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Two Phase Cooling for Superconducting Magnets

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ABSTRACT

Comments on the use of two phase helium in a closed circuit tubular cooling system and some results obtained with the TPC superconducting magnet are given. Theoretical arguments and experimental evidence are given against a previously suggested method to determine helium two phase flow regimes. Two methods to reduce pressure in the magnet cooling tubes during quenches are discussed; 1) lowering the density of helium in the magnet cooling tubes and 2) proper location of pressure relief valves. Some techniques used to protect the refrigerator from too much cold return gas are also mentioned.

INTRODUCTION

Under many circumstances forced flow two phase helium is an attractive coolant for superconducting magnets. Forced flow can utilize tubular cooling systems which can withstand higher pressures than the larger tanks of a conventional pool bath system. A tubular cooling system also reduces the helium inventory involved in quenches, and can be used to guide cooling specifically to areas where it is most needed during the cooldown. However, designing such a system involves problem areas not encountered in a pool bath system. The magnet designed for use in the TPC detector at PEP utilizes two phase cooling. Examples are given based on experience with that magnet. The emphasis will be primarily on general problems with two phase design rather than the design of a specific magnet.

THE PRESSURE DURING QUENCHES

The largest pressures a forced flow system is likely to experience will probably occur at the time when the magnet quenches. At that time LHe in the vicinity of the magnet will begin to boil rapidly until a one phase status is reached; then, the pressure will continue to rise as the He and magnet temperature rises. The calculation of the real pressure distribution in the system is very difficult because He is flowing out of the tubular part of the system. However, it is relatively simple to calculate an upper limit for the pressure by assuming that no He is leaving the tube. To compute this, one has to know the initial density and the final temperature which the magnet system will reach. From He data at that temperature, one can determine the pressure that corresponds to the initial density. That pressure is maximum if the tube is initially full of pure liquid. It would be desirable to design the cooling system to withstand this pressure. In the case of the TPC magnet, T_{max} is around 60 K and P_{max} would be 24 MPa. The finned cooling tube wrapped around the solenoid was designed to withstand this pressure, as were the transfer lines running to and from the magnet.

Unfortunately, in the TPC magnet, a part of the lines feeding He cooling to the current leads was built using flex tubing which, as it was subsequently discovered, could not withstand P_{max} with absolute certainty and these pieces of flex tubing were relatively inaccessible. Since the system does not run with only liquid everywhere in the cooling tube, and much of the He is vented out during quenches, the actual pressure exerted on the flex tubing was much less than the upper limit computed above. We increased the venting by adding an additional relief valve. Then a more realistic pressure was calculated based on compressible flow dynamics with the relief valves included. The final pressure was proportional to the assumed initial He density. Our prediction of the initial density used thermodynamic analysis of the coil and refrigerator system assuming that liquid traveled at the same velocity as gas through the system before a quench. This assumption was based on applying to He a diagram based on water and air which had been developed by Ovid Baker in 1954.[1] It indicated that, under our conditions, the flow regime should be bubble and froth, a regime where the liquid and gas flow at the same velocity. Our operating point should be far from the boundary with any other type flow on that diagram (see Table 1 for TPC magnet operating conditions). With this initial density, the pressures during quenches were predicted to be acceptable, but during tests using induced quenches at low magnet stored energies the measured pressures were found to be higher than the predictions. Then the initial density was measured and found to be higher than our prediction, and in just the right ratio to correspond to predictions of the pressure calculation. This could be explained by assuming that liquid was moving slower than gas in the tube and hence that the flow regime we were experiencing was different from that predicted by applying Baker's diagram to our two phase He. This assumption was strengthened by observations of pressure and temperature variations which could also be explained by assuming slug flow, a flow regime where liquid and gas do not necessarily flow at the same rate. We then realized that using Baker's parameters to extrapolate the water and air data to He violates the rules of dimensional analysis. Therefore, a warning is in order not to use the plot by Baker for two phase He.

