Lawrence Berkeley Laboratory - University of California **Engineering Note**		Code SR0208	Serial M7664	STAR Doc # SN0318	Page 1 of 7
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Program
Solenoidal Tracker at RHIC

Sub program
Time Projection Chamber - Cooling System

Title
Calculations for Gas to Water Heat Exchanger

1.0 Introduction

Physics considerations require that the TPC gas be maintained at a steady temperature +/- 1 degree C. A steady-state temperature of 75F was selected prior to construction and now fixed by thermal expansion considerations. Gas flowing from the mixing room travel through roughly 95 meters of metal tubing before entering the TPC. Ambient temperatu Assembly Building and Wide Angle Hall is allowed to vary between 70F and 80F. Given relatively long supply line and low heat capacity of the gas, P10 entering the TPC could significantly warmer or cooler than specified unless a point of use heat exchanger is incorporated. At the STAR Magnet the flow is split equally between inlets at the TPC's e and west Wheels (end caps). The following note covers the calculations used to design of tube-in-tube, water to gas, heat exchanger for use in the final legs of the TPC gas sup line.

2.0 Overall Heat Transfer Coefficient

2.1 Convective heat transfer coefficient for water on OD.

2.1.1 Flow velocity and heat transfer coefficients for a range of flow rates.

2.1.1.1 Heat exchanger geometry

Inner tube diameter is fixed at 2.5 " OD.

$$D_1 := 2.5 \cdot in$$

Jacket tube, limited by availability to 3 " OD with either 0.049" or 0.065" wall or sch 5 pipe (2.875" OD, 0.083" wall). Selecting the pipe minimizes the annular which minimizes the water flow for a given heat transfer coefficient.

$$D_2 := 2.875 \cdot in \qquad t_w := 0.083 \cdot in$$

Annular cross section: $$A_{an} := \pi \cdot \left[\left(\frac{D_2}{2} - t_w \right)^2 - \left(\frac{D_1}{2} \right)^2 \right]$$

Wetted Perimeter : $$P := \pi \cdot \left(D_1 + D_2 \right)$$

Hydraulic Diameter: $$D_h := \frac{4 \cdot A_{an}}{P} \qquad D_h = 0.203 \cdot in$$

2.1.1.2 Coolant flow velocity, pressure drop and Reynolds number

Calculate over a range of flow rates.

$i := 0..8$

Physical Properties of Water, 75

$\rho := 996.6 \frac{kg}{m^3}$

$\mu := 9.2 \cdot 10^{-4} \cdot \frac{kg}{m \cdot sec}$

$k_w := 0.609 \frac{watt}{m \cdot K}$

$C_w := 4179 \frac{joule}{kg \cdot K}$

$Pr := 6.31$

Flow rate range: $Q_i := 5.0 \frac{gal}{min} + i \cdot 1.0 \frac{gal}{min}$

Velocity : $v_i := \frac{Q_i}{A_{an}}$

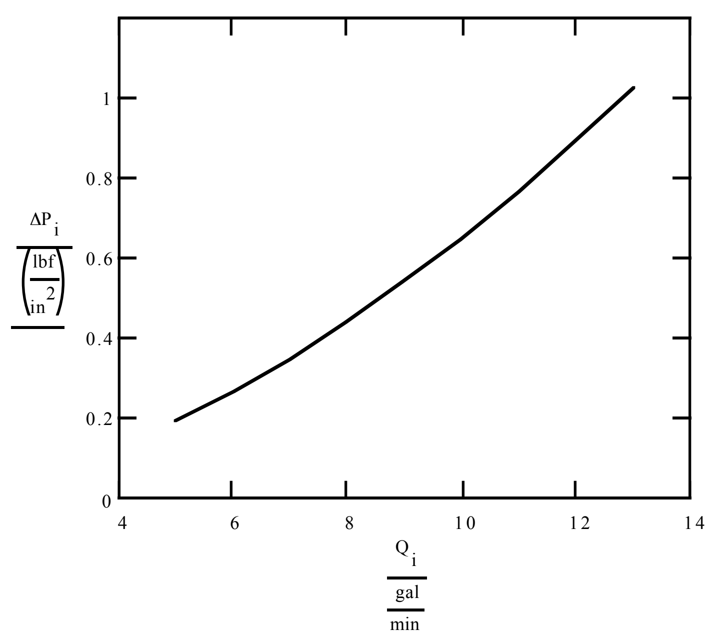
Reynolds number: $Re_i := \frac{v_i \cdot D_h \cdot \rho}{\mu}$

Friction factor for smooth pipe: $f_i := \frac{0.3164}{(Re_i)^{.25}}$

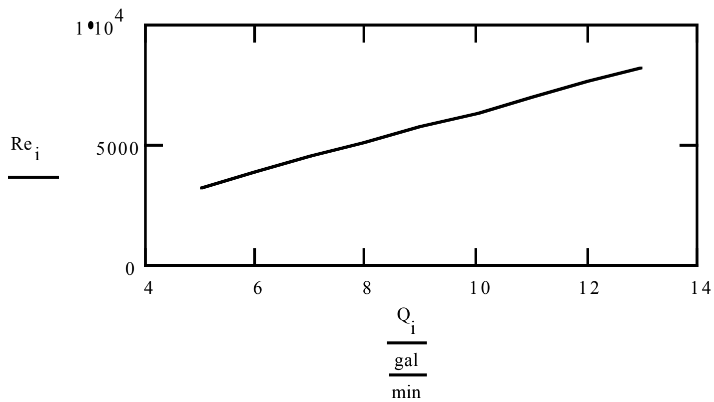
Unit length of heat exchanger: $Le := 1 \cdot m$

