Water-hammer Calculation and Protection of Condensate Pump System

Jin Jiang[1], Xiaohong Weng[2], Xinhua Zhang[3], Xuefei Jiang[4]

[1]*Key Laboratory of Hydraulic Machinery Transient, Ministry of Education Wuhan University Wuhan, China, 430072*

[2]*School of Power and Mechanical Engineering Wuhan University Wuhan, China, 430072*

[3]*Daya Bay Nuclear Power Operations and Management CO., LTD*

[4]*Faculty of Engineering McMaster University Hamilton, Ontario, Canada*

jjiang@whu.edu.cn, wengxh_2005@yahoo.com.cn, zhangjiayi731@yahoo.com.cn, jxfzxyzey_@live.cn

Abstract: In this paper, the equations describing the gas-liquid two phase transient flow are derived. The mathematical and physical models of condensate pump system are established and various boundary conditions in the system are studied. The various water hammers of the condensate pump system are simulated and analyzed. The reasons of vibration alarm are analyzed when condensate pumps stop suddenly and operation conditions shift. The protection measures against water-hammer are studied. The results of simulation show that the accumulator of water-hammer prevention is an extremely effective method to eliminate pressure surge and valve vibration.

Keywords: pump system, water-hammer

1. Introduction

In the nuclear power station, water hammer phenomenon widely exists in the secondary loop [1],[2], especially on the condensate pump. The condensate pump's function is to extract the condensate water from the condenser. The water passes through seal cooler, low pressure heater and filter to oxygen remover. The main feed pumps supply with the water in oxygen remover for recycling. A small amount of condensation water is sent to the area needed for condensate to be cooled. In a nuclear power station or a thermal power station, condensate pump system does not allow the water to reflow. The condensate pump is equipped with a check valve, which may cause to greater water hammer when operation conditions of condensate pumps shift. Since a certain nuclear power station put into operation, its condensate pump vibration alarm has occurred many times, instant outage sound having associated with water hammer in check valve. Meanwhile, the pipelines have in some degree swayed. The vibration of pumps is small in normal operation condition, but the excessive vibration will trig vibration alarm when the pump stops and its operating condition shifts. After the shift of operation condition comes to an end, the vibration will restore to its original. In each pump exists the same imagination. Instant amplitude in check valves in the outlet is much larger than the amplitude in pumps. The main reason is that when the pump stops, the rapid closure in check valve causes pressure pulse. It will produce a large vibration and even damage to pipeline. More attention should be paid to this situation.

2. Basic Equations of Water Hammer Calculation

2.1. Basic Equations of Two-Phase Transient Flow in Pipe

2.1.1 Quality Equations [3],[4]

Gas phase: $$\frac{\partial}{\partial t}(\rho_g aA)+\frac{\partial}{\partial x}(\rho_g aAv)=\dot{m}A \quad (1)$$

Liquid phase: $$\frac{\partial}{\partial t}[\rho_l(1-a)A]+\frac{\partial}{\partial x}\rho_l(1-a)Av)=-\dot{m}A \quad (2)$$

Account the gas compressibility and the elastic effect of pipe, $K=\frac{dp}{d\rho/\rho}$, consider the elastic wall of pipe, changing rate of pipeline section area is associated with pressure in pipe, i.e.: $\frac{1}{A}\frac{dA}{dt}=\frac{DC}{E\delta}\frac{dp}{dt}$, and

$$(\frac{1}{K_g}+\frac{DC}{E\delta})\frac{dp}{dt}+\frac{1}{a}(\frac{\partial a}{\partial t}+\frac{\partial a}{\partial x})+\frac{\partial v}{\partial x}=\frac{\dot{m}}{\rho_g a} \quad (3)$$

Similarly:

$$(\frac{1}{K_l}+\frac{DC}{E\delta})\frac{dp}{dt}-\frac{1}{1-a}(\frac{\partial a}{\partial t}+v\frac{\partial a}{\partial x})+\frac{\partial v}{\partial x}=\frac{\dot{m}}{\rho_l(1-a)} \quad (4)$$

