ELECTRIC ACTUATION OF A SURFACE CONTROLLED SUBSEA SAFETY VALVE

A Senior Honors Thesis

by

TIMOTHY JOSEPH BARTLETT

Submitted to the Office of Honors Program
& Academic Scholarships
Texas A&M University
in partial fulfillment of the requirements of the

UNIVERSITY UNDERGRADUATE
RESEARCH FELLOWS

April 2002

Group: Science/Engineering

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Approved as to style and content by:

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April 2002

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ABSTRACT

Electric Actuation of a Surface Controlled

Subsea Safety Valve. (April 2002)

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The growing technology associated with petroleum equipment has lead to an increase in electrical components used to measure, adjust and control many aspects of the petroleum extraction process. This increase in electrical use has lead to the conversion of many components from hydraulic operation to electric. Therefore, it is the purpose of this report to detail and propose an engineering solution to the electrical conversion of a surface controlled subsea safety valve (SCSSV).

Preliminary analysis has resulted in the following important functional requirements that the SCSSV should operate within:

- Temperatures up to 350 °F.
- Estimated operational life of 20 years.
- Peak available power of 4300 W (actual value will be much lower than this due to other power system drains).

The three conceptual designs presented in this report are the following:

- Electric motor actuator with a solenoid latching detent.
- Linear actuation with a fluid reservoir locking valve.
- Rotational engagement with an electric locking clutch.

After reviewing and comparing these concepts, the last concept was chosen to be further developed. This concept will have the least temperature sensitive components and fewer leak path problem areas.

Upon completion of the detailed design phase an electric SCSSV solution has been proposed. The following is a brief list of the performance claims:

- Power requirement to open: 0.5 W.

- Time required to open valve: 10 minutes.
- Installable inline with 4-½ tubing.
- Operating temperature range: 20-350 °F.
- H_2S service certified.
- Hold open energy requirements 0~0.2 W.

There are however practical issues that must be addressed to ensure reliable and satisfactory performance of the SCSSV. The actual efficiency of the cylindrical cam mechanism and reliability of the torque overload clutch must be empirically determined due to the number of theoretical assumptions used. Thus these items should be thoroughly tested in order to produce a reliable and satisfactory SCSSV.

It is with these claims and working issue that this electrically converted SCSSV be fully developed and installed for electrical petroleum production systems.

ACKNOWLEDGMENTS

This work was supported by FMC Kongsberg Subsea.

The author wishes to thank the faculty and fellow colleagues at Texas A&M University, specifically: Dr. Sooyong Lee and Dr. Reza Langari whom have assisted this effort.

The author also wishes to thank my fellow colleagues at FMC Kongsberg Subsea for their support and encouragement.

NOMENCLATURE

A_{ca}	Hydraulic piston area, accumulator side
A_{ch}	Hydraulic piston area, control line side
API	American Petroleum Institute
b_f	Flow tube damping coefficient
b_p	Hydraulic piston damping coefficient
b_v	Flapper valve damping coefficient
d_{Aap}	Hydraulic piston diameter, accumulator side
d_{Ach}	Hydraulic piston diameter, control line side
d_v	Flapper valve diameter
g	Gravitational constant
J_v	Flapper valve moment of inertia
k_s	Power return spring coefficient
k_v	Flapper valve coil spring coefficient
l_c	Flow tube/Flapper valve geometrical contact constraint
l_p	Flow tube displacement limit
m_f	Flow tube mass
m_p	Hydraulic piston mass
m_v	Flapper valve mass
NACE	National Association of Corrosion Engineers
$P_{acc}(x_f)$	Dynamic accumulator pressure
P_{int}	Initial accumulator pre-charge pressure
P_{pa}	Accumulator pressure
P_{ph}	Hydraulic control line pressure
P_w	Working pressure
SCSSV	Surface Controlled Subsea Safety Valve
T_{max}	Maximum extreme temperature
T_{min}	Minimum extreme temperature
TOC	Time of Command
T_w	Working temperature
V_{int}	Initial accumulator pre-charge volume
x_f	Flow tube displacement
δ_k	Power return spring pre-load
δ_{of}	Flapper valve hinge pin offset
δ_θ	Flapper valve coil spring pre-load
θ_v	Flapper valve rotation

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INTRODUCTION

Throughout the petroleum industry, a number of safety mechanisms and redundancies must exist to prevent catastrophic failures and/or situations. As the shallow reservoirs across the world are exhausted, the deeper and more difficult reservoirs must be tapped and controlled safely. The development of subsea christmas tree installations, which allow remote well production on the sea floor, has brought the capability to control multiple oil reserves from one surface production platform. As the range of these reservoirs expands, the need to control a greater number of production areas within a central control station increases. With this distance, the current hydraulic control of subsea components are compromised due to ineffective response times, and alternative methods of long distance control must be employed.

Recently, the industrial trend has focused on using electrical power to control subsea oil field equipment, as opposed to current hydraulic technology that has been proven and used throughout the history of subsea oilfield equipment. With the possible switch to electrical command power, current safety mechanisms, which solely rely on hydraulic power, must be converted and qualified for service. Surface controlled subsea safety valves (SCSSV) are the current safety barrier inside the subsurface casing that has not been thoroughly examined for alternative means of operation[1-5].

SCSSV's are hydraulically actuated flapper type or ball valve situated inline with the casing. The casing is one of the multiple concentric strings of tubing that traverses between the wellhead and the reservoir. Current SCSSV's are controlled by hydraulic line pressure originating on the production platform. This line may be considerably long; therefore, large lag times may exist in operation due to the large dynamic fluid volume. Operation of the SCSSV is controlled by a hydraulic piston, which is linked to the valve. The hydraulic piston acts against a spring, which gives the system a fail-safe operation mode. The fail-safe mechanism occurs when the loss of hydraulic pressure allows the compressed spring to close the valve, sealing the reservoir from the subsea environment.

In order to increase the reliability and decrease the developmental phase of the alternative SCSSV the task will be simplified by developing only a means of actuator replacement, without redesigning current SCSSV valve operation. This issue is being actively examined and funded by FMC Kongsberg Subsea, a leader in subsea oilfield systems.

This thesis follows the style and format of the *Society of Petroleum Engineers Journal*.

A similar approach to the Texas A&M University Institute for Innovation & Design in Engineering process will be followed throughout this report. This engineering process consists of multiple steps, the three which will be stressed through this report are the following:

- Need Analysis.
- Conceptual Design Analysis.
- Embodiment Design.

These phases will be presented in the following text, documenting and supporting the validity of the proposed engineering solution.

NEED ANALYSIS

Introduction

The purpose of the Need Analysis is to develop a function structure allowing analysis of the critical constraints and parameters. The Need Analysis is the first stage in the Texas A&M University Institute for Innovation & Design in Engineering process. Through the completion of the Need Analysis an overall understanding of the problem will have been reached, possible solution areas may be seen, and transformation of industrial requirements into technical specifications will occur. The Need Analysis will define the parameters and requirements for the forthcoming design challenge.

Need Statement

The need statement establishes the scope of the design task and identifies the primary function along with the primary constraint. The statement is formed to be solution independent, allowing unrestricted innovation.

Need Statement:

Electrically allow reservoir access while maintaining unassisted failsafe closure.

Function Structure

The premise of the function structure is to identify the real scope and magnitude of the need. The function structure consists of primary functions and secondary functions. The characteristics of the function structure are as follows: ascending in the hierarchy will show "why", while dissension in the hierarchy will show "how". All precautions have been taken to avoid any solution specific functions, constraints, or parameters. The complete function structure can be found in **Fig. 1**.

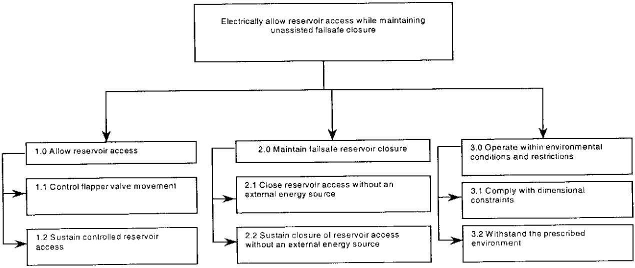

Fig. 1—Function structure.

Secondary Function Structure Expansion

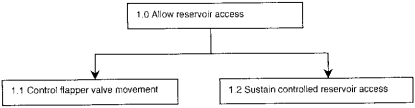

Fig. 2—Secondary function structure 1.0.

Allow reservoir access

The secondary function structure for function 1.0 can be found in **Fig. 2**. The following expansion will identify the corresponding design parameters and constraints.

1.1 Control flapper valve movement

Parameter(s): Peak current draw
Peak voltage

Constraint(s): Peak current draw below 180 A
Peak voltage below 48 V

1.2 Sustain controlled reservoir access

Parameter(s): Power required to sustain reservoir access

Constraint(s): Steady state power below 220 W

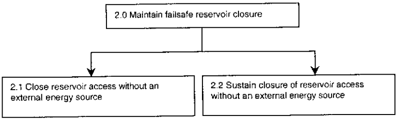

Fig. 3—Secondary function structure 2.0.

Maintain failsafe reservoir closure

The secondary function structure for function 2.0 can be found in **Fig. 3**. The following expansion will identify the corresponding design parameters and constraints.

2.1 Close reservoir access without an external energy source

Parameter(s): Failsafe closure mechanism(s)

Constraint(s): Store energy required for closure within the valve. The loss of control signal would indicate closure of the reservoir.

2.2 Sustain closure of reservoir access without an external energy source

Parameter(s): Reservoir closure

Constraint(s): When unpowered the valve must close off access to the reservoir, in the presence of reservoir pressure and hydrocarbon/gas flow.

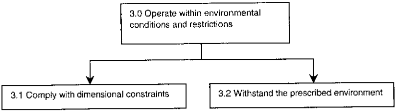

Fig. 4—Secondary function structure 3.0

Operate with environmental conditions and restrictions

The secondary function structure for function 3.0 can be found in **Fig. 4**. The following expansion will identify the corresponding design parameters and constraints.

3.1 Comply with dimensional constraints

Parameter(s): Must be installed in current well designs
Must not restrict flow more than current design

Constraint(s): Max O.D.: 5.135 or 8.875 in
Min I.D.: 2.379 or 4.625 in

3.2 Withstand the prescribed environment

Parameter(s): Temperature variations, hydrostatic installation pressure, working bore pressure, and compatible with H_2S environment

Constraint(s): Temperature variation between 20 and 350 °F, installation pressure ~ 4000 psi, working bore pressure 15,000 psi, and material must meet National Association of Corrosion Engineers (NACE) specifications for H_2S environments[6]

Design Specifications

Upon completion of function structure, a collection of design parameters can be seen in **Table 1**. These parameters will be utilized throughout the design process. These specifications outline the functional constraints and design criteria of the problem presented by FMC Kongsberg Subsea.

TABLE 1—CRITICAL DESIGN PARAMETERS

Parameter	Value	Units
Temperature Range	20-350	°F
Working Reservoir pressure	15	ksi
Bore test pressure	18.75-22.5	ksi
Min I.D.[*]	2.379 4.625	in
Max O.D.[*]	5.135 8.875	in
Design Life	20	yrs
Max outer pressure	4	ksi
Max available power[**]	4300	W
S.S. power range	220	W
Max current draw[**]	<180	A
Max voltage[**]	<48	V
Current hydraulic power req.[†]	~160	W
Test cycle life	500	Cycles
H2S resistant	NACE Specifications	

* Taken from two through bore SCSSV from various data sheet
** Values based on battery information NOTE: these values will be lower with the load of other equipment
† Order of magnitude estimation from extrapolated measurements operating with a 1000 psi pressure differential on the piston and an opening time of 5 seconds

It should be noted that due to time requirements and unavailability of items and information that all design parameters might not be met. If any parameter must be compromised a detailed explanation for its dismissal will be documented. This will ensure the possibility for future designs to incorporate features not currently available, or were unable to be fully developed.

Summary

FMC Kongsberg Subsea has provided a design challenge to electrically allow reservoir access through the production tubing, while retaining a failsafe safety barrier for emergency cases. The appropriate solution will facilitate an electric signal actuating access to the reservoir, a mechanism to allow continuous access to the reservoir without the need for large power consumption, and finally a failsafe mechanism that requires no external energy sources for operation. All these challenges must be carried out in the variance of temperature, high pressures, and dimensional constraints that have been specified in the section.

HYDRAULIC SCSSV MODELING

Detail of Safety Valve Operation

The safety valve used in this model will be a simplified flapper type valve found in most applications. The safety valve operates by moving a hydraulic piston, which acts on an internal flow tube. The motion of the flow tube causes the rotation of the flapper valve. This process will be explained in more detail in the following text and figures. The detailed explanation will be expanded in three stages: Closed, Opening, Open, and Closing. A simplified schematic of the safety valve can be seen in **Fig. 6**. This figure will be referenced in the following sections.

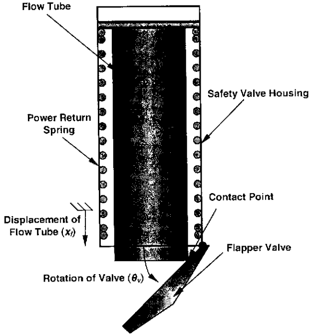

Fig. 5 –Simplified contact schematic of safety valve.

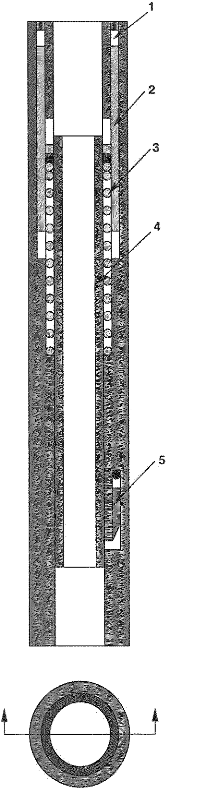

Figure 6—Simplified safety valve cross-sectional schematic.

Closed:

In the closed position, no hydraulic pressure is applied to the control line (**1**). One of the forces assisting the retracting the flow tube (**4**) by acting on the hydraulic piston (**2**) is the coil-tubing accumulator, which is not shown in the figure. The coil-tubing accumulator is acting on the opposing side of the control line (**1**), therefore the piston is slightly biased towards an upward direction from the representative figure. The majority of biased movement upward is created by the power return spring (**3**), which acts on the flow tube (**4**). The spring is the failsafe mechanism that stores the energy required to close the valve on loss of hydraulic power to the control line. When the flow tube (**4**) has retracted completely upwards, the flapper valve (**5**) is free to rotate back into the working bore. The flapper valve (**5**) is also slightly biased due to a small coil spring forcing the valve out into the working bore, thus sealing the reservoir. The flapper valve remains closed due to the reservoir pressure acting upwards on the bottom face of the valve.

