Modelling of Cryocoolers

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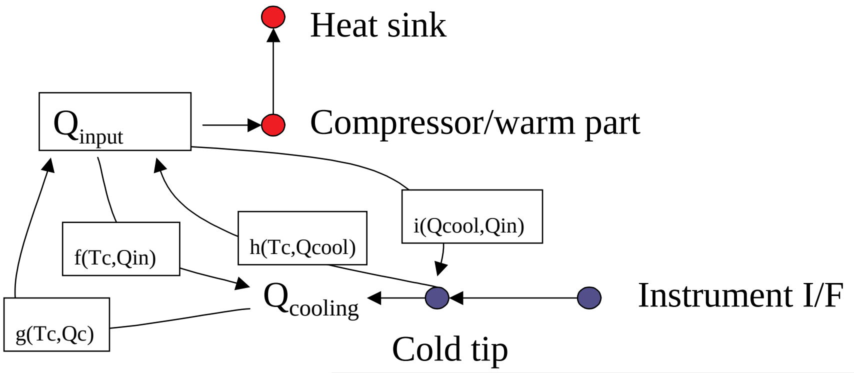
Introduction

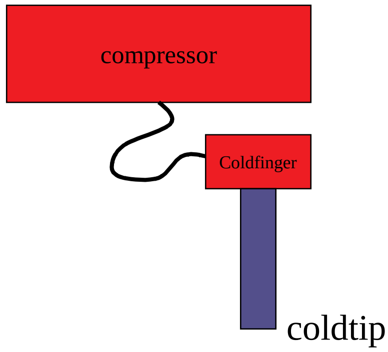
- *Current cryocooler models in ESATAN make use of polynomial fit functions in combination with boundary nodes*
- *This limits the number of free parameters (e.g heat sink temperature)*

therefore

- *Leading to a conservative design only considering worst cases*
- *It is difficult to perform sensitivities or assess the impact of sub-systems (e.g. compressor performances)*
- *A correct heat balance is not always obtained*

Stirling/ Pulse Tube models





$$Q_{input} = a\,Q^2_{cool} + b\,Q_{cool} + c$$

$h(T_c, Q_{cool})$:

$$a = a_1\,T^2_{cold\ tip} + a_2\,T_{coldtip} + a_3$$

$$b = b_1\,T^2_{cold\ tip} + b_2\,T_{coldtip} + b_3$$

$$c = c_1\,T^2_{cold\ tip} + c_2\,T_{coldtip} + c_3$$

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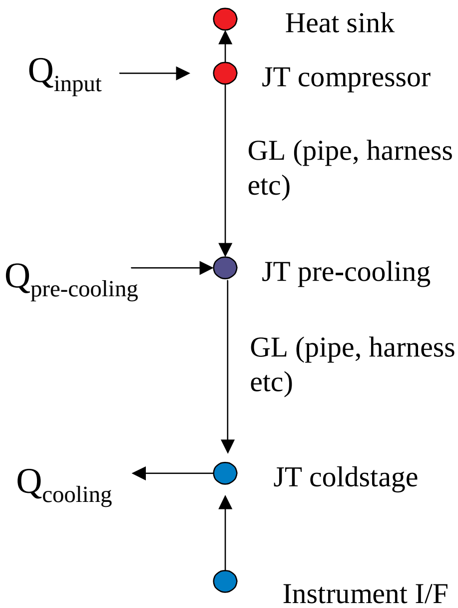
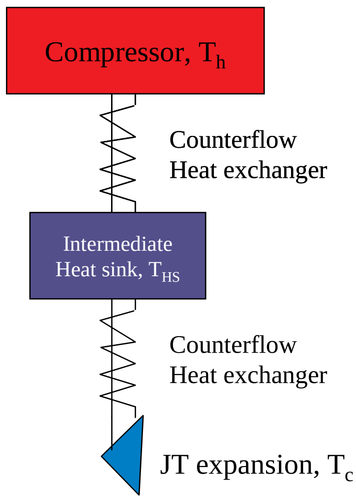
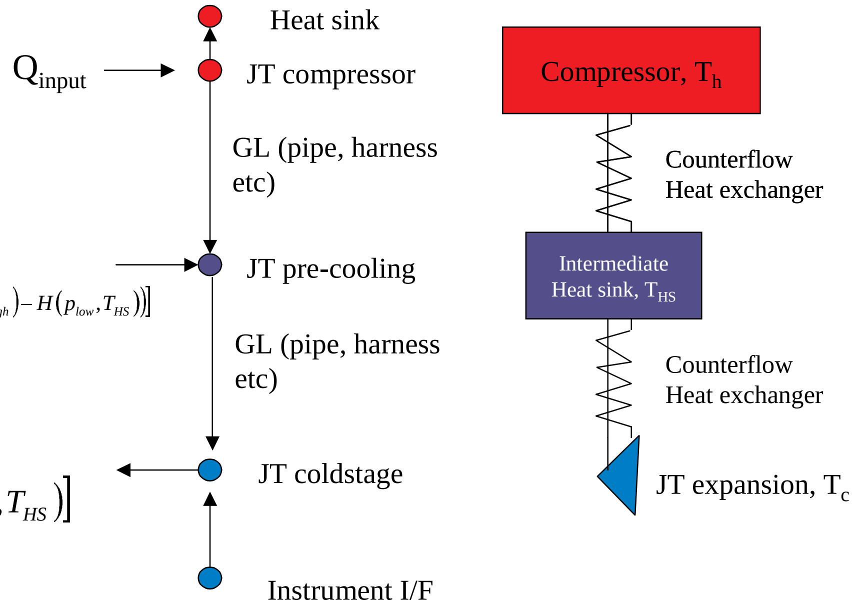
Joule Thomson coolers

