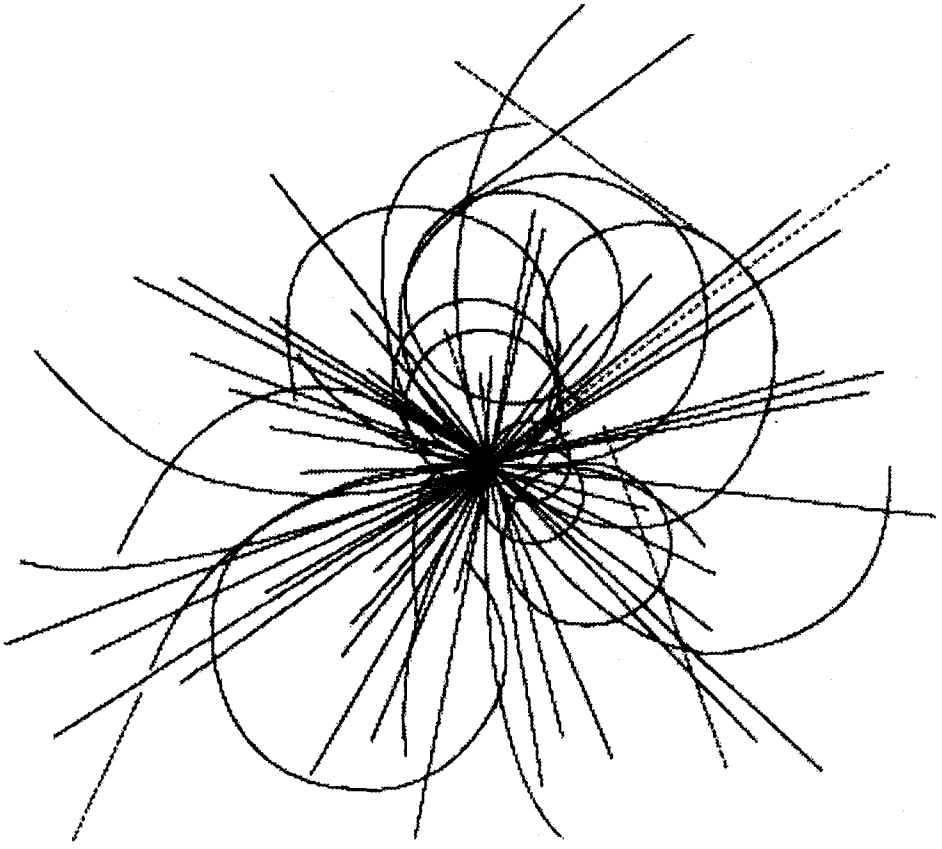
SSCL-Preprint-488

SSCL-Preprint-488
July 1993
Distribution Category: 400

V. Ganni
T. Apparao

Design Verification and Acceptance Tests of the ASST-A Helium Refrigeration System



Superconducting Super Collider Laboratory

SSCL-Preprint-488

Design Verification and Acceptance Tests of the ASST-A Helium Refrigeration System*

V. Ganni and T. Apparao

Superconducting Super Collider Laboratory†
2550 Beckleymeade Ave.
Dallas, TX 75237

July 1993

*Presented at the International Cryogenics Engineering Conference, July 12–14, Albuquerque, N.M.
†Operated by the Universities Research Association, Inc., for the U.S. Department of Energy under Contract No. DE–AC35–89ER40486.

DESIGN VERIFICATION AND ACCEPTANCE TESTS OF THE ASST-A HELIUM REFRIGERATION SYSTEM

V. Ganni and T. V. V. R. Apparao

Cryogenics Department, Accelerator Systems Division
Superconducting Super Collider Laboratory*
Dallas, TX 75237

INTRODUCTION

Three similar helium refrigerator systems have been installed at the Superconducting Super Collider Laboratory (SSCL) N15 site: the ASST-A system, which will be used for the accelerator system's full cell string test; the N15-B system, which will be used for string testing in the tunnel; and a third plant, dedicated to magnet testing at the Magnet Testing Laboratory. The ASST-A and N15-B systems will ultimately be a part of the collider's N15 sector station equipment. Each of these three systems has many subsystems, but the design basis for the main refrigerator is the same and is given in Reference 1. The flow diagram of the ASST-A system, the subject of this report, is shown in Figure 1. Each system has a guaranteed capacity of 2000 W of refrigeration and 20 g/s liquefaction at 4.5 K.

The testing and design verification of the ASST-A refrigeration system consisted of parametric tests on the compressors and the total system. A summary of the initial performance test data is given in Reference 2. The tests were conducted for two cases: in the first (Case 1), all four compressors were operating; in the second (Case 2), only one compressor in each stage was operating. In each case, tests were conducted in three modes of operation described later on.

The process design basis supplied by the manufacturers and used in the design of the main components—the compressor, and expanders and heat exchangers for the coldbox—were used to reduce the actual test data using process simulation methodology. In addition, the test results and the process design submitted by the manufacturer were analyzed using exergy analysis. This paper presents both the process and the exergy analyses of the manufacturer's design and the actual test data for Case 1. The process analyses are presented in the form of T-S diagrams. The results of the exergy analyses comparing the exergy losses of each component and the total system for the manufacturer's design and the test data are presented in the tables.

