SSC-N-100
CCI Report No. 593-103

DUAL PIPE COOLING
SYSTEM FOR SSC MAGNETS

PREPARED UNDER P.O. NO. 94362 BY
CRYOGENIC CONSULTANTS, INC.
ALLENTOWN, PA

FOR

FERMI NATIONAL ACCELERATOR LABORATORY
BATAVIA, IL

February 9, 1984

INDEX

LIST OF TABLES

LIST OF FIGURES

DUAL PIPE COOLING SYSTEM FOR SSC MAGNETS

INTRODUCTION:

The two-phase cooling system of the Tevatron magnets has performed well and seems to be fairly predictable in performance. However, the system suffers from some problems which may become serious when much longer distances between refrigeration stations are contemplated. These problems are the following:

a) Stability of the system, mainly under transient conditions.

b) Control loop problems.

c) Large variations in inventory under transient conditions.

d) High pressure drop.

This report describes a modified two-phase cooling system which appears to offer advantages over the two-phase system as presently used in the Tevatron system.

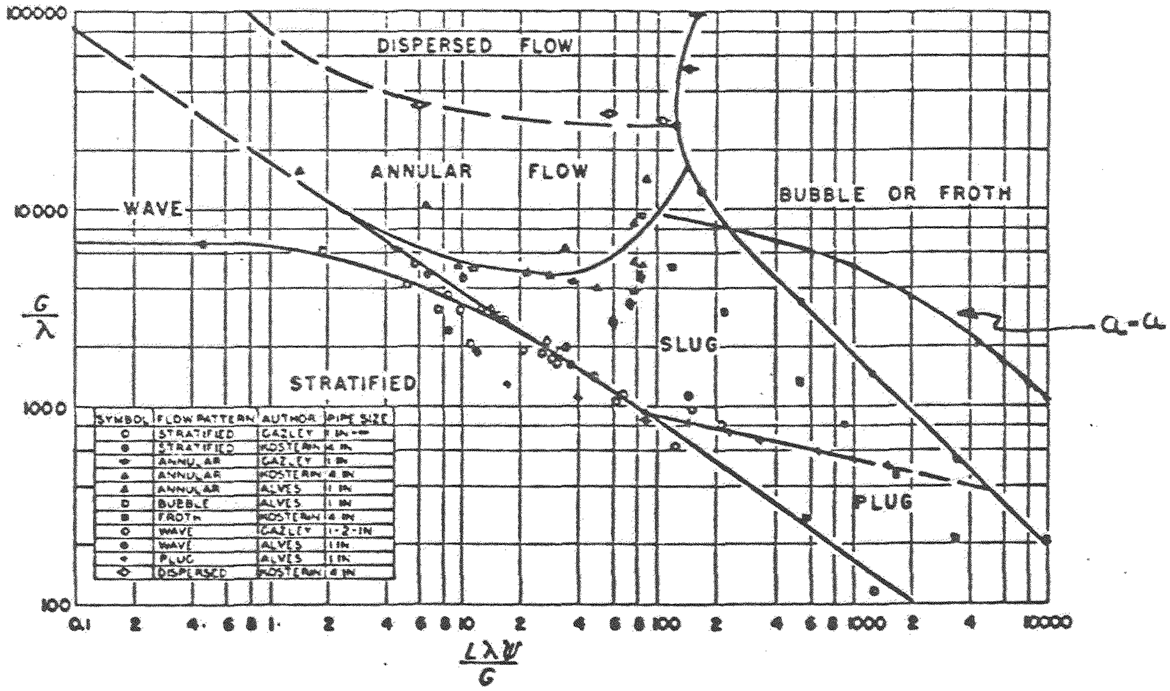
DISCUSSION:

1. Conventional Two-Phase Cooling System:

The conventional two-phase cooling system carries two-phase fluid through a pipe or jacket on the magnets. Heat transfer between single-phase fluid in the magnets and the two-phase fluid may be carried out in separate vessels, the pipe carrying the two-phase fluid, or the magnet.

1.1 Type of Flow:

Two-phase flow system behavior has been analyzed by many investigators (Ref. 1-6) because of the general interest in pipeline systems carrying liquid and gas at the same time. The so-called Baker correlation (Ref. 2) attempts to predict the type of flow as a function of liquid and gas properties and mass flow rates. Figure 1 represents this correlation:



FLOW-PATTERN REGIONS in two-phase flow may be defined in terms of mass velocity of gas phase and ratio of liquid and gas mass velocities.

