2014

Gas Pulsation Control Using a Shunt Pulsation Trap

Paul Xiubao Huang
Hi-Bar MC Tech, United States of America, paulxbh@yahoo.com

Sean Yonkers
Hi-Bar MC Tech, United States of America, syonkers@ix.netcom.com

David Hokey
GE Oil and Gas, United States of America, david.hokey@ge.com

Gas Pulsation Control Using a Shunt Pulsation Trap

Paul Xiubao HUANG[1]*, Sean YONKERS[2], David HOKEY[3]

[1, 2] Hi-Bar MC Technologies, LLC.
Fayetteville, GA, USA
(Tel: 678-817-1497, E-mail: paulxbh@yahoo.com)

[3] GE Oil & Gas
Connersville, IN, USA

* Corresponding Author

ABSTRACT

Gas pulsations commonly exist in HVACR, energy and automotive industry. They are believed to be a major source for system inefficiency, vibrations, noises and fatigue failures. It has been widely accepted that gas pulsations mainly take place at the discharge side of a positive displacement (PD) type compressor such as a screw, scroll or internal combustion engine. The pulsation magnitudes, ranging from a fraction to a few bars, are especially significant at off-design conditions of either under-compression (UC) or over-compression (OC). Traditionally, a serial pulsation dampener, often a reactive type silencer, is connected after the compressor or engine discharge. It is capable of reducing pressure pulsation by 10 plus fold; or 20 plus dB. However these dampeners are bulky and impose sizable back pressure losses.

This paper discusses an alternative method, a shunt pulsation trap (SPT), using a parallel configuration, which tackles the gas pulsations before the compressor or engine discharge. The SPT method is based on the shock tube theory (Huang, 2012a) that characterizes the nature of gas pulsation as a composition of non-linear waves, compression wave (CW) and expansion wave (EW) and induced fluid flow (IFF). The theory also predicts that the dominant source of gas pulsation is a direct result from the sudden discharging process under UC or OC condition. Two prototypes, 75 HP and 350 HP Roots type blowers, were built and tested with and without SPT. The experimental results partially validate the new theory and also indicate a 10 plus fold; or 20 plus dB pulsation reduction by using SPT under different load and speed conditions without suffering any back pressure losses.

1. INTRODUCTION

1.1 PD Compressor Classification and Under-Compression, Over-Compression Modes

PD compressors are capable of generating high pressures for a wide range of flows and are widely used in various applications. For example, pipeline transport of purified natural gas, natural gas processing plants, petrochemical plants, and large industrial plants for compressing intermediate or end product gases. Other examples include refrigeration, air conditioner equipment and many various manufacturing processes that power all types of pneumatic tools.

A positive displacement compressor converts shaft energy into velocity and pressure of a gas media. In a broader sense it includes gases and gas mixture by trapping a fixed amount of gas into a cavity then compressing that cavity and discharging into the outlet pipe. A positive displacement compressor can be further classified according to the mechanism used to move the gas. A rotary type, such as screw or scroll and a reciprocating type, like a piston or diaphragm, as shown in Figure 1.

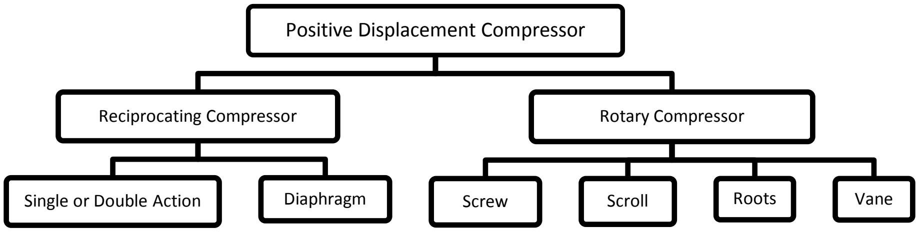

Figure 1: Conventional compressor classification

Though each type of PD compressor has its own unique shape, movements, principle, strengths and weakness, they all have in common a suction port, a volume changing cavity and a discharge port where a valve mechanism controls the timing of the release of gas media. Moreover, they are all cyclic in nature and possess the same cycle for the processed gas, that is, suction, compression and discharge. Figures 2a-2d show the compression cycle of a conventional rotary screw compressor and Figure 2e shows a serial outlet dampener connected to the discharge port. Gas flows into the compressor as the cavity on the suction side expands and then traps the media to being compression by mechanical means, say a piston or lobe, as the trapped cavity volume is reduced. After a desired compression ratio or volume reduction ratio is reached, the discharge valve or porting is opened and gas flows out of the discharge into the outlet. The inlet volume is constant given each cycle of operation and discharge volume varies according to the compression ratio as designed.

