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PROPOSED SOLUTIONS OF FOUR REFRIGERATION PROBLEMS RELATING TO SUPERCONDUCTING ACCELERATORS AND CRYOGENIC EXPERIMENTAL EQUIPMENT

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Problem One - Operation at 1.85°K

The Stanford Linear Accelerator (SLAC) at 360 pps @ 1.5μs can operate at 22 Gev with a 0.0005 duty cycle. If converted to superconducting operation at or below 1.85°K the duty cycle at 20 Gev would be 1.0 and 0.06 at 100 Gev which would be useful to high energy particle physics experimenters. The conversion would require new klystrons, a new disk-loaded wave guide of superconducting material to impart energy into the electron beam and a number of large refrigerators of novel design. Existing utilities would be adequate if 16 or 32 refrigerators were used. The use of 16 units is feasible. Fewer units are rejected because 10,000 feet of cold He_4 transfer lines would be needed and size of vacuum tanks, heat exchangers and helium vapor recovery pumps would be excessive. To reach 1.85°K where the He_4 vapor pressure is 0.0195 atm requires vacuum pumping over the liquid surface. Pumping power is 2,503xMol/sx $\ln P_2/P_1$ plus friction HP plus heating of gas . Vacuum pumps should be of the hermetically sealed, internally oil mist cooled, rotary, helical lobed type with compression ratios of up to 17, piston operated slide gates for 100% load modulation and 99.9999% or better oil separators and separate hermetically sealed lube oil pumps. Such units make it possible to use only 3 or 4 stages of compression. The first stage would require 3 units operating in parallel each handling 14,000 CFM. Each refrigerator would provide up to 930 W refrigeration at 1.85°K. Full flow filters would be provided at 302.6°K, 43.5°K and 10.0°K to remove impurities. Gas return lines would have automatic spring loaded soft seat check valves to prevent back streaming of oil. A full flow intermediate heat shield would intercept 1141 W heat leakage from ambient. It would consist of aluminum plates and tubing for helium gas backed by 70 wraps of perforated aluminized mylar. Its maximum average temperature would be 44.7°K when heat leakage to the cold end is estimated to be 45 W. Expanders would be the single stage reciprocating type to insure maximum efficiency. Heat exchangers would be the multipass aluminum platefin type arranged for counterflow service to insure compactness, economy and maximum effectiveness. Reynolds number would be held to 2000 or less to insure laminar flow and tolerable friction losses in vapor passes. Hot supply gas would be liquefied with 100% yield at the suction pressure of the top compressor stage and then subcooled to 2°K. This will assure yields of 95% or better at the final expansion to 0.0195 atm. Pressure reducing valves would be standard types. A number of cycles using 3 or 4 stages of compressors and 2 or 3 expanders and various intermediate and hot gas pressures were studied. Selected temperature differences were 0.150°K at the cold end of the subcooler, 0.100°K minimum inside the critical heat exchanger and 1.5°K at the warm end. Heat exchanger effectiveness would be 91% in the subcooler and 99% at the warm end. Core lengths would be 7'-6" maximum to allow use of 8' dia. by 12' high vacuum tanks. Minimum input KW can be attained when hot gas pressure is close to 4 atm. At lower pressure total flow increases rapidly. At higher pressure total flow falls off but input KW increases. The most promising arrangement has 3 stages of compression and 3 expanders with pressures balanced to obtain reasonably low volume ratios through expanders. Expander efficiencies are taken as 75%$\pm$ 1 for the warm end expander where irreversible heat flow effects would be encountered and 80%$\pm$ 1 for cold end expanders where heat capacity of piston walls is low. Pressures of 4.0 atm, 0.9 atm, 0.2 atm, . and 0.013 atm results in 10.5 Mol/s vacuum pump flow, 7.4 Mol/s cold end expander flows, 4.0 Mol/s warm end expander flow and 21.9 Mol/s compressor flow. Brake KW would be 288.0 for vacuum pumps, 280.8 for compressors, 7.5 for oil pumps or 576.3 total per refrigerator. Brake W/W performance would be 620 and input KW would be 648. Electrical demand at 110% load would be 10,373 KW for 16 refrigerators estimated to cost $8,923,000. Increasing the warm end temperature difference from 1.5°K to 5.0°K would increase compressor flow by 25% and electrical demand 1.2 MW. The additional power could cost more than 16 complete sets of heat exchanger cores based on 1.5°K warm end differential. Having the cold end expander discharge at 4.0°K results in good temperature differences across cold end heat exchangers. See Table I and Figures I and 2.

