AD/RHIC/RD–107

**RHIC PROJECT**

Brookhaven National Laboratory

# Performance of VJRR, Six O'Clock Valve Boxes and Circulating Compressor

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November 1996

# PERFORMANCE OF VJRR, 6 O'CLOCK VALVE BOXES AND CIRCULATING COMPRESSOR

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## ABSTRACT

During the period from July 15 through July 20 1996, the helium refrigerator for the Relativistic Heavy Iron Collider, RHIC, was used to cool to liquid helium temperature the Vacuum Jacketed Transfer Line from Refrigerator to Ring, the VJRR, two Valve Boxes in the six o'clock position at the ring, and one spool of the Vacuum Jacketed Line connecting the Valve Box and the RHIC tunnel, the VJR. The superconducting bus in the valve boxes and lines was powered to 7000 amperes. The refrigerator and cryogenic distribution system are mainly operated by personnel from the RHIC Cryogenic Section. This report presents preliminary results obtained from this test including 1) heat loads of the 55 K shields of the VJRR and the Valve Boxes, 2) 4 K heat loads of the supply and the cooldown return lines in the VJRR, and 3) the pump performance curve of the circulating compressor located in the 6 O'clock Blue Valve. Further verification of these results will be acquired during the first sextant test.

## I. Introduction

The RHIC Refrigerator is located in Building 1005 near the five o'clock location. The VJRR is used to connect the Refrigerator and the Valve Boxes at six o'clock. The VJR is used to connect the Valve Box and RHIC tunnel. There are five pipes inside the VJRR: 4 K helium supply line S, 4 K helium return line R, 55 K shield supply line HG, 55 K shield return line HR and a cooldown return line CR. There are also five pipes inside the VJR: a magnet cooling line M, 4 K helium supply line S, 4 K helium return line R, utility line U, and heat shield line H. Descriptions of the VJRR and VJR are given in the RHIC Design Manual. For this test, all five lines in the VJRR were used. Only the M and the U lines in the VJR were used to carry helium from the Yellow Valve Box to the RHIC tunnel and back. An End Can was attached at the end of the VJR in the RHIC tunnel to connect the M and U lines for this test.

In the morning of July 15, the RHIC Refrigerator was turned on. Cold helium was sent to the Valve Boxes sometime later. On July 18, the refrigerator, VJRR and the two Valve Boxes were cooled to 5 K. On July 19, the superconducting bus and its leads were powered to 7000 amperes. From the evening of July 19 to early morning of July 20, tests of the circulating compressor were performed. The heat load measurements for the Supply and the Cooldown lines in VJRR were performed in the morning of July 20. The test ended at noon time, July 20th.

## II. Shield Heat Load

The heat shield is cooled by high pressure helium at approximately 60 K from Cold Box 3 to the 6 o'clock location through the heat shield supply line, HG, in the VJRR as shown in Figure 1. At the 6 o'clock location, shield flow is split to feed the Yellow and the Blue Valve Boxes. After flowing through the heat shield inside the Valve Boxes, helium flow is recombined and returned through the heat shield return line, HR, to the RHIC Refrigerator. The locations of the temperature sensors, the pressure transducer and a venturi flow meter used for the measurements are also given in Figure 1. A cross section view of the VJRR is given in Figure 2. As seen, HR and the other three cold helium lines S, R and CR, are located inside the shield. Therefore, the shield heat load is intercepted primarily by HG.

