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A NONPROPRIETARY, NONSECRET PROGRAM FOR CALCULATING STIRLING CRYOCOOLERS

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Using entirely nonproprietary and nonsecret sources of information, a design program for an integrated Stirling cycle cryocooler has been written on an IBM-PC computer. The program is easy to use and shows the trends and itemizes the losses. The calculated results have been compared with some measured performance values. In its present form the program predicts somewhat optimistic performance. The program needs to be calibrated more with experimental measurements. As has been done before, adding a multiplier to the friction factor can bring the calculated results in line with the limited test results so far available. The program is offered as a good framework on which to build a truly useful design program for all types of cryocoolers.

Key words: Analysis; computer program; cryocooler; integral cooler; second order; Stirling cycle.

1. Introduction

When I received a request from a client to produce a cryocooler design program for the IBM-PC, which both they and I had available, I assumed that such a program would be readily available from a government agency with an interest in this technology. It would simply be a matter of adapting it to the client's computer. We found, however, that access to all these programs is restricted, usually, to U. S. companies who already have a contract with a government agency to make cryocoolers. It is perfectly legitimate to have such restrictions. However, there needs also to be available a program that can be used in schools and by companies who do not have an official need to know.

Since I had written two design manuals for Stirling engines, and since many of the equations that were used in these manuals came from earlier publications showing how to design cryocoolers, I have undertaken the process of producing a cryocooler design program analogous to my numerous Stirling engine design programs.

The previous literature will be cited. The method of analysis will be explained in general. The specific arrangement of a Stirling cycle cryocooler for which the program was written is given. A full list of input values is presented with typical input values. A sample of the full calculated output is given and explained. Limited test results are compared with calculated performance and discussed. Application areas for this type of design program are suggested. Finally, conclusions are drawn about the utility of this type of design program.

2. Literature Review

A full literature review on cryocooler analysis is not attempted. Only those antecedent publications that have a bearing on the computer program described in this paper will be given. In 1968 the author lead a group of engineers at the Donald W. Douglas Laboratories, in Richland WA in developing a Stirling engine for an artificial heart. We developed our own design method. Gradually we became aware of other methods of analysis. We had simple methods and very complicated and time consuming methods. Later the author now at the Joint Center for Graduate Study, University of Washington and sponsored by NASA-Lewis, wrote two design manuals for Stirling engines [1,2]. Also a long IECEC paper outlined in detail this design method [3]. As a result of these publications this method of design has been used widely among those who had no access to proprietary information.

There were a number of prior publications which were discovered in a literature search which had an important bearing in selecting the equations that were recommended in the design program which was presented. Crouthamel and Shelpuk [4] gave all the design equations for a VM cycle cooler. Zimmerman and Longsworth [5] contributed to our understanding of shuttle heat loss as did Rios [6]. Leo [7] supplied the equation for appendix loss. Gorring [8] supplied the equation for matrix conduction. Note that many of these publications are in the cryocooler literature.

Since the design manuals have been written the author has continued to evaluate and perfect the isothermal analysis. A 1980 IECEC paper showed that up to that time the isothermal analysis was as good as any other available [9]. A further extension of this isothermal program for large machines was done in which the effect of adiabatic hot and cold gas spaces was included for both heat engines and heat pumps. This effect is particularly serious for engines or heat pumps that operate over a small temperature ratio. The 1983 IECEC paper showed that the improved isothermal analysis agreed with some engine test data over almost the full operating range for both power output and efficiency to within plus or minus 10 percent with absolutely no adjustments [10]. Finally the isothermal analysis has been extended to predict the operation of free-piston machines and search for an optimum design [11].

3. Method of Analysis

In the literature of Stirling engine design, methods are classified as first, second, and third order. First order methods use variations of the Beale equation [12] to predict what the power can be, given the displacement, speed, pressure and temperatures. This method has been extended to large cryogenic refrigerators by Walker [13]. It does not apply to miniature cryocoolers. The method suggested in this paper is second order. It will be defined in more detail later. Second order methods are the simplest methods that take into account all the dimensions that influence performance. Third order methods use nodal analysis to simulate the physical process very closely. They are expensive to run because they use a large amount of time on the very largest and fastest computers. They are useful for special studies, but are not practical to be used in search routines to find the best design.

