D0 ENGINEERING NOTE
No. 3823.112-EN-497

D0 SILICON UPGRADE

D-Zero Silicon Cooling System

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July 14, 1998

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Status: The cooling system design is not complete. This paper lays out the general design and some of the design calculations that have been performed up to this date. Further refinement will be performed. This is especially true in the piping layout, piping insulation and detector manifold areas.

INTRODUCTION

The silicon detector is cooled by means of a coolant in the beryllium channels that also act as the primary supporting device for the silicon ladders and wedges. The coolant is water with ethylene glycol added as a freezing point depressant. The glycol concentration in the coolant is 30% by weight resulting in a freezing point of approximately -15°C . If the water/glycol is not sufficient for maintaining the desired detector temperature the concentration of the water/glycol may be changed or an alternative coolant may be used. Alternatives are being looked into.

GENERAL OPERATING SCHEME

The silicon detector is cooled by a closed loop cooling system. The pressure in the detector cooling channels will be maintained below atmospheric pressure. This will be obtained by locating an atmospheric expansion upstream and slightly lower than the detector. This expansion tank will have a bladder or membrane that separates the coolant from coming into contact with air. The gas space above this bladder will be purged with dry air or nitrogen to prevent condensation. The coolant circulation pump is located downstream from the detector so that it will further reduce the pressure in the cooling channels when the pump is operating.

The purpose behind maintaining the cooling channels below atmospheric pressure is to ensure that leaks in the cooling system will not cause the coolant to drip or squirt out of the system onto the detectors. Instead, this will result in air/nitrogen entering the cooling system. Air separators are located downstream of the detector at an elevation about the same as the detector.

The temperature of the coolant will be maintained constant for all detector operating conditions. The coolant is distributed to the cooling channels from manifolds. Since each cooling channel is sized to have similar pressure drop for the nominal flow, no balancing valves are required. This requires that the manifolds are designed to minimize mass while not starving any of the cooling channels. These two aspects of the design simplify the controls of the cooling system. The system is cooled down by reducing the setpoint of the controller in the chiller and the overall flow rate is manually set to its appropriate value. From this point on, the system needs to be monitored to ensure that

everything is operating properly. Leaks can be monitored by watching the level in the air separators, the overall flow rate and pressures within the system.

GENERAL EQUIPMENT LAYOUT

The location for the chillers is on the south sidewalk. The piping for the detector will service both the Collision and Assembly Hall locations. Most of the piping will be vacuum insulated where practicable. Valves will be utilized to select which hall the chiller will service. The piping layout has not been determined at this time but the area under the edge of the sidewalk where the calorimeter piping runs between the two halls will be used for part of the path. This is the area where the old TRD piping was located. Installation in this area will be tricky but not impossible.

The expansion tank is believed to be located near the air conditioner platform when in the collision hall position. In the assembly hall, we may have to install a temporary structure to mount the vessel.

The manifolds for the detector are currently designed to come from both the south and north ends of the detector. The manifolds will stop just short of $z=0$. This is to reduce mass at this location. The original design for the 7 barrel arrangement had the manifolds going the entire length of the silicon detector and were supplied with two lines at one end of the detector and two returns out the other end. The 6 barrel design may end up with a similar design if the flow balancing cannot be worked out with the current concept.

The piping outside of the calorimeters will be stainless steel. The piping within the central calorimeter (CC) will be plastic. The piping that passes between the CC and the end calorimeter will be vacuum insulated but there will have to be a transition from the metal piping to plastic piping before extending too far within the CC. Flow balancing between the south end and north end piping needs to be studied. This will be looked into after determining the feasibility of the current detector manifold concept.

The location of the air separators has not been determined at this time. However some concepts are under consideration. The separators may simply be a larger cross section in the piping that would be in the form of a tee where the liquid turns to go to a lower elevation and the gas would rise to the upper branch of the tee.

