A model for the transient pulsation generation at the discharge of a screw compressor by a shock tube analogy

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A model for the transient pulsation generation at the discharge of a screw compressor by a shock tube analogy

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Abstract. Although screw compressors (dry and wet) have been widely used in many applications (such as HVAC&R, petrochemical and gas transmission, plant air, etc.) for more than half a century, limited information is available regarding the genesis and mechanism of gas pulsations at pocket passing frequency (PPF) dominating at discharge side of these compressors [1]. The understanding of its physical nature, magnitude and affecting parameters, the location and moment of generation, and the velocity it propagates are all of fundamental, and will help to determine how it will interact with downstream components such as silencers and piping system, especially for accurately predicting downstream pulsation levels and excited vibration and noise at the higher multiples of the PPF.

The primary goal of this paper is to apply the shock tube theory [2] to screw compressors during the transient generation phase of gas pulsations at discharge. It will reveal that the nature of gas pulsation is a combination of large amplitude compression waves (CW) and expansion waves (EW) accompanied with an induced fluid flow (IFF). The most dominant gas pulsation at discharge is directly caused by either an over compression (OC) or a under compression (UC) suddenly discharging to compressor outlet. Therefore its exact location and moment of generation, magnitude, travelling directions and velocity can all be predicted based on design parameters and operating conditions of those machines. With this knowledge, its interactions with the compressor cavity and downstream piping and silencer are conducted for a typical UC and case, and results are in good agreement with the test data of previous researchers. The analysis suggests that the shock tube model can provide a pedagogic tool in understanding the physics of the transient phase of pulsation generation and subsequent formation. As such it can provide valuable insight to developers of more precise CFD calculations and a comprehensive algorithm of gas pulsations in the future.

1. Introduction
1.1 The myth of gas pulsations with screw compressors
During the past 60 years and especially during the last 30 years, screw compressors have become very popular because they combine the advantages of various PD compressors and can provide a wide range of operating conditions. In 1975, screw compressors gained more acceptance after the publication of “API Standard 619 (1st Edition)”. Today, screw compressors are widely used to compress a wide range of gases in many different industries including chemical and petrochemical,
food processing, pulp and paper, power generation, natural and process gas applications, refrigeration and in vapour recovery services [1]. A typical dry screw compressor is shown in Figure 1.

Figure 1. A Typical Dry Twin Screw Compressor per API 619 [1].

Screw compressors (dry and wet) do not have suction or discharge valves like reciprocating compressors. There is a common misconception that the gas is continuously “extruded” from the compressor, like a sausage grinder, without any significant pulsations [1]. Many sales brochures and instruction manuals try to create a myth that suggests that screw compressors do not generate any pulsations and, if they do, the pulsation levels are expected to be very low. Statements that screw compressors do not produce pulsation are in stark contrast to the technical literature. For example, API Standard 619 5th Edition [3] states that “In screw compressor systems, the flow is not steady, but moves through the piping in a series of flow pulses that are superimposed upon the steady (average) flow”. As observed and reported recently [1,4,5,6], the pulsation levels are minimized when the compressors are operating near the design conditions, but can be significantly increased at off design conditions, as much as 10 – 20 percent of the absolute discharge pressure. Moreover, the waveform of the pressure pulsations are first revealed by Fujiwara and Sakurai [7] and later confirmed by Mujic et al [8] as of a general sawtooth/triangular shape as shown in Figure 2.

Figure 2a. Measured pressure pulsation waveform at the screw discharge from Fujiwara et al [7].

Figure 2b. Measured pressure pulsation waveform at the screw discharge from Mujic et al [8].
The high levels of gas pulsations are harmful to the downstream piping system if left uncontrolled and could potentially damage pipe line, in-line equipments, and excite severe vibrations and noises. So API 619 requires pulsations dampeners or silencers be installed at inlet and outlet of compressor. The knowledge of their physical nature, magnitude and frequency under various operating conditions is critical to the design of silencers and downstream system.

### 1.2 Screw compression cycle and causes of gas pulsations

Both wet and dry screw compressors generate pulsations at multiples of the pocket-passing frequency (PPF), which is defined as the number of lobes on the male rotor multiplied by the compressor running speed in Hz. The maximum generated pulsation levels normally occur at 1x PPF and are generally reduced at the higher harmonics.

