CHAPTER 5

THERMODYNAMICS OF GTL TRANSPORTATION

In order to evaluate the feasibility of GTL transportation through TAPS, flow behavior of GTL products and GTL-crude oil blends needs to be accurately modeled. Hydraulic modeling of fluid flow through TAPS has been covered in the previous chapter. In this chapter, thermodynamic considerations in flow of GTL material through TAPS are studied. It is important to study the thermodynamic aspects of fluid flow through TAPS for the following reasons.

i) Variation of properties of the fluid due to temperature changes along the pipeline
ii) The temperature of the fluid coming out of the TAPS
iii) Heat loss from the fluid as it flows through 800 miles of pipe

In studying the flow of GTL products through the TAPS, in either batch or commingled mode, it will be necessary to know the expected heat loss along the entire pipeline. This heat loss is dependent on a number of factors, such as different temperatures of the fluid as it passes through different sections of the pipe, different ambient conditions to which the pipe is exposed, the location of the pipe above or below ground etc. In carrying out a proper study the various parameters that contribute to this heat loss should be examined and the methods of accounting them must be considered. Moreover, the horsepower required to pump the medium between pipeline pump stations are dependent on fluid properties which are functions of temperatures. Thus, thermodynamics of GTL flow through TAPS is important in understanding the flow behavior.

OBJECTIVE OF THIS STUDY

The objective of this study is to develop the heat transfer and fluid dynamic equations and apply them to determine the heat loss and pumping power required for different modes of crude oil and GTL transportation and compare the results. This study helps us in evaluating the capability of TAPS pumping equipment and other auxiliary components at different pump stations to transport GTL considering heat transfer and fluid dynamic aspects. The theory of heat loss from pipeline and governing equations that are used to obtain numerical values are described in the following sections.

5.1 HEAT TRANSFER ANALYSIS

5.1.1 Below Ground Pipe Line

Figure 5.1 shows the configuration of the below ground pipeline.

The heat transfer rate can be calculated by:

$$q = \frac{T_{iav} - T_{\infty}}{[1/(h_i 2\pi R_1 L)] + [\ln(R_2 / R_1)/(2\pi k_p L)] + [1/(k_s S)] + [d_2/(k_{sn} LH)] + [1/(h_o LH)]} \quad (5.1)$$

It includes inside convective heat transfer coefficient, resistance due to pipe wall, soil, snow and outside convective heat transfer coefficient. H represents the width over which heat is transmitted, here it is assumed to be fifteen times the diameter of the pipe, due to two-dimensional nature of heat flow through the soil and snow. For calculation of heat transfer in summer, resistance due to snow $d_2 / k_{sn} LH$ is set to zero.

Heat flux is obtained from

$$q'' = q / 2\pi R_1 L = U_i (T_{iav} - T_{\infty}) \quad (5.2)$$

All terms are described in the nomenclature.

Overall Heat Transfer Coefficient

The heat flux can be expressed as the product of the reciprocal of the resistance for a unit area, and an appropriate temperature difference. The reciprocal of resistance for a unit area U, is termed as overall heat transfer coefficient.

U_i based on inside surface area is given by

$$\frac{1}{[1/h_i] + [(R_1 / k_p)\ln(R_2 / R_1)] + [(2\pi R_1 L)/(k_s S)] + [(2\pi R_1 L d_2)/(k_{sn} LH)] + [(2\pi R_1 L)/(h_o LH)]}$$

(5.3)

