculations are included in Appendix D, giving a figure of magnitude, necessary to shear a solidified sodium seal.

Just above the water jacket, a stem scraper is employed. This scraper is provided to clean the valve stem as it is raised and lowered. When raised, the wiper prevents particles of sodium which may have adhered to the valve stem from being carried into the atmosphere. When lowered, this wiper gland prevents any moisture which may have accumulated on the stem from being carried down to a level where it may come into contact with the sodium.

A ball bearing thread is shown in the sketch. This type of thread is recommended because of its high efficiency in both directions and ease of turning. A gear-motor arrangement may be attached to the valve stem if power operation is desired.

Another safety feature which may be appended to this valve is an emergency seal as shown in Fig. 12. The valve stem drive shaft motion would have to be purely rotary. This is easily accomplished through a lead screw. With this seal, the valve may be positively sealed against leakage in any position desired.

Therefore, this valving device is highly recommended for the following reasons:

1. Simplicity of construction.
2. Simplicity of operation.
3. Safety features incorporated.
4. Reliability.

2. Proposal #2 - Slotted Rotor Valve

The second valving method is a slow speed cammed rotor motor (see Fig. 5).

This valve is operated in the following manner: An electric motor or a handwheel drives a ring gear which carries a series of permanent magnets. The magnetic field exerts a torque upon the slotted rotor within the valve casing. This rotor in turn actuates a lead screw, closing or opening the valve (see Fig. 5). The seal between the valve and its actuator is the can separating the rotor from the magnets. Such a seal is foolproof and leakproof. Allowances were made in the calculations for $1/4$ " gap between the rotor and the permanent magnets. Calculations pertaining to this valve are shown in Appendix E.

This valving scheme has the following advantages:

1. Compactness.
 - 2. Positive seal at all times against leakage.
 3. Ease of operation.
 4. Emergency over-ride provided (see Proposition #5).
3. Proposal #3 - Hydraulic Valve

The third proposed method is a hydraulically actuated valve utilizing a linkage mechanism for a locking device (see Fig. 6, 7, 8). This system was devised as an entirely new approach to the valve problem. The hydraulic system is powered by an electro-magnetic pump. This type of pump was used because ordinary displacement type pumps (such as gear pumps), are subject to severe wear under the galling effects of the liquid sodium. Since the flow of an electro-magnetic pump is normally unidirectional, a selector or pilot valve is employed to introduce pressure to the desired side of the power piston. This selector valve is operated by a "push-pull" solenoid. The suction side of the pump acts on the opposite side of the piston to which the pressure is applied to generate a greater differential pressure. The system is completely enclosed, operating on liquid sodium.

Examining the characteristics of electro-magnetic pumps, we find that a piston 18" in diameter will seat the valve with enough force to prevent leakage (see Appendix F).

When the valve is in either full open or full closed position, it is necessary to lock it in that position. This may be accomplished in any of three ways (see Fig. 6, 7, 8). In all methods, the links are brought into toggle by the main piston and an external means is employed to carry the linkage to and from its locked positions. In all cases, the hydraulic system takes the load while the linkages are being locked and unlocked, thus the requirements on the linkage actuators are kept to a minimum. In Fig. 7, the links are moved by a slotted rotor magnetically actuated as in Proposal #2. In Fig. 8, the linkage is moved by a "C" shaped piece engaging the protruding pins and pulling the linkage into a locked position. The motion is achieved by a "push-pull" solenoid exerting a pull on the ferro-magnetic slug. In Fig. 6, the linkages also are actuated magnetically. Permanent magnets are moved linearly up and down dragging the discs inside the valve casing. Of the three methods, Fig. 7 is perhaps the most desirable as it is the simplest and most compact. The locking device is located slightly off center to allow room for a mechanical over-ride to descend and engage the power piston in case of emergencies arising.

The valve is held in its upper position by permanent magnets around the hydraulic cylinder exerting pull on the piston (see Fig. 6).

The advantage of this type of valve are:

1. Compactness.
2. Completely enclosed.

Other pertinent calculations are shown in Appendix G.