Opening

In order to move the hydraulic piston (**1**) the differential pressure across the flapper valve should be less than 500 psi. Hydraulic pressure is applied from the control source, once this pressure reaches a specified level, one that will overcome the accumulator pressure plus the pre-load force of the return spring, the piston will begin to progress downwards. Once the piston begins to move, it contacts the flow tube (**4**). The flow tube and the flapper valve are in constant contact where the variable in the movement is the contact point. This can be seen in a simplified schematic in Fig. 5. Through this schematic, it is seen that the downward movement of the flow tube directly causes the flapper valve to rotate in a counter clockwise manner.

Open:

Once the flow tube has breached contact with the flapper valve, the flapper valve will be positioned completely out of the working bore. The hydraulic piston will press against a stop, once this position has been reached the control line pressure must remain. The control line pressure will keep the flow tube and hydraulic piston engaged with the stop, thus keeping the valve from retracting back into the working bore.

Closing:

Once the hydraulic pressure has been removed or lost from the control line, the slight bias from the coil-tubing accumulator and the large force of the power return spring will begin to retract the flow tube. Once the flow tube has retracted enough to allow the flapper valve to begin to enter the inner working bore it may be assisted by the flowing fluid. This assistance would be due to flowing fluid and/or hydrocarbons, which would force the flapper valve in a clockwise direction as seen in Fig. 5. This assistance will rapidly decrease the closing time of the safety valve. In case of unexpected safety valve closure during flowing production, the flapper valve and flow tube assembly will experience extremely high accelerations. These accelerations must be accounted for and designed around.

CONCEPTUAL DESIGNS

Introduction

In order to widen the opportunities and possibilities for unrestricted solutions, three different conceptual designs will be presented in the following section. These designs rely on different parameters for operation; therefore, will be based upon three varying physical principles. This section will introduce and discuss each concept individually. A comparison of the three concepts will then be presented at the conclusion of this section in the form a design matrix, allowing a quick concise comparison.

The three concepts that will be presented in the report, which are modifications of current valve designs, are the following:

- Electric motor actuator with a solenoid latching detent.
- Linear actuation with a fluid reservoir locking valve.
- Rotational engagement with an electric locking clutch.

Concept #1: Electric motor actuator with a solenoid latching detent

Premise

The main idea of this concept begins with a focus on the flow tube movement. Three figures are presented to illustrate this concept. The first two figures, **Fig. 8** and **Fig. 9**, show the conceptual safety valve in an open and closed position respectively, without a surrounding valve body shell. These figures do not show all the details of the valve, but only those pertaining to the actuator assembly. The third figure, **Fig. 7**, details the latching mechanism discussed later in this section.

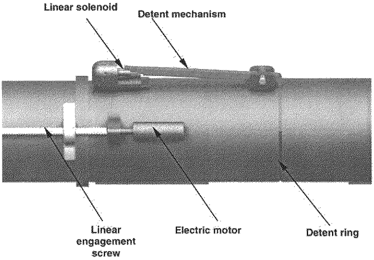

Fig. 7—Detail schematic of concept #1.

The main component of the actuator will be a long rotary screw, which is threaded through a flow tube engagement piece. This engagement piece would be constructed of similar geometry as the current hydraulic actuator piston contact zone. The rotational motion of the long lead screw will cause movement of the engagement piece either way depending on screw rotation direction. Thus in order to rotate the lead screw, an electric motor will be connected to the lead screw. This may require a gearbox to increase the output torque in order to operate the safety valve. However, this has yet to be determined at this stage, since this would rely on multiple items, such as screw pitch, type, and friction forces involved.

Now that there is a proposed method for retracting and extending the flow tube, through the use of an electric motor connected to a lead screw, a method for continuous reservoir access will be presented. In order to maintain low power requirements, a locking method has been devised as to minimize the steady state load. This method is as follows.

Once the lead screw has engaged the flow tube completely, a solenoid is in engaged into the detent mechanism, schematic shown in **Fig. 10**. This figure illustrates the forces required to hold the position of the flow tube, the required holding force value can be reduced with the addition of the detent lever shown in **Fig. 11**. This lever would act on the top portion of the detent, forcing the detent into the valley within the flow tube. This mechanism will restrict the motion of the flow tube, the lead screw will then be retracted into its original position, leaving the flow tube in the open position. In order to lower the energy requirements, a latching solenoid could be used. A latching solenoid uses a permanent magnet to maintain the desired position; therefore no electrical power would be required to maintain reservoir access. To allow the failsafe release, a low power capacitive circuit would be used to apply a small amount of stored power to the solenoid upon loss of control signal. This small electrical power would retract the solenoid from the magnets latching range and disengage the system completely.

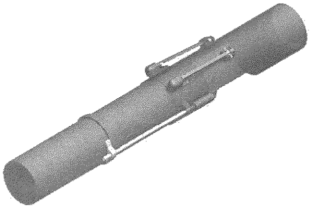

Fig. 8—Concept #1 (open position).

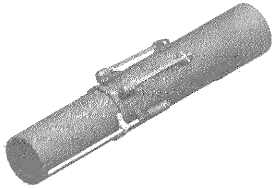

Fig. 9—Concept #1 (closed position).

The fail-safe mechanism will be the same power return spring as is used currently. The loss of a trigger signal would cause the capacitive circuit to retract the solenoid, allowing the flow tube to be retracted with the stored energy internal to the spring. Thus, no external energy sources will be required for closure of the safety valve.

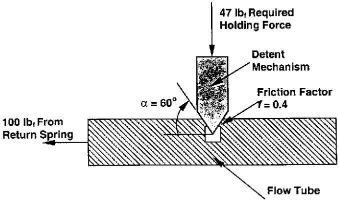

Fig. 10—Schematic of detent latching mechanism, graphically showing the location and magnitude of the required holding force to prevent movement of the flow tube. Details can be seen in Appendix B[7].

The process described, including all components would be duplicated at an offset angle to add redundancy to the system. The redundancy can be seen in Fig. 8 with the dual detent latching mechanisms. All components would be designed as if they are the only devices accomplishing the specified task. The result of this method would allow independent operation of each system, only if the opposing system has been successfully retracted, in the event on unsuccessful operation a mechanical release override may be designed into the screw mechanisms.

Preliminary calculations have been conducted on the detent latching mechanism. The result of these calculations can be found in the Appendix B. Both the detent mechanism and the detent lever have been estimated for working requirements, from these calculations it appears this concept should be feasible.

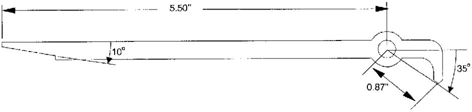

Fig. 11—Detent lever conceptual schematic.

If the detent lever has the dimensions and configuration as shown in **Fig. 11**, the required horizontal solenoid force to hold a 100 lb_f would be approximately 13 lb_f or 57 N. With this estimated force, it is clear from **Fig. 12**, that linear actuators available currently have the ability to produce the required forces.

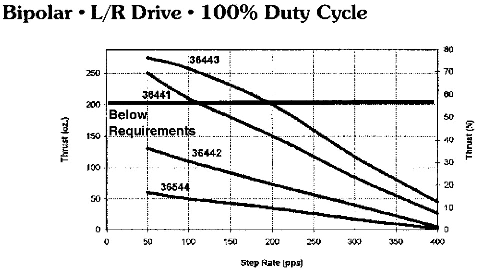

Fig. 12—Haydon Switch & Instruments linear actuator series 36000 step rate vs. thrust curves[8].

It should be reiterated once more, that these components will be exposed to extremely high temperatures, reaching as high as 350 °F. These temperatures will have to be accommodated by only using high temperature rated electrical equipment, those that meet IEEE class H temperature ratings, from manufacturers such as Haydon Switch & Instruments or BLP.

Critical Design Issues

With the electric conversion, there will be many critical parameters and issues that will need to be addressed. In presenting this concept, the following are the areas that will require critical attention:

- Minimize the required motor torque; therefore reducing the motor size.
- Prevent galling and contamination of the engagement threads.
- Minimize the detent rotation height, while maintaining an adequate retention force.
- Configuring a solenoid to latch with the required holding force.
- Possibly thermally cooling the electrical components by available means: thermoelectric coolers or miniature refrigeration system.

- Clutch mechanism within the flow tube engagement mechanism to allow consistent fail-safe operation.
- Reducing micro-wear at the tip of the latching detent during production, due to high frequency vibrations.

These issues will need to be fully developed and designed in order to accomplish the design task specified by FMC Kongsberg Subsea.

Concept #2: Linear actuation with a fluid reservoir locking valve

Premise

This concept will rely on a volume of fluid to maintain the position of the hydraulic piston while the flow tube movement will originate from an array of linear actuators. A simplified graphical schematic may be seen in **Fig. 13**.

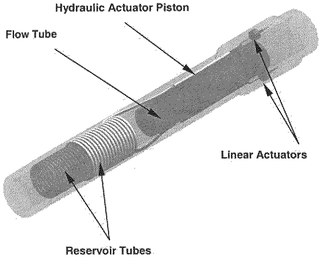

Fig. 13—Concept #2 schematic.

A similar operation, to what is being presented, can be found in any hydraulic ram or jack, where fluid is transported from one reservoir to another in order to change the position of an object. The only difference between the presented concept and a ram or jack is that the transfer of fluid from one reservoir to another will be completed by suction, similar to a syringe, from the movement of the hydraulic piston.

A reservoir containing an incompressible fluid will be inlayed into the outer wall of the safety valve; one similar to the coil tubing depicted in Fig. 13. This reservoir will be connected to the control port side of the hydraulic actuator piston. The piston will be made to current designs, with a few exceptions. The alterations to the current safety valve will be the installation of a fluid solenoid valve between the reservoir and the actuator, and the addition of an electric actuator to move the flow tube into position.

The fluid solenoid valve will be either designed or obtained to the following operational specifications:

- Application of power should result in closure of the valve.
- Loss of power should result in a fail-safe opening of the valve.
- Only require a small trigger signal to sustain closure.
  – this could be accomplished by designing a magnetic latching mechanism within the solenoid itself.

A simplified schematic of the placement and action of the fluid solenoid can be seen in **Fig. 14**. The electric solenoid will act against a return, resulting in a reliable fail-safe mechanism. A data sheet taken from BLP **(Fig. 15)** shows the force vs. stroke characteristics of the 124 Series solenoid[9]. If a spring force of 5 lb_f is reacting against the solenoid at the seated position, the solenoid force would be greater than the 5 lb_f or 2.23 kg_f used to retract the sealing needle from the seated position. A solenoid of this type would be able to overcome the spring force and seal off the two volumes. Another option could be the use of a piezoelectric actuator, in place of the solenoid.

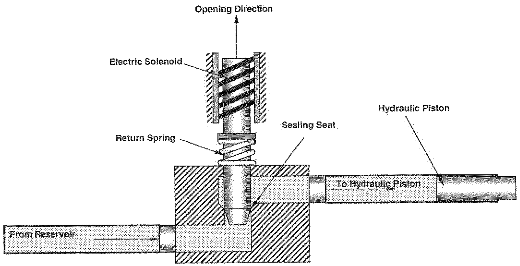

Fig. 14—Schematic of solenoid valve and positioning layout.

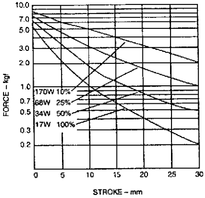

Fig. 15—BLP Black Kight Tubular solenoid series 124 force/stroke curve[9].

The electric actuator used to move the flow tube will be designed to the following specifications:

- Require less than specified current and voltage for operation.
- Operate in a multitude of contacting positions.

The actuation method may include an array of linear actuators such as those manufactured by Haydon. The typical curve for their 4600 Series actuator can be seen in **Fig. 16**[8]. As can be seen from this figure, large forces can be generated within these actuators, large enough to be suitable for safety valve use.

The combination of these elements will be completed with a digital or analog control system. The control system is required since it will be highly unlikely to completely seal the entire reservoir from the hydraulic piston; therefore, the volume holding the position of the flow tube will slowly decrease due to small leakage. This decrease in volume, while minimal, may cause unexpected safety valve closure. Thus, it is important that the electric actuator have the ability to reposition the flow tube to a desired range.

The control system will sense the flow tube position, from this position reading it will determine whether the flow tube is retracting from its desired position. The control system will only require minimal power requirements when the flow tube is in the desired range, but once the flow tube has retracted to an unsatisfactory distance, the controller will activate the electric actuator. Once the electric actuator begins to move the flow tube, the fluid solenoid will be opened allowing more fluid to be pulled out of the reservoir, again similar to the action of a syringe. Once the flow tube has been positioned, the controller closes the solenoid valve and regresses into a standby state sensing the flow tube position.

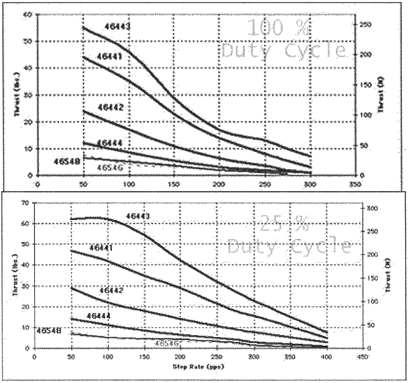

Fig. 16—Haydon linear actuator series 4600 thrust curves[8].

Critical Design Issues

The design issues that will require considerable attention for this concept are as follows:

- Sealing the fluid solenoid between the reservoir and the hydraulic piston volume.
- Reducing the expansion of the fluid due to large temperature variations.
- Minimizing the required steady state opening power.
- Control system able to withstand the specified temperatures, once again possibly requiring the use of thermoelectric coolers.
- Electric actuation method to move flow tube.

These design issues will be further developed and designed in order to fulfill the design challenge presented by FMC Kongsberg Subsea.

Concept #3: Rotational engagement with an electric locking clutch

Premise

This concept will rely on the following components, whose actions and purpose will be described in the following section:

- Low profile electric motor.
- Electromechanical brake.
- Low profile planetary gear reduction assembly.
- Speed limiting clutch.
- Cylindrical cam.

A simple conceptual schematic of this concept can be seen in **Fig. 17**. In this schematic, the positional relationship of the aforementioned components can be seen. The low profile electric motor will rotate at a high speed producing a small amount drive torque. This drive torque will be transmitted through the electromechanical brake. The brake will engage and disengage the rotational capabilities from the rest of the system with a low power electrical signal.

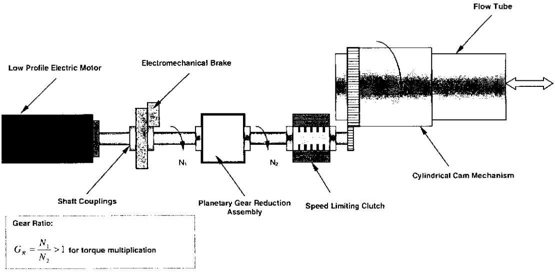

Fig. 17—Conceptual schematic for concept #3.

