# Evaporative Cooling Calculations

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# Topics

Assumptions

Coolant Flow versus Power

Two Phase Flow in Stave and Sectors

Exhaust Vapor Pressure Differences

Summary

# Assumptions:

1. Two staves or sectors in series per cooling circuit. 8.4 W per module.

2. Coolant inlet temperature = +20 C
   Coolant exhaust temperature = -20 C or -25 C

3. Coolant inlet quality factor = 0.4
   Coolant exhaust quality factor = 0.9

4. Some data for C3F8 (mostly from 3M):
   Heat of vaporization at -20 C = 94.8 J/g
   Vapor pressure at -20 C = 2.036 bar
   Vapor dynamic viscosity at -20 C = 9.e-6 Pas*s
   Liquid kinematic viscosity at +20 C = 2.3e-7 m**2/s
   Vapor Cp = 0.65 J/g, k = 1.2

5. Some data for C4F10 (mostly from 3M):
   Heat of vaporization at -20 C = 102.1 J/g
   Vapor pressure at -20 C = 0.464 bar
   Vapor dynamic viscosity at -20 C = 9.e-6 Pas*s
   Liquid kinematic viscosity at +20 C = 2.6e-7 m**2/s
   Vapor Cp = 0.79 J/g, k = 1.2

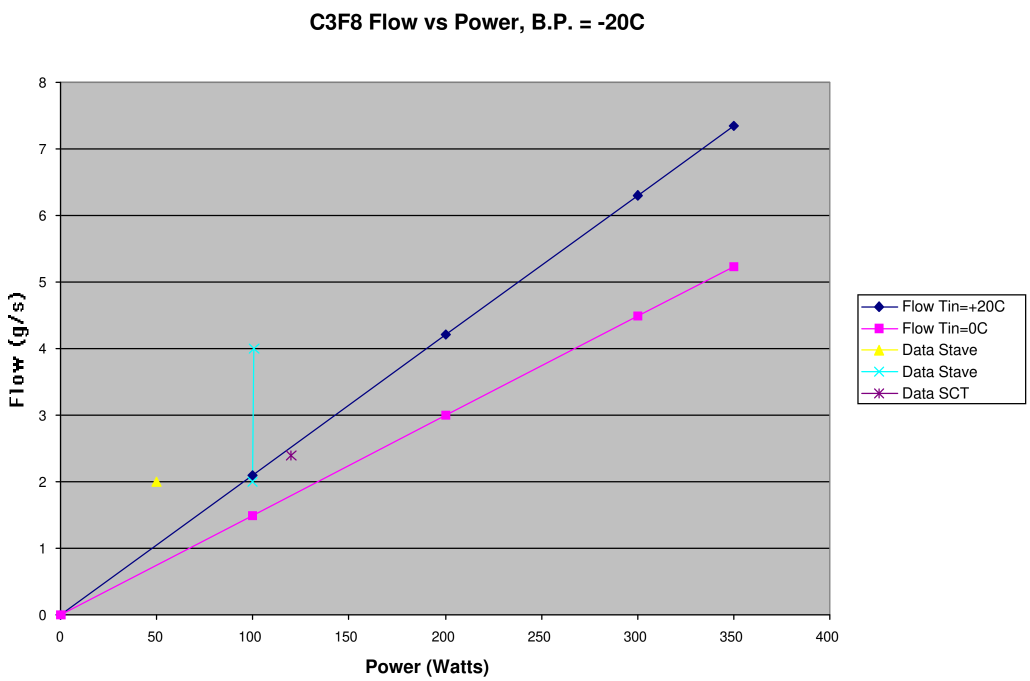
6. Piping diameters and lengths

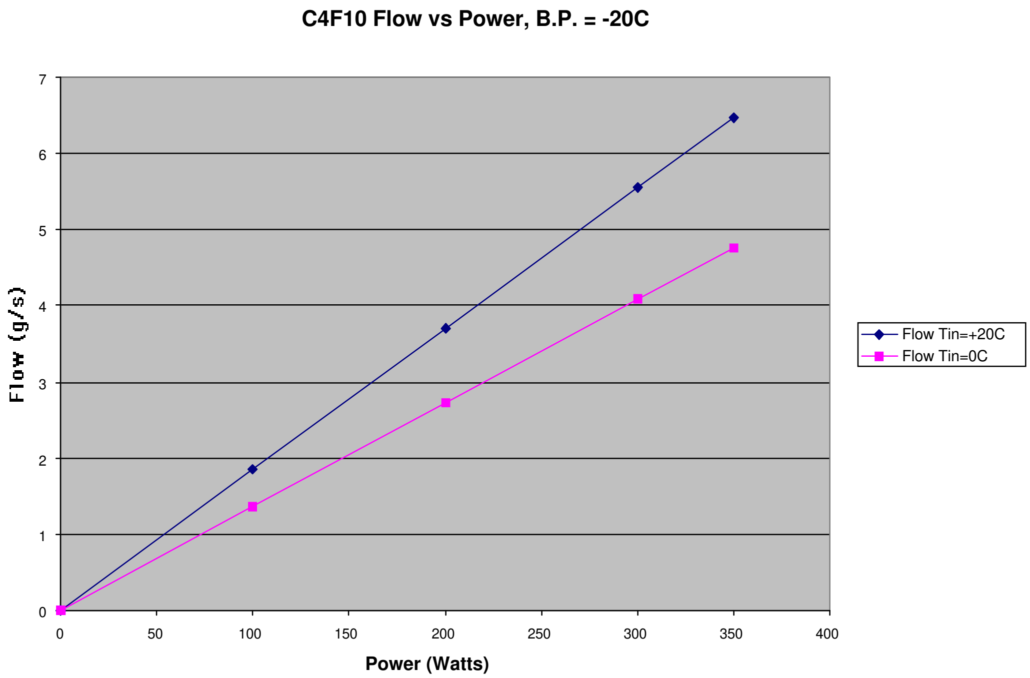
| | Min. L | Max. L | Inlet ID | Outlet ID | Nr. Bends |
|---|---|---|---|---|---|
| Stave-PPB1 | 0.99 m | 1.49 m | 1. mm | 7. Mm | 5 |
| PPB1 -PPB2 | 6.26 | 9.41 | 2. | 9. | 4-6 |
| PPB2 -PPB3 | 10.2 | 10.2 | 3. | 13. | 2 |
| PPB3 –Rack | 6.0 | 24.0 | 4. | 13. | 2 |
| Totals | 23.45 m | 45.10 m | | | |

