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Performance of RHIC Refrigerator V: Overall Refrigerator Performance

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RHIC PROJECT

Brookhaven National Laboratory

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PERFORMANCE OF RHIC REFRIGERATOR V: OVERALL REFRIGERATOR PERFORMANCE

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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). Flowmeter, 2). Turbines, 3). Cold Vacuum Compressor, 4). Heat Exchangers and 5). Refrigerator Overall Performance. This technical note presents the Overall Performance of the Refrigerator.

I. INTRODUCTION

The performance of the flowmeters, the turbines, the cold vacuum compressor and the heat exchangers has been evaluated and results given in RHIC Technical Notes 99 to 102.[1,2,3,4] With a few exceptions, the available instrumentation provides adequate measurements for pressure, temperature and flow rate. Inadequate flow readings as identified in Technical Note 99 were modified for the evaluation of the refrigerator.

The capability of the refrigerator to provide the necessary cooling for RHIC has been demonstrated but the lowest temperature in the helium pots is 0.2 K above the design value. The test results are very encouraging. The energy balance in each section of the refrigerator is good. For one particular condition analyzed, good agreement between the test results and the computer simulation was obtained.

II. REFRIGERATOR OVERVIEW

On the process control computer, the RHIC refrigerator is divided into three pages of display as shown in Figures 1 to 3. Figure 1 shows the warm end of the refrigerator including heat exchangers HX 1 through HX 4, turbines EX 1 and EX 2, and a Shield Calorimeter. Figure 2 shows the middle section of the refrigerator including heat exchangers HX 5 through HX 9 and turbines EX 3 and EX 4. Figure 3 shows the cold end of the refrigerator including the High Pot, Intermediate Pot, Low Pot, turbine EX 5, Cold Vacuum Compressor C2, the 4 K Calorimeter and a penetration for extraction of cold helium to the Thermax Heater.

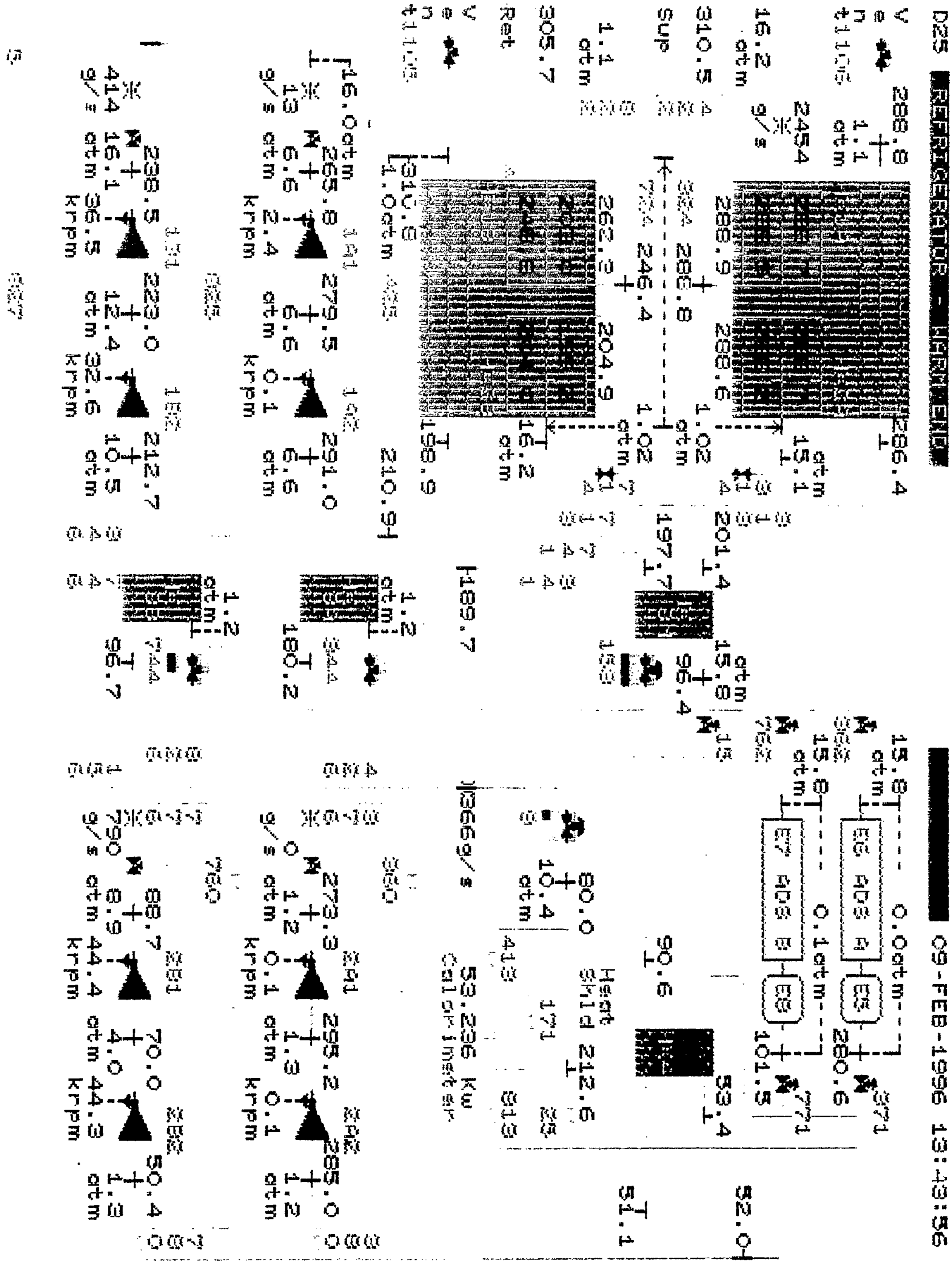
The compressor system which provides the helium to the refrigerator is given in Figure 4. In Figure 4, the twenty first stage compressors, four second stage compressors, the helium flow supply to the refrigerator, amount of by-pass flow and total electrical power are displayed.