2.1.2 Mixture momentum equation

If ρ_m represents mixture density, which is equal to:

$$\rho_m=\alpha\rho_g+(1+a)\rho_g \quad (5)$$

Mixed phase flow equation can be written in the following formula, namely:

$$\frac{\partial}{\partial t}(\rho_m vA)+\frac{\partial}{\partial x}(\rho_m v^2 A)+A\frac{\partial p}{\partial x}+g\rho_m A\sin\theta+\tau_0\pi D=0 \quad (6)$$

$$\frac{\partial}{\partial t}[(1-a)\rho_l aA)+\frac{\partial}{\partial x}[(1-a)\rho_l v^2 A)+A\frac{\partial p}{\partial x} \quad (7)$$
$$+g(1-a)\rho_l A\sin\theta+\tau_0\pi D=0$$

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$$[(1-a)\rho_l A)\frac{\partial v}{\partial t}+(1-a)\rho_l vA)\frac{\partial v}{\partial x}+A\frac{\partial p}{\partial x}+g(1-a)\rho_l A\sin\theta+\tau_0\pi D=0 \quad (8)$$

Be divided by $(1-a)\rho_l A$, then:

$$\frac{dv}{dt}+\frac{1}{(1-a)\rho_l}\frac{\partial p}{\partial x}+g\sin\theta+\frac{f}{2D}v|v|=0 \quad (9)$$

2.1.3 Two-phase transient flow equations

Equations (2.3), (2.4), (2.9) can be written as follows:

$$\begin{bmatrix}1&0&0\\C_2&1&0\\0&0&1\end{bmatrix}\begin{bmatrix}\frac{\partial\alpha}{\partial t}\\\frac{\partial p}{\partial t}\\\frac{\partial v}{\partial v}\end{bmatrix}+\begin{bmatrix}v&0&-C_1\\C_2v&v&0\\0&C_3&v\end{bmatrix}\begin{bmatrix}\frac{\partial\alpha}{\partial x}\\\frac{\partial p}{\partial x}\\\frac{\partial v}{\partial x}\end{bmatrix}=\begin{Bmatrix}b_1\\b_2\\b_3\end{Bmatrix} \quad (10)$$

In which

$C_1=a(1-a)(\frac{1}{K_g}-\frac{1}{K_l})[\frac{DC}{E\delta}+\frac{a}{K_g}+\frac{(1-a)}{K_l}]^{-1}$;

$C_2=[a(1-a)(\frac{1}{K_g}-\frac{1}{K_l})]^{-1}$; $C_3=\frac{1}{(1-a)\rho_l}$;

$b_1=\dot{m}[(1-a)\rho_l(\frac{1}{K_l}+\frac{DC}{E\delta})+a\rho_g(\frac{1}{K_l}+\frac{DC}{E\delta})$;

$[\rho_g\rho_l(\frac{DC}{E\delta}+\frac{(1-a)}{K_l}+\frac{a}{K_g})]^{-1}$

$b_2=\dot{m}[\frac{1}{\rho_g a}\frac{1}{\rho(1-a)}][\frac{1}{K_g}-\frac{1}{K_l}]^{-1}$; $b_3=-g\sin\theta-\frac{f}{2D}v|v|$.

2.2. Pump Boundary Conditions during Sudden Stop

During fluid transition caused by sudden stop of pump, the boundary condition depends on the complete characteristic curves of pumps, pump head equilibrium equation and the unit inertia equation.

2.2.1 Complete characteristic curves of pumps

Pump features in various different operating conditions can be represented by four feature parameters, i.e. the dimensionless pump head $h=H/H_R$, the dimensionless flow rate $v=Q/Q_R$, dimensionless speed $\alpha=n/n_R$, dimensionless torque $\beta=T/T_R$.

The pump characteristic curves are expressed by Suter curves. $x=\pi+\arctan(v/\alpha)$ is used for horizontal coordinates, vertical coordinates with $WH=h/(\alpha^2+v^2)$ and $WB=\beta/(\alpha^2+v^2)$. The pump transient lift can be as with Suter curves [5]:

$$H=hH_R=H_R(\alpha^2+v^2)WH(\pi+\arctan(v/\alpha)) \quad (11)$$

As WH curve is stored in the form of 89 sets of data, the linear interpolation method can be used to calculate the transient lift:

$$H=H_R(\alpha^2+v^2)[A_0+A_1(\pi+\arctan(v/\alpha))] \quad (12)$$

In which

$A_1=[WH(i+1)-WH(i)]/\Delta x$; $A_0=WH(i+1)-iA_1\Delta x$;
$i=[x/\Delta x]+1$.

