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# PROJECT REPORT 2012-002

# ENERGY CONSUMPTION AND THERMAL PERFORMANCE ISSUES OF EVAPORATIVE COOLING TOWERS OPERATING AT ZERO TOWER DISCARGE, WITH ELEVATED LEVELS OF TOTAL DISSOLVED SOLIDS, AND SOFT WATER

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## 1.0 INTRODUCTION

The Marley Cooling Tower Company presented a paper on the use of saltwater in evaporative cooling towers by Ting, et. al. [2] which draws the conclusion that for normal seawater at 34,500 ppm TDS, operating at up to 3 cycles of concentration ≈ 100,000 ppm in the recirculating water (effectively a 10% NaCl brine solution), the effects on thermal performance are negligible, roughly 2% to 5% lower than for the use of fresh water.

Cooling water systems are typically designed with between 25% and 33% excess capacity as a margin against variations in heat load, issues with the precision of the design of the piping with regard to friction losses, degradation of the system with age, and other variables beyond the designer's control. One place where you can readily observe this design strategy is in the pumps - typical systems have a spare pump installed in parallel with the operating pumps - a third or fourth pump of equal capacity to the others - to allow for fluctuations in the load place on the system by operational necessities. Up to 10% variation in the flow and heat rejection characteristics of the cooling water are well within the operational capabilities of the usual cooling system. Another key advantage of operating with the softened water used in the Zero Liquid Discharge technology is that the cooling water will tend to remove scale and bio-fouling on heat exchange surfaces, which will significantly reduce operational energy costs.

The high TDS, high pH water is bio-static, killing off and then removing biofilms. The Langelier Saturation Index (LSI) is less than -0.5, so existing lime scale and corrosion products will tend to be dissolved away from the insides of pipes, tubes, and tower splash fill, with significant benefits in fluid flow and heat transfer.

I have revisited the calculations performed by Marley's people and others, and sustain their conclusions that for an evaporative cooling tower with 100,000 ppm TDS in the circulating water (analogous to a 10% NaCl brine solution), the performance is close enough to the performance with fresh water as to make no practical difference.

**2.0 DISCUSSION**

The most noticeable change in the properties of the cooling water is the increase in density due to the dissolved solids. A tower at 100,000 ppm TDS (3 cycles of concentration for ordinary sea water, or effectively a 10% NaCl brine solution – a useful approximation for getting the physical properties for fluid mechanics and heat transfer) will have a specific gravity of ~1.1, giving a water density of ~68.6 lb/cuft, versus 62.4 for ordinary water. The other significant properties affected by the dissolved solids are viscosity and vapor pressure. Thermal conductivity of the cooling water is not significantly changed by operating at 100,000 ppm TDS.

Density directly affects the horsepower needed to pump the cooling water, and the dynamic pressure of the flowing water, which translates into the frictional losses due to flow.

As I demonstrate in the calculations in the Appendices, the pump horsepower at 100,000 ppm TDS is approximately 10% greater than for fresh water, and the frictional losses expressed as psi per 100 feet of run are ~10% higher than for fresh water. These pale in significance when compared to the effects of ~10 years accumulated fouling and tubercle growth, which can easily increase pumped water flow resistance by up to 700% when compared to a new installation!

Since most pump installations use a recycle loop to control the flow rate, 10% difference in fluid density is not really an issue in real world operations, or if it is, the operator can run the installed spare pump, and maybe add another spare pump in parallel, if the installation is close to its capacity limit.

As I've discussed, almost nobody is going to take the risk of getting a cooling tower with less than 25% excess capacity over their anticipated loads, and as the folks at Marley found out approximately 40 years ago, using seawater at 2 to 3 cycles of concentration presents no need for any special thermal concerns, and only straightforward attention to the materials of construction.

Following along on the theme of fluid mechanics, a 10% NaCl brine has a viscosity of 1.1 centipoise, compared to 1.0 for fresh water according to the graph on page A-4 in Crane's Handbook [3]. We use the viscosity to determine the Reynolds Number, which is in turn used to calculate Darcy's Friction Factor (f) from the Moody Diagram on page A-26 of Crane's. The friction factor (f) is then used in estimating the dynamic pressure loss for the flow.

