MODELING OF PURGE VALVE CLOSURE

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ABSTRACT

In some nuclear reactor containment isolation applications, butterfly valves with pneumatic actuators are used to close pipelines in the event of containment pressure transients resulting from postulated accidents. A model is developed which predicts the closure time history of such valve/actuator systems under prescribed transients. Examples are presented which show the importance of the valve and actuator characteristics. It was found that actuation delays of sufficient magnitude could allow the dynamic torque to exceed the available actuator torque thus preventing valve closure.

NOMENCLATURE

\[ \begin{align*}
A &= \text{piston area} \\
A_p &= \text{projected disc area} \\
C_v &= \text{exhaust or vent valve discharge coefficient} \\
C_T &= \text{torque coefficient} \\
d &= \text{shaft diameter} \\
D &= \text{valve diameter} \\
F &= \text{force} \\
f_f &= \text{friction factor} \\
F_F &= \text{equivalent frictional force} \\
F_V &= \text{equivalent dynamic force} \\
F_0 &= \text{spring preload} \\
F &= \text{spring constant} \\
M &= \text{mass flowrate} \\
M &= \text{equivalent mass of moving parts} \\
\mu &= \text{polytropic coefficient} \\
P &= \text{pressure} \\
\Delta P &= \text{pressure drop across valve} \\
t &= \text{time} \\
T_F &= \text{frictional torque} \\
T_V &= \text{dynamic torque} \\
t_A &= \text{actuation delay} \\
V &= \text{volume} \\
x &= \text{linear position}
\end{align*} \]

Greek

\[ \begin{align*}
\rho &= \text{density} \\
\theta &= \text{angular position}
\end{align*} \]

INTRODUCTION

During certain postulated accidents in a nuclear power generating station, the pressure inside containment rises quickly. Emergency core cooling systems (ECCS) and containment spray systems (CSS) are activated within approximately one minute and fifteen minutes, respectively, to remove heat and control the pressure transient. Containment pressure response curves, such as Figure 1, are developed thermodynamically with due consideration of the specific plant configuration. Necessarily, various pipelines connect through containment and are fitted with containment isolation valves which must close to minimize releases to the environment in the event of an accident. Such valves, their actuators, and control systems are designed for fail-safe operation.

![Figure 1 Typical Containment Pressure Response](image-url)
Butterfly valves are a class of containment isolation valves characterized by the fact that they must isolate a compressible gas flow at essentially the pressure difference between containment and the environment. Radiological considerations are used to determine the tolerable maximum closure time of such valves through consideration of total release to the environment. Typically, the valves must be fully closed within 5 to 10 seconds of the initiation of the pressure transient with the actual time being plant specific. In some applications, high-performance butterfly valves (6 to 36 inches) are used in conjunction with a pneumatic actuator and mechanism which converts rotary shaft motion to linear actuator motion. Both scotch yokes and slider-crank mechanisms are used. The actuator consists of a pneumatic cylinder with piston and piston rod. A coil spring acts on the piston rod, tending to close the attached valve. Pressurization of the pneumatic cylinder compresses the spring and opens the valve. An exhaust valve is located on the pressure side of the pneumatic cylinder. It is held closed by pilot valve pressure which is provided in response to an electrical signal.

Removal of the electrical signal at the initiation of the containment pressure transient causes relief of the pilot pressure, allowing the exhaust valve to open. As air escapes through the exhaust valve, the spring tends to close the butterfly valve. Both dynamic torque due to the imposed containment pressure transient causing flow through the butterfly valve, and torque due to seal and bearing friction must, in net, be small enough, if opposing motion, that the spring can close the butterfly valve within the prescribed time. In addition, the dynamic torque may, at times, be in such direction and have such magnitude to assist closure. At other times, however, the dynamic torque may oppose closure. In this case, it becomes very important that sufficient valve closure is achieved before the containment pressure rises to a value which produces enough flow and hence dynamic torque to stall the actuator. In addition, there may be delays in the containment pressure sensing system and pneumatic control systems which, effectively, shift the valve operation schedule further along the containment pressure curve, thus, reducing the stall margin.

