LA-UR-97- 3985

Title:

Downhole Pulse Tube Refrigerators

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Downhole Pulse Tube Refrigerators

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September 1997

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Introduction

This report summarizes a preliminary design study to explore the plausibility of using pulse tube refrigeration to cool instruments in a hot down-hole environment. The original motivation was to maintain Dave Reagor's high-temperature superconducting electronics at 75 K, but the study has evolved to include three target design criteria:

- Cooling at 30°C in a 300°C environment,
- Cooling at 75 K in a 50°C environment,
- Cooling at both 75 K and 30°C in a 250°C environment.

These specific temperatures were chosen arbitrarily, as representative of what is possible. The primary goals are low cost, reliability, and small package diameter.

Pulse-tube refrigeration is a rapidly growing sub-field of cryogenic refrigeration.^{1,2} The pulse tube refrigerator has recently become the simplest, cheapest, most rugged and reliable low-power cryocooler. We expect this technology will be applicable downhole because the ratio of hot to cold temperatures (in absolute units, such as Kelvin) of interest in deep drilling is comparable to the ratios routinely achieved with cryogenic pulse-tube refrigerators.

Pulse tube refrigerators follow the same thermodynamic cycle as Stirling refrigerators,³ but they eliminate all moving parts at low temperature, via a passive, dissipative acoustic-impedance network and the so-called "pulse tube," which mimic the dynamics of the Stirling's cold piston. The function of the pulse tube is to provide thermal isolation between the cryogenic heat exchanger and the dissipative network.

As of September 1997, we have no funding to continue this work. We would enjoy building one of these coolers; but we are sharing this information freely in the hope that more clever people will be able to improve the design and/or build one.

Important features common to all three designs

A downhole pulse tube refrigerator must be supplied with mechanical power and must reject waste heat to the well environment. It seems to us that circulating mud or other fluid pumped from the surface will be available in the most interesting situations (such as logging while drilling), so our discussion will tacitly assume use of a hydraulic motor powered by the mud to provide shaft power to the refrigerator, and will assume rejection of waste heat to flowing mud. However, other options are fully compatible with our designs. For example, if electric power or rotary power can be supplied to the unit then either could provide the shaft power; if no forced mud flow is available to carry away waste heat then natural convection of the fluid in the hole could be used, with some reduction in performance.

In our designs, a rotating shaft, driven from below at roughly 2400 rpm, moves bellows-sealed pistons longitudinally via a wobble plate, as shown in the accompanying figures. The wobble plate is the mechanism used in some torpedo engines, in about half of automobile air-conditioner compressors, and in some hydraulic equipment, so it is a fairly conventional, rugged, reliable, inexpensive technology. Each wobble plate drives 4 pistons at relative phases of 0°, 90°, 180°, and 270°; each of these pistons drives a pulse tube refrigerator. Four pistons equally spaced in phase is the minimum number yielding smooth torque on the shaft, independent of rotation frequency (recall the large flywheels needed for 1-cylinder and 2-cylinder internal combustion engines).

Our trusted colleague John Corey (CFIC, Troy NY) estimates that a simple, robust wobble plate mechanism of this size will probably be about 50% efficient at converting shaft power to pV power. With careful engineering

and light lubricants, this could be raised to almost 70%. We assume 50% for design purposes. This low mechanical efficiency simply requires more shaft power (which we expect to have in abundance!) and more capacity to reject waste heat to the bore.

DuPont markets a family of perfluoropolyether high-temperature lubricating oils under the trade name Krytox. The highest-temperature member of the family can be used to $600^{\circ}\text{F} = 316^{\circ}\text{C}$. At room temperature its viscosity equals that of 10W30 motor oil at about -30°C ; at 300°C its viscosity is about 5 times lower than that of 10W30 motor oil at 120°C . Hence, we expect that automobile-engine design practice will serve as an excellent guide for design of our wobble mechanisms with this lubricant.

