28 November 2001

ALICE SPD cooling system

Requirements:

- To remove a nominal power of 1380 W distributed per 60 staves (cooling channels) at a nominal duct temperature of 15°C [1]

- To ensure as much as possible a temperature uniformity along the stave (even if for the time being this requirement has not yet been formulated).

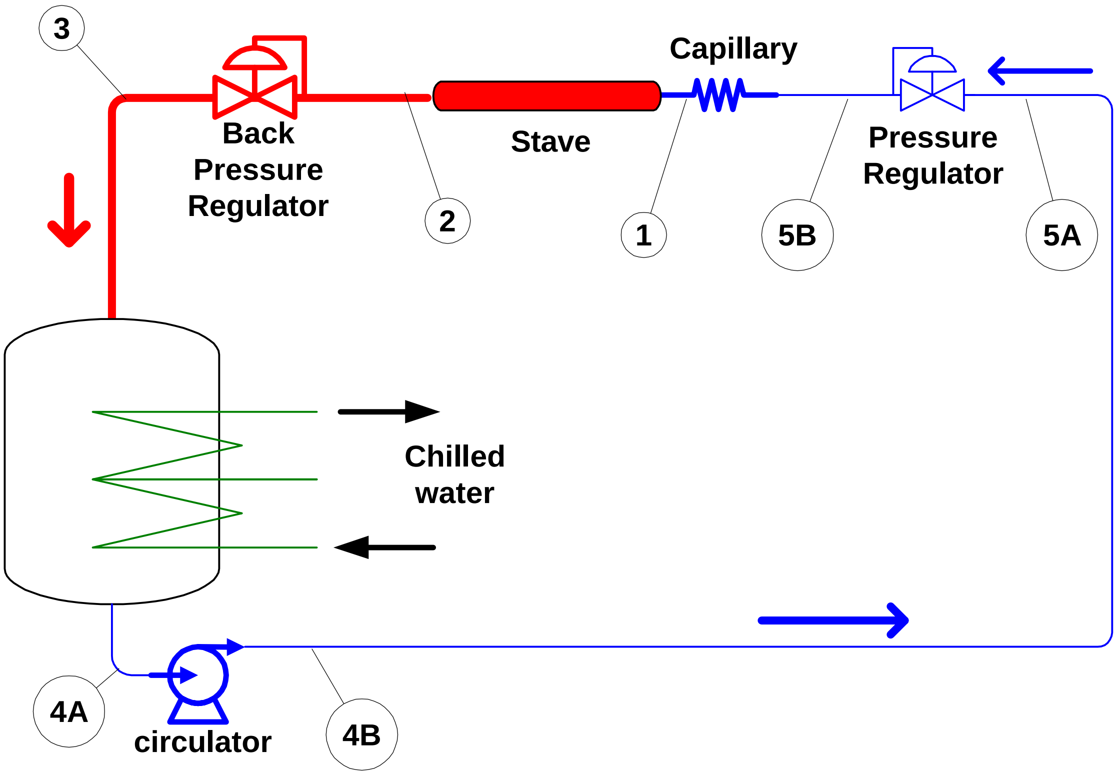
- To work at reasonable pressure (near atmospheric pressure) in order to reduce the material budget as much as possible

- To be as much as possible unlikely to leak

- To be adjustable in real-time, allowing the tuning of the cooling temperature and power removed for each individual stave

[1] The overall heat transfer coefficient between the duct inner wall and the chips is unknown at this stage. To begin with, an evaporating temperature of 15°C will be envisaged, believing that the chips' temperature will then stabilize between 20°C and 25°C.
In any case, as it will be seen later, the system is flexible enough to vary the evaporation temperature over a reasonable range, in case 15°C turns out to be too high/low.