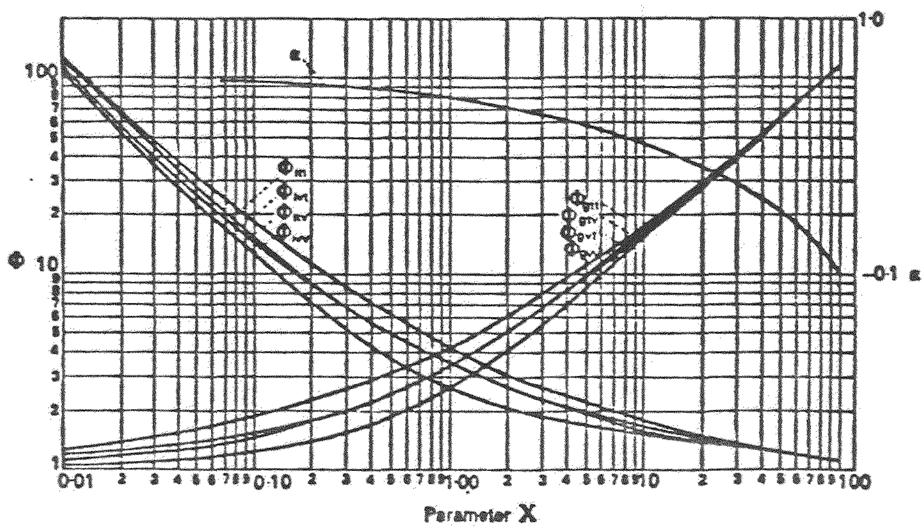
FIGURE 1

The Tevatron two-phase system operates along line a-a. At high liquid fraction the L/G is large and G is small. When liquid is vaporized in moving along the magnets on the way back to the refrigerator, the operating points move along the line until at some point (roughly 10-15% liquid) the flow regime changes from bubble or froth to slug. The location of the line relative to the line separating bubble or froth flow regime from plug or slug flow is determined by the pressure drop allowed for the system. Higher pressure drop will move the operating line to the right; lower pressure drops to the left.

In the case of the SSC it certainly will be important to lengthen the distance between refrigerators. This means a lower allowable pressure drop per unit length. Since total allowable pressure drop tends to be fixed because of limitation of the maximum boiling point, the operating line will move to the left. As a consequence, the plug flow regime will be entered at a higher liquid fraction. At this time it is not clear whether this is detrimental for proper operation of the system. Some pressure fluctuations may occur, but because of the low density of liquid helium, this may not be very important.

1.2 Pressure Drop:

The Lockhart-Martinelli correlation (Ref. 5) has been used to determine the required flow passage area for the two-phase flow. Figure 2 shows the graph used to determine the increase in pressure drop due to two-phase flow:



Correlation of Lockhart and Martinelli (1949) for frictional pressure drop.

FIGURE 2

The parameters are as follows:

$$\phi = \left[\frac{\left(\frac{\Delta P}{L}\right)_{TP}}{\left(\frac{\Delta P}{L}\right)_{1}}\right]^{1/2}$$

$$X = \left(\frac{R_{eg}}{Re_1}\right)^{0.2} \left(\frac{C_1}{C_g}\right) \left(\frac{L}{G}\right)^2 \left(\frac{\rho_g}{\rho_1}\right)$$

Where: $\left(\frac{\Delta P}{L}\right)_{TP}$ = Pressure drop per unit length for two-phase flow.

$\left(\frac{\Delta P}{L}\right)_{1}$ = Pressure drop per unit length for liquid flow only.

Re_g = Reynold's Number for gas only.

Re_1 = Reynold's Number for liquid only.

$\left(\frac{C_1}{C_g}\right)$ = 1.0 when both gas and liquid flow regimes are turbulent.

(L/G) = Mass ratio of liquid to gas.

ρ_g ρ_1 = Density of gas and liquid, respectively.

Figure 3 shows the results of applying the data of Figure 2 to a two-phase system of liquid and gaseous helium. Figure 3 shows that the pressure drop of a two-phase fluid containing 10% vapor is five times greater than when the total mass flows as liquid through the line. On average, the pressure drop is ten times as high for two-phase flow as that for 100% liquid flow.

