Northeastern University

College of Engineering

Department of Mechanical and Industrial Engineering

Energy analysis of heavy oil transportation in harsh environment

Submitted by Justin Miller

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Course Instructor Medhi Abedi

1 Problem Introduction

Oil transportation is a major part of the industry and is the backbone of the supply chain for various sectors such as trade, transportation, and manufacturing. The continental United States receives much of its oil from deposits near and in the Artic Circle in parts of Alaska and Northern Canada. Delivery of this scale and distance is very difficult, so many companies have invested in pipeline developments to facilitate consistent, high-volume delivery of crude oil to refineries located at locations farther south.

Many Alaskan and Canadian pipelines have the unique constraint of extreme weather. Temperatures in these subarctic regions consistently stay below freezing, with high winds adding an additional chill to any pipeline structures being constructed in the area. The viscosity of crude oil is very sensitive to temperature, and in these hostile conditions it becomes very difficult for the fluid to flow through the pipeline. So, it is a common practice for engineers to add heating elements at the pipelines pumping stations, increasing the temperature of the medium so that it is easier to push hundreds of miles towards the refineries. Determining the optimal amount of heat to add to these flows is of particular interest to petroleum engineers due to the high monetary costs associated with the transportation of 860,000 barrels of oil per day.

In this report, the engineering department will present a comprehensive heat model for the specified oil pipeline. This heat model will incorporate the dimensions and material composition of the pipeline and fluid as well as atmospheric conditions such as temperature, wind, and solar radiation. This heat model will be combined with a simplified fluid flow model and optimized to find the minimum amount of energy required to transport crude from the northern tip of Alberta, Canada to the US border, a total distance of roughly 1200 km.

2 Fluid Model

2.1 SS Fluid Flow – Overview

Incompressible Steady State Fluid Flow through a pipe can be modeled by Bernoulli's law.

$$z_1 + \frac{v_1^2}{2g} + \frac{P_1 \nu_1 g_c}{g} + H_p = z_2 + \frac{v_2^2}{2g} + \frac{P_2 \nu_2 g_C}{g} + Hf$$

For the scope of this project, it is assumed that the pipeline travels at a completely straight, horizontal path with no significant changes in pipe diameter. This further reduces Bernoulli's equation to an equivalence between the pump head and the head loss from friction.

$$H_p = H_f$$

This relation is in meters and is a common design parameter for industrial pump stations. The head loss term (Hf) will be discussed further in section 6.3.

2.2 Constants and Constraints

Many assumptions were made regarding the parameters of this flow, see below.

2.2.1 Pipeline Class and Strength

Pipeline class strength parameters are covered in detail in the Canadian Natural Resources Pipeline Specifications Manual[1]. Because this is a high-volume corridor, the pipeline has been constructed using Line Class EZ. This pipeline has varying MAWP values (Maximum Allowable Working Pressure) that are shown below in Table 1.

Temp. °F	-20 to 100	200	248
°C	-29 to 38	93	120
MAWP, psig	3600	3410	3335
kPag	24820	23540	23000

2.2.2 General Dimensions

Table 2 shows the general dimensions of the study pipeline.

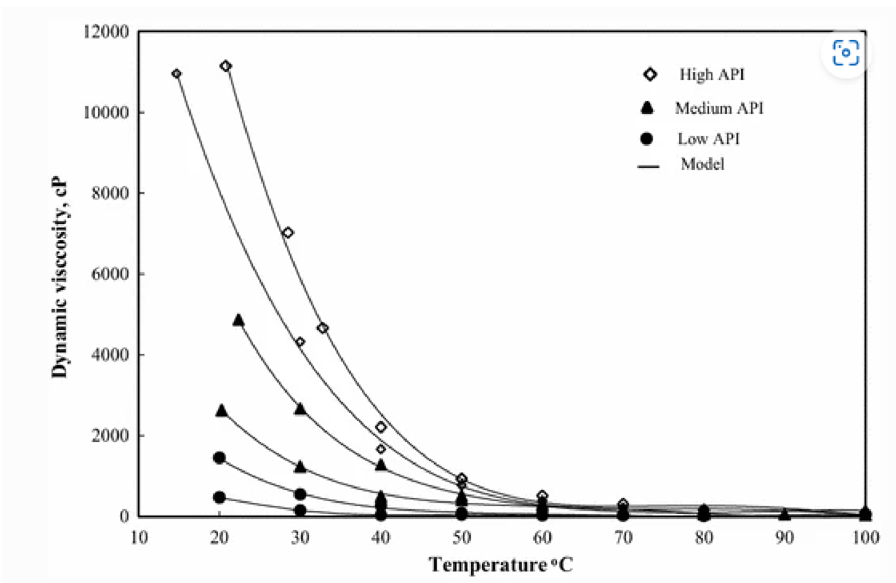
Pipeline Diameter (m)	0.762
Mass Flow Rate (kg / s)	1424.3
Fluid Flow Velocity (m/s)	3.4701

[1] 1 (cer-rec.gc.ca)

Fluid Density (kg / m^3)	900
Reynolds Number (External Flow)	257430
Reynolds Number (Internal Flow)	Variable: In Turbulent Regime
Total Oil Delivered (m^3 Yearly)	136729.074

2.2.3 Heavy Crude Viscosity Model

The viscosity of Heavy Crude Oil is highly dependent on temperature. A 2016 study (Alomair et. Al)[2] generated an empirical correlation for the temperature range of 20 to 10 degrees Celsius to use in high-temperature pipeline applications. Selected values are plotted below in intervals of 10 degrees. The interpolation was replicated in MATLAB and built into the Steady State fluid model.



2.3 Head Loss

The head loss resulting from pipe friction is defined by the Darcy-Weisbach equation.[3]

$$H_f = f\frac{L}{d}\frac{V^2}{2g}$$

Where f is the friction coefficient.