The pulsation amplitudes generated by screw compressors are affected by many variables as observed by various researchers [1,4,5,6,7,8], such as the shape of discharge port, screw profile, internal clearances and gas type. But the most significant contribution comes from operating under high pressures at off-design conditions of either an under-compression (UC) or over-compression (OC) as shown in Figures 3a and 3c. An UC happens when the pressure at the discharge line (p_{outlet}) is greater than the pressure of the compressed gas within the cavity (p_{cavity}) just before the opening. This would result in a rapid backflow into the cavity, a pulsed gas flow, according to the conventional theory. On the other hand, an OC takes place when the pressure at discharge line (p_{outlet}) is less than the pressure inside the cavity (p_{cavity}), causing a rapid forward flow of the gas into the discharge system. Screw compressors suffer from UC or OC due to the inherent mis-match of the fixed design pressure and ever-changing system pressures and operating conditions. The black and blue shaded areas in Figure 3 are the power losses due to discharge silencer (back pressure loss = p_{outlet} - p_{back}) and from UC or OC mode respectively.

![Figure 3. A P-V Diagram showing different modes of operation and power losses.](image)

Figure 4 shows a flow chart for phases of a traditional compression cycle of a screw compressor system in under compression mode (UC). The inlet volume flow Q_{inlet} goes from suction to compression, and to discharge phase when pulsation is generated as the induced fluid flow (IFF) or backflow in case of UC (This will be explained in detail by Figure 6 in paragraph 2.2), which is then dampened by a dampening device typically connected downstream of the compressor discharge.

![Figure 4. Phases of a traditional screw compression cycle under UC mode.](image)

### 1.3 The state of gas pulsation modelling and the goal of this paper

Naturally, as important matters as gas pulsations, there have been tremendous efforts from both academia and industry made to model the phenomena, especially since the late 1980s by researchers [4,5,6,8]. The approaches are mostly CFD based having different schemes solving non-linear differential equations (DE) and the results can explain many observations for various applications. However, a comprehensive algorithm is still lacking and needs to be developed for accurately predicting the pressure and flow modulation (pulsation) generated by screw
Compressors at different operating conditions, according to [1]. For example, it is difficult to accurately predict the pulsation levels, especially at the higher multiples of the PPF, in the silencer and downstream piping when the modulation produced by the compressor is not accurately known. Many compressor manufacturers do not know the exact pressure and flow modulation (pulsation) generated by the compressor and often use an approximate value, such as 10 – 20 percent of the discharge pressure, when evaluating silencer designs.

Instead of directly tackling DEs, a diversionary approach has been attempted by Huang [2,9] using a shock tube analogy to simulate the transient generation of gas pulsations at the PD compressor discharge under UC or OC at off-design conditions, hence to gain a quantitative understanding of the affecting parameters and its physical nature. This paper will explore more deeply into the shock tube model. The aim is to apply the shock tube model to the screw compressor discharge in order to get a clear physical picture of the transient generation of gas pulsations, such as the exact location and moment of its generation, magnitude, travelling directions and velocity based on the design parameters and operating conditions of screw compressors. Then this information is used to determine or derive how it will interact with a quasi 1-D compressor system consisting of the cavity, silencer and downstream piping system. In other words, this paper intends to correlate and explain how the operating conditions of UC and OC shown in Figure 3 could produce the shape of the waveforms and magnitude of pressure and flow pulsations as measured in Figures 2a and 2b. As such it is expected to provide valuable insight to developers of more comprehensive and precise CFD calculations of gas pulsations in the future.

2. Shock tube mechanism of the transient pulsation generation at the discharge of a screw

Refer to [2,9,10] for more detailed descriptions of the shock tube analogy. However, some main points and figures are taken from these references and used below in order to keep a smooth context flow. According to the transient analysis of a screw compressor discharge by the shock tube analogy, the sudden communication between the cavity and the compressor outlet at different pressures is compared with (analogous to) the diaphragm bursting of the shock tube and would trigger gas pulsation generations in the form of CW-IFF-EW, as illustrated in Figure 5 for the under-compression (UC) and over-compression (OC) modes. The main results of the shock tube analysis can be summarized by the three Transient Gas Pulsation Generation Rules below.