Helium gas at an average pressure of 40 bar = 580 psia will be the thermodynamic working substance, with the pistons causing pressure oscillations of 0.4 MPa amplitude. The lubricated space around the wobble plates and other bearings will contain gas at the same average pressure, so there will be no average pressure difference across the bellows.

For quantitative calculation of cooling power etc. for a given size, we have relied on our own code,⁴ which enjoys widespread use for pulse-tube refrigerator design and analysis.

30 Centigrade in hot well

Figure 1 illustrates our 300°C to 30°C pulse-tube refrigerator design. This figure shows a 3 inch diameter package containing 4 refrigerators operating in parallel. We will briefly outline the key aspects of this design:

Thermodynamic: Each of the four coolers comprises a stacked-screen regenerator between two drilled-copper-block heat exchangers, with a piston on the hot end and a pulse tube and inertance tube on the cold end. The piston motion on the hot end, and the acoustic impedance of the pulse tube and inertance tube on the cold end, cause the 40-bar helium gas in and near the regenerator to follow a Stirling cycle, basically: compression, displacement upwards, expansion, displacement downwards. The expansion cools the helium located above the cold heat exchanger and the subsequent displacement downwards forces the cool helium through the cold heat exchanger; this is the desired thermodynamic effect. The compression and subsequent displacement upwards rejects the cycle's waste heat at the hot heat exchanger, whence it is conducted to the pressure case and carried away by flowing mud. Our design calculations predict 36 Watts of cooling power at 30°C for this

design, requiring 140 W of pV input power and hence 280 W of shaft power.

We estimate $1/3$ of the cooling power will be devoted to heat leak, and the balance will be available for instrument cooling. A larger package diameter and other simple design changes would permit greater cooling power; a smaller package diameter might be possible.

Thermal: The hot heat exchangers, pistons and wobble mechanism, upper hot heat exchanger, inertances, and outer pressure case are the parts that are uniformly at the hot, well-bore temperature. The regenerators and pulse tubes span the temperature difference. The cold heat exchangers and the instrument platform extending above them are near 30°C . The rotating shaft, wobble plates, and pistons are steel, and much of the internal support structure is copper or aluminum to conduct heat. We chose such primitive conducting structures instead of heat pipes, for simplicity and low cost, with some degradation in performance.

Best performance in this design would be obtained using vacuum insulation with multi-layer radiation baffling, but this would be awkward if the user anticipates frequent field changes to the instrument package. In this situation, we'd opt for ceramic-fiber insulation, with air at atmospheric pressure, so the user could disassemble and reassemble the system without needing a vacuum pump.

Pressure: The fluid pressure in the well is supported by the pressure case; in this drawing it is sized for 15,000 psi = 1 kbar. A thicker case could of course support greater pressure. Inside, three separate spaces are sealed from each other: the thermodynamic helium in the regenerators, heat exchangers, pulse tubes, inertances, and piston spaces at 40 bar (with 40 Hz oscillations of 8 bar pk-pk); helium plus lubricating oil in the wobble-drive spaces, also at 40 bar; and vacuum (or 1-bar air for debugging or for easy conditions) in the insulation space just inside the pressure case. In addition to these three primary spaces, a series of seals around the shaft at the bottom allow transition from high-pressure, dirty well fluid to high-pressure clean oil to 40-bar oily helium.

Miscellaneous: Most interesting applications will use electrical power for instruments, so we've included an electric alternator in some of our drawings. Alternatively (no pun intended), battery power may be adequate in some applications.

To maintain $\lesssim 75^{\circ}\text{C}$ for some reasonable period of time while the shaft is not rotating, so that the instrument package doesn't warm up to 300°C

whenever the drilling mud stops for a few minutes, we'll incorporate a mass of Cerro alloy or Wood's metal on the cold platform, as used in today's dewar-based logging tools.

Novel: The four inertances connect at a common point, effectively a point of zero oscillating pressure. Hence, there is no need for the reservoir volumes usually seen in pulse-tube refrigerators. This point is a convenient location for dealing with the need to minimize average pressure difference across the bellows. A connection here between the two helium-filled spaces would ensure zero average pressure difference. Such a connection would have to incorporate an oil filter to keep oil out of the thermodynamic spaces. We believe the oil filters used in commercial Gifford-McMahon and Joule-Thompson cryocoolers would be suitable.

