FUTURE TREND OF PULSE TUBE CRYOCOOLER RESEARCH

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ABSTRACT

The flexibility of a pulse tube allows integrating the cryocooler with its application. However, the drawback will be a deterioration of thermodynamic efficiency because of the application oriented design philosophy. To minimize the additional thermodynamic losses, the extending cycle analysis, which can simulates the requested configuration, will be required. Therefore, one of the important subjects for the future progress of such a flexible pulse tube coolers is a substantial improvement of the software as a design tool of the cryocooler, which could be used to the actual application. Several proposals based on a basic cycle analysis are presented.

INTRODUCTION

Earlier stage of pulse tube studies was mainly focused on the improvement of thermodynamic efficiency, which results in the development of various phase-shift mechanisms. Many phase-shift mechanisms have been developed in this stage [1-3]. As the thermodynamic efficiency of the pulse tube approaches to that of Gifford-McMahon (GM) or Stirling cycle cryocoolers, the research target was set to achieve the possibility of lower cooling temperature by utilization of multistage pulse tube [4]. Efficient multistage method was also developed [5,6]. From the viewpoint of cryocooler application, the importance of the object oriented cooler development has been well recognized and some of the cryocoolers are already specialized from such a viewpoint of cost reduction, reduced mechanical vibration or extended operating lifetime.

In the case of pulse tube cryocoolers, however, non-obstructive or invisible cooler development will be possible by step into the application field furthermore. For example, the cold part of the pulse tube cooler could be utilized as a part of the cooling object such as, the supporting rod of the cold mass. The design of such an integrated pulse tube cold part without restraint of optimized cooler configuration leads to the deterioration of thermodynamic efficiency of the cooler. To minimize the additional thermodynamic losses, the extensive cycle analysis will be required to simulate the requested configuration. Therefore, one of the important issues for the future progress of such a flexible pulse tube cooler is a substantial improvement of the software as a design tool of the cryocooler, which could be applied to the actual application. This paper describes several examples of the valved and the valve-less pulse tube coolers.

BACKGROUND OF PULSE TUBE COOLER DEVELOPMENT

Cryocoolers are classified into two different types from the viewpoint of the operating gas flow patterns as shown in Figure 1. The circulating flow type composed of a turbo-expander or a low temperature valved reciprocating expander with a counter flow heat exchanger, while the oscillating flow type consists of the valve-less expander and a regenerator. Thus, the compact system can be fabricated, using the latter type cryocooler. In fact, many small-scale cryocoolers based on the oscillating flow type are applied to the wide range of application fields. It has been considered that the pulse tube cooler, as one of the oscillating flow type, has a potential to replaces the other type of coolers such as, Stirling, GM, Solvay and Vuillemier cycle. However, one of the difficulties to develop the pulse tube cooler is its systematic design; the interactions of composed components are complicated and it is hard to apply thermodynamic analyses.

In the case of circulating flow type, the function of each component, such as a compressor, a counter flow heat exchanger, an expander or a JT valve, are rather independent of each other. However, the function of regenerator for oscillating flow type cryocoolers has a strong dependency of other components.