Typically, analyses of the closure process begin by assuming a valve position vs time relationship. For example, an assumed delay and then linear closure rate over the prescribed window is often used. A table or graph of the valve torque (dynamic and friction) as a function of position was then prepared. The two curves are then compared to determine that the available actuator torque is always greater than the valve torque required. The actuator/valve combination is usually operated without any through flow to verify that the actual delay is less than that assumed and that the stroke time is less than that assumed. It is then argued that such an analysis is conservative and that the valve will close within the required time window. While this may be so in cases where the actuator torque is always considerably greater than the valve torque required, it is not clear that it is adequate when the available torque is only slightly greater than required torque. Furthermore, such analyses give no information regarding the safety margin available in terms of actual valve closure time.

In general terms, the actuator available torque is known as a function of position while the dynamic torque is known as a function of time. The assumed time-position relationship allows the position to be determined. In actuality, however, the actuator is dynamically coupled and the escape of the fluid, the actuator governed by thermodynamics. Therefore, the study of the closure process in detail, the governing equations were developed for the system components and incorporated into a model which predicted the system closure dynamically.

**MATHEMATICAL MODEL**

The valve and actuator constitute a pneumatic/mechanical system, Figure 2. The air pressure on the pressure side of the cylinder compresses the spring and holds the valve open. On opening of the quick exhaust valve, this pressure is reduced and the spring moves the piston, closing the valve through the connecting linkage. Valve torque due to the flowing fluid (dynamic torque) and due to friction transmits force to the piston through the linkage.

![Figure 2: Valve and Actuator Mechanism Schematic](image)

**FIGURE 2 VALVE AND ACTUATOR MECHANISM SCHEMATIC**

During the closure process, the spring force acts to close the valve, while the net of the gas pressure (pressure side - vent side) retards closure. Valve friction force resists closure but the force due to valve torque may aid or retard closure. By convention, negative torque coefficients hinder closure while positive ones assist closure.

The valve closure process was modeled using three differential equations: one conservation of mass equation for the pressure side, one for the vent side, and the dynamic equation, Newton’s second law. The rate of change of mass, \( \rho V_p \), in the pressure side of the cylinder, is equal to the mass flow rate, \( m \), through the exhaust valve.

\[
\frac{d(\rho V_p)}{dt} = m
\]

Similarly, the rate of change of mass, \( \rho V_v \), in the vent side of the cylinder, is equal to the mass rate of flow through the vent valve.

\[
\frac{d(\rho V_v)}{dt} = -m
\]

Appropriate signs for the mass flow rates were assigned when the flow equations for the valves were written.
Newton's second law was written in terms of equivalent linear motion of the piston rod as

\[ M \frac{d^2x}{dt^2} = P - P_v - F_v - F_s - F_v + F_f \]  

(3)

where

- \( M \) = equivalent mass of moving parts
- \( P \) = pressure force, pressure side
- \( P_v \) = pressure force, vent side
- \( F_v \) = spring force
- \( F_f \) = force due to valve torque
- \( F_s \) = force due to valve friction

The gas in the pneumatic cylinder was assumed to undergo a polytropic process so that,

\[ P/P_n = \text{const} \]  

(4)

\[ P_v/P_n = \text{const} \]  

(5)

The exhaust valve mass flow rate was calculated using

\[ m_e = 1360 \frac{C_v \rho_{STP} P Y V X_T}{P} \]  

(6)

where

- \( C_v \) = discharge coefficient of quick exhaust valve
- \( \rho_{STP} \) = standard air density
- \( X = \frac{P}{14.7/P} \)
- \( Y = \frac{1 + X}{3X_T} \)
- \( X_T = 0.75 \)
- \( X < X_T \)

The vent side mass flow rate was similarly calculated using the vent side pressure, \( P_v \), and vent discharge coefficient.

