Malfunction of Electro Hydraulic Servovalves Operating on Phosphate Ester Based Fluids

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Many of the fossil fuel steam turbine speed regulation and load controls are dependent upon electro hydraulic closed loop servosystems for proper operation. This type of system is extremely fast and accurate with resultant excellent control and speed regulation. The regulation and control is accomplished by modulating the amount of energy (steam) to the prime mover (turbine). In addition to the control and regulation function, these systems also provide a protective function which can automatically trip out the turbine if an overspeed condition occurs. From the description above one can see that proper performance of the electrohydraulic servosystem is critical to turbine operation.

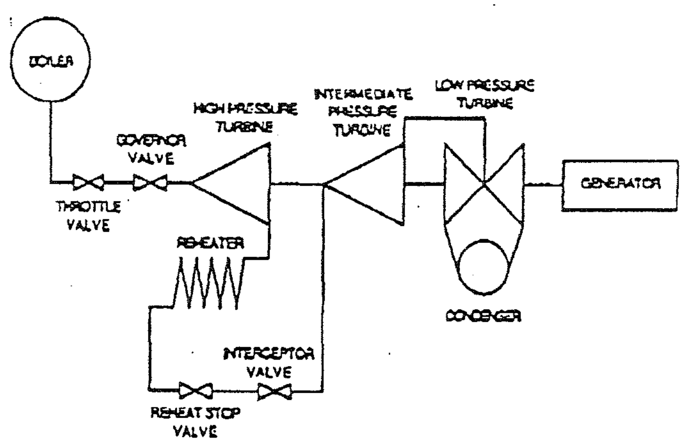


FIGURE 1. FOSSIL STEAM TURBINE FLOW PATH AND VALVING

FIGURE 1

Figure 1 (above) shows a simple representation of a steam flow path for a typical fossil fuel unit. Ahead of the turbine are modulating type throttle and governing valves which are used to control speed and load by means of closed loop electrohydraulic servosystems. The reheat and interceptor valves are basic open/close type valves which generally are not capable of modulation.

The control units micro processor receives input from the operator who sets performance parameters that are desired plus input from speed and load monitoring transducers. These inputs are compared and operated on by the micro processor on a continuing basis. A setting for the opening of the governor valve is determined and, if different than the present setting, sends a command to the electrohydraulic servovalve to flow oil to move the steam control valve to the desired position. This positioning system is interactive and continuing repositioning the valves dependent on the load on the system.

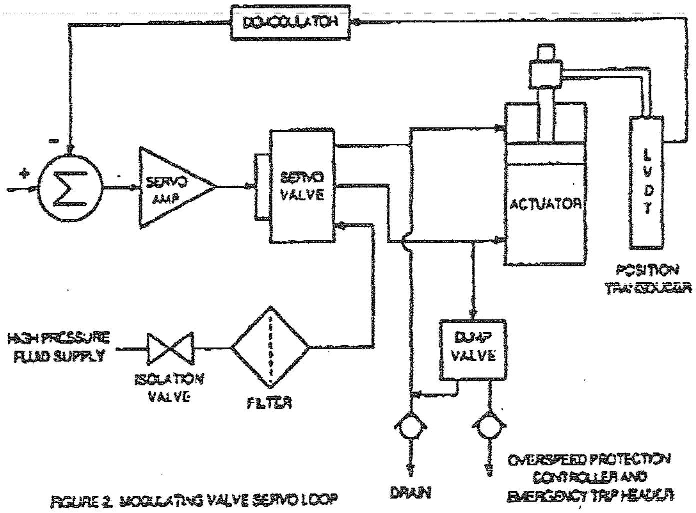


FIGURE 2

Figure 2 illustrates a typical steam valve servo loop. Valve positioning is the result of demand signals from the micro processor which are compared with actual position and if different, amplified and sent to the servovalve at each steam valve actuator. The servovalves control steam valve position by metering high pressure hydraulic fluid to the steam valves hydraulic actuator. Actual steam valve position is determined from the voltage output from a linear variable differential transformer (LVDT) attached to hydraulic actuator. The LVDT produces a D.C. voltage which is directly proportional to its position. This signal is fed into the summation junction of the amplifier and compared against the commanded position. Any differences are amplified and fed to the servovalve to reposition the actuator until desired and actual positions are the same.