7. Bends and ID changes were included in pressure difference calculations.

# Coolant Flow versus Power

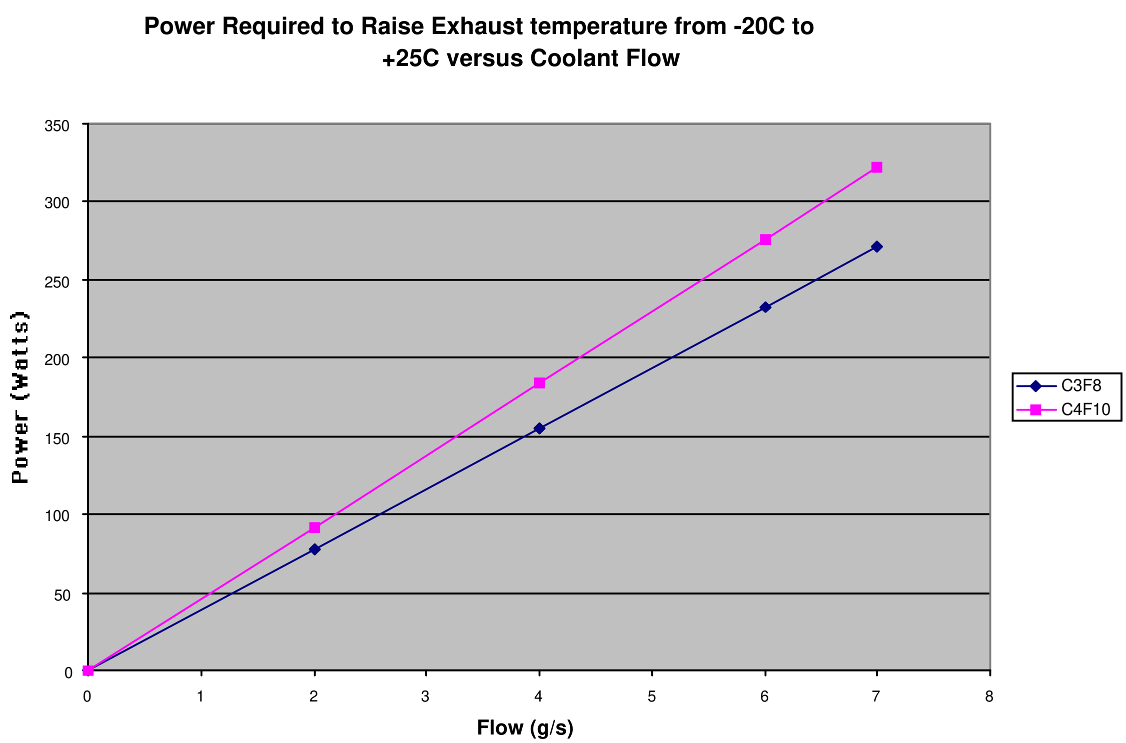
Coolant flow as a function of power for C3F8 and C4F10 have been calculated from Enthalpy tables supplied by 3M. The Enthalpy of the liquid at the assumed inlet temperature (e.g. +20C) was subtracted from the Enthalpy at the assumed boiling temperature (e.g. –20C). For an inlet temperature of +20C this implied an inlet quality of approximately 0.4. An outlet quality of 0.9 was assumed.




C4F10 Flow vs Power, B.P. = -20C
Flow (g/s)
0
1
2
3
4
5
6
7
0
50
100
150
200
250
300
350
400
Power (Watts)
Flow Tin=+20C
Flow Tin=0C