30 Centigrade in hot well—Stirling option

To illustrate the difference between Stirling refrigerators and pulse tube refrigerators, in Fig. 2 we illustrate a Stirling design that is comparable to the pulse tube refrigerator of Fig. 1. The wobble plate for the upper piston set is mounted on the shaft at a different angle than that for the lower piston set, causing the upper pistons to move with a different phase than the lower pistons. Proper gas displacement and phasing in each regenerator is caused by this phased motion of two pistons (instead of the single piston plus acoustic impedance in the pulse tube refrigerator).

This design has a serious shortcoming and a distinct advantage. The shortcoming is that all frictional heat generated in the upper wobble plate and piston set must be removed by the refrigerator. Assuming 50% mechanical efficiency as discussed above, we find that $3/4$ of the cooling power is required to service this nuisance head load, leaving almost no cooling power for the instruments. Some mechanical power is indeed fed back to the shaft by the cold wobble plate, but this reduces the total shaft power requirement by only 10%. The advantage is that the refrigerator would function equally well in any orientation, whereas the pulse tube design works best when oriented so that convective stability is maintained in the pulse tube itself.

We conclude that it would be extremely challenging to engineer sufficiently efficient mechanical components at the cold end without adopting an expensive "high-tech" approach. Hence, we believe the pulse tube refrigerator is a better choice.

For variety, we drew Fig. 2 with the alternator at 30°C instead of 300°C.

This is also an option in Fig. 1.

75 Kelvin in cool well

Our design for maintaining 75 Kelvin in a 50°C environment is illustrated in Fig. 3; the layout is identical to that of our 30°C — 300°C design, although there are many differences that don't show up in this drawing. We have designed for 8 W of cooling power, most of which will be devoted to heat leak; Reagor estimates that the heat load from his superconducting electronics will be much less than a Watt. Input pV power will be about 160 W, so the shaft power required will be about 320 W.

We believe the 75 Kelvin designs will require vacuum insulation, with some multi-layer radiation reflectors (known as "superinsulation" or "Multi-layer insulation (MLI) in cryogenic engineering).

75 Kelvin and 30 Centigrade in hot well

Figure 4 shows our design for a two-stage refrigerator, providing cooling at both 75 Kelvin (for superconducting components) and at 30°C (for a Cerro block and any conventional electronics). We have designed this one for 10 W of cooling at 75 Kelvin and 40 W at 30°C, rejecting waste heat at 250°C. After deducting for heat leak, we expect on the order of 1 W of cooling power at 75 Kelvin and 30 W at 30°C to be available for electronics. The refrigerator requires 500 W of pV power and hence about 1 kW of shaft power. Based on the 500 W input power, the design is "18% of Carnot efficiency," which is perfectly respectable for this type of refrigerator.

Removal of 500 W of frictional heat from the mechanical components may require deliberate use of the lubricating oil as coolant and heat-transfer medium.

We have struggled to keep this two-stage design inside a 3-inch overall package diameter, but it may have turned out to be so crowded that insulation is too challenging. Hence, a 4-inch package diameter might turn out to be more practical.

For those rare readers who are familiar with our thermoacoustics design code, DeltaE files⁴ for the two-stage design are attached at the end of the report.

Horizontal operation

In Figs. 1, 3, and 4, the pulse tubes *per se* rely in part on gravity for stability with respect to convection. We expect no significant degradation

in performance for operation up to 60° off vertical, but the available cooling power may be reduced by horizontal operation of these refrigerators. (The Stirling configuration of Fig. 2 doesn't have this problem.)