* Operated by the Universities Research Association, Inc., for the U. S. Department of Energy under Contract No. DE-AC35-89ER40486.

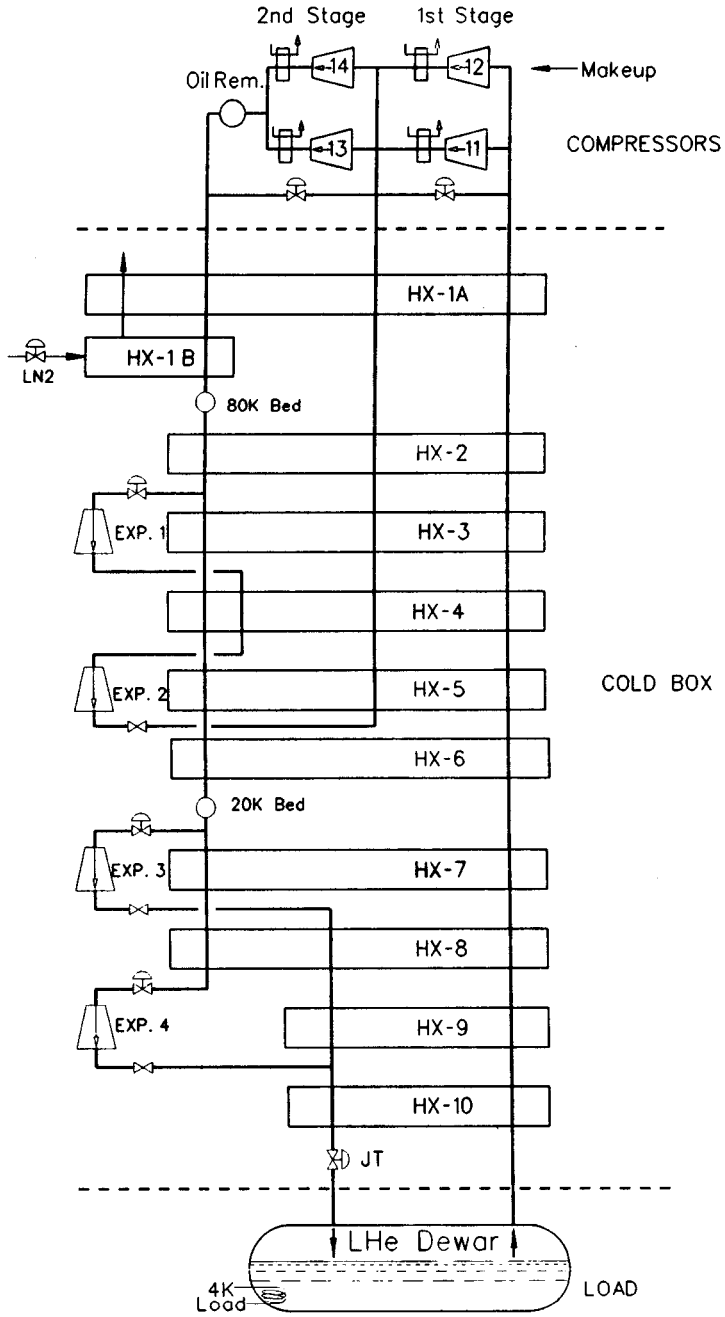


Figure 1. Refrigeration system flow diagram.

COMPRESSOR SYSTEM

The compressor system consists of two first-stage and two second-stage Sullair screw compressors. Prior to the total system test, parametric tests were conducted to map each compressor. The volumetric and isothermal efficiencies (the latter include the motor efficiency) for one first-stage and one second-stage compressor are given in Figure 2. The efficiency values provided by the manufacturer (Sullair) and used in the design are shown in Figure 2, as are the efficiency values reduced from the test data for the three modes.

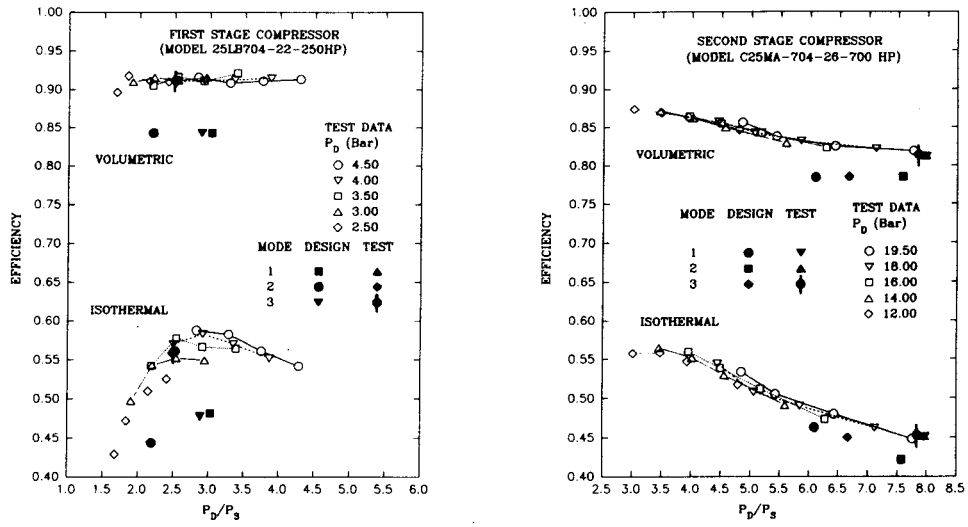


Figure 2. Compressor characteristics.