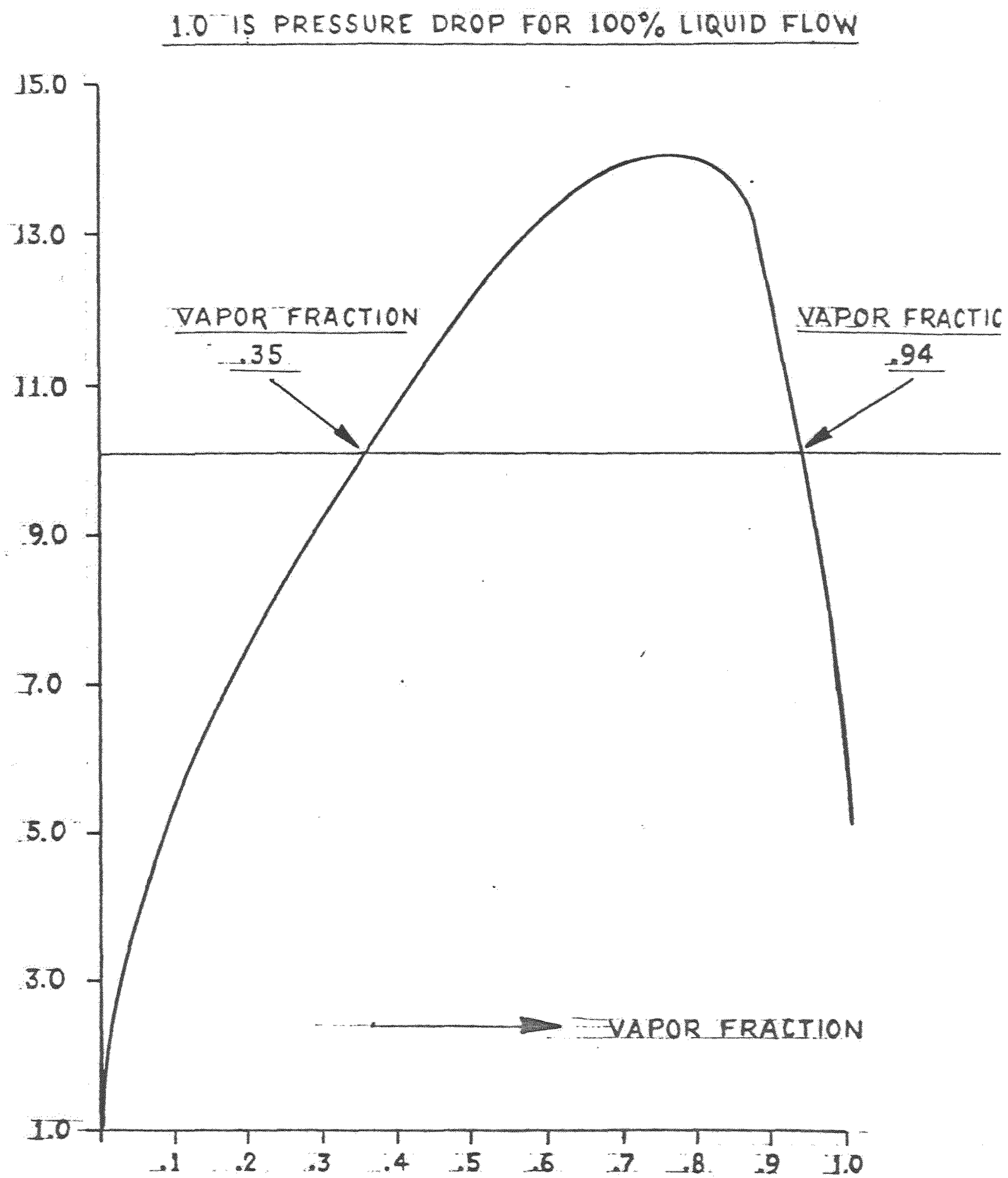

1.0 IS PRESSURE DROP FOR 100% LIQUID FLOW
15.0
13.0
11.0
9.0
7.0
5.0
3.0
1.0
VAPOR FRACTION
.35
VAPOR FRACTIC
.94
VAPOR FRACTION
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0

FIGURE 3

1.3 Inventory:

It seems that inventory of the two-phase system is not known accurately. Inventory is probably some function of flow regime. For instance, bubble flow with small amounts of vapor suspended in the liquid has a velocity determined by the liquid component. Vapor bubbles are dragged along. On the other hand, froth flow with small liquid droplets suspended probably has a somewhat higher vapor velocity than liquid velocity. Liquid droplets are dragged along by the vapor. If the droplets are very small, the difference in velocity between liquid and gas is probably slight. In slug or plug flow, vapor and liquid velocities are the same. In stratified flow, liquid velocity will be lower since driving force for a horizontal thin layer of liquid is only provided by vapor flow friction.

Variation of heat load with intermittent cooling results in substantial changes of inventory. If pressure in the single-phase system is held constant, a heating of the liquid will increase flow of the liquid through the J-T valve into the two-phase system. If the single phase inventory is rather large, ejection of 5% of the single-phase fluid in a 60 second ramp will add liquid helium to the two-phase system at a rate one-third higher than required for cooling, and the two-phase system winds up with too much liquid. Vice-versa, the heat of ramping disappears during a flat top, and fluid flow to the two-phase system will decrease substantially as long as single-phase fluid pressure is maintained. Time constants of the changes are large (of the order of 5-15 minutes) for the Tevatron. Conceivably, they will become very long for the SSC.

2. Proposed System:

Figure 4 shows the schematic of the proposed system. The temperature of the single-phase fluid rises as it flows through the magnets. At regular intervals this single-phase fluid is cooled with boiling liquid helium in a heat exchanger. At the end of the string of magnets, the single-phase flow divides between 10°K shield flow and low pressure flow. The low pressure flow is separated into gas (small fraction) and liquid in the first heat exchanger. Boiling liquid in this exchanger will generate more vapor. Vapor and liquid flow from this heat exchanger to the next in separate lines. Heat transfer in the second heat exchanger will generate more vapor, and again vapor and liquid are separated and flow through separate lines to the next heat exchanger, and so on.

Although the line sizes between heat exchanger stations need to be "tuned" by size to balance pressure drop, fairly wide variations in pressure drop between adjacent stations are permitted without losing liquid seal. These variations are a function of the height of the column between liquid and vapor takeoff points in the heat exchangers. It turns out that a column of 20 inches of height (50 cm) represents a pressure difference of .0722 psig when comparing 100% liquid with a 100% vapor column. This pressure difference is a rather large fraction of the total pressure drop permitted between individual heat exchangers. In case better tuning is required, crude "valves" may be used to change the impedance of either the liquid and/or vapor line systems.

In order to better evaluate the merits of the proposed system, some preliminary calculations were carried out on pressure drops, line sizes, and behavior under steady and non-steady state conditions.

2.1 Assumptions:

The calculations were based on the following assumptions for steady state operation.