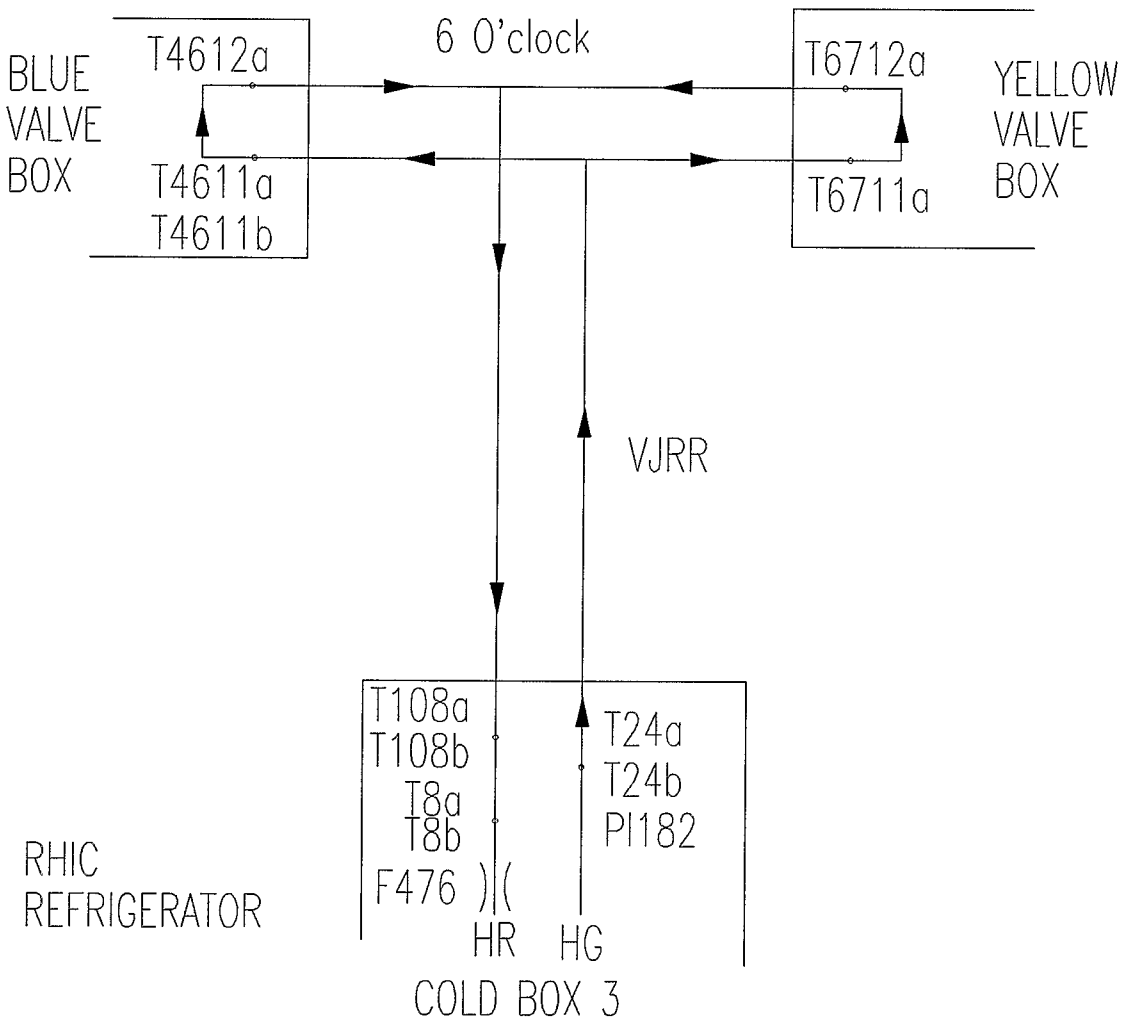


**Figure 1 Flow Path for the Heat Shield from RHIC Refrigerator to 6 O'clock**

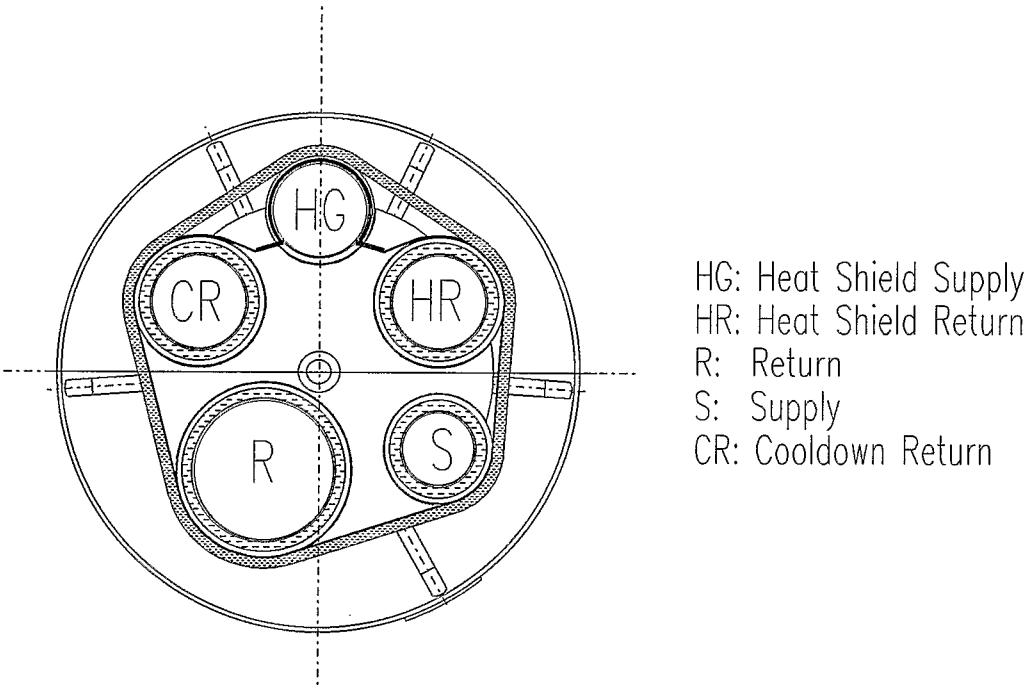


**Figure 2 Cross Sectional View of VJRR**

The temperature readings at HG and HR in Cold Box 3, T24a and T8a, for July 19 and 20 are given in Figure 3. During this period, the shield temperature drifted between 55 and 70 K. If constant shield temperature is maintained, the heat input to the system equals the product of the flow rate and the enthalpy difference between the supply and the return lines. Since the shield temperature varies with time, the heat load is not equal to the product of the flow rate and enthalpy difference. Figure 4 shows the product of the enthalpy difference and flow rate. The average shield heat load is about 3,000 watts for the VJRR and the two Valve Boxes as obtained from Figure 4 when the supply temperature is stable. The travel time of helium from Cold Box 3 to 6 o'clock location and back is approximately 10 minutes.

