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Selection Criteria of Spring Stiffness for Nozzle Type Check Valves in Compressor Station Applications

K.K. Botros
NOVA Research & Technology Center
2928 – 16 Street NE
Calgary, Alberta, Canada
Email: botrosk@novachem.com

ABSTRACT

Nozzle type check valves are often employed in compressor stations in three locations: compressor outlet, station discharge and station by-pass. The fundamental design concept of these valves is based on creating a converging diverging flow through the valve internal geometry such that a minimum area is achieved at a location corresponding to the back of the check valve disc at fully open position. This will ensure maximum hydrodynamic force coefficient which allows the valve to be fully open with minimum flow. Spring forces and stiffness determine the performance of this type of check valves and impact the overall operation and integrity of compressor station. This paper examines the effects of various spring characteristics and stiffness in relation to the compressor and station flow characteristics. The results show that when the spring forces are higher than the maximum hydrodynamic force at minimum flow, the disc will not be at fully open position, which will give rise to disc fluttering and potential for cyclic high velocity impact between components of the internal valve assembly. This could lead to self destruction of the check valve and subsequent risk of damage to the compressor unit itself. The paper also points to the fact that the spring selection criteria for a unit check valve are different than that for station and bypass check valves. An example of a case study with actual field data from a high pressure ratio compressor station employing this type of check valves is presented to illustrate the associated dynamic phenomena and fluid-structure interaction within the internal assembly of the check valve.

INTRODUCTION

In a natural gas compressor station, check valves are critical elements in the design and operation of the station. They are employed at compressor outlet (unit check), station discharge (station check) and station by-pass (see Fig. 1). The unit check valve is commonly placed on the discharge side of the compressor to prevent reverse flow that can cause serious damage to the compressor and other components such as seals and bearings. Nozzle type check valves are often selected as they offer low pressure drop, low reverse flows and low slamming upon closing. These valves are based on a moving disc that is held against an opening within the internal geometry of the valve by means of a spring. When the flow is forward, a hydrodynamic force created by the flow will force the disc to move back against the spring force and allow the flow to go forward. When the flow is stopped (or reversed), the spring will quickly close the valve by forcing the disc to go back to its closed position against a seat.

Important selection criteria of these check valves are their flow characteristics in steady flow as well as their dynamic characteristics in unsteady flow operation. For the **steady flow** aspects, the fundamental design concept of these valves is based on creating a converging diverging flow through the valve internal geometry with a minimum flow area at a location corresponding to the back of the disc at the fully open position. In this case, spring stiffness manifests itself in aspects of the design and operation related to minimum flow required to fully open the valve, and the relationship between this minimum flow to the compressor performance characteristics (map), as well as station forward flows, especially when gas recycling is involved.

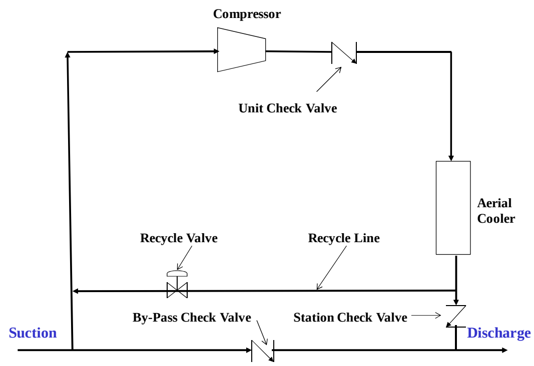

FIGURE 1: A SCHEMATIC OF A COMPRESSOR STATION LAYOUT.

The other criteria for nozzle check valve selection are the valve dynamic characteristics in **unsteady flows**. One aspect is the slamming of the disc against its front and back seats at the onset of reverse flow or during an emergency shutdown (ESD) of the compressor unit. In the case of swing type check valves, dampeners (slam retarders) and counterbalance weights are often recommended to improve the mechanical integrity of the valve and reduce any potential risk of disc damaged if slammed [1,2]. But in the case of nozzle type check valves, both viscous and frictional damping are very low due to the inherent nature of the design, shown schematically in Fig. 2, particularly in the case of a hung ring type disc, as opposed to solid round disc. Therefore, severe disc slamming can occur which could give rise to mechanical failure of the disc itself and the entire internal assembly of the valve.

The other aspect in unsteady flows is the maximum reverse flow that can occur as the valve closes due to flow deceleration. Of course, the objective is to minimize this when the valve closes. This has been studied extensively, both experimentally and numerically in incompressible flows by several authors [5-14]. A good review paper on this topic is by Thorley [15]. Two lines of research can be identified. The first is an attempt to deduce the dynamic behavior of the check valve from combining the valve geometrical and physical properties and fluid flow characteristics in developing and solving the equation of motion for the valve internal assembly. This technique has been successful for swing type check valves [e.g., 8-10]. The second technique, which was first developed by Provoost [9], 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 is revealed by these two parameters rather than a detailed account of all parameters. This latter technique was first applied to swing and ball type valves [16] and later was introduced formerly by Delft Laboratory [10] to all types of check valves, and is known as the 'Dynamic Characteristic Curve - (DCC)' of the valve. The above two parameters (V_r and dV/dt) can be described in a dimensionless manner in the form [17]:

$$V_r/V_o \;\; and \;\; \frac{D}{V_o^2}(dV/dt) \qquad (1)$$

where:

- D - valve dimensional characteristics (e.g. I.D. of the upstream pipe).
- V - mean flow velocity upstream of the valve.
- V_o - minimum mean flow velocity upstream of the valve to open the valve to a fully open position.
- V_r - maximum reverse flow velocity through the valve upon closing.
- t - time.

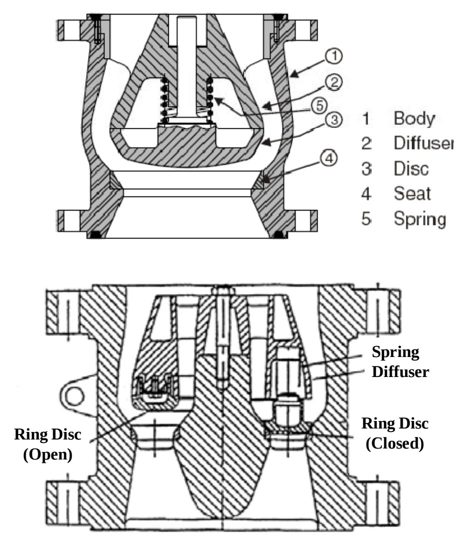

FIGURE 2: A SCHEMATIC OF A NOZZLE TYPE CHECK VALVES (TOP: FULL ROUND DISC [3], BOTTOM: RING DISC [4]).