COLDBOX SYSTEM

The coldbox contains 11 heat exchangers grouped into six brazed aluminum cores, housed together with four expanders, two 80-K beds, one 20-K bed, and associated valves, piping, and instrumentation. For comparison of the manufacturer's design with test data of the processes for the three modes, see Figures 3–5.

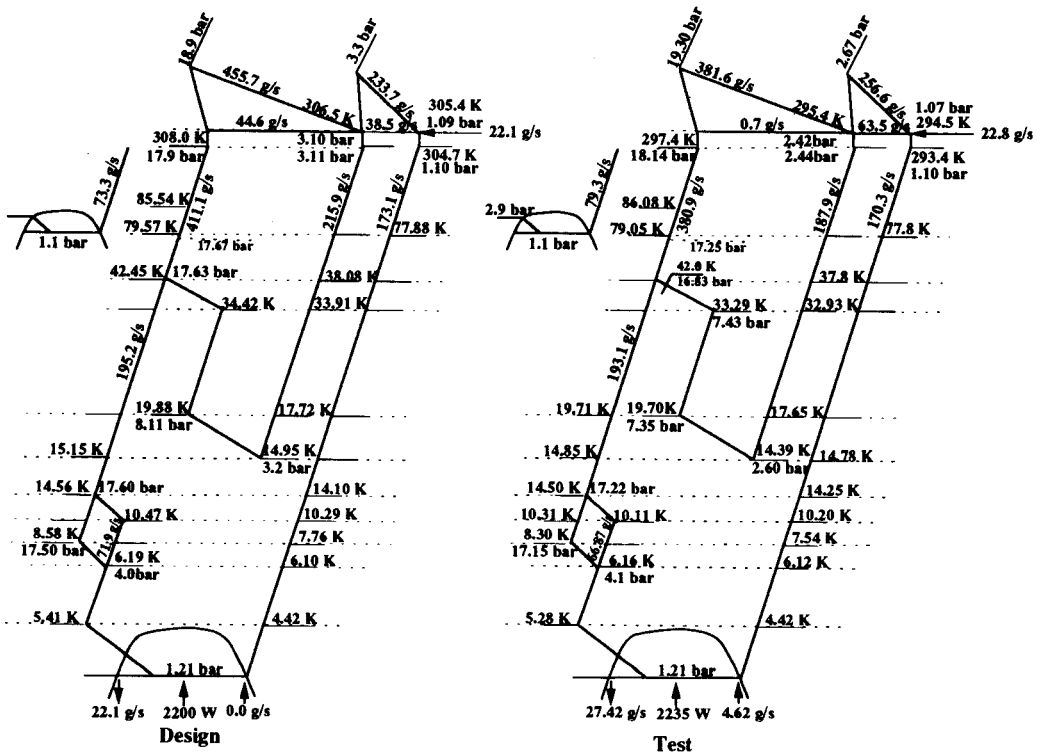


Figure 3. Mode 1 (50L-50R) process analysis (110% of guaranteed capacity).