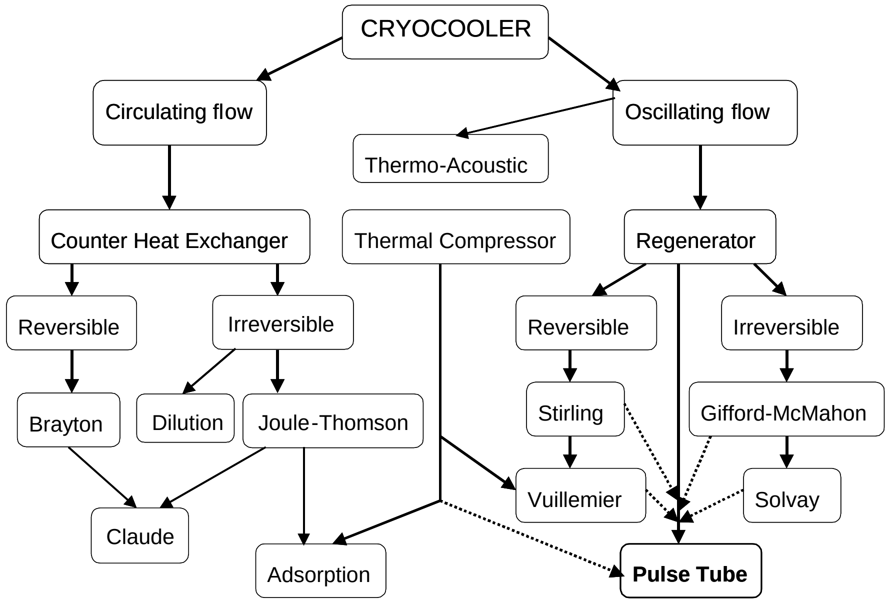


Figure 1 Classification of cryocoolers

Therefore, the design of oscillating flow type cryocooler is much more complicated than the design of circulating flow type. The pulse tube cryocooler, which is classified as one of the oscillating gas flow type, is further complicated to design the components independently each other, because it has no solid displacer or expander. The concept of equivalent PV work makes the simulation easier. The minimum requirement of each component will be predicted by means of these simplified simulation models.

GM-TYPE (VALVED) PULSE TUBE COOLER

Figure 2(a) shows generalized schematics of a valved type regenerative cryocooler. The pressure wave is generated by the combination of a compressor and switching valves consist of V1 for the high pressure intake and V2 for the low pressure exhaust.

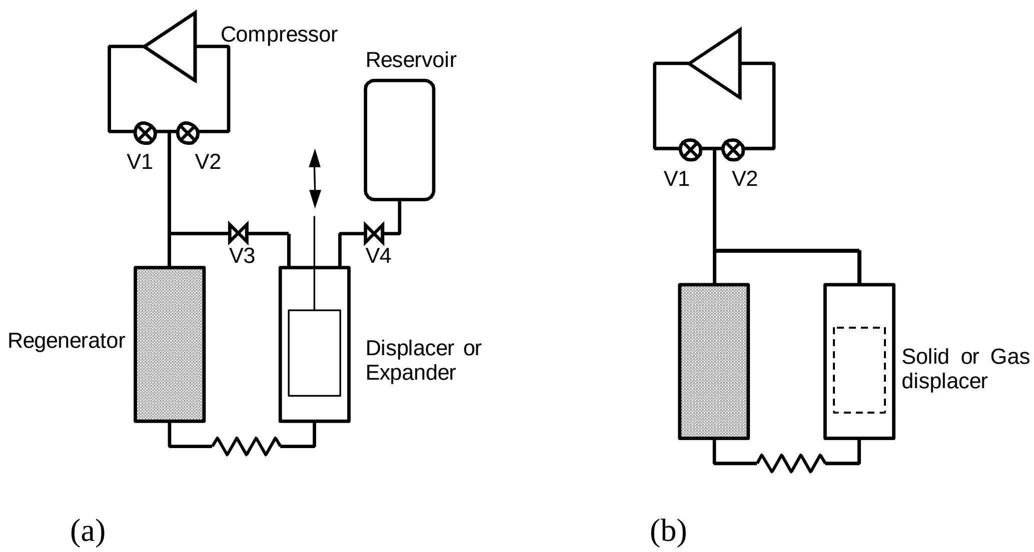
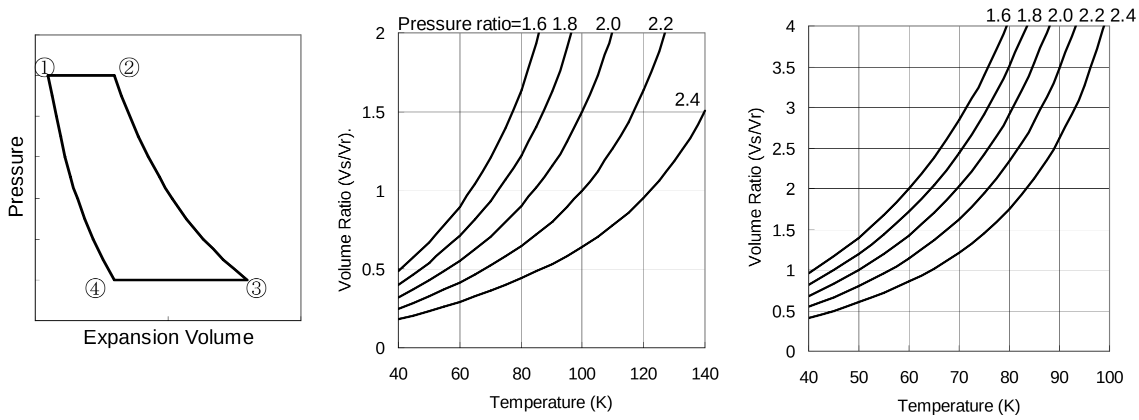