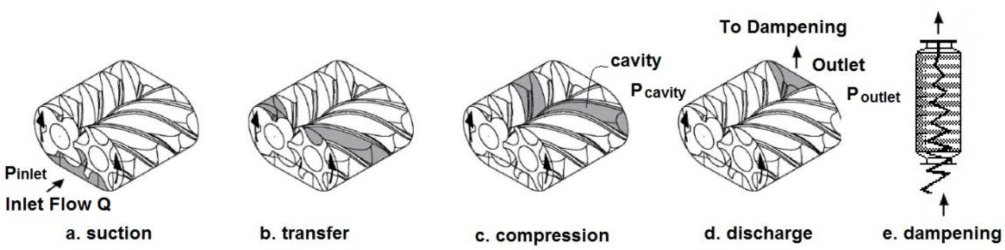

Figure 2(a-e): Conventional screw compression cycle with a serial dampener at discharge

Since all PD compressors divide the incoming gas mechanically into parcels of cavity size for delivery to the discharge, they inherently generate gas pulsations with cavity passing frequency at the discharge, and the pulsation amplitudes are especially significant under high operating pressures or off-design conditions of either an UC or OC (Stoecker, 2004), as shown in Figures 3a-3b.

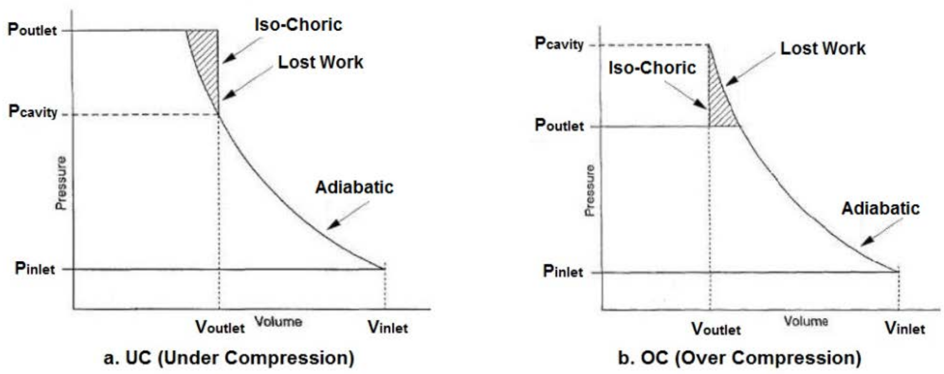

Figure 3(a-b): Screw compressor under compression or over compression processes on P-V diagram

UC happens when the pressure at the discharge opening, system back pressure, is greater than the pressure of the compressed gas within the cavity just before the opening. This results in a rapid backflow of the gas into the cavity, a pulsed flow, according to conventional theory. All fixed pressure ratio compressors suffer from UC due to varying system pressures. An extreme case is the Roots type blower where there is no internal compression at all, or UC is 100%, so that pulsation constantly exists and pulsation magnitude is directly proportional to the gas pressure rise from blower inlet to outlet. On the other hand, OC takes place when pressure at discharge opening is smaller than pressure inside the compression cavity, causing a rapid forward flow of the gas into the discharge. For most applications where the system back pressure is normally not a constant, a fixed pressure ratio PD compressor will result in either an UC or OC.

1.2 PD Gas Pulsations and Adverse Effects

It has long been observed that gas pulsations take place mainly at the discharge side of the compressor with cavity passing frequency and its amplitude is especially significant under high operating pressures; or at off-design conditions of either UC or OC as reported by Mujiü *et al*. (2007), Koai and Soedel (1990), Sangfors (1999), Wu *et al*. (2004) and Gavric and Badie-Cassagnet (2000). For example, screw pulsation levels can range from 0.02 – 2 bar (0.3 - 30 psi); or equivalent to 160-200 dB. For Roots type, pulsation amplitude is small, ranging from 0.002 – 0.2 bar (0.3 - 3 psi), but it constantly exists due to being a 100% UC. The gas pulsations generated by UC or OC are transient within the gas discharge flow line, called gas borne, and periodic in nature. They travel long distances throughout the downstream piping system and if left uncontrolled, could potentially damage pipe lines and equipment in the form of fatigue failure, and excite severe vibrations and noises as reported by Price and Smith (1999), Tweten *et al*. (2008) and Peters (2003).

In addition to severe gas pulsations, the compressor efficiency at UC or OC becomes worse than the design point as indicated by additional work, seen as the shaded areas in Figures 3a-3b. The conventional theory as exemplified by Stoecker (2004) and Huang (2012b), models the thermodynamic process of UC or OC as isochoric, deviating abruptly from the adiabatic compression curve when the discharge opening suddenly occurs as represented by a vertical rise or fall of pressure on P-V diagram as illustrated in Figures 3a-3b. For this reason, PD compressors are often cited unfavorably with high pulsation induced noise, vibration and harshness (NVH) and low efficiency at off-design conditions when compared with dynamic types like the centrifugal or axial compressor.