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[2] Heavy oil viscosity and density prediction at normal and elevated temperatures | SpringerLink

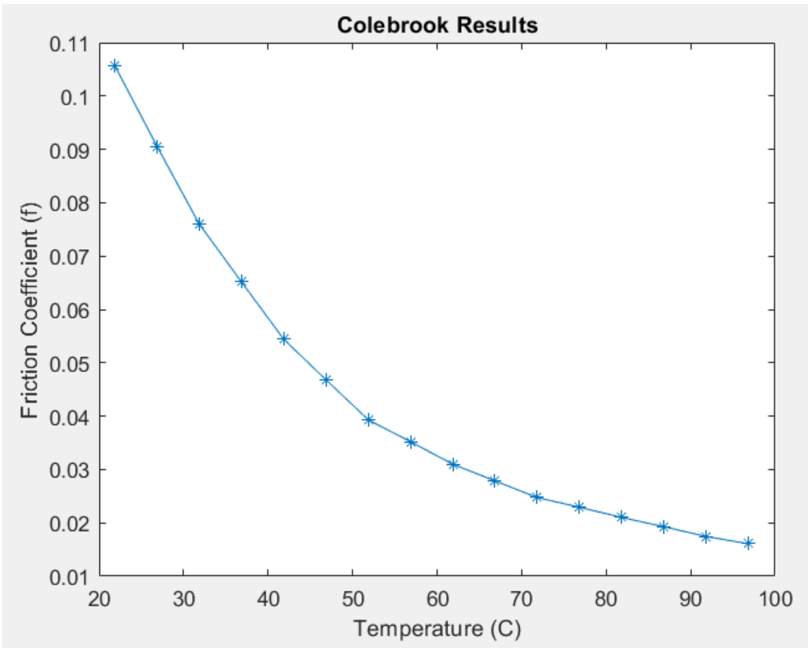
[3] Losses in Pipes (queensu.ca)

2.3.1 Friction Coefficient

For turbulent flows like in the pipeline, the friction coefficient can be calculated iteratively using the Colebrook Equation.

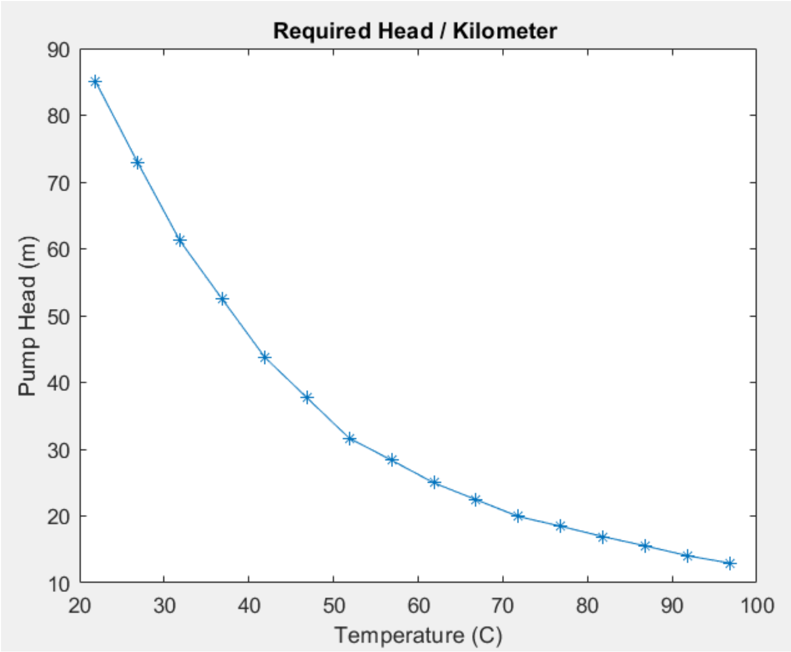
$$\frac{1}{\sqrt{f}} = -0.869 \ln\left(\frac{\varepsilon/D}{3.7} + \frac{2.523}{\mathrm{Re}\sqrt{f}}\right)$$

Where epsilon is the skin friction property, assumed to be 0.00001 for a steel pipe. The friction coefficient is variable depending on Reynold's number, which is reflected in the pipeline study due to changing Temperatures.



2.3.2 Head Loss by Temperature

Combining the Colebrook iteration with the Darcy-Weisbach equation, we get the head loss as a function of Reynold's number. Since our Reynold's number is a function of kinematic viscosity, plugging in our empirical relationship from 6.2.3 gives us a formula for required pump head as a function of temperature $\mathrm{H_f}(T)$.



2.3.3 Pump Energy Requirements by Temperature

Pump head can be converted into energy using the potential energy relationship.

$$W = \dot{m}g\frac{1}{L}\int_0^L H_f(t)\,dx$$

Where $\dot{m}$ is the rate of mass flow through the pipeline and $\frac{1}{L}\int_0^L H_f(t)\,dx$ is the averaged head friction of that section. Note that this is the work going into the fluid, if we are looking at the pump's actual energy consumption this W would be multiplied by the pump efficiency.

2.3.4 Pump Force and Pipe Pressure

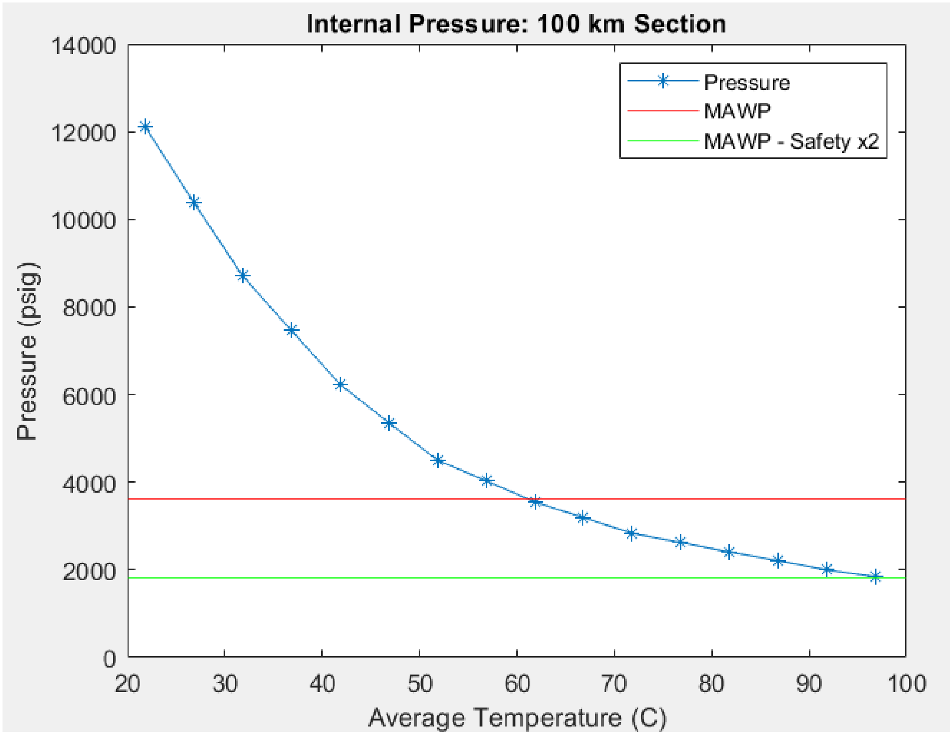
The force required at the pump follows a similar equation,

$$F = \frac{W}{L} = \dot{m}g\frac{1}{L^2}\int_0^L H_f(t)\,dx$$

Using the relationship below, we can convert the pump's head into the internal pressure.