Figure 2 Generalized schematics of valved type pulse tube cooler

If V4 is fully closed and V3 is opened, it becomes a GM cycle cooler, which requires external mechanical force to drive the displacer: controlled opening rate of V3 gives the minimum driving force of the displacer. If V3 is fully closed and V4 is opened, it becomes Solvay cycle cooler: controlled opening rate of V4 gives pneumatically controlled expander. If the solid displacer is removed, it becomes the orifice (V3 close, V4 open), the double inlet (V3 and V4 open) and the by-pass (V3 open, V4 close) pulse tube coolers, respectively. Here the third case named as the by-pass pulse tube is not used commonly, however, it would be used when the total volume of the cooler must be minimized, because it does not requires a large reservoir volume.

The valved type cryocooler is commonly used because of its flexibilities to the applications; however, it has a lower thermodynamic efficiency than that of Stirling type cryocooler, in general. This lower thermodynamic efficiency is primarily caused by the work loss at the switching valve system. Therefore, to find out the way to reduce this loss by the appropriate analytical method would be an important issue for the future development of cooler application. The loss at the valve will be classified into two groups. One is a limited size of the valve port, which gives a large pressure drop and is mainly depending on the valve design. The other is a loss due to the large pressure difference just before the valve opening and is mainly depending on the phase shifter process of the cycle. Therefore, the later one could be minimized by selecting the optimum relation of the expansion volume with the pressure ratio.



(a) (b) (c)

Figure 3 Required conditions to minimize the valve loss

In the case of GM cycle, this optimization is limited because the total volume is fixed. Figure 2(b) represents a simplified model of GM cooler having the regenerator void volume of Vr and the piston displacement volume of Vs. To minimize the work loss at V1 and V2, a PV diagram at the cold end of the piston as shown in Figure 3(a) is desirable. When the piston is located at the cold end, V1 is opened at the point ①. The piston is moved up, then the gas flow to the cold end volume with the pressure of Ph. As V1 is closed at the point ② and keep moving the piston, the pressure decreases. The pressure becomes Pl at the end of the piston stroke, the point ③. V2 is opened and the piston moves downward. V2 is closed at the point ④ and keep moving the piston, then the pressure increases. The pressure becomes Ph at the end of the stroke and the cycle is completed. These conditions are realized when the ratio of the swept volume and the regenerator void volume is selected with particular sets of pressure ratio and the temperature ratio as shown in Figure 3(b). For example, if the pressure ratio is 2 and the cooling temperature is 80 K, then the volume ratio of 0.9 should be selected to minimize the valve loss.

Modification of GM-type pulse tube cooler

Extending this consideration, optimization of GM-type pulse tube cooler will be obtained. Similar calculation has been done for the pulse tube cooler simply replacing the solid piston of Figure 2(b) to a compressible gas piston. The result is shown in Figure 3(c). Similarity in the operating condition of GM cooler can be seen, although the available pressure ratio at the given temperature ratio is more critical than the case of GM cooler.

In the case of Solvay cycle, however, the total volume can be changed by changing the expander stroke volume without any restriction discussed above. If the required cooling temperature is lowered, the regenerator performance becomes more important. In such a case, it is recommended that the Solvay cycle type phase shifter, which released from the fixed total volume specialized in the GM cycle, should be selected to realize the efficient valved pulse tube cooler.