1.3 Serial Gas Pulsation Control Method and its Limitation

To control gas pulsations, a serial dampener, also called reactive silencer or muffler, is required to be connected in series with the discharge port. A reactive type silencer works by employing plenums that consist of a number of chokes. Its effectiveness in pulsation control depends on the size and number of the plenums; or stages of dampening. By increasing the size and number of stages attenuation is increased. An ideal design can achieve a reduction of 20 plus dB, but results in a large size which creates other problems such as inducing more noises and vibration due to the increase of surface area and supported weight. Sometimes internal dampener structure fatigue fail, resulting in possible catastrophic damage to downstream components and equipment. Additional weight and space is another concern, especially for mobile applications. Figures 2-4-5 show some examples of a screw, roots and scroll compression cycles with serial silencer connected at discharge. Another familiar sight is the auto muffler used after the internal combustion engine that is in essence a reactive type silencer.

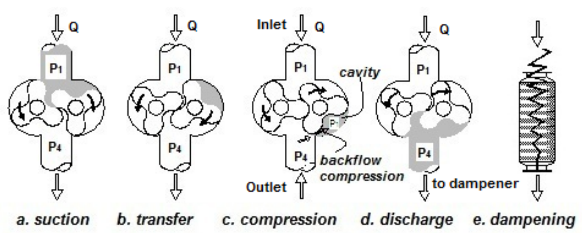

Figure 4(a-e): Conventional Roots compression cycle with backflow compression principle

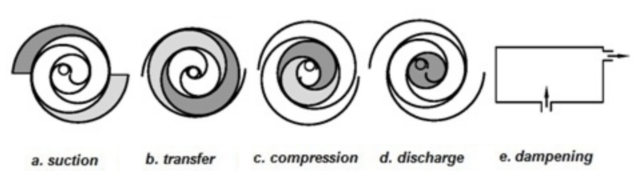

Figure 5(a-e): Conventional scroll compression cycle with a serial dampener at discharge

Another major defect associated with a serial type discharge dampener is the static pressure loss that reduces the compressors overall efficiency. The pressure loss is directly proportional to the attenuation for a fixed size silencer. For a typical silencer, they range from 0.4 psi to 5 psi or even higher.

The challenging question becomes, is there a new way to attenuate the gas pulsations as effective as a serial dampener, 20 plus dB, while staying small in size and not suffer a back pressure loss? This paper attempts to answer these questions and discuss a new pulsation control strategy with a Shunt Pulsation Trap (SPT).

1.4 Parallel Dampening Attempts and a New Shunt Pulsation Trap Approach

For over a hundred years, serial dampening configuration has been and perhaps still is the dominant method tackling gas pulsations. However, various efforts have been attempted to reduce gas pulsations without using a serially connected dampener or orifice plate at the discharge over the past 50 years. The most widely used method is based on a flow feedback principle as discussed by Weatherston (1984), Yanagisawa and Maeda (1989). The idea is to feed back a portion of the compressed gas from a modified outlet port shape or through a pre-opening slot or holes, thereby gradually equalizing gas pressure inside the cavity and reducing discharge pressure spikes compared with an abrupt lobe opening at discharge. Figure 6a is an example of a pre-opening design from Weatherston (1984). However, its effectiveness for pulsation attenuation is typically less than 2 fold pulsation reduction; or 5-6 dB.

It is the belief of the present authors that a more effective pulsation control method is only possible as a result of a better understanding of the nature of gas pulsations and addressing the fundamental questions like, what is pulsation control up against? Is gas pulsation a wave or particle movement or both? Where and what exactly triggers it? These questions have been discussed in papers (Huang, 2012a; Huang, 2012b) under the titles of "Gas pulsations: a Shock Tube Mechanism" and "Under (or Over) Compression: an isochoric or adiabatic process?".Based on the new theoretical predictions from this work, this paper will employ a different method from those by Weatherston (1984), Yanagisawa and Maeda (1989) but follow the same parallel configuration approach to tackle gas pulsations. The purpose is to build a new control strategy that gives understanding and suggestive insight rather than detailed design and calculation. It is believed that at the present state of the art, this is supplementing the CFD modeling due to its inability to describe the transient nature of the gas pulsation phenomena and the limitation of the serial dampening configuration.