4. Proposal #4 - Magnetic Disc Toggle Valve

The fourth proposal offered consists of a series of ferro-magnetic discs operated by permanent magnets to drive a valve actuating mechanism (see Fig. 10).

This type of magnetic valve actuation is a variation of a magnetic clutch. Instead of transmitting rotary motion, axial motion is transmitted. By turning a valve wheel, the magnets are moved forward or backward, dragging the inside discs along with them thus opening or closing the valve. These magnetically operated discs are connected to a shaft which operates a toggle linkage. This greatly reduces the size prime mover required. As the links come into toggle, the mechanical advantage approaches infinity. This, of course, is never attained due to friction forces present (for magnitude, see Appendix K). Besides reducing the required applied force to seat the valve properly, the linkage also serves as a locking device. As the links are carried past the toggle position, they hit a stop, thus effectively locking the valve shut. Any play or wear in the linkage pins is taken up by a heavy leaf spring at the upper end of the linkage. This spring is of sufficient strength to maintain 25,000 lbs on the valve disc when the linkage is in its locked position. When the valve is raised, it is held there by the magnets which hold the discs in position.

Due to the peculiarities of this linkage, the maximum force required to actuate it occurs slightly before seating (see Fig. 11, Appendix H). This feature is highly desirable for then additional power is available for locking and unlocking the linkage mechanism. Since the valve disc is hinged in this device, a study of valve seats is recommended. Spherical seats offer one solution to insuring proper seating around the disc periphery.

Other pertinent calculations are included in Appendix J and K.

This valve has the advantage of:

1. No external power required (other than manual).
2. Completely sealed valve system.
3. Positive locking device.

5. Proposal #5 - Mechanical Over-rides

On only two of these proposals are provisions made for a mechanical over-ride. Since this over-ride must be completely reliable and operatable at all times, the simplest design is best.

The over-ride which meets these requirements is shown in Fig. 13. Basically, its operation consists of rupturing a thin metal diaphragm to connect the over-ride shaft to the valve stem. Before operation of the over-ride, the liquid sodium is sealed in by the metal diaphragm. The over-ride stem is suspended above this diaphragm by means of a lead plug. After operation, the sodium is sealed by freezing in the over-ride stem column. Even after operation of this over-ride, it may be re-used again by breaking loose the frozen sodium seal with suitable impact force. The diaphragm may then be replaced when conditions permit.

The over-ride sketched in Fig. 13 is designed for rotary motion only. If push-pull operation is desired, the over-ride stem may be locked to the valve stem by some means such as screw threads or other such latching devices.

IV CONCLUSIONS

The problem requirements were to propose a means of valve closure, which includes a mechanical over-ride and a positive seal, other than bellows, between the valve disc and the external operator force. These requirements can be met through the application of one or more ideas or alternative combinations of ideas presented in the proposed schemes.

A particular philosophy has grown with the study of these alternate valving techniques. This philosophy points directly to the use of a valve as recommended in Proposal #1, in which the valve disc is moved by a direct physical effort of the operator. The operator of a pushbutton type of valve must rely entirely on faith that the valving mechanism obeys the signal. Since this is an emergency valve and since a mechanical over-ride is required in all the valve mechanisms proposed, the most practical means is to combine the two functions in a directly operated valve as in Proposal #1. Using this method to accomplish closure, the operator is quite convinced that the valve actually is open or closed.

A further ramification of this philosophy lies in the power requirements during so-called emergency conditions. A power operated device draws kilowatts from the auxiliary power system. When an emergency condition arises, kilowatts are most precious. Therefore, a valve mechanism operating independently of the power system is highly desirable.

If a mechanical over-ride is not required, then Proposal #2 combines the best features. In order to justify the elimination of the mechanical over-ride, tests must be made under simulated conditions to determine the reliability of the magnetic drive.

V APPENDIX

A. Valve Stem Size (Prop. #1)

Although the stem probably will not be loaded heavier than 25,000 pounds, assume a case where the valve jams and needs additional force to free it. Therefore, let the ultimate load be 40,000 lbs.

