Gas Pulsations: A Shock Tube Mechanism

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ABSTRACT

Gas pulsations are defined presently as a macro flow rate and/or pressure fluctuation with relatively low frequency and high amplitude. They commonly exist in HVACR, energy and other processing industries, and are widely accepted to be mainly caused by PD type gas machinery such as reciprocating or rotary compressors, expanders and Roots type blowers. Moreover, they are believed to be responsible for system vibrations, noises and fatigue failures.

Naturally, as important a matter as gas pulsations, there have been tremendous R/D efforts from both academia and industry focused in this area, especially since the late 1980s. The most well known works are acoustic models based on small perturbation and CFD methods aimed at solving nonlinear unsteady differential equations for pulsating flows. Both approaches have been successful in calculating gas pulsations at off-design conditions of either an under-compression, UC (over-expansion, OE) or over-compression, OC (under-expansion, UE). However, due to the transient nature of pulsation phenomena, some fundamental questions still remain to be answered, such as: What is the physical nature of gas pulsations? What exactly causes them to happen? Where and when are they generated? How are they different from acoustical waves and how to predict their behaviors such as amplitude, travelling direction and speed at source?

This paper attempts to answer these questions by taking a different approach: applying the classical Shock Tube Theory to gas pulsation phenomena. The results not only confirm the findings of the previous workers, but also reveal the nature of gas pulsations as a composition of strong bi-directional waves and an accompanying unidirectional through-flow. Moreover, the pressure pulsations consist of pressure waves (coalescing into a quasi-shockwave) and a fan of finite expansion waves travelling in opposite directions, while the flow pulsation is simply the induced unidirectional flow as these strong waves sweep across the gas. It will be further demonstrated that the most dominant gas pulsations are the direct results from either an OC (or UE) or an UC (or OE) suddenly discharging at the compressor or expander outlet. Therefore its location of generation, magnitude, travelling directions and speed can be predicted based on design parameters and operating conditions of those machines. Based on this new insight, an effective pulsation control method called Pulsation Trap can be devised that tackles the non-linear waves and strong induced flow simultaneously and right at the predicted sources.

1. INTRODUCTION

1.1 PD Type Gas Machinery: Application and Classification

PD (Positive Displacement) type compressors convert the shaft energy into the gas internal energy by trapping a fixed amount of gas into a cavity, then compressing and discharging it into the outlet pipe. They are capable of generating medium to high pressures for a wide range of flows. Therefore they are widely used in various applications, such as in pipeline transport of purified natural gas from production site to consumers thousands of miles away or in industrial plants for compressed air supply or in refrigeration and air conditioning equipment in refrigerant cycles. On the other hand, PD type expanders convert the gas internal energy back to the shaft energy by an opposite cycle of the compressor. They are gaining markets and popularity as a global energy conservation trend drives the need by replacing throttling devices to recover energy that otherwise would be exhausted. For some applications, the role of compressing or expanding can even be changed back and forth according to system needs.

There are a wide variety of PD compressors and expanders depending on the specific shape, movements and operating principles or applications. A common classification, as shown in Figure 1, is based on the mechanism used to move the gas by dividing them into two general types: a *rotary* type as is used in screw or scroll and a *reciprocating* type as is used in piston or diaphragm. In spite of the different drive motions and cavity forming shapes, they commonly possess a suction port, a volume changing cavity and a discharge port where a valve controls the timing of the release of gas media. Moreover, they are all cyclic in nature and have the same compression or expansion cycle for the processed gas. As an example, Figures 2a-2d show the compression cycle of a typical screw compressor. Gas flows into the compressor as the cavity on the suction side opens and traps the gas, which is then being compressed as the cavity volume is reduced. After a desired compression ratio or volume reduction ratio is reached, the discharge port opens and gas flows out of the discharge into the outlet system.

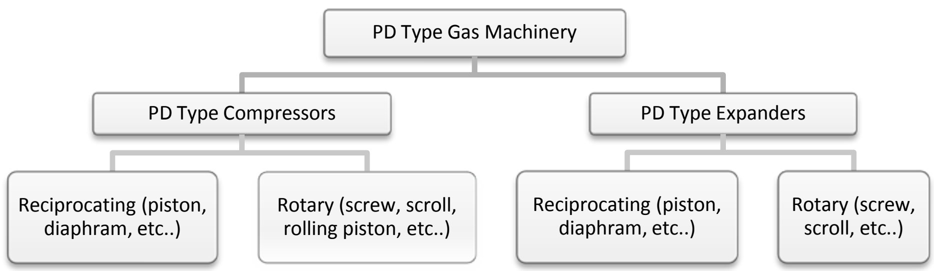
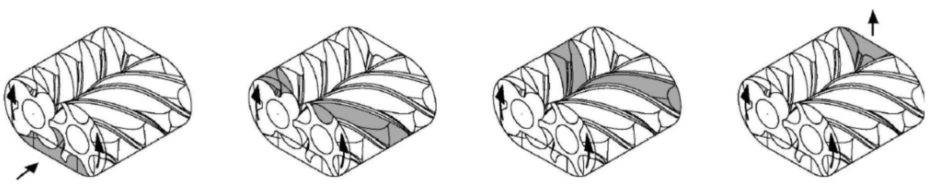