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Problem Two - A Separate Heat Shield

When operation of SLAC at 1.85°K was first studied it was estimated that heat leakage from 300°K to 70°K would be 1100 W and from 70°K to 1.85°K would be 76 W. If the heat shield temperature is lowered below 70°K radiant heat leakage to the shield holds "constant" while conduction leakage increases. If a heat shield has average temperature of 9°K heat leakage from 300°K increases to 1280 W but leakage from the shield to 1.85°K reduces to 8 W. The economics for operation of an electron linear accelerator at 1.85°K make it impractical to consider a separate heat shield but this is not true for other applications at higher temperatures or for accelerators at 1.425°K or below. A single stage compressor is adequate for a 9°K heat shield. Using two stage compressors increases both electrical demand and initial cost while leakage only drops to 5 W. One KW over 10 years at 2/3 load would cost $250 at our present rate. A 2 MW reduction between alternate approaches could justify an additional $30,000 per refrigerator. If a single stage compressor were used to obtain temperatures of less than 9°K total gas flow and compressor power increase rapidly. A refrigerator rated for 1280 W service at 9.0°K has a very simple cycle. The compressor would take suction at 1 atm and discharge hot gas at 8 atm. Hot gas would pass through a full flow filter, enter a single heat exchanger and be cooled to 14.3°, pass through a second full flow filter, and enter a single expander emerging at 7.0°K. It would then pass through the heat shield, emerge at 11.1°K reenter the heat exchanger and return to the compressor via an automatic check valve. Expander efficiency is taken as 85% ± 1. One could lower the supply and return gas pressures and improve performance but the compressor size increases rapidly. The cycle described is close to optimum and requires 102.2 KW for the compressor, 1.5 KW for the lube oil pump or a total of 103.7 KW per unit. Brake W/W performance would be 81 and input KW would be 117. Electrical demand would be 1872 KW for 16 units estimated to cost $2,068,000. System components would be similar to those described under problem one. Flow per unit would be 14.5 Mol/s or 760 CFM at 1.0 atm. Less than 16 compressors would be used to reduce cost about 30%. See Fig. 3.

Problem Three - Refrigeration for Cryogenic Equipment

This applies to large pieces of electrical equipment to be operated at 4.224°K when electrical losses are small. A large dewar containing a cryogenic magnet might contain 9,000L of LHe_4 at 1.0 atm. The heat shield load at 9.0°K is estimated at 128 W and the magnet electrical losses at 12 W. The LHe_4 at $2/L would be $36,000. A refrigerator is justified even if service is intermittent. A single stage compressor could raise 301.1°K, 1.0 atm return gas to 302.6°K and 9.0 atm at the outlet of the aftercooler. These temperatures are based on 75°F cooling water, a 10°F approach and warm end temperature difference of 1.5°K as is true for all cycles described. Supply gas would be filtered, cooled to 14.3°K in a single warm end heat exchanger, filtered again and flow would then split with 1.75 Mol/s entering a single expansion engine to emerge at 1.0 atm and 7.0°K and 0.16 Mol/s entering a small cold end refrigeration process. The expander flow would be split in turn using manual flow trimming valves to furnish 0.30 Mol/s to thermodynamically balance the warmest of the small cold end heat exchangers and 1.45 Mol/s through the heat shield. These flows would recombine at 11.0°K and reenter the large warm end heat exchanger and return to the compressor suction via an automatic check valve. A compensating dummy load would be used to prevent rise of liquid level within the dewar and obviate complicated controllers. The 12 W refrigerator and 128 W heat shield combination is estimated to cost $44,000. Brake KW would be 13.8 for the compressor, 1.5 for the lube oil pump and 15.3 total. Input KW would be 17.2, brake W/W performance would be 1275 and compressor suction flow would be 1.91 Mol/s or 99.8 CFM. Liquid yield at 4.224°K is 91.3%. System components would be similar to those described under problem one. See Figs. 4 and 5.

