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EFFECT OF CHECK VALVE DYNAMICS ON THE SIZING OF RECYCLE SYSTEMS FOR CENTRIFUGAL COMPRESSORS

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ABSTRACT

A recycle system for centrifugal compressors is designed to minimize the potential for surge, as can occur during shut down of the compressor. The recycle system volume capacitance (V) and the recycle valve capacity (Cg) are sized to give the required system dynamic response for surge prevention. Thusfar the check valve dynamic characteristic was not taken into account in the sizing of the recycle system. A numerical investigation was carried out on a scaled-down model of a compressor station to study the dynamic behavior of the check valve in relation to the recycle valve capacity (Cg), its dynamic characteristics, and the recycle system volume capacitance (V) during an emergency shutdown (ESD) of the compressor unit. A model describing the dynamic characteristics of an undamped check valve was introduced into the dynamic simulation of the entire system during ESD. The results show that the sizing of the recycle system can be optimized by also taking into account the check valve dynamic characteristics. Furthermore, it was found that Cg is proportional to the logarithm of the slope of the check valve dynamic characteristics, demonstrating that a small increase or decrease in Cg could have a dramatic effect on the maximum allowable reverse flow through the check valve without surging the compressor during ESD. The recycle piping volume (V), however, was found to be linearly proportional to the slope of the check valve dynamic characteristics resulting in a less dramatic effect. This paper presents numerical results and a discussion of these relationships.

NOMENCLATURE

Cg	-	recycle valve capacity coefficient when fully open
D	-	characteristic diameter of the check valve
H	-	centrifugal compressor adiabatic head
I	-	compressor/driver combined rotor inertia
m	-	gas mass flow rate through the compressor
N	-	compressor rotational speed
Q	-	volumetric flow rate through the compressor at inlet conditions
v	-	local mean flow velocity
v_o	-	minimum mean flow velocity required to fully open the check valve
v_r	-	reverse mean flow velocity through the check valve
V	-	piping volume capacitance upstream of the check valve
t	-	time
η_m	-	compressor mechanical efficiency
η_a	-	compressor adiabatic efficiency
α	-	slope of the check valve dynamic characteristic curve

1.0 INTRODUCTION

The unit discharge check valve is a critical element in the recycle system of a centrifugal compressor to minimize the potential for surge[1]. The check valve prevents reverse flow and reverse rotation of the compressor unit that can cause serious damage to the seals and bearings [2].

The dynamic characteristic of the unit discharge check valve in compressible flow was generally considered to be not important. The valve was therefore considered a static element and not taken into account in the sizing of the recycle system. Although check valves are typically fast closing, a certain amount of reverse flow still occurs during valve closing depending on the local flow deceleration and the dynamic characteristics of the valve. The introduction of large diameter pipeline systems, higher power compressor units, associated new technologies of lower rotor inertia's, demanded careful sizing of the compressor recycle system increasing the interest in the dynamic characteristic of check valves in gas applications.

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The traditional valve designs based on the concept of simple swinging flaps or discs (single or dual) are adequate for smaller diameter, low power high inertia compressor units. However, the new compressor technologies may require the application of more advanced check valves with faster dynamic response, such as the piston or nozzle valve designs. These designs have a spring loaded axially moving disc with a very short stroke from open to closed. At the same time these valves have very low pressure loss reducing compressor fuel cost.

The objective of the present investigation is to address the dynamic behavior of the check valve in relation to the recycle valve capacity (Cg), its dynamics, and the recycle system volume capacitance (V), during compressor ESD. Upon ESD, a signal is sent to the recycle valve to open in order to prevent the compressor from surging. Depending on the dynamic characteristics of the check valve, a reverse flow occurs during valve closing depending on the amount of flow deceleration in the vicinity of the valve. Hence comes the dynamic interaction between the deceleration rate of the compressor unit, the opening characteristics of the recycle valve, the dynamic characteristics of the check valve, and the volume capacitance (V) of the pipes between these three components. The present work points to the importance of carrying our a thorough dynamic analysis during the design stage of the interaction between check valve, compressor unit and the recycle system dyring flow transients to ensure system reliability and optimum equipment performance.