2.2.2 Pump pressure balance equation

Pump pressure shall meet the following conditions, neglecting the loss of water under the conditions of a short tube.

$$H=H_{P1}+H_f-ELS=C_M+BQ_{P1}+H_f-ELS \quad (13)$$

In which Q_{p1} is transient flow of pump: $Q_{p1}=Q_R v$; H_f is valve head loss in pump outlet, $H_f=(H_{f0}/\tau^2)v|v|$, H_{f0} is head loss when valve is completely open and the flow is $C_M=H_2-BQ_2+RQ_2|Q_2|$.

2.2.3 Pump inertia equation

During pump stopping, rotating speed of pump rotor reduces and the reducing rating depends on moment of inertia and transient torque of pump. The equation of present pump torque and speed variations is unit inertia equation. It is another characteristic equation of pump boundary conditions.

From the moment of momentum theorem, after pumps stop, the relationship between torque and rotational accelerometer can be expressed as:

$$\alpha_i-\alpha=\frac{\Delta t}{2K}\beta_i+\frac{\Delta t}{2K}(\alpha^2+v^2)[C_0+C_1(\pi+\mathrm{arctg}\,v/\alpha)] \quad (14)$$

Where: α_i,α transient dimensionless speed of beginning and end of time periods Δt, respectively; β_i, transient dimensionless torque of start of time periods Δt; K inertia constant of pump rotor, $K=\pi GD^2/(120g)$; GD^2 moment of inertia of pump rotating parts; $C_1=[WB(i+1)-WB(i)]/\Delta x$; $C_0=WB(i+1)-iC_1\Delta x$; $i=[x/\Delta x]+1$.

2.3. Other Boundary Conditions [6]

2.3.1 Check Valve

For an accurate treatment, the valve head loss as a function of discharge may be found from valve tests, then this head loss may be taken as of the beginning of Δt, using Q(1).

To find the criteria for positive flow passing through the pump and check valve, set v=0 in pump head balance equation.

$$F3=HCP-HCM+H_R\alpha^2*WH(\frac{\pi}{\Delta t}++1) \quad (16)$$

If F3 is greater than zero positive flow occurs, otherwise v=0 due to the check valve.

2.3.2 Salt Remover

Salt remover is a closed container equipped with a powdered resin. Its inner flow is complex. Salt remover is fine filters. In hydraulic calculation, it is described by resistance unit, $\Delta H = fQ^2$, f is drag coefficient.

2.3.3 Oxygen Remover

Oxygen remover is needed to remove oxygen dissolved in water supply systems and other non-condensed gas, in order to prevent equipment corrosion and heat damage to heat transfer. In the fixed pressure, water is heated to boiling point so that the steam partial pressure is almost equal to the surface pressure. It is generally lower than the saturation partial pressure of gas, dissolved in water. So other gases, dissolved in water, will be saturated and released off. Since the interval step of calculation is very short, the pipe is rather longer between condensate pump to the oxygen remover and there is a bigger connection tube in the outlet of check valve. So the rather stable pressure in oxygen remover is assumed in the calculation.

Condenser in this calculation is located in the upstream of condensate pump, the inlet pressure of the condense pump is 75mbar.

3. Water Hammer Calculation and Protection

3.1. System Components and Parameters

There are three condensate pumps in each unit. Two of them are in operation and the third one of them is spare. Specific performance parameters are:

Rated flow: 2000m^3/h, rated shaft power 1404KW efficiency of 83%, rated lift 215m;

Zero discharge water temperature of 42 degrees head 326m net positive suction head required is 5.6m speed 1482r/min Motor power 1525KW.

There are three-stage vanes in rotation parts of condensate pump. The first stage is double-suction vane used to reduce the net positive suction head. The condensate pump system for calculation includes: condense pump, check valve, control valve, pipe connection tube, Seal Cooler, low-pressure heater, filter, and oxygen remover.

3.2. Calculation of Water Hammer

The simulation of waterhammer includes stop of pump sudden and shifts of operating conditions.

The following operating conditions are simulated; the results are shown in Fig. 1- Fig. 5.