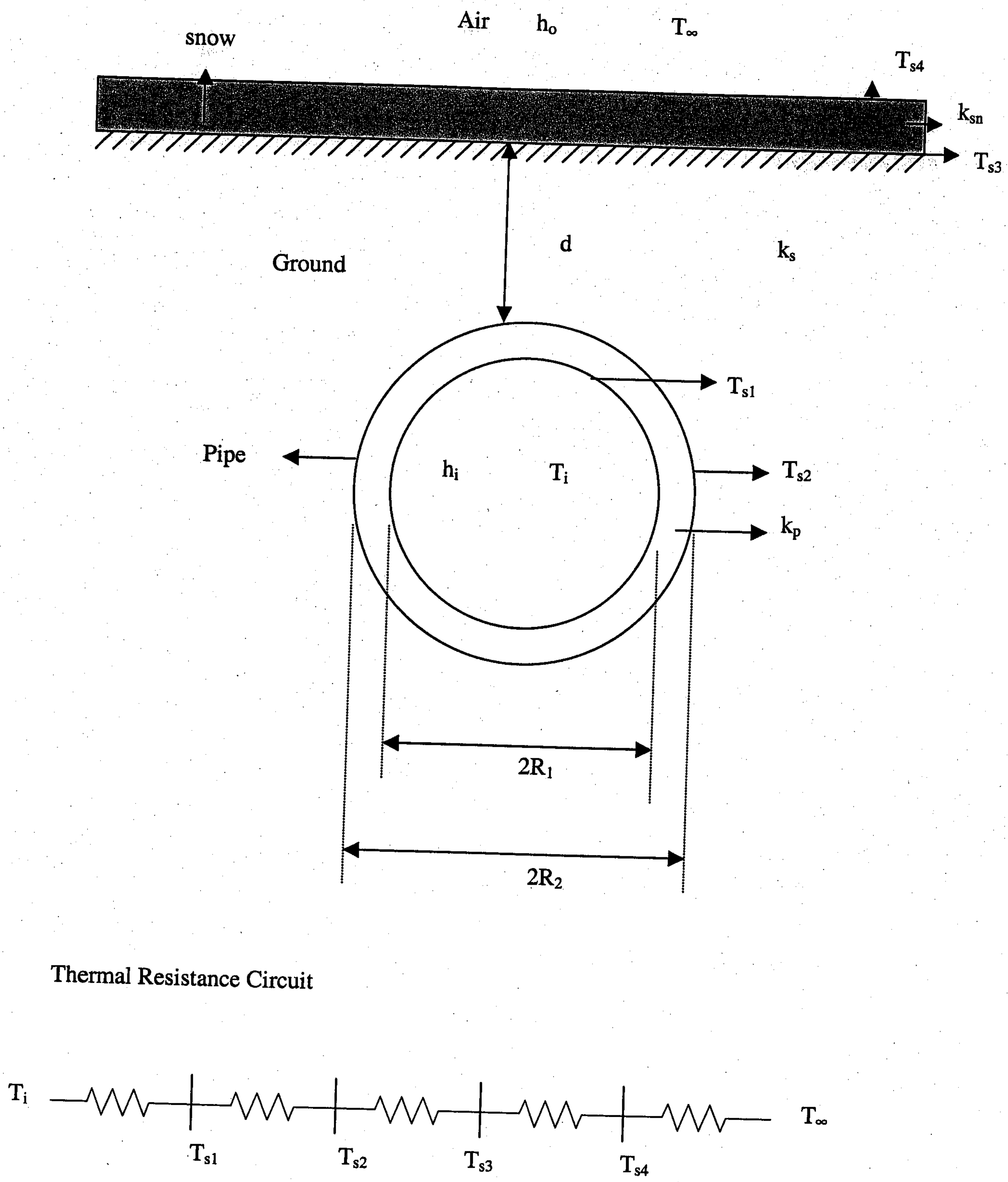


FIGURE 5.1. Below Ground Configuration of the Pipe Line

Conduction shape factor

For cases like a pipe buried underground, the temperature of the soil is two-dimensional and the resistance concept is applied by relating the heat transfer rate to the geometry and thermal conductivity of the soil medium. This is called the conduction shape factor and is denoted by S.

For a cylinder buried in a semi infinite medium the conduction shape factor can be determined from the equations presented in Suryanarayana (1995)

$$S = \begin{cases} \dfrac{2\pi L}{\cosh^{-1}(d/R)} & \text{if } L >> R \quad (5.4.a) \\[2ex] \dfrac{2\pi L}{\ln(2d/R)} & \text{if } L >> R\text{ ; } d > 3R \quad (5.4.b) \\[2ex] \dfrac{2\pi L}{\ln(L/R)[1-\ln(L/2d)/\ln(L/R)]} & \text{if } d >> R\text{; } L >> d \quad (5.4.c) \end{cases}$$

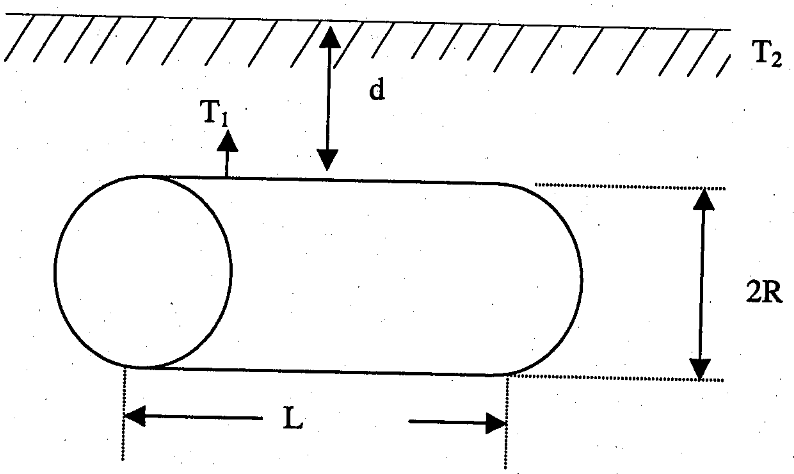


FIGURE 5.2 Pipe Orientation for Conduction Shape Factor

Resistance due to Pipe Wall:

Due to the cylindrical shape of the pipeline the resistance offered by the pipe wall is

$\ln(R_2/R_1)/2\pi k_p L$

Resistance due to the snow

Resistance due to the layer of the snow on the earth surface can be determined by

$d_2 / k_{sn} LH$, assuming heat transfer over fifteen diameters:

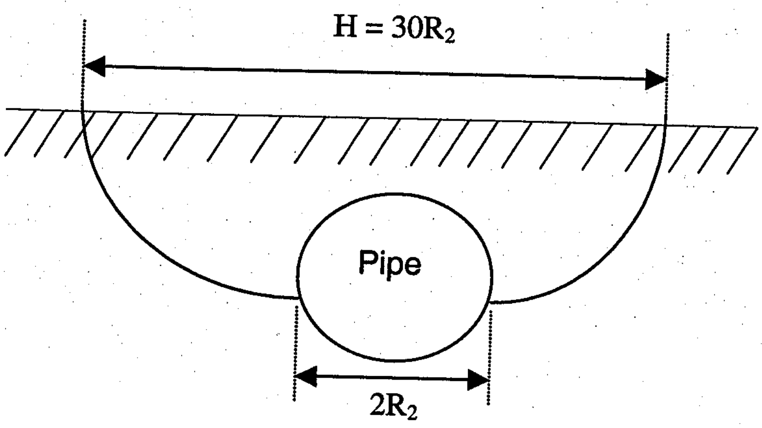


FIGURE 5.3 Assumed Zone of Heat Loss from the Pipe

Resistance due to convection

Heat transfer occurs between the oil and its boundary solid surface of the steel pipe wall. The heat flux from or to the solid surface is proportional to the difference between the surface temperature and a characteristic temperature of the fluid. The coefficient of proportionality is known as convective heat transfer coefficient.

The convective heat transfer coefficient when the heat transfer is from fluid to the surface can be determined by

$$h_i = \frac{Nu_d k_f}{2R_1} \qquad (5.5)$$

The above equation is based on the inside diameter for cylindrical passage.
Where Nu is called the Nusselt number and k_f is the thermal conductivity of the fluid flowing through the pipeline, h_i is the internal convective heat transfer coefficient

For heat transfer from the snow in winter or bare surface in summer to the outside air, the convective heat transfer is given by

$$h_o = \frac{Nu_H k_a}{H} \qquad (5.6)$$

The Nusselt number is based on the linear dimension H of a rectangular surface, h_o is the outside convective heat transfer coefficient and k_a is the thermal conductivity of air.

The equation to determine the value of the Nusselt number depends on various parameters like Reynolds number and Prandtl number. Reynolds number is the non dimensional number that is related to the motion of the fluid. It is the ratio between the inertia and viscous forces acting upon the fluid.

