Cryocourse 2016

School and Workshop in Cryogenics and Quantum Engineering

26th September - 3rd October 2016

Aalto University, Espoo, Finland

Cryocoolers

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The course is based on:

Basic operation of cryocoolers and related thermal machines
A.T.A.M. de Waele
J. of Low Temp. Physics, Vol.164, pp.179-236 (2011) (open access)

Cryocoolers
A.T.A.M. de Waele
Lectures given at Cryocourse 2013 and former ones

Cryocoolers: the state of the art and recent developments
R. Radebaugh, J. Phys., Condens. Matter **21**, 164219 (2009)

Documents from manufacturer's Web pages:

Cryomech	http://www.cryomech.com
Sumitomo	http://www.shicryogenics.com/
Thales Cryogenics	http://www.thales-cryogenics.com
Advanced Research Systems	http://www.arscryo.com/

Wikipedia: https://en.wikipedia.org/wiki/Cryocooler

Outline of the course

- Introduction
- Some thermodynamics
- Joule-Thomson coolers
- Stirling cycle
  - Stirling engines
  - Stirling coolers
- Pulse-tube coolers
  - History
  - Principles
  - Commercial coolers and Applications
- Gifford-McMahon (GM)-coolers

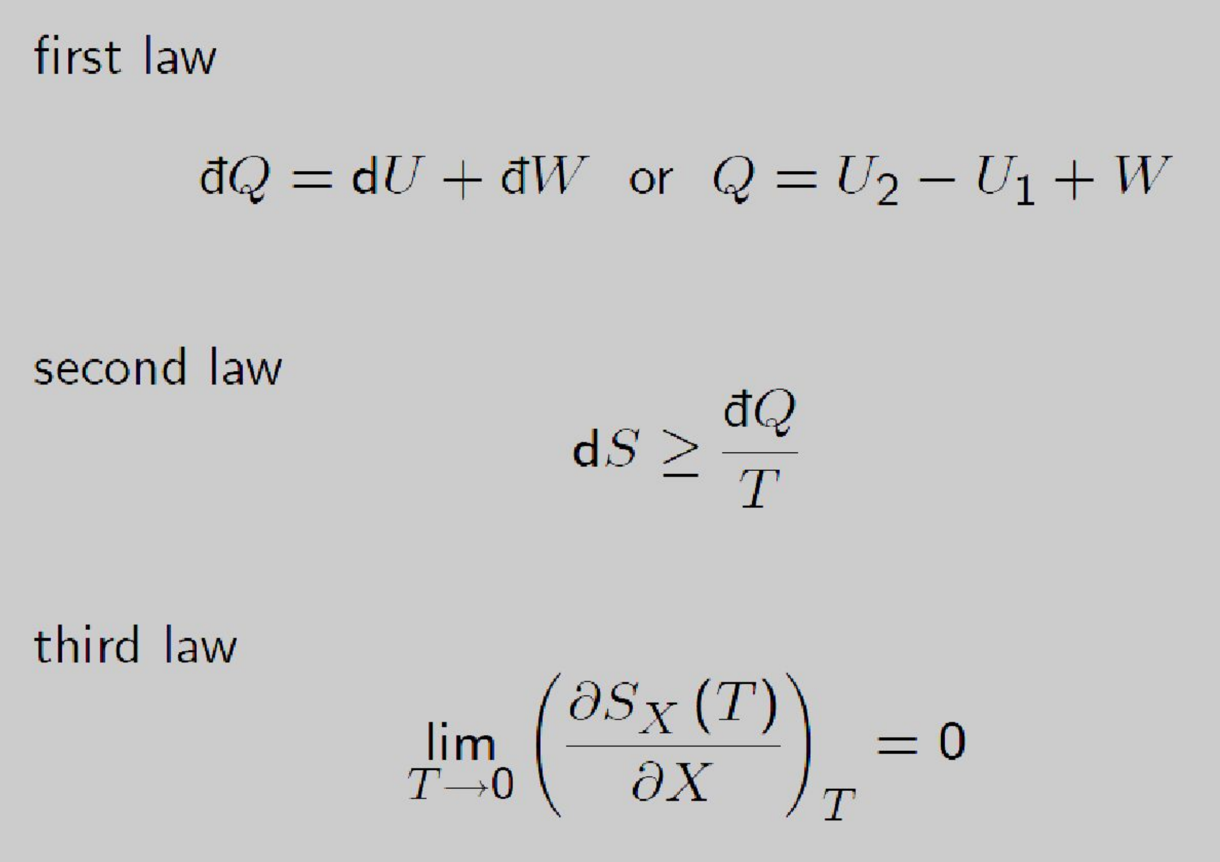
What is a « cryo-cooler »

Cryocooler
From Wikipedia, the free encyclopedia

> *A **Cryocooler** is a standalone cooler, usually of table-top size. It is used to cool some particular application to cryogenic temperatures.*
>
> A recent review is given by Radebaugh.[1]
> The present article deals with various types of cryocoolers and is partly based on a paper by de Waele.[2]

The name « cryocooler », however, is normally used to designate *cyclic thermal machines* based on periodic flow of **gases**, operated in the refrigeration mode.

Laws of Thermodynamics



first law

$$\text{đ}Q = \mathrm{d}U + \text{đ}W \quad \text{or} \quad Q = U_2 - U_1 + W$$

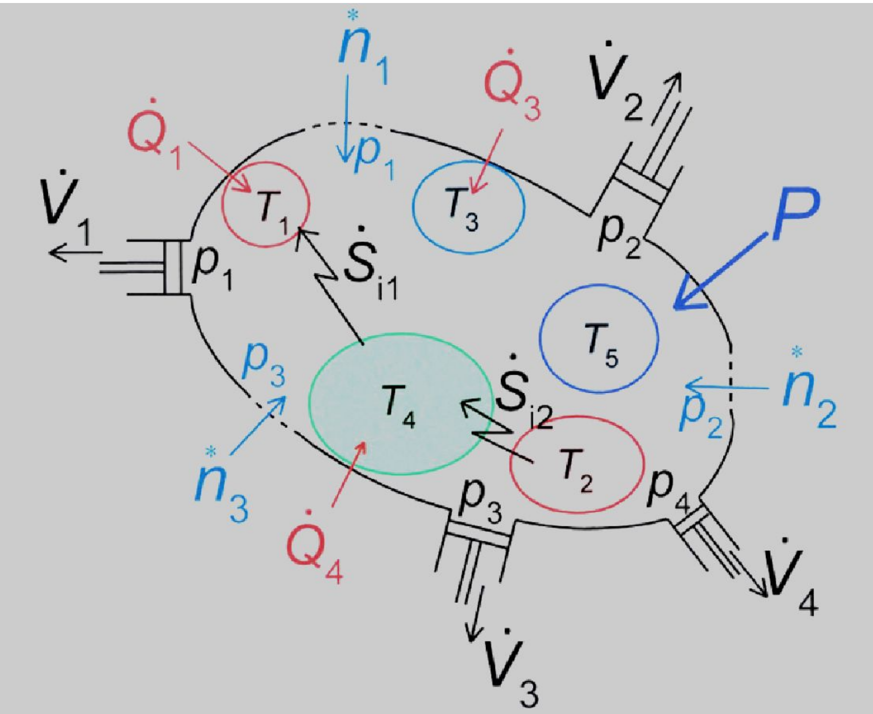
second law

$$\mathrm{d}S \geq \frac{\text{đ}Q}{T}$$

third law

$$\lim_{T \to 0} \left(\frac{\partial S_X (T)}{\partial X} \right)_T = 0$$

Open systems



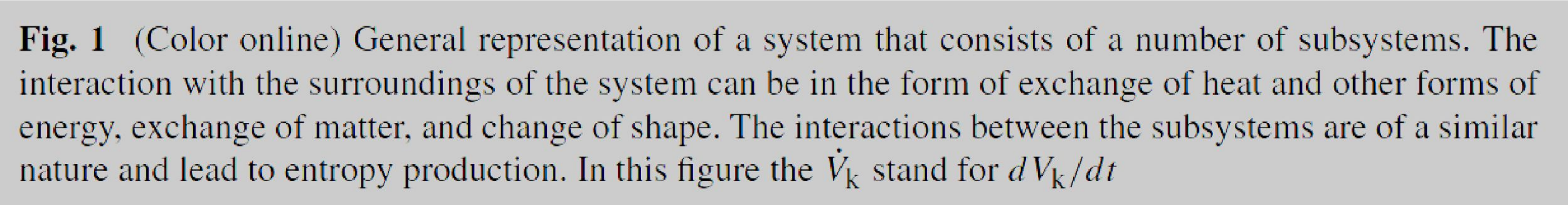
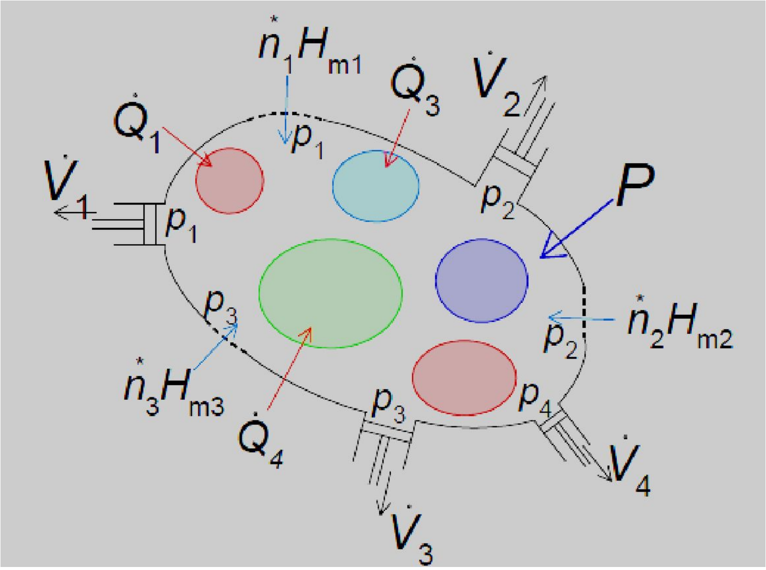


Fig. 1 (Color online) General representation of a system that consists of a number of subsystems. The interaction with the surroundings of the system can be in the form of exchange of heat and other forms of energy, exchange of matter, and change of shape. The interactions between the subsystems are of a similar nature and lead to entropy production. In this figure the $\dot{V}_\mathrm{k}$ stand for dV_k/dt

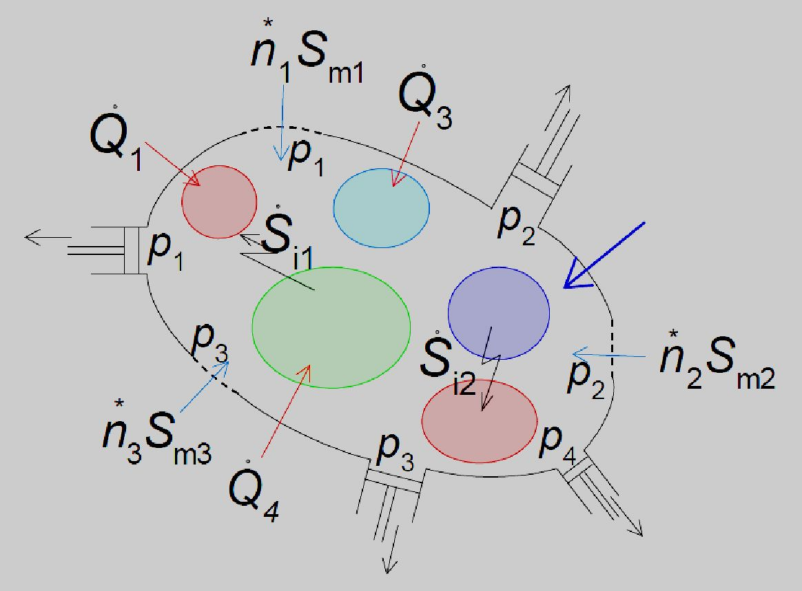
First law for open systems



$$\frac{\mathrm{d}U}{\mathrm{d}t} = \sum_k \dot{Q}_k + \sum_k \overset{*}{H}_k - \sum_k p_k \frac{\mathrm{d}V_k}{\mathrm{d}t} + P$$

$$\overset{*}{H}_k = \overset{*}{n}_k H_{\mathrm{m}k} = \overset{*}{m}_k h_k$$

second law for open systems



$$\frac{\mathrm{d}S}{\mathrm{d}t} = \sum_k \frac{\dot{Q}_k}{T_k} + \sum_k \overset{*}{S}_k + \sum_k \dot{S}_{\mathrm{i}k} \quad \text{with} \quad \dot{S}_{\mathrm{i}k} \geq 0$$

$$\overset{*}{S}_k = \overset{*}{n}_k S_{\mathrm{m}k} = \overset{*}{m}_k s_k$$

Irreversible processes

- heat flow over a temperature difference
- mass flow over a pressure difference
- diffusion
- chemical reactions
- Joule heating
- friction between solid surfaces

Heat engines

first law

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \sum_k \dot{Q}_k + \sum_k \overset{*}{H}_k - \sum_k p_k \frac{\mathrm{d}V_k}{\mathrm{d}t} - P$$

reduces to

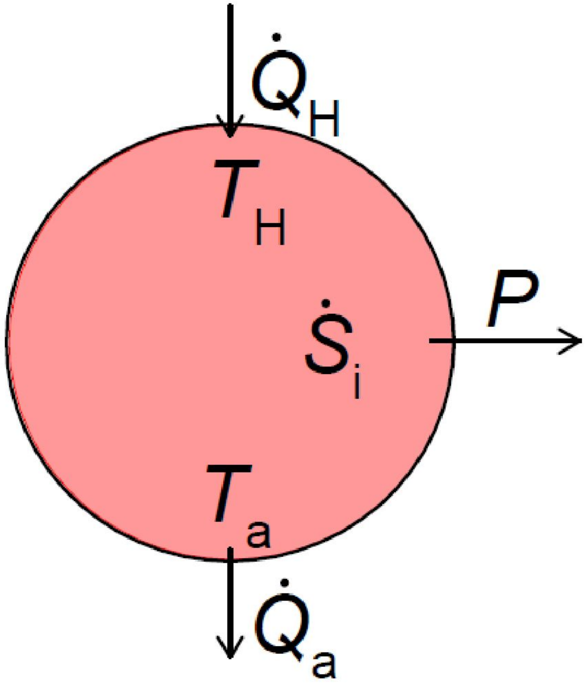
$$\dot{Q}_\mathrm{H} - \dot{Q}_\mathrm{a} = P$$

second l

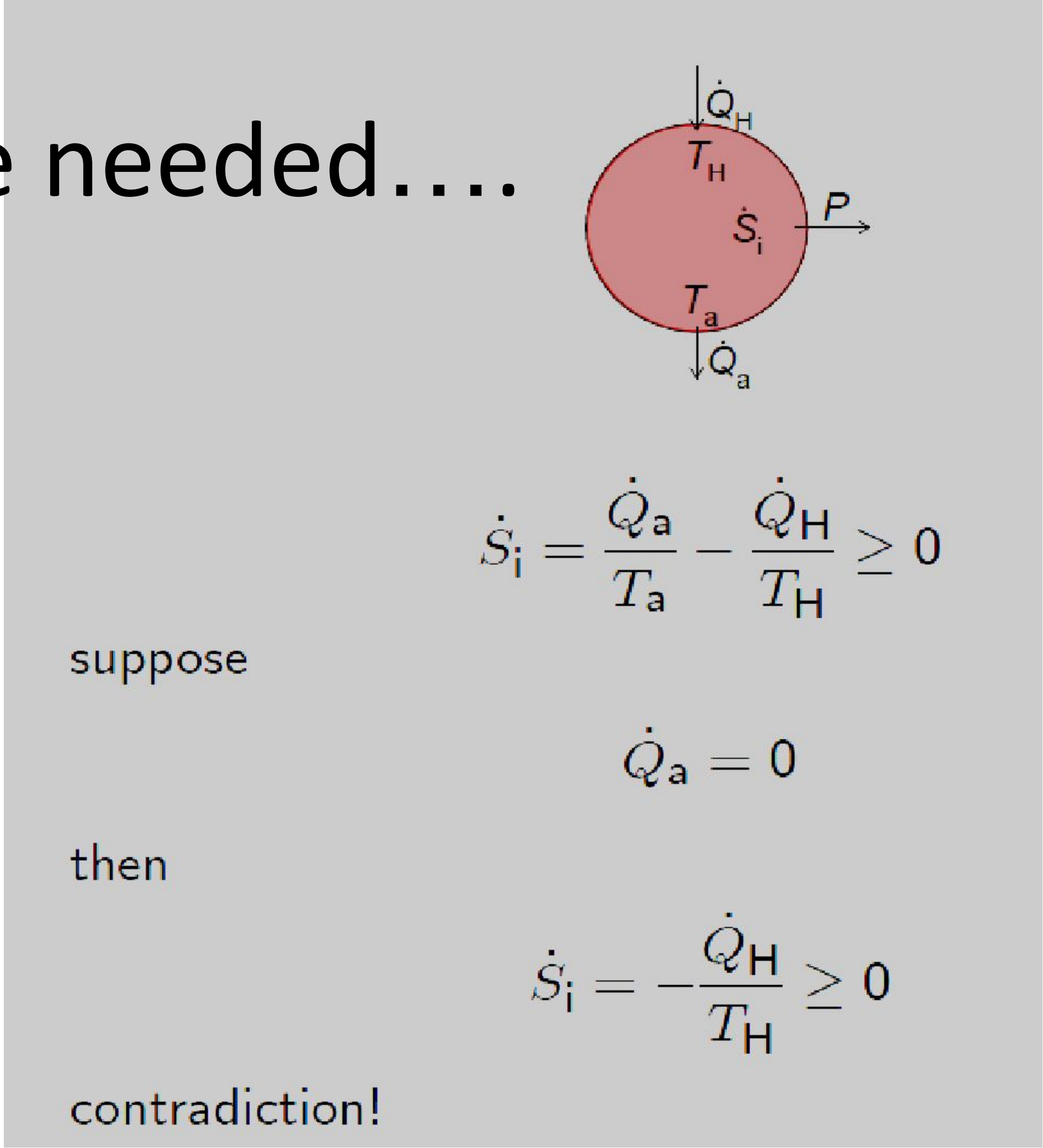
$$\frac{\mathrm{d}S}{\mathrm{d}t} = \sum_k \frac{\dot{Q}_k}{T_k} + \sum_k \overset{*}{S}_k + \sum_k \dot{S}_{\mathrm{i}k} \quad \text{with} \quad \dot{S}_{\mathrm{i}k} \geq 0$$

reduces to

$$0 = \frac{\dot{Q}_\mathrm{H}}{T_\mathrm{H}} - \frac{\dot{Q}_\mathrm{a}}{T_\mathrm{a}} + \dot{S}_\mathrm{i} \text{ with } \dot{S}_\mathrm{i} \geq 0$$



Cold source needed….



$$\dot{S}_{\mathrm{i}} = \frac{\dot{Q}_{\mathrm{a}}}{T_{\mathrm{a}}} - \frac{\dot{Q}_{\mathrm{H}}}{T_{\mathrm{H}}} \geq 0$$

suppose

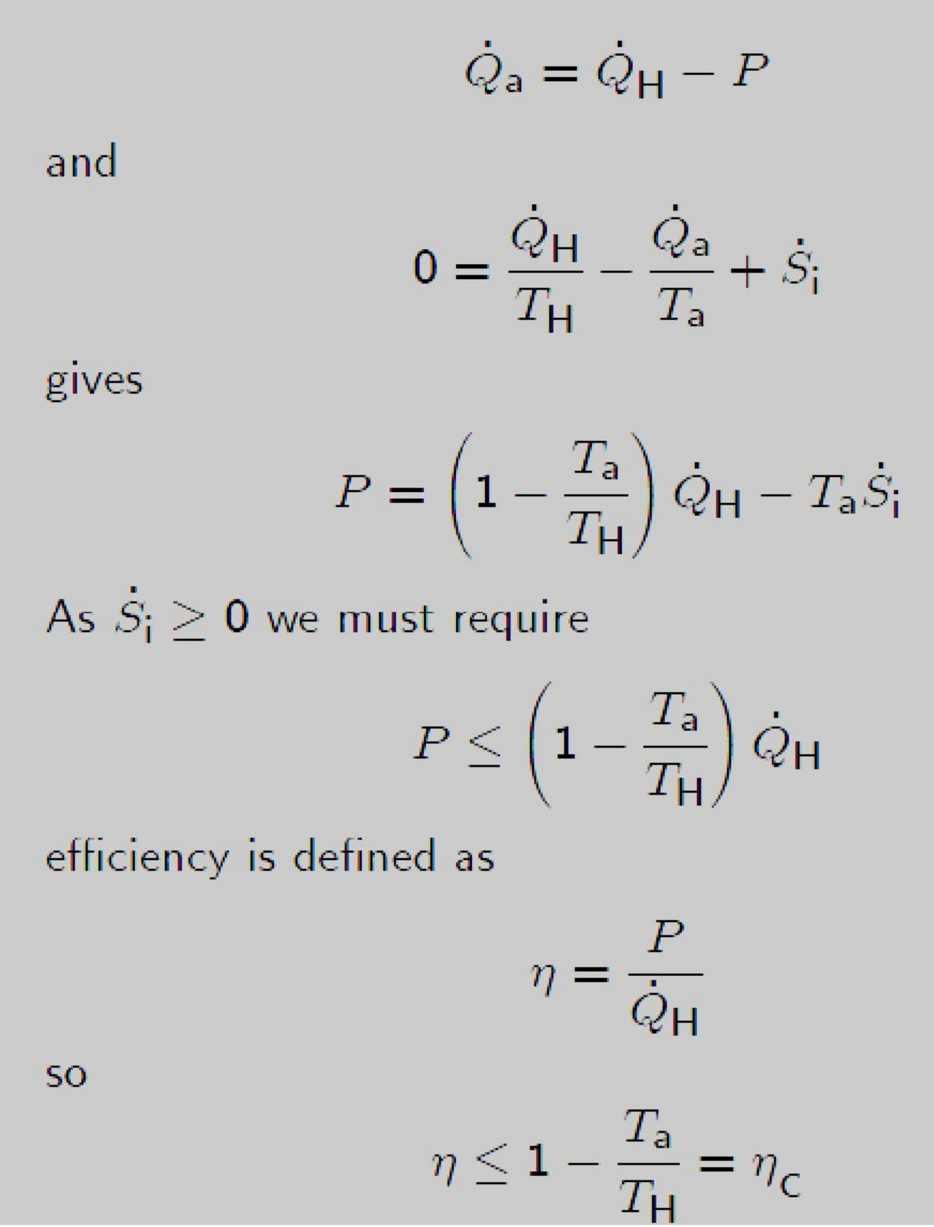
$$\dot{Q}_{\mathrm{a}} = 0$$

then

$$\dot{S}_{\mathrm{i}} = -\frac{\dot{Q}_{\mathrm{H}}}{T_{\mathrm{H}}} \geq 0$$

contradiction!