2.0 CHARACTERIZATION OF THE DYNAMIC BEHAVIOR OF CHECK VALVES:

The dynamic behavior of check valves were studied experimentally and numerically by several authors. A good reference paper of this topic is by Thorley [3]. Two lines of research can be identified. The first, an attempt to deduce the dynamic behavior of the check valve by combining the valve geometrical properties, physical properties, and fluid flow characteristics in developing and solving the equation of motion for the valve. This technique has been successful for swing type check valves [e.g. 4,5,6]. The second technique (first developed by Provoost) is based on direct measurements of the maximum reverse flow velocity (v_r) as a function of the local mean flow deceleration (dv/dt). In this technique, direct manifestation of valve components and flow characteristics are revealed by these two parameters rather than a detailed account of all parameters (i.e. focus on end behavior rather then on causes or contributing parameters). This latter technique was first applied to swing and ball-type valves [7], and was later introduced formerly by Delft Hydraulics [8]. It is known as the 'Dynamic Characteristic Curve (DCC)', an example of which is shown in Fig.1, taken from reference [3] for different undamped valve types. The above two parameters (v_r, dv/dt) can be described in a dimensionless manner in the form [9]:

$$\frac{v_r}{v_o} \quad and \quad \frac{D}{v_o^{\,2}} \cdot \left(\frac{dv}{dt}\right)$$

where:

D - characteristic diameter
dv/dt - mean fluid deceleration

This latter technique of the 'DCC' is employed in the present investigation as follows:

2.1 Dynamic Modelling of Check Valve

It has been shown that the closure behavior of undamped check valves does not influence the flow up to the moment the maximum back flow is reached [10]. Tests have also shown that for these types of valves, the valve almost momentarily closes once the maximum reversed flow corresponding to the flow deceleration is reached [10,11]. Therefore, undamped check valves can be modeled in the following way:

i) when the mean flow velocity through the valve is positive ($v>0$), the valve can be modeled as a resistive element with a pressure loss coefficient corresponding to the valve opening position determined from the local mean flow velocity.

ii) when the flow velocity through the valve is negative, the maximum reverse velocity is first calculated from the DCC curve based on the local mean flow deceleration. The calculated reverse velocity is compared with the maximum reverse velocity. If the former is found to be smaller, the valve is assumed closed (see Fig. 2) and flow velocity is set to zero, constituting closed end boundary conditions at the pipe terminals connected to the valve.

The closure behavior of damped check valves is more involved since the assumption that the valve closes momentarily when the maximum reverse velocity is reached cannot hold. Depending on the damping parameter, the valve closure is delayed and the maximum reverse flow (v_r) is larger in magnitude and duration than a corresponding undamped valve [11].

3.0 SIMULATION OF THE ESD PROCESS

The process of compressor ESD is schematically presented on an H-Q plot in Fig.3 following the trend observed both experimentally and numerically in [12]. Six phases are identified as the compressor decelerates from a steady state point (S.S.) to zero flow and zero head across the unit. These phases are described in more detail in [13], and are summarized as follows:

Phase I: The operating point of the compressor follows approximately a straight line characterized by a slope corresponding to the characteristic impedance of the mainline. The recycle valve remains closed during this phase, and for a period of time corresponding to valve pre-opening (pre-stroke) delay. The fuel is assumed to be completely shut-off to the gas turbine driver, and the compressor decelerates due to rotor inertia. The compressor unit is connected to long pipe sections both on the suction and discharge sides constituting anechoic boundary conditions.

Phase II: The recycle valve starts to open marking the beginning of this phase which is terminated when the first flow/pressure perturbation due associated with recycle valve opening arrives at the compressor.

Phase III: Flow starts to increase through the compressor due to the arrival of these perturbations at the compressor.

Phase IV: Due to gas inertia's in the recycle line and mainline, the flow through the compressor tends to overshoot followed by a short period of an undershoot (phase V) around the recycle system resistance line shown in Fig. 3.

Phase VI: This is the final phase where the compressor winds down in small over- and under-shooting around the recycle resistance line until it reaches zero flow and zero head.

As can be seen, the most critical point on the ESD path is at the end of phase II. If the time associated with phases I & II combined is long (or the recycle valve size and type limit the magnitude of the perturbations arriving at the compressor to affect an increase in its flow) the end of phase II may cross the surge limit and the compressor will experience a negative flow (surge). Additionally, the point at which the check valve closes depends on the local mean flow velocity and flow deceleration (dv/dt) at the check valve location. It was found to be along phase III or Phase IV for most cases, as will be seen later.

Dynamic simulation of the above process entails the solution of the one dimensional unsteady compressible fluid flow equations described in more detail in [14]. The method of characteristics was used to solve the above equations along positive and negative characteristics as well as path lines. Detailed treatment of the solution techniques and the formulation of the pertinent equations for other than pipe elements and end conditions can also be found in [14]. The compressor system is assumed to respond to any perturbation in a quasi-steady manner following its full characteristics, including that to the left of the surge limit [12]. Compressor/power turbine deceleration is governed by the following equation (which assumes that the fuel gas is completely shut-off to the gas turbine, allowing only the exchange of energy between the fluid flow and compressor/power turbine rotor):

$$-I \cdot N \cdot \frac{dN}{dt} = \frac{m \cdot H}{\eta_m \eta_a} \qquad (1)$$

4.0 RESULTS AND DISCUSSIONS

The above model has been applied to compressor station shutdown scenarios to study the effects of the reverse flow through the check valve on the compressor response during ESD. A low pressure scale-down air test rig model was considered in this study, a simple schematic of which is shown in Fig. 4. The particulars of this scale-down test rig, its dimensions, comparisons between measurements, and numerical results during various ESD scenarios are given in [12]. The good agreement between measurements and numerical results demonstrated in [12] furnished a necessary credibility for the solution techniques used in the dynamic model, and allowed further investigation of the present problem numerically.

