Fermilab

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REVIEW OF QUESTIONS POISED BY JEFF APPEL ON SATELLITE REFRIGERATOR DESIGN

PREPARED UNDER FERMILAB SUBCONTRACT NO. 92690
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FOR

FERMI NATIONAL ACCELERATOR LABORATORY
BATAVIA, ILLINOIS

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INTRODUCTION

On 3/7/78, Jeff Appel of Fermilab submitted the following questions to CCI with regard to the design of the satellite refrigerator cold box under Fermilab Contract No. 90398:

QUESTIONS

1. What is the temperature between exchangers II and III (high pressure gas) when the refrigerator operates in the satellite mode?

2. Will there be problems in reading the VPT's in the high pressure streams between exchangers II and III, and III and IV?

3. Line Q (80°K) passes through the shell side space between exchangers III and IV. What is the effect of heat transfer in that area?

4. Why do we use Schedule 5 - 10 piping and 2,000 lb fittings in the same assembly?

5. MV-20 is connected through an 1/8" IPS pipe; valves MV-13, MV-14, MV-60, MV-61 and PI-6 through 1/4 in. OD tubes.

6. What tolerances can one put on flattening of the copper tube of exchanger I?

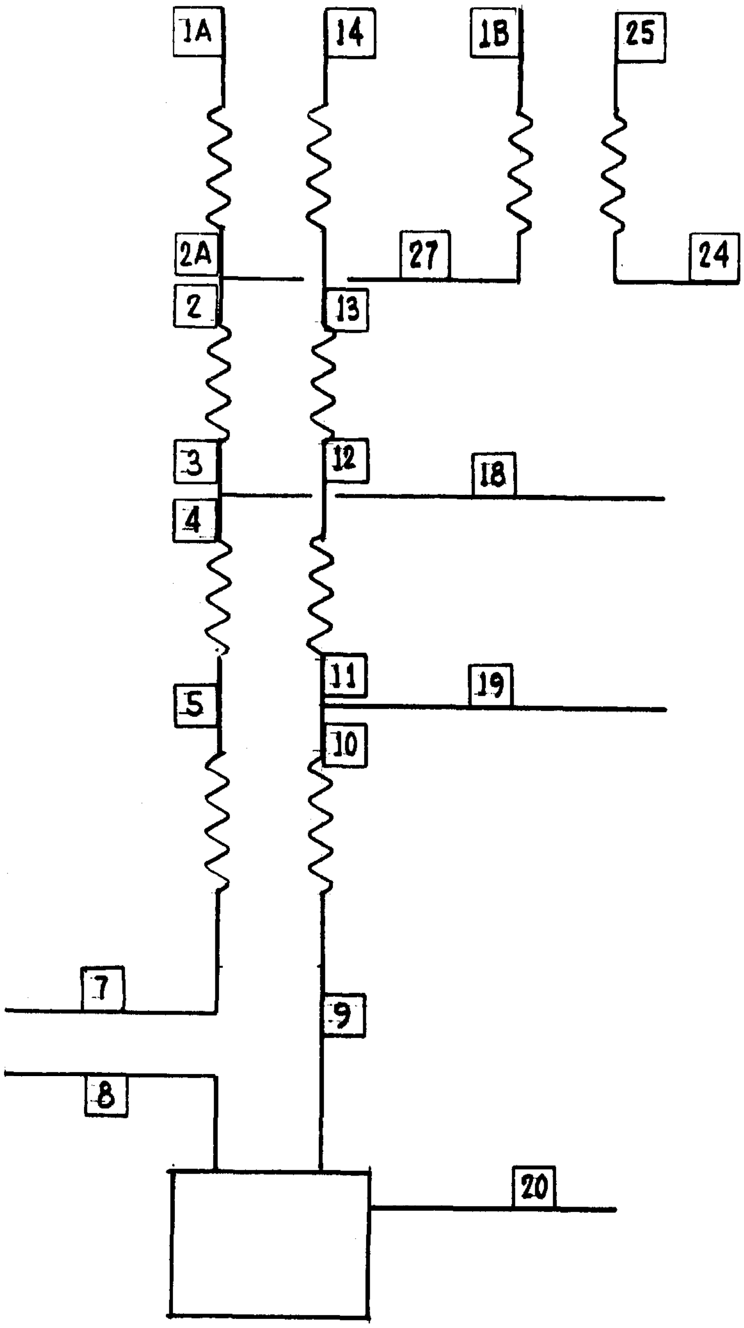
Question 1:

Temperature distribution in heat exchangers during satellite mode operation.- It is assumed that the process of the satellite refrigerator is as shown in Figure 1. Process points are as indicated in Table I:

TABLE I

Point	Fluid	Pressure atm	Temp. °K	Enthalpy J/gr	Flow Rate g/sec
1A	He	20.8	300	1579.2	37.9
1B	He	-	-	-	-
2	He	-	80	434.4	37.9
2A	He	-	80	434.4	37.9
3	He	20.06	25.07	138.8	37.9
4	He	20.06	25.07	138.8	37.9
5	He	20.02	15.0	77.76	37.9
6	He	-	-	-	-
7	He	20	5.24	21.84	37.9
8	He	1.8	4.75	13.1	37.9
9	He	1.22	4.4	29.94	41.1
10	He	1.19	13.15	81.55	41.1
11	He	-	-	-	-
12	He	1.16	23.8	137.84	41.1
13	He	1.11	76.2	410.42	41.1
14	He	1.05	278.5	1466.1	41.1
20	He	-	4.4	11.91	3.2
24	N_2	-	-	-	-
25	N_2	-	-	-	-
27	He	-	-	-	-