2.1 Rules of the transient gas pulsation generation based on the shock tube mechanism

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

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

$$CW = p_2 - p_1$$

Figure 5. Transient phase of screw compressor discharge by the Shock Tube Analogy [2].
\[ EW = p_4 - p_2 \]  
\[ \Delta U = (p_2 - p_1)/(\rho_1 \times W) \]  

where \( (p_2 - p_1) \) is the over-pressure from CW, \( (p_4 - p_2) \) is the under-pressure from EW, \( \rho_1 \) is the gas density at low pressure region, \( W \) the speed of the shock wave, \( \Delta U \) the velocity of Induced Fluid Flow (IFF);

3. Rule III: Pulsation component CW is the action by the high pressure gas \( p_4 \) to the low pressure gas \( p_1 \) while pulsation component EW is the reaction by the low pressure gas \( p_1 \) to the high pressure gas \( p_4 \) in the opposite direction, and their magnitudes are such that they approximately divide the initial pressure ratio \( p_4/p_1 \) equally into CW and EW [equation (4)]. At the same time, CW and EW induce a unidirectional fluid flow pulsation IFF in the same direction as the CW.

\[ p_2/p_1 = p_4/p_2 = (p_4/p_1)^{1/2} \]  

Rule I implies that there would be no or little pulsations during suction (if there is no inlet valve), transfer and compression (expansion) phases of a screw compression cycle because of the absence of either a pressure difference or sudden opening. Focus instead should be placed upon the discharge phase, especially at the moment of the discharge when the cavity is suddenly opened to outlet line during off-design conditions such as UC or OC.

Rule II indicates specifically the moment of gas pulsation generation as the instant of the communication between gas \( p_1 \) and gas \( p_4 \), and the location as the discharge port opening. Moreover, it defines two sufficient conditions for gas pulsation generation:

a) The existence of a pressure difference \( \Delta p_{41} \) which is a measure of how severe the UC or OC is;

b) The sudden opening of the divider separating the pressure difference, a measure of the effect of rotor speed.

Rule II also reveals the composition and magnitudes of gas pulsations at onset 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 with changing wave form during propagation. This is in direct contrast to the acoustic waves that are linear and do not induce a mean through flow. It is important to note the wholeness of three pulsation components (CW, EW and IFF) that are generated simultaneously and one cannot be produced without the others.

Rule III shows further the interactions between two gases of different pressures are mutual so that for every CW pulsation component, there is always an equal but opposite EW pulsation component in terms of pressure ratio [equation (4)]. Working together, they induce a unidirectional fluid flow pulsation (IFF) component in the same direction as the compression waves (CW). Moreover, Rule III quantitatively correlates, by equation (4), the post-discharge opening induced gas pulsations (\( \Delta p_{21}, \Delta p_{42} \)) with the pre-opening pressure difference \( \Delta p_{41} \) between the compressor cavity and discharge line.

2.2 Application of the transient gas pulsation rules to screw compressor discharge

Now let’s take a new look at the traditional compression cycle of a screw compressor operating at under compression mode (UC) in light of the Shock Tube Theory. Figure 6 illustrates the same process shown in Figure 4 again in time domain, emphasizing the events during the transient backflow phase of discharge.

The transient phase starts when the compressor cavity suddenly opens to the outlet line to expose the pressure difference between the design pressure in the compressor cavity \( P_{\text{cavity}} \) and the outlet pressure \( P_{\text{outlet}} \), i.e. \( \Delta P_{\text{UC}} = \Delta P_{41} \). According to the transient analysis [2,10] of the shock tube analogy, the sudden communication between the cavity and the outlet would trigger the generation of gas pulsations in the form of CW-IFF-EW, which originate at the discharge port as shown in Figure 5a. CW would go into the cavity to wave compress the cavity gas Q (traditionally called backflow
compression) and induce IFF, while EW comes out to the outlet pulling IFF and causes gas pulsating downstream. The magnitude of CW and EW is approximately equal by dividing the $\Delta P_{UC} (= \Delta P_{AI})$ in terms of pressure ratio. Since the waves (both CW and EW) travel much faster (10-20 times) than the screw rotor, the time it takes to wave compress and induce IFF is much shorter than the traditional rotor driven compression and discharge process (which starts as the cavity is closed to the inlet and ends until all gases are emptied into the outlet) as shown by $\Delta t_{UC}$ and the sharp rise of pressure $\Delta P_{UC}$ in Figure 6b. It should be emphasized that it is in the transient phase that the pressure in the cavity gets equalized with the outlet by incoming CW and volume flow IFF while simultaneously the induced downstream pulsating gas flow IFF and EW are dampened by the silencer if there is one. Though the transient time is short, it is finite and can be calculated approximately by