$$p = \frac{9804.139}{L}\int_0^L H_f(t)\,dx\ [Pa]\ \ or\ p = \frac{1.422}{L}\int_0^L H_f(t)\,dx\ [Psig]$$

Recalling Table 1 from section 2.2.1, the maximum pressure for our pipeline class and temperature range is 3600 psig. A sample pressure plot is shown below for a 100 km section.



Note that the plot is just an example at this point. Once the heat model is considered in section 3, we will be able to procedurally determine the safe distance (clearly, 100 km is too far apart unless we design a very hot oil stream).

3 Heat Model

3.1 *SS* Heat Balance - Overview

The general heat model from pump to pump is derived from the Steady State heat balance shown below:

$$q_{applied} = q_{conv\_forced} + q_{conv\_natural} + \cancel{q_{cond}} \cancel{+ q_{r\_emitted}} \cancel{- q_{r\_solar}}$$

The heat applied at one pump is replenished at the next to keep the temperature profile consistent even if it changes between stations. The writeup explicitly says to ignore the emitted and absorbed radiation, assuming that only convection happens on the surface of the pipeline. Note that conduction effects of the pipeline saddle are also ignored as they are negligible.

Since the applied heat is not distributed equally along the pipeline, we need to modify the energy balance above to account for the heat lost between pumping stations. This is modeled using the one-dimensional heat equation for fluid flow in a pipe.

$$\frac{T(x) - T_{atm}}{T_0 - T_{atm}} = \exp\left(-\frac{P\Delta x\bar{\mathrm{U}}}{\dot{\mathrm{m}}\mathrm{c_p}}\right)$$

This model is simplified by assuming a constant heat flux out of the pipe. This assumption is only valid if we split the pipe up into differential chunks, which can be conveniently done through the MATLAB model. Since environmental conditions are constant and internal fluid temperature changes relatively slowly Δx can be set to 100 meters to ease computation without losing accuracy.

3.2 Constants and Constraints

There are additional assumptions that need to be made in order to develop a heat transfer model. The largest one is the weather conditions, the model will use a conservative estimate of Edmonton's winter conditions[4], a city in central Alberta. These numbers are 270 K (~-3 Celsius) for ambient temperature and 5 m/s for average windspeed. By using temperatures slightly colder and windier than average, the model will err on the side of caution when describing the heat needed to keep the pipeline flowing.

3.2.1 Environmental Conditions

Prandtl (Pr) - Air	0.71
k - Air	0.0243
Prandtl (Pr) – Crude	50
k - Crude	12
Cp – Crude	2000
Ambient Temperature (k)	270
Average Wind Speed (m/s)	5

[4] Edmonton Climate, Weather By Month, Average Temperature (Canada) - Weather Spark

3.3 Forced Convective Heat Transfer

For forced convection over a smooth pipe, we use the following relationship to estimate the Nusselt number. This equation is valid for cross flow over a cylinder with a Reynolds number ranging from 40,000 to 400,000, for this particular pipeline Re=257,430. Note that we are assuming that the 5 m/s wind is flowing completely perpendicular to the line: this is the most conservative estimate under the environmental conditions.

$$\mathrm{Nu_d} = 0.027Re_d^{0.805} * \mathrm{Pr}^{\frac{1}{3}} = 546.215$$

Solving for h using the Nusselt relationship:

$$\mathrm{h_{forced}} = \frac{\mathrm{k}}{\mathrm{D}}\,\mathrm{Nu_d} = 17.419\left[\frac{W}{m^2k}\right]$$

With that h, we have the following heat equation.

$$\mathrm{q_{cv,f}} = \mathrm{hA_s}(T - T_{atm})$$

This rate format is the same for all convective terms in our heat model. The combination of this heat rate with the other components will be discussed and equated in section 3.7.

3.4 Natural Convective Heat Transfer

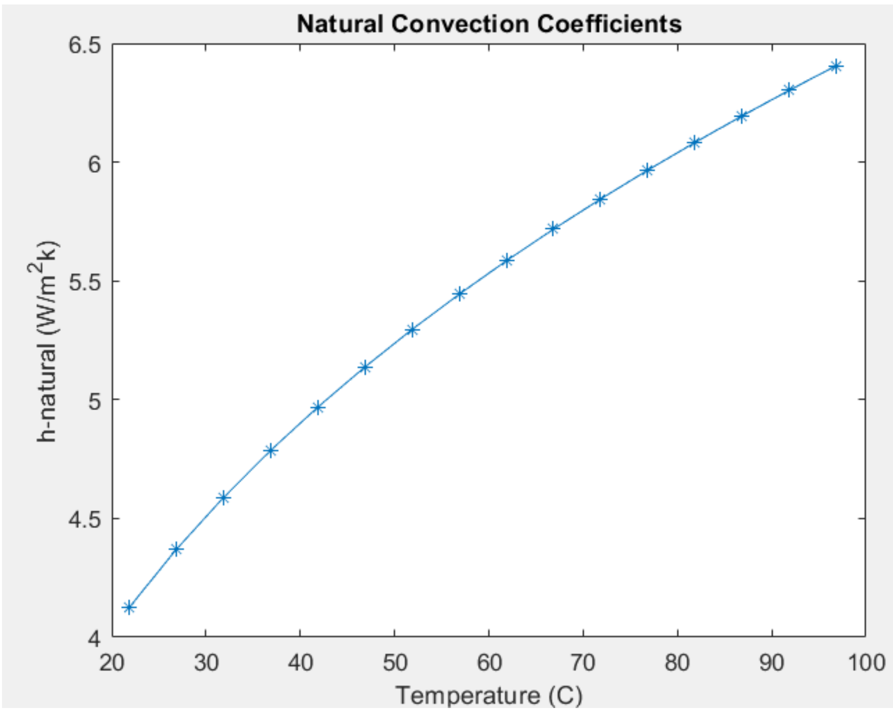
In this situation, natural (free) convection is not negligible so it’s important to factor into the heat model. The Nusselt number for natural convection is a factor of the Rayleigh Number, which is dependent on the surface temperature of the pipe. The relevant equations for a long horizontal cylinder are shown below.