- **Description of the operating principle**



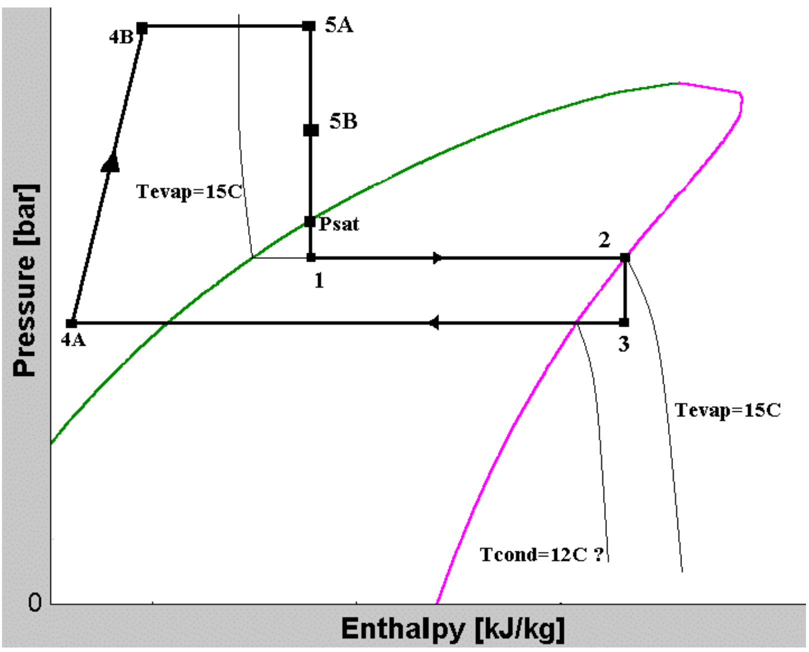


Fig 1: Conceptual circuit and Thermodynamic cycle in P-h diagram (qualitative)

The coolant enters the staves as saturated or slightly evaporated liquid (state 1). Evaporation takes place along the stave and the mass flow should be such that nearly saturated vapor ($\chi \geq 95\%$?) should leave the staves (state 2). If necessary an electrical heater can ensure that the eventual remaining liquid evaporates before exiting the SPD volume.

The pressure in the staves is set by the back pressure regulators (22). These, as well as the return piping, inevitably introduce a small head loss so the vapor reaches the condenser at state 3. Yet it should be noticed that the pressure in the staves is determined only by the set point on the back pressure regulators and is not influenced by how much lower the pressure is in the condenser. The pressure in the condenser is determined by the flow and temperature of the chilled water. Obviously this pressure should be sufficiently lower than the pressure in the staves so ensure the return of vapor (state 3).

See at:
http://www.goreg.com/tech_info/animation/bp3_anim.htm
for an animation explaining the operation of a Back Pressure regulator (Dome loaded).

Assuming that condensation takes place at 12°C, the corresponding pressure drop $2 \rightarrow 3$ for the three candidate coolants is(see P-T saturation curves in appendice):

60mbar C_5F_{12},
30 mbar forC_6F_{14}
200 mbar for C_4F_{10}

To avoid cavitation, the temperature at the inlet of the pump should be as far below 12°C as possible. ST/CV provides a network of chilled water at 6°C throughout CERN, so with an efficient heat exchange, it should be possible to lower the temperature at state 4A down to 8 °C…?

The line 4A $\rightarrow$ 4B represents the pressure rise given by the pump. The temperature rise (resulting from the thermodynamic inefficiency of the pumping process) is usually small, so, unless point 4A is at the right temperature or heat is transferred along the supply line (4B$\rightarrow$5A), the liquid will be delivered to the pressure regulators at a temperature which may not be adequate. This would result to liquid being supplied to the staves too cold to be able to start evaporating at the very beginning of the stave (point 1 located to the left of the saturated liquid curve) or in opposition, wasting latent heat of evaporation by being too much into the two phase region.Note that states 5A (inlet of pressure regulators) and 5B (inlet of capilaries) are approximately at the same temperature, i.e. equal or, ideally, slightly above the evaporation temperature, (isothermal lines are pretty much vertical on the subcooled liquid side of the P-h diagram). In this case, the whole liquid supply line would be above the dew point in the cavern, and probably would not exchange heat with the outer volumes of the detector in its routing into the SPD.