To our knowledge, the only experimental study of this issue was by Ron Ross and coworkers at JPL, reported at the 1995 pulse-tube workshop (no proceedings exist; copies of viewgraphs are in circulation). They measured convective heat transport in a pulse tube at a variety of angles with respect to gravity, under the following conditions: with the refrigerator turned off, with T_H held constant at 30°C, at various T_C 's $\gtrsim 35$ K. Convection was absent for $T_C > 200$ K. At lower temperatures, convection was greatest for horizontal orientation, carrying about 1 W of heat from 30°C to 75 K. I believe that their pulse tube was a little larger in diameter than the pulse tubes in our designs, and that their helium pressure was comparable.

To judge the implications for our work, we estimate that laminar, gravity-driven convective heat transport Q in a horizontal tube of length L and radius r is approximately proportional to

$$Q \propto \frac{\rho^2 c_p g r^5 (T_H - T_C)^2}{\mu L (T_H + T_C)}, \quad (1)$$

where ρ is the gas density, c_p is its isobaric heat capacity per unit mass, μ is its viscosity, and g is the acceleration of gravity. The strong dependence on radius comes from the viscous resistance to the flow ($\propto r^4$) and the gravitational driving head ($\propto r$). We expect less convection in our 75 K pulse tubes than was observed in the JPL work because our pulse tubes are smaller in radius. In our 300°C to 30°C design, we expect even less, because $\rho(T)$, $\mu(T)$, and $(T_H + T_C)$ also reduce Eq. 1. On the other hand, the JPL work was carried out with the refrigerator itself shut off; we might expect that they would have observed greater heat flow during operation, because the oscillatory gas motion is very effective in maintaining a small temperature difference between each end of the tube and the adjacent gas.

Based on these considerations, we have good reason to believe that our 300°C to 30°C refrigerator will work as designed, even in horizontal operation. Benchtop operation at various angles would be among the first tests we would make. If convection turned out to be more serious than we expect, it would be easy to fix by switching to smaller-radius pulse tubes. The other physical phenomena important in the pulse tube are much less sensitive to radius than r^5 , so there will be plenty of room for optimization.

The 75 K refrigerator is slightly riskier with respect to this issue, but we also expect that adjustment of pulse tube radius can reduce convection to acceptable levels.

Odd details, open questions

Spacers, not shown in the figures, will give mechanical support and thermal isolation between the pressure case and cold parts inside it, so the unit will be robust against lateral shocks.

The upper dome of the pressure case might be ceramic instead of steel, if electromagnetic radiation must pass through. The strong ceramics are stronger than steel in compression. Some are electrical insulators, but most have conductivities typical of semiconductors; in either case, kHz electromagnetic signals could pass through. It might also be possible to use the entire metal pressure case as an antenna, or mount an external antenna on the can.

We regard the shaft seal between the well and the oil-lubricated wobble-plate space as a significant challenge. Our current low-cost plan: We envision a series of two seals along the shaft. First, a seal would separate drilling mud from clean oil maintained at a slight overpressure relative to the mud. This first seal might be a ceramic face seal, similar to those used in the inexpensive clothes-washing machines in our homes. A spring-loaded bellows would maintain the slight overpressure; its volume would be sufficient to supply the slow leak of oil from this space into the well for the expected life of the refrigerator. Second, a hydraulic lip seal would separate this oil, at slightly above well pressure, from the oil in the wobble-plate space, at 40 bar. Variseal hydraulic rod seals are advertised to work up to 10,000 psi and 300°C. When running, the oil pump in the wobble-plate space would pump any leaked oil back out into the high-pressure oil space. Presumably the rod seal would leak very little when the shaft was stopped, so there would be no need for the oil pump during down time.

If this low-cost shaft-seal plan is for some reason impractical, there is a high-tech approach: magnetic coupling from the mud space into the wobble-plate space. The high-vac catalogs (e.g., Kurt Lesker, MDC, Huntington) show magnetic rotary feedthroughs, bakeable to 250°C, transmitting torque of order 1 N-m, selling for of order \$3k.