Problem Four - Operation at 1.0°K to 1.425°K

According to accelerator planners the RF losses for a superconducting SLAC might be 12000 W at 1.85°K, 825 W at 1.425°K and 80 W at 1.0°K. It will be shown that refrigerators for 1.0 to 1.425°K can cost less than units for 1.85°K service because of far lower RF losses. The need to investigate steady state refrigeration at very low temperatures is real. If we had to use only He_4 it might be feasible down to 1.425°K. At 1.425°K input KW for all He_4 refrigerators and hybrid He_3-He_4 units is much the same. At 1.425°K the vapor pressure of He_4 is only 0.002 atm. If we should use 4 stages of compression the intermediate pressure might be 0.1 atm and the first stage suction CFM would be less than for 1.85°K operation because of the far lesser refrigeration load. Fortunately, we should not have to use He_4 but should be able to use He_3 also. We cannot use He_3 in the heat shield or accelerator dewars. He_3 is never a super fluid and heat removal might be inadequate. Also He_3 has a 5000 barn cross section to thermalized neutrons. Thus we would have to locate LHe_4-II to boiling He_3 heat exchangers away from the accelerator behind earth radiation shielding. He_3 refrigerators used together with separate 9.0°K heat shields should be quite practical. Although our information on He_3 is less reliable than for He_4, the principles are the same. Also the flows are so small that efficiency is not important and only one expander would be used. As of now it is not certain that operation at 1.0°K could be justified due to lowered RF losses. Accordingly the best bet would be 16 units rated for 75 W, 1.325°K service. The temperature difference across the He_3 boiler is taken as 0.1°K. Three stages of compression would be adequate. Gas pressures would be 4.0 atm, 0.725 atm, 0.135 atm and 0.025 atm. Initial liquefaction with 100% yield would take place at 0.725 atm, the expander would discharge at 5.28°K and 0.135 atm, and vacuum pump suction would be about 0.025 atm for 1.325°K operation. Expander efficiency is taken as 78% ± 1. Yield at cold end would be 96.5%. It would be easy to operate the He_3 refrigerator at lower temperatures. At 75 W of refrigeration compression ratios are about 6.0 per stage. If the ratio is increased to 8 per stage, the first stage pressure would fall to 0.0078 atm and the pressure over boiling liquid helium three might be 0.012 atm. Thus He_3 temperature would be 1.0°K and that of LHe_4II in accelerator dewars 1.1°K. Provision would be made to use spare heat shield compressors to pump over the liquid He_4 to reduce cool down from days to hours. Brake KW would be 71.2 for He_3 compression, 102.2 for He_4 compression, 6.0 for lube oil pumping and 179.4 total per refrigerator. Brake W/W performance would be 2,392 and input KW would be 201.8. Electrical demand at 110% load would be 3229 KW for 16 refrigerators estimated to cost $3,942,000. First stage He_3 suction CFM would be 3635. The addition of a fourth stage of He_3 compression would allow operation of accelerator dewars at 1.0° K or less but this does not appear to be justified. See Figs. 6 and 7.

Conclusions and Notes

Refrigeration heat shields would be evacuated to 10^{-7} Torr using existing sputter-ion type vacuum pumps. A check on the Kapitza effect at the refrigerant-metal interfaces revealed that it is a few thousandths of 1.0°K at 1.0°K due to the necessary surface area of cold end heat exchangers. To achieve maximum LHe_4 yields at 1.85°K it would be necessary to subcool below the lambda point and exact performance is indeterminate in the absence of a working model but would not be much different from calculated values. Accelerator operating loads could vary from 0 to 100% while heat shield loads would be very steady. The cycles described can follow the wide load variation solely by varying flow through vacuum pumps and compressors. Temperature difference across the heat shield would increase from less than 3°K at 100% accelerator load to over 15°K at 0% accelerator load. Cold end temperatures would remain essentially constant during load variation. It is concluded despite a myriad of possible cycle detail each of the problems discussed has its own rather unique solution for optimum simplicity, reliability and least cost. Reducing the number of stages of compression coupled with retaining reasonable expander efficiencies resulted in selection of 4 atm high pressure and an estimated brake W/W performance of less than 620 at 1.85°K. The problem of a separate heat shield rushes one to the use of a single stage compressor and very low temperatures. To add a small refrigeration process at the cold end of such a system where electrical losses are small in comparison to heat shield load is easy and obvious. Finally, to include within the quite cold world of 9.0° a He_3 refrigerator for service at 1.325°K or below is also simple and clear.

