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Estimating Maximum Gas Velocities in Oxygen-System Valves[2]

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ABSTRACT: An earlier paper estimated the maximum velocities that can occur in oxygen piping systems, specifically within and downstream of isolation valves that are opened with differential pressure applied. The goal was to determine several applications where the use of a bypass valve (per Compressed Gas Association Pamphlet G-4.4) might be omitted. The equations that were applied raised controversy. These equations are developed and defended relative to three criticisms voiced and still appear reasonably applicable for the intended use. The original analysis is expanded to address not only adiabatic flow but isothermal flow, and the differences found are small relative to the domain of practice for these valves.

KEY WORDS: oxygen, oxygen compatibility, flammability, flammability limits, fire, velocity, bypass valves, isolation valves.

Castillo and Werley [*1*] recently published a paper on the use of bypass valves and selected circumstances in which they may not be needed even across carbon steel isolation valves. As a part of the paper, the authors estimated the "maximum" average velocity likely to be present in an oxygen valve during certain conditions of use. A specific equation was cited from Perry [*2*] and applied, but it was not derived nor justified in detail in the paper.

This equation proved to be controversial among several peers on Committee G-4. Numerous nonspecific questions (some vague) have been raised as to its origins and its suitability in comparison to other alternatives. Although seldom dealt with, it is of benefit for oxygen compatibility practitioners to have familiarity with the fluid mechanics of

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oxygen systems. Therefore, this paper elaborates on the equation and attempts to defend its use, so that G-4 peers may consider all alternatives and the best option for this purpose may be established.

Among the specific concerns raised were (1) the equation was inappropriately applied, since Perry cites the equation in a section on Pitot tube use for which the size of the Pitot tube is typically taken as a very small portion of the related pipeline cross-sectional size, (2) a bypass valve is more like an orifice and should use equations to that end, and (3) the equation should only be applied to equilibrium flowing systems rather than to the transient state during the initial opening of a valve.

The equation at issue appears in Perry [*2*], and in manipulated form in Zabrenski et al. [*3*] as:

$$P_T/P_D = [(v_D^2 \rho_D / 2g_c K P_D) + 1]^K \quad (1)$$

where: P_T is the source pressure in absolute units (psia or MPa), P_D is the downstream pressure in absolute units (psia or MPa), v_D is the maximum magnitude of velocity downstream (f/s or m/s), ρ_D is the downstream gas density (a function of downstream pressure in lb/ft^3 or kg/m^3), g_c is a dimensional constant (4636 lb in^2/lb$_f$ s^2 ft or 1 kg/N s^2), and K is given by the quantity $[\gamma/(\gamma-1)]$ where γ is the ratio of the gas's specific heat at constant pressure to constant volume (C_p/C_v).

Equation (1) can be rearranged and substituting the ideal gas expression $P_D/T_D R$ for ρ_D, yields the equation as it appears as Equation (4) in Castillo and Werley [*1*]:

$$P_D = P_T/[(v_D^2/2g_c K R T_D) + 1]^K \quad (2)$$

where: T_D is the temperature downstream in absolute units (R or K), and R is the Universal Gas Constant in consistent units (0.333 ft^3 lb$_f$ /°R lb in^2 or 26 Nm/kg°K, respectively).

Equation (2) was used to estimate ratios of upstream to downstream pressures to limit the maximum gas velocity through a valve to the values allowed by CGA Pamphlet G-4.4. Since the values allowed by G-4.4 are low (25-200 ft/sec[7.6-61 M/s]), downstream temperatures were taken as the same as the upstream temperature. However, when high velocities are estimated, then T_D is less than this estimate and lower velocities would be calculated indicating the estimate is conservative relative to this effect. The extent of this effect will be analyzed later.

Clearly, the use of this or any other specific equation depends on the assumptions applied to the specific system, and so this paper is to: (1) cite the fundamental assumptions applied, and then it attempts to: (2) defend the use of the "Pitot tube equation" and trace the origins of the equation's application, (3) defend the use of the equation in comparison to the equations for an orifice, (4) defend use of the equation even in some instances for the very first (transient flow) instant that a valve is opened, and finally: (5) indicate conditions for which the equation may not provide a practical estimate of maximum gas velocity.

Fundamental Assumptions

As reported in the previous paper and as tersely stated in CGA Pamphlet G-4.4 [*4*] bypass valve use is defined as follows:

> "Where required to minimize adiabatic compression or particle impingement, a small manual valve bypass valve should be provided around isolation valves and shall be of copper base material because of the high velocity involved."

G-4.4 contains no explicit definition of a bypass valve, presumably taking it as obvious. This description is contained within the specific definition of isolation valves, and begs for elaboration. However, in most industrial applications, a bypass valve is small in order to allow for economy as well as to allow for slow pressurization of the downstream system. In some applications the bypass valve could be the same size as the related isolation valve, but unless pressure drop were crucial, there would then be no need for the isolation valve.

Therefore, the assumptions for a bypass valve are as follows:

- The valve is inherently slow opening or is capable of slow-opening (and is used that way), meaning either it is physically small (implying a small orifice), and/or its pressure controlling element can be precisely adjusted (as in a needle valve) and the pressure drop across it allows for slow pressurization of the downstream system. When an isolation valve is being used as its own bypass-valve surrogate, it would never be opened more than a small amount at a time during surrogate bypass use, regardless of how small the differential pressure across it is.

- Its operation is analyzed only in its opening mode, during which the upstream system is basically stagnant. Bypass valves have no specific function during their closure.

Defense of the Pitot Tube Equation

Much distraction appears to have occurred because the equation used [which yields Equations. (1) and (2)] appears in Perry [*2*] in its section discussing the operation of Pitot tubes. Concerns have been expressed that Pitot tubes are required to be a tiny fraction of the size of the piping run in which they are installed and carry many other constraints on their use. One peer argued that a Pitot tube was irrelevant and that the operation of a bypass valve is more nearly like that of a orifice, and that this would imply greater velocities than estimated in the prior paper. However, in the authors opinion, the former misinterprets the significance of the equation and the latter [to be addressed later] is in error.