In principle, one can simply use the heat inputs to the refrigerator and the power input to the compressor for efficiency evaluation. In this test, data were mainly taken for the refrigerator and only a few pages were taken for the compressor. Therefore the compressor power needed to be calculated from estimated flow rates and efficiencies. In the next run, a comprehensive data acquisition system will be employed so that all available data at any instant can be saved for later analysis. The process variations resulting from data taken from individual pages can be eliminated.

For this test, the raw data used for the performance evaluation of the refrigerator were taken from printouts when the refrigerator was in "Steady" condition. The calculations for the flow through the refrigerator starts with the flow through the cold end and continue toward the warm end as in the evaluation of heat exchangers.[4] It should be noted that there is discrepancy between the total flow evaluated from the sum of individual flowmeters in the refrigerator and the flow measured by the Venturi flowmeter located at the discharge of the compressors.[1] The accuracy of the performance evaluation is thus limited by the uncertainties of the flow measurements.

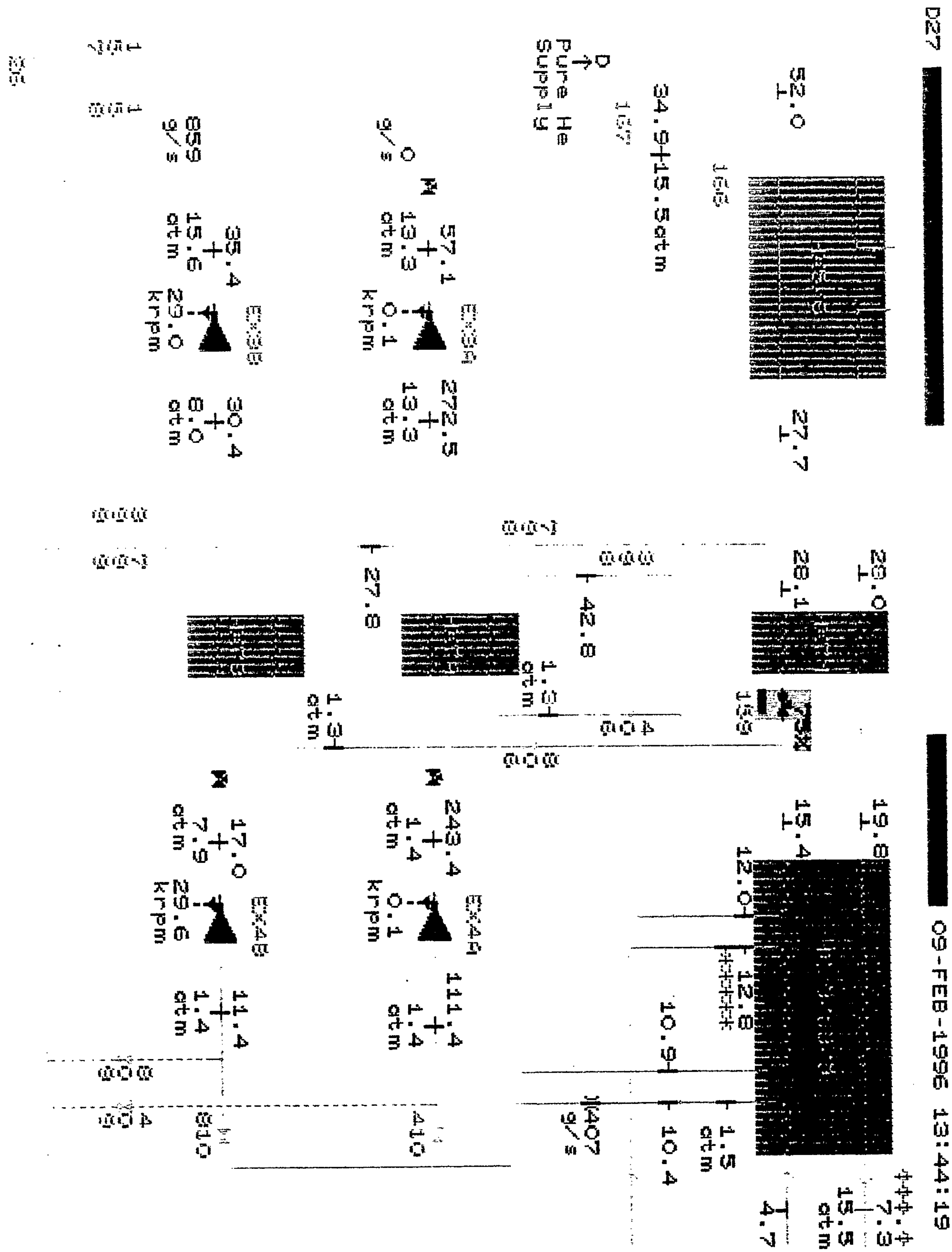
III. OPERATING CAPACITY

The heat load allowance for RHIC is given in Tables 3-1 through 3-5 of the RHIC Design Manual. The 4 K allowance is 10,342 watts and the 55 K allowance is 37,159 watts. The 55 K is the average of the supply temperature, 40 K, and the return temperature, 70 K, for the heat shield. In Table 3-1, the 4 K heat load of 10,342 watts consists of 6747 watts of refrigeration and 45 g/s of liquefaction. One gram per second of liquefaction is converted to 80 watts of 4 K refrigeration in the Design Manual. The purpose of refrigerator performance test was to demonstrate that the capacity of RHIC refrigerator meets the requirements of the Design Manual.



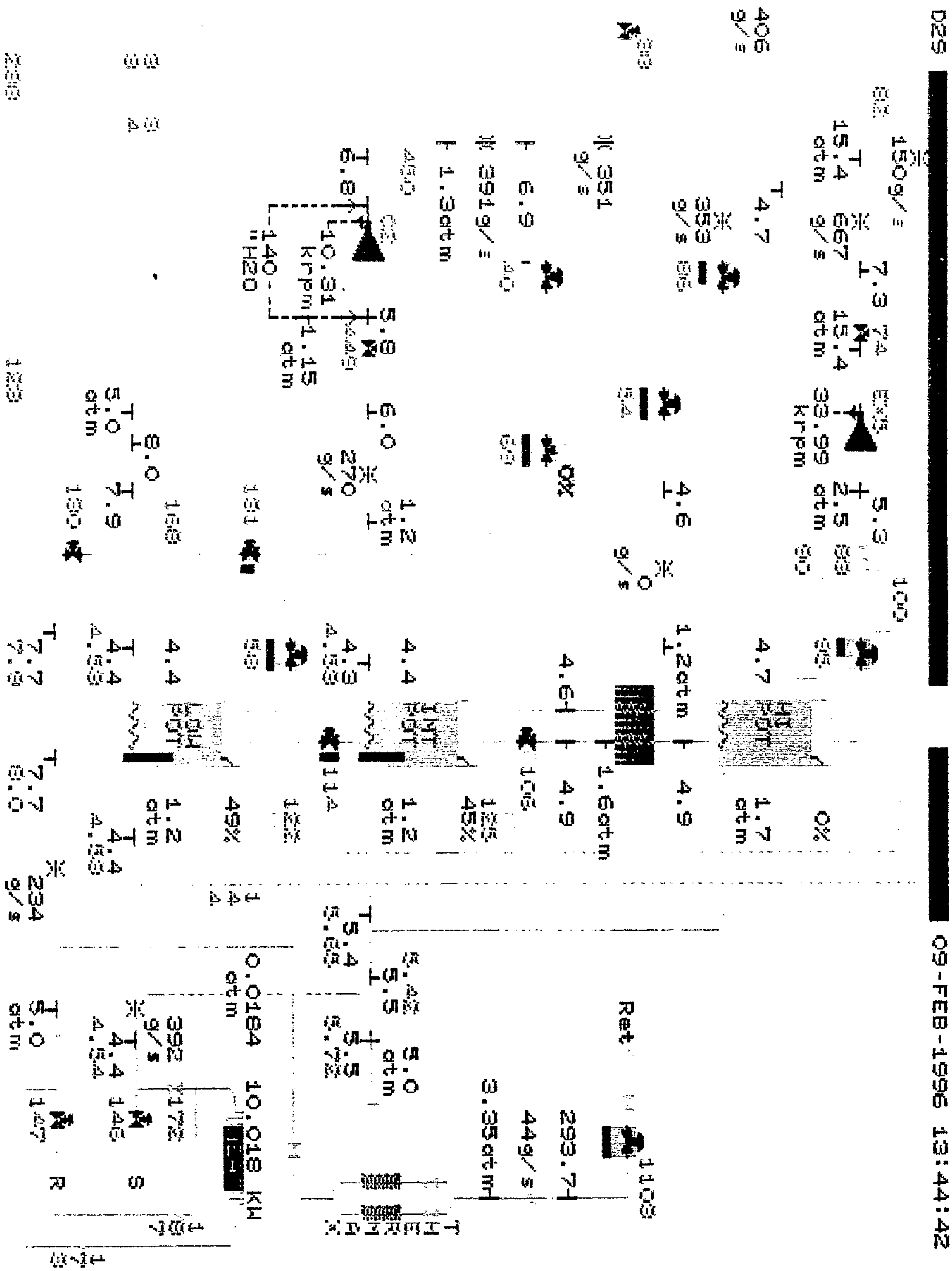
Screen 25, from CRT1, 09-FEB-96 14:02:45 Refrigerator - Warm End