$$\mathrm{Ra_D} = \frac{\mathrm{g\beta(T_s} - T_{atm})D^3}{\nu\alpha}$$

$$\mathrm{Nu_d} = \left(0.60 + \frac{0.387 Ra_D^{\frac{1}{6}}}{\left(1 + \left(\frac{0.559}{\mathrm{Pr}}\right)^{\frac{9}{16}}\right)^{\frac{8}{27}}}\right)^2$$

$$\mathrm{h_{free}} = \frac{\mathrm{k}}{\mathrm{D}} \mathrm{Nu_d}$$

This formula gives h values ranging from 4 to 6.5 $\frac{W}{m^2 k}$, which aligns with the intuition that the forced convection should be 3-4 times more effective. The values plotted against surface temperature are plotted below.



Natural convection acts in parallel to forced convection, so in the final heat model those two h coefficients will be stacked, this relationship is further discussed in section 3.7.

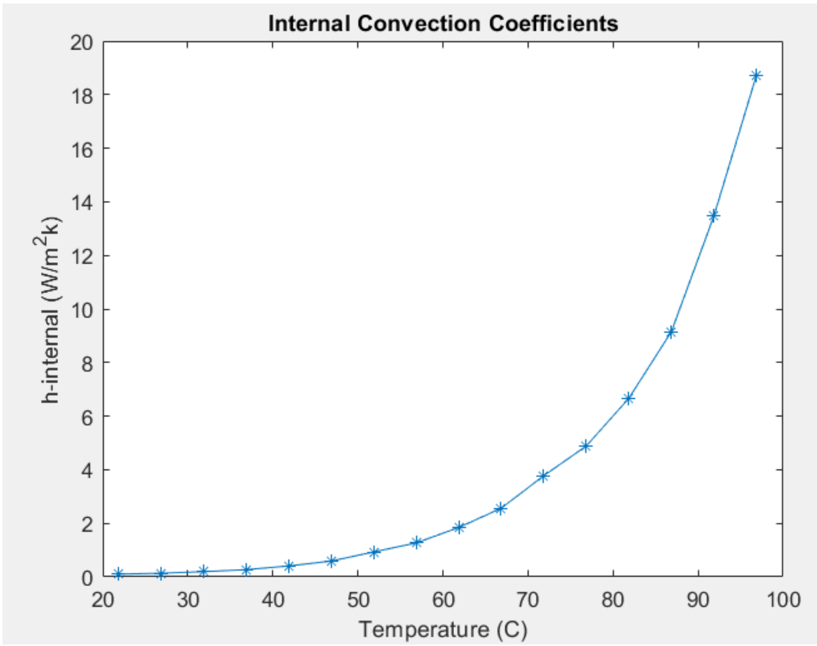
3.5 Internal Heat Transfer

At this temperature and viscosity, internal flow is fully turbulent. As such, the average Nusselt number can be determined from the following formula.

$$\mathrm{Nu_d} = 0.023 Re_d^{0.8} * \mathrm{Pr}^{0.3}$$

$$\mathrm{h_{internal}} = \frac{\mathrm{k}}{\mathrm{D}}\,\mathrm{Nu_d}$$

Since Re is dependent on temperature this coefficient varies for different fluid temperatures. These values are shown in the plot below and are incorporated into the MATLAB model.



3.6 Pipe Wall Conduction

In this situation, the thermal resistance of the insulated pipe wall is not negligible and must be factored into the heat equation. This is modeled as the thermal resistance:

$$\mathrm{R} = \frac{\mathrm{L}}{\mathrm{k}}$$

Since there is no explicit insulation width provided, we will set this at 10 mm and then explore the effects of this further in section 4.

3.7 Overall Resistance-Based Heat Transfer Relationship

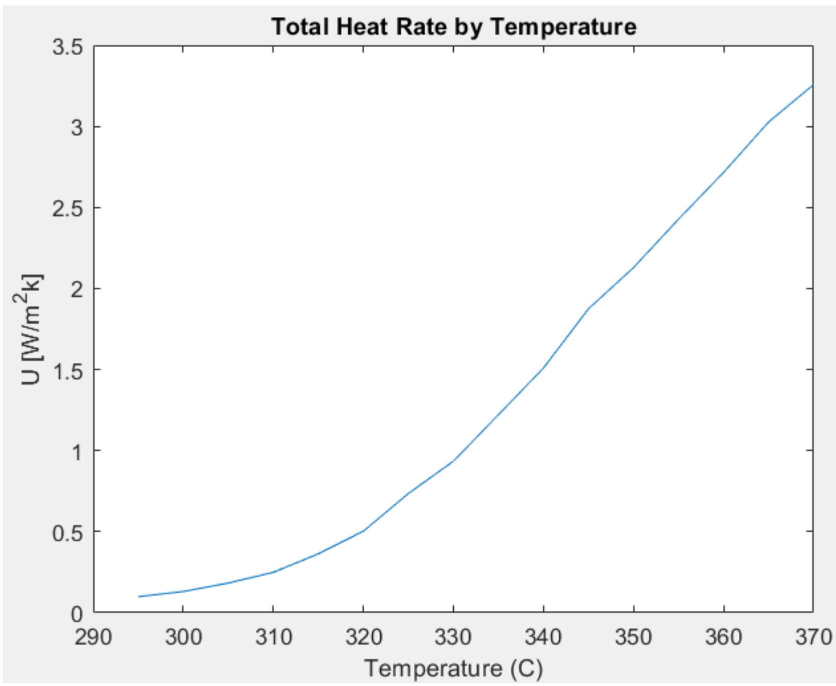
Recalling section 3.1, the temperature profile is dependent on the Ū term:

$$\frac{T(x) - T_{atm}}{T_0 - T_{atm}} = \exp\left(-\frac{P\Delta x\bar{U}}{\dot{m}c_p}\right)$$

