Contrails

EVALUATION OF CHARACTERISTICS OF LARGE SIZE FUEL SHUTOFF VALVES

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FOREWORD

This report has been prepared by the FluiDyne Engineering Corporation, Minneapolis, Minnesota, in satisfaction of the requirements of Air Force Contract No. AF 33(616)-2152. This work was accomplished under Project No. 3084, "Fuel and Propellant Component Integration." The work was administered under the direction of the Power Plant Laboratory, Wright Air Development Center, with Mr. E. Rowland acting as project engineer.

The authors were assisted in various phases of the work by members of the technical staff of FluiDyne Engineering Corporation. Several valve companies and industrial organizations furnished information pertinent to various phases of the study.

ABSTRACT

The performance characteristics of valves suitable for shutoff use in 4 to 20 inch aircraft and missile fuel and propellant lines were evaluated. The nine most promising types were selected in a preliminary study and representative designs were prepared for each type. Weight, dimensions, actuating requirements, sealing and pressure drop characteristics were evaluated for each type for 60 and 600 psi application. An operating temperature range from -65° to $450^\circ F$ was considered. Estimated weight and dimensional characteristics of the various valves vary widely, underlining the importance of type selection for large valves. When compared on the basis of equal pressure drop characteristics for equal flow capacity, the relative weight differences are very pronounced. The large weight of the valve bodies and their complex shapes produces a need for extensive experimental structural analysis to determine the lightest design of a given type.

Three valve types are recommended for use in the 600 psi range and four types for use in the 60 psi range.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.
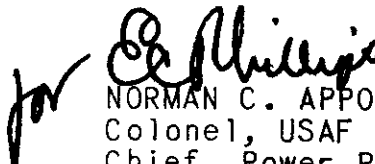
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Fuel requirements of large jet and rocket engines have seriously complicated fuel systems and their components. As a result, the problem of equipment selection has been greatly accentuated and a definite need has arisen for adequate information concerning large size fuel shutoff valves. This report presents the results of an engineering study to evaluate the performance characteristics of valve types when used as fuel shutoff valves in aircraft and missile fuel and propellant lines.

The aircraft industry has been mainly concerned with small valve sizes, under 4-inch diameter. Large size valves have been designed for use in industrial applications where the requirements are quite different. For aircraft and missile fuel systems, factors of weight, dimension, power requirements, sealing ability, and pressure losses are of a great deal of importance. Since aircraft valves are selected to meet these different requirements, consequently their design is materially affected. The purpose of the study was to evaluate and compare the performance characteristics, pertinent to aircraft and missile fuel shutoff applications, of various valve types in the size range of 4 to 20 inches in diameter. Preliminary investigation indicated that for the purpose of this study it was not necessary to differentiate between fuel and propellant valves, only on a basis of pressure class, 600 or 60 psi. The valve types on which further development work could most profitably be concentrated were to be recommended.

After a preliminary study phase, nine representative valve types, Figs. 1 through 9, were selected for more detailed study and evaluation. The following performance characteristics were determined and compared for these valves: weight, actuating requirements, dimensions, sealing and pressure drop. These various characteristics were compared for valves operating at 60 and 600 psi over a temperature range from -65° to $+450^{\circ}F$. The design and fabrication of the valve types that indicated low weight trends were investigated in more detail by considering use of more efficient materials and methods of fabrication in an endeavor to determine their extent of weight reduction possible. Consideration was also given to investigating methods and techniques which could produce more refined and detailed structural analysis of valve components. The design methods and calculations used in determining weights, actuating requirements, and dimensions are included in the appendix.

SECTION I

VALVE TYPES STUDIED

A survey was made of existing aircraft fuel shutoff valves and standard industrial type valves. Their characteristics were studied and evaluated as to their applicability for use as large size fuel shutoff valves. Manufacturers were contacted for information concerning their valves and comments concerning the study were invited. Characteristics of the basic valve types (gate, globe, plug, and butterfly) were also studied and evaluated for fuel shutoff application.

A number of valve types and modifications have been considered and preliminary layouts made and studied for those that indicated desirable characteristics. Some of these have been discussed in the progress reports. Nine types were selected for a more detailed study and evaluation. These nine types are representative of the basic valve types and possess certain advantages and characteristics that warranted their inclusion in the study. The design of each of these nine types could undoubtedly be improved upon for a specific application; however, their design is representative of a type of valve for the purpose of the study.

Pivoted Gate Valve

Design of the pivoted gate valve with actuator is shown in Fig. 1. Actuation is accomplished by means of a double acting piston type hydraulic actuator mounted as shown. The same basic design was used for the 60 psi class valves as for the 600 psi class with the exception that the rib construction was omitted in the smaller sizes for the 60 psi class valves. The flat areas and construction of this valve were ideally suited for the use of ribs. The two part flexible gate is moved by means of a carriage arm fastened to the shaft and $60°$ rotation is required from open to closed position.

Y Pattern Globe Valve

The design of the Y pattern type globe valve is shown in Fig. 2. The design of this valve is based on a liquid oxygen valve currently being manufactured and is characteristic of globe type valves in general. The valve is operated by means of line pressure using a pilot operated solenoid of the same design as the valve itself. In the smaller sizes a direct acting solenoid

is used. When the solenoid is energized the line pressure is
admitted to the top of the actuating piston. Since the area of
the piston is larger than the poppet (25% larger), sufficient
force is created to open the poppet against line pressure. The
area ratio is sufficient to keep the valve open with the pres-
sure differential across the valve. When the solenoid is de-
energized, the inlet pressure is cut off from the actuating
cylinder and the poppet closes. Dash pot action is provided by
the small orifice in the piston to minimize and control pressure
shock.

Slide Gate Valve

The design of the slide gate valve with actuator is shown
in Fig. 3. This valve is similar in design to smaller size
valves manufactured by several aircraft equipment companies. A
vane type hydraulic motor was selected for use with this valve
because of its adaptability and compact design. The actuator
mechanism is of the scotch yoke type. The resisting torque is
a minimum at the position of maximum force requirements, when
the slide is in the closed position with full line pressure
drop across the slide, because the moment arm of the resisting
force is a minimum. 180° crank rotation is required to move the
slide from fully open to closed position. The housing plates
for the 600 psi valves are curved to reduce the wall thickness
and hence the weight.

Angle Valve

The design of the angle valve with actuator is shown in
Fig. 4. Some design modifications have been incorporated in the
design of this valve in comparison to ordinary industrial types
of angle valves. For example, the poppet and seat are remove-
able by means of a snap ring from the bottom, resulting in a
saving in weight and producing a more compact design. The poppet
is guided by means of three guides which are cast integrally
with the body and actuated by means of a piston type hydraulic
actuator mounted directly to the valve. The actuator can be
double acting or single acting, as shown, with the pressure drop
across the poppet assisting the spring in providing a closing
force.

Poppet Valve

The design of the poppet type valve is shown in Fig. 5.
This valve is similar in design to a type developed by the

Zuelsdorf plant in Germany for rocket units (Ref. 10). The valve is designed for line pressure actuation; however, external hydraulic or pneumatic pressure can be used to actuate the valve. A four-way control valve is used to control the sequence of operations, the pressurized port would be open to the upstream pressure and the exhaust port open to the downstream pressure. The line pressure operation is dependent upon the pressure drop through the valve. The internal structure is supported by six supports integral with the body and equally spaced as shown in sections B-B and C-C of Fig. 5. The flow passage area is constant throughout the valve.

Butterfly Valve

The design of the butterfly valve with actuator is shown in Fig. 6. It has long been recognized that the characteristics of the butterfly type of valve result in minimum weight. It is used extensively for throttling applications but sealing of the vane is a major problem when used for shutoff applications. The design of the valve is affected by the sealing arrangement used and in the design shown, an expandable type of seal was selected because it is one of the more promising types. The sealing membrane is Teflon (well confined), which is suitable for use over the temperature range indicated, and operated with air pressure greater than operating line pressure. The seal is pressurized after the vane is in the closed position and depressurized before the vane is moved to the open position. This is accomplished by means of a three way air valve which might be controlled by the hydraulic pressure of the actuator. With the vane and offset shaft arrangement shown, the problem of sealing the shaft is minimized; the shaft is downstream of the vane and seal. This split shaft arrangement has been successfully employed by North American in the development of a 5 inch butterfly valve for liquid oxygen shutoff. A vane type hydraulic actuator was selected for use with the valve because of the mounting problems and resulting light weight and compact design.

Diaphragm Valve

The diaphragm valve shown in Fig. 7 is similar in design to a valve currently being manufactured. The valve is actuated by means of line pressure which is admitted to the chamber above the diaphragm through an orifice by means of a bleed passage. An outlet bleed passage connects the chamber to the downstream side of the valve. When de-energized, a small control valve closes the outlet bleed passage and line pressure holds the valve in the closed position. The control valve shown is pneumatic; however, an electric solenoid could be used depending on the

source of power available. With the control valve energized, the orifice in the inlet passage maintains a pressure differential across the diaphragm and line pressure holds the valve in the open position.

Eccentric Plug Valve

The design of the eccentric plug valve similar to an industrial type valve with actuator is shown in Fig. 8. There is a transition in the flow passage from round to a somewhat rectangular opening. The plug is not a solid cone but a segmental type and the center of curvature for the seat and plug are eccentric with the valve body, thus allowing a positive seating force in closing. The design requires a horizontal and a vertical rib to keep the body sections from becoming excessive. A piston type hydraulic actuator mounted as shown in used to actuate the valve.

Cone Valve

Design of the cone valve is shown in Fig. 9. The valve as designed is actuated by line pressure; however, with some modifications it could be actuated by external pneumatic or hydraulic means. Operation is dependent upon the pressure drop through the valve and a four way solenoid operated pilot valve is used to control the sequence of operations. One cone is integral with the valve body and the other cone is rotated by means of vanes on its periphery. Only a few degrees of rotation are required from open (slots aligned) to closed position.

SECTION II

WEIGHT DETERMINATIONS

A. Relative Weight Comparisons

Weights of the various valves depend on their design and construction. The scope of the study did not allow determination of the optimum design and fabrication for each valve considering all the requirements. To accomplish this would require a thorough analysis and design of each valve accompanied with a testing and development program. Therefore, an endeavor was made to

place the valves on a common basis, giving no special considera-
tion to any one valve type, and by this means obtain good rela-
tive comparisons of the various performance characteristics.
Such an approach serves the purpose of this study by providing
the necessary information for comparisons, evaluations, and
recommendations. As a result it was necessary to standardize
design in certain areas.

B. Method

The following design conditions were standardized and used
for determining valve weights:

1. Body Construction -- a flanged type of cast body con-
struction was used for each valve over the complete size range
and for both pressure classes.

2. Valve Size -- the valve size was standardized as the
line OD allowing the ID to be determined by the required wall
thickness.

3. Design Line Pressures -- 1200 psi for 600 psi operating
pressure; 120 psi for 60 psi operating pressure.

4. Design Temperature -- 450°F.

5. Materials -- Body, 356-T6 aluminum alloy; poppets,
slides, pistons, stems, shafts, seats, etc., 4130 or 4340 HT
steel.

6. Material Stresses -- the material stresses used in
design are their yield strength at 450°F. The following tensile
values were used for the various materials:

356-T6 cast aluminum	7500 psi
24-ST4 wrought aluminum	20,000 psi
61-ST6 wrought aluminum	7500 psi
4340-HT steel	65,000 psi

The ten-inch size was selected as the size to design and
lay out in detail. For reliable comparison purposes, it was
necessary to consider the complete design of each valve. Design
procedures and techniques were developed for the components that
contributed the major portion of the valve weight. For weight
comparison with line pressure actuated valves it was necessary
to consider actuators. The same basic valve design was used for
both the 600 and 60 psi pressure classes and generally over the
complete size range, 4 to 20 inch diameter. Some modifications

in material and construction were employed in the smaller sizes
of the 60 psi valves. Standard design techniques and methods
were adhered to as much as possible with special emphasis given
to aircraft design techniques. The assumptions and design methods
employed for each valve are presented in the appendix along with
a sample calculation. The design techniques and methods developed
for the 10 inch valves were employed in determining the weight
of the 4, 16, and 20 inch valves for both pressure classes. By
plotting a curve through these four weights, weight vs. size is
obtained over the complete size range of 4 to 20 inch diameter.
Dividing the size range in this manner places an emphasis toward
the lighter end of the size range which is the range of most
concern.

C. Results

Estimated weights of the nine 600 and 60 psi valve types
are compared over the size range in Figs. 10 and 11, respectively.
Weights vs. size are also compared on logrithmic coordinates in
Figs. 12 and 13 for the 600 and 60 psi valves, respectively.
The plots are straight lines of approximately the same slope
which indicates the exponent of the equation relating weight
and diameter is essentially constant for a particular valve
type and similar for all the types. The relationship can be
expressed as follows:

$$W = KD^n$$

where W = valve weight, lbs.
K = constant depending on
design
D = valve diameter, inches
n = exponent

If the exponent "n" is a constant, the log plot of W vs. D re-
sults in a straight line. Table 1 gives a comparison of the
exponent "n" determined from the log plots for the nine valve
types. It is interesting to note the uniformity of the exponent
in each pressure class. The maximum variation in "n" for the
600 psi valves is 9.63% while for the 60 psi valves it is 11.34%.
The average value of the exponent for the 600 psi valves is
higher than for the 60 psi valves which indicates a greater
effect of diameter on weight for the 600 psi valves. The 600 psi
valves are designed mainly for strength, while the 60 psi valves,
especially in the smaller sizes, have such thin sections that
other design factors than strength determine the sections in
many cases. It seems reasonable to assume that for any design
of a particular type of valve similar to one of the nine types,
that these exponents can be used to estimate weights for various
sizes (over the size range from 4 to 20 inches) if one weight is
known or determined. The logrithmic plots also clearly reveal
the weight differences between the smaller size valves which is
important.

The relation between the various weight curves is fairly uniform over the size range even though the variations in slope of the log curves is about 10%. The reason for this is the large spread in weight. In the 600 psi pressure class the butterfly valve indicates the lowest weight over the complete size range. The pivoted gate and angle types follow with the pivoted gate indicating lower weight in the larger sizes. In the 60 psi pressure class the angle and butterfly types indicate the lowest weight trends over the size range. Around the 10 inch size there is a reversal in weight trends, the angle type having lower weight above 10 inches and the butterfly type below 10 inches. The poppet and pivoted gate types follow with the pivoted gate having lower weight above the 9 inch size and the poppet having lower weight below the 9 inch size. The other types in both pressure classes are grouped reasonably close together.

The weight curves indicate considerable weight differences between the various valve types. In the 600 psi pressure class there is approximately 300% variation in weight between the lightest and heaviest valves over the size range and in the 60 psi pressure class 250% variation. This is significant and places an emphasis on the choice of a valve type for minimum weight neglecting other requirements.

Since the valves are three dimensional objects, it was expected that their weights would vary as the diameter cubed. The average exponent for the 600 psi pressure class valves is 2.81 and for the 60 psi class, 2.55. As a result, any change in diameter greatly influences the valve weight. Regardless of design or valve type, large valves will be heavy. Since the valve weight varies approximately as the diameter cubed and flow rate as the diameter squared (for equal velocities), for a given flow rate one large valve would weigh approximately twice as much as four valves of half the diameter. As an example, one 20 inch diameter valve could be replaced by four 10 inch valves providing equal flow area at approximately half the weight. The problem of substituting several smaller valves for one large valve to conserve weight introduces numerous other problems and will depend on the system requirements.

As previously mentioned, these weight curves represent relative weight comparisons and should not be used as optimum weights for the various types. Design and construction of a particular type could alter these weights by possibly as much as 25% but would not alter the results to any extent because of the large spread in weights. By a thorough design analysis and proper choice of materials and fabrication techniques, the weight

of any of these types could be reduced. In Section III, Materials, the valves that indicated low weight trends were investigated in an endeavor to determine the extent of weight reduction possible considering use of more efficient materials and design. As noted, these valve weights include the weight of the actuators. For the hydraulic type of actuator selected, the actuator weight is generally small in comparison to the valve weight. Table 1 compares the weights of the various valve types with and without actuators and percentages of actuator weight to valve weight. The weight of the actuators includes weight of all necessary mountings and supports. In the 600 psi pressure class the percentage of actuator weight to valve weight varies from 12 to 20% in the smaller sizes to 10 to 15% in the larger sizes. For the 60 psi class the percentages vary from 3 to 18%.

The larger valves are heavy and their design will be critical. A savings of only a few percent in weight will be important. The bodies of these valves constitute a major portion of their weight. A comparison of the percentages of body weight to valve weight for the various valve types is given in Table 1. In the 600 psi class the percentages vary from 50% to over 70% for the butterfly and slide gate types.

The analytical design of a valve body is difficult if not impossible because of the varying shapes, all integrally attached. Stresses occurring in one section influence stresses in adjacent sections. Minimum body section thicknesses were determined using the modified Barlow 1/ equation with a shape factor of 25% instead of 50%. This equation is universally used for valves and fittings and has proven acceptable. The equation is basically the fundamental hoop stress equation for thin walled cylinders with a shape factor of 50%. The application of the equation in determining body section thicknesses for the various valves is presented in Appendix B. The body section thickness thus determined was used as a minimum section thickness over the complete body. The bodies of the various valves have different degrees of complexity and it is obvious that the same section thicknesses are not required for each valve type; also, the same section thicknesses are not necessarily required over the body. By proper distribution of material, considerable weight savings could be realized in many cases, especially in the larger sizes.

1/ Modified Barlow formula of the A.S.M.E. Boiler Construction Code for cylindrical sections and adding 50% to the thickness thus determined to compensate for the shape of the fittings.

For optimum distribution of material some method of experimental stress analysis is required. Valve companies interested in good distribution employ experimental stress analysis and several models are built and tested before the final design is arrived at.

Use of ribs and waffle type construction could be definitely used to advantage in conserving weight in valve bodies. Analytical design is extremely difficult due to the complex shape of the bodies. Considerable development and testing is being done at the present time by several aircraft companies on waffle type of construction for fuselage and wing panels. The pivoted gate valve was ideally suited for the use of ribs with its flat surfaces. By employing the rib construction shown, Fig. 1, it is estimated that a saving of 7 to 10% in body weight resulted.

Design conditions, particularly the upper temperature limit (450°F), influence the weight. Strength of the various metals drops rapidly with an increase in temperature, as indicated in Table 2 which compares yield strength for various wrought and cast materials at room temperature and 450°F. The strength of the cast aluminum alloy (356-T6) selected for the body material at 450°F is roughly 1/4 of its strength at room temperature. This increased the weight of the bodies approximately fourfold. Steel and titanium are not affected in this temperature range to the same extent as aluminum. Our choice of materials may not have been the best from a minimum weight standpoint; however, for relative weight comparison it was not too important. The use of more efficient materials to reduce the weight of the valves will be discussed in Section III.

SECTION III

MATERIALS

A. Purpose

Different construction techniques and use of more efficient materials have been considered for the valve types that have indicated low weight characteristics in an endeavor to determine what their actual minimum weight might approach. The various valve types lend themselves to different methods of construction and design, and, as previously discussed, it was necessary to standardize design in certain areas for relative comparison

purposes. The scope of the study did not allow a detailed analysis of the valve types in an endeavor to determine their optimum design and construction. Consequently, it was felt that it would be of value to the study to investigate the effect of higher strength materials and different fabrication methods on valve weight.

B. Efficiency Comparisons

A comparison of yield strength, modulus of elasticity and strength-density ratios for various cast and wrought materials at room temperature and 450°F is given in Table 2. This table is representative of a few of the more important materials that could be used in valve construction. The yield strength-density ratio is a measure of the efficiency of a material on a strength-weight basis. Of the cast materials fiberglass and steel have the highest strength efficiency and of the wrought materials, steel and titanium are highest and approximately equal at 450°F. By employing the more efficient wrought titanium or steel instead of the cast 356-T6 aluminum alloy for a body material, considerable weight savings could be realized in many cases. It must be kept in mind, however, that other factors than strength alone enter into the design of valve bodies, and for some body types it is questionable whether a weight saving can be realized. At 450°F wrought titanium and steel are almost eight times more efficient than the 356-T6 cast aluminum and are by far the most efficient of all the materials. The strength of aluminum and magnesium drop rapidly with temperature and consequently their efficiencies are low at 450°F. The final selection of a material for a particular application will depend on additional requirements beside strength alone.

C. Results

Results of the weight determinations revealed that the valve bodies are a large percentage of the weight and use of the low strength cast aluminum resulted in heavy body weights. Consequently the design and construction of the bodies were studied for the possibility of employing a type of construction and fabrication utilizing the higher efficiency wrought materials. To use wrought materials the bodies may either be forged, machined from stock, or fabricated of simple geometric shapes welded together. Since wrought titanium indicated the highest strength efficiency, its use for valve bodies was considered.

Titanium is a relatively new metal and its development is progressing so rapidly it is difficult to keep abreast of the latest developments. It can be forged and welded under suitable conditions but as yet it cannot be cast in large sections. It has excellent corrosion and fatigue resistance to acids, especially at high temperatures. It possesses poor seizing and galling properties, but by proper design these can be minimized.

The designs of four promising valve types, the angle, pivoted gate, butterfly, and poppet valves, were reviewed in the light of these considerations of types of material and fabrication.

1. Angle Valve

A modified design of the 10 inch, 600 psi angle valve employing a wrought titanium body of welded construction is shown in Fig. 14. The body is fabricated from tubing or rolled plates with the different sections welded together. A different poppet arrangement was utilized, the poppet is a cylinder which telescopes into another cylinder upon opening. This arrangement is similar to that used for the poppet valve, Fig. 5, and eliminates large pressure loads on the poppet which results in small operating forces. The spring and small lip on the end of the poppet cylinder provide the necessary sealing forces. The valve is completely fabricated from titanium except for the actuator, spring, seat and rod. The titanium poppet and containing cylinder are plated to minimize seizing and galling or they can be fabricated from steel. The piston type hydraulic actuator could be mounted directly to the top cover plate with the spring inside the body if necessary.

The valve has the same 10 inch OD line size as before, but the thinner wall sections resulting from use of the higher strength titanium increased the flow passage area equivalent to that of a 12.5 inch OD valve having a cast aluminum body. This modified design resulted in a weight saving of 60% based on equivalent flow capacities (equal ID), comparing the weight of the modified 10 inch angle valve to that of a 12.5 inch cast aluminum body angle valve. The large weight saving was not entirely due to the use of titanium, but partially gained by employing a different design utilizing a cylindrical body and poppet. The actuator was also smaller due to the low operating force requirements of the cylindrical poppet.

2. Pivoted Gate

The basic design of the pivoted gate valve was not altered, but the body was redesigned to be fabricated from machined titanium plate stock with the various components welded together. A modified design of the pivoted gate valve is shown in Fig. 15. This valve has a flow passage area equivalent to that of a 10 inch valve having a cast aluminum body and since its OD was reduced to the equivalent of an 8 inch aluminum body valve, it is spoken of as an 8 inch valve. A weight saving of 32% resulted based on a comparison with the 10 inch aluminum body pivoted gate valve. With the same ID as the standard 10 inch valve, the gate, shafts, and actuator are identical; therefore, the entire saving in weight resulted from the use of titanium in the body.

3. Poppet Valve

Substitution of titanium for the cast aluminum in the design of the body of the 10 inch, 600 psi poppet valve, Fig. 5, resulted in a weight saving of 39%. The internal structure remained the same and the body was considered to be fabricated from a cylinder and two truncated cones welded together with welded on flanges. The flow passage area remained the same and the OD was reduced to the equivalent of an 8 inch cast aluminum poppet valve. As with the pivoted gate valve, the entire saving in weight resulted from the use of titanium in the body.

Similar results can be expected for other valve types when bodies are designed primarily on a strength basis. Since the design requirements for the body of the butterfly valve were not primarily based on strength, consideration was not given to substituting a more efficient material for the aluminum. Substituting more efficient materials would have varying effects on the different valve types, but it is doubted that the effect would alter the relationship between the weight curves to any great extent. As previously pointed out, the reason for the large weight saving for the angle type valve was partially due to the change in design and construction. The weight savings can be substantial and indicate what to expect; however, the design will have to be proven through testing and development. Each valve is an individual problem and must be handled as such.

