# Crowley's Method of Isothermal Compression and Expansion.

## Introduction

This paper describes a new (patent pending) method of isothermal compression and expansion of gases. In this paper the term isothermal means a system that approaches isothermal or is more isothermal than adiabatic. In any real machine it is not possible to have a truly isothermal or adiabatic process but this paper describes a practical method of approaching isothermal.

The paper mainly describes the isothermal process in terms of gas compression, as this will generally be the more common application but the process is equally applicable to gas expansion.

Isothermal compression is significantly more energy efficient than adiabatic compression, but with the current standard compressor technologies isothermal compression is not possible as the compression process happens so fast. In most compressors, compression is almost 100% adiabatic.

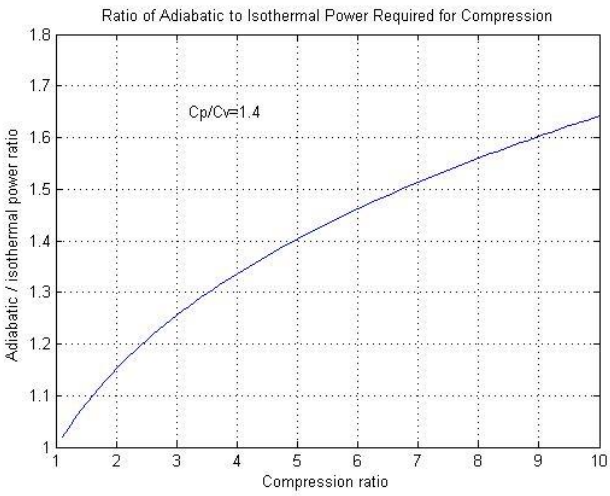


Figure 1

Figure 1 shows the additional power required to compress a gas adiabatically in comparison to isothermal compression.

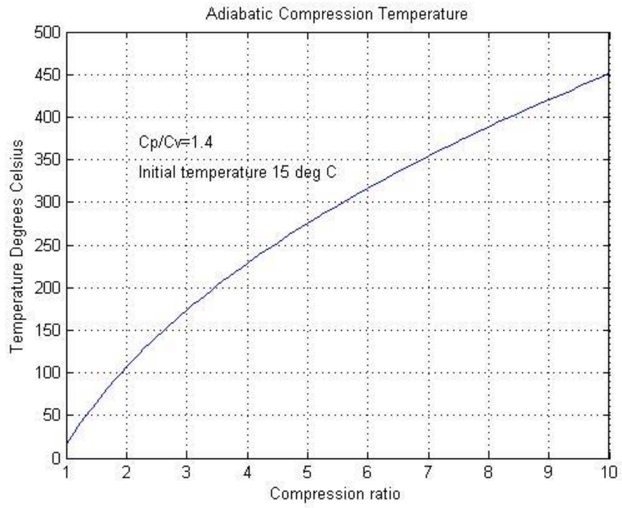


Figure 2

Adiabatic compression can also raise the temperature significantly as shown in figure 2. This figure shows how the temperature rises with increased pressure ratio when the starting temperature is 15$^{O}$C. This temperature rise can effectively limit the maximum compression ratio. For example in a gas turbine the maximum compression ratio is limited by the maximum allowable compressor blade temperature.

The most common way to negate the two effects of increased power and temperature is to use multistage compressors with inter-stage cooling, but with even relatively modest compression ratios the additional power requirements can be significant. The multi stage compression process also adds additional cost and losses, and the inter-stage gas temperature will be higher than the initial gas temperature, so limiting the efficiency gains of multistage compression.

## The Isothermal Compressor

Figures 3A and 3B below shows schematically the basic concept of the new method of compression. In figure 3A the piston is at the end of the suction stroke about to start compression and in figure 3B the compression stroke has just been completed.

