Barry L. Werley[1]

Sonic, Supersonic and Other Extreme Velocities in Oxygen Systems[2]

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ABSTRACT: A rule of thumb used in the design and understanding of oxygen systems is the condition of critical flow. During critical flow, "sonic" velocities are produced and the mass flow rate is limited (when the absolute differential pressure ratio is approximately ≥ 2:1). However, although the flow is limited, this condition does not preclude the achievement of supersonic velocities of potential concern to oxygen compatibility practitioners. Nor does it preclude sonic velocities and other disproportionate velocity levels under "subcritical" conditions in some hardware. Tutorial on the basic criteria for flow limitation and velocity development is reviewed, hopefully simplified, and related to oxygen system components.

KEY WORDS: oxygen, oxygen compatibility, flammability, shock wave, compression shock, expansion wave, pressure wave.

Flow in an oxygen system typically results when a potential-energy difference resulting from a greater pressure at one point is converted into kinetic energy in the gas. If the potential energy is sufficient, then the prospect exists that the gas can be accelerated to a velocity equal to or greater than "sonic" (the velocity at which low-level sound waves move through a system at various conditions of pressure and local temperature).

In the most basic components of piping systems (valves, orifices, diffusers, filters, etc.), the maximum "flow" through the system is limited by a conditions known as "critical" or "choked" flow. In this case, the oxygen differential pressure across a component is increased until the oxygen gas achieves local sonic velocity at the critical or choked point (in some cases, the vena contracta) and maintains that sonic velocity *at that point* regardless of further increases in the differential pressure. Indeed, once the critical

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[1]Senior Principal Project Specialist, Air Products and Chemicals, Inc. 7201 Hamilton Boulevard, Allentown, PA 18195-1501.

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point is achieved, the critical mass-flow rate will not change despite progressive decreases in the downstream pressure (but will increase if upstream pressure is raised).

Oftentimes, this condition is taken to imply that gas velocities throughout the system will not exceed sonic levels. However, supersonic velocities often obtain in regions of ordinary supercritical oxygen systems (that is, having pressure differentials equal or greater than the minimum for critical flow). More importantly, sonic and even supersonic velocities may also be present in some gas systems for which the overall differential pressure ratios are less than critical. In this paper, the conditions given in the literature for the formation of sonic and supersonic velocities are described, an attempt is made to simplify their analysis for ideal oxygen gas, and several of the equations applied in the literature are exhibited and elaborated.

What is "Sonic" Flow?

The speed of sound, or "sonic" velocity, in a gas system must always be cited with a point of reference. In reading the literature, the point of reference may often be misinterpreted. The speed of sound is given [*1*] as:

$$c = (\gamma P/\rho)^{0.5} = (\gamma RT)^{0.5} \tag{1}$$

where γ is the ratio of specific heat at constant pressure to constant volume (C_p/C_v), P is the absolute pressure, ρ is the density, R is the ideal gas constant, and T is the absolute temperature. This is called Mach one. For example, the speed of sound in atmospheric-pressure air at room temperature is 1073 ft/s. At a different temperature, the speed of sound changes significantly, and that speed is also referred to as Mach one. For an ideal gas, sonic velocity is not dependent on pressure, because density depends on pressure in an offsetting way, but remember that temperature depends on pressure during adiabatic volume changes, therefore sonic velocity has a functional indirect dependence on pressure during adiabatic events. Keep in mind that sound is taken as only relatively low intensity pressure waves. High intensity sounds, such as explosion shock waves, can move at velocities greater than the speed of "sound."

From Potential Energy to Kinetic Energy

The relation between potential energy of a pressurized gas and maximum kinetic energy of an oxygen flow are developed variously [*1-4*], and they are reviewed in an author's recent paper [*5*]. For adiabatic gas flow from a large stagnant gas source into a basic lower-pressure system (one absent converging and diverging elements, external heating or mechanical-compression equipment), the available potential energy of pressure combined with work done on and by the system would be capable of producing maximum velocities, **v**, limited to values estimated by the following equation:

$$P_T/P_d = [(v_d^2\rho_d/2g_cKP_d) + 1]^K \tag{2}$$

or,

$$\mathbf{v}_d = \left[\left(2g_c K P_d/\rho_d\right)\{[P_T/P_d]^{[(\gamma-1)/\gamma]} - 1\}\right]^{0.5} \quad (3)$$

where respectively: P_T is the total (or stagnant) 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).

The author's recent paper [*5*] on estimating the maximum velocity that might be developed between two pressure levels in a typical system, explored the several applicable energy terms. These were (1) the initial kinetic energy of the slug, (2) the work done on the gas slug by the upstream pressure, (3) the work done by the gas slug on the downstream system, and (4) the work done on the gas slug by itself by virtue of its own expansion/contraction.

These energy terms for the flow are:

The kinetic energy at the inlet = $m\mathbf{v}_u^2/2$
The work done on gas slug by the upstream system: $P_u V_u$ (max.)
The work done by slug on the downstream system: $P_d V_d$ (max.)
The adiabatic work of expansion between P_u and P_d(max.).

where V is the volume of the slug having mass, m, at pressure, P.

The last term is related in a recent paper [*6*] to the mechanical TNT equivalency of the gas in vacuum, obtained by integrating the pressure/volume relationship for adiabatic compression [*7,8*]. From that paper, the maximum for this term is:

$$E = [1/(\gamma-1)]\,[(P_u V_u) - (P_d V_d)]. \quad (4)$$

If a gas flow begins from a stagnant system, the first term for initial kinetic energy is zero and the sum of the remaining terms yields the final kinetic energy from which equations (2) and (3) are obtained. Indeed, the literature indicates that equation (3) estimates not only a limit on the maximum velocity but a good measure of the velocity that would occur as flow approaches and achieves critical conditions for many systems.

In an attempt to simplify an understanding of these events, the earlier paper [*5*] proposed viewing these events as a gas slug passing through an "expansion engine" (more correctly a "machine" since it does not produce external work) as exhibited in Figure 1.

At stage one, the slug of gas arrives at the machine bearing its initial kinetic energy. At stage two, the upstream pressure pushes the slug of gas into the machine vacuum doing work on it. At stage three, the slug expands adiabatically in vacuum doing work on itself. At stage four, the slug geometry is changed but its volume is maintained (no energy change). Finally at stage five, the slug of gas is allowed to displace gas in the downstream system (at the expense of its own energy) and enter the piping. This analysis is valid for a system starting at stagnant conditions, but is altered when velocities exceed local sonic conditions .