The layout of the compressor station laboratory model of Fig. 4 shows that the recycle valve and the check valve are located close to the compressor discharge side. This was found to be the best scenario from an ESD point of view following the recycle system investigation in [12]. For a perfect check valve (i.e. one that allows zero reverse flow), the minimum Cg coefficient for the recycle valve of 80 ms pre-stroke delay and 80 ms stroke, is 300. The recycle system resistance line with a Cg=300 recycle valve is shown in Fig. 5 to the right of the surge limit of the centrifugal compressor. Recycle system resistance lines with other Cg's (350 and 400) are also shown in the Figure for comparison, together with the recycle system resistance line with a 37.5% margin from the surge limit commonly used in NOVA in the design and sizing of recycle systems.

A computer simulation of an ESD was conducted on the above compressor station scale model where the steady state operating point was initially close to the surge limit (representing a worst case scenario). The check valve dynamic characteristic curve (DCC) was varied according to the following linear characteristics:

$$\frac{v_r}{v_o} = \alpha \cdot \frac{D}{v_o^{\,2}} \cdot \left(\frac{dv}{dt}\right) \qquad (2)$$

where α is a dimensionless constant signifying the slope of the check valve DCC.

The value of α was increased systematically (representing slower closing valves) until the compressor goes into surge at any point during ESD (along the process shown in Fig. 3). The maximum value of this parameter α is recorded, which determines the maximum tolerance allowed by the compressor/recycle system for a back flow through the check valve. For the system described above and recycle valve of Cg = 300, the maximum value of the parameter α was found to be 1.04. (Note the critical velocity of the swing type check valve used in the air test rig is approximately 40 m/s, and its characteristic diameter D is 0.0766 m).

The results of the simulation for the limiting value of α are shown in Fig. 6, where it is evident that only small reverse flow is allowed through the check valve manifested by both the velocity and mass flow curves through the check valve. Note also that the check valve closes at a point along phase V of the ESD process shown in Fig. 3.

If the capacity of the recycle valve increases to Cg=350 (representing a recycle resistance line of 30% to the right of the surge limit), the maximum value of α increases to 11.2 – an order of magnitude higher than the previous case. The results of this case is shown in Fig. 7. When the capacity of the recycle valve was further increased to Cg=400, representing 48%

recycle resistance line (see Fig. 5), the maximum permissible value of parameter α increased to 101.5, yet another order of magnitude higher than the previous case. Figure 8 shows the results of this case. Notice the higher magnitude of the allowable reverse flow through the valve and yet no sign of compressor surging during this ESD process. This finding suggests that the choice of the check valve and the associated dynamic behavior represented by its dimensionless DCC characteristics should be examined in connection with the recycle system capacity. The higher the recycle system capacity, the more tolerance of back flow through the check valves.

The results of the previous three cases are presented on the check valve dynamic plot shown in Fig. 9, showing the limiting DCC lines for the various recycle valve Cg. Therefore, proper selection of the check valve should be tied with the cost associated with recycle system capacity. A further consideration for check valves is the incremental fuel cost associated with the pressure loss across the valve. For example, since swing type check valves are generally less expensive than nozzle or piston types up to size NPS 16, a larger capacity recycle valve can be considered provided that the cost of this higher capacity is not offset by the saving in the check valve capital cost and increased compressor fuel cost associated with higher pressure loss. The opposite can be said for check valve sizes greater than NPS 16, where the piston type valves are typically less expensive and where lower reverse flow may allow for the sizing of a lower capacity recycle system resulting in further cost savings.

In summary, increasing the capacity of the recycle system allows for less stringency on the maximum allowable check valve reverse flow (Fig. 9).

To put the above results in a dimensionless form, the ratio between Cg/Cgo vs. the DCC slope α is plotted in Fig. 10. Here, Cgo is the reference recycle valve coefficient for hypothetically perfect check valves with very low backflow. The plot shows that Cg/Cgo is proportional to the logarithm of α, indicating that a small variation in Cg could have dramatic effects on the maximum permissible value of α. For example, a 15% increase (or decrease) in the recycle valve coefficient Cg could result in values 10 times higher (or lower) than the permissible α coefficient. This elucidates the importance of optimizing the selection of both the check valve and recycle valve to achieve the most cost effective equipment needed.