Acknowledgement and References

The efforts of K. G. Carney, Jr. in reducing these solutions to computer programs is highly appreciated. These confirm that cycles described are subject to minor refinement and only new approaches could allow further improvement. By August 1971 it should be clear whether we can do better. Some aspects of the accelerator refrigeration cycles were discussed in "Proposed Refrigeration

Cycles for Superconducting Accelerators at 1.85°K, 1.425°K and 1.0°K" by the author for inclusion in the IEEE proceedings of the 1971 National Particle Accelerator Conference. He_4 data is taken or interpolated from "Provisional Thermodynamic Functions for Helium 4 for Temperatures from 2 to 1500 K with pressures to 100 MN/m^2 (1000 Atmospheres)" by R. D. McCarty of NBS, Boulder Laboratories, Boulder, Colorado except the value of 3.99J/Mol liquid enthalpy of He_4 at 1.85°K was given verbally by R. D. McCarty and the value of 96.484J/Mol gas enthalpy at 1.85°K and 0.0195 atm was derived by the author. He_3 data is taken from AFML Report TR-87-175 "Thermodynamic Data of Helium-3" by R. M. Gibbons and D. I. Nathan or interpolated and extrapolated by the author. While specific points of state for helium three are subject to doubt it is not likely that the general conclusions reached are totally in error. Cost estimates are based on 6000 x No. of units x (Input KW per unit)$^{0.7}$ as published by T. R. Strobridge in the IEEE proceedings of the 1969 National Particle Accelerator Conference.

<u>Table I - Data for Helium Four Refrigerators</u>

Some data is given for 3 or 4 stages of compression, 2 or 3 expanders and hot gas supply pressure to expanders over the range 2.5 to 6.0 atmospheres. It would appear that the 930 W Refrigerator arrangement is close to optimum for 1.85°K service.

Evaporator rating, W.	916.0	918.0	922.0	925.0	928.0	930.0
Compressor disch. pressure, atm	4.0	5.5	4.5	6.5	7.5	5.5
Nominal Heat Shield Pressure	3.0	4.5	3.5	5.5	6.5	4.5
Expander inlet pressure, atm	2.5	4.0	3.0	5.0	6.0	4.0
Initial liquefaction press., atm	0.7	0.9	0.8	1.0	1.0	0.9
Expander outlet pressure, atm	0.2	0.2	0.2	0.2	0.3	0.2
Total Compression Stages	3.0	3.0	3.0	3.0	4.0	3.0
First Stage Vacuum Pump Casings	3.0	3.0	3.0	3.0	3.0	3.0
Total Casings	6.0	5.0	5.0	5.0	6.0	5.0
Expander #3 Efficiency, %	77.3	76.4	78.4	70.5	70.7	75.4
Expander #2 Efficiency, %	None	None	None	74.3	79.8	80.6
Expander #1 Efficiency, %	83.2	80.4	83.1	74.8	81.6	80.4
Average Heat Shield °K	32.0	34.8	38.9	41.3	42.4	44.7
Heat Shield Load, W	1185.8	1174.4	1161.0	1154.0	1148.0	1141.2
Expander #1 Discharge °K	4.0	4.0	4.0	4.0	4.5	4.0
Base Refrigeration Load, W.	805.0	805.0	805.0	805.0	805.0	805.0
Heat Shield Leakage, W	31.0	33.0	37.0	40.0	43.0	45.0
Spare Capacity, W	80.0	80.0	80.0	80.0	80.0	80.0
Expander #3 Mol/s	6.0	5.4	4.8	4.7	4.4	3.9
Expander #2 Mol/s	0.0	0.0	0.0	1.4	1.4	1.0
Expander #1 Mol/s	11.9	7.9	9.4	5.6	4.8	6.4
Total Engine Mol/s	17.9	13.3	14.2	11.7	10.6	11.3
Vacuum Pump Mol/s	10.4	10.4	10.4	10.5	10.5	10.5
Compressor Mol/s	28.3	23.7	24.6	22.2	21.1	21.8
Vacuum Pump Brake KW	285.7	285.7	285.7	288.0	352.1	288.0
Compressor Brake KW	341.5	303.0	301.3	293.3	257.1	280.8
Oil Pump KW	7.5	7.5	7.5	7.5	9.0	7.5
Total Brake KW-1 unit	634.7	596.2	594.5	581.3	618.2	576.3
Brake W/W Refrigeration	692.9	649.5	644.8	636.5	666.2	619.7
Input KW-1 unit	714.0	670.7	668.8	662.5	695.5	648.3
Input KW-16 units	11424.0	10731.2	10700.8	10598.4	11128.0	10373.4
Estimated $ ÷ 1000 - 16 units	9547.0	9138.0	9120.0	9058.0	9373.0	8923.0
First Stage Section MCFM	41.0	41.0	41.0	41.3	41.3	41.3