FIGURE 1. Warm End of RHIC Refrigerator



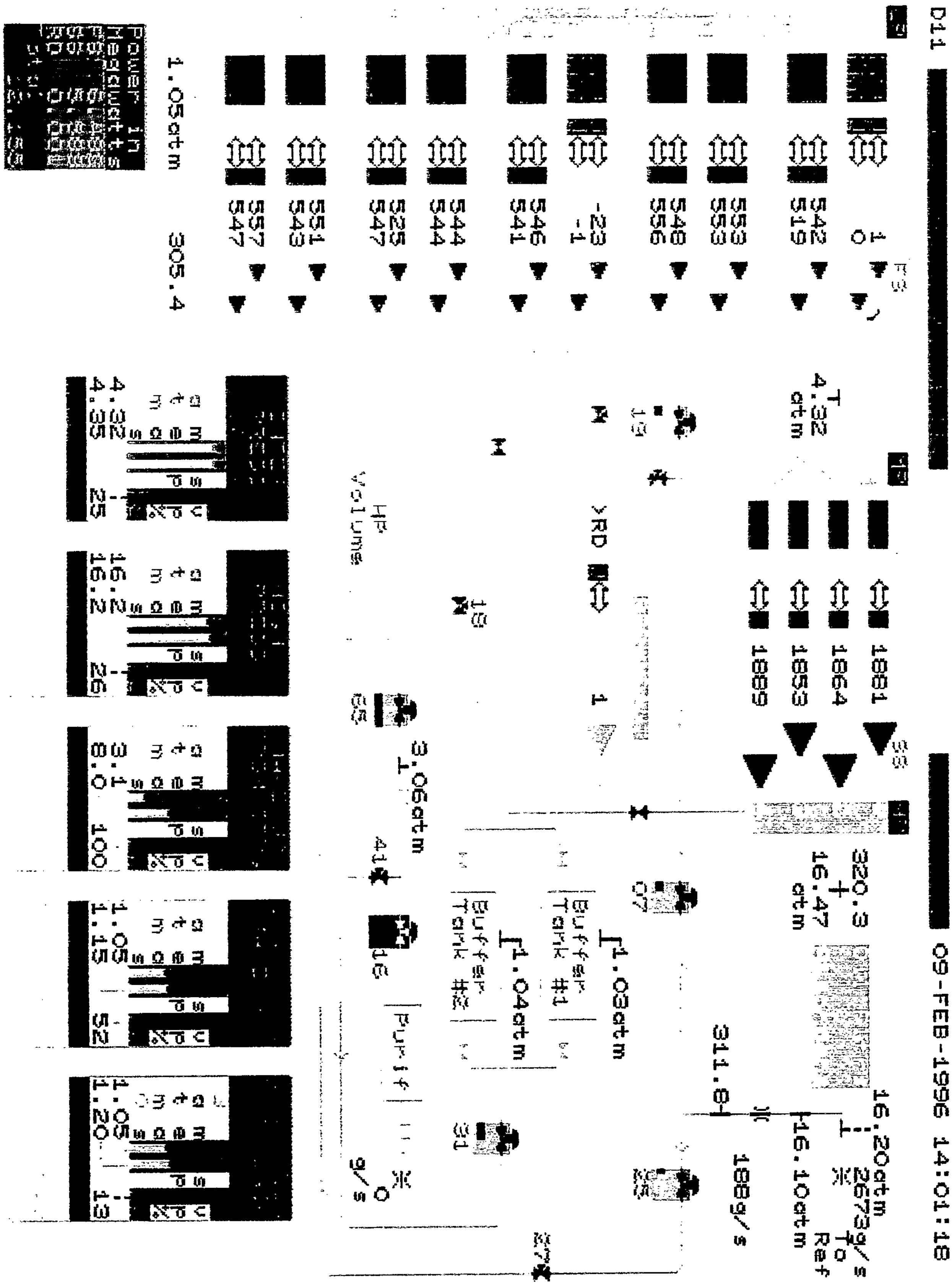
Screen 27, from CRT1, 09-FEB-96 14:03:08 Refrigerator - Middle

