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Under-Compression (Over-Expansion) – An Isochoric Or Adiabatic Process?

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ABSTRACT

It has long been established that for a Positive Displacement (PD) compressor or expander such as reciprocating or rotary screw type, the thermodynamic process is ***adiabatic*** when compressor or expander discharge pressure is equal to system back pressure (or 100% internal compression or expansion). At this design point, compressor or expander efficiency and noise are most desirable. But more often, PD compressor or expander operates at off-design points where the discharge pressure is either lower or higher than the system back pressure caused by the inherent nature of possessing a fixed built-in volume ratio. The resulting processes are often called an Under-Compression abbreviated as UC (or Over-Expansion, OE) and an Over-Compression, OC (or Under-Expansion, UE), and the thermodynamic process suddenly changes to ***iso-choric*** (constant volume). The compressor or expander efficiency and noise become worse at these off-design conditions and some type of controls are always desired such as a variable geometry in order to minimize the losses.

On the other hand, there have been test observations that seem to contradict the conventional theory, pointing to the dramatic difference of the thermodynamic processes between an UC and internal compression. So questions arise: What is the true thermodynamic process of an UC? What really does the compression at the instant when the gas is exposed to higher system pressure? How is energy exchanged during an UC cycle? And why does UC possess the unique self-adjusting capability to different system pressures?

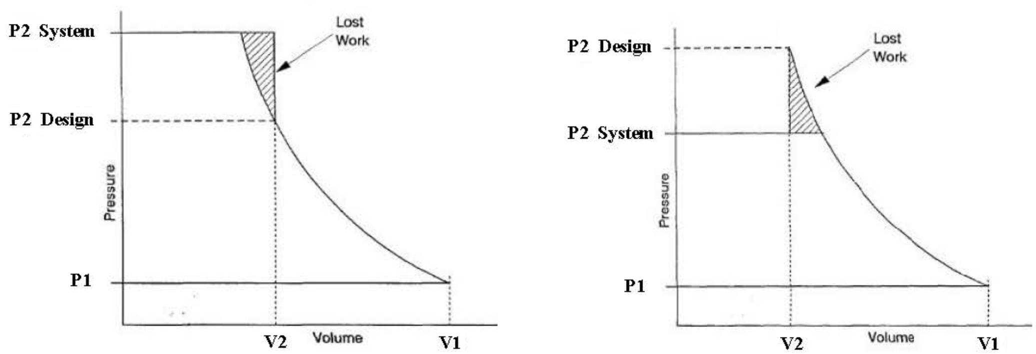
This paper attempts to re-exam these questions by applying the 1st Law of Thermodynamics to UC (or OE) mode. It will be demonstrated that an UC (or OE) may inherently be an adiabatic process with compression achieved by a "flexible" backflow rather than a "rigid" piston or lobe. The energy exchange, in addition to work input from shaft to gas, is assisted by a dynamic process of "borrow and return" with the discharge system. It will be further explored that the mechanism of backflow compression is in essence a wave compression as illustrated by the Shock Tube Theory. Potential efficiency of an UC or OE could be close to the classical internal compression if the high velocity backflow is managed properly. But the best or maybe the most overlooked property seems to be its "feedback" capability, that is, an UC is a self-correcting, negative feedback control loop capable of meeting different system back pressures without using any variable geometry. A Roots type blower is used as an extreme example of UC to illustrate the new theory.

1. INTRODUCTION

1.1 PD Under Compression (UC) and Roots Blower

All PD compressors or expanders possess a fixed built-in volume ratio or internal compression ratio. Whenever there is a mismatch between design pressure and system back pressure, the thermodynamic process changes suddenly from adiabatic to isochoric as represented by a vertical rise or fall of pressure on P-V diagram. Under these conditions, often termed Under Compression or UC (Over-Expansion or OE) and Over Compression or OC (or Under-Expansion, UE) as illustrated in Figure 1, the compressor or expander efficiency and noise become worse than the design point as indicated by consuming additional work (horn areas) and some type of controls are desired such as a variable geometry design to avoid these conditions.