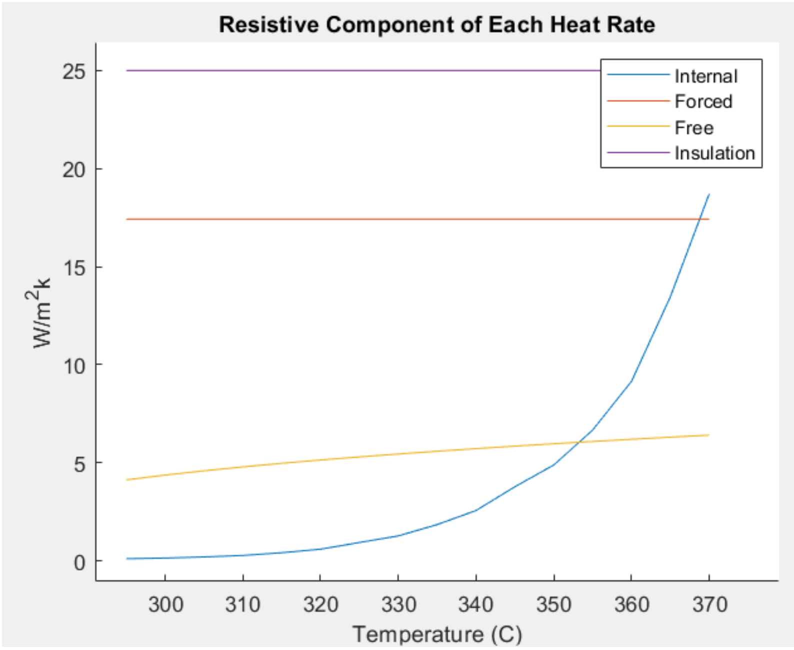
Where $\bar{U}$ is defined as the combination of all heat rates separating the fluid flow from the atmosphere. In this case:

$$\bar{U} = \left(\frac{1}{h_{internal}} + \frac{L}{k} + \frac{1}{h_{free} + h_{forced}}\right)^{-1}$$

This results in the total heat coefficient shown below.



The individual resistive components are included for comparison. The majority of the increase is being caused by an increase in internal convection as the temperature of the crude rises.

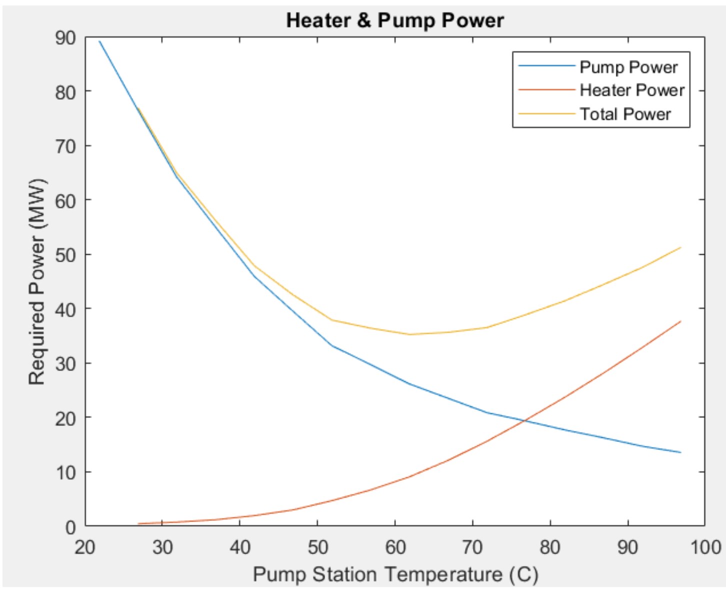


4 Analysis

Combining these two models, we can tune a few different parameters to create the best heating design given other circumstances such as insulation, distance, and pump station capabilities.

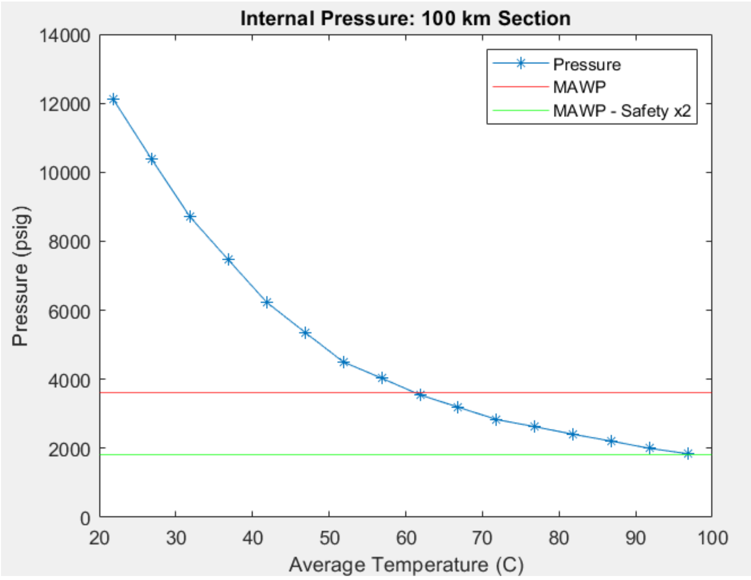
4.1 Optimal Heating Design

A pipeline in these extreme conditions requires two main power sources: energy for pumping the flow and energy for heating the flow. To maximize the company's profits, locate the minimum of the sum of these two quantities: the temperature where the lowest overall energy is needed (given that all other design requirements are met, such as pipeline pressure, volume, etc…).



A quick combination of these two models on the 100 km pipeline reveals that this optimal temperature is around 60 Degrees Celsius. Note that this comparison is using a standard pump efficiency of 0.75 and heater efficiency of 0.50, this assumption should be tuned given the exact specifications of the system.

Revisiting the pressure safety plot from section 2 (see below). Considering the energy analysis of different temperatures and the project parameters, it would be best to select a pump station temperature of 75 degrees Celsius. This provides a safety factor of greater than 1.5 on the pressure standard while still keeping energy costs relatively low and spacing out stations nicely.