$Q_{pre\text{-}cooling}$ and $Q_{cooling}$, depending on the Hx efficiency and the fluid, are provided only by polynomial fits





Joule Thomson coolers including Physics



$$Q_{pre-cooling} = \dot{m}\left[\left(H\left(p_{high},T_{high}\right)-H\left(p_{high},T_{HS}\right)\right)-\varepsilon\left(H\left(p_{low},T_{high}\right)-H\left(p_{low},T_{HS}\right)\right)\right]$$

$$Q_{cooling} = \dot{m}\left[H\left(p_{high},T_{HS}\right)-\varepsilon H\left(p_{low},T_{HS}\right)\right]$$

Requirements for Cooler models

- *In most cases the coldtip temperature is specified by the system (coming from detector needs etc)*
- *The required cooling power is obtained by the TMM*
- *For sizing the thermal system the cooler model must provide*
  $Q_{input} = f(T_{coldtip}, Q_{cooling})$
- *For transient verification, the cooler model must be able to calculate T_{cold} for a given Q_{input}. The following function is required within a TMM:*
  $Q_{cooling} = g(T_{coldtip}, Q_{input})$

Requirements for Cooler models

- *Cooler model shall consider as a minimum the following parameters:*
  - *For single stage cooler $T_{heat\ sink}$, $T_{coldtip}$, Q_{input}, $Q_{cooling}$*
  - *For multistage coolers, the influence of the intermediate cooling stages for different operating conditions needs to be implemented*
- *This can not be handled by polynomial fits or would require a large amount of data points at specific conditions (e.g. isothermal points)*

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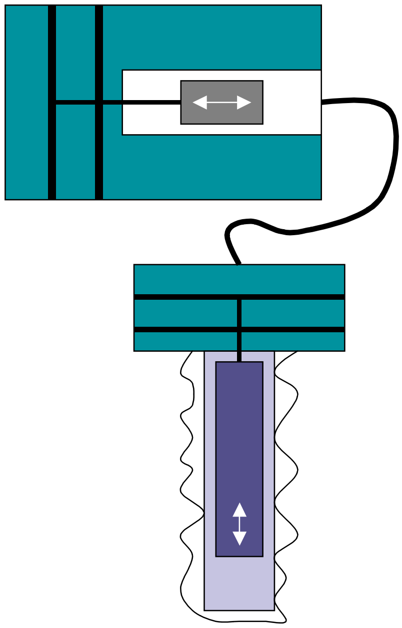
Requirements for Cooler models

- *Overall heat balance needs to be correct*
  - *Common approach is: $Q_{dissipated} = Q_{input}$*
  - *For Stirling, PT and reverse Turbo-Brayton correct approach is: $Q_{dissipated} = Q_{input} + Q_{cooling}$*
- *Use of boundary nodes shall be limited, where required link them correctly with the TMM*
- *Shall be simple, fast and robust*

Stirling cooler

- Compressor transform electrical Energy into pneumatic Work (pV work). For high efficient space coolers one can assume: $pV = Q_{in} - I^2R$
- Pressure wave generated passes through a regenerative heat exchanger (= Regenerator)
- At the cold end the pressure wave and massflow wave are shifted such, that the following cooling occurs: $q_{gross} = T_{cold}/(T_{hot} - T_{cold}) * pV$
- The available cooling power is: $q_{net} = q_{gross} - losses$
- Main losses are: conductive, radiative, regenerator and shuttle losses



