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Study on Automatic Surging Prevention of Blower*

By Shigeru TSUJI**

The author published earlier the study on surge-prevention of blowers with the method of "Surge-Preventing Valve Operation".

In this report there are shown various types of operation to prevent surge by operating the surge-preventing valve with oil hydraulic controller which is of Askania type and are given the data which are necessary for designing of preventing equipments.

There are also shown experiments on stability and adulation where an automatic surge-preventing unit is installed on the blower made for testing. The author tests how it influences upon hunting and adulation of the control unit to damp the flow in the lead pipes to the controller and to the meter. In the case of the damped flow to the controller the stability of the unit increases while adulation decreases. Throttling of the valve in order to eliminate hunting influences little upon adulation.

Introduction

The author published his study in experimental and theoretical analysis for surge-prevention of blowers before⁽¹⁾. For this practical application in industry, it is necessary to regulate automatically the surge-preventing valve and to operate the blower with speed, stability and accuracy according to variations of capacity.

In this paper the author has studied on the preventing method of surging through the operation of surge-preventing valve by Askania type controlling unit on the blower. The automatic controller made for testing of surge-prevention has been applied to the experimental and theoretical study of relationship between constants of controlling unit and characteristics for surge-prevention of the blower, and stability and adulation of the apparatus.

1. Automatic controller for surge-prevention

1.1 Automatic controller for surge-prevention

Blow off valve type is the commonest of surging controllers of blowers that have been used. The blower for this type, where the required capacity is within the surging limit, reduces the capacity to blow away into the air or into the suction pipe in order to keep any running point

on the head-capacity curve without the surging limit. This type has a more simplified design and an accurate performance, but not a good running efficiency. For this equipment it has been tested⁽²⁾ to recover the energy issued from air turbine, but this has not been so much made use of because of its complexed design.

In addition to this there are a few other examples⁽³⁾, but these have not been utilized, for some have so complicated mechanism and others have many difficulties in operation.

Therefore the method to prevent surge on the blower by means of automatic control surge-preventing valve that the author has prevented, will be expected to satisfy the needs of industry if we can solve the problems on simplicity and running efficiency of the controller.

1.2 The principles of the design of this surge-preventing automatic controller

The important things in design of automatic controller for surge-prevention are of the following five :

- (1) The over-all head-capacity characteristic curves of the blower.
- (2) Constants on the pipe line system and the position to provide the surge-preventing valve.
- (3) Revolution-torque characteristics of the motor.
- (4) Duct capacity between the surge-preventing valve and the capacity control valve.
- (5) The maximum and the minimum resistance curves for surge-prevention of the

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surge-preventing valve.

Besides we should consider the most effective of the conditions to treat the blower. The following are the details of the above problems.

(1) We can obtain experimentally the head-capacity curve by means of the method above mentioned. When we can guess accurately the head-capacity curve on the data given from the previous experiments, we may obtain an approximate value.

(2) We should pay attention to the following items: the types of blower, their pipe line systems, the positions of the loading equipments and the types of automatic controllers.

(3) The torque characteristics of the motor are particularly influenced in the range of the critical surging limit where the differential value $\partial H/\partial Q = \phi$ is small on the head-capacity curve.

(4) We should especially estimate the minimum volume required for surge-prevention in the pipe line systems. Where the duct volume is not enough, it is necessary to set an air tank on a branch pipe.

(5) The minimum resistance values for surge-prevention on the surge-preventing valve can be obtained as critical points in the stable operation in which we test to get the over-all head-capacity characteristic curve.

1.3 Automatic controller for surge-preventing

Fig. 1

Steady state can be maintained by supplying the pipe with a valve head loss h_{d1} through automatic control operation, which comes about by providing control signals in response to the conditions of the blower. In the following there are a few types of general oil hydraulic controllers.

