The origin of gas pulsations in rotary PD compressors: ΔP or ΔU?

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ABSTRACT

This paper is a continuing work from the same authors on the same topic by analogizing the transient shock tube process to the gas pulsation phenomena [1, 2]. It has been demonstrated in the previous theoretical investigation that gas pulsations can be generated either by a sudden velocity change (ΔU) say from a piston or lobe movement, or by the sudden opening of a pressure difference (Δp_{41}) such as during the discharge phase of a rotary PD (Positive Displacement) compressor (screw, scroll or Roots) under UC (Under-Compression) or OC (Over-Compression) conditions. It is further reasoned that the more dominant pulsation source should be from the Δp_{41} induced source, but without backing from experiments.

This paper is aimed at validating the theoretical predictions through experimental investigation conducted on a Roots blower, a special 100% UC case. This further clarification is important not only to prove the theory, but would pave the way for potential gas pulsation control methods if exact origin and nature of gas pulsations can be located and understood. Two schemes of tests under different load and speed conditions were carried out and results confirm the previous analytical conclusion that the dominant source of the gas pulsations for the 100% UC case is mainly from the sudden release of a pressure difference (Δp_{41}) while the non-uniform rotor movement (ΔU) induced pulsation is about one order of magnitude lower. More experimental work on other types of rotary compressors such as screw or scroll is desirable in the future.

NOMENCLATURE

CW	compression waves pulsation component in PD compressor, shock wave in shock tube
EW	expansion waves pulsation component in PD compressor and shock tube
IFF	induced fluid flow pulsation component
OC	over compression
P, p	absolute gas pressure
T	absolute temperature
UC	under compression
ΔU	piston or contact surface velocity in shock tube, IFF velocity in PD compressor
W	shockwave velocity in shock tube, CW velocity in PD compressor
ρ	gas density

Subscripts

1	initial low pressure in shock tube, or PD compressor inlet, in cavity
2	pressure after shock wave in shock tube
4	initial high pressure in shock tube or at compressor outlet or jet port
jet	Roots blower jet port
inlet	Roots blower inlet port
outlet	Roots blower outlet port

1. INTRODUCTION

Gas pulsations are a group of phenomena characterized by the simultaneous flow rate and pressure fluctuations with relatively low frequency and high amplitude. They commonly exist in HVACR, energy and other processing industries, and are widely accepted to be inherently associated with PD type gas machinery such as reciprocating or rotary compressors, expanders and Roots type blowers. Moreover, they are believed to be one of the primary sources responsible for exciting or inducing system vibrations, noises and fatigue failures.

The conventional methods of modeling gas pulsations are mostly acoustic models based on small perturbation method or using various CFD schemes aimed at solving nonlinear differential equations for unsteady flows. However, as demonstrated in Table 1 [1], the magnitude of gas pulsations generated by industrial gas machinery is typically ranging from 0.02 – 2 bar (0.3 - 30 psi) or equivalent to 160-200 dB. By comparison, the pressure fluctuations of acoustic waves are typically less than 0.0002 bar (0.003 psi) or equivalent to the pressure level of 120 dB as the limit for using the linear acoustic equations according to Beranek [3].

Table 1: Magnitude of acoustic waves in comparison with industrial gas pulsations from Huang [1]

Acoustic Waves ⟺ Gas Pulsations							
Pressure Pulsation, bar	0.000002	0.00002	0.0002	0.002	0.02	0.2	2
Pressure Pulsation, psi	0.00003	0.0003	0.003	0.03	0.3	3	30
Sound Pres. level, L_p, dB	80	100	120	140	160	180	200

In order to get a glimpse of a more realistic physical picture of the gas pulsation phenomena while avoiding solving the full set of non-linear differential equations, the present author has proposed an alternative shock tube theory [1, 2] by analogizing the transient shock tube process to the gas pulsation phenomena as occurred typically in the discharge phase of rotary PD compressors such as screw, scroll or Roots. It has been demonstrated that gas pulsations can be generated either by a sudden velocity change (ΔU) say from a lobe or valve movement, or by the sudden opening of a pressure difference ($\Delta p_{41} = p_4 - p_1$) such as during the discharge phase of a rotary compressor (screw, scroll or Roots) under an UC (Under-Compression) or OC (Over-Compression) condition. It is further argued that the more dominant pulsation source should be from the later case (Δp_{41} induced), but without giving experimental proof.