Table 1. Operating Parameters of the TPC Magnet

	All Flow through He Bath Heat Exchanger	Partial Bypass around He Bath Heat Exchanger
inlet pressure Pascal (abs)	1.7×10^5	1.6×10^5
outlet pressure Pascal (abs)	1.3×10^5	1.3×10^5
magnet temp. (K)	4.7	4.6
dewar pressure Pascal (abs)	1.3×10^5	1.3×10^5
pre-JT pressure Pascal (abs)	8.4×10^5	8.2×10^5
pre-JT temp. (K)	5.3	5.6
Baker parameter B_x	3.1×10^3	from 3.9×10^3 to 0.54×10^3
Baker parameter B_y	6.6×10^3	from 6.1×10^3 to 18.1×10^3

THE BAKER DIAGRAM

One of the more difficult problems in designing a fluid transport system with two phases flowing in the same pipe is determining the flow regime of the two phases. That is, how do the two phases coexist: as froth, slugs, plugs, stratified flow, etc.? Pressure drops and density throughout a system are dependent on this flow regime. In 1954, Ovid Baker, as part of a study on pressure drops in oil and gas lines, developed a graphical method of summarizing experimental results on flow regimes of two phases flowing together in the same pipeline. He collected data points from four experimenters [2-5] working on two phase flow of air and water, organized them on a two dimensional plot based on two parameters recommended by a major handbook in his field of study,[6] and drew lines representing boundaries between the flow regimes. This is the "Baker Diagram". The two parameters were based on several physical properties thought to be relevant. This system has been in use for many years in chemical engineering. It has been suggested that one might use this diagram with its flow regime boundaries to predict flow regimes for two phase helium [7,8]; however, we find it fails to correctly predict flow regimes that we have seen in the TPC magnet. A closer examination of the development of Baker's work shows that such a failure in predicting flow regimes for He should come as no surprise.

The two parameters of the Baker diagram are $B_x=L/G(\lambda\Psi)$ for the abscissa, and $B_y=G/\lambda$ for the ordinate. L is the liquid's mass velocity, and G is the gas's mass velocity, $\lambda\propto(\rho_G\,\rho_L)^{1/2}$ and $\Psi\propto[\mu_L/\rho_L{}^2]^{1/3}/\nu$, where ρ_L and ρ_G are the liquid and gas densities, ν is the surface tension of the liquid, and μ_L is the liquid's viscosity. This parameterization was supposed to allow use of the Baker diagram for predicting flow regimes of liquids and gasses other than water and air and in fact was applied to oil and natural gas.

The properties of helium at 4.2 K are very different from those of air and water at room temperature. (See Table 2.) To scale the experience gained with fluid flow of water and air to two phase helium at 4.2 K one would have to follow the rules of scaling given by dimensional analysis based on all relevant parameters. It can be shown that Baker's diagram does not respect the rules of dimensional analysis[9] and hence, is not a reliable tool to use for helium. Also, Baker's parameters do not depend on the gas viscosity and since most of the helium is often in gaseous form, it should be an important factor. In conclusion, there is no justification for using the Baker diagram for two phase helium flow.

LIMITING MAXIMUM PRESSURES

As mentioned earlier in this report, the density of He in the cooling tubes was found to be higher than anticipated. Since there was concern that the flex tubing connecting the current leads might be of inadequate burst strength, it was finally decided to reduce liquid density in the cooling tube to reduce the pressure there during quenches. We were able to change He density in the TPC magnet using the method of bypassing the heat exchanger. This caused the flow to cross the boundary between a pulsing slug/plug flow with Baker parameters given in column 1 of Table 1, and a smoother flow regime with Baker parameters somewhere in the range given by the values in column 2 of Table 1. This boundary point falls far from the boundary shown on the Baker diagram and further confirms that the diagram does not apply to the case of two phase He.

The He density in the cooling tubes was much higher than at the refrigerator outlet due to a heat exchanger immersed in liquid helium through which coolant flowed before entering the magnet's cooling tube (see Fig 1.). One method of reducing the density was to install a heater after the heat exchanger. While this method worked and allowed testing of the magnet to full stored energy, it was viewed as a temporary solution. It was not fail safe and reduced reserve refrigeration. An alternate method involved adding a valve which allowed a portion of the refrigerator's output to bypass the He bath heat exchanger. Operating with this valve partially opened gave a density which lead to safe pressures during quenches at full stored energy.

Table 2. Properties of Air, Water, and Helium

	Air Room temp.	Water Room temp.	He Gas 4.2 K	He Liquid 4.2 K
density (Kg/m)	1.2	1.0×10^{3}	17	125
surface tension (N/m)	-	7.3×10^{-2}	-	1.0×10^{-4}
viscosity Pa-S	1.8×10^{-5}	1.0×10^{-3}	1.3×10^{-6}	3.2×10^{-6}