$$\Delta t_{UC} = \frac{2L_c}{W_{CW}}$$  \hspace{1cm} (5)  \\
$$\Delta t_{OC} = \frac{2L_c}{W_{EW}}$$  \hspace{1cm} (6)

where $L_c$ is a characteristic length of the cavity at the moment of the discharge opening that CW or EW has to reach to be reflected and $W_{CW}$ or $W_{EW}$ is the wave velocity corresponding to CW and EW. Factor 2 means a round trip (incident and reflective waves) is needed to wave compress the cavity gas $Q$ to diminish the initial $\Delta P_{UC}$ or $\Delta P_{OC}$ to less than 5% [2,10]. For an industrial size air compressor, the magnitude of $\Delta t_{UC}$ is typically less than a fraction of one 1/1000 s as shown in Table 2 below.

![Figure 6. Phases of a screw compression cycle for traditional serial dampening under UC mode.](image)

After the transient backflow phase, the combined cavity volume flow and back flow, $Q+IFF$, is discharged into the discharge line by the positive movement of the rotors that takes much longer time as marked as discharge phase in Figure 6b. $Q+IFF$ would go into a downstream silencer (second time for IFF) to be further dampened. The silencer type used is mostly a reactive or combo type sometimes with an orifice (called Kotter plate) plate at the silencer inlet as shown in Figure 7.

![Figure 7: Example of a discharge system with silencer and Kotter plate (orifice plate)](image)
Figure 8 shows the phases of the screw compression cycle under an OC mode. Note that for OC mode, the IFF goes forward through the damper during the transient phase before the cavity flow Q, while Q flows through the same silencer after IFF passes. It is different from the UC mode in that IFF only goes through the silencer once instead of twice for UC mode.

2.3 The non-linearity and nature of gas pulsations of screw compressors
To get a feel for the magnitudes of gas pulsations as non-linear waves that induces strong fluid flow in stark contrast to the linear acoustic waves, Table 1 lists a practical range of fluctuating pressures, from 0.00002 – 2 bar (0.0003 - 30 psi) or equivalent to 100-200 dB, for comparison. The high end (2 bar or 30 psi or 200 dB) represent a typical pressure pulsations value for most of the industrial screw compressors while the low end (00002 bar or 0.0003 psi or 100 dB) represent a typical pressure fluctuation for high sound. It will be shown that, because of the orders of magnitude differences, gas pulsations and sound (linear acoustic waves) behave very differently, hence the way to model them should differ too.

Table 1. Magnitude of acoustic waves in comparison with industrial gas pulsations [2].

|                         | Sound   | Gas Pulsations |
|-------------------------|---------|----------------|
| Pressure Fluctuation,   | 0.00002 | 0.0002         |
| bar                     | 0.002   | 0.2            |
| Pressure Fluctuation,   | 0.0003  | 0.03           |
| psi                     | 0.3     | 3              |
| Sound Pressure level,   | 100     | 140            |
| dB                      | 180     | 200            |

From Pulsation Rules, the pulsation component of the compression waves (CW= p2 – p1) can be expressed by Equation (3), also known as the Shock Wave Equation, relates the abruptly arisen pressure of a shock wave with strength, ΔP21, propagating in air of density, ρ, in shockwave speed of W, and inducing a flow velocity of ΔU. When the magnitude of ΔP becomes smaller and smaller, Equation (3) is reduced to Equation (7) as follows,

\[ dp = \rho a \, du \]  

which is the classical acoustic equation, representing small amplitude pressure wave front, dp, propagating in air of density, ρ, at the speed of sound, a, and inducing an air velocity of du, (Capital P and U are used to indicate their larger magnitude). To get a sense of their quantitative differences, Table 2 lists the induced fluid velocity and other variables by the same range of pressure fluctuations listed in Table 1 for air propagating one-dimensionally in a shock tube. An on-line WiSTL Shock Tube Calculator [11] is used, from known W, to calculate the values of ΔP41 (=p4−p1), ΔP21 (=p2−p1),
speed of sound, speed of CW and EW, ΔU (IFF) under the following initial conditions: for air with the same initial temperature of 300 K on both sides and low pressure of 1.0 bar for a diaphragm-triggered shock tube. The transient time is calculated from wave velocity of CW & EW by Equation (5) assuming Lc=0.05 m. The bottom two rows of Table 2 list the calculated pressure ratios for both CW and EW gas pulsations to show the non-linearity and for validating Equation (4) in the pressure range for typical engineering applications.