Reynolds number for internal flows is given by $\mathrm{Re}_d = \dfrac{\rho V d_1}{\mu}$

We can write mass flow rate as $\dot{m} = \rho V \dfrac{\pi d_1^{\,2}}{4}$

which gives $\rho V d_1 = \dfrac{4\dot{m}}{\pi d_1 \mu}$

Therefore, Reynolds number can be written as:

$$\mathrm{Re}_d = \frac{4\dot{m}}{\pi d_1 \mu} \qquad (5.7)$$

The above equation is written taking diameter as the characteristic length for cylindrical surfaces and V is the average velocity of fluid flowing through the pipe.

Depending on the value of the Reynolds number the flow is characterized as laminar flow or turbulent flow.

For flow inside pipes, if the value of Reynolds number is less than 2100 then the flow is characterized as laminar flow and if the Reynolds number is greater than 2100 then the flow is turbulent flow.

To calculate the Reynolds number for external flows V is replaced by V_∞ which is the ambient air velocity

Therefore Reynolds number can be written as $\mathrm{Re}_H = \dfrac{\rho_a V_\infty H}{\mu}$ (5.8)

The above equation is written taking length as the characteristic length for a rectangular surface.

For external flows over rectangular surfaces, if the value of the Reynolds number is less than $5*10^5$ then the flow is characterized as laminar flow and if the Reynolds number is greater then $5*10^5$ then the flow is turbulent flow.

Prandtl number represents the relative effectiveness of molecular transport of momentum and energy within the hydrodynamic and thermal boundary layers.

$$Pr = c_p\mu/k_f \quad (5.9)$$

Where c_p is the specific heat of the fluid and μ is the coefficient of dynamic viscosity of the fluid and k_f is the thermal conductivity of the fluid, all at a particular temperature.

Depending upon the values of Reynolds number and Prandtl number for flow of the fluid inside the pipe, Nusselt number can be calculated and from Nusselt number the convective heat transfer coefficient can be determined. Equations of Nusselt number for internal flows can be found from Suryanarayana (1995).

For fully developed laminar flow $Nu_d = 3.66$ (uniform surface temperture) (5.10)

$Nu_d = 4.36$ (uniform surface heat flux) (5.11)

For turbulent flows with entry length effects the equations are

$0.5 < pr < 1.5$; $2300 < Re_d < 10^6$; $0<d/L<1$

$$Nu_d = 0.0214\,(Re_d^{4/5} - 100)\,Pr^{2/5}\,[1+(d/L)^{2/3}] \quad (5.12)$$

$1.5 < Pr < 500$; $2300 < Re_d < 10^6$; $0<d/L<1$

$$Nu_d = 0.012(Re_d^{0.87} - 280)\,Pr^{2/5}\,[1+(d/L)^{2/3}] \quad (5.13)$$

The hydrodynamic entry length equations can be found from Suryanarayana (1995).

For laminar flow $L_e/d = 0.0565\, Re_d$ (5.14)

For turbulent flow $L_e/d = 1.359\, Re_d^{1/4}$ (5.15)

The thermal entry length equations can be found from Suryanarayana (1995)

For laminar flow $L_{e,th}/d = 0.037\, Re_d\, Pr$ (uniform surface temperature) (5.16)

$L_{e,th}/d = 0.053$ (uniform heat flux) (5.17)

For turbulent flow $L_{e,th} = 10\,d$ (5.18)

For turbulent flow it is considered to be fully developed if x/d > 10, where x is the distance from the entrance.

The Nusselt number equations can be used for fully developed case by setting $d/L = 0$
Equations of Nusselt number for external flows over rectangular surfaces can be found from Suryanarayana (1995)

For Mixed flow (entry laminar flow and turbulent thereafter)

$$Nu_H = (0.037\,Re_H^{4/5} - 871)\,Pr^{1/3} \qquad 5*10^5 < Re_H < 10^7 \tag{5.19}$$

$$Nu_H = [1.967\,Re_H\,(\ln Re_H)^{-2.584} - 871]\,Pr^{1/3} \qquad 10^7 < Re_H < 10^9 \tag{5.20}$$

For fully turbulent flows

$$Nu_H = 0.037\,Re_H^{4/5}\,Pr^{1/3} \qquad Re_{cr} = 0,\ Re_H < 10^7 \tag{5.21}$$

$$Nu_H = 1.967\,Re_H\,(\ln Re_H)^{-2.584}\,Pr^{1/3} \qquad Re_{cr} = 0,\ 10^7 < Re_H < 10^9 \tag{5.22}$$

By using the above equations (1) through (22) all the heat transfer parameters can be calculated.

Exit Temperature of the Fluid

Due to the heat loss, the fluid temperature must diminish in the direction of flow (Figure 5.4).

The exit temperature of the fluid from the pipeline can be determined by using the equation presented by Suryanarayana (1995)

$$\frac{T_\infty - T_e}{T_\infty - T_i} = \exp\left(\frac{-U_i A_i}{\dot{m} c_p}\right) \tag{5.23}$$

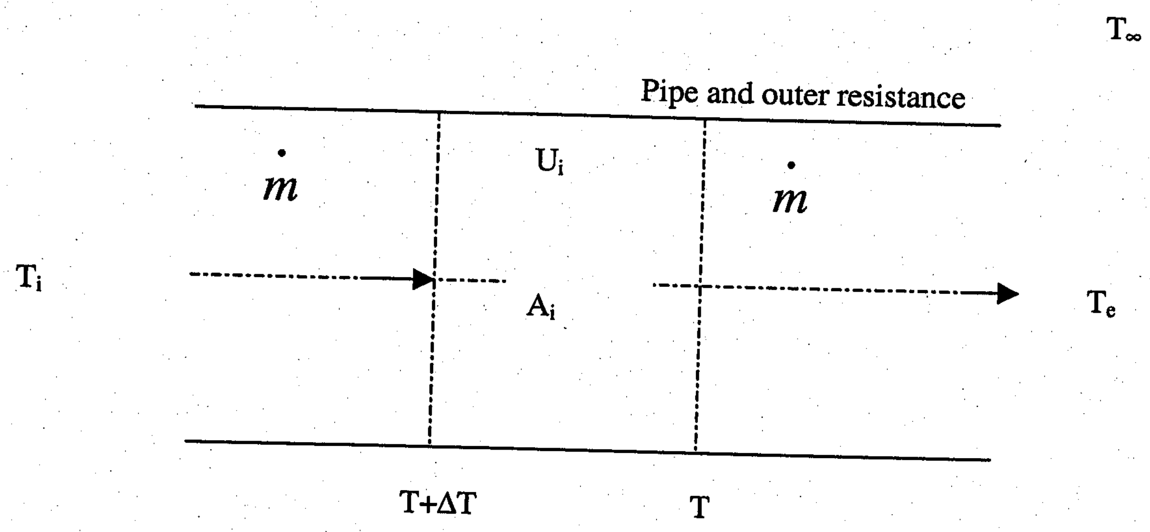


FIGURE 5.4 Relationship Between Fluid Inlet and Exit Temperatures and Overall Heat Transfer Coefficient

5.1.2 Above Ground Pipe Line

Figure 5.5 shows the configuration of the above ground pipeline.

