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An experimental investigation on pulsation & noise reduction by shunt pulsation trap with a nozzle for compressors operating in under-compression

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Abstract. This paper is a continuing work from the same authors on the same topic on gas pulsation and noise control using a shunt pulsation trap (SPT) method by Huang [1, 2, 3]. Traditionally, a serial pulsation dampener/muffler, often a reactive type, is connected AFTER the discharge of a positive displacement compressor. It has been demonstrated in the previous experimental investigations that gas pulsations from under-compression (UC) can be as effectively controlled by an alternative scheme - shunt pulsation trap (SPT), which is more compact and tackles the gas pulsations BEFORE the compressor discharge. Two dampening SPT schemes were investigated and compared: using a perforated plate (p-plate) vs. a perforated tube (p-tube). However, the pulsation induced noise is still a challenge with SPT employment alone. The focus of the present paper investigates experimentally the effect of a new dampening scheme - combining a SPT with a nozzle and an integrated absorptive silencer (IAS) to undertake both pulsation and noise problems at source. It is found that an ASME nozzle SPT with IAS scheme has superior noise reduction capability than both the p-tube + IAS scheme and the traditional scheme using premium silencers. Furthermore, the addition of an integrated absorption silencer (IAS) is not suffering as much back pressure drop associated with traditional reaction type silencers and therefore will not affect system efficiency. The integration of absorptive silencer into a nozzle or p-tube based SPT would provide an optimal design choice for size/weight and pulsation/noise reduction and potentials for energy saving for PD compressors operating in UC mode.

1. Introduction

1.1. Dichotomy for PD Compressor Design: Energy Saving or Pulsation/Noise Reduction?

Positive displacement (PD) compressors, especially for screw and Roots, are known for generating high amplitude pulsating pressure (up to 180-200 dB) when operating in the under-compression (UC) mode according to Soedel [4]. Gas pulsation induces vibration, noise and fluctuating flows which in turn demand a silencer or dampener to suppress at discharge, as exemplified in Figure 1. The conventional dampener could lead to considerable energy loss or size increase for the system, a design dichotomy. That is, the growing global demand for higher efficiency and lower noise is conflicting with the current serial dampening configuration or scheme employed for at least 100 years, where more pulsation and noise reduction is always accompanied with more back pressure losses or has to use larger size or multiple dampeners. No current technology could meet size/weight reduction, energy saving and



pulsation/noise reduction demands simultaneously due to the limitation of the current serial noise suppression methodology.

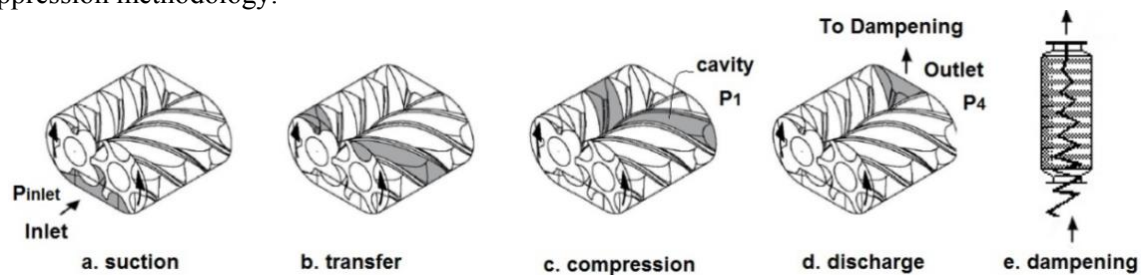


Figure 1(a-e): Screw compressor with a conventional serial dampener at discharge, Huang [3]

1.2. Review of Gas Pulsation and Noise Control by a Shunt Pulsation Trap Method in UC

It has been widely accepted that gas pulsations mainly take place at the discharge side of a positive displacement (PD) type compressor. The pulsation magnitudes, ranging from a fraction to a few bars, say for a screw air compressor, are especially significant at off-design conditions of either an under-compression (UC) or an over-compression (OC) defined as the discharge pressure either lower or higher than the system pressure, as illustrated in Figure 2. UC or OC is inherent for PD compressors with a fixed built-in internal compression ratio (without an automatic discharge valve) whenever there is a mismatch between the design pressure and various system back pressures at different operating conditions [4].