Figure 1: Classification for PD type fluid machinery



a. suction b. trapping & transfer c. compression d. discharge

Figure 2: Compression cycle of a screw

1.2 Gas Pulsations and Their Adverse Impacts

Gas pulsations are currently defined as a macro flow rate and/or pressure fluctuation with relatively low frequency and high amplitude. Since PD compressors or expanders divide the incoming gas stream mechanically into parcels of cavity size for energy transfer and delivery to the discharge, they inherently generate pulsations with cavity passing frequency at discharge, and the pulsation amplitudes are especially significant under high operating pressures or at off-design conditions of either an under-compression, UC (over-expansion, OE) or over-compression, OC (under-expansion, UE). An UC (or OE) happens when the pressure at the discharge (system back pressure) is greater than the pressure of the compressed gas within the cavity just before the opening. This would result in a rapid backflow into the cavity, a pulsed gas flow, according to the conventional theory. On the other hand, an OC (or UE) takes place when pressure at discharge is less than pressure inside the cavity, causing a rapid forward flow of the gas into the discharge system. All fixed pressure ratio compressors or expanders suffer from UC, OC or OE, UE due to the mis-match of the fixed design pressure and ever-changing system pressures and operating conditions.

An extreme case of UC is the Roots type compressor (or expander) where there is no internal compression (or expansion) at all, or a perfect 100% UC (OE). So for Roots, pulsations always exist and pulsation magnitude is directly proportional to the pressure difference between inlet and outlet. This pressure difference at discharge is believed to be responsible, at least as one of the main sources, for generating large amplitude gas pulsations in the discharge flow. They are gas borne and travel through the downstream piping system and if left uncontrolled, could potentially damage pipe line, in-line equipments, and excite severe vibrations and noises according to Price et al (1999) and Tweten et al (2008).

For this reason, PD compressors are often cited unfavorably with high pulsation induced NVH and low efficiency at off-design when compared with dynamic types like the centrifugal compressor. At the same time, the ever stringent

NVH regulations from the government and growing public awareness of the comfort level in residential and office applications have given rise to the urgent need for quieter and more efficient PD compressors.

1.3 Present Pulsation Studies and Methods

Naturally, as important a matter as gas pulsations, there have been tremendous R/D efforts from both academia and industry made in this area, especially since the late 1980s. The most well known works are acoustic models based on small perturbations and CFD methods aimed at solving nonlinear differential equations for unsteady flows.

Mujiü (2007) gives an excellent survey of the key observations since the 1980s. In summary, Koai and Soedel (1990) developed an acoustic model in which they analyzed an idealized low pulsation in a screw compressor and investigated how it was related to compressor performance. They noticed that the gas pulsations as a function of the discharge pressure have a minimum value. The results were also confirmed later by Sangfors (1999) and Wu et al (2004) using CFD modeling of gas pulsations in screw compressor discharge port. Moreover, according to Wu et al (2004), this minimum corresponds to the theoretical design point where the discharge pressure matches the compressor built-in volume ratio, while Koai and Soedel (1990) claimed that this minimum does not correspond exactly to the design pressure, and Gavric (2000) and Sangfors (1999) pointed out that it coincides with a small deviation towards under-compression. At off-design volume ratios, all the above authors reported that the pressure difference (or conditions of OC or UC) between the compressor cavity and the discharge chamber at the start of the discharge process is the most important factor in gas pulsations. Sangfors (1999) and Wu et al (2004) also observed that the amplitude of the pressure pulsations during the discharge process increases with the rotational speed.

It should be noted that the acoustic model, though partially successful as mentioned above, may over-simplify the gas pulsation phenomena by assuming that there are no mean through-flows and perturbations are relatively small in magnitude compared with mean value. On the other hand, CFD simulations that target non-linear differential equations may be too mathematically focused to reveal the physical characteristics of industrial gas pulsations. So even today, more fundamental questions remain to be answered such as: What is the true nature of gas pulsations? What exactly causes them to happen? Where and when are they generated? How to predict their behaviors such as amplitude, travelling direction and speed?

