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Performance of RHIC Refrigerator III: Cold Vacuum Compressor

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ABSTRACT

In February 1996, the RHIC Refrigerator was successfully cooled to liquid helium temperature with 10 kilowatts of heat input at 4.5 K, 53 kilowatts of heat input at 60 K and 44 grams per second of liquid extraction. A comprehensive analysis was performed to evaluate the performance of the refrigerator including the turbines, the cold vacuum compressor and the heat exchangers. Because of the amount of data and the number of charts involved, the report is divided into five technical notes on, respectively: 1). Flowmeters, 2). Turbines, 3). Cold Vacuum Compressor, 4). Heat Exchangers and 5). Refrigerator Overall Performance. This technical note describes the performance of the Cold Vacuum Compressor.

I. Introduction

The RHIC Cold Vacuum Compressor is designed to reduce the pressure and temperature in the liquid helium pots of the refrigerator. The design speeds for producing suction pressures of 0.92, 0.80 and 0.65 atm are 9,400, 11,100 and 13,000 rpm respectively. In this test, the mechanical reliability of the compressor over a range including several operating points in the surge region was demonstrated. However the suction pressure did not reach the design value for the compressor. Most of the time, the cold vacuum compressor was operated at 10,330 rpm. The lowest suction pressure obtained with a "minimum" flow was 0.8 atm. Excessive pressure drops associated with the piping and isolation valves in the suction and discharge lines of the Cold Vacuum Compressor were found. The suction pressure and temperature should approach the design values when larger size valves and pipes are installed.

The performance of the Cold Vacuum Compressor is presented recognizing that there are large uncertainties in the data. The head and flow performance curves suggest that the capacity of the pump seems, for the most part, to agree with the manufacturer's curve. However, the efficiencies obtained are not meaningful.

II. PERFORMANCE CURVE

The performance of a centrifugal compressor follows the head and flow curve for the unit. The head coefficient is defined as the head rise divided by the square of the impeller tip speed. The flow coefficient is defined as ratio of the impeller exit meridional velocity to the impeller tip speed. The impeller tip diameter and the exit blade height equal 5.375 and 0.19 inches respectively.

In this technical note, only the pressure head is used in the head coefficient calculation. The pressure and temperature at the compressor discharge are used to calculate the impeller exit meridional velocity.

The pressure transducer at the suction of the compressor, PT218H, was re-calibrated after the test. PT218H was found to read high by 0.125 atm for all operating pressures in the test. The differential pressure transmitter DPT452H was accurate. The temperature sensors, TT119H and TT120H, are mounted respectively on the surfaces of the suction and the discharge lines. Their accuracy and response may not be the "best".

Flowmeters F41H and F211H are located in series in the discharge line of the Cold Vacuum Compressor. Between F41H and F211H, there is a by pass flow from the compressor discharge, through valve H40A, to the suction of the unit for surge prevention. During most of the test, a "small" flow is used all the time to maintain the associated piping cold. This by-pass flow is believe to be small since when the by pass valve H40A is closed, the suction pressure is found to decrease by only 7 inches of water. The performance of F41H and F211H has been examined in the technical note: Performance of RHIC Refrigerator I: Flowmeters. Generally speaking, F41H reads 14% higher than F211H. Since their accuracy can not be further determined, the flow coefficients have been evaluated based on both F41H and F211H. The head and flow curves for data taken on February 7 and 9 as well as during a final pump down with minimum flow are given in Figures 1 and 2.

The surge as indicated by a reduction of head coefficient with decreasing flow coefficient is believe to occur below the flow coefficient of 0.2 in Figure 1, and at 0.17 in Figure 2. The head coefficient of 0.6 at a flow coefficient just above the surge limit is very close to the design value.