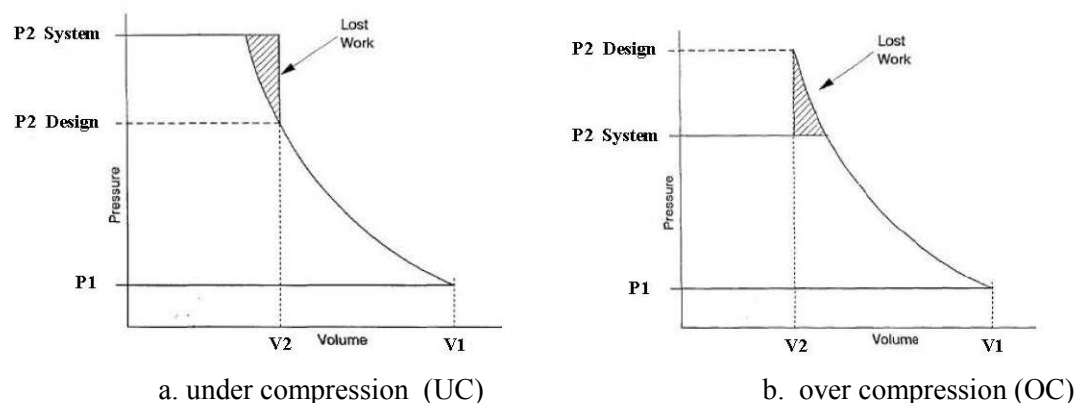


Figure 2: UC or OC of a PD compressor on P-V diagram, Huang [5]

The traditional definition of a UC is a misnomer because even though the gas may be under compressed by internal compression, it is eventually compensated at discharge by a sudden expansion of discharge gas back into the cavity until the cavity pressure is equalized with the system pressure. Hence it is also called backflow compression that demands a silencer or dampener to treat or suppress the side effects from the excitation of the sudden gas expansion process. The pulsation dampener is often a premium type combining both absorptive and reactive means for silencing as illustrated in Figure 3a, connected in series with and AFTER the compressor discharge to suppress both the broadband noise and high amplitude low frequency pulsations. It is capable of reducing pressure pulsation by 10 plus fold and noise by 20 plus dBA. However these dampeners are often bulky and impose sizable back pressure losses.

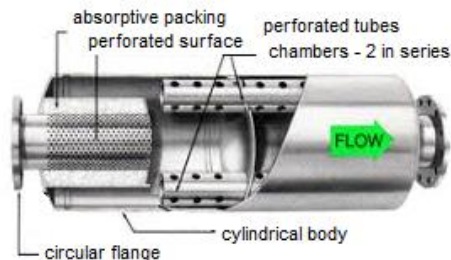


Figure 3a: Example of a combo type silencer with both reactive and absorptive silencing

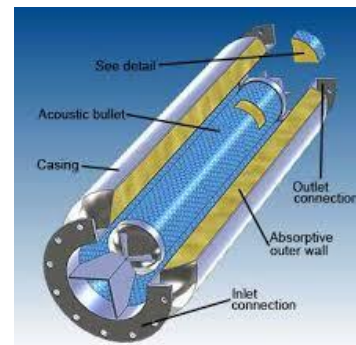


Figure 3b: Example of an absorptive silencer with absorptive material behind

In the previous investigations by Huang [2, 3], a novel method called the shunt pulsation trap (SPT) is studied that uses a parallel dampening configuration to tackle the UC induced gas pulsations BEFORE the compressor discharge. The SPT method is based on a shock tube theory proposed by Huang [1] that characterizes the nature of gas pulsation as a composition of a pair of non-linear compression wave (CW) and expansion wave (EW) plus an induced fluid flow (IFF) directly resulted from the sudden discharging process at UC condition, just like the sudden opening process of a shock tube. The same theory is also explored by Huang [5, 6] to gain insight into the UC, indicating that is inherently an adiabatic (not isochoric) process with compression achieved by a weak shock wave (=CW) at the speed of the wave, while the backflow is simply the induced fluid flow (IFF) driven by CW/EW pair. It also points out that the potential efficiency of an UC could be close to a classical internal compression if the high velocity backflow could be managed properly.

Two prototypes, 75 HP and 350 HP Roots type (100% UC) blowers, were built and tested with and without SPT for validating the concept. The experimental results confirmed that the same 10 plus fold pulsation reduction as the traditional silencers can be achieved by SPT under different load and speed condition. In addition, SPT has a power saving ranging from 5-16% when compared with the traditional serially connected silencers since it does not suffer any back pressure losses.