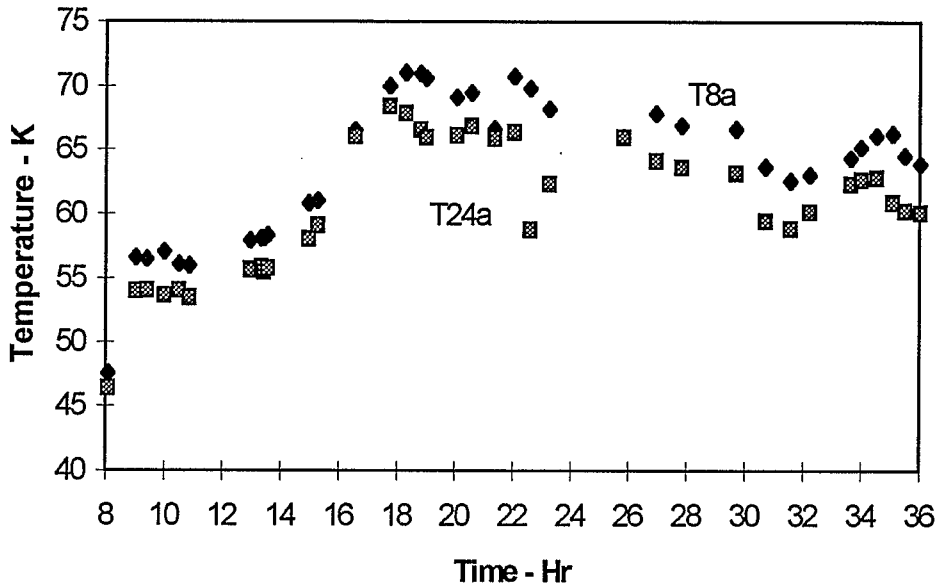


**Figure 3 Shield Supply and Return Temperatures at Cold Box 3**

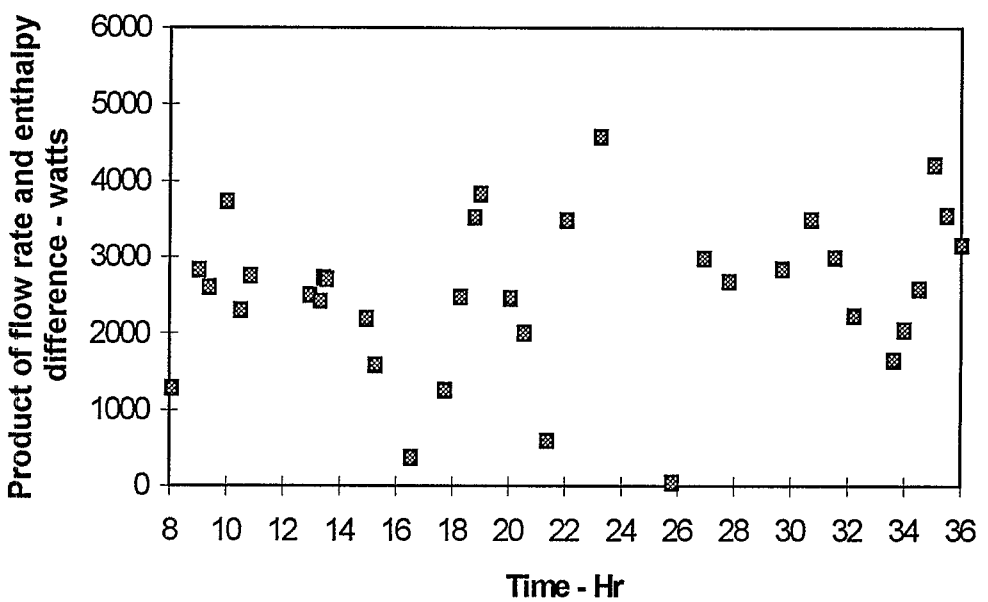


**Figure 4 Product of Flow Rate and Enthalpy Difference for the Shield**

In a similar way, the shield heat loads for the VJRR, and the Yellow and the Blue Valve Boxes were evaluated. No observable difference was found by using either of the redundant sensors. The products of flow rate and enthalpy difference, defined as apparent heat loads, are given in Figure 5. The best estimates for the shield loads in the VJRR, and the Yellow and the Blue Valve Boxes are 1100, 1200 and 700 watts respectively. The design values of these shield heat loads are 800 watts for the VJRR and 325 watts for each Valve Box. The measured heat loads are higher than the design values. Uncertainties associated with the heat load measurements maybe large but we also know the heat load in the Yellow Valve Box are substantially higher than its design value because its vacuum were in the $10^{-3}$ Torr range. At the present time, problem associated with poor vacuum in the Yellow Valve Box has been corrected. Lower heat leak is expected for the next test.

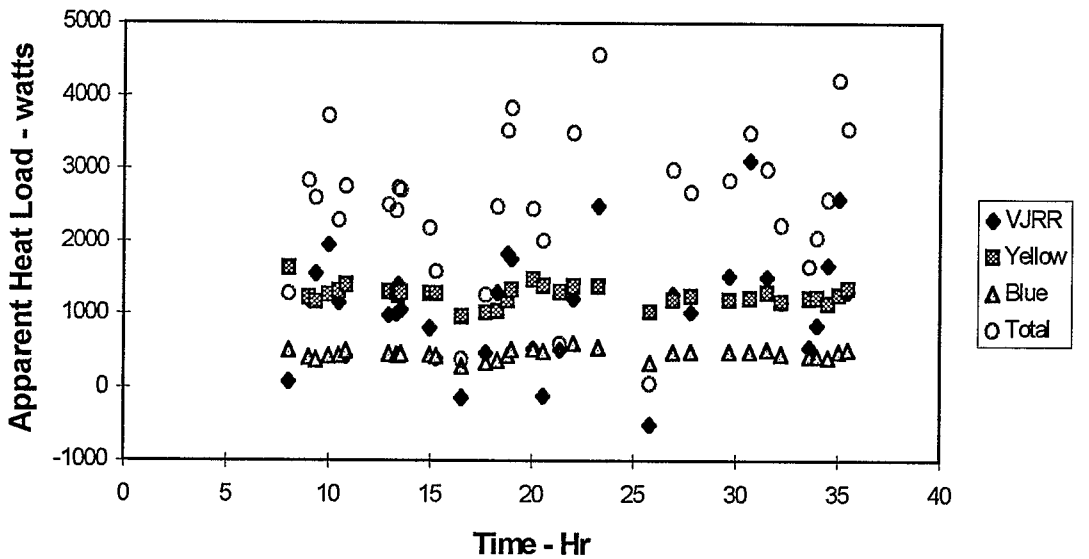


**Figure 5 Apparent Shield Heat Load in VJRR and Valve Boxes**

## III. Supply Line and Cooldown Return in VJRR

It is often difficult to calculate the heat load directly using the flow rate and the enthalpy difference since the temperature difference is small and the flow rate may not be readily available. For normal RHIC operation, the supply line is used to feed the Recooler Heat Exchangers and the power leads. Cold helium vaporizes in the Recooler Heat Exchanger and returns to the refrigerator through the Return Line inside the VJRR. Helium flow warms up through the cooling passages of the power leads and returns to the compressor suction through the warm return header. This operating configuration is not suitable for the heat load measurements.