2.1.1 Distance between refrigerators is 6 km.

2.1.2 Temperature rise of the liquid helium flowing through the single-phase magnet system is .25°K (4.45 to 4.7°K).

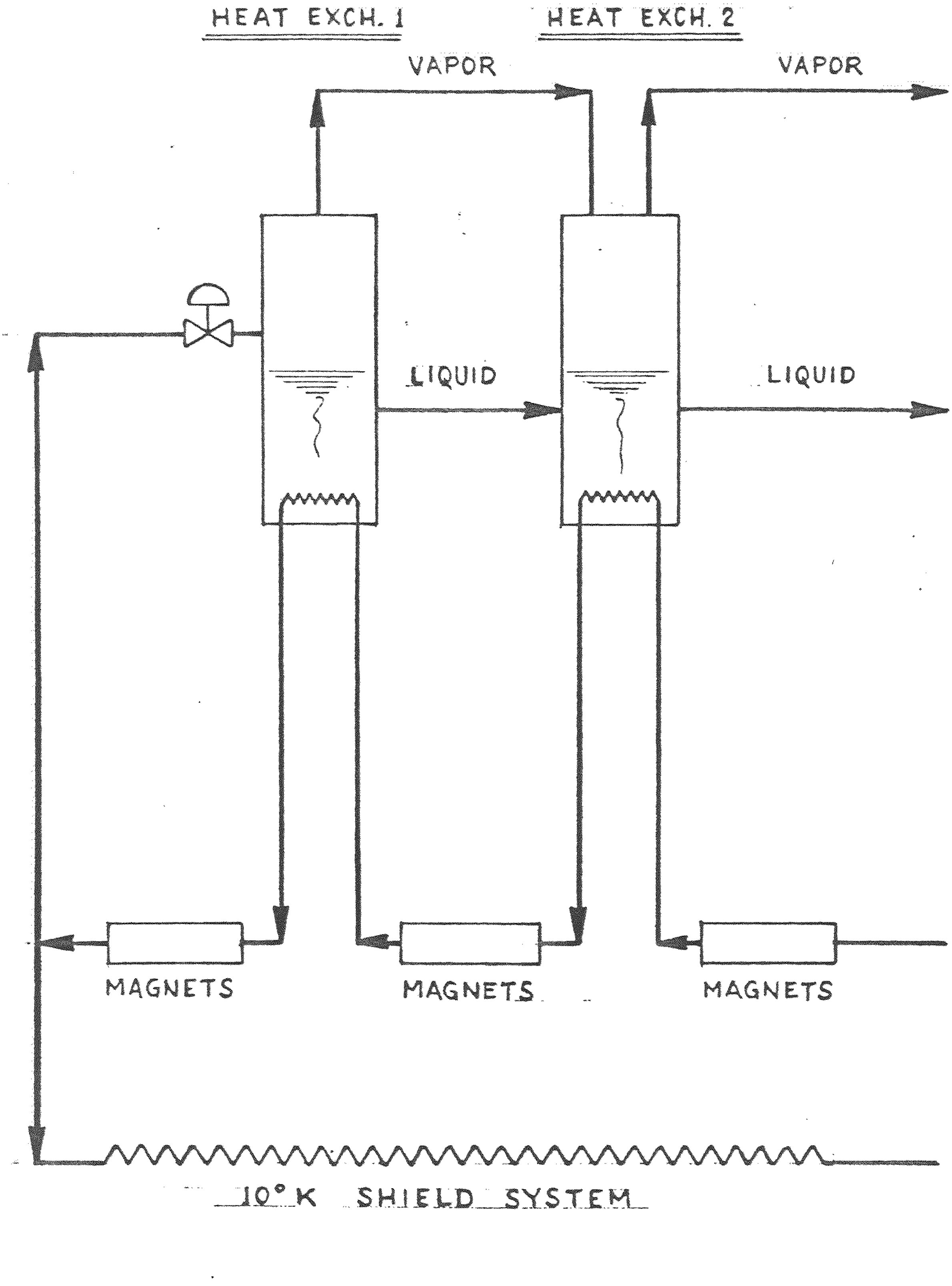

HEAT EXCH. 1
HEAT EXCH. 2
VAPOR
VAPOR
LIQUID
LIQUID
MAGNETS
MAGNETS
MAGNETS
10°K SHIELD SYSTEM

FIGURE 4

2.1.3 Single-phase flow rate is 50 g/sec.

2.1.4 Two-phase flow rate is 40 g/sec.

2.1.5 Heat load is approximately 750 W per 3 km and is uniform.

2.1.6 Pressure in two-phase system is 1.2 ata.

2.1.7 There are ten heat exchangers located at equal intervals of 300 meters.

2.1.8 Pressure drop over the distance of 3 km is .3 psig for pipe friction. Added to this are velocity heads caused by changes in flow direction and velocity.

2.2 Pressure Drop:

Pipe frictional pressure drop is given by:

$$\frac{\Delta P}{L} = \frac{f\,G^2}{193\,\rho\,d_h}$$

Where: $\frac{\Delta P}{L}$ = (psig/ft)

$f = \frac{.046}{Re^{.2}}$

G = Mass flow rate (lb/sec ft^2).