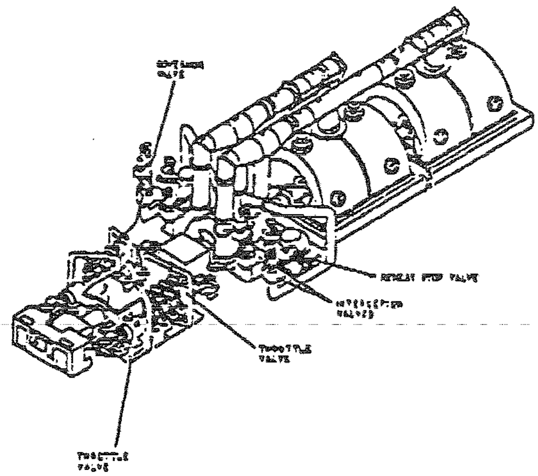


FIGURE 3. TURBINE STEAM VALVE CONFIGURATION

FIGURE 3

Turbine steam valve configuration, Figure 3, shows the physical configuration of the steam valves of a typical fossil fuel steam turbine.

There are usually 2 to 4 throttle valves and 4 to 8 governor valves on a unit.

In the control system one of, if not the single most critical component, is the electrical hydraulic servovalve. It is at the interface of the electrical/electronic portion of the servoloop and the hydraulic portion of the loop. One may think of the electronics as the brains and the hydraulics as the muscle. Figure 4 shows the inner construction of a typical servovalve. The servovalve consists of a pilot stage and an output or power stage.

Valve, Servo

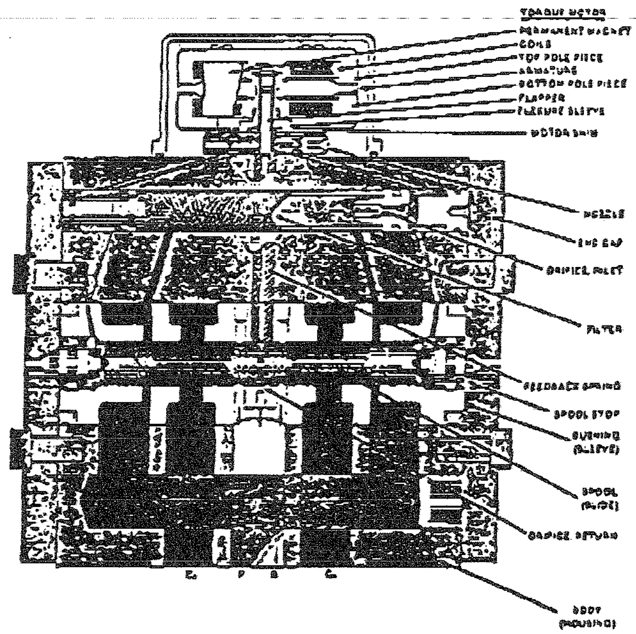


FIGURE 4

The pilot stage is made up of an electromechanical torque motor which receives the electrical signal and a hydraulic stage which converts torque motor motion to controlled hydraulic flow. The pilot stage hydraulic flow is used to position the output stage sliding spool within the stationary sleeve, thus, is able to control the direction and amount of hydraulic fluid flow from the servovalve.

The electrical magnetic torque motor consists of an armature, coils, permanent magnets top and bottom pole pieces. The armature is mounted on a hallow tube called a flexure sleeve. The flexure sleeve positions the armature between the top and bottom pole piece, serves as an isolation barrier between the pilot stage hydraulic flow and the magnetic portion of the torque motor. Most importantly it acts as a centering spring to maintain the armature in the center of the polepieces counteracting the decentering forces of the magnetic flux acting on the armature. The ends of the armature are suspended in the flux field between the upper and lower pole piece. The two servovalve coils surround the armature.

The hydraulic portion of the pilot stage consists of a dual nozzle and flapper and two orifices. The hydraulic network is a two arm, four element hydraulic bridge. Two elements are passive (inlet orifice) and two are active (nozzles/flapper). The flapper, rigidly attached to the midpoint of the torque motor armature passes between the two nozzles. This creates the two active or variable orifices between the two nozzle tips and the flapper. The fluid between the two nozzles and the two inlet orifice is applied to the ends of the spool (See Figure 5). This forms a hydraulic bridge network and by varying the nozzle/flapper orifices we are able to control fluid flow to the end of the spool and, thus, control spool motion.