Efficiency



$$\dot{Q}_\mathsf{a} = \dot{Q}_\mathsf{H} - P$$

and

$$0 = \frac{\dot{Q}_\mathsf{H}}{T_\mathsf{H}} - \frac{\dot{Q}_\mathsf{a}}{T_\mathsf{a}} + \dot{S}_\mathsf{i}$$

gives

$$P = \left(1 - \frac{T_\mathsf{a}}{T_\mathsf{H}}\right)\dot{Q}_\mathsf{H} - T_\mathsf{a}\dot{S}_\mathsf{i}$$

As $\dot{S}_\mathsf{i} \geq 0$ we must require

$$P \leq \left(1 - \frac{T_\mathsf{a}}{T_\mathsf{H}}\right)\dot{Q}_\mathsf{H}$$

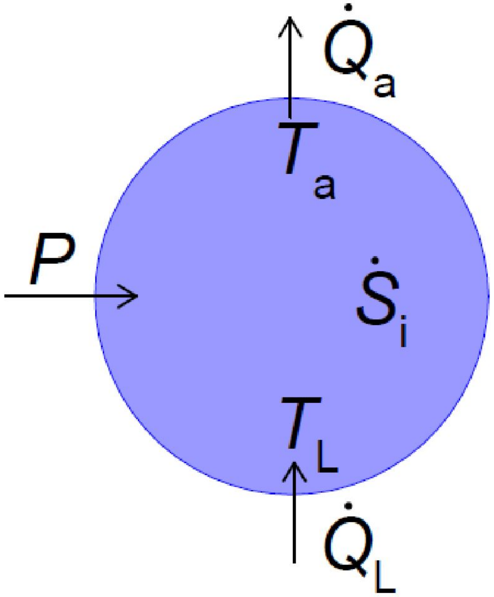
efficiency is defined as

$$\eta = \frac{P}{\dot{Q}_\mathsf{H}}$$

so

$$\eta \leq 1 - \frac{T_\mathsf{a}}{T_\mathsf{H}} = \eta_\mathsf{C}$$

Refrigerators



first law

$$\dot{Q}_a = P + \dot{Q}_L$$

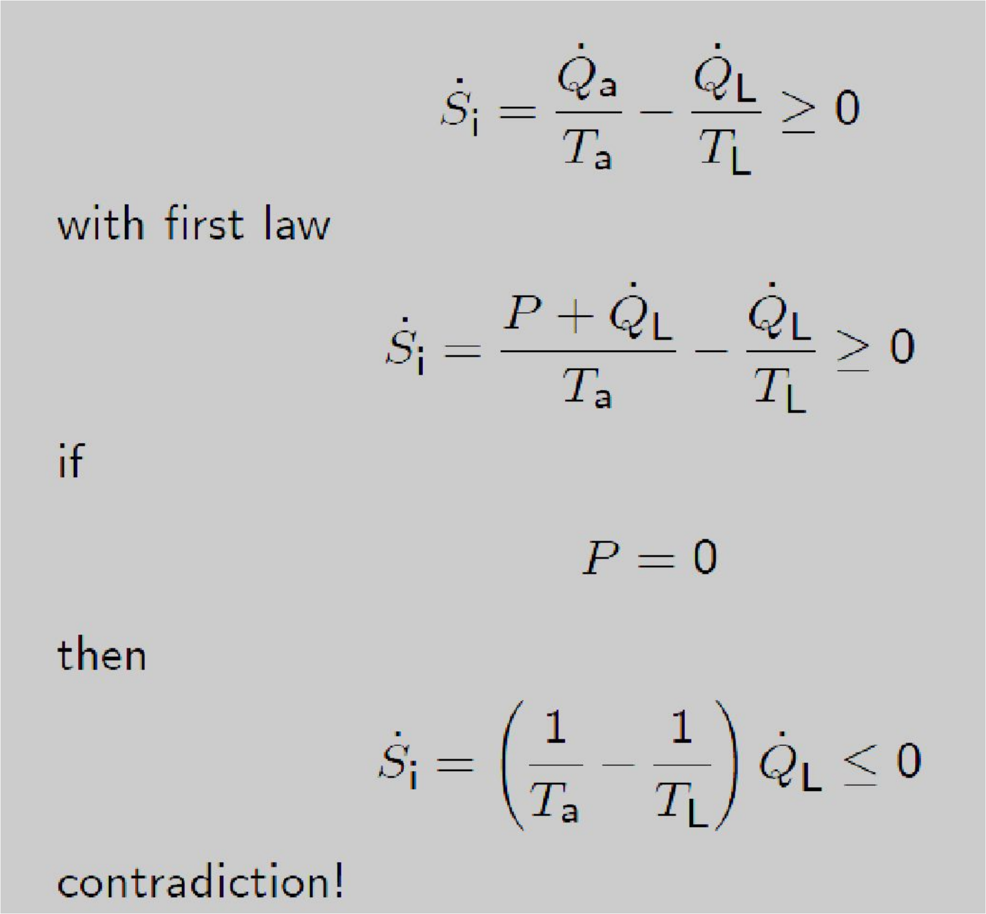
second law

$$0 = \frac{\dot{Q}_L}{T_L} - \frac{\dot{Q}_a}{T_a} + \dot{S}_i \text{ with } \dot{S}_i \geq 0$$

or

$$\dot{S}_i = \frac{\dot{Q}_a}{T_a} - \frac{\dot{Q}_L}{T_L} \geq 0$$

Need external power!



$$\dot{S}_\mathsf{i} = \frac{\dot{Q}_\mathsf{a}}{T_\mathsf{a}} - \frac{\dot{Q}_\mathsf{L}}{T_\mathsf{L}} \geq 0$$

with first law

$$\dot{S}_\mathsf{i} = \frac{P + \dot{Q}_\mathsf{L}}{T_\mathsf{a}} - \frac{\dot{Q}_\mathsf{L}}{T_\mathsf{L}} \geq 0$$

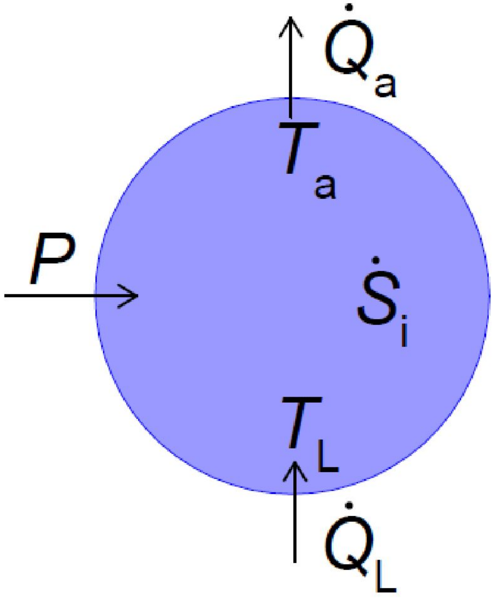
if

$$P = 0$$

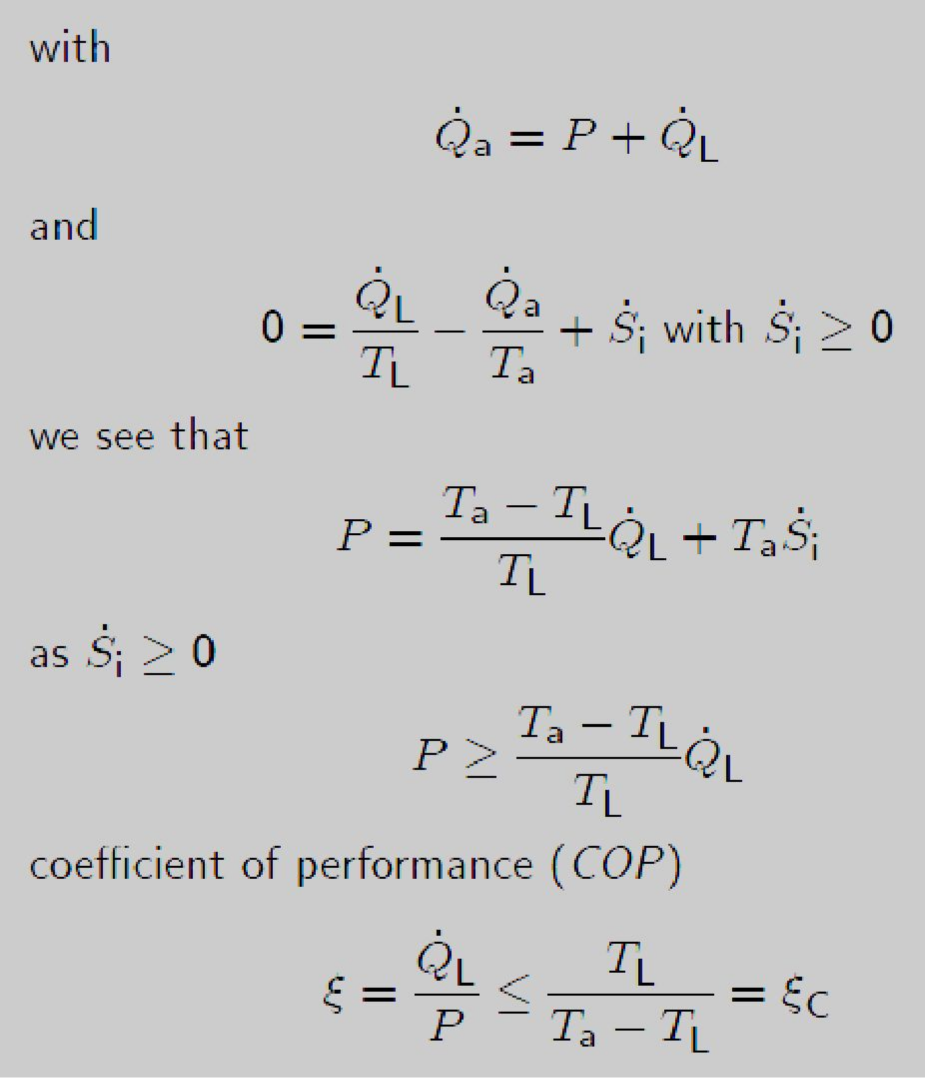
then

$$\dot{S}_\mathsf{i} = \left(\frac{1}{T_\mathsf{a}} - \frac{1}{T_\mathsf{L}}\right)\dot{Q}_\mathsf{L} \leq 0$$

contradiction!



Coefficient Of Performance (COP)



with

$$\dot{Q}_{\mathrm{a}} = P + \dot{Q}_{\mathrm{L}}$$

and

$$0 = \frac{\dot{Q}_{\mathrm{L}}}{T_{\mathrm{L}}} - \frac{\dot{Q}_{\mathrm{a}}}{T_{\mathrm{a}}} + \dot{S}_{\mathrm{i}} \text{ with } \dot{S}_{\mathrm{i}} \geq 0$$

we see that

$$P = \frac{T_{\mathrm{a}} - T_{\mathrm{L}}}{T_{\mathrm{L}}} \dot{Q}_{\mathrm{L}} + T_{\mathrm{a}} \dot{S}_{\mathrm{i}}$$

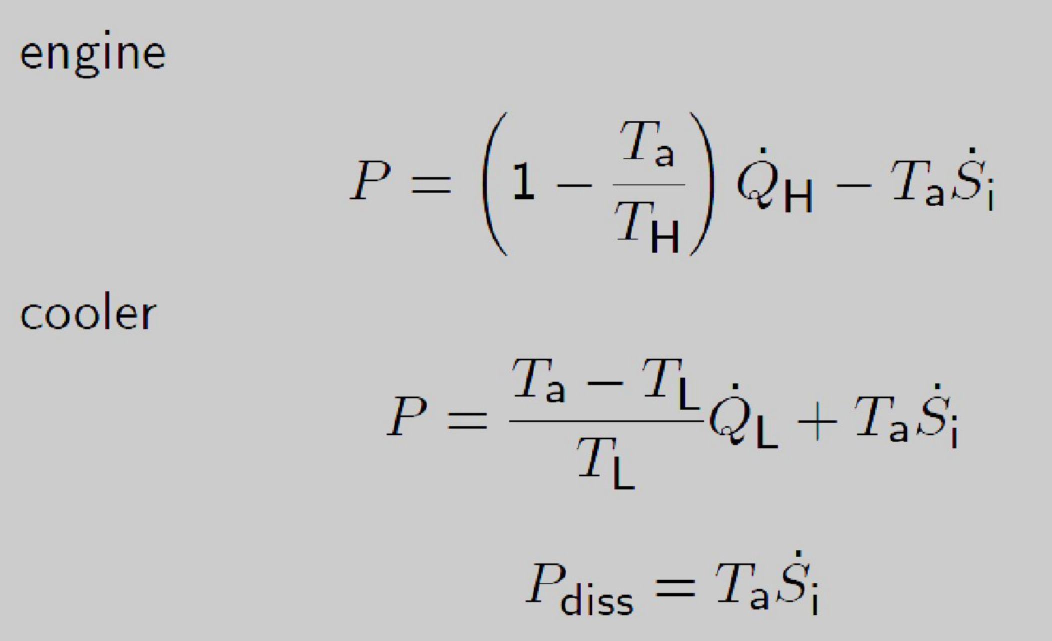
as $\dot{S}_{\mathrm{i}} \geq 0$

$$P \geq \frac{T_{\mathrm{a}} - T_{\mathrm{L}}}{T_{\mathrm{L}}} \dot{Q}_{\mathrm{L}}$$

coefficient of performance (COP)

$$\xi = \frac{\dot{Q}_{\mathrm{L}}}{P} \leq \frac{T_{\mathrm{L}}}{T_{\mathrm{a}} - T_{\mathrm{L}}} = \xi_{\mathrm{C}}$$

Dissipated power



engine

$$P = \left(1 - \frac{T_\mathsf{a}}{T_\mathsf{H}}\right)\dot{Q}_\mathsf{H} - T_\mathsf{a}\dot{S}_\mathsf{i}$$

cooler

$$P = \frac{T_\mathsf{a} - T_\mathsf{L}}{T_\mathsf{L}}\dot{Q}_\mathsf{L} + T_\mathsf{a}\dot{S}_\mathsf{i}$$

$$P_\mathsf{diss} = T_\mathsf{a}\dot{S}_\mathsf{i}$$

Different types of Cryo-coolers

Oscillating gas flow cryocoolers

- **Stirling refrigerators**
- **Gifford-McMahon (GM) refrigerators**
- **Pulse-tube refrigerators**

Constant gas flow cryocoolers

- **Joule-Thomson cooler**
- **Dilution refrigerators (yes, some of them are table-top…;-)**

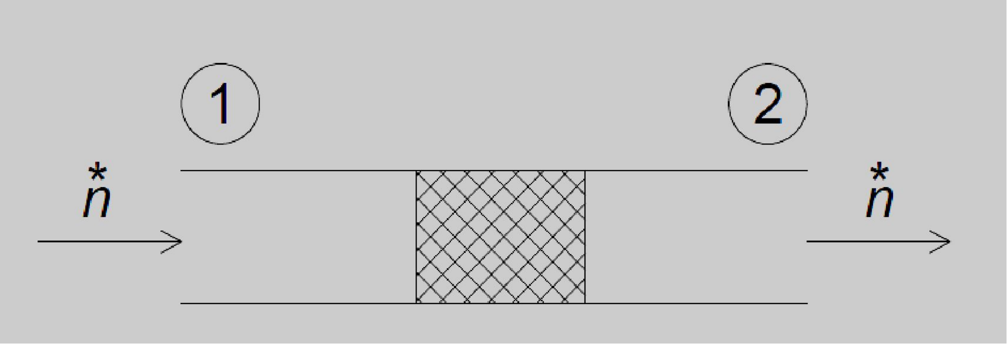
Joule-Thomson coolers

Invented by Carl von Linde and William Hampson, it is sometimes named after them.

Basically it is a very simple type of cooler which is widely applied as the (final stage) of liquefaction machines.

It can easily be miniaturized, but it is also used on a very large scale in the liquefaction of natural gas.

Joule-Thomson: thermodynamics



first law $0 = \overset{*}{n} H_{\mathrm{m1}} - \overset{*}{n} H_{\mathrm{m2}}$ so

$$H_{\mathrm{m1}} = H_{\mathrm{m2}}$$

ideal gas $H_{\mathrm{m}} = C_{\mathrm{p}} T$ so

$$T_1 = T_2$$

second law $0 = \overset{*}{n} S_{\mathrm{m1}} - \overset{*}{n} S_{\mathrm{m2}} + \dot{S}_{\mathrm{i}}$ so

$$\dot{S}_{\mathrm{i}} = \overset{*}{n} \left(S_{\mathrm{m2}} - S_{\mathrm{m1}} \right) \geq 0$$

ideal gas

$$S_\mathrm{m} = S_0 + C_\mathrm{p} \ln \frac{T}{T_0} - R \ln \frac{p}{p_0}$$

since T is constant

$$\dot{S}_\mathrm{i} = \overset{*}{n} R \ln \frac{p_1}{p_2} \geq 0$$

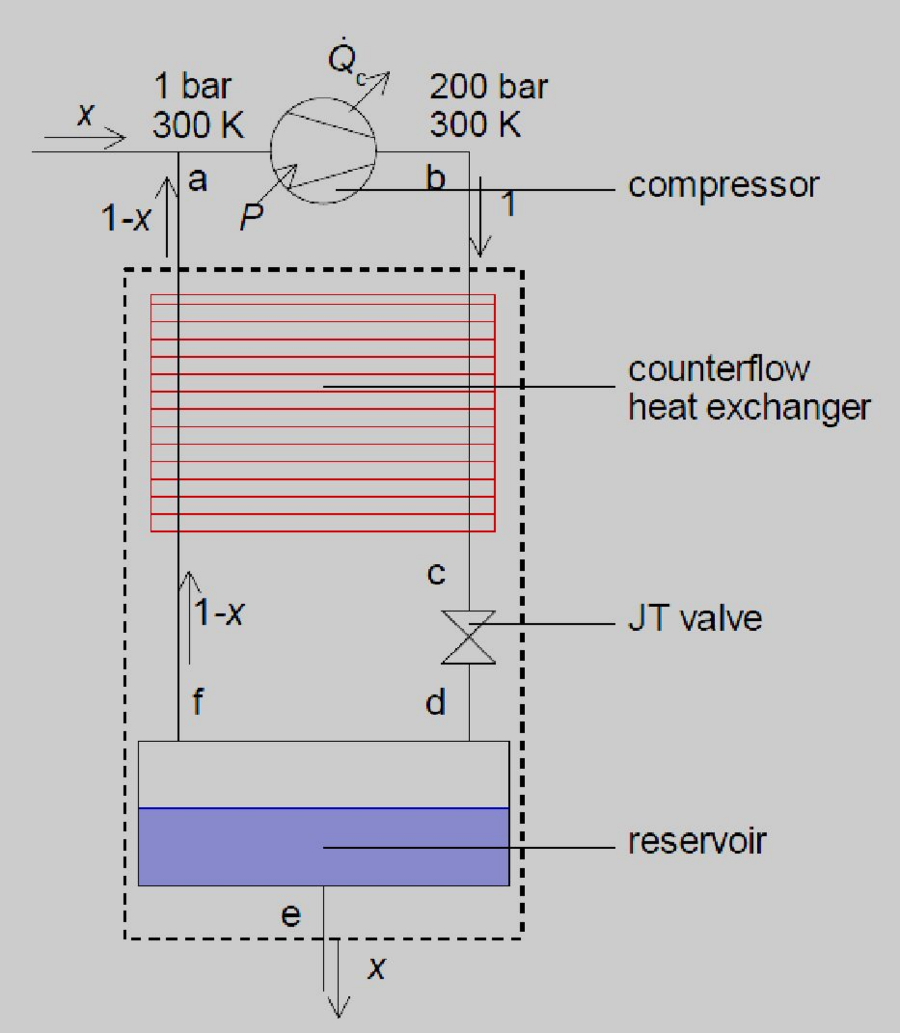
small pressure drop

$$\dot{S}_\mathrm{i} = \overset{*}{n} R \frac{p_1 - p_2}{p_0}$$

ideal gas $\overset{*}{n} = \frac{p_0 \overset{*}{V}}{RT}$ with $\overset{*}{V} = C\,(p_1 - p_2)$ we get

$$\boxed{\dot{S}_\mathrm{i} = \frac{C}{T} (p_1 - p_2)^2}$$

Joule-Thomson cooler (case of a **nitrogen** liquefier)

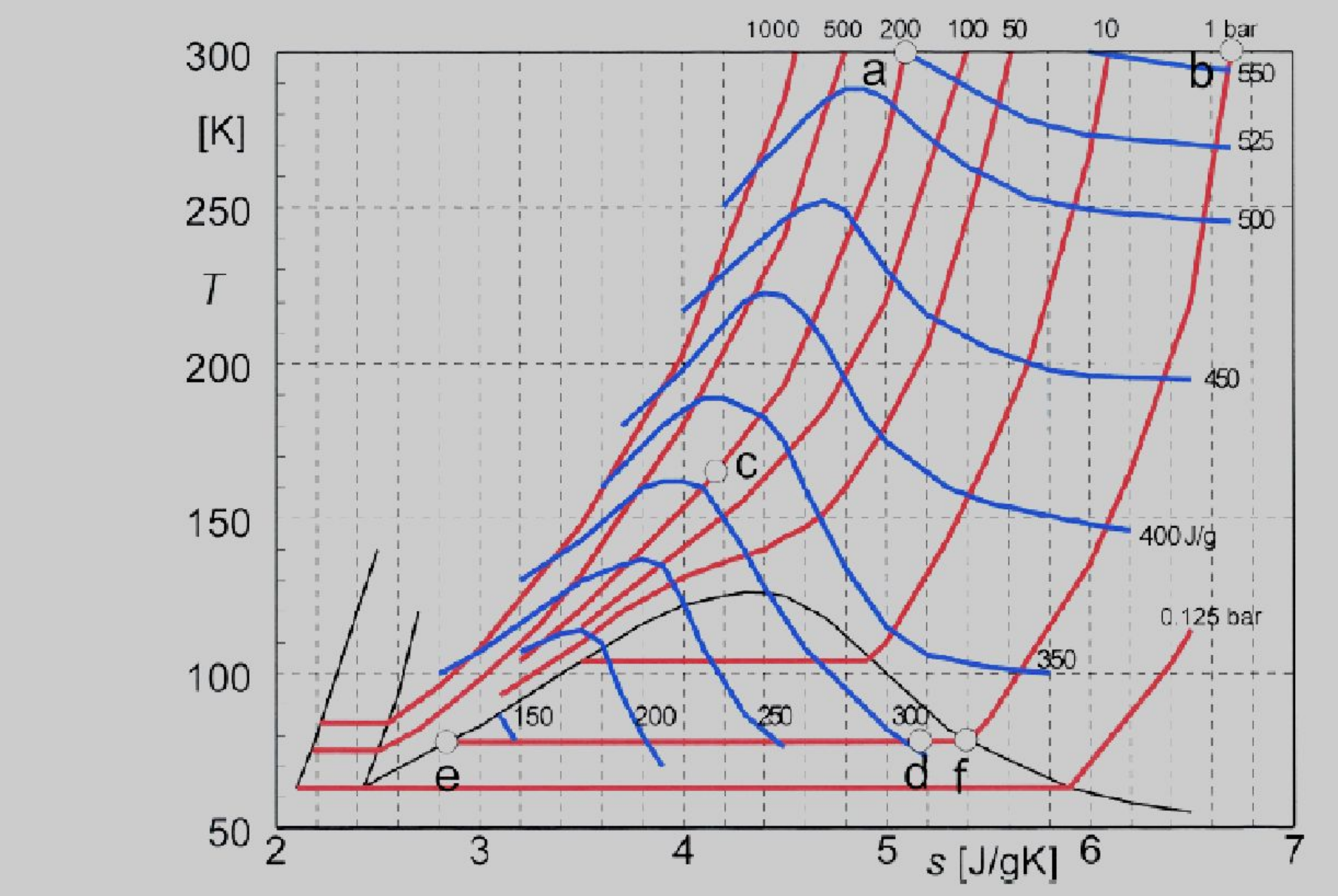


Schematic diagram of a JT liquefier

At the liquid side a fraction x of the compressed gas is removed as liquid.