1.4 Characteristics of Industrial Pulsations and Limitation of Acoustic Theory

Gas pulsations generated by industrial gas machinery can be measured as fluctuating pressures using dynamic pressure transducer. Their magnitude is typically very high, ranging from 0.02 – 2 bar (0.3 - 30 psi) or equivalent to 160-200 dB using Equation (1). By comparison, pressure fluctuations of acoustic waves are typically less than 0.0002 bar (0.003 psi) or equivalent to the pressure level of 120 dB according to Beranek (1988). Table 1 lists the corresponding values of gas pulsations and acoustic waves in Bar and dB. It can be seen that the pressure level of gas pulsations is much higher and well beyond the pressure range of Classical Acoustics. One the other hand, industrial gas pulsations have a distinguished frequency generally corresponding to cavity passing and the pulsation wave form rises and falls much faster than the idealized harmonic waves in Acoustics. That is, the "real life" gas pulsations are non-linear waves with finite amplitude and changing wave form.

$$L_\mathrm{p} = 10\,\log_{10}\left(\frac{p^2}{{p_\mathrm{ref}}^2}\right) = 20\,\log_{10}\left(\frac{p}{p_\mathrm{ref}}\right)\ \mathrm{dB} \tag{1}$$

Table 1: Magnitude of acoustics in comparison with industrial gas pulsations

Acoustics Wave ⟺ Gas Pulsations							
Pressure Pulsation, bar	0.000002	0.00002	0.0002	0.002	0.02	0.2	2
Sound Pres. level, L_p, dB	80	100	120	140	160	180	200

The implication is challenging: the linearized wave equations that allow the superposition of solutions based on small perturbations can no longer be accurately used for industrial gas pulsations, as also concurred by Tweten et al (2008). Instead, a different approach needs to be taken which should depict a more realistic physical picture of the gas pulsation phenomena and could provide some useful design guidelines and prediction formula while avoiding solving the non-linear differential equations at the same time.

2. PD PULSATIONS AND SHOCK TUBE ANALOGY

2.1 Diaphragm and Piston Triggered Shock Tubes

The shock tube, invented in 1899 by French scientist Pierre Vieille, is a device used to study the transient aerodynamic phenomena under a wide range of temperatures and pressures for a variety of gases. It has been studied extensively by researchers from the start of the last century and has effectively served the developments of the supersonic and hypersonic flights, and a range of transient devices such as rockets, pulse-jet engines and a special supercharger called Comprex as summarized by Mueller (2006). However, the physical phenomena observed in shock tube and the well established shock tube theory have thus far not served for examining hence determining the gas pulsation mechanism of a PD type compressor or expander in general.

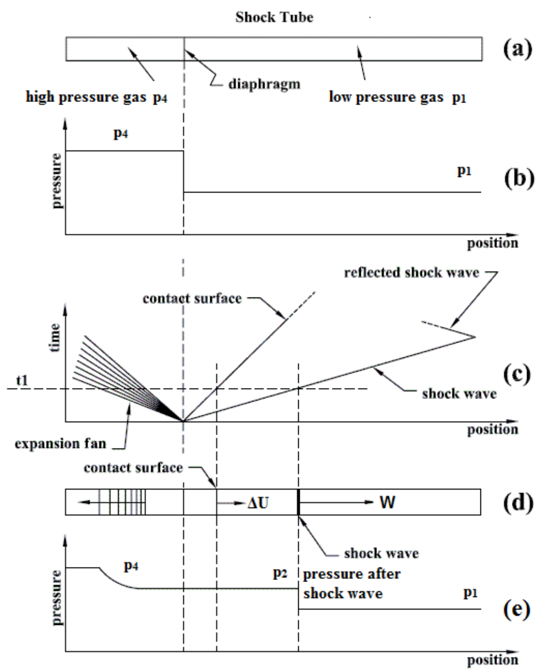


Figure 3(a-e): Diaphragm-triggered shock tube wave diagram and pressure distribution

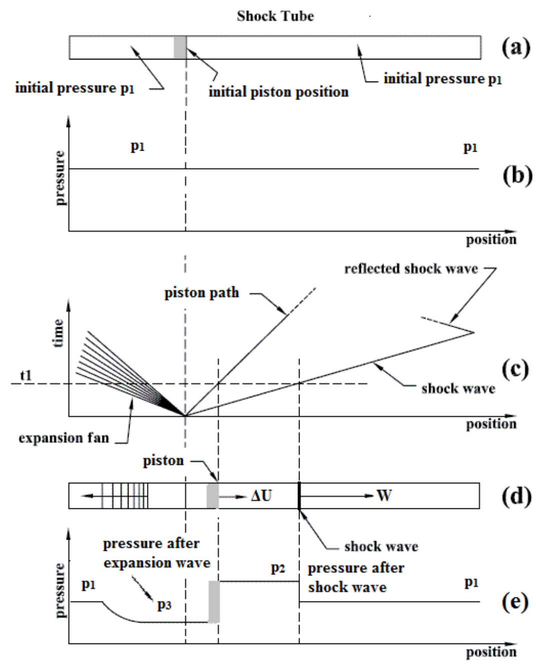


Figure 4(a-e): Piston-triggered shock tube wave diagram and pressure distribution