A flow path is set up for measuring the heat loads of the Supply and the Cooldown Return lines. Cold helium from the refrigerator flows from the S Line to the six o'clock Valve Box and then back through CR to the compressor suction through a Thermax Heater. The flow path and the location of the instrumentation are given in Figure 6. A well defined flow rate through these lines and a more readable temperature difference due to lower flow offer two advantages to this flow set up. The control valves that feed the Recoolers and the makeup valves in both the Yellow and the Blue Boxes were both closed. No flow was allowed to return from the Return Line to the refrigerator. The flow rates through the S and CR lines are the same and were given by flow meter FI1102. The RHIC refrigerator was operated as a helium liquefier. Shield flows in the system were adjusted to keep the heat load in these lines the same as in normal operation. The flow rate was maintained at about 25 gram per second. The travel time of helium was 2.5 to 4 hours from the S line to the six o'clock valve boxes and was about 1.5 hours from CR back to the Refrigerator. During the test, the Supply temperature was kept essentially constant and the product of the flow rate to the enthalpy difference can be considered as the heat load of the lines.

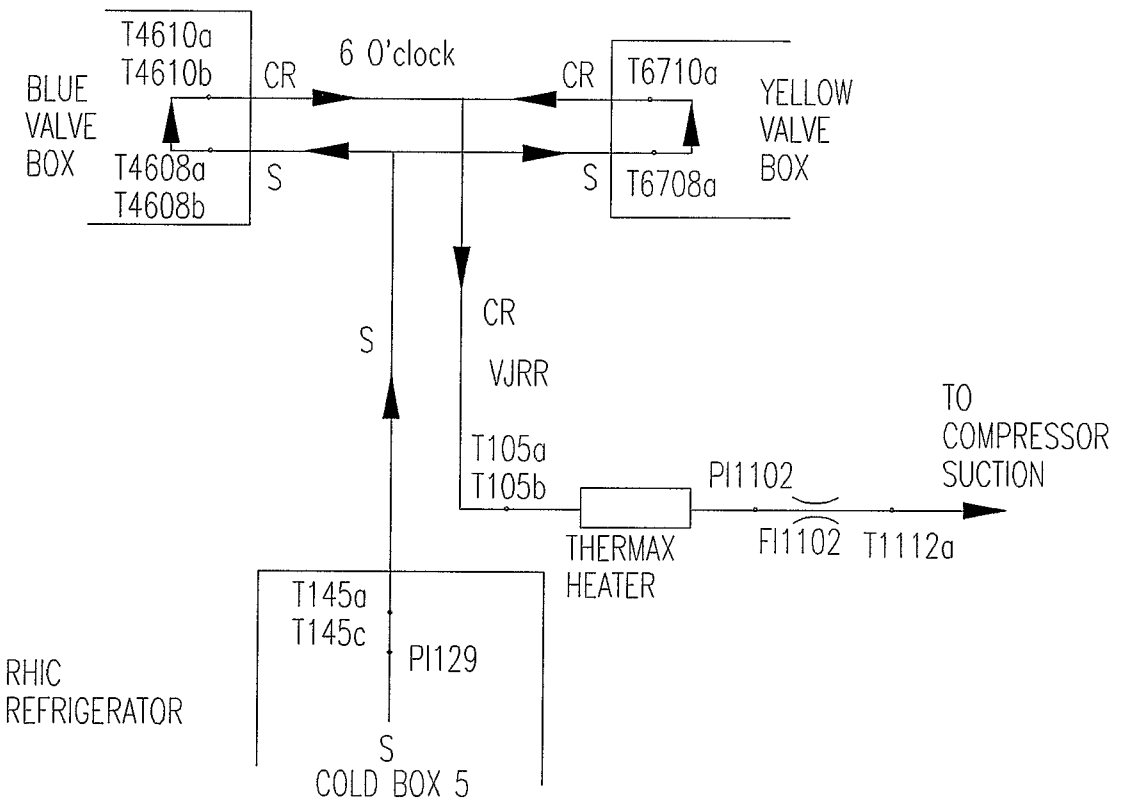


**Figure 6 Flow Path for Measurement of Heat Load in Supply and Cooldown Return**

As shown in Figure 7, the flow rate was between 24 and 26 g/s during the test. The temperature at both ends of the VJRR for the Supply Line are given in Figure 8. As shown by T145a in Figure 8, the temperature from RHIC refrigerator was stable at 4.6 K. The temperature in the Yellow Box, T6708a, was higher than that at the Blue Box, T4608a. The temperature at both ends of CR are given in Figure 9. As can be seen, the temperature at CR in the Yellow Valve Box, T6710a, was not only higher than that in the Blue Box, T4610a, but also higher than that of CR at the refrigerator, T105a. Since temperature increases with the direction of the flow, either temperature readings at the Yellow Box are inaccurate or the division of flow between the Yellow and the Blue Boxes is uneven. Assuming the temperature readings are accurate, then it appears most of the flow goes through the Blue Valve Box and only a smaller portion through the Yellow Box.