The mechanism of a UC process can be best demonstrated by a Roots type blower (or expander) where there is no internal compression (or expansion) at all, or a perfect 100% UC (OE). The compression cycle for Roots becomes a "square card" (area ABCDEA) as shown in Figure 2 according to the conventional theory. This paper uses Roots as a simplified example of an UC in order to re-exam the energy transfer process and demonstrates its unique capability of generating variable pressures without using a variable geometry. Moreover, only UC is discussed while the mechanism and conclusions can be applied to OE (over expansion) for expanders as well.



a. under compression (UC) b. over compression (OC)

Figure 1: UC or OC of PD compressors on P-V diagram

Roots blowers, also named rotary lobe or rotary piston blowers, are well known since its invention in 1860 by the entrepreneur-spirited Roots brothers of Connersville, Indiana, USA. They are well known because they have been widely used throughout the past 150 years time span in many industrial and municipal applications such as venting a mine, supplying combustion air for iron and steel smelting furnaces, etc. The latest use in 20th century includes power source for loading and unloading bulk materials or liquid for trucks or tankers, for aeration in a waste water treatment plant, or for supercharging internal combustion engines to boost output power (commonly known as Roots superchargers). The last application was exemplified by its famous inventor, Gottlieb Daimler, who in late 19th century included a Roots-style supercharger in a patented automotive engine design, making Roots the oldest supercharger designs now available.

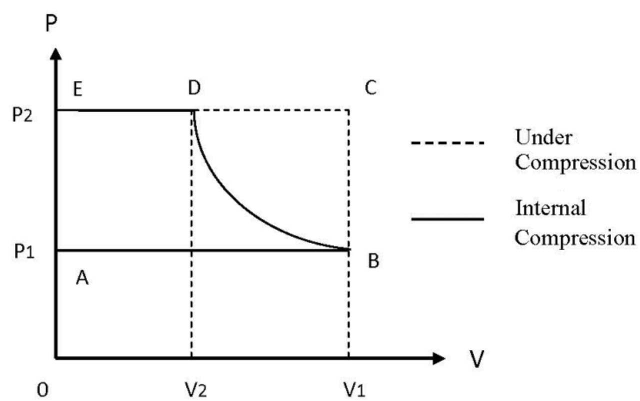
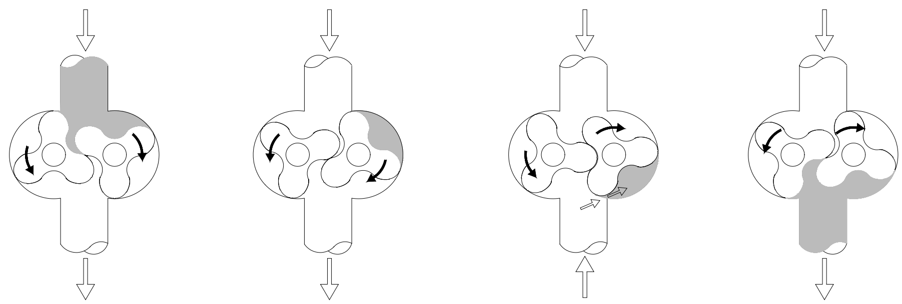


Figure 2: UC of Roots blower vs internal compression on P-V Diagram

1.2 Conventional Theory of an UC

150-years are a long time that should generate a lot of research for this topic. Yet surprisingly, the UC mechanism has been relatively little studied throughout the years by either academia or industry. The explanation could be partially due to a well entrenched but maybe over-simplified conventional theory or partially because of the inherent transient nature of the process involving complicated unsteady pulsatile flows and waves, which were too laborious or imprecise to model or calculate even with today's super computers.

Among dozens of the published papers, patents and textbooks, the consensus regarding the mechanism of UC seems to be well established and almost unanimous. For clarification, let's exam a complete cycle of a classical Roots blower as illustrated from Figures 3a to 3d by following one flow cell in a typical 3-lobe configuration. In Figure 3a, low pressure gas first enters the spaces between lobes of a pair of rotors as they are open to inlet during their outward rotation from inlet port to outlet port. At the lobe position shown in Figure 3b, the gas becomes trapped between the two neighboring lobes and casing as it is transported from the inlet to the outlet. Then the trapped gas is suddenly opened to higher pressure outlet as shown in Figure 3c.



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

Figure 3: Compression cycle of a conventional Roots blower

According to the conventional theory, a backflow is generated that would rush in compressing the gas inside the cell and equalizing pressures with the outlet as shown in Figure 3c. Since it is almost ***instantaneous*** and there is no volume change taking place, the compression is regarded as an iso-choric process (constant volume) as shown by the vertical line BC in Figure 2. After the compression, the rotors continue to move against this full pressure difference until lobes from two rotors meet again, mesh out the compressed gas to outlet port and return to inlet suction position to start the next cycle again, as shown in Figure 3d. The work consumed in the process is equivalent to isochoric work as indicated by area ABCDEA on P-V diagram in Figure 2.