This paper is aimed at validating the above theoretical predictions by the shock tube theory through experimental work. This would pave the way for devising new gas pulsation control strategy if the exact source and nature of gas pulsations can be located and understood. A brief review of the shock tube theory will be conducted first and followed by experimental investigation. The results are then compared between theory and test and conclusions are drawn.

2. GAS PULSATION SOURCES: A SHOCK TUBE THEORY

2.1 Brief Review of the Shock Tube Theory and Results

Refer to Huang [1, 2] for more detailed descriptions of the shock tube analogy with gas pulsations of PD compressors. However, highlights are summarized below in order to provide necessary bases for experimental comparison later. In general, two types of primary sources of gas pulsations [4] can be correlated to two types of shock tubes shown in Figures 1 & 2. The first type is Δp induced gas pulsation simulating the suddenly burst diaphragm shock tube and the second type is ΔU induced gas pulsation simulating a suddenly accelerated piston shock tube. An analogy is established between the shock tube at the moment just before and after the sudden diaphragm bursting and the discharge process of a rotary screw compressor under off-design conditions of an UC or OC, and the analogous equivalents are correlated in Table 2 and Figure 3.

Table 2: Analogy – diaphragm shock tube vs. rotary screw discharge under UC or OC

Device: shock tube	Device: rotary screw outlet
moment: before ($t = 0^-$) and after ($t = 0^+$) diaphragm opening	moment: before ($t = 0^-$) and after ($t = 0^+$) lobe opening
location: near front and back of the diaphragm with Δp_{41}	location: opening lobe experiencing Δp_{41}
high pressure region: p_4	outlet side pressure (UC): p_4
low pressure region: p_1	cavity side pressure (UC): p_1

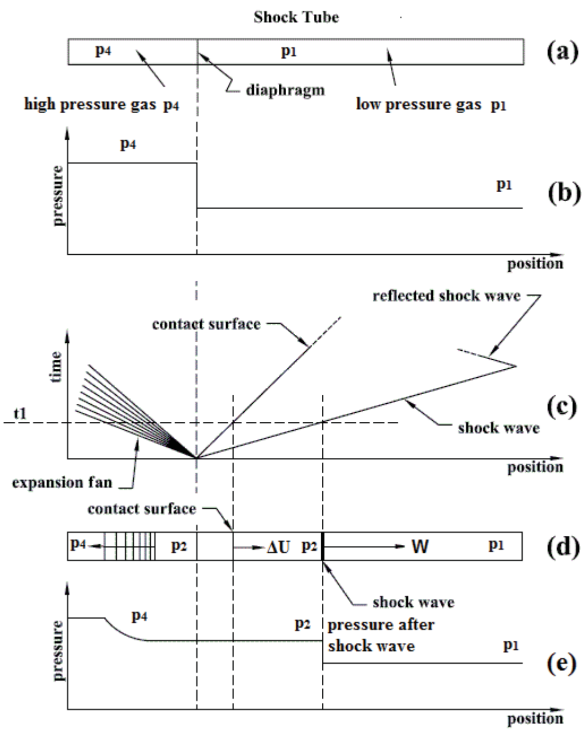

Figure 1(a-e): Diaphragm-triggered shock tube wave diagram from Anderson [4]

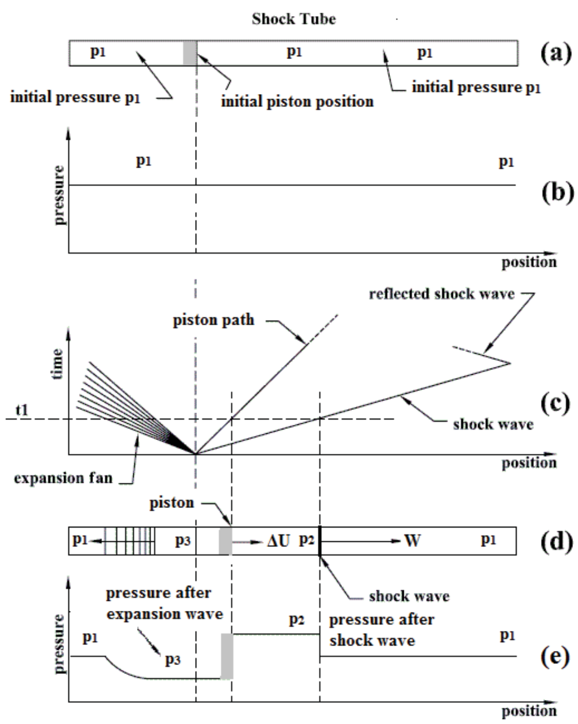

Figure 2(a-e): Piston-triggered shock tube wave diagram from Anderson [4]