Pitot tube hardware is, indeed, irrelevant to this analysis. However, Pitot tubes are used to measure total pressure and/or equivalently gas velocity. Therefore, the equation used to relate a Pitot tube response to a piping system velocity is the same equation the authors sought to analyze the bypass valve situation. In the earlier publication by Zabrenski et al. [*2*], this equation was first cited to infer what the total pressure might be on a metal component situated in a flowing oxygen stream. It treated a flowing stream as a

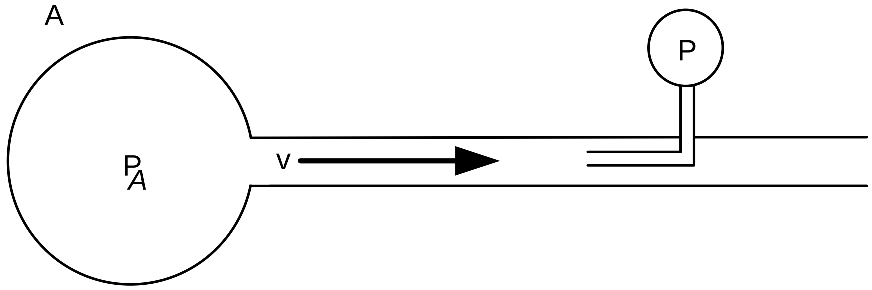

FIG. 1—*Pitot tube.*

more severe environment than the environment that a line pressure gage suggests, because at points of stream impingement, the kinetic energy in the stream can be converted back into potential energy and exhibit a greater "stagnation" pressure. Therefore, it proposed that in some cases, those who use metals in flowing streams may wish to limit the total pressure (rather than the line gage pressure) to a value less than the threshold at which the metal is know to sustain combustion propagation (with an applied safety factor). This has often been Air Products' strategy. Given that a gas stream is at a given velocity, one could find the total pressure by installing a Pitot tube and measuring it, or one can insert the known maximum gas velocity into the Pitot tube equation and solve for the total pressure.

In preparing the more recent paper [*1*], it was noted that the conversion of kinetic energy (resulting from the oxygen velocity) into potential energy (resulting in an increase of pressure) is a symmetrical, reversible process. Fig. 1 illustrates this point. A vessel, A, at an approximately stagnant pressure P_A supplies flow through a lossless pipeline which contains a Pitot tube attached to a small second vessel. Since A is stagnant, its line pressure is its total pressure.

As oxygen flows from A into the pipeline, its potential energy is converted into kinetic energy (developing its velocity). When the gas stream impinges on the Pitot tube, a portion passes into the tube and, since it is forced to halt, develops pressure in the second vessel which will build until it's stagnant. At that point, therefore, the kinetic energy would have reverted back into potential energy, and in an adiabatic lossless system, the line gage pressure in Vessel B would match that in the vessel A. Note that this process may be reversed. If Vessel A is initially empty and the gas stream is reversed into A then for an initial instant (until its pressure falls) the vessel on the Pitot tube will contribute its contents to A, in the process generating a velocity V in its own line that would be the negative (same magnitude, opposite direction) of the velocity that charged the vessel. Therefore, it was concluded on the basis of this symmetry argument that the Pitot tube equation was appropriate for estimating the magnitude of the velocity, **v**, from a stagnant source for flow in either direction. In effect, we were visualizing the use of a micro-Pitot tube in a bypass valve.

This can be shown from a more detailed analysis that will follow, however, this is how the Pitot tube equation from Perry [*2*] came to be cited in both of the previous papers [*1,3*]. The Pitot tube equation by symmetry predicts the relation between the pressure in Vessel A and the velocity it produces in the oxygen stream. Indeed, the conditions that a

Pitot tube be a small fraction of the internal piping area is merely to preclude its causing a choke point and, consequently, an artificially elevated velocity that can lead to errors.

Origins of Equation (1)

The first analysis presented here is patterned after the thermodynamics and fluid mechanics presented by the ASME in its publication on fluid meters [5]. Fig. 2 illustrates gas flow at two points in a system separated by a boundary. The boundary is shown as a reduction in piping diameter but might be a valve, an orifice plate, a Pitot tube, an obstruction, etc. As the gas moves across the boundary, its condition changes. The pressure (P), temperature (T), volume (V), velocity (**v**), flow area (A), etc. may all be different on the opposing sides of the threshold.

The left side will be taken as the upstream condition and the right side as the downstream condition, meaning the gas is moving from left to right. To the left of the transition the conditions of pressure, temperature, volume, velocity, and area are designated respectively as: P_u, T_u, V_u, $\mathbf{v}_u$, and A_u, where the subscript, *u*, indicates upstream. Similarly, the downstream conditions would exhibit a subscript: *d*.

The ASME analysis argues that:

> "Since the assumption has been made that the fluid neither receives nor imparts energy from or to surrounding bodies, the total energy at the two sections will be equal..."

Therefore:

$$u_{iu} + u_{ku} + P_u V_u = u_{id} + u_{kd} + P_d V_d \quad (3)$$

Where, u_{iu} and u_{id} are the upstream and downstream internal energies, u_{ku} and u_{kd} are the upstream and downstream root-mean-square kinetic energies, and $P_u V_u$ and $P_d V_d$ are the mechanical work done on (or by) the upstream and downstream flows.

The ASME analysis is characteristic of thermo- and fluid-dynamics analysis in textbooks, and it lumps all of the transitions that occur into this single overall energy transaction (balance). The total energies (kinetic plus potential plus external actions) on each side of the transition are established, and then they are equated.

However, for a hopefully simplified understanding, the terms of this equation may be resolved into equivalent mathematical constituents and each may be analyzed physically.