The above views have been widely accepted from both the academia and industry as shown in handbooks (Stoecker, 2004; McCulloch, 2003; Hanlon, 2001), technical papers (Stone, 1988; Uthoff, 1987; Vincent, 1963) and patents (Uthoff, 1987). For instance, Industrial Refrigeration Handbook (Stoecker, 2004) describes UC mechanism for a screw compressor as "The compressed refrigerant has not yet reached the discharge line pressure when the discharge port is uncovered, so there is a sudden rush of gas from the discharge line into the compressor that almost ***instantaneously*** increases the pressure. Thereafter, the continued rotation of the screws expels this gas as well as the refrigerant ready to be discharged". The same reference also states that "the horn in P-V diagram indicates nonproductive work in the UC process caused by unrestrained expansions at the moment of opening". In another instance, Uthoff (1987) states that "the Roots type supercharger uses a backflow compression and is essentially a constant volume process...". The implied energy difference, for example for a case of 2:1 pressure ratio compression, is that almost 20% of more power has to be supplied to an isochoric UC process than to an adiabatic one (shown as area ABCDEA for isochoric and ABDEA for adiabatic on P-V diagram in Figure 2, or as shaded horn areas of lost work on P-V diagram in Figure 1).

1.3 Test Observation and Contradiction

However, a quick comparison could contradict the above theory and raise some puzzling questions. For example, there is a major discrepancy of the outlet temperature between the prediction by conventional theory and the test observations from Roots blowers. According to Amonton Law that governs the iso-choric process, the compression is achieved not by adiabatically decreasing volume as inside a piston compressor, but by increasing its temperature from heat added during the process so that the absolute pressure ratio is the same as the absolute temperature ratio.

$$T_{outlet}/T_{inlet} = P_{outlet}/P_{inlet} \tag{1}$$

For the case $P_{outlet}/P_{inlet} = 2$ and inlet air temperature T_{inlet} = 68 F = 528 K, the predicted outlet air temperature would be T_{outlet} = 1056 K = 596 F as shown in Table 1. On the other hand, the measured outlet temperature of a typical Roots blower is more likely around 250-300 F (121-149 C), a discrepancy of almost 300 F (149 C). In addition, the iso-choric compression process would also predict a much lower efficiency compared with an ideal adiabatic process - isentropic process. For the same case $P_{outlet}/P_{inlet} = 2$, the difference is about 23% (see Table 1). But in reality, the measured efficiency is much closer to the adiabatic efficiency than to an iso-choric one. Graphically, the additional work needed is shown as area BCDB in Figure 2. It is this dramatic difference between the predicted temperature rise and work input with the test observations that prompt the following questions: What is the true thermodynamic process of an UC? What is the compression mechanism at the instant when the gas is exposed to higher system pressure? How is energy exchanged during an UC cycle? And why does UC possess the unique capability of self-adjusting to different system pressures?

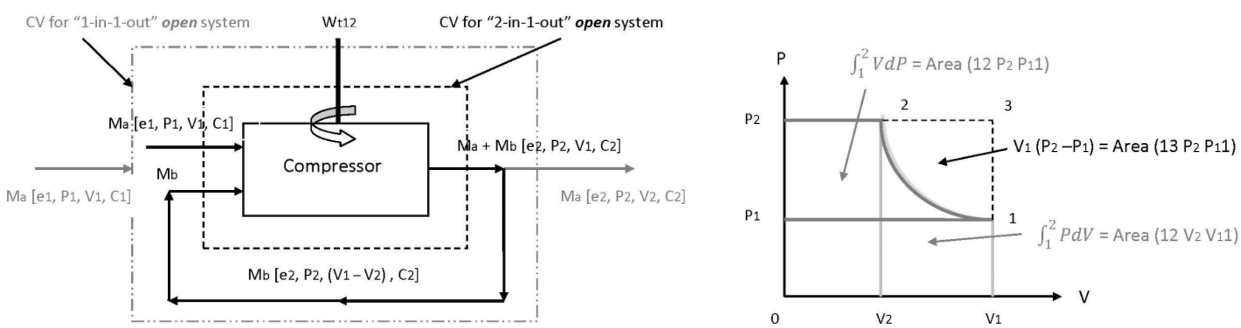
Table 1: Outlet temperature and efficiency calculations for different thermodynamic processes

Gas Type	Air	Air	Air
Ratio of specific heats, k	1.4	1.4	1.4
Inlet Temperature, F/C	68/20	68/20	68/20
Pressure Ratio: Poutlet/Pinlet	1.5	2	2.5
Outlet Temp, F/C - isentropic	147/64	184/84	242/117
Outlet Temp, F/C - shockwave	148/65	190/88	258/126
Outlet Temp, F/C - isochoric	351/177	596/313	892/478
Efficiency, % - isentropic	100	100	100
Efficiency, % - shockwave	98	95	91
Efficiency, % - isochoric	86	77	70