The power or output stage is a conventional four way sliding spool design. Output flow from the valve is proportional to spool position. A centilevered feedback spring fixed to the flapper engages a slot in the center of the spool. Displacement of the spool deflects the feedback spring which creates a force on the armature and flapper assembly.

In the neutral position the flapper imposes an equal restruction to flow through both nozzles and there is no flow or differential pressure to cause spool displacement. On electrical input signal induces a magnetic charge in the motor armature and causes a deflection of the armature and flapper towards one of the two nozzles. This action produces a differential pressure from one end of the spool to the other and results in spool displacement.

The spool displacement causes a counteracting force in the feedback spring to oppose the original input signal force in the feedback spring to oppose the original input signal torque. When these forces equalize, the flapper returns to neutral position and the differential pressure across the spool goes to zero. The spool will remain in its new position until the electrical signal is changed. In essence, the torque generated by the electrical input is counter balanced by the torque generated by spool motion times feedback wire spring (F) times the moment arm length (L). Torque is force (F) times length (L) (T=FL).

The performance requirement of the servosystem is such that the power stage spool bushing assembly has to be different than what is normally expected of a standard type flow control valve. The servovalve spool is expected to be able to respond with a change of output flow with the slightest change in input signal. This means that the spool flow lands are neither overlapped or underlapped but are critically lapped such that the edges of the external bushing and internal spool are critically lapped, coincident to each other within twenty or thirty millionths of an inch (.5 to .75 microns). Thus, when the valve is in the zero flow position, the slightest spool motion will direct flow to the cylinders to correct any mispositioning or drift. Figure 6 is a sketch of a critically lapped spool/bushing assembly.

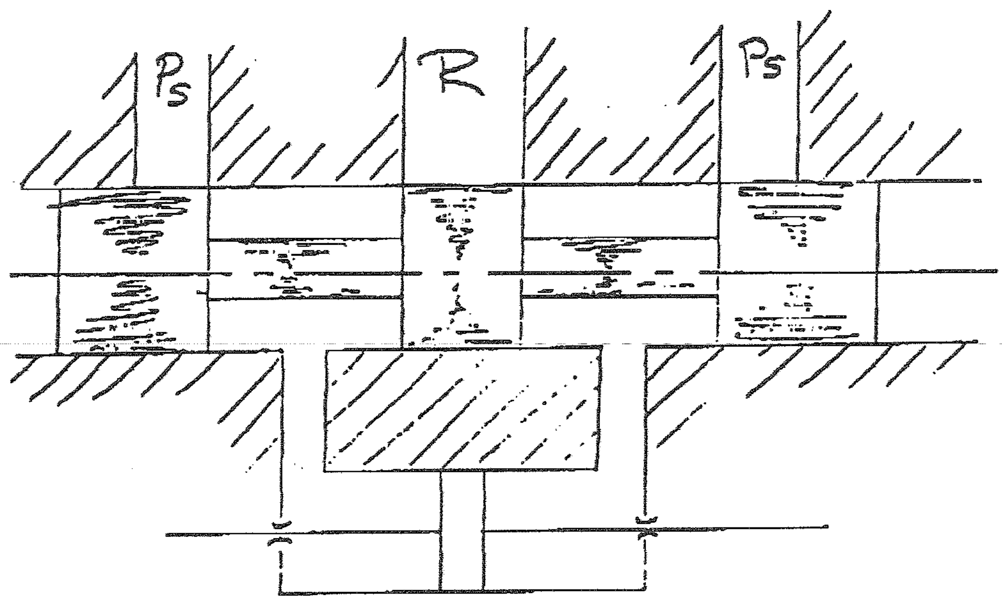


FIGURE 5
SPOOL / BUSHING ASSEMBLY

Again, the nature of the closed loop electro hydraulic servosystem places demands for quick response, fast reaction to changes on the overall system. This means that the hydraulic system has to be a closed center system with the required operating pressure always present to the servovalve. This allows the servovalve to react instantly to any change with a controlled amount of hydraulic fluid in order to correct any mispositioning or variations in the load on the generator.