Fig. 2 illustrates the upstream and downstream flow at a square-edged change in piping dimension. Elements (slugs) of flow are illustrated in both streams and each contains the same mass of oxygen. For every slug entering the upstream inlet, S_u, there is a slug of equal mass, S_d , that exits the downstream outlet (Note: that the molecules comprising an incoming slug may not be the same collection of molecules that comprise an outgoing slug).

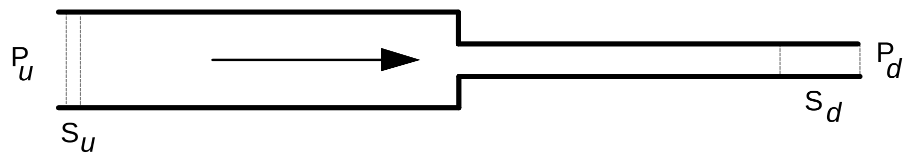

FIG. 2 ¾ *Flow into square-edged piping transition.*

Consider just one such slug of gas, S_u, moving from left to right and having a kinetic energy of u_{ku}. Consider the piping to be lossless and the flow to be stable and in equilibrium.

Consider next a hypothetical expansion engine inserted at the transition in diameter as in Fig. 3. The suction side of this engine takes in slugs of gas at the diameter of the upstream piping and discharges them into the downstream piping. Fig. 4 illustrates the internal workings of this engine in magnified form and shows its various stages of operation

As a slug, a, moves to the inlet of the expansion engine, its volume does not change, because the pressure on both sides of it do not change and it incurs no friction losses or energy changes. Therefore its velocity and kinetic energy are constant. This is stable equilibrium flow.

Ultimately this upstream slug arrives at the expansion engine as shown in Fig. 4. As it arrives at the engine, Stage 1, we will assume the engine encapsulates it in a cylindrical vessel of zero mass and zero heat capacity and allows the higher upstream pressure to push the slug into the engine into a vacuum as shown in Stage 2 of Fig. 4. When this has happened, the vessel containing the gas at upstream pressure will acquire kinetic energy equal to the net work, $\int \mathbf{F} \bullet d\mathbf{l}$, done on it:

$$\int \mathbf{F} \bullet d\mathbf{l} = \int P dV = P_u \Delta V = P_u V_u. \tag{4}$$

This slug of gas at stage 2 would be moving faster than it was moving upstream of the engine. If these encapsulated slugs were released into a downstream pipe of corresponding dimension, the greater velocity would imply gaps between the slugs would be present.