The heat transfer rate for above ground pipeline can be determined by:

$$q = \frac{T_{iav} - T_\infty}{[1/(h_i 2\pi R_1 L)] + [\ln(R_2/R_1)/(2\pi k_p L)] + [\ln(R_3/R_2)/(2\pi k_i L)] + [1/(h_0 2\pi R_3 L)]} \qquad (5.24)$$

Heat flux is defined as the heat transfer rate per unit area, which is given as

$$q'' = \frac{q}{2\pi R_1 L} = U_i(T_{iav} - T_\infty) \qquad (5.25)$$

Where U_i can be written based on the inner surface of the pipe

$$U_i = \frac{1}{[(1/h_i)] + [\ln(R_2/R_1)(R_1/k_p)] + [\ln(R_3/R_2)(R_1/k_s)] + [(1/h_o)(R_1/R_3)]} \qquad (5.26)$$

The outside film coefficient h_o can be determined from the Nusselt number

$$h_o = \frac{Nu_D k_a}{2R_3} \tag{5.27}$$

Nusselt number for external flow over cylinders can be found from the equations summarized by Suryanaraya (1995)

$$Nu_D = 0.3 + \frac{0.62 \mathrm{Re}_D^{1/2} \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\mathrm{Re}_D}{282000}\right)^{5/8}\right]^{4/5} \qquad \mathrm{Re}_D > 400000 \tag{5.28}$$

$$Nu_D = 0.3 + \frac{0.62 \mathrm{Re}_D^{1/2} \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\mathrm{Re}_D}{282000}\right)^{1/2}\right] \qquad 20000 < \mathrm{Re}_D < 400000 \tag{5.29}$$

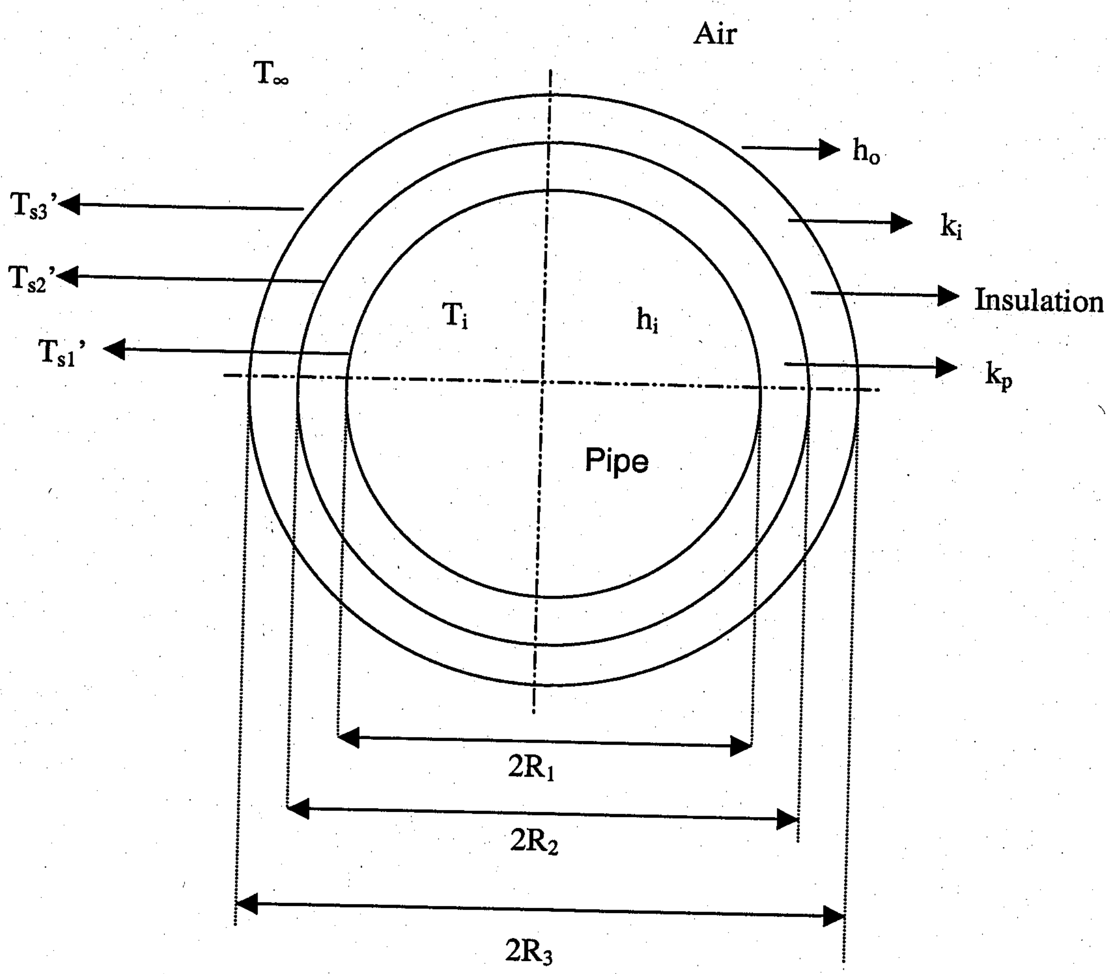
$$Nu_D = 0.3 + \frac{0.62 \mathrm{Re}_D^{1/2} \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \qquad \mathrm{Re}_D < 20000 \tag{5.30}$$

For all of the above equations, the product $\mathrm{Re}_D\mathrm{Pr} > 0.2$ must be satisfied.

In the previous three equations the Reynolds number is based on the wind velocity across the pipeline and can be obtained from the relation

$$\mathrm{Re}_D = \frac{2V_\infty R_3 \rho_a}{\mu} \tag{5.31}$$

where R_3 is the outer radius of the pipe with insulation.