TERMS USED IN THE FIGURES

- ACV Automatic Check Valve
- EXP Reciprocating Expansion Engine
- FTV Flow Trimming Valve
- GIC Gas Intercooler
- GAC Gas Aftercooler
- HEX Heat Exchanger
- JTV Throttling Valve
- PUR Full Flow Purifier
- RVP Rotary Vacuum Pump
- RCU Rotary Compressor Unit

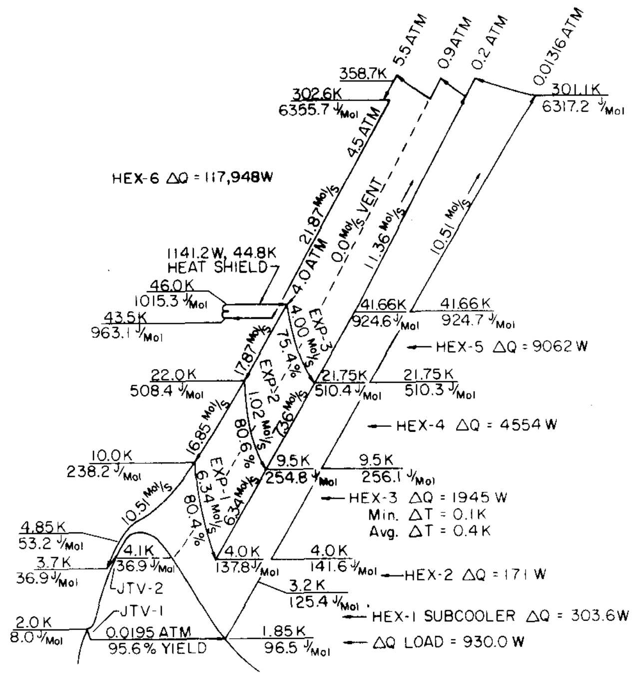

Fig. I T-S DIAGRAM FOR A 930W He_4 REFRIGERATOR FOR 1.85° K SERVICE

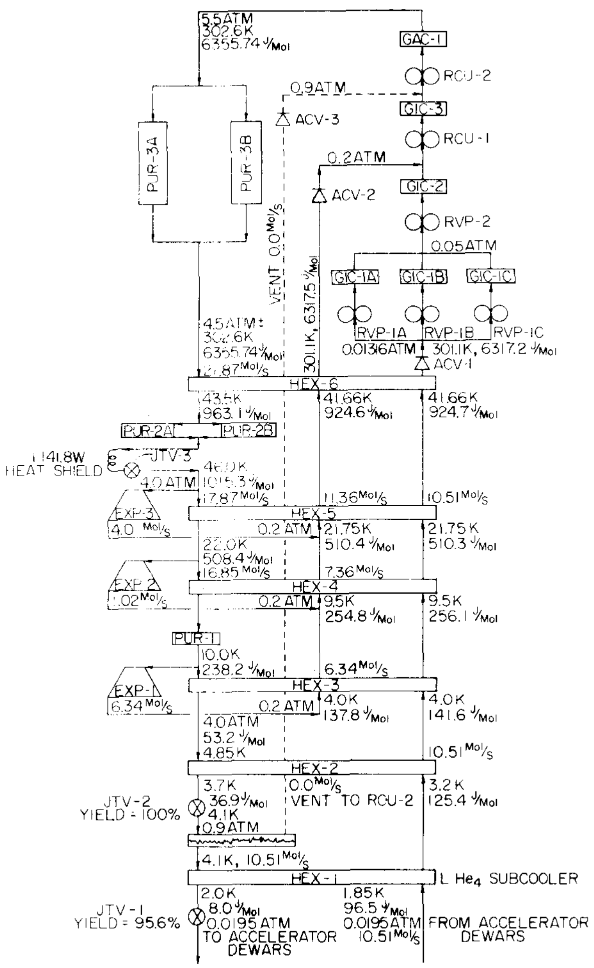