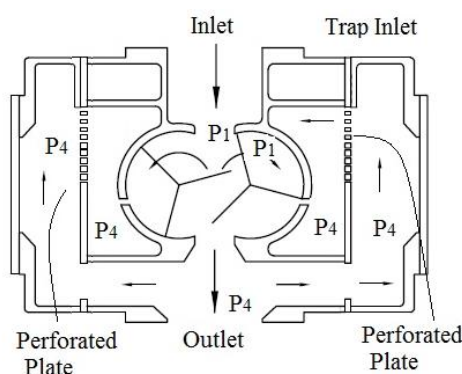


Figure 4a: SPT scheme with perforated-plate, Huang [2]

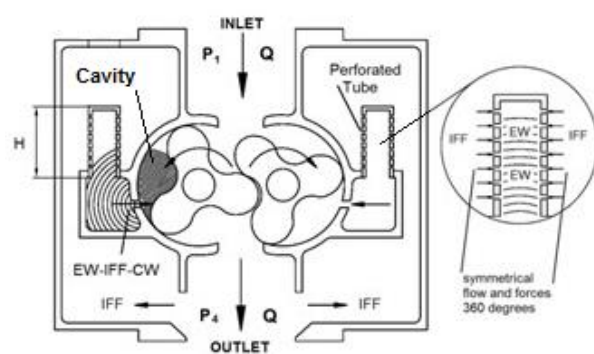


Figure 4b: SPT scheme with perforated-tube, Huang [3]

However, the noise reductions are not satisfactory with the two SPT schemes alone, either using a perforated plate (p-plate) or perforated tubes (p-tubes) as shown in Figures 4a-4b, when compared with traditional scheme using premium combo type silencers (Figure 3a) connected at blower inlet and outlet. To treat the broadband noises while preserving energy saving feature of SPT, it was decided to try an absorption type silencer (as shown in Figure 3b) at discharge for the p-tube SPT scheme. The preliminary results show that it is capable of reducing the noise level by 10-15 dBA while not suffering

the sizable back pressure drop associated with reaction and combo type silencers and therefore will not affect system efficiency. However, integrating a traditional absorptive type silencer with SPT is too bulky in size due to its cylindrical shape and circular flange connections.

1.3. Motive for the Present Research: Pulsation/Noise Control by SPT Nozzle and IAS in UC

It is well known that aeroengine noise problem has been enormously eased by the introduction of the turbofan with the reduced jet velocity: the dominant noise source for turbojet engines because the acoustic power increases as the 6-8th power of the jet velocity according to Beranek [7]. It is observed that the pressure ratio (p_4/p_1) for a Roots blower operating under deep vacuum can be as high as 10/1, which is capable of inducing high velocity jet flows through the trap inlet or the perforated orifices if equipped with SPT as illustrated in Figure 4. The presence of high jet velocity during the transient phase of discharging as demonstrated by Huang [6] warrants that SPT induced secondary broadband noises need to be addressed for compressors operating under severe UC.

On the other hand, for a PD compressor operating in UC mode, a bonus could be reaped for noise reduction and for potential energy saving if the high velocity jet flow (IFF in Figure 4b) was positioned and directed into the compressor cavity and onto the rotor in its rotating direction during the compression phase. In another words, using a nozzle at the trap inlet, either a ASME flow nozzle or de Laval, would isolate and trap the high velocity jet noises inside the cavity before discharging as long as the nozzle throat is choked so that no sound (jet noises inside the cavity in this case) could escape or propagate upstream through the nozzle throat where the jet velocity reaches the speed of sound. On the other hand, the velocity field on the inlet side of the nozzle that is opened to atmosphere or downstream is of much lower velocity, hence the flow induced noises.

2. New Control Scheme = SPT (Nozzle) + IAS

2.1. Introduction to Supersonic and ASME Nozzles

A de Laval nozzle (or convergent-divergent nozzle) was independently developed by German engineer and inventor K rting in 1878 and Swedish inventor de Laval in 1888 for use on a steam turbine. It is a tube that is pinched in the middle, making a carefully balanced, asymmetric hourglass shape. It is used to accelerate a heated and/or pressurized gas passing through it to a higher supersonic speed in the axial direction, by converting the heat and/or pressure energy of the flow into kinetic energy as shown in Figure 5a. It can be seen that both pressure and temperature decrease in flow direction as gas accelerates from subsonic to supersonic velocity.