The electromechanical brake will control and hold the position of the flow tube. By engaging the brake with a low powered electrical signal, the brake will have the ability to maintain the position of the flow tube and all rotational components. The brake will not require a large amount of power to hold its position because the torque transmitted to the brake is low due to the location and gearing of the system. Therefore to release the flow tube, the brake would be disengaged and the flow tube would retract with the assistance of the power return spring, working on the same principle of current designs.

The torque from the electric motor will then be transmitted from the brake through the low profile planetary gear reduction assembly. This assembly will transfer the low torque, high speed input into a low speed, high torque output for axial movement of the flow tube. The speed limiting clutch is placed prior to engagement with the flow tube. The speed limiting clutch will be centrifugally engaged to add damping and rotational friction to the system to slow the angular velocity of the rotating components. This placement is crucial to slow the gear assembly and rotational system from overrunning when the power return spring retracts the flow tube.

When the power return spring retracts the flow tube, high speeds could be seen throughout the gear assembly, this situation could result in failure of the gears and other components. Thus, in order to prevent failure the speed limiting clutch would be installed nearest to the source of retracting force, and possibly disconnect the rotating components from the rest of the retracting system.

The cylindrical cam is used to transfer the rotational movement of the shaft assembly into a linear motion to extend the flow tube, a simple schematic of this can be seen in **Fig. 18**. Pins would be located along the outer circumference of the flow tube, engaging with helical groves within the inner surface of the cam mechanism. These locations could be reversed depending on the stresses involved.

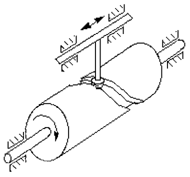

Fig. 18—Cylindrical cam schematic.

The results of the order of magnitude calculation on the gear ratio requirements of this system can be found in detail in Appendix B. The result of this calculation is shows if an electric motor capable of producing 1 oz-in of torque is used to move the flow tube, reacting with a 100 lb_f, from the return spring the gear ratio required would need be greater than 800:1.

Critical Design Issues

In order to successfully embody this design concept a few key issues will have to be addressed. The essential issue that must be addressed is the size constraint. The motor, gear assembly, brake, and clutch must be very small in cross-section, or diameter. This must be accomplished to allow installation within the walls of the safety valve. In order to the reduce the cross-sectional area of these components it may be necessary to lengthen the current dimensions of the safety valve to allow each component to expand axially.

To prevent failure of the gear assembly by experiencing high rotational speeds caused by the sudden and forceful retraction of the flow tube, the speed limiting clutch must be carefully chosen and designed. This clutch should be designed to dampen and create a frictional resistance approaching higher speeds, thus impeding any increase into an unwanted velocity range. The clutch may be designed to disengage from the cylindrical cam completely, rendering the rotational components unaffected by the sudden forceful retraction of the flow tube. However, the clutch should also remain unobtrusive when operating at much lower speeds, so the drag forces do not consume any appreciable amounts of power.

In order to meet the design specifications presented, these aforementioned issues must be addressed and effectively compensated.

Conceptual Design Evaluation

It will be the purpose of this section to briefly compare the three presented concepts in a clear and concise manner. Each concept is ranked according to a weighted criteria in **Table 2**. The design matrix is used to compare various conceptual designs according to specific function or critical indexes or requirements against the specified datum. The first presented concept has been chosen to be the datum for all comparisons. The matrix is organized to give a concise comparison of the three presented concepts to the following criteria against the datum:

Simplicity

In order to have a reliable and failsafe design, the greater number of components will increase the complexity; therefore, decreasing the reliability. Reliability is an important issue in this design, since the safety valve will not have the ability to be serviced at any short-term intervals, due to location inline with the tubing.

Steady State Power Requirement

Given the location and time required to sustain reservoir access, it has been decided that minimizing the power requirements will directly lead to a superior final design.

Lifetime

In order to avoid large production down times due to safety valve maintenance or replacement, the safety valve should be designed with lifetime greater then the expected useful production life of the reservoir. The lifetime may be assumed for comparison to be proportional to the number of wearable components and/or high stress loading mechanisms. Therefore, the preferable design will be one with fewer wearable components and low stress loading mechanisms.

Cost

As with any design solution, the overall cost of manufacturing and design is always a driving factor. Therefore, the preferable design will have fewer manufacturing specific details, such as costly tolerances or exotic materials.

TABLE 2 - CONCEPTUAL DESIGN COMPARISON MATRIX

Weight 100		Concept #1 **Electric motor actuator with a solenoid latching detent**	Concept #2 **Linear actuation with a fluid reservoir locking valve**	Concept #3 **Rotational engagement with an electric locking clutch**
30	Simplicity	DATUM	+	+
20	Steady State Power Requirement		S	+
20	Lifetime		-	+
30	Cost		S	S
	Total +	0	30	70
	Total S	100	50	30
	Total -	0	20	0

Legend:
+ The conceptual design should perform better than the datum
- The conceptual design should perform worse than the datum
S The conceptual design should perform equally well as the datum

As can be seen from the design matrix, the preferable design is concept #3. The lifetime and simplicity of this concept should easily be achieved with particular attention to the high stress areas caused when the safety valve is closed suddenly under the aid of reservoir pressure. After comparison of all concepts and discussion with FMC Kongsberg Subsea, the third concept shall be pursued.

Selected Detail Design

After meeting with FMC Kongsberg Subsea and discussing all the aforementioned concepts, the chosen design is the rotational engagement with an electromechanical locking brake. This concept has been chosen for the following reasons:

- No fluid containment issues.
- Low power requirements.
- Ease of installation within safety valve.
- Self-sustained actuator for hydraulic replacement.
- No alteration to current fail-safe mechanism.

The other concepts have been overlooked because of some key critical issues. The first concept, the latching solenoid detent, is unable to be completely fail-safe in the proposed method. This period is between the start of movement of the flow tube and the final positioning. During this period the thread engagement piece could possibly seize, resulting in failure of the safety valve. The addition of a clutch mechanism within the thread engagement ring could accommodate this but space would be the deterrent when considering this option. Due to this issue, the first proposed concept has been overlooked for further detailed design.

The second concept presented, the fluid solenoid valve had two critical issues that would be difficult to accommodate. The first issue is the large fluid expansion due to large temperature variations. This expansion could result in unwanted flow tube movement, which could result in unexpected operation. The last issue with the second presented concept is the need to contain hydraulic fluid. Due to the unavoidable occurrence of leakage, this concept could be rendered inoperable after a unknown period of time due to lack of operating fluid. Therefore, for these two critical issues this concept has been overlooked for further detailed design.

Continuing on to the next design phase with the third concept will entail addressing the following issues:

- Reducing the stress caused by high rotational speeds.
- Reducing the dimensions of rotating components to fit in specified area.
- Determining the cam dimensions to allow the system to back-drive.
- Preventing the system from retracting at unwanted speeds, i.e. add dampers or disengagement points within the rotating system.

During the next design phase, these issues will be addressed and accommodated, producing an electric actuator for use in a surface controlled subsurface safety valve.

EMBODIMENT DESIGN

Introduction

This portion of the report will cover the detailed description of the overall actuator operation, design requirements of key components, and descriptions of selected components. The embodiment design phase entails designing and detailing all key portions of the conceptually presented design of the previous section. The rotational engagement actuator with an electromechanical locking brake has been further developed and will be presented in the following section, along with recommended part selections.

Detail of Operation

An assembly drawing of the entire actuator assembly including an acceptable housing can be seen in sheet drawing number ESCSSV-A001, which can be found in Appendix C. Each component has been exploded to show their relative placement, while **Fig. 20** displays a transparent view of the actuator within the safety valve housing. Finally, a simplified operational schematic of the SCSSV can be seen in **Fig. 19**, which will be referenced in the following paragraphs detailing the actuator operation.

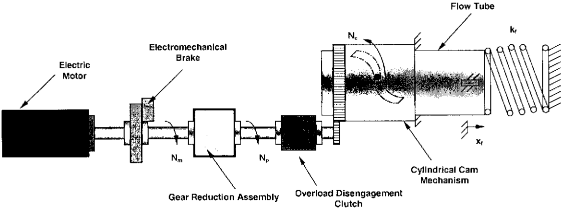

Fig. 19—Operational schematic of the electric SCSSV.

To operate the SCSSV from the closed position to the open, the pressure differential across the flapper should be lowered to a maximum of 500 psi. This is done to reduce the energy requirements to rotate the flapper valve to the open position.

The electric motor, which will transmit a low torque high speed rotation through the electromechanical brake into a gear reduction assembly, will be powered from a surface or subsea power source. The electromechanical brake will be disengaged in the opening sequence, and will only be relevant in the hold open sequence to be discussed later in this section.

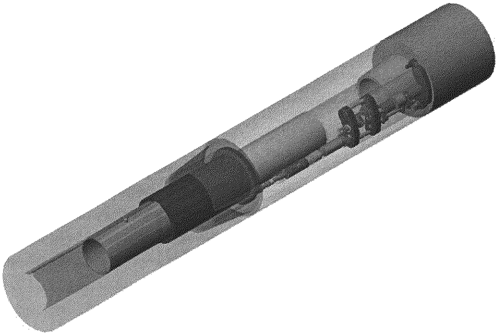

Fig. 20—Electric SCSSV overall assembled model.

The gear reduction assembly will convert the low torque, high speed input from the electric motor to a low speed, high torque output capable of operating the valve. A sheet drawing of the gear reduction assembly can be found in the Appendix C, ESCSSV-A004. From the output of the gear reduction assembly, the torque is transmitted through an overload clutch. The overload clutch is used to protect the gear reduction assembly and the cylindrical cam drive gear from damage during accidental closure in the presence of flowing hydrocarbons and/or gas. This situation may arise when the SCSSV is instructed to close when there is still a flow-rate within the bore, such as during the production stage, causing the valve to abruptly close at an extreme velocity due to the large momentum of the flowing fluids. This large energy must be dissipated and will occur at the weakest components; therefore, the clutch is used to remove any energy dissipation abilities from the actuator, focusing all dissipation to the flapper valve component itself. This is done to insure operation of the actuator and the valve since most valves of current technology are designed to withstand this closing force and thus will survive high velocity closure.

The torque will be transmitted from the overload clutch to the cylindrical cam drive gear, which rotates the cylindrical cam. The cylindrical cam has a threaded profile inlayed on the inner diameter of the cam. The cylindrical cam will transfer the rotational motion of the actuator to the required translational motion required to open the valve. The cam operates in conjunction with the flow tube, which has three pins projecting radially outward engaging with the inlayed profiles of the cylindrical cam. The rotation of the cam causes the pins of the flow tube to follow the profile, causing the flow tube to extend to the right as shown in **Fig. 19**. This operation is similar to a jack or power screw.

The flow tube will move against a large compression spring, which is used to close the SCSSV in any condition. Once the flow tube has been positioned, and the valve has been opened, the electromechanical brake is engaged and power supplied to the electric motor is removed with the use of an on-board high temperature control board. The electromechanical brake will now hold the SCSSV open, until any power loss or closure command is issued. Once a closure command or power loss is issued, the on-board control system transmits a release signal to the electromechanical brake allowing the entire system to retract with the aid of the power return spring.

Detail of Torque Overload Clutch Operation

A model of the torque overload clutch can be found in **Fig. 21**. The operation of the overload clutch[10] begins with the gearbox output torque being transferred from the input shaft to the input drive pin. The input drive pin rotates into the clutch sleeve, which is located with bearings on both shafts allowing free rotation and axial displacement from both shafts; therefore, the drive pins locate the clutch sleeve within the clutch. This interaction causes a frictional contact between the pin and sleeve. The friction force causes rotation of the clutch sleeve only if the torque being transmitted is lower than a pre-selected value. The stored spring force in the overload adjustment spring dictates this pre-selected value. Assuming the torque is at a low value, the sleeve will continue to rotate in sequence with the input shaft.

The output side of the sleeve will contact the output drive pin in the same manner as the input drive pin, both relying on the spring force biasing the pins to the lowest point on the profile. The output drive pin will be forced to following the motion of the sleeve, only if the torque applied to the shaft is below the pre-selected value. Once the torque overload value has been reached in either direction or side, both pins will push the clutch sleeve against the overload adjustment spring, causing the sleeve to retract, disconnecting the output drive pin from contacting the clutch sleeve. This operation can be seen in **Fig. 22**, where the output drive pin is clearly disengaged from the clutch sleeve. This disengagement allows the output shaft to rotate freely with a minimal drag resistance.

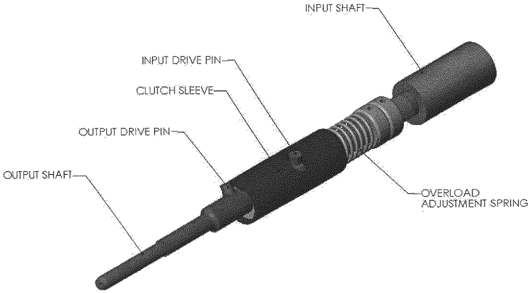

Fig. 21—Torque overload clutch representation.

A step-by-step illustration of the overload clutch disengagement can be seen in **Fig. 23**. In this figure, the engaged position can be seen at the top, while the disengaged position is seen at the bottom of this figure. The transition position, shown in the middle, displays the location of the pins as the clutch sleeve begins to disengage from the output drive pin.

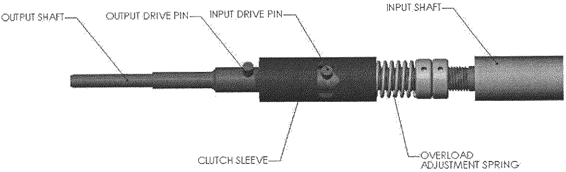

Fig. 22—Top view of torque overload clutch in disengaged position.

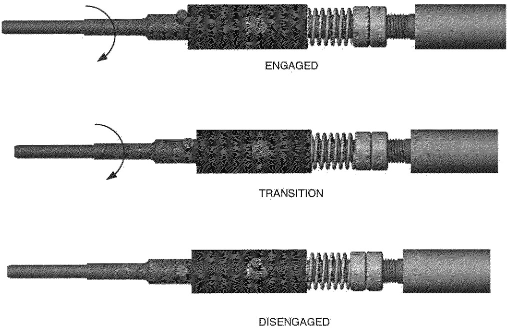

Fig. 23—Torque overload clutch disengagement illustration.

In order to reengage the overload clutch, the input shaft must be reversed allowing the input drive pin to retract into the lowest recess. Once the pin retraction occurs the overload clutch is re-activated. The advantage of this type of clutch as opposed to a slip or other type of disengagement clutch is once an overload occurs reengagement is not initiated until done so by the input shaft. This method of disengagement prevents any reengagement overloads that cause a cyclical fatigue load. A cyclical fatigue load will quickly deteriorate most components to failure. The corresponding drawing for the torque overload clutch can be found in the Appendix in ESCSSV-A003.