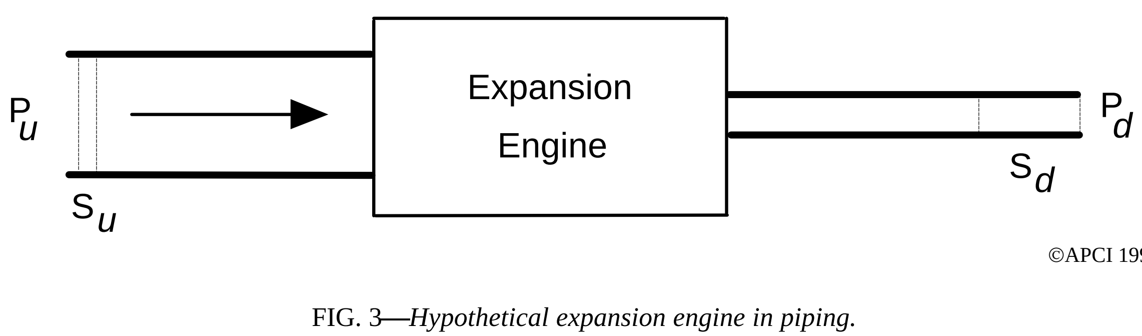

FIG. 3—*Hypothetical expansion engine in piping.*

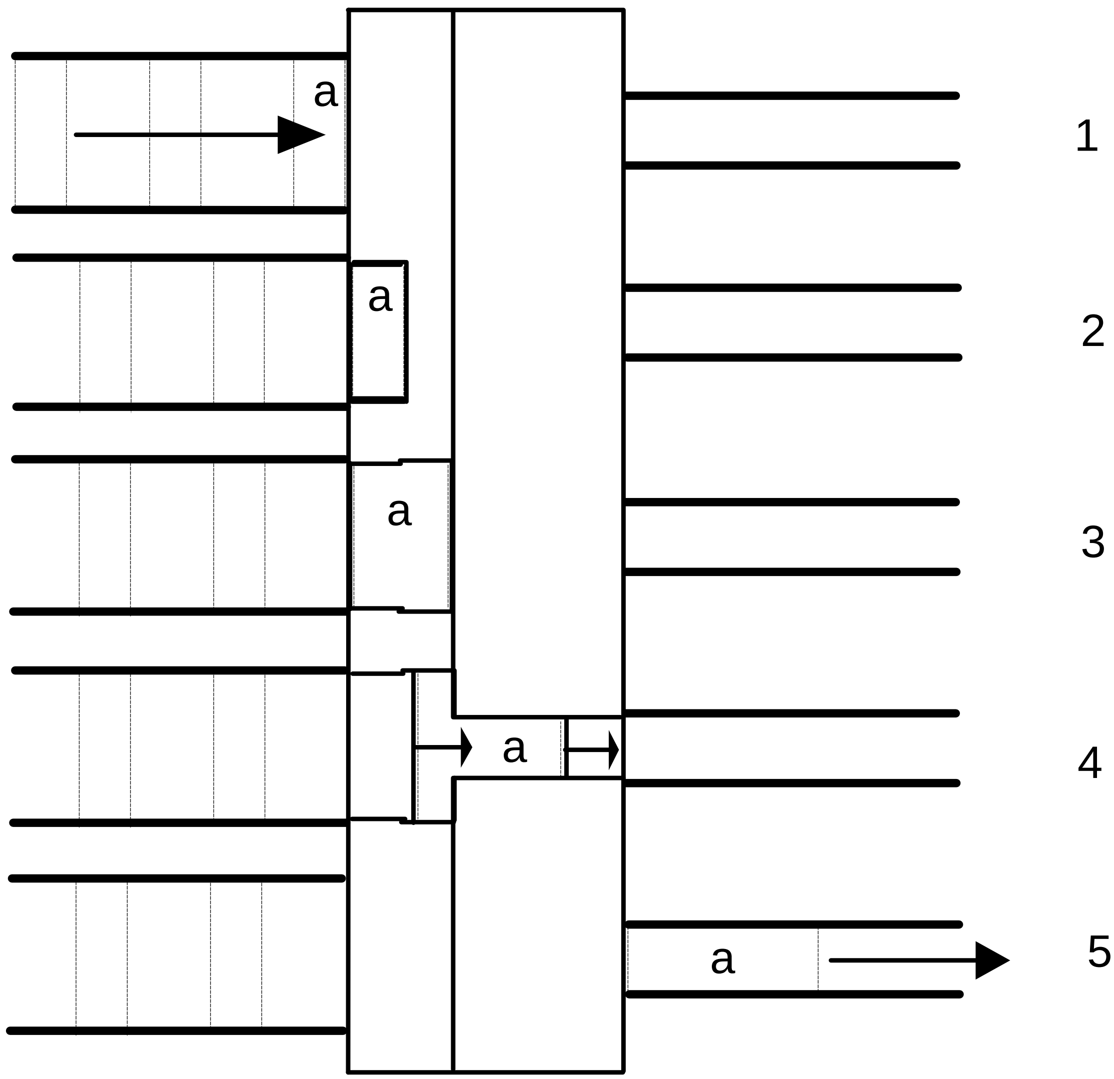

FIG. 4—*Hypothetical expansion engine.*

The expansion engine now allows the vessel to expand (telescope) and the gas to accomplish work on itself, Stage 3, until the pressure matches that of the downstream system. The work it accomplishes is once again:

$$\int \mathbf{F} \bullet d\mathbf{l} = \int P dV \tag{5}$$

This analysis is performed for a boundary adiabatic expansion and is analyzed in a recent paper [*6*] and in earlier work by others [*7,8*]. The work it accomplishes is equivalent to the adiabatic mechanical TNT equivalency of the pressurized gas. In typical flowing streams (such as gas flow through a valve or orifice plate or Pitot tube), the expansion would be taken as rapid and, therefore, approximately adiabatic expansion is a reasonable expectation.

Reference [*6*] indicates adiabatic work done on the slug by itself in vacuo which would add to its kinetic energy is given by:

$$E = [1/(\gamma-1)]\,[(P_u V_u) - (P_d V_d)] \quad (6)$$

Next the expansion engine transfers the gas at constant pressure into a vessel of smaller diameter but greater length without changing its volume (Stage 4). For this the net work is again:

$$(\int PdV = P_d \Delta V) \quad (7)$$

This work is zero, because $\Delta V=0$.

Finally, the engine allows the slug to pass through its outlet into the downstream piping (under the influence of its own inertia) and the containment vessel is removed (Stage 5). In this case, negative work is done by the slug on the gas downstream, given as before by:

$$\int PdV = P_d \Delta V = - P_d V_d \quad (8)$$

Therefore, the kinetic energy of the final slug is its initial kinetic energy plus the work energy performed by the upstream pressure ($P_u V_u$) plus the expansion work performed on it (its TNT equivalency), plus the negative work energy it performs on the downstream system in passing through the engine, or:

$$u_{kd} = u_{ku} + P_u V_u + [1/(\gamma-1)]\,[(P_u V_u) - (P_d V_d)] - P_d V_d \quad (9)$$

This equation may be rearranged to the form of Equation (4) as it appears in the ASME analysis. Note that under the assumptions for (1) a slowly-opened bypass valve or (2) use of a low-differential-pressure valve as a "bypass valve surrogate," the cross-sectional area of the upstream piping is much greater than the downstream area (the flow-limiting region between the valve control members). This implies that upstream velocity starts out as zero and never achieves significant magnitudes during the downstream pressurization. Therefore:

$$u_{ku} = [m/2]\,|\mathbf{v}_u|^2 \approx 0 \quad (10)$$

Also:

$$u_{kd} = [m/2]\,|\mathbf{v}_d|^2 = [m/2] v_d^2 \quad (11)$$

Using Equation (10) and (11) in Equation (9) and some algebraic manipulation yields:

$$u_{kd} = [m/2] v_d^{\,2} = P_u V_u + [1/(\gamma-1)]\,[(P_u V_u) - (P_d V_d)] - P_d V_d \quad (12)$$

Or:

$$v_d = \left[2\{P_u V_u + [1/(\gamma-1)]\,[(P_u V_u) - (P_d V_d)] - P_d V_d\}/m\right]^{0.5} \quad (13)$$

Equation (13) may be manipulated to:

$$v_d = \{(2/m)(P_u V_u - P_d V_d)[\gamma/(\gamma-1)]\}^{0.5} \qquad (14)$$

This relates the maximum velocity to upstream and downstream pressures and upon further manipulation yields Equations (1) and (2). Note that this is the same equation that is developed by the ASME [5] for Pitot tubes, because *the Pitot tube has no net throughput*, hence one of its kinetic energy terms (downstream) is similarly zero.

Note that there is a minimum dimension the downstream piping must have in order to accommodate the train of gas slugs that is being discharged from the expansion engine. Piping dimensions greater than this will result in some dissipation of the kinetic energy and lower-than-predicted velocity after the flow "smoothes." However, the velocity estimate above is an upper limit ***at every instant*** on the basis of energy balance, and the dimension above is the minimum that will correlate to the downstream pressure. If piping smaller than the minimum above were to be provided, then the slugs would not all "fit" into the downstream piping at the rate they are being introduced and some of the kinetic energy would be converted into potential energy reflected as a higher pressure.

Because energy is conserved, this is the maximum velocity that the energy available could produce, and the estimate should be conservative for the assumptions given. Indeed, in a real-world system, friction and other dissipative losses would tend to produce lower velocities, adding to the conservatism of the estimate *for the assumed adiabatic system*.