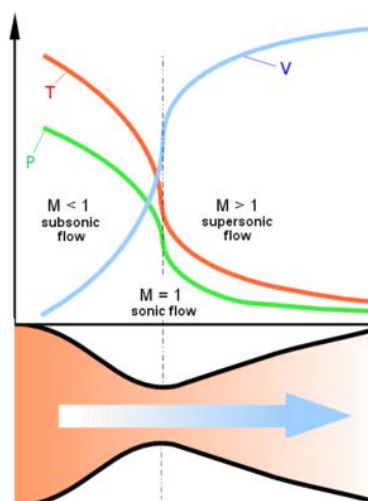


Figure 5a: A supersonic de Laval nozzle with pressure, temperature and velocity distribution. Anderson [8]

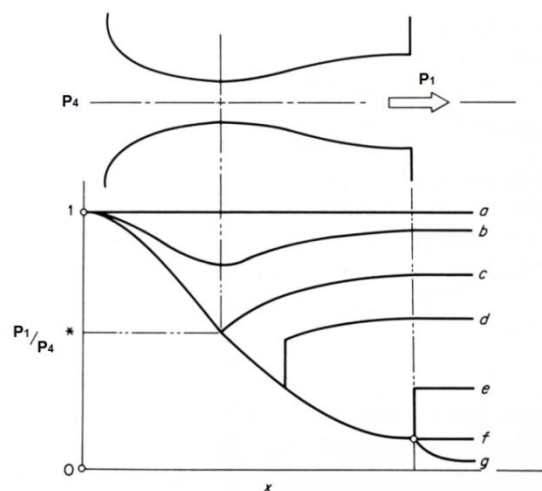


Figure 5b: Pressure distribution for subsonic & supersonic flows under different pressure ratios, Anderson [8]

The pressure ratio (p_1/p_4) across the nozzle determines whether it operates at subsonic or supersonic speeds as illustrated in Figure 5b for pressure distribution under different back pressures ranging from a-g. In a subsonic flow, sound will propagate upstream through the gas. On the other hand for a supersonic flow when the gas velocity reaches sonic (Mach number = 1.0) at the “throat” where the cross-sectional area is at its minimum, a condition called choked flow, a sound wave will not be able to propagate upstream through the gas as viewed in the frame of reference of the nozzle.

The analytical solutions and experimental confirmations for the supersonic nozzle are available more than a century ago, and the results are summarized by Anderson [8] and shown in Figure 5b. It can be seen that the critical pressure ratio (p_1/p_4) needed for gas to reach supersonic is $0.528 = 1/1.89$. A special case of the de Laval nozzle is an ASME flow nozzle when only the convergent section exists without the divergent section, a more compact and economic design. The choked flow is established at the throat as long as the pressure ratio across the nozzle satisfies $(p_4/p_1) \geq 1.89$.

2.2. Pressure Ratios for Roots Type Blowers for Nozzle Applications

It is known that a Roots blower operates at 100% under-compression. The pressure ratio it is capable of developing is governed by the discharge temperature it is able to withstand without suffering a mechanical failure. There are two configurations that a Roots can operate as discussed in detail in [9].

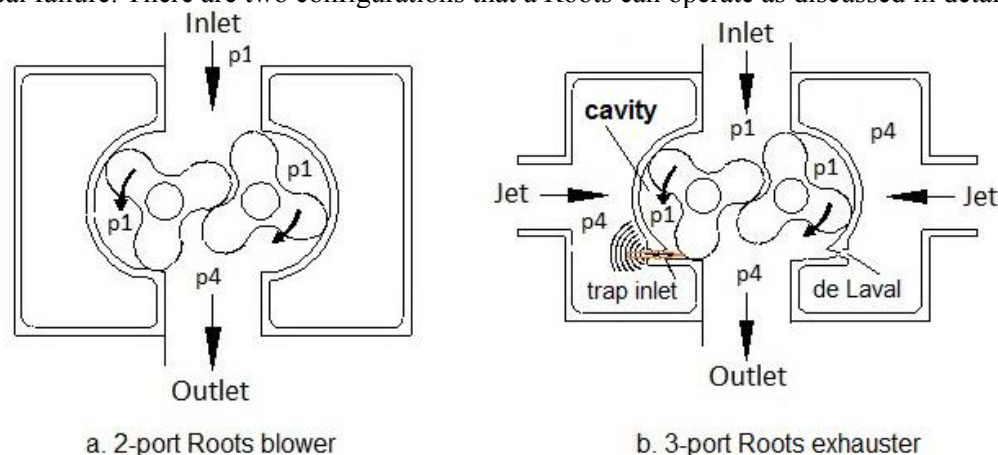


Figure 6(a-b): Examples of 100% UC: two & three port Roots with different compression capability, Huang [9]