Thermal Resistance Circuit

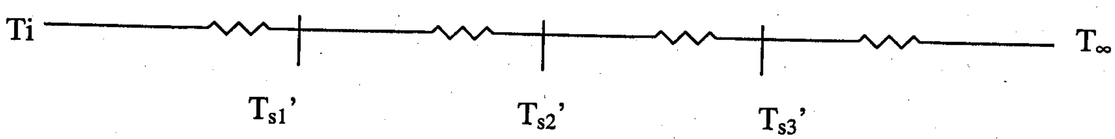


FIGURE 5.5 Above Ground Configuration of the Pipe Line

5.2 FLUID DYNAMIC ANALYSIS

Pressure drop in the pipeline is due to various factors like friction and hydrostatic head.

Pressure drop due to frictional head is given by White (1986)

$$\Delta p = \rho g h_f \qquad (5.32)$$

where h_f is the head loss due to friction.

The head loss in terms of friction factor is given by White (1986)

$$h_f = \frac{fLV^2}{2gd_1} \qquad (5.33)$$

Where,

f is called the Darcy friction factor,
V is the average velocity of the fluid flowing through the pipe,
d_1 is the inside diameter and
L is the length of the pipe.

Friction factor f for turbulent flow depends upon Reynolds number and the roughness of the pipe. The friction factor f is given by Haaland (1983)

$$\frac{1}{f^{1/2}} = -1.8\log\left[\frac{6.9}{\mathrm{Re}_d} + \left(\frac{\varepsilon / d_1}{3.7}\right)^{1.11}\right] \qquad (5.34)$$

Velocity of the fluid flowing through the pipe is given by

$$V = \frac{\dot{m}}{\rho A}$$

Where A is the cross sectional area of the pipe which is given by $\frac{\pi}{4}d_1^2$ and $\dot{m}$ is the mass flow rate of the fluid flowing through the pipe.

The pressure difference due to hydrostatic head is given by

$$\Delta p_h = \rho g (z_2 - z_1) \qquad (5.35)$$

The hydrostatic head is given by $(z_2 - z_1)$, where this term represents the elevation difference between two pump stations.

The pressure loss in pipe fittings between pump stations has been presented by Akwukwaegbu (2001)

$$\Delta p_m = \rho g h_m \tag{5.36}$$

where h_m is the head loss due to pipe fittings.

The minor losses can be written as

$$h_m = k\frac{v^2}{2g} \tag{5.37}$$

Where,

k is a constant and

v is the velocity of the fluid

h_m is constant for all fluids flowing through the pipe as the volumetric flow is taken to be 1.1 MMBPD, the velocities are the same for all the fluids. So head loss due to pipe fittings is constant.

Finally, the power required to pump the medium between the pipeline pump stations against these heads is given by Thomas (1993).

$$P = \dot{m} g[h_f + (z_2 - z_1) + h_m] \tag{5.38}$$

5.3 PIPELINE SPECIFICATIONS AND FLUID PROPERTIES

The application of governing equations to determine the heat transfer and fluid dynamic parameters for the fluid flowing through Trans Alaska Pipeline requires the basic knowledge of the pipe specifications and the current operating conditions of the pipe. It is also important to know the properties of the fluid like viscosity, thermal conductivity, density, specific heat of the fluid as well as the properties of the air to determine heat loss to the surroundings.

5.3.1 Pipe Specifications

The heat transfer and fluid dynamic models work on certain basic information of the Trans Alaska Pipeline System (TAPS). The TAPS is an 800.302 miles long pipeline with 48 inch outer diameter. The pipeline is made of steel with thickness of 0.462 inches in some sections and 0.562 inches in other sections. The pipeline starts at Prudhoe Bay and ends at the Valdez terminal. Some sections of the pipeline are above ground and the other sections are below ground. The above ground sections have 3.5 inch thick insulation and the below ground sections do not have any insulation. The current crude oil flow rate in TAPS is approximately

1.1 million barrels per day (MMBPD). The pipeline has 12 pump stations (PS) numbered from PS1 to PS12, in which PS11 was deemed unnecessary and was never built. Pump station 5 is a relief station with no pumping capacity. Pump stations 2,6,8 and 10 have been placed in standby mode. Therefore the pipe is divided into 7 major sections for the purpose of thermodynamic calculation. These sections are listed below and a schematic of the pipeline along with the pump stations is shown in Figure 5.6.

i) PS1-PS3

ii) PS3-PS4

iii) PS4-PS5

iv) PS5-PS7

v) PS7-PS9

vi) PS9-PS12

vii) PS12-VALDEZ

Certain thermodynamic properties associated with the calculations don't show a significant change in the values with variation of temperature. Therefore, these are assumed to be constant throughout the length of the pipe as shown in Table 5.1

TABLE 5.1
Constant Properties

Thermal Conductivity of Insulation	0.0462 W/m K	(Chrisman, 2001)
Thermal Conductivity of Snow	0.19 W/m K	(Thomas, 1993)
Thermal Conductivity of Pipe	60.5 W/m K	(Suryanarayana, 1995)
Thermal Conductivity of Gravel	2 W/m K	(Andersland & Anderson, 1978)