Multi-staged pulse tube cooler

Figure 4 Displacers with no pressure seal for 4K GM cooler (Photo is offered by SHI)

In the case of multi-staged pulse tube, especially 2-staged 4K pulse tube cooler, is always compared with a 4K GM cooler. Thermodynamic performance of 4 K GM cooler is better than that of 4K pulse tube cooler, in general. Demerits of 4K GM cooler are frequently regarded as the large mechanical vibration, a short maintenance interval due to the existence of the solid displacer and its pressure seal at the low temperature part. The recent progress of 4K GM cooler, however, does not require any low temperature seal as shown in Figure 4. Therefore, if the thermodynamic efficiency and the cooler setting orientation are the most important issue, the 4K GM cooler still has superiority over the 4K pulse tube cooler. The merit of 4K pulse tube will be appeared when the single stage pulse tube using He3 is precooled by another single stage pulse tube cooler. This method could give an efficient cooling performance at the temperature below 4K, using the minimum amount of He3 [7]. It is noted that the configuration of 4K GM displacer without pressure seal resembles well with the double inlet line of the pulse tube cooler as shown in figure 2(b).

STIRLING TYPE (VALVE LESS) PULSE TUBE COOLER

Stirling type pulse tube cooler has been developed as an alternative of the Stirling cooler. The flexure spring supported a linear compressor, first developed at Oxford University [8], has been successfully used for Stirling cooler and now it applied to the pulse tube coolers. With this compressor, which is now frequently called as a pressure wave generator, a single stage small-scale pulse tube cooler is already replaceable with Stirling cooler even from the viewpoint of thermodynamic efficiency. Recently, a medium or a large scale Stirling type pulse tube cooler becomes an important issue for application to the superconducting system of electric power field.

To realize such a cooler with a higher thermodynamic efficiency, however, the extensive analyses of each component and its interrelation is essential. This requires user-friendly software as a design tool of the cooler. Analysis of Stirling type pulse tube cooler always requires momentum equation because of its higher driving frequency up to a few tenth hertz. This induces more complicated simulation program to have optimum sizes of critical components.

The Stirling type pulse tube cooler utilizes a linear driven pressure wave generator as a work source and an inertance tube phase shift mechanism as a work receiver, which is shown schematically in Figure 5. To predict the inertance effect as a phase shifter, the numerical analysis has been executed [3]. An example calculated result in comparison with experimental result is shown in Figure 6(a) and (b). The boundary condition is selected as the piston movement of the pressure wave generator because it can also be obtained from the experiment. The measured data is based on the inertance tube of 10.7mm inner diameter and 1.9 meters length. Driving frequency was 49 Hz and the PV work of 1.5 kW, dashed line in Figure 6(b), was required when the cold end temperature is 80 K with heat load of 75 watts. Calculated PV work corresponding this operating condition is shown by doted line. Work flow <W> and enthalpy flow <H> through the regenerator, a pulse tube and an inertance tube are calculated as shown in Figure 6(a). Work flow is consumed within the inertance tube.