SECTION IV

ACTUATING REQUIREMENTS

A. Actuators

For weight comparison purposes between line pressure actuated valves and valves requiring an external actuator, it was necessary to consider actuators. Preliminary investigations indicated that hydraulic actuators were the most feasible for the high power requirements of the larger valves. Hydraulic actuators are simple in construction, reliable, possess low weight characteristics in the larger sizes, and are adaptable to a variety of mounting conditions. It was hoped that standard manufactured types could be applied, but this proved impractical due to the high power requirements of the larger valves and mounting problems. Consequently, it was necessary to design the actuators. For simplicity the same type of actuator was used over the complete size range and for both pressures. An endeavor was made to incorporate in their design minimum weight consistent with good design techniques. A thorough analysis may have revealed that some other type of actuator might have been more practical in the smaller sizes, but for relative weight comparison, this was not important. Valve weights are presented with and without actuators, Table 1, for comparison purposes with other actuator types if desired.

Two types of hydraulic actuators were employed: (1)piston type, and (2) vane type. The piston type is illustrated in Figs. 1, 4, and 8 and the vane type is illustrated in Figs. 3 and 6. Selection of either type depended on mounting problems, space requirements and the required actuation motion. Their design was based on an operating hydraulic pressure of 3000 psi with a design hydraulic pressure of 2.5 times the operating pressure. For the smaller sizes the operating pressure was reduced to 1500 psi. The actuators were designed for maximum force or torque requirements and since the design line pressure was twice the operating line pressure, they are capable of producing double the maximum operating requirements, which should be adequate for low temperature operation. Their design and weight determination are presented in the appendix along with each valve type. Estimated weights of the actuators are compared in Table 1 and a percentage of actuator weight to valve weight is also presented for the various sizes. Actuator weights include the weight of all necessary mountings. The percentage of actuator weight to valve weight is highest in the smaller sizes and varies from 19 to 10% in the 600 psi class and 18 to 3% in the 60 psi class.

The vane type hydraulic actuator is very flexible and possesses small size for large torque outputs. It can easily be adapted for any rotation from a few degrees to a full 360°. A small four way solenoid valve controls the sequence of operations allowing pressure to be admitted to either side of the vane causing rotation through the desired angle. Operating time and pressure shock can be controlled by variable restrictions in the line.

B. Actuating Force and Torque Requirements

Maximum actuating requirements for all valves except the butterfly valve were assumed to occur during the initial stages of opening with full line pressure across the closure member. With the closure member in the seated position, the combination of pressure forces and coefficient of friction are a maximum. For relative comparison purposes, a metal-to-metal seating arrangement was assumed in all cases with a static coefficient of friction (μ) of 0.25. Maximum torque requirements for the butterfly valve are produced by aerodynamic flow forces similar to those on an airfoil and were assumed to occur at an angle of 70° from the closed position. Actuating requirements were also determined for the poppet and cone valves which were designed for line pressure actuation since they can be actuated by external means without modification if desired. Actuating requirements for the Y pattern globe and diaphragm valves were omitted, since their design was based on line pressure actuation to a great extent. The assumption and methods used in determining the actuating requirements for each valve are presented in Appendix B.

Opening time is governed by a number of time factors involved in supplying hydraulic fluid to the actuating mechanism in sufficient quantities to cause movement of the closure member. The time factors involved for the line pressure actuated valves are similar to those for valves with actuators. These time factors are governed by: (1) hydraulic supply pressure, (2) size of hydraulic lines, (3) volume behind piston or vane, (4) leakage past piston or vane seals, (5) spring forces to be overcome, (6) time cycle of control valve, and (7) overall friction. These factors can usually be varied within limits through actual test data to give a desired time cycle. Reasonable operating times would be as follows:

4 inch	1-2 seconds
10 inch	2-3 seconds
20 inch	4-8 seconds

It is impossible to establish a definite operating time for a particular valve type and size by an analytical analysis alone. These values are an indication of the operating times that can be expected in these size ranges.

Maximum estimated actuating torques for the butterfly, slide gate, pivoted gate, eccentric plug, and cone valves are compared over the size range in Figs. 16 and 17 for the 600 and 60 psi valves, respectively. The butterfly, slide gate, and plug types indicate lower torque requirements over the size range in both pressure classes. The reason for the high torque requirements for the pivoted gate in comparison to the slide gate is due to the lever arm of the resisting force. The scotch yoke mechanism of the slide gate valve results in a minimum resisting lever arm at the position of maximum resisting force, closed position.1/ The gate carriage arm of the pivoted gate valve is considerably larger and remains constant. The high torque requirements of the cone valve are due to the taper of the cone which results in large normal forces and hence, large resisting friction forces.

Maximum estimated actuating force requirements for the poppet and angle valves are compared over the size range and for both pressure classes in Fig. 18. The actuation requirements for the poppet valve are considerably less due to the small area of the poppet exposed to the pressure differential in the seated position.

SECTION V

DIMENSIONS

In the design of the various valves, dimensions were kept to a minimum consistent with good design practice. Relationships were determined for the various dimensions from the layout and design of the 10 inch valves and were used in estimating the dimensions for the 4, 16, and 20 inch valves for both pressure classes. The following dimensions were determined for each valve: "A", flange to flange; "B", overall height (maximum dimension perpendicular to pipe axis -- includes actuator where necessary); "C", overall width; and "D". The relationships of the dimensions to the various valves are indicated in the valve layouts, Figs. 1 through 9. Dimension "A" for the angle and diaphragm valves is

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1/ See torque determinations for the slide gate valve in Appendix B.

not a flange to flange dimension because of the angle configuration. The various dimensions are compared over the size range in Figs. 19 through 22 for the 600 and 60 psi pressure classes. Standardizing the descriptive dimension of valve size as the outside diameter resulted in very small dimensional differences between the two pressure classes. The butterfly and gate type valves have minimum flange to flange dimensions (dimension "A") as expected, and the gate types have large dimensions perpendicular to the pipe axis (dimension "B"). The butterfly valve which indicates the lowest weight trend also has minimum dimensional characteristics. The flange to flange dimension ("A") has an influence on the overall system weight. A large dimension "A", which is characteristic of some valve types, takes up more length along the pipe and consequently less pipe is needed. This could conceivably influence the selection of a valve, but would depend on the system requirements.

It should be emphasized that these dimensions are estimated dimensions based on the designs shown and can be affected by change in construction or use of a different actuator; but, as with the weight computations, they represent good relative comparisons.

SECTION VI

SEALING INVESTIGATION

Three general types of sealing problems were posed in the design of the large size aircraft valves and investigated during this study. Included are static seals -- where the seal is used to prevent leakage across a joint in which there is no relative movement between the members involved, such as line joints or bonnets and cover plates; dynamic seals used to prevent the flow of fluid between members that move relative to one another. The problem of the dynamic seal affects all valves using a shaft, stem, or rod to move the shutoff member. The third problem concerns the sealing action between the moving valve closure member (gate, poppet, or plug) which obstructs the flow and the surface or seat on which this member rests.

Static Seals

The "O" ring is in general proposed for use as a static seal. There is a good deal of published information regarding

its principle of operation and efficiency when so applied. It offers a cheap, simple, effective, and re-usable type seal that does not require excessive bolting and consequently smaller flanges.

A possible disadvantage might be in obtaining a suitable material to operate satisfactorily consistent with the design considerations. The silicone rubbers give good indication of satisfying the temperature range requirements, but react unfavorably with all of the fluids except liquid oxygen and liquid ammonia.

The fluoro-plastics (Teflon and Kel-F) seem well suited for application with all of the fluids and Teflon possesses adequate temperature resistance through the range of interest. An important aspect of design and possibly worthy of developmental effort is the matter of proper groove dimensions that the mechanical properties of the "O" ring materials might impose.

Dynamic Seals

All of the valves with the exception of the diaphragm valve require some sort of rod seal. The motions involved are intermittent and either reciprocating or rotary. Since the shaft or rod speeds are relatively low, a satisfactory solution might apply to both. From a mechanical standpoint, the "O" ring presents a satisfactory answer; however, difficulties arise in proper selection of materials. No known elastomer compatible with any or all of the fluids would be suitable for prolonged use at the upper limit of the design temperature. Teflon is suitable as a material but certain mechanical properties such as a "cold flow" tendency and low resiliency reduce its sealing efficiency. A-N type "O" rings (BUNA) have been used in hydraulic components tested at 450°F (ref. 30) but the testing time was relatively short (one hour) and deterioration and permanent set had taken place in the "O" rings. They were not re-usable. "O" rings using a silicon rubber core and Teflon coating might yield a satisfactory solution. Other methods of rod packing that might be used to better utilize the advantages of "Teflon" are vee or chevron packing, lip or flange seals, and "U" packing.

Vee Ring

Vee packings are illustrated in Figs. 6, and 8 and are generally stacked with the number of rings depending upon the internal pressure (3 for 600 psi). The legs face the pressure. Rings bear against adjacent rings and exert pressure on the valve stem. Force is applied by movement of a gland not against the ring set. Spring loading of "Teflon" packing is suggested

for gland seals that must operate over large temperature ranges. The spring will accommodate the large thermal expansion of the "Teflon" without exerting excessive pressure against the shaft or stem.

Lip or Flange Seals

Lip seals are illustrated in Figs. 2, 4, and 10. The lips face in the direction of the applied pressure and the pressure forces the lips to hug the surface causing a seal.

"U" Packing

The U-packing also is installed with lips facing the applied pressure and expanders are used to keep the lips in contact with the stem.

Valve Seats

A. Desirable Qualities

The third and most important sealing problem is that of valve seating. In general, a type of seating would be selected to provide the benefits chiefly desired for each application. However, there are certain common qualities that a satisfactory seat should possess with regard to the following:

1. Shutoff -- Due to the toxic and/or explosive nature of the fluids involved, zero or small leakage only can be tolerated. Leakage is caused (disregarding material failure) by misfitting of mating parts. Such misalignments may be caused by unequal thermal expansions and contractions, vibrations, or improper installation. If pressure forces drop below minimum values, leakage may occur between seat and closure member due to improper sealing forces. In any event, to compensate for such variation, some flexibility must be designed into the seating arrangement by use of a relatively soft resilient material, by the use of springs, or by the use of joints in which some movement is allowed.

2. Service Life -- Common to all of the large fast operating valves to a somewhat varying degree are large seating forces and impact loads; choice of tough durable materials of construction is most desirable for extended service. Rugged construction will reduce deflection and reduce to some extent the necessity for flexibility.

3. Repair and Maintenance -- Possibly in compromising durability to obtain flexibility and tight shutoff, seat service life will be affected to a point where frequent replacements will be a necessity. Seats should be readily accessible for rework or replacement.

B. Seat Types

Valve seats can be divided basically into two types: soft seat and hard seat. For flexibility and tight shutoff the soft seat is the best (Ref. 36); however, materials available for its application consistent with design pressures, temperatures, and fluids are very limited. "Teflon" is the only material that could be utilized as a soft seat and remain serviceable throughout the temperature range. It also has excellent resistance to all of the fluids, with a limited flexibility in the presence of liquid oxygen as its only disadvantage in this respect. "Teflon" must be kept confined or supported as much as is practicable to combat an inherent "colf flow" or thermo-plastic condition that becomes increasingly worse at high temperatures. Soft seat types are illustrated in Figs. 2, 3, and 7. Metal-to-metal or a hard seat is the other classification. High strength durable materials are available for use throughout the temperature range and suitable for use with all of the fluids. Without exception, excellent machine finished (ground and polished) and very close tolerances in hard seat applications would be required. It is doubtful whether tight shutoff could be effected with this seating method for extended periods of time because of wear accompanying valve operation and tending to damage the well machined surfaces, especially for the gate and plug types.

C. Seating Methods

1. Gate Valves -- There are four basic configurations for the "gate" or flow obstructing member in this valve. Either a soft seat or hard seat could be utilized. The following advantages and disadvantages are characteristic of gate valves in general:

General Advantages

(1) Seals equally well in either direction
(2) Tendency for errosion of seating surfaces is small

General Disadvantages

(1) Tendency for seizing and galling
(2) Seating surfaces subjected to wear by nature of movement

a.   Single Slide -- This is the simplest of the gate designs and is illustrated with the sliding gate valve (Fig. 3). This design employs a hardened steel slide seated on "Teflon" ring seals. The "Teflon" seals are used with a metallic contour ring and a resilient backing. The backing could be either a single flat metal spring or a number of small helical springs spaced around the circumference of the seal ring. This type seating arrangement would also be adaptable to the pivoted gate valve shown in Fig. 1.

<u>Advantages</u>

(1) Flexibility over the temperature range. Lends itself reasonably to seat angle change. This would be contingent upon the resiliency of the backing.
(2) Although wear due to abrasion is definitely a disadvantage, the good lubricity of "Teflon" would reduce this effect.
(3) Teflon minimizes seizing and galling
(4) Seals are easily replaceable, although valve would have to be removed from the line to replace them.

<u>Disadvantages</u>

(1) There is some question as to length of service life. Although the "Teflon" is used with a contour ring and confined to some extent in a groove, the abrading and twisting action incited during opening and closing coupled with large loads transmitted by gate to seals could be detrimental.

b.   Two Piece Slide -- The pivoted gate valve, Fig. 1, incorporates a two piece slide with a cast in hardened steel seat. The slides are free to rotate during opening and closing and also to move independently of the carriage arm. Line pressure provides the sealing force with a spring assisting at low pressures.

<u>Advantages</u>

(1) Should compensate for misalignments due to temperature variation, line strains, and vibrations.

<u>Disadvantages</u>

(1) Seats not too accessible for regrinding.
(2) Nicks and scratches on seating surfaces would cause leakage. Use of "Teflon" insert in the seating surfaces of the gates could enhance the sealing ability but would require larger seating faces due to the lower compressive strength of the softer material. In the event of failure, replacement of

the entire disc would be a possibility as a
permanent bond between the disc and insert
would be required in order to avoid extrusion
of the insert.

(3) Disc chatter during opening and closing is a
possibility with high line velocities.

c. Solid Wedge Gate -- A metal-to-metal method of
seating. Consists of a single gate machined to a wedge
shape with seats arranged to receive it. This type of
construction offers good performance if not subjected to
transverse loads, which would destroy proper seat contact.

d. Split Wedge Gate -- The two faces of the disc are
separate pieces and are forced into a proper closing by a
suitable spreading arrangement which is loaded by the valve
actuating mechanism; in opening, the spreading force is
released, permitting disc halves to collapse slightly.
Any change in seat angle is readily followed by the split
wedge disc. The split wedge like the solid wedge is
adaptable only to valves having a straight path of open-
ing and closing as the sliding gate. The wedge arrange-
ments would improve the gate valve hard seat life because
of a decided reduction in wear, due to the disc pulling
away from the seat at the time of opening. Seizing and
galling would be a tendency and quite likely breakout loads
would be greater than with the parallel arrangements.

2. Globe Valves -- The globe valves employ three basic
methods of seating. They are: flat disc, plug or semi-plug,
and bevel type seating. All could utilize either a soft or
hard seat with the possible exception of the bevel type. The
following advantages and disadvantages are characteristic of
globe valves in general:

General Advantages

(1) Little wear
(2) No tendency toward seizing and galling
(3) Tight shutoff (small leakage)

General Disadvantage

(1) Seating surfaces subjected to errosion by fluid flow.

a. Flat Disc -- The "Y" pattern globe valve (Fig. 2)
and diaphragm valve (Fig. 7) incorporate flat disc seating
arrangements. The Y pattern globe valve employs an all-
metal disc with a removable metal seat ring containing a
"Teflon" insert. The diaphragm valve employs an insert
in the disc.

Advantages:

(1) Seats more readily than metal-to-metal contact. In closing the soft seat yields and adjusts itself to a small projection on the upper face of the disc or seat ring as applicable.

(2) Foreign matter on the seat which might cause leakage in a metal disc valve often will imbed itself in a soft material without impairing seat tightness.

(3) Seat can be removed, repaired, and replaced without removing valve from the line.

Disadvantage

(1) The "Teflon" insert offers a source of early seat failure.

b. Plug or Semi-Plug -- This method of seating is analagous to the action of a glass bottle and stopper. The semi-plug is differentiated from the plug by a shallower cone depth. The former is shown on the line pressure actuated poppet valve (Fig. 5). Either soft or hard seats can be utilized. In the case of a soft seating, the soft member can be contained in either the poppet or the seat ring. The plug type seat offers the most favorable application of "Teflon" as a soft seat.

Advantage

(1) Nicks and scratches compensated for by large seating area.

Disadvantage

(1) The design shown does not indicate a great amount of flexibility; however, dimensional instability in the axial direction should be adequately compensated for due to the flexibility of poppet movements in this direction. This type seating arrangement could be used with any of the globe type valves and in a valve using a stem more flexibility could be obtained by using a joint, stem to poppet, that would allow some movement.

c. Bevel Type Seat -- This seating arrangement is a modification of the ball disc providing a virtual line contact between seat and disc. Various angles of bevel can be used. This method is used on the angle valve shown in Fig. 4.

Advantages

(1) High unit sealing pressures developed because of small area of contact.

(2) Seats easily replaced or reworked; however, valve
must be removed from line.

<u>Disadvantages</u>

(1) Little flexibility
(2) Nicks and imperfections will cause leakage. This
seating method could be used with any of the globe
type valves.

3. Plug Valves -- There are various methods now in use
for seating plug valves. The seating method is dependent upon
the valve design. The following advantages and disadvantages
are characteristic of plug valves in general:

<u>General Advantages</u>

(1) Seating surfaces usually not in contact with fluid
(2) Seals equally well in either direction

<u>General Disadvantages</u>

(1) Tendency for seizing and galling
(2) Poor sealing without use of lubricants
(3) Accurate fit of plug to seat requires close tolerances
and excellent surface finish

a. Cone Plug -- This is the method commonly used for
plug valves. A cone shaped plug cock seats directly
against the inside surface of revolution of the body.
Essentially, this is the type seat shown for the cone
valve in Fig. 9.

<u>Disadvantages</u>

(1) Multiple ports to be sealed offers more chance
for leakage.
(2) Not much flexibility. The only alternative would
be to heavily rib the body to minimize distortion
of the taper bore.

b. Segmental Plug -- Another application of cone plug
seating is in the eccentric plug valve (Fig. 8). The valve
has a segmental eccentric plug and an eccentric raised seat
in the body; being eccentric the plug contacts the seat
only in the full closed position.

<u>Advantages</u>

(1) Slight opening rotation of the plug totally re-
leases the face from contact with the body seat,
eliminating to some extent binding, seizing, and
scoring.
(2) Ultimate wear is compensated for by plug taking a
more advanced position.

Disadvantages

(1) Little flexibility.
(2) Seats subjected to errosion by fluid flow.
(3) Difficult to machine perfectly matched pairs.

To combat the inherent seizing and galling in conical plug valves a lift-turn-reseat operating principle is used, in which the plug is cam lifted from the seat, rotated to the desired open or closed position, and then reseated by the cam mechanism. One plug valve manufacturer, to improve sealing without the use of viscous lubricants, incorporates a "floating" seat which is independent of the body. The seat amounts to a metal liner sealed against the inside surface of revolution of the body by an "O" ring at each port. A primary metal-to-metal seal between plug and seat is accomplished by a wedge fit. Pressure forces add to tightness by forcing the liner into more intimate contact with the plug. This arrangement has an advantage of flexibility but is limited as to service by the use of the "O" rings.

Another alternative on the plug cock or conical plug design is the use of a reinforced "Teflon" sleeve to separate plug and body. This method has been utilized successfully in small sized valves, and has furnished positive sealing requiring no lubrication. There is some question as to the adaptation to large sizes and high pressures because of the characteristics of "Teflon".

"Teflon" ball seals have been successfully used with spherical plug valves at the temperatures and pressures stipulated for the design study, but again there is no data available to indicate their use in larger sizes.

4. Butterfly Valves -- Four methods are indicated as possibilities for the seating of the shutoff member or vane on this valve. Two of the methods, expandable gland and lip seal, have been successfully used with pressures in excess of 600 psi and temperature limits of -300° to 165°F (Ref. 8).

a. Expandable Gland -- This design is incorporated on the butterfly valve shown in Fig. 6. When the vane is moved to the closed position, the gland is pressurized and forced against the vane for shutoff. Before the valve is to be opened, the pressure in the gland is relaxed.
Advantages

(1) Can be fabricated from materials compatible with all fluids and serviceable throughout the temperature range.
(2) Seal does not contribute any resistance to valve opening.

(3) Should be flexible enough to compensate for nicks and scratches in the vane.

Disadvantages

(1) Problem of adequately containing the "Teflon"
(2) Since the seat depends upon pressurization, any susceptibility of the pressurizing system to malfunction introduces more of a chance for valve failure.

   b.   Lip Seal -- Also developed for the valve mentioned in Ref. 8 was a plastic type (Kel-F) lip seal. Presumably the seal is of a "Z" cross section, with one leg of the "zee" fixed in the body and the other leg overlapping the vane as a lip on the pressure side. Pressure forces cause intimate contact of the lip with the gate. In opening, half of the vane withdraws from the lip, half passes through it, and deformation occurs at the axis of rotation. This type seat allows a greater deflection in the vane without loss of sealing efficiency.

   There are other methods of butterfly valve seating, but they have limiting factors due to pressure, temperature, or nature of fluid handled. Use of an "O" ring in either the vane or the body is limited by materials to approximately 350°F. By proper groove design the likelihood of "O" ring "blowout" can be minimized but indications are that maximum pressure is about 300 psi. Use of a soft resilient liner in the body has proved successful for efficient seating, but again the service would be in the lower pressure class and upper temperature limit well below the 450°F.

   In general it is not possible to predict the service life of a given seating arrangement nor its sealing efficiency without proper testing. Due regard must be paid to proper choice of materials, quality of workmanship, and proper installation. Cognizance of these factors could probably produce tightness in all of the valves at the time of manufacture; however, operating conditions would impair the seating surfaces until leakage developed. Some seating designs would be adversely affected more quickly than others. The globe valves quite possibly would give the tightest seating and longest service life. They particularly offer better opportunity for the use of "Teflon" as a soft seat. A table summary with a generalized comparison relative to shutoff, fabrication and maintenance, and expected service life is presented in Table 4. Shutoff considerations include effects due to variations in pressure and temperature and effect of distortion

due to vibration and other means. Fabrication and maintenance
might include difficulty of fabrication due to complexity of
shape or tolerances and material finishes required; and diffi-
culty of maintenance due to inaccessibility and tools required.
Service life encompasses the tendency toward wear and how it
might effect shutoff and consequently repair and maintenance.

SECTION VII

PRESSURE DROP

Pressure drop characteristics of valves are difficult to
analyze and compare without actual testing of the valves.
Approximate comparisons can be achieved by employing use of a
valve resistance coefficient or equivalent length. Experiments
have shown that the pressure drop caused by a valve can be re-
presented by an equation of the form:

$$h_L = K \frac{v^2}{2g} \tag{2}$$

where
h_L = pressure drop in feet of fluid

v = mean velocity in pipe, fps
g = acceleration of gravity

The coefficient "K" is known as the valve resistance coefficient
and is practically constant for any given valve over a large
range of Reynolds number, providing the flow is turbulent. In
general, the higher the resistance of the valve, the more nearly
independent the coefficient is of valve diameter. A comparison
of the valve resistance coefficient "K" for the various valve
types gives a good approximation of pressure drop characteristics.

Valve resistance coefficients have been determined and
estimated for the various valves and are compared in Table 4.
Actual flow data were available for determining "K" values for
the pivoted gate, slide gate, butterfly, and eccentric plug
valves; and "K" values for the other valve types were determined
from handbook data or estimated. These "K" values in some cases
may not be representative of our exact design but are reasonably
close enough for comparison purposes. Test results indicate
that globe type valves with a stem angle of 60° have "K" values
about 4, which is in the same range as for angle type valves.
Since the diaphragm valve (Fig. 7) has a larger total deflection
angle for the fluid path, its resistance coefficient is higher

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than for the angle valve. A plug valve with a straight through concentric flow passage having a minimum flow resistance, equivalent to that of the pipe, and a weight approximately the same as the eccentric plug valve, is used as a standard for pressure drop comparison purposes.