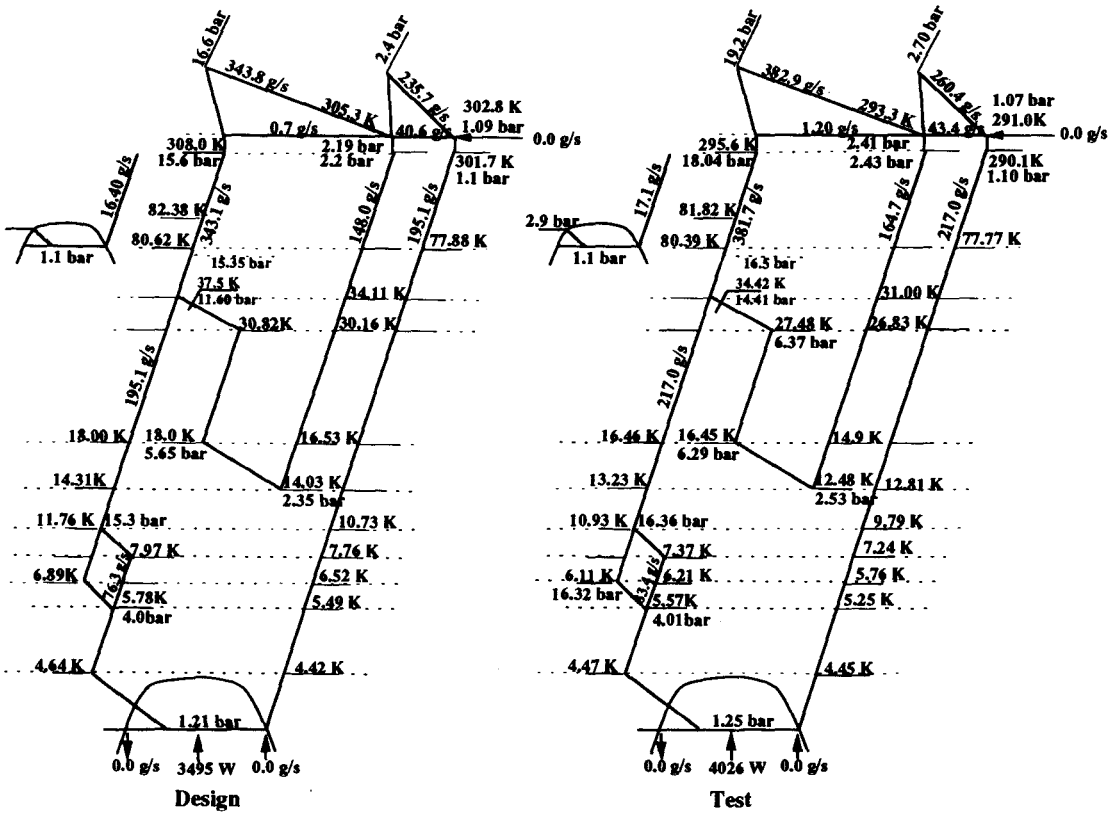


Figure 4. Mode 2 (100R) process analysis.

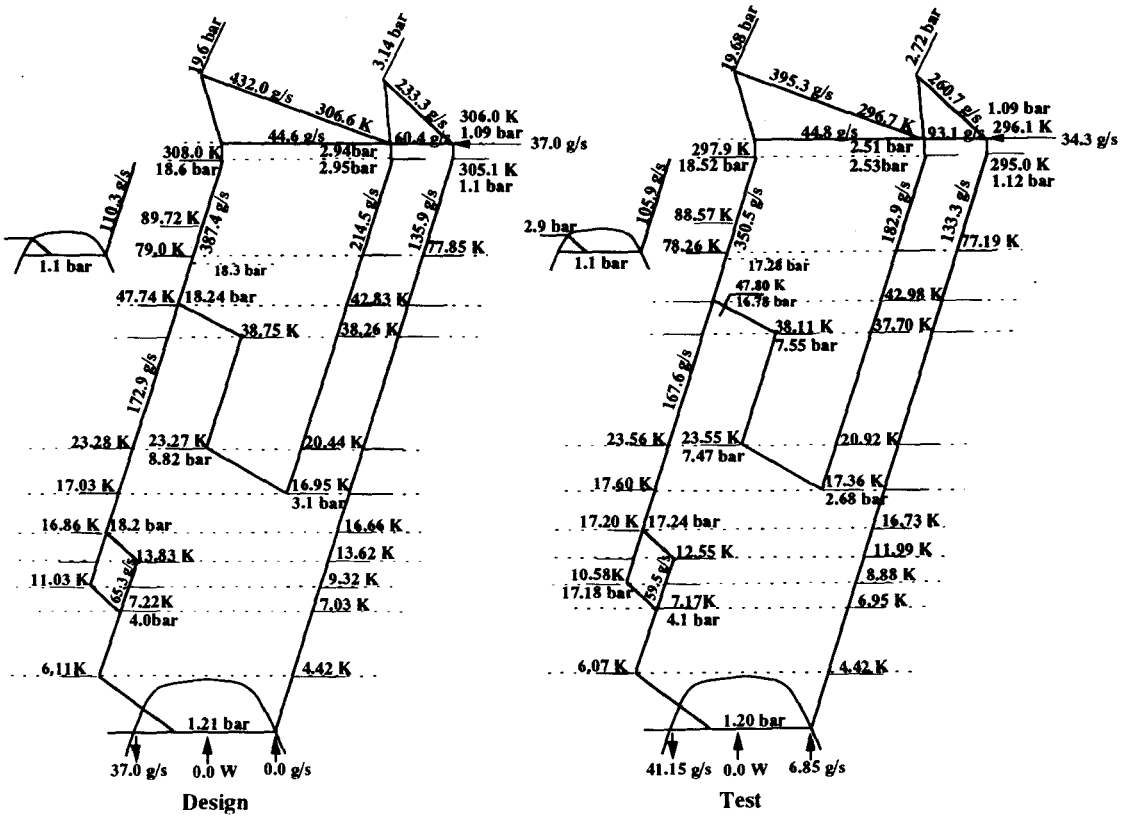


Figure 5. Mode 3 (100L) process analysis.

LOAD

The refrigeration load is applied in the 40,000-L liquid helium dewar with electric heaters. The liquefaction rate is measured by makeup gas and is verified with the depletion of the helium gas from the gas tanks and observation of the accumulated liquid in the dewar.