The basic assumption of the isothermal, second order analysis is that at each point in the cycle the pressure throughout the working gas volume is the same for each instant in time. In reality there are pressure differences between the different parts of the working gas due to flow friction. In a well designed machine these differences are small in comparison to the pressure changes due to expansion and compression. Neglecting the pressure differences at this stage greatly simplifies and speeds the calculation with little loss in accuracy.

It is also assumed that an effective gas temperature can be identified for each part of the working gas volume which holds for the full cycle. This assumption is easy to defend in the regenerator where gas is in close thermal contact with the solid. It is not true and requires corrections [10] for the hot and cold spaces of large machines. Careful measurements of the pressure and total gas volume during one cycle in a Stirling engine for artificial heart power showed that on this scale the isothermal assumption was better than the adiabatic assumption. The artificial heart engine is on the same order of size as the miniature cryocoolers.

Using the two above assumptions, it is a simple matter to calculate the pressure during the cycle usually using the perfect gas law. If the pressure of the working gas is plotted against the volume of gas in the cold space, a closed curve is produced. The area inside this curve is equal to the heat absorbed by the working gas in one cycle. When multiplied by the frequency we call this the thermodynamic heat input. If the pressure of the working gas is plotted against the total working gas volume, a closed curve is also produced. The area inside this curve is equal to the work required to be applied in a perfect machine during one cycle. When multiplied by the frequency, we call this the thermodynamic power input.

These basic thermodynamic values must be modified by the many losses that occur in a real cryocooler. To the basic thermodynamic power must be added the flow losses of the working gas, chiefly in the regenerator matrix. Also the seal friction, the mechanical friction, and the electric motor losses must be added to obtain an estimate of the power required.

From the basic thermodynamic cooling effect must be deducted all the thermal conduction and radiation heat losses. Also the flow losses in the cold part are converted to heat which must also be deducted. The matrix heat transfer is not perfect. Therefore, the gas returning back to the cold space is at a higher temperature than when it left and must be recooled. Finally, there is a loss involving the gap between the displacer and the cold finger cylinder. Heat is transferred by this shuttle loss only when the displacer oscillates.

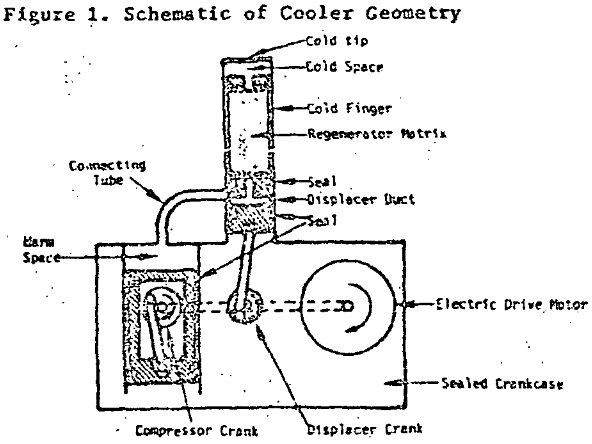
Each one of these losses mentioned above is calculated by an equation. In the second order method there is assumed to be no interaction between the various loss mechanisms. In a miniature cryocooler accurate estimation of these loss mechanisms is especially important. We find that in most cases the mechanical and electrical losses are more important than the gas flow losses and the thermodynamic power in determining the electric power requirement. On the cooling side, we find that the thermal losses consume most of the thermodynamic cooling effect. Therefore the net cooling effect is subject to relatively large errors because it is the small difference between large numbers.