DESIGN SPECIFICS

In this section, the current details of the design are laid out. There are many features being designed into the cooling system to enhance the inspection, operation and coolant quality. Whenever possible, equipment will be located near the chillers in the assembly hall in order to provide ease of access and reduce the need for accesses to the collision hall.

The nominal system flow rate for the system is about 40 lpm. This assumes a 4 psi pressure drop over the cooling channels. The anticipated coolant temperature is between -10 and -5°C .

All process transmitters in the system will be connected to the cryogenic controls system if possible. I have not contacted Dan Markley or Kurt Kempetz about this plan. The information for these devices can be made available for the data stream through this controller. A parallel electrical signal may be provided directly to the data stream if necessary.

Chiller Area Equipment

The area around the chillers will contain most of the instrumentation and coolant conditioning equipment. Immediately downstream of the chiller pump is the coolant conditioning package. This is where coolant is sterilized, filtered and de-ionized. Downstream of this package is the instrumentation package where the coolant properties are monitored and the flow rate is measured.

Chiller and Circulation Pump

The chiller is made by Applied Process Equipment Corp and is a modified version of their model AC3. The cooling capacity of the unit is between 4.5 and 6.6 kW. The testing has been completed but I need to get the paperwork to confirm the actual value. If necessary, I have the equipment to retest the capacity of the equipment. The circulation pump is located within the chiller unit. It is a centrifugal pump with a head of about 35 psid for 40 lpm of -10°C water/ethylene glycol (30%w). Since the pump is centrifugal, the head pressure can be controlled by controlling the flow through the pump. This will be performed by a manual bypass valve that goes between the chiller supply and return piping. Additional valves on the supply and return piping are being installed for additional pressure control if necessary. These valves mainly act as the way to isolate the chiller when changing out the chiller for the spare chiller. The chiller's piping connections are 1" pipe on the supply side and 1-1/4" pipe on the return side. The supply piping in the chiller area will be 1" but it may be increased to 1-1/2" depending upon convenience. It is not a pressure drop issue. The return piping in this area will be 1-1/2".

In the event that the chiller capacity is determined to be undersized, the spare chiller will be installed in parallel with the primary chiller. This will require the addition of check valves in the system which will effectively decrease the pressure at the return side of the pump. If this pressure drops below the Net Positive Suction Head required (NPSHr), the pump will cavitate and will be destroyed. A booster pump may be required when adding the check valves. A more detailed analysis of this will be required if both chillers are installed.

Pressure instrumentation will be installed on the supply and return piping of the chiller. This is required to monitor these pressures during start-up and while monitoring operation of the system. Both a pressure transmitter and a dial gauge will be installed at these locations. Additionally, the pressure will be monitored in the instrumentation package since there may be significant pressure drop between the chiller and the hall selection valve.

Coolant Conditioning and Instrumentation Packages

The coolant needs to be maintained within specified parameters to reduce the potential for corrosion and scaling. We do not currently have a specification for

corrosion but there are plans to develop one based upon studies being performed by Packard Engineering. In addition to this, the coolant must be monitored for electrical conductivity to prevent the fluid from being a ground path for the detector. If we use a magnetic flow meter in the system, there will be a minimum conductivity requirement as well. Since the glycol concentration has a significant effect on the pressure drop in the system, this will also be monitored. The coolant conditioning area is located by the chillers and is where all of this monitoring and conditioning will occur.

The properties that we are likely to measure are the electrical conductivity, pH and refractive index. The electrical conductivity measurement is quite straight forward in its application. A de-ionizer will be provided on a small bypass leg of the main flow line. This de-ionizer may be used to reduce the conductivity of the coolant. Since it is a small flow rate, the change when it is turned on will be slow.

The pH meter which can also monitor the oxidation potential is the probable device for monitoring the corrosiveness of the coolant. Information from Packard Engineering should give us an idea of the ideal range for this reading. How to make the adjustments to the pH have not been determined at this time. If an inhibitor must be used, we may either have to install a device for monitoring its concentration or we may have to take samples on a periodic basis.