# Power Required to Increase Temperature of the Exhaust from –20C to +25C

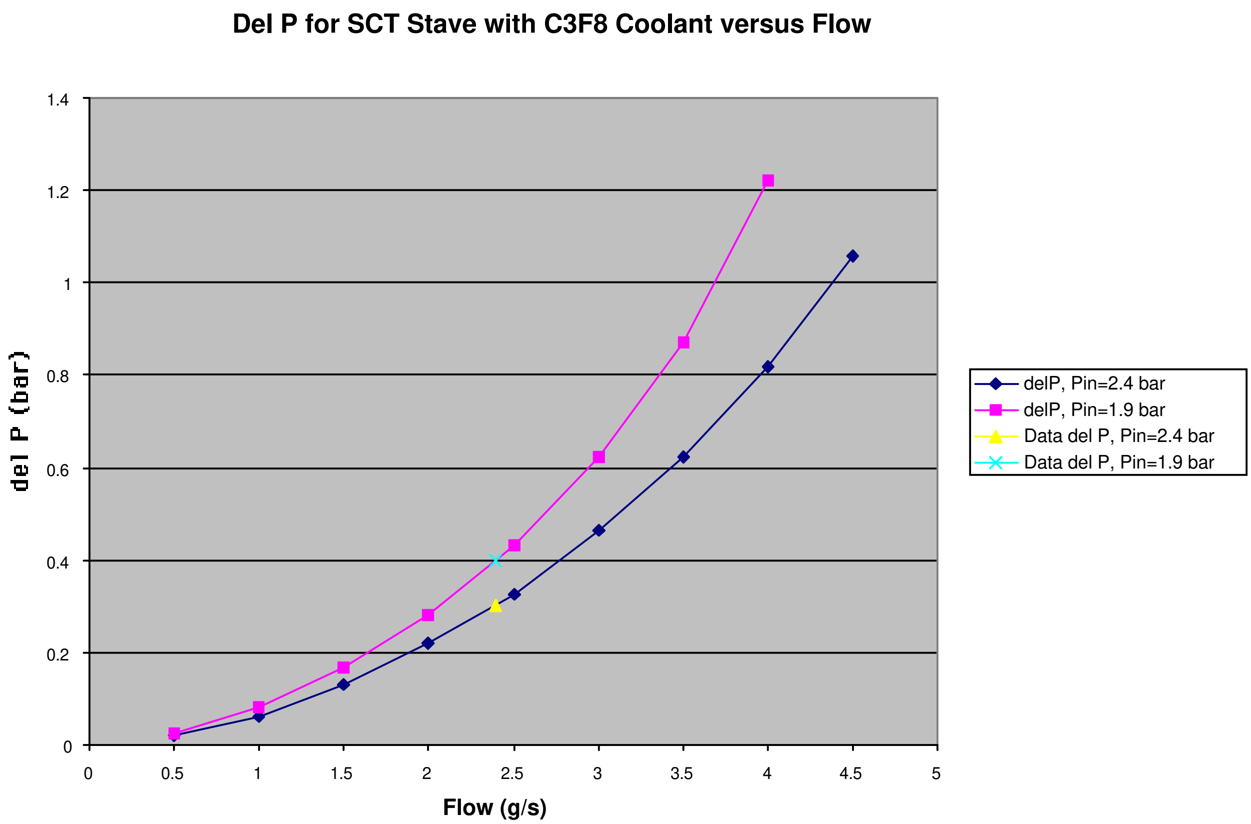
Using an outlet quality of 0.9 and the heat capacity of the coolant vapor at constant pressure (Cp) the power required to increase the temperature of the coolant exhaust was calculated.



# Quasi Two Phase Calculation of Pressure Drop in Staves and Sectors

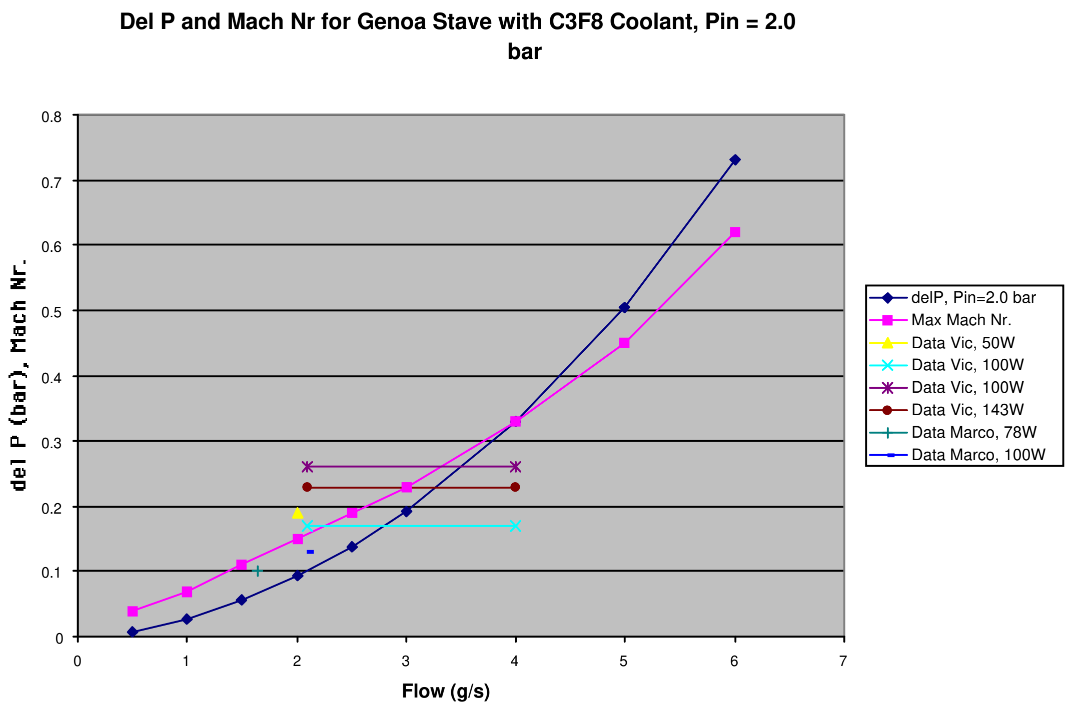
Based on following:

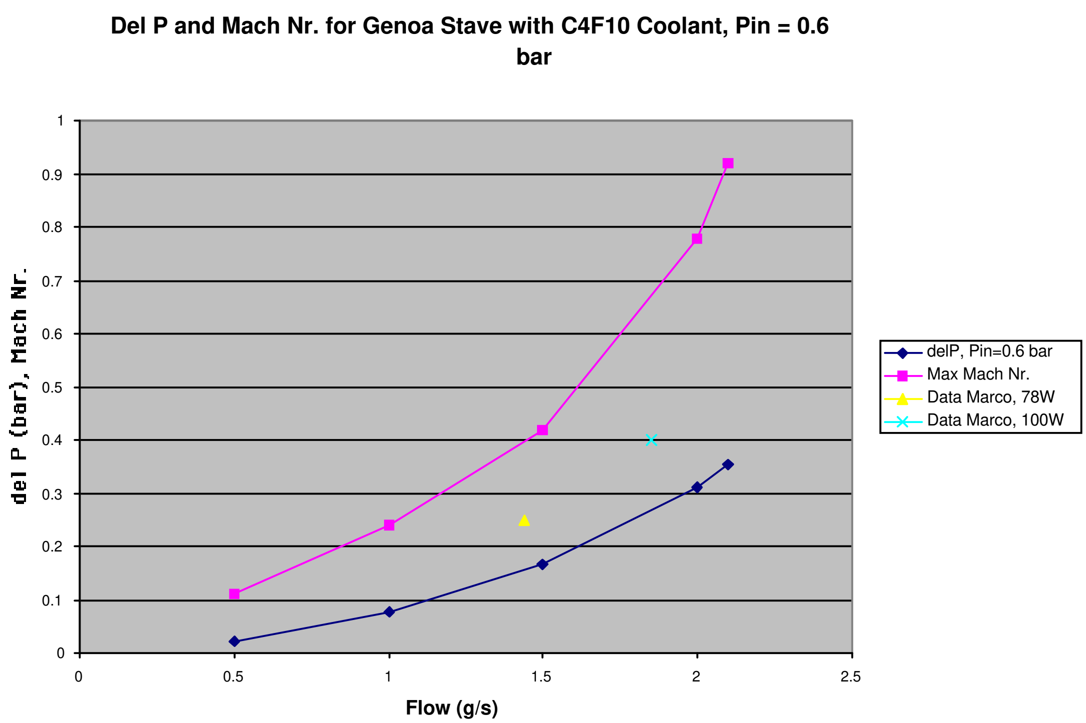
> The evaporating liquid coolant in a stave or sector occupies approximately 5% of the channel volume if there is no accumulation or pooling.
>
> Assume the pressure drop is entirely due to vapor flow at isothermal conditions
>
> Assume the liquid is evaporated uniformly along the length of the coolant channel.
>
> Assume the liquid reduces the hydraulic diameter by 5%



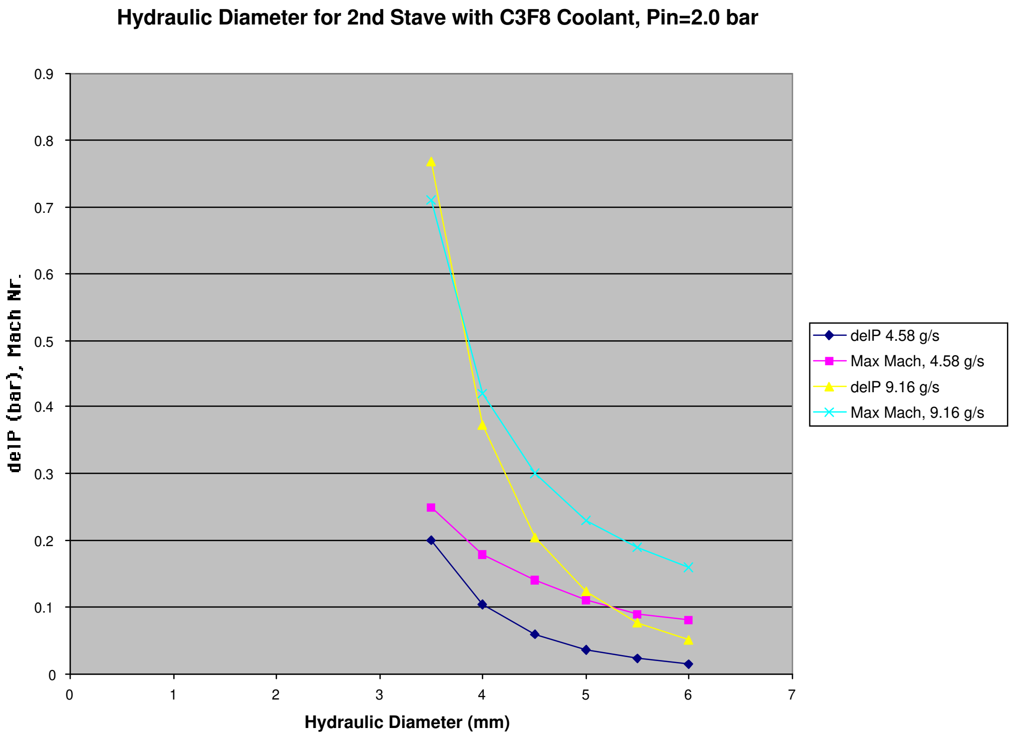
# Calculation versus data for Genoa Stave