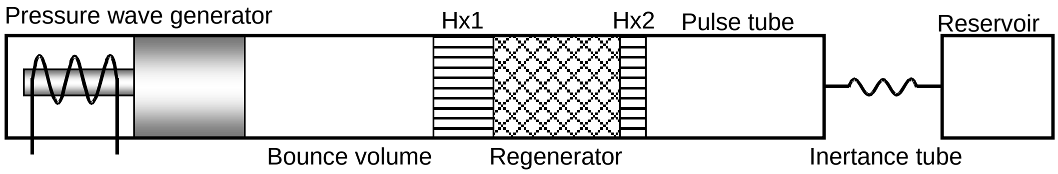


Figure 5 Valve-less pulse tube cooler

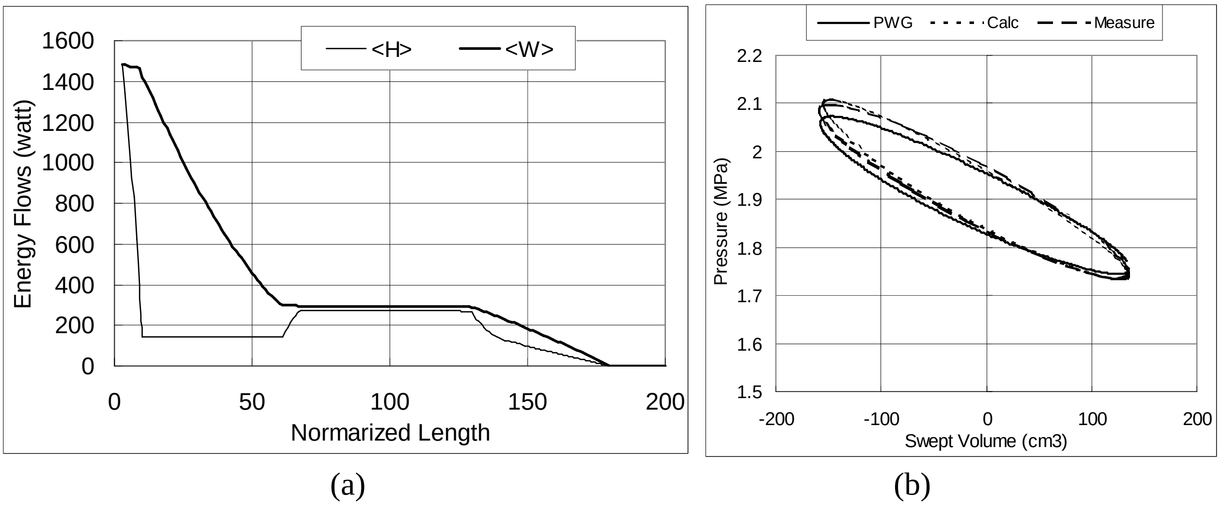


Figure 6 Calculated result of Figure 5 based on compressor swept volume in comparison with experimental data

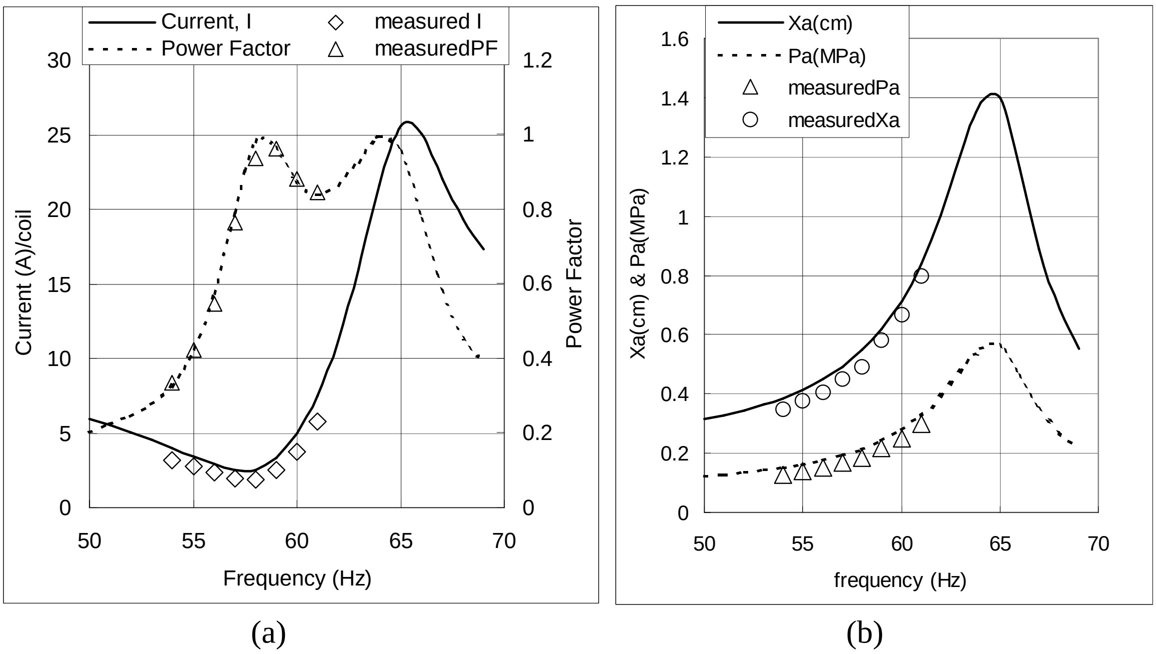
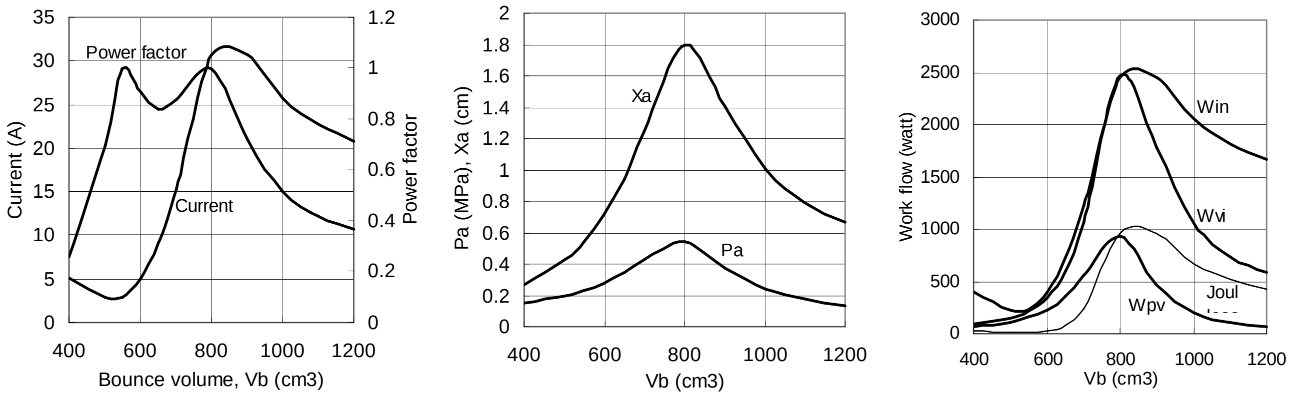