The refractive index will give us a good measure of the glycol concentration of the coolant. Specific gravity is not being measured for this since inhibitors, if added, significantly effect the specific gravity. Also, inhibitors generally deplete over time so the specific gravity would be changing over time. The glycol concentration will only change if we must periodically remove gas from the system due to a leak because the gas will be saturated with moisture. Water can be added by opening a valve on the return side of the pump and slowly sucking the water from a water supply. This valve is provided on the vessel upstream of the chiller pump.

The pH of the fluid is not the only concern for corrosion and scaling. Bacteria is a common problem in water based cooling systems. An ultraviolet sterilizer is being planned for the system to kill any bacteria that may be in the coolant. A biocide should be added to the system at the time when the coolant is added in order to kill any bacteria that may be in areas of the piping that will not get into the fluid to pass through the UV sterilizer. It may be possible to eliminate the UV sterilizer if the biocide is effective for long periods of time and will not interact with the materials in the system. As an added precaution, the system will be made light tight wherever possible.

To monitor the corrosiveness of the coolant, an inspection spool piece will be installed. I envision that this section will contain a piece of beryllium that is electrically insulated from the piping. It would be ideal if this piece is at the same potential as the cooling channels. This would give us an accurate visual and physical check on the state of corrosion in the system. The resistance of the beryllium piece can be periodically monitored for changes which are inevitable with corrosion. A sight glass will be installed to view the beryllium piece. A light blocking cover will be removed to view the piece. The spool piece can also be bypassed for removal and closer inspection.

For added protection of the components, filters are being installed in the main flow line after the UV sterilizer and after the de-ionizer. A bypass is provided around the UV sterilizer, main filter and inspection spool piece to facilitate replacing the filter elements and removing the UV sterilizer if necessary. The filter may only have to be

replaced after the first time the system is filled with coolant and circulated for a few hours. It is more likely that the filter will never be changed. A differential pressure gauge is in the design to monitor the resistance of the filter package. As the filters trap more particles, the elements will become more resistive and the pressure reading will give the best indication of the time to change the filters. The filter media will be made from pleated polypropylene fabric.

The flow rate of the coolant will be measured in the instrumentation package. The goal is to use a flow meter with no moving parts. A flow meter with moving parts greatly increases the probability that maintenance will be required. Additionally, we should get a flow meter that has a large range where the meter is accurate. A typical cost effective flow meter is an orifice or a venturi with a differential pressure meter. With an orifice meter, the minimum reading is about one third of the full scale reading (3:1) and is easily corrected for fluid properties. A venturi meter has a slightly lower minimum but requires a calibration curve for each fluid condition. These meters also have a pressure drop of about 2 to 4 psi although the venturi meter is typically less than an orifice meter. Instead of these, magnetic, vortex or coriolis flow meters can be used. They have less pressure drop and larger range of accuracy. I have looked into the magnetic flow meter and it has a ratio of at least 8:1 (depending upon ID of device). The magnetic flow meter actually measures the velocity of the fluid which can be converted to a volumetric reading. The vortex flow meter measures the volumetric flow rate. Coriolis flow meters measure the mass flow rate of the fluid. Manufacturers of the coriolis flow meter state that it can also measure the specific gravity of the fluid. Of these three devices, the coriolis flow meter costs the most (> \$4k) and the vortex flow meter is the least expensive (> \$1.4k). I need to look into the vortex flow meter to see how it compares to the magnetic flow meter but I am now leaning toward the magnetic flow meter.

Additional Equipment

In addition to the items listed above, there are a couple more features being designed into the system. We need to make sure that we do not get air into the circulation pump. It will greatly reduce the life of the pump. To prevent this, a final air separation vessel will be installed just upstream of the chiller. This vessel also serves as the location for filling the system with the coolant. A level gauge will be installed to monitor the level within the vessel.