Stirling cooler losses

- Conductive/radiative losses are equal to parasitic loads of a non-operating cooler.
- Regenerator losses: $Q_{regenerator} = dm/dt * c_p * (T_{hot} - T_{cold})(1-\varepsilon)$ with $dm/dt \sim pV/T_{hot}$
- Shuttle losses $Q_{shuttle} \sim (T_{hot} - T_{cold})/stroke$

Stirling cooler equation

- *It should be possible to describe the Stirling cooler with the following equation:*

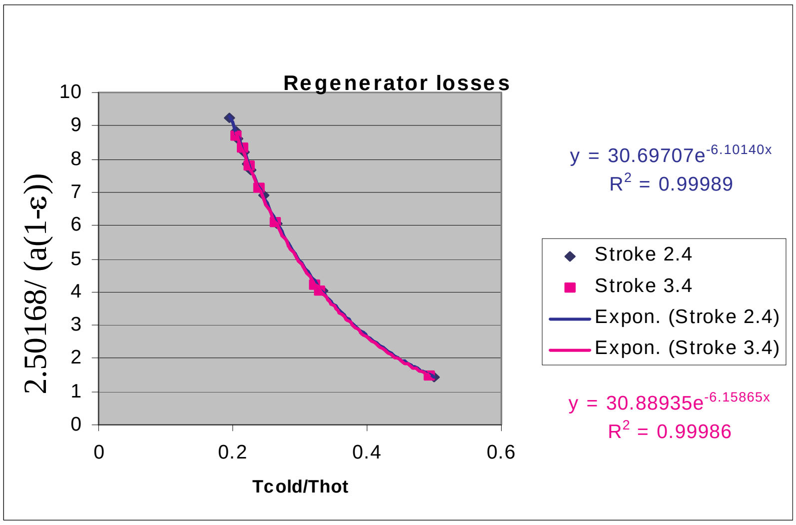
$$\frac{T_{cold}}{T_{hot} - T_{cold}} pV - c(T_{hot} - T_{cold}) -$$

$$a(1 - \varepsilon_{reg}) \frac{pV}{T_{hot}} (T_{hot} - T_{cold}) - b \frac{T_{hot} - T_{cold}}{stroke} - q_{net} = 0$$

- *Approach has been tested with MIPAS 50-80k Astrium cooler data, assuming a constant compressor efficiency (due to the lack of I^2R data)*

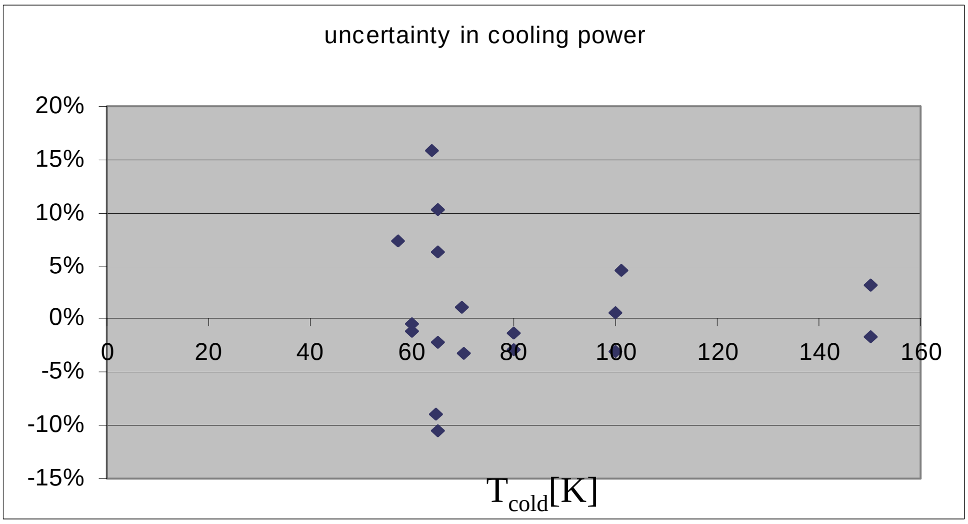
Stirling cooler extrapolation

- *Dataset includes 18 measurements from 57-150K and -10°C to 40°C heat sink*
- *$pV = 0.8 * Q_{in}$*
- *C = 220 mW /193K from parasitic measurements, refined during interpolation*
- *Distinguishing between regenerator and shuttle loss not possible, but regenerator loss as a function of displacer stroke*