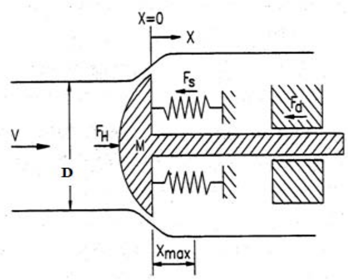

FIGURE 3: BALANCE OF FORCES ON A NOZZLE TYPE CHECK VALVE.

The present paper addresses both set of criteria (related to steady and unsteady flows) for selecting the appropriate spring stiffness for nozzle type check valves specifically when they are employed in compressor stations. Emphasis are placed on the effects on disc opening (in steady flow), and disc slamming and fluttering (in unsteady flow). Impact of the functional nature and difference between compressor unit check valves versus station check valves are investigated. Steady and dynamic simulations are conducted on a Case study of a single unit compressor station used on an existing high pressure natural gas transmission system in North America to demonstrate the interactions between the various parameters involved. In this paper, all of the steady and dynamic analyses were based on the balance of forces on the disc which follows the fundamental equation of motion in the form:

$$M\ddot{X} = -F_o - KX + C_D(X)D^2\rho V^2 - \alpha\ \rho_d D^2 \dot{X}\left|\dot{X}\right| \qquad (2)$$

The left hand side of this equation represents the inertia of the moving elements. The terms on the right hand side represent the external forces due to spring forces, fluid forces, (pressure and fluid drag) and damping (see schematic of Fig. 3).

where:

- M - equivalent moving mass (=disc mass + 1/3 spring mass).
- X - disc position (x=0 at fully closed position).
- F_o - spring force when disc at fully closed position.
- K - stiffness of all springs acting on the disc.
- D - valve dimensional characteristics (e.g. I.D. of the upstream pipe).
- V - mean flow velocity upstream of the valve.
- F_H - *hydrodynamic* force.
- C_D - *hydrodynamic* drag coefficient, which is function of disc position (*X*) and *Re.*
- α - *damping* coefficient.
- ρ - *fluid* density.
- ρ_d - *damping* fluid density.

The DCC characteristics of these valves, disc location and dynamic response to unsteady flows can be determined from the solution of the above equation of motion along with the fundamental flow equations describing the instantaneous gas dynamics through the valve. This will be elaborated on later in the paper.

HYDRODYNAMIC FORCE ON VALVE DISC

The hydrodynamic force on the disc in a nozzle type check valve depends on the flow and pressure field around the disc at any given position from very close to a fully closed position to a fully open position. This is akin to the drag force imposed on a bluff body placed in a moving fluid. This drag force is a result of two components: frictional drag and pressure drag. Frictional drag comes from friction between the fluid and the surface of the body over which it is flowing which is associated with the development of boundary layers, and it scales with Reynolds number (Re). Pressure drag, which is also function of Re (although to a lesser extend), comes from the flow field around the body giving rise to uneven pressure distribution acting normal to the surface of the body at every point, and is obtained by integrating the pressure distribution over the entire surface of the body (front and back). This drag is also associated with the possible formation of a wake due to flow separation at certain location on the surface of the body. If the fluid is inviscid, then there will be no frictional drag, but there will be still pressure drag. The distinction between the two is useful because the two types of drag are due to different flow phenomena.

This hydrodynamic or fluid drag, F_H, is normalized by the fluid density and square of the approach velocity (typically taken as the mean flow velocity in the attached upstream pipe), via the hydrodynamic drag coefficient C_D defined as:

$$C_D = \frac{F_H}{\rho V^2 D^2} = f(\mathrm{Re}, Geometry) \qquad (3)$$

At any given position of the disc in, the hydrodynamic drag is balanced by the spring force when the disc is held stationary and according to Eq. (2):

$$F_H = F_o + KX = F_S \qquad (4)$$

where F_S, is the spring force from all springs acting on the disc due to their compression corresponding to the disc position.

In the design of nozzle type check valves, it is essential to optimize the internal geometry of the valve such that the highest possible hydrodynamic drag coefficient is realized when the disc is at its fully open position. The two fundamental reasons for this are:

i) The valve will be maintained at a fully open position with minimum flow; and
ii) The valve will be quick to close once the forward flow slightly drops below this minimum flow. In this case the flow field will be altered in a way which will reduce C_D and hence the spring force will take over to quickly push the disc toward the closing position.

The above can be attained by varying the contour of the inner surface of the valve body as well as the disc/back seat geometry such that the local minimum static pressure occurs *exactly* at the back side point of the disc when it is at a fully open position.

In order to illustrate this, the flow fields around the disc in the two types of nozzle checks valves shown in Fig. 2 [3,4] were investigated via Computational Fluid Dynamics (CFD) simulations of the respective geometry when the disc is at fully open position in both. The CFD simulations were carried out using Fluent software [18], where a two-dimensional, axi-symmetric, steady incompressible form of the flow equations was modelled. Although the region of interest in the present analysis is the flow through the check valve itself, 20 diameters of straight pipe upstream and 40 diameters of straight pipe downstream are taken to allow for the flow to fully develop, particularly upstream, as well as to check the overall pressure loss coefficient (as a side result). The fluid considered is water at 20 ^oC, and the upstream mean flow velocity is taken as 3.16 m/s. A total of 50,653 quadrilateral cells are used to model the detail geometry of the valve internals, with a maximum equisize skew = 0.496. The flow Reynolds number corresponding to pipe diameter and mean flow velocity of 3.16 m/s in water is 2.75 x 10^6. Turbulence model used is Realizable k-ε, with standard wall functions assuming smooth wall roughness. Discretization of the momentum equation is 2nd Order Upwind and similarly the k-ε equations are also described using 2nd Order Upwind.

To ensure proper resolution of the near-wall region for implementing the enhanced wall functions of the turbulence model, a near-wall grid region was developed based on initial estimates of the expected non-dimensional normal distance, y^+ from the tube wall, where:

$$y^+ = \frac{\rho\, y\, U_\tau}{\mu} \tag{5}$$

and the friction velocity U_τ is related to the local wall shear stress τ_w by:

$$U_\tau = \sqrt{\frac{\tau_w}{\rho}} \tag{6}$$

It was determined after several iterations that the grid was such that the next-to-wall grid locations ensured that the y^+ values at these nodes were generally less than 100 to 200 when in fully turbulent flow and 1 or lower when in slow or laminar flow regions (it is common to consider fully turbulent flow for $y^+ \geq 30$, viscous sublayer flow for $y^+ \leq 5$ and transition to turbulence for y^+ in between). Figure 4 shows the respective values of the near wall dimensional distance y^+ along different solid boundaries of the valve internals. It is shown that the values of y^+ are generally above 11.25, which is the minimum required to apply standard wall functions. This is an important criterion to ascertain the accuracy and credibility of the computed flow fields.