An ideal diaphragm-triggered shock tube as shown in Figures 3a-3b, from Anderson (1982, p180), is a tube in which gas at low and high pressure regions are separated by a diaphragm. This diaphragm then suddenly bursts open which produces a series of pressure waves, each increasing the speed of sound behind them so that they quickly coalesce into a shock wave propagating into the low pressure gas, as illustrated in x-t wave diagram and pressure distribution in Figures 3c-3e. The shock wave increases the temperature and pressure of the low pressure gas adiabatically when sweeping through at shockwave velocity W and also induces a flow in the same direction with a slower velocity ΔU. Simultaneously, a fan of rarefaction (expansion) waves travel back into the high pressure gas decreasing its pressure and temperatures. The interface separating low and high pressure gases is referred to as the contact surface which travels at the same velocity ΔU as the induced flow. Another way to generate a similar shock wave is shown in Figures 4a-4e by Anderson (1982, p204). In this case, a piston is suddenly accelerated from zero to some finite velocity ΔU producing a series of pressure waves, each increasing the speed of sound behind them so that they quickly coalesce into a shockwave propagating through the stationary gas at velocity W, just like the diaphragm shock tube. Simultaneously, a fan of rarefaction (expansion) waves travel back in the opposite direction decreasing its initial pressure P_1 and temperatures, as illustrated in Figure 4e.

The analytical solutions for shock wave and shock tube were available more than a century ago, thanks to independent efforts by W.J.M. Rankin and Pierre Henry Hugoniot, etc. The key results are re-derived and summarized by Anderson (1982) as below:

$$\frac{p_4}{p_1} = \frac{p_2}{p_1}\left[1 - \frac{(\gamma_4 - 1)(c_1/c_4)(p_2/p_1 - 1)}{\sqrt{2\gamma_1[2\gamma_1 + (\gamma_1 + 1)(p_2/p_1 - 1)]}}\right]^{-2\gamma_4/(\gamma_4 - 1)} \quad (2)$$

$$p_2 - p_1 = \rho_1 \, W \, \Delta U \quad (3)$$

Where γ is the ratio of specific heats, c is the speed of sound that is equal to $\frac{\sqrt{\gamma p}}{\rho} = \sqrt{\gamma RT}$. Equation (2), known as the Shock Tube Equation, gives the incident shock strength p_2/p_1 as an implicit function of the diaphragm ratio p_4/p_1 for known gas types. While Equation (3), known as the Rankine-Hugoniot Equation, relates the abruptly arisen pressure of the shock strength (p_2-p_1) to shockwave speed of W, initial gas density ρ_1 and induced flow velocity of ΔU. Moreover, if the gas types are the same and temperatures are within 100 C on both sides of the diaphragm, the magnitudes of the simultaneously generated pressure waves (PW) and expansion waves (EW) can be simplified for engineering applications as:

$$p_2/p_1 = p_4/p_2 = (p_4/p_1)^{1/2} \quad (4)$$

Note that the strength of PW, or (p_2-p_1), or EW in (p_4-p_2) is the maximum possible strength that could be achieved assuming zero generation time, and is independent of tube length according to Equation (2).

For the piston-triggered shock tube, the strength of the shock wave (PW) is the same as equation (3) while the magnitude for EW could be approximated by a similar equation as equation (4), but expressed differently due to a different initial pressure:

$$p_2/p_1 = p_1/p_3 = (p_2/p_3)^{1/2} \quad (5)$$

To get a sense in numbers for typical engineering applications, Table 2 lists, for known W, the calculated values of Δp_{41} (=p_4- p_1), Δp_{21} (=p_2-p_1) and ΔU (IFF) under the following conditions: for air with the same initial temperature of 300 K and pressure of 1.0 bar on both sides of the diaphragm-triggered shock tube. Please note that the ΔU value in Table 2 can also be interpreted as the velocity of a piston-triggered shock tube that generates a PW gas pulsation with magnitude of Δp_{21} (=p_2-p_1). The bottom two rows of Table 2 list the calculated pressure ratios for both PW and EW gas pulsations using the WiSTL Shock Tube Calculator. The results validate the assumptions made for Equation (4) in the pressure ranges for typical engineering applications.

Table 2: Comparison of pressure difference induced ΔP_{41} with piston movement induced ΔU gas pulsations