Component Requirements

The following section will detail the requirements of key components, along with suggested components to use for optimum service. Each component will have the following requirements as set forth in the Need Analysis, taken from Table 1:

- Operate in temperature range between 20 – 350 °F.
- Require no more than 48 VDC.
- Total of electrical load no more than 4300 W.

Each component will be discussed and detailed as required. More detailed information can be found in the Design Documentation section with the Appendix.

Actuator Housing

The actuator housing has been chosen to be installed inline with 4-½ tubing; therefore, the housing dimension should be no larger in outer diameter than 7.875 inches and no smaller in inner diameter then 3.562 inches as dictated by API standards[11]. The material for the housing should be a high yield, Hydrogen Sulfide resistant metal, such as AISI 8630. AISI 8630 has a yield strength of 93 ksi, along with a tensile strength of 110 ksi[12]. With this yield strength the required wall thickness of the housing to hold a 15 ksi internal pressure load can be found by using equation (1) for a thin-walled cylindrical pressure vessel[13].

$$(1) \qquad t = \frac{P_{\text{int}} d_{id}}{2(S - 0.6P_{\text{int}})}$$

where P_{int} is the internal acting pressure, d_{id} is the internal diameter, and S is the allowable stress within the housing. The allowable stress for this material can be found by using (2)[13].

$$(2) \qquad S = \frac{2}{3} 0.92\sigma_y$$

where σ_y is the material yield strength. Therefore, the allowable yield strength of AISI 8630 is:

$$(3) \qquad S = \frac{2}{3} 0.92(93ksi) = 57.04ksi$$

using result (3) in equation (1), the required wall thickness is:

$$(4) \qquad t = \frac{(15ksi)(3.562in)}{2((57.04ksi) - 0.6(15ksi))} = 0.56in$$

Therefore, the housing must have a continuous outer diameter of 4.67 inches to withstand the internal acting pressure, when the inner diameter is 3.562 inches. This leaves an annular thickness of 1.60 inches to place the actuator within the wall of the housing. Thus, as a conservative dimension, all actuator components should be no greater than 1.40 inches in height or diameter.

In order to prevent wall collapse, there should be a pressure equalization device that will equalize the actuator housing bore pressure with the surrounding environments. This device should operate without allowing fluid or material transfer into the actuator housing bore. To accomplish pressure equalization, an incompressible fluid must be present in the actuator installation bore to allow a variation in pressure due to a volume change; therefore, an incompressible fluid such as silicone or any oil that is a dielectric should be used. The volume change will be achieved through the use of a diaphragm acting as a barrier between the outside environment and the internal actuator installation bore. As the external pressure increases, the diaphragm will contract into the actuator installation cavity, causing a reduction in volume and an increase in internal pressure. The reverse is true if the external pressure is lowered, then the diaphragm expands away from the actuator installation bore increasing the volume and decreasing the pressure.

The presence of an incompressible fluid within the actuator installation bore will aid in extending the lifetime of the actuator components by lubricating all the mechanical components. If chosen correctly the fluid will lubricate the gear train, while also allowing even temperature variations preventing uneven expansion between interacting components. Due to the dielectric properties of the fluid, all electrical components should be unaffected in operation.

Electric DC Motor

The electric motor has been selected according to the dimensional and temperature constraints, thus the use of a small low powered motor operating through a large gear reduction assembly. The electric motor must develop a minimum of 0.125 in-lbf of torque.

A high temperature DC motor has been chosen as model number S/V-LD011 from Empire Magnetics Inc[14], **Fig. 24**. This motor develops 0.125 in-lbf (~2 oz-in) in the middle power band as can bee seen in **Fig. 25** when wired in series.

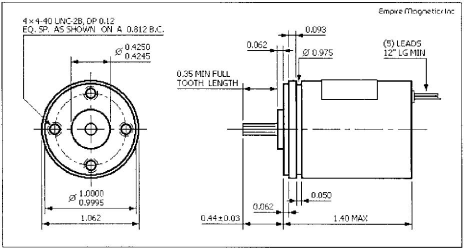

Fig. 24—Empire Magnetics extended temperature DC motor, model no. S/V-LD011[15].

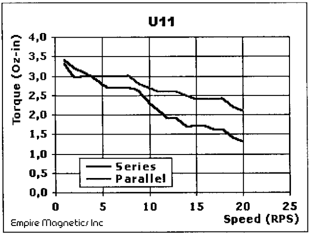

Fig. 25—Electric motor torque vs. speed[16].

The maximum outer diameter of this motor is 1.062 inches; thus it will be suitable to be installed within the limited annulus area.

Electromechanical Brake

The electromechanical brake has been selected to require the least power requirements in the engaged mode, while also having a low profile. Electroid Company has a collection of bi-stable brakes that require no current to operate, requiring only a 100 millisecond DC current pulse of proper polarity to either engage or disengage the brake. This property is an excellent choose for this actuator since the brake will be engaged for a long lengths of time, resulting in large power savings. Electroid has proposed to manufacture a special high temperature version of their BSB-1 model for use within this actuator. The holding power of this brake is 0.75 in-lbf, which is larger than the output of the electric motor adding a majority of control area.

Planetary Gearbox

The planetary gearbox will multiply the electric motor torque with a ratio of 25:1, obtaining an averaged output torque of 3.125 in-lbf. To successfully transmit the rotational energy, the gearbox should be efficient. The gearbox should be at least 90% efficient in operation to maintain actuator performance. A suitable gearbox is one manufactured by Empire Magentics, model number 11P:25[17], seen in **Fig. 26**.

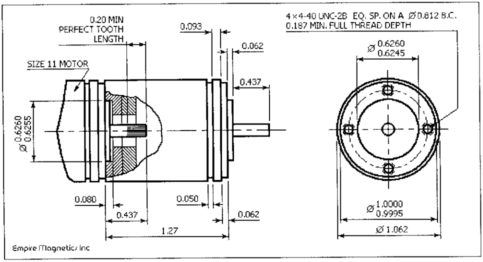

Fig. 26—Empire Magnetics planetary gearbox, model no. 11P:25[18].

Gear Reduction Assembly

A figure displaying the important information of the primary gear reduction assembly can be seen below in **Fig. 27**. The gear assembly will consist of six gears, resulting in a combined reduction ratio of 16:1. The final gear rating will limit the maximum rated output torque of the gear reduction assembly, which is 89 in-lbf of torque. This rated torque is well above the actual applied torque of 50 in-lbf as seen in Fig. 27. The gear reduction assembly will require bearings capable of operating up to an ambient temperature of 350 °F, Boston Gear's Bost-Bronz Oil-Impregnated Sintered Bronze Bearings[19] should be sufficient for these purposes. Although, oil-impregnated bearings are not required since the surrounding environment will be filled with a lubricant.

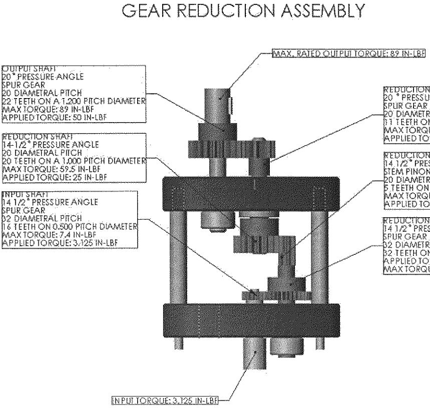

Fig. 27—Schematic representation of the gear reduction assembly.

The shafts within the gear reduction assembly will be experiencing a dominant shear torque, as opposed to a combination of shear and axial stresses due to the use of spur gears. Therefore, to determine the required shaft diameter due to an applied torque, the following equation can be used (5)[20].

$$(5) \qquad d_{\min} = \left(32 \frac{n\tau}{\pi\sigma_y}\right)^{\frac{1}{3}}$$

where d_{min} is the minimum shaft diameter, n is the requested safety factor, τ is the applied torque and σ_y is the material yield strength. Therefore, if $n = 2$, $\tau = 50$ in-lbf, and $\sigma_y = 50{,}000$ psi, then the required minimum shaft diameter is (6):

$$(6) \qquad d_{\min} = \left(32 \frac{(2)(89in - lbf)}{\pi(50{,}000 psi)}\right)^{\frac{1}{3}} = 0.33in$$

The gear reduction assembly will operate at relatively low speeds; therefore, surface velocity calculations will not be required in selecting bearings. The bearings will be used for locational purposes only, not for load bearing purposes.

All gears used in this assembly shall have a minimum rated torque as shown in **Fig. 27**. The specifications set forth in the figure are a minimum in order to operate without failure. Each gear during selection should be compared to the max torque section in each note associated with the gears.

Torque Overload Clutch

The torque overload clutch will be designed to disengage the entire rotational system at a torque of 80 in-lbf. This torque is chosen to be slightly less than the rated output torque of the gear reduction assembly in order to prevent premature wear or failure of any component. From the detailed derivation of the clutch functionality, a relationship between the applied torque and the required adjustment spring force to maintain transmission has been determined in the Design Documentation found in the Appendix, summarized in (7). A free body diagram of the clutch sleeve and drive pin can be seen in **Fig. 28**. This figure references the variables used in the following equations for visual reference.

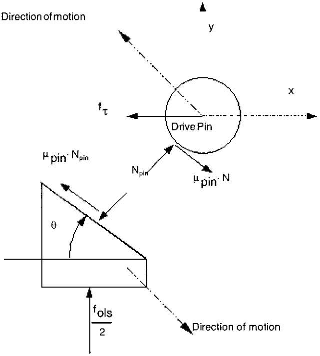

Fig. 28—Free body diagram of the drive pin and clutch sleeve interaction.

$$(7) \qquad f_s = -4\frac{\tau_{\max}}{d_{pin}}\left(\frac{-\cos(\theta)+\mu_s\sin(\theta)}{\sin(\theta)+\mu_s\cos(\theta)}\right)$$

where f_s is the required spring force, τ_{max} is the applied torque, d_{pin} is the drive pin contact diameter, θ is the incident angle of the pin and indexed clutch sleeve, and μ_s is the static friction between the clutch sleeve and drive pin. If τ_{max} = 80 in-lbf, d_{pin} = 0.9 in, θ = 70 deg, and μ_s = 0.15 (mild steel on mild steel when lubricated), then the spring force should be less than:

$$(8) \qquad f_s = -4\frac{80in-lbf}{0.9in}\left(\frac{-\cos(70)+(0.15)\sin(70)}{\sin(70)+(0.15)\cos(70)}\right) = 72lbf$$

The drive pins should be manufactured with the intent to create a rough surface finish. The higher the coefficient of friction between the pins and the clutch sleeve, the lower the spring force must be to prevent disengagement.

The input and output shafts of the overload clutch assembly will be restricted from relative axial movement. This restriction will be accomplished by restricting the output shaft with a spring-loaded c-clip. The c-clip will be installed against a flanged bearing, located in the inner diameter of the input shaft, allowing a low resistance to relative rotational displacements, but preventing any axial displacement. The details of this setup can be seen in ESCSSV-A003.

The output shaft must seal against a 15 ksi differential in order to prevent debris and other fluids from coming in contact with the actuator components. Thus, a dynamic rod seal has been chosen from Parker Seals[21]. The seal belongs to their FlexiSeal series, specifically part number VS-11-3-109-S-106. This part number has been selected using Parker Seals three step approach. The first step is to determine the required shaft hardness for the given temperature range. The suitable temperature range for this application is between –250 to 575 °F. From the temperature range, the minimum shaft surface hardness from the Parker Seal literature is 35 Rockwell C; therefore, the shaft should be manufactured out material with this surface hardness such as aged hardened INCONEL 725. The last step in seal determination is to determine the dynamic surface speed on the sealing surface. This can be found by using equation (9).

$$(9) \qquad V_s = \frac{\pi d_s N_p}{12}$$

where d_s is the shaft diameter, N_p is the shaft angular velocity in RPM. For the output shaft of the overload clutch, the shaft diameter, d_s = 0.313 inches and N_p = 2200 RPM during an extreme closure rate, thus the shaft surface velocity is:

(10) $$V_s = \frac{\pi(0.313in)(2200RPM)}{12} = 15\frac{\text{ft}}{\text{min}}$$

Thus, with these criteria the required material for the FlexiSeal is Parker Seal material #106, which is a Polymer/PTFE (Polytetrafluoroethylene).

Cylindrical Cam

The rotational energy from the gear reduction and torque overload assembly will be transfered through a spur gear acting on a meshing gear affixed to the cylinder itself. The gear reduction ratio from the output of the gear reduction assembly to the cam, should be no less than 9.15:1. This final gear reduction gives an overall rotational gear reduction of (11):

(11) $$GR = 50 \times 16 \times 9.15 = 7320:1$$

The rotational motion is transferred into a linear extension by means of inlayed threaded profiles that engage with pins extended from the surface of the flow tube. The flow tube is restricted from rotational motion; therefore, behaves similar to a nut and bolt. The bolt corresponding to the flow tube, and the nut is the cylindrical cam. Their interaction will produce the required linear motion to operate the safety valve.

The inlayed profiles within the cylindrical cam, seen in **Fig. 29**, must have the following properties:

- Smooth radii on all edges to prevent seizure.
- Tight tolerance between O.D. of flow tube and I.D. of cam.
- Pitch of the cylindrical cam must be greater than or equal to 4 inches per revolution.
- Teflon coating used on threads, along with the flow tube pins, resulting in a static/dynamic coefficient of friction <= 0.1.

The pitch is key for proper operation of the safety valve, if the pitch is to low, the power return spring may not be able to retract the system. If the pitch is to high, there might not be enough torque to rotate the cam and extend the flow tube. Thus, it is pertinent that any changes in cam pitch should require an in depth evaluation of all components of the safety valve.

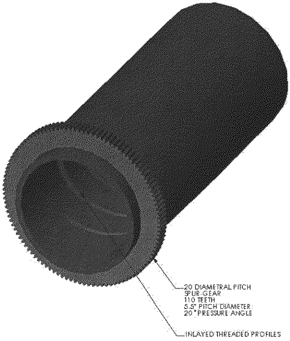

Fig. 29—Model representation of the cylindrical cam.

Flow Tube with Cam Pins

The flow tube will be a thin-walled non-pressure retention tube that will have three pins extending from the outer diameter that will contact the cylindrical cam, along with a single pin extending from the outer diameter for anti-rotation purposes, **Fig. 30**. The pins should be Teflon coated to reduce surface friction between components. The outer surface of the flow tube should also be Teflon coated as well, reducing rotational resistance.

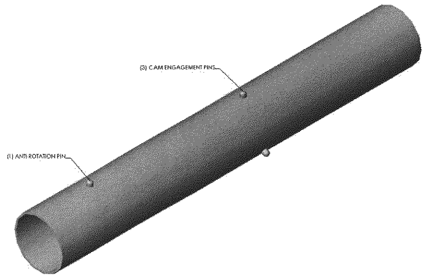

Fig. 30—Model representation of flow tube with cam pins.

The internal diameter of the flow tube should be no less than 3.562 inches, while the outer diameter should be any typical wall thickness for pipe dimensions, for instance in this design case the outer diameter is 3.80 inches.