Assume: 18-8 high chromum-nickel-iron alloy, cold drawn

Material properties:

Ultimate strength = 105,000 to 300,000 psi

Yield point = 60,000 to 250,000 psi

Stress causing 1% creep in 10,000 hours

At 1000° F = 17,000 psi

Allowable stress = 15,000 psi

$$S = \frac{F}{A} \quad ; \quad A = \frac{F}{S}$$

$$A = \frac{40,000}{15,000} = 2.67 \text{ sq. in.}$$

$$D^2 = \frac{4A}{\pi} = \frac{(4)(2.67)}{\pi} = 3.39 \quad ; \quad D = 1.842$$

Therefore, stem diameter = 1.875"

This calculation offers a dimension for the diameter of the valve based upon pure axial loading, either compression or tension. Since the stem is substantially supported over its entire length, buckling of the stem need not be considered.

B. Stem Column Size

Assuming a majority of the load is transferred to the valve seat through the column surrounding the stem, the column area must be 2.67 square inches.

For a hollow column $A = \frac{\pi}{4} (D_o^2 - D_i^2)$

Solving for D_o :

$$D_o^2 = \frac{4(2.67) + \pi (1.875)^2}{\pi} = 7.24$$

$$D_o = 2.695''$$

Therefore, column outside diameter = 2.75"

$$\begin{aligned} \text{Wall thickness} &= \frac{2.75 - 1.875}{2} = 0.4375'' \\ &= \frac{7}{16}'' \end{aligned}$$

C. Air Convection Freezing

Knowing the valve stem size and its surrounding column size, the formulae and curves found in General Electric's Design Data Book, Section G-504.2 (for hc) 505.2 (for hr), and 508.3 (for to) can be applied to solve for natural air convection cooling.

Assume: $l = 1.5'$, Mat'l = 18-8 S.S.
 $K = 9 \text{ Btu/hr/ft}^2/\text{°F/ft.}$

$$\begin{aligned} h_c &= 1.5 \\ h_r &= 3.4 \\ h_t &= 4.9 \end{aligned} \quad a = \sqrt{\frac{A h}{K A_c b}} = \sqrt{\frac{48 D_o h}{K (D_o^2 - D_i^2)}}$$

$$a = \sqrt{\frac{48 (2.75) (4.9)}{9 (7.56 - 3.51)}} = 4.21$$

$$ab = al = 4.21 (1.5) = 6.31$$

$$t_o = 180 + \frac{670}{\cosh ab} = 180 + \frac{670}{275}$$

$$t_o = 182.44^\circ \text{F}$$

Similarly:

	t_o	
17"	183.44°	F
1.3"	185.68°	F
1.2"	188.70°	F
1.1"	193.07°	F
12"	199.85°	F
11½"	203.80°	F
11-1/32"	208.00°	F

Hence, from the preceeding calculations, freezing by using the ambient 180° F air in the chamber is quite feasible, using a proper length valve stem (see Fig. 3).

D. Breaking Sodium Seal

Since the sealing is to be effected by freezing a thin annulus of sodium around the valve stem, determine the force necessary to break the seal by a hand operated valve. With a 12" air cooled stem, approximately 1" of stem will be frozen (see Fig. 3). Sodium has strength characteristics somewhat like soft solder; however, accurate data is not available, particularly near its melting point. Since conditions are very near the melting point, assume its strength in shear $S_s = 500$ psi which is about 20% of its strength at room temperatures:

$$\text{Area to be sheared} = \pi D h = \pi (1.875)(1) = 5.89 \text{ in.}^2$$

$$\text{Force necessary} = S_s A = 500(5.89) = 2945 \text{ lbs.}$$

Assuming a $2\frac{1}{2}$ ft. diameter wheel acting through a 5:1 gear reduction:

$$\text{Gear eff.} = 98\%$$

$$\text{Thread eff.} = 80\%$$

$$F_1 d_1 = F_2 d_2 \times \text{gear reduction} \times \text{gear eff.} \times \text{thread eff.}$$

$$F_2 = \frac{(2945) \left(\frac{48.75}{2} \right)}{(15)(5)(.98)(.80)} = 47 \text{ lbs.}$$

This is a reasonable figure.