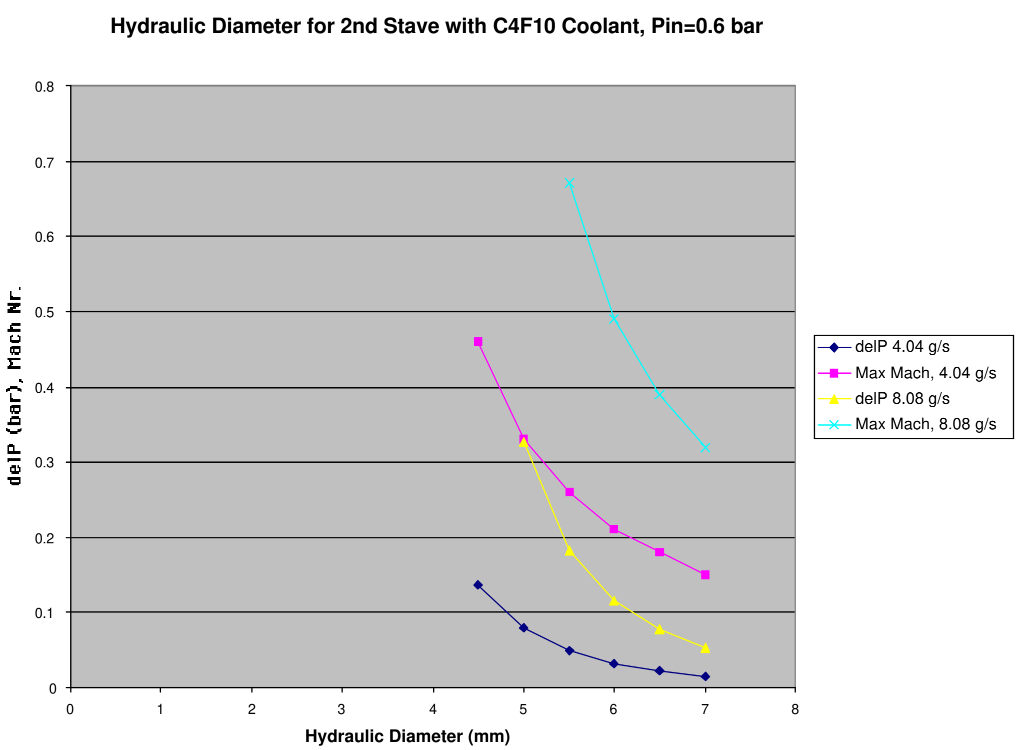
Not all data parameters available for a given test.  Assumed average hydraulic diameter = 2.9 mm from a measurement at LBNL.



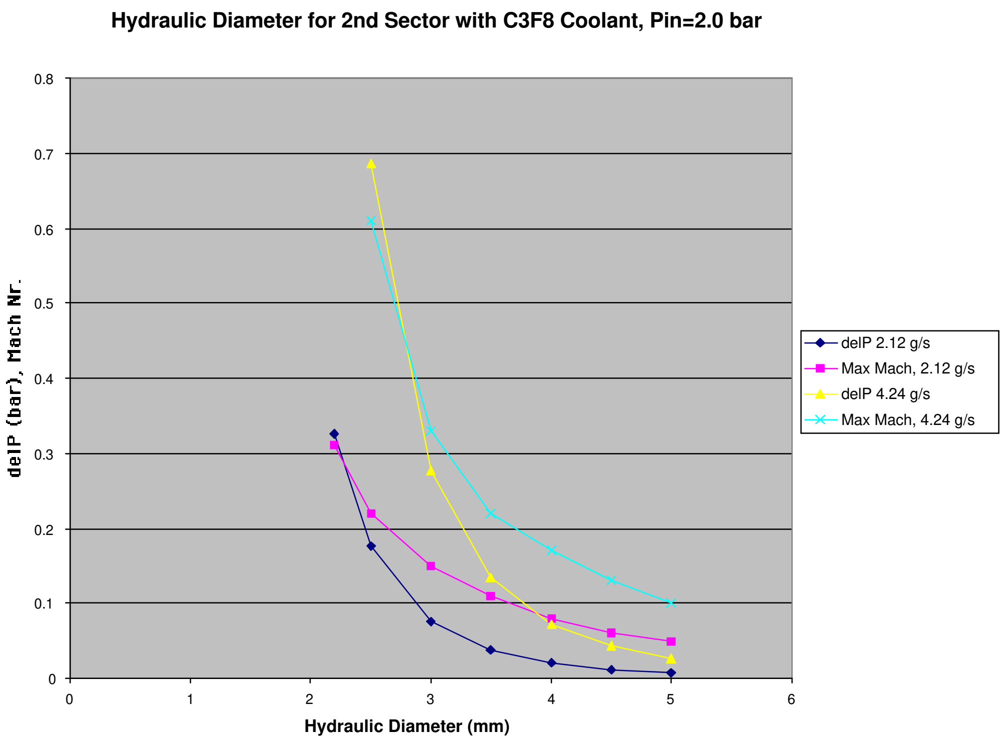



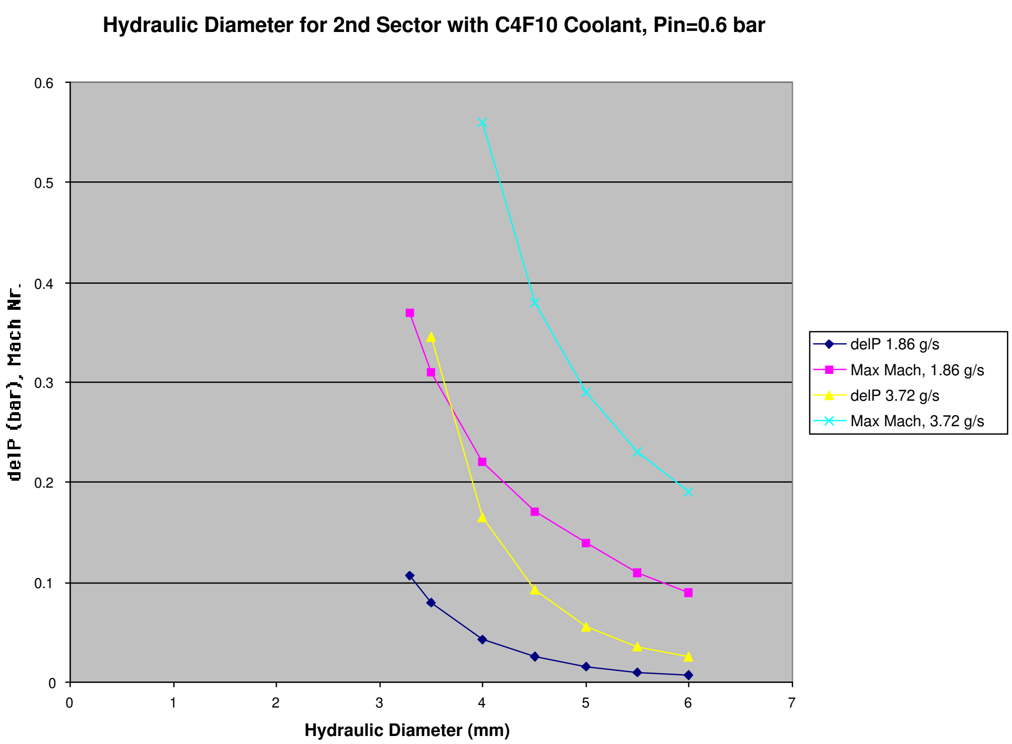
# Hydraulic Diameter of Staves for Second Stave at Nominal and Two Times Nominal Coolant Flow.