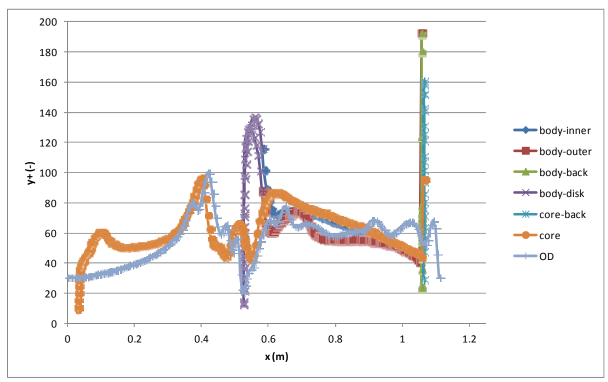

FIGURE 4: NEAR WALL NON-DIMENSIONAL DISTANCE Y+ ALONG DIFFERENT SOLID BOUNDARIES (X-DIRECTION) OF THE VALVE INTERNALS.

Ring Disc Check Valve:

The results of the CFD for this type of nozzle check valves are shown in Fig. 5 in terms of the magnitude of the flow velocity, and in Fig. 6 in terms of the static pressure field. The internal geometry of this valve was taken exactly as that found in ref. [4]. Note the stagnation zone on the disc front corresponding to zero velocity, and the lower pressure behind the disc due to the acceleration of the flow through the two annular diffuser channels around the ring disc (see Fig. 2-bottom). Although the disc is at a fully open position, the metal to metal contact between the disc and the tip of the diffuser does not provide a perfect seal, hence pressure is allowed to communicate between the inner lip and the outer lip on the back side of the disc. This was effected by maintaining a small (1 mm) gap between the back of the disc and the back seat (diffuser) which is extended a distance equals to the thickness of the outer wall of the diffuser.

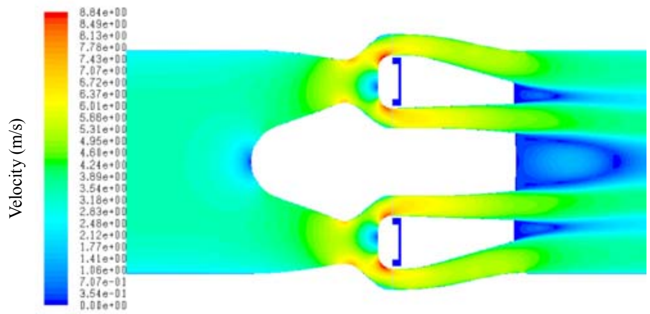

FIGURE 5: VELOCITY FLOW FIELD IN A NOZZLE CHECK VALVE (RING DISC) – APPROACH WATER VELOCITY = 3.16 M/S (INTERNAL GEOMETRY WAS TAKEN FROM REF. [4]).

Fluent also performs an integration of both the pressure and viscous forces on a solid element in the flow (which is the disc being the body of interest in the present case) in the axial direction. The resulting pressure and viscous forces on the front of the disc are shown in Table 1, as well as the pressure force on the back of the disc (no viscous force behind the disc). Manual integration of the axial component of the pressure distribution (shown in Fig. 7) was also conducted to check the value obtained by Fluent's integration of its predicted pressure distribution. The results are also included in Table 1, which agree very well with Fluent. The total net hydrodynamic force on the disc is shown to be = 0.796 $\rho V^2 D^2$, hence the hydrodynamic drag coefficient (C_D) coefficient as defined in Eq. (3) is equal to 0.796.

CFD data were also found in the literature on this ring type disc check valve at fully open position [4]. Figure 8, reveals the pressure isobar contours around the disc at fully open position which is consistent with the pressure field shown in Fig. 6.

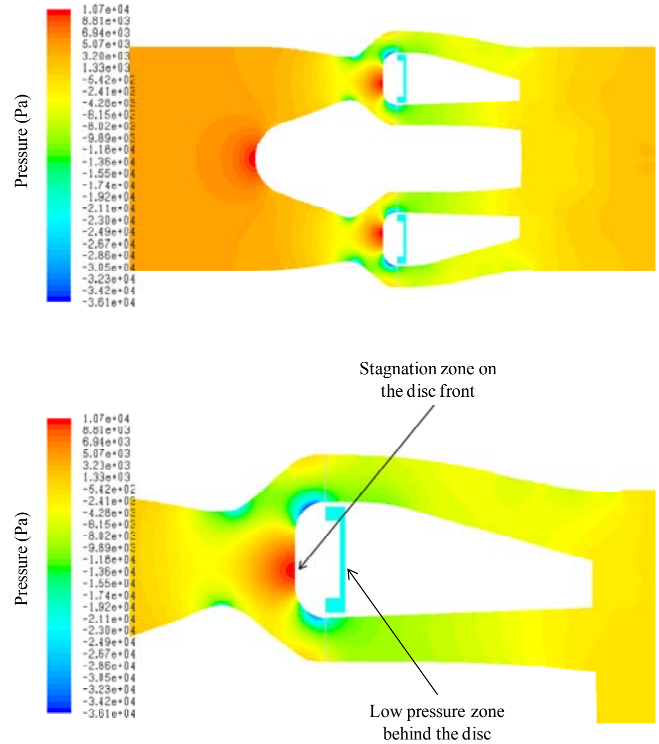

FIGURE 6: PRESSURE FIELD IN A NOZZLE CHECK VALVE (RING DISC) – APPROACH WATER VELOCITY = 3.16 M/S (INTERNAL GEOMETRY WAS TAKEN FROM REF. [4]).