The friction factor is only part of the calculation - the dynamic friction losses are proportional to the kinetic energy of the flowing fluid, which is proportional to the square of the velocity in ft/s (fps) in the pipes and tubes. Since water is an incompressible fluid, the velocity will vary as the square of the inner diameter of a pipe. The pipe shown in Figure 1 below has been constricted to approximately 60% of its original cross-sectional area by tubercles, a corrosion product often caused by corrosion in place and/or biological fouling and corrosive attack of the pipe walls under the biofilm (or similar) deposits (under deposit attack), especially in iron or steel pipes. The image in Figure 1 is fairly typical for a system with several years of active service.

**Figure 1: Typical Constriction of Pipe due to Tubercle Buildup**
**Flow area is constricted to ~60% of clean pipe, roughness: ε/d ≈ 0.12**

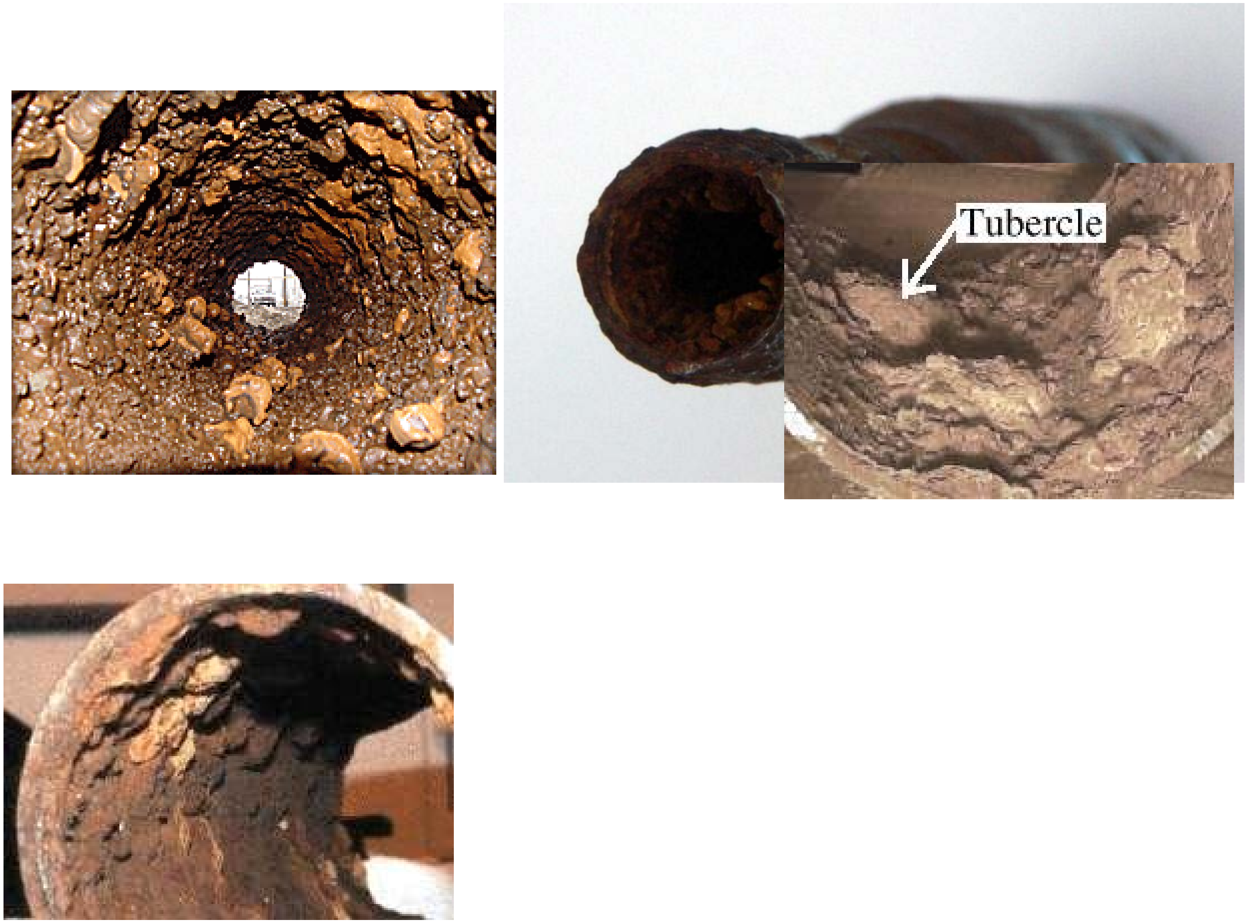


**Figure 2: Assorted Photographs of Tubercle formations due to Corrosion in iron/steel pipes**

The tube in Fig. 1 has approximately 60% of its original cross sectional area open for the flow of cooling water. This translates to 1/0.6 = 1.7 times greater velocity for the same flow of cooling water. To keep the water flowing 70% faster for the same volumetric flow in gpm, the frictional losses are ~700% of those for the originally installed pipe!