We have been working on improvements to pulse-tube-refrigerator technology under DOE/Fossil support for a few years. Our first two improvements are incorporated into these designs. These are the use of inertance

tubes at the upper ends of the pulse tubes to provide beneficial phase shifts to gas motion⁵ (this was invented simultaneously at several labs) and the use of slight taper (not shown in the figures) to suppress streaming-driven convection in the pulse tubes.⁶ These improvements, incorporated into the designs presented here, reduce the required shaft power by about 35% compared to earlier pulse-tube technology. In the next several years, with continued DOE/Fossil support, we expect to learn how to implement two additional efficiency improvements, each reducing the shaft power required by roughly 20%, and reducing package diameter by roughly 10%. We have not counted on these future improvements in the current design.

Acknowledgments

We thank Don Dreesen, Jim Blacic, and Dave Reagor, who are among our local potential users and downhole experts, for giving this work a preliminary reality check. They pointed out the great importance of horizontal operation, which we discussed above, and raised many of the issues discussed below.

Next steps

We should do a literature search, to learn the difference between wobble plate and swash plate mechanisms, and to learn standard practice in the torpedo and auto air-conditioner industries. Mechanical design details, including lubrication issues and bellows lifetime, are the riskiest aspects of this development, so we should focus initial attention on them.

Sizing of Cerro needs to be worked out; a two-hour hold time would be good. We might also consider some cryogenic thermal mass for the 75 K cooler, to provide several minutes of hold time (e.g., for gravity logging).

We should try a quick literature search to learn the quantitative realities of insulation (both multi-layer/vacuum and fiber/air).

We should be more careful to estimate temperature differences across copper or aluminum conduction paths and between the pressure-vessel case and the surrounding mud. Thus far, we've simply made order-of-magnitude estimates, showing that these differences are not prohibitively high. Case-to-mud heat transfer should be estimated both for vigorous mud flow (during drilling) and for slow mud flow (e.g. conventional well logging with the tool moving around 100 feet per minute).

The designs presented here are roughly $\pm 25\%$ accurate with respect to powers and cross-sectional areas. We should do a more detailed numerical design before beginning fabrication drawings.

JB and DD suggested that reduction in package diameter from 3 inches to under 2 inches would yield huge dividends, because then an array of coolers and sensors could be disposed around a 6-inch drill collar, leaving a hole in the center for mud on its way to the drill. The sensors would then be as close to the borehole wall as possible, and the coolers would be out where the return mud flow could carry away the waste heat. Such an array of sensors could detect azimuthal asymmetries in the well. We should keep this issue in mind at all times, looking for all opportunities to shave a little off the package diameter.

The one-stage 75 K cooler should be redesigned for $T_H = 125^\circ\text{C}$ instead of $T_H = 50^\circ\text{C}$, to match a greater percentage of real-world wells. This temperature span seems to be on the edge of what is efficient with one stage, so a two-stage design like that of Fig. 4 may be necessary. These options should be compared numerically before choosing between them.

Finally, we should build and test either of the one-stage units. For initial testing, we can omit the outer pressure case and shaft seal, and supply shaft power to the refrigerator with an electric motor.

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!Created@ 9:58: 5 26-Sep-97 with DeltaE Vers. 3.9b2 for the IBM/PC-Compatible