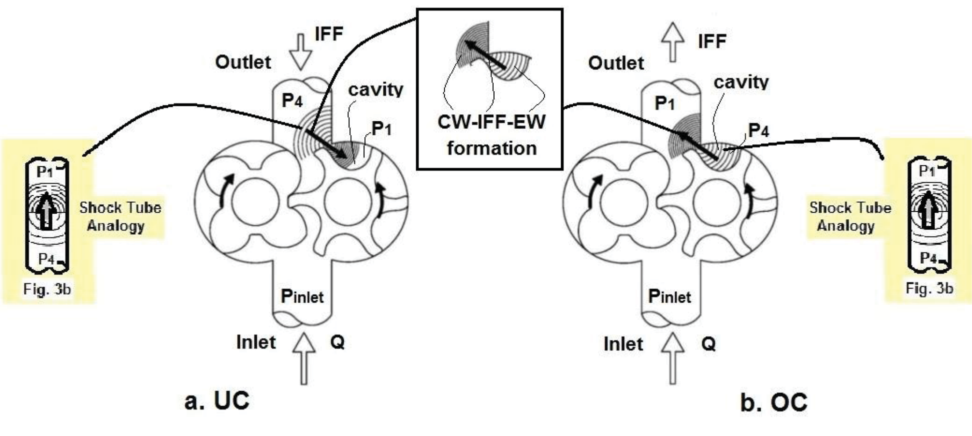

Figure 3(a-b): Gas pulsation sources of a screw compressor according to the shock tube theory from [2]

The results of the diaphragm shock tube analogy can be best summarized by the Gas Pulsation Rules [1, 2] as quoted below. In theory, these rules are applicable to different gases and for gas pulsations generated by different types of PD gas machinery, such as engines, expanders, pressure compressors and vacuum pumps, reciprocating or rotary.

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

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2. Rule II: If, at an instant, the divider between the high pressure gas p_4 and the low pressure gas p_1 is suddenly removed in the direction of divider surface, gas pulsations are instantaneously generated, at the location of the divider and at the instant of the removal, as a composition of a fan of Compression Waves (CW) or a quasi-shockwave, a fan of Expansion Waves (EW) and an Induced Fluid Flow (IFF) with magnitudes as follows:

$$CW = p_2 - p_1 = p_1 [(p_4/p_1)^{1/2} - 1] = (p_4 \times p_1)^{1/2} - p_1 \quad (1)$$

$$EW = p_4 - p_2 = CW * (p_4/p_1)^{1/2} = p_4 - (p_4 \times p_1)^{1/2} \quad (2)$$

$$\Delta U = (p_2 - p_1)/ (\rho_1 \text{ x } W) = CW/ (\rho_1 \text{ x } W) \quad (3)$$

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

3. Rule III: Pulsation component CW is the action by the high pressure (p_4) gas to the low pressure (p_1) gas while pulsation component EW is the reaction by low pressure (p_1) gas to high pressure (p_4) gas in the opposite direction, and their magnitudes are such that they approximately divide the pre-open pressure ratio p_4/p_1, that is, $p_2/p_1 = p_4/p_2 = (p_4/p_1)^{1/2}$. At the same time, CW and EW pair together to induce the third pulsation component, a unidirectional fluid flow IFF in a fixed formation of CW-IFF-EW.

In general, Gas Pulsation Rules explain the relationship between an UC (or OC) and gas pulsations as a cause (pre-opening pressure difference $p_4 - p_1$) and the effect (post-opening results) that are the two aspects of the same phenomena. Moreover, Rules I & II give the two sufficient conditions that link the UC (or OC) and gas pulsation events:

a) The existence of a pressure difference $p_4 - p_1$ from either an UC or OC;
b) The sudden opening of the divider separating the pressure difference $p_4 - p_1$.