In a miniature cryocooler there is so much heat transfer area for so little volume that one would be tempted to assume that the heat transfer is nearly perfect. If tests show that this is so, calculation of the performance would be speeded and simplified. We did not make this assumption. We calculated a heat transfer coefficient for the hot space and the cold space and also decided on a heat transfer area. By trial and error we found an effective cold gas temperature lower than the specified cold metal temperature such that the heat that can be transferred based upon the calculated area and heat transfer coefficient is equal to the heat that needs to be transferred. At the same time by trial and error we found an effective warm space temperature that is higher than the specified warm metal temperature such that the heat that can be removed from the warm space is equal to the heat that must be removed. When the fractional error in the heat balance at both ends is less than the convergence criteria for two trials in a row, the calculation ends and the results are printed out.

4. Cooler Geometry

Although this program can be adapted to any type of Stirling cryocooler, it is particularly set up for an integral cryocooler shown schematically in figure 1. An hermetically sealed electric motor operates two cranks. One crank operates the displacer in the normal manner. The other crank operates the displacer piston in the inverted manner. The cold finger containing the displacer is in the same plane as the compressor piston. The two cranks are offset 90 degrees to provide the proper phase relationship. The program takes into account the slightly non-sinusiodal nature of the piston and displacer motion due to the crank geometry.

Figure 1. Schematic of Cooler Geometry



5. Input Values

The program stores all the input data on a disk file. The user has the option of starting with the set of input data which was used last which is on this disk file, or using base case input data which is written into the program. The operator makes this decision to start the program. Then the main menu is displayed on the screen (table 1). The operator can start calculating the next case by keying in a 1 or look at one of seven submenus by keying in 2 thru 8.

Table 1. Main menu for input.

```
MAIN MENU

1 — Start calculation next case.
2 — Program control parameters.
3 — Engine operating conditions.
4 — Pistons and connecting rods and crank case.
5 — Warm flow passages.
6 — Regenerator properties.
7 — Cold flow passages.
8 — Heat cond., seals and motor.
Your choice is — (Key number and enter) —5
```

Program control parameters. Table 2 shows the menu for all the program control parameters. On the left hand side is the menu number. In the middle is the definition. On the right hand side is the current value. Note that at the bottom of the table a menu number is called for. The operator puts in an integer between 1 and 8 (No decimal point.). If a real number with a decimal point, or an integer outside of the specified range is entered the program displays the appropriate error message and asks for the menu number again. If 1 is entered the program displays the master menu again.

Table 2. Submenu for Program Control Paramenters

```
Program control parameters:
1 — Return to main menu.
2 — Convergence criteria. ______________________ .0010
3 — Case number defined by operator. ___________ 31
4 — Graphic option — 0=no, 1=yes. ______________ 1
5 — Printing option — 0=no, 1=yes. _____________ 0
6 — Number of time steps per cycle. _____________ 24
7 — Pressure drop option ________________________ 1
        0 = Calculate from pressure drop correlation.
        1 = Calculate from pressure drop test.
8 — Reheat loss option _________________________ 0
        0 = Calculate from heat transfer correlation.
        1 = Calculate from pressure drop test
            using Reynolds analogy.
9 — Multiplier for friction factor _____________ 1.00
Menu number? 1
```

If 2 is entered, the program then asks that a real number be entered with a decimal point. This convergence criteria is compared with the fractional change from one trial to the next of the thermodynamic power input and the thermodynamic heat input. This criteria must be a small positive real number. The program checks to see if the number input is between 0.0 and 0.1. If it is, the convergence criteria is changed to the new value, and the menu is displayed again with the new value in place. If the number that is input is not within the specified range, an error message is displayed, and the menu is redisplayed without change.

If 3 is entered, the program asks for an integer. If this case number is negative, the program rejects it and asks for another. Otherwise the case number can be set to any number desired. The case number is automatically indexed without operator attention.

If 4 is entered, the graphic option can be changed. Some IBM-PC's have the ability to draw graphs on the screen. Others do not. The program can be used either way. If the computer does not have the ability to do graphics and the graphics option is set at 1, the computer will probably hang up. The program will have to be restarted.

If 5 is entered, the printing option can be changed. Since some computers are equipped with printers and others are not, the program can be used either way. If a printer is called for but is not connected, the program will stop.

If 6 is entered, the number of time steps per cycle can be changed. More time steps leads to higher accuracy at the expence of speed of computation. Twenty-four has been found to be a good compromise. A formula has been developed to compensate for a small number of time steps [1]. However, is has not been used in this particular program.