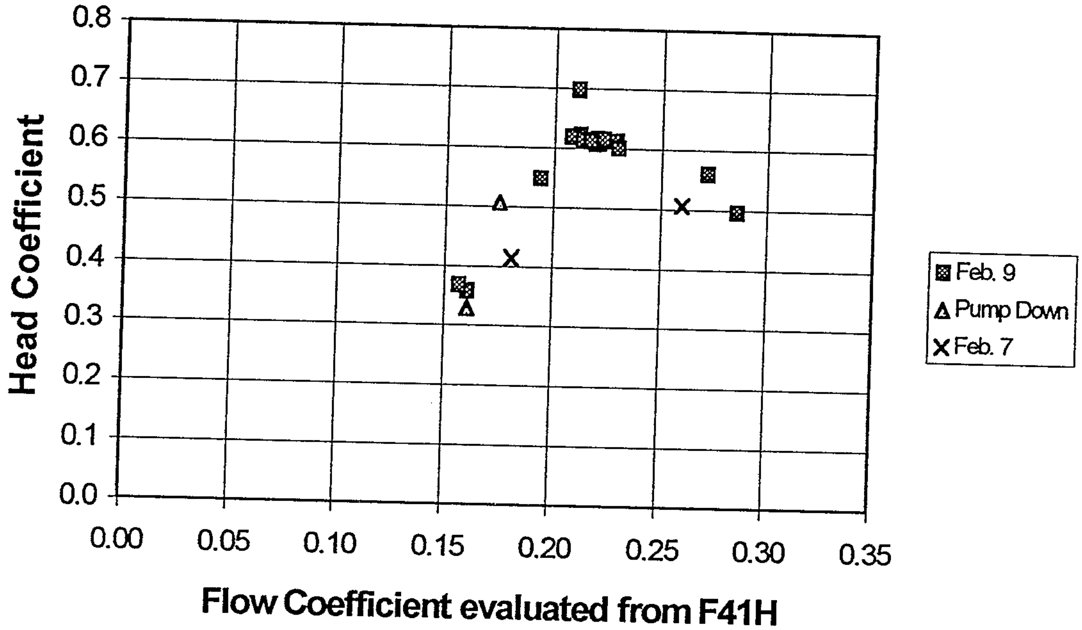

Figure 1 Head and Flow Curve evaluated from F41H for RHIC Cold Vacuum Compressor

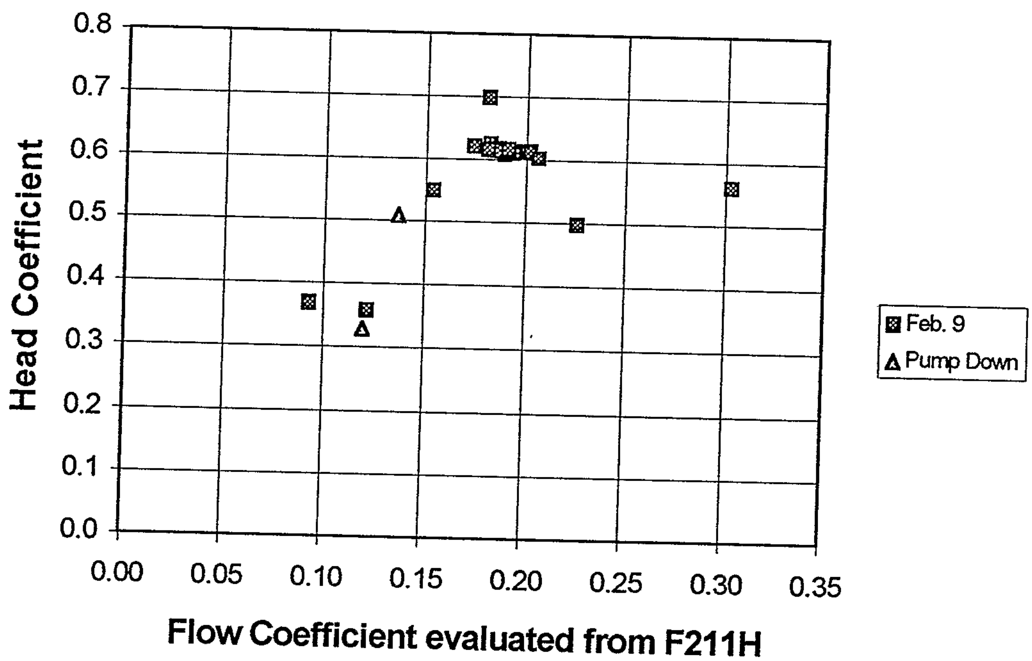

Figure 2 Head and Flow Curve evaluated from F211H for RHIC Cold Vacuum Compressor

III. Adiabatic Efficiency

The compressor efficiency is defined the ratio of enthalpy increase in an ideal process to that of actual process. The efficiency is plotted against the flow coefficient in Figures 3 and 4. As shown, there are great uncertainties in these results due primarily to insufficient resolution in the temperature read outs. The efficiencies obtained in the ambient air test range from 66 to 88 %. Heat leaks will reduce these efficiencies to some lower values perhaps close to those presented in Figures 3 and 4. For the time being Figures 3 and 4 should not be taken too seriously. For the next refrigerator run, two digits decimal display will be implemented and efficiencies for this compressor will be re-evaluated.