The dimensions of the cam pins should be determined using the following equation, (12), assuming that the pins are loaded similarly to a cantilever beam[20]:

$$\sigma_b = \frac{64 N_p h_{pin}}{4\pi d_{pin}^3} \tag{12}$$

Therefore, if the material yield strength is 40 ksi, a safety factor of 2, a pin reaction force $N_p = 340$ lbf, height of the pin $h_{pin} = 0.4$ in, then the minimum required diameter of the pin should be $d_{pin} = 0.325$ in.

Return Spring

The power return spring in previous safety valves has been selected to suit to the depth of installation, allowing the proper closure pressure, with the replacement of hydraulic actuation with electrical, this is no longer necessary. The only criteria for the return spring is that it exerts enough force to close the safety valve in all conditions. For this valve, a spring that exerts a 300-lbf resultant force at approximately 8 inches will be suitable at any depth.

High Temperature Control Box

The electric motor and electromechanical brake will require some means of electronic control. In order to reduce the number of wires required extending to the safety valve from the surface, a simple control scheme has been devised. This scheme consists of a common ground wire, a signal return wire, and a power signal wire. The polarity of the power signal will dictate the response and operation of the safety valve. A flow diagram of the control system can be seen in **Fig. 31**

From Fig. 31, the presence of a positive polarity power signal will trigger the opening sequence of the valve, while negative polarity will engage the brake. Although it should be noted, that any lose in the power signal causes a direct closure of the safety valve.

The control box should be constructed of components, which operate within a temperature range of 20 to 350 °F. Also, high vibrations can be present during operation; therefore, this device should not be susceptible to these vibrations. Hybrid Design Associates manufactures high temperature memory modules and controllers and would be an exceptional source for the electronic controller.

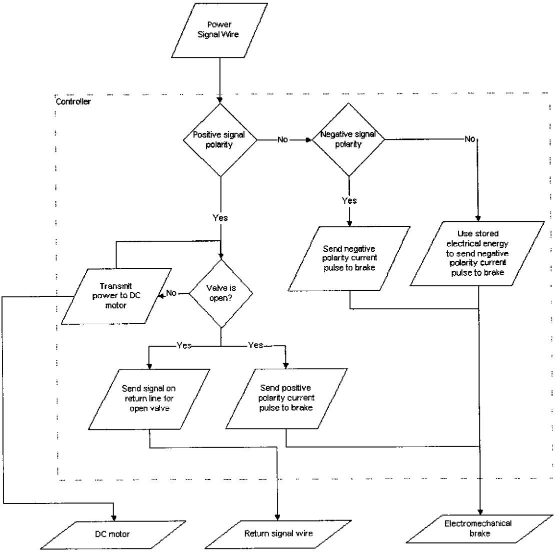

Fig. 31—Controller logic flow diagram.

Manufacturing and Materials

This section will summarize the requirements and recommendations for each component. The requirements for yield strength and possible manufacturing techniques for each part can be found in **Table 3**.

Part	Required σ_y (ksi)	H_2S Service	Manufacturing Possibilities
Actuator Housing	93	Y	Forged AISI 8630, machined
Lower ESCSSV Housing	93	Y	Forged AISI 8630, machined
Cylindrical Cam	50	Y	Investment cast
Flow Tube with Cam Pins	50	Y	Machined
Actuator Plug Cap	40	N	Machined
Upper ESCSSV Connection	93	Y	Forged AISI 8630, machined
Actuator Alignment Shaft	40	N	Machined
All Alignment Plates	30	N	Machined or cast
Electric Motor Drive Shaft	50	N	Machined
Overload Clutch Output Shaft	50	Y	Machined aged hardened Inconel 725
Overload Clutch Input Shaft	50	N	Machined
Indexed Clutch Sleeve (Both Pieces)	50	N	Machined
Overload Clutch Shaft Coupler	60	N	Machined
Overload Adjustment Nut	40	N	Machined
Bushing Alignment Block Spacer	30	N	Machined
Gear Reduction Assembly Output Shaft	50	N	Machined
Gear Reduction Intermediate Shafts	50	N	Machined
Upper & Lower Bearing Alignment Bushings	30	N	Machined or cast

TABLE 3—MANUFACTURING AND MATERIAL RECOMMENDATION AND REQUIREMENTS

Valve Performance Specification Claims

The electric safety valve performance claims will be presented in the section. These values have been estimated from the operating properties set forth in the previous sections, along with other environmental factors associated with the safety valve.

The first performance evaluation to be considered is the amount of energy required to open the safety valve against a maximum 500 psi pressure differential. This will be estimated by determining the potential energy stored within the return spring, then this energy divided by the overall system efficiency will result in an electrical energy requirement.

The stored energy change within the return spring can be defined:

$$E_{\text{spring}} = \frac{1}{2}k\left((x_f + x_i)^2 - x_i^2\right) \tag{13}$$

where x_f is the final flow tube position, x_i is the spring pre-load distance, and k is the spring constant. If the spring constant is 25 lbf/in, the pre-load distance is 4 inches, and the flow tube travels 8 inches, then the energy stored within the spring is (14):

$$(14) \qquad E_{\text{spring}} = \frac{1}{2}(25lbf/in)\left((8in + 4in)^2 - (4in)^2\right) = 1600lbf \cdot in = 180.78J$$

Therefore, if the overall system is $\eta = 60$ % efficient taking into account the large inefficiencies in the conversion of electrical energy to mechanical, then the electrical energy required to open the safety is (15):

$$(15) \qquad E_e = \frac{E_{\text{spring}}}{\eta} = \frac{180.78J}{60\%} = 301J$$

With the required energy (15), an estimated time required to open the safety can be used to determine the power that must be present in the electrical umbilical. To determine the time required to open the safety valve, the number of rotations required for the motor to open the valve is required. This value has been determined by evaluating the overall system ratio, resulting in 7337 revolutions from the closed to open position. If the motor speed is estimated from the torque curve, Fig. 25, to approximately $\omega_m = 12$ RPS, then the time required to open the safety value is:

$$(16) \qquad t_{\text{open}} = \frac{\theta_{\text{m}}}{\omega_{\text{m}}} = \frac{7337rev}{12RPS} = 10.2\min$$

Therefore the average power required to open the valve is defined as (17):

$$(17) \qquad P_{\text{open}} = \frac{E_{\text{e}}}{t_{\text{open}}} = \frac{301J}{10.2\min} = 0.5W$$

This value is relatively low since there is such a large time for closure due to the limited size of the electric motor. In order to allow the valve to operate quicker the motor should be replaced with one capable of delivering a larger torque output. But as of this case study, the motor presented appears to be suitable and obtainable.

A summary of the performance criteria of the developed electric SCSSV can be found in **Table 4**.

TABLE 4—ELECTRIC SCSSV PERFORMANCE CAPABILITIES

Parameter	Value	Units
Installation tubing size	4-1/2	in
Temperature range	20-350	°F
H_2S Certified	Yes	
Working reservoir pressure	15	ksi
Min I.D.	3.562	in
Max O.D.	7.875	in
Hold open energy requirement*	0~0.2	J
Opening time	10.2	min
Opening power requirements	0.5	W
Energy required to open	301	J

* Due to no current requirements on electromechanical brake in the engaged mode but small trickle current in order to maintain controller operation

CONCLUSION

Upon completion of this extended case study a feasible method for electronic conversion of an SCSSV has been developed. The proposed method combines a small low powered DC electric motor, bi-stable electromechanical clutch, low-profile gear reduction assembly, torque overload clutch, and a cylindrical cam mechanism to produce a linear force, which is required to open a standard flapper type safety valve. This method has the following proposed performance capabilities, summarized from:

- Operate in an environmental temperature range of 20 to 350 °F.
- 0~0.2 W energy requirement to sustain an open position of the valve.
- Opening time of approximately 10 minutes.

With these outstanding capabilities, there are a few issues that must be addressed to determine actual and proper operation. These issues are:

- Actual efficiency of the cylindrical cam mechanism.
- Reliability of torque overload clutch disengagement point.

The cylindrical cam mechanism may have to high of an energy loss to be effective; therefore, a prototype of this mechanism should be considered and tested to determine actual operation. If the efficiency of this device is to low, other alternatives should be considered. These alternatives may be the use of ball bearings at the contact locations of the pins and the cam, or possible a bronze bearing. These alternatives have not been selected since they add complexity and parts to the system. Since this safety valve must be reliable, a low part count and simple system approach has been taken.

The torque overload clutch also has been analyzed in a pure theoretical manner. This analysis has showed a high sensitivity to frictional variations. Due to this sensitivity a prototype of this mechanism should also be considered to test the actual sensitivity of this device.

In completion of the detailed design phase of the electric actuator for a typical safety valve, the design challenge has been meet. All requirements have been achieved in a satisfactory manner, with minimal but pertinent empirical testing issues. The proposed actuator replacement operates with minimal power requirements for sustained access along with low energy requirements to open the flapper valve. It is the recommendation of this report that this engineering solution be utilized in future production safety systems when considering electric actuators for a surface controlled subsea safety valve.

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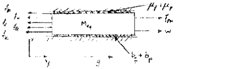
APPENDIX A—HYDRAULIC SCSSV MODEL

HYDRAULIC ANALYSIS (SCSSV)

FIG 1: SIMPLIFIED SCHEMATIC OF FLOWTUBE & ACTUATOR →

FIG 1



WHERE:

f_V → VALVE REACTION FORCE → $\begin{cases} f_V & 0 \le X_p < \bar{X} \\ 0 & X_p \ge \bar{X} \end{cases}$ WHERE $\bar{X}$ IS A GEOMETRICAL CONSTRAINT TO BE DEFINED LATER...

f_{PA} → ACCUMULATOR REACTION FORCE ①

f_d → VISCOUS DRAG FORCE ②

f_L → POWER RETURN SPRING FORCE ③

f_{fr} → FRICTIONAL DRAG FORCE ④

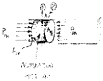
f_H → HYDRAULIC CONTROL FORCE ⑤

W → EQUIVALENT WEIGHT ⑥

DEFINITIONS: →

f_V → UNKNOWN VALVE-FLOWTUBE REACTION FORCE

① f_{PA} → ASSUMING IDEAL GAS MODEL FOR NITROGEN WITH AN INITIAL PRECHARGE PRESSURE (P_i) AND VOLUME (V_{INIT}), THE TOTAL REACTING PRESSURE WILL BE: →



① - INITIAL VALUES
② - NEW POSITION/VALUES

$$V_1 = V_{INIT}$$
$$V_2 = V_{INIT} - A_c \cdot x_p$$

IDEAL GAS MODEL:

$$P_1 V_1 = mRT = P_2 V_2$$

$$P_2 = \frac{P_1 V_1}{V_2} = P_{INIT}\left(\frac{V_{INIT}}{V_{INIT} - A_c\, X_p}\right) = P_2 = P_{PA}$$ P_{PA}

$$\text{THUS } f_{PA} = P_{PA} \cdot A_{CA} = \boxed{P_i\left(\frac{V_{INIT}}{V_{INIT} - A_c X_p}\right) \cdot A_{CA}}$$ f_{PA}

AND $A_{CA} = \frac{\pi}{4} D_{AP}^2$ WHERE D_{AP} → ACTUATOR PISTON DIAMETER ON ACCUMULATOR SIDE.

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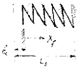
② $f_d \rightarrow \boxed{f_d = \dot{x}_f (b_f + b_p)}$ ← f_d

WHERE $b_f \rightarrow$ VISCOUS DAMPING DUE TO FLOW TUBE INTERACTIVE WITH EXTERIOR WALLS AND POSSIBLE PRODUCTION FLUID.

$b_p \rightarrow$ VISCOUS DAMPING DUE TO ACTUATOR PISTON WITH EXTERIOR WALLS.

③ $f_k \rightarrow f_k = k_s X_{s}$



WHERE → $L_s \rightarrow$ NATURAL SPRING LENGTH

$d_s \rightarrow$ INSTALLED COMPRESSION LENGTH

$f_k = k_s X_f + d_s k_s = \boxed{k_s (d_s + x_f) = f_k}$ ← f_k

④ $f_{fr} \rightarrow f_{fr} = (\mu_f \cdot N_f + \mu_p N_p)$ WHERE $\mu_f \rightarrow$ COEFFICIENT OF FRICTION BETWEEN WALL AND FLOW TUBE

$\mu_p \rightarrow$ COEFFICIENT OF FRICTION BETWEEN WALL AND ACTUATOR PISTON

$N_f \rightarrow$ CONTACT FORCE BETWEEN FLOW TUBE AND WALL

$N_p \rightarrow$ CONTACT FORCE BETWEEN ACTUATOR PISTON AND WALL

$\boxed{f_{fr} = \mu_f N_f + \mu_p N_p}$ ← f_{fr}

[illegible] IF ASSUME [illegible] MOVEMENT $\frac{d}{f_{fr}} \neq 0$

⑤ $f_{ph} \rightarrow$ ASSUMING SMALL VOLUME ADDITION [illegible] WILL REMAIN AT CONTROLLED. THUS $f_{ph} = f(t)$ AND NOT x_f DEPENDENT

$\boxed{f_{ph} = P_{cv}(t) \cdot A_{ap}}$ ← f_{ph}

WHERE $A_{ap} = \frac{\pi}{4} D_{ap}^2$ $D_{ap} \rightarrow$ ACTUATOR [illegible] [illegible]

⑥ $W \rightarrow \boxed{W = g(m_f + m_p)}$ ← W

WHERE $m_f \rightarrow$ MASS OF FLOW TUBE

$m_p \rightarrow$ MASS OF ACTUATOR PISTON.