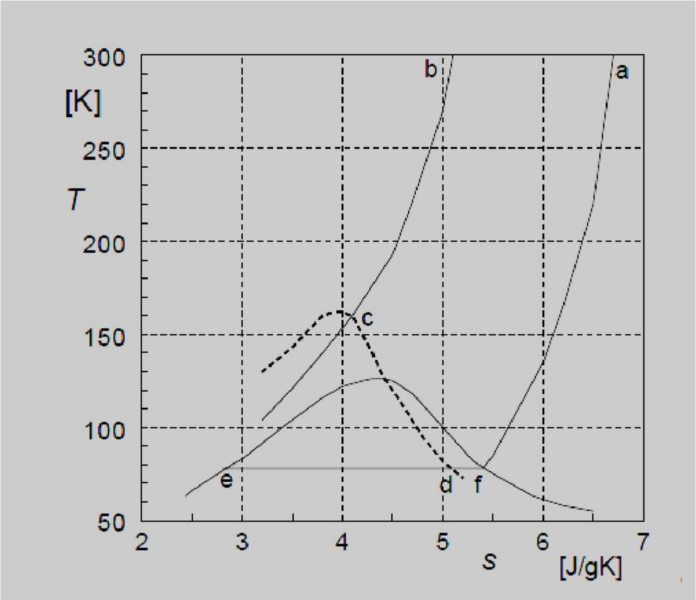
At room temperature it is supplied, so that the system is in the steady state.

The symbols a…f refer to points in the Ts - diagram.



Ts-diagram of nitrogen with isobars, isenthalps, and the lines of coexistence. The pressures are given in bar, the specific enthalpy in J/g.

Ts-diagram of nitrogen with isobars at 1 and 200 bar, the coexistence line and the isenthalp of the JT-expansion indicated.



$$h_b = x h_e + (1-x)\, h_a$$

or

$$x = \frac{h_a - h_b}{h_a - h_e}$$

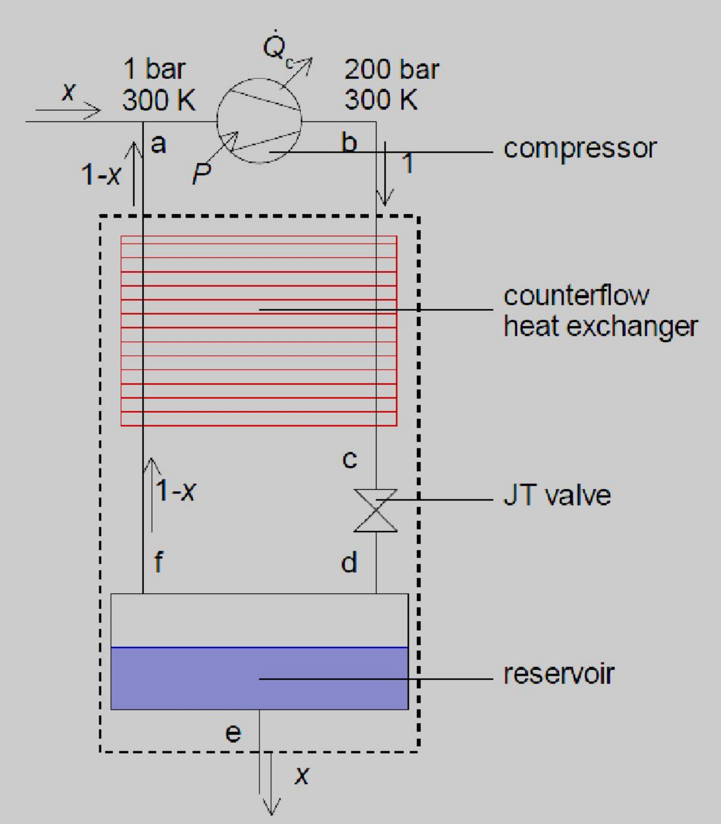
liquefaction if $x > 0$. As $h_a > h_e$ this means

$$h_a > h_b$$

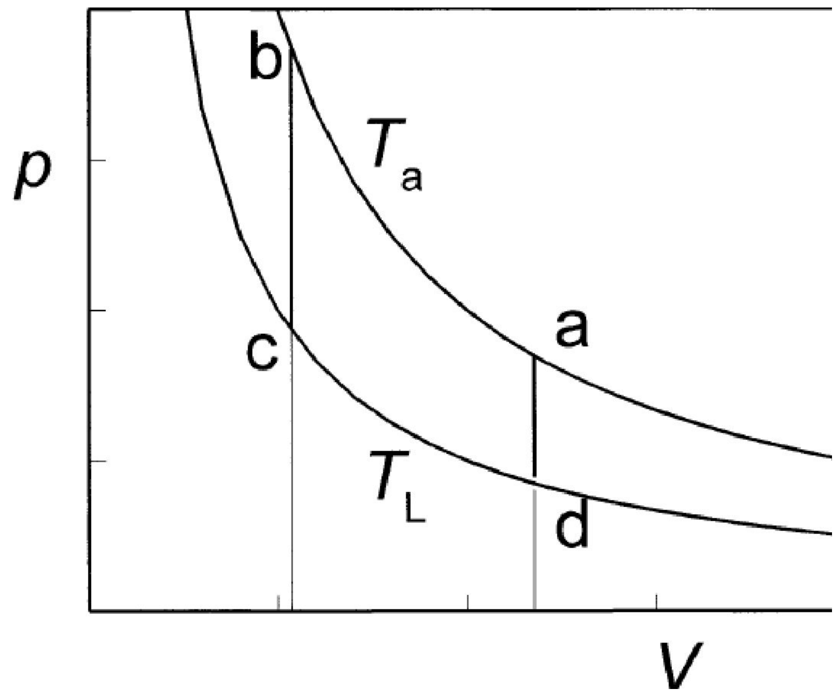
$$x = \frac{555 - 525}{555 - 130} = 0.07$$

$$h_d = x h_e + (1-x)\, h_f = 307 \text{ J/g}$$

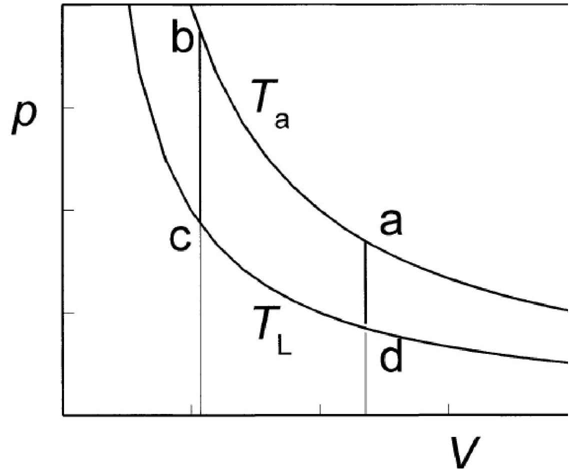
	p (bar)	T (K)	h (J/g)	s (J/gK)
a	1	300	555	6.7
b	200	300	525	5.1
c	200	(165)	(307)	(5.2)
d	1	78	(307)	(4.2)
e	1	78	130	2.8
f	1	78	320	5.4

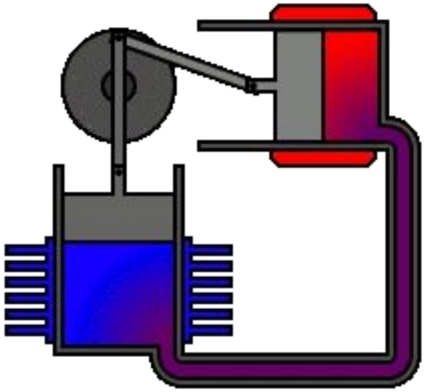


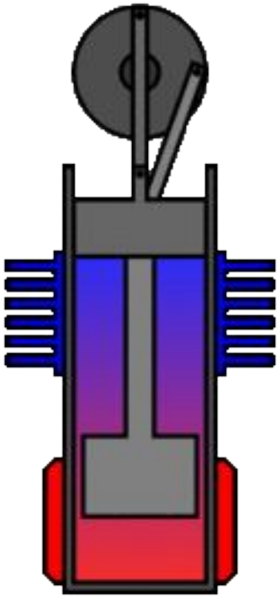
Stirling cycle



Stirling cycle and Stirling engines

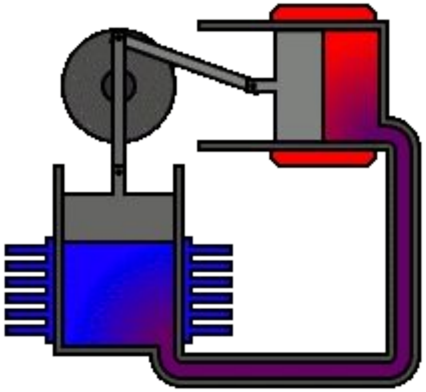


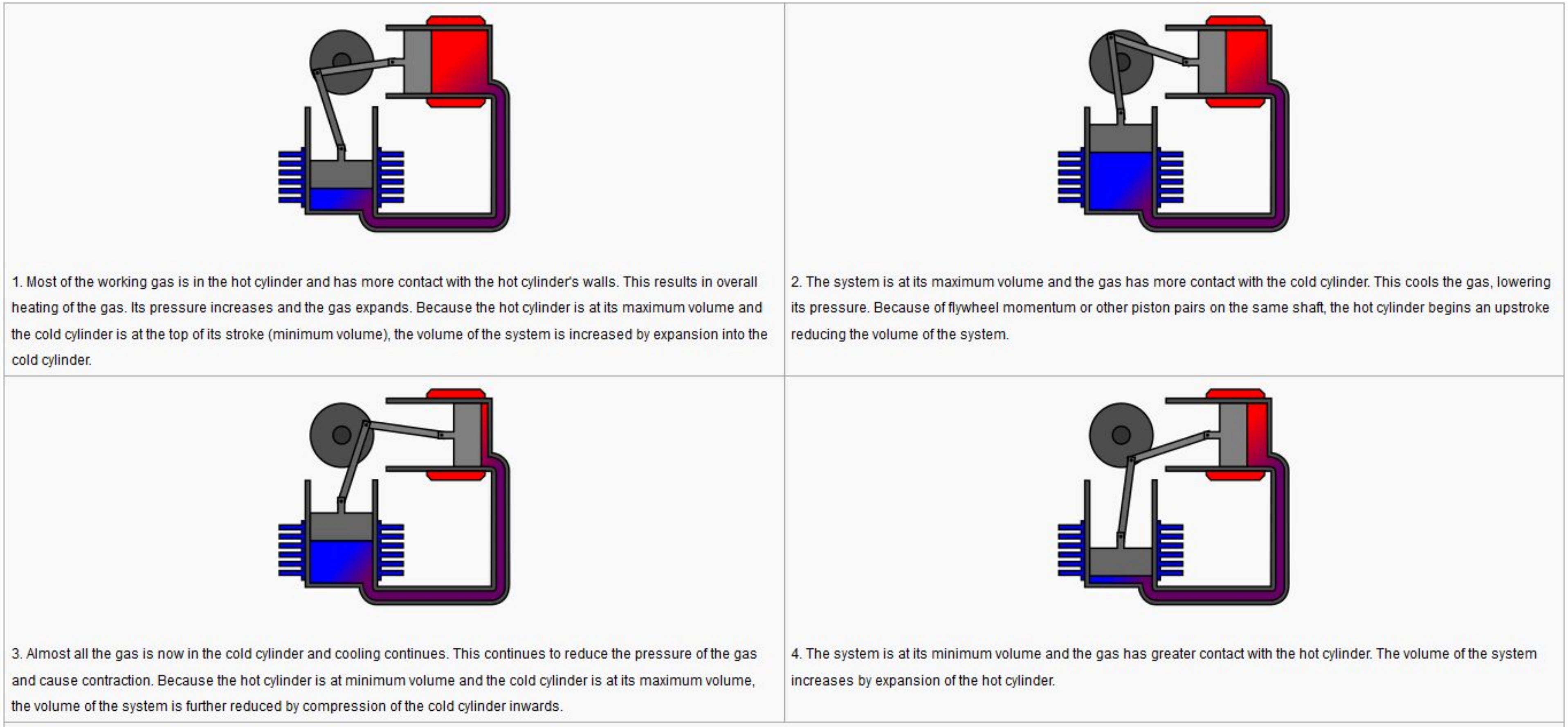




Stirling alpha engine

https://en.wikipedia.org/wiki/Stirling_engine

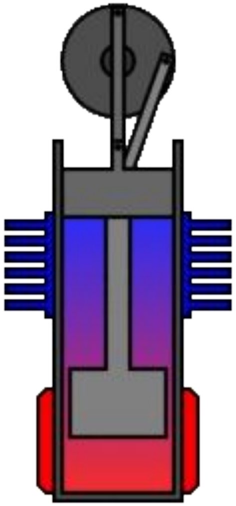


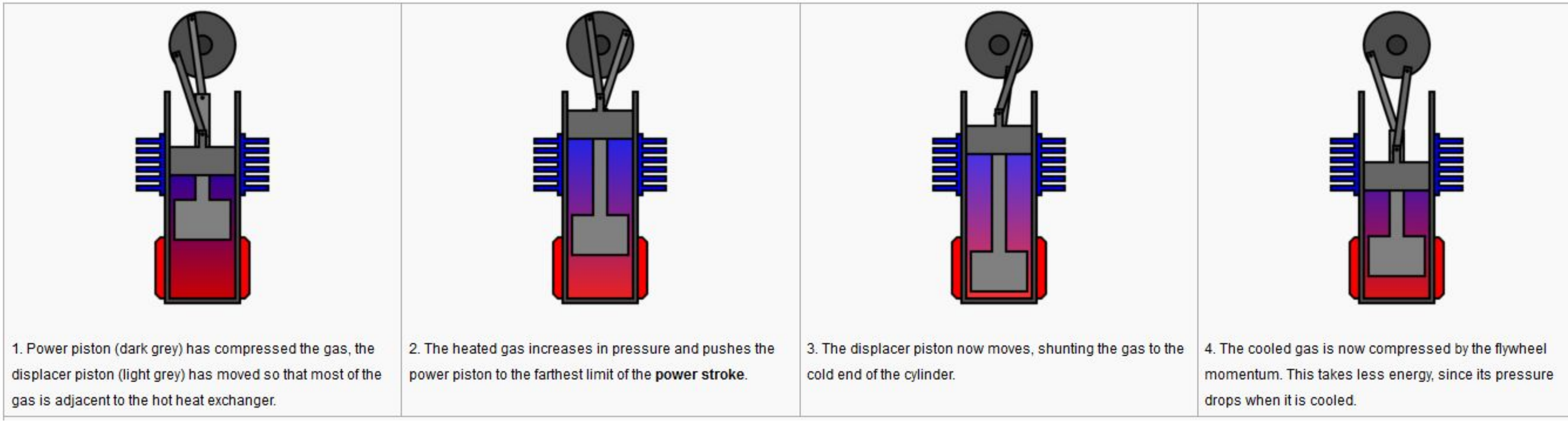


1. Most of the working gas is in the hot cylinder and has more contact with the hot cylinder's walls. This results in overall heating of the gas. Its pressure increases and the gas expands. Because the hot cylinder is at its maximum volume and the cold cylinder is at the top of its stroke (minimum volume), the volume of the system is increased by expansion into the cold cylinder.	2. The system is at its maximum volume and the gas has more contact with the cold cylinder. This cools the gas, lowering its pressure. Because of flywheel momentum or other piston pairs on the same shaft, the hot cylinder begins an upstroke reducing the volume of the system.
3. Almost all the gas is now in the cold cylinder and cooling continues. This continues to reduce the pressure of the gas and cause contraction. Because the hot cylinder is at minimum volume and the cold cylinder is at its maximum volume, the volume of the system is further reduced by compression of the cold cylinder inwards.	4. The system is at its minimum volume and the gas has greater contact with the hot cylinder. The volume of the system increases by expansion of the hot cylinder.

Stirling beta engine

https://en.wikipedia.org/wiki/Stirling_engine





1. Power piston (dark grey) has compressed the gas, the displacer piston (light grey) has moved so that most of the gas is adjacent to the hot heat exchanger.

2. The heated gas increases in pressure and pushes the power piston to the farthest limit of the **power stroke**.

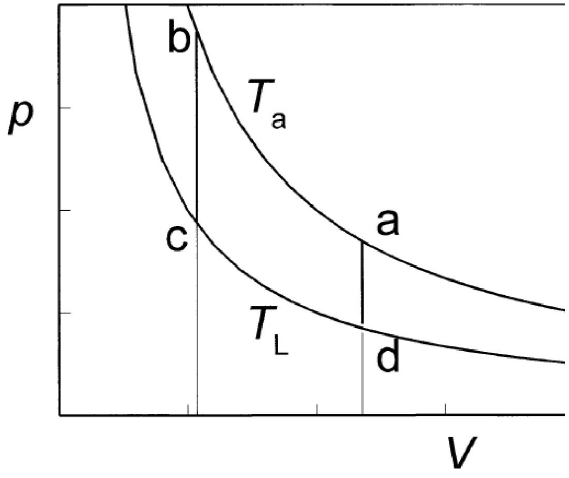
3. The displacer piston now moves, shunting the gas to the cold end of the cylinder.

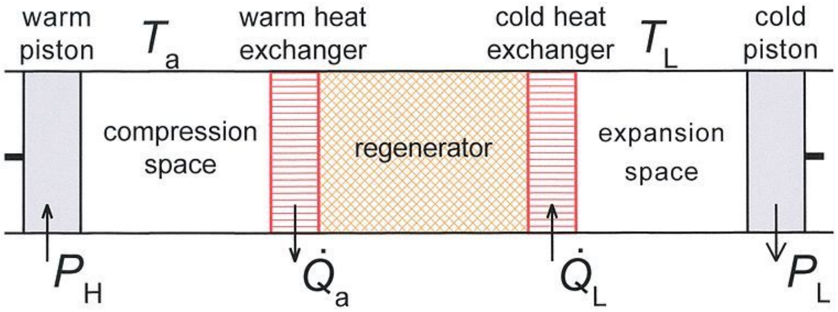
4. The cooled gas is now compressed by the flywheel momentum. This takes less energy, since its pressure drops when it is cooled.