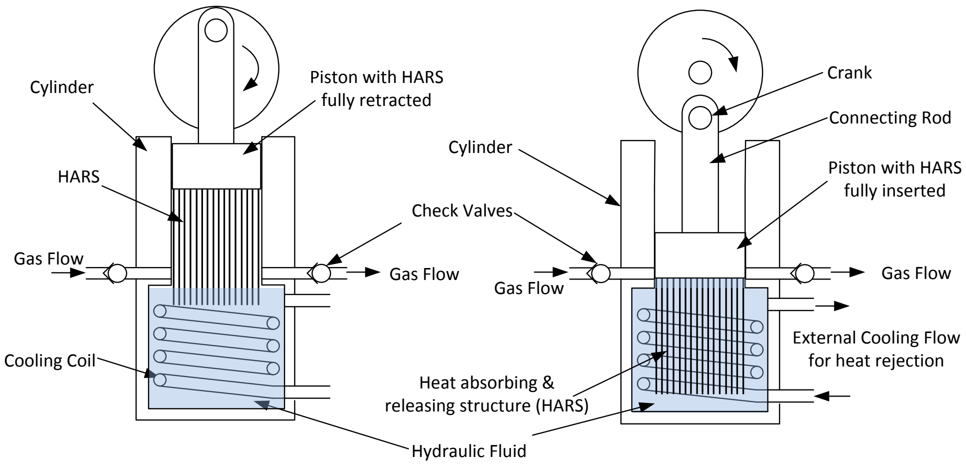


Figure 3A

Figure 3B

The essential new feature to make the compressor isothermal is a heat absorbing and releasing structure (HARS) attached to the piston which is inserted into the hydraulic fluid (usually water) during the compression stroke.

Figure 4

Figure 4 above shows a typical arrangement of the HARS. This specimen was used on Fluid Mechanics test rig.

In figure 4 the HARS is made from 0.15mm sheets of aluminium. Each sheet is at 2mm spacing and is held in position on the end of the piston with an epoxy resin moulding. Each of the sheets of aluminium is curved around the same centre, as this gives them much higher stiffness, so they are not deflected by hydraulic or other loads as they move in and out of the hydraulic fluid at speed.

During compression the gas is contained inside the HARS. The HARS has a large surface area and in the configuration shown in figure 4 the average distance the gas molecules are from the HARS surface is only 0.5mm. The thermal heat capacity of the HARS is significantly higher than the gas, so as the gas is compressed the heat of compression is rapidly transferred into the HARS, thus stabilising the gas temperature and providing isothermal compression.

The HARS is subsequently cooled by the hydraulic fluid in the bottom of the cylinder and then an external cooling circuit is used to cool this fluid.

Unlike other isothermal compressor with this new method the fluid is almost stationary and held in the bottom of the cylinder by gravity, and it is the HARS attached to the piston which moves in and out of the fluid. The HARS is designed such that the sheets that make up the HARS are parallel to the direction of travel of the piston, this allows the free flow of the hydraulic fluid in and out of the HARS with the minimum hydraulic resistance. These two design features allow the device to operate at relatively high speeds. The piston shown in figure 4 has been successfully operated at speeds up to 1500rpm using some of the additional design features described below.

## Design Enhancements to Operate at Speed

Increasing speed or reduced spacing between the HARS sheets, will cause increased turbulence in the hydraulic fluid. Eventually (with the configuration shown in figure 3) the fluid turbulence increases to the point where the gas and fluid mix. Gas bubbles are formed at the bottom of the cylinder and fluid is lost from the cylinder as a mist with the compressed gas. Loss of fluid from the cylinder and fluid gas mixing can significantly reduce the efficiency of the device.

Figures 5A/B below shows schematically how the compressor can be arranged to address these issues. It shows an air compressor where the working hydraulic fluid is water, but it could be an oil or any liquid.