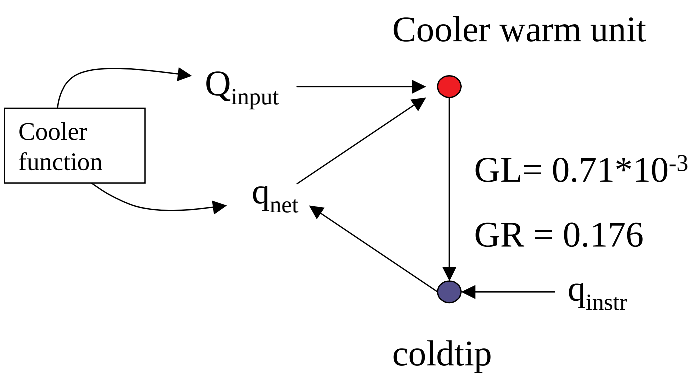


Stirling cooler extrapolation uncertainty

Strong deviations from measurements above 150K, additional correction for these temperatures required



Stirling cooler model



Result TMM:

(300mW q_{instr},Cooler Wu 263K)

T_{cold} = 64.9K -> 12.19W input

Q_{in} = 11.17W -> 66.1K T_c

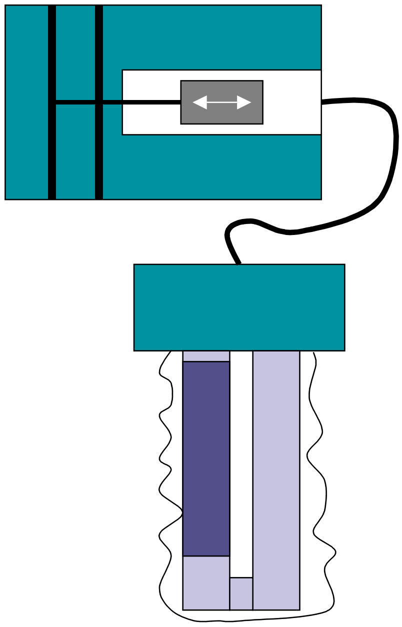
Cooler function:

$$0.8 * Q_{input} \left[\frac{T_{cold}}{T_{hot} - T_{cold}} - \frac{2.50168}{30.697} e^{6.101\, T_{cold}/T_{hot}} \frac{1}{T_{hot}} (T_{hot} - T_{cold}) \right] - q_{net} = 0$$

Note: not valid for temperatures below 50K and above 150K

Pulse Tube cooler

- *Compressor transform electrical Energy into pneumatic Work (pV work). For high efficient space coolers one can assume:*
  $pV = Q_{in} - I^2R$
- *Pressure wave generated passes through a regenerative heat exchanger (= Regenerator)*
- *At the cold end the pressure wave and massflow wave are shifted such, that the following cooling occurs:*
  $q_{gross} = T_{cold}/T_{hot})*pV$
- *The available cooling power is:*
  $q_{net} = q_{gross} - losses$
- *Main losses are:*
  conductive, radiative, regenerator and pressure losses