3/4

THE EQUATION OF MOTION RESULTING FROM FIG 1.

$$\sum F_{x_f} \xrightarrow{+} = M_{eq}\ddot{x}_f = f_{m} + W - \{ f_{PA} + f_v + f_d + f_{fc} + f_{k} \} \qquad (1)$$

THE UNKNOWN VARIABLE IN (1) IS (f_v) → SOLVE FOR f_v

$$f_v = f_{pr} + W - \{ M_{eq}\ddot{x}_f + f_{PA} + f_d + f_{fc} + f_k \} \qquad (2)$$

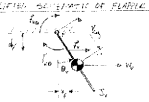
REPLACING f TERMS WITH DEFINITIONS.

$$f_v = P_{pr}(t)\cdot A_{cp} + M_{eq}g - \Big\{ M_{eq}\ddot{x}_f + P_A \left(\frac{V_{ist}}{V_{ist} - A x_f} \right) A_{cA} + \dot{x}_f(b_f + b_t) + \mu_f N_f + \mu_p N_p \ldots + k_s(\delta_s + x_f) \qquad (3)$$

$f_v(x_f)$

FIG 2 SIMPLIFIED SCHEMATIC OF FLAPPER VALVE

FIG 2



WHERE: →

$f_{k\theta}$ → RETURN COIL SPRING TORQUE ①

$f_{b\theta}$ → VISCOUS DRAG TORQUE ②

W_v → WEIGHT TORQUE (NON-LINEAR) ③

J_v → MOMENT OF INERTIA OF VALVE

δ_θ → SPRING PRELOAD

DEFINITIONS →

① $f_{k\theta} = k_v(\theta_v + \delta_\theta)$ →

$f_{k\theta}$

WHERE k_v → COIL SPRING COEFFICIENT

② $f_{b\theta} = b_v\dot{\theta}_v$ ←

$f_{b\theta}$

WHERE b_v → VISCOUS DAMPING COEFFICIENT

EQUATION OF MOTION IN (θ_v) FROM FIG 2. →

$$\sum M_{cg} \circlearrowleft = J_v\ddot{\theta}_v = f_v\cdot d_{cg} + X_{cg}W_v\cos\theta_v - f_{k\theta} - f_{b\theta} \qquad (4)$$

REPLACING f TERMS WITH DEFINITIONS.

$$J_v\ddot{\theta}_v = f_v d_{cg} + M_v g X_{cg}\cos\theta_v - k_v\theta_v - b_v\dot{\theta}_v - k_v\delta_\theta \qquad (5)$$

SOLVING FOR (f_v) →

$$f_v(\theta_v) = \{ J_v\ddot{\theta}_v + b\dot{\theta}_v + k_v(\theta_v + \delta_\theta) - M_v g X_{cg}\cos\theta_v \} / d_{cf} \qquad (6)$$

$f_v(\theta_v)$

NOW HAVE $f_v(x_f)$ & $f_v(\theta_v)$, NOW NEED A KINEMATIC CONSTRAINT FROM FIG 2 :→

δ_{of} = CONSTANT & x_f → TRACKING VARIABLE

THUS: $\frac{x_f}{\delta_{of}} = \tan\theta_v$ OR $\boxed{x_f = \delta_{of}\tan\theta_v}$ ← KINEMATIC

IF $x_f = \delta_{of}\tan\theta_v$, WOULD LIKE TO SOLVE FOR x_f

$$\theta_v(x_f) = \tan^{-1}\left(\frac{x_f}{\delta_{of}}\right) \qquad (7)$$

$$\dot\theta_v = \frac{-\dot x_f/\delta_{of}}{1+\left(x_f/\delta_{of}\right)^2} = \frac{-\dot x_f}{\delta_{of}}\left(1+\left(x_f/\delta_{of}\right)^2\right)^{-1} \qquad (8)$$

$$\ddot\theta_v = \frac{-\ddot x_f/\delta_{of}}{1+\left(x_f/\delta_{of}\right)^2} + \frac{\dot x_f}{\delta_{of}}\left(1+\left(x_f/\delta_{of}\right)^2\right)^{-2}\left(\frac{\dot x_f}{\delta_{of}}\right)$$

$$\ddot\theta_v = \frac{-\ddot x_f}{\delta_{of}\left[1+\left(\frac{x_f}{\delta_{of}}\right)^2\right]} + \left(\frac{x_f}{\delta_{of}}\right)^2\frac{1}{\left[1+\left(x_f/\delta_{of}\right)^2\right]^2} \qquad (9)$$

PUT $f_v(\theta_v)$ → IN TERMS OF x_f WITH (7), (8), & (9)

$$f_v(\theta_v) \to \frac{1}{\delta_{of}}\left[J_v\left[\frac{\dot x_f^2}{\delta_{of}^2\left[1+\left(x_f/\delta_{of}\right)^2\right]^2} - \frac{\ddot x_f}{\delta_{of}\left[1+\left(x_f/\delta_{of}\right)^2\right]}\right] + b_v\frac{-\dot x_f}{\delta_{of}\left[1+\left(\frac{x_f}{\delta_{of}}\right)^2\right]} + k_v\delta_e + k_v\tan^{-1}\left(\frac{x_f}{\delta_{of}}\right)\ldots\right.$$

$$- m_v g k_a \cos\left(\tan^{-1}\left(\frac{x_f}{\delta_{of}}\right)\right) = f_v(x_f) \qquad (10)$$

EQUATING (9) & (10) WILL RESULT IN A 2^{ND} ORDER NON-LINEAR DIFFERENTIAL EQUATION IN (x_f) ... REDUCING THE ORDER TO TWO 1^{ST} ORDER EQUATIONS IT CAN BE SOLVED IN MATLAB.

USING MATLAB TO SOLVE THE VARIABLES.

$$\begin{cases} x_f = v \\ \dot x_f = \dot v \end{cases}$$

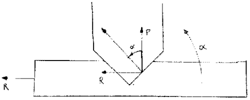
THE TWO EQUATIONS WILL BE

$$\begin{cases} \dot x_f = v \\ \dot v = \ldots \quad f(\theta_v) = f(x_f) \end{cases}$$

TOPS PRINTING

APPENDIX B—CONCEPTUAL DESIGN INFORMATION

Latching detent pin —



WHERE: R - IS THE HOLDING FORCE, OR IN THIS CASE THE LOWER RETURN SPRING FORCE AT THE OPEN POSITION.
P - IS THE REQUIRED HOLD DOWN FORCE.
f - COEFFICIENT OF FRICTION BETWEEN PIN AND SURFACE

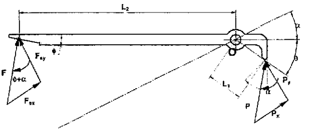
$$P = \frac{R}{\tan\alpha + f}$$

IF... $R = 100\ lb_f$, $\alpha = 60°$, $f = 0.4$ → $P = \frac{100\ lb_f}{\tan(60°) + 0.4} = \boxed{46.9\ lb_f = P_{60°}}$

IF... $\alpha = 45°$ → $P_{45} = \frac{100\ lb_f}{\tan(45°) + 0.4} = \boxed{71.43\ lb_f \quad P_{45°}}$

Concept # 1 (Calculations) | Tim Bartlett 1-17-05

Locking Detent Mechanism: →



Assume a slow rotational motion about point O →

Summation of moments →

$$\sum M_O = 0 = -L_1 P\cos\alpha\cos\theta - L_1 P\sin\theta\sin\alpha + F_s L_2 \sin(\phi+\alpha)\sin(\alpha) + \ldots$$
$$+ F_s L_2 \cos(\phi+\alpha)\cos(\alpha) = 0$$

Solving for F →

$$F_s = \left(\frac{L_1}{L_2}\right)\frac{\cos\theta\cos\alpha + \sin\theta\sin\alpha}{\sin(\phi+\alpha)\sin(\alpha) + \cos(\phi+\alpha)\cos(\alpha)} \cdot P$$

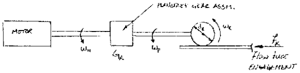
[illegible] → $F_{sx} = F_s\cos(\phi+\alpha)$ → [illegible]

$$F_{sx} = \left(\frac{L_1}{L_2}\right)\frac{[\cos\theta\cos\alpha + \sin\theta\sin\alpha]\cos(\phi+\alpha)}{\sin(\phi+\alpha)\sin(\alpha) + \cos(\phi+\alpha)\cos(\alpha)} \cdot P$$

[illegible]
$\theta = 35°$
$\phi = 10°$
$\alpha = 3°$
$P = 100\ lb_f$
$L_1 = 0.87$ in
$L_2 = 5.75$ in

$F_{sx} = 0.127\,P = \boxed{12.7\ lb_f = F_{sx}}$

For an increased mechanical advantage:
↑ $\frac{L_2}{L_1}$ while also an ↑ in [illegible]



IF THE POWER SPRING EXERTS $100\ lb_f = f_R$ ON THE ROTATING SYSTEM.

d_R WILL NEED TO BE ~ $\underline{d_R = 0.50\ in}$ ← FOR SPACE REQUIREMENTS.

IF ASSUME NO GEAR REDUCTION BETWEEN ω_p & ω_R → $\omega_p = \omega_R$

THEREFORE THE TORQUE TRANSMITTED THROUGH THE PLANETARY GEAR ASSM. →

$$T_p = d_R \cdot f_R = 0.50 \cdot 100\ lb_f\ in = 50\ in \cdot lb_f = T_p \text{ or } 800\ in \cdot oz$$

IF THE MOTOR COULD PRODUCE A NOMINAL TORQUE OF $T_M = 1\ oz \cdot in$ THE REQUIRED GEAR RATIO G_R WOULD BE... →

$$G_R \rightarrow \qquad T_M \cdot G_R = T_p$$

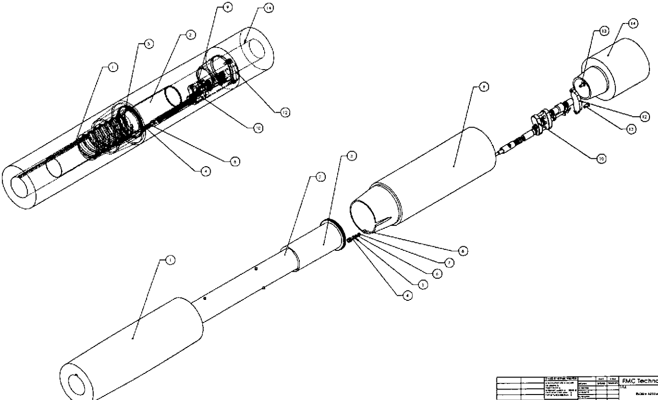
$$G_R = \frac{T_p}{T_M} = \boxed{800 = G_R}$$

APPENDIX C—SHEET DRAWINGS

TABLE 5—ATTACHED SHEET DRAWING LIST

Drawing No.	Description
ESCSSV-A001	Overall ESCSSV Assembly
ESCSSV-A002	Primary Actuator Assembly
ESCSSV-A003	Torque Overload Clutch Assembly
ESCSSV-A004	Gear Reduction Assembly

ITEM NO.	QTY.	MANUFACTURER	P/N	DESCRIPTION
1	1	FMC TECHNOLOGIES		LOWER ESCSSV HOUSING
2	1	FMC TECHNOLOGIES		FLOW TUBE WITH CAM PINS
3	1	FMC TECHNOLOGIES		CYLINDRICAL CAM
4	1	BOSTON GEAR	YA12 5/16 (46128)	20 DIAMETRAL PITCH SPUR GEAR W/ 12 TEETH ON 20 DEG P.A.
5	1	ROTORCLIP	HO-58	SPRING RETAINING CLIP
6	1	FMC TECHNOLOGIES		SEAL RETAINING WASHER
7	1	PARKER	VS 112 109-E-012	PIPE FLEXISEAL 3/8 I.D.
8	1	WM. BERG	701-00006-012	3/8 THIN WALLED BEARING
9	1	FMC TECHNOLOGIES		ESCSSV ACTUATOR HOUSING
10	1	FMC TECHNOLOGIES		PRIMARY ESCSSV ACTUATOR ASSEMBLY
12	1	FMC TECHNOLOGIES		ESCSSV ACTUATOR CAP PLUG
13	2	MISC		SAE 3/8 X 24 X 1/2 CAP HEAD BOLT
14	1	FMC TECHNOLOGIES		UPPER ESCSSV HOUSING CONNECTOR

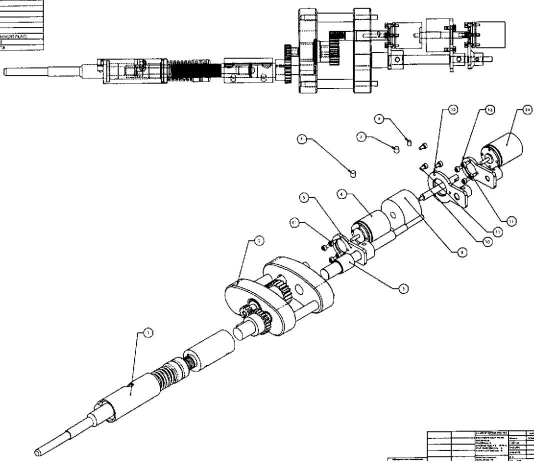


FMC Technologies

TITLE: ESCSSV ASSEMBLY

SIZE D | DWG. NO. ESCSSV ASSY | REV A

ITEM NO.	QTY	MANUFACTURER	P/N	DESCRIPTION
1	1	FMC TECHNOLOGIES		TORQUE OVERLOAD CLUTCH ASSEMBLY
2	1	FMC TECHNOLOGIES		ECOSSV GEAR REDUCTION ASSEMBLY
3	1	FMC TECHNOLOGIES		ALIGNMENT SHAFT
4	1	FMC TECHNOLOGIES		GEARHEAD ALIGNMENT PLATE
5	1	EMPIRE MAGNETICS	11P50	GEARHEAD
7	2	MISC		10-32 X 1/8 SET SCREW
8	1	ELECTROID		ELECTROMECHANICAL BRAKE
9	1	MISC		6-32 X 1/8 SET SCREW
10	1	FMC TECHNOLOGIES		ELECTRIC MOTOR DRIVE SHAFT
11	1	MISC		UNF 4-48 X 3/16 CAP HEAD BOLT
12	1	FMC TECHNOLOGIES		ELECTROMECHANICAL BRAKE ALIGNMENT PLATE
13	1	FMC TECHNOLOGIES		ELECTRIC MOTOR ALIGNMENT PLATE
14	1	EMPIRE MAGNETICS	11-SV-LOW	ELECTRIC HIGH TEMPERATURE MOTOR



FMC Technologies

TITLE:

PRIMARY ACTUATOR ASSEMBLY

SIZE D

DWG. NO.