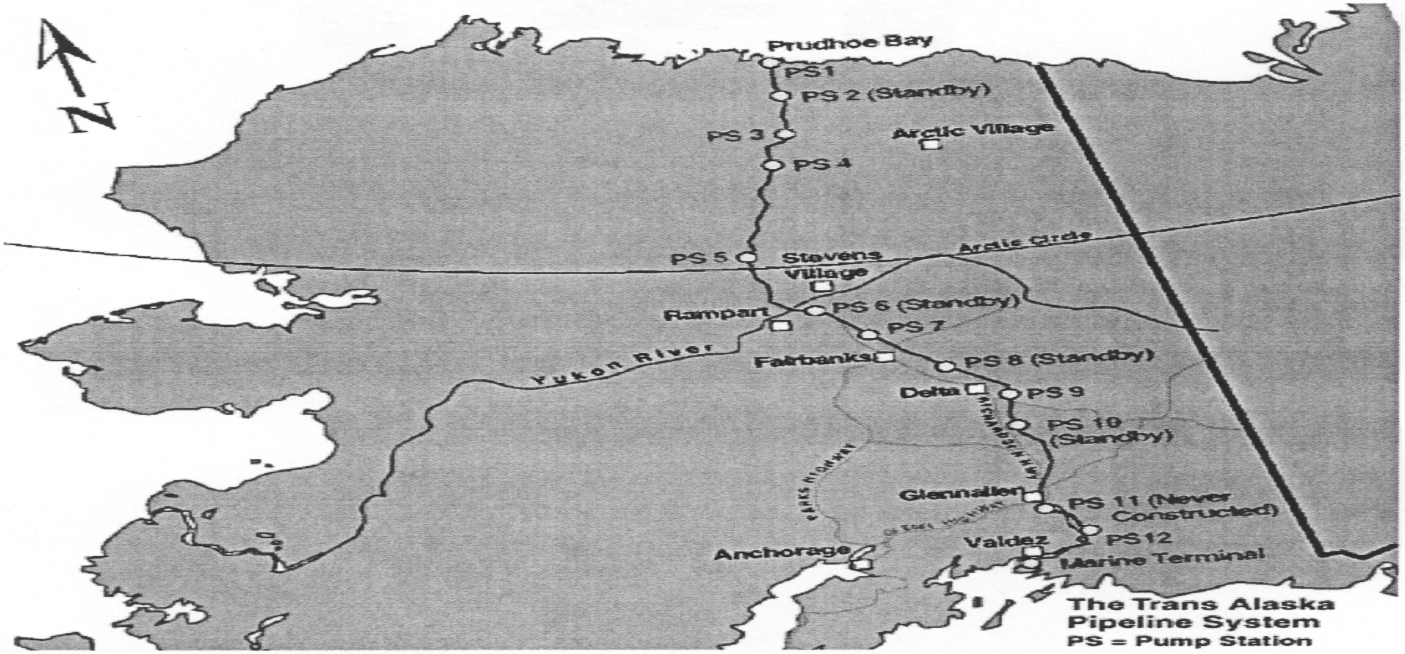


FIGURE 5.6 The Trans Alaska Pipeline System and the pump stations

5.3.2 Fluid Properties

The crude oil is pumped from pump station 1 approximately at temperature of 115.7 °F (Chrisman, 2001). As the oil flows through the 800 miles pipe, there is some heat loss from the oil to the surroundings. Because of heat loss, there will be a decrease in temperature, which in turn affects certain properties of the oil.

The properties of the fluid that are sensitive to the temperature change are:

i) Density of fluid
ii) Thermal Conductivity of fluid
iii) Viscosity of fluid

Since we will have to consider the flow of crude oil, GTL and a mixture of oil and GTL called commingled mixture properties variation for these three different types of fluids needs to be determined. Densities and viscosities of crude oil, GTL, and GTL-crude oil blends as a function of temperature are already described in Chapter 3. Thermal conductivities of these fluids are discussed below.

In the present study, three different types of fluids are considered: crude oil, GTL, and a 3:1 blend of crude oil and GTL. Variation of thermal conductivity of crude oil and GTL with temperature is shown in the following Table 5.2 (Dandekar, 2001).

TABLE 5.2
Variation of Thermal Conductivity with Temperature

Temperature (^oF)	Thermal Conductivity (W/m K)	
	Crude Oil	GTL
120 (48.88 °C)	0.1418	0.1299
100 (37.78 °C)	0.1442	0.1325
80 (26.67 °C)	0.1466	0.1351
60 (15.56 °C)	0.1491	0.1377
40 (4.45 °C)	0.1503	0.1432
20 (-6.67 °C)	0.1529	0.1485

The thermal conductivity of the GTL-crude oil mixture can be determined by the mass average method.

$$k_{mix} = \frac{m_g k_g + m_o k_o}{m_o + m_g} \qquad (5.39)$$

We know that one kg of mixture contains 0.7789 kg of crude oil and 0.2210 kg of GTL. Thus,

$$kmix = \frac{0.2210(0.1322) + 0.7789(0.1440)}{1}$$

$$k_{mix} = 0.14139 \text{ W/m K}$$

Variation of thermal conductivity of the fluids under consideration with temperature is shown in Figure 5.7.