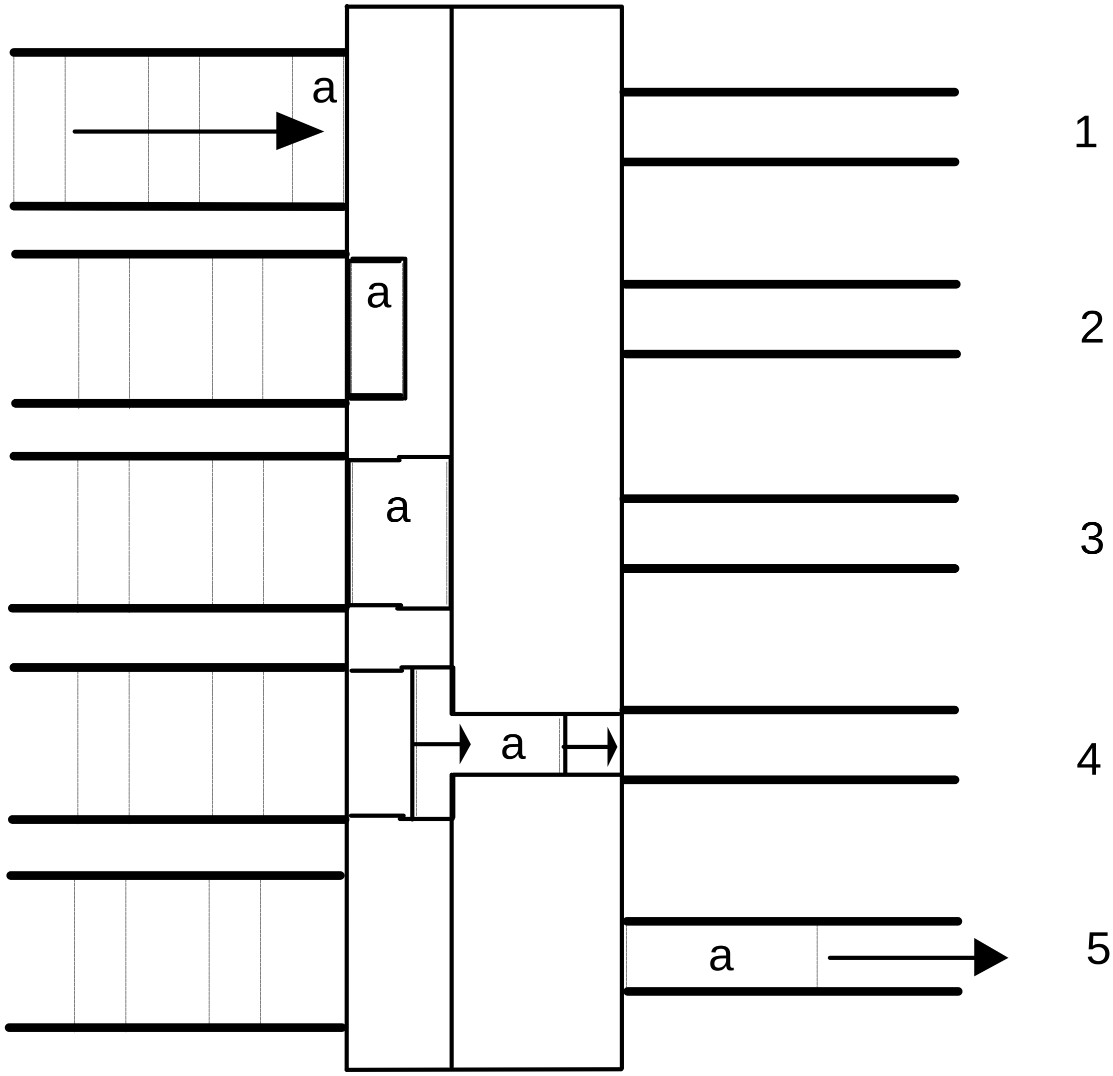

FIG. 1—*Hypothetical expansion machine.*

When the energy changes for the five stages are summed (assuming the incoming slug is stagnant) the resulting relationship yields the equations (2) and (3). The energy estimate of any flow would then be:

$$m\mathbf{v}_d^2/2 = m\mathbf{v}_u^2/2 + P_uV_u + [1/(\gamma-1)]\,[(P_uV_u) - (P_d\,V_d)] - P_dV_d \qquad (5)$$

for which $m\mathbf{v}_u^2/2$ equals zero.

Figure 2 exhibits this upper limit on velocity for various conditions of oxygen pressure

These curves are all multiples of each other and may be standardized in terms of the absolute-pressure-ratio variable, $P_d\,/P_u$, as shown in Figure 3. Figure 3 also exhibits the effect of temperature and γ (the ratio of specific heat at constant pressure to specific heat at constant volume) for an ideal gas. Values of γ were chosen that cover the principal

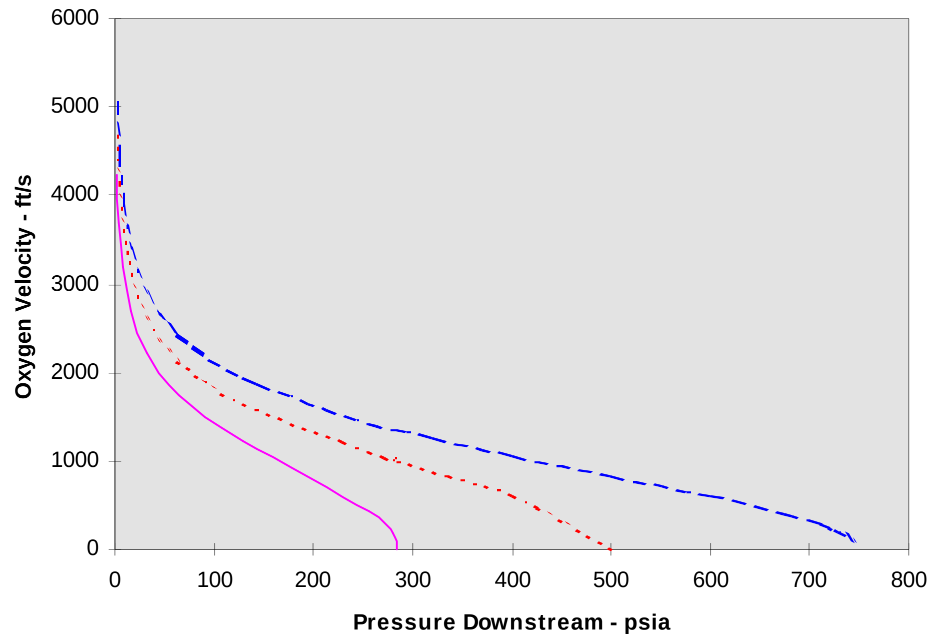

FIG. 2 ¾ *Maximum estimated gas velocities produced at various downstream pressures, basic case for Cp/Cv = 1.403 and T= 530R.*

range of interest for most gases (from unity to two). Notice that at higher values of γ, (C_p/C_v) or higher temperatures, higher maximum velocities are possible. This effect of C_p/C_v is because an adiabatic system is assumed, and the higher C_p/C_v relates to greater cooling as pressure decays, moving from right to left, along each curve, therefore reflecting increasing energy that has been converted into velocity.

These velocities are the maximum that could obtain for each case since they represent a complete conversion of the potential energy into kinetic energy without loss and include mechanical work done *on* the gas and *by* the gas. Hence they represent a condition of constant total system energy (potential plus kinetic). Real piping systems may not be so efficient. Further, although these conditions may apply for an initial system at rest, they do not apply precisely to sonic and supersonic systems.

Critical or Choked Flow

Knowing the maximum velocity of a gas flow though a component, one can calculate the maximum volume or mass-flow rate per unit area by multiplying the respective gas density and the maximum velocity:

$$[\text{mass}/\text{length}^3]\text{x}[\text{length}/\text{second}]=\text{mass}/\text{length}^2\text{second}.$$