Using these "K" values, a diameter was calculated for each valve type based on equivalent flow capacity and pressure drop as for the 10 inch plug valve having minimum pressure drop characteristics. Valve weights were determined and estimated for these various diameters using the 600 psi weight curves. Ratios of the various valve weights to the weight of the plug valve having minimum pressure drop characteristics are compared in Table 4 and provide a basis for comparison of valve weights based on equivalent pressure drop and flow. The gate types (slide and poppet), poppet, and butterfly types indicate low weight for a given flow and pressure drop. Assuming a line velocity of 30 fps and a hydrocarbon fuel (s.g. 0.8), the additional pumping horsepower required by the flow loss through the valve was determined for the various 10 inch valve types and compared in Table 4. These horsepower values are directly proportional to the "K" values and consequently the valve types having low "K" values require low pumping power.

The pressure drop characteristics of the valve designs represented in Figs. 1 through 9 could be improved upon without weight penalty in several instances by rounding off sharp corners on the poppets and seats and applying more efficient aerodynamic shaping of the flow passage and flow obstruction member.

SECTION VIII

SUMMARY

Characteristics of nine types of valves suitable for shut-off use in 4 to 20 inch aircraft and missile fuel lines have been evaluated. Nine types for detailed investigation were selected in a preliminary study and representative designs were prepared for each type. Weight, dimensions, actuating requirements, sealing and pressure drop were evaluated for 60 and 600 psi application with an operating temperature range from -65° to 450°F.

This study indicates that the weight of a 10 inch, 600 psi valve can be expected to vary between approximately 85 and 260 pounds, depending on the type of valve selected. Weight variation of a similar magnitude may be expected between types in other sizes or when designed for 60 psi service. The body weights alone amount to 50% or more of the total valve weight. Design methods for analyzing the strength of complex shapes such as valve bodies are very inadequate and use of experimental stress techniques is indicated for large size aircraft valve development. A means of estimating the weight of a valve of given type when the weight of a different size is known is presented.

The adverse temperature conditions associated with high speed flight make material selection and type of construction an important factor. When designing for the upper temperature limit weight reduction of as much as 30% can be realized through material and fabrication selection.

The type of seal to be employed on various types of valves has been investigated; however, there is very little information available on the quantitative behavior of the various seal arrangements and only qualitative conclusions can be drawn.

The butterfly, pivoted gate, and angle types of valves are superior for both pressure classes of service. The butterfly type valve is far superior in most respects and warrants considerable attention to the problem of effective sealing at higher pressures and temperatures.

SECTION IX

CONCLUSIONS

1. The butterfly, pivoted gate, and angle type valves indicate lower weight trends in the 600 psi pressure class. These three types, together with the poppet type, indicate lower weight trends in the 60 psi class. In both pressure classes large weight differences exist between the various valve types.

2. The weight of the valve bodies constitutes from 50 to 80% of the total valve weight. In the development of specific valves substantial weight savings may be obtained by applying experimental stress analysis techniques.

3. The upper operating temperature limit should be very carefully selected because of the great effect of temperature on weight and material selection. For example at 450°F calculations indicate that a wrought titanium valve body will be as much as 30% lighter than the same article in cast aluminum.

4. To handle a given fuel flow four valves will produce a lesser total valve weight (and probably cost) than one valve of twice the diameter. For example, four 10 inch, 600 psi valves might weigh approximately 560 pounds; in contrast one 20 inch, 600 psi valve which would handle the same flow might weigh 1000 pounds.

5. The globe and poppet type valves offer the most promising sealing characteristics. However, available seal information is sketchy and at best qualitative. A definite need exists for the development of quantitative seal information.

6. Pressure drop appears to be important primarily as it influences the valve size and therefore weight of a given installation, i.e., different sizes of various valve types will be necessary if a given system and therefore valve pressure drop is to be maintained. Comparing all valves to the minimum possible pressure drop indicates an extremely wide variation in weight.

7. Hydraulic actuators appear to be the most feasible for large power requirements and are not a determining factor in the valve weight (Table 1). The vane type hydraulic actuator as illustrated in Figs. 3 and 6 is very flexible and possesses small size for large torque outputs.

SECTION X

RECOMMENDATIONS

1. Results of this study have indicated that future development work can most profitably be concentrated on the butterfly, pivoted gate, and angle type valves for both 600 and 60 psi pressure classes of service over the size range of 4 to 20 inch diameter. The poppet type valve should also be considered for 60 psi application. For a specific application, various design configurations of these types should be thoroughly investigated in order to obtain the optimum configuration resulting in best overall performance.

2. Body design and fabrication of the various valve types should be thoroughly investigated for application of wrought materials to conserve weight.

3. Since accurate analytical design of valve bodies is difficult if not impossible in most cases, it is recommended that experimental stress design techniques be applied. Better material distribution and lower valve weights should result where these techniques are feasible.

4. In order to provide a basis for evaluating and comparing the sealing characteristics of the various seating arrangements for the basic valve types, it is recommended that information on leakage and service life be developed for various operating pressures and temperatures. The effects of material, surface finish, hardness, and other similar variables should be considered.

5. Since the butterfly valve is definitely superior in most respects, the problems connected with sealing the vane at high pressures and temperatures should receive concentrated attention.

6. The problems arising from substituting several smaller valves for one large valve to conserve weight should be carefully investigated and evaluated.

7. The use of high line velocities in order to reduce line and valve size seems worthy of investigation.

TABLE 1 -- COMPARISON OF ESTIMATED WEIGHT CHARACTERISTICS -- 600 PSI VALVES

Valve Type	Size	Valve Weight Without Actuator- Lbs.	Valve Weight With Actuator- Lbs.	Actuator Weight[c] Lbs.	Valve Weight / Actuator Weight	Body Weight Lbs.	Valve Weight / Body Weight	$W = KD^n$ "n" Values
Pivoted Gate	4	8.25	10.3	2.05	19.9	6.10	59.2	2.70
	10	133.0	153.0	20.0	13.0	96.2	62.8	
	16	400.0	462.0	62.0	13.4	263.0	56.9	
	20	765.0	865.0	100.0	11.6	514.7	58.3	
Slide Gate	4	12.6	14.44	1.84	12.7	10.9	75.4	2.96
	10	170.6	200.0	29.4	14.7	144.0	72.0	
	16	644.6	772.0	127.4	16.5	556.0	72.0	
	20	1249.6	1509.0	259.4	17.2	1031.0	68.3	
Y-Pattern Globe[a]	4		20.9			11.35	54.3	2.71
	10		263.1			144.3	54.8	
	16		1030.7			574.0	55.7	
	20		1927.3			1023.0	53.0	
Angle	4	10.28	11.74	1.46	12.4	5.58	47.5	2.76
	10	132.5	148.9	16.4	11.0	73.0	49.0	
	16	492.4	555.6	63.2	11.4	277.6	49.7	
	20	868.4	1003.8	135.4	13.5	487.2	48.5	
Poppet[a]	4		14.62			8.8	60.2	2.92
	10		222.0			127.4	57.5	
	16		859.0			523.0	60.9	
	20		1616.0			949.0	58.8	
Butterfly	4	6.15	6.98	0.83	11.9	5.29	75.8	2.76
	10	75.4	83.9	8.5	10.1	60.2	71.7	
	20	532.8	591.8	59.0	10.0	401.4	67.7	
Diaphragm[a,b]	4		18.3					2.83
	10		239.9			159.6	66.4	
	16		935.0					
	20		1755.0					
Eccentric Plug[b]	4		19.3					2.78
	10	207.6	242.9	35.3	14.5	141.2	58.1	
	16		951.0					
	20		1780.0					
Cone[a,b]	4		21.0					2.85
	10		263.9			199.4	75.5	
	16		1030.0					
	20		1930.0					

Average Value 2.81

Valve Type	Size	Valve Weight Without Actuator- Lbs.	Valve Weight With Actuator- Lbs.	Actuator Weight[c] Lbs.	Valve Weight / Actuator Weight	Body Weight Lbs.	Valve Weight / Body Weight	$W = KD^n$ "n" Values
Pivoted Gate	4	4.83	5.5	0.67	12.2	3.72	67.7	2.47
	10	48.3	52.0	3.7	7.1	31.5	60.6	
	16	157.0	168.0	11.0	6.6	95.6	56.9	
	20	246.0	265.0	19.0	7.2	127.0	47.9	
Slide Gate	4	6.5	8.0	1.5	18.7	5.54	69.3	2.57
	10	74.0	79.0	5.0	6.3	60.2	76.2	
	16	271.0	293.0	22.0	7.5	205.0	70.0	
	20	536.5	577.0	40.5	7.0	423.0	73.2	
Y-Pattern Globe[a]	4		6.49			3.08	47.4	2.61
	10		67.2			25.4	37.8	
	16		236.8			73.4	31.0	
	20		426.8			135.6	31.8	
Angle	4	3.62	4.06	0.44	11.8	2.26	55.7	2.38
	10	31.0	34.0	3.0	8.8	18.3	53.8	
	16	101.4	109.7	8.3	7.6	57.7	52.6	
	20	182.2	195.5	13.3	6.8	98.8	50.6	
Poppet[a]	4		4.7			2.86	60.8	2.62
	10		52.6			24.1	45.8	
	16		173.7			70.8	46.0	
	20		335.0			155.5	46.4	
Butterfly	4	2.92	3.17	0.25	7.9	2.55	80.4	2.65
	10	33.3	34.5	1.21	3.5	21.3	61.7	
	20	224.9	231.8	6.9	3.0	125.3	54.1	
Diaphragm[a,b]	4		5.9					2.58
	10		63.5			27.3	43.0	
	16		209.0					
	20		382.0					
Eccentric Plug[b]	4		6.6					2.53
	10	59.6	67.7	8.1	12.0	37.0	54.6	
	16		239.5					
	20		431.0					
Cone[a,b]	4		6.8					2.56
	10		70.0					
	16		247.5					
	20		445.0					
Average Value								2.55

a – line pressure actuated valves
b – valve weights for the 4, 16, and 20 inch sizes were estimated
c – includes weight of mountings

TABLE 2 -- STRENGTH EFFICIENCY AT ROOM TEMPERATURE AND 450°F FOR VARIOUS MATERIALS

Material	Designation	Modulus of Elasticity PSI x 10^6	Density lb/cu.in.	Compressive Yield Stress		Compressive Yield Strength/Density	
				Room Temp. ksi	450°F ksi	Room Temp.	450°F
			WROUGHT ALLOYS				
Steel	SAE 4340	30	0.283	200	165	706	583
Aluminum	75 S-T6	10.5	0.10	78	15	780	150
Aluminum	24S-T4	10.5	0.10	48	29	480	290
Magnesium	AMC 58S-T5	6.5	0.065	37	10	570	154
Titanium	RC-130B	17.7	0.17	152	100	894	588
Stainless	18-8	28	0.286	160		560	
			CAST ALLOYS				
Steel		28	0.283	120	100	423	352
Aluminum	356-T6	10.3	0.10	27	7.5^a	270	75
Aluminum	122-T551	10.3	0.10	35	16^a	350	160
Magnesium	AR 240-T61	6.5	0.065	22	7.4^a	338	114
Magnesium	AM 246-F	6.5	0.065	15.6	8.3^a	240	128
Titanium[b]							
Fiberglass	Phenolic	2.5	0.065	40	20^c	616	308

a - after 10,000 hours of heating/the other values are for short duration heating
b - Titanium not castable in large sections
c - after 200 hours of heating

TABLE 3 -- GENERAL SUMMARY OF SEALING CHARACTERISTICS

Valve Type	Sealing Configuration	Seat Type	Shutoff	Fabrication & Maintenance	Service Life	Overall Efficiency
Gate	Parallel	(Soft (Hard	Good Fair	Very good Fair	Good Fair	Good
	Wedge	(Soft (Hard	Very good Good	Fair Fair	Good Good	
Globe	Plug	(Soft (Hard	Excellent Very good	Very good Very good	Very good Excellent	
	Bevel	Hard	Very good	Good	Fair	Very good
	Flat	(Soft (Hard	Very good Fair	Very good Very good	Good Very good	
Butterfly	Expandable Gland		Good	Poor	Very good	
	Lip		Good	Good	Good	Fair to good
	"O" ring or liner		Poor	Good	Poor	
Cone	Rotating[a]		Good	Poor	Fair	
	Lift-Turn-Reseat		Good	Good	Good	Good
	Teflon Sleeve		Very Good	Fair	Very good	
Eccentric	Segmental		Good	Fair	Very good	

a - No sealing lubricant

TABLE 4 — PRESSURE DROP CHARACTERISTICS

Valve Type	Valve Resistance Coefficient "K"	Diameter for Equivalent Flow and Pressure Drop as 10" Plug Valve	Weight of 600 psi Valve	$\dfrac{\text{Weight of Valve}}{\text{Weight of Plug Valve}}$	Pumping HP Due to Pressure Drop Thru 10" Valve
Slide Gate	0.09	12.2	364	1.49	1.12
Pivoted Gate			244	1.0	
Poppet[a]	0.3	16.6	960	3.92	3.74
Y-Pattern Globe	4.0	31.6	6000[c]	24.5	49.9
Angle			3300[c]	13.5	
Diaphragm[a]	6.0	34.9	8000[c]	32.7	74.8
Butterfly	0.6	19.7	572	2.33	7.48
Eccentric Plug	1.90	26.3	3300[c]	13.5	23.7
Cone[a]	2.5	28.1	5300[c]	21.6	31.2
Plug[b]	0.04	10.0	245	1.0	0.50

a – "K" values estimated

b – Plug valve with straight thru concentric flow passage having minimum flow losses

c – Interpolated

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A	Area of cross section, square inches
b	Width of sections; subscript "bending"
c	Fixity coefficient for columns; distance from neutral axis to extreme fiber; subscript "compression"
D	Diameter
d	mathematical operator denoting differential
E	Modulus of elasticity in tension; average ratio of stress to strain for stress below proportional limit
F	Allowable stress
f	Internal (or calculated) stress
F_{su}	Ultimate stress in pure shear. This value represents the average shearing stress over the cross section.
F_{ty}	Tensile yield stress at which permanent strain equals 0.002 (from tests of standard specimens)
G	Modulus of rigidity
I	Moment of inertia
J	Torsion constant ($=I_p$ for round tubes)
K	A constant, generally empirical
L	Length; subscript "lateral"
M	Applied moment or couple, usually a bending moment
P	Applied load (total, not unit load)
psi	Pounds per square inch
r	Radius
T	Applied torsional moment torque
t	Thickness
w	Specific weight, lb/cu. in.
Z	Section modulus, I/y
δ	Deflection
ρ	Radius of gyration
μ	Poisson's ratio
∅	Coefficient of friction
m	Reciprocal of μ (Poisson's ratio)
c_t	Coefficient of torque
Δp	Pressure drop, psi
W	Flow rate, lb/sec
c_f	Coefficient of flow
p	Unit load, psi
m.s.	Margin of safety
i	Inside
o	Outside

GENERAL DISCUSSION OF VALVE STRUCTURAL ANALYSIS

It will be noticed that the stress analysis utilized in the design of the various valve parts for weight investigation exhibits considerable variation in exactness from part to part. In the design of the actuators and of the internal parts of the valves, for the most part the expressions used may be expected to give stress values quite close to the true values for the design conditions. It is true that in certain cases, the exact boundary condition was somewhat in doubt; in such an instance, the more conservative of the limiting conditions was used. However, with this reservation, the bulk of the expressions may be said to be exact.

However, in the case of the valve body itself, the same cannot be maintained. In a rather large number of cases, the cause of inexactness may be said to be twofold. In the first place, the various surfaces of the valve body were short enough so that very few parts may be said to be free from the effects of relatively distant parts, which added to the uncertainty concerning the boundary conditions. The principal difficulty arose from the fact that in general, the valve bodies are a complex of geometrical shapes for which exact theoretical stress expressions simply do not exist. In particular, such expressions are unavailable which describe the stress conditions near and at the intersection of two such shapes.

The reason for the absence of such expressions is of course the mathematical complexity involved. Industrial valve companies go as far as they can in the design of the valve body by utilizing the few relatively simple equations that do exist and then resort to some form of experimental stress analysis to complete the design.

As a result, in the design of the valve bodies for weight comparisons in the present study, simplifying assumptions were necessarily made for some cases; the resulting expressions are, however, sufficiently accurate for weight comparison purposes.

In a very few cases, more or less common to all the valves, more exact expressions than those used above do exist; it remains to check the results obtained by the use of these equations with those obtained from the simpler expressions actually used in the valve design.

Flanges

Bolted flanged connections contribute significantly to valve total weight (10 to 20% for the valves studied) indicating a need for a relatively accurate design method. Methods have been developed (1) for flange design in which the gasket or contacting flange surfaces are entirely within the bolt circle. However, the flat-faced flange with the accompanying "O"-ring gasket was adopted for use in the design of the valves studied and when surface contact occurs beyond the bolt circle, the resulting moment tends to help prevent the flange ring from rotating, and smaller flange dimensions may be used.

As yet no acceptable solution has been developed for this particular case. One approach is to consider the case of the full face gasket with flat flanges (35). The results may also be expected to be on the conservative side.

For comparison purposes, design of the flanges was effected using this method, and was compared with the results of the method actually used in the valve design. Such comparisons indicated that the percent deviation between results of the two methods increased with increasing valve size. As an illustration, for the 10 inch, 600 pound valve flange, a ring thickness of 1.66 inches was determined using the actual design method. By the full-faced gasket method, a ring thickness of 1.80 inches was required.

Since the latter method is known to be conservative, the actual design method, although less rational, is felt to be approximately correct.

It may be mentioned that other approaches to the problem of flange design exit (37) (38) but none are directly applicable to the actual design used.

Cylinder Wall Thickness

In nearly all the valves, there occur one or more body parts which may be considered as basically a cylindrical pressure vessel with a circular section. In the design of these parts the modified Barlow equation was used with an additional factor of 25% as compensation for the shape effect of adjacent parts.

Computations were carried out to determine the true shape factor, using a more rational expression for cylindrical pressure vessel thickness than that afforded by the modified Barlow equation. For a 10 inch, 600 pound valve, the arithmetic mean of the values for thickness determined by the Lami, maximum strain-energy (Ref. 39) and maximum distortion energy (Ref. 40) methods was 0.74 inches; the design thickness of 1.12 inches therefore included a true shape factor of 50%. Similarly, for a 20 inch valve, the true shape factor was also 50%, while for a 4 inch, 600 pound valve, the true shape factor was 83%. Thus, 50% rather than 25% represents the minimum factor allowed to provide for the largely indeterminate end condition, local stress concentration, and corrosion effects.

Pressure Vessel Closure

It should be mentioned that one of the few parts of a valve body for which relatively exact theoretical expressions exist is that of a flat head closure to a cylindrical pressure vessel. Watts and Long (41) have obtained expressions for stresses in such a case, applicable however, only to relatively thin-walled vessels. A similar group of expressions appears in Roark (42). A check of the results obtained in the above design of the valves for weight comparison reveals that for the given condition, including cylinder wall thickness, design by the assumption of a clamped uniformly loaded plate give results approximating those of the more exact method. An accurate comparison cannot really be made, however, due to the unknown thick-wall effect.

It might also be mentioned that a number of the plate equations used in the above design have been verified experimentally. In (43), (44), and (45), normally loaded clamped edge rectangular plates, circular plates without, and circular plates with central holes, respectively, were tested; the results were found to agree within a few percent of the corresponding theoretical values.

It has been mentioned previously that little theoretical information is available concerning stress conditions at the junctures of complicated shapes such as are found in the valve bodies of the present study. For the purposes of weight comparison, it was deemed unnecessary to attempt to utilize the experimental information obtained by industrial valve companies, particularly since the valve designs of the present study differed from their commercial counterparts, sometimes markedly, rendering such information of, at best, only qualitative value.

It was recognized from relatively idealized studies (46), (47), that stress concentration factors of the order of 4 or 5 or perhaps even larger may be present at the intersection of surfaces; in the weight design of the valve bodies, generous fillets and rounds were employed to relieve the unfavorable stress conditions while adding negligibly to the weight.

It was also realized that certain other factors exist, dependent upon service requirements, which will influence valve design to a more or less indeterminate extent. For the most part, insufficient data on service conditions has prevented a real evaluation of the extent of the influence of each. Nevertheless, in the design of a particular valve for a particular set of conditions, these factors, listed below, may each play a part.

Thermal Stress

For the temperature range given by the design conditions, thermal stress may or may not be important, depending upon resulting temperature gradient and end restraint which is itself partly a function of temperature. In all probability, lateral restraint will be slight, depending to a large extent upon the method of valve support. Axial restraint, however, is a decided possibility and in valve design for a particular system, would have to be considered.

Cyclic Loading and Fatigue

All parts of each of the valves are to some extent subjected to cyclic loading by repeated operation of the valve, hence the possibility of fatigue failure must be considered.

The nature of the loading varies from part to part. It may be stated that for the most part, complete stress reversal occurs in only a relatively small number of the valve components, i.e., moving parts of the actuators and some actuating shafts. For the rest of the valve there is stress repetition but not reversal. Also, the range of stress varies from a very small value in the case of the valve body to an appreciable magnitude in the case of parts of the actuators.

Fortunately, those parts most subject to fatigue failure have been designed, for the most part, by considerations other than strength, resulting in a generous allowance for repeated stress. As a result, a detailed analysis of the loading cycle

for such parts has been unnecessary, particularly in view of the relatively few number of design cycles (100,000).

As a final note, it may be stated that cyclic loading (vibration) may conceivably occur in the valve installed in the system, due to outside forces, and set up large stresses in any part of the valves.

Creep

No allowance was made for creep in the selection of the allowable stress in the above design of valves for weight for the following reasons:

Creep for a given material is basically dependent upon duration of loading and temperature. Now the maximum duration of loading is unknown; it would appear to be no more than perhaps a few hours. With respect to temperature, it was felt that for the metallic materials used, the maximum design temperature (450°F) was not large enough to produce, in conjunction with the above assumed duration of loading, a creep problem. For the non-metallics used, creep can unquestionably occur. However, if the questionable material is carefully confined, creep should be no problem, even for this case.

Corrosion Fatigue

Corrosion fatigue failure not only requires that a metal be subjected to repeated alternate loading, but also be exposed to a corrosive environment. Although producing a failure similar to ordinary fatigue, this failure can be produced at a much lower stress than the endurance limit of the material due to the added influence of corrosion. Corrosion fatigue cannot always be minimized by elimination of radical changes in shape and avoiding areas of stress concentration because pitting which can accompany corrosion can provide enough discontinuity in geometry to initiate stress concentration and ultimate fatigue failure.

Valve stems, bolts, poppets, and other load bearing components are more susceptible to this type failure than the valve walls. Adequate corrosion fatigue endurance limit information will be required in the design of these large valves.

Surge Pressure

Pressure increase due to valve closure is not dependent upon working pressure, but upon velocity. Maximum increase in pressure will occur when the total flow is stopped within a period of time equal to that required for the pressure wave to travel from the point of closure to the opposite end of the line and return. This time can be expressed

$$t = \frac{2L}{a} \qquad (3)$$

where
- t = time in seconds for valve closure
- L = length of line in feet
- a = wave velocity, fps

The wave velocity is determined by

$$a = \frac{12}{\frac{w}{g}\left(\frac{1}{k} + \frac{D}{Ee}\right)} \qquad (4)$$

where
- g = acceleration due to gravity
- w = fluid density
- D = ID of pipe
- c = thickness of pipe wall
- k = voluminal modulus of elasticity of fluid
- E = modulus of elasticity of pipe material
- v = fluid velocity

and the maximum pressure rise in psi:

$$P = \frac{aV}{g} \times .433 \qquad (5)$$

Using the 10 inch line used in this study for the 600 psi range and assuming the fluid to be water, the maximum pressure rise above static is equal to 1870 psi. The time of valve closure required to produce this rise, assuming a length of line of 5 feet is t = .0022 seconds. It can be seen that to produce a maximum pressure rise the valve closure must be virtually instantaneous. Valve closure in time greater than 2L/4 may also cause pressure surges, but the magnitude is much less than with instant closure. Moreover, surges can arise from the opposite action of acceleration such as valve opening or pump starting.