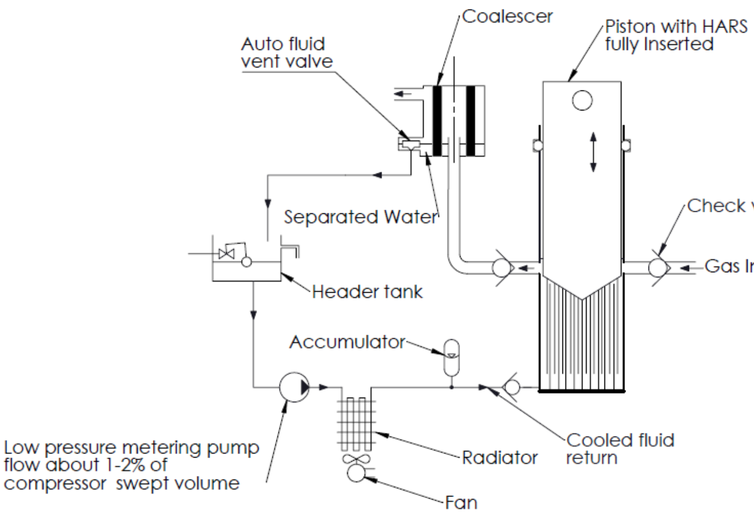


Figure 5A

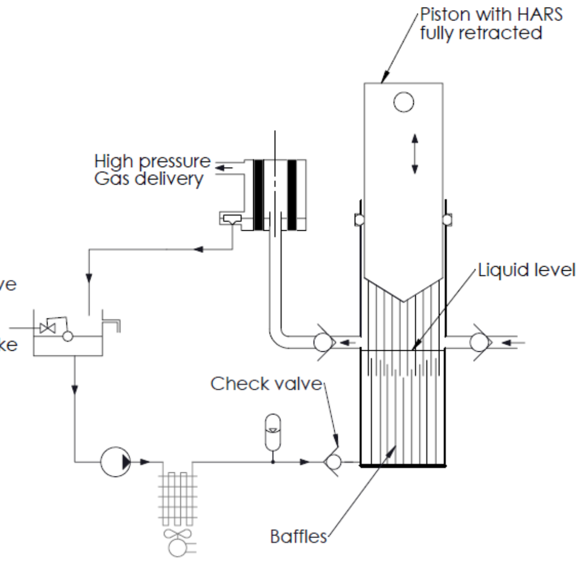


Figure 5B

Figure 5B shows the compressor with the piston and attached HARS retracted at the end of the suction stroke and figure 5A shows the compressor with the piston and attached HARS fully inserted at the end of the compression stroke.

Fixed to the bottom of the cylinder are a series of baffles which are interleave between the sheets that make up the HARS. The baffles have the same physical form as the HARS. These baffles significantly reduced turbulence and prevent air bubbles penetrating below the water surface. The baffles resist the movement of the water in the bottom of the cylinder and hold the working fluid in place. This allows the compressor to operate at significantly higher speeds.

## Enhanced External Cooling

For larger compressors it can be difficult to get the required heat flow through the external cooling circuit shown in figure 3. As it is difficult to get the required coil surface area inside the cylinder. Additionally tests have shown that with the configuration in figure 3, a small amount of water mist (less than 1% by volume) is expelled from the cylinder with the compressed gas on each compression stroke. Both of these issues are resolved with the arrangement shown in figure 5.

In this arrangement the fact that some fluid is lost from the cylinder as a mist is enhanced and taken advantage of for external cooling. A low pressure metering pump continually circulates a small volume of fluid (about 1-2% of compressor swept volume) around an external cooling circuit. A check valve in the circuit allows fluid to flow into the cylinder on the suction stroke and prevents reverse flow on the compression stroke. As fluid is added to the bottom of the cylinder the same volume will be expelled with the compressed gas on every compression stroke. Test have shown the fluid tends to be expelled as a fluid mist.

The liquid is then separated from the high pressure gas in a coalescer and then recirculated back to the metering pump via a header tank.

The gas leaving the coalescer will have 100% humidity which will take water out of the system. At the same time the gas entering the compressor will also have a humidity but probably less than 100% but the volume flows are higher. So there will be a balance between water vapour entering and leaving the system, this usually results in a net flow of water in or out.

The system shown schematically in figure 5 has been designed to accommodate either a loss or gain in net water. As the header tank has an automatic top up and an overflow to drain.