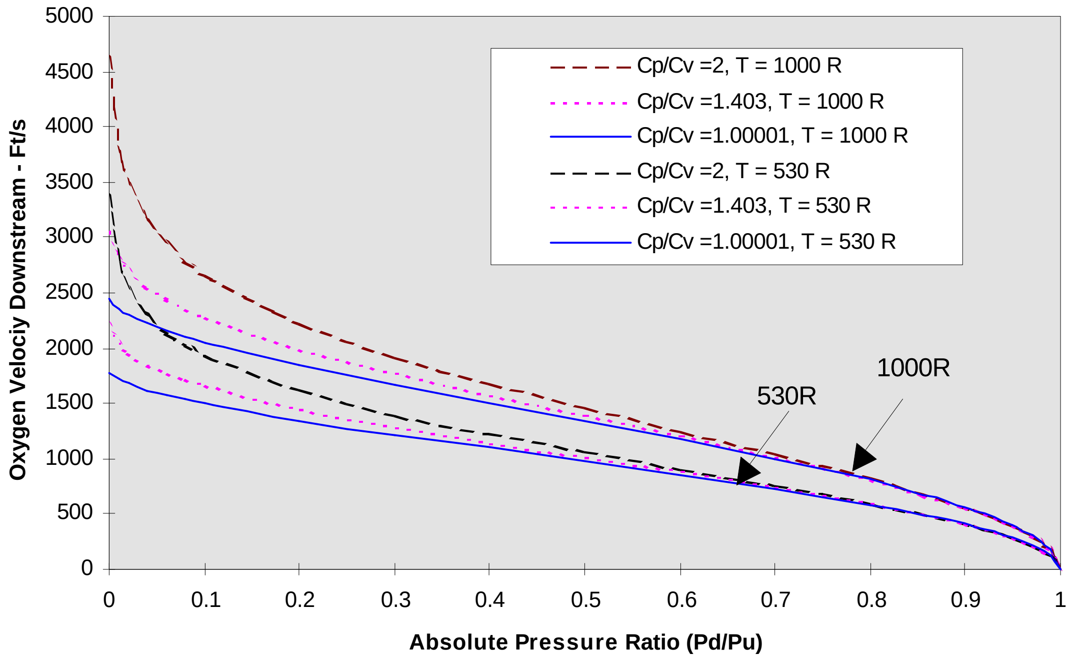

FIG. 3—Estimated g*as velocities produced at normalized pressure ratios, basic case.*

The density of an ideal gas is proportional to n/V= P/RT Therefore, the relative density of an adiabatically expanded ideal gas at any point where the pressure downstream is P_d will be given by :

$$\rho_d/\rho_u = [n/V_d]/[n/V_u] = [P_d/RT_d]/[P_u/RT_u] = [P_d/P_u][T_u/T_d] \quad (6)$$

For an adiabatic expansion, $[T_u/T_d] = [P_d/P_u]^{(1-\gamma)/\gamma}$. Therefore:

$$\rho_d/\rho_u == [P_d/P_u][T_u/T_d] = [P_d/P_u]\,[P_d/P_u]^{(1-\gamma)/\gamma} = [P_d/P_u]^{1/\gamma} \quad (7)$$

This is plotted in Figure 4 for the range of γ used in Figure 3.

The maximum mass flow possible for each pressure-ratio condition is given by the product of Equation (3) and Equation (7), which are the curves of Figure 2 and Figure 4, respectively. Since the density is zero for pressure ratios of zero (meaning the product is zero), and the velocity is zero for pressure ratios of unity (meaning the product is zero), then the product must exhibit a maximum. Figure 5 exhibits these product curves for the three values of C_p/C_v used.

To find the maximum of these maxima, one takes the derivative of this product equation and equates it to zero. Tietjens [*4*] works through the full math and provides that the maximum occurs where:

$$P_d/P_u = \left(2/(\gamma+1)\right)^{\gamma/(\gamma-1)} \quad (7)$$

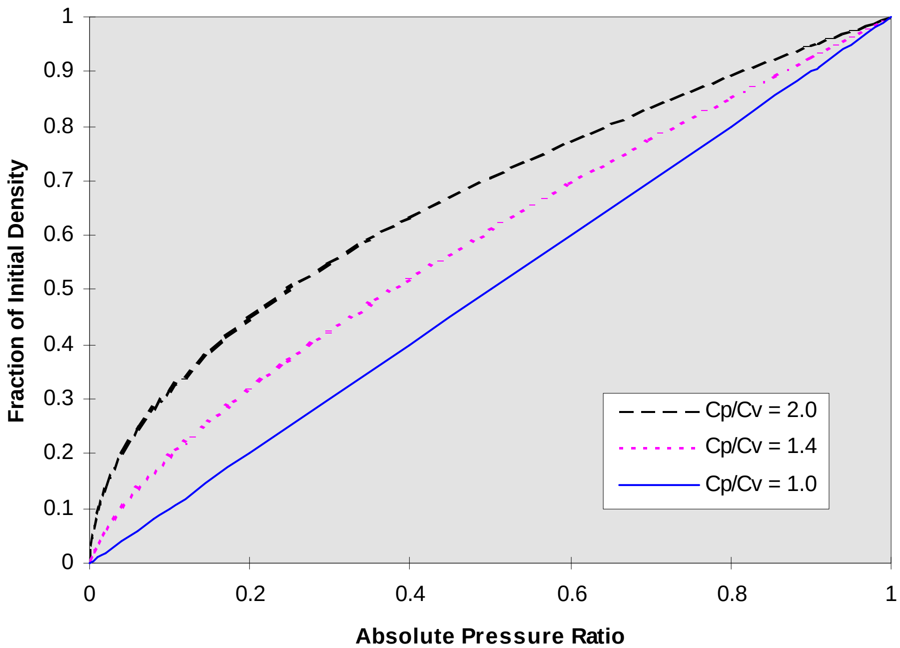

FIG. 4—*Variation in the density of adiabatically expanded gas.*

Figure 6 exhibits how this critical pressure ratio varies with γ. Note that typically, γ varies only over a range of from unity to about 2 for real gases. Notice also that the critical ratio is independent of temperature for an ideal gas, but that gamma, itself, may be a function of temperature for a real gas. The critical pressure ratio for a gas with a γ of 1.0 is ≈0.606 (a limit value), for a γ of 1.4 is 0.528, and for a γ of 2 is 0.444. Also note that if these γ are for a gases with equal initial densities, then the gas of largest γ will exhibit the highest mass flow rate possible. That is, a vessel containing this gas of larger γ will empty faster than one containing a gas of smaller γ (other things equal).

Figure 7 exhibits the maximum velocity estimates as in Figure 3 plus local sonic velocity [per Eq.(1)] for adiabatic ideal oxygen gas having an initial density of 0.0828 lb/ft^3 at STP but with three different ratios of C_p/C_v. Clearly as the pressure downstream decreases, the adiabatic gas temperature decreases and the local speed of sound follows suit. This figure illustrates that the maximum of the maxima occurs where the local sonic velocity obtains as indicated at points a, b, and c, which highlight the intersection of the local sonic velocity curves with their respective estimated gas velocity curves (that is, at the critical points).

Figure 8 exhibits (1) the velocities of sound for ideal oxygen gas having an initial density of 0.0828 lb/ft^3 and C_p/C_v of 1.403 at STP as well as (2) the maximum gas velocities, both at three initial temperatures (530, 1000 and 1500R). Notice regardless of temperature, that these local sound-velocity curves intersect their respective gas-velocity curves at the same pressure ratio (P_d/P_u = 0.528) as predicted by Eq. (7).

The proof that this condition is, indeed, sonic velocity is also provided in Tietjens [*4*] and elsewhere and is recommended reading for the oxygen-system design community.