Another factor that was examined is the capacitance of the piping system contained between the check valve, the compressor, and the recycle valve. For this reason, a similar configuration to the above compressor station model was considered. This is shown in Fig. 11, which exhibits the same piping lengths, compressor, and recycle system characteristics – the only difference being that the recycle valve and recycle line was mirrored so as to have the recycle valve closer to the suction side of the compressor. Hence, a larger piping capacitance upstream of the check valve was allowed. In comparison to the original configuration of Fig. 4, the piping volume capacitance in this case is 12 times that of the original configuration. The recycle valve capacity was set at the lowest value of Cg=300 (determined earlier for safe ESD without compressor surging). In this case, the maximum permissible back flow represented by parameter α of the DCC characteristics increased to 29.0 from 1.04 of the previous case of Fig. 6. This increase in the allowable reverse flow was affected by the increased capacitance of the piping system upstream of the check valve. Although the two configurations of Fig. 4 and Fig. 11, are equivalent in terms of protecting the compressor unit from surge during ESD (as concluded in ref. [12]), it seems that the second configuration places less stringent demands on the reverse flow through the check valve, which may be preferable from this view point. The relationship between piping volume capacitance and α is shown to be linear as shown in Fig. 12, indicating a lesser effect than that of the recycle valve capacity.

5.0 CONCLUSIONS

The following two main conclusions can be drawn from the present work:

1. Reverse flow through check valves in compressor stations during emergency shutdowns can be tolerated by a large capacity recycle valve (Cg coefficient). The higher the recycle valve Cg coefficient, the higher the maximum allowable reverse flow through the check valve without the risk of surging the centrifugal compressor (reverse flow through the compressor). It was also found that Cg is proportional to the logarithm of the slope of the check valve DCC curve α, indicating that a small variation in Cg could have dramatic effects on the maximum permissible reverse flow through the check valve. This elucidates the importance of optimizing the selection of both the check valve and recycle valve to achieve the most cost effective equipment needed.

2. The same is applied, but to a lesser degree, on the volume capacity of the compressor station piping bounded by the compressor discharge side, the recycle valve and the check valve. The larger this piping volume capacitance is, the more tolerable the reverse flow through the check valve can be without causing reverse flow through the compressor (surge) during ESD. However, the relationship between the piping volume capacitance and α is rather linear indicating a lower effect than that of the recycle valve Cg coefficient.

6.0 ACKNOWLEDGMENT

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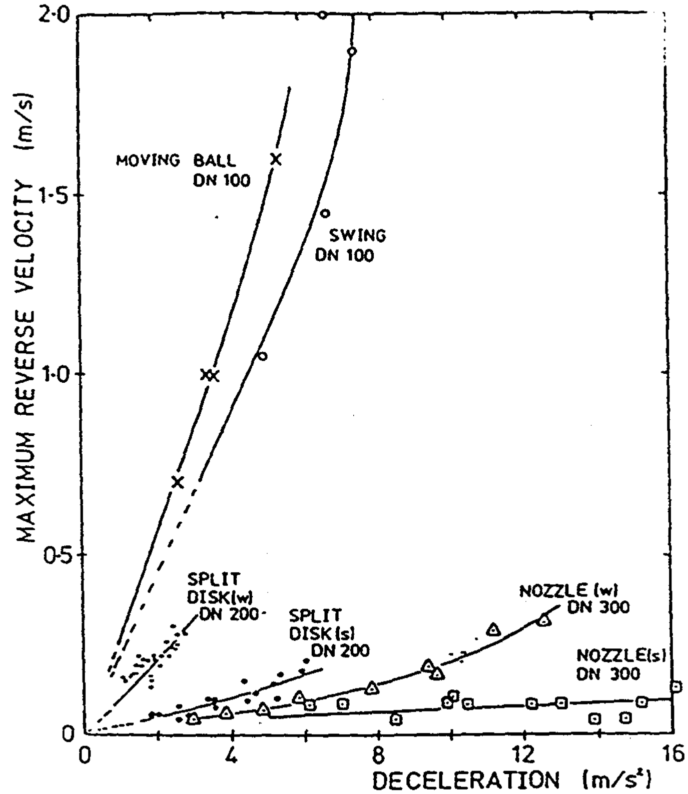

Fig. 1: Dynamic Characteristic Curves for different types of check valves (from ref [3]).