Head loss due to friction: $hl_i := f_i \cdot \left[\frac{(v_i)^2}{2 \cdot g} \cdot \frac{Le}{D_h}\right]$

Conversion from head to pressure drop: $\Delta P_i := hl_i \cdot \rho \cdot g$



$$Q = \begin{bmatrix} 5 \\ 6 \\ 7 \\ 8 \\ 9 \\ 10 \\ 11 \\ 12 \\ 13 \end{bmatrix} \cdot \frac{gal}{min}$$



$$Re = \begin{bmatrix} 3.187\cdot 10^{3} \\ 3.824\cdot 10^{3} \\ 4.462\cdot 10^{3} \\ 5.099\cdot 10^{3} \\ 5.736\cdot 10^{3} \\ 6.374\cdot 10^{3} \\ 7.011\cdot 10^{3} \\ 7.648\cdot 10^{3} \\ 8.286\cdot 10^{3} \end{bmatrix}$$

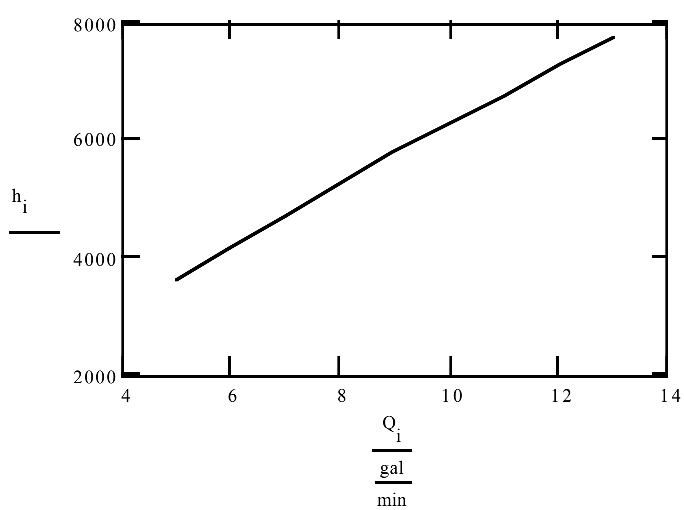
2.1.1.3 Convective heat transfer coefficient for turbulent flow.

Transition from laminar to turbulent flow in smooth pipes generally occurs at Reynolds numbers between 2300 and 4000. For the given geometry, flow ra about 5 gpm or more will result in turbulent flow.

Heat transfer coefficients for turbulent flow in pipes can be calculated from relation recommended by Dittus-Boelter for fully developed flow, where Nu is Nusselt number:

$$Nu_i := 0.023\cdot\left(Re_i\right)^{.8}\cdot Pr^{.4}$$

Solving for the heat transfer coefficien $h_i := \frac{k_w}{D_h}\cdot Nu_i$



$$h = \begin{bmatrix} 3.612\cdot 10^{3} \\ 4.179\cdot 10^{3} \\ 4.727\cdot 10^{3} \\ 5.26\cdot 10^{3} \\ 5.78\cdot 10^{3} \\ 6.288\cdot 10^{3} \\ 6.786\cdot 10^{3} \\ 7.275\cdot 10^{3} \\ 7.757\cdot 10^{3} \end{bmatrix}\cdot\frac{watt}{m^{2}\cdot K}$$

2.2 Convective heat transfer coefficient for P10 gas at tube ID

2.2.1 Geometry and parameters

Physical Properties of P10 Gas

$\rho_{P10} := 1.513 \frac{kg}{m^3}$ $\qquad$ $k_{P10} := 0.14 \frac{watt}{m \cdot K}$

$\mu_{P10} := 1.934 \cdot 10^{-5} \cdot newton \cdot \frac{sec}{m^2}$ $\qquad$ $C_{P10} := 599 \cdot \frac{joule}{kg \cdot K}$ $\qquad$ $Pr_{P10} := \frac{C_{P10} \cdot \mu_{P10}}{k_{P10}}$

Total gas flow through the TPC is fixed at 600 liters per minute with equal flow t end cap. Flow through each end of the TPC, Qgas is thus:

$$Q_{gas} := \frac{600}{2} \cdot \frac{liter}{min}$$

Inner tube is 304 Stainless Steel, 0.049" wall.

$$D_{gas} := D_1 - 2 \cdot 0.049 \, in$$

$$A_{gas} := \pi \cdot \left(\frac{D_{gas}{}^2}{4} \right)$$

$$v_{gas} := \frac{Q_{gas}}{A_{gas}}$$

$$Re_{gas} := \frac{v_{gas} \cdot D_{gas} \cdot \rho_{P10}}{\mu_{P10}}$$

$Re_{gas} = 8.163 \cdot 10^3$ $\qquad$ Turbulent flow.