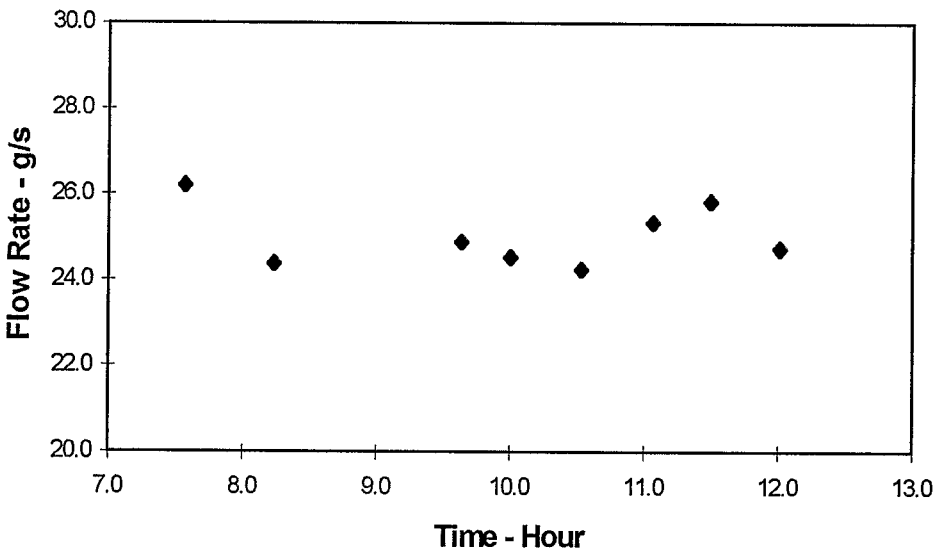


**Figure 7 Flow Rate through S and CR**

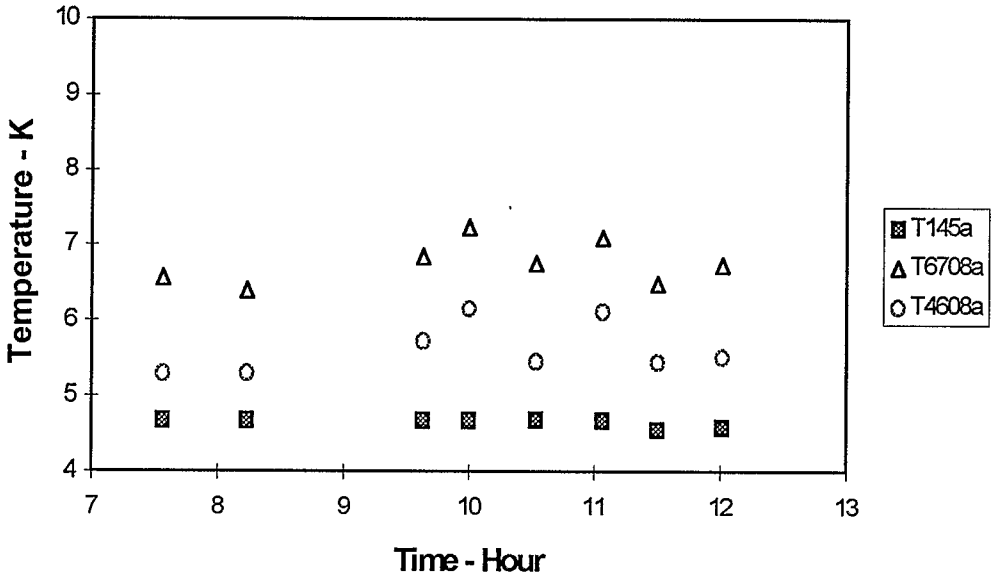


**Figure 8 Temperature at Ends of Supply Line**

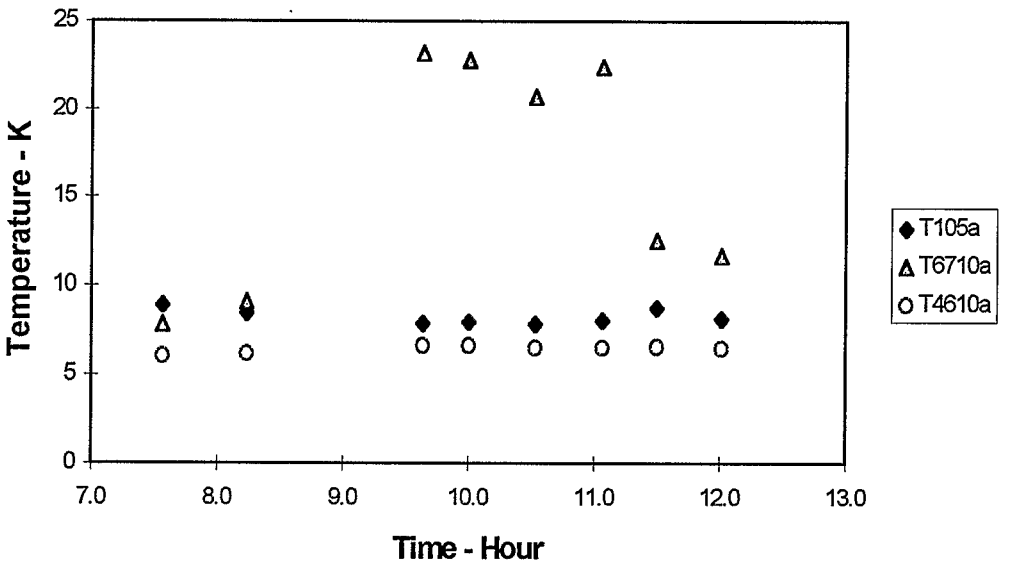


**Figure 9 Temperature at Ends of Cooldown Return**

Using the temperature readings from the Blue Valve Box, the heat loads in the S and CR lines are given in Figure 10 and 11. The average heat loads for S and CR are 134 and 479 watts respectively. If adjustment for an unequal division of flow is made, then the heat load will be higher for S and lower for CR. The estimated heat loads for S and CR are 180 and 260 watts respectively if these heat loads are proportional to the surface areas of CR and S. It must be emphasized that uncertainties associated with these results could be large since only eight sets of data were taken over a five hour period. The design heat load is 80 watts for the S line. The heat load for the CR line is not specified because it is not critical to the operation of RHIC.