This also means that there is a constant flow of hydraulic fluid from a high pressure to a low pressure (from pump to tank) through the two hydraulic sections of the servovalve. There is a constant flow of hydraulic fluid through the pilot stage and through the power stage when the valve is at the zero flow condition. This flow of hydraulic fluid is energy supplied by the hydraulic system pump and dissipated across the servovalve without doing any useful work. It is completely wasted energy and is the penalty or cost of fast accurate controls. No one ever said that servosystems were efficient.

Because of existing operating conditions it was felt that phosphate esters or chlorinated hydrocarbon fluids containing small amounts of phosphate esters were suited for steam turbine applications. These fluids are adequate with regard to fire resistance, heat transfer properties, bulk modules, oxidation and thermal stability, foaming and air entrainment characteristics toxicological properties lubricity and filterability. Their low temperature viscosity and rust protection properties are matters of concern for steam turbine applications. Both of these types of fluids have been used to lubricate boiler feed pump drives and gas turbines. Because of their hydrolytic oxidation and thermal stability, the chlorinated hydrocarbon type was chosen initially for turbine electrohydraulic control evaluation.

Initial test on prototype systems using a chlorinated hydrocarbon fluid as a hydraulic fluid as a medium was very successful. Then disaster struck. The fluid broke down forming hydrated iron oxide. Millions of particles, approximately 1 micron in size, would be generated in the system which would plug filters and silt up servovalve spools until they could not be moved under any conditions.

Eventually the decision was made to convert all hydraulic systems over to phosphate ester based fluids and again the initial results were very satisfactory. Filter plugging and spool/bushing friction was eliminated, but after eight weeks of operation the servovalve leakage rate started to increase. This increase in leakage continued and eventually became so large that the leakage rate exceeded the pump capacity and the entire control system had to be shut down. This leakage in a operational system can be observed by measuring the pump motor current.

The total system leakage and power loss across the servovalve are kept to a minimum by use of small orifices and clearances in the pilot stage hydraulic bridge network and by maintaining closely fit diametrical clearance between the sliding spool and the stationary bushing plus always maintaining a slight overlap on the spool flow edges in relationship to the bushing flow edges. These concerns for maintaining the non-useful flow through the servovalve to an absolute minimum plus other system requirement such as pressure gain and response dictate the design requirements for the power stage bushing/spool assembly.

Again to re-emphasize the critical nature of the power stage spool/bushing design, I would like to point out that the output flow of a servovalve is typically stated at a pressure drop of 500 psi across each of the flow metering orifices. Thus, flows up to 15 GPM can be obtained with an effective orifice diameter of .17 inch. That is a great deal of flow, approximately 60 in^3/sec. through a very small hole. We can do this because the pressure drop is very high.

When the use of electrohydraulic servosystems for control of a steam turbine speed governing system were first thought of in the early to mid fifties, a great deal of time was spent in consideration of the proper hydraulic fluid. The first and most natural consideration was to use mineral based hydraulic fluid, but this was quickly discarded because of the threat of serious fire if a pressure hydraulic line were to break or a pin hole leak develop with the hydraulic oil being ignited by a hot steamline or other heat sources.

Use of waterbase fluids were also considered early in the program but were discarded due to concerns about the ability to maintain the proper water mix concentration over a longtime span.

It was determined by inspection of the affected servovalves that the spool bushing metering edges were being attached and material was being removed allowing leakage flow across the bushing/spool assembly from high pressure to drain. This leakage flow could be as great as the rated flow of the servovalve. Figure below shows the edges affected.