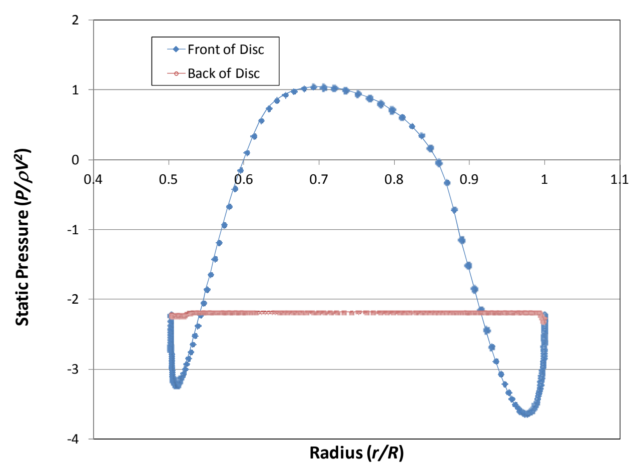

FIGURE 7: NORMALIZED PRESSURE DISTRIBUTION ON THE DISC SURFACE OF THE NOZZLE CHECK VALVE FROM FIG. 6 (R IS THE OUTER RADIUS OF THE DISC).

TABLE 1: RESULTS OF THE HYDRODYNAMIC FORCES CALCULATED FROM THE CFD ANALYSIS OF FLOW THROUGH A RING DISC TYPE CHECK VALVE.

Calculated axial force on front of disc	$\rho V^2 D^2$
Pressure Force	-0.372
Viscous Force	0.005
Total axial force, front-side	**-0.367**
Check via ∫ P.dA	-0.373
Calculated axial force on back of disc	
Pressure Force	1.163
Viscous Force	0.000
Total axial force, back-side	**1.163**
Check via ∫ P.dA	-1.163
Total Force on disc	**0.796**

The important point to notice is the location of the minimum pressure zones along the surface of disc. Both Fig. 6 and Fig. 8, as well as the pressure distribution of Fig. 7, clearly show that the minimum pressure zones on the inner and outer radii of the ring disc are not in line with the back of disc. Recall that the best valve design would ensure that the minimum pressure zones are located at locations corresponding to the back of the disc for maximum C_D.

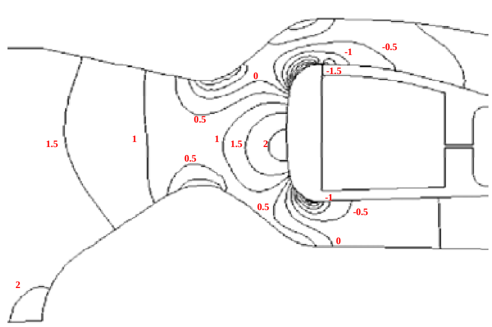

FIGURE 8: PUBLISHED PRESSURE ISOBARS OF FLOW FIELD OF A RING DISC TYPE VALVE IN A FULLY OPEN POSITION (Isolines range from $-5\,\rho V^2$ to $+2\,\rho V^2$, step of ½ ρV^2) – [ref. 4].

Solid Disc Check Valve:

The above results are contrasted with similar CFD results conducted by Roorda on a solid disc type check valve akin to that shown in Fig. 2 (top) [19]. These results in terms of velocity and static pressure fields are shown in Fig. 9 and the resulting pressure distribution along the disc surface is shown in Fig. 10. Here the minimum pressure zone is clearly located at the back of the disc, i.e., at the dimensionless radii of $r/R = \pm 1$, where R is radius of the solid disc. From the point of view of maximum C_D, clearly the design of the internal geometry of this valve is better optimized. The resulting C_D coefficient for this valve in [19] can be obtained by integration of the pressure distribution in Fig. 10, and was found to be equal to 1.814, which is much higher than that of the ring disc type valve design evaluated here.

The significance of accurately determining the C_D coefficient is that one can determine the minimum flow velocity to fully open the same valve for a given spring force $(F_s)_{max}$ and in different fluid applications. For example, if the fluid is natural gas of density 52.65 kg/m^3 and the same spring forces are used, the minimum velocity to fully open the valve is 13.76 m/s. The corresponding Reynolds number will be 0.5 x 10^6, which is still high enough to maintain turbulent flow.

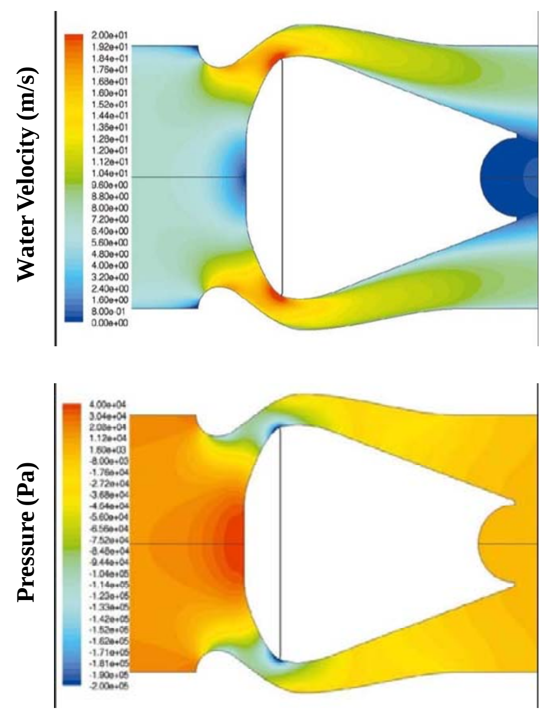

FIGURE 9: VELOCITY AND PRESSURE FLOW FIELD IN A NOZZLE CHECK VALVE (FULL DISC) – REF. [19].

As the valve closes due to lower flow velocity, the flow passage geometry will change, and flow will be more prone to separation, which in turn will give rise to higher pressure drag. Additionally, the gap between the disc and the valve seat will be smaller which will cause the flow to accelerate faster causing higher local frictional drag. The combined effect will result in a higher C_D coefficient as the valve closes. When the valve disc is at near closed position, the major contributor to the hydrodynamic drag will be mostly due to the pressure drag arising from the stagnation pressure on the frontal surface of the disc and static pressure on the back side.

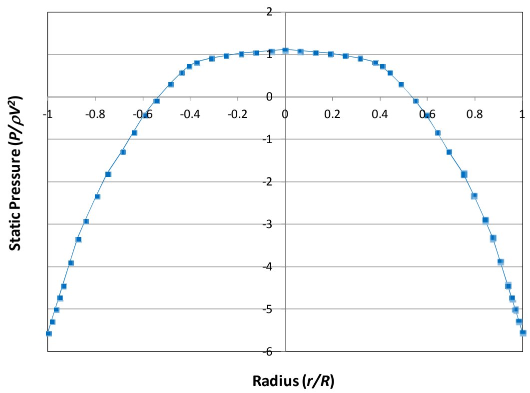

FIGURE 10: NORMALIZED PRESSURE DISTRIBUTION ON THE DISC SURFACE OF THE NOZZLE CHECK VALVE OF FIG. 9.