Figure 7 Basic performance of linear compressor

Performance of the linear motor for the pressure wave generator is separately calculated as shown in Figure 7(a) and (b). Frequency dependency of the performance is well described with the experimental data. It is noted that a bounce volume inserted between the compressor piston and the regenerator gives an important effect to the performance. Figure 8 shows the effect of the bounce volume at the fixed frequency. It indicates the similar effect of frequency dependency. This implies that an optimum volume for the driving frequency is existed.

If the inertance part is replaced by a set of solid piston and mechanical spring as shown in Figure 9 schematically, required momentum equations reduces to only two sets. Figure 10 shows the result of example calculation of this method. Figure 10(a) and (b) well represent Figure 7(a) and (b) qualitatively. Distinctive feature of this method is that work flows at each important section are able to evaluate as shown in Figure 10(c).



(a) (b) (c)

Figure 8 Effect of bounce volume at the fixed driving frequency (Xa: piston stroke, Pa: pressure amplitude, Wpv: PV-work at the piston head, Win: effective input power, Joul: joul loss of the winding, Wvi: total input power)

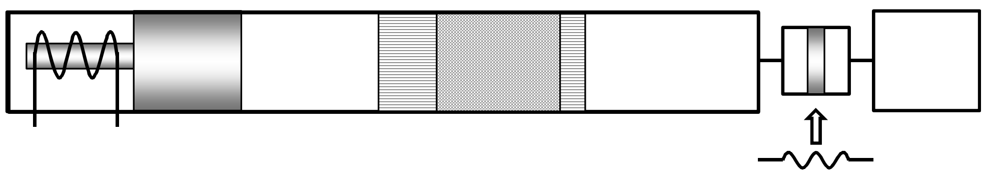
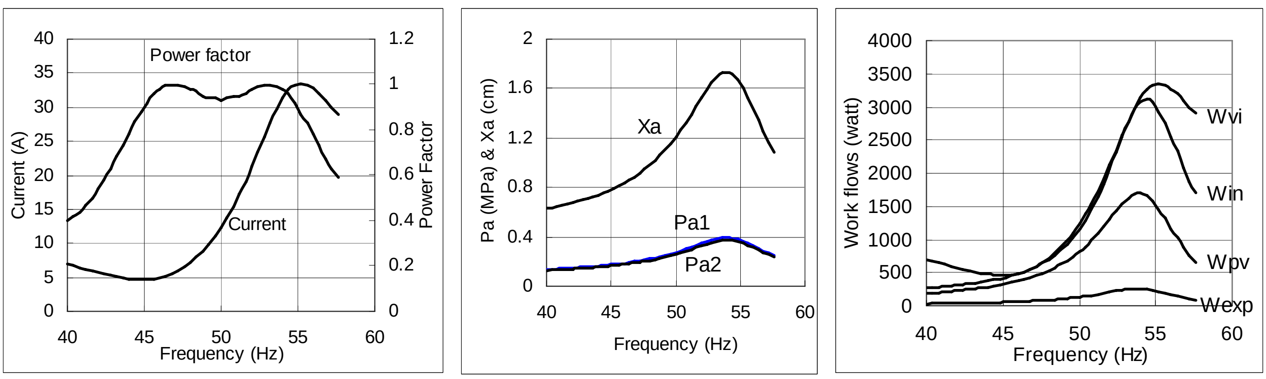