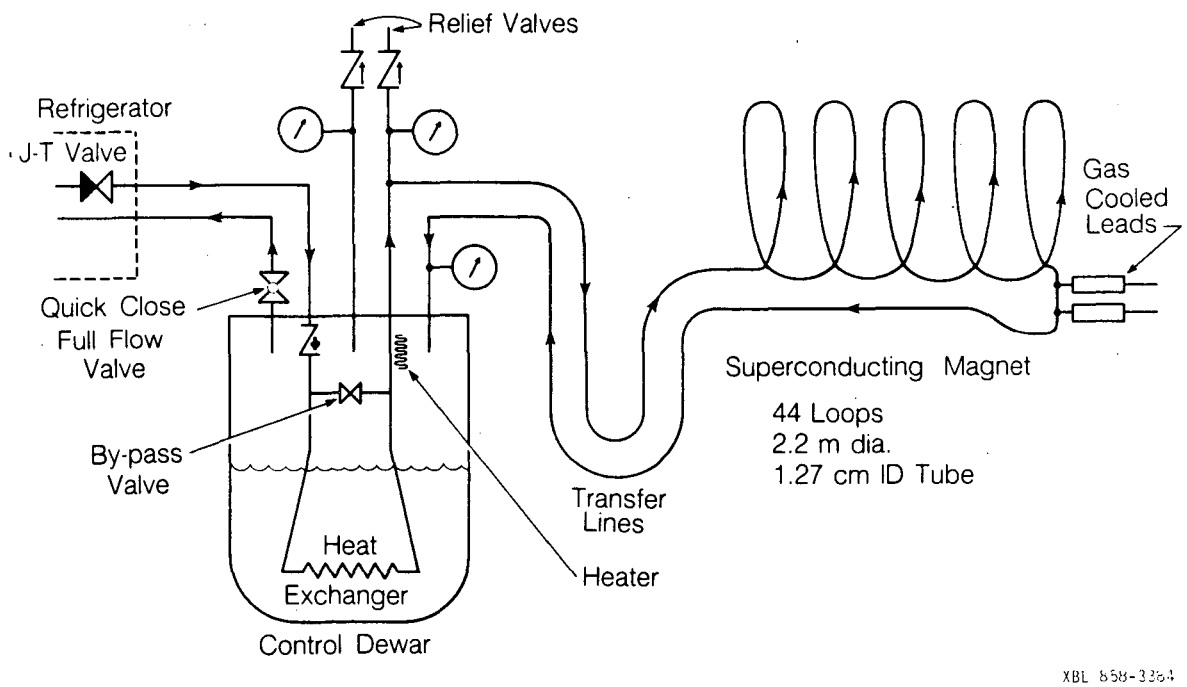

Fig. 1. Helium flow circuit--TPC magnet.

ACTIVE VALVES FOR THE PROTECTION OF THE SYSTEM

The final relief valve system used to minimize pressures during quenches utilized two automatic safety relief valves, one at either end of the cooling tube. These valves are of the same design used on the Fermilab beam magnets[10] and have proven to be very satisfactory. They open to full flow quickly, can operate in a partially opened mode, and reseat reliably.

The refrigerator for the TPC magnet is a CTI model 2800 with two turboexpanders. These turboexpanders rotate at high speeds, and their support bearings are easily damaged, particularly in the event of sudden temperature and pressure changes which accompany quenches. In order to protect the turboexpanders and thin walled heat exchangers of the refrigerator, a valve (V_{ret}) on the transfer line returning to the refrigerator can be closed when required. We have chosen to close this valve on any of several situations: 1) a quench is detected, 2) the pressure in the dewar rises above 2.0×10^5 Pa (abs), 3) the pressure difference between supply and return transfer lines rises above 1.0×10^5 Pa, 4) a refrigerator shutdown. A check valve prevents flow in the reverse direction back into the supply transfer line. Automatic controllers have been added to regulate temperatures at the refrigerator and permit it to continue running, even when flow to and from the magnet is interrupted.

If cooling is interrupted, the magnet warms slowly. In the case of the TPC magnet, T_c is reached about ten minutes after the loss of cooling, and the magnet quenches. Heat leaks in the supply transfer line cause the helium in the line to rise in temperature faster than the magnet. Thus, if one attempts to restart the flow of coolant, it is likely that warm He in the transfer line will raise the magnet temperature and cause a quench. If the refrigeration is lost, we turn off the magnet current supply in order to drop the magnet's stored energy. This leads to lower temperatures when the magnet quenches, and reduces recovery time.

CONCLUSION

While two phase forced flow cooling may be considered as an attractive technique, it involves design problems which differ in many cases from those encountered with pool bath designs. In particular, the prediction of flow regimes, pressure drops, and He density are complicated by the lack of an appropriate design guide. It would be of great benefit for the cryogenics community if such a design tool specifically for He near 4.2 K were developed.

