Performance characteristics comparison of major types of mechanical cryocoolers for aerospace applications

W Chen
Jet Propulsion Laboratory, California Institute of Technology, Pasadena, CA 91109
USA
Email: weibo.chen@jpl.nasa.gov

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Abstract. The three major types of mechanical cryocoolers in aerospace applications, namely reverse-Brayton cryocoolers, Stirling/pulse tube cryocoolers and Joule-Thomson cryocoolers, have significantly different performance characteristics. Some of these differences are due to the nature of their thermodynamic cycles; others come from their drastic differences in mean operating pressures and pressure ratios. This paper first provides an overview of the performance characteristics of these three types of cryocoolers, their unique performance benefits and main applications. It then derives an analytical expression for a cryocooler’s Carnot efficiency, considering the effects of the system pressure ratio drop from the warm end to the cold end and heat leak due to the convective flow and axial conduction. The convective heat leak is related to the pressure drop and the figure of merit of the heat transfer surface using the Colburn-Chilton analogy. Based on this result, the paper discusses the key control parameters affecting the size and mass of the heat exchanger and thus the overall cooler system in each type of cryocooler.

1. Introduction
Mechanical cryocoolers are a critical enabling technology for many aerospace imaging and sensing applications. They reduce IR emission from optical systems and enhance detector sensitivity by reducing dark currents. Mechanical coolers are also used as upper stages for sub-kelvin cooling systems needed for detectors used in long-infrared, X-ray and cosmic microwave background space missions [1-3]. Long term cryogenic propellant zero boil-off storage also requires mechanical coolers with large cooling capacity to intercept heat leak into propellant tanks. The main types of mechanical cryocoolers for these applications are Stirling/pulse tube, Reverse-Brayton (RB), and Joule-Thomson (J-T) cryocoolers. Each type of cryocooler has its distinctive performance characteristics that makes it uniquely suitable for a particular application.

All these cryocoolers employ a similar process to achieve cooling: first compressing gas and rejecting the compression heat to a heat sink, passing the compressed gas through a Heat Exchanger (HX) to cool it to a temperature very slightly above the cold end temperature, expanding the compressed cold gas in an expander/throttle to drop its temperature, absorbing heat from a target, passing back into the HX to warm up the gas to the heat sink temperature, and finally returning to the compressor. In practices, the thermodynamic efficiency and specific mass of these coolers, however, can differ from each other significantly. Their performance differences are partially caused by the heat exchanging process employed between the flows going to and returning from the cold end (i.e. recuperative or regenerative...
process), by their flow configurations (i.e. alternating flow or direct flow compressors), and by their gas expansion processes at the cold end. Their differences are also partially caused by their drastic differences in the mean operating pressures and pressure ratios, and the associated compressor configurations employed. The impact of the system pressure, pressure ratio and associated compressor configurations on the cryocooler performance are the focuses of this paper.

2. Aerospace mechanical cryocooler configurations and their performance features

2.1. Stirling/Pulse Tube Cooler

This type of cryocoolers consists of a reciprocating positive displacement compressor (wave generator) on the warm end, a regenerator, and a mechanical expander (Stirling) or an acoustic expander (pulse tube) on the cold end (figure 1a). The compressor introduces a pressure wave in the system by actively modulating the active volume in the closed system. The pressure wave in turn induces a mass flow in and out of the expander at the cold end, performing mechanical work. This causes the flow enthalpy to drop and thus allow the flow exiting the expander to provide cooling. The exacted mechanical work is used to facilitate the gas compression process in the warm end in a Stirling cooler, but it is dissipated as waste heat at the warm end of a pulse tube cooler. The regenerator between compressor and the expander cools the gas entering the expander by indirectly using the subsequent returning cool flow exiting the cold end. Because the mass flow is induced by the compression and expansion of the gas volume in a closed volume, the mass flow has a waveform similar to a sinusoidal wave (i.e. an AC current flow) and its amplitude varies along the flow direction. There is no net mass flux passing any given cross-section over a cycle period. The working fluid in this type of cooler is invariably helium regardless their cooling temperatures because helium has a higher thermal conductivity than any other gases except flammable hydrogen and helium is an inert gas.