2. NEW THEORY OF UNDER COMPRESSION

2.1 Analysis of Energy Transactions of UC

To demonstrate the thermodynamic and energy transfer process for a complete cycle of UC, the 1st Law of Thermodynamics is used to view the global transactions of the work and energy involved. In thermodynamics, it is customary to analyze a process or a cycle using a control volume (CV) where the Laws of Thermodynamics can be applied. It has also been well accepted that the same physical phenomena can be analyzed with different choices of system boundaries and surroundings without affecting its results.



a. systems and control volumes selection b. various works on P-V diagram

Figure 4: Selection of control volumes and works of under compression

There are two ways to define the CV for a Roots compression as an extreme case of UC. As illustrated in Figure 4a, in a close view, the flow cell in and out of the system during one compression cycle can be modeled as a steady one-dimensional "2-in-1-out" ***open*** system to account for two inflows from both inlet and backflow and one outflow to outlet. Refer to Figures 3a-3d again, an ideal UC cycle is defined as following phases by following one flow cell: a suction phase with flow coming in from inlet with mass flow of M_a, a compression phase by backflow with mass flow of M_b, a discharge phase with combined mass flow of (M_a + M_b) when lobes return to inlet without any carry-over mass flow. The parameters shown inside the parentheses in Figure 4a, say for Ma [e1, P_1, V_1, C_1], are the corresponding thermodynamic properties at those states for this "2-in-1-out" CV. On the other hand, as illustrated in Figure 4a in a far view, the local backflow is "disappeared" by selecting a CV further downstream; hence the system becomes "1-in-1-out" for one inflow from inlet with mass flow of M_a and one net outflow to system with the same mass flow of M_a. Both CVs are open systems with a technical work input W_{t12} from shaft balanced by flow work (PV) and energy change accompanied by mass flows in and out of the CV. It is further assumed that there is no heat transfer across the CV boundary and velocities of inflow and outflow are about equal in magnitude ($C_2 = C_1$).

Apply the 1st Law of Thermodynamics first to "2-in-1-out" CV for one complete cycle so that the net mass and energy change inside the CV is equal to zero. Also note that M_b represents the mass backflow and the corresponding flow work done by the backflow is $P_2(V_1 - V_2)$, both of which do not exist for conventional internal compression model. Summing up the total work (shaft input and flow works) and energy change in and out of the CV results the following equation:

$$W_{t12} + P_1V_1 + P_2(V_1 - V_2) - P_2V_1 = (M_a + M_b)(e_2 + C_2^2/2) - M_a(e_1 + C_1^2/2) - M_b(e_2 + C_2^2/2) \tag{2}$$

Or re-arrange the left side of the equation for flow work:

$$W_{t12} + P_1V_1 + P_2(V_1 - V_2) - P_2V_1 = W_{t12} + P_1V_1 + P_2(V_1 - V_2) - P_2(V_1 - V_2) - P_2V_2 \tag{3}$$

Since $C_1 = C_2$, all kinetic energy and $P_2(V_1 - V_2)$ terms cancelled out, resulting a simplified equation:

$$W_{t12} = M_a(e_2 - e_1) + P_2V_2 - P_1V_1 = M_a(h_2 - h_1) = \int_1^2 VdP \tag{4}$$

While for the alternative "1-in-1-out" model in Figure 4a, applying the 1st Law of Thermodynamics again results the following equation:

$$W_{t12} = P_2V_2 - P_1V_1 + M_a(e_2 + C_2^2/2) - M_a(e_1 + C_1^2/2) = M_a(h_2 - h_1) = \int_1^2 VdP \tag{5}$$

Where the term $M_a(e_2 - e_1)$ is the change of internal energy and $M_a(h_2 - h_1)$ the enthalpy change. Another way to look at energy transactions of "1-in-1-out" model is applying the 1st Law of Thermodynamics to the equivalent ***closed*** system (CV following constant mass flow of M_a), resulting in the following equation:

$$W_{12} = M_a(e_2 - e_1) = \int_1^2 PdV \tag{6}$$

or

$$W_{t12} = \int_1^2 PdV + P_2V_2 - P_1V_1 = \int_1^2 VdP = M_a(h_2 - h_1) \tag{7}$$

Note that Equations (4) (5) (7) are the same even though CVs are selected differently. Moreover, the equation forms are identical to the form for the classical internal compression for an adiabatic process. Physically, the equation can be illustrated on P-V diagram as shown in Figure 4b where the inflow work P_1V_1 is equal to Area ($1P_10V_11$), outflow work P_2V_2 is equal to Area ($2P_20V_22$) while the Area ($12V_2V_11$) is the work done by internal compression like a piston, or expressed in technical work W_{t12}, the Area ($12\,P_2P_11$). By comparison, the isochoric theory would predict a technical work as $V_1(P_2 - P_1)$, or Area ($13P_2P_11$) in Figure 4b that is larger than the adiabatic work or Area ($12\,P_2P_11$) by additional Area (3213). The corresponding discharge temperature for isochoric process would also be much higher due to this additional work input as discussed in section 1.3.