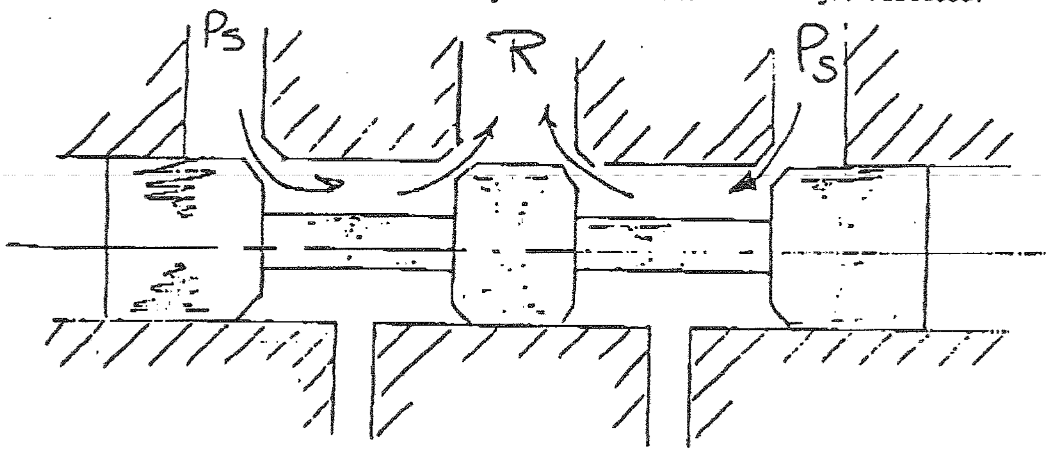


FIGURE 6
SPOOL / BUSHING EDGE REMOVAL

All the possible mechanisms for edge removal were considered. Hard particle erosion, cavitation damage, eletro-chemical corrosion damage and improper material. But, none of these mechanisms could completely explain what was occurring.

The reason for this erosion was diagnosed as being associated with the chemistry of the fluid, although the exact mechanism was not known at that time. Therefore, a continuous by-pass filtration system, involving the use of fuller's earth filter media, was adapted to continually absorb any active chemical species in the fluid. This filter system proved to be 100% effective.

Several controlled tests were run where the system leakage rate was monitored while a fuller's earth filtration system was added or removed from the systems. The leakage rate increased when the fuller's earth filtration was removed and stopped increasing when the filtration system was added back to the system. Thus, confirming that a fuller's earth filtration system could control the mechanism of the servovalve wear.

This spool/bushing, orifice erosion was not limited to just turbine controls systems but affected and affects any hydraulic system that uses phosphate ester based fluid as a hydraulic media. This includes commercial aircraft which use a specialized phosphate ester fluid with a viscosity improver added. Commercial aircraft use phosphate ester fluid because of their fire retardant characteristic. The commercial aircraft industry was having a great deal of problems with increased leakage with resultant loss of system control. This was determined to be a very serious problem by Boeing Aircraft , thus, a full fledged investigation by the Boeing Science Research Laboratories, Seattle, Washington was initiated. The results of this exhaustive investigation has been published and titled "Corrosion of Servo Valves by an Electrokinetic Stream Current" authored by T. R. Beck, D. W. Mahaffey and J.H. Olsen, dated September 1969. In essence, T. Å. Beck, et al, stated the mechanism that eroded all orifices within the hydraulic system was "a current flow normal to the wall produced by the generation of electrical streaming current in an accelerating fluid flow and causes electrochemical corrosion damage of the metal".

They go on to further state that the damage is a consequence of a chain of events that may be eliminated by breaking the chain at any one of its links. The use of fuller's earth filters is an effective method of breaking the chain of events because fuller's earth filters remove the products of hydrolysis which generate the current within the fluid.

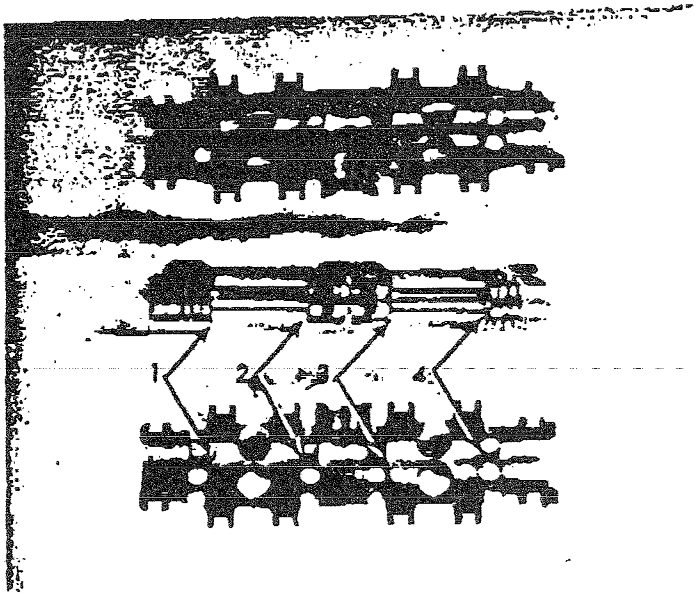
Almost immediately upon the decision to use phosphate ester fluids in turbine controls systems, it was found that serious material removal problems were encountered that would affect the ability of the system to perform its function. This was in the early sixties. Although the mechanism that caused this problem was not identified, it was determined that this problem could be controlled by using a continuous by-pass filtration system that incorporated a fuller's earth filter. Later in the late sixties the mechanism "wall current generated by an electron streaming current in an accelerating fluid flow cause electrochemical corrosion damage of the orifice edges was the most of problem. Again it was identified that use of fuller's earth filters would prevent the action from occurring.