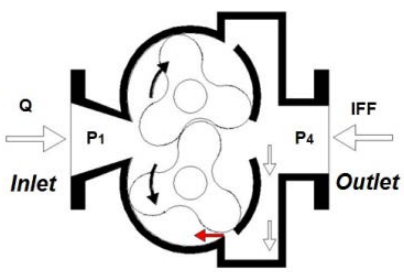

Figure 6a: A Roots blower with pre-opening and backflow compression

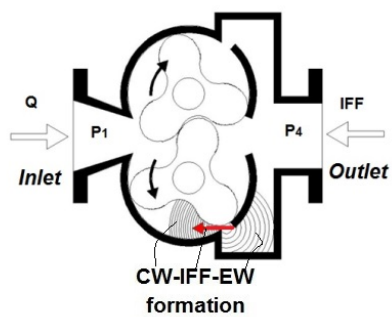

Figure 6b: A Roots blower with a pre-opening and wave compression

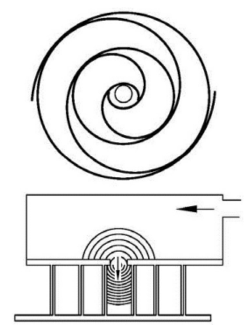

Figure 6c: A scroll compressor with wave compression at UC mode

2. GAS PULSATION CONTROL: A SHUNT PULSATION TRAP METHOD

2.1 Summary of Shock Tube Mechanism: Gas Pulsation Generation Trigger and Physical Nature

Refer to Huang (2012a) for a more detailed description of the shock tube analogy with the UC or OC induced gas pulsations illustrated in Figure 7. The essence of the theory is best summarized by the Gas Pulsation Rules that are quoted below for a smooth context flow. In principle, these rules, if validated by experiments, 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.

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.
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 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-trigger 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 relationship of CW-IFF-EW.

Rule I implies that there would be little or no gas pulsations during the suction, transport and internal compression phases of a cycle if there is no pressure difference and abrupt opening. The focus instead should be placed upon the sudden discharge phase, especially at the instant when the cavity is suddenly opened to a different outlet pressure during the off-design conditions like UC or OC.

Rule II indicates specifically that the ***moment*** triggering the gas pulsation generation is the instant the lobe separating p_4 and p_1 suddenly opens and that the ***location*** is at the opening lobe. Moreover, it defines the ***two sufficient conditions*** to generate a gas pulsation as follows:

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

Because all PD compressors have a "sudden" opening to Δp_{41} under UC or OC for each cycle, both sufficient conditions are satisfied at the moment of the cavity opening to the discharge, thus the reason gas pulsations are inherent at off-design conditions for PD type gas machinery.

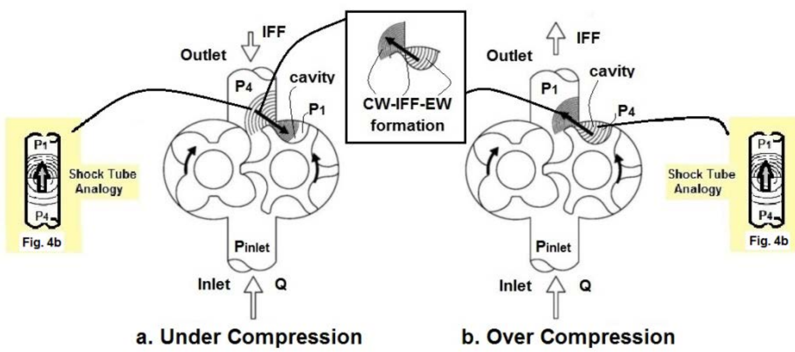

Figure 7(a-b): Screw compressor under UC and OC modes according to the shock tube analogy

The Gas Pulsation Rules predict a new physical picture for the gas pulsation, the sudden opening at the discharge under either UC or OC with a pressure difference Δp_{41} triggers simultaneously the generation of a composition of compression waves (CW) or a shockwave, a fan of expansion waves (EW) and an induced reverse fluid flow (IRFF) as a homologous whole and in an inseparable formation of CW-IRFF-EW. This image is daunting, the gas pulsation is not just a back flow anymore, but always accompanied with a pair of strong moving waves that will propagate in both directions at the speed of the waves, and along the way induce fluid flow and reflect upon hard surfaces and potentially cause structural damage. Hence, any pulsation control strategy should deal with all three components of the gas pulsation, CW-IRFF-EW, before they get into the pipe line. It is believed that this formation reflects the dynamics of the transient gas pulsation event with the wave fronts CW and EW as the forces driving the induced fluid flow IFF in between. In turn, the source of the post-trigger CW and EW is simply a re-distribution of the pre-trigger potential energy Δp41 of UC or OC that is now being suddenly released and turned into a moving force driving the flow in the pressure gradient Δp_{41} direction. Figures 6b-6c-7a-7b illustrates this new physical picture for different types of compressors for UC mode.

In light of the shock tube theory, let's take a new look at Figures6a-6bwhich illustrate the same under compression phase but interpreted by the new pulsation generation mechanism. It can be seen from Figure 6b that the EW pulsation component is left loose at the outlet side even though the pre-opening makes the opening more gradual. This could explain the limited success of the pre-opening method, just 2-fold reduction; or 5-6 dB. So something extra is needed that would not only eliminate the pressure difference Δp_{41}before discharge but also control the EW pulsation component, or CW component in the case of an OC, at the same time. This new method is called the shunt pulsation trap (SPT) in this paper and relevant references (Huang, 2009; Huang, 2011; Huang, 2013).