Based on these two conditions, it can be determined that the location and moment that trigger the under compression action and gas pulsation generation are at the discharge and at the instant when the discharge port suddenly opens. Because all PD compressors or expanders convert energy between shaft and gas by dividing incoming continuous gas stream into parcels of cavity size and then discharges each cavity separately at the end of each cycle, there always exists a “sudden” opening at discharge phase to return the discrete gas parcels back to a continuous gas stream again. Therefore both sufficient conditions are satisfied at the moment of the discharge opening if compressor operates at off-design conditions such as an UC or OC.

Rule I also implies that there would be no or little gas pulsations during the suction, transfer and internal compression phases of a cycle because of the absence of either a pressure difference ($p_4 - p_1$) or a sudden opening. The focus instead should be placed upon the discharge phase, especially at the moment when the discharge port is suddenly opened and under off-design conditions like either an UC or OC.

Rule II also reveals the nature and composition of gas pulsations as a combination of large amplitude Compression Waves (CW) or a quasi-shockwave, a fan of Expansion Waves (EW) and an Induced Fluid Flow (IFF). These waves are non-linear waves with ever changing wave front during propagation. This is in direct contrast to the acoustic waves that are linear in nature and wave fronts stay the same and do not induce a mean through flow. It is also noted that the three different components (CW, EW and IFF) are generated as a homologous, inseparable whole simultaneously and in a fixed formation CW-IFF-EW. It is believed that this formation reflects the dynamics of the transient UC (or OC) and pulsation events with the wave fronts CW and EW as the moving forces driving the induced fluid flow IFF in between. In turn, the source of CW and EW is simply a re-distribution of the pre-opening UC (or OC) pressure difference Δp_{41} that is now being suddenly released and turned into a moving force pushing the flow (IFF) at front by CW and pulling the flow (IFF) from behind by EW at the same time. This new physical picture implies that gas pulsations would be difficult to control because it’s not one component (just IFF as suggested by the conventional theory) but all three components have to be dealt with as a whole.

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

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2.2 Quantitative Comparison of Pulsation Strength of ΔP Induced with ΔU Induced

To get a quantitative idea for the two primary pulsation sources, Δp_{41} (=p_4- p_1) and ΔU, Table 3 lists some typical values of industrial strength pulsation components: CW, EW and IFF, corresponding to Δp_{21} (=p_2-p_1), Δp_{42} (=p_4-p_2) and ΔU respectively from Equations (1)-(3). According to Equation (3), ΔU value in Table 3 can also be interpreted as the velocity of a piston-triggered shock tube that generates a CW gas pulsation with magnitude of Δp_{21} (=p_2-p_1) on the pressure side and an EW gas pulsation with magnitude of Δp_{42} (=p_4-p_2) on the vacuum side.

Table 3: Comparison of gas pulsation strengths: ΔP_{41} vs. ΔU

W in Mach Number	1.03	1.06	1.09	1.12	1.15	1.18	1.21
p_4/ p_1	1.146	1.306	1.480	1.666	1.870	2.080	2.312
Δ p_{41} (=p_4 - p_1), bar	0.146	0.306	0.480	0.666	0.867	1.083	1.310
EW, Δ p_{42} (=p_4 – p_2), bar	0.071	0.162	0.259	0.369	0.491	0.625	0.770
CW, Δ p_{21} (=p_2 - p_1), bar	0.075	0.144	0.220	0.297	0.376	0.458	0.542
IFF, ΔU, m/sec	17.1	33.7	50.0	65.7	81.1	96.2	111

From Table 3, it can be seen that a sudden release of a pressure difference Δp_{41} (=2.2 psi=0.146 bar) would generate an induced flow with velocity of ΔU (=17.1 m/s). On the other hand, it can be interpreted that an abrupt velocity change ΔU (=17.1 m/s) from a piston or lobe or valve movement would generate a CW pulsation front Δp_{21} (=1.1 psi=0.075 bar) and an EW pulsation front Δp_{42} (=1.0 psi=0.071 bar). But for rotary type PD compressors such as screw or scroll or Roots, it was argued in [1] that it is common to see Δp_{41} range from a few psi up to tens of psi but rarely see an abrupt rotary lobe velocity change (acceleration or deceleration) more than 17 m/s. This is due to the nature of smoother kinematics of rotary movement in contrast to reciprocating type.