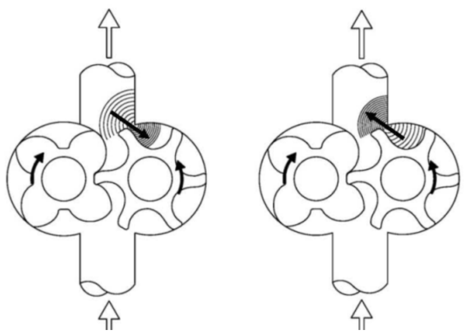
W in Mach Number	1.03	1.06	1.09	1.12	1.15	1.18	1.21
Δp_{41} (=p_4 - p_1), bar/psi	0.15/2.2	0.31/4.5	0.50/2.3	0.70/10.1	0.92/13.3	1.17/17.0	1.21/20.3
Δp_{21} (=p_2 - p_1), bar/psi	0.07/1.0	0.14/2.0	0.22/3.2	0.30/4.4	0.38/5.5	0.46/6.7	0.54/7.8
ΔU, m/sec, IFF	17.1	33.7	49.9	65.7	81.1	96.2	111
p_4/p_2, EW	1.07	1.14	1.22	1.30	1.38	1.48	1.59
P_2/p_1, PW	1.07	1.14	1.22	1.30	1.38	1.46	1.54

2.2 Analogy of Shock Tube to PD Compressor or Expander

It has long been observed that gas pulsations for PD gas machinery take place mainly at suction or discharge process and their magnitudes could be large (not the small disturbances as assumed by acoustics), especially for screw or scroll types that don't have automatic valves and operate under UC and OC conditions. Three years ago during a research project with injection cooling for rotary blower, we accidently observed that there are also strong pulsating waves propagating in opposite directions that accompany the strong pulsating flows.

To explain the newly observed phenomena while not resolving to deal with the full set of non-linear differential equations for unsteady flow, we turn to Shock Tube theory which has a close similarity with the discharge process of PD type gas machinery: both are transient in nature and driven by the same forces either from a suddenly exposed pressure difference or from a sudden movement from hardware. Using this analogy, the well established results of the Shock Tube accumulated over the past 100 years, though limited somehow by the one dimension assumption, can be readily applied to examine hence reveal the gas pulsation mechanism of a PD type compressor or expander without the need to solve any differential equations. The connecting point for the analogy is: the sudden opening of a pressure difference for a PD compressor or expander at suction or discharge is the same phenomena as what happens for a suddenly burst diaphragm or a suddenly accelerated piston inside a shock tube.

In light of this revelation, let's review the discharge process of the screw compressor shown in Figure 2 during an under-compression. As the low pressure gas first enters the cavity between lobes of a pair of rotors as they are open to inlet, gas *gradually* becomes trapped as it is transported from inlet to outlet. Then it is compressed to design pressure followed by a sudden opening to the outlet with a higher back pressure (UC) as shown in Figure 5a.



a. UC b. OC

Figure 5: Discharge process of a screw according to the shock tube theory

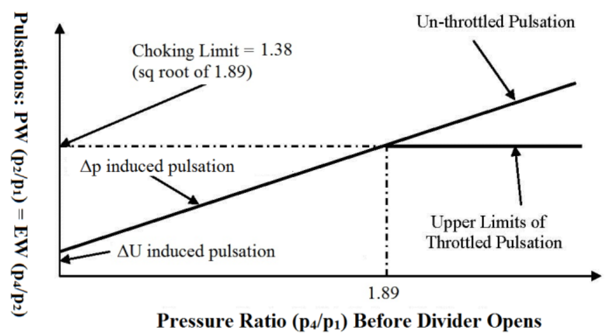


Figure 6: Gas pulsations as a function of deviated pressure ratio due to UC or OC

According to the conventional theory, a backflow would rush in due to the pressure difference at UC. However, by the shock tube theory, the lobe opening as shown in Figure 5a resembling the diaphragm bursting of a shock tube as shown in Figures 3a-3e would generate a series of pressure waves into the cavity between lobes (Note: the three dimensional geometry will slow down the pressure wave re-enforcing effect so that only a quasi-shockwave or just pressure wave is generated). The wave front sweeps through the low pressure air and compresses it at the same time at the speed of sound. Simultaneously, a fan of expansion waves is generated into higher pressure outlet. Both pressure and expansion waves induce a unidirectional fluid flow between them, or backflow in conventional theory. For the case of an OC (or UE), the wave and flow directions are reversed from UC as shown in Figure 5b.

In summary, it is concluded that gas pulsations for PD type gas machinery consist of strong bi-directional waves and an accompanying unidirectional through-flow. They are generated at places where there is a sudden opening with either the presence of a pressure difference (Δp induced) or a sudden hardware movement (ΔU induced). Since all PD compressors or expanders mechanically divide the incoming continuous gas stream into parcels of cavity size for delivery to the discharge, it inherently generates pulsations with valve opening frequency at the moment of valve opening when there is a pressure difference at the valve. As for ΔU induced pulsations, it will be discussed below that their magnitudes are typically much smaller than Δp induced pulsations even though there are accelerated lobe or piston movements during suction and discharging phases inside PD gas machinery.

2.3 Comparison of ΔP Induced Pulsation with ΔU Induced Pulsation

The above results show that two types of sources of gas pulsations can be correlated to two types of the shock tubes. The first type is Δp induced simulating a suddenly burst diaphragm and the second type is ΔU induced like a suddenly accelerated piston. Let's compare their magnitudes to decide which type is the more dominant mode inside PD compressors or expanders.