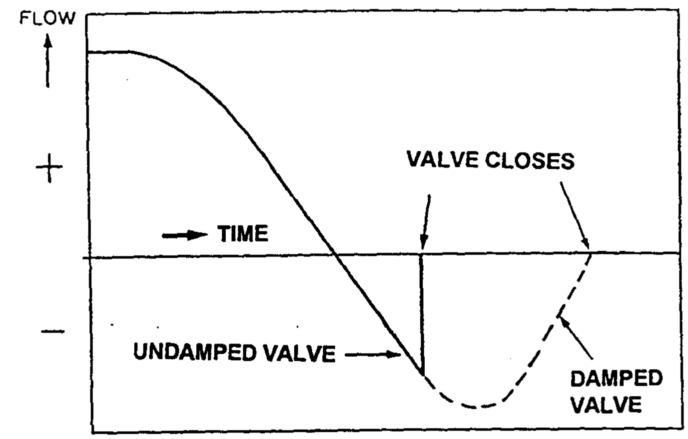

Fig. 2: **Concept of the dynamic model of a check valve in decelerating flows.**

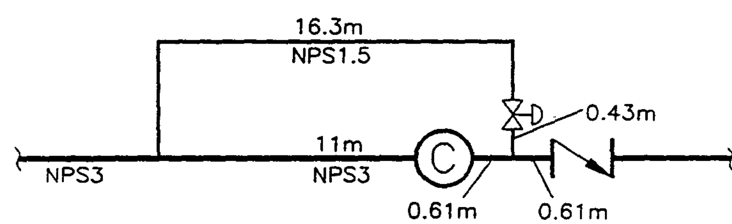

Fig. 4: **Schematic of the compressor/recycle/check valve laboratory test rig considered in the present work.**

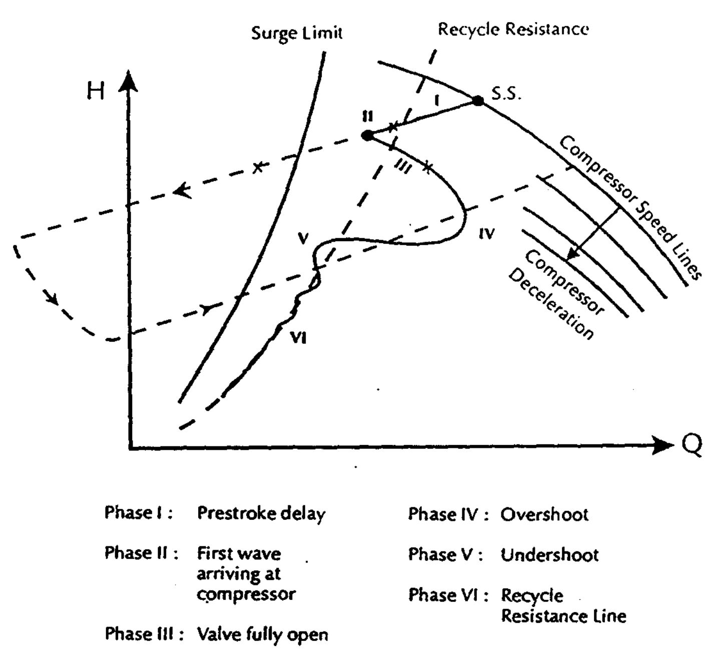

Fig. 3: Typical ESD process on H-Q plot.

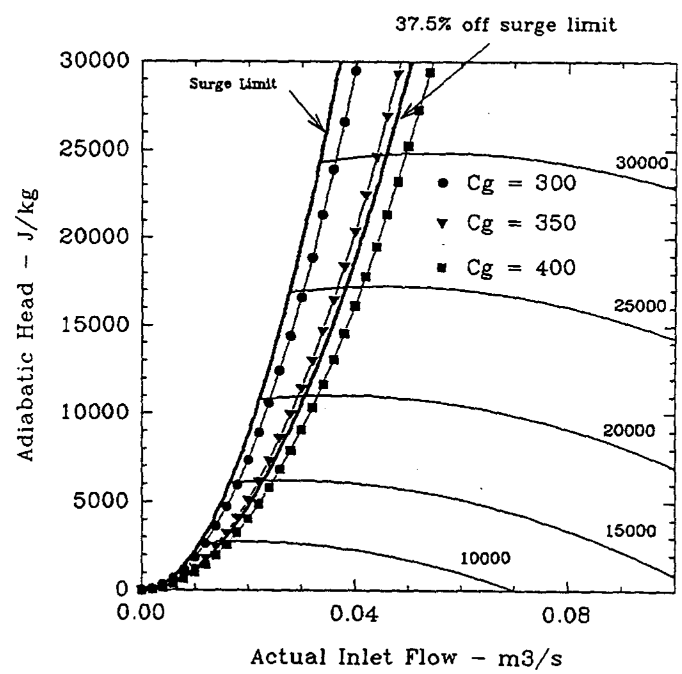

Fig. 5: **Centrifugal compressor performance characteristics showing various recycle system resistance lines.**