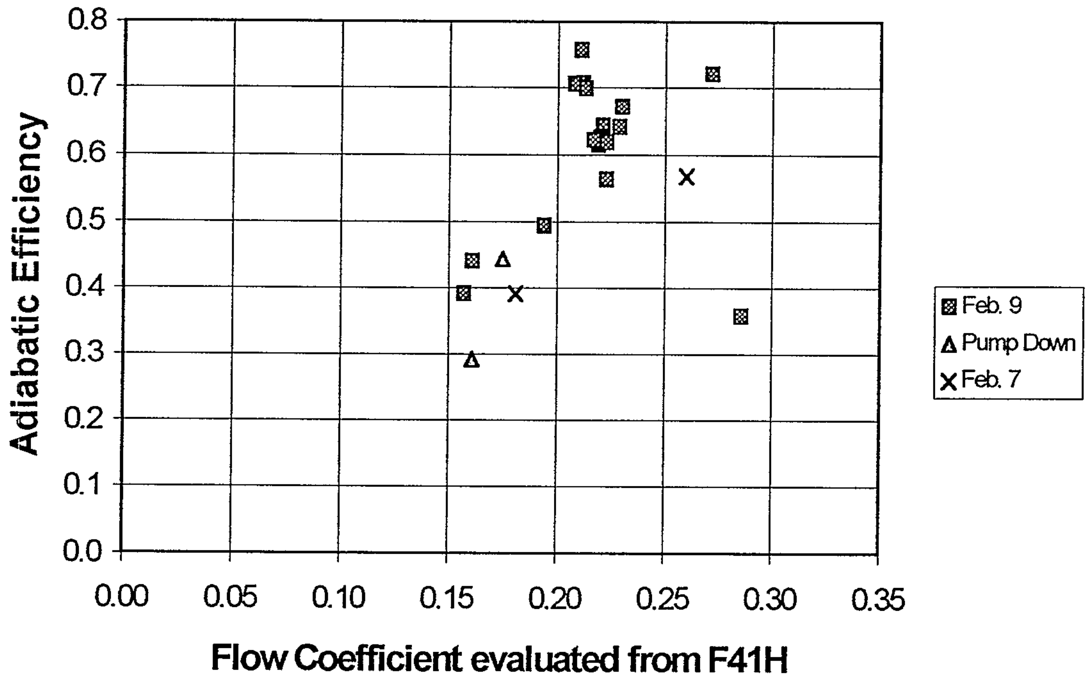

Figure 3 Efficiency as a function of flow coefficient based on F41H

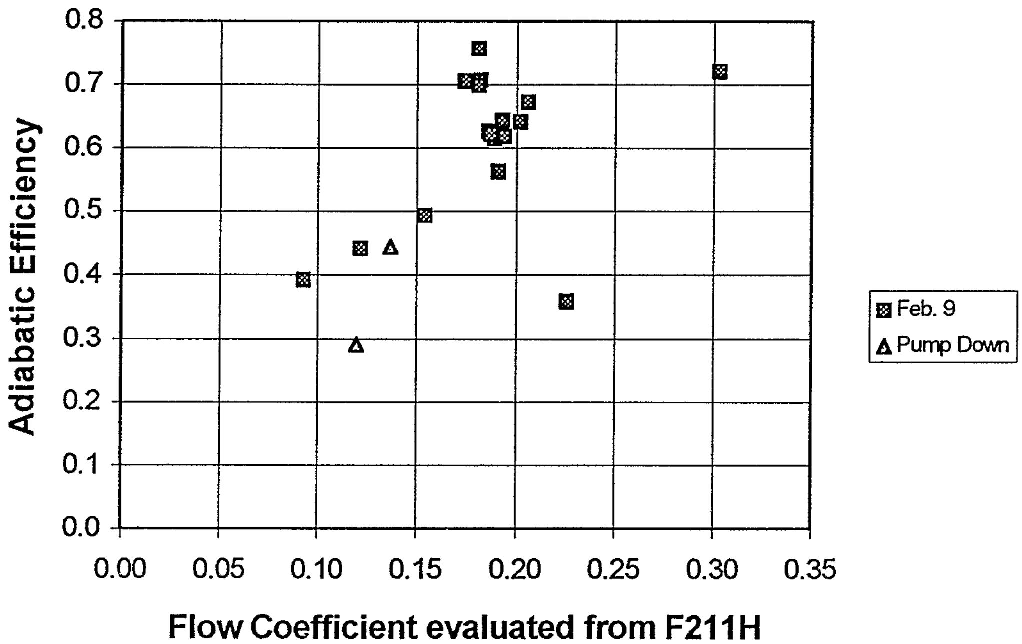

Figure 4 Efficiency as a function of flow coefficient based on F211H

IV. Pressure Drops

When pressures in the helium pots did not reach the design values, the pressure drops associated with the valves and piping in the suction and the discharge lines were investigated.

The compressor was running at 10,230 rpm with a suction temperature and pressure of 5.6 K and 1.04 atm. The compressor produced a pressure rise equivalent to 165 inches of water across the unit. The pressure drop from the discharge of the compressor to the next down stream pressure tap PT45 including a 2 inch isolation valve was equivalent to 62 inches of water. The pressure drop from the compressor suction to the pressure tap located up stream of PT63H including a 2 inch isolation valve and a filter was equivalent to 126 inches of water. If the 2 inch isolation valves are replaced by 3 inch valves and a larger filter is used, then the pressure drop will become equivalent to 15 inches of water at the discharge and 32 inches of water at the suction. The pressure drop in the helium pots will be reduced by about 0.35 atm.