Figure 6a shows a typical two-port Roots blower with a discharge to inlet pressure ratio up to 2.5-3.0 while Figure 6b is a typical three-port Roots exhauster operating under inlet vacuum with a discharge to inlet pressure ratio up to 10. The dramatic increase of the pressure ratio for the 3-port is due to the addition of a third port, Jet-Port, that sucks in atmospheric air for cooling, similar to the economizer scheme used for screw compressors in refrigeration per Stoecker [10].

It can be seen that both two-port and three-port Roots possess the maximum pressure ratio (p_4/p_1) to drive a supersonic flow if equipped with a properly designed de Laval nozzle at the trap inlet as illustrated in Figure 6b. However, the pressure ratio (p_4/p_1) will decrease rapidly and transit into subsonic flow during the cavity fill-in process until the pressure ratio across the nozzle is close to 1. On the other hand for an ASME nozzle at the trap inlet, a **choked throat** is established as long as the pressure ratio across the nozzle satisfies $(p_4/p_1) \geq 1.89$.

2.3. A New Pulsation and Noise Reduction Scheme: SPT with Nozzle and IAS in UC

In addition to the primary Δp induced pulsation from UC with low frequency and high amplitude, there are also secondary rotor meshing and interacting pulsations called ΔU induced pulsations that induce broadband noises which need to be addressed by different means. The present paper investigates a new combo dampening scheme - using a SPT nozzle (ASME or de Laval nozzle) to manage the dominant Δp induced discrete pulses in parallel (before discharge) and an integrated absorptive silencer (IAS) to suppress the ΔU induced noises after discharge (in series).

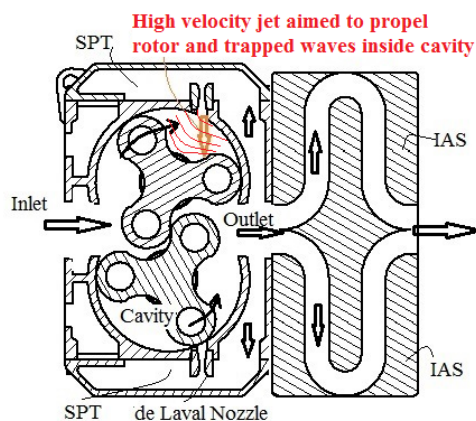


Figure 7: A schematic of a new scheme: SPT-Nozzle with discharge IAS for two-port

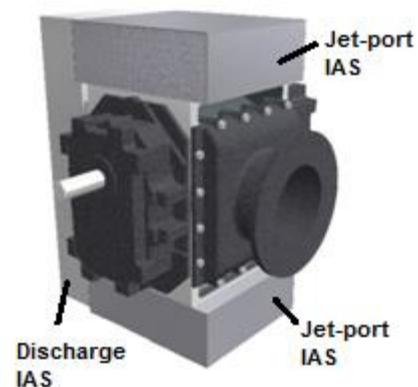


Figure 8: Exterior view of SPT-Nozzle scheme with both Jet-port and discharge IAS for three-port Roots

Figure 7 shows a schematic of the new combo scheme for a 2-port Roots blower as an example of references [11, 12]. There are several advantages associated with this scheme when compared to a p-tube SPT with a serially connected traditional absorptive dampener discussed in [3]. First of all, the higher speed jet flow and induced jet noises are contained (trapped) inside the compressor or blower cavity in such a way that it is closed to both the blower inlet and outlet during the compression phase and unable to propagate upstream through the choked throat because the over-critical pressure ratio ($p_4/p_1 \geq 1.89$) operation occupies most of the filling interval. Secondly, the velocity field on the inlet side of the nozzle that is opened to atmosphere or downstream is of much lower velocity, with lower potential for flow induced noises. Thirdly, due to the higher velocity achieved by using a more efficient nozzle vs. an orifice, the “starved” or under-compressed cavity fill-in time is much faster. This makes it possible for the scheme to work for high speed PD compressors that have seen quantum leaps in rotational speed increases during the last few decades. On the other hand, an IAS in a serpentine shape as shown in Figure 7 would complement the SPT by tackling the wideband noises leaked from nozzle throat when flow velocity is below sonic, or induced noises from rotor meshing or interacting with flows and other flow turbulences. The IAS is much more compact than traditional cylindrical shaped silencer by folding the flow path into a S-shape that further prevent direct wave propagation from the inner core. Figure 8 shows a SPT-Nozzle scheme for a three-port Roots with both Jet-port and discharge IASs. The big question is: will the experiment support the above analysis and predictions?