It has also been noted that the resistivity of the phosphate ester fluid is reduced when the fluid is in a condition to react with orifice edges. Thus, it should be possible to identify fluids that have the capabilities of eroding metering orifice edges by monitoring the resistivity of the fluids.

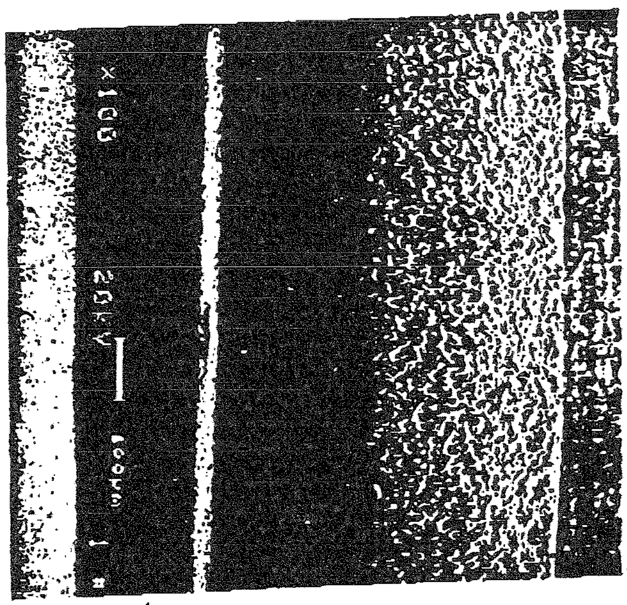
Sadly, today because so much of the original work on turbine control was done twenty to thirty years ago, we continue to see evidence of units that are operating on phosphate ester fluids being returned to the manufacturer for refurbishment with extreme cases of material erosion.

It epears tht the original information on the effects of phosphate ester fluids were not passed onto today's users.

See Figure 7 and Figure 8 for examples of the conditon of recently returned units. Note the great amount of erosion of the spool/bushing metering edges.



The damaged metering edges on the spool
and on the bushing interior.
1X



The spool metering land No. 1 (above)
showing damage on the right edge.
SEM 100X

The cause for the badly damaged control system components are well understood as stated above. The elimination of the damaging mechanism is also understood and easily implemented. Also, great strides have been made in filter media from what was available in the early sixties to what is available today. Today we can easily and cost effectively obtain filter media that can remove hard particle particulate down to .5 micron. Also, the adsorbent filter media has improved. In many cases activated alumina has proven to be more effective than fuller's earth as a media for removal of acid and products of hydrolysis from phosphate ester fluids.

It must be understood that erosive damage to turbine control components is of serious concern because of the potential for the loss of control which will result in an outage. But it is important to note that the potential erosive conditions of phosphate ester fluids can be easily controlled by using a continuous filtration scheme that incorporates a fuller's earth or activated alumina filter. Use of a continuous filtration system with the adsorbent filter media will eliminate the possibility of control component damage and system outage.

REFERENCES

(1) G. F. Wolfe and M. Cohen "Fire Resistant Fluids for Steam Turbine Electro-hydraulic Control Application" Paper No. 91, ASTM Annual Meeting, June (1967)

(2) T. R. Beck, D. W. Mahaffey and J. H. Olsen "Corrosion of Servovalves by an Electrokinetic Streaming Current", Boeing Scientific Laboratories Document D1-62-1839 September 1969

(3) S. Sadigh "Failure Analysis of Servovalve, Flapper, and Spool/Bushing Assembly" Moog Interdepartmental Document July 19, 1988

(4) C. Staley and B. McGuigan "The European Use of Phosphate Esters in Steam and Gas Turbines, Lubrication Engineering, Volume 33, 10 PP 527 - 534

(5) J. Qureshi "Failure Analysis of Moog Servovalves from Byron No. 2 Commonwealth Edison" November 25, 1989