FIGURE 2. Middle Section of RHIC Refrigerator



Screen 29, from CRT1, 09-FEB-96 14:03:32 Refrigerator - Cold End

FIGURE 3. Cold End of RHIC Refrigerator



Screen 11, from CRT1, 09-FEB-96 14:20:07 Compressor Room Control

FIGURE 4. RHIC Compressor System

As shown in Figure 3, a refrigeration capacity of 10,018 watts and 44 g/s of liquefaction at 4.4 K were obtained. At the same time, a cooling capacity of 53,236 watts at about 65 K for the heat shield were measured as given in Figure 1. These capacities are about 2,000 watts or 30 % more than the 4 K load and 43 % more than the 55 K load. The refrigerator still has excess capacity because both the turbines and the compressors were not fully loaded.

The capacities as a function of time for the February 9 run are given in Table 1. In the morning, the cold helium extraction and the heat input to the 4 K Calorimeter were increased gradually. After a 40 g/s liquefaction rate and 10,000 watts refrigeration capacity were reached, heat was gradually applied to the 55 K Shield Calorimeter electrically. As can be seen from Table 1, the refrigerator was operated above the RHIC requirement for at least two hours between 10:58 and 13:44. After 13:44, the compressor system tripped a few times due to insufficient helium gas inventory. The test run was terminated at about 16:00 to be ready for the scheduled laboratory-wide power shut down at 18:00.

Table 1. Capacities obtained on February 9, 1996

Time hr	min	4 K Qcal watt	Lead Flow g/s	Total 4 K Load watt	55 K Qshld kilo-watt	Modified F1 g/s
0	15	0	19	1520	0	2363
0	34	1300	29	3620	0	2340
0	53	3810	37	6770	0	2419
1	3	5568	41	8848	0	2375
1	20	8272	37	11232	0	2311
1	29	9042	40	12242	0	2152
1	42	10183	40	13383	0	2166
2	38	10159	39	13279	0	2278
3	18	10134	40	13334	0	2322
3	55	10061	40	13261	12.82	2274
5	5	9976	40	13176	12.747	2266
5	38	9908	40	13108	14.921	2307
6	37	9927	40	13127	16.19	2348
9	28	10037	40	13237	33.626	2205
10	19	10079	40	13279	38.535	2216
10	58	10061	39	13181	48.449	2236
13	44	10018	44	13538	53.236	2812
15	27	10085	30	12485	34.847	2744

IV. TEMPERATURE REACHED

The temperatures reached in the Low and the Intermediate Pot of the refrigerator are about 4.3 K and are higher than their design values by approximately 0.2 degree Kelvin. This is an indication that either the Cold Vacuum Compressor was not able to reduce the pressures in these pots or the pressure drop is excessive in the refrigerator. Near the end of the run, a test was performed to determine the pressure drops in the suction and the discharge lines of the Cold Vacuum Compressor.