(1) A single pump runs and stops suddenly;

(2) A single pump runs and stops suddenly; meanwhile the other one starts;

(3) Two pumps run and one of them stops suddenly;

(4) Two pumps run and one of them stops suddenly, while the other one starts;

(5) Two pumps run and both of them stop simultaneously.

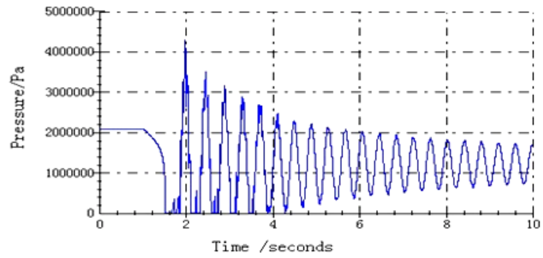

Figure 1. A single pump runs and stops suddenly

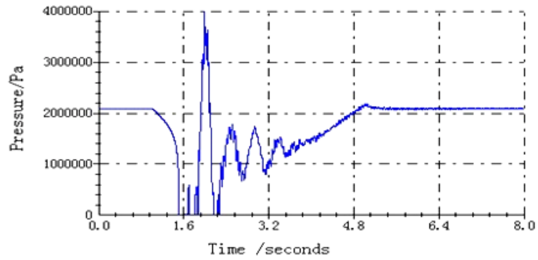

Figure 2. A single pump stops and the other one starts

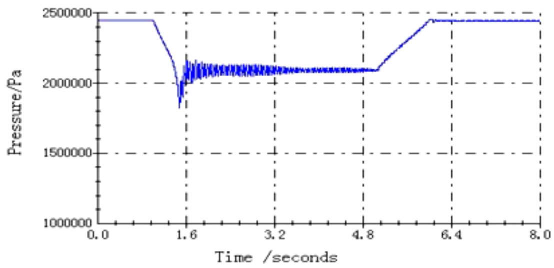

Figure 3. Two pumps run and one of them stops

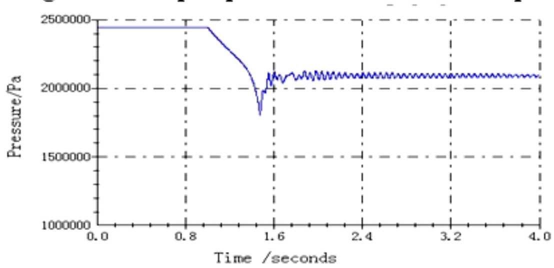

Figure 4. Two pumps run and one of them stops suddenly, while the other one starts

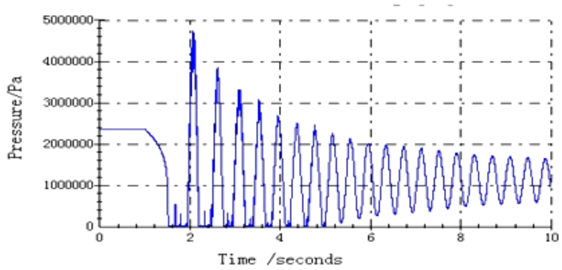

Figure 5. Two pumps run and both of them stop simultaneously

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When one pump stops suddenly, the greater water hammer pressure will occur. The max pressure reaches 4.3MPa in check valve and a more intense pressure oscillation. For conditions (3) and (4), there is mild pressure surge. For the last condition in which two pumps stop suddenly, it is the most dangerous situation. The max pressure will arise to 4.8MPa and more severe pressure oscillation will occur.

3.3. Water Hammer Protection

The rather high pressure surge in condense pump system is unfavorable for safe and stable operation of the system. Some protection measures must be taken to eliminate larger severe water hammer pressure and pressure oscillations.

To reduce the harm to condensate pump system done by water hammer, a new check valve that performs programming control can be used. Besides cost, the valve structure is rather complicated and its usage reduces system reliability. If the control system is out of order and causes pump to reverse runaway, the secondary damage will be done [7].

As the pressure surge arises in condense pump system, considering safety and reliability requirements, an accumulator shall be used for water hammer protection. Accumulator elements can effectively reduce the water hammer, stabilize pipe pressure and reduce variation.

The air vessel is connected to the main pipeline by a short pipeline. In this analysis the lumped inertia model, including friction, is used in the connector. Compatibility equation is applied to the connector.