The recovery of even a portion of this 700% far outweighs the 10% increase in frictional resistance due to the increased density and viscosity due to operating with water at 100,000 ppm TDS @ 100 °F.

A similar improvement is observed due to the dissolution of accumulations of scale and fouling on the splash fill of the cooling tower when operating with the Zero Liquid Discharge technology.

Some have asserted that the depression of the vapor pressure of the cooling water due to the TDS is an issue. Vapor pressure is reduced in an ideal solution by the mole fraction of the species multiplied by its saturation vapor pressure as a pure liquid. In the case of seawater, we can model the dissolved species as common salt, NaCl, and be correct to within 1%. I selected 100 °F as a fairly common return water temperature for cooling water in air conditioning service.

**Table 1: Effect of TDS on Vapor Pressure of Water: Ideal Solution Method**

| TDS (ppm) | mol Fraction of $H_2O$ | $P_v$ @100 °F (psia) | |
|---|---|---|---|
| 0 | 1.0000 | 0.95 | [4] |
| 34,500 | 0.9895 | 0.94 | |
| 100,000 | 0.9702 | 0.92 | |
| 150,000 | 0.9559 | 0.91 | |

It can be seen from the above table and Figure 3 below that the vapor pressure of the 100,000 ppm TDS solution (~10% NaCl brine) is not significantly different from the saturation pressure of plain water, or roughly 0.9 psia. Thus, the evaporation of water to reject heat is substantially the same for the Zero Liquid Discharge cooling tower operation when compared to the use of plain water.