Again refers to Table 2, it can be seen that pressure difference Δp_{41} (=2.2 psi=0.15 bar) would generate a suddenly induced flow velocity of ΔU (=17.1 m/s). On the other hand, we can also say that a sudden velocity change ΔU (=17.1 m/s) from hardware would generate a PW pulsation Δp_{21} (=1.0 psi=0.07 bar). In reality, it is common to see the discharge Δp range from a few psi up to hundreds of psi at off-design conditions but the lobe or piston velocity change (acceleration or deceleration) are usually well below 17 m/s. For that reason, Table 2 can also be used to determine quantitatively whether velocity change is high enough as a major source or not based on known kinematics.

Hence, it is concluded that Δp induced pulsations are dominant pulsations for gas machinery while ΔU induced pulsations are much smaller in magnitude. Moreover, the maximum Δp induced pulsations take place at discharge under all off-design conditions and its magnitude is directly proportional to UC (OE) or OC (UE). On the other hand, ΔU induced pulsation could happen under both design and off-design conditions and everywhere inside PD gas machinery due to constant interaction between gas and hardware, and its magnitude is proportional to the velocity change of the hardware (valve, piston or lobe). In other words, ΔP pulsations are related to unbalanced aerodynamic conditions while ΔU pulsations are the results of unbalanced kinematics.

Figure 6 plots both Δp and ΔU induced pulsations in terms of pressure ratio, for the sake of convenience, as a function of deviated pressure ratio at discharge in logarithm scale so that the relationship is linear. It is most suited for rotary type gas machinery such as a screw or scroll where no valves used at inlet and outlet and the dominant pulsation mode is Δp induced while ΔU induced pulsation is secondary.

2.4 Pulsation Rules for PD Compressor or Expander

For the convenience of effectively using the above results for industrial applications, the following pulsation rules are summarized as a simplified way to determine: where is the source of gas pulsation and when does it happen, what are the sufficient conditions for its generation, how to predict quantitatively its magnitude and travel direction at source. In principle, these rules are applicable to different gases and for gas pulsations generated by any industrial PD type gas machinery or devices such as engines, expanders, or pressure compressors, vacuum pumps and valves.

1. Rule I: For two divided compartments (either moving or stationery) with different gas pressures p_4 and p_1, there will be no or little gas pulsations generated if the two compartments stay divided;

2. Rule II: If the divider between the high pressure gas p_4 and the low pressure gas p_1 is suddenly removed, gas pulsations are generated at the location and moment of the opening as a composition of a fan of Pressure Waves (PW) or a quasi-shock wave, a fan of Expansion Waves (EW) and an Induced Fluid Flow (IFF) with magnitudes as follows:

$$PW = p_2 - p_1 \tag{6}$$

$$EW = p_4 - p_2 \tag{7}$$

$$\Delta U = (p_2 - p_1)/(\rho_1 \times W) \tag{8}$$

   Where ρ_1 is the gas density at low pressure region, W the speed of the lead pressure wave, ΔU the velocity of Induced Fluid Flow (IFF);

3. Rule III: Pulsation PW is the action by the high pressure gas p_4 to the low pressure gas p_1 while pulsation EW is the reaction by the low pressure gas p_1 to the high pressure gas p_4 in the opposite direction, and their magnitudes are such that they equally divide the initial pressure ratio p_4/p_1 [equation (4): $p_2/p_1 = p_4/p_2 = (p_4/p_1)^{1/2}$]. At the same time, both PW and EW pulsations induce a unidirectional fluid flow pulsation IFF in the same direction as the PW.

Rule I implies that there would be no or little pulsations during suction (if there is no inlet valve), transfer and compression (expansion) phases of a PD cycle because of the absence of either a pressure difference or sudden opening. The focus instead should be placed upon the discharge phase, especially at the moment of the discharge when divider is suddenly opened and during off-design conditions like UC (OE) or OC (UE).

Rule II indicates specifically the ***moment*** of pulsation generation as the instant the divider separating p_4 and p_1 opens and the ***location*** as the divider. Moreover, it defines ***two sufficient conditions*** for gas pulsation generation:

a) The existence of a pressure difference Δp_{41};
b) The sudden opening of the divider separating that pressure difference.

Because all PD gas machinery converts energy between shaft and fluid by dividing incoming continuous fluid stream into parcels of cavity size and then discharging each cavity separately at the end of each cycle, there always exists a "sudden" opening at discharge phase to return the discrete parcels back to a continuous stream again. So both sufficient conditions are satisfied at the moment of discharge opening if there exists a pressure difference between the cavity and outlet it is opened to. For compressors and expanders operating at off-design points with a fixed volume ratio, this pressure difference is either an over-compression (under-expansion) or under-compression (over-expansion).