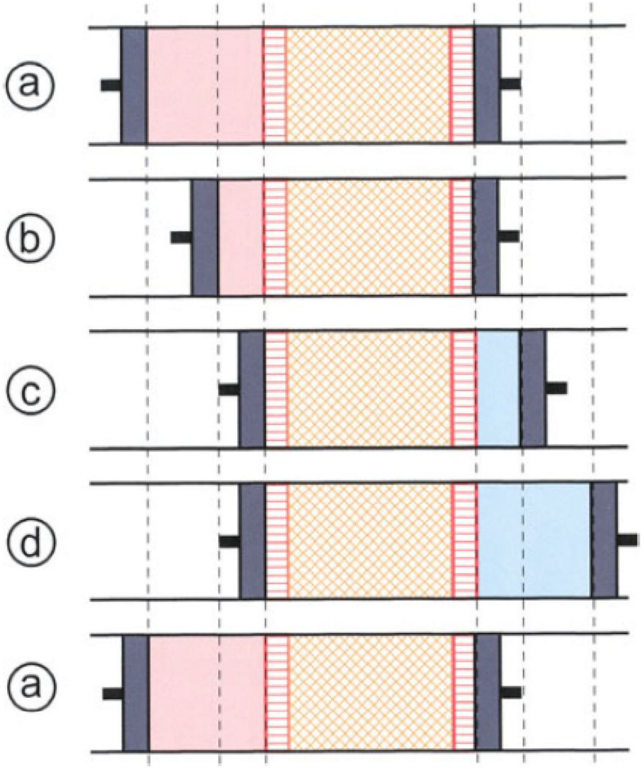
https://en.wikipedia.org/wiki/Stirling_engine

Stirling Coolers

$$\xi = \frac{T_\mathsf{L}}{T_\mathsf{a} - T_\mathsf{L}}$$







The thermal contact with the surroundings at the temperatures T_a and T_L is supposed to be perfect so that the compression and expansion are isothermal

https://en.wikipedia.org/wiki/File:Schematic_Stirling_Cooler.jpg

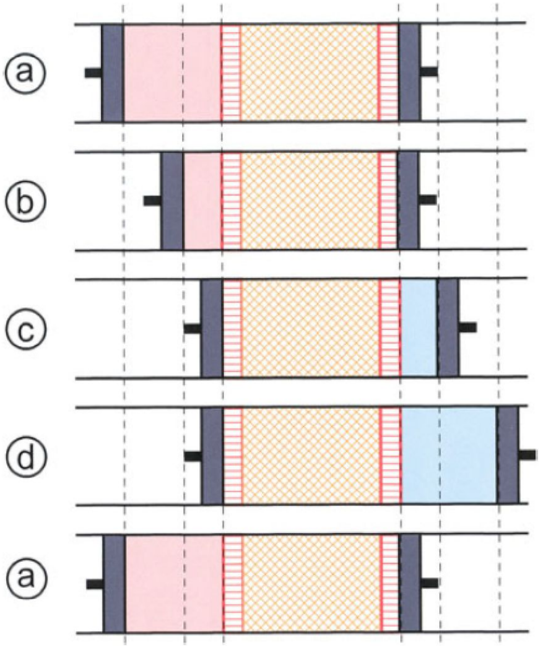
Stirling Coolers

1. From a to b. The warm piston moves to the right over a certain distance while the position of the cold piston is fixed. The compression at the hot end is isothermal by definition, so a certain amount of heat *Q*a is given off to the surroundings at temperature *T*a.

2. From b to c. Both pistons move to the right so that the volume between the two pistons remains constant. The gas enters the regenerator at the left with temperature *T*a and leaves it at the right with temperature T_L. During this part of the cycle heat is given off by the gas to the regenerator material. During this process the pressure drops and heat has to be supplied to the compression and expansion spaces to keep the temperatures constant.

3. From c to d. The cold piston moves to the right while the position of the warm piston is fixed. The expansion is isothermal so heat *Q*L is taken up from the application.

4. From d to a. Both pistons move to the left so that the total volume remains constant. The gas enters the regenerator at the right with temperature T_L and leaves it at the left with *T*a so heat is taken up from the regenerator material. During this process the pressure increases and heat has to be extracted from the compression and expansion spaces to keep the temperatures constant. In the end of this step the state of the cooler is the same as at the start.



Stirling cooler

$$\frac{\dot{Q}_a}{T_a} = \frac{\dot{Q}_L}{T_L}.$$

$$\xi = \frac{\dot{Q}_L}{P_H - P_L} = \frac{T_L}{T_a - T_L}.$$

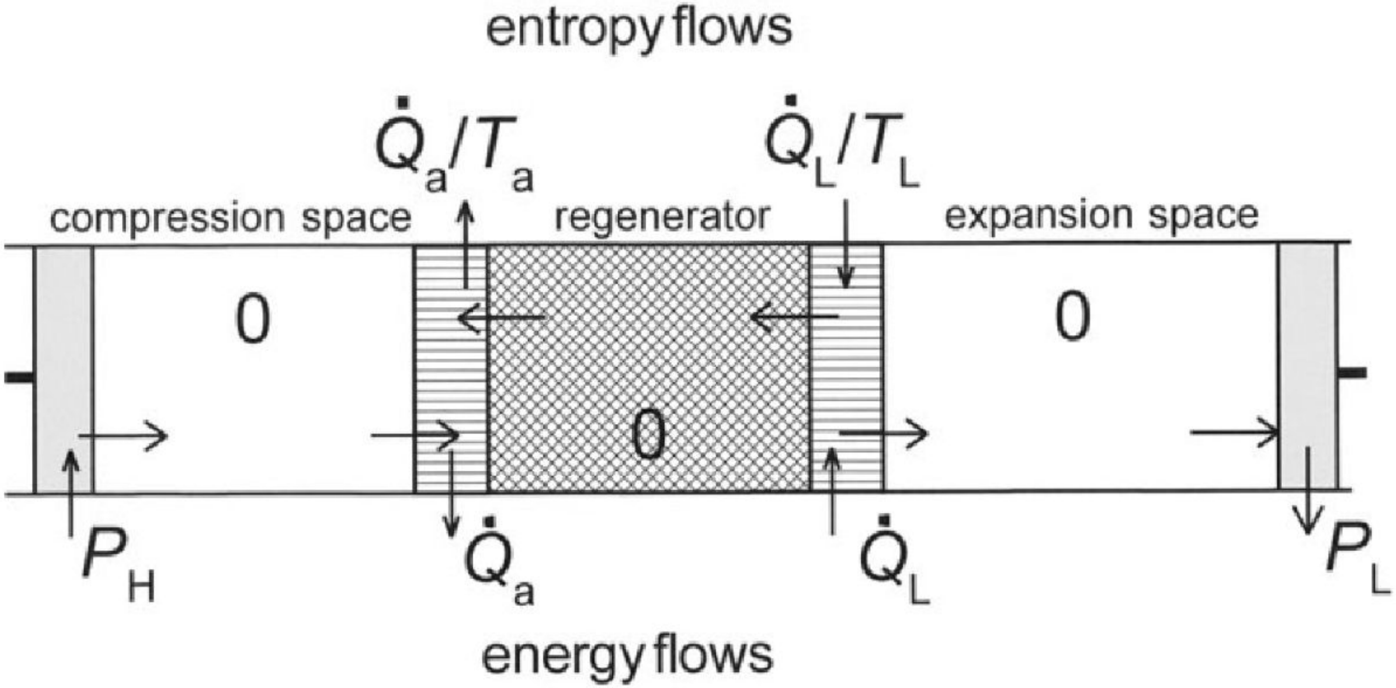
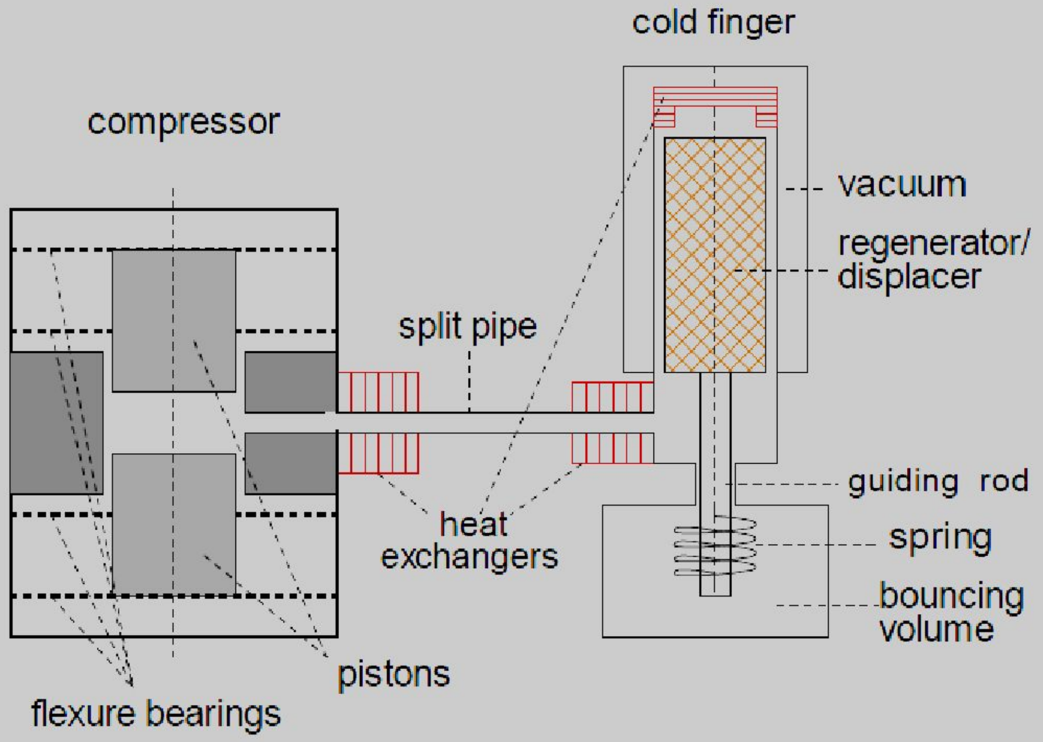
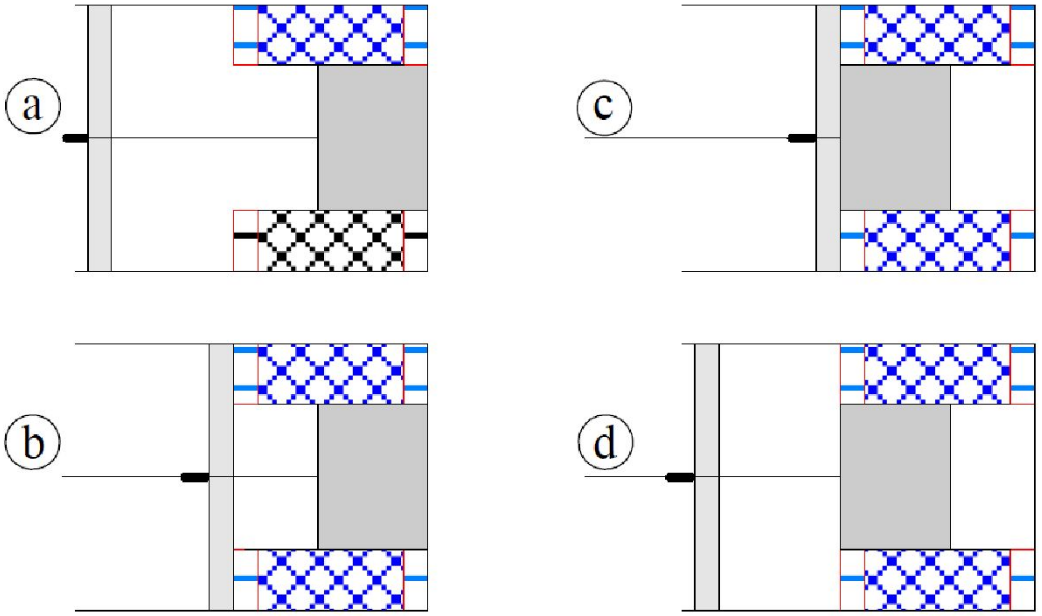


Fig. 9 Schematic diagram of a Stirling cooler. The system has one piston at ambient temperature T_a and one piston at low temperature T_L. The upper half shows the entropy flows and the lower half the energy flows

free-piston Stirling cooler



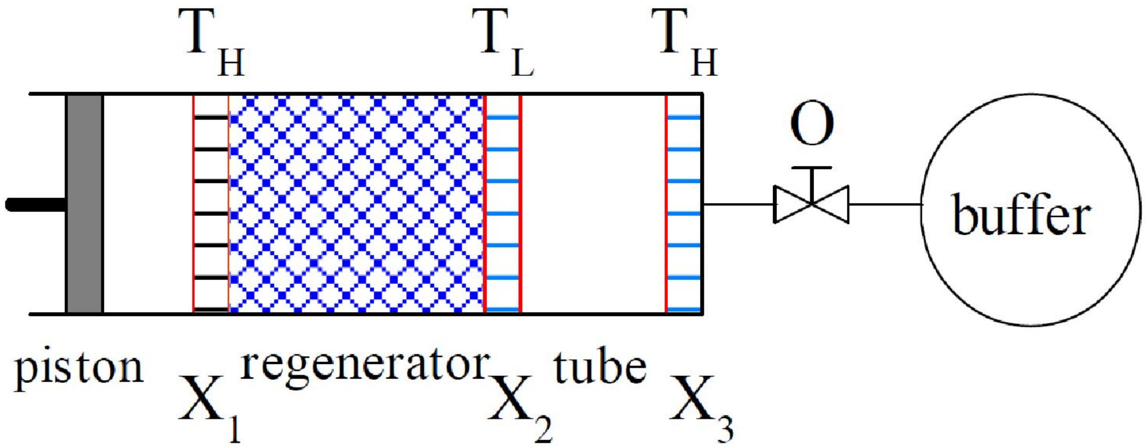
Displacer-type Stirling coolers



Modified Stirling cycle. The cold piston is replaced by a displacer.

PULSE-TUBE REFRIGERATORS (PTRs)

Stirling type single-orifice PTR



From left to right the system consists of a compressor with moving piston (piston), the after cooler (X_1), a regenerator, a low-temperature heat exchanger (X_2), a tube (tube), a second room-temperature heat exchanger (X_3), an orifice (O), and a buffer.
The cooling power is generated at the low temperature T_L. Room temperature is T_H.

In this Section all flow resistances are neglected except from the orifice.
The system is filled with helium at an average pressure of typically 20 bar.
The part in-between the heat exchangers X_1 and X_3 is below room temperature.
It is contained in a vacuum chamber for thermal isolation.

Some remarks…

The piston moves the gas back and forth and generates a varying pressure in the system. The pressure varies smoothly.

The operating frequency typically is 1 to 50 Hz.

Acoustic effects, such as travelling pressure waves, or fast pressure changes (pulses), are absent.
The operation of PTR's has nothing to do with "pulses“… Wrong name!!!!

In the regenerator and in heat exchangers the gas is in good thermal contact with its surroundings *while in the tube the gas is thermally isolated*.

Thermodynamics…

Gas elements inside the tube are compressed or expanded adiabatically and reversibly, so their entropy is constant.

Using the expression for the molar entropy S_m of the gas

$$T\mathrm{d}S_\mathrm{m} = C_\mathrm{p}\mathrm{d}T - T\alpha_\mathrm{V} V_\mathrm{m}\mathrm{d}p$$

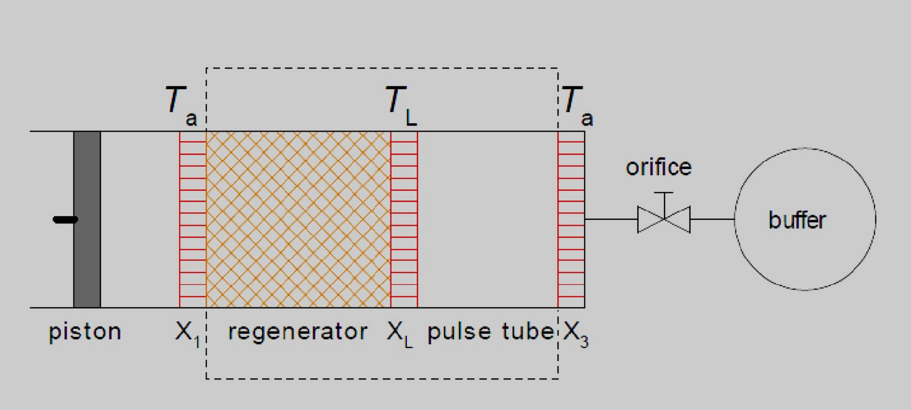
with T the temperature, C_p the molar heat capacity at constant pressure, $\boldsymbol{\alpha_V}$ **the volumetric thermal expansion coefficient** given by

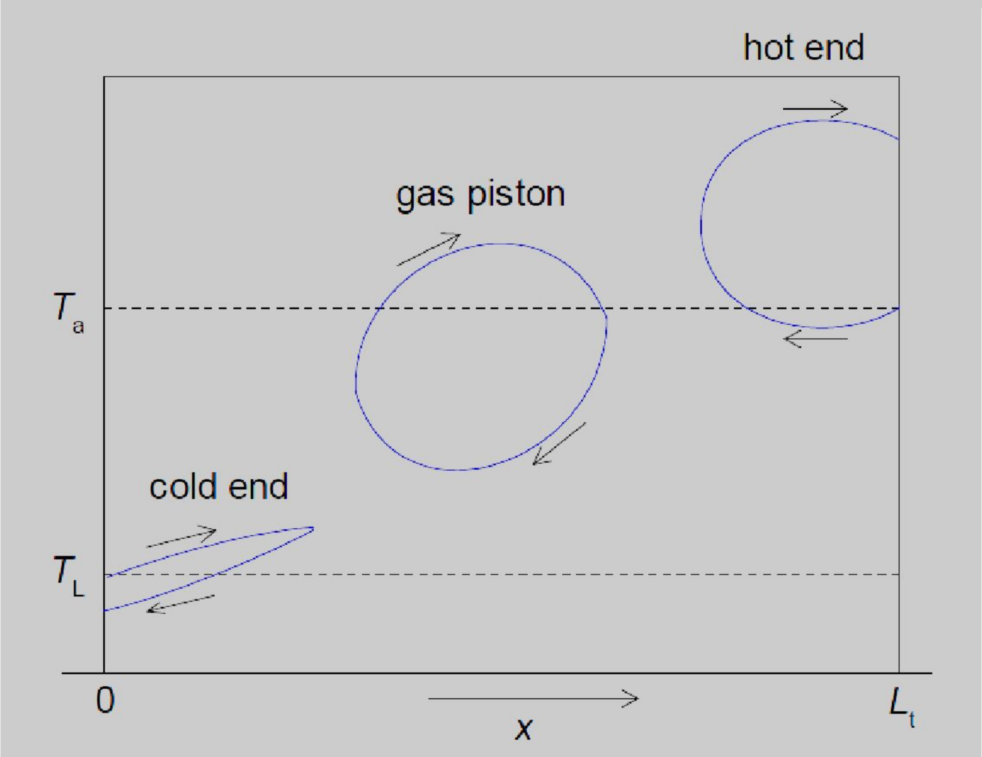
$$\alpha_\mathrm{V} = \frac{1}{V_\mathrm{m}}\left(\frac{\partial V_\mathrm{m}}{\partial T}\right)_\mathrm{p}$$

V_m the molar volume, and p the pressure. From Eq.(1), with $\delta S_m = 0$, we see that the temperature variation δT is related to a pressure variation δp according to

$$\delta T = \frac{T\alpha_\mathrm{V} V_\mathrm{m}}{C_\mathrm{p}}\delta p \quad (S_\mathrm{m}\ \text{constant})$$

Usually $\boldsymbol{\alpha_V > 0}$. This well-known fact means that **compression leads to heating and expansion to cooling**. This fact is the basis for the operation of many types of coolers.





Temperature-position curves of two gas elements (one at the cold end and one at the hot end)

Left : a gas element enters the tube at temperature T_L and leaves it at a lower temperature hence producing cooling.
Right : a gas element enters the tube at temperature T_H and leaves it at a higher temperature producing heating.

At the **hot end** gas flows from the buffer via the orifice into the tube with a temperature T_H if the pressure p_t is below the pressure in the buffer p_B ($p_t < p_B$).

If $p_t = p_B$ the gas at the hot end comes to a halt.
If $p_t > p_B$ the gas moves to the hot end of the tube and through the heat exchanger X and the orifice into the buffer.

So gas elements enters the tube if $p_t < p_B$ and leaves the tube if $p_t > p_B$.
So the final pressure is larger than the initial pressure.
Consequently the gas leaves the tube with a temperature higher than the initial temperature T_H.
Heat is released via the heat exchanger X3 to the surroundings and the gas flows to the orifice at ambient temperature.

At the **cold end** of the tube the gas leaves the cold heat exchanger X and enters the tube when the pressure is high and temperature T_L. It returns to X when the pressure is low and the temperature is below T_L. Hence producing cooling.

The analysis of the situation at the cold end is a bit more complicated due to the fact that the velocity at the cold end is determined by the velocity of the gas at the hot end and by the elasticity of the gas column in the tube. Still the situation is basically the same.

Ideal regenerators

The thermodynamic and hydrodynamic properties of regenerators usually are extremely complicated.
In many cases it is necessary to make simplifying assumptions.
The degree of idealization may differ from case to case.
In its most extreme form in an ideal regenerator:

1. the heat capacity of the matrix is much larger than of the gas;
2. the heat contact between the gas and the matrix is perfect;
3. the gas in the regenerator is an ideal gas;
4. the flow resistance of the matrix is zero;
5. the axial thermal conductivity is zero;
6. sometimes it is also assumed that the void volume of the matrix is zero.