In order to monitor the flow balancing between the north and south sets of manifolds, differential pressure transmitters are designed into the system to monitor the pressure drop between the bottom of the air separators and the point where the two piping runs are connected. These two lines will merge near the chiller area so as to provide a large enough pressure drop to measure accurately. The pressure drop versus flow rate will be characterized for each section in order to get an approximation of the differences between the two piping runs. For a more accurate flow reading, the low pressure taps could be moved to just before the piping runs connect.

Detector Area Equipment

The equipment near the detector is fairly straight forward. I must point out that I do not include the detector manifolds or piping around the CC in this statement. The equipment in the detector area consists of the expansion tank, air separators and their associated instrumentation and valving. The other items will be discussed in a later section.

Expansion Tank

Exactly how the expansion tank should be used has not been determined. There are a couple of scenarios that need to be examined before the concept is worked out. It will in all likelihood be a commercial expansion tank with small modifications. An expansion tank used for well systems typically has a schrader valve used to set the system pressure. There is a bladder within the vessel to separate the water from the gas volume. In our case, we may simply remove the core of the valve. Tests will need to be performed to ensure that the time response of the system will not be hampered by this small orifice. Also, I need to find out what materials are in contact with the liquid. Carbon steel will not hold up to an uninhibited water/ethylene glycol coolant. The consequences of stopping the flow when the coolant is cold need to be examined. In principle, the expansion tank is 100% full and at atmospheric pressure when the system is at room temperature. After cool-down, the density of the fluid has increased so the level in the tank will have dropped. When the flow is stopped, the level in the tank and the level in the cooling channels would reach the same level if there were holes in the cooling channels. In principle, there won't be any holes therefore the pressure in the cooling channels will drop. Thus an equilibrium will be reached. I have not confirmed this by calculation but I plan to look at this.

It should be possible to measure the liquid level in the expansion tank with a differential pressure transmitter or a gauge pressure transmitter. Local indication is not normally readable at this location so no pressure gauge will be provided.

Air Separator

Details of the air separator need to be worked out. I believe that the air separators should have a gas volume at the top and the liquid level will be monitored. As the gas volume increases due to a leak into the system, the liquid level will drop. The gas may then be evacuated by means of a solenoid valve at the air separator that is connected to a vacuum pump at a lower elevation. This solenoid valve will be controlled manually however it could be automated since all of the equipment necessary will be in place. A good location for the vacuum pump would be near the chillers for easy access. A vessel near the vacuum pump could be used to act as a buffer in case all of the gas gets pulled out of the air separator and coolant gets pulled into the lines. This vessel would be equipped with a level switch that would isolate the vacuum pump to prevent pump damage.

Detector Manifolds

As mentioned earlier, the manifolds are currently desired to come from both the south and north ends of the detector. This is to eliminate the mass of the coolant and

piping at $z=0$. The flow balancing for this arrangement is possible if the manifolds are uniform in pressure. Since there is an additional requirement for minimizing the mass of these devices, the manifolds will be stepped down if possible. A test on whether a straight manifold will work will be conducted in the near future. If this test fails to balance the flow, we will need to determine if we can revert to a design that feeds from one end and returns the flow to the other end of the detector therefore placing some mass at $z=0$.

The connections on the cooling channels have not been completely finalized at this time. The connection location on the bulkhead has been finalized and any change is highly undesirable. The F-disk and H-disk connections can be changed if necessary since fabrication of these cooling channels has not begun. It appears that the best connection on the barrel F-disks is straight out in z toward the bulkhead it is affixed to. The end F-disks are very tight but it looks possible to install the fittings in a similar manner to the barrel F-disks. The H-disk fittings have more space so there are more options available.

The manifold connections are best pointed radially inward. This makes the final connection difficult. This forces the tubing to be placed on the cooling channels first, then fed through the half cylinder (without the manifolds), connecting the tubes to the manifolds and then pushing the tubes back into the half cylinder and mounting the manifolds. The clearances between the tubes and the detector need to be carefully understood to avoid disturbing the detector. A mockup of this will be highly beneficial.