2.2 Principle of Shunt Pulsation Trap Method

As a brief description of the principle of a shunt pulsation trap, Figure 8 shows again a Roots type blower, PD compressor at 100% UC, but with an addition of a shunt pulsation trap near the compression cavity. In principle, a SPT is used to both TRAP and ATTENUATE of all three components of the gas pulsation CW-IFF-EW at the same time. For comparison, a traditional discharge dampener is connected in series with the cavity, compression chamber, after the discharge port, as shown in Figure 2e and Figure 4e, and through which both the cavity flow Q and the gas pulsation flow IFF pass. While a shunt pulsation trap is connected in parallel with the cavity before the discharge port through which only the gas pulsation flow IFF passes, shown in Figure 9b.The phases of flow suction, transfer and internal compression for both serial and parallel configuration are still the same as shown in Figures9a-9b. But during the discharge phase, only UC mode is discussed here, as illustrated in Figure 8, instead of waiting for the lobe opening at discharge as a conventional PD compressor, the flow cavity is pre-opened to a port termed "trap inlet". Hence the trap inlet is connecting, before discharge, the cavity to the pulsation trap in parallel, which in turn is also communicating with the compressor outlet through a feedback port, termed "trap outlet". Between the trap inlet and trap outlet and within the shunt pulsation trap, there is a pulsation dampening means to control the EW pulsation component that was left loose in the pre-opening method by Weatherston (1984), Yanagisawa and Maeda (1989).

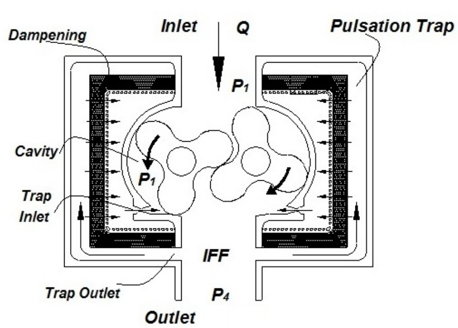

Figure 8: Roots blower with a Shunt Pulsation Trap for UC mode

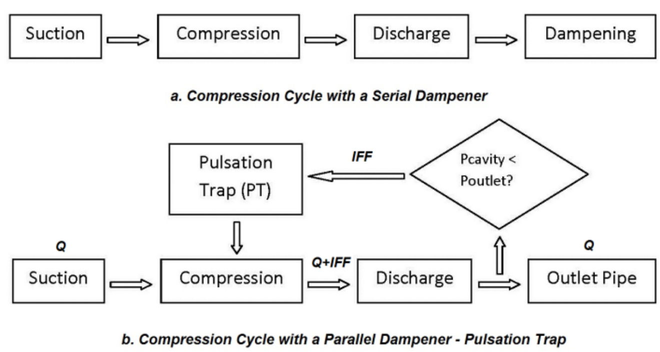

Figure 9(a-b): Comparison of a Shunt Pulsation Trap with the conventional serial configuration for UC mode

The difference between serial and parallel configurations is fundamental and significant here as there are several distinct advantages associated with the shunt pulsation trap when compared with the traditional serially connected pulsation dampener. First of all, the attenuation of the pulsation gas flow IFF is separated from the main cavity gas flow Q so that an effective IFF dampening will not affect the main cavity gas flow Q, resulting in both higher system efficiency, no discharge pressure loss is associated with a parallel dampener, and maintain attenuation efficiency. In a traditional serially connected dampener, both the pulsating gas flow IFF and cavity gas flow Q travel mixed together through the dampener where a better attenuation always comes at a cost of higher static pressure losses. So a compromise is often made in order to reduce the losses by sacrificing the degree of pulsation dampening or have to employ a very large volume and costly dampener in a serial setup. Secondly, the parallel pulsation trap attenuates the gas pulsation closer to the source of pulsation generation and in a shape more compact and conformal to the cavity shape than a serial one, resulting in smaller size and less weight. The key question is, will the actual testing support the shunt pulsation trap concept designed according to the Gas Pulsation Rules rather than the conventional backflow theory?

2.3 Applications of Shunt Pulsation Trap to Different Compressors

The shunt pulsation trap concept can be applied to different types of PD compressors as shown in Figure 10 for the case of UC mode; OC case is similar but dealing with CW wave instead.