Finally, note that the kinetic energies above are calculated on the basis of root-mean-square (RMS) velocities. That is, on the basis of the square root of the average of the squares of the velocities over each element of area over the piping cross-section. RMS velocities are always greater than or equal to mean velocities cited in CGA G-4.4. Theoretical development of equations in the ASME Manual treats the flow as uniform (across the cross-section as well as in surging), but in real systems it is not. In the limit as the velocity approaches zero, the velocity distribution over the piping area approaches a paraboloid. A paraboloid distribution has an RMS velocity that is about 15% greater than its average velocity. Hence even at the low velocities allowed by G-4.4, the variation of velocity across the bore of the piping should be small and this approximation is therefore conservative by up to 15%. At high velocities the error should be even smaller.

Fig. 5 exhibits the maximum velocities the equivalent equations (1), (2) and (9) allow for two upstream pressures, assuming the expanded gas is at room temperature (solid lines) and the more precise case where temperature decreases adiabatically (dotted lines). Note that at the pressures of interest to bypass-valve use (<200 f/s[<61 m/s]), the two curves are virtually identical. The plot illustrates the extreme gas velocities that would be possible with the energy available. Note that at higher velocities, these estimates would break down, because, turbulence, friction and viscosity would be significant factors. In many instances, the formula is not valid (and is extremely conservative) as velocity cannot exceed the local speed of sound (horizontal line).

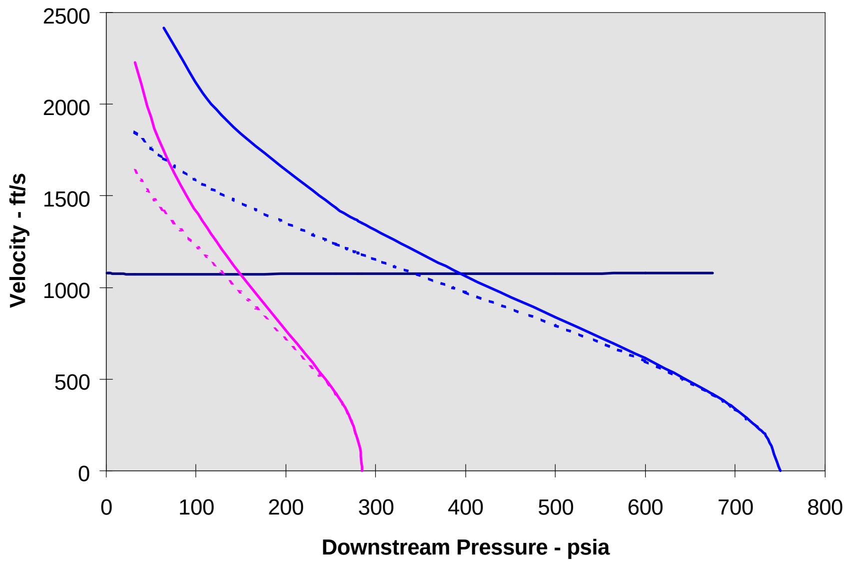

FIG. 5 ¾ *Estimates of maximum velocities achievable on energy basis, adiabatic case.*

Defense of Pitot Tube Equation Vs Orifice Equation

Interestingly, in the ASME Manual [*5*] orifices are analyzed with the same equations as Pitot tubes. Indeed, two classes of orifices are described: rounded-edge and square-edge. In the case of the latter, the exact same equation for a stagnant upstream is derived as Equation 1. In the case of rounded edge orifices, greater restriction is encountered, and contrary to peer concern, *lower velocities would be predicted in some modes of operation*.

Also interestingly, it is common practice to think of the velocities of the gas through and emanating from an orifice (or nozzle) as being limited to sonic regardless of the pressures involved. However, although the gas velocity in the bore of these devices (at the vena contracta) is limited to sonic levels, further downstream the velocity can and often is supersonic (and the pressure may fall below the downstream pressure). Notwithstanding, these high velocities are only achievable if there is sufficient energy available as predicted by Fig. 5.

Defense of Equation (1) for the Very Initial Opening of a Valve

This concern seems to originate from the general (and correct) understanding that pressure waves move through a gas system at the local speed of sound, which is always a much greater velocity than CGA G-4.4 recommends for oxygen in steel piping. Whenever a valve is opened, even slowly, there is an initial transient pressure wave that flows through the valve into the downstream system, and it moves at the speed of sound. This initial transient concerns some peers.

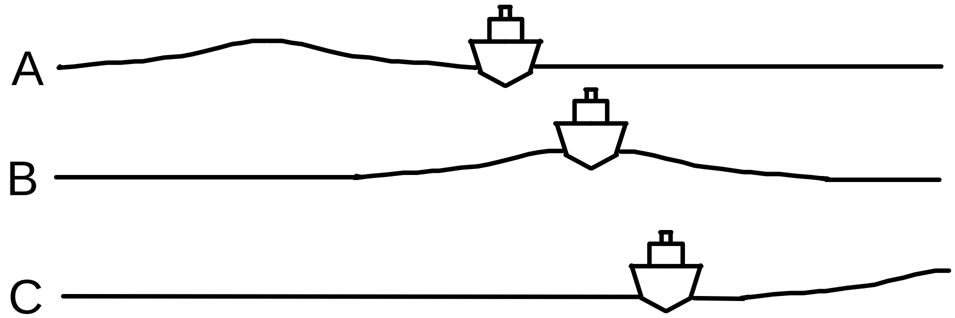

FIG. 6 ¾ *Ship and tidal wave.*

This is a subtle issue. Engineers who design pipelines are accustomed to using "Rules of Thumb." One of them deals with assumptions about pressure waves moving through systems. Indeed, "sonic" flow and "critical" flow are often discussed.

This seems to be the origin of the concern over the initial opening stage of a valve. Might there be transient velocities in excess of the ultimate steady state velocities? The author does not feel this is a valid concern.