ρ = Fluid density (lb/cft).

d_h = Pipe hydraulic diameter (inches).

When equating expressions for pressure drop of liquid and gas, we find:

$$\frac{D_g}{D_l} = 1.398\left(\frac{F_g}{F_l}\right)^{.3749} \qquad (1)$$

Where: D_g and D_l are pipe internal diameters (ft) for gas and liquid, respectively; and F_g and F_l are gas and liquid flow rates (lbs/hr).

Diameter for liquid and vapor flow is given by:

$$D = \left[\frac{(2.37 \times 10^{-12})\ F^{1.8}\ (\mu)^{.2}}{\rho\ (\Delta P/L)} \right]^{.2083} \qquad (2)$$

Where μ = fluid viscosity, lbs/ft hr.

Tables I and II show the results of the calculations for 40 g/sec (317.2 lbs/hr) of flow rate at 1.2 ata. Table I shows a decreasing pipe diameter from 1.77 to .776 inches for liquid flow, and an increasing diameter of 1.08 to 2.47 inches for vapor flow. Note that:

a) Total pressure drop for a distance of 3,000 meters is only .3 psig.

b) Equivalent liquid inventory in the liquid and vapor spaces of the two-phase system is of the order of 3,500 liters or 1.17 liters per meter.

TABLE I

Liq. Frac.	F_1 g/sec	F_1 lbs/hr	d_1 in.	v_1 ft/sec	H_{v1} psi	A_1 in^2	V_1 liters
0.9	36	285.5	1.77	0.616	.00031	2.46	476
0.8	32	253.7	1.69	0.601	.00029	2.24	434
0.7	28	222.0	1.61	0.579	.00027	2.03	393
0.6	24	190.3	1.52	0.557	.00025	1.81	350
0.5	20	158.6	1.42	0.532	.00023	1.58	306
0.4	16	126.9	1.30	0.507	.00021	1.33	257
0.3	12	95.2	1.17	0.470	.00018	1.08	209
0.2	8	63.4	1.01	0.420	.00014	0.80	155
0.1	4	31.7	0.776	0.356	.00010	0.47	92
							2672

H_{v1} = One velocity head, psi.

A_1 = Pipe cross sectional flow area, in^2.

V_1 = Liquid inventory in pipe, liters.

d_1 = Pipe internal diameter, in.

TABLE II

f_g	F_g g/sec	F_g lbs/m	d_g in.	v_g ft/sec	H_{vg} (psi)	A_g in.^{2}	V_g liters
0.1	4	31.7	1.08	1.08	.00016	0.923	179
0.2	8	63.4	1.41	1.27	.00022	1.553	301
0.3	12	95.2	1.64	1.41	.00027	2.106	408
0.4	16	126.9	1.82	1.53	.00032	2.613	506
0.5	20	158.6	1.98	1.61	.00036	3,088	598
0.6	24	190.3	2.12	1.69	.00039	3.540	685
0.7	28	222.0	2.25	1.75	.00042	3.974	769
0.8	32	253.7	2.36	1.82	.00046	4.392	850
0.9	36	285.5	2.47	1.87	.00048	4.799	929
							5225

The calculated pipe sizes are smaller than the 3.011 inch pipe diameter required to carry 40 g/sec of two-phase fluid in a single pipe with the same pressure drop. Consequently, there does not seem to be a penalty in terms of more mass for two parallel piping systems.

Tables I and II show that each set of magnets between heat exchangers contains different pipe sizes. This complicates the fabrication process and also increases the number of spares required. To solve this problem, it certainly is possible to reduce the number of different sizes by providing a magnet assembly with the maximum size pipe and to provide inserts for these pipes to "tune" pressure drop. It also should be noted that the total pressure drop of a string of magnets is the significant parameter. A 12 meter long magnet in a string of 300 meter length, equipped with the wrong size pipe, will not affect pressure drop all that much.

2.3 Heat Exchangers:

Figure 5 is a conceptual drawing of the heat exchanger. Liquid level may vary by 20 in. between vapor takeoff and liquid in and outlet. There are six lines in and out of the exchanger. Single-phase helium (lines 1-2) flows through the finned tubes and is cooled by boiling liquid helium. Lines 3 and 4 represent low pressure helium liquid in and out. Lines 5 and 6 are vapor lines in and out. It should be noted that the liquid level will not drop below the coiled exchanger as long as there is an adequate supply of liquid. At the lowest liquid level, some vapor will be drawn into line 4. At the highest liquid level some liquid will spill over into the vapor line and be carried to the next exchanger with the vapor.

2.4 System Control and Stability:

Adequate cooling of the single-phase liquid in the magnets is assured as long as heat exchanger coils are submerged in liquid. Liquid level and pressure should be measured in each exchanger. As long as liquid level is located between liquid and vapor lines, system inventory can be calculated with great accuracy. Rate of rise or fall of liquid level in each exchanger will indicate the condition of each section of magnets between exchangers.

It appears relatively simple to tune the system for steady state operation. The tuning may be done through rather crude "valves" which can increase or decrease the impedance of the individual pipes. Even without these devices, the worst unbalance between a vapor and liquid line between exchangers is self-correcting. For instance, low pressure drop in a vapor line will ultimately result in filling the upstream heat exchanger and flow some liquid with the vapor through the vapor line. This quickly increases the pressure drop of this line to that of the liquid line.