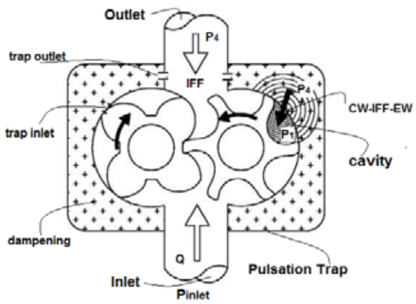

Figure 10a: Screw compressor with a Shunt Pulsation Trap for UC mode

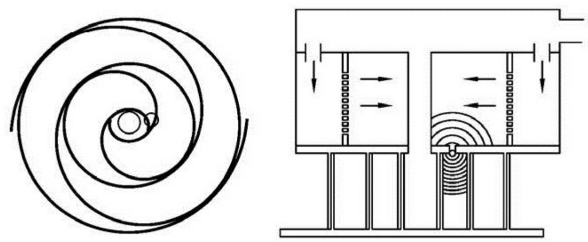

Figure 10b: Scroll compressor with a Shunt Pulsation Trap for UC mode

The key issue is the location and size of the trap inlet. In principle, the suddenly generated CW waves travel into the cavity, compressing the gas inside with the assistance of IFF. At the same time, the simultaneously generated EW waves at the trap inlet are propagating into the pulsation trap, and therein are being contained and attenuated. Because waves travel at the speed of sound, about 5-20 times faster than the piston or rotor tip speed, the trap inlet should be designed so that both CW and EW pulsations could be well settled down after a few reflections before the piston or lobe tip reaches the outlet, hence discharging a pulsation-free flow. Thus, there is no more need for a serial dampener. For a wide range of operating pressures, the desired trap inlet of a pulsation trap is so designed that the compressor will operate under a partial internal compression and partial UC mode, but never at OC mode in order to maximize the average system efficiency and minimize the gas pulsations and induced NVH.

3. VALIDATION TESTING OF SHUNT PULSATION TARP (SPT)

3.1 Prototype I - 75 HP Test, Results and Discussion

To validate the shunt pulsation trap concept experimentally, a special case of 100% UC of a 75HP Roots type blower is prototyped with a SPT designed and built according to the Gas Pulsation Rules. For comparison, base tests are first conducted using two conventional designs, one with a serial dampener at discharge and another with a pre-opening injection port called WhispAir (Roots, 1980). For consistency, the same test setup and instruments are used again to test and measure prototype I with a SPT. The test setup follows ASME PTC-9 specification and blower steady state performance data were measured at different speeds and pressure rises. In addition, gas pulsations are measured using piezoelectric high response dynamic pressure transducers at the blower inlet and discharge, before and after the serial or parallel dampener for the whole range of design pressure and speed. Both CSI signal analyzer and National Instruments NI-DAQ system are used and compared for accuracy of the data acquisition system for the dynamic pressure measurement. A sample discharge pulsation measurement data in Figure 11 show pressure pulsation traces, confirming the large pulsation amplitude, pulsation unit are psi, and the dominant cavity passing frequency of a gas pulsation from Roots type compression with 100% UC.

The first validation step is to demonstrate whether a SPT could really lure the gas pulsation into the trap so that discharge would be pulse free and no serial dampener is needed at discharge. The blower is tested under vacuum mode so that discharge and trap are both open to atmosphere, but isolated from each other. The result is shown in Figure 12, indicating indeed that the gas pulsation can be triggered to take place earlier at trap inlet, before discharge port, upon the sudden opening to discharge pressure. This confirmed what Weatherston (1984) has found using pre-opening method and is also consistent with the prediction from the sufficient conditions for pulsation generation according to the Gas Pulsation Rules. Moreover, the measurement shows that pulsations inside the trap is relatively high, 0.5-0.9 psi, while discharge pulsation is low, < 0.15 psi. In other words, all the pulsations are lured into the trap for subsequent treatment so that outlet becomes pulse free; hence there is no need for a silencer. With the measured pressure pulsations high inside the trap, it also proves, at least partially, that pulsations are not just a reverse flow phenomena as theorized conventionally but possess multiple pulsation components with CW and IFF pulsations going into the cavity, and EW component coming out into the trap, as depicted in Figures 6-7. It could be this EW pulsation component that explains why pre-opening only method by Roots (1980) only achieves a 2 fold pulsation reduction; EW pulsation component exists and needs to be controlled too.

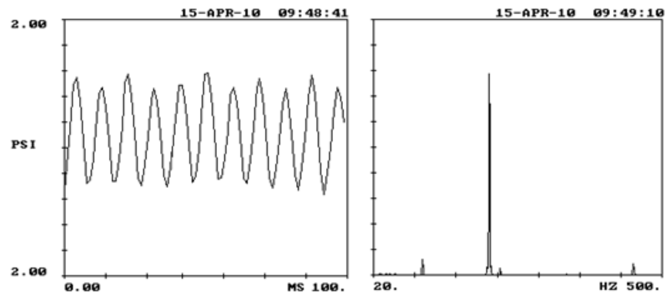

Figure 11(a-b): A typical pulsation measurement at Roots discharge with data taken by a CSI analyzer