To analyze the above result and how energy is transacted in the cycle, pay attention to the discharge flow work P_2V_1 in Equation (2) that seems to indicate the isochoric work because the outlet volume is the same as the inlet volume V_1. But by incorporating backflow M_b into the "2-in-1-out" model, the P_2V_1 contained in back flow work can be cancelled out by the same term in flow work from discharge, resulting in Equation (4) that is identical to the work input for the classical internal compression as expressed by equations (5) and (7). Physically, the "cancel out terms" could be interpreted as the energy transactions for an UC as follows: first flow work $P_2(V_1 - V_2)$ is "borrowed or loaned" from outlet (system) in the form of backflow M_b, which after doing the compression is "returned" to outlet (system) as lobe is pushing against the "full" system back pressure P_2 and discharged. The end result is a net discharge flow work P_2V_2, identical to the discharge flow work for the classical internal compression. Here backflow M_b can be considered as a re-cycled energy that is exchanged back and forth locally between the flow cell and outlet (system), but globally does not show up. It is interesting to compare this energy exchange strategy with a classical PD that gets the work input 100% from piston or squeezing lobes without any interaction with outlet system.

In light of this revelation, the contradictions between measurements and conventional UC theory can be explained now. The seemly more work needed as the lobes push against the full outlet pressure P_2 actually exist as P_2V_1 during an UC, which is, however, being cancelled out by part of the work done by the back flow: $P_2(V_1 - V_2)$. It could be the neglection of the existence of backflow term that results in the conclusion of the isochoric compression in conventional theory. By including the work done by the backflow modeled as "2-in-1-out" process, the work input for UC is exactly the same as the work input for the classical internal compression, that is, adiabatic work expressed as $\int_1^2 VdP$, not the "square card" work $V_1(P_2 - P_1)$ as predicted by isochoric compression. The adiabatic nature of UC as expressed by Equations (4) (5) (7) also predicts a temperature rise that is in the close proximity of testing. It should be pointed out the above result is based on the assumption that the "borrowed" amount is the same as "returned". But in reality, there are losses during these transactions, resulting in a lower efficiency and higher temperature rise than that predicted from the classical internal compression. This prompts the need to take a closer look at the internal mechanism for a UC.

2.2 Shock Tube Mechanism of UC

The above analysis only gives a ***global*** view of energy transfer process for a complete cycle of an UC at steady state. In order to exam the non-steady ***transient*** characteristics of UC, the Shock Tube theory is applied by zooming in at the instant of the sudden lobe opening that triggers the UC.

The shock tube is invented in 1899 by French scientist Pierre Vieille and is a device used to study the transient aerodynamic flow and waves under a wide range of temperatures and pressures. It has been studied extensively by researchers from the start of the last century and has effectively served the developments of the supersonic and hypersonic flights, and a range of transient devices such as rocket and pulse-jet engines and a special supercharger called Comprex as summarized by Mueller (2006). However, the physical phenomena observed in shock tube or other impulse devices have thus far not served for examining hence determining the UC mechanism of PD type gas machinery. This paper will show that, according to the shock tube theory, UC could be achieved by pressure-waves generated by suddenly exposing low pressure gas in cavity to higher pressure outlet and exchanging energies directly between two gases. Unlike the gradual mechanical compression inside a piston compressor, UC is achieved by elusive waves in a transient process with constant oscillatory and pulsatile gas motions.