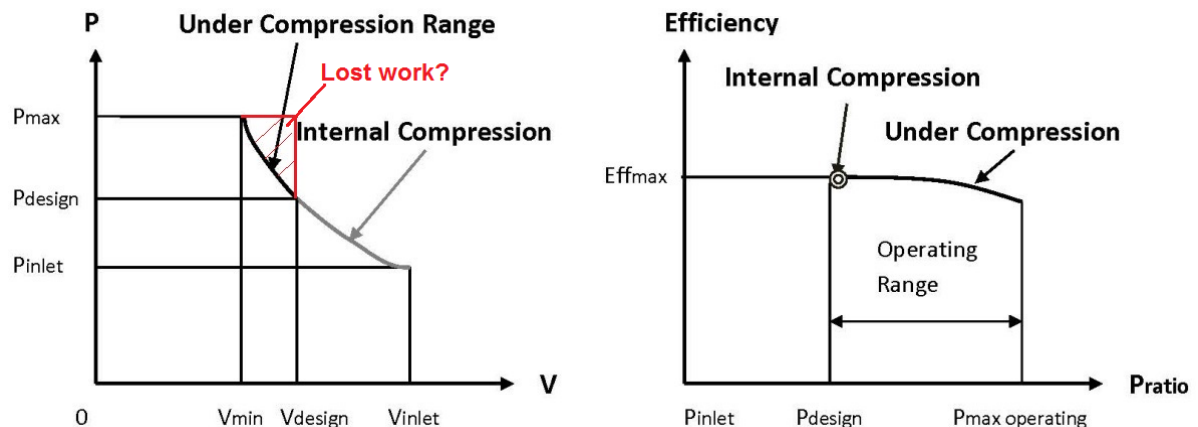


Figure 9: A composite compression process with an initial internal compression followed by a UC, Huang [5]

2.4. A Composite Design Scheme for PD Compressors: Internal Compression Compensated by UC

However, a pre-requisite condition for the above scheme to work for other type of PD compressors is the existence of UC. Fortunately, this is the preferred way for screw or other compressor types that do not use automatic valves at discharge. As recommended by Soedel [4] and Stoecker [10], the best design strategy for PD compressors with a wide range of operating conditions is to have a combined internal compression and UC such that it would never operate under an Over-Compression mode. Figures 9a & 9b show the two segments of this composite compression process with an initial internal compression targeting the minimum operating compression ratio, then joined or compensated by an under-compression for the rest of the operating compression ratios. The RED-shaded area is the “lost” work associated with UC that could be potentially recovered (instead of being wasted as for a traditional design) and is the focus of our future work as to be discussed in the last section of this paper: future work.

3. Experimental Validation

3.1. A subsection

For experimental validation, a 75 HP Roots type exhaustor with the three-port configuration as shown in Figure 6b is tested to exploit its maximum pressure ratio and size/weight saving potential. Two prototypes were built and comparison tested under the same conditions and measured by the same instruments against the base test.

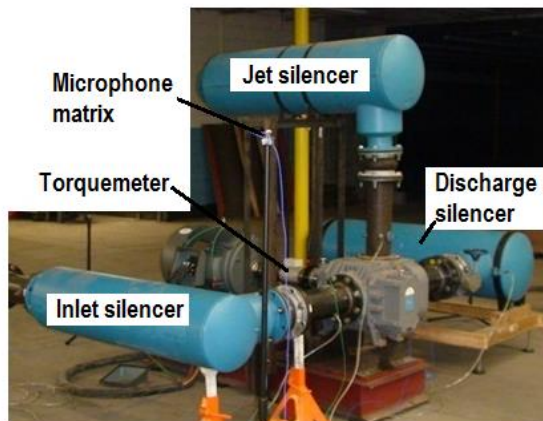


Figure 10: Base test with premium grade combination type silencers at inlet, jet port and discharge