Manufacturer's data for valve actuators usually consist of torque vs. \( \theta \) and torque factor \( F(\theta) \) due to the spring force and mechanical linkage. The force due to the dynamic torque on the valve disk was calculated as,

\[ F_v = TV/F(\theta) \]  

(7)

where

\[ TV = CT(\theta) \Delta P D^3 \]  

(8)

and

- \( CT \) = torque coefficient
- \( \Delta P \) = containment pressure drop across valve
- \( D \) = nominal valve diameter

The axial force due to bearing friction was similarly calculated using

\[ F_f = \Delta P f_f d/2 \]  

(9)

and

\[ F_f = TF/F(\theta) \]  

(10)

where

- \( TF \) = torque due to friction
- \( \Delta P \) = project disk area
- \( f_f \) = friction factor
- \( d \) = shaft diameter

The spring force was evaluated using

\[ F_S = Kx + F_O \]  

(11)

where

- \( K \) = spring constant
- \( F_O \) = spring preload

Using this relation in conjunction with the torque factor, and torque data vs. \( \theta \), the relationship between displacement, \( x \), and angular position, \( \theta \), was found, viz:

\[ Kx + F_O = TV(\theta)/F(\theta) \]  

(12)

or

\[ x = f(\theta) \]  

(13)

as was the spring constant and preload. Alternatively, this relationship could be determined from the mechanical linkage design, but the above information is more readily available.

The forces on the piston due to the pressures in the pneumatic cylinder were evaluated as:

\[ P = P_a A \]  

(14)

\[ P_v = P_a A \]  

(15)

where

- \( P \) = pressure, pressure side
- \( P_v \) = pressure, vent side
- \( A \) = piston area

Differential equations 1, 2, and 3 were simultaneously integrated through time using a FORTRAN computer code and 4th order Runge-Kutta integration scheme.

The computer code allowed input of all pertinent data including initial conditions and subsequently simulated the dynamic process of valve closure including position vs. time. Input data consisted of physical parameters, manufacturer's torque factor curves, the prescribed containment pressure transient, and torque coefficient curves. Auxiliary information such as dynamic torque or cylinder pressure could be similarly listed as well. Delay between the beginning of the containment pressure excursion and opening of the quick-exhaust valve was determined separately and input to the program as a parameter.

TORQUE COEFFICIENTS

Data describing most of the system components were readily available. It was found, however, that torque coefficient curves for butterfly valves in compressible flow differ from those for incompressible flow. Additionally, compressible flow data are quite scarce. This problem is compounded by the fact that torque coefficient curves can be seriously affected by valve orientation when elbows or other fittings are in close upstream proximity. For purposes of example in this paper, torque curves were adapted from the available literature as well as those deduced from blowdown tests. Additionally, attention was confined to those configurations where the valve shaft was oriented towards the downstream piping because this orientation results in negative (valve tends to open) torque coefficients especially near the fully open position (90 degree).
Figure 3, curve A, shows torque coefficients for an 18 inch valve operating in compressible flow in a straight pipe run. These coefficients were determined from an analysis of test data obtained at Wyle Laboratories (2) using a nitrogen blowdown facility.

A 18 INCH VALVE BLOWDOWN TESTS
USED FOR 20 INCH (t/c = 0.20)
IN STRAIGHT PIPE
B USED FOR 6 INCH (t/c = 0.29) IN
STRAIGHT PIPE
C USED FOR 20 INCH WITH
UPSTREAM ELBOW
D USED FOR 6 INCH WITH UPSTREAM
ELBOW
E TYPICAL FOR 20 INCH INCOMPRESSIBLE
FLOW IN STRAIGHT PIPE

CURVE A CALCULATED USING DATA FROM
REFERENCE 2. UNCERTAINTY UNKNOWN.

FIGURE 3 TORQUE COEFFICIENT VS. VALVE DISC POSITION

In these tests, a pressure regulator was used to simulate a typical containment pressure response curve. The angular position of the valve disc vs. time during closure was recorded. Although the inlet pressure was recorded, the mass flow rate was not, so that the inlet Mach numbers could not be found. The fluid dynamics of butterfly valves is dependent on the thickness of valve disc in relation to its diameter (aspect ratio). For this particular valve the ratio was t/c = 0.20. The torque coefficient curve for a similar valve in incompressible flow is shown as curve E. Distinct differences are seen in that for compressible flow, the values are always negative (valve tends to open) while for incompressible flow significant opening torque is only found near fully opened angles.