Figure 1. Cycle schematics for Stirling/pulse tube, reverse Brayton and Joule-Thomson coolers

Stirling/pulse tube coolers are relatively compact and can achieve very high thermal efficiency when the required compressor power is below a few hundred watts and the cooling temperature is above 20 K. All but a few aerospace cryocoolers are this type of coolers. This type of coolers is used to provide cooling for a small area located near the cold tip, such as a focal plane array (FPA) and possibly its optics, by using a flexible thermal strap to transfer heat from the cooling target to the cold tip. The flexible thermal strap provides damping to reduce vibrations transmitted to the detector due to exported
vibrations from the cooler. Because of their compactness and high efficiency at cooling temperatures above 20 K, these coolers are also used as the upper stage in a hybrid cooling system for the lower J-T stage [4] or for the reverse Brayton stage [5]. The shortcomings of this type of cooler include its inability to provide remote, distributed cooling over a large area. This is because the amplitude of the pressure wave in the system is directly controlled by the compressor displacement mass during each stroke and the mass of gas in the system. Additional transfer lines can significantly reduce the pressure wave amplitude and increase flow pressure drop, and thus the cooler performance. For below 20 K cooling, this type of coolers also faces an additional challenge that is unique to regenerative coolers: most materials used for constructing a regenerator matrix with microchannel configurations needed for low-temperature helium flow do not have an adequate volumetric heat capacity to stabilize the regenerator temperature. Consequently, exotic rare-earth materials are needed to enhance the regenerator heat capacity by utilizing the high magnetic entropy change during the paramagnetic-ferromagnetic phase transition. However, the poor fabricability of these rare earth materials make it challenging to develop high performance low-temperature regenerators.

2.2. Reverse Brayton Cooler

This type of cooler consists of a DC flow compression system, a recuperator, and a mechanical expander at the cold end (figure 1b). Cooling at the cold end is achieved by expanding the compressed gas through a DC flow expander, which is typically a miniature turbine, to extract mechanical work from the gas and thus reduce the gas temperature at the expander outlet. The extracted mechanical work is converted to electrical power in a generator attached to the shaft of the turbine wheel. Typically, the working fluid is a gas with the highest molecular weight (and thus the lowest sound speed) but with a normal boiling point slightly below target cooling temperature. Using a heavier gas allows centrifugal compressors to achieve a higher pressure ratio without the need to operating at excessively high speeds. For example, the NICMOS cryocooler for the Hubble Space Telescope uses neon to provide cooling at 65K.

The DC flow allows long transfer lines between the expander exit and load HX to provide remote distributed cooling with virtually no thermal efficiency penalty except a small pressure drop in the transfer lines. Furthermore, because this type of cryocooler uses miniature turbomachines operating at rotational speeds higher than 1 kHz, the exported vibrations from RB coolers are much lower than cryocoolers using positive displacement compressors with speeds below 200 Hz. This feature makes it very attractive for applications that are very sensitive to mechanical disturbance. The RB coolers are also suitable for applications requiring a relatively high compressor input power (more than a couple of hundred watts) to achieve a relatively high cooling, such as cooling for a large orbital propellant depot [6]. However, when the cooling power requirement is modest, RB coolers tend to be larger and heavier than other types of cryocoolers, for the reasons discussed later on.

2.3. Joule-Thomson Cryocoolers

The cycle schematic of this type of recuperative coolers is similar to an RB cooler except that the expander at the cold end of an RB cooler is replaced with a simple flow restrictor (figure 1c). The expansion process is isenthalpic and no mechanical work is extracted by the flow. Cooling at the cold end is achieved by the temperature drop of a real gas during an isenthalpic process (Joule-Thomson effect). Using an isenthalpic process has the benefit of having no moving parts on the cold end. This type of coolers typically operates with a very high pressure on the high-pressure side to achieve strong non-ideal gas effects to enhance its specific cooling (cooling power per unit mass flow rate) and thermal efficiency. The pressure ratio in this type of coolers is also typically much higher than other types of coolers to allow the use of a very small recuperator to achieve a high cooling power. The working fluid in a J-T cooler typically is a gas (or a gas mixture) that has a saturation temperature at the low-side pressure just below the target cooling temperature to achieve the highest J-T effects, which also tends to lead to a large enthalpy difference between the high and low pressure streams at the warm end (more precisely, at the precooling temperature). Similar to an RB cooler, the DC flow in a J-T cryocooler also enables it to provide remote redistributed cooling over a large area. Furthermore, because the gas
densities in a J-T cooler typically are very high and the mass flow rate typically is low, flexible transfer lines with very small diameters can be used to allow the cooling target to move relatively to the stationary cooler compressor.