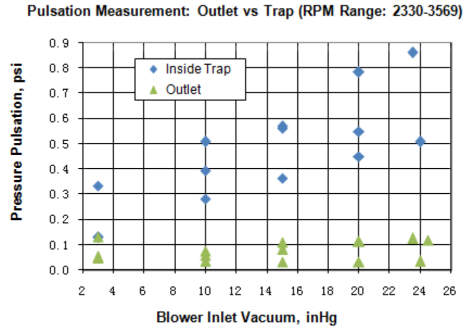

Figure 12: SPT luring capability: from outlet to inside the trap

The second validation step for prototype I is to demonstrate whether the lured-in gas pulsation inside the SPT could be killed or attenuated and how effective the SPT is when compared with a serial dampener. The blower is tested under the pressure mode which means the inlet is open to atmospheric pressure and Figure 13 shows the measured discharge pulsations under different pressure rises and speeds of a SPT in comparison with conventional serial dampening and pre-opening methods. They are all compared with the same pressure pulsations before any dampening at the same operating conditions, flow, pressure rise and RPM.

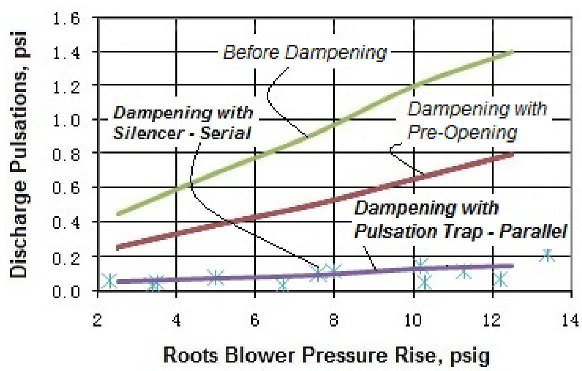

Figure 13: Comparison of pulsation measurements for different dampening methods under UC mode, speed range: 2330-3569 RPM

Test data first confirms the observed results that the gas pulsation from UC takes place mainly at discharge with magnitude ranging from 0.01 – 0.1 bar (0.15 – 1.5 psi) in Figure 13 and is directly proportional to the pressure difference of a UC Roots blower, up to 1 bar (15 psi), while the inlet side pulsation is at least one order of magnitude lower, less than 0.01 bar (0.15 psi). Secondly, it demonstrates the effective dampening by a traditional serially connected dampener, 5-10 fold pulsation reduction or 10-20 dB at discharge, but with a dampener static pressure loss ranging from 0.01-0.03 bar (0.15-0.45 psi). Thirdly, it further confirms the level of dampening by a pre-opened feedback design, about 2 fold pulsation reduction at discharge or 5-6 dB. Lastly, 10 fold pulsation reduction is achieved at the high end of pressure rise with a SPT without using a discharge dampener, hence resulting in no dampener related losses. It should be pointed out that the above pulsation dampening result of SPT is achieved from a single design across the whole working range of pressure rises and operating speeds of the blower.

The above data showing effective pulsation attenuation by the SPT can also be interpreted as a partial validation for the new pulsation theory because it addresses EW pulsation in addition to CW and IFF pulsation components. It is the additional control specifically targeting the CW and EW, wave control in addition to flow control that differentiates the SPT with the pre-opening method, though both are in parallel configuration.

3.2 Prototype II - 350 HP Test, Results and Discussion

With the promising results achieved on the 75 HP prototype, the question is whether the same principle could be applied to a much larger machine with a different speed range. Moreover, how much shaft power (BHP) could be really saved by replacing a traditional serial dampener with a SPT with the same level of pulsation attenuation while delivering the same flow and pressure? The second prototype of a 350HP Roots type blower with a SPT is built and tested. The shaft power measurement employs a Himmelstein torque meter between the motor and the blower. The same test procedure is followed for comparison purpose by first conducting base tests with a conventional serial dampener at discharge. Then the same test setup and instruments are used to test and measure the prototype II equipped with a SPT. Because the prototype is much larger, its speed is lower in the range of 900-2000 RPM.

The test results from the Prototype II show the same trend and the level of pulsation reduction, 10 fold plus, as the prototype I even though the size and speed range are quite different. At the same time, for the same PTC9 performance (pressure rise and flow rate) as the base test, shaft power is dropped considerably, 6-15%. In addition, the prototype size and weight reduction by the SPT is also significant. This proves the point that pulsation dampening by SPT does not have the side effects of static pressure loss and increased package size and weight. In other words, SPT is smaller and more efficient than a serial dampener while achieving the same level of dampening.

4. CONCLUSIONS

The control of large amplitude gas pulsations and the induced NVH have been a continuing challenge for over a hundred years due to its importance to the reliability and efficiency for the widely used internal combustion engines and PD type compressors. Instead of following the path of conventional serial configuration, an alternative parallel method called SPT (shunt pulsation trap), is devised and partially validated with experiments in this paper based on a new pulsation generation theory, the shock tube mechanism. It is concluded:

1. SPT may be feasible for PD compressors to pro-actively tackle the main dynamic source of the transient gas pulsations, Δp_{41} from UC or OC, by trapping and attenuating both pulsating waves and flow, or CW-IFF-EW formation, <u>before</u> discharge phase.
2. Two Roots blower prototypes rated at 75 and 350 HP are tested as a special case of the 100% UC that demonstrates a 20 dB plus pulsation reduction across different load and speed conditions without a serially connected discharge dampener and associated dampener losses. In other words, the test results show that it is feasible to attenuate gas pulsations by a SPT as effective as a serial dampener, 20 plus dB reduction, while being small in size and does not suffer any back pressure losses.