From Gas Pulsation Rules I & II, Δp_{41} induced gas pulsations take place only at the moment of the discharge phase when the compression cavity is suddenly opened to an outlet pressure difference, a transient (point) source. On the other hand, ΔU induced gas pulsations occur all the time due to the constant interactions between gas and positive displacement driver surrounding the compressor cavity. Figure 4 plots both Δp and ΔU induced gas pulsations in terms of pressure ratios (p_2/p_1) or (p_4/p_2) as a function of UC or OC pressure ratio (p_4/p_1) of a compressor on logarithm scale so that an equal EW and CW relationship exists according to Pulsation Rule III.

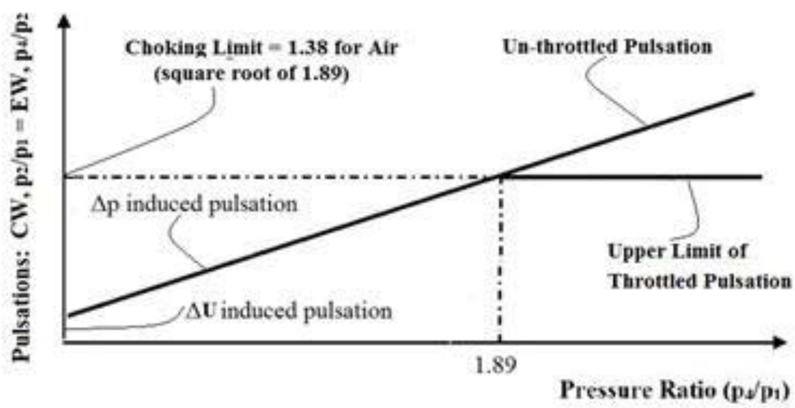


Figure 4: Gas pulsations CW, IFF and EW as a function of pressure ratio of UC or OC

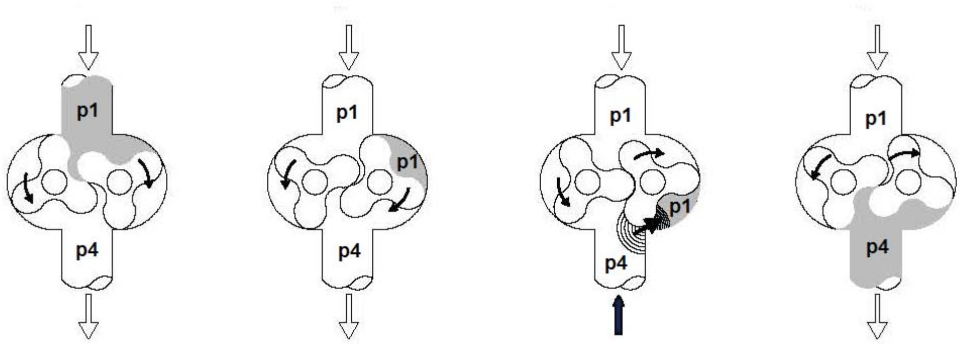
It can be seen from Figure 4 that ΔU is predicted to be small when compared with Δp_{41} for rotary type PD compressors such as screw, scroll or Roots. However, this prediction is based on pure theoretical speculation. The challenging question this paper tries to answer is: will experiments support this speculation?

3. GAS PULSATION SOURCES: EXPERIMENTAL INVESTIGATION

3.1 Test Objective and Test Machine Selection

The objective of experimental investigation is to find out which source, ΔU or Δp_{41}, is a dominant source either at the inlet or outlet of a rotary PD compressor that could influence upstream and downstream flow. In practice, it is difficult to distinguish the effect of ΔU from Δp_{41} due to the fact that two are mingled together. It was reasoned above that unlike the Δp_{41} induced gas pulsations that take place only at the moment when the compression cavity is suddenly opened to an outlet pressure difference, ΔU induced pulsations occur all the time because of the inherent nature of internal compression for most of the positive displacement type compressors such as screw or scroll. However, Roots type compression is an exception, the only one that possesses NO internal compression (or 100% pure UC), while has a rotary kinematic motion and rotary discharge valve that are similar to those of a screw or scroll.