This type of coolers is used for applications where eliminating the technical risk of implementing a mechanical expander at the cold end or minimizing the mass of the recuperator and cold end has a higher priority than achieving high thermal efficiency. J-T cryocoolers are ideal for applications requiring remote distributed, low-power cooling (~100 mW), especially at very low temperatures (<10 K). Example applications include the James Webb Space Telescope Mid InfraRed Instrument (MIRI) cooler [6] and the Planck 4 K J-T cooler. For this type of application, it is very challenging for an RB cooler to achieve a net efficiency significantly higher than a J-T cooler and thus it is difficult to justify the additional complication associated with installing an extremely small turbo-expander. While Stirling/pulse coolers not only lack the remote distributed cooling capability to meet the application requirements, they also have relatively low efficiency at this temperature range.

Indeed, J-T coolers can achieve very respectable thermodynamic efficiency for certain low-temperature applications. The specific cooling power that can be achieved through an isenthalpic expansion process can be a very large fraction of an isentropic expansion process. For example, for 6-K cooling, a J-T expansion process similar to that in the MIRI cooler can achieve a cooling power about 45% of an isentropic expansion (figure 2). For lower temperature cooling at 4 K, however, this percentage drops to about 15%. This efficiency drop is caused by the lowered helium pressure on the low-pressure side of about 0.8 bar to prevent condensation of helium. At such a low pressure, the non-ideal gas effect is relatively weak and the efficiency of the J-T process is low.

Other emerging applications for J-T coolers includes imaging systems on gyro stabilized gimbal platforms where the compressor of a J-T cooler can be located off the platform to minimize the mass of the moving platform. J-T coolers are also very attractive for applications that require rapid cooldown because their recuperator and cold end are very lightweight.

One of the main disadvantages of a J-T cooler is its high vulnerability to system contamination. The flow rate through a J-T throttle typically is very low and the pressure drop across the throttle is very high. Consequently, the flow cross-sectional area in the J-T throttle must be very small. The small flow passages are sensitive to volatile contaminants that can be frozen and accumulate at the entrance of the throttle device, plugging up the flow passage. Therefore, ultra-high cleanliness for the surfaces inside the coolers is a top priority to ensure the reliable operation of a J-T cooler.

Figure 2 Equivalent isentropic efficiency of J-T expansion process for $^4$He. The precooling temperature is fixed at 18K.

3. Effects of system pressure and pressure ratio on cryocooler HX size
Besides the differences in their cycle configurations, the other main contributors to the performance difference among the three type of cryocoolers are the compressor configuration differences, and the resulting differences in the system pressures and pressure ratios. As alluded to previously, the system mean pressure and the pressure ratio have a strong impact on the size of the cryocooler HX. The rest of this paper will first analyze the impact of these parameters on the efficiency of the cooler, as well as their effects on the HX size. It then discusses the typical pressure and pressure ratio in the three types of coolers based on the compressor configuration employed.

3.1. Net Cooling Power

For a recuperative cryocooler, the net cooling power produced in a cooler $Q_{c,net}$ is the difference between the gross cooling power generated in the expander and the losses in the HX due to limited convective conductance $\Delta Q_{c,UA}$ and axial conduction heat leak $\Delta Q_{c,cond}$:

$$Q_{c,net} = mR_T\ln(P_{Ratio,exp})\eta_{exp, isothm} - \Delta Q_{c,UA} - \Delta Q_{c,cond}$$

(1)

The first term on the right is the gross cooling power expressed as the product of isothermal expansion cooling power $mR_T\ln(P_{Ratio,exp})$ and the expander efficiency $\eta_{exp, isothm}$, where $m$ is flow rate, $R$ is the universal gas constant, $T_c$ is the cooling temperature, and $P_{Ratio,exp}$ is the pressure ratio in the expander. The pressure ratio across the expander is smaller than that in the compressor $P_{Ratio,comp}$ due to pressure drop in the recuperator (neglecting the pressure drops in the aftercooler and the cold end interface HX for simplicity here). Assuming the ratios of the pressure drop $\Delta P$ to the flow pressure $P$ on the high- and low-pressure sides are the same, both equal to $\alpha$, then the isothermal cooling power can be expressed as the ideal cooling power with no pressure drop subtracted by the loss of cooling power due to pressure drop:

$$mR_T\ln(P_{Ratio,exp}) = mR_T\ln\left(\frac{P_{max,comp}}{(1+\alpha)P_{min,comp}}\right) \approx mR_T\ln(P_{Ratio,comp}) - 2\alpha$$