DESIGN CALCULATIONS

The design calculations and assumptions used in the design of the various valves for determining weights and actuator requirements are presented for the more important components of each of the nine valve types. These calculators are representative of how the various parts were analyzed; however, for brevity, many incidental calculations are omitted. Parts common to the valves that were analyzed in the same manner (bodies, flanges, and plates) are presented in a general section.

The sample calculation for the 10 inch, 600 psi angle valve presented in Appendix C is representative of the type of analysis used for each valve.

I. GENERAL

A. Body

1. Minimum shell thickness, t: modified Barlow formula as specified by the American Standards Association B16e (1939) for flanged fittings including valves.
for diameters less than 5 inches

$$t = \frac{(p+125)D}{2F} + 0.065 \tag{6}$$

for diameters greater than 5 inches

$$t = \frac{pD}{2F} + .1 \tag{7}$$

where D = outside diameter, inches
    p = design pressure, psi
To the thickness thus determined a factor of 25% was added to compensate for shape. This calculated thickness is used throughout the body unless otherwise noted.

B. Flanges

1. Bolt number and size

a. Number of bolts, N -- same as the number of bolts in a standard ASME flange of comparable size and pressure class.

b. Assumed m.s. $= \dfrac{\text{allowable load}}{\text{design load}} - 1 = 5 \tag{8}$

c. Hydrostatic end load $P = \dfrac{\pi D_i^2 p}{4}$ (9)

d. Allowable load per bolt $= \dfrac{6p}{N}$ (10)

e. Standard AN bolt size selected   .

2. Dimensions

where D_F = flange OD

D_o = valve OD

D_i = valve ID

e = 2 x bolt diameter
c = 1/2 bolt head +
     wrench clearance +
     fillet radius

$D_F = D_{BC} + 2e$ (11)

3. Determination of flange thickness, t_f

a. Assumptions -- the following conditions
exist on the flange ring: (1) full fixation at the
bolt circle; and (2) P distributed uniformly about D_i.

b. $t_f = \sqrt{\dfrac{3P}{2\pi F_{ty}}\left[1 - \left(\dfrac{2mr_i^2 - 2r_i^2(m+1)\ln r_o/r_i}{r_o^2(m-1) + r_i^2(m+1)}\right)\right]}$

      (12)

Unless otherwise noted, this method is used for all flanges.

C. Circular Flat Plates

Case 1. Edges simply supported, uniform load over
entire surface.

a. Maximum stress at center
$$t = \frac{3pr^2}{8F_{ty}}(3+u)$$ (13)

b. Maximum deflection at center
$$\delta = \frac{2pr^4}{3Et^3}$$ (14)

Case 2. Uniform load over entire surface, edges
fixed (Note: The assumption was made that bolting as
well as integral construction produced fixity)

a. Maximum stress at edge in the radial direction:

$$t = \sqrt{\frac{3pr^2}{4F_{ty}}}$$ (15)

b. Maximum deflection at center

$$\delta = \frac{pr^4}{6Et^3}$$ (16)

D. Circular Flat Plates with a Hole at the Center

Case 3. Inner edge simply supported, uniform load over entire actual surface.

Case 4. Inner edge fixed, uniform load over entire actual surface.

Case 5. Outer edge simply supported, uniform load over entire actual surface.

Case 6. Outer edge fixed, uniform load over entire actual surface.

All of the foregoing involve rather lengthy expressions for minimum thickness and deflection determination, but are characterized by the general forms

a. $$t = \sqrt{K \frac{pr_o^4}{F_{ty}}}$$ (17)

b. $$\delta = \frac{K_1 pr_o^4}{Et^3}$$ (18)

Numerical values for K and K_1 for various values of the ratio r_o/r_i and $u = 0.3$ for the different conditions were obtained from Ref. 38 either from tables or by evaluating the appropriate expression. For Case 4, K and K_1 were evaluated using an expression to be found in Ref. 42.

II. VALVES

PIVOTED GATE (Figure 1)

A. General Assumptions -- valve overall dimensions were established by gate travel from open to closed, with the radius of rotation of the midpoint of the gate equal to the valve OD.

B. Components

1. Gates -- gate thickness was established by consideration as a circular plate uniformly loaded with edges

supported, Case 1. Maximum load conditions occurring
with valve in closed position.

2. Gate Carriage Arm

moment at section 1-1, $M = PL'$ (19)

where $P = \dfrac{\pi D_G^2}{4} \times p \times \emptyset$; D_G = gate OD   (20)

maximum stress at section 1-1

$f = \dfrac{Mc}{I}$ (21)

approximate maximum stress in ring section 2-2

$f = \dfrac{P'}{A} + \dfrac{Mc}{I}$ (22)

where $M = \dfrac{P'D}{2}$

   A = cross-sectional area of ring

  $P' = P/2$

3. Shaft

torsional moment: $\quad T = \dfrac{PL}{}$

Round section: $\quad D_S = \sqrt[3]{\dfrac{16T}{\pi F_{su}}}$ (23)

Square section: $b = \sqrt[3]{\dfrac{4.8T}{F_{su}}}$ (24)

where b = length of side of square

4. Body Sections: The lateral flange on the body
is assumed to divide it into two distinct parts. A strip
section is taken at the center of the bonnet and assumed
to act as a beam uniformly loaded and fixed at both ends.
The maximum moment is obtained by the expression

$M = \dfrac{pL^2}{12}$ (25)

where L = unsupported length of the strip. Ribs were
added to gain a more economical distribution of material.
The strip then has a "T" cross section and is designed to
resist the maximum moment

$\dfrac{I}{Y} = \dfrac{M}{F_{ty}} = Z$ (26)

The ribs were spaced alternately with bolts in the lateral flange.

C. Maximum Operating Torque -- Maximum resistance is experienced at the point of breakout or initial opening of the valve with full pressure drop across the gate. The coefficient of static friction ($\emptyset$) is assumed to have a value of .25. The torque may be expressed:

$$T = PL \tag{27}$$

where P and L have the same values as in item B-2.

D. Actuator

1. With the torque known and selection of suitable lever arm consistent with other valve dimensions, the thrust for the actuator can be determined

$$P' = \frac{PL}{L_A} \tag{28}$$

2. Piston area: (assuming 3000 psi operating pressure)

$$A_p = 1.1(\frac{P'}{3000})(10\% \text{ allowance for friction}) \tag{29}$$

3. Piston diameter: $D_p = \sqrt{\frac{4A}{\pi}}$ \hfill (30)

4. Cylinder wall thickness

$$t_c = \frac{P_B D_p}{2F_{ty}} \quad \text{where } P_B = \text{burst pressure} \tag{31}$$

5. Piston rod cross-sectional area

$$P = \frac{F_{ty}A}{1+(\frac{L}{e})^2 \frac{F_{ty}}{\pi^2 CE}} \tag{32}$$

6. Cylinder closures -- Assume as circular plates fixed and supported at edges both with and without a hole in the center.

Y-PATTERN GLOBE (Figure 2)

A. General Assumptions
    Poppet lift -- 50% of valve OD
    Poppet diameter -- 10% greater than valve OD
    Piston area -- 25% greater than area of poppet

B. Components

1. Poppet -- analyzed as a circular plate with a hole,

Case 3. Maximum stress occurs at inner edge just as pop-
pet leaves seat against full line pressure. Thickness
determined using equation (17) and deflection checked
using equation (18).

   2. Piston -- analyzed as a circular plate with a hole,
Case 2. Thickness determined using equation (15).

   3. Stem -- analyzed as a short column with $L/e \leq 100$.
Maximum stress occurs when hydraulic force on piston
balances pressure load on poppet.

      a. Diameter determined from equation (32).

   4. Cylinder Cover Plate -- analyzed as a uniformly
loaded circular plate with a hole, Case 4. Same number
of bolts used as for a comparable flange and bolt size
determined from pressure load with a m.s. of 2.

      a. Cover Plate OD = bolt circle diameter +
2 x 15 (bolt diameter)

      b. Thickness determined using equation (17).

   5. Spring

      a. Assumptions:
         mean coil diameter, D_{mc} = 2.75 inches
         500 lb. spring force in the open position
         50 lb. spring force in the closed position
         material -- chrome silicon alloy steel
         design stress -- F_s = 130,000 psi

      b. Wire diameter $D_w = \sqrt[3]{\dfrac{2.55 \times P \times D_{mc}}{F_s}}$ (33)

      c. Number of coils $N = \dfrac{\delta \times G \times D_w^4}{8 \times P \times D_{mc}^3}$ (34)

SLIDING GATE VALVE (Figure 3)

   A. General -- Line flange dimensions were determined by
the method described in Section I-B with the exception of thick-
ness. The total flange thickness or dimension "A" was held con-
sistent with the height of the curved portion of the valve.
This radius of curvature was assumed equal in all cases, to the
line diameter. The chord length of the arc was numerically equal
to the slide width.

B. Components

1. Slide -- Maximum loading condition occurs with slide in closed position with full pressure differential across it. Although not a circular plate, it is loaded over a circular area and is closely approximated by using the expressions for a circular plate uniformly loaded and simply supported, Case 1.

2. Crank -- Analyzed as a cantilever beam with maximum moment expressed by equation (19) and minimum section determined from equation (26).

3. Body Section Thickness -- A section thickness was determined for the curved areas by consideration of only the tensile or so called membrane stresses by:

$$t = \frac{pr}{F_{ty}} \tag{35}$$

where r = radius of curvature.
This thickness was maintained for the body flanges. The method is strictly an approximation as bending stresses occurring at the change in curvature at the flange are disregarded and assumed to be compensated for by an increase in section thickness through the addition of a generous fillet.

4. Shaft -- The shaft was proportioned through the use of expression (23).

C. Maximum Operating Torque -- The torque on this valve is measured as the product of gate resisting force times and effective length of the crank. Conservatively, a crank rotation of 30° is chosen as the maximum point. Since full pressure drop still exists across the slide (further rotation would increase the lever arm but decrease the pressure drop and the slide resistance), the conservatism arises from the choice of the coefficient of friction as being that at breakout. The resisting force can be expressed by equation (20). The length of effective lever arm:

$$L_E = L_c \sin\theta \tag{36}$$

where θ is the angle of rotation.

D. Actuator -- Analyzed the same as the vane type actuator for the butterfly valve with the exception of top and bottom plates. These plates were approximated by considering them as rectangular flat plates with fixed edges

$$M_{max} = BpL^2 \tag{37}$$

where B is a numerical factor depending upon the ratio of the sides of the rectangle.

POPPET VALVE (Figure 5)

A. General Assumptions -- The flow passage area is constant throughout the valve. The piston area is 25% greater than the area of the poppet exposed to line pressure.

B. Components

1. Body -- Analyzed as a thick walled cylinder. The Barlow equation (7) was used in determining the wall thickness without the 25% shape factor.

2. Inner Cylinders -- Analyzed as cylinders subjected to external pressure.

a. Thickness

$$p = \frac{t}{F} \times \frac{F_{cy}}{1+4\ \frac{F_{cy} \times r^2}{E \times t^2}} \tag{38}$$

3. Inner Cones -- Analyzed as cylinders subjected to external pressure.

4. Pistons -- Design determined by sealing requirements.

C. Actuating Force -- Although designed for line pressure actuation, external hydraulic or pneumatic means could be used. The maximum force occurs in the closed position with full line pressure drop across the poppet.

$$P = \frac{\pi}{4}\ (D_1^2 - D_2^2) \times p \tag{39}$$

BUTTERFLY VALVE (Figure 6)

A. General Assumptions -- The offset split shaft arrangement was patterned after a design developed by North American for an oxygen fuel shutoff valve. The offset shaft minimizes the problem of shaft sealing and simplifies the sealing of the vane.

B. Components

1. Vane -- Analyzed as a circular flat plate uniformly loaded and supported at two points on periphery. Maximum stress occurs at supports with vane in the closed position. An allowable deflection was assumed at outer edge of vane for efficient sealing.

a. Thickness $t = \sqrt[3]{\dfrac{0.371\ pr_o^4(12)(1-\mu^2)}{E\delta}}$ (40)

Maximum stress checked for this thickness.

2. Shaft -- Analyzed as a cantilever beam with one end fixed in the bearing and supporting half the vane load.

a. Diameter $D = \sqrt[3]{\dfrac{M}{.0982 \times F_{ty}}}$ (41)

b. Shear Stress -- check shear stress in shaft due to torque $f_s = \dfrac{Tc}{J}$ (42)

C. Actuating Torque -- Sum of the aerodynamic and bearing friction torques (plus an additional 25%). Maximum aerodynamic torque assumed to occur at a vane angle of 70° from the closed position.

1. Aerodynamic Torque $T_A = C_t D^3 \Delta p$ (43)

A flow velocity of 60 fps was assumed for determining Δp. C_t for a vane angle of 70° = 0.2

$$W = 75.5 \, C_t D^2 \sqrt{W \Delta p}$$ (44)

2. Bearing friction torque $T_{Bf} = P x r_b x \emptyset$
A coefficient of running friction equal to 0.1 was used.

D. Actuator (vane type) -- A mechanical efficiency of 60% was assumed for this type of actuator. A design torque of twice the operating torque and a hydraulic design pressure of 2-1/2 times the operating pressure was used in design.

1. Vane size $T = \displaystyle\int_{r_1}^{r_2} pbdLL$ (45)

2. Vane thickness -- analyzed as a cantilever beam uniformly loaded

$$t = \sqrt{\dfrac{M}{\dfrac{bF_{ty}}{6}}} = \sqrt{\dfrac{3pL}{bF_{ty}}}$$ (46)

3. Body -- The body and cover plates analyzed as plates with fixed edge conditions.

a. Side Plates -- analyzed as flat rectangular plates. The maximum bending moment occurs at the middle of the longer sides and is determined from equation (37).

$$t = \dfrac{m \times 6}{L \times F}$$ (47)

b. Bottom and Cover Plate -- analyzed as a triangular plate with fixed edge conditions. The maximum bending moment is somewhat less than half that for a square plate.

DIAPHRAGM VALVE (Figure 7)

A. General -- An assumed proportioning of the diaphragm established a maximum diameter through the valve. With this diameter and the Barlow equation a maximum body thickness was determined.

B. Components

1. Flange -- Flange dimensions determined by the method described in Appendix B-1.

2. Poppet -- Assumed that the poppet takes all of the load with the valve in the closed position. Analyzed as a circular plate uniformly loaded and simply supported at the edges, Case 1.

3. Cover Plate -- Considered as a circular plate with clamped edges, Case 2.

4. Diaphragm -- Little is known about the analytical approach to sizing this member. Experimental development is used by manufacturers. The diaphragm is subjected to pressure on both surfaces at all times with the maximum differential being at the instant the valve is to be opened. Since experimental development would be required on this member, a thickness is approximated within practical limits for flexibility and strength. This member could very well establish the maximum practical size for this type valve.

ECCENTRIC PLUG VALVE (Figure 8)

B. Components

1. Valve Body -- With ribs as shown in Fig. 8, central horizontal strip analyzed as uniformly loaded T-beam with fixed ends, with a span length equal to one-half the horizontal distance, flange to flange.

2. Cover Plate -- analyzed as a uniformly loaded circular plate with a hole in the center, clamped at the outside edge, Case 6.

3. **Valve Barrel** -- Check assumed thickness, considering as thin-walled circular cylinder, clamped at ends and subjected to internal pressure.

$$f_{max} = \frac{6\left[\frac{p}{2\beta^2} X_2(2\alpha)\right]}{t^2} + \frac{pr_i}{2t} \tag{48}$$

$$\text{where } \beta = \left[3\frac{(1-\mu^2)}{r_i^2 t^2}\right]^{1/4} \tag{49}$$

$$2\alpha = \beta L$$

$X_2(2\alpha)$ = trigonometric function of 2α

4. **Bottom of Valve Barrel** -- Analyze as a uniformly loaded circular plate, clamped at the edges, Case 2.

C. **Maximum Operating Torque** -- Maximum resistance is experienced at instant of valve opening with full pressure drop across plug. Coefficient of static friction ($\emptyset$) is assumed to have a value of 0.25. Torque may be expressed: $T = PL\emptyset$ (50)
where $P = p \times$ (area of port opening + area of seating surface)
L = radius, plug shaft center to seating surface

1. **Plug shaft** -- Design criteria are bending and torsion; considered as acting independently.

a. **Bending** -- Consider plug and shaft as uniformly loaded beam with edges fixed at bearing edges

$$f = \frac{32\left(\frac{PL}{12}\right)}{\pi D^3} \tag{51}$$

where L = distance, edge to edge, of shaft bearing surfaces
$P = p \times$ projected area of plug body and ends
D = shaft diameter

b. **Torsion** $f = \frac{16(PL\emptyset)}{\pi D^3}$ (52)

2. **Plug Body Ends** -- Design criteria: bending with bending moment same as PL/12 in (a) above.

3. **Plug Body** -- Design criteria: assumed as uniformly loaded beam with fixed ends

$$f = \frac{6}{t^2}\left[\left(\frac{PL}{12}\right)\right] \tag{53}$$

where t = plug body thickness

      P = p x projected area of plug body

      L = length of plug body

Deflection δ (max. at center)

$$\delta_{max} = \frac{PL^3}{384EI}$$ (54)

where P and L are same as in 3 above.

   D.   Actuator -- Assumed a torque efficiency of 80%

Design Torque = $\frac{PL\phi}{0.8}$, where PLø as in C above   (55)

Piston, piston rod, cylinder, wall, and cylinder closures designed as in D-2, 3, 4, 5, and 6, Pivoted Gate Valve. Connection between piston rod and crank arm -- pin diameter assumed and checked for double shear. Crank arm considered as twin-cantilever beam with concentrated force acting at end. Pivot point bracket designed as shown in Fig. 8. Checked for bending at critical section (double T-beam).

   CONE VALVE (Figure 9)

   A.   General Assumptions

      Slot area = pipe area

      Cone height assumed equal to 1.865 base diameter
         (valve ID)

      Assume 8 equal slots, each of 15° arc in width, equally
         spaced around conical surface; seating surface is
         this $7\text{-}1/2^o$ of arc.

      Assume minimum seating width = 1/8 inch.

   B.   Components

      1.   Inner cone bars (solid sections between slots) -- Most critical of several conditions checked occurs when slot in outer cone is just covered. Condition approximates that of a uniformly loaded rectangular plate with two opposite edges simply supported, a third edge free, and the fourth edge clamped.

$$f = \frac{6(\beta p L^2)}{F_{ty}}$$ (56)

where L = length of longer side

     β = factor dependent on a/b

     ℓ = length of shorter side

Since plate is actually tapered, maximum value of β for any condition was selected, to be on conservative side.

Check for deflection $\quad \delta_{max} = \dfrac{2pL^4\gamma}{\pi^5}$ (57)

where δ = deflection

γ = factor dependent on b/a

For same reason as above, maximum value of γ was used.

2. Outer cone bars -- analyzed by considering as a uniformly loaded, simply supported trapezoidal beam (valve is completely closed).

$$f = 6(0.078 \ bpL^2)/F_{ty}$$ (58)

where b = maximum width of beam

L = beam span

In order to avoid excessive weight, a rib was added, making in effect a "T"-beam. Deflection was checked by

$$\delta = \dfrac{-5PL^3}{384EI}$$ (59)

where P = total load on beam

L = beam span

3. Vane Chamber Endwalls -- Can roughly approximate as containing an elliptical hole subjected to internal pressure. With outer boundaries at an infinite distance,

$$f_{max} = p \ (\frac{2a}{b} - 1) \ \left[\frac{d^2 + (a/2)^2}{d^2 - (a/2)^2} \right]$$ (60)

with a, b, and d as shown in sketch.

4. Vane Chamber outerwall -- analyze as a short fixed-ended cylinder subjected to internal pressure. Check for tangential stress by Barlow equation.

5. Limit Stops (statronam vanes) -- consider just as uniformly loaded fixed ended beam.

$$f_{max} = \frac{6\left(\frac{pL^2}{12}\right)}{t^2}$$ (61)

where t = thickness of stop
L = span

Consider also as uniformly loaded rectangular plate simply supported on two opposite sides, fixed on the third and free on the fourth

$$f_{max} = \frac{6\left(\beta pL^2\right)}{t^2}$$ (62)

where L = span of plate (direction normal to "a")
β = factor dependent on b/a

The first consideration gives the thickness at the free edge; the second, that at the clamped edge opposite to the free edge.

C. Maximum Operating Torque -- Maximum resistance is experienced at instant of valve opening with full pressure drop across cone. Coefficient of static friction (∅) is assumed to have a value of 0.25. Torque may be expressed:

$$T = \frac{\pi}{2} \frac{\emptyset p}{\sin\alpha} (r_0+t)^3 + 2\pi(r_0+t)^2 pL\emptyset$$ (63)

where α = one-half the cone angle
r_0 = radius of base of inner cone
t = thickness of inner cone
L = length of cylindrical portion of inner cone.

SAMPLE DESIGN CALCULATION -- 10 INCH, 600 PSI ANGLE VALVE

Design Conditions

    600 psi operating pressure
    1200 psi design pressure
    450°F design temperature

Design Calculations

I. Body

    Material: 356-T6 cast aluminum alloy
    Design stress: F_{ty} at 450°F = 7500 psi

    A. Determine minimum body thickness

$$t = \frac{pD_o}{2F_{ty}} + 0.1 \text{ (modified Barlow equation)} \qquad (64)$$

where p = design pressure, psi
    D_o = actual outside diameter, inches

$$t = \frac{1200 \times 10}{2 \times 7500} + 0.1 = 0.9 \text{ inches}$$

Add a 25% shape factor 1.25t = 1.125 inches
This calculated thickness is used throughout the body for weight estimations.

II. Flanges

    Material: 356-T6 cast aluminum alloy

    A. Determine flange thickness t_f

$$t_f = \sqrt{\frac{3P}{2\pi F_{ty}} \left[1 - \left(\frac{2mr_i^2 - 2r_i^2(m+1)\ln\frac{r_o}{r_i}}{r_o^2(m-1) + r_i^2(m+1)} \right) \right]} \qquad (65)$$

where P = hydrostatic end load, $\pi/4\, D_i^2 P$, pounds
    r_i = inside flange ring radius, inches
    r_o = bolt circle radius, inches

$$t_f = \sqrt{\frac{3 \times \pi \times (7.75)^2}{2 \times \pi \times 4 \times 7500} \left[1 - \left(\frac{2 \times 3.33 \times 15 - 2 \times 15 \times 4.33 \times 0.482}{39.7 \times 2.33 + 15 \times 4.33} \right) \right]}$$

        = 1.66 inches

    B. Determine size of bolts
        Assume 16 bolts
        Pressure force = $\pi/4\, (7.75)^2 \times 1200$ = 56,600 lbs.

$$\frac{56,600}{16} = 3540 \text{ lbs/bolt}$$

A safety factor of 6 is used for line flanges
6 x 3540 = 21,240 lbs/bolt
Use 16 - 9/16 AN bolts (23,600 lbs/bolt)

C.   Determine flange dimensions
Assume bolt edge distance (e.d.) = 1.5 x bolt diameter
       = 1.5 x 0.56 = 0.84
Flange OD = 10 + 2(e.d. + 1/2 bolt head + wrench clearance +
       fillet radius)
       = 10 + 2(0.84 + 0.46 + 0.25 + 0.56) = 14.22 inches

III.   Poppet

   Material: S.A.E. 4340 steel
   Design Stress: F_{ty} at 450°F = 165,000 psi
   Poppet diameter -- The poppet will be beveled at a 45°
angle to rest on the seat (Fig. 3). Minimum poppet diameter has
been assumed the same as the valve ID. The maximum poppet diameter
then becomes:
$$D_{max} = \text{Valve ID} + 6.5\% \text{ Valve ID}$$
$$= 1.065 (7.75) = 8.25 \text{ inches}$$

   A.   Determine thickness -- analyze as a uniformly loaded
circular plate with a hole in the center, simply supported at
inner radius r_i
$$t^2 = \frac{Kpr_o^2}{F_{ty}} \tag{66}$$

   t = thickness, inches
   p = uniform load, psi
   r_o = outer radius, inches
   K = factor depending on r_o/r_i and Poisson's ratio
Assume for the moment $r_i = 0.62$
Then $r_o/r_i = 4.1/0.62$ and K = 5.9
$$t^2 = \frac{5.9 \times 1200 \times (4.1)^2}{165,000} = 0.72$$

   t = 0.85 inches

   B.   Check deflection in closed position (seated) -- Can be
conservatively analyzed as a uniformly loaded circular plate
with a hole supported at the outer radius, r_o.

$$\delta_{max} = \frac{K_1 pr_o^4}{Et^3} \tag{67}$$

   δ_{max} = maximum deflection (at r_i)

   E = modulus of elasticity, psi

$$t = \text{poppet thickness, inches}$$
$$K_1 = \text{factor depending on } r_o/r_i \text{ and Poisson's ratio}$$

for $r_o/r_i = 4.1/0.62$, $K_1 = 0.80$

$$\delta_{max} = \frac{0.8 \times 1200 \times (4.1)^4}{29 \times 10^6 \times (0.85)^3} = 0.0153 \text{ inches}$$

This deflection should not prevent good seating.