## Ullage and Volumetric Efficiency

An additional advantage of the configuration shown in figure 5 is that it virtually eliminates the ullage or unswept volume inside the cylinder. This can significantly improve the volumetric efficiency of the cylinder.

It is not recommended that this improved volumetric efficiency together with isothermal compression are used to increase the compression ratio of the cylinder. It is recommended that compression ratios are limited to a maximum of 4:1 for each stage, because if the ratio is too high the time available for heat transfer between the gas and HARS is reduced which reduce isothermal efficiency.

## System Efficiency

We define system efficiency (z) in terms of how far the compression process has been transformed from a purely adiabatic to Isothermal process. So zero efficiency (z) would be a totally adiabatic process and an efficiency (z) of 1 would be fully isothermal.

$$z = \frac{\Delta T_\gamma - \Delta T_n}{\Delta T_\gamma}$$

Where

| | |
|---|---|
| z | Efficiency |
| $\Delta T_{\Upsilon}$ | Temperature rise with adiabatic index $\Upsilon$=Cp/Cv |
| $\Delta T_n$ | Temperature rise with actual polytropic index n |

This definition of efficiency allows you to directly calculate how much power is required to compress the gas knowing the isothermal and adiabatic power requirements for a given compression ratio.

$WD_n = WD_I + (1\text{-}z)\ (WD_{\Upsilon}\text{-}WD_I)$

Where

| | |
|---|---|
| $WD_n$ | work done at polytropic index n |
| $WD_I$ | Work done for isothermal index of 1 |
| $WD_{\Upsilon}$ | Work done for adiabatic index |

So by way of an example, if the compression ratio was 9 from figure 1 it can be seen the adiabatic compression requires 60% more power than the isothermal compression. However if we had a system efficiency of say 90% then the additional power required over the isothermal compression would only be 6%.

The temperature of the gas inside the compressor Tn (used to calculate $\Delta T_n$) is the average temperature. The temperature of gas between the sheets of HARS will vary. The gas molecules in intimate contact with the sheets of HARS will be at the same temperature as the sheets and during compression the temperature profile between the sheets will increase to a maximum at the midpoint then reduce again towards the next sheet. This temperature profile is needed to transfer the heat from the gas into the HARS.

The transient temperature profile between the sheets of the HARS can be calculated using the techniques defined by Yunus A Cengel in "Heat Transfer A Practical Approach" and then knowing the temperature profile it is possible to calculate the average temperature Tn and then the compressor efficiency (z).

It can be shown that

$$z = 1 - 0.81e^{4.93\left({}^{1}/_{\gamma}-1\right)K.Nu.T/(P.Hz.G^2)}$$

When Fourier number $\frac{2(1-\frac{1}{\gamma})NuKT}{P.Hz.G^2} > 0.2$

Where

| | |
|---|---|
| K | gas thermal conductivity |
| Nu | Nusselt Number |
| T | Gas temperature (isothermal) |
| P | Maximum Gas Pressure (isothermal) |
| Hz | Speed of compressor in hertz |
| G | Gap between sheets of HARS |

Tests at Fluid Mechanics Ltd has shown reasonable correlation with this formula.

From inspection of the formula it can be seen that the exponent of e will always be negative because 1/ϒ is less than 1, so for optimal efficiency K, Nu and T need to be as large as possible and P, Hz and G should be as small as possible.

For any given gas compressor the requirements for compression ratio or maximum pressure will be fixed. As will the gas inlet temperature and gas thermal conductivity. The reason gas temperature and pressure are important quantities in the formula is because they fix the gas density, which then determines the thermal capacity of the gas.

At the design stage the Nusselt number, cycle speed and gap between HARS sheets can be optimised to achieve the required system performance. It is fairly obvious that if the cycle time is increase (speed of machine is lower) then the thermal efficiency will improve. Additionally if the gap between the sheets is reduced then this will improve thermal efficiency. However reduced speed will require a larger compressor for a given flow rate and reduced gap will also reduce the maximum working speed at which the hydraulic fluid is still stable at the bottom of the cylinder, so reduced gap could also increase compressor size.