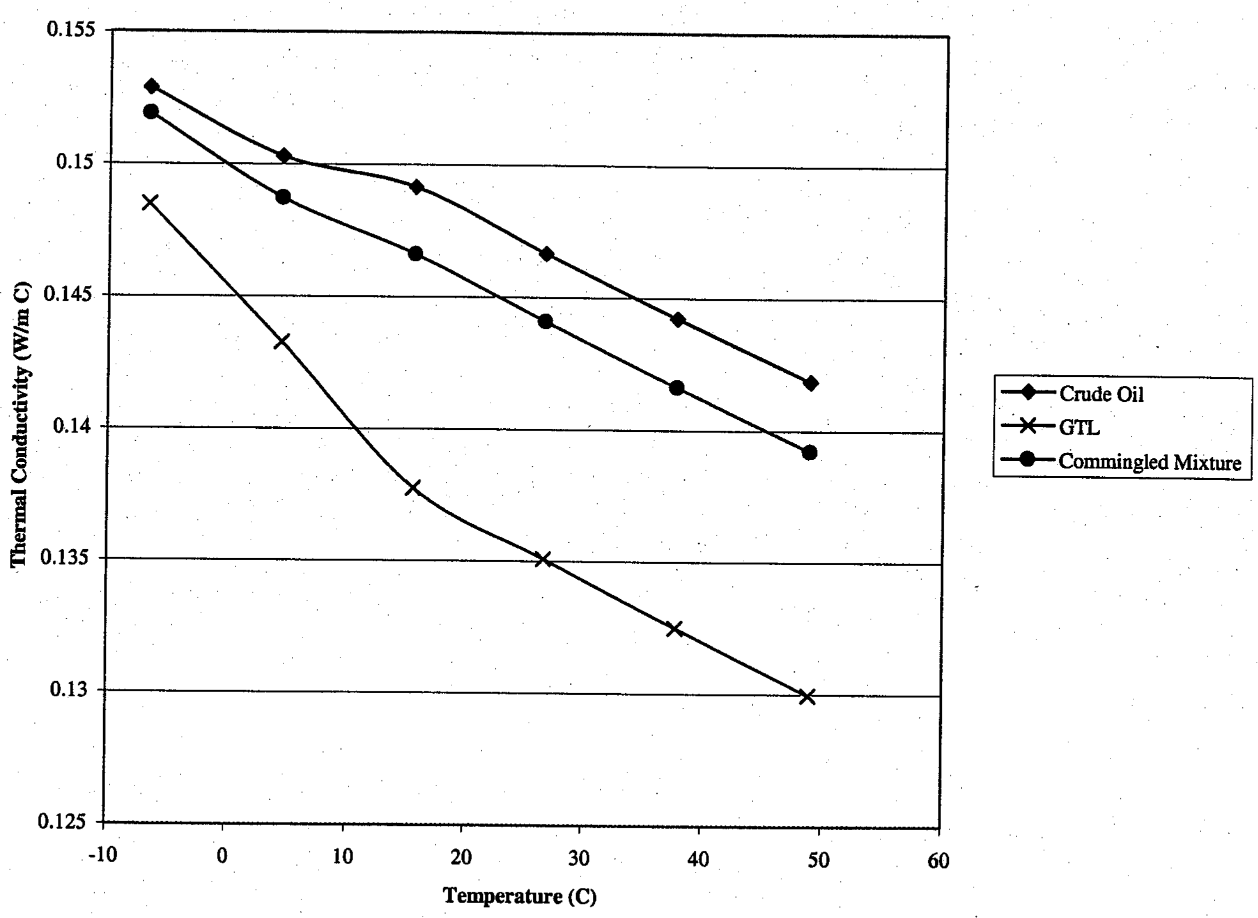


FIGURE 5.7 Variation of Thermal Conductivity of Fluids with Temperature

5.3.3 Properties of Air

The fluid flowing through the Trans Alaska Pipeline System loses heat to the outside air by convection. The outer convective heat transfer coefficient of air depends on Nusselt number and Reynolds number, which in turn depends on the viscosity and the density of air. The ambient air temperature along TAPS varies for different months and for different locations of the pipe. So the variation of the properties of the air needs to be determined with temperature.

The properties of the air that vary with temperature are:

i) Density of air
ii) Viscosity of air
iii) Thermal Conductivity of air
iv) Prandtl number of air

The variation of properties of air at different ambient temperatures are shown graphically in Figures 5.8 through 5.11 (Suryanarayana, 1995).