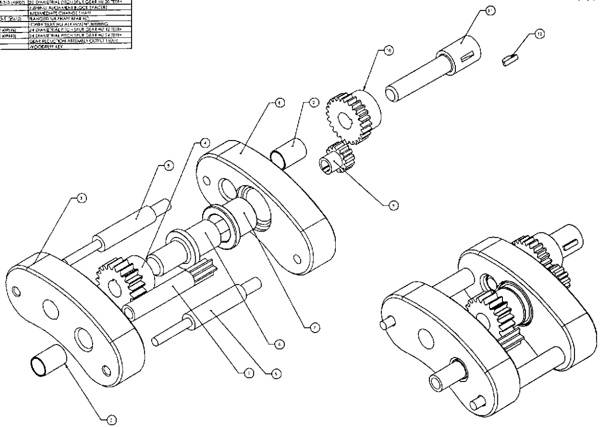
REV A

SCALE 1:1

WEIGHT

SHEET 1 OF 2

ITEM NO.	QTY.	MANUFACTURER	P/N	DESCRIPTION
1	1	BOSTON GEAR	NA8S [09654]	20 DIAMETRAL PITCH STEM PINION W/ 8 TEETH
2	2	BOSTON GEAR	B67-5 [13579]	5/8 SHAFT BEARING
3	1	FMC TECHNOLOGIES		UPPER BEARING ALIGNMENT BUSHING
4	1	BOSTON GEAR	NA20B-3/8 [46007]	20 DIAMETRIAL PITCH SPUR GEAR W/ 20 TEETH
5	2	FMC TECHNOLOGIES		BUSHING ALIGNMENT BLOCK SPACERS
6	1	FMC TECHNOLOGIES		INTERMEDIATE CHANGE SHAFT
7	1	BOSTON GEAR	FB-012-5 [36612]	FLANGED 5/8 SHAFT BEARING
8	1	FMC TECHNOLOGIES		LOWER BEARING ALIGNMENT BUSHING
9	1	BOSTON GEAR	H2412 [09576]	24 DIAMETRIAL PITCH SPUR GEAR W/ 12 TEETH
10	1	BOSTON GEAR	H2424 [09610]	24 DIAMETRIAL PITCH SPUR GEAR W/ 24 TEETH
11	1	FMC TECHNOLOGIES		GEAR REDUCTION ASSEMBLY OUTPUT SHAFT
12	1	MSC	#300	WOODRUFF KEY

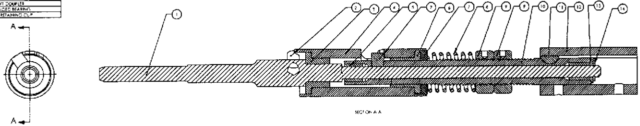


FMC Technologies

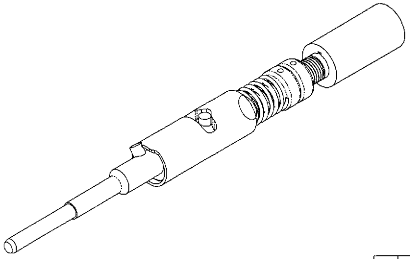
TITLE: GEAR REDUCTION ASSEMBLY

SIZE D | REV A

ITEM NO.	QTY.	MANUFACTURER	P/N	DESCRIPTION
1	1	FMC TECHNOLOGIES		OUTPUT SHAFT
2	2	FMC TECHNOLOGIES		SHAFT PIN
3	1	BOSTON GEAR	FB610-4	3/8 FLANGED BEARING
4	1	FMC TECHNOLOGIES		INDEXED CLUTCH SLEEVE
5	1	W.M. BERG	701-00004-092	3/8 THIN-WALLED BEARINGS
6	1	BOSTON GEAR	FB810-5	1/2 FLANGED BEARING
7	1	W.M. BERG	BRS-3	1/2 I.D. ROLLER THRUST BEARING
8	1	FOX VALLEY SPRING	0845-072-02	OVERLOAD SPRING
9	2	FMC TECHNOLOGIES		OVERLOAD ADJUSTMENT BOLT
10	1	FMC TECHNOLOGIES		INPUT SHAFT
11	1	MISC.	#303	WOODRUFF KEY
12	1	FMC TECHNOLOGIES		3/4 SHAFT COUPLER
13	1	BOSTON GEAR	FB46-4	1/4 FLANGED BEARING
14	1	ROTORCLIP	BE-25	SPRING RETAINING CLIP



SECTION A-A



FMC Technologies

TITLE: OVERLOAD CLUTCH ASSEMBLY

SIZE D | DWG. NO. | REV A

APPENDIX D—DESIGN DOCUMENTATION

The following additional theoretical information has been included for the readers understanding only, since there is information that has been removed for confidentiality reasons.

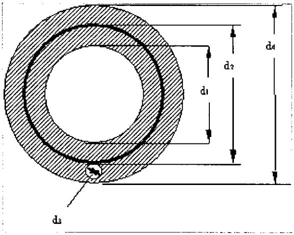
SCSSV Environmental Design Specifications

Below is a list of the SCSSV Design Parameters to be used throughout the following calculations. These values have been chosen and presented according to recommendations and evaluations presented in the Design Report corresponding to this project.

$P_{int} := 15000psi$	*Internal Working Pressure*
$P_{test} := 18750psi$	*Internal Test Pressure*
$d_1 := 3.562in$	*Internal Drift Diameter*
$d_4 := 7.875in$	*External Drift Diameter*
$T_{max} := (350 + 459.6)R$	*Maximum Operating Environmental Temperature*

SCSSV Dimensional Derivation:

The purpose of these calculations are to show the required minimum and maximum allowable diameters to be used in the electric SCSSV. A cross-sectional view of the SCSSV can be seen below and will be used in defining all further variables. The lower circular hole associated with the dimension d_3, will be used to install the electric actuator within the SCSSV, therefore is the main interest of these following calculations



Assumptions:

- Internal Pressure of $P_{int} = 15000psi$ within d_1
- No external pressure on d_4, e.g. a $\Delta P = P_{int}$
- Uniform material properties
- δ_{wall} will be the required thickness of the valve body
- Calculations will be conducted according to ASME Boiler Code Pressure Vessels VIII-I

$\sigma_y := 93 \cdot 10^3 \cdot \text{psi}$ *Yield Strength of Material*
-AISI 8630 Special (110/93) API 6A & H_2S Service

$R_y(T) := 1$ *Ratio of average temperature dependent trend curve value of yield strength to the room temperature yield strength*

$S := \left(\frac{2}{3}\right) \cdot 0.92 \cdot \sigma_y \cdot R_y(T_{max})$ *Allowable internal stress above room temperature associated with ferrous and nonferrous structural quality material for pressure retention*

$S = 57040.00\text{psi}$

Calculations:

Internal hoop stress associated with an internal pressure acting on an enclosed cylinder can be found by the following equation for a thin walled cylindrical pressure vessel, Guidebook for the Design of ASME Section VIII Pressure Vessels. The requirements to use this approximation is that the thickness to internal radius dimension be less than 0.5. If this value is larger than 0.5 the thick walled cylinder approximation should be used instead, but since using a high strength material it should be thin-walled.

Thin-walled Cylindrical Pressure Vessel

$$t(P_{int}, R, S) := \frac{P_{int} \cdot R}{S - 0.6 \cdot P_{int}}$$

where: R is the ID radius
P_{int} is internal pressure
S is allowable stress
t is thickness of material

Resulting in a required wall thickness of:

$$\delta_{wall} := t\left(P_{int}, \frac{d_1}{2}, S\right)$$

$\delta_{wall} = 0.56\text{in}$

if $P_{int} = 1.50 \times 10^4 \text{psi}$

$\frac{d_1}{2} = 1.78\text{in}$

$S = 5.70 \times 10^4 \text{psi}$

Check to make sure this result is in the valid range for the given equation, $\frac{t}{R} < 0.5$

$$\frac{2 \cdot \delta_{wall}}{d_1} = 0.31$$

Therefore the required O.D. of the SCSSV is as follows:

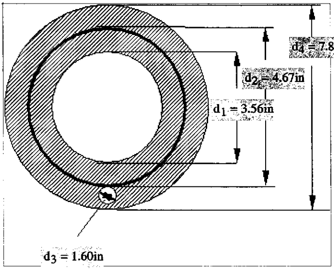
$$d_2 := d_1 + 2 \cdot \delta_{wall}$$

$d_2 = 4.67\text{in}$

Allowable dimensional diameter for the actuator installation:

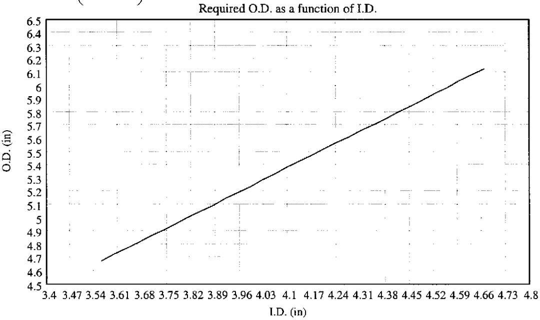
$$d_3 := \frac{d_4}{2} - \frac{d_2}{2}$$

$$d_3 = 1.60\text{in}$$



Below is a list of outer diameter requirements as a function of inner diameters to be used a design reference:

$$od(id) := id + 2 \cdot t\left(P_{int}, \frac{id}{2}, S\right) \qquad id := d_1, d_1 \cdot 1.01 .. d_2$$



Cylindrical Cam

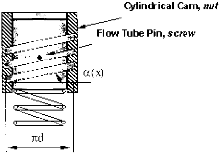
The cylindrical cam mechanism is used to transfer the rotational torque and motion to a linear force and translation. The cam will have an outer ring, *nut*, that has 3 identical profiles ground on the i.d. of its surface. These profiles will be equally spaced around the I.D. The internal flow tube, *bolt*, will have 3 pins located equidistant around the O.D. along the same vertical plane, if viewed in the position given in the schematics below. These pins will engage with the profiles on the outer ring. The outer ring will be held in position from translating to the right or left, while the internal flow tube be held from rotation. Thus, when the outer ring is rotated the internal flow tube is forced to follow the ground profile. The concerns in designing the cam profile are the following:

- profile must be large enough to allow spring retraction
- profile should minimize the required input torque
- designed so that the pins can withstand the resulting stresses

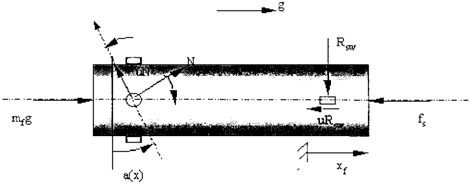
Company: FMC Technologies
Model:
Dia:

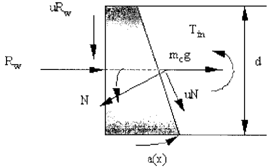
$p(x) := (4in)$ *Pitch of thread (distance / revolution)*
$d := 4.15in$ *Mean diameter of thread*

The following figures detail the forces involved with the flow tube pins and cylindrical cam interaction, the derivation of the cam torque and pin reaction forces can be found below the diagrams. The cam, *nut*, will rotate from left to right as shown below, causing the flow tube to translate downward.



Below is the associated free body diagram for the cylindrical cam mechanism. The flow tube with the (3) attached pins is shown first, while below that a section of the outer cam ring. A few things to note, a(x) is the angle associated with the current pitch of the cam, since the pitch may change according to the cam's position. Another is that through this analysis, the input torque will be determined as a function of the flow tube position.

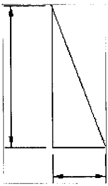




Assumptions:

- All components will be considered rigid bodies
- Analysis will take place at a quasi-static position, e.g. marginal acceleration of bodies
- All pins incur the same load, but will be designed as to accommodate singular loading of an individual pin
- The reaction forces, R_{sw} and R_w will be sustained by anti-rotational and drift bearings, respectively
- The contact locations will be Teflon-on-Teflon, $\mu := 0.1$, taken greater than documented to account for inconsistencies within the surfaces.
- The contact locations for the reaction forces will be assumed to have the following properties, $\mu_{sw} := 0.04$ (some form of bearing), and $\mu_w := 0.10$ (Teflon-Teflon contact)
- The pitch of the cam has been previously defined and will be used in the following analysis
- The mass of each object will be assumed negligible, $m_f := 0lb$ and $m_c := 0lb$

Use geometric relations to determine a(x) in terms of p(x) and d, if the cam is unwrapped one full rotation, it could be approximated as seen below, where the adjacent line is the circumference of the triangle while the opposite is the pitch.



$$\tan(\alpha(x)) = \frac{p(x)}{\pi \cdot d} \quad \text{thus} \qquad \alpha(x) := \operatorname{atan}\left(\frac{p(x)}{\pi \cdot d}\right)$$

Equations of Equilibrium:

For the flow tube direction (x_f):

$$0 = m_f \cdot g - \mu \cdot N \cdot \sin(\alpha(x_f)) + N \cdot \cos(\alpha(x_f)) - \mu_{sw} \cdot R_{sw} - f_s(x_f)$$

And the perpendicular direction for equilibrium:

$$0 = \mu \cdot N \cdot \sin(\alpha(x_f)) + N \cdot \cos(\alpha(x_f)) - R_{sw}$$

For the cylindrical cam summation of torques:

$$0 = \mu_w \cdot R_w \cdot \frac{d}{2} + N \cdot \sin(\alpha(x_f)) \cdot \frac{d}{2} + \mu \cdot N \cdot \cos(\alpha(x_f)) \cdot \frac{d}{2} - \tau_{cam}$$

And equation of equilibrium:

$$0 = R_w - N \cdot \cos(\alpha(x_f)) + \mu \cdot N \cdot \sin(\alpha(x_f))$$

Solving the above equations:

$$\tau_{cam}(x_f) := \frac{1}{2} \cdot \left[f_s(x_f) \cdot d \cdot \frac{(\mu_w \cdot \mu \cdot \sin(\alpha(x_f)) - \mu_w \cdot \cos(\alpha(x_f)) - \sin(\alpha(x_f)) - \mu \cdot \cos(\alpha(x_f)))}{(\mu \cdot \sin(\alpha(x_f)) - \cos(\alpha(x_f)) + \mu_{sw} \cdot \mu \cdot \sin(\alpha(x_f)) + \mu_{sw} \cdot \cos(\alpha(x_f)))} \right]$$

$$N(x_f) := \frac{(-f_s(x_f))}{(\mu \cdot \sin(\alpha(x_f)) - \cos(\alpha(x_f)) + \mu_{sw} \cdot \mu \cdot \sin(\alpha(x_f)) + \mu_{sw} \cdot \cos(\alpha(x_f)))}$$

$$R_{sw}(x_f) := (-f_s(x_f)) \cdot \frac{(\mu \cdot \sin(\alpha(x_f)) + \cos(\alpha(x_f)))}{(\mu \cdot \sin(\alpha(x_f)) - \cos(\alpha(x_f)) + \mu_{sw} \cdot \mu \cdot \sin(\alpha(x_f)) + \mu_{sw} \cdot \cos(\alpha(x_f)))}$$

$$R_w(x_f) := (f_s(x_f)) \cdot \frac{(\mu \cdot \sin(\alpha(x_f)) - \cos(\alpha(x_f)))}{(\mu \cdot \sin(\alpha(x_f)) - \cos(\alpha(x_f)) + \mu_{sw} \cdot \mu \cdot \sin(\alpha(x_f)) + \mu_{sw} \cdot \cos(\alpha(x_f)))}$$

Drive Shaft Requirements:

It will now be the purpose to determine the required material and dimensions of the output drive shaft to withstand the maximum applied torque. This will be conducting using the maximum shear stress theory. The appropriate diameter of the shaft is as follows, Mechanical Engineering Design, Shigley and Mischke.