A capillary represents a fixed pressure drop in which the flow is laminar and thereby proportional to the driving pressure difference applied at its ends (Pressure difference P_{5B} - P_1). The pressure at the capillary outlet (P_1) is set by the back pressure regulators and is the pressure at which evaporation takes place in the staves. On the other hand, pressure regulators introduce an adjustable pressure drop, so one can modify the pressure at the capillary inlet P_{5B} in order to obtain the desired flow rate.

The higher the pressure available upstream of the capillary, the wider the available proportional range of flow through it to accommodate stave heat load variation (e.g. varying numbers of powered silicon modules). Hence, the pressure at the pump outlet should be high enough to allow for this proportional fluid flow tuning process. The pump speed can be regulated with a frequency drive in order to obtain the desired outlet pressure. The lower limit to this flow variation happens when the pressure 5B reaches the saturated liquid pressure Psat. For lower Pressure Regulator outlet pressures, evaporation would take place upstream of the capillary thereby obstructing it with vapor bubbles.

• Preliminary design parameters

An evaporating temperature of 15ºC shall be assumed in the staves. A dissipation of 23W/stave will be assumed

The corresponding vapor pressures for the three coolant candidates are:
C_5F_{12} (FC-87 from 3M) Psat@15ºC= 577 mbara
C_6F_{14} (FC-72 from 3M) Psat@15ºC= 189 mbara
C_4F_{10} (PF-5030 from 3M) Psat@15ºC= 1.92 bara

The corresponding latent heat of vaporization is:
C_5F_{12} , ΔHfg @15ºC =104 618 J/kg
C_6F_{14} , ΔHfg @15ºC =107 958 J/kg
C_4F_{10} , ΔHfg @15ºC =91 008 J/kg

Mass and volumetric flow needed

	Density at 15ºC Kg/m3		Flow / stave			6 staves (one sector)			60 staves		
	Sat liquid	Sat vapour	g /s	L/min		g /s	L/min		g /s	L/min	
				in	out		in	out		in	out
C_5F_{12}	1623.8	7.23	0.22	0.01	1.82	1.32	0.05	10.95	13.19	0.49	109.47
C_6F_{14}	1747.2	2.72	0.21	0.01	4.70	1.28	0.04	28.2	12.78	0.44	281.97
C_4F_{10}	1536	20.75	0.25	0.01	0.73	1.52	0.06	4.38	15.16	0.59	43.85

Note: this is assuming saturated liquid at the staves inlet and saturated vapor at the outlet, i.e., full advantage is taken of the latent heat (ideal situation). In reality, the flow needed may be up to 1.5 times higher (corresponding to 2/3 of the max latent heat being used).
For selecting the pump, a nominal flow rate of 0.75 L/min and ΔP=5 bar shall be assumed.

Estimation of heat transferred at the condenser (for 60 staves) and water flow rate required:

	Cp (J/Kg. ºC) Liq @ 10 ºC	g /s	ΔHfg @12ºC (J/kg)	Qcond (W)	To subcool to 8ºC (W)	Water flow rate required L/min (Tin=6ºC, Tout=10ºC)
C_5F_{12}	1029	13.19	105 607	1393	54	5.2
C_6F_{14}	1029	12.78	108 875	1391	52	5.2
C_4F_{10}	1056	15.16	92 079	1396	64	5.2

For 6 staves (one sector):

	Qcond (W)	To subcool to 8ºC (W)	Water flow rate required L/min (Tin=6ºC, Tout=10ºC)
C_5F_{12}	139	5	0.5
C_6F_{14}	139	5	0.5
C_4F_{10}	139	6	0.5

SYSTEM CONTROL

Similarly to other LHC experiments, in the final installation pressure regulators are likely to be located in the experimental cavern, where the magnetic field might be beyond the acceptable limit for standard industrial control electronics. This will require the use of analog air pressure-piloted regulators driven by remote electro-pneumatic converters.

To ensure that the correct flow rate is being sent into each stave, the pressure regulators will be individually tuned via feedback control according to the circuit load variation.

A temperature sensor on the exhaust tubing sends a feedback signal to a commercial PID controller (model G9FTE-R*E1R-88-N Mfr: RKC Instruments). The PID controller sends an analog signal to an electro-pneumatic converter, which in turns sends an analog pressure signal to a (several) pressure regulators.