If 7 is entered, the option on the way flow losses are calculated can be changed. For the 0 option, the flow loss is determined based upon a correlation by Kays and London as simplified by Martini [2]. For the 1 option, the flow resistance test used to qualify the displacer for use is employed instead. Other options can be added as needed.

If 8 is entered, the option on the way the heat transfer coefficient for the regenerator is calculated can be changed. For the 0 option, a simplified Kays and London correlation is used [2]. For the 1 option, the heat transfer coefficient is derived from the flow friction test using Reynolds analogy. Other options can be added as needed.

If 9 is entered, a multiplier on the friction factor can be changed. If the Kays and London correlation is used, this factor is multiplied by the calculated friction factor to give the friction factor used to calculate flow loss. If the pressure drop test data is used, the computed pressure drop is multiplied by this factor.

Engine operating conditions. Table 3 shows the menu for the engine operating conditions. The menu is shown twice to illustrate how it is changed. The inputs are self explanatory. The program adjusts the working gas inventory for the first 4 cycles to make the time averaged working gas pressure equal to the input value. After that leakage continues to operate between and crank case and the working gas. An effort has been made to ask for the input values in customary units. They are then converted to a consistent set of units in the program.

Table 3. Submenu for Engine Operating Conditions

```
Engine operating conditions:
1 -- Return to main menu.
2 -- Average working gas pressure, psia ____________ 460.00
3 -- Motor speed, rpm ______________________________ 1200.00
4 -- Metal temperature in cold space, K ____________ 70.00
5 -- Metal temperature in warm space, F ____________ 80.00

Menu number? 4
Value? (with decimal point)80.
Engine operating conditions:
1 -- Return to main menu.
2 -- Average working gas pressure, psia ____________ 460.00
3 -- Motor speed, rpm ______________________________ 1200.00
4 -- Metal temperature in cold space, K ____________ 80.00
5 -- Metal temperature in warm space, F ____________ 80.00

Menu number?
```

Pistons -- connecting rods -- crankcase Table 4 shows the menu which allows values connected with the mechanical motion to be changed. The same organization is used as with previous tables. There is enough space to clearly define each input value and give the units. At present the volume of the crank case space is not known very well and is not used in the calculation. Strictly, h ever, it should be used in leakage calcul ns and in kinetic simulations of the cooler.

Table 4. Submenu to change inputs related to pistons, connecting rods and crankcase

```
Pistons and connecting rods and crank case.
1 -- Return to main menu.
2 -- Crank angle, degrees __________________________ 90.00
3 -- Diameter of compressor piston, inch ___________ .5620
4 -- Seal diameter at displacer, inch ______________ .1875
5 -- Radius of compressor piston crank, inch _______ .0520
6 -- Radius of displacer drive crank, inch _________ .0550
7 -- Length of compressor piston conn. rod, inch ___ .6183
8 -- Length of displacer drive conn. rod, inch _____ 1.2187
9 -- Volume of crank case space, cu. inches ________ 2.0000

Menu number? 1
```

Warm flow passages Table 5 shows the submenu which allows values connected with the warm flow passages to be changed. Most of the entries are self explanatory by reference to figure 1. The warm displacer duct is the hole thru the seal and bearing part of the displacer to allow gas to pass from the connecting tube to the warm end of the regenerator matrix. The number of velocity heads are used to account for the flow resistances of entrances, exits and bends in the gas flow passages. Standard handbook values were used [13]. The dead volume is an estimate of the dead volume in the warm space in addition to that in the piston end clearance, connecting tube, and the warm displacer duct.