The pressures in these two pots are about 1.1 atm and are higher than their design values by 0.2 to 0.3 atm (80 to 120 inches of water). At the operating speed of 10,000 rpm, the typical pressure rise across the Cold Vacuum Compressor equals 0.4 atm. The pressure drop in the suction line including a 2 inch globe valve and a filter equals 0.3 atm. The pressure drop in the discharge line including a 2 inch globe valve is 0.15 atm. These pressure drops consume all the pressure head the Cold Vacuum Compressor produces. If these pressure drops were eliminated, the temperatures in the liquid helium pots would decrease accordingly. Currently 4 inch pipes and valves have been installed in both the suction and the discharge lines and the temperatures in the liquid helium pots should approach the design values in the next test.

V. ENERGY BALANCES

The calculation of the energy balance is of great interest in establishing the confidence in the data analysis. In the present set up it is convenient to calculate the energy balance in the cold, the middle and the warm sections of the refrigerator respectively.

The rate of energy entering each refrigerator section, Qin, equals the enthalpy flux entering the section plus the electric heat input minus the turbine work extraction. The rate of energy leaving a refrigerator section, Qout, equals the enthalpy flux leaving that section. Qin, Qout and the ratio of Qin to Qout for the warm, the middle and the cold sections are given in Tables 2 to 4.

In Table 2, Qin and Qout are within a few per cent of each other suggesting that the measurements and the assumptions used in the analysis are adequate in the warm section.

In Table 3, Qin and Qout are within a few per cent of each other suggesting again that the measurements and the assumptions used in the analysis are also adequate in the middle section.

Table 2. Energy balance in the warm end of the refrigerator

Energy balance in the warm end of the refrigerator

Time		based on	T2	
hour	min	Qin	Qout	Qin / Qout
		watts	watts	
0	14	4162164	4262662	0.976
0	33	4124562	4225919	0.976
0	52	4219400	4307735	0.979
1	3	4062820	4169862	0.974
1	19	3985417	4077686	0.977
1	28	3682029	3775977	0.975
1	41	3660200	3752569	0.975
2	37	3969301	4051787	0.980
3	17	3883356	3989394	0.973
3	53	3903118	3974153	0.982
5	5	3841226	3913419	0.982
5	36	3961599	4054511	0.977
6	36	4223488	4309113	0.980
9	27	3751526	3823305	0.981
10	15	3814198	3847583	0.991
10	49	3821880	3885268	0.984
13	43	5397889	5446557	0.991
15	26	5124390	5198628	0.986

Table 3. Energy balance in the middle section of the refrigerator

Energy balance in the middle section of the refrigerator

Time				
hour	min	Qin	Qout	Qin / Qout
		watts	watts	
0	14	426561	441609	0.966
0	33	426135	443332	0.961
0	52	445376	459996	0.968
1	3	426218	437084	0.975
1	19	414917	425197	0.976
1	28	362833	372019	0.975
1	41	345389	354982	0.973
2	37	378923	397541	0.953
3	17	345254	361336	0.955
3	53	389436	402275	0.968
5	5	383125	395861	0.968
5	36	380554	392726	0.969
6	36	418572	429117	0.975
9	27	345007	354464	0.973
10	15	340832	352131	0.968
10	49	340145	351343	0.968
13	43	659550	665134	0.992
15	26	589175	592804	0.994

In Table 4, Qin and Qout vary from a few per cent to 27 % suggesting that either the measurements may not be as accurate as or assumptions may not be as applicable to those used for the warm and the middle sections. Some of the contributing factors to these discrepancies may be: 1) the one digit decimal display on the temperature and the pressure readings in this test, 2) the overcorrecting of turbine flow and work to EX 5 and 3) inability to account for liquefaction in the flow rate. While one can not do much about item 3, two digit decimal displays have been implemented for all temperatures measurements below 10 K and for all pressure measurements below 5 atm. A transducer sufficient for the EX 5 flow measurements has been installed to prevent the transducer from exceeding its full scale. These changes should provide more accurate data for the energy balance calculation in the future.