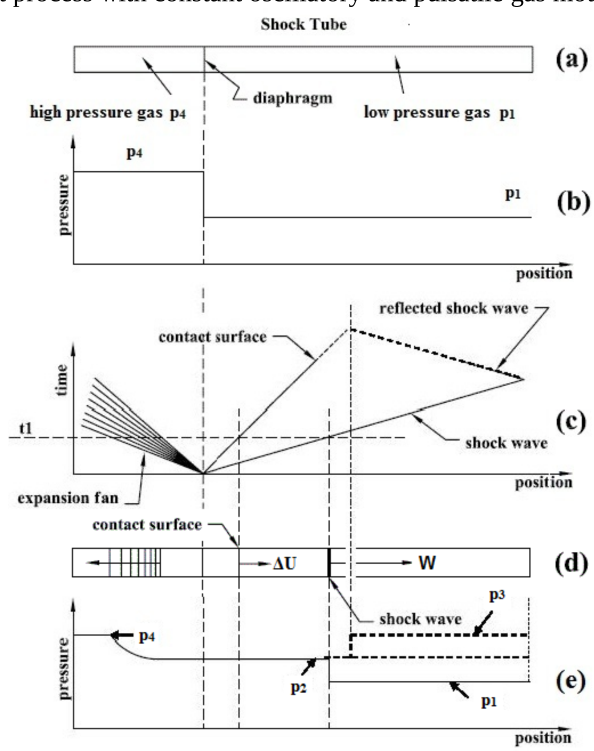
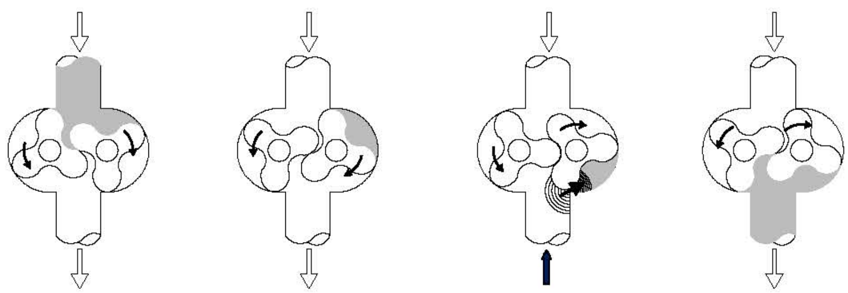


Figure 5(a-e): Diaphragm-triggered shock tube wave diagram and pressure distribution

A simple shock tube (Anderson, 1982), as shown in Figures 5a-5b, is a tube in which a gas at low pressure and a gas at high pressure are separated by a diaphragm. This diaphragm then suddenly bursts open which produces a series of pressure waves, each increasing the speed of sound behind them so that they coalesce into a shock wave propagating through the low pressure gas, as shown in Figures 5c-5e (A reflected shockwave shown as dashed line is generated upon hitting the end of pipe producing a higher pressure p_3 which is just slightly less than initial p_4). The shock wave increases the temperature and pressure of the low pressure gas and induces a gas flow in the direction of the shock wave but at lower velocity than the lead wave. Simultaneously, a fan of rarefaction (expansion) waves travels back into the high pressure gas decreasing its pressure and temperatures. The interface separating low and high pressure gases is referred to as the contact surface that follows the lead wave at a lower velocity.

To understand the UC mechanism in light of the shock tube theory, let's re-exam the cycle of the conceptual Roots blower as illustrated from Figures 6a to 6d. The suction and transport phases are still the same as the conventional theory as shown in Figures 3a to 3b. Then the trapped gas is suddenly opened to higher pressure gas at outlet as shown in Figure 6c. According to the conventional theory, a backflow would rush in compressing the gas inside the cell at this point as shown in Figure 3c. However, according to the shock tube theory, the lobe opening phase in Figure 6c resembling the diaphragm bursting of a shock tube as shown in Figure 5d would generate a series of compression waves or a coalesced shock wave into the cavity between lobes. The wave front sweeps through the low pressure gas and compresses it at the same time at the speed of wave. This results in an almost instantaneous wave compression well before the backflow (behind the contact surface) could arrive because wave travels much faster than the contact interface, as illustrated by the wave diagram in Figures 5c-5e. In this view, the pressure waves or a coalesced shock wave are the primary drivers for the Roots compression while the back flow is simply an induced gas flow behind the wave after compression takes place. According to the shock tube theory, this shock wave compression process is adiabatic thermodynamically and governed by Rankine-Hugoniot Equation (Anderson, 1982), not the Amonton Law for an isochoric process.



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

Figure 6: Compression cycle of a conceptual Roots blower according to the shock tube theory

Figure 7 shows the comparison of the calculated compression curves for isentropic, shock wave and iso-choric processes on P-V diagram. It can be seen that the shockwave deviates just slightly from the ideal isentropic process due to entropy rise across the shockwave while the iso-choric curve is much further away. The areas underneath the curves are indication of work needed for the same compression ratio and are proportional to compression efficiency and temperature rises. Table 1 calculates the normalized efficiencies (assume the isentropic process is 100%) of an iso-choric and shockwave adiabatic process as a function of UC pressure ratio. It can be seen that the efficiency by a shockwave compression is very compatible with the classical internal compression or even dynamic type compressor, just less than 5% difference with an ideal isentropic process for pressure ratio of 2:1. This is much better than the efficiency predicted by an iso-choric process with a 23% gap. For pressure ratio higher than two-to-one, the shock loss will increase and efficiency would decrease more. But remember, most PD compressors or expanders have UC pressure ratios well under 2:1. In addition, a comparison of discharge temperature would shed more lights onto previously discussed differences between measurement and prediction from the conventional theory. According to the Shock Tube theory, UC is an adiabatic process and its temperature rise is governed by the Rankine-Hugoniot Equation (Anderson, 1982) which, when applied to the same example shown in Table 1, results an outlet temperature T_{outlet} = 190 F (88 C). For comparison, the outlet temperature for an ideal isentropic compression is 184 F (84 C) while the isochoric theory predicts an outlet temperature of 596 F (313 C). The shock tube result is much closer to the test data of 250-300 F (121-149 C) that would also include several major losses such as from leakage and dynamic energy exchange.