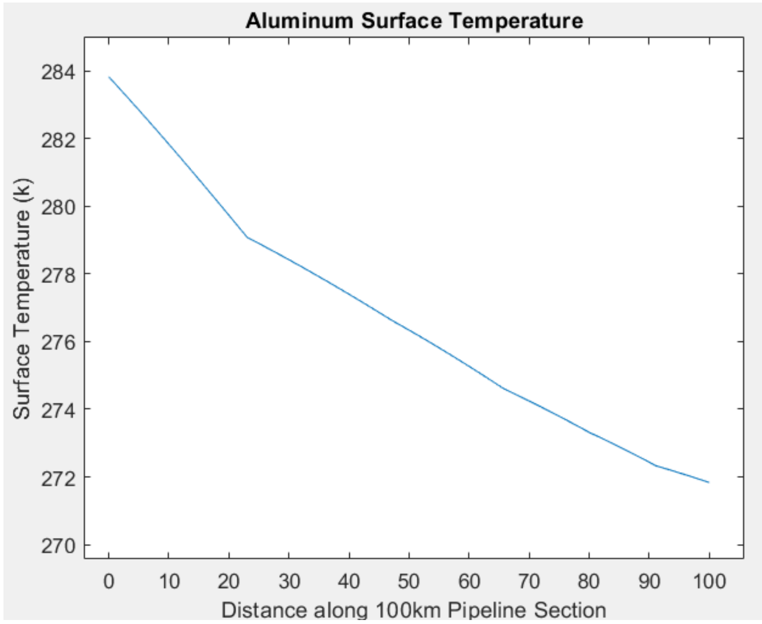


4.1.1 Surface Temperature Along Pipe

The surface temperature along the pipe can be calculated using a similar thermal circuit, this time considering the 1 mm of aluminum skin. Since the main heat model provides the fluid temperature, using the one-dimensional conduction rate formula can find us the temperature just under the aluminum skin. Since that layer is very thin and conductive, we can assume that the temperatures are the same.

$$q'' = \bar{U}(T(x) - T_{atm})$$

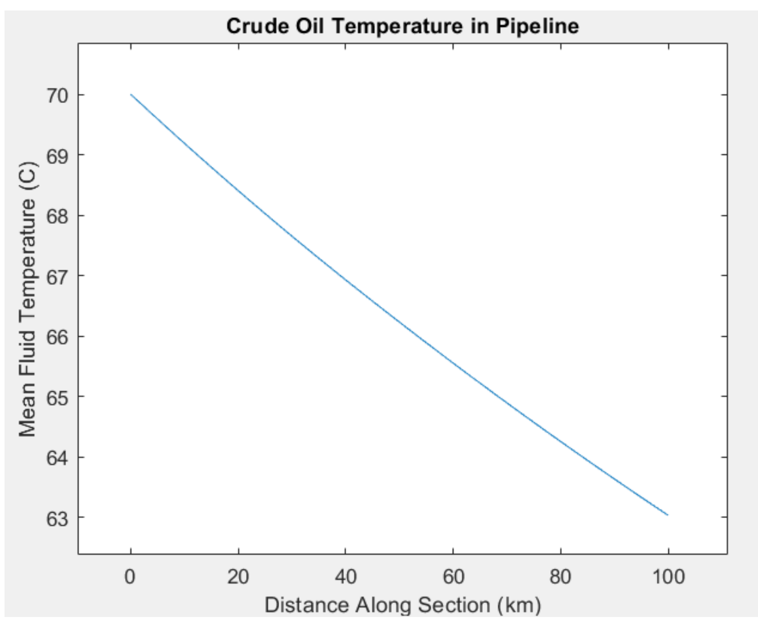
Plugging in our total heat passing through the surface, the plot below is generated.



As the crude oil in the pipe begins to cool down, the heat transfer is less effective which explains why the surface temperature towards the cooler end of the pipe is so much lower than at the start, even though the internal temperature doesn't vary that much (see section 4.1.2 below).

4.1.2 Output Temperature Between Pump Stations

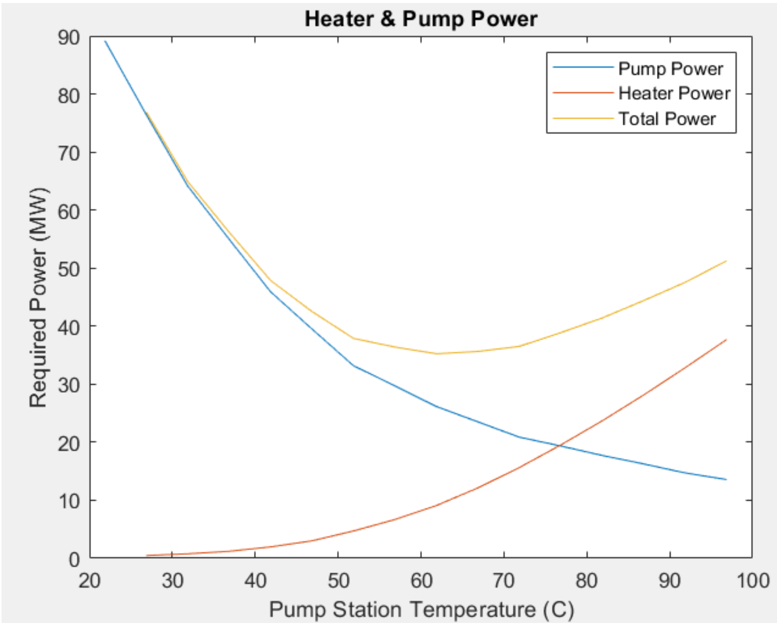
With only 1 cm of insulation, the harsh weather conditions take a significant amount of thermal energy out of the pipeline. In section 4.1 it was decided that the pump station temperature should be 75 degrees Celsius. The plot below shows the fluid temperatures inside a 100km section when the initial flow is at 75 degrees Celsius.



There is a 7 degree drop over the course of the pipeline. The outlet temperature at the oil stream is around 63 degrees Celsius.

4.1.3 Heating Load Between Pump Stations

The total heating load lost between pump stations is equal to the heat lost to convection. Revisiting our heat and pump power plot, at our station temperature of 75 Degrees this is around 20 MW for each 100 km section.