Alternatively, PID control can be implemented using a BridgeView PID extension toolkit, with WAGO DAC modules to pilot the electro-pneumatic converters.

A hard-wired thermal interlock system triggered by negative temperature coefficient sensors (NTC) automatically cuts power to individual silicon modules should their temperature exceed safe values (in case of coolant flow or de-lamination of a particular module from its cooling channel, etc)

Pressure at the pump outlet (state 5A) can be regulated via PID variation of the pump speed.

• Which liquid to choose?

The ideal would be to evaporate at a pressure below atmospheric in order to avoid leaks, yet not too low so that thinner cooling ducts can be used.

On the other hand, the pressure drop in the return line must be lesser or equal to the difference between evaporation and condensation pressure (if it is lesser, additional head losses can be added).

An estimation will be made for the pressure drop endured by the coolant from the moment it enters the stave as saturated liquid at 15°C to the reservoir where condensation takes place.

a) estimation of pressure loss in one stave

Relevant Properties for head losses calculation:

	Density at 15ºC Kg/m3		Kinematic viscosity m^2/s	
	Sat liquid	Sat vapour	Sat liquid	Sat vapour
C_5F_{12}	1623.8	7.23	3 x10^{-7}	5 x 10^{-7}
C_6F_{14}	1747.2	2.72	4.37x10^{-7}	5 x 10^{-7}
C_4F_{10}	1536	20.75	1.62x10^{-7}	5 x 10^{-7}

The equivalent hydraulic diameter for the cooling ducts is

$$D_H = 4.\frac{Area}{wettedperimeter} = 4.\frac{(2.8-0.6)\times 0.6+p\times\frac{0.6^2}{4}}{(2.8-0.6)\times 2+p\times 0.6} = 4.\frac{1.6027}{6.285} = 1.02mm$$

Theoretical estimation of frictional pressure drop for two-phase flow is not only complex but also of uncertain accuracy so it will not be attempted here. Instead, estimations were made for two hypothetical cases: (i) assuming 100% of the mass flow as liquid all along the stave and (ii) 100% of the mass flow as vapor all along the stave. As can be seen below, due to the fact that pressure losses depend loosely from the square of the velocity, the flow would be laminar in case (i) and turbulent in case (ii). Since the staves are of reduced length (L=288 mm) the approximation adopted will be to take the average between (i) and (ii).

	Reynolds - if only liquid-	Reynolds - if only vapor-	pressure loss - if only liquid- (mbar)	pressure loss - if only vapor- (mbar)	Average (liq+vap)/2 (mbar)	Corresponding T glide along the stave °C
C_5F_{12}	287 (laminar)	38731 (turb.)	3.7	81.8	42.7	2
C_6F_{14}	175 (laminar)	98271 (turb.)	5.1	164.5	84.8	12
C_4F_{10}	639 (laminar)	15335 (turb.)	2.3	44.3	23.3	0.5

Note : Pressure losses were calculated with:

$$\Delta P = \frac{64}{\mathrm{Re}} \frac{1}{2} V^2 \frac{L}{D} r \quad \text{Laminar flow}$$

$$\Delta P = 0.184\,\mathrm{Re}^{-0.2} \frac{1}{2} V^2 \frac{L}{D} r \quad \text{Turbulent flow}$$

b) Pressure loss estimation for one vapor return line (segment within the detector)

The diameter within the detector has to be reduced to the bare minimum, so three values were considered: 10, 12 and 15 mm. The length of the pipe is not known yet, nor is the number of elbows. For one elbow, the equivalent length in pipe diameters can be assumed as 20 ~40. This means that, for instance, it would take 10 elbows to introduce an additional equivalent length of 1 m in linear losses for a 10mm pipe.