For clarification, let's exam a complete cycle of a classical Roots blower as illustrated from Figures 5a to 5d by following one flow cell in a typical 3-lobe configuration. In Figure 5a, low pressure gas first enters the space between lobes of a pair of rotors as they are open to inlet during their outward rotation from inlet port to outlet port. At the lobe position shown in Figure 5b, the gas becomes trapped between the two neighboring lobes and casing forming a cavity as it is transported from the inlet to the outlet. Then the trapped gas in cavity is suddenly opened to higher pressure outlet as shown in Figure 5c. According to the shock tube theory for UC [5], the lobe opening phase in Figure 5c resembling the diaphragm bursting of a shock tube as shown in Figure 1d would generate a series of compression waves that sweep through the low pressure gas inside the cavity and compresses it at the same time at the speed of wave. After the compression, the rotors continue to move until lobes from two rotors meet again, mesh out the compressed gas from the cavity to outlet port and return to inlet suction position to start the next cycle, as shown in Figure 5d.



a. Suction **b. Transfer** **c. Compression** **d. Discharge**

Figure 5: Compression cycle of a classical Roots according to the shock tube theory

The Roots cycle is unique because there is NO cavity volume variation during the transport and compression phases. That is, there is NO internal compression like the constant meshing of twin screw rotors or scroll disks. The meshing lobes of Roots only interact together with gas during the suction and discharge phases while during the transfer and compression phases, the gas stays inside the fixed volume cavity formed by the two neighboring lobes and casing.

According to Gas Pulsations Rules I & II, the Roots transfer phase should have NO pulsations because of the absence of any pressure difference or lobe meshing. Lobe interaction only takes place during the Roots suction and discharge phases that could induce ΔU type pulsations. Moreover, there should exist strong Δp_{41} type pulsations during the Roots discharge phase because it is a case of 100% UC. It can be further anticipated that the larger the pressure rise from Roots inlet to outlet, the higher Δp_{41} pulsations would be induced according to Rule II. The resulting gas pulsations are a composition of pressure waves (CW, EW) and induced backflow (IFF) as shown in Figure 5c. Because of its unique features, the Roots compression is selected as the test bed and a 75HP Roots blower is prototyped first for this purpose.

3.2 Test Schemes for Source Differentiation

Two test schemes are devised to differentiate the contribution between ΔU and Δp_{41}. Scheme A is to measure directly the gas pulsations at the conventional blower inlet and outlet as shown in Figure 6a under different speed (relate to ΔU effect) and pressure rise (relates to Δp_{41} effect) conditions. The strategy is, since it is expected that the inlet pulsation has dominant contribution from ΔU (little Δp effect) while outlet pulsations has dominant contribution from Δp_{41}, the comparison of the pulsation magnitude of the inlet and outlet pulsations would tell which effect is dominant at the inlet or outlet. Note that the case for No load condition (when there is no pressure rise from inlet to outlet, or $\Delta p_{41} = 0$) is special because it can reveal the full effect of ΔU pulsation without any contribution from Δp_{41}.