Measurements of torque coefficient were made by NASA/Langley (3) for 6 inch (t/c = 0.29) valves fed by upstream elbows. Some of the data are plotted in Figure 4, out-of-plane elbow feed, and Figure 5, in-plane elbow feed. Several curves are shown on each figure corresponding to inlet pressure. Although it was possible to estimate inlet Mach numbers from the measurements, no systemic dependence could be found. These data are similar to those for compressible flow in Figure 3 in that the torque coefficient is always negative. Unfortunately, measurements were not made for straight pipe runs so direct comparison cannot be made. They do show the same general shape but with slightly greater magnitudes at low angles and significantly greater magnitudes at large angles.

In general, it seems, that, for compressible flow, increasing the t/c ratio results in more negative torque coefficient curves and that feeding the valve with an elbow results in more negative torque coefficient valves at large angles as compared to a valve with the shaft located downstream in a straight pipe run. Based on this argument, several torque coefficient curves were developed for calculational pur-
poses. Curve A, of Figure 3, was used for straight pipe runs with valves having \( \frac{t}{c} = 0.20 \). Curve C, of Figure 3, was taken as 1.5 times curve A and used for valves of \( \frac{t}{c} = 0.29 \) in straight pipe runs. The magnitudes of these curves was substantially increased at angles between 70 degrees and 90 degrees, similar to Figures 4 and 5, to account for upstream elbows. These are shown as curves B (\( \frac{t}{c} = 0.20 \)) and D (\( \frac{t}{c} = 0.29 \)) in Figure 3.

RESULTS

Closure time histories were obtained for a 6 inch and 20 inch valve equipped with pneumatic actuators. The associated parameters are given in Table 1. In both cases the cylinder was pressurized to 100 psig at 70°F and the expansion process was assumed to proceed isothermally, \( n = 1 \). Figure 6 shows several closure time histories for the 6 inch valve while Figure 7 shows similar calculations for the 20 inch valve.

In all cases an initial vent time is required for the pressure to drop sufficiently that the valve actuator begins moving. Once this point is reached, valves usually close quickly owing to the negative slope of the coefficient curves. The initial vent time is directly related to the CT(90) values. As this increases in magnitude, the increasing torque to open due to the containment pressure transient causes the valve to hold-off time before the actuator moves.

Table 2 summarizes the sample calculations by giving the vent times and closure times as a function of torque curve and actuation delay. For the 6 inch valve, the effect of the torque curve assumed (i.e., whether straight or elbow feed) is not substantial. This happens because the actuator is relatively large for the valve. Actuation delays, as might be encountered in signaling, are nearly additive to the basic response for the same reason. For the 20 inch valve, however, the actuator is not substantially oversized. The base curve in Figure 7, \( \text{CT} \) = \( \text{"A"} \) and \( t_d = 0 \), shows the effect of disc and mechanism inertia both at the beginning and end of the stroke. The effect of upstream elbows is significant in delaying initial venting and closure. Actuation time delays are critical: a 0.25 sec delay almost doubles closure time while a 0.50 sec delay causes the actuator to stall so that the valve does not close.

CONCLUSIONS

For butterfly valves equipped with pneumatic actuators in service as containment isolation valves, several
factors are of primary concern when considering the adequacy of such valves to close under prescribed containment pressure transients. Actuation delays should be minimized and shown not to have significant impact on closure ability. The initial vent time is dependent on the initial pressurization of the actuator, the torque coefficient values near the fully opened position, and the exhaust valve capacity. It is obvious that the initial pressure and magnitude of the torque coefficient should be minimized and that the exhaust valve capacity maximized. Furthermore, the dynamic response of the valve/actuator system should be considered, as well, and the best available torque coefficient curves for the intended service, used.

REFERENCES