Table 2. Comparison of behaviours between linear sound and non-linear gas pulsations.

|                              | Acoustic Waves | Gas Pulsations |
|------------------------------|----------------|----------------|
| CW Strength, Δp_{21}, bar    | 0.00002        | 0.2            |
| EW Strength, Δp_{42}, bar    | 0.00002        | 0.3            |
| UC or OC, Δp_{41}, bar       | 0.00004        | 0.5            |
| CW speed in Mach Number      | 1.000009       | 1.09           |
| IFF, AU, m/s                 | 0.0052         | 47             |
| Speed of sound (a_1), m/s    | 347            | 347            |
| Speed of CW, m/s             | 347.4          | 348            |
| Speed of EW (a_2), m/s       | 347.4          | 348            |
| Temp behind CW (T_2), K      | 300            | 302            |
| Transient time, Δt_{UC}, mSec| 0.288          | 0.285          |
| Transient time, Δt_{OC}, mSec| 0.288          | 0.280          |
| p_{4}/p_{1}, CW              | 1.00002        | 1.02           |
| p_{4}/p_{2}, EW              | 1.00002        | 1.02           |

It can be seen that below the fluctuating pressure of 160 dB (in RED in Table 2) in the domain of Classic Acoustics, CW strength and velocity are almost the same as EW, the IFF velocity ΔU is very low (< 0.52 m/s) and is linearly proportional to the strength of CW and EW. On the other hand, above 160 dB in the domain of gas pulsations, CW strength and velocity are very different from EW (CW is always faster than EW), and the IFF velocity ΔU is very high (302 m/s at 200 dB), much higher than the average flow velocity of screw compressors. This embodies the non-linearity or nature of the gas pulsations that the transient effect is dominant and demands an aerodynamic approach, such as shock tube theory or characteristics method [2,5,6] rather than traditional Thermodynamic approach.

The transient times in Table 2 also show that the time it takes to wave compress and induce IFF is much shorter than the time it takes to rotor compress (also called internal compression) the gas. In other words, transient events are driven by CW/EW waves as dynamic forces while internal traditional compression cycle is driven by rotors which are typically much slower. For example, for a screw running at high RPM of 15000 with a 4x6 lobe combination, the pocket passing time is 1 mSec (=1/(15000x4/60)), > 0.175 mSec shown in Table 2.

3. Dynamic interactions of discharge pulsation – a quasi 1-d model

From the above analysis, we obtain the needed knowledge of gas pulsations at its generation source (= location of discharge valve opening): its physical nature and the magnitude, the location and moment of generation, and the velocity it propagates both upstream and downstream. Now we are going to use this information to determine or derive how it will interact with compressor cavity and downstream components such as piping system, silencers as shown in Figure 7 in order to produce the shape of the waveforms and magnitude of pressure and flow pulsations. The results are then compared with measurement data by various researchers as shown in Figures 2a and 2b.

3.1 The making of the discharge gas pulsation in a free downstream system – model Y

The simplest case of the discharge system for the pulsation making process is modelled as the following as illustrated in Figure 9a for UC mode (ΔP_{UC} = Δp_{41}): the process starts with the sudden
valve opening at point A as the compressor cavity with low gas pressure $p_1$ is opened to the downstream piping with high gas pressure $p_4$. There isn’t any obstruction downstream. The cavity displacement (modelled by a piston moving always to the right shown as yellow in Figure 9a) velocity $U_0$ and volume are assumed to be much smaller than the speed of sound of the gas at discharge ($U_0 << a$) and the downstream volume ($L_c << L_d$). A pressure tab is installed at point B downstream to sense the local pressure and flow variations within one period (= t\_cycle\_ from the moment of one valve opening to the next opening).