Table 4. Energy balance in the cold end of the refrigerator

Energy balance in the cold end of the refrigerator

Time hour	min	Include Qin watts	F229 Qout watts	Qin / Qout
0	14	26825	37040	0.724
0	33	30932	35912	0.861
0	52	30238	36259	0.834
1	3	30246	36367	0.832
1	19	32787	37455	0.875
1	28	29637	33344	0.889
1	41	29460	33433	0.881
2	37	28604	32287	0.886
3	17	24273	25116	0.966
3	53	23898	25126	0.951
5	5	23998	24942	0.962
5	36	23611	24724	0.955
6	36	24836	28331	0.877
9	27	25063	25679	0.976
10	15	24552	26215	0.937
10	49	25031	26565	0.942
13	43	45714	51563	0.887
15	26	45054	51245	0.879

VI. CARNOT EFFICIENCY

The Carnot Efficiency is defined as the ratio of the theoretical minimum work to the compressor input power. The theoretical minimum work is calculated from the exorgy changes of helium through the load. Exorgy equals the product of ambient temperature and entropy minus enthalpy at a given temperature. The total exorgy change for helium in this test equals the sum of exorgy changes from the 4 K heat load of the main Calorimeter, the cold helium extraction and the 55 K heat input of the Shield Calorimeter.

The estimated power inputs are calculated from the isothermal compression of helium with an assumed 50 % efficiency. The flow rate is based on the modified flow reading from flowmeter F1. It should be remembered that the modified flow from F1 is about 7 % less than the total flow evaluated from the sum of individual flowmeters in the refrigerator.[1] In addition, the compressor by-pass flow which accounts for 5 to 7 % of the total flow is neglected in this study. Thus the measured power input could be 12 % to 14 % higher than that calculated. As seen from Table 5, the Carnot efficiency is low when the refrigerator is operated at low capacity. As the load increases, the efficiency increases. Near the two RHIC design load conditions, the calculated efficiencies are 10.7 % and 13.1 %. Note that the present configuration introduces the 4 K heat input at 5 atm pressure. Therefore the Carnot efficiency is slightly lower than that of a conventional refrigerator/liquefier.

Table 5. Summary of Carnot work and estimated refrigerator efficiency

Time hour	min	4 K Load watts	Carnot Liquefaction watts	work 55 K Load watts	Total watts	Calculated Input power watts	Carnot efficiency %
0	14	0	130363	0	130363	8254044	1.6
0	33	77711	189620	0	267331	8173986	3.3
0	52	193732	221362	0	415094	8162894	5.1
1	3	271079	236597	0	507676	8012087	6.3
1	19	380969	197821	0	578790	7799411	7.4
1	28	420042	213464	0	633507	7261920	8.7
1	41	448880	205123	0	654003	7308322	8.9
2	37	439732	198727	0	638459	7687273	8.3
3	17	441585	204027	0	645612	8109939	8.0
3	53	438404	204027	141745	784176	7619823	10.3
5	5	436764	204908	148705	790377	7592939	10.4
5	36	433787	204908	162928	801623	7731201	10.4
6	36	434619	204908	165504	805031	8148321	9.9
9	27	435312	203165	238573	877050	7389385	11.9
10	15	437133	203165	264942	905240	7427791	12.2
10	49	448796	201105	329343	979244	7493082	13.1
13	43	472645	243464	292129	1008238	9424922	10.7
15	26	464166	170538	265050	899754	9194484	9.8

VII. COMPARISON WITH SIMULATION

The results of a calculation for the refrigeration process with a heat load of 10,000 watts on the 4 K calorimeter, 44 g/s of cold helium extraction and 55 kilo-watts on the shield calorimeter is given in Figure 5. This is approximately the same load as the condition of the refrigerator at 13:43. The calculated process needs 2801 g/s compressor flow with 9.92 Megawatts input power. A comparison between the calculated and the test results is given in Table 6.

Table 6. Comparison between the calculated results and measurements at 13:44

	Calculation	Test Values
Load		
4 K Calorimeter	10,000 watts	10,018 watts
	(4.2 - 6.73 K)	(4.4 - 7.7 K)
Lead Flow	44 g/s	44 g/s
	(6.73 K)	(7.7 K)
55 K Calorimeter	55,000 watts	53,236 watts
	(40 - 70 K)	(53 - 80 K)
Flow through		
EX 2	820 g/s	790 g/s
EX 3 and 4	935 g/s	859 g/s
EX 5	523 g/s	1030 g/s*
Flow through 4 K Calorimeter	601 g/s	353 g/s
Flow leakage in the cold end	0 g/s	150 g/s
Sum of above flows	2801 g/s	3182 g/s*
Modified F1		2812 g/s
Compressor by-pass flow	0 g/s	188 g/s
Power input	9.92 Mw	12.2 Mw

In Table 6, the flow through EX 5 is extrapolated from the speed of the turbine. The original data corresponded to a differential pressure greater than the full scale. The raw flow data is 667 g/s and the extrapolated flow is 1030 g/s. By using the extrapolated value for flow through EX 5, the test flow is about 13 % higher than that calculated. If the actual flow is less than the extrapolated value of 1030 g/s, the test results would be closer to the calculated results. Elimination of flow leakage in the cold end of the refrigerator could also improve the plant efficiency. Current new valve seats have been installed on all leaky valves identified.