IV. Piston

Material: 4340 steel

A. Determine diameter, d

Pressure force on poppet = $\pi/4 \ (8.25)^2 \times 1200 = 64.200$ lbs. Assume a 10% friction force due to packing. Now assume use of 3000 psi hydraulic pressure.

$$1.1 \times 64,200 = 3000 \times \frac{\pi}{4} \ (d^2 - (1.24)^2)$$
$$d^2 = 31.44$$
$$d = 5.61 \text{ inches -- use 5.7 inches}$$

B. Determine thickness, t

Analyze same as poppet -- $r_o/r_i = 2.85/0.62 = 4.6$; $K = 4.75$ Assume 2.5 times the working pressure as the design pressure for hydraulic actuators

$$2.5 \times 3000 \text{ psi} = 7500 \text{ psi}$$
$$t^2 = \frac{4.75 \times 7500 \times (2.85)^2}{165,000} = 1.76$$
$$t = 1.33 \text{ inches}$$

V. Cover Plate

Material: 4340 steel
Design Stress: F_{ty} at $450^\circ F = 165,000$ psi

A. Determine size of bolts and bolt radius R

Pressure force on plate (conservative) = $\pi/4 \times (5.7)^2 \times 1200$
= 30,700

30,700/12 = 2560 lbs/bolt

Use 12 - 3/8 AN bolt
$$\text{m.s.} = \frac{10,100}{2660} - 1 = 2.95$$
$$R = 2.85 + 0.25 + 0.375 + 0.25 + 0.19$$
$$= 3.96$$

Use R = 4.0 inches

B.   Determine cover plate OD

OD = 2R+2(1.5d) = 2 x 4.0 + 2(1.5 x 0.375) = 9.125

C.   Determine thickness, t

Analyze as a uniformly loaded circular plate with a hole in the center, fixed at the outer radius r_o

$r_o/r_i = 4.0/0.62 = 6.45$; K = 1.4

$$t^2 = \frac{Kpr_o^2}{F_{ty}} = \frac{1.4 \times (7500-1200) \times (4.0)^2}{165,000} = 0.855 \qquad (68)$$

t = 0.925 inches   Use 0.95 inches

VI.   Cylinder Housing

Material: 4340 steel
Design Stress: F_{ty} at $450^{\circ}F$ = 165,000 psi

A.   Determine Wall thickness, t

Assume OD = 6 inches

$$t = \frac{pD_o}{2F_{ty}} = \frac{7500 \times 6}{2 \times 165,000} = 0.136 \text{ inches} \qquad (69)$$

Use t = 0.15 inches
Actual OD = 5.7 + 2(0.15) = 6.0 inches

B.   Determine height

Height = bottom clearance + piston thickness + lift + shaft offset thickness + clearance on top.
         = 1 + 1.33 + 5 + 0.45 + 0.12 = 7.90 inches
Note: all values except piston thickness assumed

VII.   Cylinder Cover Plate
Material: 4340 steel

A.   Determine size of bolts and bolt radius, R

Cover plate is sized for actual 3000 psi hydraulic pressure.
Pressure force = $\pi/4$ $(5.7)^2$ x 3000 = 76,600 lbs.
76,600/16 = 4780 lbs/bolt
Use 16 - 5/16 AN bolts   m.s. = $\frac{6500}{4780}$ -1 = 0.36

R = 2.85 +0.15 +0.12 +0.31 +0.12 +0.15 = 3.70 inches

B.   Determining cover plate OD

OD = 2 x 3.7 + 2(1.5 x 0.31) = 8.33 inches

C.   Determine thickness, t

Analyze as a uniformly loaded circular plate fixed at outer radius, r_o.

$$t^2 = \frac{3}{4} \frac{pr_o^2}{F_{ty}} = \frac{3}{4} \times \frac{3000 \times (3.7)^2}{165,000} = 0.187 \qquad (70)$$

$t = 0.433$ inches   Use 0.45 inches

Note: No hydraulic fluid above piston -- allowance of 3000 psi for leakage.

VIII. Stem

Material: 4340 steel
Maximum load occurs with valve fully opened
Total load on piston $= 7500 \times \pi/4 \left[(5.7)^2 - (1.24)^2 \right] =$
$$= 182,500 \text{ lbs.}$$

A. Determine stem diameter, d -- assuming $d = 1.24$ inches
Tension on stem $= P/A = 182,500 / \frac{\pi}{4} (1.24)^2 = 150,500$ psi
Since $150,500$ psi $< 165,000$ psi, use diameter $= 1.24$ inches.

B. Determine stem offset thickness, t
Design Stress: F_{su} at $450^\circ F = 105,000$ psi

$\pi Dt \times 105,000$ psi $= 182,500$ lbs.

$$t = \frac{182,500}{\pi \times 1.24 \times 105,000} = 0.445 \text{ inches}$$

IX. Stem Guide and Stop

Material: 4340 steel

A. Determine wall thickness, t
Analyze as a compression member

$$\pi/4 \, (OD^2 - \text{stem diameter}^2) \times F_{cy} = 182,500 \text{ lbs.}$$

$$OD^2 = 4/\pi \times \frac{182,500}{165,000} + (1.24)^2 = 2.945$$

$OD = 1.72$ inches

$$t = \frac{1.72 - 1.24}{2} = 0.24 \text{ inches} \qquad \text{Use 3/8 inch to}$$

give adequate bearing and support for stem.

X. Spring

Material: chrome silicon - alloy steel
Design Stress: $F_{sy} = 130,000$ psi
Assumptions: mean coil diameter, $D_{mc} = 2.5$ inches

500 lbs. spring force in the open position and
50 lbs. force in the closed position

A. Determine wire diameter, D_w

$$D_w^3 = \frac{2.55P \times D_{mc}}{F_{sy}} = \frac{2.55 \times 500 \times 2.5}{130,000} = 0.0245 \qquad (71)$$

$D_w = 0.29$ inch

Nearest standard gauge -- 0.307 inch

B. Determine number of coils, N

$$N = \frac{\delta G D_w^4}{8PD_{mc}^3} = \frac{5.55 \times 11,500,000 \times (0.307)^4}{8 \times 500 \times (2.5)^3} = 9.08 \qquad (72)$$

Spring index $= \dfrac{500-50}{5} = 90$ lbs/inch

$\delta = \dfrac{500}{90} = 5.55$ inch total deflection

Use 10 active coils
Assume one dead coil on each end, giving a total of
12 coils.

Weight

Based on the design calculations presented, the weight
of the component parts contributing the major portion of the
weight for the 10 inch, 600 psi valve are:

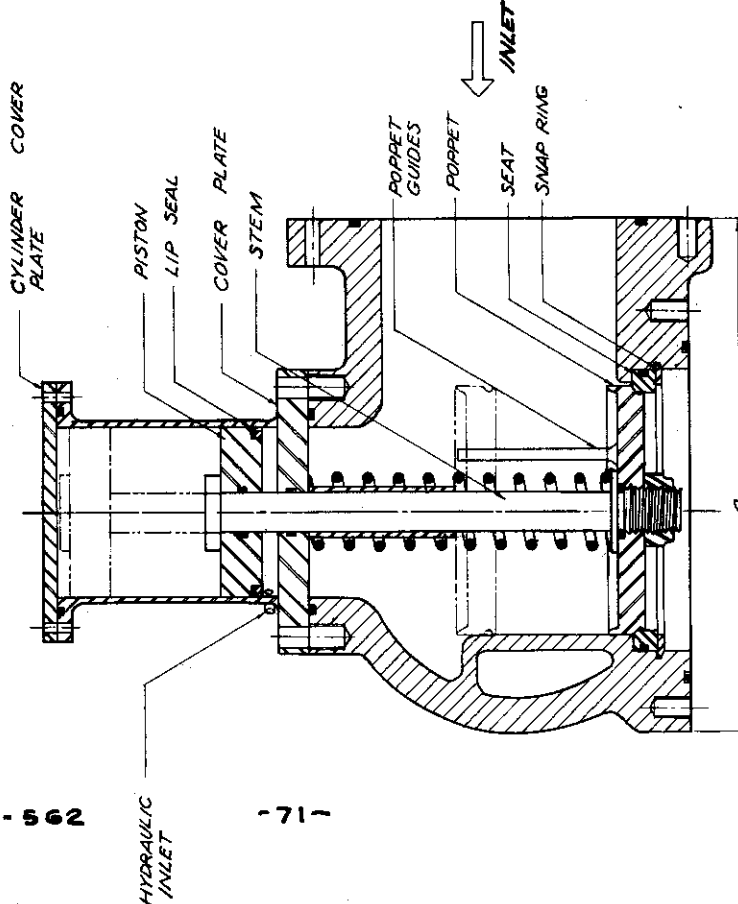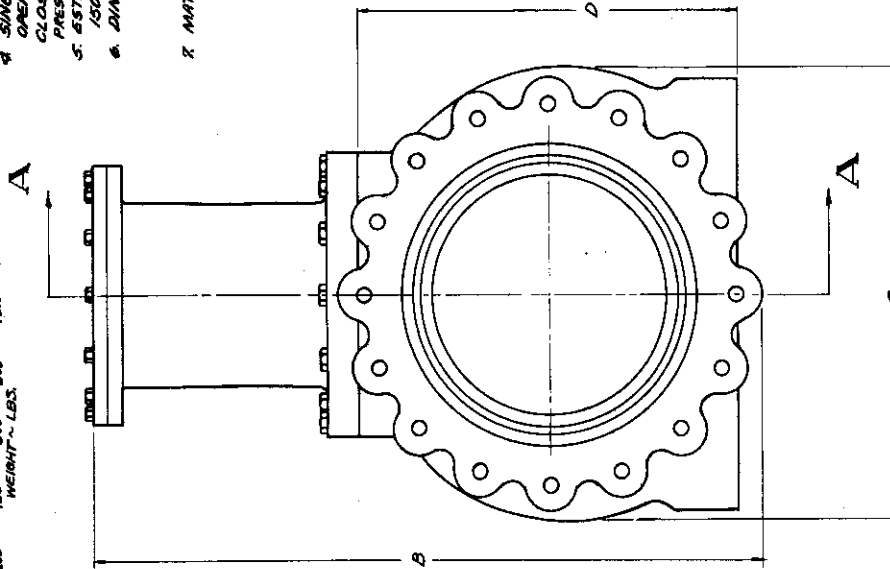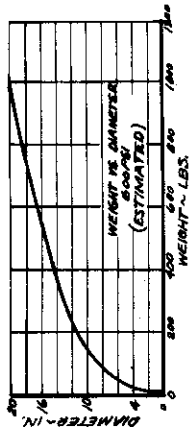|	
Body and flanges	72.98 lbs.
Poppet	13.04 lbs.
Seat	3.73 lbs.
Cover Plate	22.00 lbs.
Piston	11.67 lbs.
Actuator	16.40 lbs.
Stem	5.30 lbs.
Spring & Stem Guide	3.81 lbs.
	148.93 lbs.

NOTES:

1. PIVOTED GATE TYPE SHUT-OFF VALVE.
2. LINE SIZE 10"QA. PORT DIAMETER 175"
3. OPERATING PRESSURE 600 PSI DESIGN PRESSURE 1200 PSI
4. DOUBLE ACTING PISTON HYDRAULIC ACTUATOR — HYDRAULIC OPERATING PRESSURE 3000 PSI
5. ESTIMATED WEIGHT WITH ACTUATOR 145 LBS.
6. DIMENSIONS:

VALVE DIA	A	B	C	D
10"	12	14½		

7. MATERIAL KEY: ▨ ALUM ▧ STEEL

FLUI DYNE ENGR. CORP.
MINNEAPOLIS, MINNESOTA

MOUNT

A
D
C
SHAFT

A
B
A

DOUBLE ACTING PISTON ACTUATOR

CARRIAGE

SEAT

TWO PART FLEXIBLE GATE

SECTION A-A

WEIGHT VS DIAMETER 600 PSI VALVES (ESTIMATED)

DIAMETER—IN.
WEIGHT—LBS.

FIGURE I PIVOTED GATE VALVE

WADC TR 54-562

-68-
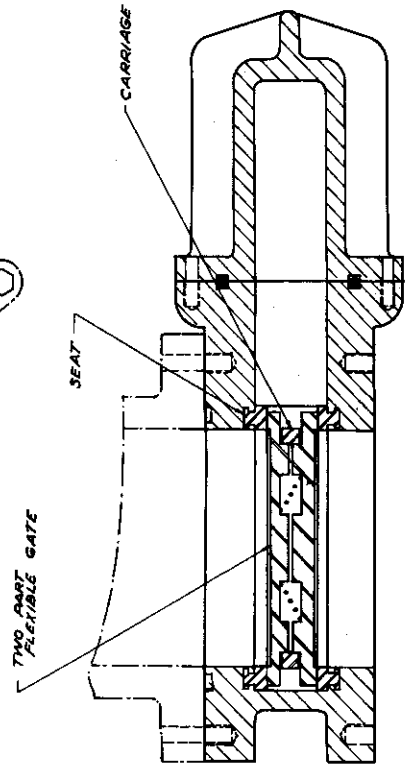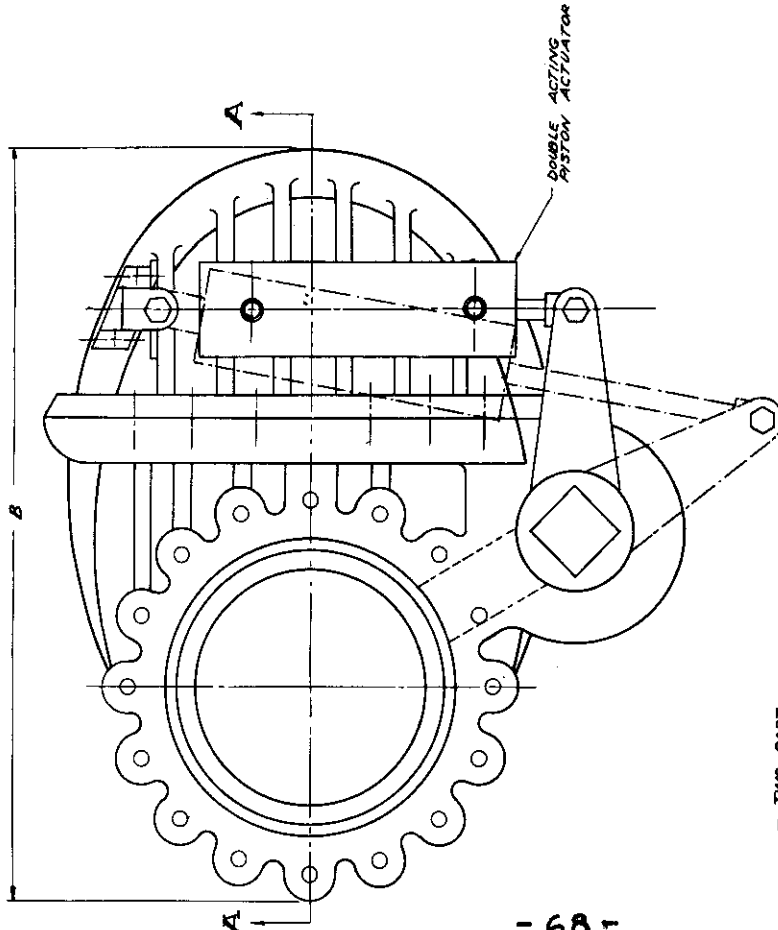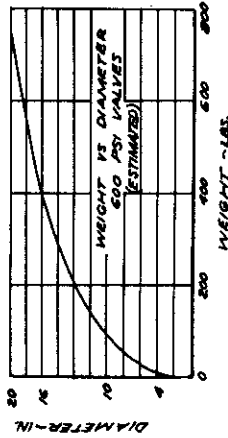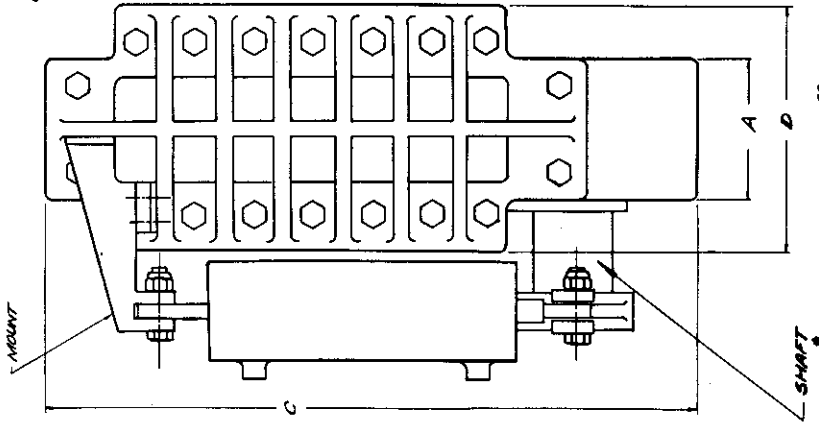
NOTES:

1. "Y" PATTERN GLOBE TYPE SHUT-OFF VALVE.
2. LINE SIZE 10" O.D. PART DIA. 7.75
3. OPERATING PRESSURE 600 PSI
   DESIGN PRESSURE 1200 PSI
4. LINE PRESSURE ACTUATED
5. ESTIMATED WEIGHT 213 LBS.
6. DIMENSIONS: VALVE DIA. 14.81C
7. MATERIAL KEY: ALUM STEEL

WEIGHT VS. DIA.
600 PSI
(ESTIMATED)

SOLENOID OPERATED PILOT CONTROL VALVE

ORIFICE

PISTON

SEAT RING

POPPET

SECTION AA
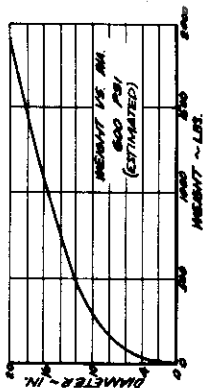
INLET

FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA
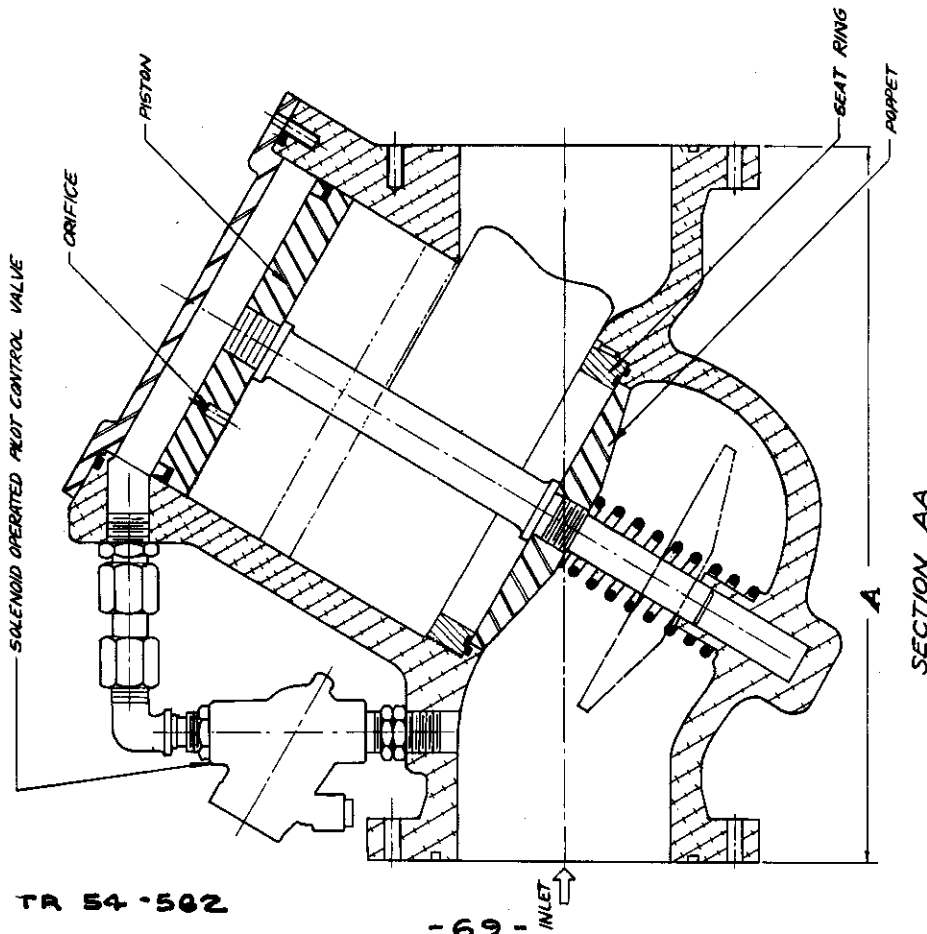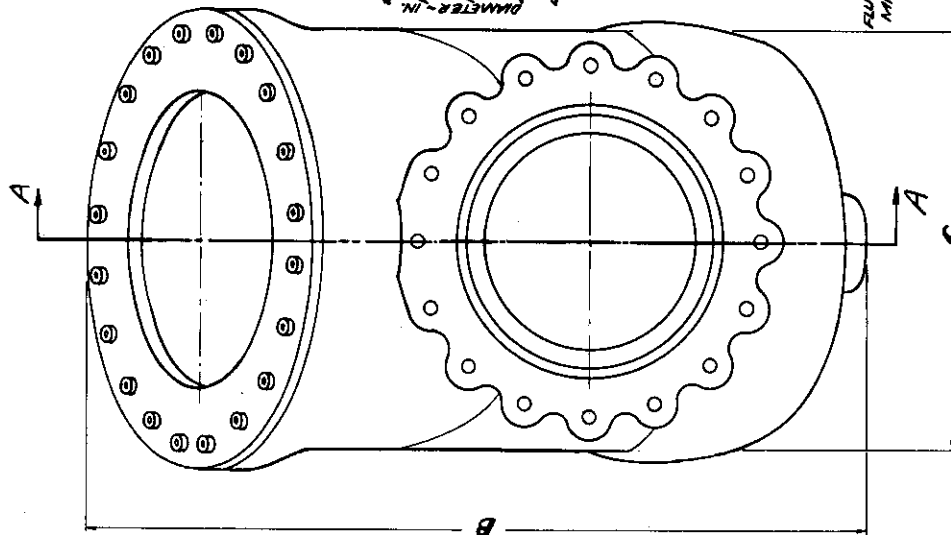
FIGURE 2 GLOBE VALVE

WADC TR 54-582

-69-
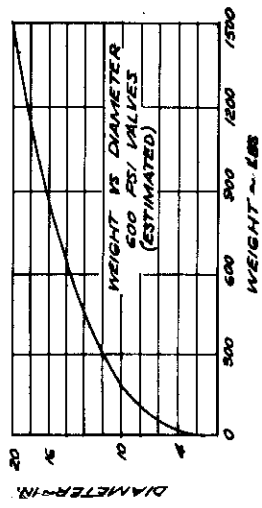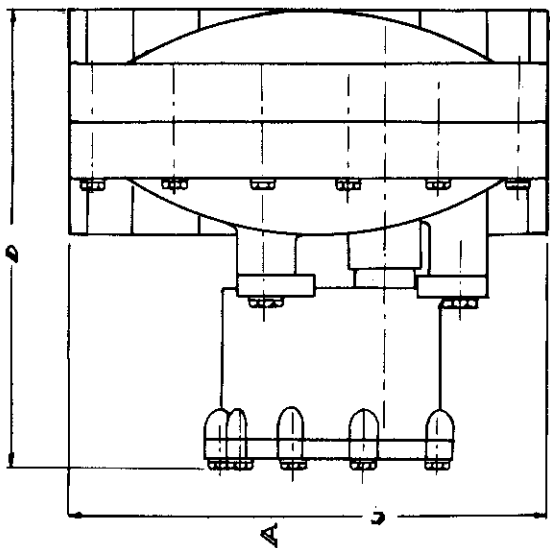
NOTES:

1. SLIDING GATE TYPE SHUTOFF VALVE
2. LINE SIZE : 10" O.D. 7.75 PORT DIA.
3. OPERATING PRESSURE 600 PSI
   DESIGN PRESSURE 1200 PSI
4. HYDRAULIC VANE MOTOR
   ACTUATOR - HYDRAULIC OPERATING
   PRESSURE - 3000 PSI
5. ESTIMATED WEIGHT WITH ACTUATOR
   200 POUNDS
6. DIMENSIONS

VALVE DIA 10"	A	B	C	D
	6.5	30.8	14.2	13.4

7. MATERIAL KEY : ▨▨▨ ALUMINUM ▥▥▥ STEEL

FLUI DYNE ENGR CORP.
MINNEAPOLIS , MINNESOTA

CRANK
HYDRAULIC VALVE MOTOR

SCOTCH YOKE MECHANISM

SLIDE

SLIDE SEAL
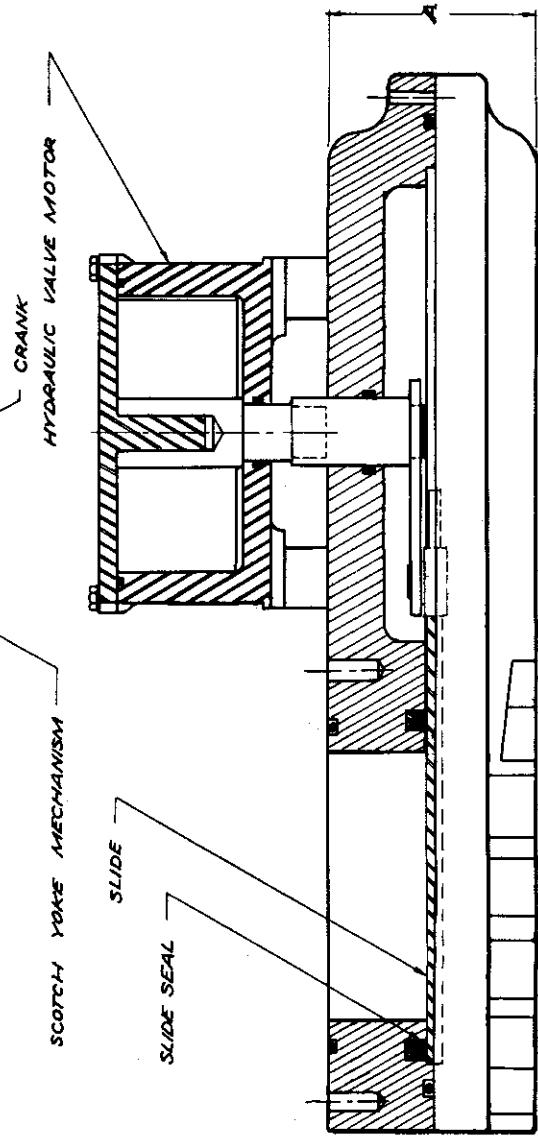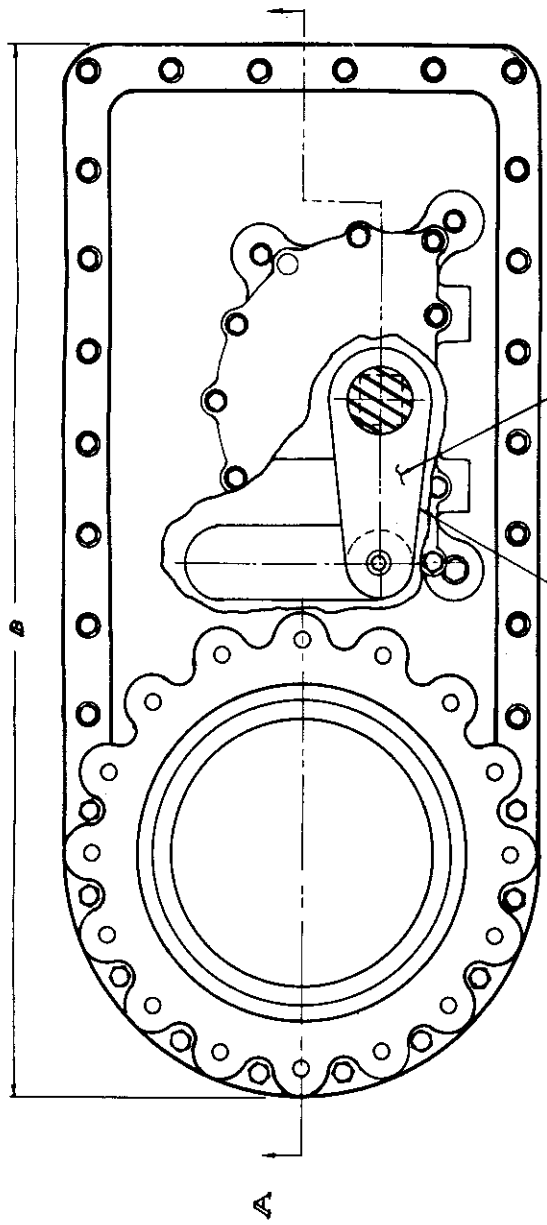
SECTION A A

FIGURE 3   SLIDING GATE VALVE

NOTES:

1. ANGLE TYPE SHUT-OFF VALVE
2. LINE SIZE-10" OD, PORT DIAMETER 7.5"
3. OPERATING PRESSURE 600 PSI.
   DESIGN PRESSURE 1200 PSI.
4. SINGLE ACTING PISTON ACTUATOR TO
   OPEN, SPRING AND LINE PRESSURE TO
   CLOSE; HYDRAULIC OPERATING
   PRESSURE 3000 PSI
5. ESTIMATED WEIGHT WITH ACTUATOR
   150 LBS.
6. DIMENSIONS

X MATERIAL KEY

VALVE DIA	A	B	C	D
10"	X MATERIALS

ALUM   STEEL

FLUIDYNE ENG. CORP
MINNEAPOLIS, MINNESOTA

WEIGHT VS DIAMETER
600 PSI
(ESTIMATED)

DIAMETER - IN.

WEIGHT ~ LBS.

INLET

CYLINDER COVER PLATE
PISTON
LIP SEAL
COVER PLATE
STEM
POPPET GUIDES
POPPET
SEAT
SNAP RING

HYDRAULIC INLET

SECTION A—A

FIGURE 4   ANGLE VALVE

FIGURE 5 POPPET VALVE
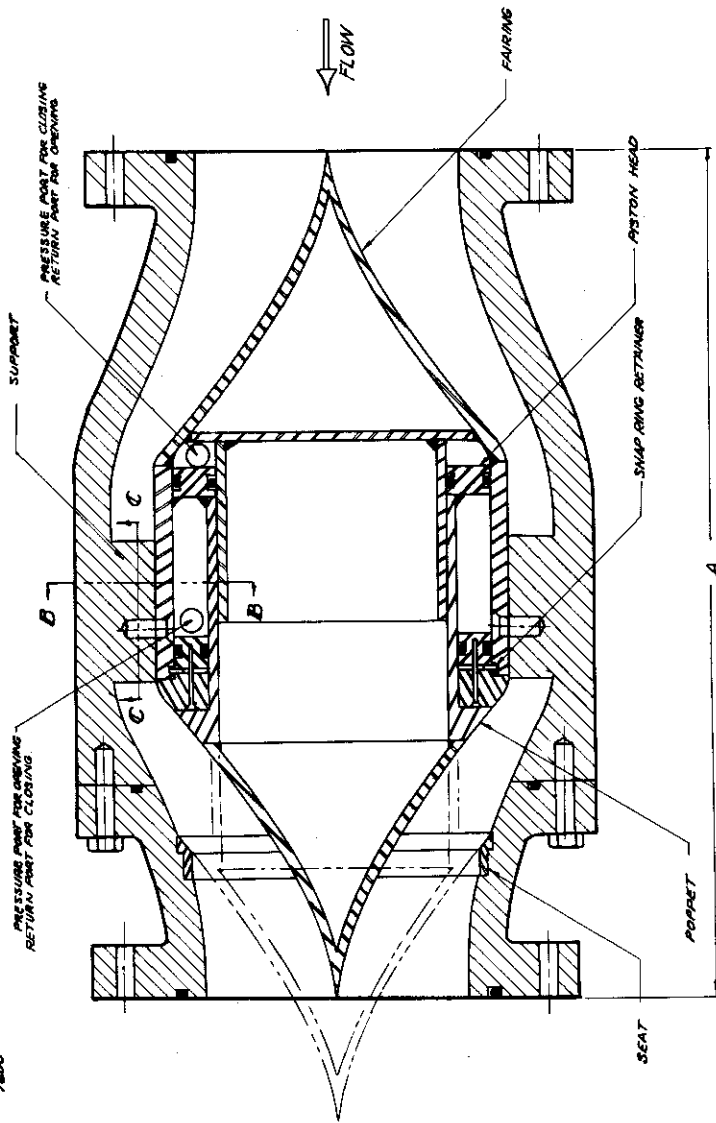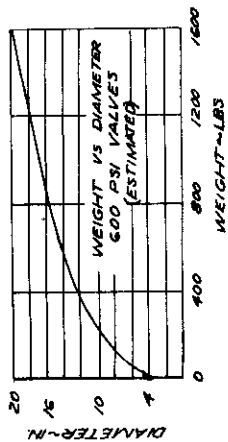
FLUI DYNE ENGINEERING CORP.
MINNEAPOLIS, MINNESOTA

NOTES:

1. POPPET TYPE SHUT-OFF VALVE

2. NORMAL OPERATING PRESSURE 600 PSI
   DESIGN PRESSURE 1200 PSI

3. LINE SIZE: PORT DIAMETER 7.75 IN. 10"O.D.

4. OPERATION: TWO POSITION OPERATION
   ACCOMPLISHED BY LINE FLUID PRESSURE
   WITH FOUR WAY VALVE FOR EXTERNAL
   HYDRAULIC PRESSURE WITH FOUR WAY
   VALVE

5. WEIGHT 220 POUNDS (ESTIMATED)

6. DIMENSIONS:

| VALVE | DIMENSION | |
DIAMETER	A	B
10'	24.3	15.3

7. MATERIAL KEY: ▨ ALUMINUM, ▨ STEEL

SECTION AA

SECTION CC

SECTION BB

WEIGHT VS DIAMETER
600 PSI VALVES
(ESTIMATED)

WADC TR 54-562
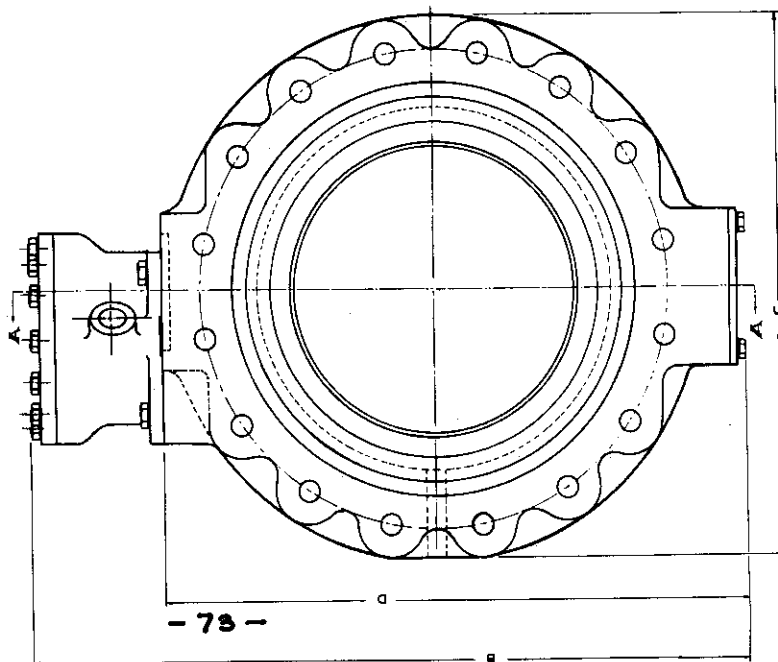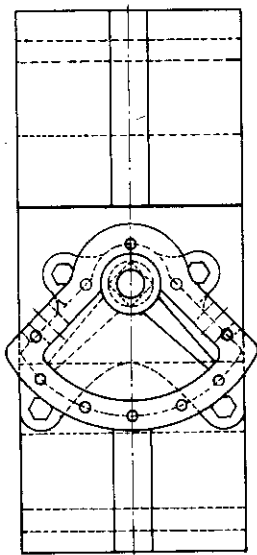
NOTES:

1. BUTTERFLY TYPE SHUT OFF VALVE
2. LINE SIZE 10" Ø PORT DIAMETER 7.75"
3. OPERATING PRESSURE 600 PSI
4. HYDRAULIC VANE MOTOR ACTUATOR
5. HYDRAULIC OPERATING PRESSURE ~3000 PSI
6. ESTIMATED WEIGHT WITH ACTUATOR ~85 LBS.
7. DIMENSIONS:

MATERIAL KEY: ▨ ALUM ▦ STEEL

FLUIDYNE ENG. CORP
MINNEAPOLIS, MINNESOTA

WEIGHT IS DIAMETER
600 PSI VALVES
(ESTIMATED)

DIAMETER ~ IN.

WEIGHT ~ LBS.

SECTION A-A

HYDRAULIC ACTUATOR

V SHAFT PACKING
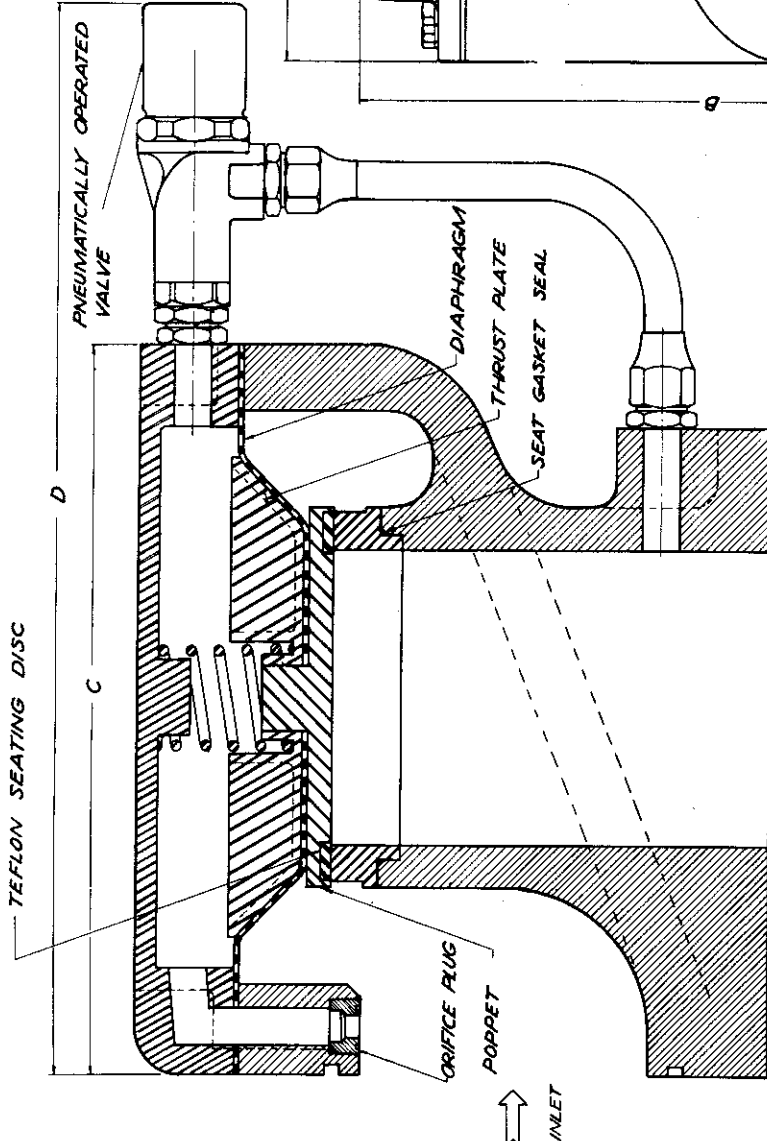
BEARING

EXPANDABLE SEAL

VANE
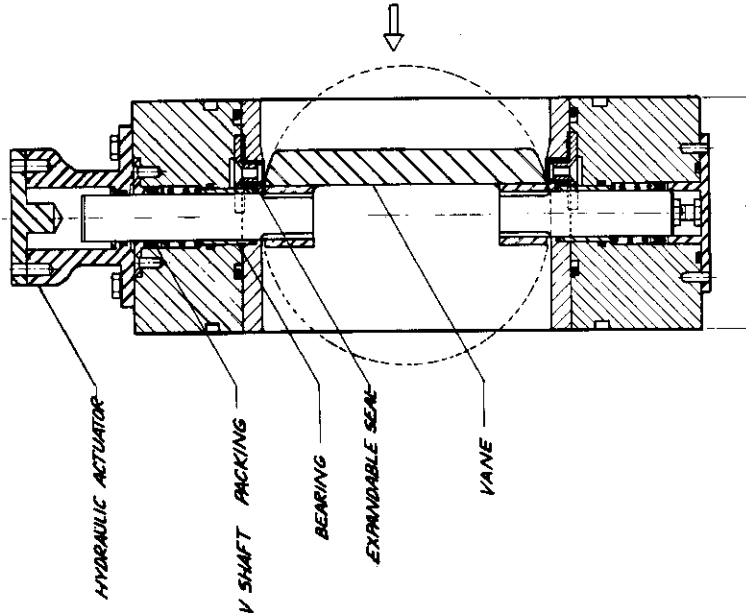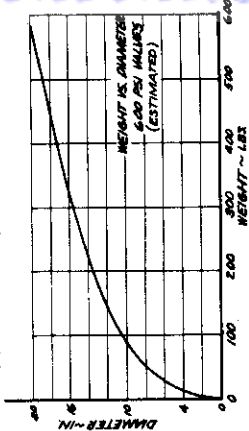
FIGURE 6 BUTTERFLY VALVE

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WADC TR 54-562

NOTES:

1. DIAPHRAGM TYPE SHUT-OFF VALVE
2. LINE SIZE 10" PORT DIAMETER 7.75
3. OPERATING PRESSURE 600 PSI
   DESIGN PRESSURE 1200 PSI
4. LINE PRESSURE ACTUATED
5. ESTIMATED WEIGHT - 240 LBS
6. MATERIAL KEY

	ALUMINUM		STEEL	
VALVE DIA	A	B	C	D
10"	6	7.91	14.21	15.5

7. DIMENSIONS

PNEUMATICALLY OPERATED
VALVE

TEFLON SEATING DISC

DIAPHRAGM
THRUST PLATE
SEAT GASKET SEAL

D
C

ORIFICE PLUG
POPPET

INLET

WEIGHT VS DIAMETER
600 PSI
(ESTIMATED)

DIAMETER-IN.
WEIGHT LBS
500 1000 1500 2000
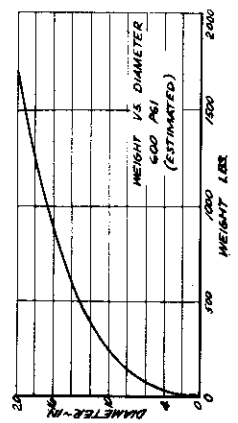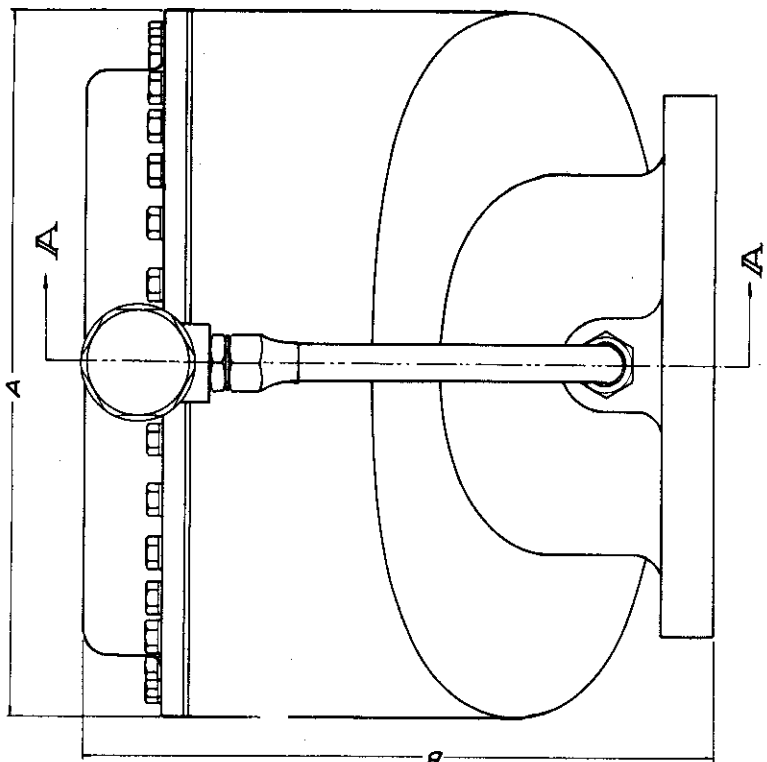
SECTION A-A

FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA

FIGURE 7 DIAPHRAGM VALVE

NOTES:

1. ECCENTRIC PLUG TYPE SHUT-
   OFF VALVE.
2. LINE SIZE 10"OD. PORT DIAMETER
   7¾"
3. OPERATING PRESSURE 600PSI/
   DESIGN PRESSURE 1200 PSI
4. DOUBLE ACTING PISTON ACTUATOR
   HYDRAULIC OPERATING PRES. 3000 PSI.
5. ESTIMATED WEIGHT – 249 LBS.
6. DIMENSIONS :

	VALVE DIA.	A	B	C	D
	10"				

7. MATERIAL KEY: ▨ ▨
   ALUM STEEL

ACTUATOR MOUNT

PLUG SHAFT

FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA

HEIGHT VS. DIAMETER
600 PSI
(ESTIMATED)

DIAMETER – IN.