2.2.2 Calculation of convective heat transfer coefficient for gas flow.

Following the steps in 2.1.1.3

$$Nu_{gas} := 0.023 \cdot Re_{gas}^{.8} \cdot Pr_{P10}^{.3}$$

The 0.3 exponent for the Prandel numbe consistent with the gas is being cooled.

$$h_{gas} := \frac{k_{P10}}{D_{gas}} \cdot Nu_{gas}$$

$$h_{gas} = 33.671 \cdot \frac{watt}{m^2 \cdot K}$$

2.3 Conduction through tube wall

Heat transferred from the gas to the liquid must pass through the tube wall. The thermal resistance of this element is determined by the conductivity of the meta Stainless Steel) and the wall thickness. The conduction term is calculated belo the summation of terms.

Selected Physical Properties of 304 Stainless Steel

$$k_{304} := 16.3 \frac{watt}{m \cdot K} \qquad C_{304} := 460 \cdot \frac{joule}{kg \cdot K} \qquad \rho_{304} := 7817 \frac{kg}{m^3}$$

2.4 Summation of terms for overall heat transfer coefficient, U

2.4.1 Reference geometry.

Using the inner tube OD and 1 meter length as the reference dimensions :

$$L_1 := 1 \cdot m \qquad h_w := h_0 \qquad h_w = \cdot \frac{watt}{m^2 \cdot K}$$

$$A_o := \pi \cdot D_1 \cdot L_1 \qquad r_o := \frac{D_1}{2} \qquad r_o =$$

$$A_i := \pi \cdot D_{gas} \cdot L_1 \qquad r_i := \frac{D_{gas}}{2} \qquad r_i =$$

2.4.2 Calculation of thermal resistance terms using the values derived above.

$$term1 := \frac{A_o}{A_i} \cdot \frac{1}{h_{gas}} \qquad term1 = 0.031 \cdot m^2 \cdot \frac{K}{watt}$$

$$term2 := \frac{A_o \cdot \ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot k_{304} \cdot L_1} \qquad term2 = 7.789 \cdot 10^{-5} \cdot m^2 \cdot \frac{K}{watt}$$

$$term3 := \frac{1}{h_w} \qquad term3 = 2.769 \cdot 10^{-4} \cdot m^2 \cdot \frac{K}{watt}$$

Note term 2 and term 3 are negligible so relatively large changes in the water flow wall thickness or material will have a very small affect on the overall heat transfer coefficient..

2.4.3 Calculation of overall heat transfer coefficient

$$U := \frac{1}{term1 + term2 + term3}$$

$$U = 31.984 \cdot \frac{watt}{m^2 \cdot K}$$

3.0 Heat Transfer and Heat Exchanger Length

3.1 Quantity of heat transferred.

Assuming the gas is at 80F, how much energy is transferred to reduce the temperature to 75F?

$$\Delta T := \frac{5}{1.8} \cdot K$$

$$q_0 := Q_{gas} \cdot \left(\rho_{P10} \cdot C_{P10} \cdot \Delta T\right) \qquad q_0 = \cdot watt$$

3.2 Log mean temperature differential.

The heat exchanger is assumed to be run counter flow with the gas passing from 1 to point 2 and the coolant flowing from 2 to 1. Th is the gas temperature and the coolant temperature. The gas is allowed to exit the heat exchanger 0.33 de C warmer than the inlet water temperature.

$$Th_1 := \frac{(80-32)}{1.8} \cdot K \qquad Tc_2 := \frac{(75-32)}{1.8} \cdot K \qquad Tc_2 = 23.889 \cdot K$$

Select the lowest water flow considered: $i := 0$

$$Tc_1 := \left(\frac{q_0}{Q_i \cdot \rho \cdot C_w}\right) + Tc_2 \qquad Tc_1 = 23.898 \cdot K$$

$$Th_2 := Tc_1 + .33 \cdot K$$

$$\Delta T_m := \frac{\left[\left(Th_2 - Tc_2\right) - \left(Th_1 - Tc_1\right)\right]}{\ln\left[\frac{\left(Th_2 - Tc_2\right)}{\left(Th_1 - Tc_1\right)}\right]} \qquad \Delta T_m = 1.157 \cdot K$$

3.3 Calculation of net heat transfer and heat exchanger length.

3.3.1 Heat transfer per meter of heat exchanger:

$$q := U \cdot A_o \cdot \Delta T_m \qquad q = 7.385 \cdot watt$$

3.3.2 Minimum length of heat exchanger:

$$L_{hx} := \frac{q_0}{q} \cdot 1 \cdot m \qquad L_{hx} =$$

4.0 Conclusion

Fabricating a 1.7 meter long heat exchanger with a 2.5" diameter x .049" wall inner tube and 2.5" nominal schedule 5 pipe outer cylinder provides a rate of heat transfer that is appropriate for this application. Gas supply pressure loss, calculated separately, is minimal. The required flow rate of 5 gallons per minute and pressure loss of less than 1 psi are easily achieved.