Though tests so far have only been conducted for the special case of a 100% UC, the Roots type blower, it is believed that the principle of the Gas Pulsation Rules and SPT control strategy can also be applied to other types of PD (Positive Displacement) compressors such as piston, screw and scroll, or PD gas machinery such as internal combustion engines, gas expanders, vacuum pumps and superchargers.

Finally, we hope, in our continued search to understand and control gas pulsations, more experiments based on other types of PD compressors incorporating SPT principle can be conducted under both UC and OC conditions. Interested parties may contact the lead author for collaboration details. With more research and development resources devoted by both academia and industry, it is anticipated that future generation of PD compressors and Internal Combustion Engines can be designed to be even simpler in structure, smaller in size and smoother in running than those used today.

NOMENCLATURE

CW	compression wave pulsation component	IFF	induced fluid flow pulsation component
EW	expansion wave pulsation component	OC	over compression

p	gas pressure pulsation
P	absolute gas pressure
PD	positive displacement
Q	inlet flow rate
SPT	shunt pulsation trap
UC	under compression
ΔU	IFF velocity
V	PD compressor volume
W	CW or shock wave velocity
ρ	gas density

Subscripts

1	initial low pressure in shock tube, or in cavity in screw compressor during UC
2	pressure after shockwave in shock tube, pressure after CW in PD compressor
4	initial high pressure in shock tube or in cavity in screw compressor during UC
Cavity	compressor cavity
Inlet	compressor inlet
Outlet	compressor outlet

REFERENCES

Gavric, L., Badie-Cassagnet, A., *Measurement of gas pulsations in discharge and suction lines of refrigerant compressors*, The 2000 International Compressor Engineering Conference at Perdue, p. 627-634, 2000.

Huang, P., *Gas Pulsations: A Shock Tube Mechanism.* The 2012 International Compressor Engineering Conference at Perdue, 2012a.

Huang, P., *Under Compression: An Isochoric or Adiabatic Process?* The 2012 International Compressor Engineering Conference at Perdue, 2012b.

Huang, P., *Rotary Lobe Pump or Vacuum Pump with Shunt Pulsation Trap*, US2011/0300014 A1, 2009.

Huang, P., *Screw Compressor with a Shunt Pulsation Trap*, US2012/171069 A1, 2011.

Huang, P., *Screw Pulsation Generation and Control: A Shock Tube Mechanism.* The 2013 International Compressor Engineering Conference at City University, 2013.

Koai, K., Soedel, W., *Determination of port flow and interpretation of periodic volume source*, The 1990 International Compressor Engineering Conference at Perdue, p. 369-377, 1990.

Mujiü, E., Kovaþeviü, A., Stošiü, N., Smith, I., *The influence of port shape on gas pulsations in a screw compressor discharge chamber*, International Conference on Compressors and their Systems 2007, London, UK, p. 213-223, 2007.

Peters, R., *Damping of low frequency noise in piping systems by means of perforated plates*, International Conference on Compressor and Their Systems, London, p.545-558, 2003.

Price, S., Smith, D., *Sources and Remedies of Hi-Frequency Piping Vibration and Noise*, Proceedings of The 28thTurbomachinery Symposium, p. 189-212, 1999.

Roots, Dresser Industries, Inc., *Noise Control for Rotary Blowers*, Connersville, Indiana, 1980.

Sangfors, B., *Computer simulation of gas-flow noise from twin-screw compressors*, International Conference on Compressor and Their Systems 1999, London, p.707-716, 1999.

Stoecker, W., *Industrial Refrigeration Handbook, 2nd Edition*, McGraw-Hill, p. 125-139, 2004.

Tweten, D., Nored, M., Brun, K., *The Physics of Pulsations*, Gas Machinery Conference, 2008.

Weatherston, L., *Pulse-free blower*, US Patent# 4,215,977, 1984.

Wu, H., Xin, Z., Peng, X., Shu, P., *Simulation of discharge pressure pulsation within twin screw Compressors*, International Conference on Compressors and their Systems 2007, London, UK, 2004.

Yanagisawa, K., Maeda, M., *Screw Compressor*, US Patent# 5,051,077, 1989.

ACKNOWLEDGEMENT

The authors would like to thank Dr. Joost Brasz for his encouragement at the early stage of the project, and whose influence has always proved to be valuable. Most importantly, we would like to thank GE Oil and Gas, whose special grant made the prototype testing possible.