V. Pressure Rise and Speed

The pressure rise across the compressor as a function of speed is given in Figure 5. As can be seen, the pressure rise is small when the speed is less than 8,000 rpm. The compressor design requires operating speeds of 11,000 and 13,000 rpm to reach suction pressures of 0.8 and 0.65 atm. The head rise as extrapolated from Figure 5 is 170 and 250 inches of water respectively for 11,000 and 13,000 rpm. To confirm the performance of the Cold Vacuum Compressor, it is necessary to operate this compressor at speeds from 10,000 to 13,000 rpm during the next refrigerator run.

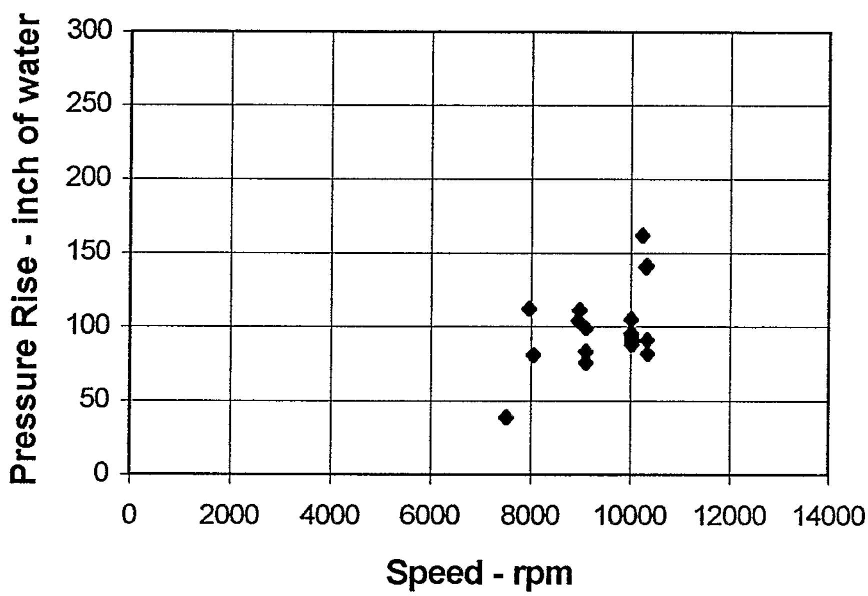

Figure 5 Pressure Rise for RHIC Cold Vacuum Compressor

VI. Summary

The performance of the Cold Vacuum Compressor has been investigated. The pressure drops associated with the suction and discharge lines are excessive and will be corrected. Better measurements shall be performed during the next refrigerator run.

ACKNOWLEDGMENT

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