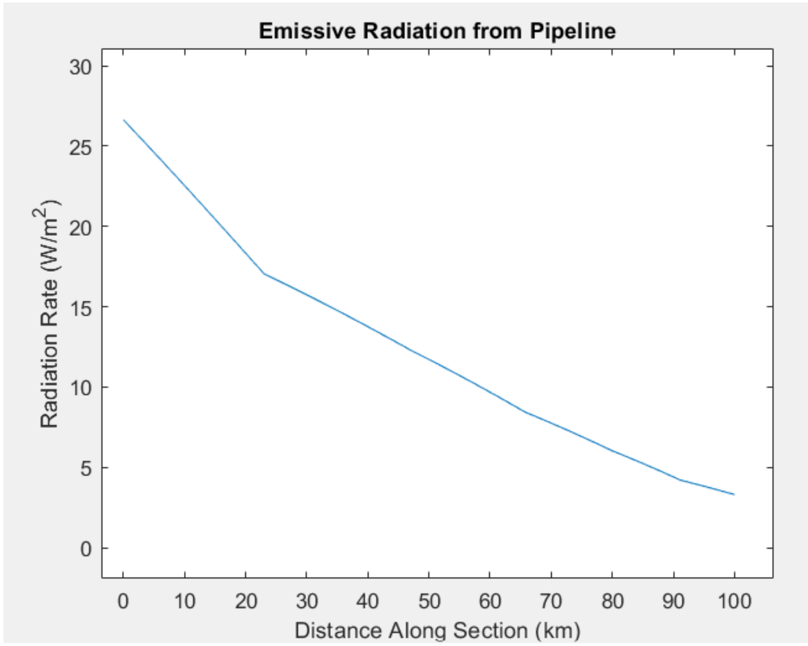


4.1.4 Emissive Radiation Loss

Emissive radiation rate is calculated by the following formula.

$$q''_{\mathrm{r}} = \varepsilon\sigma(T_s^4 - T_{atm}^4)$$

Using an emissivity of 0.4 due to coverage and corrosion of the aluminum skin, we get the plot below showing radiation along the length of the 100km pipeline section. Note that this plot uses the surface temperature of aluminum as calculated in 4.1.1.



These radiation rates integrated over 100 km of pipeline surface area sum to 2.989 MW, which is around 15% of the convective cooling losses. This number is not negligent and should be factored into future versions of the heat model.

5 Conclusions / Future Work

Overall, comparing the fluid and heat models for the pipeline opens the door for further tuning. By changing insulation, station distancing and starting temperatures the client can adjust the model based off what best fits their needs. Further research would be helpful to better see the effect of different heating components on the pipeline's losses.

6 Appendix

6.1.1 Fluid & Heat Model Code

```
% Pipeline V1
%% CTEs
% Physical Constants
sigma = 5.67e-8;

% Dimension Constants
g = 9.81;
height = 1.5;  % m
diameter = .762;
thickness_al = .001;
thickness_ins = .1;
daily_v = .86e6 * 0.158987; % Barrels --> m^3
P = pi * diameter;
distance = 1200e3;  % m

% Shell Constants
k_ins = .25;
emmissivity = .4;
pipe_epsilon = .01e-3; % https://www.engineeringtoolbox.com/surface-roughness-ventilation-ducts-d_209.html

% Fluid Constants
cp = 2 * 1000;
density = 900;
viscosity = 1;

% Atmospheric Constants
T_atm = 270;   % (-3.15 C)

% Solving for m_dot & related
v_dot = daily_v / (24 * 60 * 60);
v_fluid = v_dot / (pi * (diameter / 2)^2);
m_dot = v_dot * density;  % kg / s

% Reynolds
vsc_air = 1.48e-5;
wspd = 5;  % m/s
Re = diameter * wspd / vsc_air;

% Convection helpers
% Internal Forced
k_oil = 12e-3;
Pr_oil = 50;
Nu_i = 0.023 .* Re_pipe.^.8 .* Pr_oil.^0.3;
h_i = k_oil / diameter .*Nu_i;

% External Natural
Pr_air = 0.71;
k_air = 24.3e-3;
beta_air = 1/T_atm;  % (1/T_film)
alpha_air = 19.1e-6;
Ra = g .* beta_air .* (T_i - T_atm) .* diameter .^3 / (vsc_air .* alpha_air);
Nu_o_n = (0.60 + (0.387 .* Ra.^(1/6))./((1 + (0.559 ./ Pr_air) .^ (9/16)).^(8/27))).^2;
h_o_n = k_air / diameter .*Nu_o_n;

% Convection constants (temperature independent)
h_o = 17.419;  % Calculated by hand

% Running pipe head model
section_L = 100e3;  % m
T_i = [295:5:370];
vsc = get_vsc(T_i);
Re_pipe = v_fluid .* diameter ./ vsc;

% One off run to get fluid temp over time for target station temp
% T_list_70 = heat_model(70 + 273.15, km, h_i, 0.1, k_ins, h_o, h_o_n, T_atm, P, m_dot, cp, T_i);

syms f
f_list = [];
for i=1:length(Re_pipe)
    Re_d = Re_pipe(i);
    ls = 1/sqrt(f);
    rs = -0.869 * log((pipe_epsilon / diameter) ./ 3.7 + 2.523 ./ (Re_d .* sqrt(f)));
    f_val = double(solve(ls == rs));
    f_list = [f_list, f_val];
end
hf = f_list .* section_L / diameter * v_fluid^2 / (2*g);
q_pump = m_dot * g * hf;

% Converting to pascals at each temperature
p_pa = hf .* 9804.13943;
p_psig = p_pa ./ 6894.75728;

% Running heat model
km = section_L / 1000;
Q_list = [];
for i=1:length(T_i)
    T_list = heat_model(T_i(i), km, h_i, 0.01, k_ins, h_o, h_o_n, T_atm, P, m_dot, cp, T_i);
    % T_list_no_ins = heat_model(T_i, km, h_i, 0, k_ins, h_o, h_o_n, T_atm, P, m_dot, cp);
    q_delta = m_dot * cp * (diff(T_list));
    q_heater = sum(q_delta);
    Q_list = [Q_list, q_heater];
end

%% Raw Viscosity Data
% function [vsc] = get_vsc(T_i)
%     temp = [0 20 40 50 60 70 100 135 177] + 273.15;
%     visc = [10000 500 35.89 28.32 24.77 22.09 17.77 14.10 10.83] * 1e-6;
```

```
%     vsc = interp1(temp, visc, [T_i]);
% end

function [vsc] = get_vsc(T_i)
    % https://link.springer.com/article/10.1007/s13202-015-0184-8
    temp = [20 30 40 50 60 70 80 90 100] + 273.15;
    visc = [11000 5000  2000 700 300 120 60 25 10] * 1e-6;
    vsc = interp1(temp, visc, [T_i]);
 end

% function [vsc] = get_vsc(T_i)
%     % https://link.springer.com/article/10.1007/s13202-015-0184-8
%     density = 1000e-3; % g/cm^3
%     if (T_i <= 273.15 + 100)
%         a = 10.76;
%         b = 275.3;
%         c = 107.8;
%     else
%         a = 7.93;
%         b = 309.60;
%         c = 61.51;
%     end
%     vsc = exp(a + b / (T_i - 273.15)^2 + c * density^2 * log(density));
% end
%% Heat Components
function [T_list] = heat_model(T, km, h_i, thickness_ins, k_ins, h_o, h_o_n, T_atm, P, m_dot, cp, T_i)
   T_list = [T];
   dx = 100;
   steps = km * 10;
   for dl=1:steps

      syms T_o
      % U Values
      hi_local = interp1(T_i, h_i, [T]);
      h_o_n_local = interp1(T_i, h_o_n, [T]);
      U_cv = (1 / hi_local + thickness_ins / k_ins + 1 / (h_o + h_o_n_local) )^-1;
      % U_cvn = (1 / h_i + thickness_ins / k_al + 1 / h_o_n)^-1;
      % U_r is same for emissivity and absorptivity
      % U_r = (1 / h_i + thickness_ins / k_al + 1 / (sigma * emmissivity * (T_i^2 + T_o^2) * (T_i + T_o)))^-1;

      % T_im lineup
      % T_im = ((T_i - T_atm) - (T_o - T_atm)) / log((T_i - T_atm) / (T_o - T_atm));

      % Temperature equations
      %q = (U_cv) * P * dx * T_im;
      %q_delta = m_dot * cp * (T_i -  T_o);  % EQ 8.34

      % Solving one dx
      % T_i = solve(q == q_delta);
      % T_list = [T_list, T_i];

      % OR
      T = T_atm + (T - T_atm) * exp((-1 * P * dx * U_cv) / (m_dot * cp));
      %q_delta = m_dot * cp * (T_i -  T_o);  % EQ 8.34
      T_list = [T_list, T];
      %Q_list = [Q_list, q_delta];

   end
end
```