According to the Transient Pulsation Generation Rule II, the moment of the discharge valve opening (DVO) triggers the generation of gas pulsations in the form of CW-IFF-EW at the point A as shown in Figures 9a-9b at $t=0^+$. A fan of the incident CW (ICW) would sweep into the cavity at the speed of wave $W_{ICW}$ to compress the cavity gas from pressure $p_1$ to $p_2$ and then it is reflected at $t=t_1$ upon reaching the end wall (where $U=U_0$, displacement velocity), producing a reflective CW (RCW) that further compress the gas from pressure $p_2$ to $p_4$. While the generated fan of the incident EW (IEW) sweeps downstream at the speed of wave $W_{IEW}$ causing a backflow into the cavity at velocity $\Delta U (=U_0-U_2)$ and reducing the downstream pressure to $p_2$. Both downstream going IEW and RCW are then sensed by the pressure tab at point B at the moments of $t=t_2$ and $t=t_3$ respectively, as shown in Figures 9a-9b-9c during $t= 0$ to $t_3$. The RCW would follow IEW travelling downstream, increasing the downstream pressure back to $p_4$ and stalling the flow to velocity $U=U_0$ as sensed by the pressure tab at point B at. For the purpose of analyzing and comparing with experimental data, the pressure and flow velocity history at point B is also expressed in time domain, as illustrated in Figure 9d, where a single pressure pulse produced by the combined actions of the IEW and RCW waves induces a single flow pulse in sync.

The time for fast moving waves to travel from $0 \rightarrow 1 \rightarrow 2 \rightarrow 3$ in model-Y can be calculated by the following formula:

$$ t_1 = \frac{L_c}{W_{IEW}} $$  \hspace{1cm} (8)

$$ t_2 = \frac{L_d}{W_{ICW}} $$  \hspace{1cm} (9)

$$ t_3 = t_1 + \frac{(L_c+L_d)}{W_{RCW}} $$  \hspace{1cm} (10)

The same discharge system (Model Y) can be also modeled for an OC mode ($\Delta P_{OC} = \Delta p_{41}$), as illustrated in Figure 10a where the gas pressure in cavity $p_4$ is higher than the downstream piping pressure $p_1$. Using the same Transient Pulsation Generation Rule II, a fan of the incident EW (IEW) would sweep into the cavity at the speed of wave $W_{IEW}$ to expand the cavity gas from pressure $p_4$ to $p_2$ and then it is reflected at $t=t_1$ upon reaching the end wall (where $U=U_0$), producing a reflective EW
(REW) that further expand the gas from pressure $p_2$ to $p_1$. While the generated fan of the incident CW (ICW) sweeps downstream at the speed of wave $W_{ICW}$ causing a forward flow out of the cavity at velocity $\Delta U = (U_2 + U_0)$ and increasing the downstream pressure to $p_2$. Both downstream going ICW and REW are sensed by the pressure tab at point B at the moments of $t=t_2$ and $t=t_3$ respectively, as shown in Figures 10a-10b-10c during $t=0$ to $t_3$. The REW follows ICW traveling downstream, decreasing the downstream pressure back to $p_1$ and stopping the flow to velocity $U=U_0$ as sensed by the pressure tab at point B. The pressure and flow velocity history at point B is also expressed in time domain, as illustrated in Figure 10d, where a single pressure pulse produced by the combined actions of the ICW and REW waves induces a single flow pulse in sync.

Figure 10. The making of the discharge gas pulsation for Model Y - OC mode.

3.2 The making of the discharge gas pulsation in an obstructed downstream – model Z

Figure 11. The making of the discharge gas pulsation for Model Z - OC mode.
The Model-Y assumption that there isn’t any obstruction downstream of the compressor discharge isn’t always true. Figure 7 shows an example of the reality with a downstream Kotter plate followed by a silencer. Kotter plate is basically a perforated plate that is often used to dampen the gas pulsations by blocking (or reflecting) a portion of the waves going downstream and transforming low frequency flow pulse into multiple high speed jets with a wide range of higher frequency noises. The selection from a wide range of the open-to-close area ratio determines the degree of the wave reflection and jet velocity. The silencer behind is often a reactive or combo type that can be regarded as a second stage dampening equipped with perforated tubes, reflective chambers and absorptive materials tackling both low and high frequency noises at the same time. For simplicity, only a Kotter plate (K-plate) is incorporated into model-Z, as shown in Figure 11a for the OC mode while all other assumptions for model-Y remain the same.