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Appendix A

Density Measurement, the Ratio of Gas to Liquid Velocity, and Pressures During Quenches

Measurement of the average density of helium in the cooling tubes of the TPC magnet involved estimating the mass of helium in the magnet. The changes in mass of helium in parts of the system other than the cooling tubes was measured by recording the level of liquid in the control dewar and the pressures and temperatures at various places in the system. We started a measurement with cooling flowing normally through the tubes and the parameters were recorded. Then we bypassed the cooling fluid directly into the control dewar, without letting it flow through the cooling tube and waited for the magnet to warm above the boiling point of helium in the tubes. The mass of gas in the tubes plus the change in mass in the system outside the cooling tubes constituted our measure of the original mass of helium in the tubes. From this mass and the known volume of the system we determined the average density.

We also know the average operating pressure of the fluid while normal cooling is going on, and therefore the density of gas and the density of liquid in the two phase mixture. This allows us to calculate the ratio of mass of gas, m_{gas}, to mass of liquid, m_{liquid}, in the tubes on the average as

$$\frac{m_{gas}}{m_{liquid}} = \frac{\frac{1}{\rho} - \frac{1}{\rho_{liquid}}}{\frac{1}{\rho_{gas}} - \frac{1}{\rho}} \tag{A.1}$$

This measurement can be used to estimate the ratio of gas velocity, v_{gas}, to liquid velocity, v_{liquid}, if we have a measure of the mass flow in gas phase and the mass flow in liquid phase during normal cooling since

$$\dot{m}_{gas} = m_{gas}\frac{v_{gas}}{\ell} \quad \text{and} \quad \dot{m}_{liquid} = m_{liquid}\frac{v_{liquid}}{\ell}$$

where ℓ is the total length of tubing. We can estimate the average enthalpy per gram flowing through the cooling tubes, h, by measuring the heat leak of the system (80 watts), the total change in enthalpy per gram of fluid delivered by the refrigerator in passing through the system and by knowing where the heat is deposited in the system. The effect of the heat exchanger is taken into account by assuming it reduces the temperature of fluid to the temperature of the liquid in the control dewar. If the heat exchanger is less than 100 percent efficient, this will induce an error in our measurement. Again, we know the average operating pressure of the two phase mixture and therefore the enthalpy per gram of the gas, h_{gas}, and the enthalpy per gram of the liquid, h_{liquid}. We then estimate

$$\frac{\dot{m}_{gas}}{\dot{m}_{liquid}} = \frac{h - h_{liquid}}{h_{gas} - h} \tag{A.2}$$

Thus we estimate

$$\frac{v_{gas}}{v_{liquid}} = \left(\frac{\dot{m}_{gas}}{\dot{m}_{liquid}}\right) / \left(\frac{m_{gas}}{m_{liquid}}\right) \tag{A.3}$$

A.1 Operation with heater in the magnet supply line

Table 3 shows the results of applying this technique. The first column represents a first set of measurements made with the original operating condition and shows $v_{gas}/v_{liquid} = 1.3$, in disagreement with our expectations of equal velocity from the location on the Baker diagram which predicted bubble and froth which would imply $v_{gas}/v_{liquid} = 1$. The second column represents an operating condition arrived at by heating the helium fluid after it passed through the heat exchanger but before it entered the cooling tubes. The density was reduced and we estimate $v_{gas}/v_{liquid} =$

1.2, which is nearer to 1. It is by no means certain that liquid and gas are flowing at the same speed. We did notice a smoother flow of gas up our current leads and less pulsing of the readings on a temperature diode in the return line from the magnet. Peak pressures experienced in quenches at 1800 A are also included in Table 3 for the two cases, illustrating the desired drop in peak pressure with decreased density. This decrease in pressure allowed us to test the magnet at currents up to 2000 A before installing a bypass valve around our heat exchanger.