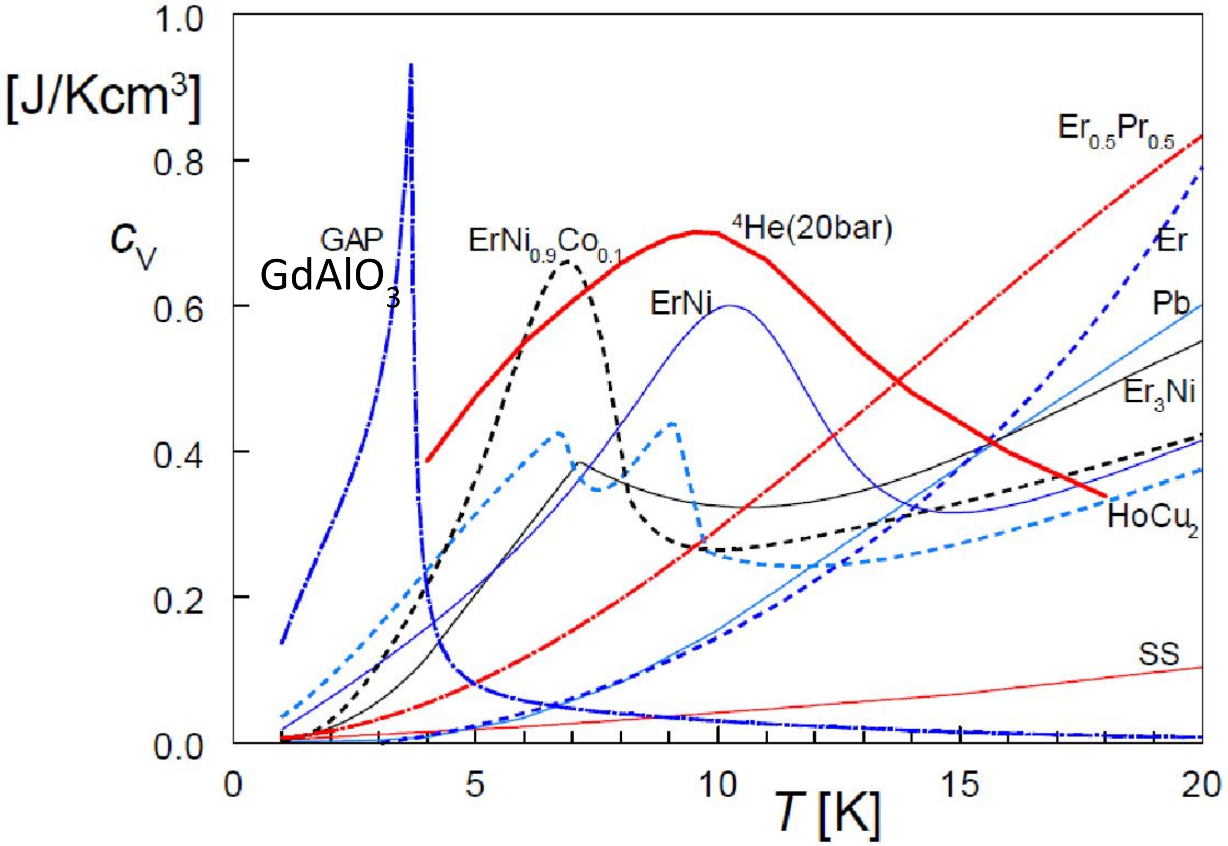
Depending on the situation one or more assumptions may be dropped. Usually it is replaced by another assumption with a less rigorous nature.

If conditions 1 and 2 are satisfied then the gas temperature at a certain point in the regenerator is constant.

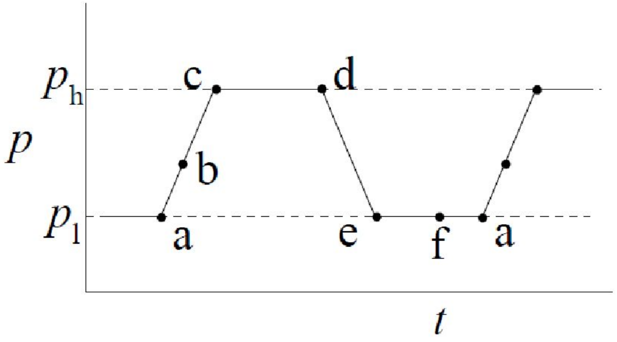
If, in addition, condition 3 is satisfied as well then the average enthalpy flow in the regenerator is zero.

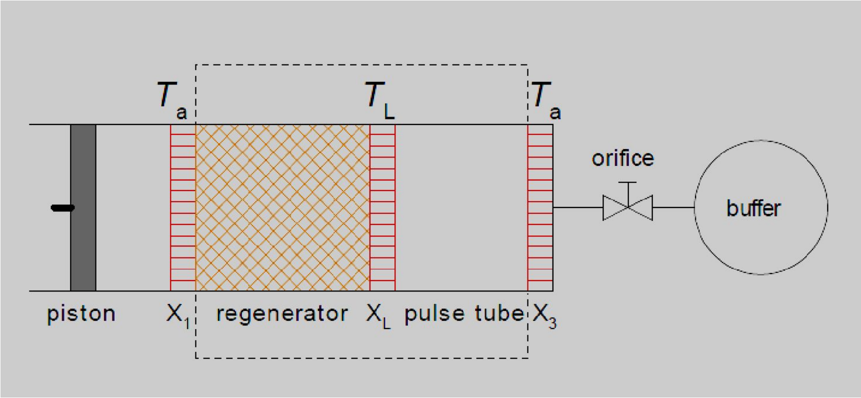
If conditions 2, 4,and 5 are satisfied there are no irreversible processes in the regenerator.

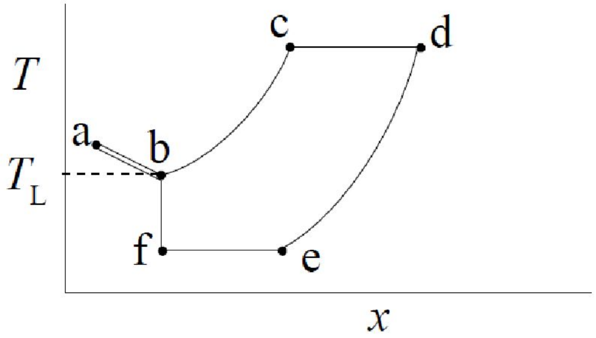
Regenerator: materials



An idealized cycle







The figure illustrates the cooling process at the cold end in a somewhat idealized cycle.
The pressure in the tube is assumed to vary in four steps:

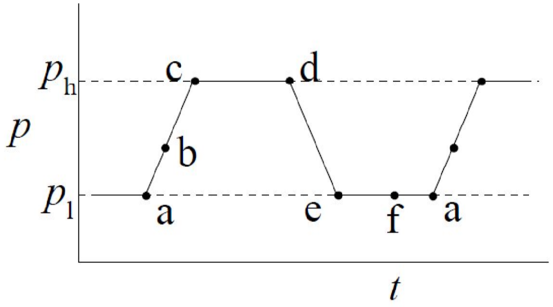
1. **from a via b to c**. The piston moves to the right with the orifice is closed. The pressure rises.

2. **c to d.** The orifice is opened so that gas flows from the tube to the buffer. At the same time the piston moves to the right in such a way that the pressure in the tube remains constant.

3. **d to e.** The piston moves to the left with the orifice is closed. The pressure drops.

4. **e via f to a**. The orifice is opened so that gas flows from the buffer into the tube. At the same time the piston moves to the left so that the pressure in the tube remains constant.

Now we follow a gas element that is inside the regenerator at the start of the cycle (point (a)).



a to b: When the pressure rises the gas element moves to the right but its temperature remains at the local temperature due to the good heat contact with the regenerator material.
At point (b) our gas element leaves the regenerator and X_2 and enters the tube with the temperature T_L of the heat exchanger X_2. The pressure is p_b.

b to c: Now the gas element is thermally isolated and its temperature rises together with the pressure
while it moves to the right.

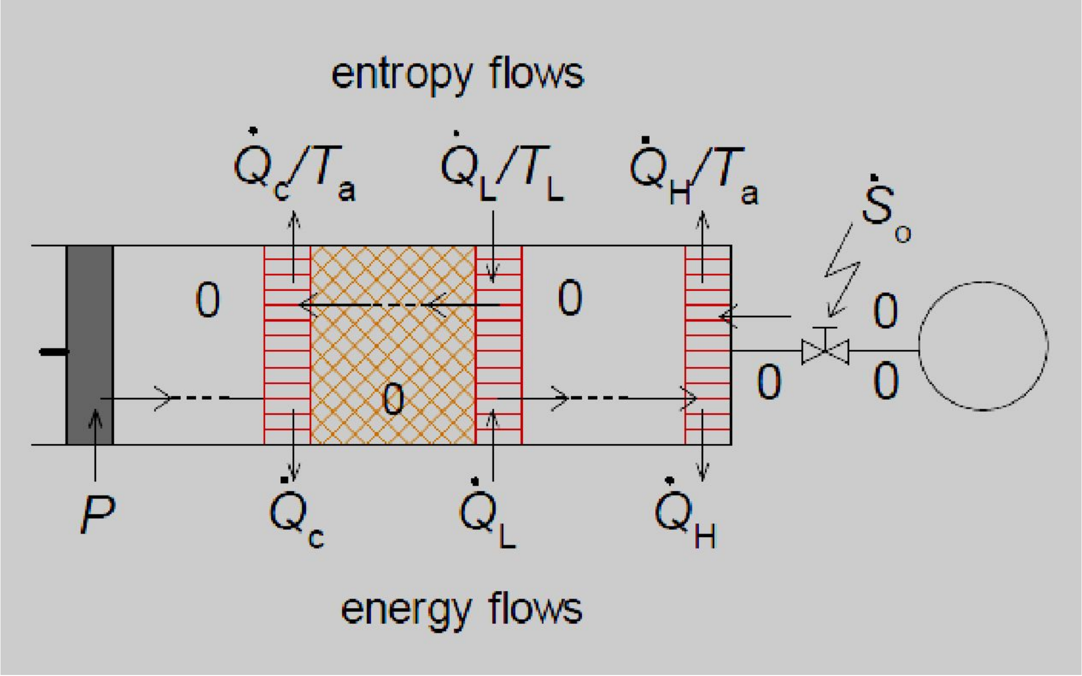
c to d: The gas element moves to the right. The pressure is constant so the temperature is constant.

d to e: When the pressure drops the gas element moves to the left. As it is thermally isolated its temperature drops to a value below T_L since $p_e < p_b$:

e to f : The gas element moves to the left. The pressure is constant so the temperature is constant. At point (f) the gas element enters the heat exchanger X2. In passing X2 the gas extracts heat (produces cooling) from X2. The gas element warms up to the temperature TL.

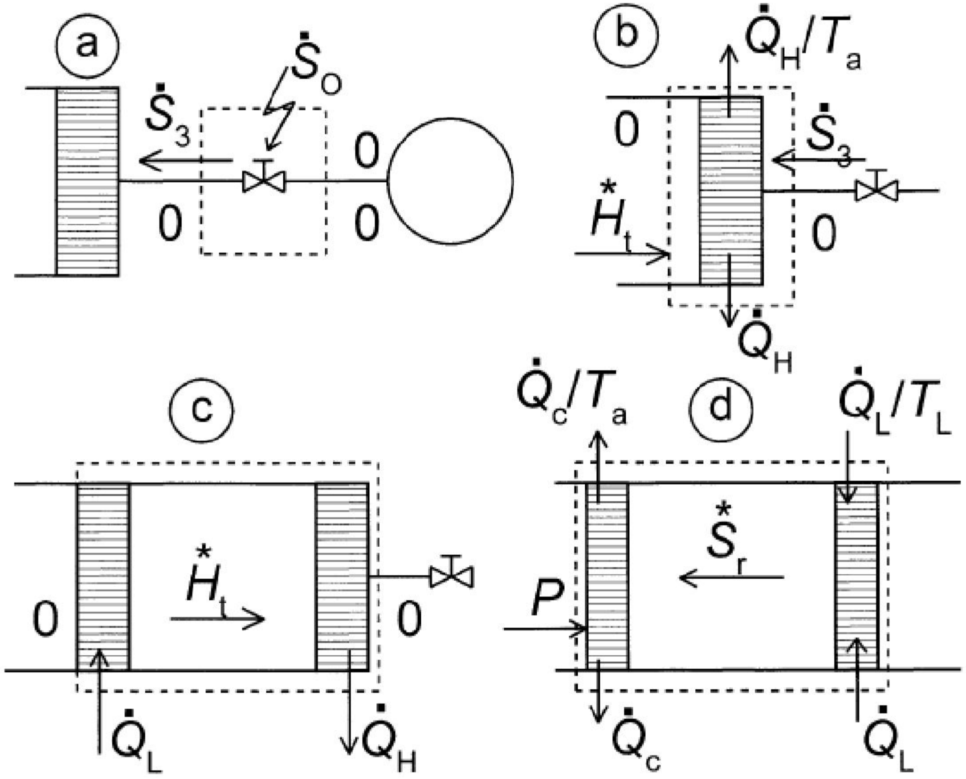
f to a: The gas element is inside the regenerator and moves with the local temperature back to its original position.

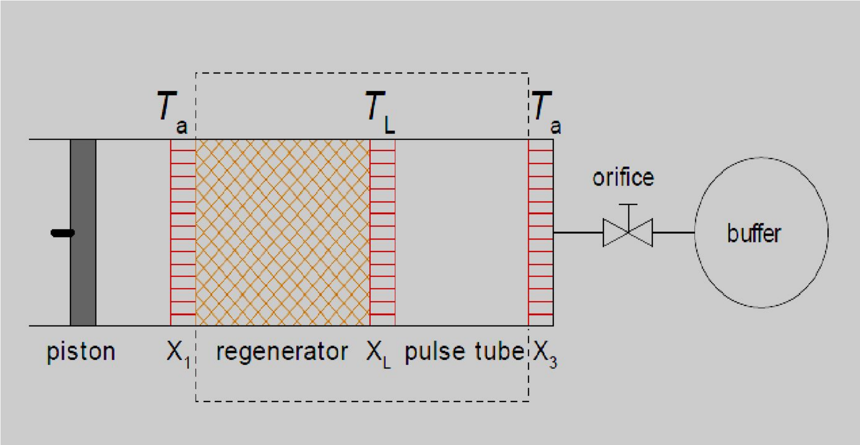
Thermodynamics of PTR’s



Ideal PTR: dissipation only occurs in the orifice

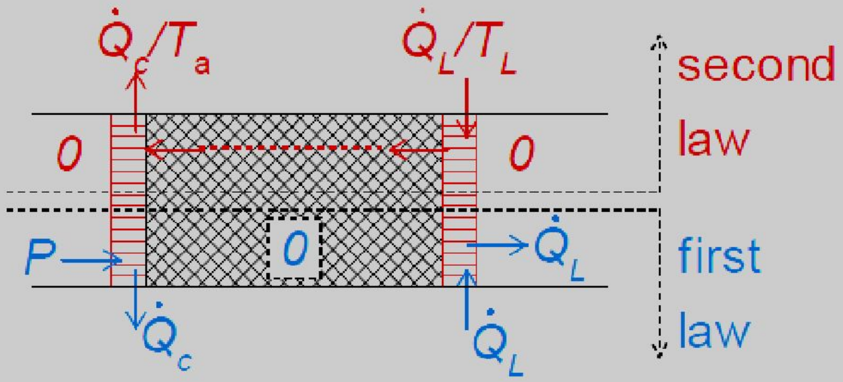
Thermodynamics of PTR’s





Thermodynamic systems containing the orifice (a), the heat exchanger X3 (b), the pulse tube and its heat exchangers (c), and the regenerator and its heat exchangers (d) average enthalpy flow in the regenerator

$$\overline{\overset{*}{H}}_{\mathrm{r}} = 0$$

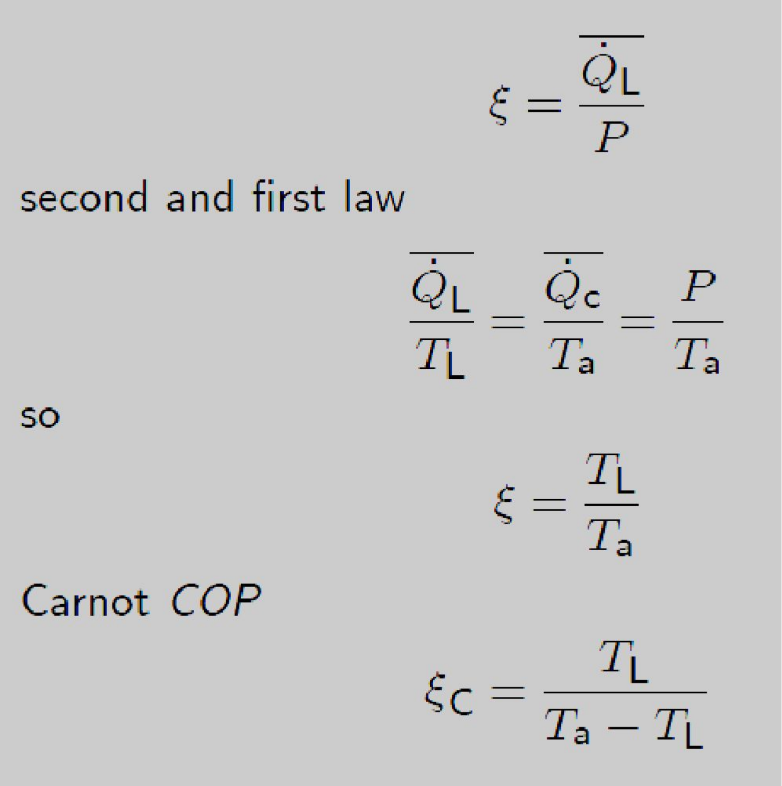


$$\overline{\overset{*}{S}}_{\mathrm{c}} = \overline{\overset{*}{S}}_{\mathrm{t}} = 0$$

ideal case

$$\frac{\overline{\dot{Q}_{\mathrm{L}}}}{T_{\mathrm{L}}} = \frac{\overline{\dot{Q}_{\mathrm{c}}}}{T_{\mathrm{a}}}$$

Coefficient Of Performance (COP)



$$\xi = \frac{\overline{\dot{Q}_{\mathrm{L}}}}{P}$$

second and first law

$$\frac{\overline{\dot{Q}_{\mathrm{L}}}}{T_{\mathrm{L}}} = \frac{\overline{\dot{Q}_{\mathrm{c}}}}{T_{\mathrm{a}}} = \frac{P}{T_{\mathrm{a}}}$$

so

$$\xi = \frac{T_{\mathrm{L}}}{T_{\mathrm{a}}}$$

Carnot *COP*

$$\xi_{\mathrm{C}} = \frac{T_{\mathrm{L}}}{T_{\mathrm{a}} - T_{\mathrm{L}}}$$

PULSE-TUBE REFRIGERATORS: first machines

Pulse-tube refrigerators have their origin in an observation that W. E. Gifford made, while working on the compressor in the late 1950's. He noticed that a tube, which branched from the high-pressure line and was closed by a valve, was hotter at the valve than at the branch.

He recognized that there was a **heat pumping mechanism** that resulted from **pressure pulses in the line**. In 1963 Gifford together with his research assistant R. C. Longsworth introduced the Basic Pulse-Tube Refrigerator (BPTR).

The BPTR has not so much in common with the modern PTRs. The cooling principle of the BPTR is the surface heat pumping, which is based on the exchange of heat between the working gas and the pulse tube walls.

The lowest temperature, reached by Gifford and Longsworth was 124 K with a single-stage PTR and 79 K with a two-stage PTR.

The PTR has no moving parts in the low-temperature region, and, therefore, has a **long lifetime and low mechanical and magnetic interferences**.

A typical average pressure in a PTR is 10 to 25 bar, and a typical pressure amplitude is 2 to 7 bar.

A piston compressor (in case of a *Stirling type PTR*) or a combination of a compressor and a set of switching valves (*GM type PTR*) are used to create pressure oscillations in a PTR.

The main breakthrough came in 1984, when Mikulin and his co-workers invented the Orifice Pulse Tube Refrigerator (OPTR) [6]. A flow resistance, the orifice, was inserted at the warm end of the pulse tube to allow some gas to pass to a large reservoir. With a single-stage configuration of the OPTR Mikulin achieved a low temperature of 105 K, using air as the working gas.

Soon afterwards R.Radebaugh reached 60 K with a similar device, using helium [7]. For the first time since the invention of the PTR its performance became comparable to the Stirling cooler.

In 1990 Zhu et al. connected the warm end of the pulse tube with the main gas inlet by a tube, containing a second orifice [8]. Thus, a part of the gas could enter the pulse tube from the warm end, by-passing the regenerator. Because of this effect such a configuration of the PTR was called the Double-Inlet Pulse-Tube Refrigerator (DPTR).

In 1994 Y. Matsubara used this configuration to reach a temperature as low as 3.6 K with a three-stage PTR [9].

In 1999 with a three stage DPTR a temperature of 1.78 K was reached at the Low Temperature Group of Eindhoven University of Technology [10].

In 2003 the group of Prof. G. Thummes from Giessen University developed a double-circuit 3He/4He PTR that achieved 1.27 K [11].

Adapted from: PhD Thesis
Low-temperature cryocooling / by Irina Tanaeva. -
Eindhoven : Technische Universiteit Eindhoven, 2004. –
ISBN 90-386-2005-5

REFERENCES

1. McClintock, P. V. E., Meredith, D. J., Wigmore, J. K., “Matter at low temperatures”, John Wiley & Sons, New York, 1984.
2. Good, J., Hodgson, S., Mitchell, R., and Hall, R., “Helium free magnets and research systems”, Cryocoolers **12**, 2003, pp. 813-816.
3. Walker, G., “Cryocoolers”, Plenum Press, New York and London, 1983.
4. Gifford, W.E. and Longsworth, R. C., “Pulse tube refrigeration”, Trans. ASME, 1964, pp. 264-268.
5. Longsworth, R. C., “An experimental investigation of pulse tube refrigeration heat pumping rates”, Advances in Cryogenic Engineering **12**, 1967, pp. 608-618.
6. Mikulin, E.I., Tarasov, A.A., and Shkrebyonock, M., P., “Low-temperature expansion pulse tubes”, Advances in Cryogenic Engineering **29**, 1984, pp. 629-637.
7. Radebaugh, R., Zimmerman, J., Smith, D., R., and Louie, B., “Comparison of three types of pulse tube refrigerators: New methods for reaching 60 K”, Advances in Cryogenic Engineering **31**, 1986, pp. 779-789.
8. Zhu, Sh., Wu, P., and Chen, Zh., “Double inlet pulse tube refrigerators: an important improvement”, Cryogenics **30**, 1990, pp. 514-520.
9. Matsubara, Y. and Gao, J., L., “Novel configuration of three-stage pulse tube refrigerator for temperatures below 4 K”, Cryogenics **34**, 1994, pp. 259-262.
10. Xu, M. Y., Waele, A. T. A. M. de, and Ju, Y. L., “A Pulse Tube Refrigerator Below 2 K”, Cryogenics **39**, 1999, pp. 865-869.
11. Jiang, N., Lindemann, U., Giebeler, F., and Thummes, G., “A $_3$He pulse tube cooler operating down to 1.27 K”, Cryogenics **44**, 2004, pp. 809-816.
12. Zia, J. H., “Design and operation of a 4 kW liner motor driven pulse tube cryocooler”, Advances in Cryogenic Engineering **49**, 2004, pp. 1309-1317.