One of the more subtle concepts in the basic physics of wave motion is the distinction between wave motion and medium motion. Because it is so easily overlooked, two examples pertinent to this concern will be drawn from basic physics.

Fig. 6 exhibits a ship at open sea that is being approached by a tidal wave. Tidal waves are known for their destructive power and fury. However, in open sea, a tidal wave behaves merely as a swell. At point A, the wave moving from left to right at great speed approaches the ship. At point B, the ship has been raised slightly as the wave passes through, and at point C the ship has been moved only slightly to the side after the wave has passed. The tidal wave has passed through at great speed, but the elements of the sea and the motions of "particles" in the sea (specifically the ship) as witnessed by the lateral motion of the ship have moved only slightly and at quite low velocity. The key factor in this is the depth of the sea. In a shallow sea, the lateral motion would be vastly greater.

When the tidal wave (or even an ordinary wave) approaches shore, it breaks, and the velocity of its elements and "particles" its carries (such a surfers) is much greater. Surfers can achieve wild rides on tidal waves near shore, but one could not "catch" a tidal wave near its origin and surf across the Pacific. In a tidal wave, the depth of the sea is analogous to the pressure in a gas system.

Fig. 7 exhibits a second, much smaller scale, example of a multiple-ball pendulum. This example is much more nearly like that of the opening valve, because the balls may be taken to analogize molecules and/or particles as in the kinetic theory of gases. A series of ball bearings is suspended by strings from above. When ball 1, upper portion of Fig.7, is raised to position, a, and released, it falls through positions: b, c, and d, until it impacts the aligned balls 2-6. Rapidly the energy passes through the balls, and ball 6 swings along an arc through positions e, f, g, and h. Similarly, when a valve is opened, a pressure wave is produced and energy is introduced into a collection of gas molecules (the falling ball is analogous to the pressure wave). When the wave enters the downstream medium (when the falling ball impacts the row of balls/molecules), it travels at the speed of sound through

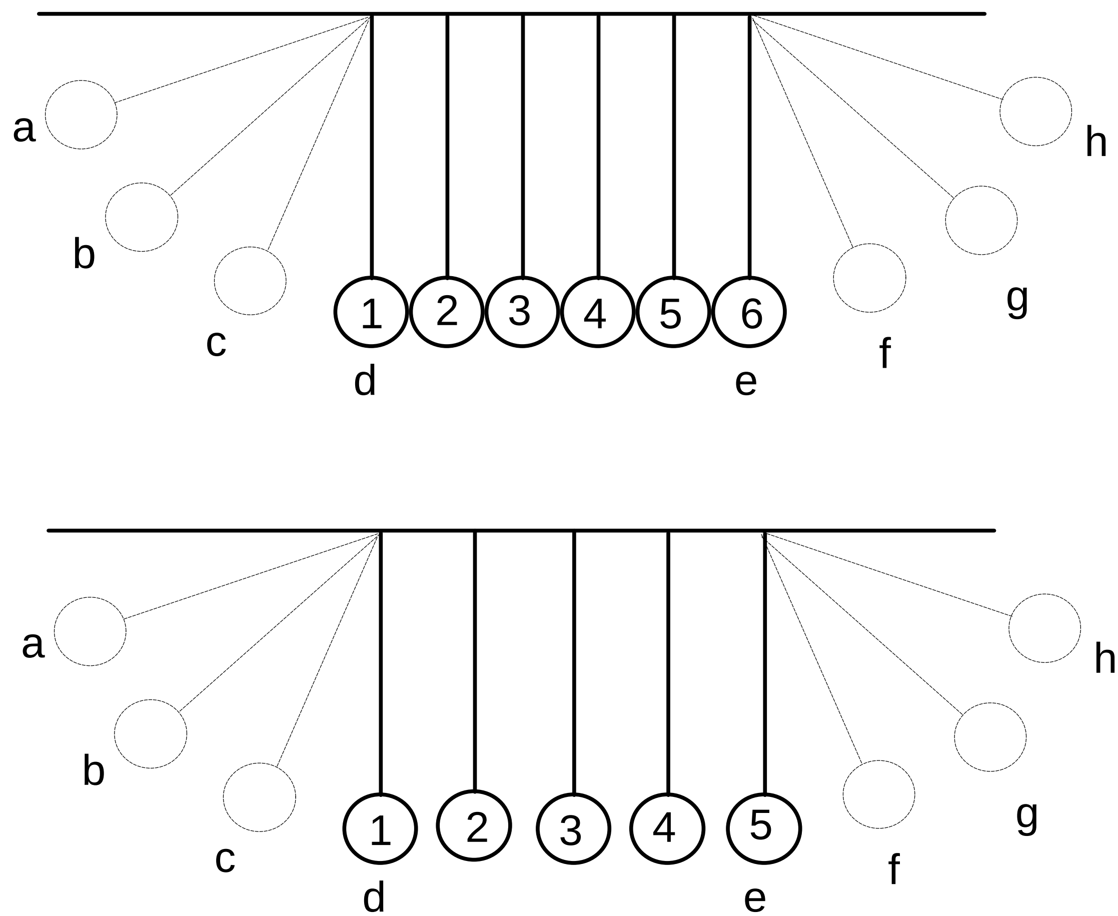

FIG. 7—*Multiple-ball pendulum.*

the gas (along the row of balls) and exits the valve (the rising ball/molecule). However, the molecules within the valve (the row of balls, one of which may be thought of as a "particle") actually move very little and achieve only a small portion of the velocity of the wave.

Notice on the lower portion of Fig. 7, that if the balls have spaces between them (corresponding to a lower pressure) the pressure wave will move more slowly through the medium, but the medium itself will move faster. In a ball pendulum, the string suspending the balls constrains the maximum size gap that can be accommodated, but if the mounting of the string at the top were allowed to move, and if there were only one ball in the device, then when the falling ball were at position d, it would move to position e at constant velocity which would be both the wave velocity and the medium velocity.