= start making this a 2 stage integral unit. =-

frequency= 40.000Hz mean pressure= 4.000E+06Pa

T(K)	p(Pa)	U(m ³ /s)	hdot(W)	wdot(W)
525.1	400000.0	0.0	0.00063	0.00059
125.29				125.29
ISODUCT dummy				
Heat extracted: 3.100E-09 Watts				
525.1	400000.0	0.0	0.00063	0.00059
125.29				125.29
SXFRST regen hot hx				
Heat = -115.532 (W) metal temp= 525.000 Kelvin				
525.1	397486.5	-2215.5	0.00062	0.00046
9.76				122.79
STKSCREEN HOTTER REG				
293.8	363723.8	-22099.3	0.00034	0.00011
9.76				60.86
TXMIDL middle T hx				
Heat exch wavvec =(0.284 , -3.806E-02) m ⁻¹				
Heat = 10.000 (W) metal temp= 300.000 Kelvin				
293.8	363723.8	-22099.3	0.00034	0.00010
19.76				60.58
HBRAN to 1st pt				
TTTTTTTTTTTTTTTTTT Branching into Tee Level= 1 TTTTTTTTTTTTTTTTTT				
293.8	363723.8	-22099.3	0.00010	-0.00002
16.54				19.14
STKDUCT another pulse tube				
513.7	363694.4	-22160.7	0.00010	-0.00008
16.54				18.98
HXLAST another hx				
Heat exch wavvec =(0.209 , -2.426E-02) m ⁻¹				
Heat = 2.368 (W) metal temp= 525.000 Kelvin				
513.7	363691.2	-22163.6	0.00010	-0.00008
18.91				18.91
ISODUCT inertance				
Re=0.41E+05, r/dn= 10.5, m= 4.1025, m-prime=0.8576 at start;				
Re=0.61E+05, r/dn= 10.5, m= 5.6269, m-prime=0.8467 at end of this segment.				
Heat extracted: 18.9 Watts				
513.7	0.0	0.0	0.00007	-0.00018
0.00				0.00
SOFTEND the end				
impedance (p A/rho a U)=(-4.476E-14, 4.674E-12)				
513.7	0.0	0.0	0.00007	-0.00018
0.00				0.00
TTTTTTTTTTTTTTTTTT Returning to Trunk Level= 0 TTTTTTTTTTTTTTTTTT				
293.8	363723.8	-22099.3	0.00023	0.00011
3.22				41.44
STKSCREEN regenerator				
75.0	281101.4	-51084.7	0.00006	0.00000
3.22				8.34
SXMIDL cold heat exch				
Heat = 2.500 (W) metal temp= 75.000 Kelvin				
75.0	280901.6	-51102.3	0.00006	0.00000
5.72				8.13
STKDUCT the pulse tube				
525.0	280822.4	-51316.3	0.00005	-0.00004
5.72				8.24
SXLAST p.t. hot h.x.				
Heat = 2.440 (W) metal temp= 525.000 Kelvin				
525.0	279950.4	-50551.1	0.00005	-0.00005
8.16				8.16
ISODUCT inertance				
Re=0.27E+05, r/dn= 8.1, m= 3.6821, m-prime=0.8223 at start;				
Re=0.38E+05, r/dn= 8.1, m= 4.7669, m-prime=0.8094 at end of this segment.				
Heat extracted: 8.16 Watts				
525.0	0.0	0.0	0.00003	-0.00009
0.00				0.00
SOFTEND the end				
impedance (p A/rho a U)=(-2.004E-12, 3.600E-11)				
525.0	0.0	0.0	0.00003	-0.00009
0.00				0.00