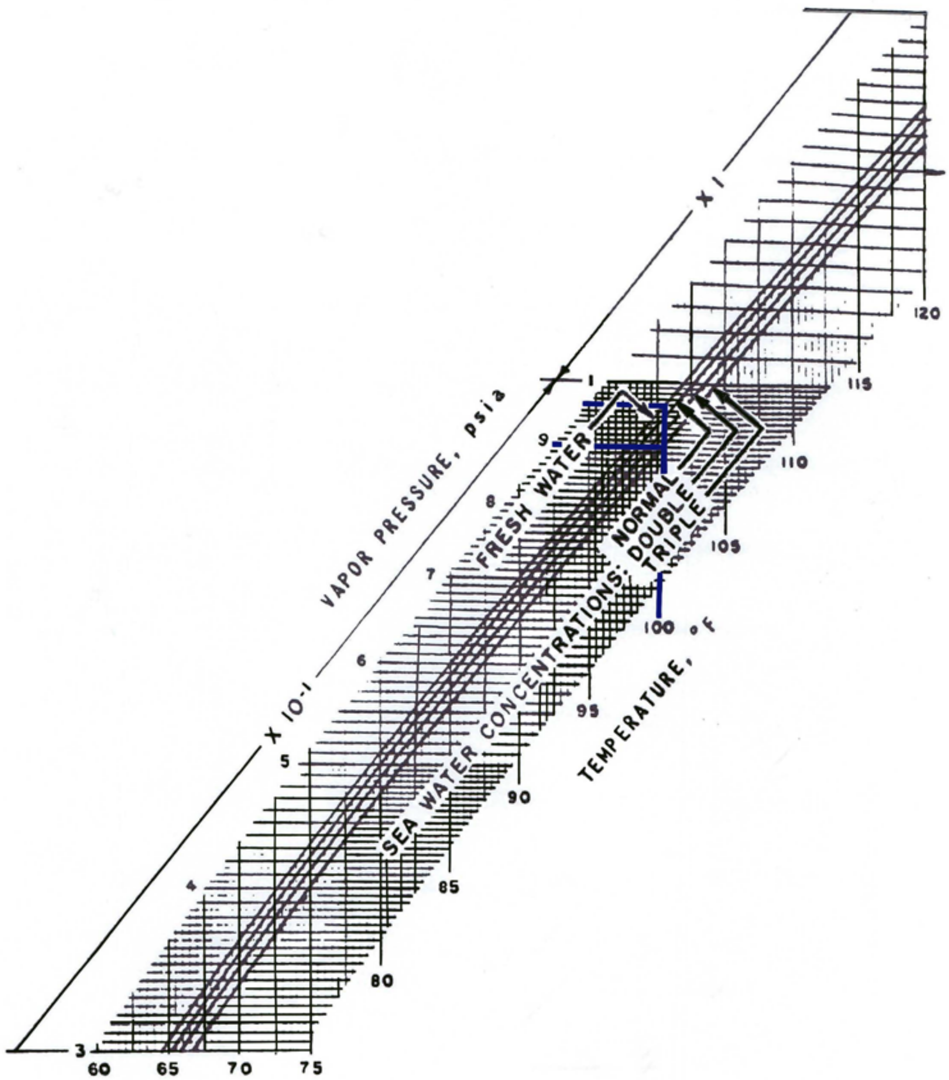


**Figure 3: Vapor Pressure of Salt Water & Brine at 100°F, Data from Ting [2]**
**(Blue lines indicate the two TDS conditions at 100 ºF)**

**REFERENCES**

1. Long, C. Evaluation Series: Environmental Side Effects of Water Treatment Programs: Part 1, Quarter IV, 2011, Aqua Green Technologies, Inc.

2. Ting, Bing-Yuan, and D.M. Suptic. 1991. The Use of Cooling Towers for Salt Water Heat Rejection. The Marley Cooling Tower Company.

3. Flow of Fluids Through Valves, Fittings, and Pipe, Technical Paper 410, Engineering Department, Crane Company, 2010

4. ASME Steam Tables, 1967

**APPENDIX**

Calculations: Example Case: 2,000 gpm system, 100 ft head resistance to pumping.

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| PUMP POWER vs. SPECIFIC GRAVITY | | | | | | | |
| | | | | | | | |
| 2,000 | gpm | | 100 | ft of head | | | |
| | | | | | | | |
| hp = | QxHxS.G. | | Perry's Handbook (1999) pg. 10-23 | | | | |
| | 3960 | | | | | | |
| | | S.G. | hp | bhp | | | |
| Fresh Water | | 1.00 | 50.51 | 67.34 | | | |
| 100,000 ppm TDS | | 1.10 | 55.56 | 74.07 | | | |
| | | | | | | | |
| bhp = hp/$\eta$ | | $\eta$ = | 0.75 | typ. From pump curve. | | | |
| | | | | | | | |
| Crane's Handbook TP-410 (2010), page B-7 | | | | | | | |
| | | | | | | | |
| 500 | gpm of water | | 100 | ft of head | | | |
| hp = | 12.63 | x4 | 50.52 | hp to pump 2,000 gpm of water | | | |
| | | x1.10 | 55.57 | hp to pump 2,000 gpm of 10% NaCl Brine | | | |
| | | | | | | | |
| | | bhp = | 67.36 | | | | |
| | | | 74.10 | | | | |

## Frictional Losses in Pipes:

Reynolds Number = $D v \rho / \mu$

D = 10" ID
$v$ = 8.14 fps x 3,600 s/hr = 29,304 ft/hr
$\rho$ = 62.4 lb/cuft fresh water, 68.6 lb/cuft @ 100,000 ppm TDS [3]
$\mu$ = 1.0 cP for fresh water, 1.1 cP for 100,000 ppm TDS [3] (Fig. A-1
1 cP = 2.4 lb/ft-hr

$Re_{Fresh}$ = (10/12) ft x 29,304 ft/hr x 62.4 lb/cuft /(1.0 X 2.4 lb/ft-hr) ≈ 635,000

$Re_{Brine}$ = (10/12) ft x 29,304 ft/hr x 68.6 lb/cuft /(1.1 X 2.4 lb/ft-hr) ≈ 635,000

From the Moody Diagram, Re = 635,000 in 10" pipe gives f ~0.0145 [3] (Fig. A-2)

$\Delta P/L = 1.295 \times 10^{-3} \times f \rho v^2/D$ [3]

Fresh Water: $\Delta P/L = 1.295 \times 10^{-3} \times 0.0145 \times 62.4$ lb/cuft x $(8.14 \text{ fps})^2$/10.02" x 100' = <u>0.732 psig/100'</u>