PhD Thesis
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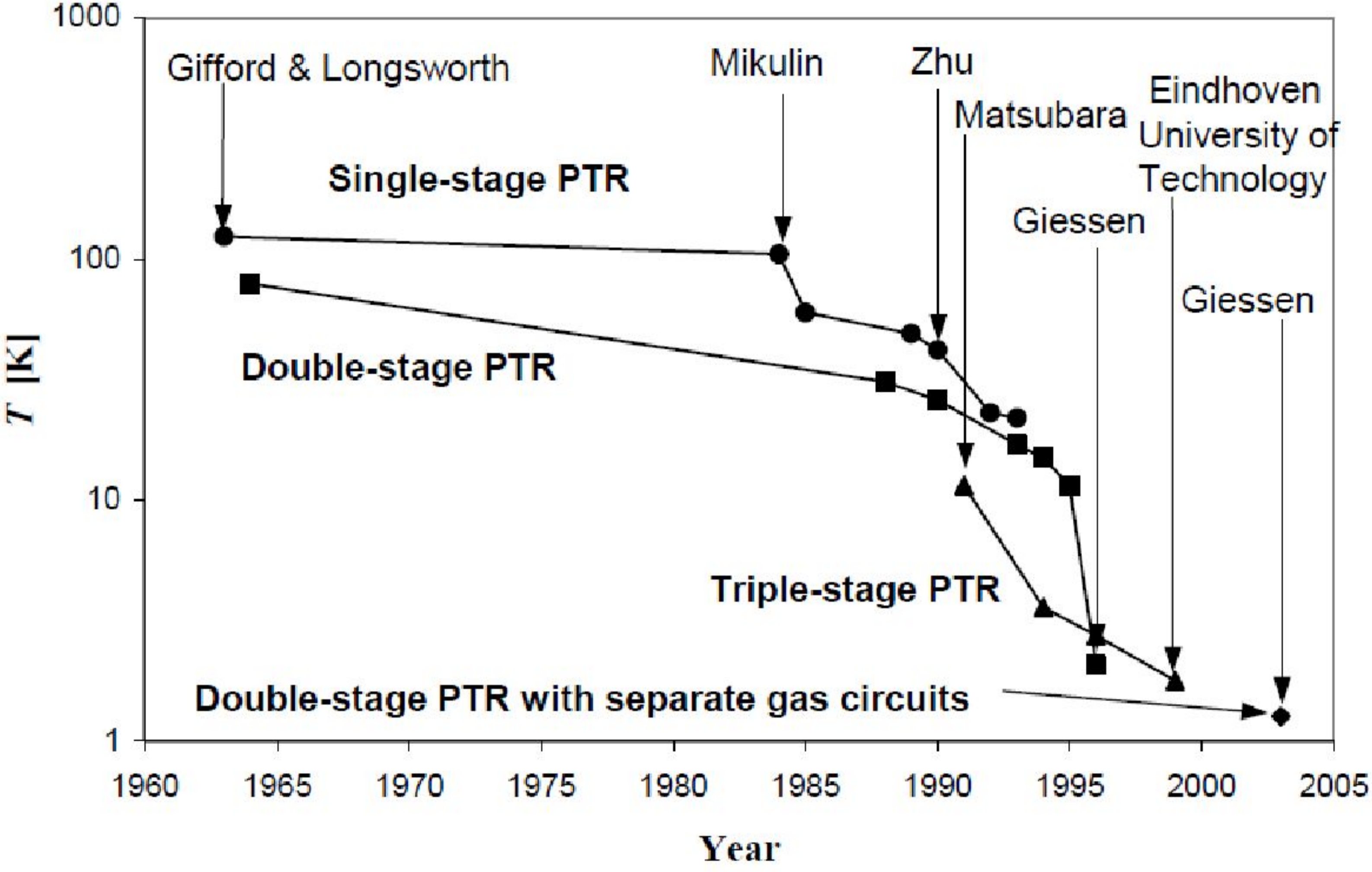
Low temperatures achieved by PT coolers



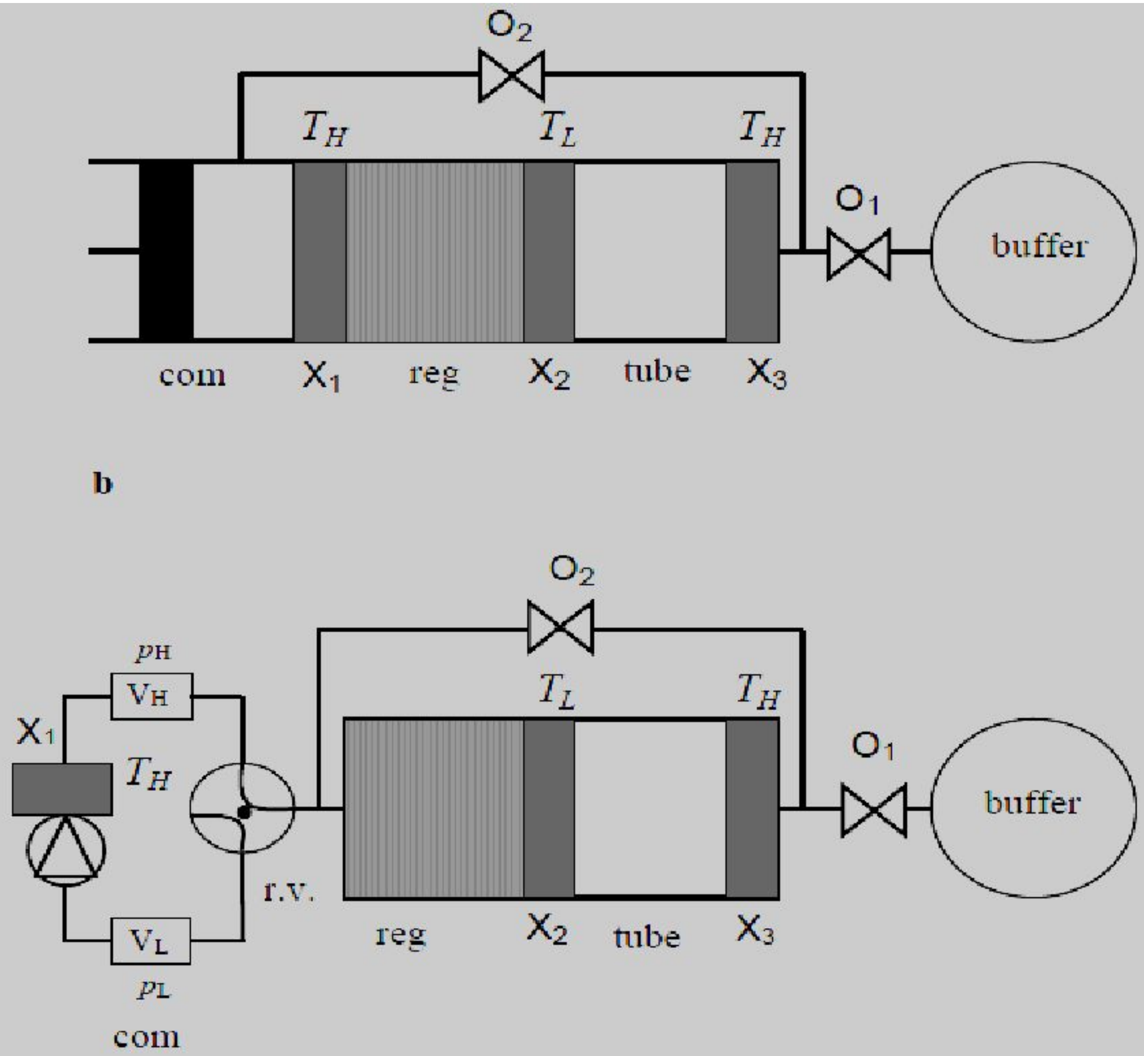


Figure 2. 1. Two types of the PTR. (a) A Stirling-type PTR. From left to right it consists of a compressor (com), an aftercooler (X_1), a regenerator (reg), a cold heat exchanger (X_2), a pulse tube (tube), a hot heat exchanger (X_3), an orifice (O_1), and a buffer. Orifice O_2 connects the hot end of the regenerator and the hot end of the pulse tube. (b) A GM-type PTR. Except for the compressor - rotary valve (r.v.) combination, the main components of the GM-type PTR are the same as of the Stirling PTR.

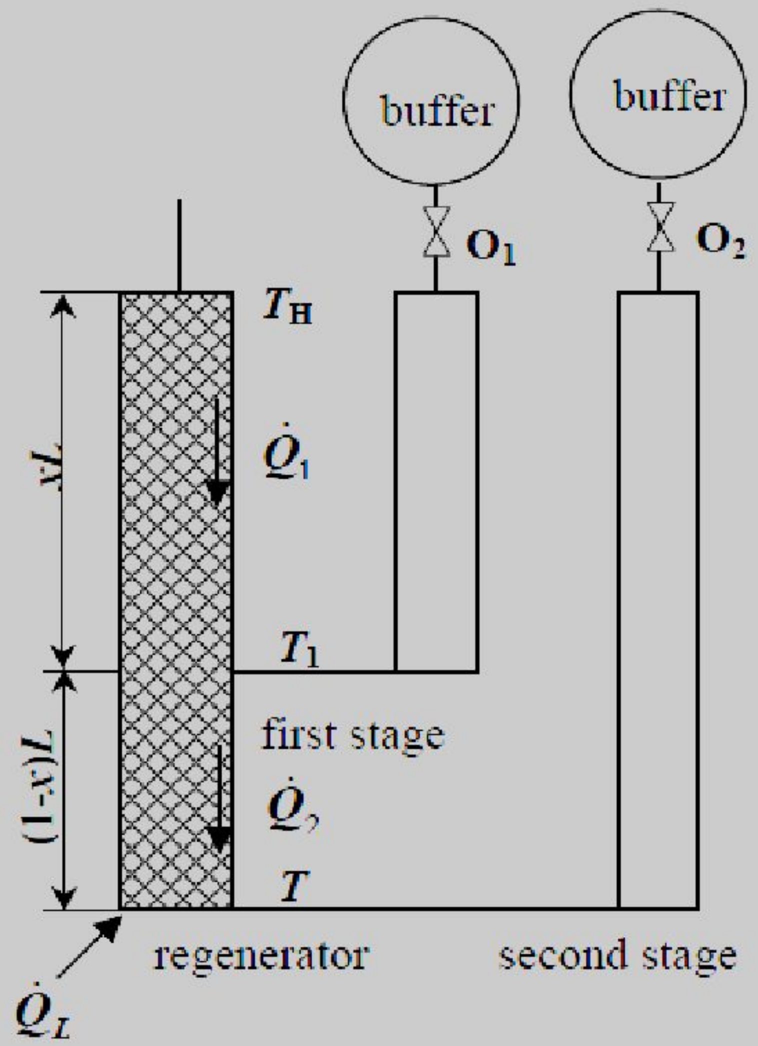


Figure 2. 7. A schematic diagram of a two-stage single-orifice PTR. The PTR has a cooling power $\dot{Q}_L$ at a temperature T_2. The heat flows $\dot{Q}_1$ and $\dot{Q}_2$ are caused by the heat conduction.

Additional cooling power

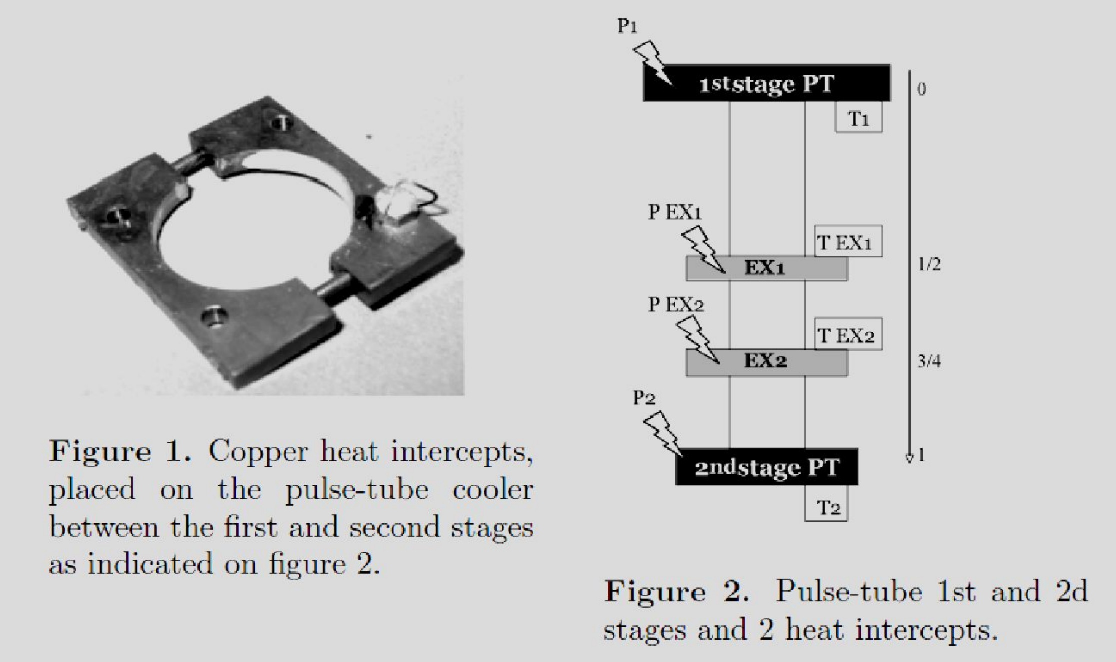
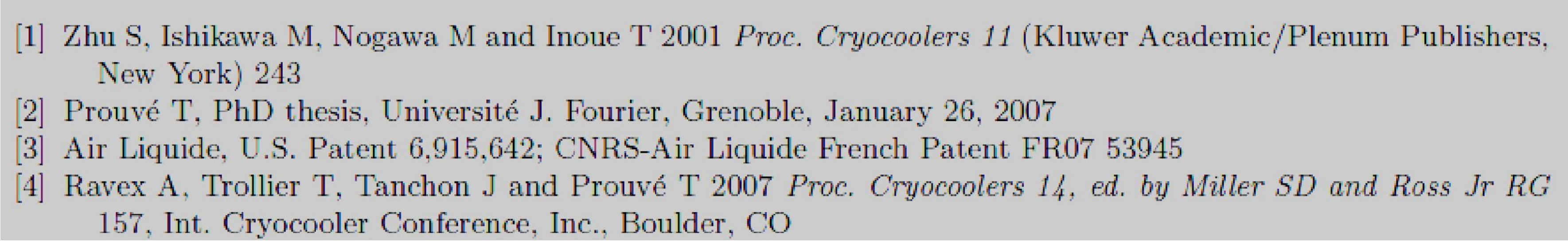


Figure 1. Copper heat intercepts, placed on the pulse-tube cooler between the first and second stages as indicated on figure 2.

Figure 2. Pulse-tube 1st and 2d stages and 2 heat intercepts.


[1] Zhu S, Ishikawa M, Nogawa M and Inoue T 2001 *Proc. Cryocoolers 11* (Kluwer Academic/Plenum Publishers, New York) 243
[2] Prouvé T, PhD thesis, Université J. Fourier, Grenoble, January 26, 2007
[3] Air Liquide, U.S. Patent 6,915,642; CNRS-Air Liquide French Patent FR07 53945
[4] Ravex A, Trollier T, Tanchon J and Prouvé T 2007 *Proc. Cryocoolers 14, ed. by Miller SD and Ross Jr RG* 157, Int. Cryocooler Conference, Inc., Boulder, CO

[5] Experimental results on the free cooling power available on 4K pulse tube coolers
T. Prouvé, H. Godfrin, C. Gianèse, S. Triqueneaux, A. Ravex
J. of Phys. : Conference Series 150, 012038 (2009).

Additional cooling power

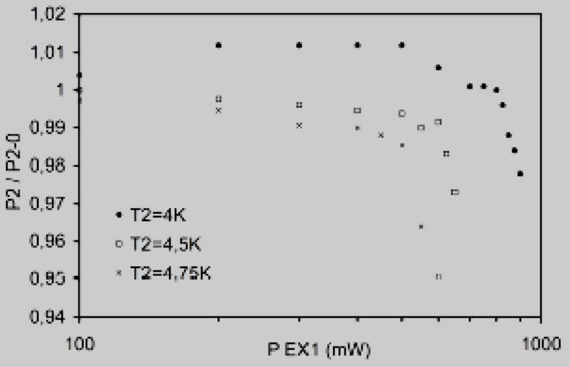


Figure 3. Normalized cooling power of the second stage, as a function of the power P EX1 applied to the the first intercept, for different temperatures T2 of the second stage.

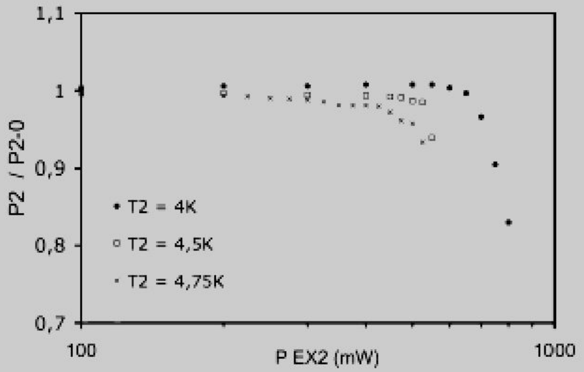


Figure 4. Normalized cooling power of the second stage, as a function of the power P EX2 applied to the the second intercept, for different temperatures T2 of the second stage.

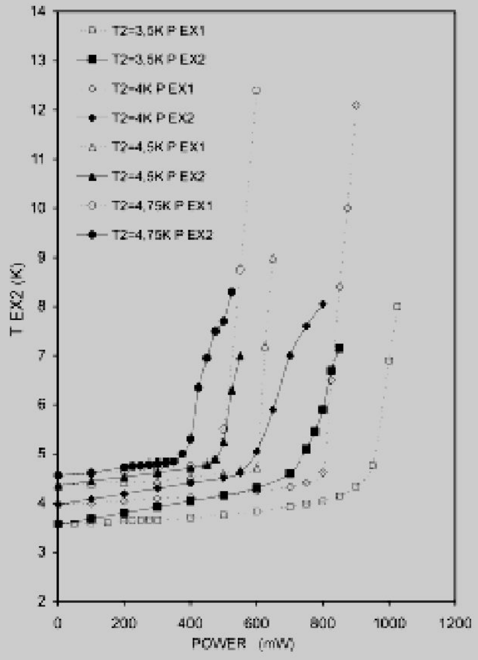


Figure 6. Temperature of the second intercept T EX2 as a function of the power PEX2 applied to this intercept (solid lines and symbols), or to the first intercept P EX1 (dashed line and open symbols) for different temperatures T2 of the second stage.

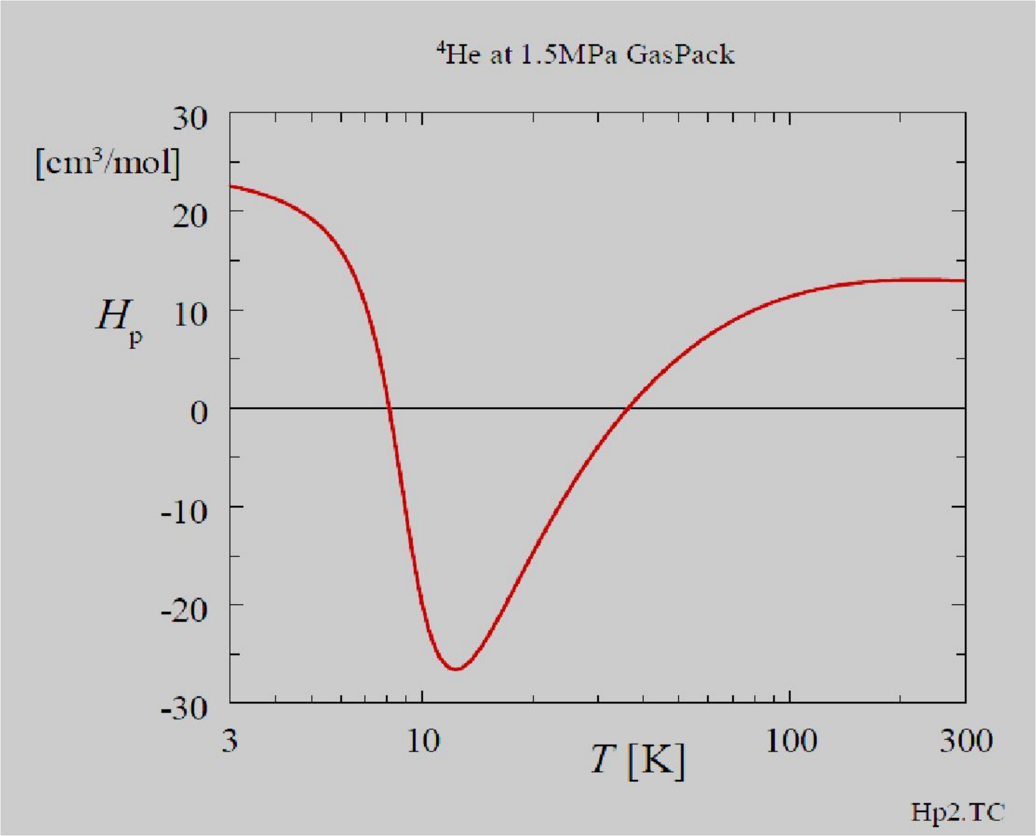
See article below for complete characterization of the cooling power as a function of the heat applied to all exchangers:

Experimental results on the free cooling power available on 4K pulse tube coolers
T. Prouvé, H. Godfrin, C. Gianèse, S. Triqueneaux, A. Ravex
J. of Phys. : Conference Series 150, 012038 (2009).