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----- 0 -----
BEGIN the setup
4.0000E+06 a Mean P Pa 525.08 A T-beg G( 0c) P
40.000 b Freq. Hz 8.6112E-04 B |U|a0 G( 0f) P
525.08 c T-beg K G 43.323 C Ph(U)0 G( 0g) P
4.0000E+05 d |p|a0 Pa -115.53 D HeatIn G( 2e) P
0.0000 e Ph(p)0 deg 3.4447E+09 E Re(Zb) G( 5a) P
8.6112E-04 f |U|a0 m^3/s G 2.8365E+08 F Im(Zb) G( 5b) P
43.323 g Ph(U)0 deg G 0.8370 G H frac G( 5c) P
helium Gas type 2.1314E-06 H Area G( 8a) P
ideal Solid type 4.2631 I Length G(14c) P
----- 1 -----
ISODUCT dummy
0.1000 a Area m^2 4.0000E+05 A |p| Pa
1.0000E-04 b Perim m -8.2707E-11 B Ph(p) deg
1.0000E-07 c Length m 8.6101E-04 C |U| m^3/s
0.0000 d Srough 43.316 D Ph(U) deg
125.29 E Hdot W
125.29 F Work W
-3.1004E-09 G HeatIn W
----- 2 -----
SXFRST regen hot hx
sameas 3a a Area m^2 3.9749E+05 A |p| Pa
0.6540 b VolPor -0.3193 B Ph(p) deg
2.0000E-02 c Length m 7.7337E-04 C |U| m^3/s
8.0000E-05 d r_H m 36.660 D Ph(U) deg
-115.53 e HeatIn W G 9.7588 E Hdot W
525.00 f Est-T K = 2H? 122.79 F Work W
-115.53 G Heat W
525.00 H MetalT K
copper Solid type
----- 3 -----
STKSCREEN HOTTER REG
3.9112E-04 a Area m^2 3.6439E+05 A |p| Pa
0.6700 b VolPor -3.4769 B Ph(p) deg
3.2691E-02 c Length m 3.6028E-04 C |U| m^3/s
1.6000E-05 d r_H m 18.526 D Ph(U) deg
0.2500 e KsFrac 9.7588 E Hdot W
60.861 F Work W
525.08 G T-beg K
293.76 H T-end K
-61.927 I StkWrk W
stainless Solid type
----- 4 -----
TXMIDL middle T hx
1.0000 a Area m^2 3.6439E+05 A |p| Pa
1.0000 b GasA/A -3.4769 B Ph(p) deg
1.0000E-06 c Length m 3.5332E-04 C |U| m^3/s
1.0000E-03 d radius m 16.295 D Ph(U) deg
10.000 e HeatIn W 19.759 E Hdot W
300.00 f Est-T K = 4H? 60.579 F Work W
10.000 G Heat W
300.00 H MetalT K
copper Solid type
----- 5 -----
HBRAN to 1st pt
3.4447E+09 a Re(Zb) Pa-s/m^3 G 3.6439E+05 A |p| Pa
2.8365E+08 b Im(Zb) Pa-s/m^3 G -3.4769 B Ph(p) deg
0.8370 c H frac G 1.0543E-04 C |U| m^3/s
-8.1842 D Ph(U) deg
16.539 E Hdot W
19.144 F Work W
41.435 G Work_T W
----- 6 -----
STKDUCT another pulse tube
9.4500E-05 a Area m^2 3.6437E+05 A |p| Pa
3.4461E-02 b Perim m (-2) -3.4868 B Ph(p) deg
4.6856E-02 c Length m 1.2732E-04 C |U| m^3/s
1.0000E-05 d WallA m^2 -38.588 D Ph(U) deg
16.539 E Hdot W
18.978 F Work W
293.76 G T-beg K
513.72 H T-end K
-0.1656 I StkWrk W
stainless Solid type
----- 7 -----
HXLAST another hx
sameas 6a a Area m^2 3.6437E+05 A |p| Pa
1.0000 b GasA/A -3.4873 B Ph(p) deg
3.0000E-03 c Length m 1.2952E-04 C |U| m^3/s
1.0000E-03 d y0 m -40.238 D Ph(U) deg
  