The fastening method and tubing material are not finalized. We need to find a tubing that will not creep and a clamp that will maintain its grip on the tubing. The experience we have had with vinyl tubing shows that this material creeps and will not maintain a leak tight connection. Polysulfone has been quoted as being a good candidate for this tubing although it may collapse under vacuum especially when bent. Further work will be done in this area.

Piping Configuration

As mentioned earlier, most of the piping in the system will be vacuum insulated. The purpose is to keep the piping heat load lower than the detector heat load. Due to the cooling capacity of the chillers, this is now very desirable if we wish to avoid using both chillers at once. It's mainly the long runs of piping that will be insulated in this fashion. The remainder of the valves, instrumentation, vessels and equipment will be foam insulated. The additional heat load due to using foam in these areas should be modest and will save significant cost and effort.

All of the supply piping between the chiller area and the expansion tank will be 1" pipe. The return piping from the air separators near the detector and the final air separator near the chiller will be 1" pipe. Where these two return lines combine the piping will increase to 1-1/2" pipe. The piping between the final air separator and the chiller will also be 1-1/2" pipe.

System Pressures

This discussion will be more focused on pressure drops rather than actual pressure. The calculations that I have performed are mainly concerned with the dynamic pressure drop in the system. The static pressure has not been considered at this time. This process actually leads to a conservative pressure at the inlet to the chiller's pump. The system pressure drop that I calculated is important to insure that the pump can deliver this head. Since the elevations cancel out for this number, it is less conservative. Table 1 lists what the initial estimates are for the pressure drop for various components in the system.

System Heat Loads

The heat load for this design is assumed to be 640 mW per SVX2 chip. It is likely that the chip power will be significantly less than this. Maybe 500 mW per chip. Therefore, this system is being designed for a 3962.88 Watt detector heat load using the 640 mW value. Table 2 is a summary of each barrel and disk heat load due to the chips.

Table 2. Barrel and Disk Heat Loads

Inner 4 barrels:	36	6-chip ladders = 216 chips = 138.24 watts
	36	9-chip ladders = 324 chips = 207.36 watts
		Total = 345.60 watts divided between 2 passages.
Outer 2 barrels:	36	3-chip ladders = 108 chips = 69.12 watts
	36	9-chip ladders = 324 chips = 207.36 watts
		Total = 276.48 watts divided between 2 passages.
F-disk:	12	14-chip wedges = 168 chips = 107.52 watts.
H-disk:	48	6-chip wedges = 288 chips = 184.32 watts.

Table 1 lists the heat loads considered for this system. The table is not complete but it does include what are assumed to be the major heat sources. The heat loads total about 1.4 kW bringing the total system heat load to about 5.4 kW.

I should note a few items of importance when calculating these heat loads. Only 0.5 inches of foam is assumed for the foam insulated sections and the lengths are considered to be conservative. Adding an additional 0.25 inches of foam would only decrease the heat load by about 80 watts. The vacuum insulation assumes that the emissivity of the inner pipe is 0.1. A clean stainless pipe is about 0.22 and about 0.17 is polished. Covering the inner pipe with aluminum tape may reduce the emissivity to 0.07. There are a couple of sections of piping that I have calculated the heat load based upon foam insulation that we could change to vacuum insulation but this would drop the heat load by about 160 watts. Using multi-layer insulation in the vacuum would help reduce the heat load by about a factor of 3 or more from the vacuum insulation with aluminum tape alone. The trade-off to increase the cooling capacity margin by increasing installation effort seems worth while unless the chip power decreases significantly. After a quick assessment of the changes mentioned in this paragraph, the piping heat load as noted in Table 1 may drop by 500 W. Thus the overall system heat load could be about

4.9 kW assuming the 640 mW chip power. Again, this does not take into account every item that will be in the system nor the ambient heat load on the detector.