6.1.2 Plot Code

```
%% Plotting minimum total power use
pump_efficiency = 0.75;
heater_efficiency = 0.5;
hold off
plot(T_i - 273.15, q_pump .* pump_efficiency ./ 10^6);
hold on
plot(T_i - 273.15, -Q_list .* heater_efficiency / 10^6);
plot(T_i - 273.15, q_pump .* pump_efficiency / 10^6 - Q_list .* heater_efficiency / 10^6);
legend("Pump Power", "Heater Power", "Total Power");
title("Heater & Pump Power")
xlabel("Pump Station Temperature (C)")
ylabel("Required Power (MW)")


%% Colebrook
hold off
plot(T_i - 273.15, f_list, "*-");
title("Colebrook Results")
xlabel("Temperature (C)")
ylabel("Friction Coefficient (f)")

%% Head
hold off
plot(T_i - 273.15, hf ./ 100, "*-");
title("Required Head / Kilometer ")
xlabel("Temperature (C)")
ylabel("Pump Head (m)")

%% Required internal pressure
hold off
plot(T_i - 273.15, p_psig, "*-");
title("Internal Pressure: 100 km Section")
xlabel("Average Temperature (C)")
ylabel("Pressure (psig)")
yline(3600, "r")
yline(1800, "g")
legend("Pressure", "MAWP", "MAWP - Safety x2")

%% H_internal coefficients
hold off
plot(T_i - 273.15, h_i, "*-");
title("Internal Convection Coefficients")
xlabel("Temperature (C)")
ylabel("h-internal (W/m^2k)")
```

```matlab
%% H Natural Convection coefficients
hold off
plot(T_i - 273.15, h_o_n, "*-");
title("Natural Convection Coefficients")
xlabel("Temperature (C)")
ylabel("h-natural (W/m^2k)")

%% Plot convection resistances by each other,
% Grabbing each piece
hold off
ins_len = .01;
k_ins = .25;
internal = 1 ./ h_i;
forced = 1 ./ h_o * ones(1,length(T_i));
free = 1 ./ h_o_n;
ins = ins_len ./ k_ins * ones(1,length(T_i));
total = (internal + forced + free + ins).^-1;
hold on
plot(T_i, internal.^-1)
plot(T_i, forced.^-1)
plot(T_i, free.^-1)
plot(T_i, ins.^-1)
legend("Internal", "Forced", "Free", "Insulation")
title("Resistive Component of Each Heat Rate")
xlabel("Temperature (C)")
ylabel("W/m^2k")

%% Plot total conductivity
hold off
plot(T_i, total)
title("Total Heat Rate by Temperature")
xlabel("Temperature (C)")
ylabel("U [W/m^2k]")

%% Plot Surface of aluminum temp
k_al = 240;
len_total = diameter + .01;
U = interp1(T_i, U_cv, T_list)
q = U .* (T_list-T_atm);
T_surface = q .* U + T_atm;
hold off
plot(100:-.1:0, T_surface)
hold on
title("Aluminum Surface Temperature")
xlabel("Distance along 100km Pipeline Section")
ylabel("Surface Temperature (k)")

%% Output Temperature along pipeline
hold off
plot(0:.1:100, T_list_70-273.15)
hold on
title("Crude Oil Temperature in Pipeline")
xlabel("Distance Along Section (km)")
ylabel("Mean Fluid Temperature (C)")

%% Heating Load along pipeline
hold off
rad = emmissivity * 5.67e-8 * (T_surface.^4 - T_atm.^4);
plot(100:-.1:0, rad)
title("Emissive Radiation from Pipeline")
xlabel("Distance Along Section (km)")
ylabel("Radiation Rate (W/m^2)")
```