It should be emphasized that the magnitude of induced gas velocity by a UC can be much higher than values from classical quasi-static theory of Thermodynamics, possibly reaching or even exceeding sonic according to shock tube theory. Imagine the losses when an induced backflow travelling at speed of sound collides head-on with gas inside the cell. So a different design is needed to properly address and manage this "explosive" backflow if the full potentials of UC are to be realized. The resulting efficiency, if successful, could be very close to that achieved by internal compression but without the structural complexity. Based on this insight, future designs could combine an UC into the internal compression in such a way as entirely avoiding the Over-Compression and achieving the maximum overall system efficiency over the desired working pressure range.

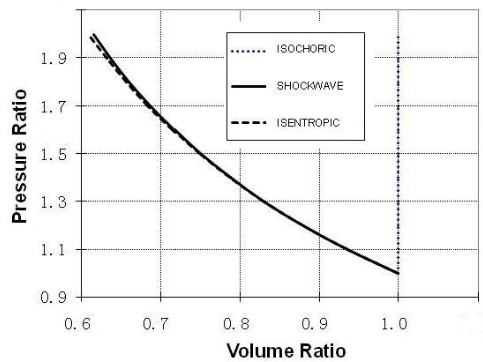


Figure 7: Thermodynamic processes of isentropic, shockwave and isochoric processes on P-V diagram

2.3 Feedback Control Analogy of an UC

In light of the above discussion, we can take another look at the unique characteristics of an UC that is capable of meeting different system back pressures without employing a variable geometry. It is recognized that the backflow in UC could simulate a negative feedback control loop that part of the output (discharge flow) is brought back by a force in such a way as to partially compensate the input. This force, in case of an UC, is the pressure difference between cell and outlet. As shown in Figure 8, when there is no pressure difference, or a perfect internal compression, the backflow or feedback flow is zero. But as soon as ΔP increases from zero, the backflow starts accordingly and behaves in a way as always to reduce and diminish this pressure difference – achieving the same control capability for variable outlet pressure without the use of a variable geometry. This explains why Roots blower can meet different system back pressures because of the existence of this natural feedback loop for an UC. Another well known example is the screw liquid pump that uses 100% UC to handle variable discharge pressure applications.

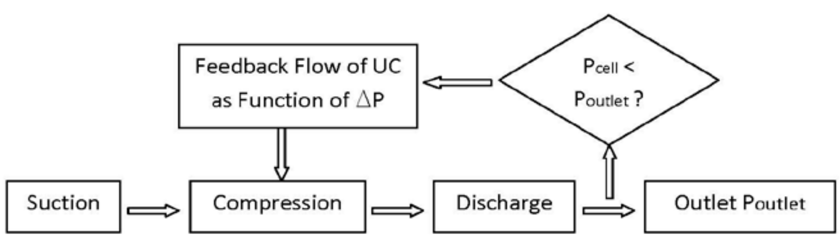


Figure 8: Negative feedback loop of UC

In contrast, most PD compressors achieve a pre-determined pressure ratio by internal compression and the outlet system has no self-correcting capability at all. A variable geometry design is often needed to create a variable internal compression as to meet the variable external system pressures. Though it can effectively handle a range of

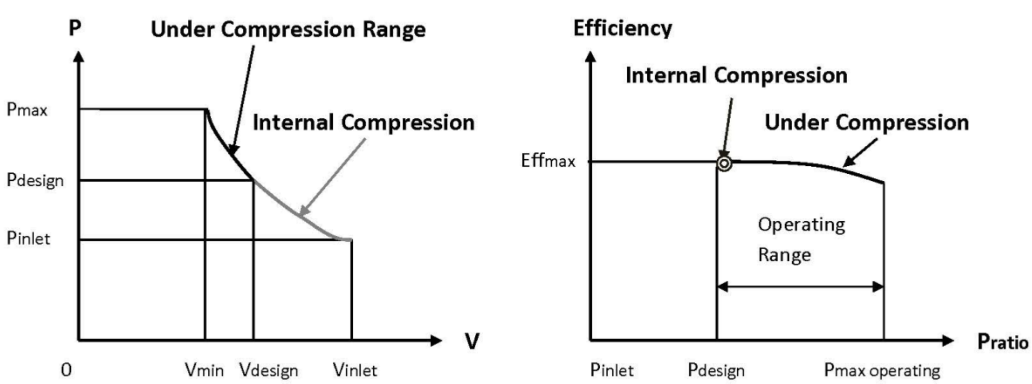
UC, the increased cost and structural complexity prohibit wider applications. It is also handicapped by the slow response time inherent for any hardware based control system. By comparison, UC relies on the system back pressure as a feedback signal to initiate the compression process and determine the actual compression ratio as needed. In this regard, it is self-adjusting or self-adapting to variable external pressure demands in the same feedback control scheme as a variable geometry, but without additional hardware and associated electronics. Therefore, UC possesses a unique advantage over the classical internal compression: maintain a good average efficiency for a wide range of system pressures without complicated external controls.