Table 5. Submenu for Warm Flow Passage Values

Warm flow passages:

1 — Return to main menu.	
2 — Diameter of connecting tube, inch	.16[illegible]
3 — Diameter of warm displacer duct, inch	.6[illegible]
4 — Effective compressor piston clearance, inch	.0[illegible]
5 — Length of connecting tube, inch	2.2[illegible]
6 — Length of warm displacer duct, inch	.4[illegible]
7 — Number of velocity heads in connecting tube	2.25
8 — Number of velocity heads in displacer duct	2.24
9 — Dead volume at warm end of displacer, cu. inch	.0[illegible]

Menu number? 1

Regenerator properties Table 6 shows the submenu which allows values connected with the regenerator to be changed. The length of the regenerator matrix includes space for the two coarse screens on each end. The porosity of the matrix is computed from the total volume of the fine matrix, the weight of the fine screens, and the density of stainless steel (7.817 g/cc)

Table 6. Submenu for Regenerator Properties

Regenerator properties:

1 — Return to main menu.	
2 — Inside diameter of displacer cylinder, inch	.10[illegible]
3 — Diameter of regenerator matrix, inch	.12[illegible]
4 — Wire diameter in regenerator matrix, inch	.0[illegible]
5 — Square mesh of screens in reg. matrix, wires/in.	5[illegible].[illegible]
6 — Number of fine screens in reg. matrix,	[illegible]
7 — Length of regenerator matrix, inch	2.[illegible]
8 — Weight of fine regenerator screens, grams	1.318[illegible]
9 — Number of coarse end screens in matrix	4
10 - Thickness of each coarse end screen, inch	.01[illegible]
11 - Test flow rate of N2 thru displ., std cc/min.	710.[illegible]
12 - Pressure drop at test flow rate, psi	10.[illegible]

Menu number? 1

Cold flow passages Table 7 shows the submenu which allows values connected with the cold flow passages to be changed. In this case there is a single orifice which squirts gas into the cold space. The correlation by Hauser [16] was used to calculate the heat transfer coefficient in the cold space.

Table 7. Submenu for Cold Flow Passages

Cold flow passages:

1 — Return to main menu.	
2 — Diameter of cold orifice, inch	.0[illegible]
3 — Effective displacer clearance at cold end, inch	.0[illegible]
4 — Length of cold orifice, inch	.0[illegible]
5 — Number of velocity heads in cold orifice	1.5[illegible]

Menu number? 1

Heat conduction, seal, and motor properties Table 8 shows the submenu which allows values connected with heat conduction, seal, and motor properties to be changed. All the input values in this list, except for the inch dimensions are difficult to determine. The emmissivities can vary over a wide range depending upon the exact properties of the surfaces. However the heat loss thru the vacuum insulation is small in any case, so these errors are unimportant. The efficiency of the drive motor and the seal and mechanical fricition are very important. They must be measured by separate tests. In this case the efficiency of the motor was estimated from the motor specifications. The seal and mechanical friction was determined by adjusting it so that the electric power demand at the very beginning of a cool down test was about right. The speed of the motor during this test was not recorded and had to be estimated. In reality the seal and mechanical friction should depend upon engine speed and working gas pressure. More realistic friction can be added when the information is available.

Table 8. Submenu for Heat Conduction, Seal and Motor Values.

Heat conduction, seal, and motor values:

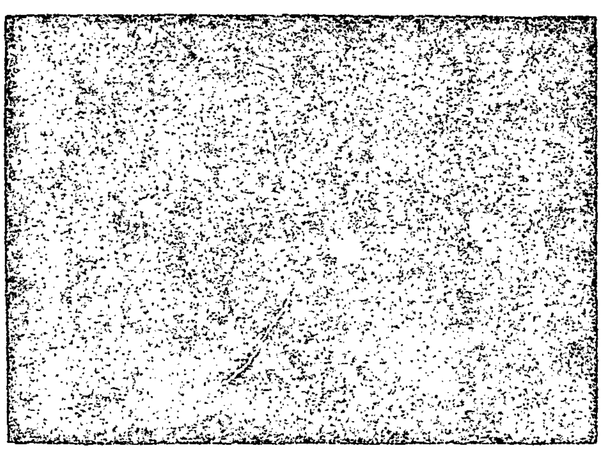
1 — Return to main menu.	
2 — Diameter of vacuum insulation chamber, inch	.8[illegible]
3 — Efficiency of drive motor, %	[illegible]
4 — Emissivity of cold finger	.0[illegible]
5 — Emissivity of vacuum insulation chamber	.0[illegible]
6 — Combined seal leakage, cu.inch/sec/psi	.0[illegible]
7 — Seal and mechanical friction, watts	7.[illegible]
8 — Thickness of cold finger wall, inch	.0[illegible]
9 — Gap between cold finger wall and displacer, inch	.0[illegible]