1.3.1 Flow rate type

This type operates the controller by measuring the variable volumetric rate of flow in accordance with the opening of the capacity control valve. Whatever deviation of the resistance may happen to the system of loads, the surge-preventing valve can be run properly to capacity fluctuation, so it will be operated accurately regardless of the kind and performance of the load. Therefore it is useful in capacity control with several valves or in controlling individually the capacities where a pipe in a blower is branched into many kinds of pipings to the load. Arrow heads of the scheme in Fig. 1 show the directions of the operations in which the capacity reduces and the opening of the surge-preventing valve decreases. The surge-preventing valve may be controlled in order to obtain the values resumed concerning the capacity, which is the static pressure in the inlet nozzle.

1.3.2 Constant pressure difference type

This is the method to control to the predetermined values of the pressure difference between both sides of the surge-preventing valve.

Fig. 2

As this type is illustrated in the previous report, it is necessary to take a curve, $a'b'c'$ or $a''b''c''$, which stands for any curve in the range between the minimum resistance value of surge-preventing valve, abc , and the maximum one, de , in Fig. 2. Where $a''b''$ can be given as a resistance curve in the condition of the constant opening of the valve, and running on the line $b''c''$ can be considered as an operation of constant pressure difference type that keeps a constant resistance through the surge-preventing valve. Even if this constant pressure difference type is applied without surging zone, there will be no inconvenience, for not only $h_{d\min}$ is small but also head loss of the capacity control valve is large.

Fig. 3 illustrates the hydraulic mechanism of the controller of the surge-preventing valve in order to trace the curve $a''b''c''$ shown in Fig. 2. These equipments which the author has designed for testing will be described in the following chapter.

2. Automatic surge-preventing controller for testing

The author has planned a controller to keep constant pressure difference on both sides of the surge-preventing valve, by applying Askania type oil hydraulic controlling unit to Blower B that was used in the previous experiments. The blower, the pipe line, the surge-preventing valve and the

Fig. 3

set position, the measurements of simultaneous pressure and the oscillogram are the same as used in the previous report. The scheme of the apparatus is shown in Fig. 4. The control leads the pressure on both sides of the valve to the diaphragm chamber of the meter where midway on the pipe a pinch cock is installed in order to regulate stability of the meter. A jet pipe is moved by the pressure difference between both sides of the diaphragm chamber at balancing with the spring, and moves servomotor through oil pressure difference between receiver ports, and then actuates the operating lever of surge-preventing valve in order to keep the set values.

3. Results of experiments

We have experimented on adulation, stability and hunting characteristics of the controller in each case of the variations of set points of the surge-preventing valve. To solve these problems we have experimented the influence upon the constants of the controller by throttling the pinch cock in each lead piping.

3.1 Adulation

(a) As a result of experiments on stability and adulation, actuating automatic controller while surging phenomenon was occurring in the blower and pipe system, we could accomplish the surging prevention accurately with about five second delay. This fact shows that this control unit makes it possible to operate surging prevention without disturbing the capacities of the meter even if there is external disturbance of surge-

Fig. 4

pressure fluctuation.

(b) Here is an experiment on adulation in the case where running state of the blower is controlled so quickly and falls in surging limit capacity, that is, the capacity decreases from 16 m³/min to zero point for a second or two with the capacity control valve where it takes several seconds to increase the opening of the surge-preventing valve to suitable valve, though during that time there occurs surging for a while. The lag time influences much upon the valve resistances on the lead pipe to the actuator. A small throttle of the cock brings a good adulation but tends to make hunting. Fig. 5 shows a comparatively typical example.

(c) We have experimented on the set point of the surge-preventing valve which will vary the adulation. As Fig. 6 shows, in the case of the suction pipe line, it takes two seconds while on the delivery side it takes four (Refer to Fig. 5). Accordingly where the surge-preventing valve is set on the suction side, the operation has more

adulation and stability because of low working pressure.

(d) In damping the flow as surging is eliminated, it is an important problem how long it takes for possible operation to occur. The experiment has been made with attention to these problems. Where we regulate the operating speed, lest surging should occur, we have attained this object by operating the capacity control valve for 6.3 seconds as Fig. 7 shows; such an operating speed can be considered to satisfy the practical conditions in use on valve operation of blowers.