# Hydraulic Diameter of Sectors for Second Sector at Nominal and Two Times Nominal Coolant Flow.

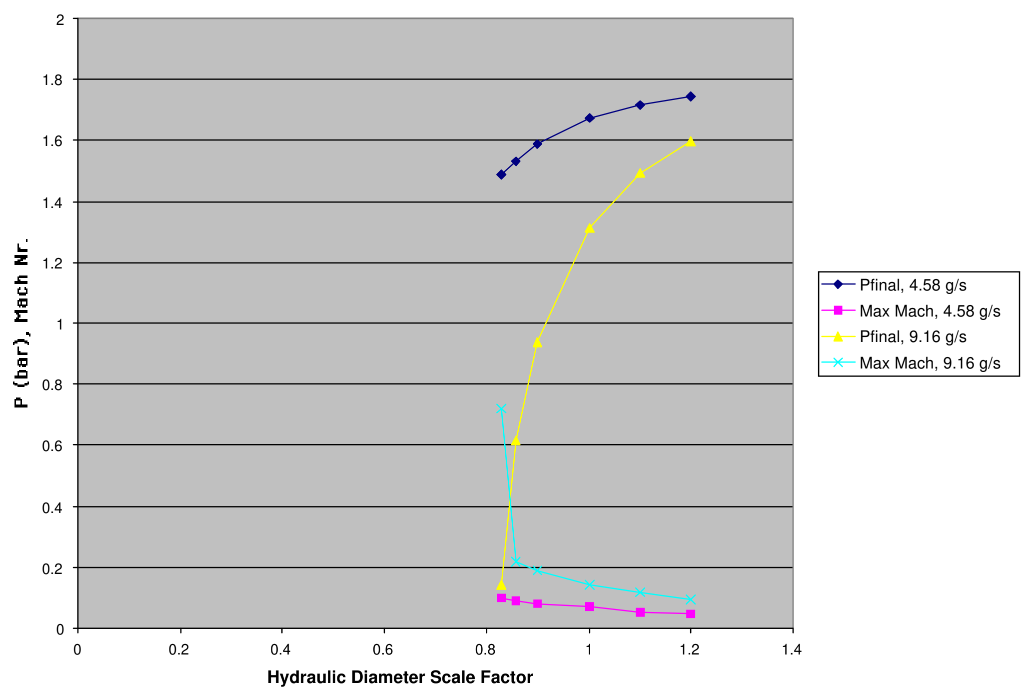


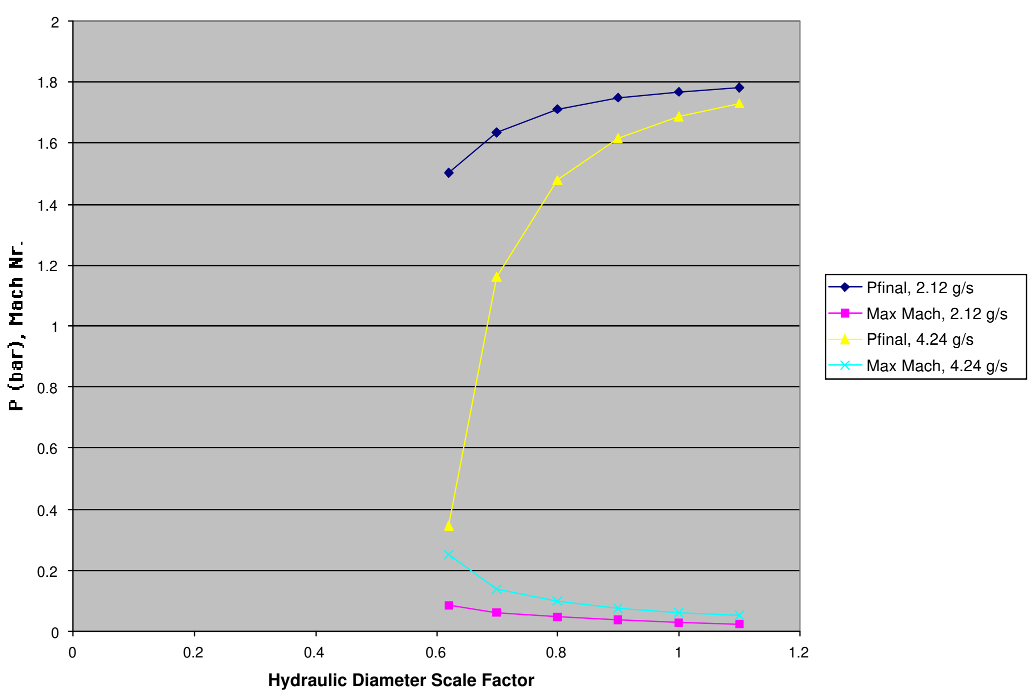
# Exhaust Tubing Sizes for C3F8.