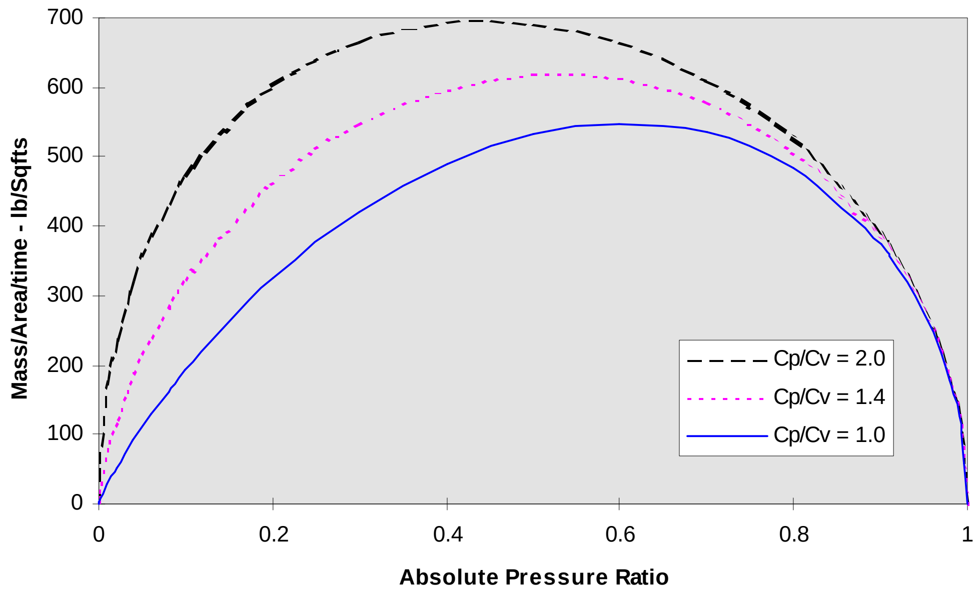

FIG. 5—*Variation in the maximum mass flow for various Cp/Cv (g).* Assuming O_2 gas having a density of 0.0828 lb/ft^3 at STP

Notice that once the critical flow condition is achieved at any point in the system, any reduction of pressure downstream does not result in an increase in either velocity or mass-flow *at the critical-flow point*. The critical point is already conveying the maximum flow possible. This feature is often exploited (very effectively) to ensure a constant mass flow rate into a experimental system. However, although this critical pressure ratio is a *sufficient* condition to ensure sonic velocity at the choke point, it is not a *necessary* condition and other circumstance for which sonic velocity may occur with smaller overall pressure drops will be covered later.

More importantly, *the limitation on sonic velocity at the choke point does not preclude greater gas velocities still farther downstream*. Indeed, velocities greater than that of sound are common in many systems, but the achievement of greatest gas velocity may be dependent upon very specific values of upstream and downstream pressures as well as specific geometries in the piping system. And supersonic velocity may be variable in the extent to which it appears. Nonetheless, particles which pass through a supercritical oxygen system when these specific combinations of pressures are present, are capable of achieving much greater velocities than at other times, and therefore, they would pose a greater risk of ignition.

Also of importance is that in certain subcritical-velocity systems, the maximum velocity between two upstream and downstream pressure points may also achieve and exceed the velocity predicted by equation (3). These particular circumstances are of interest to the oxygen-system designer.

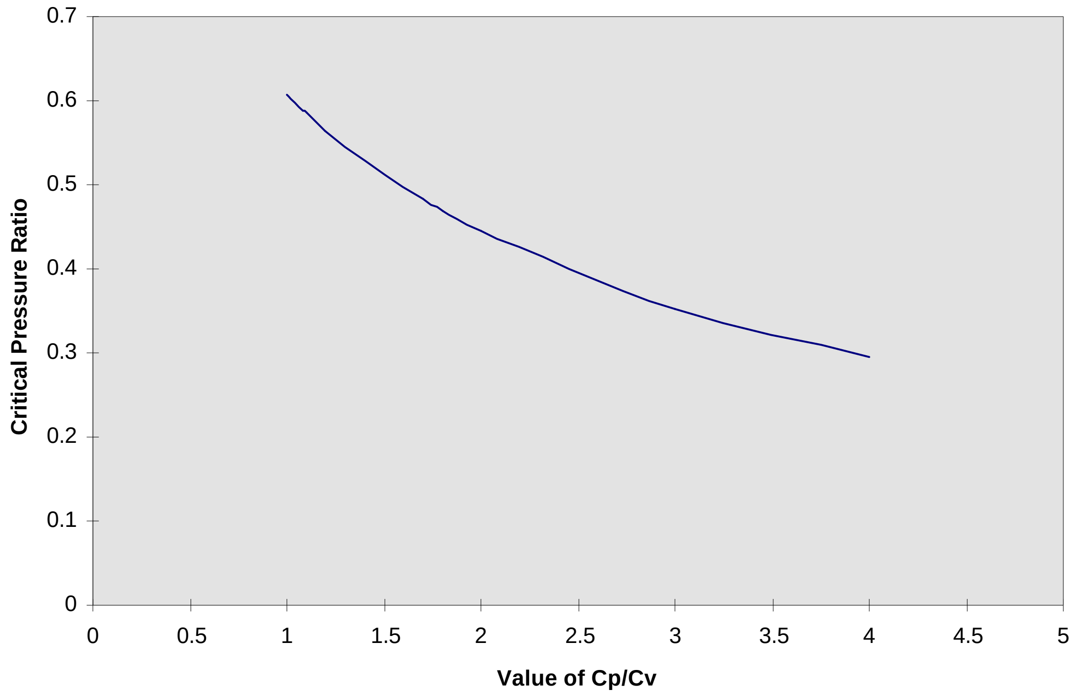

FIG. 6—*Variation in the critical absolute pressure ratio as a function of Cp/Cv (g).*

Supersonic Flow

Expanding Flow

We have seen the approach for analyzing the maximum gas velocity in a system such as at the top of Figure 9 and found that critical flow occurs at predictable ratios of P_d / P_u. Next we consider the more complex piping geometry as at the bottom of Figure 9.

When pressure downstream, P_d, produces (falls below) critical conditions (that is, becomes supercritical), the mass rate of flow (predicted by Figure 5) cannot increase at the choke point and the pressure developed there is constant regardless of further reductions in downstream pressure. However, assume the cross-sectional area of the system changes at that point. In this case, a critical or supercritical system may be viewed as two systems, using the "expansion-machine" simplification on each. One system is stagnant gas entering a first machine and exiting at sonic velocity and critical pressure, then entering a second machine at critical pressure and sonic flow and exiting at a point further downstream at a different final pressure (P_d / P_u such as in Figure 9, bottom).

In this example, the interconnecting piping downstream of the midpoint is shown diverging at a small angle. This analysis applies only for small angles of divergence, however, the maximum size of the angle will not be indicated. The maximum angle is of great interest to oxygen compatibility practitioners and bears on how far downstream the gas velocities may be a potential problem, but that analysis is very complex and is left to a different paper. For this paper the angle will simply be "small."