3.2 Hunting

(a) We can reduce hunting by damping the flow with the cock on the lead pipe to the meter; as a typical example, Fig. 8 shows these relations. But where there is no throttle in the meter in the case of the unit II, The fluctuation is 3° and the period is 1.1 seconds. In the case of the unit I, which has different values of diameter or spring constants on the diaphragm from those of unit II, the fluctuation is 2° and the period is 1.6 seconds.

Fig. 5

Fig. 6

Fig. 7

Fig. 8

(b) Where we damp the flow with the cock on the lead pipe to the actuator, we can eliminate hunting, but at the same time we can't prevent adulation from reducing. Now we throttle more and more the cock of the actuator system in which hunting is occurring at capacity $3 \text{ m}^3/\text{min}$ till hunting almost decreases. Fig. 9 shows the records concerning these conditions. We gradually throttle the cock, where hunting of the surge-preventing valve is occurring with the amplitude 4° in the state of the full-opened cock and the amplitude 1.5° in half-opened state, and then there occurs no hunting where the opening is one forth of it.

(c) A performance of hunting is related to the capacity at the equivalent point within the surging limit. Fig. 10 shows the case where the capacity lowers quickly from $16 \text{ m}^3/\text{min}$ to $3 \text{ m}^3/\text{min}$. As the diagram shows, hunting does not occur at capacity $16 \text{ m}^3/\text{min}$ (the opening of the preventing valve is 55°), but there occurs hunting at capacity $3 \text{ m}^3/\text{min}$ (the opening is about 20°). This is indeed caused not by the external disturbance for there is no surging in the case, but by the stability of the control system itself. The results of the experiment show that it is stable at more than $10 \text{ m}^3/\text{min}$. The fact is explained theoretically in the following chapter.

(d) We set the resistance value h_d of surge-preventing valve too small, where running

points of the blower tend to cause a little surging periodically, while hunting occurs on the preventing valve, whose conditions are illustrated in Fig. 10. But if such a condition happens where the critical period is very different from the surging one, the control is not so much influenced and surging will not occur suddenly.

4. Theoretical analysis of the automatic surge-preventing controller

4.1 System of process

With the surge-preventing valve provided on the delivery pipe line, we use the unit II as a controller. Thereupon we can introduce the equation of the pressure fluctuation according to the opening variation of surge-preventing valve (Fig. 11).

P_i : pressure in i th system

x : coefficient proportional to opening of surge-preventing valve

q : capacity

where suffix 0 is the value at each equilibrium state and α or β is a certain coefficient.

$$q_0 = \alpha \cdot x_0 \sqrt{P_1 - P_0} = \beta \cdot y_0 \sqrt{P_0 - P_n}$$

Now if we assume deviation from x_0 to x , we can consider nearly constant P as there is a little variation in the flow.

The fluctuation of inlet capacity q_{in} is

On delivery pipe line,
Throttling cock in the oil pipe to $1/4$ turn from $7\frac{1}{2}$ turn,
Throttled valve in the measuring lead pipe at 1 turn,
Capacity $16 \text{ m}^3/\text{min} \rightarrow 3 \text{ m}^3/\text{min}$
Set point 40 mmAq

Fig. 9

On delivery pipe line,
Capacity $3 \text{ m}^3/\text{min}$,
Throttled valve in the measuring lead pipe at $3/4$ turn,
Set point 40 mmAq

Fig. 10

Fig. 11

obtained by the equation,

$$q_{in} = \alpha \cdot x \sqrt{P_1 - P_0} - \alpha \cdot x_0 \frac{P}{2} \frac{1}{\sqrt{P_1 - P_0}}$$