The Nusselt number is a measure of the enhancement to gas thermal conductivity by the movement of the gas. If the gas is still, the Nusselt number is 1, however, if the flow is highly turbulent (high Reynolds number) then Nu could be over 50. For a HARS of the form shown in figure 4, the gas velocity and gap size will probably result in the gas flow being laminar (Reynolds number less than 3000). For laminar flow between parallel plates Yunus A Cengel suggests a suitable Nusselt number to be 7.54

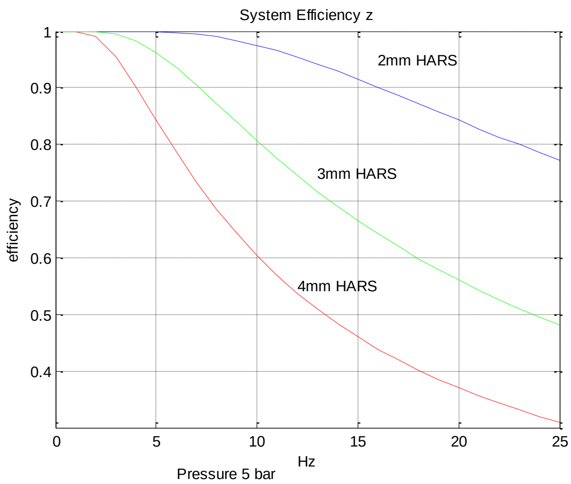


Figure 6.

Figure 6 shows expected system efficiency at varying speeds and HARS sheet spacing. The assumed outlet pressure is 5bar and Nusselt number was 7.54 (laminar flow).

## System Improvement

Fluid Mechanics are currently investigating a number of geometry changes to improve system efficiency (z).

Figure 6 shows that by reducing the gap between the HARS sheets this significantly improves system efficiency, but a reduced gap can also limit the maximum speed of operation. If alternative sheets have staggered lengths as shown in figure 7. Then the shorter lengths are only in the hydraulic fluid for a reduced part of the cycle which allows the system to operate faster. This has been shown to give a good compromise between sheet gap and system speed.

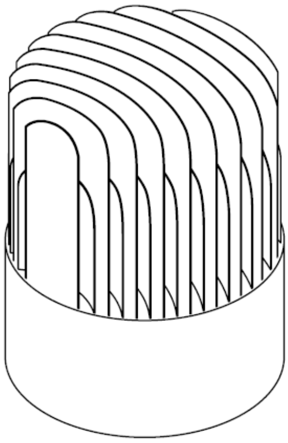

Figure 7

If the gas velocity in the HARS is increased so the flow is turbulent (Reynolds number above 4000) then the Nusselt number will increase. This will then improve efficiency (z). The gas velocity can be increased by using a spiral HARS as shown in figure 8 below.