THEORY

The process analysis was performed using the characteristics of the subcomponents of the system supplied by the manufacturers. The present work uses the theory of exergy analysis as applied to refrigeration systems.[3] The theory shows the importance of irreversibilities and how to minimize them. It shows the importance of dt/t and dp/p parameters in designing these systems. The maximum available specific work function or exergy is defined as:

$$de = dh - t_0 \times ds,$$

where e is exergy (j/g), h is enthalpy (j/g), t_0 is sink temperature (K), and s is entropy (j/g – K).

An example of application of this theory to helium systems is given in Reference 4.

TEST MODES OF THE SYSTEM

Preliminary tests were conducted to ascertain that the system has the guaranteed capacity, and no attempt was made to find the maximum capacity or the efficiency of the system so long as it exceeded the guaranteed capacity and operated close to the design basis. During the process simulations, small adjustments were made to the test data when they did not satisfy the mass and energy balances and/or the expander flow characteristics, to compensate for instrument resolution and accuracy limitations and errors. In some modes of operation, additional cooling occurred in the outdoor oil removal system due to low ambient temperature, and therefore the cooling water temperature was not used as the sink temperature, t_0. Instead, t_0 was assumed to be the average of the coldbox warm end inlet and outlet temperatures for the process analysis and exergy calculations.

For the compressor system, the exergy losses due to motor inefficiencies, non-ideal compression, and the cooling provided in the aftercooler and oil cooler are treated as a single loss. The exergy loss due to the pressure drop across the first-stage aftercooler and the second-stage aftercooler and oil removal system are treated separately to show the pressure drop effect on the system input power.

In the coldbox, the losses are: (1) the heat exchangers, whose losses are due to temperature differences, dt, between streams and heat leak; (2) the expanders, whose losses are due to the inefficiencies in expansion and wasted expander output; and (3) miscellaneous losses, which include nitrogen system losses, pressure drops, JT expansion, and heat leaks. During testing, there was an approximately 1-bar pressure drop across the 80-K adsorber bed filter. Tests were conducted with this pressure drop and modifications to the filter are under way to reduce it. The exergy analysis shows the effect of this pressure drop on the input power.

Mode 1 (50L-50R)

This mode is the system design basis. In this mode the system is required to produce 2000 W of refrigeration and 20 g/s of liquefaction at 4.5 K—approximately 50% liquefaction and 50% refrigeration load on the exergy basis. The system is designed with a 10% margin.

Figure 3 and Table 1 compare the results of manufacturer's design with the actual test data for this mode of operation.

Mode 2 (100R)

In this mode all the available system capacity is used for 4.5-K refrigeration. The expected design refrigeration capacity is given as 3495 W and the system was tested at 4025 W. The higher capacity was achieved, due primarily to the compressors having higher than assumed volumetric efficiency and available motor capacity, resulting in higher discharge pressure and flow. Figure 4 and Table 2 compare the results of the manufacturer's design with the actual test data.

Mode 3 (100L)

In this mode all the available system capacity is used for liquefaction. The expected design liquefaction capacity is given as 37 g/s and the system was tested at 34.3 g/s. The system is expected to produce the predicted capacity once the 80-K bed filter pressure drop is minimized and the second-stage compressor is operated at a lower suction pressure to minimize the bypass flow. Figure 5 shows the process analysis and Table 3 provides the exergy analysis for this mode.

Table 1. Mode 1 (50L-50R) exergy analysis.