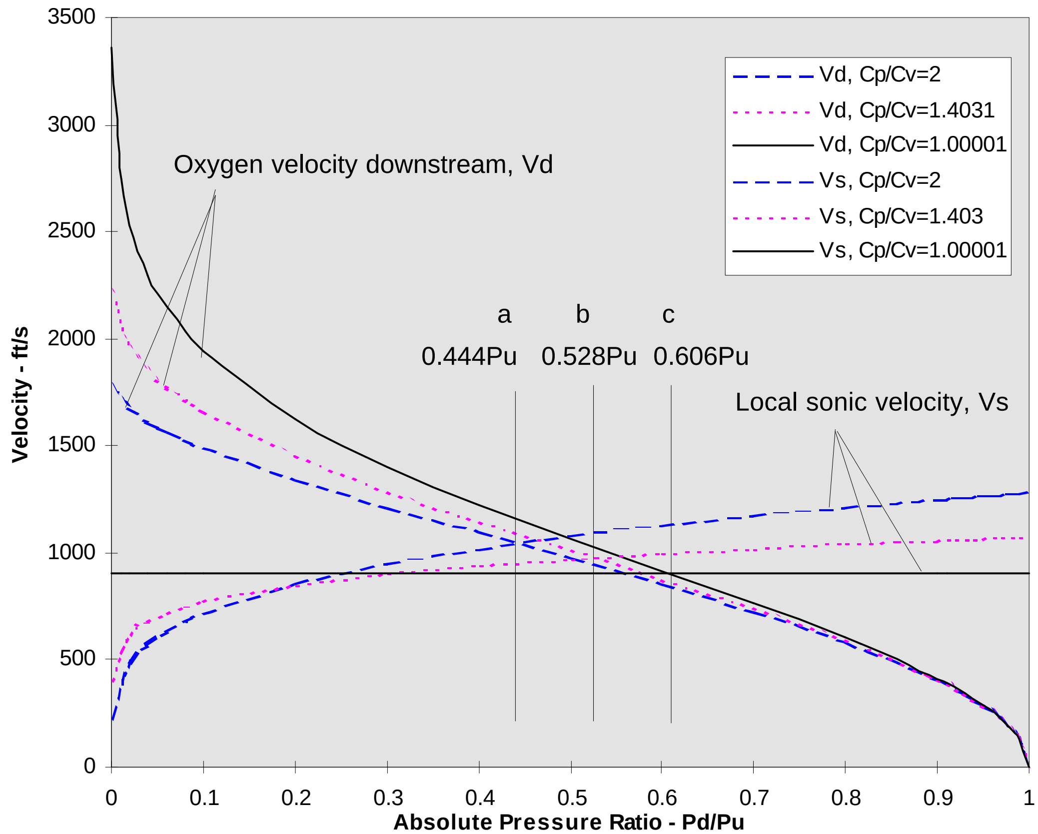

FIG. 7 ¾ *Critical flow point for various Cp/Cv.*

To the external observer, only the pressures, P_u and P_d, are measured and known. The system prior to inserting the diverging pipe is taken as critical, which will occur if the ratio of P_d/P_u is smaller than the critical ratio (but also under certain piping conditions to be reviewed later). The system will still be critical after the diverging pipe is installed, but indeed, as will be shown later, may be critical at larger P_d/P_u ratios.

Therefore, the pressure at the inlet to the second machine will be the outlet pressure of the first machine, which is the critical pressure, C_rP_u. An emerging gas slug from the first machine is moving at the local speed of sound. Therefore, small pressure changes in the upstream system are not able to "catch up" with the slug and act upon it. That is the upstream pressure is not able to push on the downstream slug. Therefore, the upstream system can no longer accomplish work on the downstream system and the P_uV_u term in the second stage of the second "machine" becomes zero.

Similarly, downstream pressure changes cannot act on any downstream slug because intervening slugs are moving at the speed of sound and so pressure pulses moving opposite to them can never traverse them. Therefore the P_dV_d term in the fifth stage of the second "machine" is also zero. The only potential source of work on the slug is from itself through expansion (represented by the TNT equivalency term in stage three of the

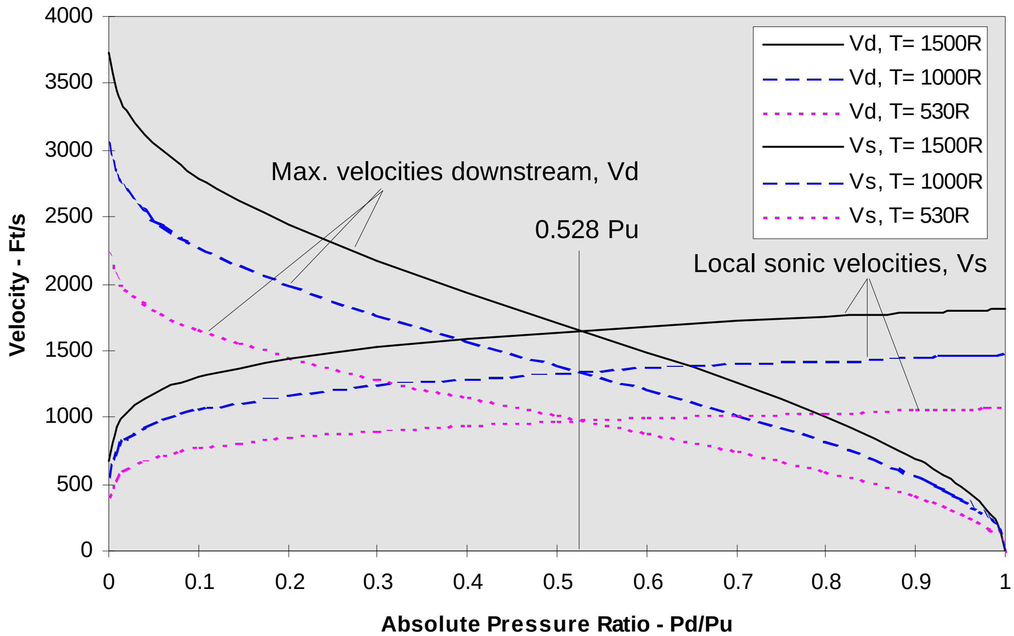

FIG. 8 ¾ *Critical Flow as the intersection of maximum gas velocity criteria with local sound velocity* (C_p/C_v *= 1.403).*

second "machine"). Therefore, the slug will tend to accelerate as a function of its own pressure, P_s, gaining kinetic energy given by:

$$E = [1/(\gamma-1)]\,[(C_r P_u V_u) - (P_s\, V_s)] \tag{8}$$

The energy of the slug (subscript, s), from which the downstream slug velocity may be calculated is given by:

$$m\mathbf{v}_s^2/2 = m\mathbf{v}_m^2/2 + [1/(\gamma-1)]\,[(C_r P_u V_u) - (P_s\, V_s)] \tag{9}$$

where $m\mathbf{v}_m^2/2$ is the midpoint energy calculated with equation (4) taking the midpoint conditions as the downstream terms.

At present, the pressure at various points downstream is not measured or known, but since it is decreasing through expansion, the velocity and mass-flow-per-unit-area of piping (the product of velocity and density) can be plotted in relation to the maximum estimates of Figure 3 and this is shown in Figure 10. Note that the velocity and mass flow per unit area are somewhat less than were predicted for a subsonic system.

However, we are assuming the flow expands to the diameter of the divergent pipe at each point (if the angle is small enough to allow it). Therefore, the volume of the slug that passes through any point is related to the pipe area, A, therefore diameter, d, of that point. Further, in stable flow, what goes in per unit time must match what comes out (continuity principle). Therefore, in any period:

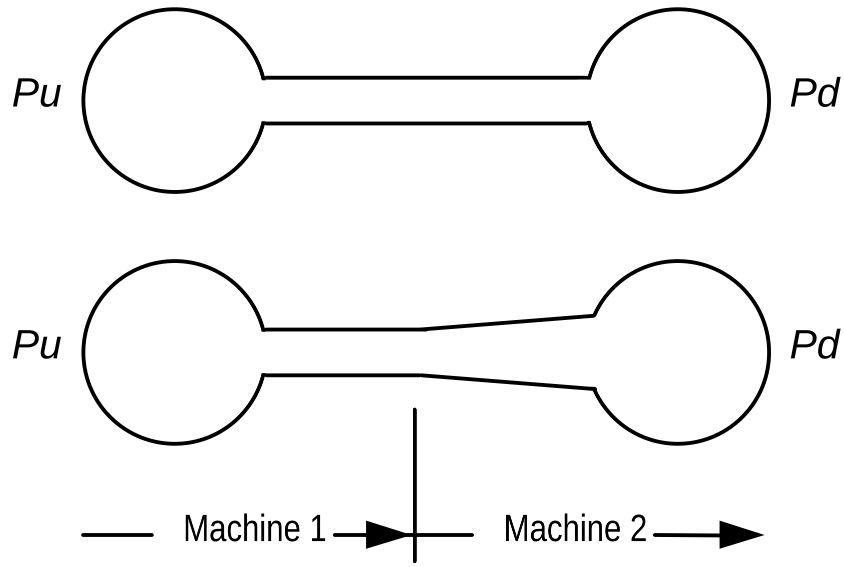

FIG. 9—*Sequential expansions*.

$$\text{mass/second} = \rho_u \mathbf{v}_u A_u = \rho_d \mathbf{v}_d A_d \qquad (10)$$

So that,

$$A_d/A_u = \rho_u \mathbf{v}_u/\rho_d \mathbf{v}_d \qquad (11)$$

This equation allows a plot of the piping area and diameter to be exhibited on Figure 10. Indeed, this relation allows for the prediction of downstream pressure during expansion to be determined as a function of position within the diverging piping (subject to other phenomena of "compression shock" to be discussed later).