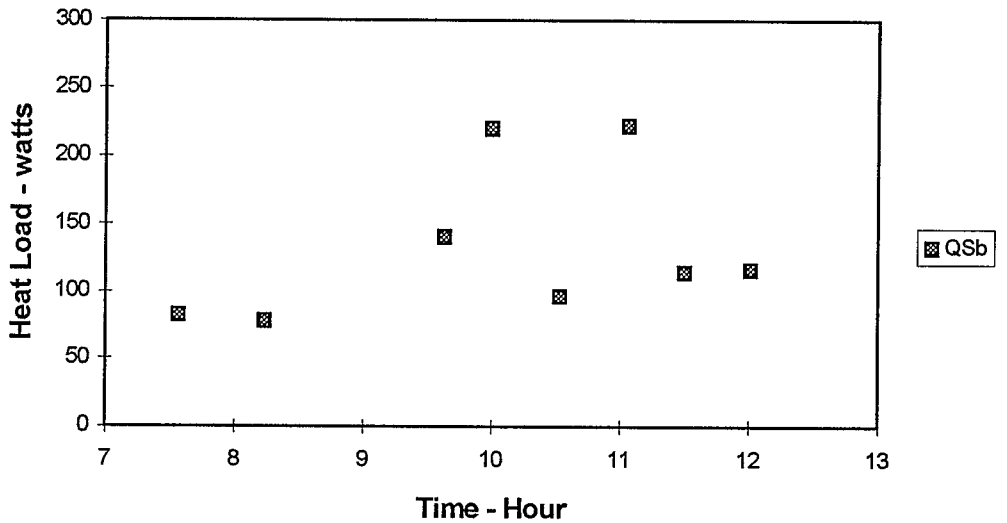


**Figure 10 Heat Load of Supply Line Using Temperature Sensor on Blue Box**

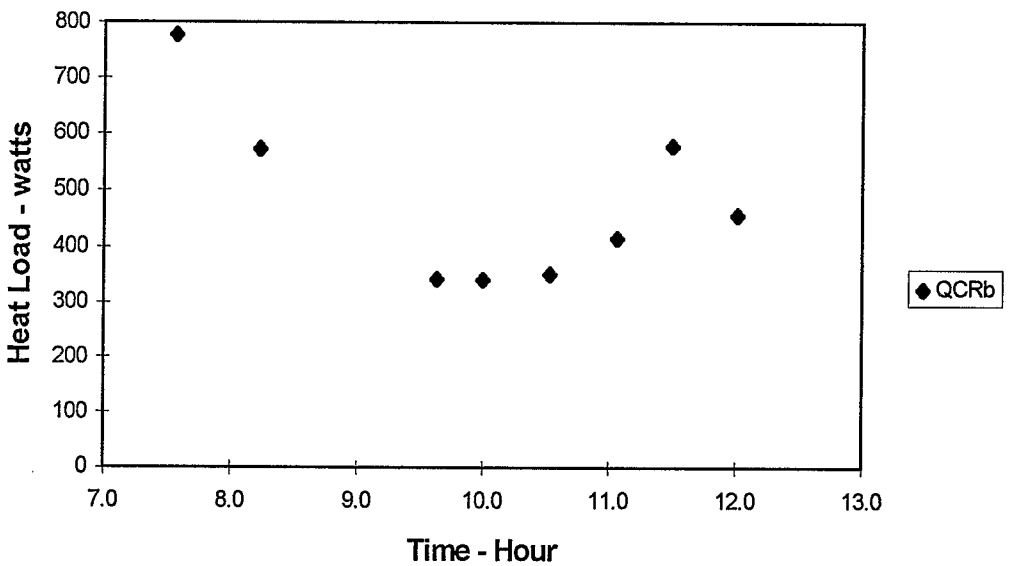


**Figure 11 Heat Load on CR Using Temperature Sensor on Blue Box**

## VI. Pump Curve for the Circulating Compressor

During the evening of July 19 and the early morning of the 20th, a test was setup with the circulating compressor running on a small bypass loop in the Blue Valve Box as shown in Figure 12. Several loop resistance were obtained using different amounts of shimming in the control valve. The compressor was operated over a speed range from 3,000 to 11,000 rpm.