(2)

The approximation in the equation above is valid when the value of $\alpha$ is small, which is the case in most cryocoolers. Plugging Eqn. 2 into Eqn. 1, and recasting the convection loss and conduction loss in terms of process parameters yield:

$$Q_{c,net} = mR_T\ln(P_{Ratio,comp}) - 2\alpha - mC_p\Delta T_{HX} - \frac{kA_{HX}}{L_{HX}}(T_H - T_L)$$

(3)

where $C_p$ is the specific heat of the flow entering the expander, $\Delta T_{HX}$ is temperature difference between the high-pressure and low-pressure streams at the cold end of the HX, $kA_{HX}/L_{HX}$ is the effective axial thermal conductance along the length of the HX, including contributions from the shell, and internal walls and the gas flows, and $T_L$ is the HX warm end temperature.

The first term on the right side is the gross cooling power produced in the expander after considering pressure ratio drop in the HXs and transfer lines, as well as the losses in the expander. The second term is the loss due to the limited thermal conductance between gas flows and heat transfer walls, which causes the gas entering the cold end to be slightly warmer than the gas leaving the cold end. The last term is the loss due to thermal conduction heat leak into the cold end through the HX housing wall, the core and gas in the HX.

The loss due to the limited thermal conductance between gas flows is a strong function of the allowable pressure ratio drop in the HX, and the system pressure $P$, as detailed below.

3.2. Loss due to Limited Convective Conductance in HX

The temperature difference $\Delta T_{HX}$ between the gas streams at the HX cold end is proportional to the HX load and inversely proportion to the thermal conductance between the gas streams. Neglecting the conduction resistance across the walls and assuming that the conductance in the low-pressure and high-pressure streams are the same (i.e. balanced thermal resistance), then $\Delta T_{HX}$ can be expressed as

$$\Delta T_{HX} = \frac{Q_{HX}}{U_A_{overall}} = \frac{2mC_p(T_H - T_L)}{U_A_{overall}}$$

(4)

where $UA$ is the thermal conductance of each flow rate. This value is twice the overall thermal conductance value $U_A_{overall}$. Using Chilton-Colburn analogy between heat transfer and momentum
transfer, the flow stream differential conductance $dU/A$ in a differential control volume $dV$ can be related to the differential pressure drop for a given flow passage configuration. Considering the flow on one side of the HX (can be either the high- or low-pressure side), $dU/A$ in a differential control volume with a length of $dl$ and heat transfer surface area $dA_s$ can be expressed as

$$d(UA) = h dA_s = (j p u c_p P_r^{-2/3}) dA_s$$  \hspace{1cm} (5)

where $h$ is the heat transfer coefficient, $j$ is a dimensionless factor for heat transfer, and $P_r$ is Prandtl number. $dU/A$ can be related to the local pressure drop by dividing the equation above by $dP = f \frac{dl}{D_h} \frac{1}{2} \rho u^2$, where $D_h$ is the hydraulic diameter and $f$ is the Darcy friction factor:

$$\frac{d(UA)}{dP} = 2 \frac{1}{f} \frac{1}{D_h} \frac{1}{2} \rho u c_p P_r^{-2/3} D_h dA_s = 8 \frac{1}{f} \frac{1}{D_h} \frac{P_A}{m RT} c_p P_r^{-2/3} dV$$  \hspace{1cm} (6)

where $Ac$ is the flow cross-sectional area. In the last step of the equation, the definition of hydraulic diameter $D_h = \frac{4Ac}{dA_s}$ and the identity $u = \frac{m}{nA_c} = \frac{mRT}{P_A}$ were applied, assuming ideal gas law. Since $dV = Ac dl$, Eqn. (6) can be rewritten as

$$d(UA) = 8 \frac{1}{f} c_p P_r^{-2/3} \frac{A_c^2}{m RT} \frac{dp}{P}$$  \hspace{1cm} (7)

The $j/f$ is a figure of merit for the heat transfer surface, a non-dimensional ratio of heat transfer performance to pressure drop. Its value is function of the channel geometry. It is weakly dependent on the flow Reynolds number and can be treated as a constant for typical laminar flows in cryocoolers. $dU/A$ and $dP$ are functions of local temperature; integrating over the length of the HX yields

$$UA = 8 \frac{1}{f} c_p P_r^{-2/3} \frac{A_c^2}{m RT} \frac{dp}{P}$$  \hspace{1cm} (8)