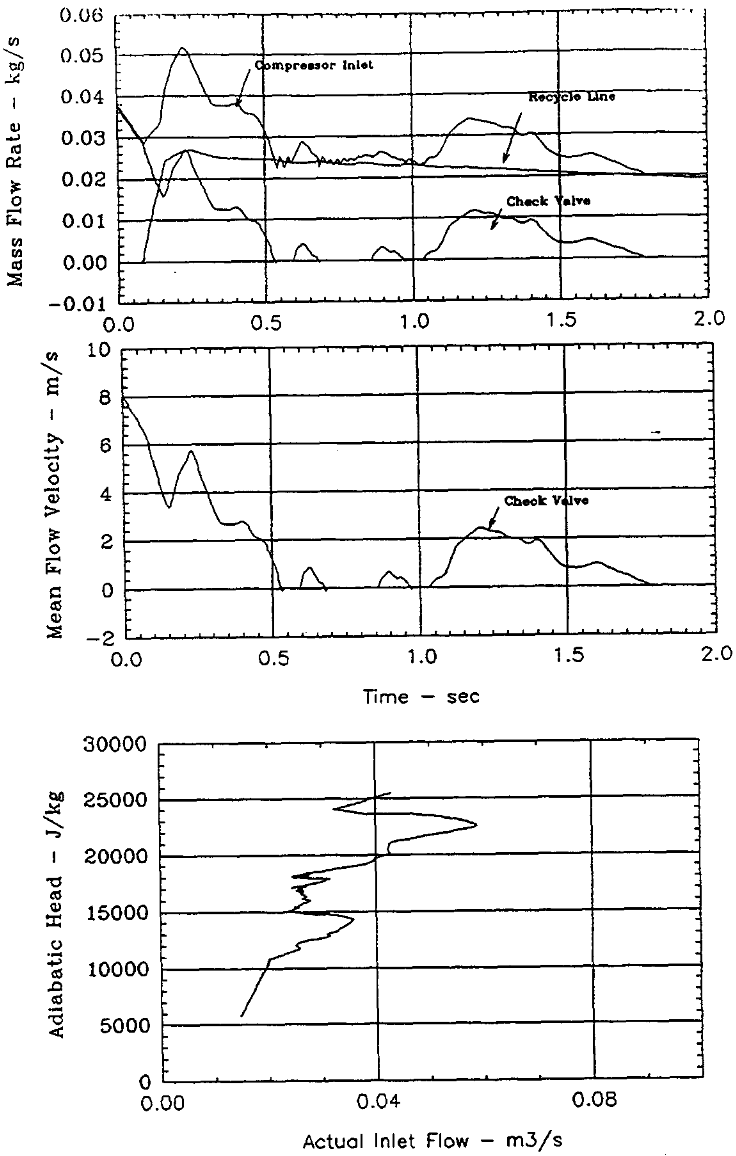

Fig. 6: Simulation results of ESD process of the system shown in Fig. 4 and a check valve of $\alpha = 1.04$