For Model-Z, the gas pulsation interaction with the compressor cavity and downstream K-plate has the same process during t=0-t2 when the ICW passes through the point B and before hitting the K-plate as shown in Figures 11a-11b-11c. Because of the presence of K-plate, portion (say 50%) of the ICW is reflected at t=t3 upon reaching the plate. Due to the obstruction of the K-plate, the induced flow is slowed down (say by half) at K-plate, producing a weakened reflective ICW (RICW) represented by the dashed wave fronts in Figure 11a that further compress the gas from pressure p2 to p3 and slow down the induced flow behind the wave front before colliding with the rightward going REW. The wave collision (refer to [12] for details) would strengthen the rightward moving REW (due to slowed flow by RICW) while weaken the leftward going RICW, as shown in Figures 11a-11b-11c during t= t3-t4. The resulting pressure and flow velocity history sensed by the pressure tab at point B can be illustrated in time domain in Figure 11d.

3.3 Features of the dynamic process of gas pulsation making
Though Model-Y is a very simple case, it demonstrates some of the fundamental features of a dynamic gas pulsation making process for a typical screw discharge system. First of all, the source of ICW and
IEWS is simply the re-distribution of the pre-opening UC pressure difference \( \Delta p_{41} \) that is being suddenly released at the discharge opening at point A and split into a pair of moving forces as IEW and ICW that are recombined (IEW and RCW) and teamed together to move and stop the flow when they sweep through downstream, resulting in a pulse flow (IFF). Secondly, the synchronization of the pressure and flow pulses in Figure 9d illustrates the dynamics of the transient production of the gas pulsation with the wave fronts IEW and RCW acting as the motive forces driving and stalling the instantly induced flow IFF between two wave fronts when they sweep through in the form of IEW-IFF-RCW. Thirdly, the repeat of the dynamic cycle shown in Figure 9d will produce a train of the pressure and flow pulses that ride on a main flow moving downstream at average velocity of \( U_0 \) (rotor displacement velocity) with the ICW and IEW pair shuttling through the main flow at the speed of waves to maintain the pulsed flow energized. This is in direct contrast to a steady state pipe flow where a static pressure difference stationary to the wall (that is, the driving force is stationary) is driving a continuous flow downstream. The last but not the least insight is that the transient gas pulsation event after initiation would be very difficult to control because there are three fast moving targets (CW, IFF, EW) bouncing around in different directions at high velocities that have to be dealt with.

3.4 Magnitude and waveform of the gas pulsation in comparison with previous investigators

From the above analysis and the Transient Pulsation Generation Rule III, it is shown that the magnitude of gas pulsations at the source is directly proportional to the pressure offset between the compressor cavity and outlet as expressed by the UC or OC or mathematically related to \( \Delta p_{41} \) by Equations (1) to (3). While the measured gas pulsations downstream of the discharge are related to its source value, they are affected by a few other design parameters nevertheless. In other words, the split of \( \Delta p_{41} \) into three different components at the source, CW-IFF-EW, so that each will interact with downstream system or each other by the ways of reflecting, inducing, colliding or merging, resulting in different types of gas pulsations in magnitude and waveform. For example, depending on whether the operating discharge conditions is a UC or OC, the split of the same magnitude \( \Delta p_{41} \) into CW and EW could have different induced flow directions as shown by the difference in Figures 9 and 10. Or the degree of the downstream obstruction determines whether the original CW and EW will be passed downstream or partially reflected to further strengthen the pulse amplitude, as demonstrated by the different magnitude between Figures 10 and 11 for the same OC mode. Another important parameter (not discussed in this paper) is the area expansion ratio from cavity to outlet that will weaken the strength of gas pulsations from the source to downstream pipeline. This is the reason the above models are called Quasi-1D as it can incorporate cross area changes when waves or flow travelling downstream. For a typical industrial screw with a “discharge port to outlet pipe” area expansion ratio around 3-5 times, plus the split of \( \Delta p_{41} \) into CW and EW, the downstream pulsation level is reduced 6-10 times from its generation source, and is roughly in the range of 10-15% of the source strength defined by \( \Delta p_{41} \).

![Figure 13. The pulses of the discharge gas pulsation for Model Z at higher frequency.](image)
The geometrical parameters, such as locations of the pressure tab and K-plate, will affect the shape of the pulses while the speed of the compressor decides the duration of the cycle period or pulse frequency in relation to the pulse shape. For example, when compressor speed is increased so that more pulses are produced per period of time, the waveforms shown in Figures 11d and 12d for Model-Z will become pulses shown in Figures 13a and 13b for UC and OC modes respectively. It is interesting to compare the derived waveforms shown in Figures 13a and 13b with the measured waveforms in Figures 2a and 2b. The predicted waveform is basically a saw-tooth shape, same as those from both Fujiware [7] and Mujic [8]. However, we know now from the above analysis that the rising side of the pressure pulse corresponds to a CW action with a steeper slope (faster velocity) while the falling side of the pressure pulse corresponds to an EW action with a gentle slope when these waves sweep by the fluid at the pressure tab location at the speed of waves. There are also pressure ripples on different sides of the pulse from both the prediction and Mujic results, which can be seen as the secondary interactions such as reflection, colliding/merging from various boundaries inside the discharge system under UC or OC mode. However, the 1D assumption neglects a lot of influences from the reality and can be improved by the more precise CFD modelling in the future.