3. CONCLUSIONS & DISCUSSIONS

Based on the above analysis and comparisons between the new and conventional theories and illustrating examples, the following preliminary conclusions can be drawn for an UC mechanism:

1. The thermodynamic process of an UC may not be isochoric but adiabatic with potential efficiency just slightly less than the efficiency of the classical internal compression;
2. The UC has a unique energy exchange process of "borrow and return" with the outlet system, in contrast to the strategy used by internal compression of relying 100% on "hardware compression";
3. The UC is possibly achieved not by slow moving hardware lobes or piston, but with pressure waves at the speed of wave;
4. The UC has a unique "feedback control" capability, that is, it is a self-correcting, negative feedback control loop adaptable to different system back pressures without a variable geometry control.

Conclusion 1 indicates that UC may inherently be an adiabatic thermodynamic process, not an iso-choric one. Since most PD compressors operate in a wide range of system pressures, a design scheme can be used so that the compressor will work either under internal compression or UC, but never under Over-Compression in order to maximize average system efficiency and minimize pulsations and noises. This strategy happens to be in agreement with recommendations by Stoecker (2004). Figure 9a shows the two segments of this composite compression process with an initial internal compression then joined by an under compression, and Figure 9b shows the corresponding efficiency curve using this scheme. Both processes are adiabatic in nature but with a slight difference in efficiency (slope of the curve on P-V diagram). The down-side of employing this composite process is the generation of gas pulsations during an UC as discussed in the companion paper "Gas Pulsations: A Shock Tube Mechanism" by the same author. To battle that effect, a new control method called Pulsation Trap (the subject of a sequel paper) can be devised that tackles strong gas pulsations in a parallel way and right at its source.



a. composite process on P-V diagram b. efficiency of composite process

Figure 9: Composite process by combining internal compression with UC

Conclusions 2 & 3 indicate that the UC is like NEITHER conventional PDs with internal compression through actions of a piston or rotary screw threads NOR the dynamic type through actions of fast rotating blades, instead it achieves the same compression by a series of pressure waves generated by the sudden opening of the trapped cavity to higher outlet pressure. It is distinguished with the transient and ultra-fast compression velocity that can be as fast as the speed of wave. The same unique wave compression principle is also demonstrated by the Comprex supercharger developed by Brown Boveri and used successfully in automotive industry. In terms of energy exchange, the UC demonstrates a "short-cut" way between two fluids without involving hardware piston or rotating blades. Their major benefits are their potential to generate large pressure changes in short time or within small distance with a low speed driver, thus reducing the size and weight of the compressor. Using Roots as an example, the hardware of a Roots blower is just functioning as a continuous wave generator where two rotors act like both a rotary valve and a rotary seal for moving a fixed volume of gas from low pressure inlet to high pressure outlet in a non-stop batched duty manner. So the structure of a Roots blower is much simpler than traditional PDs because no internal compression is needed. The future designs could take advantage of wave compression without the burdens of hardware associated inertial, cost and weight.

Conclusion 4 shows that UC is ideal for variable demand applications such as in municipal wastewater treatment aeration tanks where water levels change constantly or for automotive supercharging at boosting to different pressure levels while maintaining a good efficiency throughout the range. In the same way, the UC principle can also be used in refrigeration and air conditioner cycle due to the ever-changing system requirement and surrounding conditions.

Finally, this paper by no means is conclusive, but an attempt to stimulate more interests from both academia and industry to re-exam the mechanism of an Under Compression or Over Expansion that seems to be long-due. It is hoped that the full potentials of UC or OE or maybe its overlooked merits could be explored so that future PD gas machinery could be designed to be simpler in structure, smaller in size and smoother in running.

NOMENCLATURE

C	velocity of gas	P	static pressure of gas		**Subscripts**
e	internal energy of gas	T	static temperature	1	inlet
h	enthalpy of gas	V	Gas volume	2	outlet
M	gas mass flow	W	technical work input	t	technical work

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