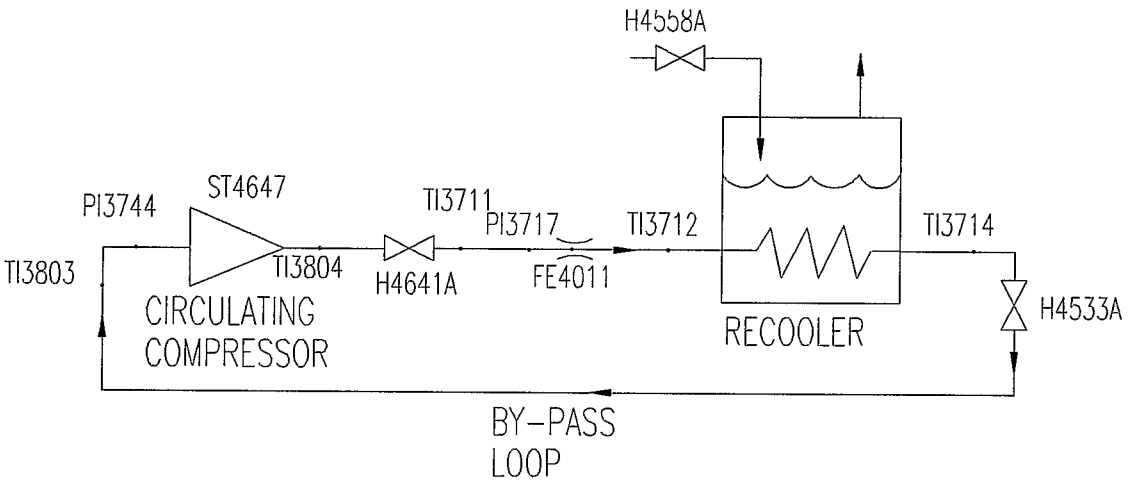


**Figure 12 Test Loop for Circulating Compressor**

The flow rate as a function of speed at different loop resistance is given in Figure 13. As can be seen, the flow increases linearly with speed for a given loop resistance. The pressure rise as a function of speed is given in Figure 14. The pressure rise is independent of the loop resistance and is proportional to the square of the pump speeds as show with a second order curve. The pressure rise as a function of flow rate at different speed and loop resistance is given in Figure 15. Over the operating range, the pressure rise depends mainly on the speed. Values of the head coefficient measured in these test as a function of the flow coefficient are shown in Figure 16 together with a curve obtained by the manufacturer using methanol. As shown, the test results agree well with the manufacturer's results until the flow coefficient reaches a value of about 1.1. According to the manufacturer, the maximum flow coefficient is about 1.4. Therefore the data point at the flow coefficient of 1.86 in Figure 16 is questionable. The head coefficient is rather flat for flow coefficients from 0.2 to 1.0. The efficiencies of the compressor were not evaluated because the temperature readings at the suction and discharge of the compressor were inaccurate.