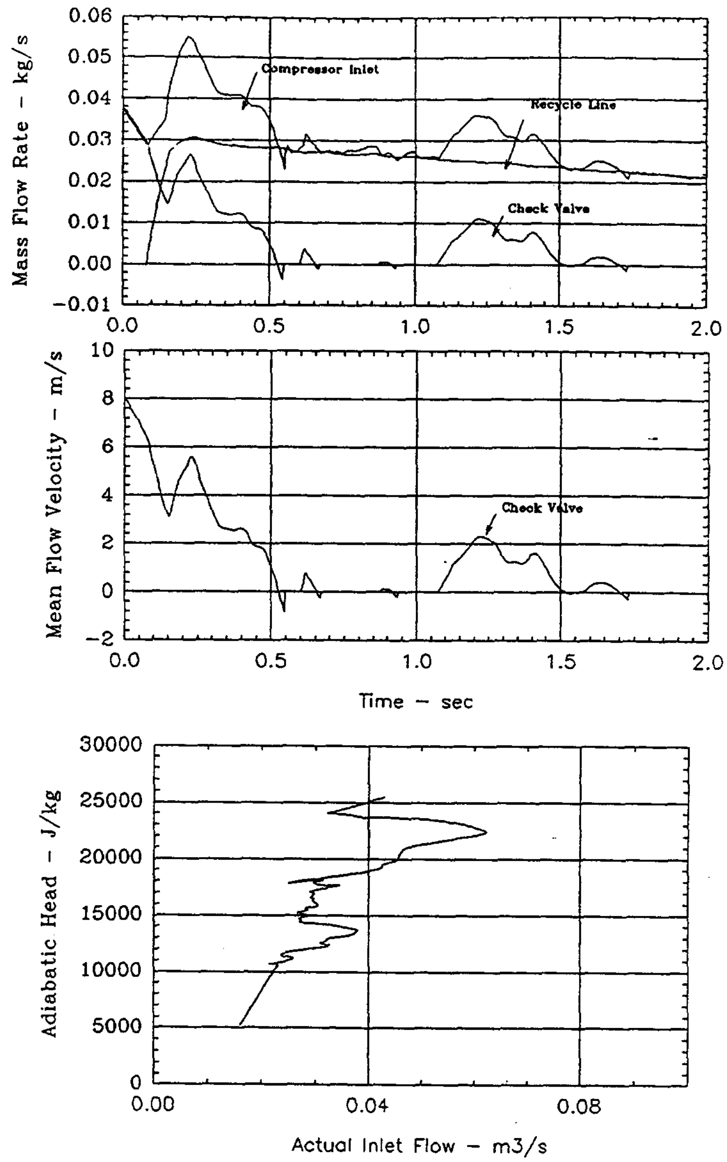

Fig. 7: Simulation results of compressor ESD process of the system shown in Fig. 4 and a check valve of $\alpha = 11.2$

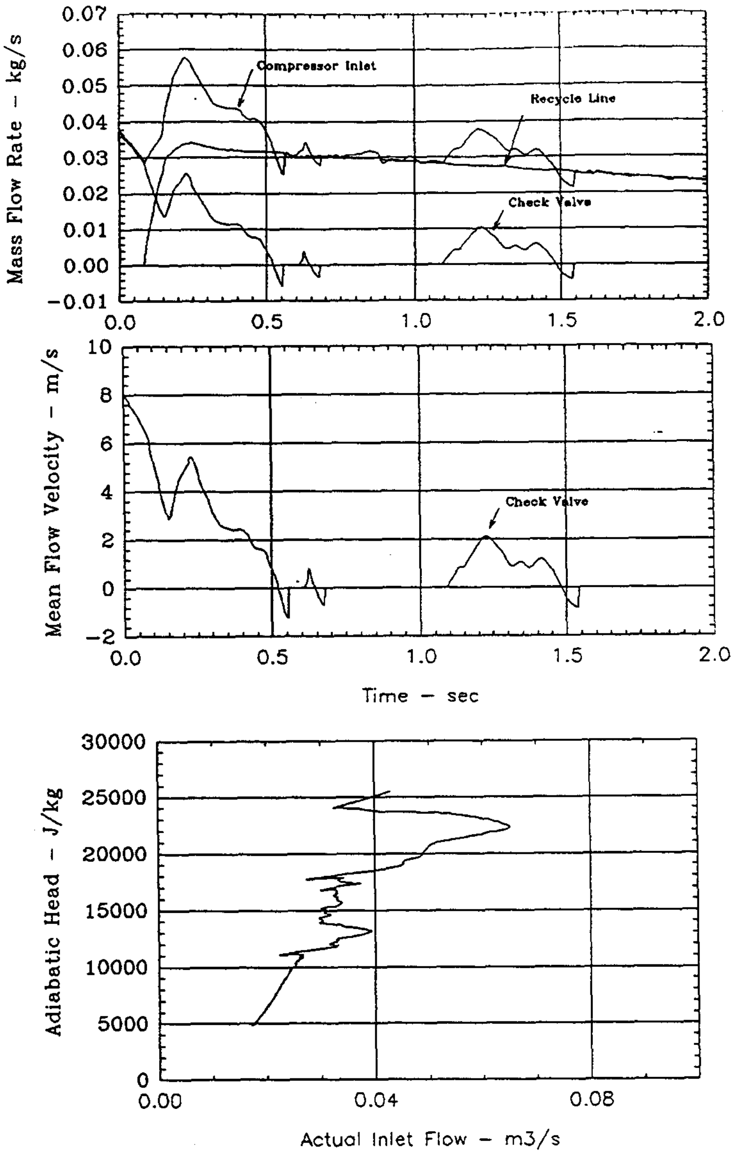

Fig. 8: Simulation results of compressor ESD process of the system shown in Fig. 4 and a check valve of $\alpha = 101.5$