EFFECTS OF SPRING STIFFNESS

With the aid of the C_D coefficient, it is possible to assess the effects of spring forces and stiffness on the check valve performance in compressor station applications. For given suction and discharge conditions of the gas flow across the compressor, the minimum flow to fully open a given check valve can be determined via the balance of forces of Eq. (2), written in the form:

$$C_D \rho V_o^2 D^2 = (F_s)_{\max} = F_o + K X_{\max} \tag{7}$$

In the above equation, both gas density (ρ) and mean flow velocity (V_o) in the pipe correspond to the local gas condition immediately upstream of the check valve. For a compressor unit check valve, the gas condition would be that at the discharge of the compressor unit. Once the mean flow velocity (V_o) is determined from Eq. 7, and for a given spring force and stiffness, the mass flow rate can be calculated and related back to the suction condition of the compressor in terms of actual inlet flow rate. This allows superimposing the check valve flow characteristics at a fully open position on the compressor performance characteristics. An example of this is shown in Fig. 11, for the ring disc type valve discussed earlier with C_D = 0.796, $(F_s)_{max}$ = 2.549 kN and stiffness K = 15.833 kN/m. The compressor performance characteristics are that of a Nuovo Pignone PCL 603 that is driven by a LM2500 gas generator driving a PGT25 power turbine ISO rated at 23.3 MW. The unit check valve is 750 mm (30 inches) in size. Suction pressure ranges between 7-8 MPag while discharge pressure ranges between 11-12 MPag. The maximum adiabatic head of this compressor unit is 60 kJ/kg, and the pressure ratio is ~ 1.6, hence the compressor is equipped with two centrifugal impellers.

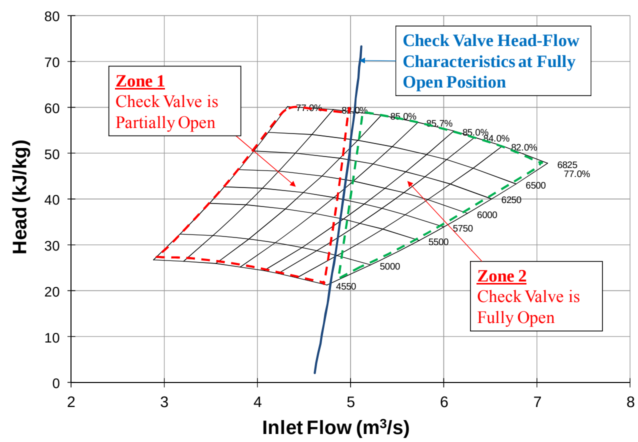

FIGURE 11: EXAMPLE OF THE RING DISC CHECK VALVE FULL OPEN CHARACTERISTICS ON COMPRESSOR PERFORMANCE CHARACTERISTICS.

In this example, it is shown that the check valve flow characteristics cut across the compressor performance map as shown in Fig. 7. The significance of this is that, depending on the operating point on the compressor map, there will be some conditions where the check valve will be partially open (Zone 1 to the left of the check valve characteristic line, and other conditions where the check valve will be fully open (Zone 2 to the right of the check valve characteristic line). Figure 12 shows an example of one month worth of operating data on an actual compressor station employing this compressor and check valve type. It appears that more than 50% of the time, the unit check valve was partially open.

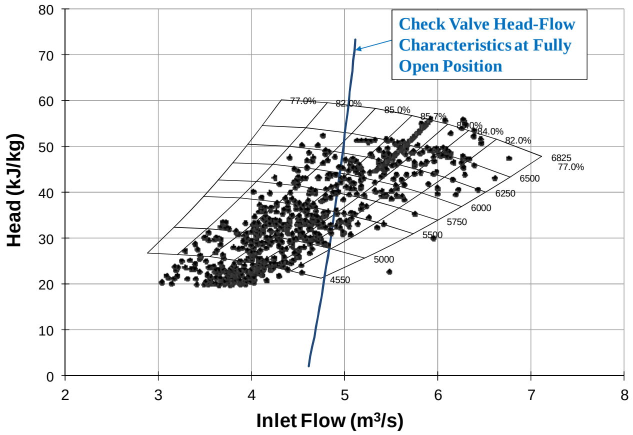

FIGURE 12: ACTUAL OPERATING DATA OF A COMPRESSOR STATION ON COMPRESSOR PERFORMANCE MAP.

The implication of a partially open nozzle type check valve on valve integrity and performance can now be discussed. In the case of ring disc type valves, the disc will be hung on its radial guides by guide arms somewhere between fully open and fully closed positions, as shown in Fig. 13 [20-23]. With the disc in this position, it will be subjected to flow disturbances, turbulence and oscillations commonly found in gas flows at the compressor discharge. This will give rise to forced disc oscillation (fluttering) which could lead to damage to internal components or dislodging of the disc/radial guide/spring assembly.

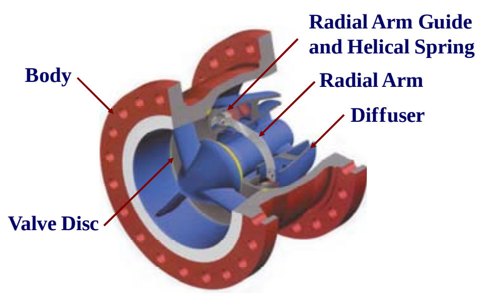

FIGURE 13: SCHEMATIC OF A RING DISC NOZZLE TYPE CHECK VALVE SHOWING THE RADIAL GUIDES AND GUIDE ARMS AND SPRINGS – REF. [20].

In order to investigate this, a two degree of freedom forced vibration model for the internal valve assembly for this valve was developed. This model is shown schematically in Fig. 14. Stiffness provided by the radial guide arms was neglected. The equations of motions representing the forced vibration of this system can be expressed as follows:

Disc equation of motion:

$$M_1\ddot{X}_1 = -K_1(X_1 - X_2) - C(\dot{X}_1 - \dot{X}_2) + A_o e^{i\omega t} \qquad (8)$$

Radial guides equation of motion:

$$M_2\ddot{X}_2 = +K_1(X_1 - X_2) + C(\dot{X}_1 - \dot{X}_2) - K_2 X_2 \qquad (9)$$

where:

X_1 = displacement of the disc
X_2 = displacement of the radial guides
M_1 = mass of the disc
M_2 = mass of all radial guides + 1/3 of springs' mass
K_1 = total spring stiffness of the springs
K_2 = very small value of stiffness (~ 0) to represent connection between M_1 and M_2.
C = damping coefficient arising from the friction forces between radial guides and disc rings.
$A_o e^{i\omega t}$ = hydrodynamic oscillation force of amplitude F_o and frequency ω.