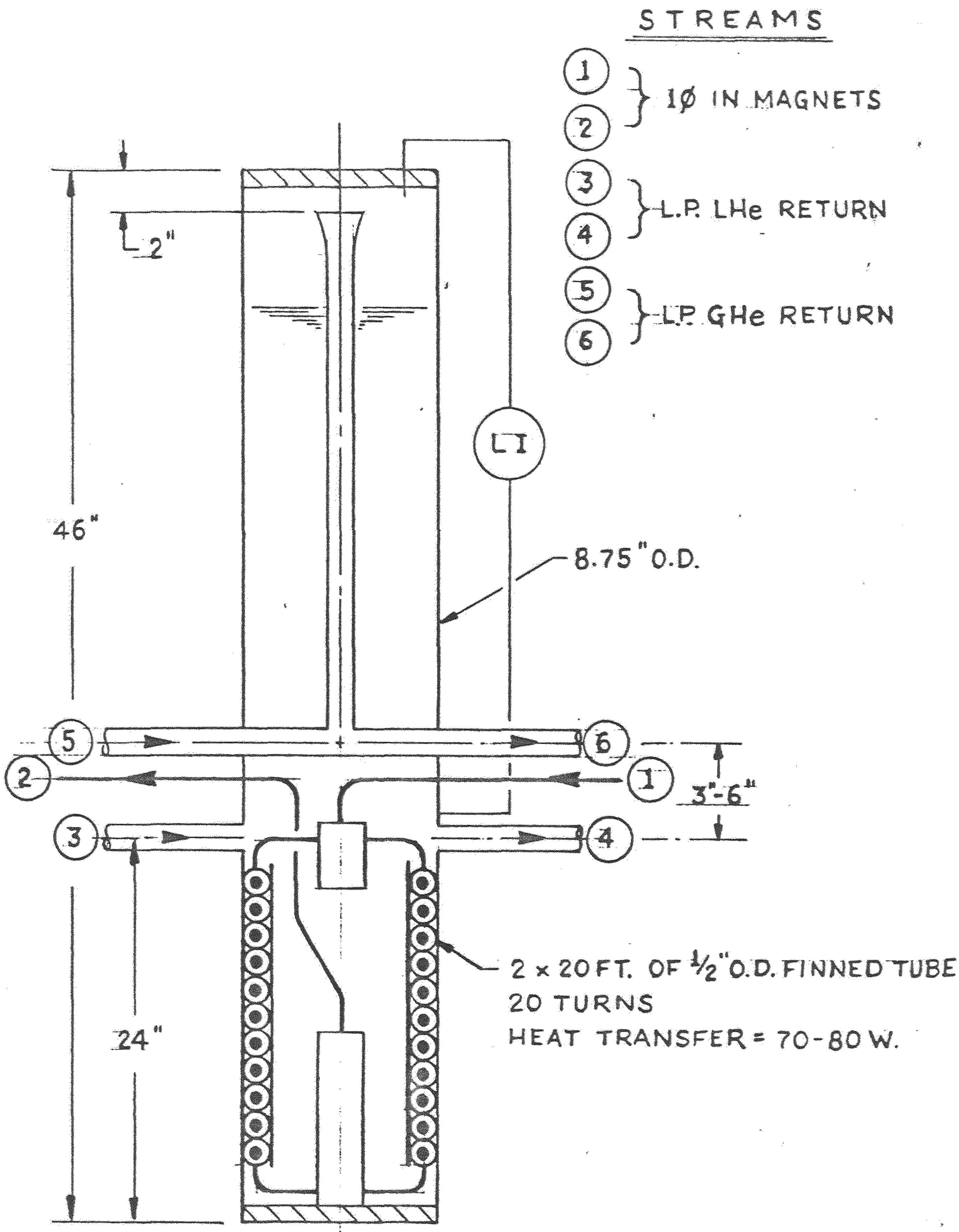


FIGURE 5

DESIGN OF EXCHANGER

2.5 Transient from Energizing of Magnets:

When the ring is cold and maintained in the superconducting state without beam, refrigerator requirements are minimal. This implies that the pressure drop in the two-phase system will be low and that the two-phase cooling system inventory is at a low temperature. If a high flow rate of liquid is maintained through the magnet system (by means of a pump), then the mass of the single flow system will be low in temperature. Inventory in single-phase system is then high, and the two-phase system will operate with excess liquid, relatively high liquid flow pressure drop, and containers filled with liquid in the heat exchangers.

If the flow rate through the magnet system has been matched to the minimal loss requirement, single-phase flow will still have a gradient of roughly the same magnitude as the one experienced at full load.

At high flow rate, when the ramp starts, the fluid in the single-phase magnet system will increase in temperature. Heat transfer to the liquid in the exchangers will raise vapor generation and pressure drop in the vapor piping of the two-phase system.

If pressure in the magnet single-phase system is kept constant, flow to the two-phase system will increase because of volume expansion of the heated single-phase fluid. This in turn increases pressure drop in the liquid lines of the two-phase system. The increase in average temperature of the single-phase fluid may be as much as .15°K. This in turn means an increase in specific volume of 4-5%.