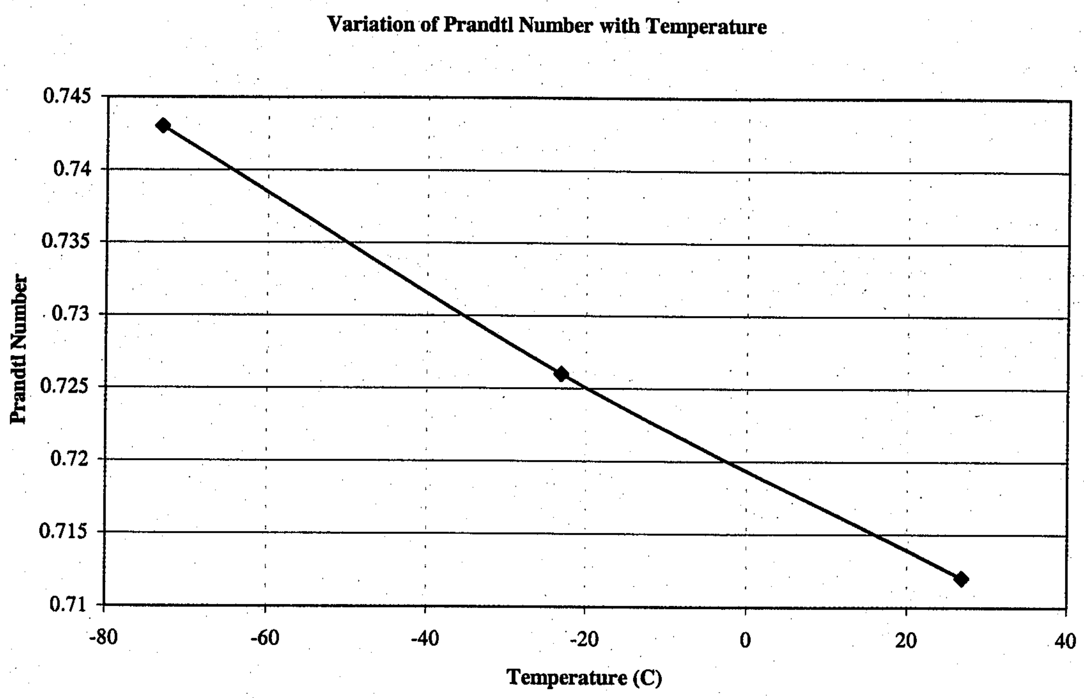


FIGURE 5.8 Prandtl Number Variation for Air With Temperature

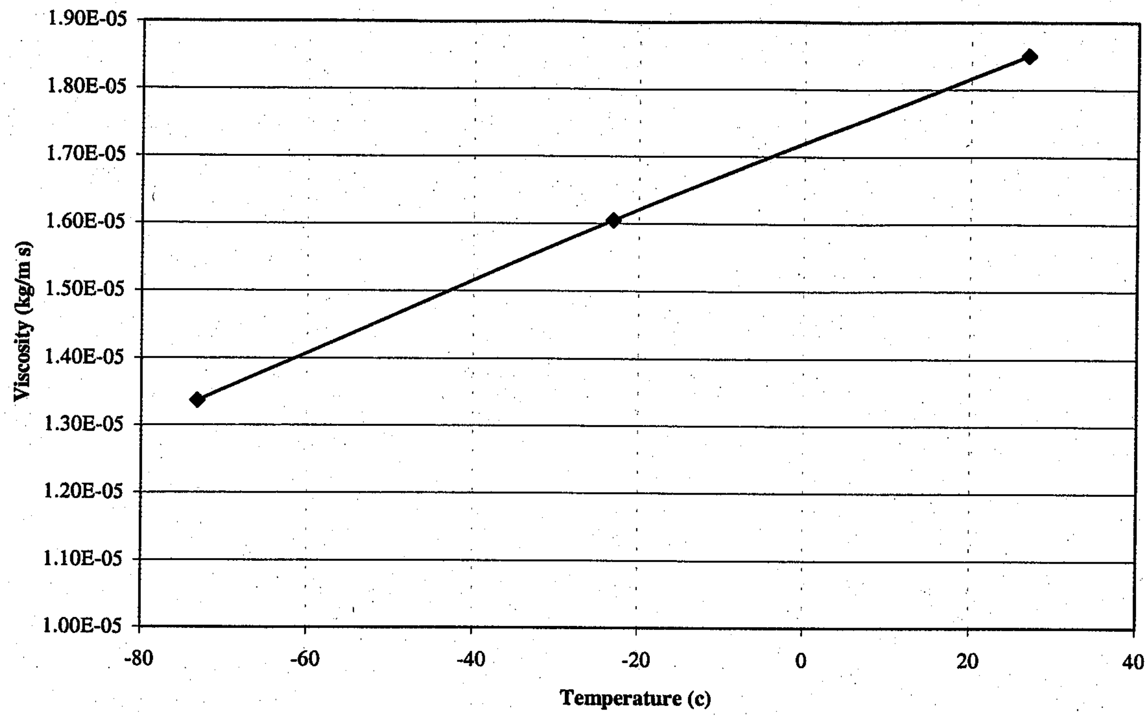


FIGURE 5.9 Viscosity Variation for Air with Temperature

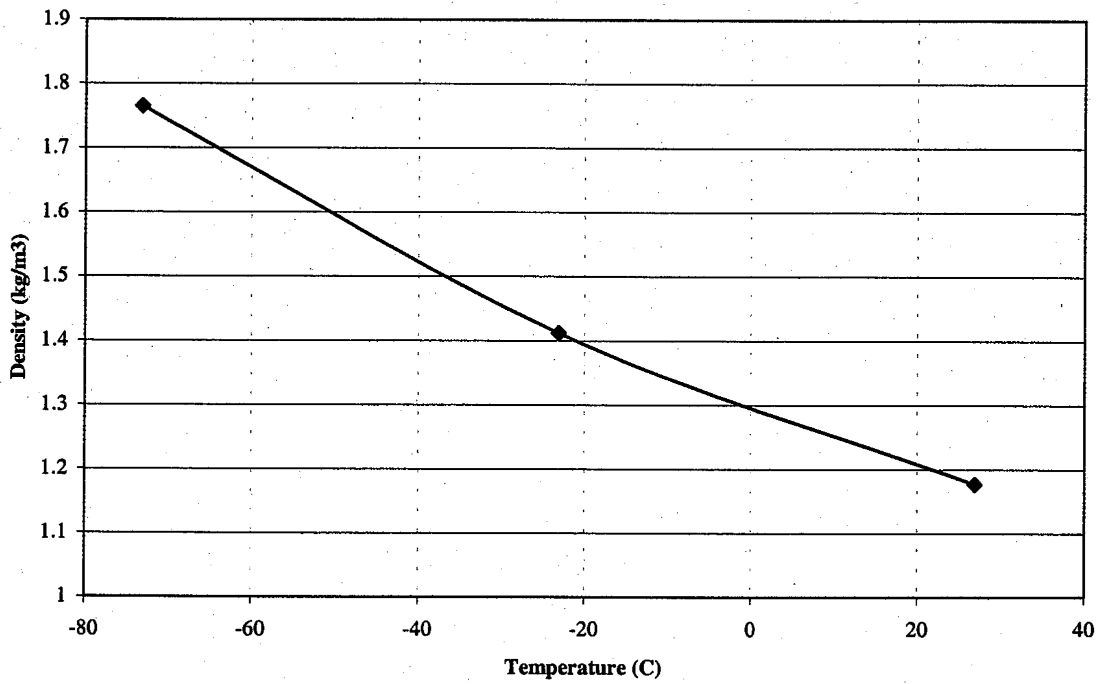


FIGURE 5.10 Density Variation for Air with Temperature

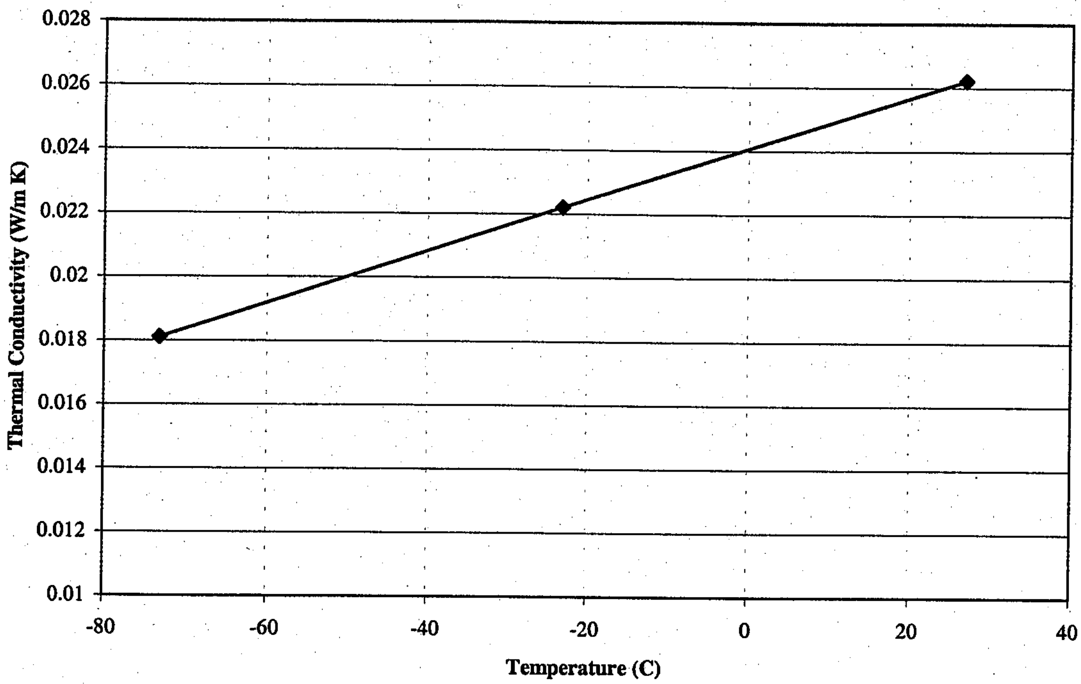


FIGURE 5.11 Variation of thermal conductivity of air with temperature