Figure 9 Equivalent model of figure 5

It is noted that the relation of the total input work (Wvi), the effective input work (Win), PV work at the compressor head and the work flow within the pulse tube (Wex) are well explained. The best operating frequency of this particular case is predicted as about 47 Hz, which gives the maximum of Wex/Wvi. This kind of simplified simulation method will be effective to optimize the each component of pulse tube cooler driven by the linear motor system.

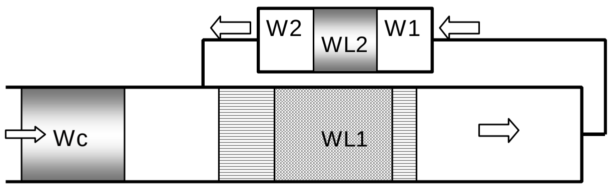


(a) (b) (c)

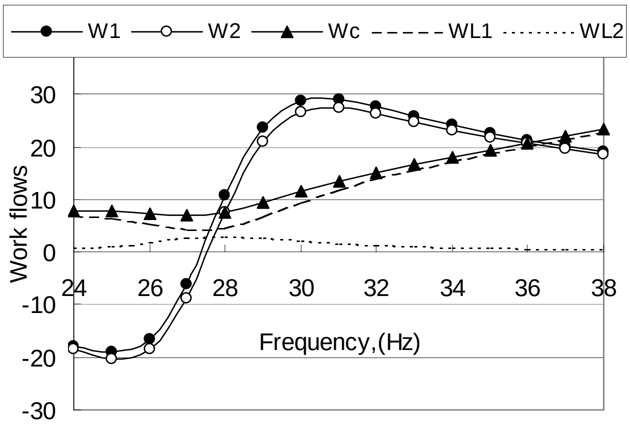
Figure 10 Performance of resonating warm expander with a linear compressor

Modification of warm expander method

Simulation program used for Figure 9 is applicable to other type of phase shift mechanisms. Figure 11(a) shows one of the examples. The warm expander of Figure 9 becomes a resonant warm displacer and the calculated result is given in Figure 11(b). In this example, the same temperature at the both end of the regenerator has been used. This result indicates the possibility of a phase shift mechanism without using the buffer volume. It is found that the work flows though the pulse tube (W1) is recovered as (W2) through the warm displacer, where (WL1) and (WL2) are work loss at the regenerator and the displacer respectively. Therefore, the required work input to the pressure wave generator is smaller than the work input to the regenerator. This means the theoretical limit of the thermodynamic efficiency of this method approaches to Carnot efficiency.



(a)



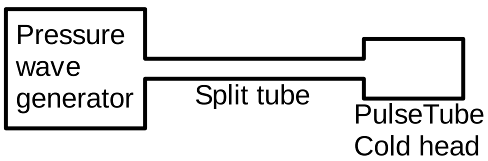
(b)

Figure 11 Frequency dependency of feedback type resonating displacer

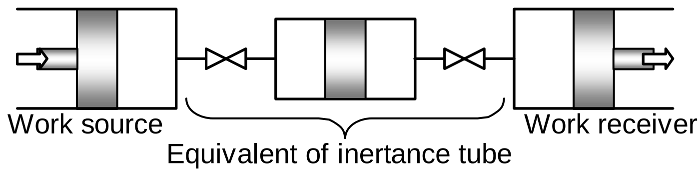
Modification of split-Stirling type pulse tube cooler

One of the disadvantages of the Stirling type pulse tube cooler is the mechanical vibration caused by the integrated compressor unit to the cold head. The split-Stirling type, as shown in figure 12(a), has been developed to compensate this particular disadvantage. Separation of the cold head gives user convenience, however, the degradation of the cooler efficiency is proportional to increase the split tube length due to the pressure drop within the tube.