Assume a total single-phase inventory of 380,000 grams for 3 km and two magnet strings (flow area $\sim 5\ cm^2$). At 100 g/sec flow rate for two strings, mass moves through one section between exchangers in 380 seconds. The temperature rise and extra expulsion of liquid takes place over this length of time. The amount expulsed is then some 17,000 grams or 8,500 grams per system. This is only 68 liters of liquid. The expulsion of single-phase flow occurs at a rate of 44.7 g/sec or 22.3 g/sec per string. This represents a large percentage increase in flow rate.

Since the liquid levels in the heat exchangers are high, additional liquid cannot be stored in the two-phase liquid system. Liquid will spill over into the vapor return line, and sôme liquid will be passed into the refrigerator system. This liquid needs to be temporarily stored in order to prevent a major upset condition of the refrigerator.

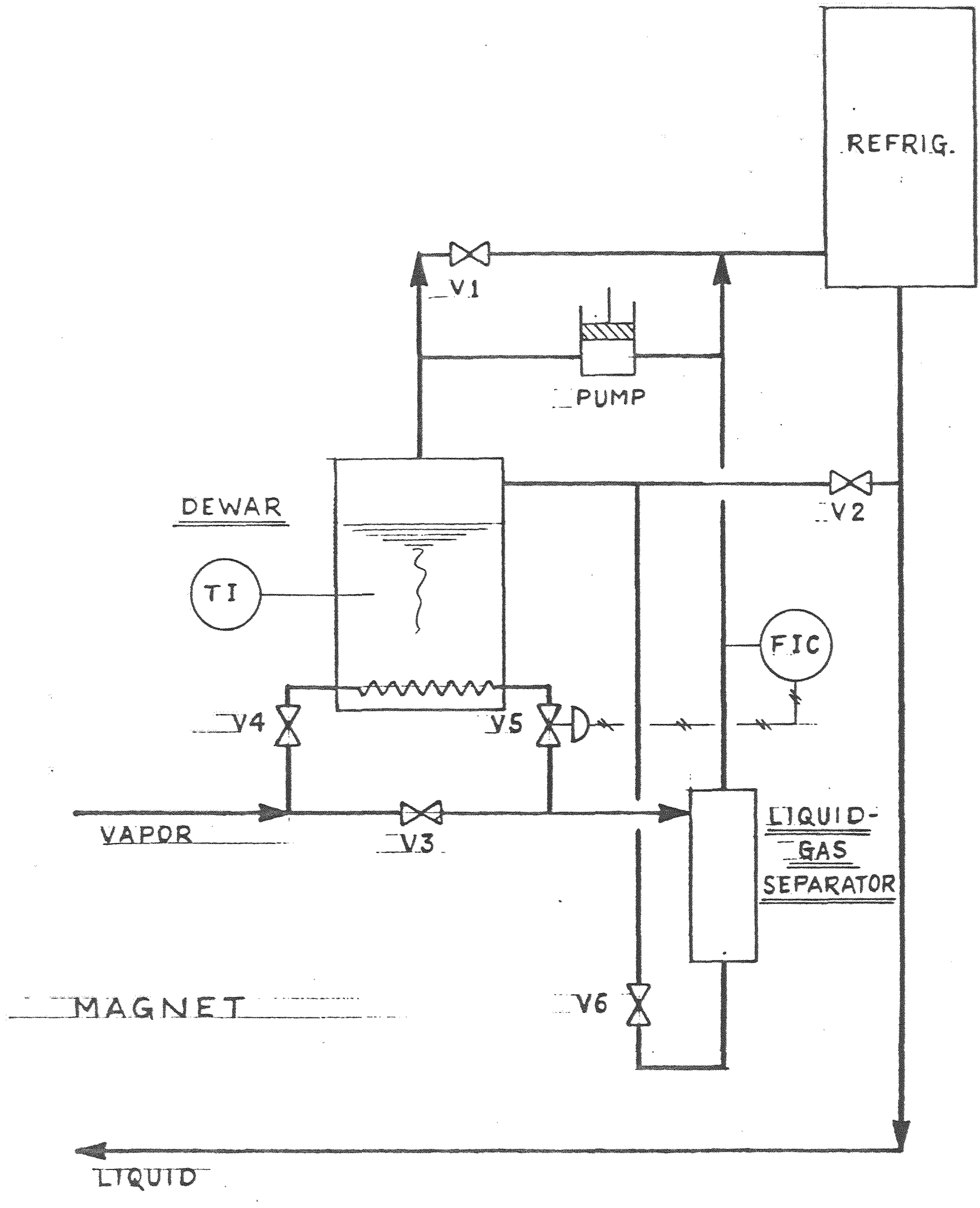
Total heat added to the system during a ramp is 100 Joules/meter per string or 600,000 Joules for a length of 3 km. We will store approximately half of this through temperature rise in the single-phase fluid. After 380 seconds the full rate is being transferred to the two-phase system, and the balance of the ramping heat vaporizes liquid and generates gas which flows to the refrigerator. Rate of ramping will determine the instantaneous load on the refrigerator. If the refrigerator has been operating at a low level, the sudden increase in load may provide too much of a transient. Section 2.6 discusses a system which permits temporary storage of heat in the refrigerator system.