PROGRAM RHIC

CALCULATE PERFORMANCE OF RHIC HELIUM REFRIGERATOR OPERATED
WITH CALORIMETERS AND VAPORIZER.
5 EXPANDERS AND 1 COLD VACUUM COMPRESSOR ARE USED.

SUMMARY OF SYSTEM PARAMETERS

REFRIGERATION-WATTS		MASS FLOW-G/S
Qcal	QSHLD	F74
10000.	55000.	44.

ESTIMATED HEAT LEAKS IN THE HEAT EXCHANGERS - WATTS

HX1	HX2	HX3	HX4	HX5	HX6	HX7	HX8	HX9	H. POT	I. POT	L. POT
950.	3240.	2670.	1880.	240.	410.	600.	520.	180.	340.	290.	290.

HEAT EXCHANGER PARAMETERS

HEAT EXCHANGER	HIGH P FLOW G/S	LOW P FLOW G/S	CMAX/ CMIN	EFFECTIVENESS RATIO	REQUIRED AU KW/K	NTU	DESIGN AU KW/K
1.0	2801.4	2757.4	1.017	.977	459.5	32.09	684.0
2.0	2405.5	2757.4	1.134	.936	106.8	8.54	183.1
3.1	2405.5	2366.4	1.023	.980	411.9	33.50	533.8
3.2	395.9	390.9	1.023	.980	66.8	32.90	103.8
4.0	2405.5	2757.4	1.116	.949	135.5	10.60	280.4
5.0	1981.2	1937.2	1.079	.925	91.4	9.03	201.2
6.0	1046.1	1937.2	1.625	.801	15.4	2.55	58.4
7.1	1046.1	1050.6	1.168	.917	38.1	6.78	127.8
7.2	935.1	886.6	1.168	.917	32.0	6.75	148.5
8.1	1046.1	1937.2	1.465	.331	3.5	.46	33.7
8.2	1046.1	1373.8	1.079	.954	54.8	7.66	61.8
9.0	1046.1	438.8	1.553	.947	12.1	3.87	11.5

EXPANDER PARAMETERS

TURBINE	PIN ATM	POUT ATM	TIN K	TOUT K	FLOW G/S	ETA	WORK W
1.0	16.23	9.00	180.00	152.55	396.	.73	57394.
2.0	8.86	1.30	64.61	38.69	820.	.75	111643.
3.0	15.58	8.00	25.00	21.01	935.	.65	18512.
4.0	7.92	1.41	12.01	7.50	935.	.63	15439.
5.0	15.49	2.50	5.85	5.21	523.	.50	2543.

COMPRESSOR PARAMETERS

COMP.	ISOTHER. EFF.	ADIABATIC EFF.	PIN ATM	POUT ATM	TIN K	TOUT K	FLOW G/S	WORK KW	WORK H.P.	IN VOL FL ACFM	IN DENSI. G/CC	PRES RATIO
MAIN	.50		1.05	17.25	302.0	305.0	2801.	9918.	13295.			
COLD		.60	.92	1.40	4.14	5.13	563.3	2.584	3.464	77.0	.0155	1.523

ESTIMATED OPEATING SPEED FOR COLD VACUUM COMPRESSOR 10186. RPM
ESTIMATED SURGE SPEED IS 14036. RPM
Flow Coefficient = .207 Head Coefficient = .519

LOAD SUMMARY

	PRIMARY LOAD		SECONDARY LOAD	
	SUPPLY	RETURN	SUPPLY	RETURN
FLOW RATE-G/S	523.12	479.12	424.34	424.34
PRESSURE-ATM	5.01	5.00	15.67	9.67
TEMPERATURE-K	4.20	6.73	40.00	64.58
ENTHALPY-J/G	11.04	30.15	222.00	351.61

Figure 5. Process Calculation of RHIC Refrigerator with the same Loads as the Test

VIII. SUMMARY

The capacity of the RHIC refrigerator has been shown to meet the RHIC heat load allowance. The efficiency has been calculated. Good agreements between the calculations and tests have been obtained. A few problems associated with the refrigerator have been identified and corrected. Improvements on plant efficiency are expected prior to the next refrigerator run.

ACKNOWLEDGMENT

The author would like to express his appreciation for the help provided by M. Iarocci and all personnel in the RHIC Cryogenic Section, and the valuable discussions and comments from A. Prodell.

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