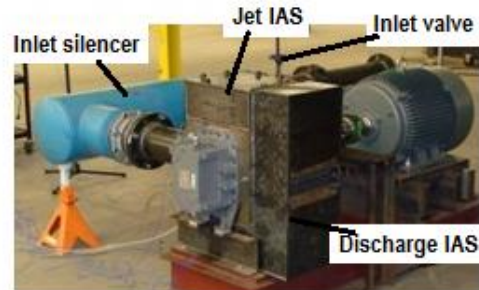


Figure 11: Exterior view of a SPT with IAS at jet port and discharge

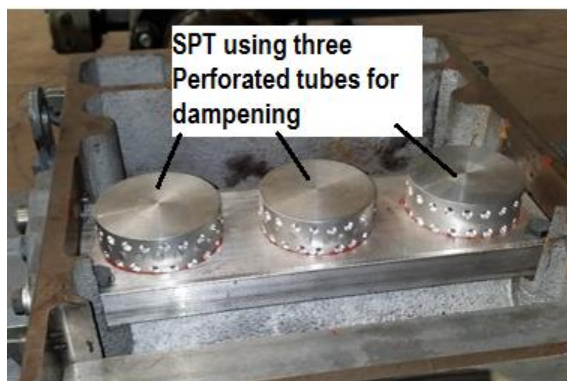


Figure 12: Interior view of a SPT with perforated tubes and IAS (not shown)

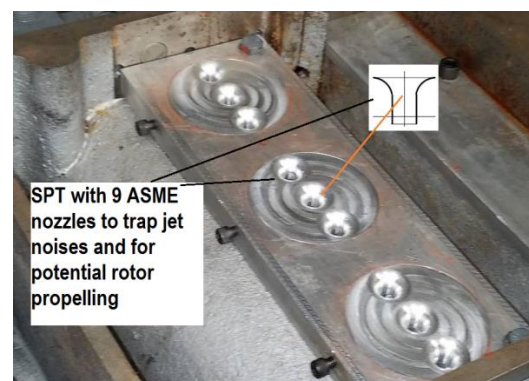


Figure 13: Interior view of a SPT with ASME nozzles and IAS (not shown)

1. Base test: a traditional Roots type exhauster equipped with three premium grade combination type silencers at inlet, jet port and discharge as shown in Figure 10. A torquemeter is shown located between the motor and blower to verify the power savings from getting rid of the back pressure from previous study [2,3];
2. SPT with p-tube + IAS: a 3-lobe Roots type exhauster with a SPT/p-tube and IAS at jet port and discharge, shown in Figures 12 & 11 respectively. Figure 11 has the same inlet silencer as Figure 10 in order to contain the throttling noise from the inlet valve while jet port and discharge silencers are replaced by IAS;
3. SPT with ASME flow nozzle + IAS: the same IAS and 3-lobe Roots exhauster modified with a SPT/nozzle as shown in Figures 13 & 11 respectively. The ASME flow nozzle is firstly investigated due to its lower cost and compact size while possessing the same noise choking capability as a de Laval nozzle.

The test setup and measurement strictly follow the ASME PTC-9 for piping and ISO 2151 for noise. Tests are conducted over usable speed range (2000-4000 rpm) under a deep vacuum with pressure ratio about 6/1 ($\approx 30/(30-25)$) and a typical noise data in dBA is plotted in Figure 14 for all 3 prototypes for comparison.

3.2. Test Results and Discussion

The following results are obtained and compared:

1. On size/weight comparison, Figures 10 & 11 show visibly the size difference is about 3-4 fold between using the IAS and 2 traditional silencers replaced. The weight saving is about the same order. The size/weight/cost saving is even more significant when considering that the blower package is put into a much smaller sound enclosure for further noise reduction;
2. On energy saving comparison, the back pressure measured by a water monometer on discharge IAS is compatible with previously measured pressure loss from a cylindrically shape traditional absorptive silencer: about 5-10% of the traditional reactive or combo type silencers it replaces throughout the operating range, which translate into power saving of about 5-8%, consistent with the torquemeter measurement. Since the present paper focuses on noise reduction, the detailed energy saving data and discussion from getting rid of the back pressure can be found in reference [2, 3];
3. On noise comparison, Figure 14 shows the dBA level for the sound pressure as measured per ISO 2151 around the exhaustor with the SPT/Nozzle+IAS having the lowest noise level. It is no surprise that it is better than SPT/p-tube+IAS scheme because of the use of nozzle that delivers higher speed jet flow inside the cavity while reducing flow velocity outside the nozzle. As the RPM increases, the difference becomes smaller and smaller until all three have about the same dBA level. It is believed that higher RPM induced flow and ΔU become dominant. The base test shows a high peak that could be a resonating frequency that pulsation excites with the traditional silencers used.