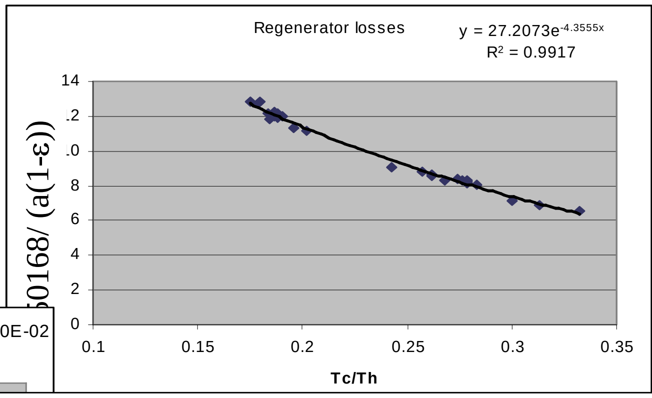


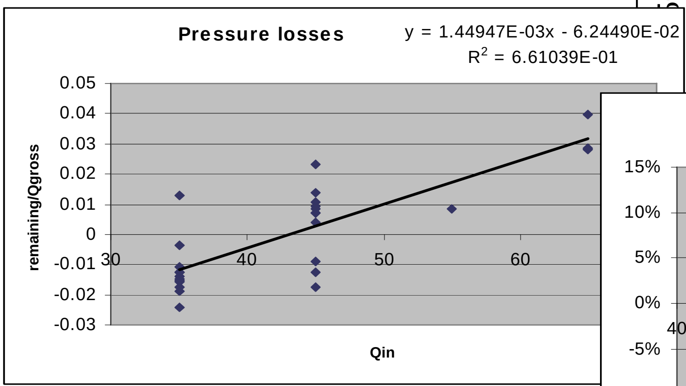
Pulse Tube cooler losses

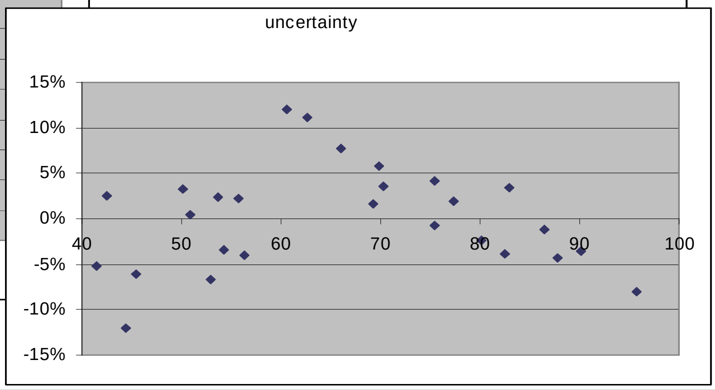
- *Conductive/radiative losses are equal to parasitic loads of a non-operating cooler.*
- *Regenerator losses:*
  $Q_{regenerator} = dm/dt * c_p *(T_{hot} - T_{cold}) (1-\varepsilon)$
  with $dm/dt \sim pV/T_{hot}$
- *Pressure losses* $Q_{press} \sim f(p_{ampl}) *T_c/T_h*pV$
  $p_{ampl} \sim Q_{in}$
- *Data taken from MPTC Air liquide TRP cooler*

Pulse Tube cooler extrapolation

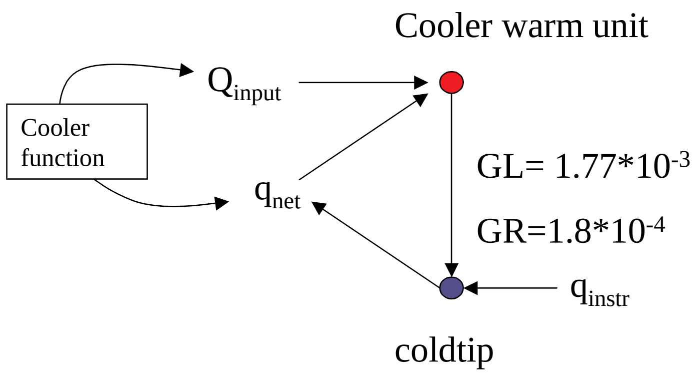
- Dataset includes 23 measurements from 40-95K and -40℃ to 40℃ heat sink
- $pV = 0.7 * Q_{in}$
- C = 465 mW /208K from parasitic measurements







Pulse Tube cooler model



Result TMM:

T_{cold} = 60.63K , T_{hot}=310K, q_{instr} =0

Q_{in} = 45W -> 58.7 K T_c

T_{cold} = 75.43K , T_{hot}= 273K, q_{instr} = 1W -> 36.3 W

Q_{in} = 35W -> 76.3 K T_c

Cooler function:

$$0.7 * Q_{input}\left[\frac{T_{cold}}{T_{hot}}\left(1.0637 - 1.479 * 10^{-3} * Q_{input}\right) - \frac{2.50168}{27.1518} e^{4.327\, T_{cold}/T_{hot}} \frac{(T_{hot} - T_{cold})}{T_{hot}}\right] - q_{net} = 0$$

Note: function not valid for input powers below 30W and $T_{coldtip}$ above 100K, not verified for high $T_{coldtip}$

Conclusion

- *Performance predictions of Stirling and PT coolers is feasible by interpolating the various loss mechanisms, requiring much less input data than polynomial fits*
- *Some additional effort is required for high cold tip temperatures, mainly for cool down predictions*
- *In addition to the classical thermal parameters, the compressor efficiency also needs to be measured*

Future work

- *Distinguish between Compressor and warm part of coldfinger and predict dissipated power on each*
- *Modelling of the gas temperature as a function of gross cooling power and I/F temperature (to improve high temperature performance)*
- *Extend model to multistage Stirling and Pulse Tube cooler*