4. Conclusions and recommendations for future work

The use of sections to divide the text of the paper is optional and left as a decision for the author. Where the author wishes to divide the paper into sections the formatting shown in table 2 should be used.

4.1 Conclusions

The understanding of the physics and mechanism for the generation of gas pulsations and subsequent interactions with compressor system is fundamental and will help to provide valuable insight to developers of more precise CFD calculations and a comprehensive algorithm to predict gas pulsation and excited vibration and noise at the higher multiples of the PPF. Instead of using traditional thermodynamic approaches, this paper applies the transient aerodynamics from the shock tube theory to the transient discharge phase of the screw compressors to derive various scenarios of gas pulsations and compare them with the results of other researchers. It is concluded:

1. The most dominant source of gas pulsation at the screw discharge is directly caused by either an over compression (OC) or a under compression (UC) suddenly opening to compressor outlet, resulting in a combination of large amplitude compression waves (CW) and expansion waves (EW) accompanied with an induced fluid flow (IFF) in an inseparable formation of CW-IFF-EW.
2. The making of the discharge gas pulsation are demonstrated from the moment that the split of $\Delta p_{i}$ into three different components at the source, CW-IFF-EW, through its formation process in which each component will interact with downstream system or each other by ways of reflecting, inducing, colliding or merging, resulting in different types of gas pulsations as observed downstream. These dynamically formed pressure and flow pulsations are comparable with measurement data by previous researchers with good agreement regarding the shape of the waveforms and order of magnitude.
3. The pressure and flow pulses are always in sync which illustrates the dynamics (Newton’s Law of Motion) of the transient production of the gas pulsation with the wave fronts EW and CW acting as the motive forces driving and stalling the instantly induced flow IFF between two wave fronts when they sweep through in the form of EW-IFF-CW.
4. The methodology of the transient analysis with the shock tube analogy provides a pedagogic tool in understanding the physics of the transient phase of gas pulsation generation and subsequent formation downstream. As such it can provide valuable insight to developers of more precise CFD calculations and a comprehensive algorithm of gas pulsation predictions in the future.
4.2 Future work

The most important takeaway of this paper is the realization that the transient gas pulsation event after initiation would be very difficult to control because there are three fast moving targets (CW, IFF, EW) bouncing around in different directions and at high velocities within the system. Based on this insight, a new pulsation control method called Shunt Pulsation Trap or SPT [13,14] is proposed to target and take advantages of the fast moving waves in a parallel configuration that is advantageous to the traditional serial dampening schemes as analyzed in [15]. It is hoped that the principle of the SPT control strategy can be applied to the screw compressor design so that future generations of screw compressors can be simpler in structure, smaller in size and smoother in running than those of today.

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**Notation**

- **a**: speed of sound
- **CW**: compression wave, pulsation component
- **DVC**: discharge valve closed
- **DVO**: discharge valve open
- **EW**: expansion wave, pulsation component
- **IFF**: induced fluid flow, pulsation component
- **IEW**: incident EW
- **ICW**: incident CW
- **L**: length
- **OC**: over compression
- **ρ**: gas density
- **p**: gas pulsation pressure
- **Δp_{41}**: gas pulsation source or measure of OC/UC
- **P**: absolute gas pressure
- **PD**: positive displacement
- **Q**: volume flow rate
- **REW**: reflected EW
- **RCW**: reflected CW
- **SPT**: shunt pulsation trap
- **t**: time
- **UC**: under compression
- **U**: fluid flow velocity
- **ΔU**: IFF velocity
- **V**: PD compressor volume
- **W**: wave velocity

**Subscripts**

- **1**: initial low pressure in shock tube
- **2, 3**: post opening pressures in shock tube
- **4**: initial high pressure in shock tube
- **A, B…**: locations
- **Inlet**: compressor inlet
- **Outlet**: compressor outlet