The possibility to extend the split tube length by introducing the effect of acoustic resonation is considered. The split-tube is replaced by an inertance tube with the minimum acoustic resistance. Based on the modified computer program used for an inertance tube phase shifter [3], an example calculation has been conducted. The tube length is 1 meter and its inner diameter is 16.5 mm. When the frequency is 50 Hz, the work transfer ratio, defined as a work output divided by a work input, of 95 % has been predicted for the case of 1.5 kW work-flow. To understand this phenomenon more clearly, a simplified equivalent model was introduced as shown in figure 12(b).



(a)



(b)

Figure 12 (a) Schematics of split-Stirling type pulse tube cooler, (b) Modified split method.

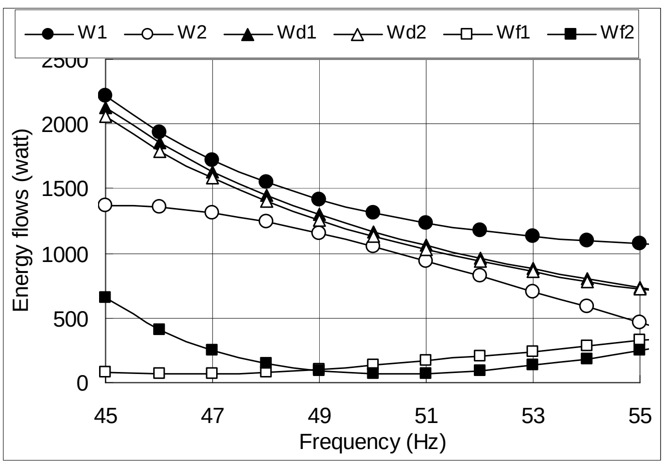


Figure 13 Performance of modified split-Stirling type pulse tube cooler, W`s indicate the workflow, 1 is for the work source and 2 is for the work reciever While, d1 and d2 are for the left and the right side of displacer, respectively.

Figure 13 shows an example calculation of the fixed diameter and the length of the transfer tube. The workflows are plotted as a function of driving frequency. Wf1 and Wf2 are the work loss at the valve mark in the figure 12(b), which represent the loss due to the flow friction at the tube wall. It is noted that the pressure ratio at the regenerator side is much larger than the compressor side. It shows the optimum frequency is a function of insert volume at the inlet of the transfer tube. The maximum work transfer efficiency was 81 % at 50 Hz. This result indicates the possibility of an efficient split-Stirling type pulse tube cooler by use of a resonant transfer tube.

CONCLUSIONS

There are many analytical methods of pulse tube coolers. Some of them are precise analysis of a particular component; such as a regenerator, a pulse tube phase shifter and others are the simplified analysis of total system. Most of them, however, are not easy to use it extensively for the new users. In other words, the developed numerical codes are generally not easy to understand. In these situations, there are still some rooms to develop the user-friendly software, which is able to design the pulse tube cooler to meet the requirement for the particular application. However, the additional thermodynamic losses could not be avoided in most cases.

To keep the thermodynamic efficiency high enough for the applications, several approaches have been described based on the simplified computer analysis. For valved type pulse tube coolers, the operating condition to minimize the valve loss was given.

The use of resonant displacer as a phase shifter for valve-less pulse tube cooler gives Carnot efficiency as its maximum efficiency. Possibility of such a phase shift mechanism without use of an orifice and a reservoir was introduced by a simplified calculation. An efficient split-Stirling type pulse tube cooler is proposed by the use of resonating tube and a bounce volume.

Further analytical study to overcome the various configurations requested from the application fields should be performed, which makes the best use of the basic advantages of the pulse tube cooler.

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