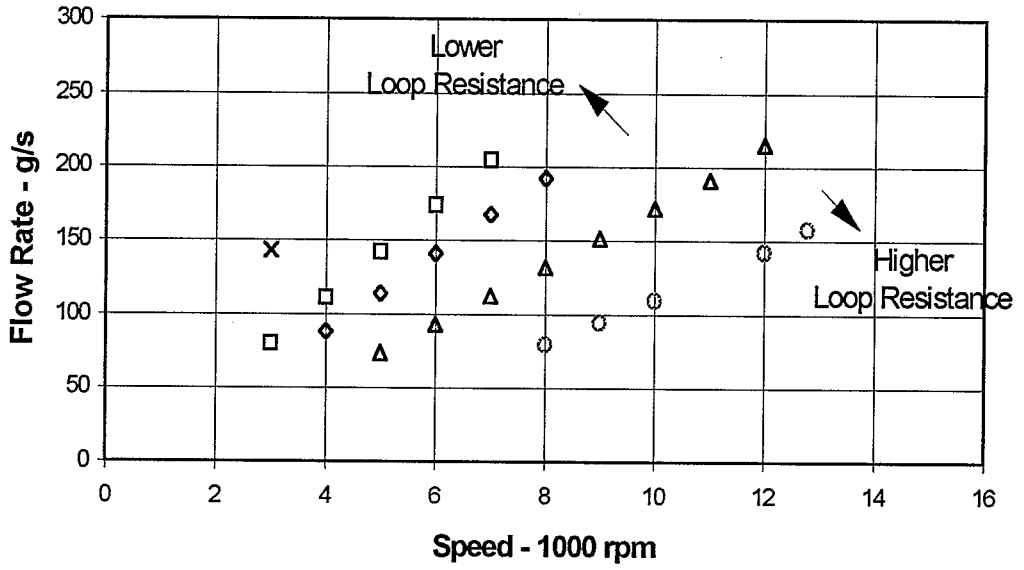


**Figure 13 Flow Rate as a Function of Compressor Speed at Various Loop Resistance**

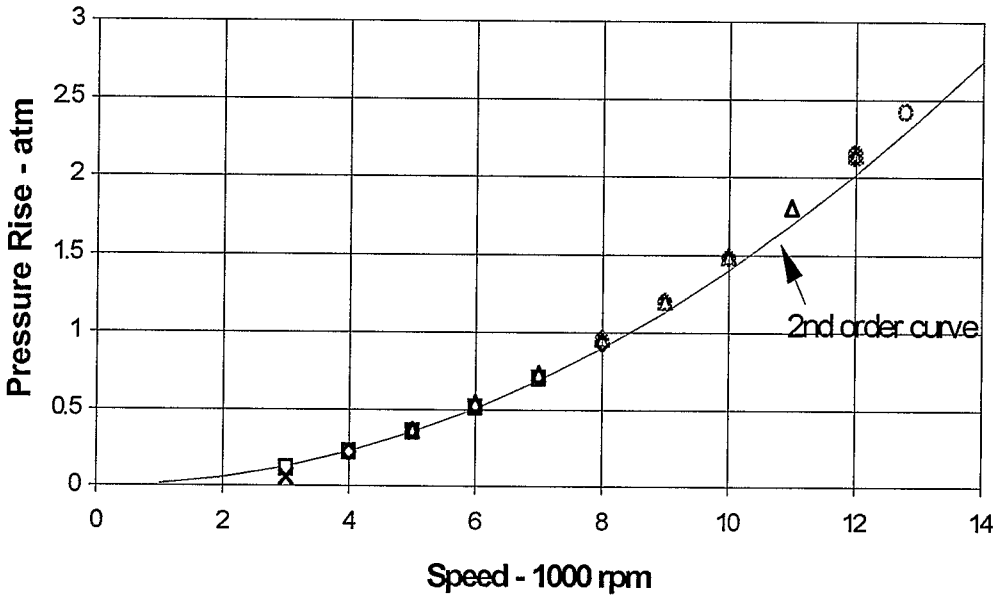


**Figure 14 Pressure Rise across the Compressor as a Function of Speed at Various Loop Resistance**

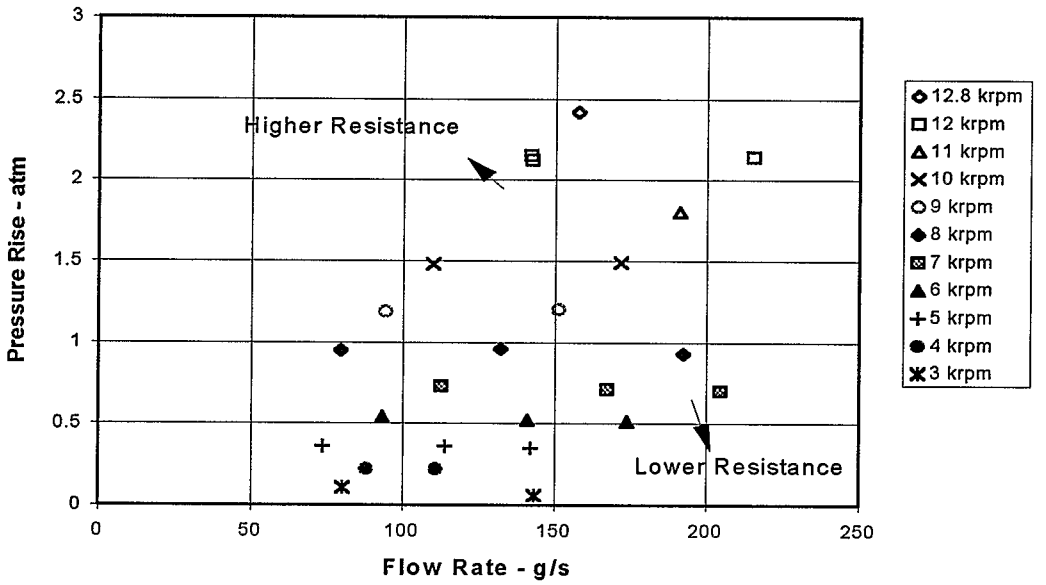


**Figure 15 Pressure Rise versus Flow at Various Speed and Resistance**

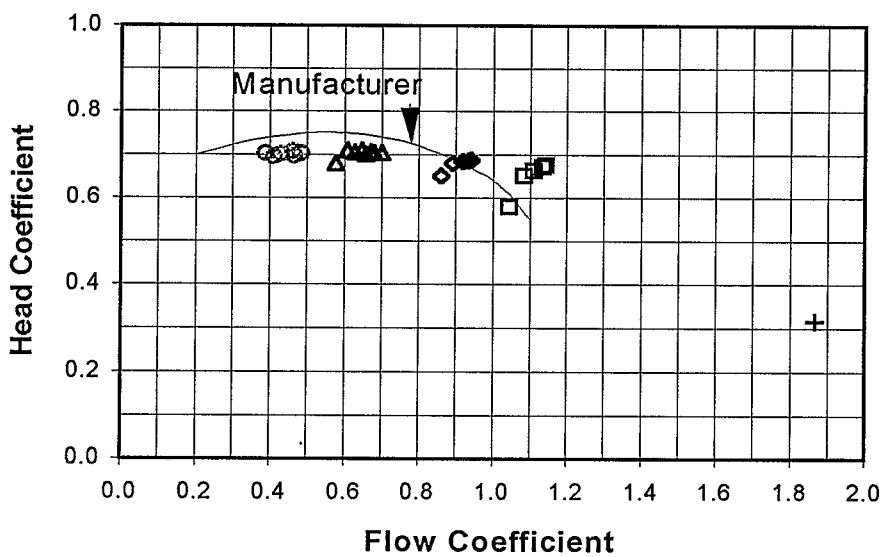


**Figure 16 Head Coefficient versus Flow Coefficient for the Circulating Compressor**

The performance curve obtained from the small by-pass loop agrees well with the manufacturer's results. However, the performance of the circulating compressor for actual circulation of cold helium in the larger loop inside the Blue Valve Box or from the Yellow Box to the RHIC tunnel is not consistent with that in the small loop. The pump performance in the large loop is more nearly represented by the data point with the flow coefficient of 1.86 in Figure 16 and appears to result from the smaller loop resistance of the system. Further investigation of the compressor performance will be made.

## V. Summary

Preliminary results for the heat loads of the shield and the S and CR lines have been presented. General speaking, the heat loads are greater than their design. More accurate measurements will be made in the first sextant test to verify the present results.

The pump curve obtained from the by-pass loop is in good agreement with the vendor's results. This suggests that the pressure and flow measurements are appropriate. However, the temperature measurements were not adequate to determine the compressor efficiency. The pump behavior in the actual loop was not consistent with the manufacturer's results and was probably due to the very low loop resistance in the system. The first sextant test will have a higher loop resistance and should provide a better test environment for performance evaluation.

## ACKNOWLEDGMENT

The author would like to thank A. Prodell for the valuable discussions and comments.