Rule II also reveals the composition and magnitudes of gas pulsations as a combination of large amplitude Pressure Waves (PW) or a quasi-shockwave, a fan of Expansion Waves (EW) and an Induced Fluid Flow (ΔU). These waves are non-linear waves with changing wave form during propagation. This is in direct contrast to the acoustic waves that are linear and wave fronts stay the same and do not induce a mean through flow. It is interesting to note the wholeness of three different pulsations (PW, EW and IFF) that are generated simultaneously and one cannot be produced without the others. This makes gas pulsations very difficult to control because it's not one but all three effects have to be dealt with. To calculate the theoretical magnitudes of gas pulsations PW, EW and IFF, an on-line Shock Tube Calculator (from WiSTL Gas Dynamics Lab) can be used if initial conditions and gas types are known.

Rule III shows further the interactions between two gases of different pressures are mutual so that for every PW pulsation, there is always an equal but opposite EW pulsation in terms of pressure ratio ($p_2/p_1 = p_4/p_2$). Together, they induce a unidirectional fluid flow pulsation (IFF) in the same direction as the pressure waves (PW).

2.5 Corrections for Three Dimensional and Finite Opening Time Effects

The Pulsation Rules assume a one dimensional tube for PW, EW and IFF to generate and propagate, so it only predicts pulsations at source before it has any interaction with the piping system. Moreover, it is assumed that the time it takes either for diaphragm opening or piston accelerating is infinitely small, and the tube length is relatively long so that pressure waves have time to coalesce into a real shock wave. But reality inside PD type gas machinery is quite different: the source is a gradually (finite time) enlarging opening from zero to finite area and waves and induced flow propagate into 3-D spaces on both sides of the source. This is especially true on cavity side due to the short length, so the generated pressure waves will probably remain as pressure waves or approximated by a quasi-shockwave. Therefore, certain correction factors are needed to modify each component of the maximum possible pulsations for PW, EW and IFF in Pulsation Rule II as follows:

$$F = F_t \times F_s \tag{9}$$

Where F_t is a time correction factor to modify the effect of finite opening speed of discharge process while F_s is a space correction factor taking account the 3D effect which in general will disperse the intensity of pulsations from the source. The combined correction factor is F (=$F_t \times F_s$) that can be obtained from testing and applied to equations (6), (7) and (8) for PW, EW and IFF respectively for specific geometry and machine types. It should be noted that the commonly measured pressure pulsations are not necessarily the true source pulsations since pressure measurement taps are always some distance away from the pulsation source: usually at valve opening or a throat.

The fact that discharge area is opened from zero (closed) to finite indicates that a throttling or choking effect of the induced fluid flow could take place at source, limiting the maximum IFF to sonic velocity at throat. This corresponds to the upper limit of the maximum pressure ratio for generating pulsations to 1.89 according to Anderson (1982, p132). For most compressors and expanders, the deviated pressure ratios at discharge due to UC (OE) or OC (UE) are typically well below 1.89. But the Roots blowers and vacuum pumps are an exception where pressure ratios can range from 1 to 10. The choking effect is reflected in Figure 6 as an upper limit of pulsations for the throttled opening. Figure 6 also shows a small pulsation component due to the effect of a suddenly moved hardware (like lobe, valve disk) with velocity impulse of ΔU that co-exists with ΔP induced pulsations.

3. EXAMPLES USING PULSATION RULES FOR PD GAS MACHINERY

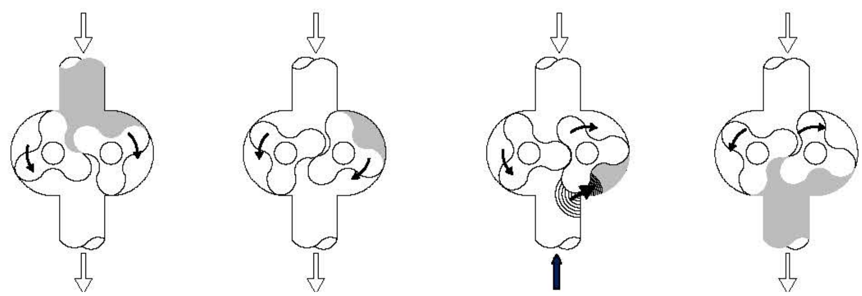
3.1 Explanation of Valve Induced Pulsations from a Screw, Scroll or a Recip

Valves generate pulsations because of the existence of a pressure difference and a sudden opening. There are generally two types of valves, so called *gate* type as is used in screw, scroll and certain type of rotary vane compressors, and *automatic* type as is used in piston type compressors. Gate valve opens or close at a pre-determined interval (by volume ratio) that is independent of pressures difference across the valve, therefore it generates conditions of UC or OC. On the other hand, automatic valve like floater type, spring-loaded reed or plate valve only opens passively on demand by pressure difference across it when exceeding the force of the spring.