This indicates that in the supersonic downstream region, pressure progressively *tends* to fall and velocity *tends* to progressively increase towards a limit. The use of the word *tends* in the preceding sentence is with great care, because the "tendency" is not always realized. William Royals noted, during the presentation of this paper, that these limit velocities can be very difficult to achieve. A phenomena referred to in the literature as boundary layer separation may prevent the gas stream from expanding with the diverging piping. Great skill and precise system geometries may be needed to realize these extreme velocities. This harkens to the dilemma oxygen compatibility practitioners face in interpreting autoignition temperatures reported in the literature. Those who seek successful combustion conduct tests for and report *minimum* temperatures that ensure ignition every time, while oxygen compatibility proactitioners are interested in maximum temperatures below which ignition will *not* occur every time. However, the key observation for oxygen compatibility practitioners is that gas velocity is allowed to increase above local sonic conditions downstream of the critical point, even if it does not exceed it to the extent of the limit values estimated here.

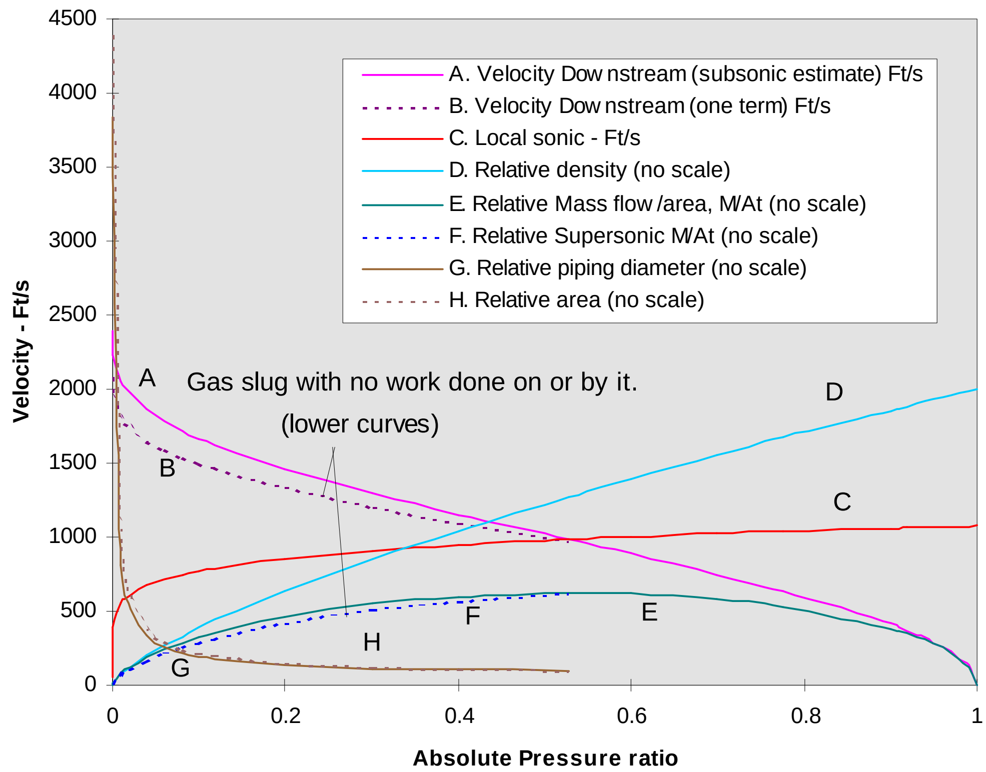

FIG. 10—*Estimated effect of divergent piping downstream of a critical flow.*

Supersonic Expansion

In Figure 9, assume a diverging pipe is abruptly increased in diameter at the point where it is connected to the downstream receiver of pressure P_d. The ambient (stagnant pressure) in this large receiver can play a role in the velocity that exists in the divergent pipe. If the ambient pressure is equal to or less than the pressure predicted at the exit by Figure 10, then the velocity profile of Figure 10 applies. The higher-pressure supersonic stream will burst into the open receiver as a "free jet." It does not immediately expand to the full diameter shape but rather expands in a small-angle "funnel" shape (very likely of larger angle than the diverging piping previously analyzed) of complex structure involving substantial eddies that dissipate energy. Figure 11 depicts a "free" jet that has illustrated this effect in textbooks for several decades.

However if the pressure at the outlet of the diverging pipe is greater than that predicted by Figure 10, then the lower-pressure stream will not emerge. In this second case, the pressure is higher at the outlet than would *tend* to be the pressure of the emergent stream. Since pressure waves flow from gases at higher pressure into gases at lower pressure, then a pressure wave will attempt to flow *against* the emergent flow despite its greater velocity.

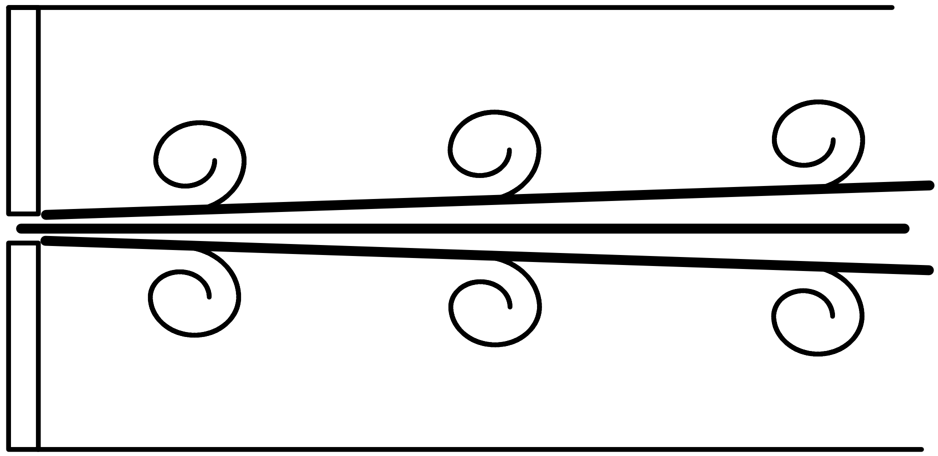

FIG. 11—Expansion of a "free" jet.

Pressure waves passing into lower pressure systems have a tendency to "sharpen" as they flow. What results is a "shock" wave: a propagating pressure front that is extremely thin, one side of which is at a much lower pressure than the other. This same effect happens when high-velocity low-density steams collide with low velocity high-density streams flowing in the opposite direction. So a shock wave attempts to propagate upstream into the diverging piping. The piping is flowing oxygen in the opposite direction, and the shock wave ultimately arrives at a location where its velocity equals the velocity of the opposing stream and it becomes a standing wave (a stationary “wall”). Oxygen flowing into the divergent piping is progressively accelerated above the speed of sound until it abruptly "hits this wall" (arrives at the shock wave), at which place it is abruptly compressed to a higher pressure adiabatically and its velocity abruptly falls below sonic. Thereafter, its character changes, and as it flows the remaining distance, its pressure thereafter *increases* and its velocity *decreases*. Estimates [*3*,*4*] of the distance across which this "compression shock" occurs are as little as 10^{-5}-in. [0.00025-mm].

Therefore, at higher downstream pressures, this supersonic velocity will not emerge from the divergent region of the piping. For lower downstream pressures, supersonic velocities will emerge and project their hazard farther downstream as a supersonic "free jet". However, these emergent supersonic jets can also experience the establishment of multiple standing shock fronts.