The amplitude A_o is related to hydrodynamic drag coefficient C_D via perturbation of Eq. (3), i.e.

$$A_o = 2C_D \rho V_o . \delta V_o D^2 \qquad (10)$$

or

$$\frac{A_o}{F_H} = 2\frac{\delta V_o}{V_o} \qquad (11)$$

Amplitudes of flow oscillations can be determined from the compressor flow suction/eye signal and application of Fourier transform (FT) to these signals. Four flow signals were taken while the compressor was operating at different flow conditions. The corresponding amplitudes vs. frequency obtained by FT are shown in Fig. 15. It is shown at a frequency of 1 Hz, the amplitude of flow oscillations is around 2% of mean flow.

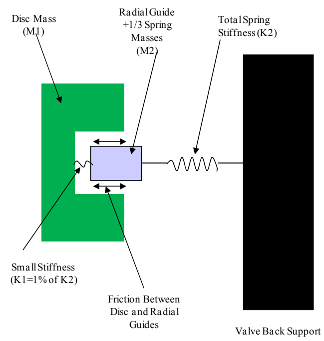

FIGURE 14: SCHEMATIC OF TWO-DEGREE OF FREEDOM VIBRATION MODEL OF THE RING DISC TYPE CHECK VALVE INTERNAL ASSEMBLY.

The damping coefficient, *C,* was assessed based on the friction force between two sliding steel surfaces of the radial guides and the ring disc having a kinematic coefficient of friction (dry) = 0.4. Masses and spring stiffnesses used in the analysis are given in Table 2. The above two equations of motion were solved simultaneously using Runge-Kutta (4^{th} order) method and the resulting oscillations are presented in Fig. 16 in terms of relative displacement and velocity between the ring disc and radial guides for an amplitude of flow oscillations of 2% of mean flow. The results indicate that relative displacement between the disc and the radial guide is as high as 0.1 m. It also indicates that: i) the radial guides can lose contact with the disc, and ii) there is also possibility of the disc slamming into the front and/or back seat depending on the disc position, given that the full stroke of the disc is limited only to 0.0914 m in this valve. The relative velocity between the radial guide and the ring disc is shown to be on the order of 0.25 m/s which could lead to continuous gouging between the two elements. These results are a clear manifestation of the lack of damping which is inherent in this ring disc type valve.

For the above reason, it is very important to select spring stiffness such that the disc would be at the fully open position at the minimum expected flow. This would be the flow corresponding to minimum compressor speed and flow corresponding to a point close to the surge limit at this speed. For this valve, a spring stiffness such that total $(F_s)_{max}$ = 0.91 kN would ensure that the check valve flow characteristics are located to the left of the entire compressor performance map as shown in Fig. 17. Conversely, a spring of total $(F_s)_{max}$ = 5.4 kN would be very detrimental as the valve flow characteristics would be located to the right of the compressor performance map, as also shown in Fig. 17.

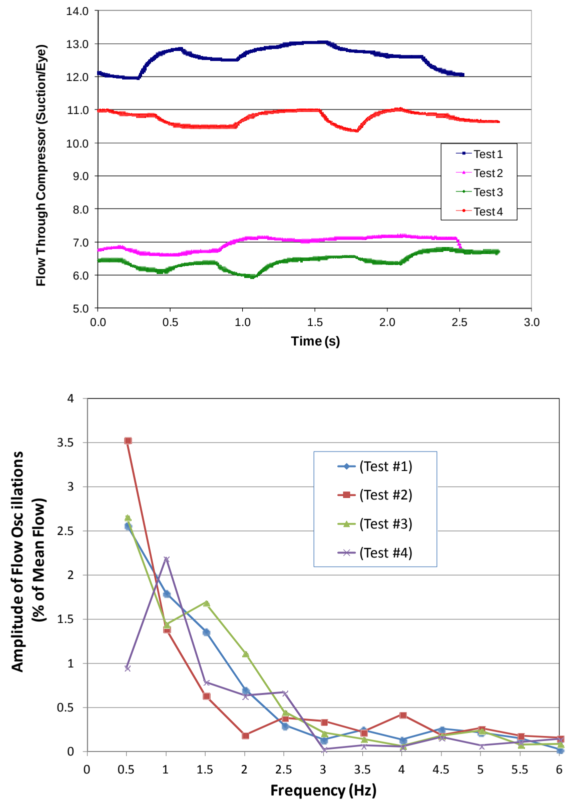

FIGURE 15: MEASURED FLOW OSCILLATION AT COMPRESSOR SUCTION AND CORRESPONDING AMPLITUDE SPECTRA.

TABLE 2: RING DISC TYPE CHECK VALVE DATA USED IN THE PRESENT CASE STUDY.

Disc mass	84	kg
Guide Mass	2.7	kg
No. of guides	3	
Total Guide Mass	8.1	kg
No. of springs	3	
Mass of one spring	0.75	kg
Total mass of springs	2.25	kg
Effective Total Mass	92.85	kg
Stiffness of one spring	5278	N/m
Total Stiffness of springs (k)	15833	N/m
Length of one spring	0.267	m
length of spring (when valve is fully closed)	0.1974	m
length of spring (when valve is fully open)	0.106	m
Stroke (closed to open)	0.0914	m
Force to fully open valve	2549	N

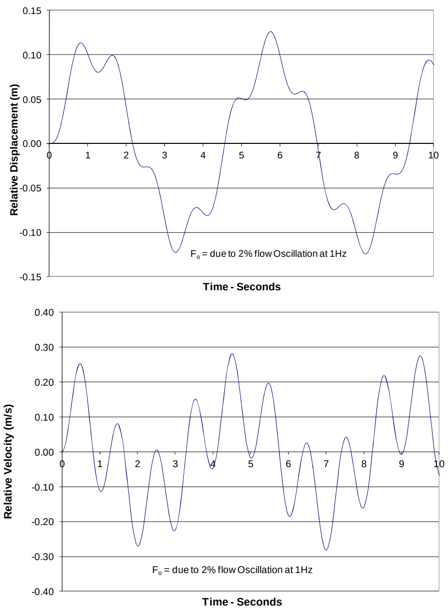

FIGURE 16: DISC FLUTTERING DUE TO FLOW OSCILLATIONS AT 1 HZ.