	DESIGN		TEST	
4.5 K Refrigeration load (W)	**2200.0**		**2235.0**	
Liquefaction load (g/s)	**22.1**		**22.8**	
Reliquefaction load (g/s)	**0.0**		**4.62**	
Coolant reference temperature (K)	306.4		295.4	
INPUT	**kW**	**%**	**kW**	**%**
First-Stage Compressor	340.6	20.9	257.1	17.2
Second-Stage Compressor	1132.3	69.7	1078.5	72.3
LN_2 System (Eff. Carnot = 0.35)	152.6	9.4	156.9	10.5
INPUT EXERGY TOTAL	**1625.5**	**100.0**	**1492.5**	**100.0**
OUTPUT	**kW**	**%**	**kW**	**%**
First-stage: compressor and motor	176.3	10.8	113.0	7.6
First-stage bypass	26.6	1.6	34.2	2.3
First-stage suction mixing	4.7	0.3	7.7	0.5
First-stage aftercooler DP	7.4	0.5	13.4	0.9
First-stage subtotal	**214.9**	**13.2**	**168.3**	**11.3**
Second-stage compressor and motor	607.9	37.4	592.0	39.7
Second-stage bypass	51.0	3.1	0.8	0.1
Second-stage suction mixing	4.4	0.3	4.0	0.3
Second-stage AC and oil rem. DP	11.9	0.7	11.6	0.8
Second-stage subtotal	**675.3**	**41.5**	**608.4**	**40.8**
COMPRESSORS - SUBTOTAL	**890.2**	**54.8**	**776.7**	**52.0**
Heat exchanger 1A	47.1	2.9	47.3	3.2
Heat exchanger 1B	6.5	0.4	6.3	0.4
Heat exchanger 2	26.4	1.6	22.3	1.5
Heat exchanger 3	4.7	0.3	4.4	0.3
Heat exchanger 4	23.0	1.4	19.1	1.3
Heat exchanger 5	7.0	0.4	7.1	0.5
Heat exchanger 6	0.9	0.1	0.6	0.0
Heat exchanger 7	2.4	0.1	1.5	0.1
Heat exchanger 8	5.6	0.3	5.0	0.3
Heat exchanger 9	2.3	0.1	2.2	0.1
Heat exchanger 10	20.3	1.2	18.9	1.3
Heat exchangers - subtotal	**146.2**	**9.0**	**134.7**	**9.0**
Expander 1	35.4	2.2	28.0	1.9
Expander 2	33.4	2.1	32.3	2.2
Expander 3	37.7	2.3	27.0	1.8
Expander 4	29.5	1.8	26.5	1.8
Expanders - subtotal	**135.9**	**8.4**	**113.8**	**7.6**
LN_2 system	99.6	6.1	102.4	6.9
80-K bed	0.5	0.0	10.6	0.7
DP I/O expanders	0.1	0.0	3.0	0.2
JT	37.5	2.3	33.9	2.3
Transfer line	3.5	0.2	3.4	0.2
Dewar heat leak	0.0	0.0	0.7	0.0
Calculation error	9.6	0.6	10.1	0.7
Miscellaneous - subtotal	**150.8**	**9.3**	**164.0**	**11.0**
COLDBOX - SUBTOTAL	**433.0**	**26.6**	**412.5**	**27.6**
EXERGY LOSS - TOTAL	**1323.2**	**81.4**	**1189.3**	**79.7**
Refrigeration load	149.9	9.2	146.9	9.8
Liquefaction load	152.4	9.4	150.4	10.1
Reliquefaction load	0.0	0.0	5.9	0.4
EXERGY USEFUL - TOTAL	**302.3**	**18.6**	**303.3**	**20.3**
OUTPUT EXERGY TOTAL	**1625.5**	**100.0**	**1492.5**	**100.0**

Table 2. Mode 2 (100R) exergy analysis.

	DESIGN		TEST	
4.5 K Refrigeration load (W)	**3495.0**		**4026.0**	
Liquefaction load (g/s)	**0.0**		**0.0**	
Reliquefaction load (g/s)	**0.0**		**0.0**	
Coolant reference temperature (K)	304.9		292.9	
INPUT	**kW**	**%**	**kW**	**%**
First-Stage Compressor	262.1	19.5	259.6	19.0
Second-Stage Compressor	1046.1	77.8	1073.5	78.6
LN2 System (Eff. Carnot = 0.35)	36.4	2.7	33.2	2.4
INPUT EXERGY TOTAL	**1344.6**	**100.0**	**1366.3**	**100.0**
OUTPUT	**kW**	**%**	**kW**	**%**
First-stage: compressor and motor	145.6	10.8	113.8	8.3
First-stage bypass	19.6	1.5	23.4	1.7
First-stage suction mixing	3.3	0.2	6.5	0.5
First-stage aftercooler DP	11.1	0.8	15.9	1.2
First-stage subtotal	**179.6**	**13.4**	**159.6**	**11.7**
Second-stage compressor and motor	606.8	45.1	591.2	43.3
Second-stage bypass	0.9	0.1	1.5	0.1
Second-stage suction mixing	3.3	0.2	3.9	0.3
Second-stage AC and oil rem. DP	10.6	0.8	11.6	0.8
Second-stage subtotal	**621.6**	**46.2**	**608.2**	**44.5**
COMPRESSORS - SUBTOTAL	**801.2**	**59.6**	**767.7**	**56.2**
Heat exchanger 1A	35.8	2.7	37.4	2.7
Heat exchanger 1B	3.6	0.3	3.9	0.3
Heat exchanger 2	26.7	2.0	28.6	2.1
Heat exchanger 3	5.7	0.4	5.6	0.4
Heat exchanger 4	15.8	1.2	20.1	1.5
Heat exchanger 5	4.8	0.4	9.0	0.7
Heat exchanger 6	5.0	0.4	7.1	0.5
Heat exchanger 7	7.7	0.6	9.7	0.7
Heat exchanger 8	2.3	0.2	3.6	0.3
Heat exchanger 9	3.8	0.3	4.6	0.3
Heat exchanger 10	11.1	0.8	7.0	0.5
Heat exchangers - subtotal	**122.2**	**9.1**	**136.5**	**10.0**
Expander 1	21.9	1.6	26.4	1.9
Expander 2	26.2	2.0	25.1	1.8
Expander 3	23.0	1.7	24.8	1.8
Expander 4	22.8	1.7	25.7	1.9
Expanders - subtotal	**94.0**	**7.0**	**102.1**	**7.5**
LN2 system	23.7	1.8	21.7	1.6
80-K bed	0.5	0.0	19.0	1.4
DP I/O expanders	27.2	2.0	19.7	1.4
JT	29.5	2.2	30.8	2.3
Transfer line	3.5	0.3	3.3	0.2
Dewar heat leak	0.0	0.0	0.6	0.0
Calculation error	5.9	0.4	4.7	0.3
Miscellaneous - subtotal	**90.2**	**6.7**	**99.9**	**7.3**
COLDBOX - SUBTOTAL	**306.4**	**22.8**	**338.5**	**24.8**
EXERGY LOSS - TOTAL	**1107.7**	**82.4**	**1106.2**	**81.0**
Refrigeration load	236.9	17.6	260.1	19.0
Liquefaction load	0.0	0.0	0.0	0.0
Reliquefaction load	0.0	0.0	0.0	0.0
EXERGY USEFUL - TOTAL	**236.9**	**17.6**	**260.1**	**19.0**
OUTPUT EXERGY TOTAL	**1344.6**	**100.0**	**1366.3**	**100.0**