Similarly, we obtain the fluctuation of outlet capacity q_{out} ;

$$q_{out} = \beta y \cdot \frac{P}{2} \frac{1}{\sqrt{P_0 - P_n}}$$

Pressure rise causes the difference between the inlet and the outlet. Then we have

$$\frac{V}{P_0} \frac{dP}{dt} = q_{in} - q_{out}$$

where V is the volume between both valves,

Then

$$\frac{\frac{V}{P_0}}{\frac{\alpha \cdot x_0}{2\sqrt{P_1 - P_0}} + \frac{\beta \cdot y_0}{2\sqrt{P_0 - P_n}}} \cdot \frac{dP}{dt} + P = \frac{\alpha \sqrt{P_1 - P_0}}{\frac{\alpha \cdot x_0}{2\sqrt{P_1 - P_0}} + \frac{\beta \cdot y_0}{2\sqrt{P_0 - P_n}}} \cdot x$$

The time constant T of the fluctuating pressure P to x is

$$T_p = \frac{\frac{V}{P_0}}{\frac{\alpha \cdot x_0}{2\sqrt{P_1 - P_0}} + \frac{\beta \cdot y_0}{2\sqrt{P_0 - P_n}}} = \frac{2V}{P_0 q_0} \cdot \frac{(P_1 - P_0)(P_0 - P_n)}{(P_1 - P_n)}$$

Then substituting the actual values on the apparatus into this equation, we have

$$T_p = \frac{2 \times 0.29}{0.2 \times 11930} \cdot \frac{100 \times 1600}{1700} = 0.023 \text{ (sec)}$$

where

$$V = 0.29 \text{ m}^3$$

if the opening of the surge-preventing valve is assumed about an angle of 30° , we have

$$q_0 = 0.2 \text{ m}^3/\text{sec}$$

$$p_0 = 10330 + 1600 \text{ mmAq}$$

$$p_1 = 10330 + 1700 \text{ mmAq}$$

$$p_n = 10330 \text{ mmAq}$$

Then it is found that the time constant in this case is a very small value. And the proportional sensitivity K_p is

$$K_p = \frac{\alpha \sqrt{P_1 - P_0}}{\frac{\alpha \cdot x_0}{2\sqrt{P_1 - P_0}} + \frac{\beta \cdot y_0}{2\sqrt{P_0 - P_n}}} = \frac{2\alpha \sqrt{P_1 - P_0}(P_1 - P_0)(P_0 - P_n)}{q_0(P_1 - P_n)} = 1.28 \times 10^3 \text{ mmAq}$$

By Laplas change the process gain $G(s)$ is shown as follows :

$$G_{1(s)} = \frac{K_p}{1 + T_p \cdot S}$$

4.2 Controller gain

Approximate values of the velocity of the manipulating cylinder for the pressure difference between both sides of the surge-preventing valve, are shown as the following equations :

$$\frac{dz}{dt} = \pm 30.6X \quad |\Delta P| > 2 \text{ mmAq}$$

$$\frac{dz}{dt} = 30.6 \frac{\Delta P}{z} X \quad |\Delta P| < 2 \text{ mmAq}$$

where the dimensions of the manipulating cylinder are as follows : inner diameter 50 mm, stroke ($z_{\max}$) 125 mm, maximum oil capacity about $60 \text{ cm}^3/\text{sec}$, maximum velocity 30.6 mm/sec , and coefficient variable from 0 to 1 with the throttle valve.

The experiment shows that the controller delay is about 0.01–0.02 sec, and K_c is calculated 1.01, then the controller gain $G_{2(s)}$ is given by :

$$G_{2(s)} = \frac{K_c}{S}$$

4.3 Meter of the diaphragm type

We now take up the problem of the delay caused from the lead pipe from the pressure pick-up to the diaphragm. We may get the equation in condition of equilibrium on the outside term :

$$P_{out} \cdot V_{out} = (P_{out} + dP_{out})(V_{out} - dV_{out})$$

where

P_{out}, P_{in} : pressure

V_{out}, V_{in} : volume

q_{out}, q_{in} : capacity

"Out" or "in" of the suffix corresponds to "outside" or "inside" of the diaphragm chamber. Therefore

$$V_{out} \cdot dP_{out} = P_{out} \cdot dV_{out} = P_{out} \left(q_{out} \cdot dt - \frac{dP_{out} - dP_{in}}{S} A^2 \right)$$

where

$$\frac{V_{out}}{P_{out}} \cdot \frac{dP_{out}}{dt} = q_{out} - \frac{A^2}{S} \left(\frac{dP_{out}}{dt} - \frac{dP_{in}}{dt} \right)$$

