Investigation of butterfly check valve with optimum closing performance

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ABSTRACT

This paper introduces the investigation of butterfly check valve with optimum closing performance, which was developed by Shanghai Petrochemical Complex and Tsinghua University for some feed water systems of power plant and Chemical plant. The valve is consisted of valve body, eccentric shaft disk with D_N 1200 diameter, governor balance weight hammer and swing damping oil cylinder. All parameters of it have been obtained from optimum $\tau \sim t$ curve for preventing the destruction of systems from waterhammer. The software of optimum calculation has been complied for solving and treating many complex equations and boundary conditions. The paper includes how to determine water impulsive force and momentum on the disk when power fails and pump stops. It is very important and difficult. The experiments of valve turning without the water and valve on industrial systems have been completed and coincide very well with calculation.

KEYWORDS

Butterfly valve, Waterhammer, Optimum performance.

NOMENCLATURE

A	= area of pipe	f	= Darcy–Weisbach friction factor
A_d	= area of opening of a valve	F_f	= friction force on disk
B	= pipeline constant $B = a / gA$	g	= gravitational acceleration
C_d	= orifice discharge coefficient	G_g	= weight of balance weight
d	= disk diameter	G_v	= weight of valve
D	= pipe diameter	H	= instantaneous piezometric head
e	= eccentricity of disk	H_R	= rated pressure head of turbomachine
		I	= polar moment of inertia of rotating parts
		L_v	= arm of force G_v

- m = mass
 M_F = friction moment on disk
 N_R = rated speed of turbomachine
 P_{s1}, P_{s2} = pressure head
 Q_R = rated discharge of turbomachine
 R = resistance coefficient of pipe
 t = time
 T_R = rated torque of turbomachine
 V = instantaneous velocity
 WR^2/g = polar moment of inertia of rotating parts
 τ = dimensionless number describing opening of valve
 δ = closing angle of valve disk
 α = pipe slope

INTRODUCTION

Check valve is a one of the important equipments of industrial pipeline systems. It is located on the output of pump. It can prevent reverse flow and protect pump against destruction when pump stops. General check valves are of big resistance, short service life and without closing governor, so they often cause waterhammer and destruction. For many years, we have been investigating valves which can prevent system damage from waterhammer. After swing check valve with property of preventing waterhammer being investigated, recently years the butterfly check valve with $D_N 1200$ has been developed for feed water of power plant coolant systems and chemical plant. The investigation of it is introduced in this paper.

The valve consists of valve body, streamlined disk with eccentric shaft, governor balance weight hammer and swing damping oil cylinder. The disk is designed thinner and stronger for smaller resistance.

After comparing with the scales of eccentricity $\frac{d}{4}, \frac{d}{5}, \frac{d}{6}$, by experiment, for getting smaller resistance, $\frac{d}{4}$ eccentricity has been chosen.

All parameters determined according to the optimum $\tau \sim t$ curve obtained from solving and treating many complex equations and boundary conditions. The paper introduces how to determine water simultaneous impulsive force and moment on the disk when power failure and pump stoping, which is a very important and very difficult problem.

The software has been compiled for calculation. The valve has been manufactured and installed on the systems. It can open rapidly when pump starts and has a less resistance when it operates and realizes two steps adjusted closing speed automatically when pump stops and power fails. The valve can prevent the systems destruction from waterhammer. When valve shuts up its strength and rigidity are safety under the systems pressure. The results of the experiments of both valve on manufacture and valve on industrial pipeline system are coincide very well with the results of calculation.

THE INTRODUCTION OF THE CONSTRUCTURE OF VALVE AND SYSTEMS

The construction of the valve is shown in Figure 1. (a), (b). The damping oil cylinder with swing cylinder body, is of adjustable damping rod and hole.

The configuration of pipeline systems in which the valve is installed is shown in Figure 2.

ESSENTIAL EQUATIONS AND BOUNDARY CONDITIONS

Basic Differential Equations for Transient Flow

Differential Equations:

$$gH_x + VV_t + V_t + \frac{f|V|V|}{2D} = 0$$

$$VH_x + H_t - V\sin\alpha + \frac{a^2}{g} V_x = 0$$

(a) front view

(b) side view

Figure 1. Configuration of the construction of valve. (a) is front view (b) is side view.

Figure 2. Configuration of pipeline valves installed in pipeline systems

Characteristics Equations:

$$C^+ : \begin{cases} \frac{g}{a} \frac{dH}{dt} + \frac{dV}{dt} + \frac{f V|V|}{2D} = 0 \\ \frac{dx}{dt} = +a \end{cases}$$

$$C^- : \begin{cases} -\frac{g}{a} \frac{dH}{dt} + \frac{dV}{dt} + \frac{f V|V|}{2D} = 0 \\ \frac{dx}{dt} = -a \end{cases}$$

Finite-Difference Equations:

$$C^+ : H_{Pi} = C_P - BQ_{Pi}$$

$$C^- : H_{Pi} = C_M - BQ_{Pi}$$

in which:

$$C_P = H_{i-1} + BQ_{i-1} - RQ_{i-1}|Q_{i-1}|$$

$$C_M = H_{i+1} - BQ_{i+1} + RQ_{i+1}|Q_{i+1}|$$

Equation for Pumps in Parallel.The Complete Characteristics of the Pump:

$$\begin{cases} WH(x) = h / (\alpha^2 + v^2) \\ WB(x) = \beta / (\alpha^2 + v^2) \\ x = \pi + \tan^{-1} \left(\frac{v}{\alpha} \right) \end{cases}$$

$$\text{in which } \alpha = \frac{N}{N_R}, v = \frac{Q}{Q_R}, \beta = \frac{T}{T_R},$$

$$h = \frac{H}{H_R}.$$

WH(x) and WB(x) are the complete characteristics. Pumps are of the same WH(x) and WB(x) when they are with the same specific speed n_s , ($n_s = 3.65 N_R \sqrt{Q_R} / H_R^{3/4}$).