Pressure losses in mbar

Inner Diameter	10 mm ID		12 mm ID		14 mm ID	
Equivalent length	1m	25m	1 m	25m	1m	25 m
C_5F_{12}	26.4	660	11	275	5.3	131.2
C_6F_{14}	54.5	1363	23	568	10.8	271
C_4F_{10}	14.6	365	6	152	2.9	72.5

c) Pressure loss estimation for vapor return line outside the detector

Once the pipe exits the detector, the diameter can be enlarged to reduce pressure losses to negligible values (note that pressure losses are loosely proportional to $1/\varnothing^{4.8}$ for turbulent regime). Each group of N individual lines will then converge into a collector which in its turn leads to a back pressure regulator. Finally, all the 60/N back pressure regulator's outlets will converge to a single line returning to the condenser.

Why evaporate?

6 reasons:

1. To enhance the heat transfer

The heat transfer coefficient changes progressively as the refrigerant evaporates along the tube. The vapor fraction increases and the agitation thereby caused enhances the heat transfer coefficient. When the refrigerant is nearly all vaporized, the coefficient drops off to the magnitude applicable to vapor transfering heat by forced convection.

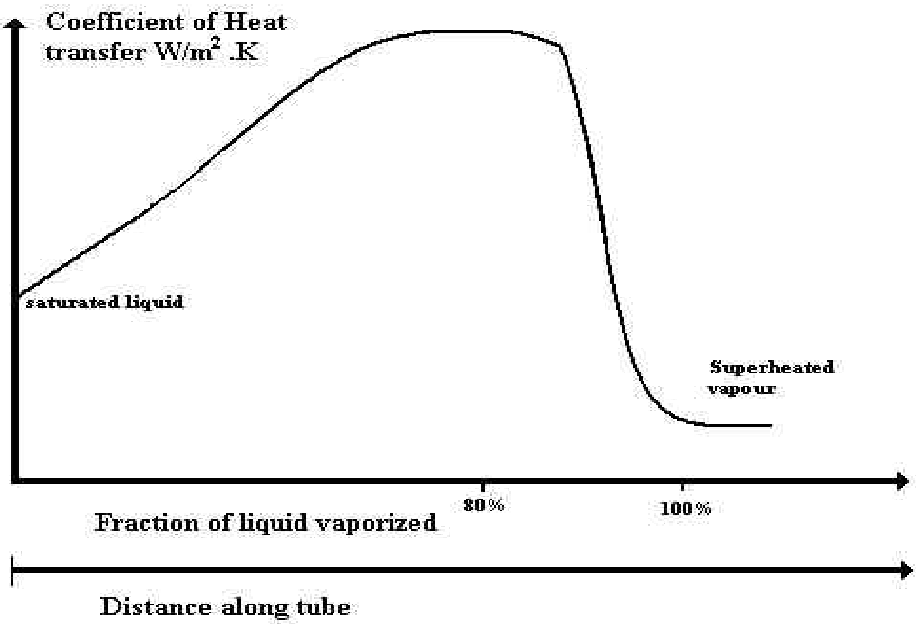


Fig 1 shows very clearly that the heat transfer coefficient increases significantly from liquid convection (left of the curve) to liquid boiling (peak of the curve)

V. Vacek and Greg Hallewell measured coefficients in the order of 2000 to 5000 W/m^{2}K for similar perfluorocarbons evaporating along a 3.6mm ID capillary.

2. To reduce the temperature increase along the stave

The heat transferred to the liquid is used for phase change and not to increase its temperature, so the temperature increase along the stave is very small. Temperature uniformity is a common requirement for HEP detector cooling

3. To reduce the circulating coolant mass

The latent heat of evaporation is much higher than the sensible heat.
example: for C3F8 at -30°C
latent heat = 104 kJ/kg
sensible heat = 1 kJ/kg per K of liquid temperature increase

so
1 kW can be removed at -30°C by evaporating 9.62 g/s of liquid C3F8
or…
by increasing 1 K the temperature of a 1000 g/s flow of liquid C3F8

4. To reduce pipe dimensions (lower material budget)

Smaller flow rates ⇒ smaller pipe diameter.
Smaller flow rates ⇒ smaller pressure drops ⇒ thinner pipes may be used.

5. Possibility to have parallel channels at different temperatures

By evaporating at different pressures (ex. by means back pressure regulators). Such control is impossible in a parallel liquid cooling system with a single liquid supply temperature…)

6. Little pipe insulation required

Insulation is only needed downstream of the expansion device, which usually is very close to the detector.