Baseline tubing sizes assumed and then scaled. Effects of bends and ID changes included in calculation. Temperature of vapor beyond PPB1 is +25C.

**P at Rack for C3F8 Stave Exhaust vs Tube ID, Stave Pout=1.8 bar**

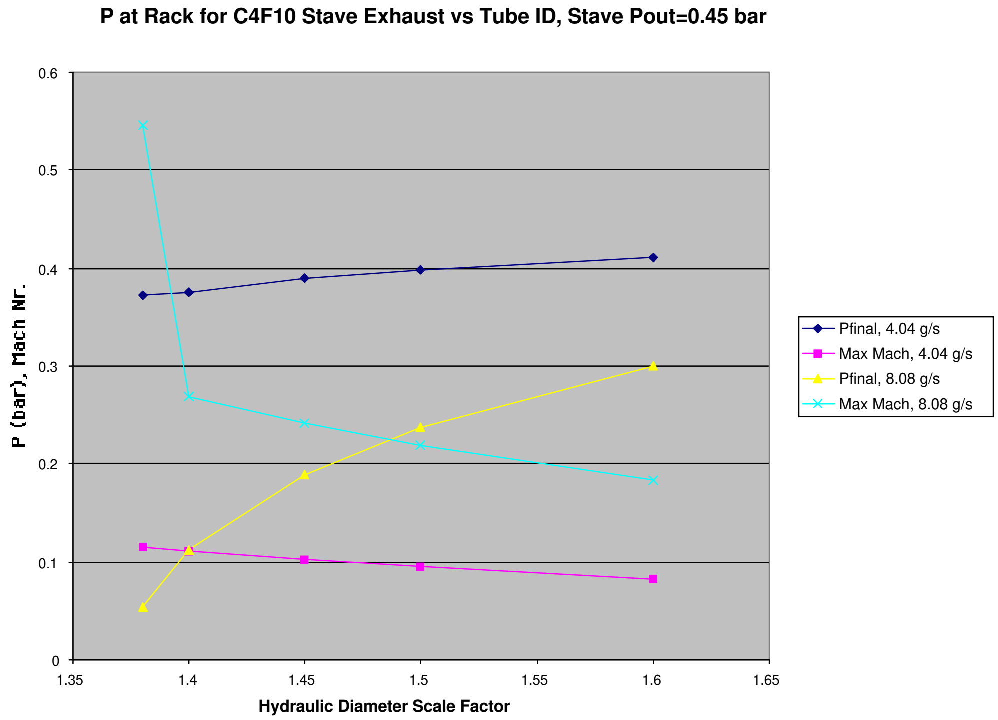


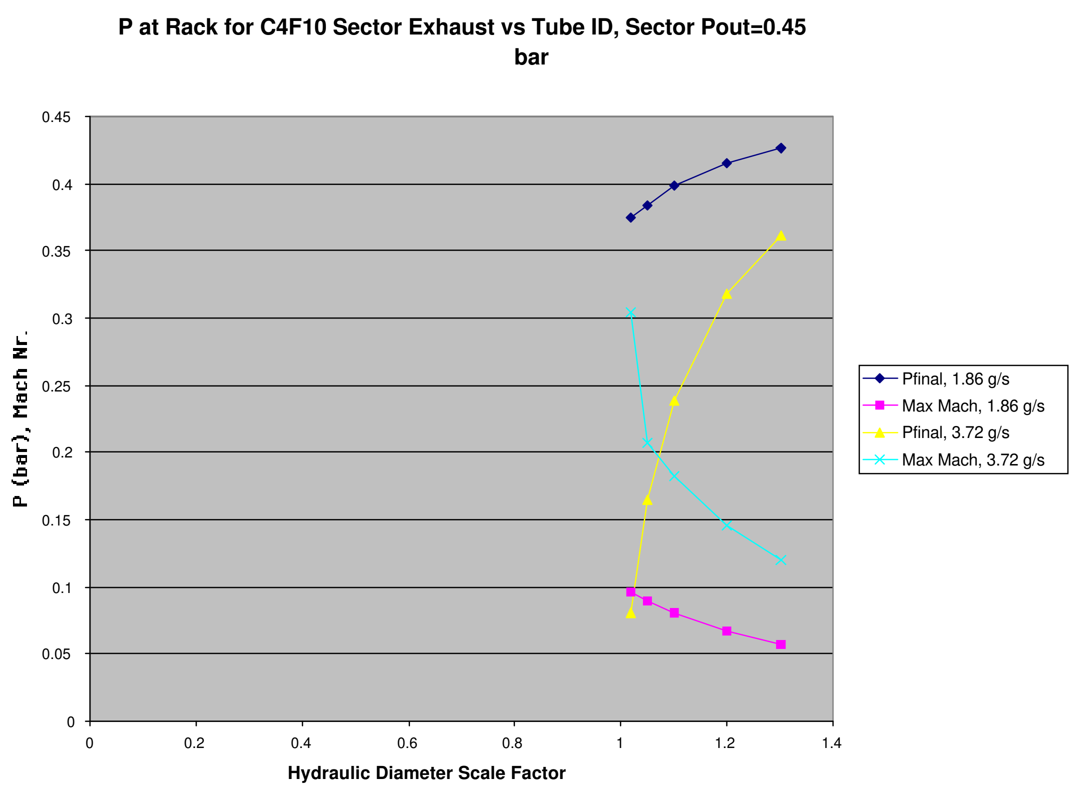

**P at Rack for C3F8 Sector Exhaust vs Tube ID, Sector Pout=1.8 bar**



# Exhaust Tubing Sizes for C4F10.

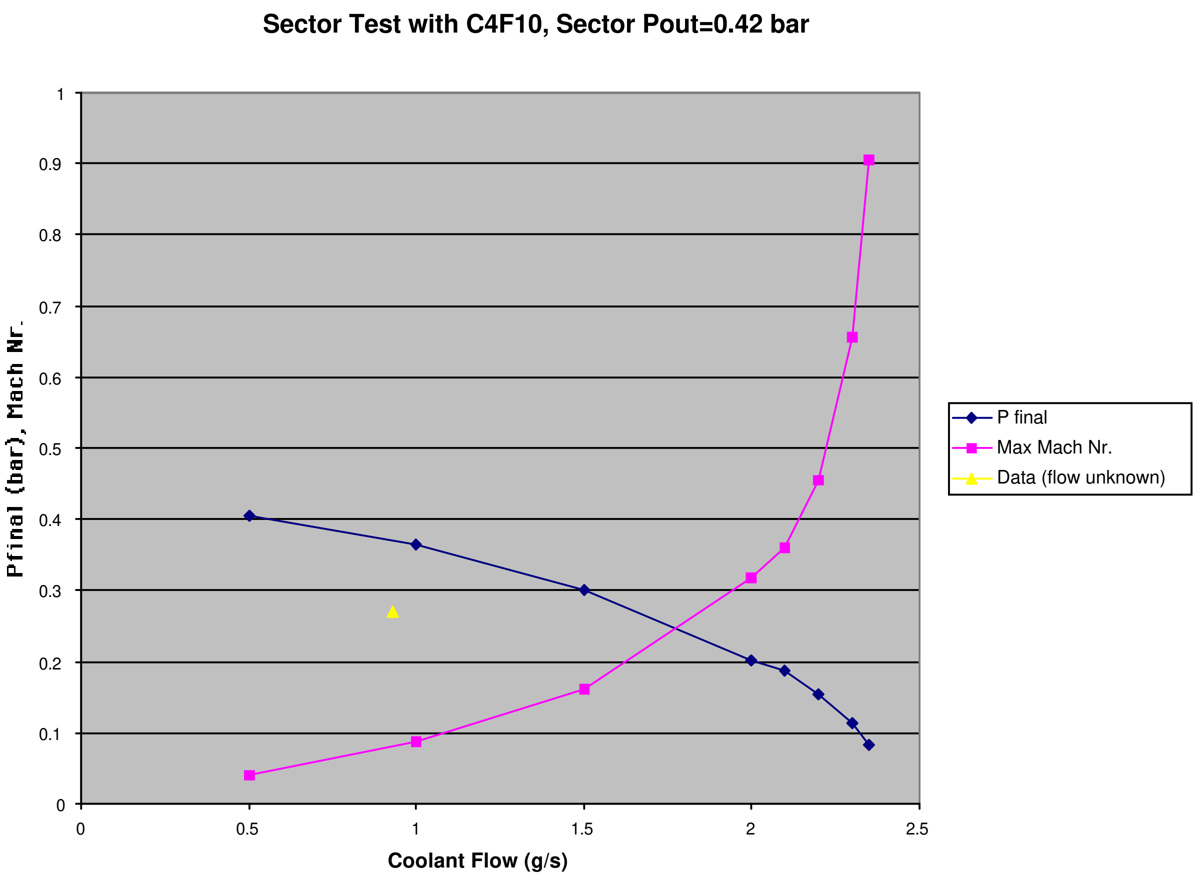
Baseline tubing sizes assumed and then scaled. Effects of bends and ID changes included in calculation. Temperature of vapor beyond PPB1 is +25C.



# Sector Test with C4F10.

Eric Anderssen tested a sector with C4F10 and observed a sudden rise in sector out pressure when increasing coolant flow. The exhaust tube ID was 6 mm and its length estimated as 6.7 meters. The pressure at the buffer (P final) was 0.27 bar. The power on the sector was 50 Watts which implies a coolant flow of 0.93 g/s. The Mach Number may have exceeded 1 when Eric increased the flow thus causing the sudden rise in sector out pressure.



# Summary

1. More comparison with data is needed.

2. Hydraulic Diameters of second stave or sector in series estimated as follows for a factor of two flow contingency (8.4 W/module), for maximum Mach number less than 0.5 and for a pressure difference across the two staves or sectors less than approximately 0.3 bar for C3F8 or 0.2 bar for C4F10.

| Device | Coolant | Hydraulic Diameter |
|---|---|---|
| Stave | C3F8 | 4.5 mm |
| Stave | C4F10 | 5.8 mm |
| Sector | C3F8 | 3.3 mm |
| Sector | C4F10 | 4.3 mm |

3. For C3F8 coolant:
   Present stave exhaust tubing IDs approximately correct for factor of two flow contingency.
   Present sector exhaust tubing IDs could be reduced by approximately 15% to 20%.

   For C4F10 coolant:
   Present stave exhaust tubing IDs would need to be increased by approximately 45% for factor of two flow contingency.
   Present sector exhaust tubing IDs would need to be increased by approximately 10%.

4. Inlet pressure differences from Rack to start of the expansion device are approximately 1.2 bar for C3F8 and 1.0 bar for C4F10 for the long inlet route and for nominal flow.