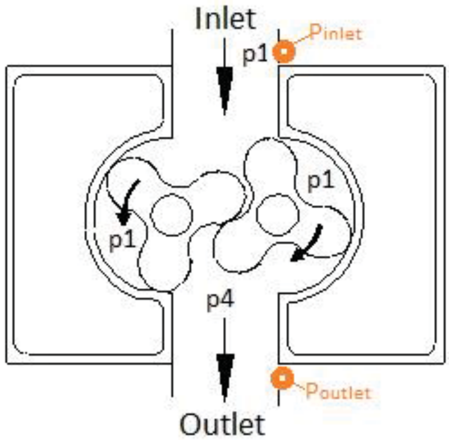


Figure 6a: Test scheme-A setup

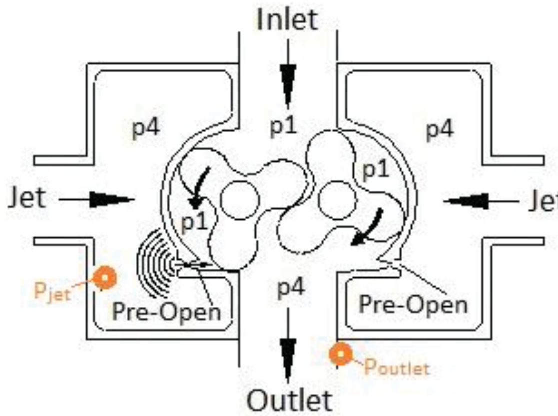


Figure 6b: Test scheme-B setup

An alternative way or scheme B is devised to totally separate Δp_{41} effect from ΔU effect by measuring gas pulsations at a non-conventional jet port connected to blower cavity through a pre-opening as shown in Figure 6b. The position of the pre-opening is such that it is at least 120 degrees away (for 3-lobe) or 180 degrees away (for 2-lobe) from the inlet opening, but long before the discharge opening. Remember that during the transfer phase of Roots cycle, there is neither rotor meshing effect nor cavity volume variation. Moreover, lobe is rotated at a constant speed with no acceleration or deceleration until it rotates through the pre-opening and the cavity is suddenly opened to the pressure difference Δp_{41}. In another word, this location is expected to “feel” the full Δp_{41} effect but little ΔU effect. To physically isolate Δp_{41} effect measured at the jet port from ΔU effect measured at the outlet, the test is setup as an exhauster so that the inlet is subjected to a vacuum load (Δp_{41}) while both jet and outlet ports are separately open to the same atmospheric pressure.

3.3 Test Results and Discussion

The measured gas pulsations at different speeds and loads are shown as colored points in Figures 7a and 7b respectively for schemes A and B. For consistency, both tests use the same dynamic pressure probes and recording instruments, and the test piping setup follows ASME PTC-9 specification (not shown in Figures 6a & 6b).

Test results of scheme A in Figure 7a show that the inlet pulsation is low and almost at a constant level around 0.2-0.4 psi, independent of either the rotor speed (range from 2400-3600 RPM, nominal) or pressure rise Δp_{41} (range from 1-16 psi). While the outlet pulsation is much higher, ranging from 0.4–2.4 psi and approximately proportional to the pressure rise from inlet to outlet. The blower speed does not show much effect at the outlet either as at the inlet. It is also interesting to note that an extended line from the average of the outlet pulsation data almost intersects with the extension line from the average of the inlet pulsation data at the point where $\Delta p_{41} = 0$. This suggests that the ΔU effect from the outlet side is almost the same as the inlet side, agreeing with predictions made from Pulsation Rule II that there would be no or little gas pulsations at both the suction (inlet) and discharge (outlet) under the no load condition ($\Delta p_{41} = 0$). The trend and order of magnitude of outlet gas pulsation shown in Figure 7a are also consistent with theoretical predictions in Figure 4 that stronger gas pulsation takes place during the compression and discharge phase when lower pressure cavity is suddenly opened to higher pressure outlet.

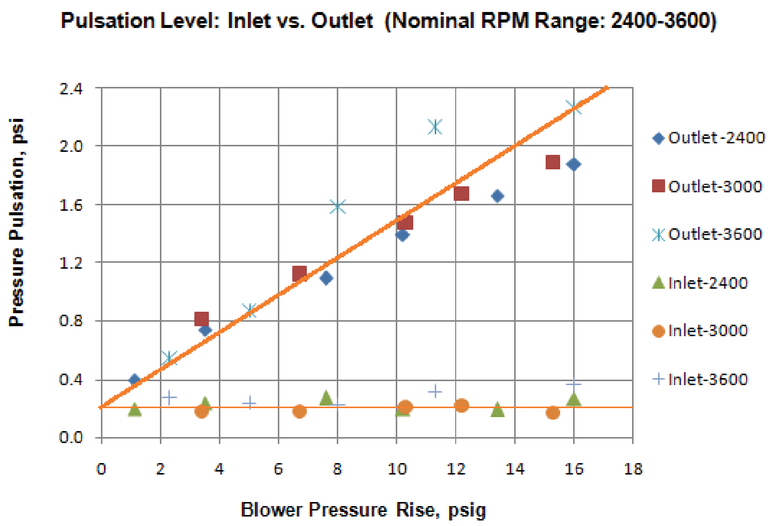


Figure 7a: Results of test scheme-A

Test results of scheme B in Figure 7b show that gas pulsations can be triggered to take place before the outlet upon the sudden pre-opening of the low pressure cavity to higher pressure jet port shown in Figure 6b. Because test is conducted under an exhauster mode so that outlet and jet ports can see the same atmospheric pressure, the level of absolute pressure rise from inlet to outlet is smaller than the blower mode as shown for scheme A with the same pressure ratio. The result in Figure 7b shows that pulsation at the jet port is high ranging from 0.2 to 0.9 psi while varying proportionally with pressure rise (Δp_{41}), the same trend as the outlet pulsation in Figure 7a. However, the outlet pulsation is now low, less than 0.15 psi and is almost a constant, which is the same trend as the inlet pulsation in Figure 7a and is also independent from the exhauster pressure rise (Δp_{41}) and speed. It is interesting again to note that an extended line from the average of the scattered jet pulsation data intersects with the zero point of both pulsation and pressure rise where $\Delta p_{41} = 0$. This suggests that the jet port does not see much of the ΔU effect, unlike the outlet gas pulsations that have both Δp_{41} and ΔU effect mingled as shown in Figure 7a. This observation is consistent with the theoretical prediction discussed in paragraph of “3.2 Test Schemes for Source Differentiation” that the jet port is the transient source “point” expected to “feel” 100% Δp_{41} effect so that outlet has only 100% ΔU effect left for test setup shown in Figure 6b.