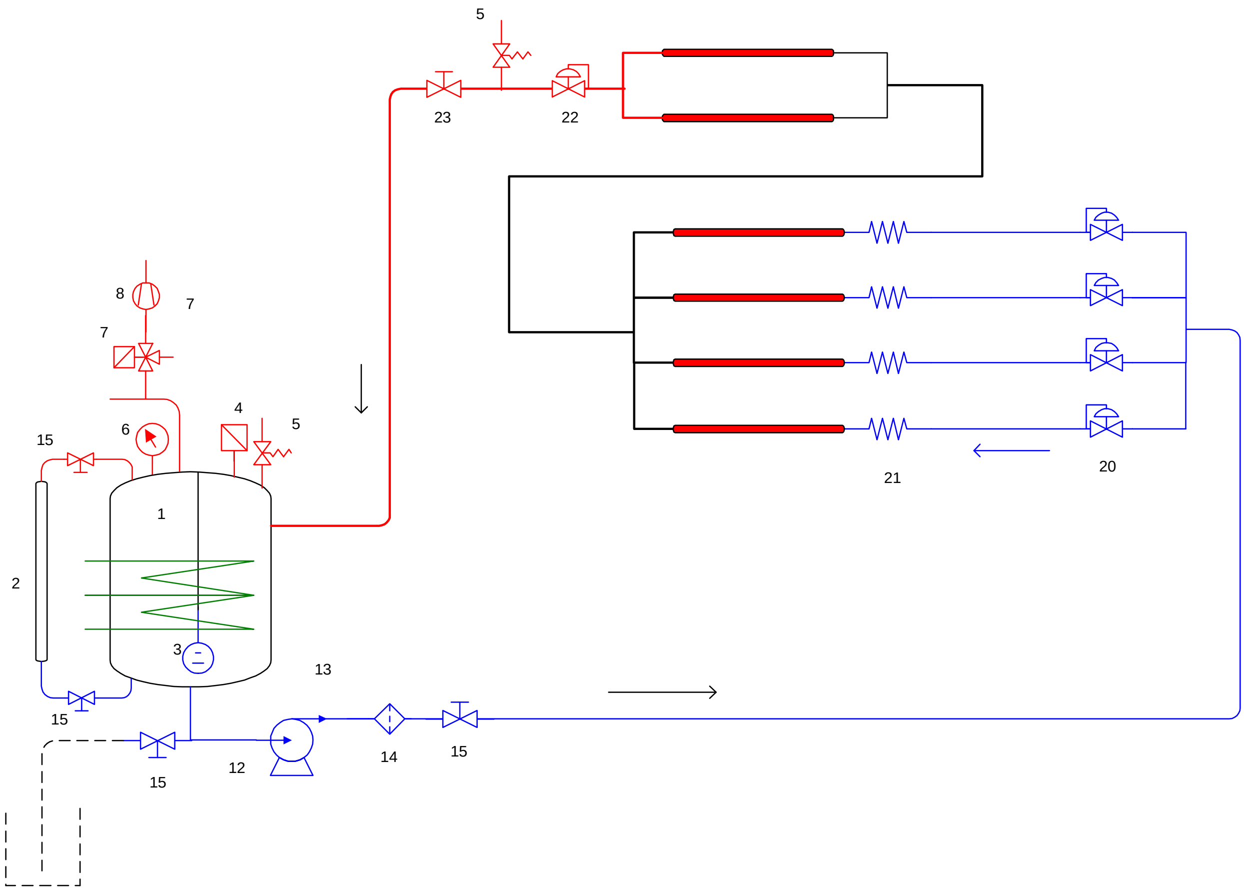
Price estimation

a)Test Unit

aims:

- To determine an overall heat-transfer coefficient for the cooling duct-chips.
- To determine the ideal evaporation temperature to obtain 20 °C on the chip surface.
- To determine the pressure loss along the cooling ducts
- To compare two configurations of access to SPD barrel

Config 1: service the SPD barrel from one side



Config 2: service the SPD barrel from both sides