$$\mathrm{d}H_\mathrm{m} = C_\mathrm{p}\mathrm{d}T + H_\mathrm{p}\mathrm{d}p$$

ideal gas: $H_\mathrm{p} = 0$

"free" cooling power for precooling



Commercial pulse-tubes

Standard 4K Cryomech Single-Stage Pulse Tube Cryorefrigerators

All models have remote-motor options available

PT 10

- 12W @ 80K
- Air or Water Cooled

PT 60

- 60W @ 80K
- Air or Water Cooled

PT 63

- 23W @ 40K
- Air or Water Cooled

PT 90

- 90W @ 80K
- Air or Water Cooled

Standard 4K Cryomech Two-Stage Pulse Tube Cryorefrigerators

All models have remote-motor options available

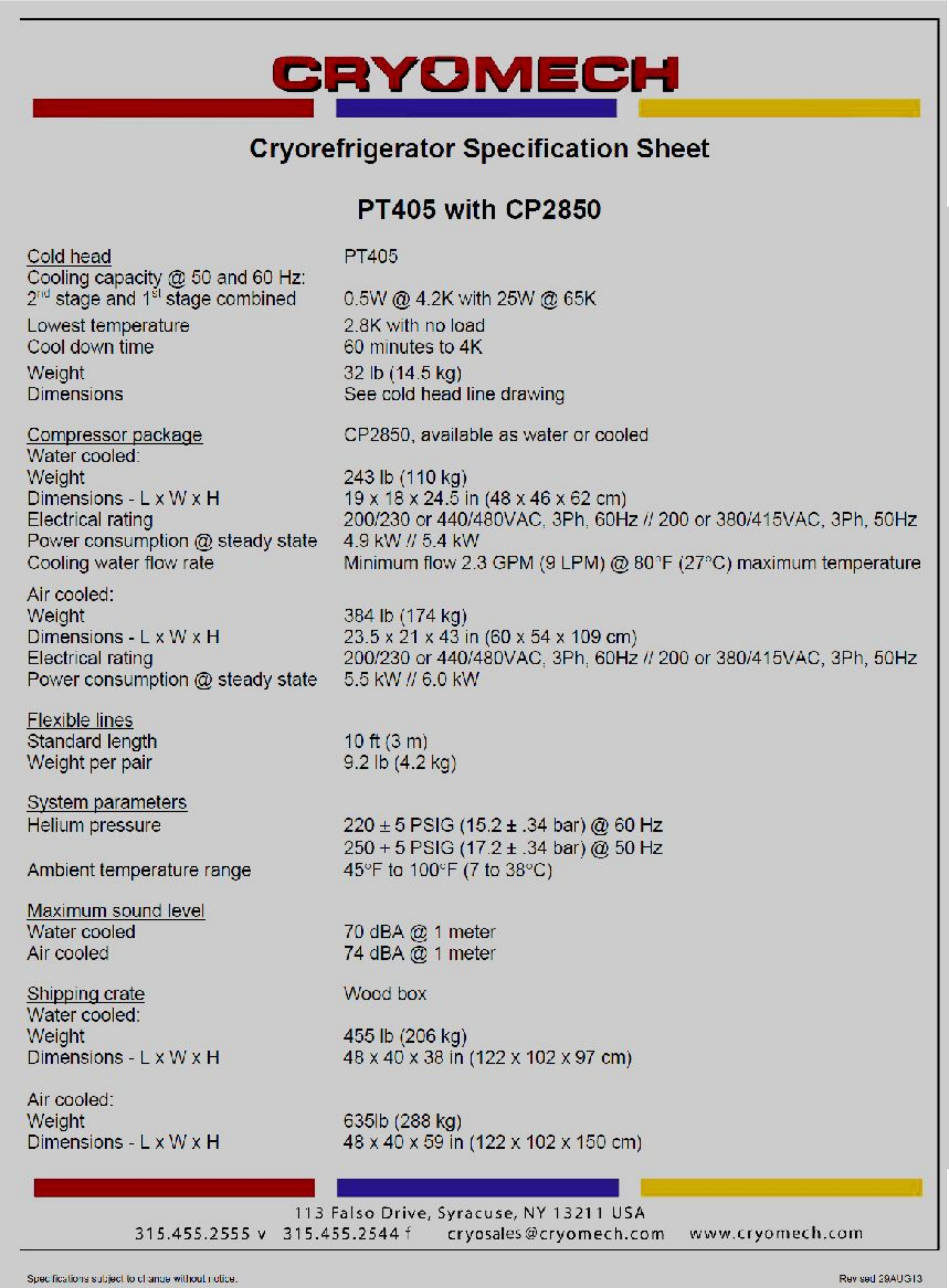
PT 403
First Stage 7W @ 65K
Second Stage 0.25W @ 4.2K
Air or Water Cooled

PT 405
First Stage 25W @ 65K
Second Stage 0.5W @ 4.2K
Air or Water Cooled

PT 407
First Stage 25W @ 55K
Second Stage 0.7W @ 4.2K
Air or Water Cooled

PT 415
First Stage 40W @ 45K
Second Stage 1.5W @ 4.2K

PT 405



CRYOMECH

Cryorefrigerator Specification Sheet

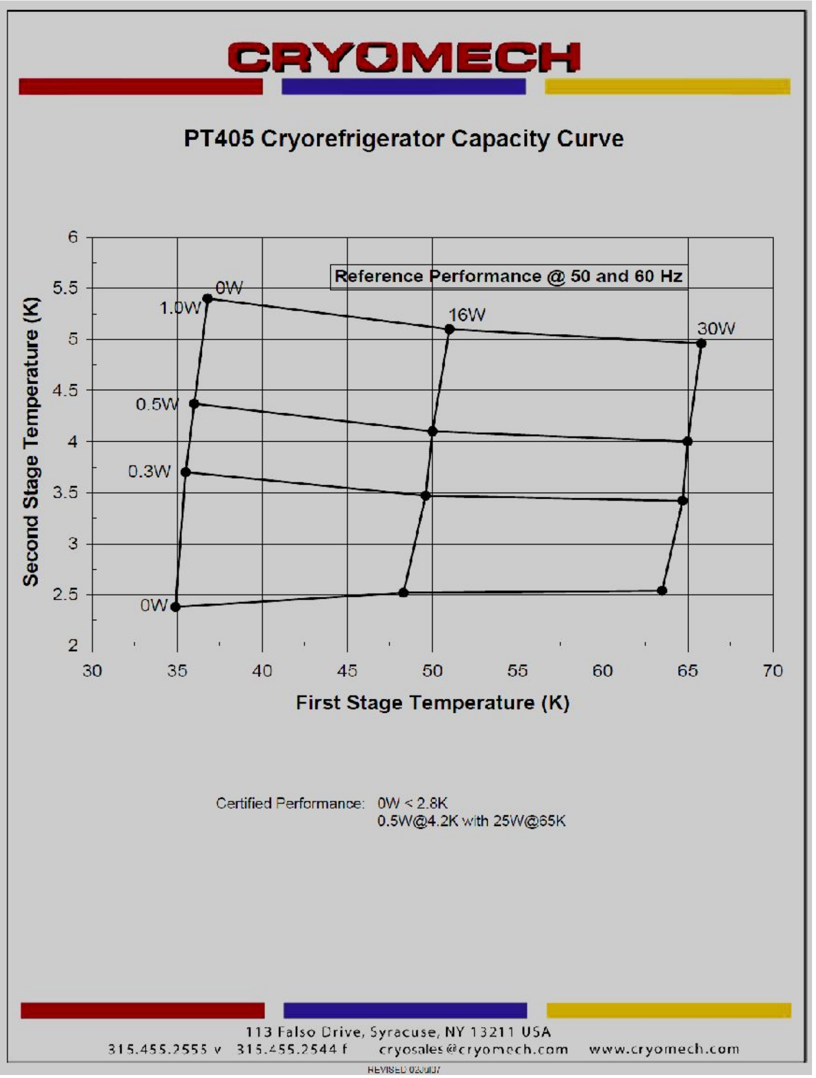
PT405 with CP2850

Cold head	PT405
Cooling capacity @ 50 and 60 Hz: 2nd stage and 1st stage combined	0.5W @ 4.2K with 25W @ 65K
Lowest temperature	2.8K with no load
Cool down time	60 minutes to 4K
Weight	32 lb (14.5 kg)
Dimensions	See cold head line drawing
Compressor package	CP2850, available as water or cooled
Water cooled:	
Weight	243 lb (110 kg)
Dimensions - L x W x H	19 x 18 x 24.5 in (48 x 46 x 62 cm)
Electrical rating	200/230 or 440/480VAC, 3Ph, 60Hz // 200 or 380/415VAC, 3Ph, 50Hz
Power consumption @ steady state	4.9 kW // 5.4 kW
Cooling water flow rate	Minimum flow 2.3 GPM (9 LPM) @ 80°F (27°C) maximum temperature
Air cooled:	
Weight	384 lb (174 kg)
Dimensions - L x W x H	23.5 x 21 x 43 in (60 x 54 x 109 cm)
Electrical rating	200/230 or 440/480VAC, 3Ph, 60Hz // 200 or 380/415VAC, 3Ph, 50Hz
Power consumption @ steady state	5.5 kW // 6.0 kW
Flexible lines	
Standard length	10 ft (3 m)
Weight per pair	9.2 lb (4.2 kg)
System parameters	
Helium pressure	220 ± 5 PSIG (15.2 ± .34 bar) @ 60 Hz 250 + 5 PSIG (17.2 ± .34 bar) @ 50 Hz
Ambient temperature range	45°F to 100°F (7 to 38°C)
Maximum sound level	
Water cooled	70 dBA @ 1 meter
Air cooled	74 dBA @ 1 meter
Shipping crate	Wood box
Water cooled:	
Weight	455 lb (206 kg)
Dimensions - L x W x H	48 x 40 x 38 in (122 x 102 x 97 cm)
Air cooled:	
Weight	635lb (288 kg)
Dimensions - L x W x H	48 x 40 x 59 in (122 x 102 x 150 cm)

113 Falso Drive, Syracuse, NY 13211 USA
315.455.2555 v 315.455.2544 f cryosales@cryomech.com www.cryomech.com

Specifications subject to change without notice.

Revised 29AUG13



CRYOMECH

PT405 Cryorefrigerator Capacity Curve

Certified Performance: 0W < 2.8K
0.5W@4.2K with 25W@65K

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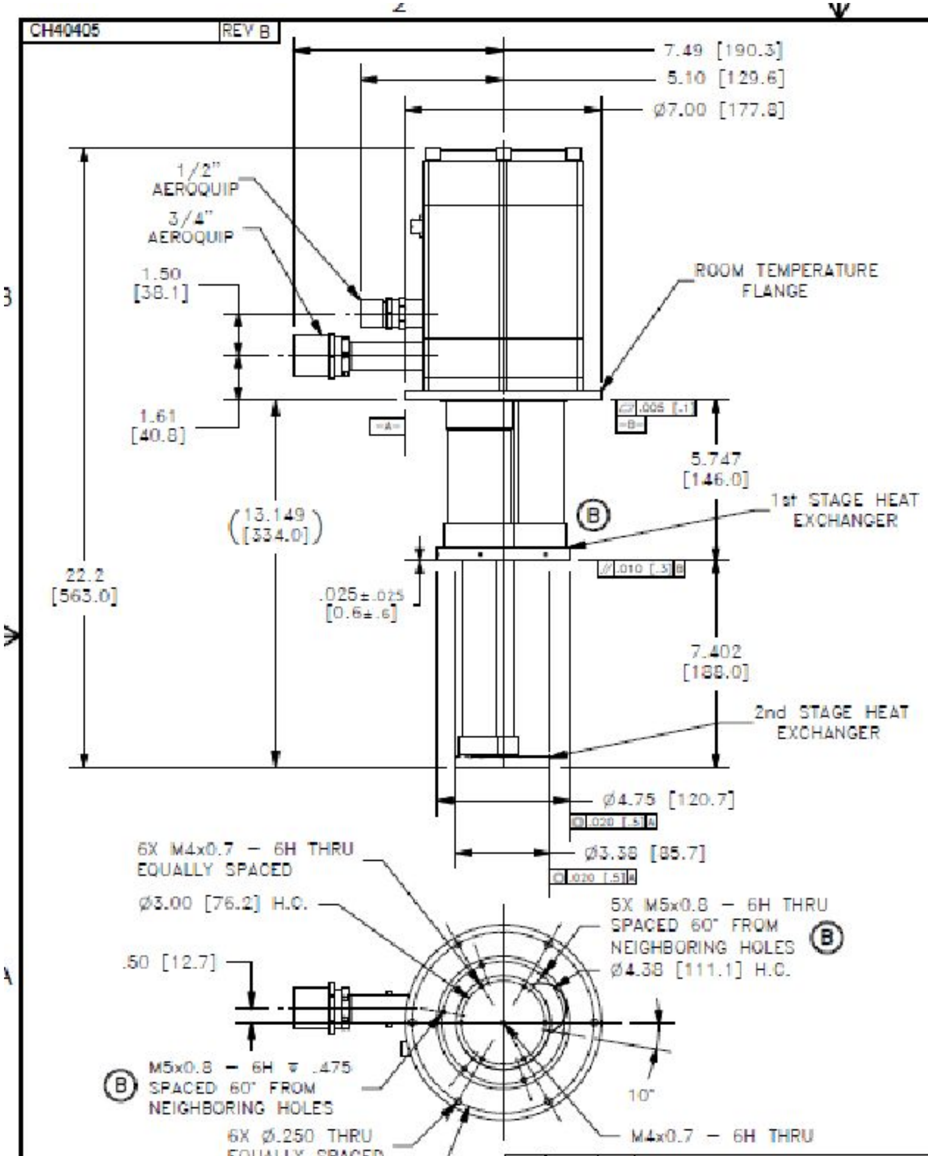
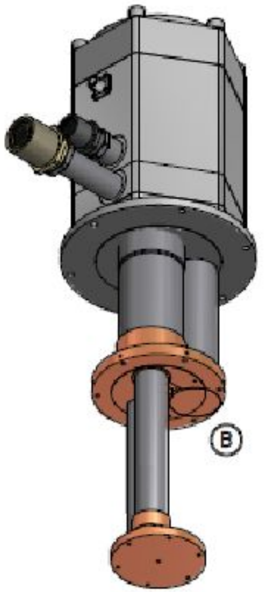
CH40405 REV B





SPECIAL NOTE: COLDHEAD MUST BE OPERATED COLD END DOWN

THIS DRAWING SUPERSEDES CHPT405

ECN	ZONE	REV	DESCRIPTION	DATE
2673	–	B	UPDATED BTA:RESURRECT 4pc 1st STAGE HT EX	19AUG11
2252	–	A	UPDATED BTA:ADDED 1pc 1st STAGE HT EX	05OCT10
1100	–	–	INITIAL RELEASE	25MAY06

THIRD ANGLE PROJECTION

CRYOMECH, INC
113 FALSO DRIVE SYRACUSE, N.Y. 13211
Tel: (315)455-2555 Fax:(315)455-2544

ALL DIMENSIONS ARE IN INCHES
TOLERANCES: FRACTIONS ±1/32 DECIMALS XXX=±.005 XX =±.01 X=±.1
ANGLES ±.5° ALLOVER FINISH 25 Ra
CONCENTRICITY .005 TOTAL INDICATOR RUN OUT
PERPENDICULAR ±.002 (UNLESS OTHERWISE SPECIFIED)

NAME: CH40405 COLDHEAD OUTLINE			
DWG #: CH40405		SHT 1 OF 1	
MATERIAL: 103-18			
DWN BY: KDJ	DATE: 09JAN03	SCALE: .175	DWG SIZE: A
1ST CHK: KAH	DATE: 09JAN03	2ND CHK: RED	DATE: 09JAN03
CUST: -		BTA: BTA-PT405/PT407	

CH60405 REV D

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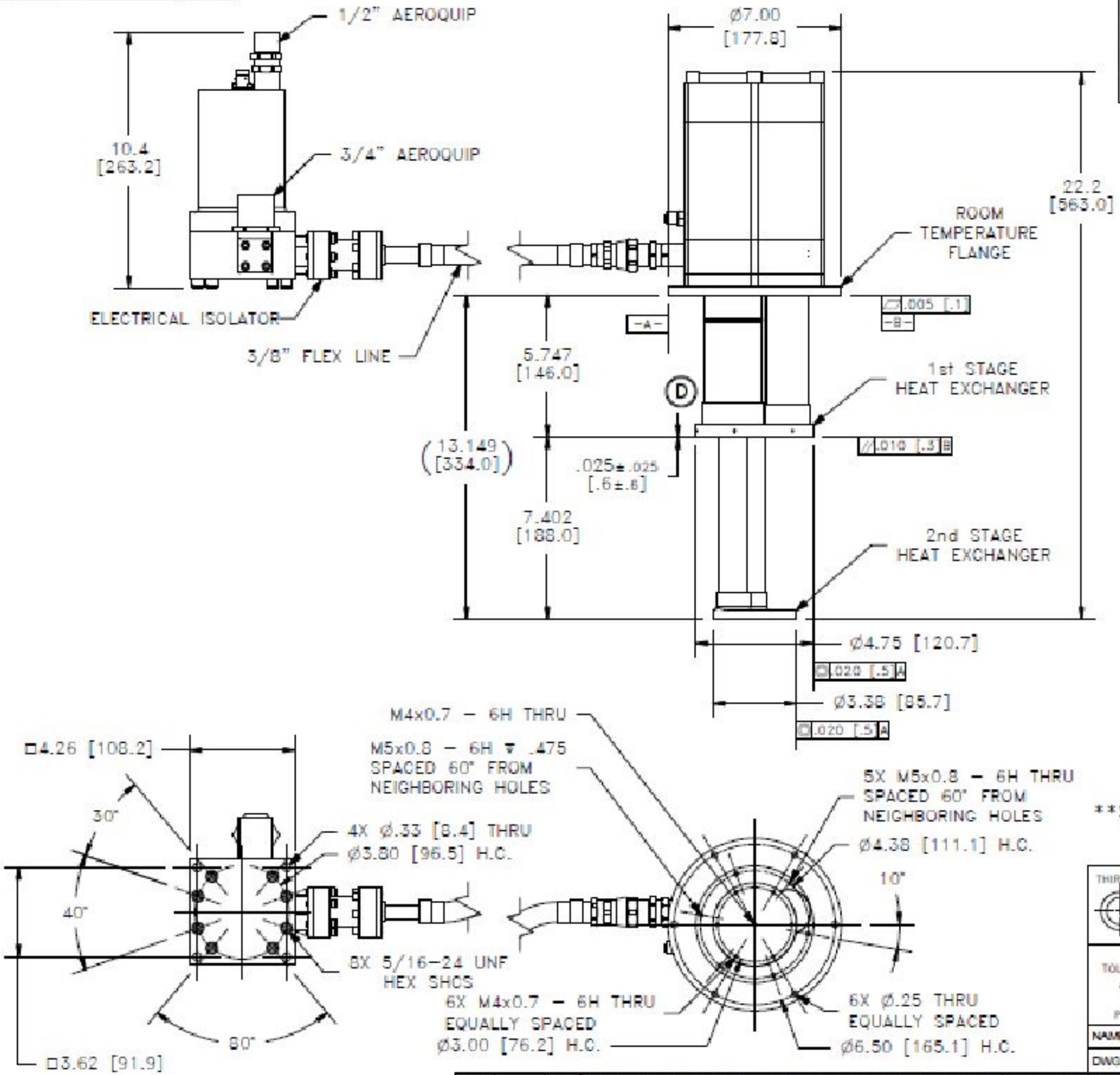
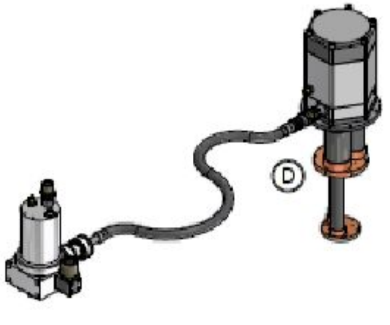




SPECIAL NOTE: COLDHEAD MUST BE OPERATED COLD END DOWN

THIRD ANGLE PROJECTION

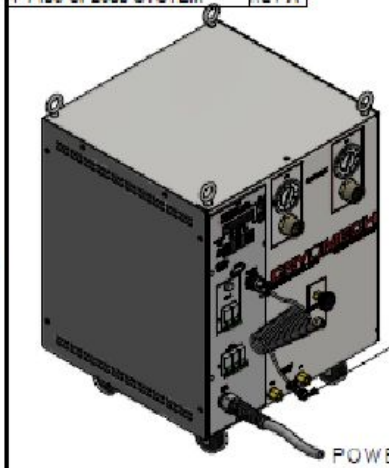
CRYOMECH, INC
113 FALSO DRIVE SYRACUSE, N.Y. 13211
Tel: (315)455-2555 Fax:(315)455-2544

ALL DIMENSIONS ARE IN INCHES
TOLERANCES: FRACTIONS ±1/32 DECIMALS XXX=±.005 XX =±.01 X=±.1
ANGLES ±.5° ALLOVER FINISH 25 Ra
CONCENTRICITY .005 TOTAL INDICATOR RUN OUT
PERPENDICULAR ±.002 (UNLESS OTHERWISE SPECIFIED)

NAME: PT405-RM COLDHEAD OUTLINE
DWG #: CH60405 SHT 1 OF 1
MATERIAL: 103-31
DWN BY: KDJ DATE: 17MAR04 SCALE: .16 DWG SIZE: A
1ST CHK: CW DATE: 23MAR04 2ND CHK: BZ DATE: 23MAR04
CUST: - BTA: BTA-CH40405/407

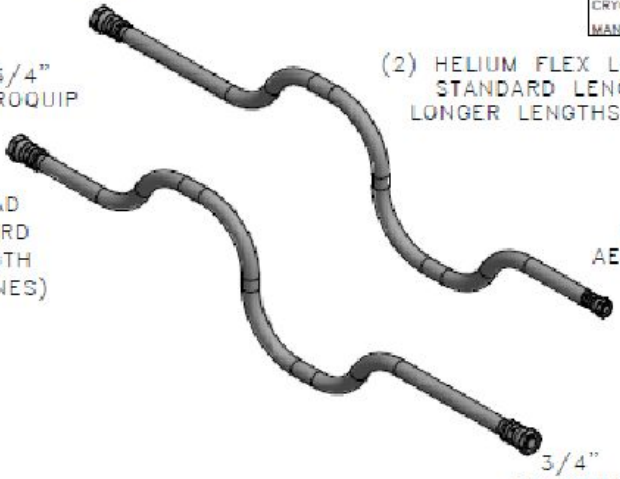
ECN	ZONE	REV	DESCRIPTION	DATE
2673	–	D	UPDATED BTA: RESURRECT 4pc 1st STAGE HT EX	19AUG11
2286	–	C	UPDATED BTA W/ 1pc 1st STAGE HT EX	04NOV10
1669	–	B	ADDED DIMS. TO R.M.	12SEP08