As a further refinement and perhaps improvement on this problem, an increased annulus thickness at the frozen section would have the effect of permitting greater relative motion between the stem and its column and make fracture of the seal easier.

E. Slotted Rotor Size (Prop. #2)

Calculations show that to apply 30,000 lbs. along a 1.5" diameter valve stem through a 0.266" pitch screw thread, 2265 lbs. are required acting at a 3/4" radius.

Therefore, since $F_1 r_1 = F_2 r_2$ the required force may be reduced by increasing the radius:

Let rotor diameter = 6"

$$(2265) (.75) = F_2 (3)$$

$$F_2 = 566 \text{ lbs.}$$

Assume: Screw efficiency = 80%

Windage loss = 5%

$$\text{Therefore } F_2 = 566(1.2)(1.05) = 714 \text{ lbs.}$$

Formula supplied by K. E. Wakefield of General Engineering Laboratory for magnetically dragged components is:

$$f = \frac{1}{8\pi} B^2 t l \left[1 - \left(\frac{4t}{4t + \pi(x-w)} \right)^2 \right]$$

Assume: $B = 10^4$ gauss
 $t = \frac{1}{4}'' = .635$ cm.
 $l = \text{length in cm}$
 $x = \frac{1}{2}'' = 1.27$ cm
 $w = \frac{1}{3}x = .423$ cm
 $f = 1600$ lbs. $= 7.1 \times 10^8$ dynes

Solving for l ;

$$l = \frac{(3.165)(10^8)(8\pi)}{(10^8)(.635)(.761)}$$

$$= 164.8 \text{ cm} = 64.9 \text{ in. for a single pole}$$

Using an 8 pole rotor, rotor length $= \frac{64.9}{8} = 8.125''$

Let rotor length $= 8\frac{1}{2}''$ for added safety factor.

F. Hydraulic Piston Size (Prop. #3)

Using an electro-magnetic pump with characteristics of 100 psi at 10 gpm and required force = 25,000 lbs. max.

$$F = PA, \quad A = \frac{25,000}{100} = 250 \text{ in}^2$$

$$D^2 = \frac{4A}{\pi} = \frac{(4)(250)}{\pi} = 318.5$$

$$D = 17.82"$$

$\therefore$ Let D of master piston = 18"

G. Closing Time Hydraulic Valve

If total travel = valve lift = 4"

$$\begin{aligned} \text{Displacement} &= 254.5(4) = 1018 \text{ in}^3 \\ &= 4.42 \text{ gallons} \end{aligned}$$

$$\text{At 15 gpm, } t = \frac{4.42(60)}{15} = 17.7 \text{ seconds.}$$

H. Toggle Force-Displacement Curve (Prop. #4)

The curve shown in Fig. 11 was obtained from a graphic layout of the linkage system. Since the maximum forces on the valve disc occurred at seating, the force diagrams were produced for the last 2° of disc angle. Forces acting on the disc were taken from Fig. 2, and the required input force to the toggle linkage was graphically determined.

J. Toggle Drive-Disc Size

As shown in Appendix H, the maximum required force to actuate the valve toggle linkage is 500 lbs.

Therefore, using the same formula as found in Appendix E, and assuming the same valves,

$$L = 45.5''$$

$$\text{Let } d_o \text{ of discs} = 4.875'' \quad C = \pi d = \pi (4.875) = 15.32''$$

$$\text{With three discs } L = 3(15.32) = 45.96''$$

This gives a margin of $1/2''$ extra L .

If distance between discs = 10 times thickness = $10(1/2) = 5''$,
or, center to center = $5-1/2''$. Total length = $3(5-1/2) + 9-1/2''$
travel = $21''$ cylinder.

K. Toggle Linkage Friction Study

As a further refinement in the development of those valve proposals utilizing toggle linkages, the effects of friction are here studied.

Proposals using this linkage (see Fig. 9 and 10) were calculated neglecting this effect.