$T_m$ is mean temperature defined by $T_m = \frac{\Delta P}{\ln(P_{Ratio})}$. Plugging this expression into Eqn. (4) yields

$$\Delta T_{HX} = \frac{\dot{m}^2 (T_H - T_L)(RT_m) P_r m^{2/3}}{4 \frac{A_c^2}{m RT} \frac{dp}{P}} = \frac{G^2 (T_H - T_L)(RT_m) P_r m^{2/3}}{4 \frac{A_c^2}{m RT} \frac{dp}{P}}$$  \hspace{1cm} (9)

where $G$ is the mass flux $\dot{G} = \frac{m}{A_c}$. Substituting this equation into Eqn. (3) yields

$$Q_{c,net} = \dot{m}RT_L \ln(PR) \left(1 - \frac{2 \frac{\Delta P}{P}}{\ln(P_{Ratio})} \eta_{exp, isothm} - \frac{\dot{G}^2 (T_H - T_L)(T_H) P_r m^2}{4 \frac{\Delta P}{P} \ln(P_{Ratio})} \frac{1}{RT_L \ln(P_{Ratio})} \right)$$  \hspace{1cm} (10)

### 3.3. Cooler Carnot Efficiency

Considering the loss in the compression process, the cooler Carnot efficiency can be expressed as

$$\eta_{Carnot} = \frac{(T_H - T_L)}{T_L} \frac{Q_{c,net}}{W_{comp}} = \eta_{comp, isoth} \frac{(T_H - T_L)}{T_L}$$

$$\left[1 - \frac{2 \frac{\Delta P}{P}}{\ln(P_{Ratio})} \eta_{exp, isoth} - \frac{\dot{G}^2 (T_H - T_L)(T_H) P_r m^2}{4 \frac{\Delta P}{P} \ln(P_{Ratio})} \frac{1}{RT_L \ln(P_{Ratio})} \right]$$  \hspace{1cm} (11)

In this equation, $\eta_{comp, isoth}$ is the efficiency of the compressor relative to an isothermal compression process. Here, for simplicity, it is assumed that the mechanical work extracted in the expander is not used to reduce the compressor power input to simplify the analysis.

Eqn.11 is derived for an RB cryocooler with ideal gas flows. In a J-T cooler, the recuperator loss is slightly more complicated because the flow heat capacitances of the high- and low-pressure streams are not balanced due to real gas effect. Consequently, the temperature difference between the streams varies significantly along the HX length. In a regenerative cooler, the regenerator loss is further complicated by the variation of gas density and mass flow rate through a cycle, the effect of regenerator void volume, and the regenerator matrix temperature swing due to its limited heat capacity. Nevertheless, the general trends predicted by this equation are still applicable to J-T and regenerative coolers.
3.4. Implication for cooler design and optimization

Eqn. 11 shows that the overall cooler efficiency is strongly dependent on the pressure drop normalized by system pressure and the system pressure ratio, $\Delta P / P / \ln(P_{\text{Ratio}})$. Increasing the value of this term reduces the pressure ratio in the expander and thus the gross cooling power, but allows a higher pressure drop in the HX to increase the mass flux and thus reduce HX size and mass, as shown in the second term inside the square bracket. The HX size and mass have a strong impact on the overall cooler size and mass. Therefore, the design of cryocoolers invariably involves proper balance between increasing the system efficiency and reducing the HX size and mass, especially for RB coolers (note that the void volume effect in a regenerator complicates the tradeoff and a larger regenerator does not necessarily reduce the regenerator loss).

The loss due to the limited convective heat transfer is proportional to $\dot{G}^2 / \dot{m}$ and $1/P^2$. This loss is inversely proportional to $P^2$ because (1) a higher pressure allows a proportionally higher pressure drop and thus enables a higher heat transfer performance, and (2) increasing the pressure reduces flow velocity and the thermal conductance for the same pressure drop, as shown in Eqn. (6). The loss decreases with $\dot{G}^2$ because increasing mass flux proportionally increases heat load in HX while reduces $UA$ for a fixed pressure drop allowance.

For the same cooling power and same normalized pressure drop, $\Delta P / P / \ln(P_{\text{Ratio}})$, increasing the system pressure would allow the use of a smaller HX cross-sectional area with proportionally higher mass flux, while increasing the pressure ratio would reduce the mass flow rate and thus allow the use of a proportionally smaller HX to maintain the same mass flux. Both effects will lead to a smaller HX and a smaller cooler.