Table 3. Mode 3 (100L) exergy analysis.

	DESIGN		TEST	
4.5 K Refrigeration load (W)	**0.0**		**0.0**	
Liquefaction load (g/s)	**37.0**		**34.3**	
Reliquefaction load (g/s)	**0.0**		**6.85**	
Coolant reference temperature (K)	306.6		296.4	
INPUT	**kW**	**%**	**kW**	**%**
First-Stage Compressor	328.1	18.9	262.0	16.6
Second-Stage Compressor	1159.5	66.8	1108.1	70.0
LN_2 System (Eff. Carnot = 0.35)	247.7	14.3	212.3	13.4
INPUT EXERGY TOTAL	**1735.3**	**100.0**	**1582.4**	**100.0**
OUTPUT	**kW**	**%**	**kW**	**%**
First-stage: compressor and motor	171.1	9.9	115.3	7.3
First-stage bypass	39.8	2.3	50.1	3.2
First-stage suction mixing	5.6	0.3	8.9	0.6
First-stage aftercooler DP	7.8	0.5	10.8	0.7
First-stage subtotal	**224.3**	**12.9**	**185.1**	**11.7**
Second-stage compressor and motor	636.8	36.7	605.3	38.3
Second-stage bypass	53.6	3.1	56.4	3.6
Second-stage suction mixing	4.2	0.2	4.3	0.3
Second-stage AC and oil rem. DP	10.8	0.6	11.7	0.7
Second-stage subtotal	**705.3**	**40.6**	**677.8**	**42.8**
COMPRESSORS - SUBTOTAL	**929.7**	**53.6**	**862.9**	**54.5**
Heat exchanger 1A	61.0	3.5	47.3	3.0
Heat exchanger 1B	9.9	0.6	8.4	0.5
Heat exchanger 2	19.3	1.1	15.7	1.0
Heat exchanger 3	4.1	0.2	3.8	0.2
Heat exchanger 4	22.1	1.3	17.6	1.1
Heat exchanger 5	7.3	0.4	6.3	0.4
Heat exchanger 6	0.4	0.0	0.3	0.0
Heat exchanger 7	0.6	0.0	2.4	0.2
Heat exchanger 8	7.4	0.4	7.1	0.4
Heat exchanger 9	5.3	0.3	5.0	0.3
Heat exchanger 10	21.9	1.3	22.7	1.4
Heat exchangers - subtotal	**159.5**	**9.2**	**136.7**	**8.6**
Expander 1	28.5	1.6	26.6	1.7
Expander 2	37.4	2.2	31.5	2.0
Expander 3	49.6	2.9	27.5	1.7
Expander 4	35.2	2.0	33.2	2.1
Expanders - subtotal	**150.6**	**8.7**	**118.8**	**7.5**
LN_2 system	161.0	9.3	138.5	8.8
80-K bed	0.6	0.0	14.0	0.9
DP I/O expanders	0.1	0.0	3.5	0.2
JT	62.3	3.6	57.1	3.6
Transfer line	3.5	0.2	3.4	0.2
Dewar heat leak	0.0	0.0	0.7	0.0
Calculation error	12.6	0.7	10.9	0.7
Miscellaneous - subtotal	**240.1**	**13.8**	**228.1**	**14.4**
COLDBOX - SUBTOTAL	**550.2**	**31.7**	**483.6**	**30.6**
EXERGY LOSS - TOTAL	**1479.8**	**85.3**	**1346.5**	**85.1**
Refrigeration Load	0.0	0.0	0.0	0.0
Liquefaction Load	255.4	14.7	227.0	14.3
Reliquefaction Load	0.0	0.0	8.9	0.6
EXERGY USEFUL - TOTAL	**255.4**	**14.7**	**235.9**	**14.9**
OUTPUT EXERGY TOTAL	**1735.3**	**100.0**	**1582.4**	**100.0**

DISCUSSION AND CONCLUSIONS

The refrigeration system was designed to be 50% refrigerator and 50% liquefier and should operate over a wide range of load conditions from 100% refrigeration to 100% liquefaction. The process and exergy analyses are useful tools for comparing the system behavior and performance at design and off-design conditions.