Fig. 2 SCHEMATIC DIAGRAM FOR A 930W He_4 REFRIGERATOR FOR 1.85°K SERVICE

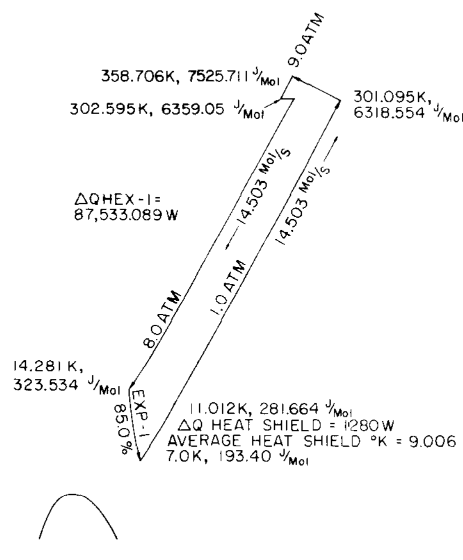

Fig. 3 T-S DIAGRAM OF A 1280W He 4 REFRIGERATOR FOR 9.006 °K SERVICE

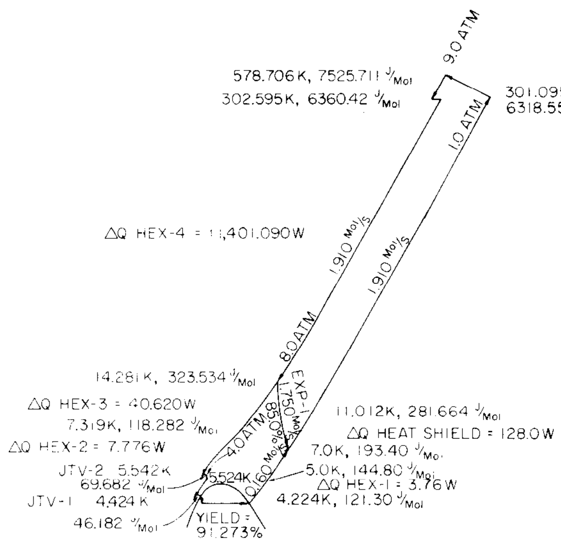

Fig. 4 T-S DIAGRAM OF A 12.0W REFRIGERATOR FOR 4.224°K SERVICE

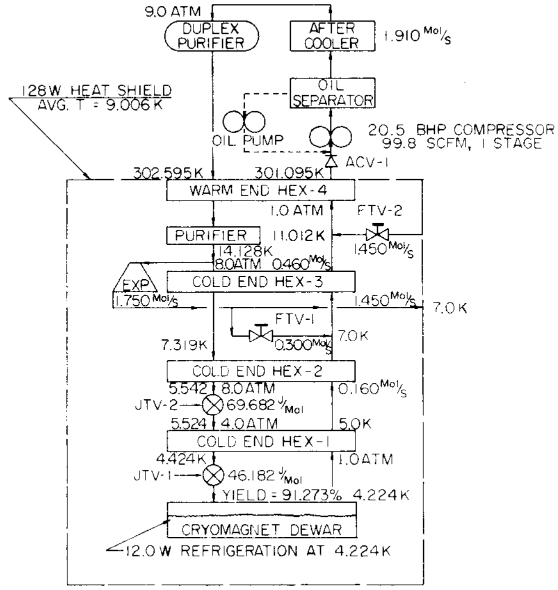

Fig. 5 SCHEMATIC DIAGRAM OF A 12.0W REFRIGERATOR FOR 4.224°K SERVICE.