Menu number? 1

6. Calculated Outputs

During solution two outputs are possible. If the graphic option is on, a display is shown on the screen like that shown in figure 2. This display is very useful to show at a glance whether the solution is converging and where some of the problems in design are. The ellipses at the right show the gas pressure plotted against the total volume. The first cycle is on top. The other 6 cycle are below as the gas pressure is adjusted. The ellipses at the left are the cold gas volume plotted against the working gas pressure. The sine curve at the top is the displacer motion for one cycle. The sine curve at the bottom is the compressor motion for one cycle. The nearly horizontal line at the top left records the trials of the effective warm space temperature. Full vertical scale is 0-400 K. Note that most of the adjustment is made after the first cycle. The nearly horizontal line at the bottom left records the trials for the effective cold space temperature. Note that there was little change.

Figure 2. Graphical Display during Solution



If the graphic option is off, Table 9 is displayed on the screen. The numbers in the first two columns are the numbers that are compared with the convergence criteria which is shown in the top line of Table 9. These fractional changes for cycle 6 and 7 were all under 0.001. Note that the work in and the heat out are now only changing in the fourth significant figure. The last column, the gas inventory, has unusual units. It is the gas inventory in gram moles multiplied by the universal gas constant, 8.314 J/(gmol*K).

Table 9. Progress to convergence table

Convergence criteria is .00100 for Gas 0 31

Cycle Numb.	Change Power In Frac.	Change Cool In Frac.	Work In Joules	Cooling In Joules	End Pressure MPa	Time Step Msec.	Gas Invent. J/K
1	1.031759	.955576	[illegible]	.054755	[illegible]	1.5915	[illegible]
2	.873493	.199875	.172291	.052871	2.6343	1.5915	.0128819
3	.052767	[illegible]	.163234	[illegible]	2.6235	1.5915	.0128738
4	.024125	.056879	.162500	.050451	2.6256	1.5915	.0126912
5	[illegible]	.051617	.162599	.055325	2.6226	1.5915	.0128933
6	[illegible]	.050137	.152743	.050959	2.6275	1.5915	.0128377
7	[illegible]	.058443	.162581	.055987	2.6255	1.5915	.0128923

Speed of Solution On the IBM-PC equipped with the 8087 coprocessor, and running Microsoft FORTRAN-77, the solution time for the above same case with 7 iterations is 6 seconds without graphics and 20 seconds with graphics. This does not include the time for printing out or displaying the solution. The display of the output to the screen takes 2.4 seconds additional. The printout takes much longer and depends upon the type of printer that is being used. With some type of spooler, printing can lag behind. Meanwhile the operator may investigate new possibilities using the information available at the monitor.

Calculated Performance At the end of the solution, the table of calculated performance is displayed on the monitor and is printed out as well, if the print option is on. Table 10 is the calculated performance for the input values given in Tables 2 to 8. The first thing to notice is that some losses are very important and some are negligible. However, it is dangerous to neglect the negligible losses in the design calculation. If you do your search for a better design will lead you into an area where these negligible losses are no longer negligible.

Table 10. Calculated Performance

Martini Eqn. Analysis of Stirling Cycle Cryocooler
Integral Geometry. **CALCULATED PERFORMANCE**