S : spring constant

A : effective area of diaphragm

And similarly, the condition on the inside may be written as follows :

$$\frac{V_{in}}{P_{in}} \cdot \frac{dP_{in}}{dt} = q_{in} - \frac{A^2}{S} \left(\frac{dP_{in}}{dt} - \frac{dP_{out}}{dt} \right)$$

and

$$q_{out} = \frac{P_1 - P_{out}}{R_{out}}, \quad q_{in} = \frac{P_0 - P_{in}}{R_{in}}$$

where R_{out} and R_{in} are concentrative constants of damping factor in the lead pipe.

Substituting q_{out} and q_{in} in the equations above, and if we take only a deviating value, we have

$$\begin{aligned} \delta(P_{out} - P_{in}) &= \frac{\left(1 + SR_{out} \cdot \frac{V_{out}}{P_{out}}\right) \delta P_0}{1 + S \left[R_{in} \left(\frac{V_{in}}{P_{in}} + \frac{A^2}{S} \right) + R_{out} \left(\frac{V_{out}}{P_{out}} + \frac{A^2}{S} \right) \right] + S^2 R_{out} R_{in} \left[\frac{V_{in} V_{out}}{P_{out} P_{in}} + \frac{A^2}{S} \left(\frac{V_{out}}{P_{out}} + \frac{V_{in}}{P_{in}} \right) \right]} \\ &\equiv \frac{1 + CS}{1 + B_1 S + B_2 S^2} \delta P_0 \end{aligned}$$

therefore the gain $G_{(s)}$ is as follow :

$$G_{3(s)} = \frac{1 + CS}{1 + B_1 S + B_2 S^2}$$

At this operating point, we could obtain each value by substituting actual values in the equation.

$$C = 0.006, \quad B_1 = 0.952, \quad B_2 = 0.00568$$

Fig.12 shows the curve which is the plotting of each value of gain or phase angle on Bode diaphragm. As shown in Fig.12 in the frequency response at the meter of the diaphragm type, the magnitude curve crosses the phase angle of 90 degree line at frequency of 5.7 rad/sec.

It has been made clear that this fact agrees approximately with the fluttering frequency 5.7 rad/sec given from the experiment, and that the cause of time lag of this controller consists in the diaphragm.

Fig. 12

4.4 Limit cycle

For the process of this control system, the comparative relationship between the piston position and pressure difference has a higher gain, so that the system tends to become unstable. The block diagram for this system is shown in Fig. 13.

where

$$T_p = 0.023, \quad C = 0.006 \text{ sec}, \quad B_1 = 0.952 \text{ sec}, \\ B_2 = 0.0067 \text{ sec}$$

therefore T_p , C , B_2 may be neglected, leaving B_1 . This process has control system of saturation performance with a fast order delay. By Paynter's method⁽⁴⁾, $T_s/T_m = 1$

if an optimum gain is resumed 30% overshoot, we obtain

$$\lambda = \frac{T}{T_m} = 0.6$$

where

T_s = dispersion time

T_m = mean delay

T = time constant

therefore $T = 0.6 \times T_m = 0.6 \times 0.952 = 0.571$

so the loop gain must be $1/T = 1.75$

Since the loop gain of this equipment is the process gain multiplied by the controller, it is shown that

$$\left. \begin{aligned} \frac{dx}{dt} &= 11 \cdot \Delta P X \\ \frac{\Delta P}{x} &= 1.5 \text{ mmAq/deg} \end{aligned} \right\}$$

then

$$\therefore \frac{\frac{dx}{dt}}{x} = 16.5X$$

Where X is 1, the loop gain becomes 16.5, so it is too large for the optimum value and the stability can't be obtained unless the value is throttled considerably. We calculated it with large capacity of the blower, and obtained:

$$\frac{\Delta P}{x} \leq 0.159$$

So it will be stable at a small value of the preventing value gain $\Delta P/x$, where no hunting develops. We know that the hunting ceases where the minimum value on the valve is more than 30 mmAq and the capacity is also more than 10 m³/min of blowers.