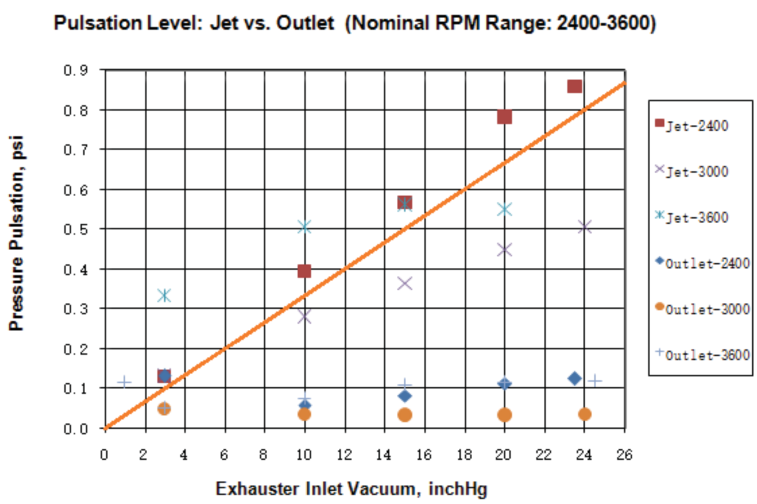


Figure 7b: Results of test scheme-B

More important is the implication from the fact shown by scheme B and Figure 7b that the dominant Δp_{41} source can be actually separated from the ΔU source and directed away from the outlet to a different location that could be eventually treated and controlled. In fact, this finding or insight is the basis for a new pulsation control method, called shunt pulsation trap that is beyond the scope of this paper. However, interested readers can refer to references [2, 6] for more details.

4. CONCLUSIONS

Pressure pulsations of a Roots blower (100% Under Compression) are measured at inlet/outlet for scheme A and at outlet/jet port as scheme B under different speed and load conditions. Though there are some scattering among the dynamic pressure data due to the wave nature of pressure pulsations, conclusions can be drawn based on the average behaviors from these tests as follows:

1. The ΔU gas pulsation mainly takes place at the inlet and outlet of the blower where two rotors are engaged in constant meshing interaction. However, the pulsation level due to ΔU effect is low (about an order of magnitude lower) compared with Δp_{41} gas pulsation and is independent of rotor speed and blower loads (pressure rises Δp_{41});
2. The Δp_{41} gas pulsation mainly takes place at the outlet of the blower triggered by the sudden opening of the low pressure cavity with $p_1$to the higher pressure outlet with p_4. Moreover, the pulsation level due to Δp_{41} ($=p_4-p_1$) is almost one order of magnitude higher than the ΔU effect, and is approximately proportional to the magnitude of blower load Δp_{41};
3. The Δp_{41} gas pulsation (a point source) can be separated from the ΔU gas pulsation (a constant source at inlet and outlet from rotor pair and gas interacting) by a pre-opening or jet port.

Conclusions 1-2 validate experimentally some of the theoretical predictions from the previous papers [1, 2], that is, ΔU pulsation is taking place throughout the rotor intermeshing process at the inlet and outlet while Δp_{41} pulsation takes place at a single point and moment when and where the Δp_{41} is suddenly exposed. This insight leads to Conclusion 3 that is the foundation for a new pulsation control method, called Shunt Pulsation Trap, as detailed in previous papers [2, 6].

Though the tests conducted in this work is on a Roots type blower that is operating under 100% under-compression (UC), the general principle should also apply to other types of rotary PD compressors such as screw or scroll with a partial UC or OC. However, direct experiments are more desirable for rotary PD compressors that possess internal compression as well as UC and OC. With a better understanding of the nature of gas pulsations, it is anticipated that future generation of rotary PD compressors can be designed to be even simpler in structure, smaller in size and smoother running than those used today.

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