POWER REQUIREMENT, WATTS		COOLING EFFECT, WATTS	
Thermodynamic Power	4.5165	Ther.Dyn.Cooling Eff.	1.6127
Cld. Orfc. F.L.	.0231	Cld. Orfc. Fric. Heat	.0051
Reg. Flow Loss	1.2555	Regen. Fric. Heat	.6183
Disp. Duct.F.L.	.0067	Shuttle Heat Loss	.1257
Comp. Tube F.L.	.0027	Reheat Loss	.1129
Mech. Friction	7.[illegible]	Appendix Loss	.0520
Elect. Mtr. Loss	9.5585	Temp. Swing Loss	.0585
		Cld. Finger Wall Cond.	.1335
Elect. Pwr. In	22.6759	Disp. Wall Cond.	.0123
		Reg. Mtx. Cond.	.0077
Eff.Cld.Ga.Temp.,K	78.1233	Vac. Insul. Heat Loss	.0311
Eff.Wrm.Ga.Temp.,K	324.5169	NET COOLING EFFECT	[illegible]

On the power requirement side, note the mechanical friction and the electrical motor losses are the two big ones accounting together for 74.4 % of the power requirement. The flow loss thru the regenerator accounts for 5.5 % of the power requirement. All the other flow losses account for only 0.2 %, leaving 19.9 % for the necessary thermodynamic power. Obviously the place to start with improvements is to find some way to reduce the mechanical and electrical losses.

On the cooling effect side, it appears that only 26.2 % of the thermodynamic cooling effect survives the gauntlet of losses to become net cooling effect. In this analysis it is assumed that all the cold orifice flow loss and half the regenerator flow loss must be deducted as heat loss from the thermodynamic cooling effect.

The shuttle heat loss and the appendix loss must be considered together, because as one increases the other decreases. Shuttle heat loss occurs by radial conduction across the gap between the cold finger cylinder and the displacer. Because of the longitudinal temperature gradient along the cold finger cylinder and along the displacer, when the displacer is at the warm end of its stroke it is colder than the cold finger cylinder all the way along, so it heats up. When the displacer is at the cold end of its stroke, it is warmer than the cold finger cylinder, so it cools down. Each cycle this process moves the heat that has been transferred one stroke length toward the cold end. This important heat loss can be minimized by using low conductivity cylinder walls such as plastic or glass or ceramic. The loss can be minimized by increasing the gap and increasing the length of the regenerator.

Appendix loss occurs as the gas is packed into and unpacked from the appendix gap between the displacer and the cold finger wall. This gap is closed off at the warm end with a sliding seal and is open at the cold end to the cold space. As the pressure increases cold gas is packed into this space. As the pressure decreases not so cold gas flows back into the cold space. Increasing the thickness of the gap reduces shuttle loss but increases the appendix loss.

A number of equations have been published for shuttle heat loss and for appendix heat loss. As far as is known there has been no experimental confirmation published of any of these equations.

The reheat loss should, for a cryocooler, be called the recool loss. Both due to pressure changes and flow, a large amount of heat must be transferred each cycle in the regenerator. No matter how good the heat transfer, the gas always re-enters the cold space warmer than when it left. The reheat loss can be made small at the expense of making the flow heating or the heat conduction loss or both too large. A careful balancing of dimensions is needed.

The temperature swing loss is a correction to the reheat loss to compensate for the heat capacity of the regenerator being less than infinite. In this case it is negligible, but it can be important.

The last 4 losses are heat conduction or radiation losses that go on whether the displacer moves or not. By far the largest is the cold finger wall since this must be thick and metal to hold back the pressure and contain the helium. The displacer wall conduction is much less because this is made of plastic. The displacer matrix conduction is even smaller because the metal screens are divided and make contact only in a few spots. Vacuum insulation loss is negligible.

Record keeping The operator can continue to experiment with the computer program to see how different inputs affect the output. He can do this without waiting for the printer on line at all. However with a spooler of some sort, the printer can keep track of all the inputs and all the outputs for further study. It may be running many cases behind what the operator is considering. After the output of table 10, the full input values are printed out. These values are printed out in the same order as they are shown in tables 2 to 8.

7. Results and Discussion

Figure 3 shows how the computed performance without any adjustmentof the calculation proceedure with test results. The net cooling effect is plotted against the cold finger temperature. Note that if the flow resistance data is used for both the flow loss and the reheat that the net cooling is too optimistic by 0.2 to 0.3 watts. If the Kays and London correlations are used, the net cooling is optimistic by 0.12 watts. From the data that so far has been obtained, it appears that when the cryocooler motor speed was not measured that it was running at 1600 to 1700 rpm. Figure 3 shows that the calculated and measured performance shows the same trend.