4.5 Relation between the automatic control system and surging period

When there is occurring a surging on the system, if we run the controller, surging can be eliminated in spite of putting force of the periodic

surge-pressure.

Assuming a linear system for it, this relation is obtained through the following

$$M = \frac{G(s)}{1 + G(s)}$$

Fig. 13 shows the relation that M is amplified to 20 db if the surging frequency grows to the resonance frequency of the system. Therefore attention should be paid to the relation between the surging frequency and the resonance frequency of the controller.

According to the results of this experiment, the surging frequency is 0.2 sec, about $\omega = 30$ rad/sec, so $M = 0$ db, and the system will keep steady without hunting under the surging influence.

Conclusion

We have applied automatic control to operation of surge-preventing valve, and have studied its effects on practical equipments. We may draw the following conclusions from the experimental analysis of this automatic control surging-preventing system, in which "Constant pressure difference type" was studied as a typical example:

1. The controller system for this type is the simplest and most practical for varying the rate of flow of a blower and a compressor. We could prevent surging at stable operation and hold energy loss to relatively small value.

2. As for the relation between hunting and adulation of the control system, it has been shown that the pinch-cock set on the metering lead pipe damps the flow and prevents hunting without any deduction of adulation.

3. We should design the controlling equipment to adjust a time constant to the relation between the surge frequency and the resonance one of the system. By theoretical analysis on the system, we

Fig. 13

Fig. 14

found that M was amplified to 20 db at the resonance frequency of the system.

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Some Fundamental Natures of Resonant Surge

(1st Report, Experimental Results and Analysis of Oscillating Systems)*

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A resonant surge of small fan duct systems has been investigated to clarify the fundamental natures of the oscillating phenomenon of this kind. The first report gives the details of our experimental apparatus and also of the results obtained. The values of frequency of oscillation measured have been used to analyse our systems to provide the equivalent lumped models for use in the theoretical analysis of the surge described in the following second report.

Introduction

In the pioneering studies of surging, G. V. Brooke⁽¹⁾, S. A. Moss⁽²⁾, I. Watanabe⁽³⁾ observed that the volume of duct had an effect on the oscillation. R. O. Bullock and others⁽⁴⁾ recognized vaguely that the surging is an oscillation on which the characteristics of compressor, volume of duct, inertia of air and pressure drops of throttles having some effects respectively. S. Fujii⁽⁵⁾ was the first one who discussed the surging phenomenon correctly and he introduced fundamental equations including the inertances, capacitances and resistances of oscillating systems consisting of water pump and tanks. Using these equations, he also explained theoretically various phenomena appearing in the surging. H. Pearson and others⁽⁶⁾ discussed the problem of surge points of compressors with use of inertances, capacitances and resistances derived by an analogy to electrical

oscillating systems. H. W. Emmons and others⁽⁷⁾, studying the surging of compressors, derived their equations through the equilibrium of energy, but their method does not differ principally from those abovementioned. H. Kusama and others⁽⁸⁾ discussed the effects of variation of revolving speed of machines.

It is noticed, however, that every recent work above-mentioned is mere theoretical one unaccompanied by experimental proofs. The present work has been carried out to investigate the fundamental natures of resonant surge from the standpoint of both theory and experiment.

1. Fan-Duct systems investigated

Particular fan duct combinations which have been used consist of three different arrangements shown in Fig. 1. All of them have a small single-stage axial-flow fan mounted on the end of a 5 inch ID duct. The fan, with no intake guide vanes, has a rotor with eighteen blades and a stator with sixteen blades. The rotor is driven by a small direct current motor enclosed in the stator

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