Applying the Pulsation Rules to screw or scroll type compressors with gate valves, strong pulsations are expected to be generated whenever there is a large pressure difference (Δp_{41} is large) as in UC or OC when discharge valve opens suddenly by high rotational speeds, while relatively little pulsations would exist at inlet side due to slow opening process and little pressure difference across the valve. At design condition when compressor discharge pressure is exactly equal to system pressure (Δp_{41}=0), pulsation is little at discharge side even though valve still opens suddenly. As for piston or diaphragm type using automatic valves, it can be approximated by Δp induced pulsations because the valve movement is driven primarily by Δp_{41} across it and is known. In this case, both inlet and outlet side can have pulsations due to the presence and sudden opening of the valves, but their magnitudes are relatively small since the Δp_{41} to overcome the spring inertial effects and spring opening force is typically small. More specifically, the inlet side pipe should see expansion waves and outlet pipe compression waves.

3.2 Explanation of Observations of Roots Blowers

Applying the above Pulsations Rules to Roots blower, it can be predicted that suction and transfer phases should have little pulsations because of the absence of any pressure difference. But during discharge phase, there always exist strong pulsations because of the 100% under compression. It can be further anticipated that the larger the pressure rise from inlet to outlet, the higher Δp pulsations induced according to Rule II. Moreover, the timing of pulsation generation takes place at the instant as the lobe suddenly opens to outlet and the location is at the opening area between lobe and casing. The resulting gas pulsations are a composition of pressure waves and induced flow going into the cavity from the opening and a fan of expansions waves travelling in opposite direction into a discharge silencer downstream as shown in Figure 7.



a. suction b. transfer c. compression d. discharge

Figure 7: Compression cycle of a classical Roots according to the shock tube theory

3.3 Explanation of Other Authors'' Observations from Screw and Scroll Compressors

Since screw compressor has a built-in pressure ratio, there would be no pressure difference induced pulsations when exactly operating at this pressure ratio (design condition) according to Rule II. This means that Pulsation Rules can predict the minimum discharge gas pulsations to take place at design point, which agrees with the observations by Koai and Soedel (1990), Sangfors (1999), and Wu et al (2004). Moreover, the Rules would also predict less dominant pulsations from lobe movement, which have its effects felt at the moment of lobe opening to outlet pressure. Because gas cavity volume is reducing at this instant, it sends out a faster travelling flow with a small dynamic pressure (compression waves) that can only be cancelled out by a higher outlet system pressure at under-compression. This could explain observation made by Gavric (2000) and Sangfors (1999) that minimum pulsation does not correspond exactly to the design pressure but coincides with a small deviation towards under-compression. Moreover, the RPM effect as reported by Wu et al (2004) can be interpreted as affecting discharge opening time or the value of time correcting factor F_t so that a faster RPM corresponds to a shorter discharge opening time that makes pulsations more impulsive (stronger) than a slowly opened discharge.

4. CONCLUSIONS

The application of the classical Shock Tube Theory to the transient process of PD type gas machinery reveals several interesting results. It is shown that gas pulsations are not equivalent to small disturbances described by linear acoustic waves, but are a composition of strong bi-directional waves and an accompanying unidirectional through-flow. Moreover, the pressure pulsations consist of strong pressure waves or a quasi-shockwave and a fan of finite expansion waves travelling in opposite directions, while the flow pulsation is simply the induced unidirectional flow as these strong waves sweep across the gas at the speed of waves. It is further demonstrated that the sources of the most dominant gas pulsations are either an OC, over-compression (or UE, under-expansion) or an UC, under-compression (or OE, over-expansion) suddenly discharging at the compressor or expander outlet. Therefore its location of generation, magnitude, travelling direction and speed are predictable based on design parameters and operating conditions of those machines. Pulsation Rules are inducted to predict the source and characteristics for industrial gas pulsations, which are also validated by existing observations.

Finally, we hope, in lights from the above revelations and discussed pulsation rules, experiments can confirm the existence of pressure waves (or a quasi-shockwave) and expansion waves at discharging process of PD gas machinery. Better CFD solvers can be designed and built for more accurate prediction. Most importantly, a more effective control method called Pulsation Trap (The subject of a sequel paper) can be devised that tackles the non-linear waves and strong induced flow simultaneously and right at the predicted sources by Pulsation Rules. With more research and resources devoted by both academia and industry, it is anticipated that future generations of PD gas machinery can be designed to be simpler in structure, smaller in size and smoother in running.

NOMENCLATURE

- C speed of sound
- F correction factor for Pulsation strength
- L_p sound pressure level
- M Mach number
- p static gas pressure
- R gas constant
- T static temperature
- ΔU piston or contact surface velocity
- W shockwave velocity
- ρ static density
- γ ratio of specific heats

Subscripts

- 1 initial low pressure
- 2 pressure after shockwave
- 3 pressure after expansion wave
- 4 initial high pressure
- s space correction factor
- t time correction factor

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