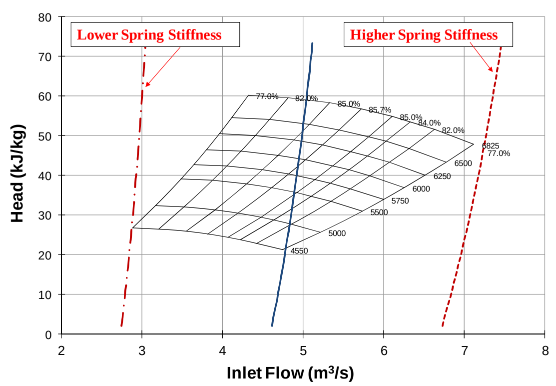

FIGURE 17: EFFECTS OF SPRING STIFFNESS ON CHECK VALVE FLOW CHARACTERISTICS w.r.t. COMPRESSOR PERFORMANCE MAP.

DIFFERENCE BETWEEN UNIT AND STATION CHECK VALVES

The flow conditions at the station check valve are different than that at the unit check valve in two ways. First, the amplitudes of pressure and flow oscillations at the station check valve are different than that at the unit check valve. An acoustic model was developed based on the transfer matrix theory in [24]. The model includes all piping elements, fittings and, in particular, the volume capacitance brought about by the aerial cooler 9see Fig. 1). The acoustic model predicts the ratios of both pressure amplitudes and velocity amplitudes at the two check valves. The results are shown in Fig. 18, which indicate that there is pulsation suppression of velocity amplitude (which range between 0.5 to 1.0) in the frequency range of 0-3 Hz (which is the range where maximum amplitude of flow oscillations were found (Fig. 15). Since it is the flow velocity oscillations (and not the pressure oscillations) that cause the disc to vibrate due to the hydrodynamic forces exerted, as discussed earlier, the station check valve will be subjected to less vibration than the unit check valve. This flow pulsation damping and suppression is brought about by the acoustic transmission loss and volume capacitance of the aerial cooler upstream of the station check valve.

The second difference between the two check valves lies in the fact that a station check valve could experience lower flow rates than a unit check valve. This is due to possibility of flow recycling when the station is put on a part-load mode of operation as part of the overall pipeline control scheme. In fact, a station check valve can experience extremely low flow that is far left of the compressor performance map. For this reason, a station check valve could be operating at partially open position more often than a unit check valve, and hence its spring stiffness should be selected with this in mind. That is, if part-load mode of operation of the station is expected to be an accepted operation for a long time, lower spring stiffness should be employed in this valve than that in the unit check valve.

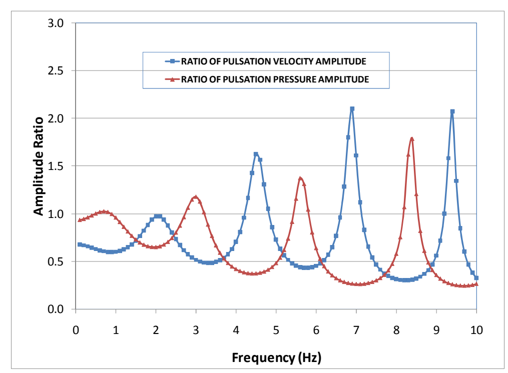

FIGURE 18: RATIO OF PRESSURE AND FLOW VELOCITY AMPLITUDES OF STATION TO UNIT CHECK VALVES.

DISC SLAMMING VELOCITIES

It is well known that compressor units of high head and low rotor inertia are prone to surge upon emergency shutdown (ESD), especially with only one recycle system that includes the aerial cooler in its path as that shown in Fig. 1. The question then is: “How do check valves (unit and station) react to an ESD operation in this case. Measurements and dynamic simulations were conducted on this system which incorporated the compressor unit and ring disc type valves as unit and station check valves. The results of both can be found in [25]. The following are results of further analysis focused on the dynamics of the internal assembly of the check valves (disc/radial-guides) to the flow conditions developing during an ESD situation.

The measured data of an ESD operation are plotted in Fig. 19 on the compressor performance map. The flow through the compressor is also shown in terms of the compressor suction/eye signal. Since flow through the unit check valve was not measured, it is assumed that the flow through the unit check valve is close to that measured through the compressor by the suction/eye signal. The hydrodynamic force on the disc can then be calculated via Eq. (3) and using the C_D coefficient determined earlier. The next step is to apply this hydrodynamic force, which is function of time, to the equation of motion of the disc/radial guide/spring assembly, assuming a one-degree of freedom model, as follows:

$$M_1 \ddot{X}_1 + C\dot{X} + K_1 X_1 = F_{HD} \tag{13}$$

The above equation is again solved using the Runge-Kutta 4th order method from an initial condition of disc location that is the steady state flow prior to ESD, and zero disc velocity. The results are shown in Fig. 20 in terms of disc displacement and velocity. Positive velocities are those associated with the disc moving forward (i.e., closing), and negative velocity corresponds to the disc moving backward (i.e. opening). The results show several slamming (pounding) at the front and back seats of the valve on the order of 1.0-1.6 m/s.

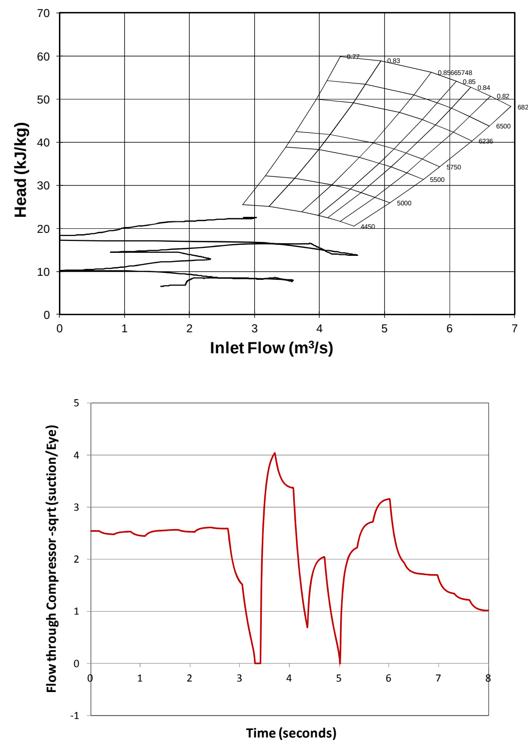

FIGURE 19: MEASURED FLOW CHARACTERISTICS DURING COMPRESSOR ESD.