10% Brine: $\Delta P/L = 1.295 \times 10^{-3} \times 0.0145 \times 68.6$ lb/cuft x $(8.14 \text{ fps})^2$/10.02" x 100' = <u>0.805 psig/100'</u>

0.805/0.732 ≈ 1.1, or 10% greater frictional losses for flow

Tuberculation:

$A_{rough}$ = 60% of $A_{clean\ pipe}$ => $v_{rough}$ = 1/0.6 ≈ 1.7 x $v_{clean}$

Re ≈ 1.7 x 635,000 ≈ 1,100,000
$\varepsilon/D \approx 0.12$
f ≈ 0.1 (off the top of the Moody diagram)

Fresh Water: $\Delta P/L = 1.295 \times 10^{-3} \times 0.1 \times 62.4$ lb/cuft x $(8.14 \text{ fps})^2$/10.02" x 100' = 5.34 psig/100' !!!

5.34 / 0.732 x 100 = 730% of the frictional losses in clean pipe.

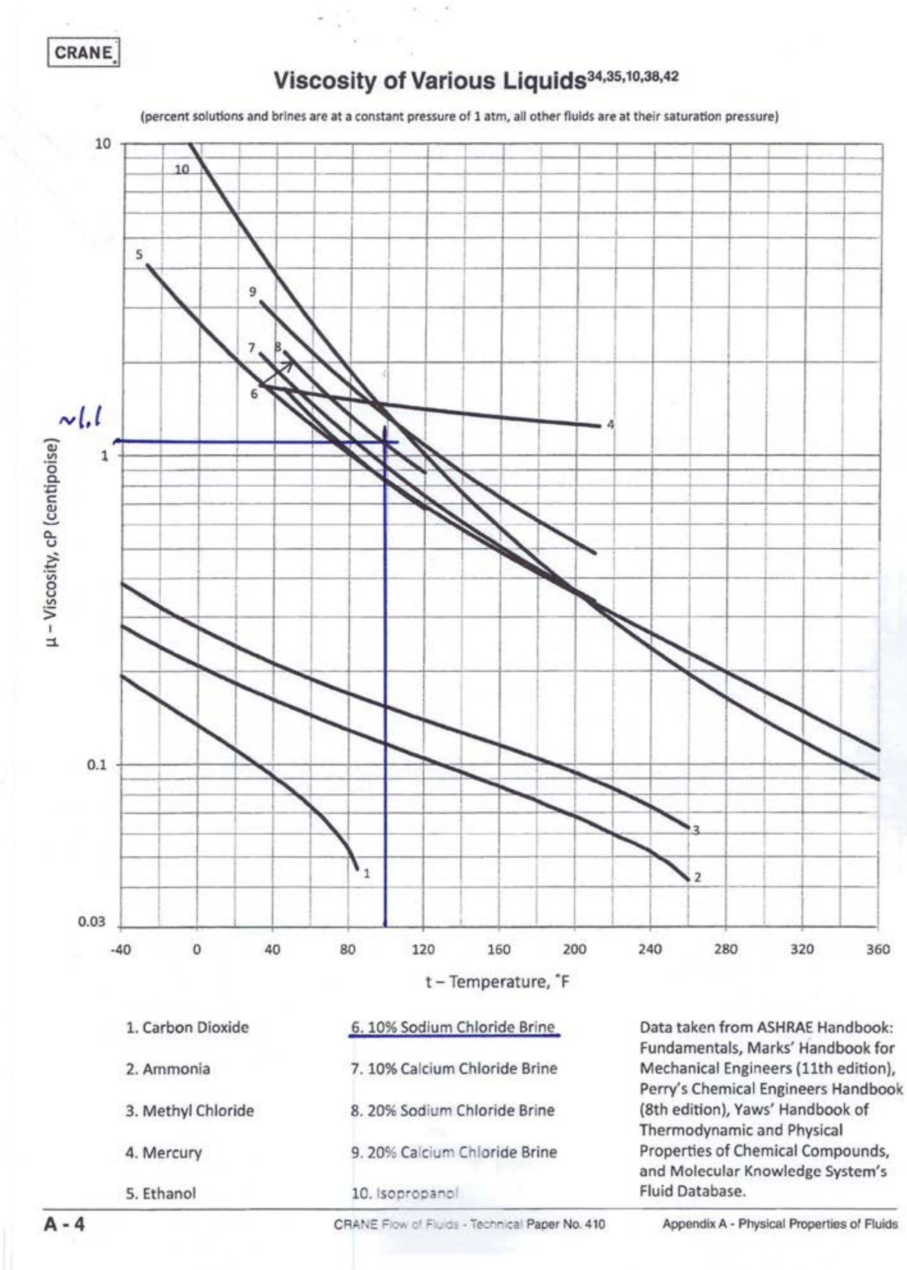


Fig. A-1, Viscosities of Fluids

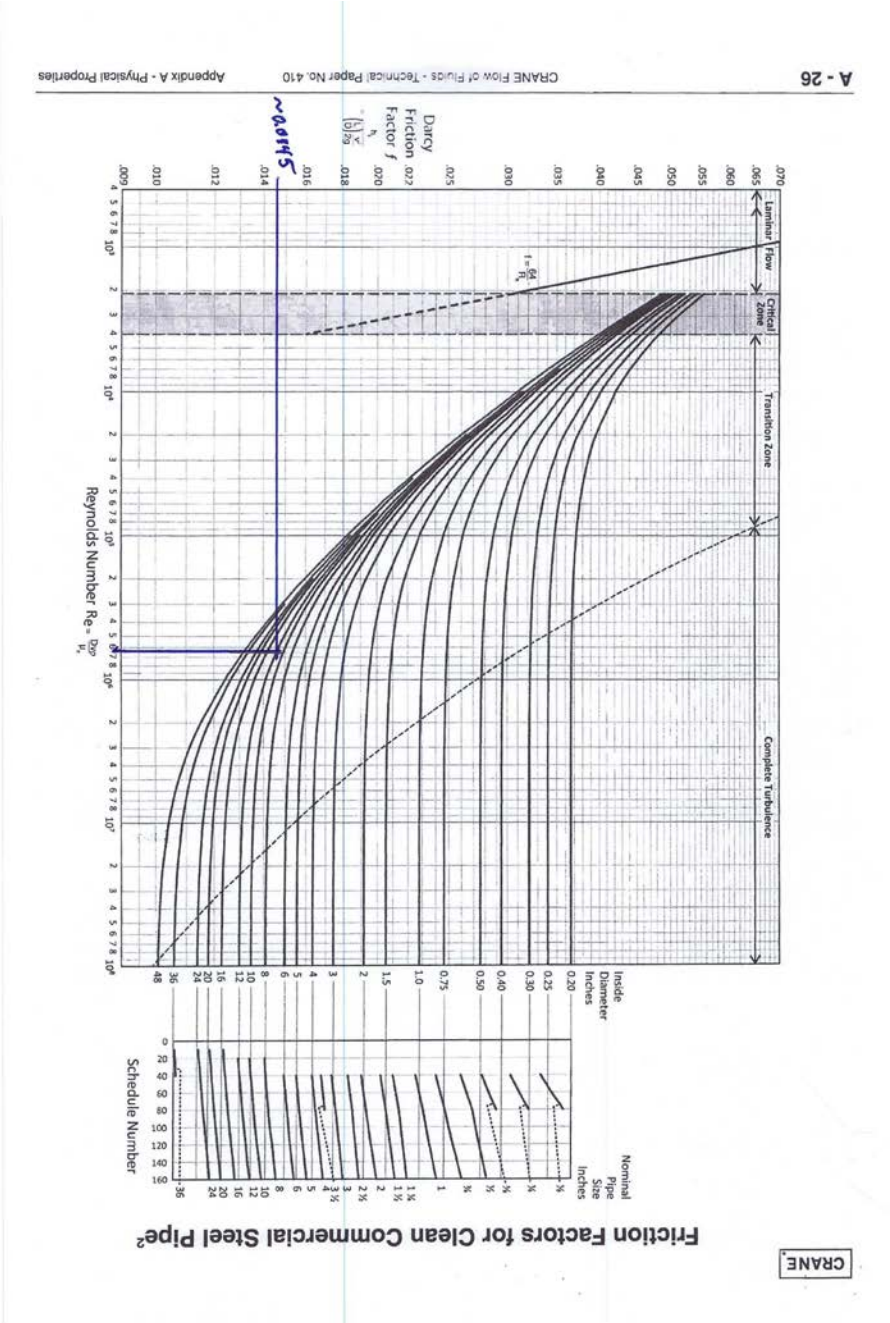

Fig. A-2: Moody Diagram