Notice however, that this latter case (one ball only) is equivalent to the case of a downstream system containing a vacuum, for which the velocity of the wave through the gas would be lower, but the velocity of the medium would higher and equal to the velocity of the wave.

This same property applies to pressure waves in gases. When a valve is initially opened, a transient pressure wave moves through the gas at the local speed of sound. If

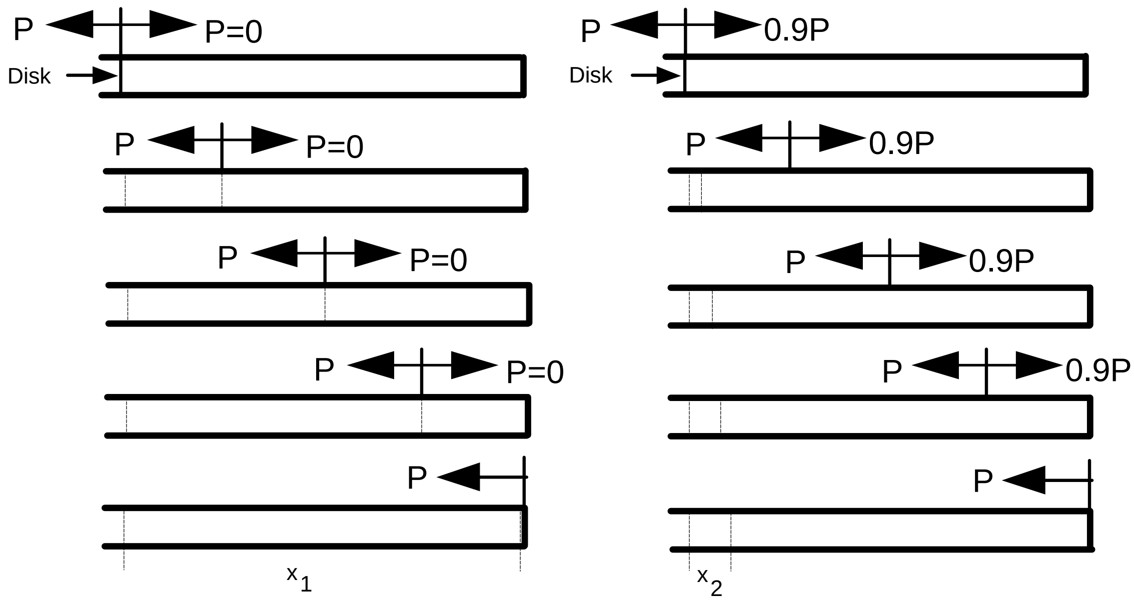

FIG. 8—*Velocity of pressure waves and medium.*

the downstream system contains a vacuum, the molecules forming the pressure wave (the medium) will also be moving at the local speed of sound. However, if the downstream pressure is close to the upstream pressure, the wave will move at the speed of sound, but the molecules of the medium will move at only a small fraction of the speed of sound. Similarly particles in the system will not be accelerated in the latter case.

This transient case may be depicted (in a simplified form) as in Fig. 8. In this case, two gas pipes connected to large-volume, large-dimension gas sources are pressurized by a frangible disk that is abruptly opened. Both pipes have a pressure, P, upstream of the disk. Downstream of the disk, one pipe is at vacuum, and one is nearly at the pressure of the upstream system, 0.9P. In both cases, the pressure wave (i.e. the boundary between the regions at pressures P and 0.9P) moves through the system at the local speed of sound. Shown above the pipes are the locations of the pressure wave at four equal intervals. Also shown (as dotted lines within the pipes) are the corresponding boundaries between the incoming gas and the downstream system (assuming no mixing occurs).

Note that in the case of a vacuum downstream, the location of the pressure wave and the incoming gas boundary are the same. The incoming gas is moving at the "local speed of sound," but note that some references indicate this may initially be more than twice the 1100 ft/s speed of sound in stagnant oxygen.

However, notice that in the case of the high initial downstream pressure, the location of the pressure wave and the incoming gas boundary (reflecting the velocity of the medium) are much different. The incoming gas has a much lower velocity. And although it is shown as moving at a constant rate would move more rapidly initially. Indeed, in some cases, the two boundaries would not reach their end points at the same time.

In the final instant, notice that the gas boundary has moved a great distance, x_1, (high gas-medium velocity) in the low downstream-pressure case, but it has moved only a short distance, x_2, in the low gas-medium velocity and high downstream-pressure case. This latter case corresponds to the case of a bypass valve opening with small differential pressure. The transient has not induced a particle impact concern.

Conditions Where the Pitot-Tube Equation May Not Apply

Note that analysis in this and the author's earlier papers thus far has treated the flow through a bypass valve as adiabatic. Most references consulted treat this as a reasonable and practical assumption to make.

However, one can make the argument that in some cases, the expansion of the gas might be isothermal. For this to occur in a system for which the gas velocities are substantial (G-4.4 addresses 25 to 200 f/s [7.6-61 M/s] depending on pressure), the interaction of the gas with the valve body must be great. That is, gas molecules must interact with the metal valve at a great relative rate. This implies an extremely small flow through the valve. Therefore, the width of the opening would have to small in comparison to the length of the opening. In other words, the bypass valve would be in a nearly closed position. In this case, the only particles that could be accelerated would be those that are on the order of the size of gap between the valve control member (e.g. plug and seat). G-4.4 does not address this scale of particle. Indeed, the practical experience base for G-4.4 is based upon industrial systems which may contain filtration to the 30-100 mesh level. Further, if such particles were to be a concern, then every leak in a steel piping system would contain velocities that were unacceptable.