Convergent/Divergent (Contracting/Expanding) Flow

Convergent/divergent flow can produce extremal velocities before achieving an overall critical-flow criteria. Here again, the rate at which the flow contracts and expands (the angle of the converging/diverging piping) is important. It must again occur at a sufficiently "slow" rate (small angle) for the contracting and expanding gas to track the volume changes.

A convergent/divergent pipe is shown in Figure 12, and is known as a Venturi tube (especially when certain geometries are observed). Venturi tubes have been known of and

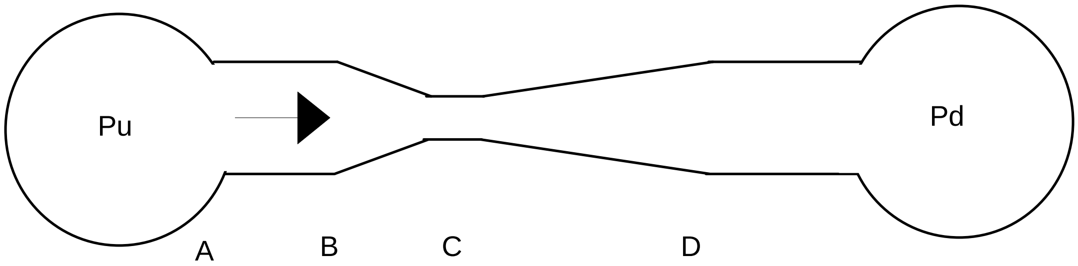

FIG. 12 ¾ *Converging/Diverging piping (Venturi).*

used since early Rome. In an 1887 paper, Herschel [*9*] surmised that they were used in early Rome to cheat on the amount of water that could be extracted from the aqueduct system.

Venturis can produce surprisingly high velocities at surprisingly small pressure differentials. Sonic and supersonic velocities can be produced in systems for which the overall upstream inlet and outlet pressures do not meet critical-flow criteria. For downstream pressures sufficiently close to the upstream pressure, the flow, m/t, that would occur in a pipe of constant diameter equal to the downstream leg at D can be calculated. The flow equations (10) and (11) apply for both the converging and diverging portions. Therefore, at the minimum-area point:

$$A_u/A_{min} = A_d/A_{min} = \rho_{min}\mathbf{v}_{min}/\rho_d\mathbf{v}_d \qquad (12)$$

Therefore:

$$\mathbf{v}_{min} = A_D\rho_D\mathbf{v}_D/(A_{min}\rho_{min}) = A_B\rho_B\mathbf{v}_B/(A_{min}\rho_{min}) \qquad (13)$$

The implications of these equations are shown on Figure 13. The points: A, B, C, D for the flow system as on Figure 12 correspond to points A, B, C, D on Figure 13. As a result, a gas slug that enters the flow system leg A-B and achieves a velocity at B, which would equal $\mathbf{v}_d$, (the ultimate velocity downstream of the venturi at point D) if the piping diameter were the same as at D. The slug accelerates as it flows through the converging leg B-C. Then it decelerates in the diverging leg C-D. Finally it achieves its ultimate downstream velocity, as predicted by Equation (3) and (4). In this case, the diameter of the piping at D is taken to be smaller than at B, yielding a higher velocity.

The previous analysis assumes that either the pressure differential between upstream and downstream are small (P_d is close to P_u) or that the ratio of minimum venturi-area-to-downstream-piping area is large. As the downstream pressure is reduced further, or as the minimum area of the venturi is decreased, the point, C, will move farther up the mass/time curve until it reaches the maximum at the local sonic point and for ideal oxygen, local pressure of 0.528 P_u. However, notice that the overall ratio of P_d/P_u in this

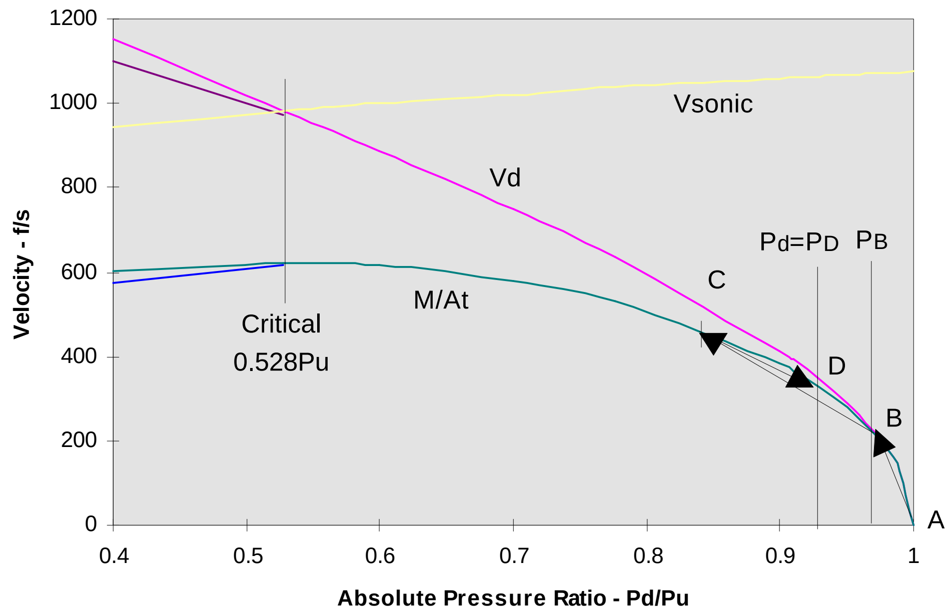

FIG. 13 ¾ *Flow through a Venturi.*

case is greater (that is of smaller pressure differential) than the critical value predicted by equation (7).

Therefore, it is possible to achieve sonic velocities by decreasing the venturi area at its narrowest point or by increasing the pressure differential. If either (or both) of these conditions is sufficient, and the flow goes "over the top," then it will tend to expand supersonically in the diverging section, as described previously. Further, if the downstream pressure is sufficiently high, then a compression shock may establish within the diverging portion. If the downstream pressure is sufficiently low, then a supersonic jet will emerge from the diverging piping. Therefore, if a venturi is sufficiently long, so that both a small angle and large area reduction are possible, sonic or supersonic velocities could be achieved with rather small pressure drops (for ideal gases and frictionless flow).

General Flow in Convergent and Divergent Piping

The final concept that will be reviewed here is for that of generic behavior of flows in convergent and divergent piping. We have already seen that diverging supersonic flow (that is "over the hump") is expanding at the expense of internal energy with no external work being done *on* it or *by* it. This yields an increasing velocity and decreasing pressure. However, when the flow entering the diverging portion of a venturi tube was subsonic and these external forces were operating, we saw that the reverse occurred: pressure increased and velocity decreased.

Similarly, we saw that when a subsonic stream which experiences and effects external work, enters the converging section of a venturi (converging pipe), its velocity increased and its pressure decreased. This begs the question of what would happen to a supersonic flow entering the same convergent piping?

In this case again, the external work *on* and *by* the flow would not be experienced. As a slug entered the converging pipe, its volume would have to decrease. That is, it would be compressed. The work of compression would have to be at the expense of its internal energy, since the external forces are not operating. Therefore, counter-intuitively to many, its pressure would increase, and its velocity would decrease. However, unlike expanding flows, which experience compression shocks (in which supersonic streams abruptly compress to produce subsonic streams), there is no analog known for expansion "shocks." Supersonic flows that enter converging piping have not been observed to experience any sudden decreases in pressure and increases in velocity. A summary of these various flow behaviors is recapitulated in Table 1.