Transient behavior will be somewhat different when flow rate through magnets and refrigerator were matched to the low standby level before ramping. In that case, temperature gradients in the single-phase system are in place. The start of the ramp immediately puts the full load on the two-phase system. To keep maximum single-phase temperature from exceeding the desired level, single-phase flow rate will be increased to a high level at the start of ramping. Vapor generation rate will be high, and the refrigerator may or may not be able to handle the load.

2.6 System for Storing Heat at 4.4°K:

Figure 6 shows the schematic of a system which will provide heat storage capability of 200-300 Joules per liter of volume. The system operates as follows:

The dewar is filled with subcooled liquid helium during steady state operation of the magnet system. This subcooled liquid has been generated by pumping with the pump in the vapor line between dewar and



REFRIG.
V1
PUMP
DEWAR
V2
TI
FIC
V4
V5
VAPOR
V3
LIQUID-
GAS
SEPARATOR
MAGNET
V6
LIQUID

FIGURE 6

refrigerator. After the desired temperature of the stored liquid has been reached, the pump stops, and valve V1 is opened to pressurize the dewar, mainly to keep pressure above atmospheric. During steady state operation valves V1 and V3 are open, and valves V2, V4, V5, and V6 are closed. When the ramp occurs valves V4 and V5 will be opened. To maintain the same flow of vapor to the refrigerator, valve V5 is controlled by a flow controller. Vapor flowing through the coil in the dewar is condensed in the heat exchanger. Condensate is stored in the liquid-gas separator. When the event is over, valves V4 and V5 are closed, and liquid from the separator may be drained into the liquid line or large dewar. Re-establishment of the subcooled state may be accomplished when desired.

Some numbers are instructive. Assume that the dewar contains 1,000 liters of subcooled liquid at 1.2 ata and 4.0°K. Mass of liquid is 131,062 grams, and amound of subcooling (below 4.42°K) is 268,940 Joules. When this liquid has been warmed to 4.42°K at constant pressure, liquid volume is 1,085.5 liters. To maintain constant pressure the refrigerator had to remove 85.5 liters of vapor or 1,755 grams. Heat added to the dewar has condensed 14,000 grams of vapor. This vapor has displaced 2,380 grams of vapor in the separator. This means that 14,000 grams of vapor flow normally flowing to the refrigerator has been reduced to 4,135 grams of vapor. Almost 10,000 grams of vapor from the magnet system has been stored. It should be noted that the numbers above are based on $\Delta T = 0°K$ at the end of the condensation cycle. This is not achievable. However, the numbers are valid if one assumes starting from a lower temperature of the subcooled liquid.

The generation of subcooled liquid helium is straight forward. The work of compression changes with time and ratio of compression. The liquid in the dewar is cooled by internal evaporation, and to reach a condition of 3.866°K and .7 ata, initial liquid mass is approximately 14% larger at 1.2 ata. Total amount of heat to be removed by the refrigerator is then some 450,000 Joules of which 20% is for compression of the cold vapor. If carried out over a period of

4 hours, rate is 31 Watts. Vacuum pump flow rate is then 1.25 g/sec, and displacement is of the order of 100 cc/sec. The pump is small, and impact on the refrigerator is small.

It appears then that the rate of ramping may be varied. The rate may be increased beyond the level at which the refrigerator can keep up with heat removal as long as a heat storage system is installed. Ramping rate then becomes a function of single-phase flow rate through the magnets (maximum temperature limitation).

3. CONCLUSIONS:

A system as described in this report has the following features:

3.1 Operates a two-phase cooling system at low pressure drops without the need to increase pipe diameter to large values.

3.2 Provides an accurate accounting of fluid inventories of all parts of the system.

3.3 Provides a system which operates satisfactorily without controls or at most will require very crude controls.

3.4 Provides very short time constants for controlling transients.

3.5 Provides the possibility of increasing distance between refrigerators.

3.6 Smaller inventory of the overall system.

3.7 Provides storage capability for excess heat generated during a ramp and reduces the transient of the refrigerator system during the ramp.

REFERENCES

(1) Anderson, R. J. and Russell, T. W. F.: Designing for Two-Phase Flow, Parts I, II, and III, Chemical Engineering, December 6, 1965, pp. 139-144; December 20, 1965, pp. 99-104; and January 3, 1966, pp. 87-90.

(2) Baker, O.: Multiphase Flow in Pipelines, The Oil and Gas Journal, November 10, 1958, pp. 156-167.

(3) DeGance, A. E., and Atherton, R. W.: Chemical Engineering Aspects of Two-Phase Flow, Part 1 - Part 8, Chemical Engineering Series Beginning March 23, 1970, pp. 135-139 and ending February 22, 1971, pp. 125-132.

(4) Hewitt, G. F., and Hall-Taylor, N. S.: Annular Two-Phase Flow, Pergamon Press, 1970.

(5) Lockhart, R. W., and Martinelli, R. C.: Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes, Chemical Engineering Progress, Vol. 45, No. 1, pp. 39-48, 1949.

(6) Rohsenow, W. K., and Harnett, J. P., Editors: Handbook of Heat Transfer, McGraw-Hill, 1973.