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0.0000	e HeatIn	W	(t)	18.907	E Hdot	W
525.00	f Est-T	K	= 7H?	18.907	F Work	W
sameas 0	Gas type			2.3680	G Heat	W
copper	Solid type			525.00	H MetalT	K
----- 8 -----						
ISODUCT inertance						
2.1314E-06	a Area	m ²	G	2.1224E-06	A p	Pa
1.6473E-03	b Perim	m	(-2)	20.206	B Ph(p)	deg
4.5605	c Length	m		1.9359E-04	C U	m ³ /s
1.0000E-04	d Srough			-70.343	D Ph(U)	deg
				-1.9671E-12	E Hdot	W
helium	Gas type			-1.9671E-12	F Work	W
ideal	Solid type			-18.907	G HeatIn	W
----- 9 -----						
SOFTEND the end						
0.0000	a Re(Z)		= 9G?	2.1224E-06	A p	Pa
0.0000	b Im(Z)		= 9H?	20.206	B Ph(p)	deg
				1.9359E-04	C U	m ³ /s
				-70.343	D Ph(U)	deg
				-1.9671E-12	E Hdot	W
				-1.9671E-12	F Work	W
				-4.4757E-14	G Re(Z)	
sameas 0	Gas type			4.6741E-12	H Im(Z)	
ideal	Solid type			513.72	I T	K
----- 10 -----						
STKSCREEN regenerator						
1.0945E-04	a Area	m ²		2.8571E+05	A p	Pa
0.6700	b VolPor			-10.300	B Ph(p)	deg
4.9156E-02	c Length	m		5.9286E-05	C U	m ³ /s
1.2000E-05	d r_H	m		-0.2204	D Ph(U)	deg
0.2500	e KsFrac			3.2199	E Hdot	W
				8.3384	F Work	W
				293.76	G T-beg	K
sameas 0	Gas type			74.960	H T-end	K
stainless	Solid type			-33.097	I StkWrk	W
----- 11 -----						
SXMIDL cold heat exch						
sameas 10a	a Area	m ²		2.8551E+05	A p	Pa
sameas 2b	b VolPor			-10.311	B Ph(p)	deg
4.0000E-03	c Length	m		5.7299E-05	C U	m ³ /s
sameas 2d	d r_H	m		-4.1266	D Ph(U)	deg
2.5000	e HeatIn	W		5.7199	E Hdot	W
75.000	f Est-T	K	=11H?	8.1322	F Work	W
sameas 0	Gas type			2.5000	G Heat	W
copper	Solid type			75.000	H MetalT	K
----- 12 -----						
STKDUCT the pulse tube						
4.1288E-05	a Area	m ²		2.8547E+05	A p	Pa
2.2778E-02	b Perim	m	(-2)	-10.356	B Ph(p)	deg
8.0625E-02	c Length	m		6.6981E-05	C U	m ³ /s
1.0000E-05	d WallA	m ²		-40.867	D Ph(U)	deg
				5.7199	E Hdot	W
				8.2368	F Work	W
				74.960	G T-beg	K
sameas 0	Gas type			524.97	H T-end	K
stainless	Solid type			0.1046	I StkWrk	W
----- 13 -----						
SXLAST p.t. hot h.x.						
sameas 12a	a Area	m ²		2.8448E+05	A p	Pa
sameas 11b	b VolPor			-10.236	B Ph(p)	deg
1.0000E-02	c Length	m		6.9287E-05	C U	m ³ /s
sameas 11d	d r_H	m		-44.343	D Ph(U)	deg
1.5760	e HeatIn	W	(t)	8.1601	E Hdot	W
sameas 2f	f Est-T	K	=13H?	8.1601	F Work	W
sameas 0	Gas type			2.4402	G Heat	W
copper	Solid type			525.00	H MetalT	K
----- 14 -----						
ISODUCT inertance						
1.3175E-06	a Area	m ²		1.3102E-05	A p	Pa
1.2951E-03	b Perim	m	(-2)	20.794	B Ph(p)	deg
4.2631	c Length	m	G	9.6808E-05	C U	m ³ /s
1.0000E-04	d Srough			-72.391	D Ph(U)	deg
				-3.5239E-11	E Hdot	W
sameas 0	Gas type			-3.5239E-11	F Work	W
ideal	Solid type			-8.1601	G HeatIn	W
----- 15 -----						
SOFTEND the end						
0.0000	a Re(Z)		=15G?	1.3102E-05	A p	Pa
0.0000	b Im(Z)		=15H?	20.794	B Ph(p)	deg
				9.6808E-05	C U	m ³ /s

sameas	0	Gas type
ideal		Solid type

[illegible]

I. Figure Captions

1. Scale drawing of pulse-tube cooler to maintain 30°C in a 300°C well. The design for 75 Kelvin in a 50°C well looks virtually identical.

2. Scale drawing of Stirling cooler to maintain 30°C in a 300°C well.

3. Scale drawing of pulse-tube cooler to maintain 75 Kelvin in a 50°C well.

4. Scale drawing of two-stage pulse-tube cooler to maintain platforms at both 75 Kelvin and 30°C in a 250°C well.

Fig. 1

Fig. 2

Fig. 3

Fig. 4