Two kinds of accumulators are used for protection against water hammer. The large accumulator has better protective effect, but the small one can also meet the requirements. The diameters of two accumulators are respectively 0.5 m, and 1 m and the heights of two accumulators are respectively 1m and 2m. The referent water depths of accumulators are 75% - 80% of total heights.

The calculation results of small accumulators are illustrated as shown in Fig.6-Fig. 10.

4. Conclusion

The check valves of condensate pumps cause higher water hammer pressure and intense vibration and even the pipes to rock. The vibration may not cause the damage to pump and valve, but it would arouse the vibration alarm. And the max water-hammer pressure does not exceed the design pressure, given condensate pump is safe according to the design. But after a long operation and pipe aging, the pipe performance would degrade. Therefore, the protection measure should be taken to reduce water-hammer pressure and vibration. The accumulators prove to be effective on reduction of the harm.

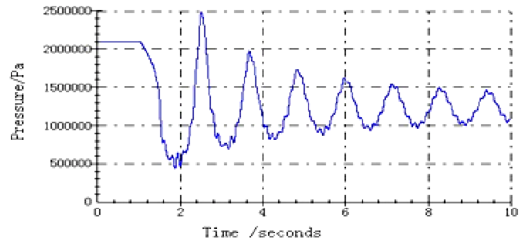

Figure 6. A single pump stops suddenly with accumulator

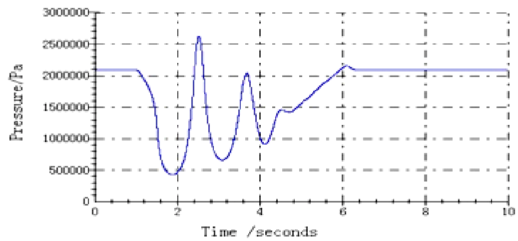

Figure 7. A single pump stops and the other one starts with accumulator

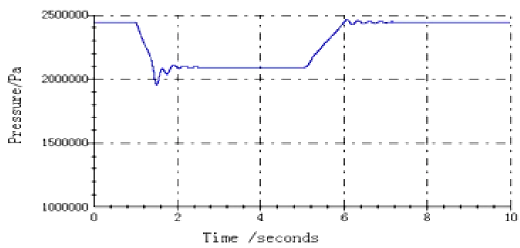

Figure 8. Two pumps one of them stops with accumulator

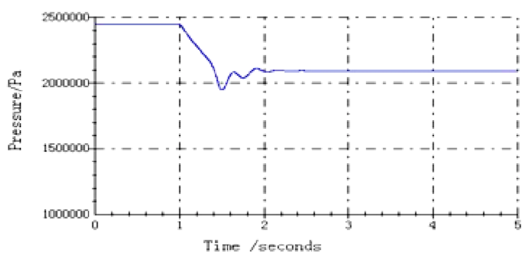

Figure 9. Two pumps run and one of them stop suddenly, while the other one starts with accumulator

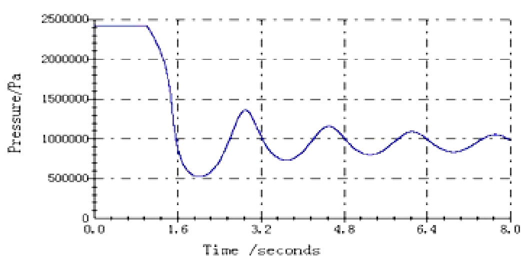

Figure 10. The two pumps run and both of them stop simultaneously with accumulator

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Nomenclature

A	area of pipe
α	void rate
α_w	wave velocity
C	pipeline capacitance
D	pipe diameter
E	modulus of elasticity
f	Darcy-Weisbach friction factor
g	gravitational acceleration
h	dimensionless pressure head
H_R	rated pressure head of turbomachine
K	bulk modulus of elasticity
$\dot{m}$	rate of mass of gas released
p	pressure
T	instantaneous torque on pump
T_R	rated torque on pump
v	dimensionless velocity
$WB\ WH$	dimensionless turbomachine characteristics
x	distance along pipe from left end
α	dimensionless speed ratio
β	dimensionless torque ratio
δ	wall thickness
θ	pipe slope
ρ	mass density
τ	dimensionless open of valve
τ_0	wall shear stress

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