Implications for Oxygen Systems

This paper has reviewed a number of basic fluid-mechanical principles, extracted from standard texts and simplified. To the oxygen practitioner, a number of piping situations to which this material is relevant are worth citing. In these cases, the behavior of the oxygen gas may be significantly different than anticipated on the basis of simpler reasoning.

Piping reducers/"upducers"

In many piping systems, conical transition pieces may be used to convert between assorted piping sizes (Figure 14, A). These transitions are converging or diverging sections. In many instances, the flow would be very limited and they might not act to abnormally accelerate or decelerate the oxygen. But suppose there were a rupture downstream of a divergent section. It might provide supersonic velocities that might disperse hitherto static particles. The prospects of a fire may be increased.

Bent Pipe

In this case, "bent pipe" is meant to include several different piping situations. Often large radius-of-curvature elbows will be bent (formed) on a mandrel to keep pressure loses to a minimum or to reduce piping costs. If the diameter of the piping is reduced as the pipe is being stretched on the mandrel, then the elbow will act as a converging/diverging (Venturi) tube (Figure 14, B). Velocities in the turn may be significantly increased to the point of being sonic or supersonic. The more the tubing is stretched and narrowed, the greater will be the Venturi effect. Here again, abnormal flow rates as in a upset or rupture downstream could produce extremal velocities in the elbow.

When bending tubing, a kink can develop (Figure 14, C) that can narrow the diameter greatly in one direction. Indeed if installed piping shifts, is deflected or if it expands due to temperature, elbows may be stressed and lead to a kink. This narrowing in one direction can reduce the cross-sectional area of the tubing and form a Venturi effect with all the consequential risks.

TABLE 1—*Gross behaviors of flow in piping.*

	Converging	Diverging
Subsonic	• Contracts and does progressively less work on downstream. • Pressure decreases. • Velocity increases	• Expands and must do progressively greater work on downstream. • Pressure increases. • Velocity decreases.
Supersonic	• Contracts at expense of its own kinetic energy. • Pressure increases. • Velocity decreases.	• Expands to do work on itself. • Pressure decreases. • Velocity increases

Indeed, piping that has been in use may be crushed for a variety of reasons. Accidents, hammering on piping to free ice, driving over inadequately supported piping, etc., may all produce unintended "Venturis" and unintended high velocities, especially during system upsets or failures.

Valve Inlet/Outlet Shapes

Often valve bodies are cast. This can make it relatively easy to produce complex shapes that twist a flow around to mate with downstream piping. If the shape of the conversion is one of changing area, then the gas will be accelerated or decelerated. If the casting is imperfect, it too can create an unusual modification of the flow velocities. Indeed, if debris accumulates and bonds (for example, liquid freezes to the inside of piping), it can produce a gradual narrowing, a Venturi effect, and lead to abnormal velocities (Figure 14, D).

Valve Control-Element Shapes

On a small scale, valve flow-control elements may produce extreme velocities, beyond normal expectations. Consider Figure 14, E. This exhibits a fine metering valve stem. Notice how the area of the flow path downstream of the maximum constriction slowly increases in area. Despite being an annular-shaped region, a Venturi effect is still present (dependant only on area relationships) and supersonic velocities should be anticipated in this region, *even though the valve may not be operating at overall critical pressure drops*.

Also consider the closing edge of a gate valve. Here a beveled gate creates a small amount of divergence and convergence. If operated at a small opening, this nozzle might produce local high velocities.

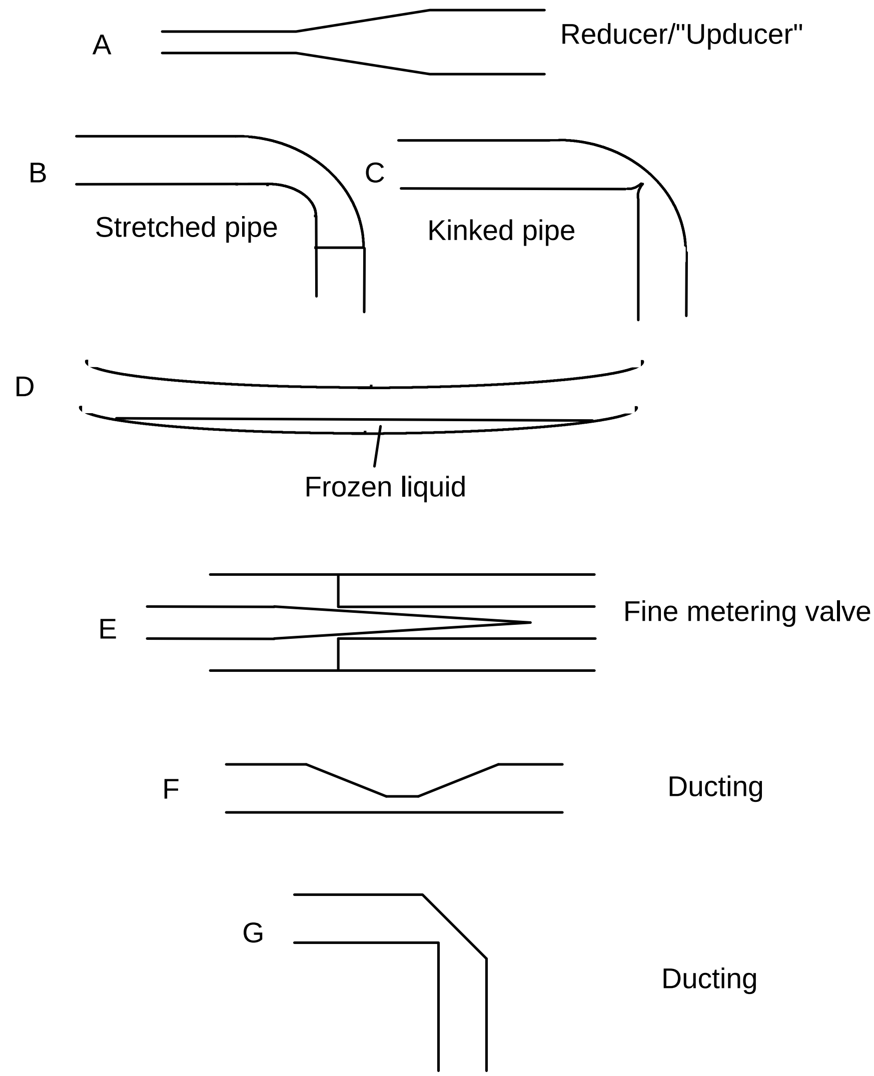

FIG. 14—Piping geometries with potential venturi effects.

Piping Constrictions

It is important to remember that the analysis of velocity production in this paper focuses on changes in cross-sectional area, combined with a gradual nature (small angles). In this regard, the system geometries of Figure 14, F and G, should all be viewed as a venturi tube much the same as if it exhibited lateral symmetry. Each of these would be easily fabricated in flat-sided ducting.

Further Work

Subsequent to presentation of this paper (but before its distribution), Koeller [*10*] prepared an analysis of the effect of real oxygen on oxygen compatibility calculations.

Oxygen is not perfectly ideal and therefore there are numerous conditions in which the analysis of this paper would be altered. An extension of this analysis to seek cases where one can simplify and explain the effect of nonideality relative to piping systems would be a benefit to the oxygen compatibility practitioner community.

Summary

A number of fluid-mechanical principles have been collected, reviewed and simplified. These principles illustrate how unexpectedly high gas velocities may form in an oxygen system. “Sonic” systems may often have regions of supersonic gas velocities within them. In some cases sonic velocities may form in systems that do not have critical pressure drops overall. In particular, events related to system upsets or failures are of interest to the oxygen compatibility practitioner, because these may lead to sonic, supersonic or other extreme velocities that increase ignition risk. Finally, wear, failure, distortion, contaminants, and other factors that can alter piping geometries, may produce elevated, sonic or even supersonic velocities where not expected.

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