System Temperatures

This discussion will assume that the coolant temperature at the cooling channels is -10°C . The heat loads for the system are estimated based upon what I believe is the worst case condition. Since I do not currently know the true piping layout I have used what I believe will be conservative values. Table 1 lists the heat loads of the components and temperature rises of the coolant passing through the components. Components such as elbows and tees are not included since they will not have a significant effect on the overall heat load estimate. The average temperature rise on the fluid due to the detector heat load is roughly 1.68°C using the 640 mW chip power. This gives an overall fluid temperature rise of 2.24°C .

SYSTEM SAFETY

The cooling channels are pressure tested to only 25 psig. Assuming the channel were to fail with any addition of pressure, this could be considered a burst pressure. A typical pressure vessel safety factor on a burst pressure is on the order of 4 to 5. Piping systems typically have a slightly lower safety factor. On the other hand, bridges have a safety factor of possibly 2. Bill has chosen to treat the cooling channels like pressure vessels. Therefore, the pressure rating is chosen to be 5 psig. There is good reason for this decision. The cooling channels have not been rigorously modeled and tested to know how good they really are. Therefore the uncertainty in the long term strength of the devices is quite high. Thus choosing a safety factor of 5 is justifiable considering the consequences of a failure. Protecting the cooling channels from exceeding 5 psig is not trivial.

As long as the expansion tank operates at atmospheric pressure, everything seems fine. There are cases where the expansion tank can exceed this pressure. When the system is at room temperature, the bladder in the expansion tank is at its highest position. When the system is flowing under these conditions, everything may seem fine but the system no longer has its buffer and it could be over-pressurized. Therefore, a relief valve must be installed. Since we cannot just dump ethylene glycol anywhere we want, we must pipe the exhaust of the relief valve into a spill tank. This can be simply a 55 gallon barrel in an unused corner or mounted under the expansion tank. The system volume is going to be greater than this barrel but it will be enough to lower the pressure and protect the cooling channels. Calculations may show that we can get by with a smaller tank.

The major pressure source in the system is the pump in the chiller. Devices down stream of the chiller need to be protected from over-pressurization. Based on the probable components for the system, a relief valve on the supply piping of the chiller will be set at no higher than 70 psig.

In addition to pressure safety, we need to make sure that we do not create mixed waste. Since the coolant will be exposed to radiation and the coolant is toxic, Fermilab

Safety is requesting that we periodically take samples to send out for measuring the amount of tritium in the coolant. We are requested to change the coolant before we reach mixed waste levels. It is not permitted for us to dilute the coolant when we are done using it even though the result would be the same. Based upon radiation estimates and calculations that were performed, we will not have to replace the coolant.

Table 1. Coolant Pressure Drop, Heat Load and Temperature Rise for the Silicon Detector Cooling System