Equations for Pumps in Parallel: Pumps in parallel is shown in figure 3.

Figure 3. pumps in parallel

For line ①:

$$F_{11} = HPM + BSQv + H_{R_1}(\alpha_1^2 + v_1^2)WH(x_1) - \frac{\Delta H_1 v_1 |v_1|}{\tau_1^2} = 0$$

$$F_{12} = (\alpha_1^2 + v_1^2)WB(x_1) + \beta_{01} - c_{311}(\alpha_{01} - \alpha_1) = 0$$

for line ②:

$$F_{21} = HPM + BSQ \cdot v + H_{R_2}(\alpha_2^2 + v_2^2)WH(x_2) - \frac{\Delta H_2 v_2 |v_2|}{\tau_2^2} = 0$$

$$F_{22} = (\alpha_2^2 + v_2^2)WB(x_2) + \beta_{02} - c_{312}(\alpha_{02} - \alpha_2) = 0$$

in which: $v = v_1 + v_2$, $HPM = C_P - C_M$, $BSQ = BQ_R$

$$c_{31} = \frac{WR^2}{g} \frac{N_R}{T_R} \frac{\pi}{15\Delta t}, \quad B = \frac{a}{gA},$$

a: pressure wave speed

Equations for Valve Moving

The valve could open, move or close according to $\tau \sim t$ curve by the action of momentums on disk shaft, such as water impulsive momentum M_H , weight hammer gravity momentum M_G , disk gravity momentum M_v , frictional momentum M_F and piston-rot resistance mo-

mentum M_T . Forces acting on the valve are shown in Figure 4.

Figure 4. Forces acting on the valve

Continuity Equation of Damping Oil Cylinder:

$$A_{s1} \frac{dy}{dt} = C_d A_d \left[\frac{2}{\rho} (p_{s2} - p_{s1}) \right]^{\frac{1}{2}}$$

Equation of Piston Motion

$$m \frac{d^2 y}{dt^2} = A_{s2} p_{s2} - A_{s1} p_{s1} + F_F - F_T - mg \cos \theta$$

Equation of Disk Rotation

$$I \frac{d^2 \delta}{dt^2} = M_v + M_G + M_H - M_F - M_T$$

in which:

$$M_v = L_v \cdot G_v \cdot \cos \delta$$

$$M_G = L_g \cdot G_g \cdot \sin(\delta + \alpha_g)$$

$$M_T = L_T \cdot F_T \cdot \sin(\delta + \alpha_T - \theta)$$

$$F_H = [1.12|(\delta - 0.175) / 0.689|^{2.1} + 0.54] \cdot \rho V^2 D^2$$

$$M_H = [-0.004|(\delta - 1.047) / 0.523|^{2.942} + 0.024] \rho V^2 D^3,$$

in which:

M_H gotten from the experiments for symmetry butterfly valve.

For butterfly valve with eccentricity shaft disk, M_H is as follows:

$$\text{advance flow: } M_H = (e - L) \cdot F_H$$

$$\text{reverse flow: } M_H = (e + L) \cdot F_H$$

in which: L is a acting point of F_H ,
 e is eccentricity.

Replanishing Equations

Sixteen equations of condenser and two equations of branch pipes are included.

Boundary Conditions

Up Stream: sea input pool level

Down Stream: (1) the level of power plant output pool. (2) the level of chemical plant output pool.

AIM OF OPTIMUM $\tau \sim t$ CURVE

The aim of optimum $\tau \sim t$ curve is as follows:

The valve could open rapidly when pump starts and is of a smaller resistance when it operates. It has a optimum $\tau \sim t$ closing curve with smallest waterhammer pressure. The softwares of optimum $\tau \sim t$ calculation compiled for solving the equations and treating the boundary conditions with FORTRAN LANGUAGE. The "Predetermine-Adjust" method has been used to solve equations for obtaining $\delta \sim t$. The $\tau \sim t$ curves are obtained from $\delta \sim t$ curve, $\xi \sim \tau$ relation and $\xi \sim \delta$, Re curve which

is gotten from experiments. One $\tau \sim t$ curve is obtained from one group parameters of valve. According to this $\tau \sim t$, groups of $H \sim t$ at every calculation point of pipes could be got. Comparing many $\tau \sim t$ and $H \sim t$ curves, choosing the $H_{\min}$ to search the $\tau \sim t$. That is a optimum $\tau \sim t$ curve, then all parameters of the valve can be determined.

COMPARE WITH EXPERIMENT

1. The $\tau \sim t$ curves of the experiments without the water at manufactory coincide with calculations very well

Figure 5. $\tau \sim t$ curve

Figure 6. $p \sim t$ curves

2. The $\tau \sim t$ curves of experiments on industrial pipeline systems with pumps coincide with calculations very well showing in Figure 5.

The $p \sim t$ curves of section 1 and section 2 are shown in Figure 6.

CONCLUSION

1. The investigation and the manufacture of the butterfly check valve are very successful.

2. The method of the optimum $\tau \sim t$ curve is of general significance.

3. The determination of F_H and M_H is very important and successful.

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