Similar analyses were conducted on data obtained from dynamic simulations of an ESD reported in [25] and are shown in Fig. 21. Here the mean flow velocity data are obtained precisely at the check valve location, and are used to estimate the hydrodynamic forces on the disc. Figure 22 shows the results in terms of disc displacement and forward and backward velocities. It indicates forward and backward slamming of the disc several times, the highest slamming velocity being 1.6 m/s in the forward motion, and 0.9 m/s in the backward motion. Therefore, nozzle check valves should be designed to withstand these slamming velocity magnitudes and frequencies over the life of the station. Their function is primarily to protect the compressor units from sustained back flow (surge) and not to be subject to self damage themselves. The station check valve slams shut once upon compressor ESD and, hence, it repeated slamming is not a concern during ESD.

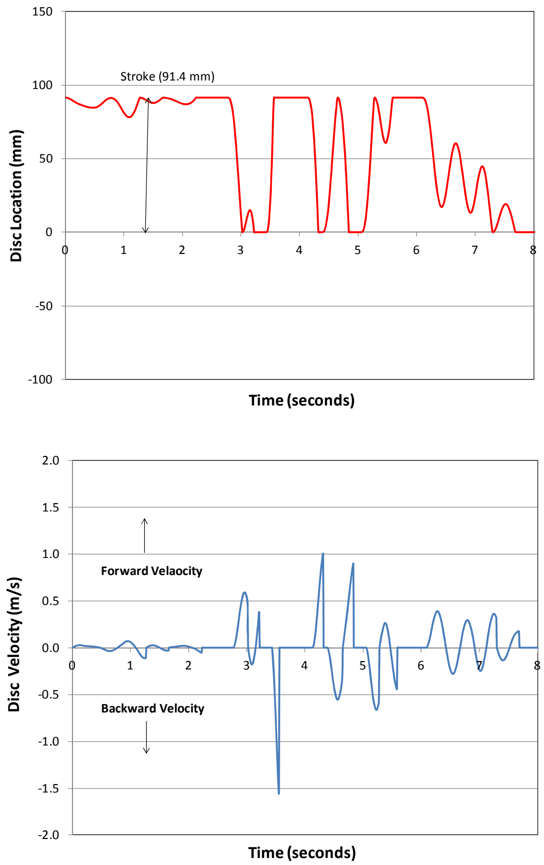

FIGURE 20: RESPONSE OF THE UNIT CHECK VALVE TO COMPRESSOR ESD OF FIG. 19.

CONCLUDING REMARKS

The internal geometry of a nozzle check valve should be optimize such that the highest possible hydrodynamic drag coefficient is realized when the disc is at its fully open position. This can be achieved by varying the contour of the inner surface of the valve body as well as the disc/back seat geometry such that the local minimum static pressure occurs *exactly* at the back side point of the disc when it is at a fully open position.

If nozzle check valves are employed in compressor station applications, spring forces and stiffness should be selected on the basis of the relationship between the valve flow characteristics at fully open position in relation to the compressor performance map. In order to avoid a partially open valve, the valve flow characteristics at fully open position should be located to the left of the entire compressor performance map. Higher spring forces and spring stiffnesses result in the check valve being partially open which could lead to disc fluttering even during normal steady flow condition. Disc fluttering is detrimental to the valve integrity as it can lead to high impact velocities of the disc against its seat and subsequent self destruction.

Spring forces and stiffness for station check valves should be lower than that for unit check valves, especially if part-load mode of operation is expected for an extended period of time.

It is important to recognize that the main function of the check valves is primarily to protect the compressor unit from damage due to reverse flow (surge). Therefore, the check valve should be designed to withstand high slamming velocities (or to be non-slamming) during compressor ESD, and not be subject to self destruction themselves.

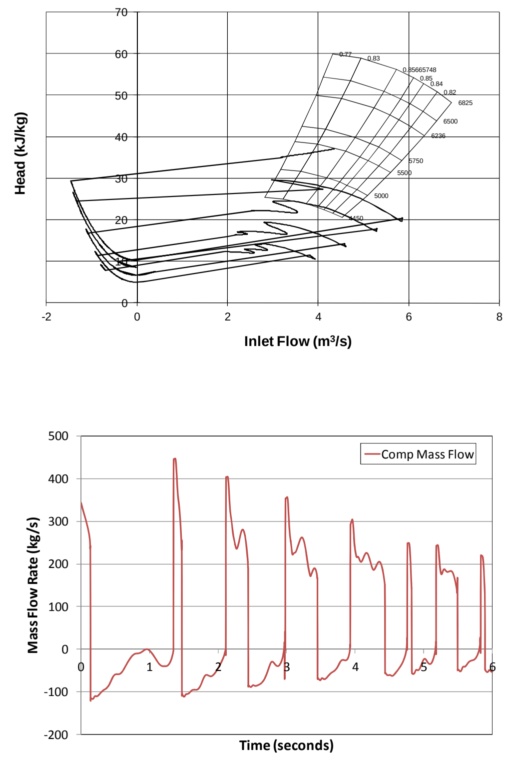

FIGURE 21: SIMULATED FLOW CHARACTERISTICS DURING COMPRESSOR ESD.

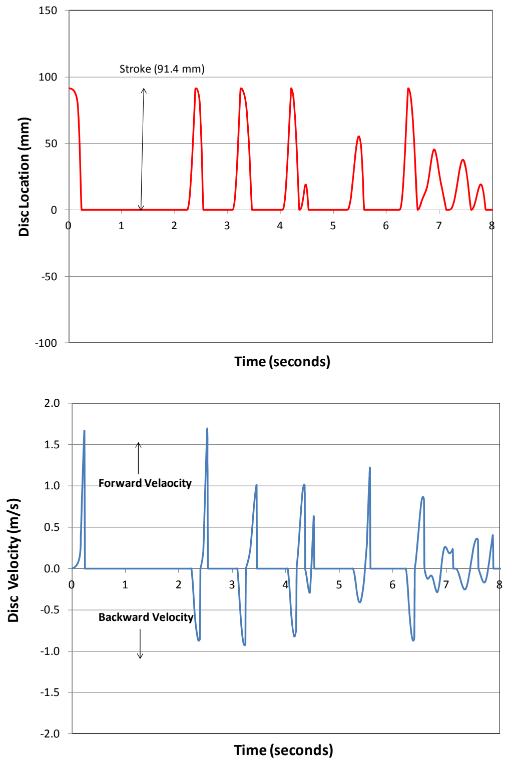

FIGURE 22: RESPONSE OF THE UNIT CHECK VALVE TO COMPRESSOR ESD OF FIG. 21.

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