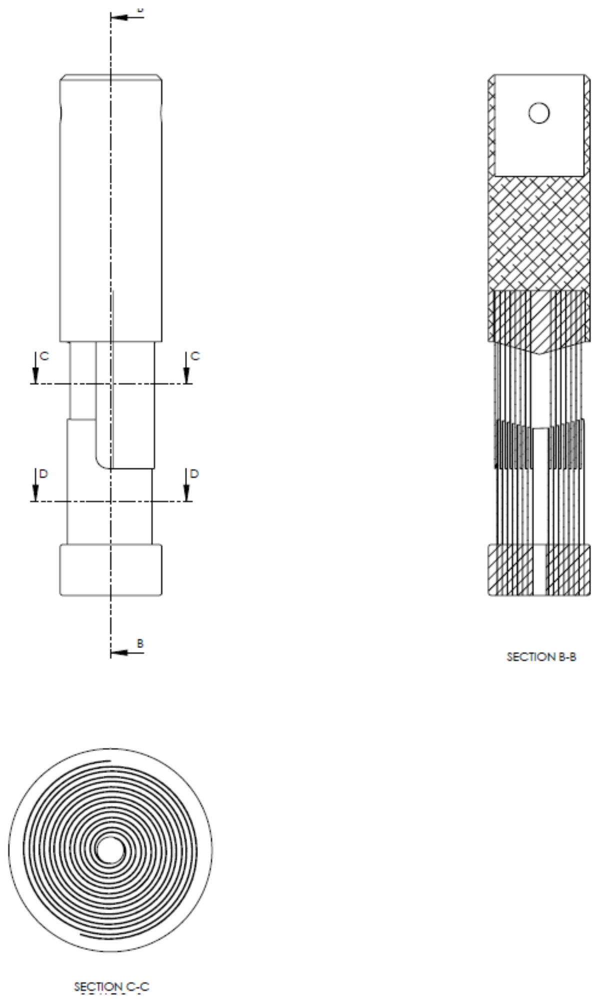


Figure 8

In this configuration the gas in the centre of the spiral has to travel a significantly greater distance for every stroke of the compressor compared to the configuration shown in figure 4, so the average gas velocity in the HARS is significantly increased. However the velocity of insertion and removal of the HARS from the hydraulic fluid is unchanged.

## Commercial opportunities for isothermal compressor

Crowley's method of compression described in this paper can significantly reduce the power required for compressing gases. Gas compression is a widely used industrial process. There are many plants which have significant power requirements such as liquefied nitrogen, oxygen and LNG. This technology will help reduce operator costs and reduce $CO_2$ emissions for large industrial compressor users.

With increased use of intermittent renewable energies, there is now significant interest and investment opportunities in energy storage systems. Compressed air energy storage (CAES) is considered by a number of companies as a cost effective way of storing energy. Compressed air can be stored in underground caverns such as disused salt mines, or in above ground pressure chambers. With conventional compressor and expander technologies the round trip efficiency can be relatively low, as the gas temperature rise during compression is lost during the energy storage stage, and subsequently during expansion and energy recovery the gas temperature drops further so not all the stored energy can be recovered from the gas. This new method of isothermal compression and expansion has the potential to significantly improve the round trip efficiency of CAES systems.

The isothermal compressor could be used as the compression stage in an Otto cycle engine. In many engines the compression ratio is restricted by the maximum allowable temperature rise (figure 2). Using this new method of isothermal compression it will be possible to use a higher compression ratio and so improve the system efficiency. This type of cycle could be used in combination with a CAES system described above to boost the power to the electric grid when power demand is high. Dresser-Rand have developed such a system using conventional compressors which could be improved upon with Crowley's method of Isothermal compression. Follow link for video explaining how system works. http://www.dresser-rand.com/industries/energy-environment/compressed-air-energy-storage/

Liquid air energy storage is a new emerging technology and companies such as Dearman have developed an engine which runs on liquid air and Highview Power Storage have a system which uses liquid air for grid energy storage. However, these companies use liquid air or liquid nitrogen provided by the big gas suppliers. If there is any significant growth in liquid air, as an energy storage vector, then using Crowley's method of isothermal compression in the manufacture of the liquid air would be more cost effective and would improve total system efficiency.

The Stirling engine and heat pump was invented in 1816 but it has never had much commercial success because the performance of real machines has never got close to the machines theoretical potential. The theoretical Stirling engine/heat pump has isothermal compression and expansion of the working gas and it also has high heat transfer from the working gas across the walls of the compressor and expander chambers. In practise neither of these requirements can be achieved with current compressor and expander technology.

Crowley's method of isothermal compression and expansion has the potential to be used in a Stirling heat pump as a more efficient alternative to the vapour compressor technology commonly used in refrigeration, air conditioning and heat pumps today. The technology can enable an almost isothermal process in the expander and compressor chamber of a Stirling heat pump. It will also enable the rapid transfer of heat between the working gas and external environment.

Crowley's method of compression and expansion is better suited to a Stirling heat pump because inside the hot and cold chambers in addition to the working gas (normally helium) will be a hydraulic fluid. In the case of a Stirling engine, fluid on the hot expander side can be transferred as vapour to the cold compressor side so reducing system efficiency. Unwanted vapour transfers do not occur in a heat pump as the expander is now on the cold side and any transfer of vapour from the expander to the compressor side aids the heat pumping.

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