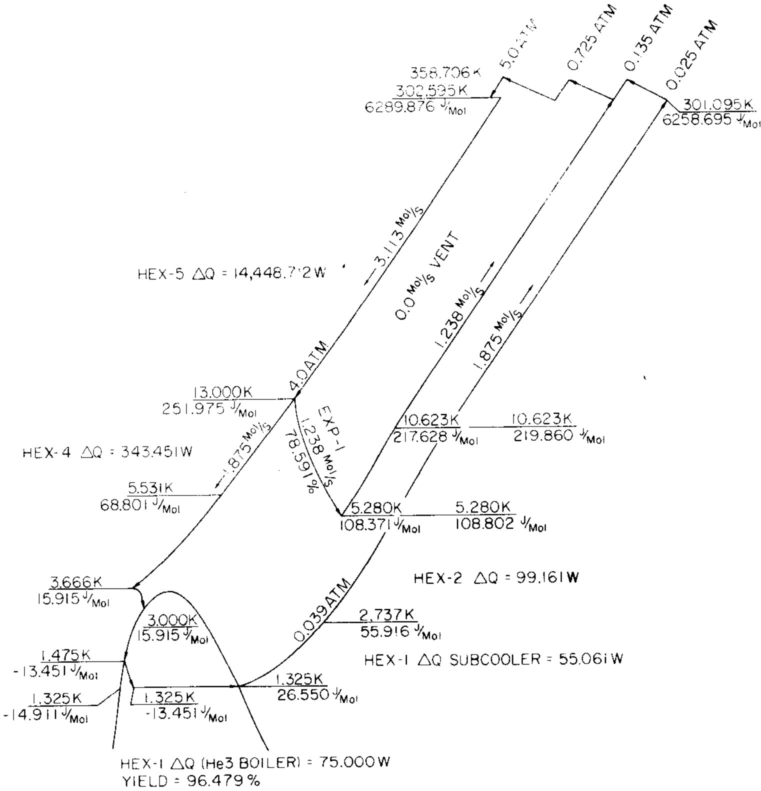

Fig. 6 T-S DIAGRAM FOR A 75.0W He3 REFRIGERATOR TO OPERATE AT 1.325°K AND MAINTAIN SUPERFLUID He4 - II AT 1.425°K. SEPARATE HEAT SHIELD IS SHOWN IN Fig. 7.

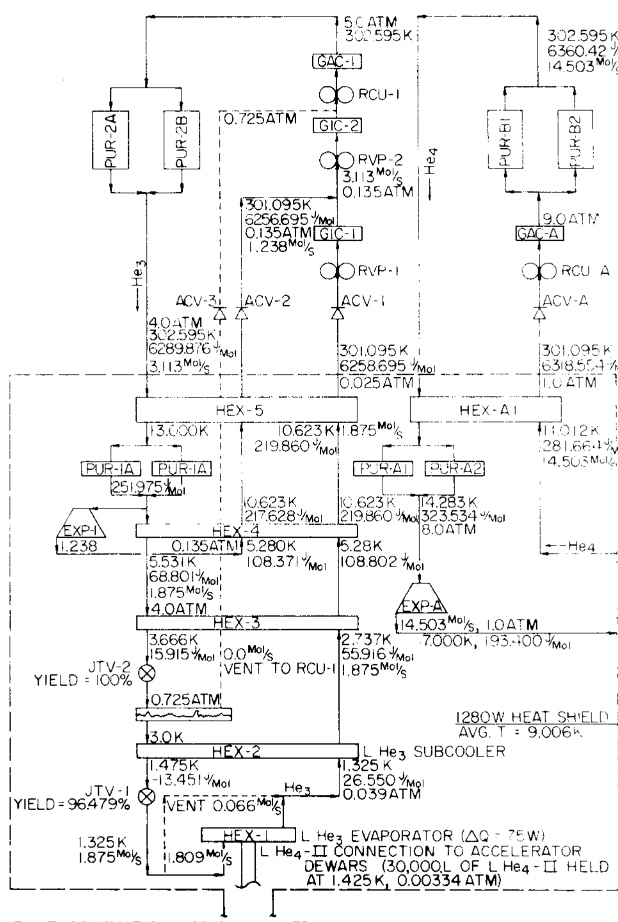

Fig. 7 SCHEMATIC DIAGRAM FOR A 75W He_3 REFRIGERATOR FOR 1.325°K SERVICE TOGETHER WITH A SEPARATE He_4 HEAT SHIELD REFRIGERATOR