Assumptions:

- Pure torsion acting through the drive shaft
- Uniform material properties
- Solid circular cross-section
- No appreciable deflection and deformation
- Material obeys Hooke's Law

$$d_{shear}(n, S_y, T) := \left(32 \cdot n \cdot \frac{T}{\pi \cdot S_y}\right)^{\left(\frac{1}{3}\right)}$$

where:

$n := 2$ *Factor of safety*

$S_y := 50000psi$ *Yield Strength*

$T := \tau_{planetary\_max} \cdot 1.5$ *Maximum applied torque*

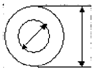
$T = 58.20in \cdot lbf$

$d_{shear}(n, S_y, T) = 0.287in$ therefore this is the minimum diameter required to have the specified safety factor if a ***solid circular cross-section***

if the drive shaft is hollow, i.e. portions which are used to couple to the electric motor the following properties are required:

$d_i := 0.125in$

$$J_{hollow}(d_o) := \frac{\pi \cdot \left(d_o^{\,4} - d_i^{\,4}\right)}{64}$$

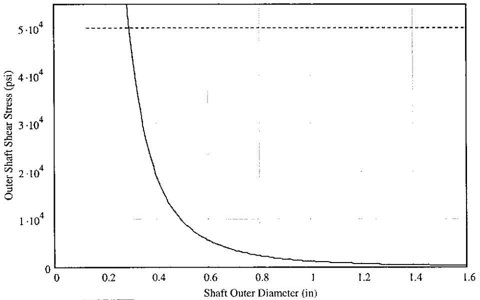


$$\tau_{shaft}(d_o) := n \cdot T \cdot \frac{\left(\frac{d_o}{2}\right)}{J_{hollow}(d_o)}$$

$d_{shaft} := d_i, d_i + 0.01in .. d_3$

since nonlinear, the shear function will be plotted, note that the safety factor has already been accounted for.

The figure below, has the shear stress plotted against the shaft outer diameter, along with the specified yield strength in the dashed line. Any point below this dashed line will meet the safety factor requirements

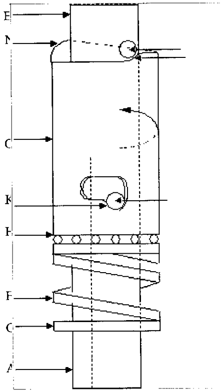


therefore a diameter of $d_o := 0.3\text{in}$ with an I.D. of $d_i = 0.125\text{in}$

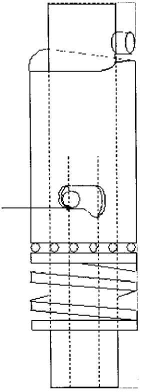
$$\frac{\tau_{shaft}(d_o)}{n} = 22637.84\text{psi} \quad \text{which is below} \quad S_y = 50000.00\text{psi}$$

Torque Overload Clutch Parameters:

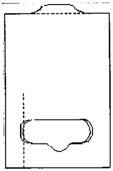
The purpose of these calculations is to determine the spring properties and clutch dimensions to release and disconnect the two drive shafts at a specified overload torque. A schematic of the clutch can be seen below, noting that this schematic only shows a single release direction while the final implementation will have a bi-directional release path. The calculations for either case are the same, it is only a dimensional design feature.



Driving Position



Release Position



Bi-directional overload sleeve, to be used instead of illustrated (C) to the left

Overload Clutch Operation:

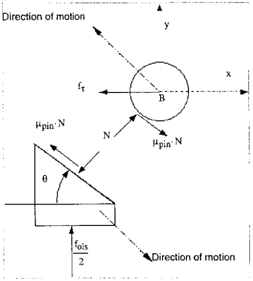
This mechanism has been modified from Ingenious Mechanisms for Designers and Inventors Jones.

> *The motion is transmitted through the hollow drive shaft **A** to pin **B**, sleeve **C**, and pin **D** to shaft **E**, finally driving the load. The hollow shaft A fits over an extension on shaft **E**, and is held against endwise movement by a thrust collar.*
>
> *Pin **B** engages a slot in sleeve **C**, this slot being similar in form to the well-known bayonet lock. Pin **D** in shaft **E** engages a cam surface formed on the end of sleeve **C**. The sleeve is held firmly against pin **D** by spring **F**, the tension of which may be adjusted by changing the position of collar **G**. A ball thrust bearing **H** is located between the spring and sleeve **C**.*
>
> *Normally, the entire assembly rotates in the direction indicated by the arrow, and it will be evident that the sleeve **C** is prevented from sliding downward by the upward pressure of spring **F**.*
>
> *If the torque reaches the danger point, the spring thrust is overcome and sleeve **C** moves downward far enough to release pin**D**, which moves along a slight rise and finally disengages pin **B** from slope **K**, thus allowing the level surface **M** to engage pin **B**, which now rests against the end of the slot. Pin **B** holds sleeve **C** in this released position, and rotation of **A** and **C** continues while shaft **E** remains free.*
>
> *To re-engage the drive, the drive shaft **A** is reversed a few turns. As the shaft **A** revolves sleeve **C**, due to the upward pressure on pin **B** and the directional resistance between **B** and **C**, surface **N** comes into contact with pin **D**. At this point any further turning movement would force sleeve**C** downward. However, the resistance between pin **B** and surface **M** is slight in comparison with the force required to rotate the surface **N** beneath pin **D**. Therefore, shaft **A** and pin **B** turn until the pin engages the lower part of this slot and is again in the driving position.*

Assumptions:

For the purpose of evaluation the following assumptions will be made:

- spring force will be spread equally between the two pins
- all reactions will be taken as seen in the free body diagram below
- pin will remain a rigid body
- once pin has begun to move, i.e. the system is in equilibrium this will determine the required preload
- coefficient of friction between the pin and cam surface, mild steel on mild steel, $\mu_{pin} := 0.74$



$\tau_{max} := \tau_p$ *Maximum transmitted torque (in-lbf) -* ***2 X Max Applied Torque***

$d_{pin\_shaft} := 1.0in$ *Pin contact diameter (in)* $d_{pin\_shaft} = 1.000in$

$\theta := 45 \cdot \left(\frac{\pi}{180} \right)$ *Pin contact angle*

$f_\tau := 2 \cdot \frac{\tau_{max}}{d_{pin\_shaft}}$ *Transposed force acting from maximum torque*

Summation in y-direction on sleeve ***C***

$$0 = \frac{f_{ols}}{2} + \mu_{pin} \cdot N \cdot \sin(\theta) - N \cdot \cos(\theta)$$

Summation in x-direction on pin ***B***

$$0 = -f_\tau + N \cdot \sin(\theta) + \mu_{pin} \cdot N \cdot \cos(\theta)$$

resulting in

$$f_{ols} := -2 \cdot f_\tau \cdot \frac{(-\cos(\theta) + \mu_{pin} \cdot \sin(\theta))}{(\sin(\theta) + \mu_{pin} \cdot \cos(\theta))}$$

$$N := \frac{f_\tau}{(\sin(\theta) + \mu_{pin} \cdot \cos(\theta))}$$

The resulting required spring force should be no greater

$f_{ols} = 53.201bf$

And a resulting reaction force

$N = 144.671bf$

Velocity Calculations

The following will determine the associated shaft surface velocities, along with the radial velocities of key components. This will result in a value for the length of time required to open the safety valve.

From the required motor torque, $\tau_{motor}(x_{max}) = 0.120\text{in}\cdot\text{lbf}$ which corresponds to a rotational motor velocity of:

$$\omega_{motor} := 12\frac{1}{\text{sec}} \quad \text{in RPS}$$

Therefore the rotational speed at the output of the gear reduction assembly is:

$$\omega_{gear\_reduction} := \frac{\omega_{motor}}{n_p} \qquad \omega_{gear\_reduction} = 1.80\frac{1}{\text{min}} \quad \text{in RPM}$$

With an output shaft diameter of $d_{shaft\_out} := \frac{5\text{in}}{16}$, this results in a surface speed of:

$$v_{shaft\_out} := \frac{\pi \cdot d_{shaft\_out} \cdot \omega_{gear\_reduction}}{12}$$

$$v_{shaft\_out} = 0.01\frac{\text{ft}}{\text{min}}$$

Then, the rotational speed of the cylindrical cam is:

$$\omega_{cam} := \frac{\omega_{motor}}{n_p \cdot n_c} \qquad \omega_{cam} = 0.20\frac{1}{\text{min}} \quad \text{in RPM}$$

with an outer diameter of $d_{cam} := 4.60\text{in}$, resulting in a surface speed of:

$$v_{cam} := \frac{\pi \cdot d_{cam} \cdot \omega_{cam}}{12} \qquad v_{cam} = 0.02\frac{\text{ft}}{\text{min}}$$

These speeds are at normal opening conditions, **not** at the extreme conditions of closure. These values are extremely low for bearing criteria and seal criteria; therefore surface speed factors should not be an issue during the selection process.

The time required to open the safety valve can be found as presented below, using the number of required gear reduction shaft rotations and the input motor velocity:

$\theta_p = 115.19$ *gear reduction output rotations to open position, (rad)*

$\theta_m := \theta_p \cdot n_p \qquad \theta_m = 4.61 \times 10^4$ *(rad)*

Therefore, the time required to open the safety valve is:

$$t_{open} := \frac{\theta_m}{\omega_{motor} \cdot 2 \cdot \pi} \qquad t_{open} = 10.19\text{min}$$

Shaft Sealing Calculations

The following calculations have be presented by Parker Seals, for the FlexiSeal product group in determining the proper seal material and size.

1.) Determine shaft surface hardness:
Must be greater than 35 Rc for temperature -250 to 575 F
Rc=39 for Aged Hardened INCONEL 725

2.) Determine required temperature range:
-20 F to 350 F

3.) Dynamic surface speed on sealing surface:
found when in closing mode, in previous section on sleeve bearing calculations the gear reduction output shaft rotational speed is:
$N_p = 36.67\text{Hz}$ *, resulting in a surface speed on a* $d_{shaft\_out} = 0.313\text{in}$*shaft*

$$V_s(d_{shaft\_out}) = 15.00\frac{\text{ft}}{\text{min}}$$

Therefore, with these requirements, the proper material should be #106, a Polymer/PTFE.

Back Drive Ability

In order to assure that the flow-tube and cylindrical cam system will return itself into the starting position without any external aid, it will be shown that once the holding torque is removed from the equations of motion, the system will be biased towards the starting position. This will be shown by indicating a change in flow tube acceleration direction. Using the kinematics previously defined and evaluating all functions of x at the extended position, the initial acceleration can be determined, along with the initial reaction forces.

Equations of Motion:

The following equations are taken from the previous equations of motion, with the exception that these values are not neglecting the acceleration terms

For the flow tube direction (x_f):

$$m_f \cdot ddx_f = \mu \cdot N_1 \cdot \sin(\alpha) + N_1 \cdot \cos(\alpha) + \mu_{sw} \cdot R_{sw1} - f_{smax}$$

And the perpendicular direction for equilibrium:

$$0 = R_{sw1} - \mu \cdot N_1 \cdot \cos(\alpha) + N_1 \cdot \sin(\alpha)$$

For the cylindrical cam summation of torques:

$$J_c \cdot 2 \cdot \frac{\pi \cdot ddx_f}{p_1} = \left(\mu \cdot N_1 \cdot \cos(\alpha) - \sin(\alpha) + \mu_w \cdot R_{w1}\right) \cdot \frac{d}{2}$$

And equation of equilibrium:

$$0 = R_{w1} - N_1 \cdot \cos(\alpha) + \mu \cdot N_1 \cdot \sin(\alpha)$$

where subscript (1) is used to use an undeclared variable
the values taken at the extended position are the following:

$$f_{smax} := f_s(x_{max})$$

$$p_1 := p(x_{max})$$

$$\alpha := \alpha(x_{max})$$

and the mass moment of inertia of the cylindricl cam can be estimated by the following:

$$m_c := 15lb$$

$$J_c := \left(\frac{1}{2}\right) \cdot m_c \cdot \left[\left(\frac{d}{2}\right)^2 - \left(\frac{d_1}{2}\right)^2\right]$$

$$m_f := 25lb$$

First, to evaluate the flow tube acceleration due to a 10% larger torque input than required:

$$\tau_{in} := \tau_{cam}(x_{max}) \cdot 110\% \qquad \tau_{in} = 371.66 in \cdot lbf$$

Solving for the flow tube acceleration and substituting numerator and denomentator

$$B(\tau_{in}) := -4 \cdot J_c \cdot \pi \cdot \cos(\alpha) + 4 \cdot J_c \cdot \pi \cdot \mu \cdot \sin(\alpha) + 4 \cdot J_c \cdot \pi \cdot \mu_{sw} \cdot \mu \cdot \sin(\alpha) + 4 \cdot J_c \cdot \pi \cdot \mu_{sw} \cdot \cos(\alpha) \ldots$$
$$+ -\mu_w \cdot d \cdot p_l \cdot \cos(\alpha) \cdot m_f + \mu_w \cdot d \cdot p_l \cdot \mu \cdot \sin(\alpha) \cdot m_f - \sin(\alpha) \cdot d \cdot p_l \cdot m_f - \mu \cdot \cos(\alpha) \cdot d \cdot p_l \cdot m_f$$

$$A(\tau_{in}) := 2 \cdot \cos(\alpha) \cdot \tau_{in} - 2 \cdot \mu \cdot \sin(\alpha) \cdot \tau_{in} - 2 \cdot \mu_{sw} \cdot \mu \cdot \sin(\alpha) \cdot \tau_{in} - 2 \cdot \mu_{sw} \cdot \cos(\alpha) \cdot \tau_{in} \ldots$$
$$+ -f_{smax} \cdot \mu_w \cdot d \cdot \cos(\alpha) + f_{smax} \cdot \mu_w \cdot d \cdot \mu \cdot \sin(\alpha) - f_{smax} \cdot \sin(\alpha) \cdot d - f_{smax} \cdot \mu \cdot \cos(\alpha) \cdot d$$

$$ddx_f(\tau_{in}) := -p_l \cdot \frac{A(\tau_{in})}{B(\tau_{in})}$$

$$ddx_f(\tau_{in}) = 314.24 \frac{in}{sec^2}$$

Noting that this value is positive, thus it will be accelerating towards the right into the power return spring, but if now the input torque is reduced to $\tau_{in} := 0 in \cdot lbf$

$$ddx_f(\tau_{in}) = -3.14 \times 10^3 \frac{in}{sec^2}$$

Noting, again there is a sign change in the acceleration, to where the flow tube is now accelerating with the aid of the power return spring. Therefore, the safety valve ***will*** retract itself without any external aid.

VITA

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Bachelor of Science in Mechanical Engineering — August 1998 to May 2002
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Work Experience:

FMC Technologies, Houston, Texas — 2001 to Present
Student Researcher (Fall 2001-Spring 2002)
Responsible for research and design of an electric safety valve actuator.
Test Engineer (Summer 2001)
Responsible for testing and refurbishing of offshore oilfield controls equipment.
Refurbished/upgraded hydraulic and topside control systems.
Conducted functional tests of subsea override tools.
Support for system integration tests of subsea control systems.

Humanetics II, Carrollton, Texas — 1999 – 2000
Database and Information Analyst (Summer 2000, Summer 1999)
Responsible for implementation and design of database driven web environments and applications.
Automated daily task to allow efficient time usage of employees.
Pioneered major database use for the company.

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