Item #	Item Description	Pressure Drop	Heat Load	Temperature Rise	Comment
1	Supply Isolation Valve	1.600 psid			
3	Sterilizer Fore Valve	0.015 psid			
4	Sterilizer	2.000 psid	240.00 W	0.102 C	This is the power consumption of the sterilizer
5	Particulate Filter	4.000 psid			Estimated dP for Changing Filter
6	Inspection Diverter Valve	0.178 psid			
7	Inspection Isolation Valve	0.015 psid			
8	Instrumentation Diverter Valve	0.178 psid			
9	Flow Meter	3.000 psid			
10	Instrumentation Isolation Valve	0.015 psid			
11	Hall Selection 3-Way Valve	0.178 psid			
15	Main Control Valve	5.000 psid			Assumed Value
20	Chiller Supply Elbows	0.548 psid			
21	Chiller Supply Tees	0.438 psid			
22	Chiller Supply Straight Pipe	0.378 psid	79.50 W	0.034 C	Foam Insulated (1/2")
30	Skid Elbows	0.091 psid			
31	Skid Tees	0.181 psid			
32	Skid Straight Pipe	0.068 psid	103.42 W	0.044 C	Foam Insulated (1/2")
40	Supply Elbows	0.286 psid			
41	Supply Tees	0.095 psid			
42	Supply Straight Pipe	1.324 psid	53.40 W	0.023 C	
50	Valve Before Expansion Tank	0.007 psid			
51	Expansion Tank Inlet	0.079 psid			
52	Expansion Tank Exit	0.005 psid			
55	Post Expansion Tank Elbows	0.008 psid			
56	Post Expansion Tank Tees	0.015 psid			
57	Post Expansion Tank Straight Pipe	0.030 psid	28.60 W	0.012 C	
58	Manifold Supply Elbows	0.033 psid			
59	Manifold Supply Tees	0.033 psid			
60	Manifold Supply Straight Pipe	0.181 psid	63.01 W	0.054 C	2 pipes therefore heat load needs to be doubled
61	Manifold	2.000 psid	23.62 W	0.040 C	Assumed dP Value; 8 pipes therefore needs 8x
62	Cooling Channels	4.000 psid			Anticipated Operating dP
63	Manifold Return Elbows	0.066 psid			
64	Manifold Return Tees	0.100 psid			
65	Manifold Return Straight Pipe	0.407 psid	141.77 W	0.120 C	2 pipes therefore heat load needs to be doubled
66	Valve Before Air Separator	0.025 psid			
67	Air Separator Inlet	0.020 psid			
68	Air Separator Outlet	0.015 psid			

Table 1. Coolant Pressure Drop, Heat Load and Temperature Rise for the Silicon Detector Cooling System

Item #	Item Description	Pressure Drop	Heat Load	Temperature Rise	Comment
69	Return Elbows	0.083 psid			
70	Return Tees	0.028 psid			
71	Return Straight Pipe	0.662 psid	53.40 W	0.045 C	2 pipes therefore heat load needs to be doubled
72	Hall Selection return 3-Way Valve	0.044 psid			
74	Pre-Final Air Separator Elbows	0.009 psid			
75	Pre-Final Air Separator Tees	0.018 psid			
76	Pre-Final Air Separator Straight Pipe	0.034 psid	51.71 W	0.022 C	Foam Insulated (1/2")
77	Final Air Separator Inlet	0.014 psid			
78	Final Air Separator Outlet	0.011 psid			
80	Chiller Return Isolation Valve	0.015 psid			
81	Pre-Chiller Elbows	0.091 psid			
82	Pre-Chiller Tees	0.072 psid			
83	Pre-Chiller Straight Pipe	0.068 psid	103.42 W	0.044 C	Foam Insulated (1/2")
	Total	27.764 psid	1365.39 W	0.540 C	Temperature Rise Does Not Include the Detector

Symbol Key for Piping and Instrumentation Diagram

AT	Analysis Transmitter pH, Electrical Conductivity, Refractive Index, etc.	LI	Level Indicator Digital Readout or Computer Reading
FE	Flow Element Mass Flow Meter, Magnetic Flow Meter, etc.	LSH	Level Switch High A Contact in Either a Level Switch or a Transmitter
FI	Flow Indicator Digital Readout or Computer Readout	LT	Level Transmitter Usually a Pressure Transmitter Or on a Level Element (Photo Cell etc.)
FSL	Flow Switch Low A Contact in Either a Flow Switch or a Transmitter	PDT	Differential Pressure Transmitter
FT	Flow Transmitter Usually on a Flow Element	PDI	Differential Pressure Indicator Analog Gauge, Digital Readout, Or Computer Readout
HS Momentary	Hand Switch Simply an Electrical Switch Here it is a Momentary Switch	PI	Pressure Indicator Analog Gauge, Digital Readout, Or Computer Readout
M	This is a Symbol for a Magnetic Flow Meter	PT	Pressure Transmitter
		TE	Temperature Element RTD, VPT, Thermocouple, etc.