WEIGHT – LBS

SECTION B.B.
SHOWN TO INCLUDE ACTUATOR

PLUG

PLUG

SEAT

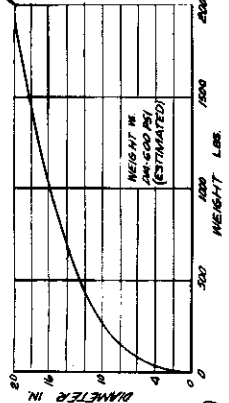DOUBLE ACTING PISTON ACTUATOR

INLET
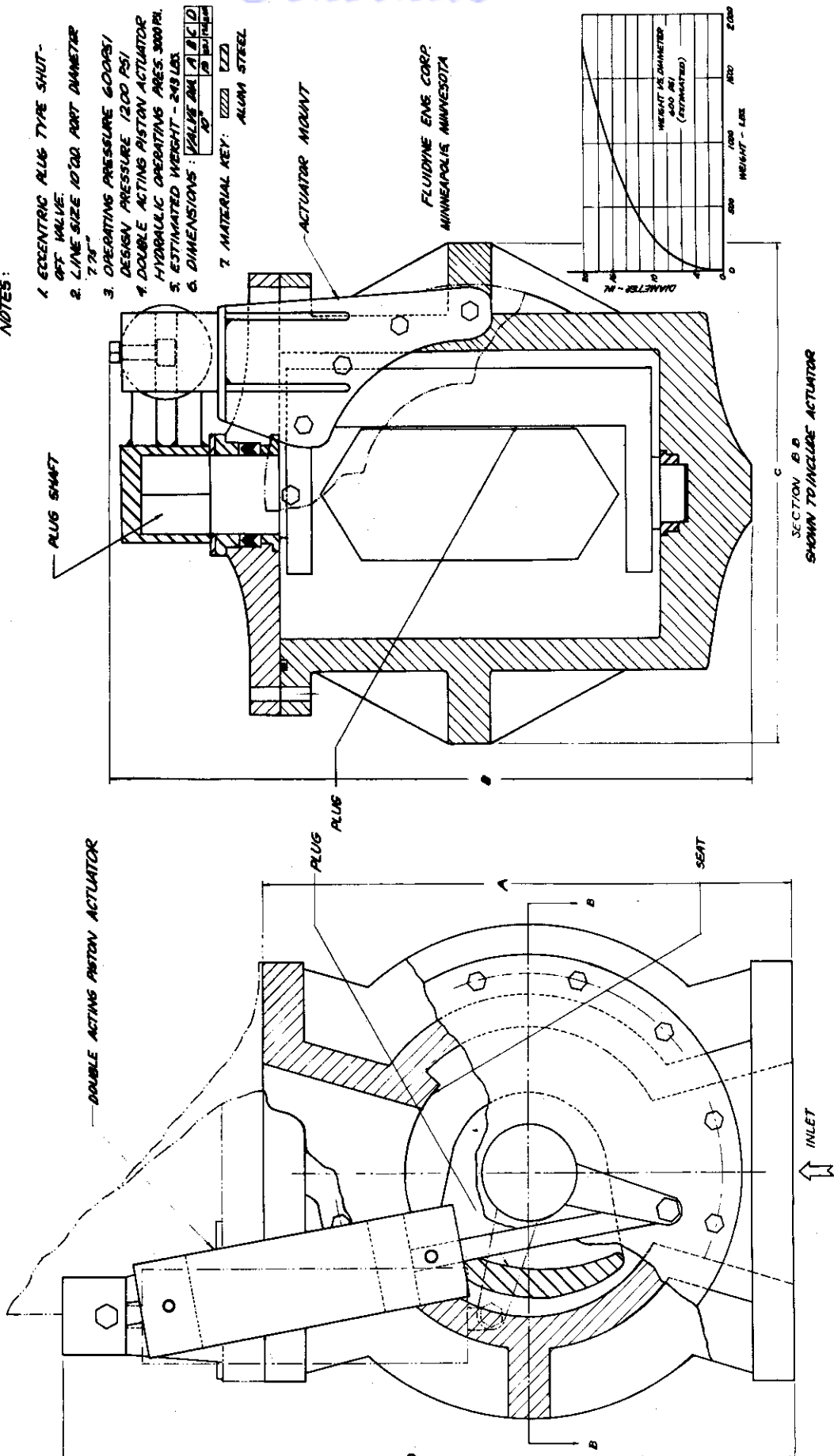
FIGURE 8 PLUG VALVE

FIGURE 9 CONE VALVE

NOTES:
1. CONE TYPE SHUT-OFF VALVE
2. LINE SIZE 10" .775" PORT DIAMETER
3. OPERATING PRESSURE 600 PSI ~
   DESIGN PRESSURE 1200 PSI
4. VANE TYPE-ACTUATOR INTEGRAL WITH VALVE
   CONSTRUCTION AND UTILIZING LINE PRESSURE
5. ESTIMATED WEIGHT - 265 LBS
6. MATERIAL KEY   AL ▨ STL ▨
7. DIMENSIONS

VALVE DIA.	A	B	C
10'	302	20	16

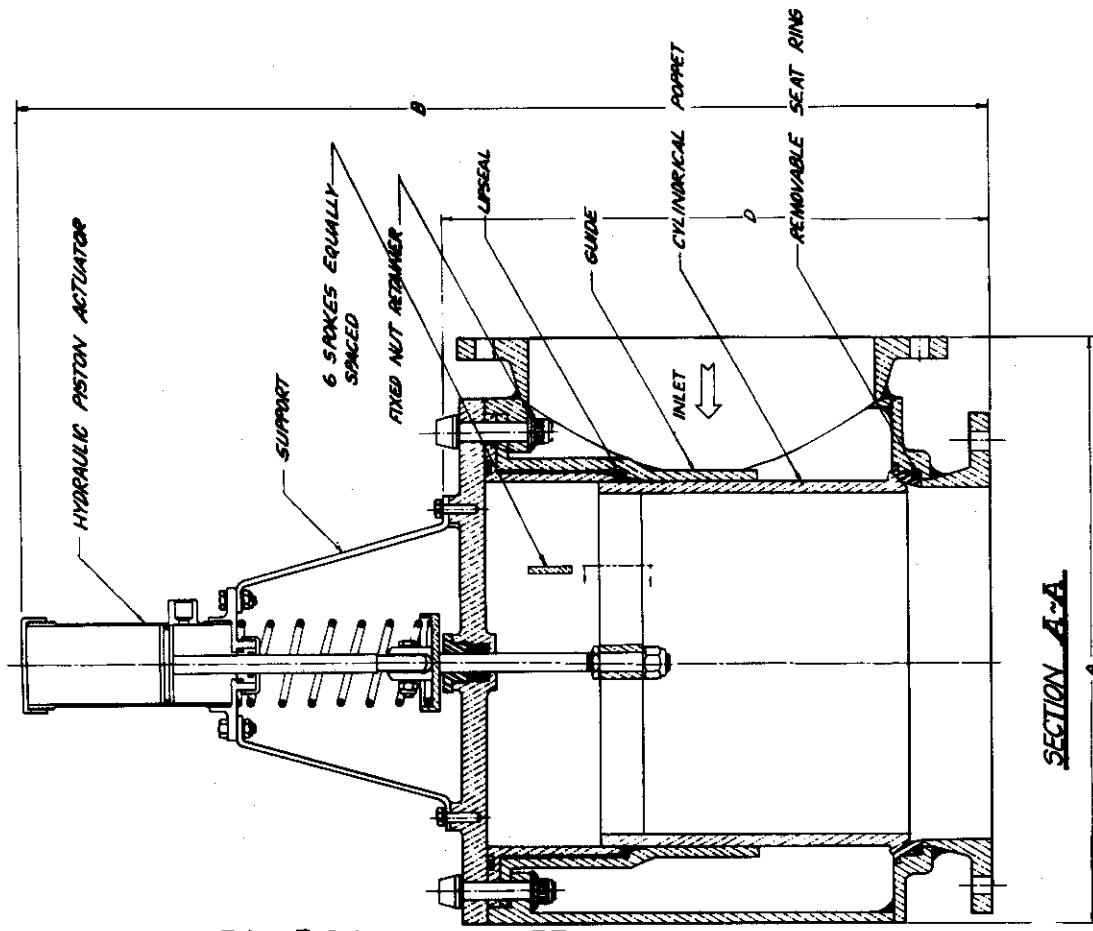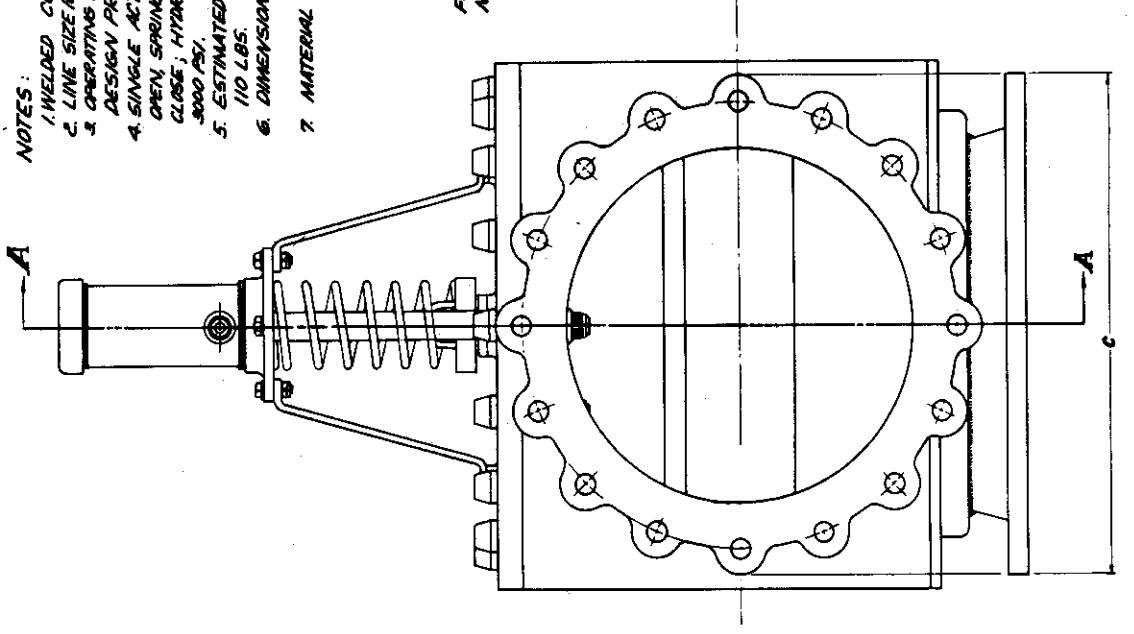FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA
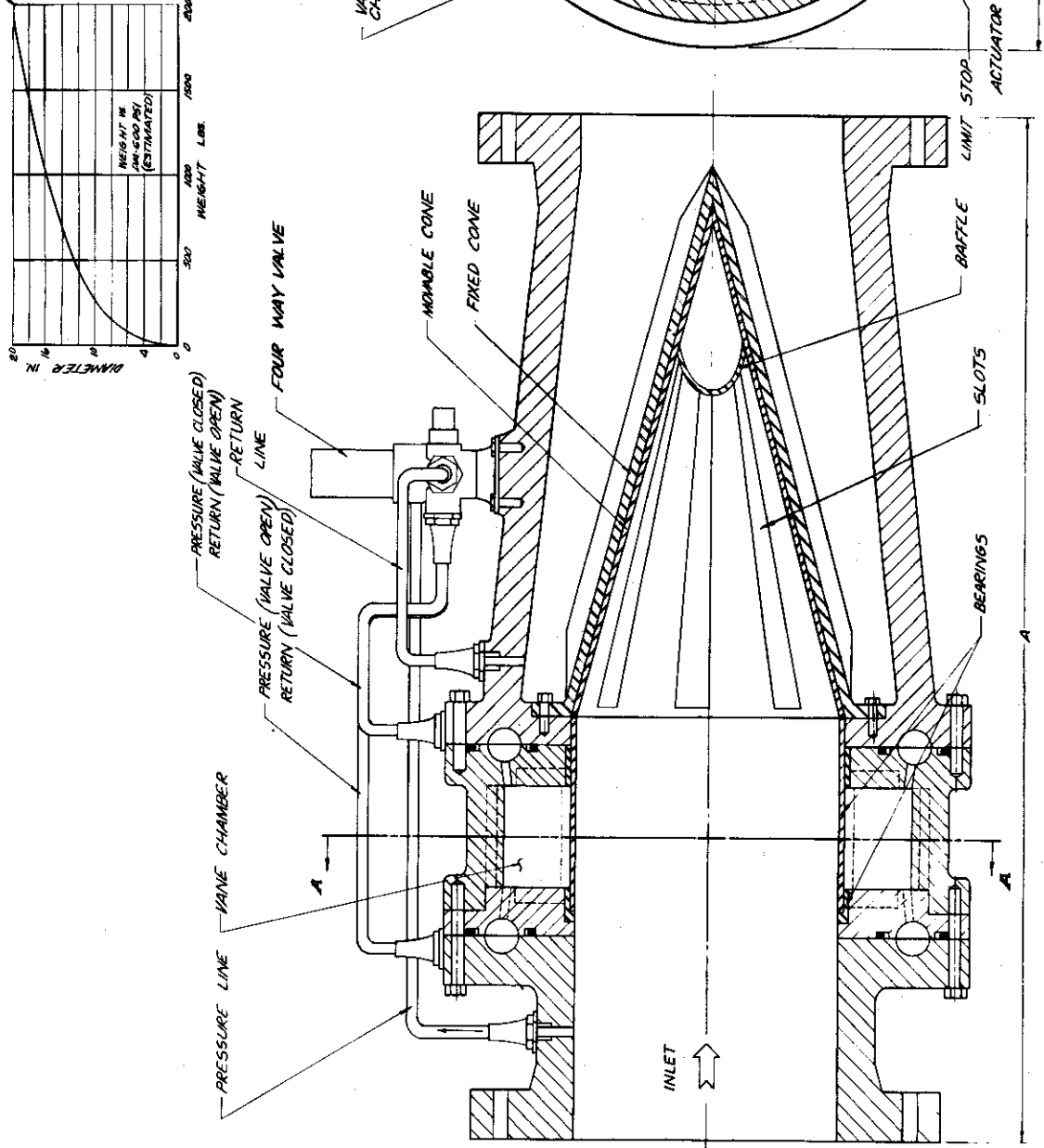
SECTION A-A

SECTION B-B

NOTES:
1. WELDED CONSTRUCTION.
2. LINE SIZE 10.00 PORT DIAMETER 9.60".
3. OPERATING PRESSURE 600 PSI.
   DESIGN PRESSURE 1200 PSI.
4. SINGLE ACTING PISTON ACTUATOR TO
   OPEN; SPRING AND LINE PRESSURE TO
   CLOSE; HYDRAULIC OPERATING PRESSURE
   3000 PSI.
5. ESTIMATED WEIGHT WITH ACTUATOR
   110 LBS.
6. DIMENSIONS

VALVE DIA.	A	B	C	D
10"	13.90	14.46		

7. MATERIAL KEY ▨ ▨
   TITANIUM   STEEL

FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA

HYDRAULIC PISTON ACTUATOR

SUPPORT

6 SPOKES EQUALLY
SPACED

FIXED NUT RETAINER

LIPSEAL

GUIDE

CYLINDRICAL POPPET

REMOVABLE SEAT RING

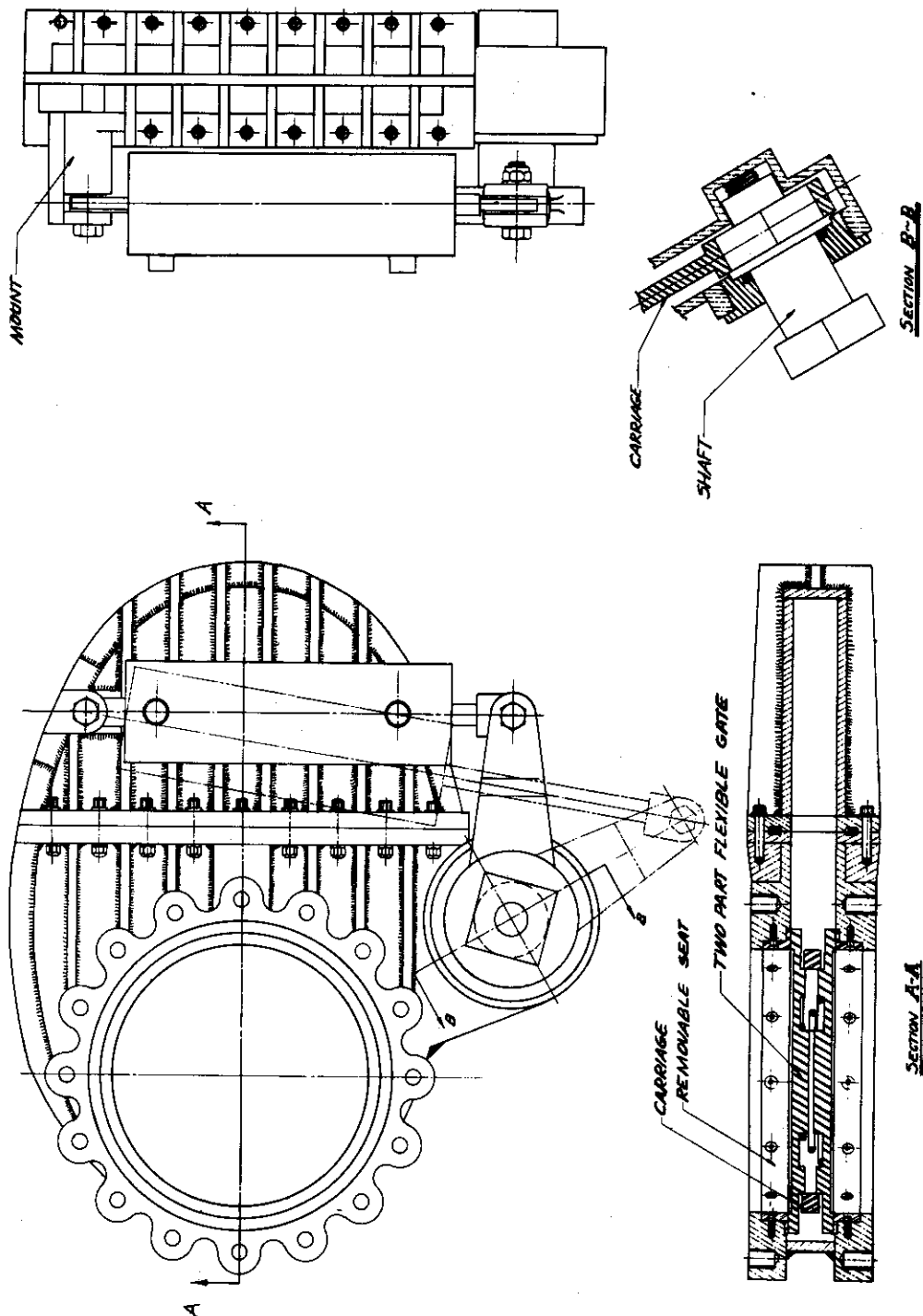INLET

SECTION A-A

FIGURE 10   MODIFIED ANGLE VALVE

NOTES:

1. PIVOTED GATE TYPE SHUT-OFF VALVE.
2. LINE SIZE 8"Ø PORT DIAMETER 7.5".
3. OPERATING PRESSURE 600 PSI. DESIGN 1200 PSI.
4. DOUBLE ACTING PISTON HYDRAULIC OPERATING PRESSURE 3000 PSI.
5. ESTIMATED WEIGHT WITH ACTUATOR 102 LBS.
6. DIMENSIONS [___]
7. MATERIAL KEY: [___] TITANIUM STL

FLUIDYNE ENG. CORP.
MINNEAPOLIS, MINNESOTA

MOUNT

CARRIAGE

SHAFT

SECTION B-B

CARRIAGE

REMOVABLE SEAT

TWO PART FLEXIBLE GATE

SECTION A-A

FIGURE-11   MODIFIED PIVOTED GATE VALVE

WADC TR 54-562   - 78 -

SYMBOL NOTATION

VALVE TYPE

Symbol	Type
⊟	PIVOTED GATE
◁	Y- PATTERN
⊡	SLIDE
▷	ANGLE
△	POPPET
⊡	BUTTERFLY
◇	DIAPHRAM
○	PLUG
◇	CONE

FLUI DYNE ENGR. CORP
MINNEAPOLIS, MINNESOTA

VALVE WEIGHT (LBS.)

VALVE DIAMETER (INCHES)

FIG. 12 ~   WEIGHT VS. DIAMETER ~ 600 PSI VALVES

FIG. 13 ~ WEIGHT VS. DIAMETER ~ 60 PSI VALVES

SYMBOL NOTATION

VALVE TYPE	
PIVOTED GATE	⊡
Y-PATTERN	△
SLIDE	▢
ANGLE	▷
POPPET	△
BUTTERFLY	⊡
DIAPHRAGM	◇
PLUG	○
CONE	○

FLUI DYNE ENGR CORP.
MINNEAPOLIS, MINNESOTA

VALVE WEIGHT (LBS.)

VALVE DIAMETER (INCHES)

WADC TR 54-562 -80-

FIG. 14 ~ LOG PLOT OF~WEIGHT VS DIAMETER~600 PSI VALVES

VALVE WEIGHT ~LBS (INCLUDES ACTUATOR)

VALVE DIAMETER(INCHES)

SYMBOL NOTATION

VALVE TYPE	
PIVOTED GATE	⊡
Y- PATTERN	△
SLIDE	□
ANGLE	▽
POPPET	△
BUTTERFLY	⊡
DIAPHRAGM	◇
PLUG	○
CONE	⊘

FLUI DYNE ENGR CORP
MINNEAPOLIS, MINNESOTA

FIG.15 ~ LOG PLOT OF~WEIGHT VS. DIAMETER~60 PSI VALVES

VALVE WEIGHT~LBS (INCLUDES ACTUATOR)

VALVE DIAMETER (INCHES)

SYMBOL NOTATION

VALVE TYPE	
PIVOTED GATE	□
Y-PATTERN	△
SLIDE	□
ANGLE	▷
POPPET	△
BUTTERFLY	□
DIAPHRAGM	◇
PLUG	○
CONE	○

FLUI DYNE ENGR. CORP.
MINNEAPOLIS, MINNESOTA

WADC   TR 54-562   -82-

NOTE:

COEFFICIENT OF STATIC (BREAKOUT) FRICTION ~ 0.25

SYMBOL NOTATION

VALVE TYPE
PIVOTED GATE
SLIDE
BUTTERFLY
PLUG
CONE

FLUIDYNE ENGR. CORP.