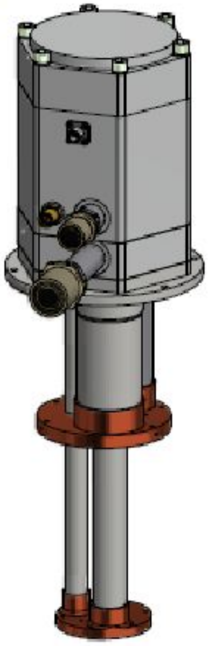
PT405 CP2800 SYSTEM	REV A



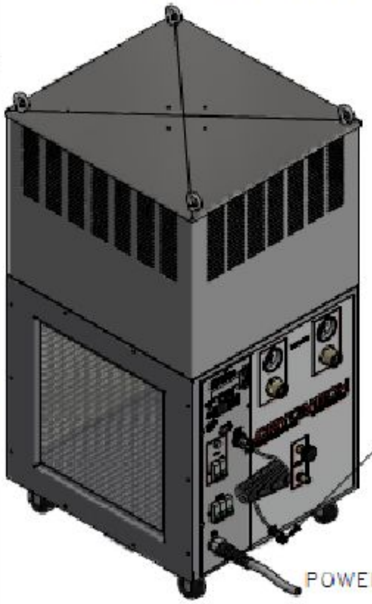
CP2850 COMPRESSOR PACKAGE (WATER COOLED)

(2) HELIUM FLEX LINES 3/4" ID
STANDARD LENGTH: 10ft
LONGER LENGTHS AVAILABLE



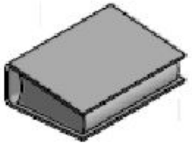


PT405 COLD HEAD

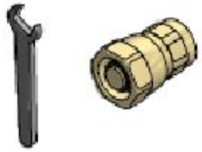


CP2850 COMPRESSOR PACKAGE (AIR COOLED)

MANUAL



TOOL KIT



ECN	ZONE	REV	DESCRIPTION	DATE
5072	–	A	ADDED AIR COOLED COMPRESSOR	21MAY13
1436	–	–	INITIAL RELEASE	14AUG07

THIRD ANGLE PROJECTION

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ANGLES ±.5° ALLOVER FINISH 25 Ra
CONCENTRICITY .005 TOTAL INDICATOR RUN OUT
PERPENDICULAR ±.002 (UNLESS OTHERWISE SPECIFIED)

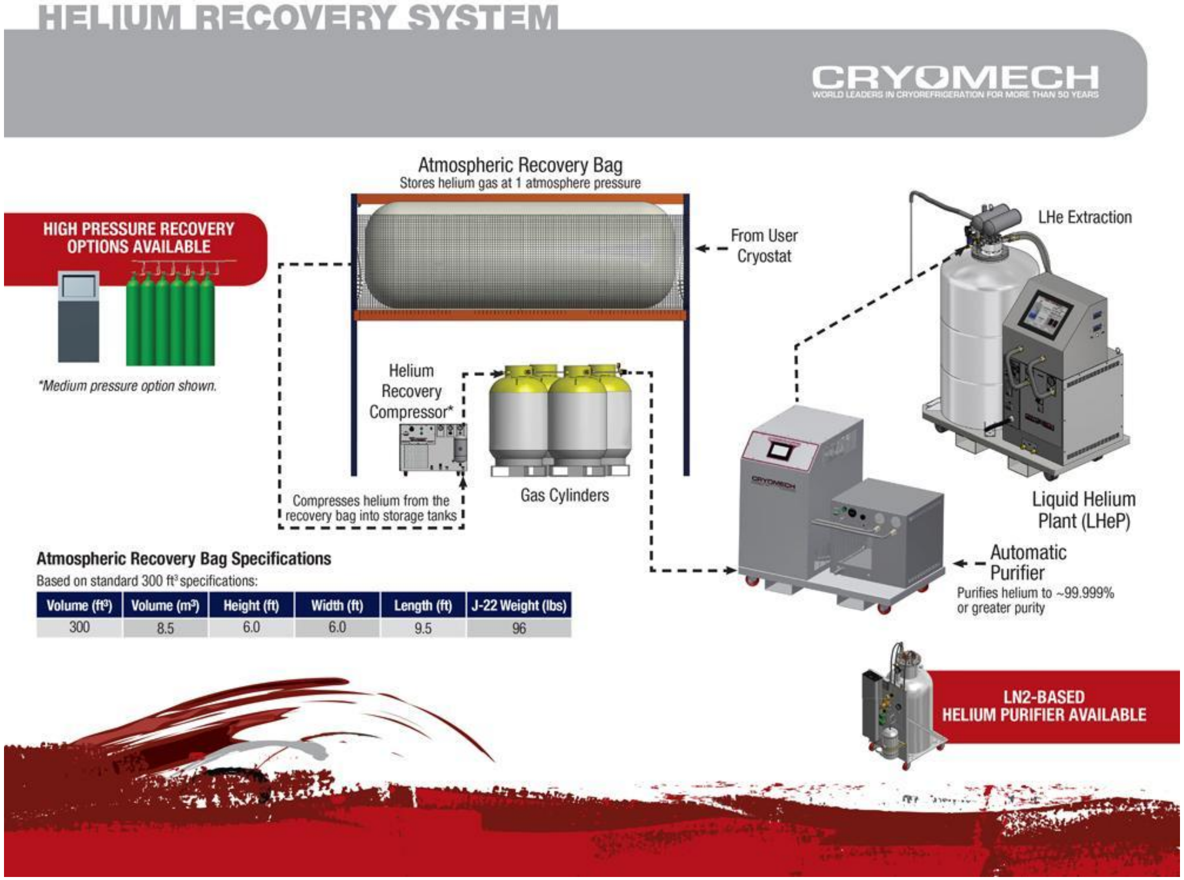
NAME: PT405 CP2800 CRYOREFRIGERATOR

DWG #: PT405 CP2800 SYSTEM		SHT 1	OF 1
MATERIAL: AS STATED			
DWN BY: BMR	DATE: 14AUG07	SCALE: TO FIT	DWG SIZE: A
1ST CHK: RED	DATE: 21AUG07	2ND CHK: AO	DATE: 21AUG07
CUST:		BTA:	

Features of Pulse Tube Cryorefrigerators

- Long mean time between maintenance
- Minimal general maintenance
- Ideal for vibration sensitive applications
- Directly liquefy helium gas and recondense boil-off in liquid cryostat
- Direct conductive cooling in dry cryostats (including low vibration options)

Liquid Helium Plants and Recovery Systems



Atmospheric Recovery Bag Specifications

Based on standard 300 ft³ specifications:

Volume (ft³)	Volume (m³)	Height (ft)	Width (ft)	Length (ft)	J-22 Weight (lbs)
300	8.5	6.0	6.0	9.5	96

Liquefaction rates from 6-60 liters per day

Helium Reliquefiers

Sumitomo pulse-tubes

SHI Cryogenics Group

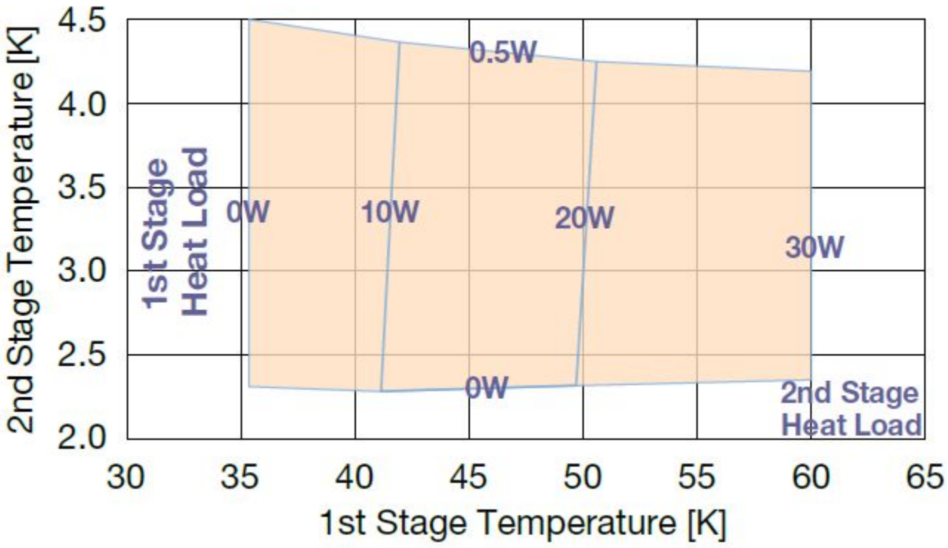
Specifications

Cold Head Model		RP-062B	RP-062BS	RP-082B2	RP-082B2S
1st Stage Capacity	50 Hz	30 W @ 65 K	25 W @ 65 K	45 W @ 45 K	35 W @ 45 K
	60 Hz	30 W @ 65 K	25 W @ 65 K	45 W @ 45 K	35 W @ 45 K
2nd Stage Capacity	50 Hz	0.5 W @ 4.2 K	0.4 W @ 4.2 K	1.0 W @ 4.2 K	0.9 W @ 4.2 K
	60 Hz	0.5 W @ 4.2 K	0.4 W @ 4.2 K	1.0 W @ 4.2 K	0.9 W @ 4.2 K
Minimum Temperature[1]		<3.0 K	<3.0 K	<3.0 K	<3.0 K
Cooldown Time	50 Hz	<100	<100	<80	<90
	60 Hz	<90	<90	<80	<90
Weight		23.2 kg (51.2 lbs.)	23.5 kg (51.8 lbs.)	26.0 kg (57.3 lbs.)	26.0 kg (57.3 lbs.)

RP-062B 4K Pulse Tube Cryocooler Series

SHI Cryogenics Group

SRP-062B Pulse Tube Capacity Map (50 Hz)

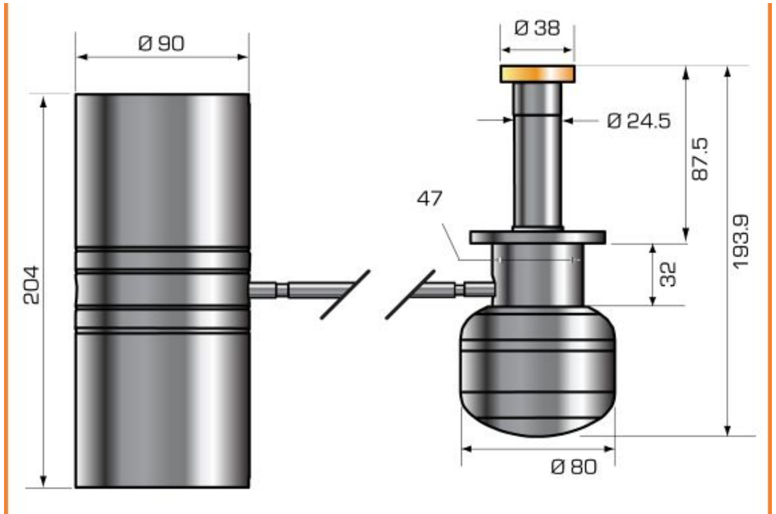


http://www.shicryogenics.com/products/pulse-tube-cryocoolers/rp-062b-4k-pulse-tube-cryocooler-series/

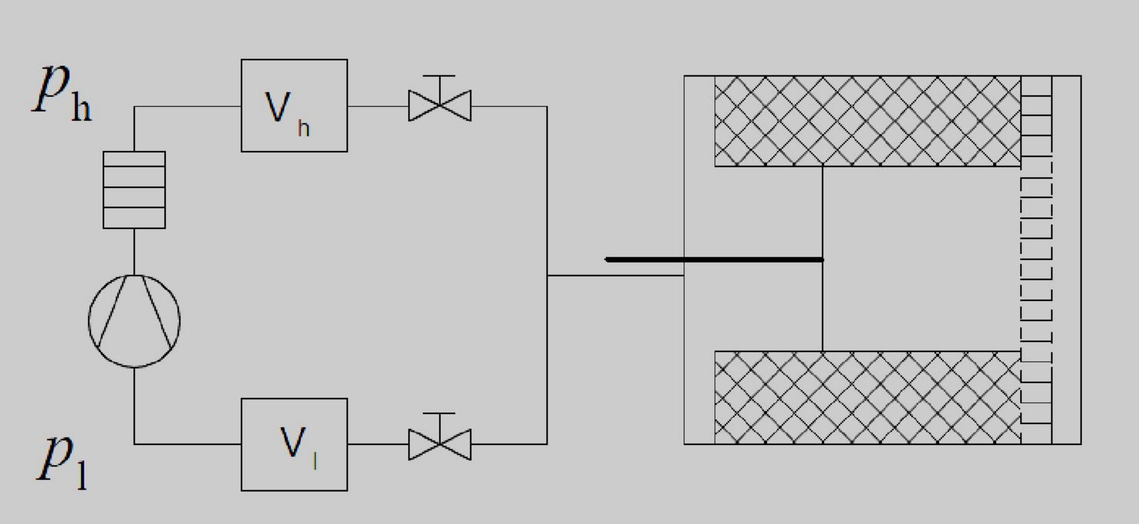
Other manufacturers

Advanced Research Systems (ARS)
http://www.arscryo.com/

Thales Cryogenics
http://www.thales-cryogenics.com



Gifford-McMahon (GM)-coolers

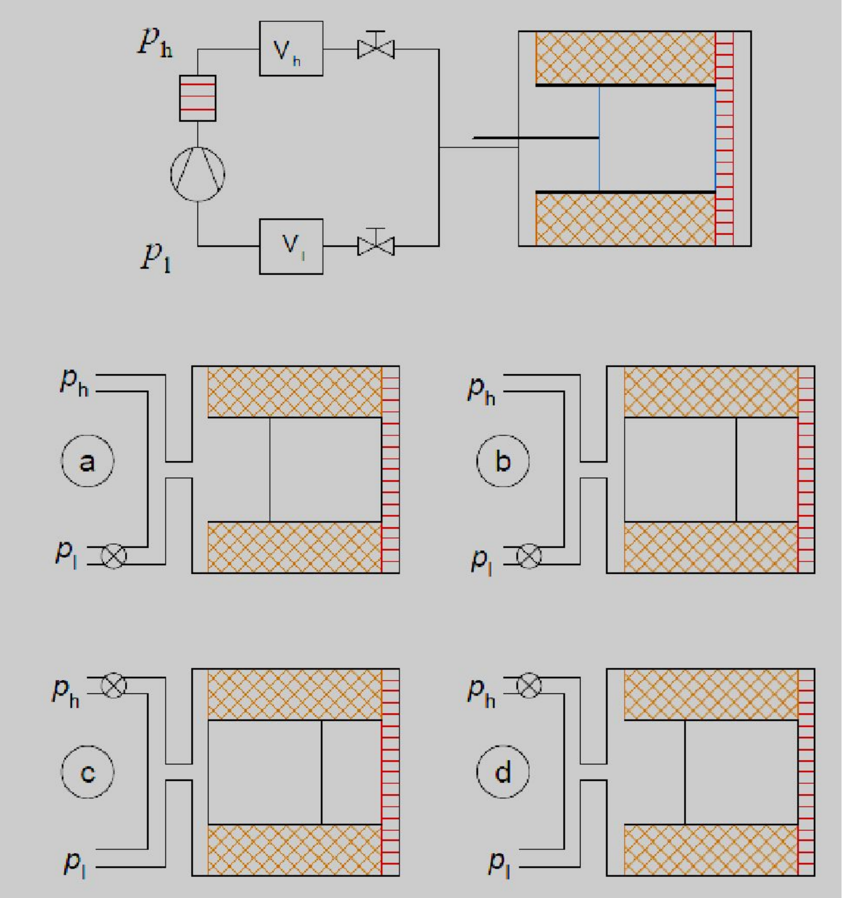


Schematic diagram of a GM-cooler. V_l and V_h are buffer volumes of the compressor.

The two valves alternatingly connect the cooler to the high- and the low-pressure side of the compressor.

Usually the two valves are replaced by a rotating valve.

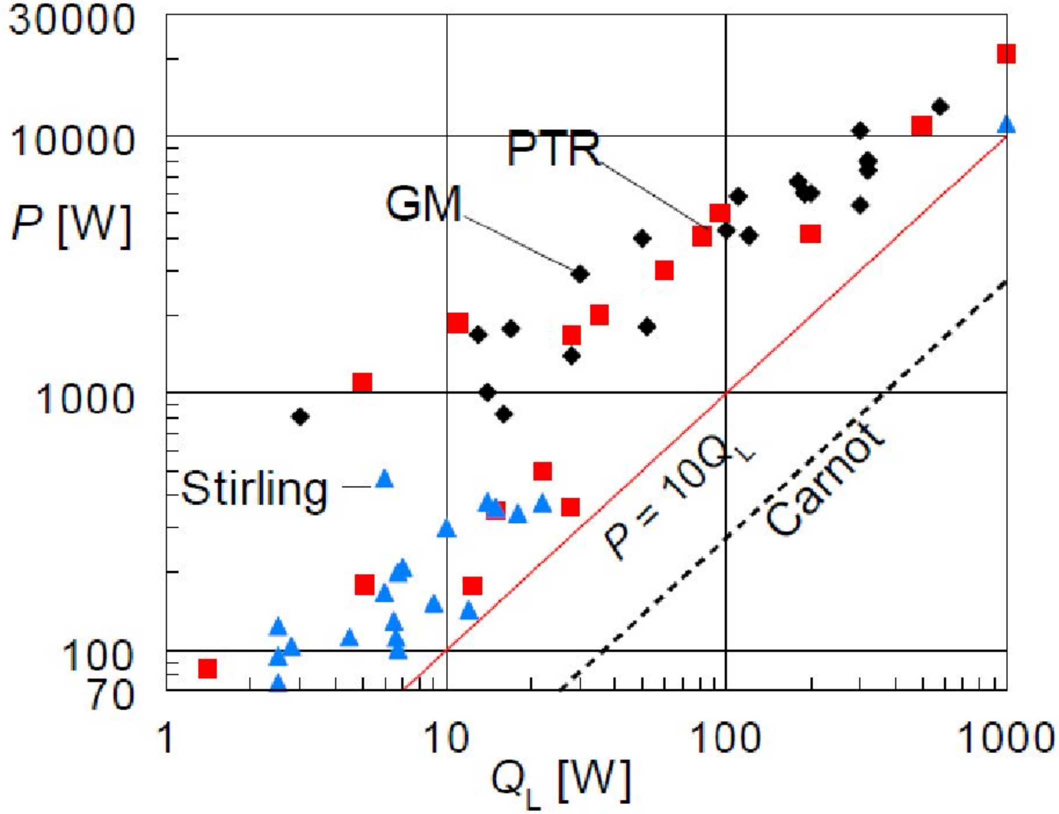
Gifford-McMahon (GM)-coolers

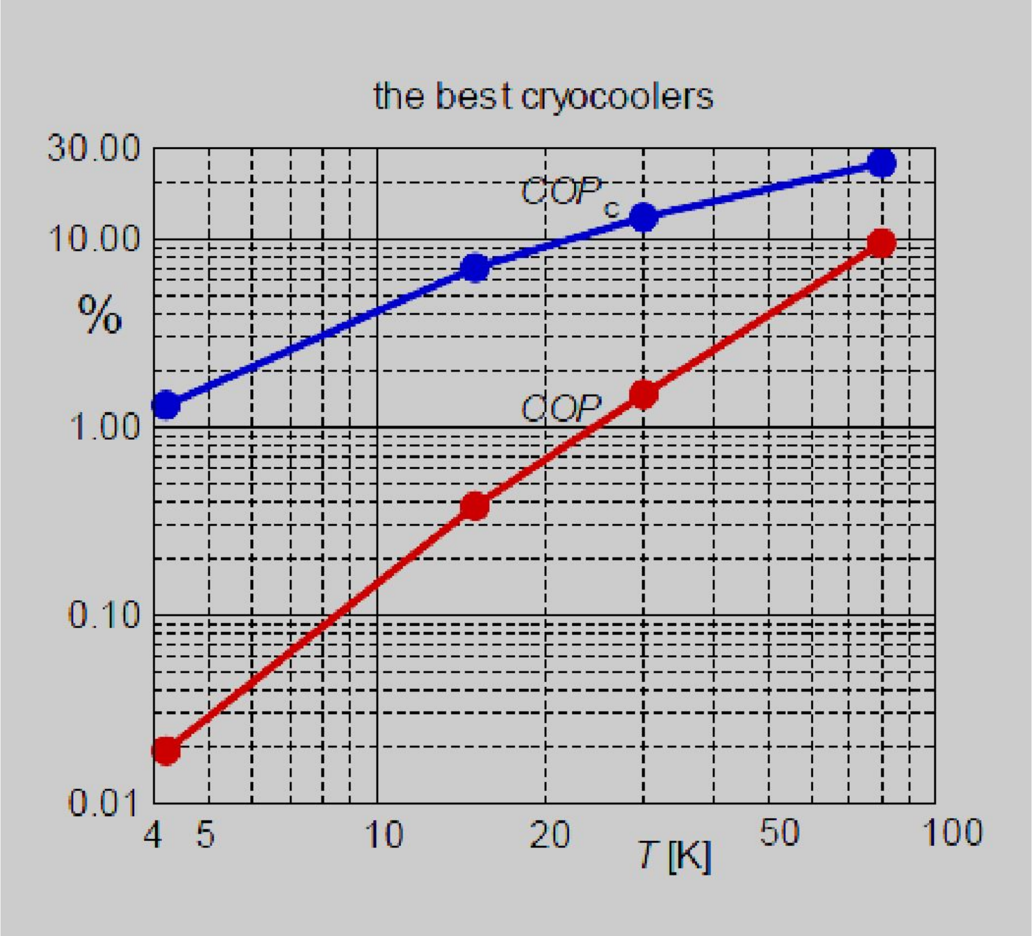


In reality rotary valves are used

Gifford-McMahon (GM)-coolers

values at 80 K




the best cryocoolers
30.00
10.00
1.00
0.10
0.01
%
COP_c
COP
4
5
10
20
50
100
T [K]

Fons’s wise words

The invisible cooler:

- no cost
- no maintenance
- no noise
- no vibrations
- no EM interference
- no space
- no weight
- no water, ice, ..
- no vacuum pump, cooling water,...
- no.... alternative