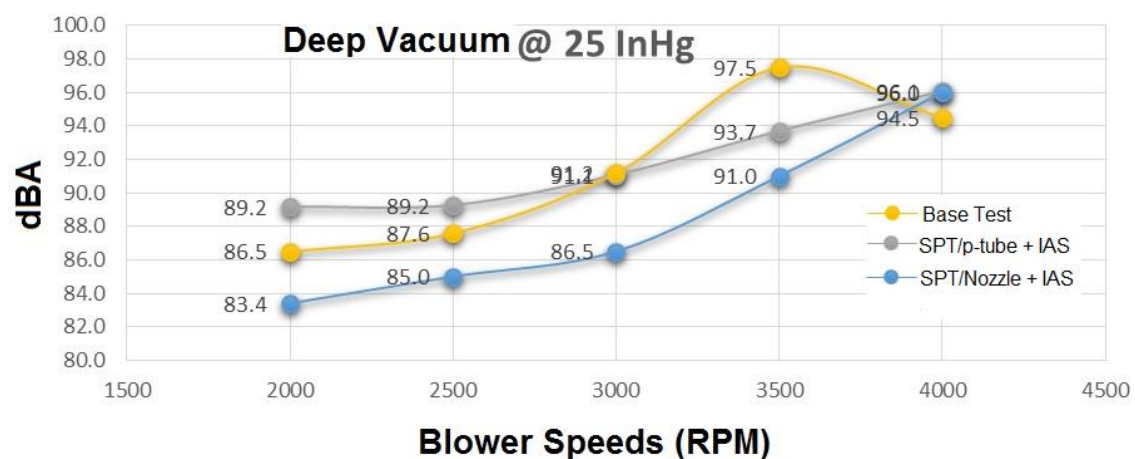


Figure 14: Comparison of noise measurements for different dampening methods

4. Conclusions

Instead of treating both gas pulsations and induced noises the traditional way AFTER the discharge, SPT elects to control UC induced pulsations BEFORE the discharge and broadband noises AFTER the discharge with an integrated absorptive silencer (IAS). This paper investigates experimentally the effect of this new scheme with a novel SPT design that employs a nozzle to manage both gas pulsations and induced noises for the case of 100% UC. The following conclusions are drawn from the analysis:

1. Compared with traditional way employing discharge silencers, the flow nozzle SPT scheme with an IAS have a superior pulsation and noise reduction capability plus a significant size/weight reduction and energy bonus due to the lack of back pressure losses.
2. The flow nozzle SPT scheme with an IAS has a superior pulsation and noise reduction capability than previously investigated SPT/p-tube and SPT/p-plate schemes by directing and trapping the high energy jet flow inside the compressor cavity.
3. Further energy savings by converting the potential energy Δp from UC to do useful work will be investigated in another paper outlined as follows.

5. Future Work

From energy conservation point of view, the traditionally lost work associated with UC, as shown in Figure 9a as the RED shaded area, could be used (instead of being wasted as implied by the widely used word “dampening”) if the potential energy from the high amplitude Δp can be effectively converted into a high velocity jet inside the compressor cavity in such a way as to assist to propel or impulse the rotor in its rotating direction as shown in Figure 7, like a Pelton Wheel. In a conventional serially connected dampening scheme shown in Figure 1, the backflow jet energy associated with the sudden expansion of discharge gas back into the cavity is generally in the direction against the rotor rotating, resulting in doing negative work for compressor system.

Our next step is to focus on the energy recovery aspect by conducting an experimental investigation using a de Laval or ASME nozzle on the above Roots prototype with detailed shaft power measurement to prove that there are indeed energy savings by recovering part of the potential energy Δp from UC. We would further proceed to use the same ASME and de Laval nozzles on a screw compressor incorporating the above proposed composite compression scheme. In our continued efforts to understand the PD compression and pulsation/noise generation mechanism, it is anticipated that better control schemes can be found and applied to future generations of PD compressors that have even more saving in energy consumption, smaller in size and smoother in running.

6. Acknowledgment

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