It should be realized that the process points of Table I do not necessarily occur at the warm and cold ends of the heat exchangers. The heat exchangers of the satellite refrigerator were designed to operate with the following values of heat transfer coefficients (U), surface areas (A) and product (UA) (Table II):



Flow Sheet

FIGURE 1

TABLE II

Exchanger	II	III	IV	V
U_{AV}, Btu/hr ft^2 °F	140	200	200	170
A, ft^2	63.04	63.04	13.56	13.56
UA, Btu/hr °F	8,825	12,600	2,712	2,305

Available in the complete train of heat exchangers is then:
UA = 26,442

The process conditions of Table I require the following performances of the heat exchangers (Table III):

TABLE III

Exchanger	II	III	IV	V
U_{AV}, Btu/hr Ft^2 °F	140	200	200	170
A, ft^2	63.04	63.04	13.56	13.56
UA_{Req}, Btu/hr °F	8,095	10,673	2,750	1,913
$UA_{Available}$	8,825	12,600	2,712	2,305

The Table III data indicate that for exchanger II $\frac{Q}{\Delta T_m}$ will be approximately 10% larger than the value calculated from Table I. When the temperature of Point 2 drops, duty Q of the heat exchanger will increase, while ΔT_m will decrease. To obtain a change of 10%, Q will increase by 4% and ΔT_m will decrease by 6%. Then the new temperature of Point 2 will be approximately 71°K.

It should be realized that small changes in flow ratio between streams 1 and 14 will have a significant change on the performance requirements of the individual heat exchangers. For this reason, control of the cold end of exchanger II probably can be exercised with the same vapor pressure thermometer (nitrogen charged).

Question 2:

The VPT installation in the tubes and pipes connecting exchangers II and III, and III and IV is not very good. Insulation on the capillary will not do a lot of good. The addition of a copper sleeve stationed to the temperature to be measured plus teflon insulation is necessary for accurate temperature measurement.

It appears that with time, the present installation will show zeroΔT between high and low pressure streams. In the future, it will be advisable to modify the installation as suggested.

The present installation will provide the following:

a) TIC-2 measures correctly and is used for control purposes.

b) TI-4 will probably drift to indicate a temperature close to that of TIC-2.

c) TI-11 and TI-10 will drift to the temperature indicated by TIC-2.

d) TIC-2 will read correctly.

Question 3:

a) Surface area for heat transfer in area between exchangers III and IV is approximately .15 ft^2 (14 in. long tube, 1/2 in. OD).

b) For a mass flow rate of 2 g/sec through the tube, heat transfer coefficient in the tube is approximately 80 Btu/hr ft^2 °F.

c) Assume the coefficient on the shell side is of the order of 20 Btu/hr ft^2 °F.

d) Then overall coefficient, based on surface area of OD of tube, is: 15 Btu/hr ft^2 °F and heat transferred may be of the order of 245 Btu/hr (70 W).

The heat transferred is fairly substantial. It represents approximately 3.5% of that available from the warm expander. If flow during the tube only occurs during non-steady state conditions (such as magnet cooldown), the loss of refrigeration may be tolerable.

Question 4:

We use 2,000 lb socket weld fittings because these are the lowest pressure wrought fittings available as standard items. Our choice of using them is often based on which designer does the job. In non-vacuum and thin wall applications, we will use butt welds; also, in areas where socket welds make for inaccessibility for welding.

Question 5:

We do not have an answer.

Question 6:

The heat exchanger consists of one 1/2 in. OD tube, 60 ft long. Length of the coil is 26 in. for 50 turns. This allows an increase in diameter of .020 in. or 4% before the tubes touch from turn to turn. It is possible to allow bundle length to grow by at least 1 in. and allow an increase in diameter of 1/32 in. Some spacing should be left between turns so that nitrogen vapor bubbles can rise through the coil.

Pressure drop is not much of a factor, since only some 6 g/sec flows through the tubes in the liquefaction mode. For cooldown it is somewhat more, but pressure drop is not a consideration in that case. Pressure drop under steady state conditions at maximum liquefaction mode is of the order of 1-2 psig. A restriction of the flow area by severe flattening may at worst double this pressure drop. Length of tube may be reduced somewhat from 60 ft, without impairing heat transfer greatly. Surface area for heat transfer is some 7.5 ft^2. Heat transferred is of the order of 10,000 Btu/hr (65 liters/hr of liquid nitrogen consumption).

With an overall coefficient of 100, we find $\Delta T_m = \frac{10000}{100 \times 7.5} =$ 13.3°F. Actual ΔT_m is of the order of 30-40°F and we have a large safety factor. This means that the helium flowing through the tube will approach the liquid nitrogen temperature to within 1.5°K. Shortening the tube by two turns (4%) will not make much difference in this approach since ΔT_m changes from 13.3 to 13.8°F and only the safety factor is less.

Based on the above considerations we could tolerate two fewer turns (48 instead of 50) and allow the dimension of the tubes along the axis of the exchanger to grow by some .040 in. to .045 in. Prime criterion is to make sure that there is daylight between turns of the coil.