This prospect has been alluded to previously in this paper, and a prior paper [*6*] presents the equation that would predict the kinetic energy gain in both the ASME analysis and at stage 3 in the hypothetical expansion engine (implying the expansion engine must heat the oxygen from its adiabatic condition to the initial temperature). From reference [*6*], the isothermal expansion (TNT equivalence) equation in vacuo is:

$$E = P_u V_u \ln [P_u / P_d] \tag{15}$$

Using this to replace the adiabatic expansion energy term in Equation (9) yields:

$$u_{kd} = u_{ku} + P_u V_u + P_u V_u \ln [P_u / P_d] - P_d V_d \tag{16}$$

Here again the upstream kinetic energy, u_{ku}, for a closed or slightly open bypass valve is zero, and very significantly, for an isothermal ideal gas model:

$$P_u V_u = P_d V_d, \tag{17}$$

Therefore, the kinetic energy downstream is the same as the TNT equivalency. Algebraic manipulation analogous to Equations (12), (13), and (14) yield velocity predictions as follows:

$$v_d = \{(2/m)\,(P_u V_u) \ln(P_u / V_u)\}^{0.5} \tag{18}$$

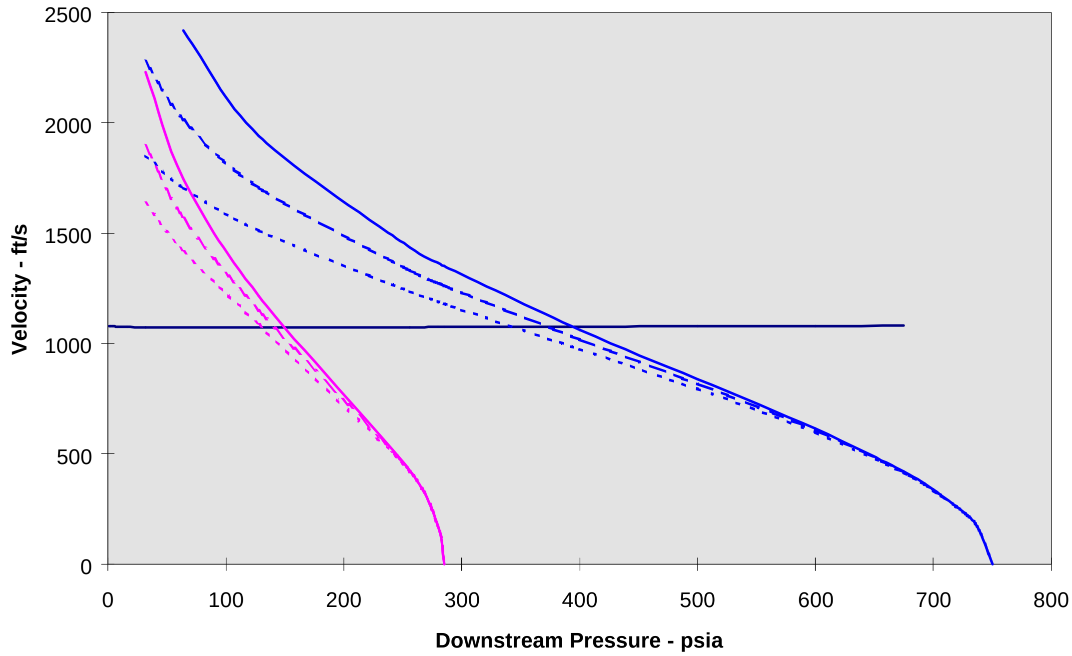

FIG. 9—*Estimates of maximum velocities achievable on energy basis, isothermal case.*

Fig. 9 exhibits the velocities this equation predicts for a two upstream pressures (dashed lines) shown falling between the curves of Fig. 5. Very interestingly, although the differences between the TNT equivalencies of adiabatic and isothermal pressure releases are typically large, the differences in predicted velocities in piping are much smaller, and the estimates of Zabrenski et al. [*3*] (top curve of Fig. 5) using equation (1) and assuming also that the downstream gas density does not change appear conservative even compared to an isothermal case. Indeed, at the velocities of interest to bypass valve use (<200f/s[<61m]), all three curves collapse to approximately the same values. Table 1 exhibits the reason for this by showing the component energies for both equations (14) and (18).

Note, in Table 1, that when the pressure drop is large (200 psig to 100 psig), the expansion energy for the isothermal case is significantly larger than for the adiabatic case (138 versus 90, a net gain of 48), but because the volume of the warmer downstream slug is greater in the isothermal case, the work done by the isothermal slug on the downstream system is also larger (-200 versus -164, a net loss of 36) and offsets much of the increase in expansion work. As a result, the velocity, which is to the square root of the kinetic energy downstream, is only about 5% greater. At the level of small pressure drops that would be associated with bypass valves, the error in maximum estimated velocity is therefore even smaller.

TABLE 1—*Energy terms for adiabatic and isothermal expansions.*[a]

P_u/P_d	Stage 2 P_uV_u	Stage 3 Expansion	Stage 5 P_dV_d	Final Energy U_{kd}
200:100	200.0	138.6 (isothermal)[b]	-200.0	138.6
200:100	200.0	89.8 (adiabatic)[c]	-164.1	127.5
200:150	200.0	57.5 (isothermal)[b]	200.0	57.5
200:150	200.0	39.5 (adiabatic)[c]	-184.2	55.3
200:190	200.0	10.3 (isothermal)[b]	-200.0	10.3
200:190	200.0	7.27 (adiabatic)[c]	-197.1	10.2

[a]For pressures in psia, assumed volume of 1.0 ft^3, energies in ft-lb.
[b]Per equation (15), isothermal case.
[c]Per equation (6), adiabatic case with $\gamma = 1.4$.

Summary

The Pitot-tube equation and several alternatives have been developed. Arguments that appear to defend use of the "Pitot-tube equation" for estimating the gas velocity in typical bypass valves during many types of typical operation have been presented. Cases where the equation does not apply and several error sources have been recognized, although the practicality of the alternates and the scale of errors identified to date are not felt to make arguments for their use compelling. Perhaps flaws will be identified with this analysis, but at present, it appears this conventional application of energy constraints to the fluid dynamics of oxygen systems may be more conservative than needed to allow carbon steel isolation valves to be used without bypass valves under suitably small differential pressure and with slow opening, and further papers on this topic and analysis are encouraged.

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