MINNEAPOLIS, MINNESOTA

TORQUE (THOUSANDS INCH POUNDS)

VALVE DIAMETER (INCHES)

FIG 16 ~ MAXIMUM ACTUATING TORQUE ~ 600 PSI VALVES
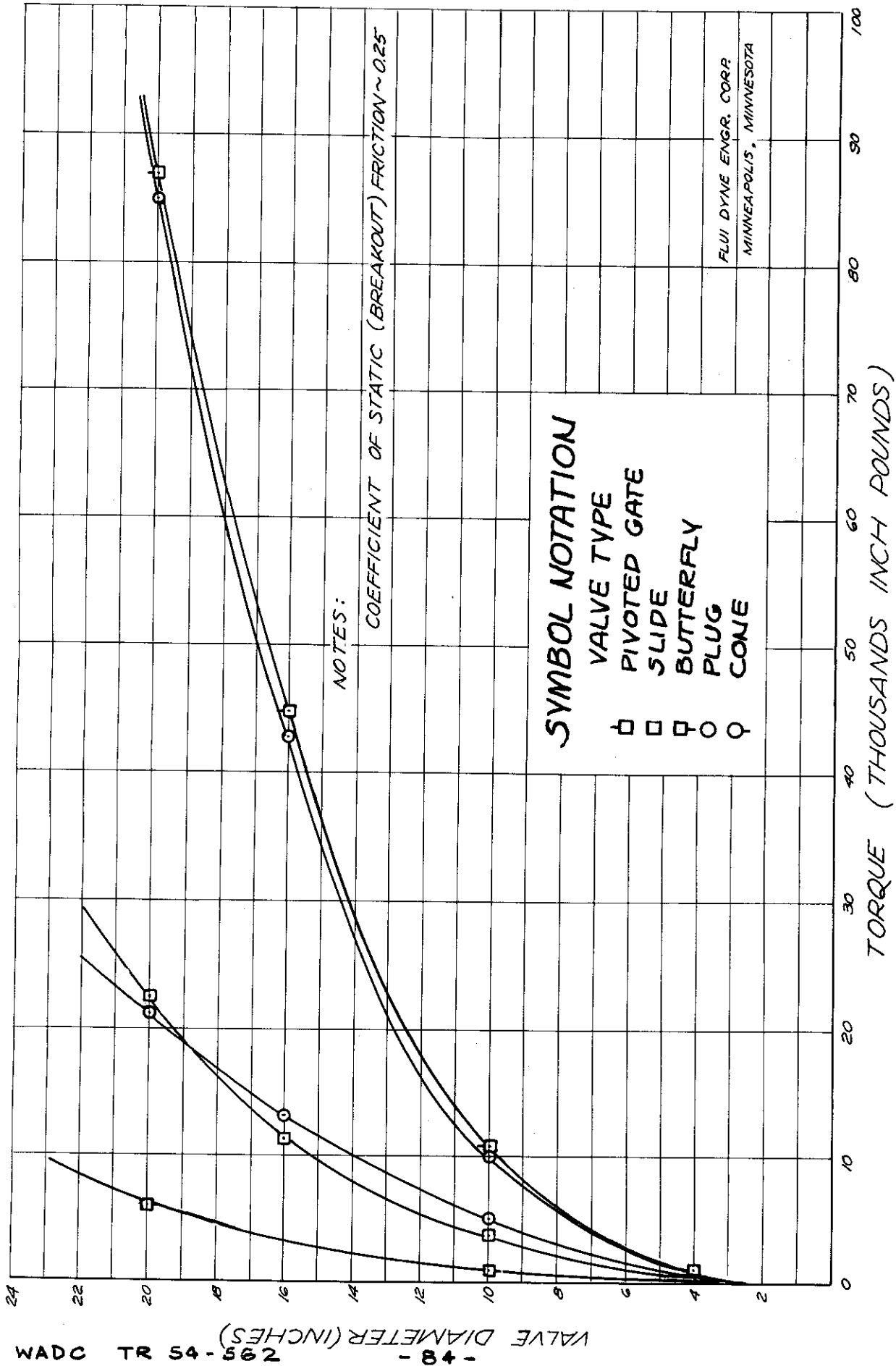
FIG 17 - MAXIMUM ACTUATING TORQUE ~60 PSI VALVES

NOTES:

COEFFICIENT OF STATIC (BREAKOUT) FRICTION ~0.25

SYMBOL NOTATION

VALVE TYPE
PIVOTED GATE
SLIDE
BUTTERFLY
PLUG
CONE

FLUI DYNE ENGR. CORP.
MINNEAPOLIS, MINNESOTA

TORQUE (THOUSANDS INCH POUNDS)

VALVE DIAMETER (INCHES)
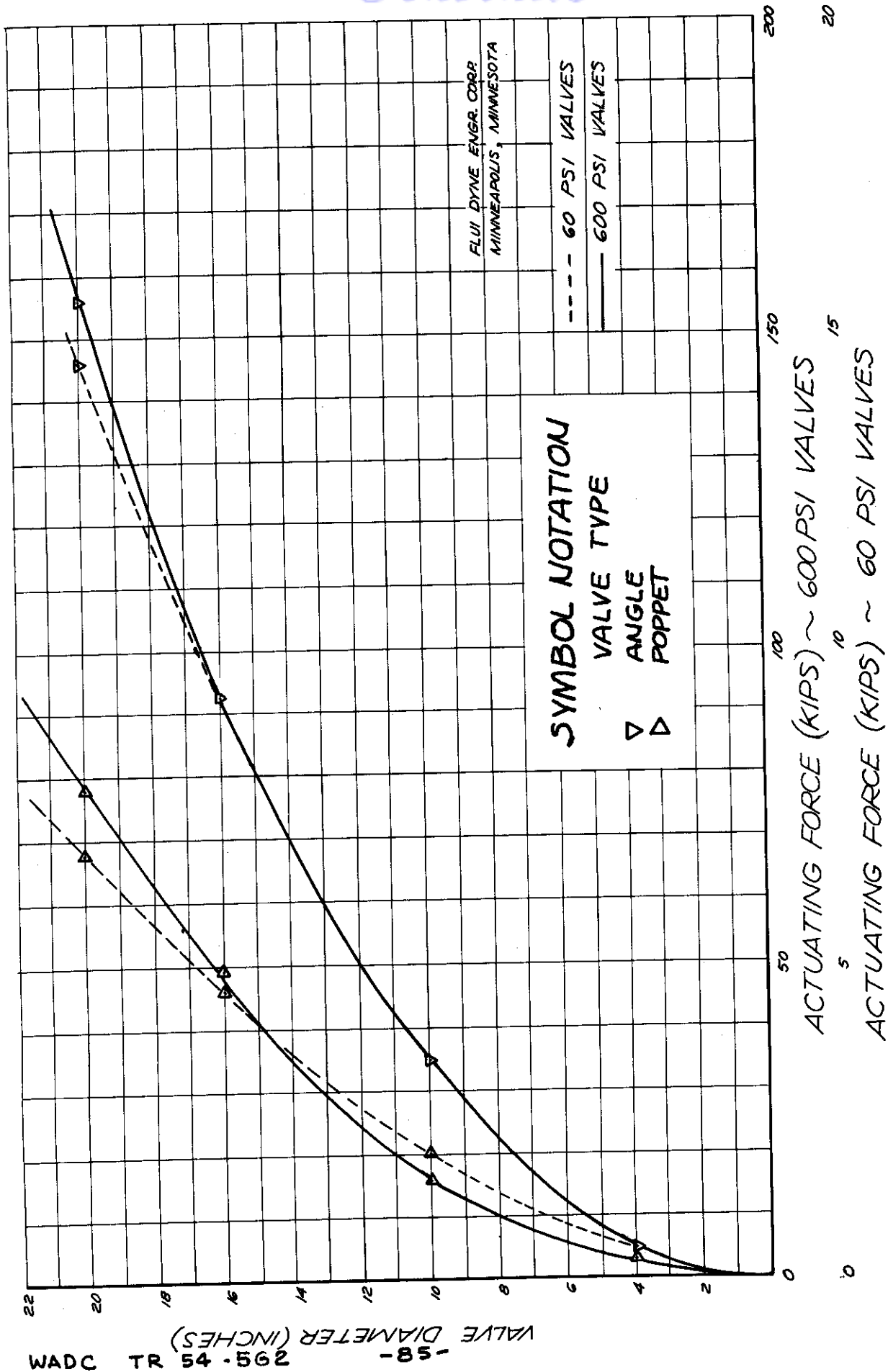
FIG 18   ACTUATING FORCE VS. DIAMETER

SYMBOL NOTATION

VALVE TYPE	
ANGLE	▽
POPPET	△

FLUI DYNE ENGR. CORP.
MINNEAPOLIS, MINNESOTA

- - - - 60 PSI VALVES
———— 600 PSI VALVES

ACTUATING FORCE (KIPS) ~ 600 PSI VALVES

ACTUATING FORCE (KIPS) ~ 60 PSI VALVES

VALVE DIAMETER (INCHES)
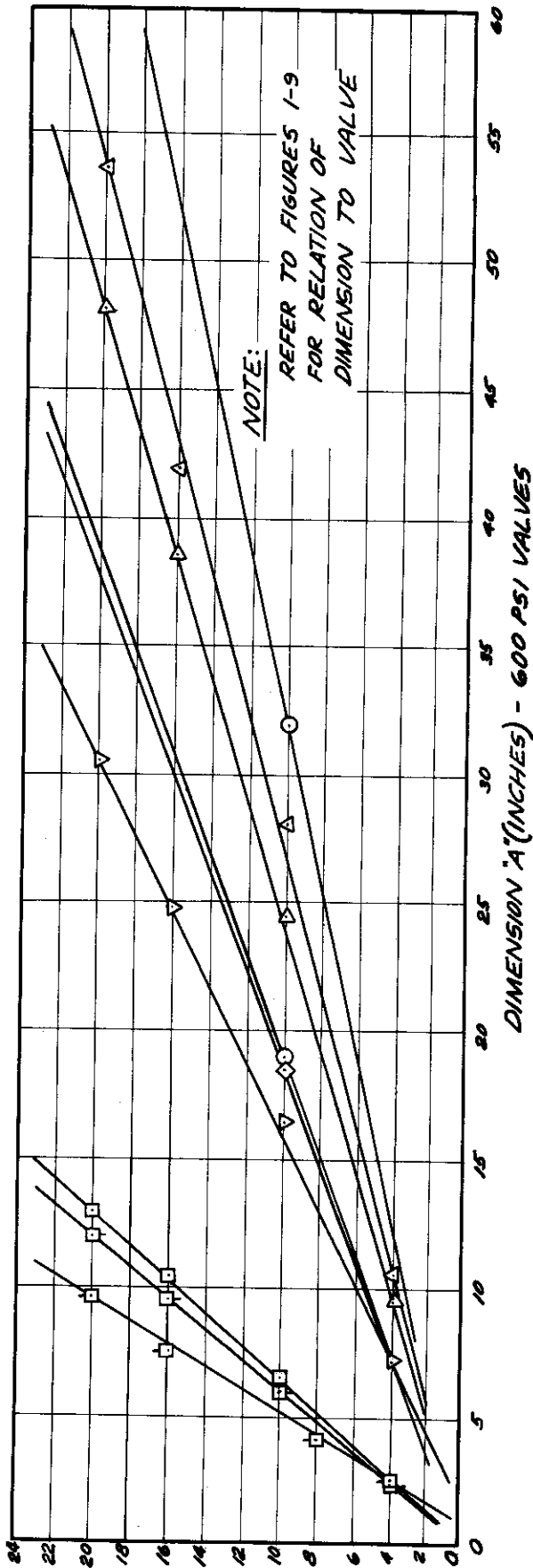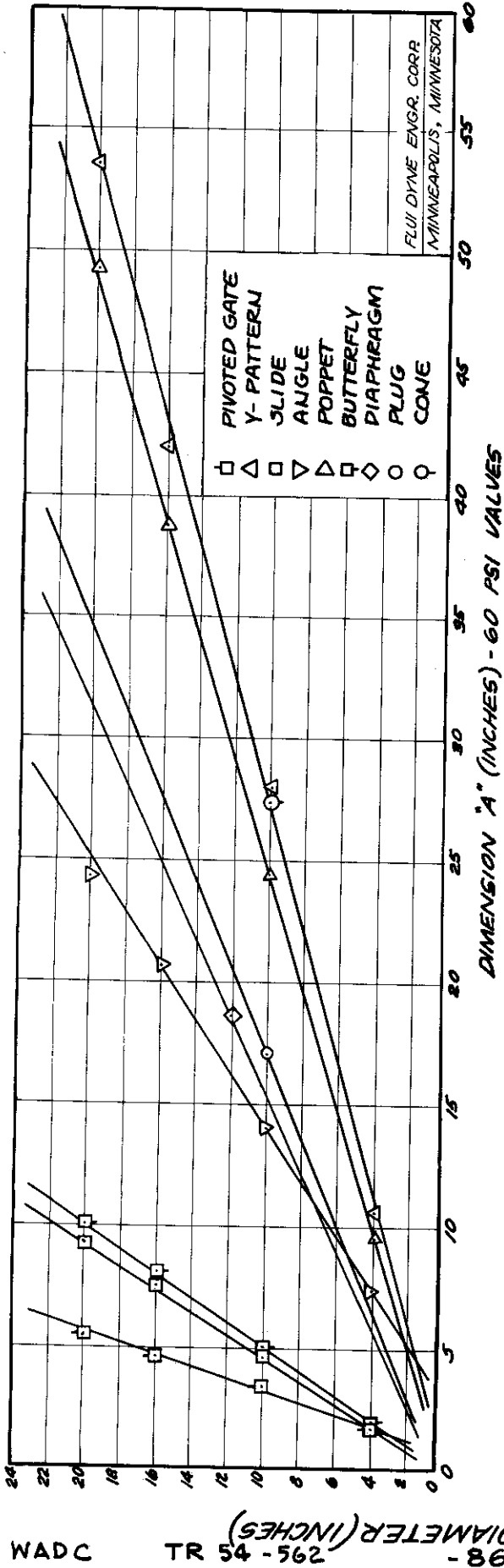
FIG. 19 ~ FLANGE TO FLANGE DIMENSION "A" VS DIAMETER
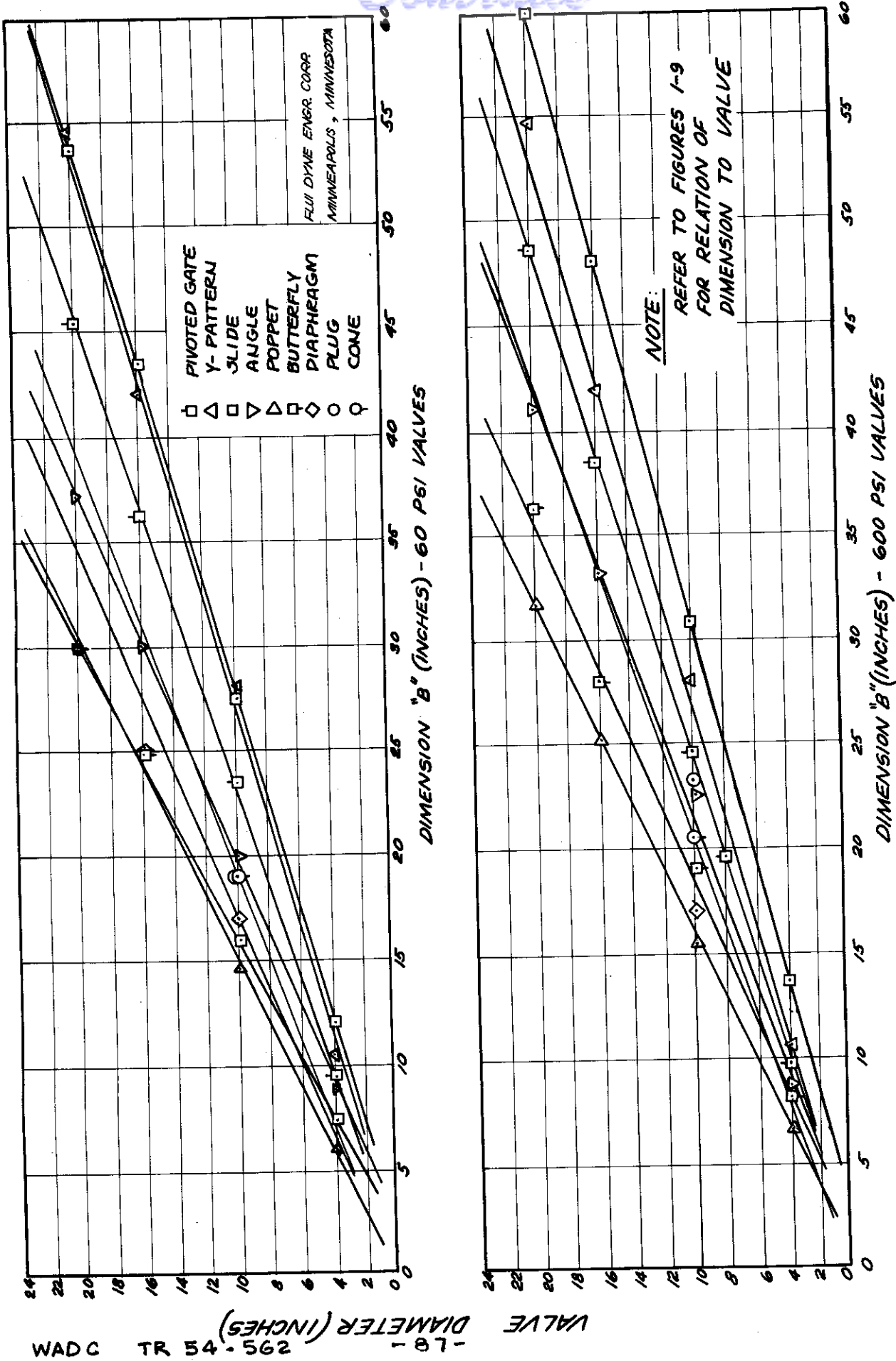
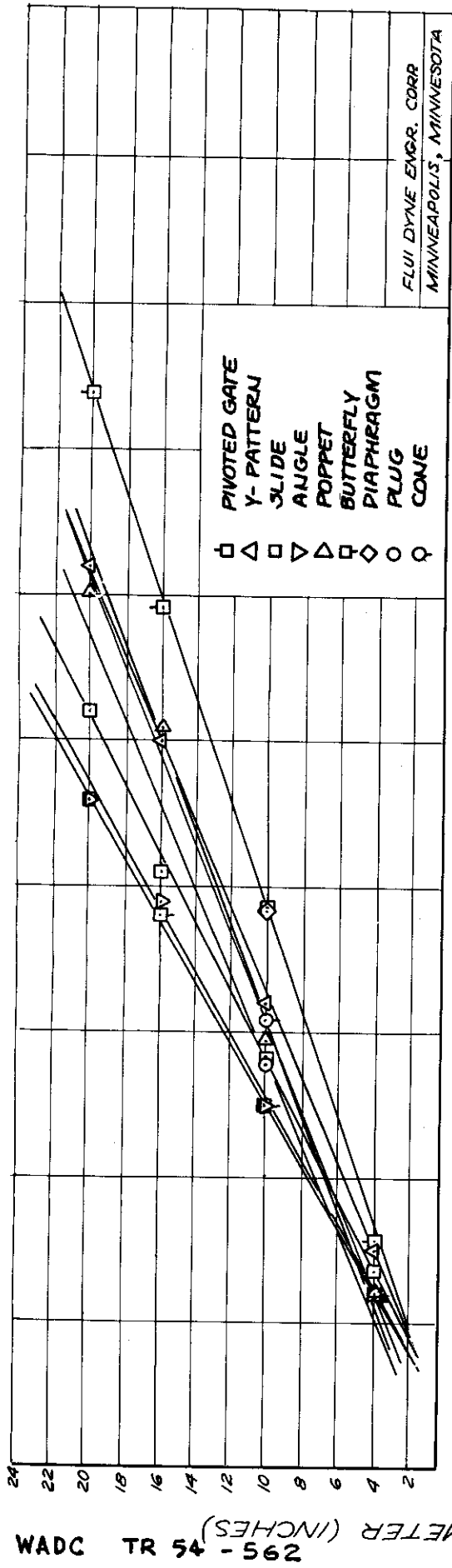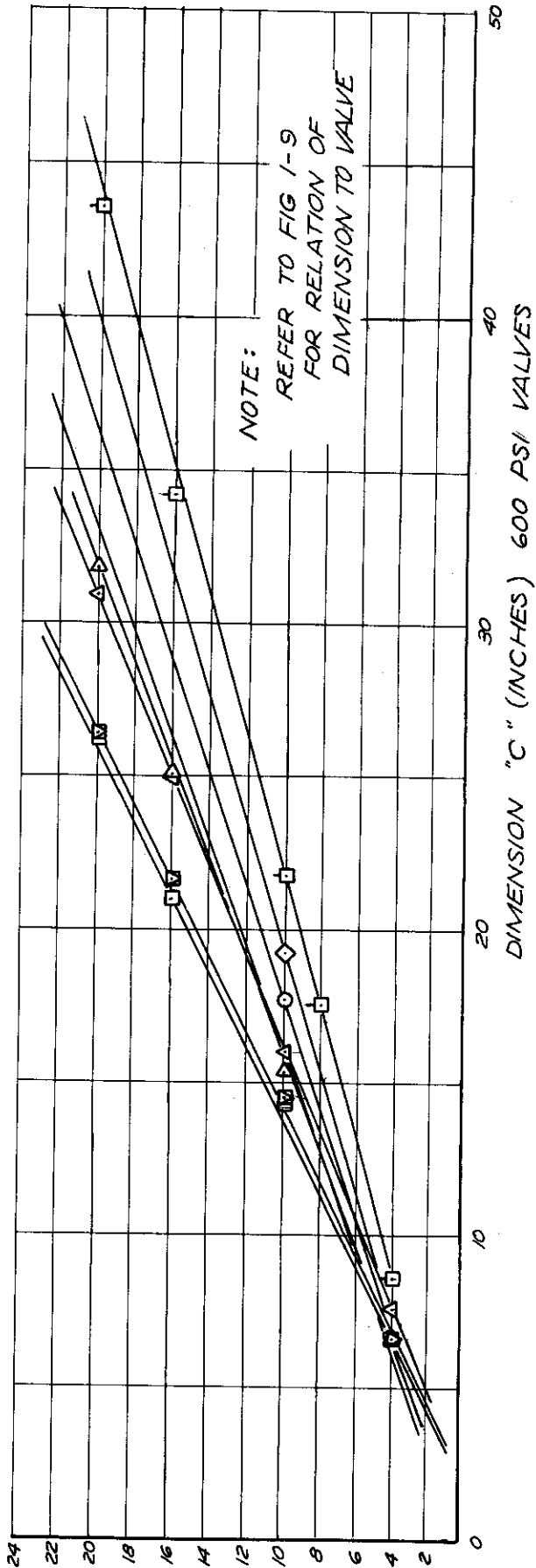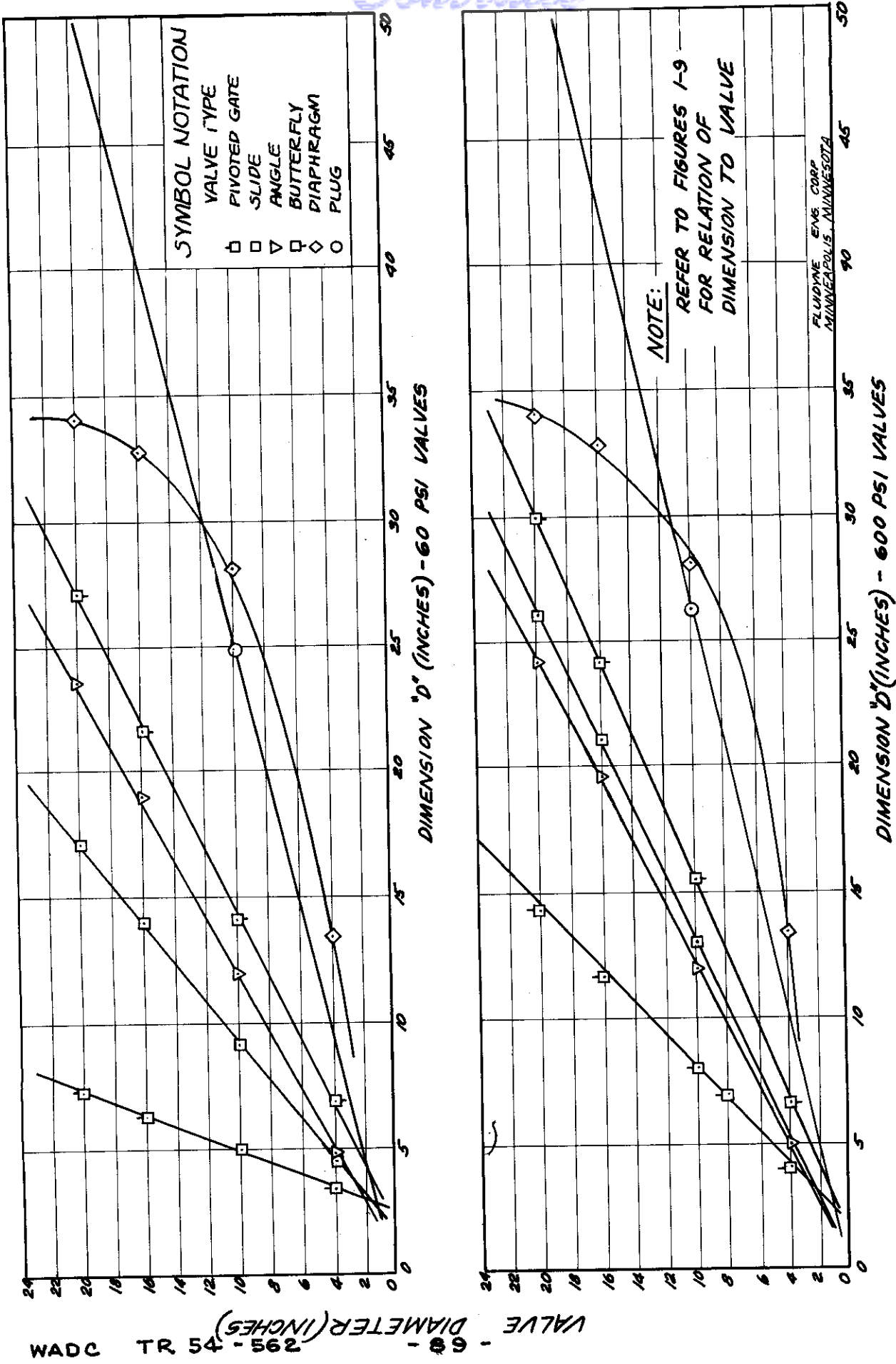FIG. 20 ~ OVERALL HEIGHT DIMENSION "B" VS. DIAMETER

FIG. 21 OVERALL WIDTH DIMENSION "C" VS DIAMETER

FIG. 22 ~ DIMENSION "D" VS. DIAMETER

SYMBOL NOTATION

VALVE TYPE	
PIVOTED GATE	⊟
SLIDE	☐
ANGLE	▽
BUTTERFLY	⬠
DIAPHRAGM	◇
PLUG	○

DIMENSION "D" (INCHES) - 60 PSI VALVES

DIMENSION "D" (INCHES) - 600 PSI VALVES

VALVE DIAMETER (INCHES)

NOTE:
REFER TO FIGURES 1-9
FOR RELATION OF
DIMENSION TO VALVE

FLUIDYNE ENG. CORP
MINNEAPOLIS, MINNESOTA