A.2 Operation with the heat exchanger bypass

A bypass valve was installed so we could cool the magnet without heating the helium, lowering the density in the cooling tubes instead by decreasing the effectiveness of the heat exchanger. It was not possible to adequately cool the magnet with the heat exchanger bypass valve completely open, so our data under those conditions is sparse and we have no data on pressures during quenches for that condition. In addition, the density measurement using the technique described earlier did not give reliable measurements when the bypass valve was open due to uncertainties in how much fluid was in the heat exchanger before and after bypassing the cooling fluid into the control dewar. Nevertheless we have been able to reduce the density in the cooling tubes by running with the heat exchanger bypass half open as evidenced by the pressure measurements during quenches given in Table 4. The first column also corresponds to measurements made under conditions similar to the original but at a different time than the one indicated in Table 3. The second column corresponds to operation with bypass valve half open. The liquid level in the control dewar was higher for measurements of Table 4 than for those of Table 3. The density measurement for the second column is estimated from the maximum pressure in an 1800 A quench and is used to imply a velocity ratio range shown in the Table. The higher value is obtained by assuming the heat exchanger has no effect and the lower value by assuming the heat exchanger is still 100 percent effective. The truth is somewhere between the two, and is compatible with $v_{gas}/v_{liquid} = 1$. The ratio of gas to liquid velocity is sensitive to small changes in density and the ratio 3.8 shown in Table 4 for the original condition is much larger than the ratio 1.3 shown in Table

3. This may be a reflection of the variability we have in our operating conditions. The operation with the bypass valve half open was found to be much smoother than with it closed, just as we found when we heated the fluid.

A.3 Effect of a heat load downstream of the magnet

We adopted the half open bypass valve as standard running condition. With this condition, we tested to find out how the density of helium in the cooling tubes was affected by a heat load downstream of the magnet, by turning on a heater in the control dewar. Due to our mode of refrigerator operation, this forced the refrigerator to deliver more refrigeration, thus more He mass flow so as to supply still some liquid downstream from the magnet. That operation was expected to augment the average amount of liquid in the magnet. Although our absolute density measurements were subject to systematic errors, these errors should be the same for all conditions of heat in the control dewar and we were able to measure density changes and to predict increments in P_{max} during quenches for various conditions. Table 5 shows the results of these tests. It is clear that we could not risk having an additional heat leak greater than 50 or 60 watts downstream of the magnet without increasing the peak pressures during quench to dangerous values.

Table 3. Flow characteristics without bypass; change of flow by heating fluid before sending it to the magnet cooling coils.

Heat added to fluid just after heat exchanger and before supply lines to the magnet cooling tubes	0 W	75 W
Pressure differential across the magnet plus transfer lines [Pascal]	3.2×10^4	3.2×10^4
Average operating pressure in magnet cooling tubes [Pascal-absolute]	1.49×10^5	1.58×10^5
Magnet plus transfer lines heat load	80 W	80 W
Measured average fluid density in cooling tubes [g/cc]	0.064	0.047
m_{gas}/m_{liquid} in cooling tubes	0.30	0.84
Mass flow [g/s]	6.0	11.1
Estimated average enthalpy per gram of fluid flowing through the magnet cooling tubes [J/g]	17.2	21.1
$\dot{m}_{gas}/\dot{m}_{liquid}$ in cooling tubes	.40	1.01
v_{gas}/v_{liquid}	1.3	1.2
"R_e" effective Reynolds No.	3×10^5	6×10^5
P_{max} @ 1800 A [Pascal-absolute]	$1.4 \times 10^{6*}$	1.0×10^6

*Extrapolated from quenches at 1200 A and 1600 A.

Table 4. Flow characteristics with bypass; change of flow by changing setting of the bypass value.

Bypass value position (closed = no bypass)	closed	1/2 open
Pressure differential across the magnet plus transfer lines [Pascal]	3.2×10^4	2.1×10^4
Average operating pressure in magnet cooling tubes [Pascal-absolute]	1.49×10^5	1.45×10^5
Magnet plus transfer line heat load	80 W	80 W
P_{max} @ 1800 A [Pascal-absolute]	$1.41 \times 10^{6*}$	8.1×10^5
Estimated or measured average fluid density in cooling tubes [g/cc]	0.084	0.049
m_{gas}/m_{liquid} in cooling tubes	0.12	0.60
Mass flow [g/s]	6.1	6.9
Estimated average enthalpy per gram of fluid flowing from the magnet cooling tubes [J/g]	17.3	16.2 to 24.0
$\dot{m}_{gas}/\dot{m}_{liquid}$ (range)	0.42	0.31 to 2.3
v_{gas}/v_{liquid} (range)	3.6	0.5 to 3.8

*Extrapolated from quenches at 1200 A and 1600 A.

Table 5. Change in density with additional heat load in control dewar.

Additional heat in control dewar	7 W	20 W	23 W	60 W
Mass flow [g/s]	6.3	5.9	6.5	8.2
$\Delta\rho$ [g/cc] (Increase over density with no added heat in control dewar)	.003	.017	.015	.040
Predicted increase in maximum pressure [Pascal] during a quench. (Increase over maximum pressure during quench with no added heat in the control dewar)	4×10^4	2.4×10^5	2.1×10^5	5.7×10^5

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