The nominal system pressure in a regenerative cooler is 10 bar, about 10X that in an RB cooler, while the pressure ratios in these coolers are similar. As a result, a regenerator is much smaller than the corresponding recuperator in an RB cooler with the same cooling performance. While the nominal system pressure in a J-T cooler is typically more than 5x of an RB cooler, and the logarithm pressure ratio in a J-T cooler can be more than 4x that of an RB cooler (e.g. a pressure ratio of 3:1 in a J-T cooler vs. a pressure ratio of 1.3 in an RB cooler). Consequently, a J-T recuperator is much smaller than a corresponding RB recuperator, and even a regenerator with the same cooling performance.

4. Differences in system pressure and pressure ratio among mechanical cryocoolers

The optimal pressure and pressure ratio in a cryocooler are mainly controlled by the compressor employed. Two types of compressors are used in mechanical coolers: centrifugal compressors and positive displacement compressors such as linear piston compressors. The former type of compressor is used in RB coolers while the latter type is used in regenerative coolers and J-T coolers. The pressure ratios in these coolers are results of compromise between compression efficiency and cooler size and mass. Increasing the pressure ratio in general reduces the efficiency because of a higher gas temperature rise during the compression and large compressor internal losses (e.g. internal leakage losses), or requires a larger multistage compression system with intercoolers. However, a higher pressure ratio can significantly reduce the size of the HX, as discussed previously. The optimal system pressure, on the other hand, is most controlled by the type of compressors employed. The compressor technologies and their performance characteristics are discussed below.

4.1. Centrifugal Compressor

This type of compressor has a very high mechanical work output power density because their operating speeds are typically above 1 kHz. The volumetric flow rate through this type of compressors typically are much higher than positive displacement compressors. Current RB cryocoolers designed for aerospace applications exclusively employ this type of compressors. The cooling power needed for typical aerospace application is low, and the corresponding mass flow rates in these coolers are small for a conventional centrifugal compressor and the turbine. To avoid the need for excessively small impeller/turbine wheels when the gas volumetric flow rates are low, and to reduce internal leakage losses across clearance shaft seals and to decrease shaft windage, a low operating system pressure in these
compressors is preferred. Typically, this pressure is just slightly above 1 atm on the low-pressure side to prevent ambient air from leaking into the system during initial ambient testing. The low operating pressure, coupled with a low-pressure ratio in each compression stage when helium is used due to its very high sound speed, lead to recuperators that are significantly larger and heavier than HXs in other coolers. However, as the required cooling capacity increases, the size and mass of RB coolers will be more competitive because the compressor sizes become smaller than those in other types of coolers with the same cooling performance. Furthermore, the compression efficiency will also further improve as internal parasitic losses in larger compressors become a smaller fraction of their useful work output.

4.2 Piston Compressors

This type of compressors typically has a lower mechanical work output power density than a centrifugal compressor because its operating speed is much lower, typically below 200 Hz even for small coolers, and thus is more suitable for applications with low to medium cooling power requirements. This type of compressor is invariably used in regenerative cryocoolers and J-T coolers (with addition of check valves to rectify the flow). This type of compressor can achieve a high force and thus suitable for a high-pressure system with a large pressure ratio (i.e. large pressure amplitude). For this type of compressors, as the diameter of the piston increases, efficient heat removal during the compression process becomes more challenging and the compression efficiency may drop accordingly. This, along with the lower resonant frequency in larger compressors, limits the maximum capacity of this type of compressor in practical applications.

5. Conclusion

This paper describes the general performance characteristics of three main types of mechanical cryocoolers in aerospace applications. An analytical expression for a cryocooler’s Carnot efficiency is derived using the Colburn-Chilton analogy. To achieve the same thermodynamic efficiency, the analytical result shows that the allowable pressure drop in a cooler is proportional to the flow pressure and the logarithm of the system pressure ratio, $P_{\text{in}}(P_{\text{ratio}})$, when their effects on the efficiency of the compressor and expander are neglected. Increasing the system pressure or pressure ratio allows a higher pressure drop in the HX to keep the relative drop in the expander pressure ratio unchanged, while a higher system pressure also reduces the flow velocity in the HX. Both effects allow the use of smaller flow passages in the HX to significantly reduce its size and mass. The three types of mechanical coolers operate at significantly different system pressures and pressure ratios due to the differences in the compressor technologies employed. Consequently, their HX sizes and therefore the overall cooler sizes in most extent are quite different from each other.

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