- Assume:
- (1) ordinary journal bearings
 - (2) carboloy on carboloy materials
 - (3) temperature 800° F
 - (4) pressure on bearings = 25,000 lbs.
 - (5) coefficient of friction = 0.30

Drawing the force diagram, we have:

Examining the bearing itself;

Equating moments;

$$\begin{aligned} T_f = T_x &= r F \mu = \frac{15}{16} (25,000) (.30) \\ &= 7040 \text{ in. - lbs.} \end{aligned}$$

Taking bearing (1) $F_1 (12) = 7040$, $F_1 = 586 \text{ lbs.}$

Taking bearing (3) $F_3 (6) = 7040$, $F_3 = 1172 \text{ lbs.}$

Taking bearing (2) $F_2 (4) = 7040$, $F_2 = 1760 \text{ lbs.}$

Therefore,

$$F_f = 1760 + 1172 + 586 = 3518 \text{ lbs.}$$

Hence, with ordinary journal bearings, the size of actuating mechanism would have to be considerably increased to take care of the friction forces. Calculations show that the hydraulic piston diameter in Fig. 9 would have to be increased from 3" to 6½" and the magnetic disc diameter in Fig. 10 would have to be increased to around 10" with 15,000 gauss magnetic field.

Obviously, the best way to overcome this large loss in friction is to reduce the coefficient of friction. This might be done by utilizing needle bearings whose coefficient of friction is in the neighborhood of .005. If such toggle linkages are employed in valving mechanisms, needle bearings should be investigated to avoid expending large amounts of effort in overcoming the forces of friction.

FIGURE #2

FORCE - DISPLACEMENT
CURVE

FOR

8" GLOBE VALVE DISC ACTING
AGAINST LIQUID SODIUM FLOW
OF 3000 GPM UNDER A MAX-
IMUM STATIC PRESSURE OF
60 POUNDS/INCH²

(ROTARY MOTION)

R. H. White
HJM
2-8-52

34

R. H. Hirt
Nem
2-5-52

35

GENERAL ELECTRIC

REV
NO.

TITLE

CONT ON SHEET

SH NO.

DIRECTLY OPERATED VALVE

CONT ON SHEET

SH NO.

FIRST MADE FOR

} BALL-BEARING THREAD

} STEM COUPLING
} STEM CLEANER

} WATER JACKET

} AIR CONVECTION
} COOLING SURFACE

FIGURE # 4

REVISIONS

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CONT ON SHEET

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GENERAL ELECTRIC

CONT ON SHEET

SH NO.

REV.
NO.

TITLE

SLOTTED ROTOR VALVE

CONT ON SHEET

SH NO.

FIRST MADE FOR

OVERRIDE APPLIED
HERE

FIGURE #5

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GENERAL ELECTRIC

CONT ON SHEET

SH NO.

REV
NO.

TITLE

HYDRAULIC VALVE

CONT ON SHEET

SH NO.

FIRST MADE FOR

FIGURE # 6

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REV
NO.

TITLE

HYDRAULIC PISTON LOCKING DEVICE

CONT ON SHEET

SH NO.

FIRST MADE FOR

HYDRAULIC PISTON

FIGURE # 7

REVISIONS

PRINTS TO

BEARINGS

STOP

MAGNETICALLY DRAGGED
SLOTTED ROTOR

MADE BY

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HYDRAULIC PISTON LOCKING DEVICE

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FIGURE # 8

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REV NO.	TITLE	CONT ON SHEET	SH NO.
	HYDRAULIC TOGGLE VALVE		
CONT ON SHEET	FIRST MADE FOR

FIGURE # 9

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41

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	MAGNETIC DISC TOGGLE VALVE		
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FIGURE #10

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42

FORCE - DISPLACEMENT CURVE
for
VALVE TOGGLE LINKAGEFIGURE # 11APPLIED FORCE TO TOGGLE LINKAGE,
POUNDS

500

400

300

200

100

0

0.2

0.4

0.6

0.8

1.0

1.2

1.4

1.6

1.8

2.0

DESIGN FIGURE

NOTE: FRICTION FORCES NEGLECTED

ANGLE "α" - DEGREES

25000

1850

1120

420

FORCE ON VALVE DISC - POUNDS

Norman J. McLean
JUL 25, 1952

43

REV NO.	TITLE	CONT ON SHEET	SH NO.
	EMERGENCY SEAL		
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FIGURE #12

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FIGURE # 13

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