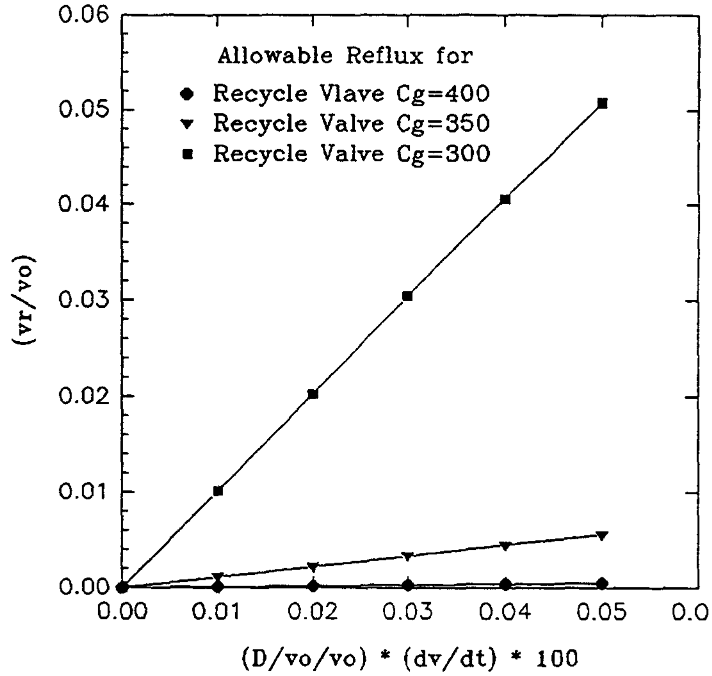

Fig. 9: Relationship between check valve dynamic characteristics and recycle valve capacity Cg.

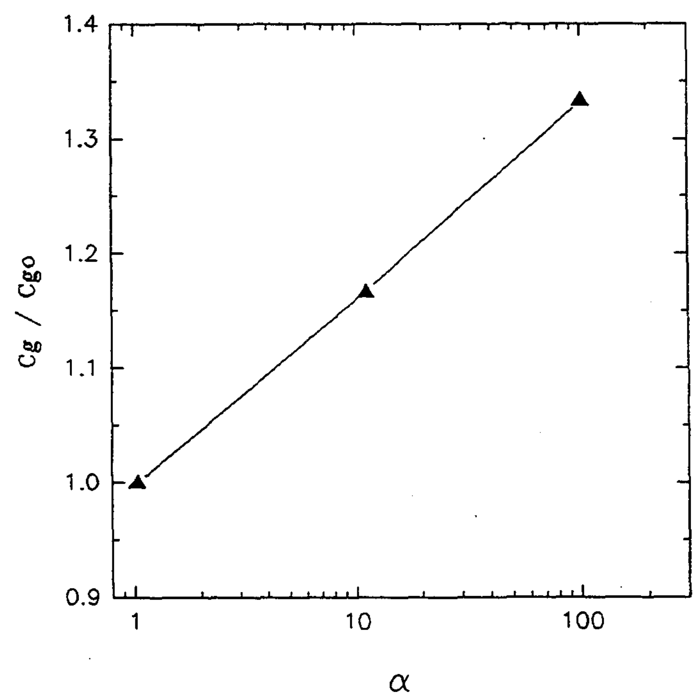

Fig. 10: Dimensionless recycle valve capacity required for a given check valve dynamic characteristic parameter α.

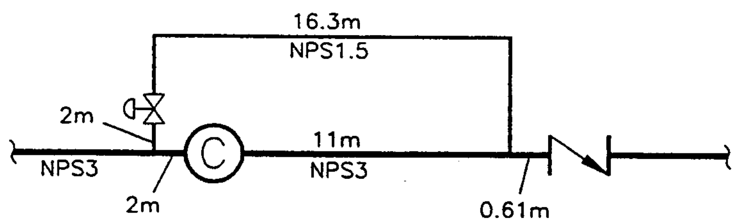

Fig. 11: Schematic of the compressor/recycle/check valve laboratory test rig with more piping capacitance upstream of the check valve.

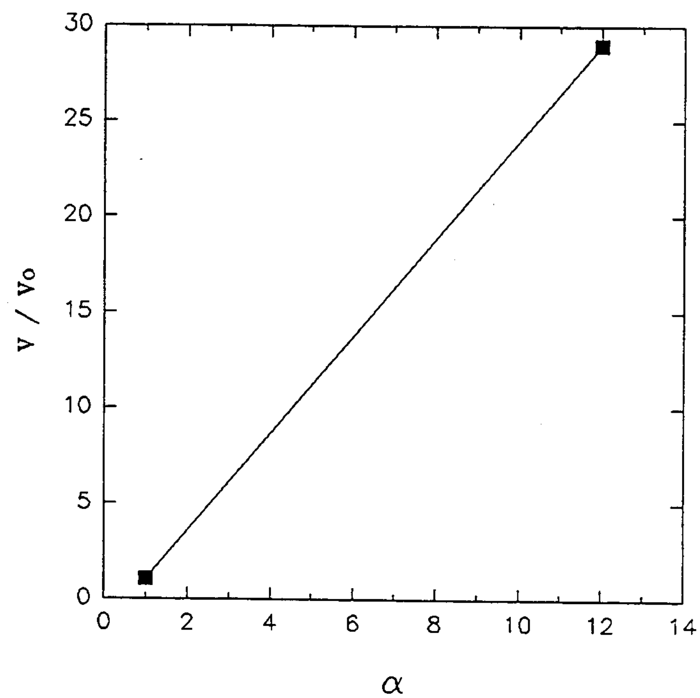

Fig. 12: Dimensionless piping volume capacitance required for a given check valve dynamic characteristic parameter α.