The detailed exergy analyses presented in Tables 1–3 and in Figure 6 show that the miscellaneous losses and compressor bypass flow losses are a smaller fraction in the refrigeration mode than in the liquefaction mode. In the liquefaction mode nitrogen precooling dominates the miscellaneous losses. Some capacity can be gained by operating the second stage at a lower suction pressure, thereby reducing the bypass flow losses. During the tests conducted subsequently, the system was operated with a low setpoint (2.2 bar) for the second-stage compressor suction. Depending on the mode, this minimized or eliminated the second-stage bypass flow. The system behavior was very stable over a wide range of load levels, for all types of load—refrigeration, liquefaction, and mixed. The system pressure levels adjusted automatically to match the load and thus minimized the input power. Controlled measurements will be made in the future with this self-adjusting control system in place, to find the system maximum capacity and efficiency. Operating the system in some modes at a higher-than-design second-stage pressure ratio and discharge pressure resulted in higher capacities; however, this resulted in reduced isothermal efficiencies.

Exergy analysis is a very strong tool for explaining how input energy is used by the individual components and by the load. Energy optimization involves minimization of input power for a required output power, and the normal question raised is: What is the economic optimum for the components in the system? The process industry uses a constant for the ratio of monetary value per unit of input power (e.g., $1000/kW). This constant is difficult to apply to the individual components. The exergy loss analysis can readily be applied for the economic optimization of each component and its effect on the total system. As an example, the first-stage aftercooler pressure drop (~ 0.3 bar) uses approximately the same amount of energy as the combined pressure drops across the second-stage aftercooler and the oil removal system (~1 bar). This is because the sum of the products of the mass flow and the dp/p for each of these latter items is approximately equal to that of the former.

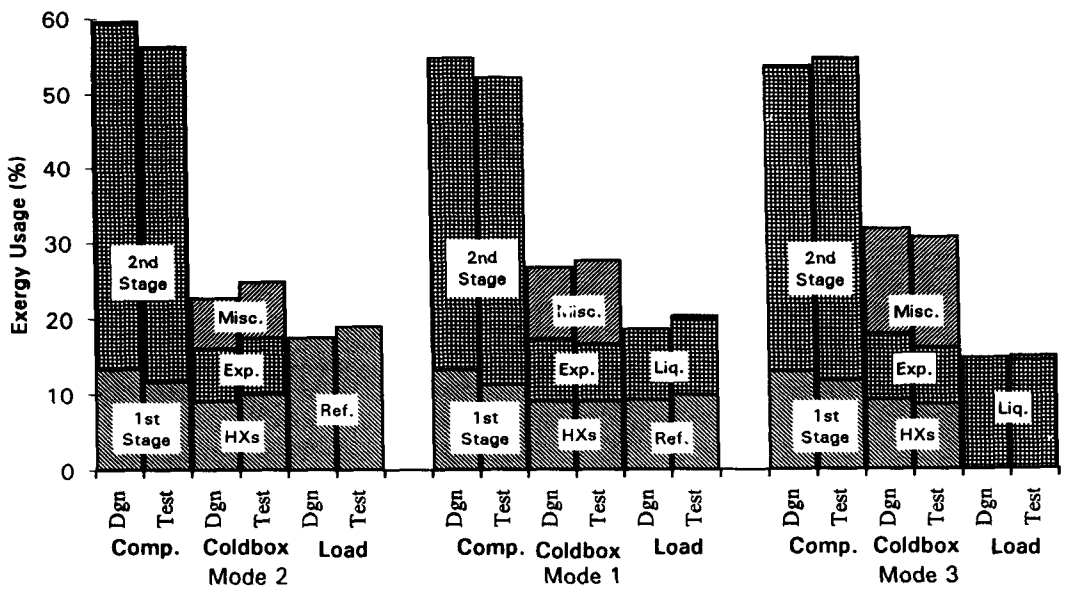


Figure 6. Summary of the exergy analysis.

To enhance the system efficiency, improvements could be made to the second-stage compression process either by designing the system with reduced pressure ratio for the second stage (the increased compressor efficiency would more than compensate for the increased coldbox losses) or by using a different type of compressor. In summary, the exergy analysis comparison of the manufacturer's design with the actual test for the three modes of operation provides a clear insight into the component and system losses and behavior.

REFERENCES

1. U. Wagner and W. Keyer, "Process design features of the SSC MTL cryogenic system," in *Supercollider 3: Proceedings of the 3rd International Symposium on the Super Collider*, Atlanta, GA, March 1991, New York: Plenum Press, 1992.
2. T. Kobel and R. Than, "Initial operation and performance test results of the ASST cryogenic system," to be published in *Supercollider 5: Proceedings of the 5th International Symposium on the Super Collider*, San Francisco, CA, May 1993, New York: Plenum Press.
3. C. Trepp, "Refrigeration systems for the temperatures below 25 K with turboexpanders," in *Advances in Cryogenic Engineering*, Vol. 7, New York: Plenum Press, 1962, p. 251.
4. B. Ziegler, "Second law analysis of the helium refrigerators for the HERA proton magnet ring," in *Advances in Cryogenic Engineering*, Vol. 31, New York: Plenum Press, 1985, p. 693.