For the flow rate that results in a 10 psi pressure drop thru the displacer, the Kays and London correlation calculated 18.4 psi. Also a number of investigators have found that the flow resistance calculated for steady flow must be multiplied by a factor to make the measurements and the predictions for engines agree more closely. At one time the author used a factor of 2.9 to make the calculated output agree with measurements for both helium and hydrogen [17]. Tew [18] used a factor of 4 for hydrogen and 3 for helium to make his calculations agree with test results. A recent test reported by Taylor and Aghili [14] showed that oscillating flow in tubes increased the friction factor by a factor of about 3. Possibly finding the right multiplier between reversing flow and steady flow resistance and the right way to apply this multiplier could bring the experimental and test results into agreement.

Figure 4 shows the first attempt at adjusting the program to fit the current experimental data. When the flow resistance test data are used, a mutiplier on the computed pressure drop of 3.3 to 3.6 is needed to bring the computed results in line with the measurements. The measurements are consistant in that they show a trend with very little scatter. The computed results derived from flow resistance test data do not show quite the same trend.

When the Kays and London flow correlation is used, the multiplier only has to be 1.25. Also the trends in the data and the adjusted calculation are the same. The correlation shows a transition from pressure drop proportional to flow rate to pressure drop proportional to the square the the flow rate in a smooth manner over a factor of about 10 in flow rate. Possibly the pressure drop is not directly proportional to the flow rate.

These exercises in adjustment have been successful in showing that adjustments can be made. However, when we have more information, particularly the motor speed, for each point, our conclusions may be quite different than they are now. However, the isothermal method has an important advantage in that it itemizes the losses and therefore gives guidance about where to make improvements.

8. Application Areas

Although the specific cryocooler design program described in this paper is for an integral Stirling cryocooler, The same programing concept can be applied to split Stirling and free-piston Stirling machines. A free-piston engine has been calculated using this type of analysis [11]. The speed for calculating each case is fast enough so that once the design method is calibrated, an optimization search can be programed to search for the best design automatically. A thorough optimization search is already programed [11].

9. Conclusions

Using only open sources of information, a cryocooler design program has been written which gives reasonable results. It requires inputs of cooler dimensions, measured flow losses, mechanical friction, and electric motor losses. Since the current error between calculated and measured performance is not large, and since the calculated performance shows the same trend as the measured performance, we expect that this computer program can be adjusted to model a real cryocooler quite exactly. The program needs to be adapted and customized for each user since there are so many ways to build a cryocooler. Also skill is needed in making the proper adjustments so that the program will be accurate over a wide range of design options. In itemizing the losses the program shows clearly where the greatest improvements in design can be made.

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Figure 3 Comparison of calculated cooling effect without adjustments with measurements.

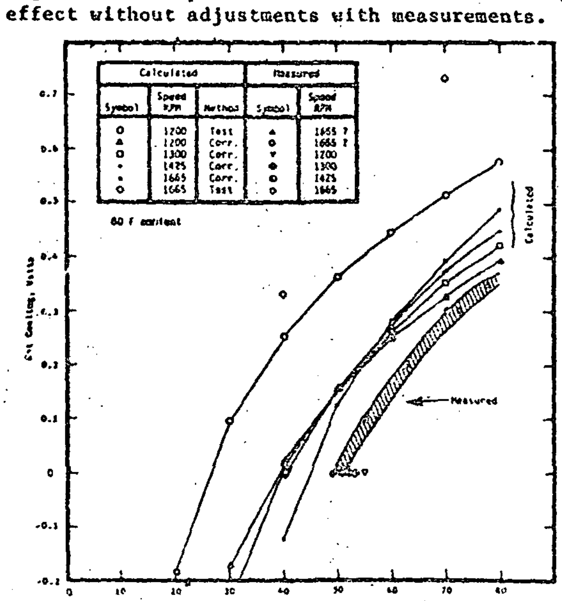

Figure 4. Comparison of calculated cooling effect with adjustments with measurements