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Design Considerations of Subsea Cooling Spool

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Abstract
Thermal management of subsea pipeline and systems operating in HT/HP fields is major challenge facing the offshore oil and gas industry. Subsea cooling spool is one of the possible solutions to battle this challenge. However, this is an immature technology and more investigations are required to improve its design for future operations. This paper investigates the input variables in order to define the design envelope in terms of practical benefit, input temperature vs. throughput vs. spool length, and verify this by simulations with respect to given design parameters. A mathematical model was developed based on heat transfer theory and simulated using MATLAB. The results indicated that 4.36m of pipe is needed for generating 1℃ lower temperature output with a gas inlet temperature of 150℃.
Keywords: Subsea, Cooling spool, Heat transfer, High temperature, High pressure

1. INTRODUCTION

Recent discoveries of high temperature/high pressure (HT/HP) fields require thermal management systems to minimise the effect of HT on subsea pipelines and other production systems. Controlling the temperature of HT/HP fields would minimise global buckling of subsea pipelines(lateral and upheaval buckling), excessive stresses in pipelines, excessive loading on terminals at hot ends, de-rating of the pipeline carbon steel material, higher rates of internal corrosion, and lesser efficiency of cathodic protection systems (Azouz, 2010; Marsh, Zvandasara, Macgill, & Kenny, 2010). Paul (2007) reported that oil and gas recovered from HP/HT wells required a cooling system to reduce the temperature before separation process equipment could be operated. Subsea cooling spools are one of the technologies capable of performing this task. This technology is required to reduce the temperature of fluid hydrocarbon from HT subsea production well before they are transported in pipelines. The two common types of cooling spools used in the offshore oil and gas industry are the linear spool and integrated modular spool.

According to Azouz (2010) subsea cooling spools are part of the new innovative technology for rapid reduction of temperature to facilitate the processing of oil and gas in a more efficient and cost effective manner. However, this is an immature technology and more investigations are required to improve on its design for future operations.

Earlier application of cooling spools revealed design deficiencies such as lack of operational intervention and excessive weld cracks (Paul, 2007). Advance knowledge in fluid mechanics, thermodynamic and heat transfer is required to study and design subsea cooling spools.

Snake- lay technology and high pressure protection system (HIPPS) is another means of controlling the temperature of HT/HP field (Bai, Qi, & Brunner, 2009; Hooper, Maschner, & Farrant, 2004). However, these approaches have higher cost. Thus subsea cooling spools presents a great opportunity of reducing the higher cost associated with thermal management of HT/HP fields.

This paper investigates the input variables in order to define the design envelope in terms of practical benefit, input temperature vs. throughput vs. spool length, and verify this by simulations.

2. DESIGN EVELOPE

The open arrangement was investigated. This means that the water temperature profile is a constant value in the model. Factors considered in the model are highlighted in figure 1. The open arrangement is divided into three subsections; natural, forced and combined natural and forced convection. The combined natural and forced convection is considered for this study.

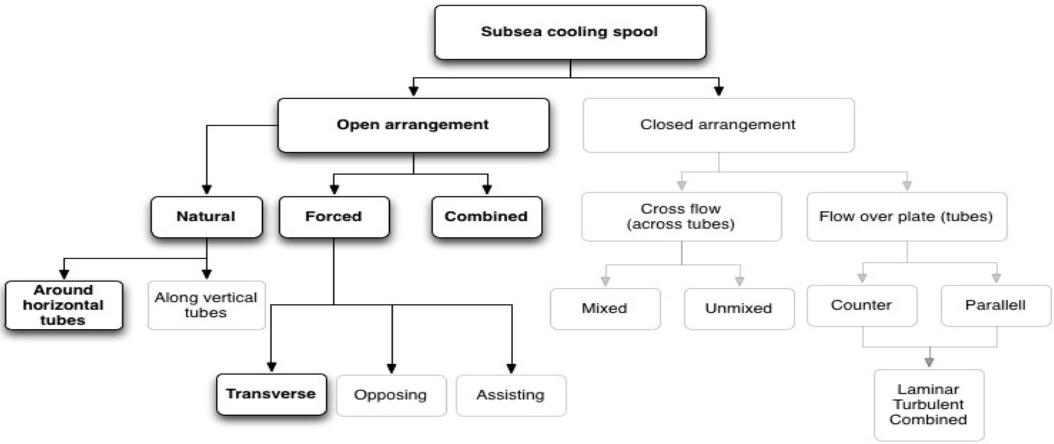

Figure 5: Overview of modelling arrangement

2.1Combined natural and forced convection

Cooling spool works as a heat exchanger and therefore, natural convection will always be present to some degree (see figure 2). However, as the velocity increases on the outside medium, the forced convection takes over. Due to the seawater depth used in the modelling, both natural and forced convection will be present. The ratio of Gr/Re^2 was employed to indicate the relative strength of both natural and forced convection.

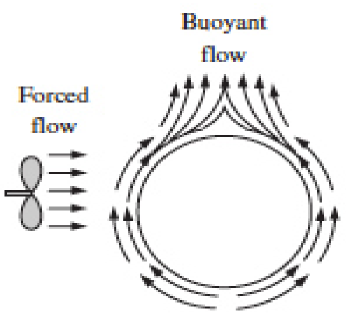

Figure 6: Combined convection (Cengel, 2003)

Cengel (2003) and Incropera & Dewitt (2002) proposed the following expression to combine the two HTC (heat transfer coefficient) into one:

$$Nu_{combined} = \left(Nu_{forced}^{n} \pm Nu_{natural}^{n}\right)^{\frac{1}{n}} \quad \textbf{(1)}$$

An assumption is made that natural convection is valid when seawater velocity is below 0.1m/s while forced convection is considered when seawater velocity is over 0.2m/s. Methane gas was used to test the model and assumed to be in single-phase along the whole pipe. Single-phase HTCs are expressed with the Re and Pr number. A simple and often used correlation for single-phase HTC is the Dittus-Boelter correlation.

For cooling expression:

$$Nu = 0.023Re^{0.8}Pr^{0.3} \quad \textbf{(2)}$$

This correlation is valid for Re numbers above 10 000 (turbulent flow) and for Pr numbers between 0.7 and 60. However, the Dittus-Boelter correlation may give errors as large as 25%. Gnielinski correlation is the most accurate and valid in a wider range of different flow regimes and is considered for the model. This correlation is mostly used and recommended for practical application (Cengel, 2003; Hewitt & Barbosa, 2008; Incropera & Dewitt, 2002)

$$Nu = \frac{\left(\frac{f_{smooth}}{8}\right)(Re-1000)Pr}{1+12.7\left(\frac{f_{smooth}}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}}-1\right)} \quad \textbf{(3)}$$

Where f_{smooth} is the friction factor which could be calculated or extrapolated from a moody diagram. .

3. MODELLING

This section described the general steps involved in obtaining the model. The linear type of cooling spool was considered in the modelling and the software used for the calculation and programing was MATLAB.

3.1. Overview of the modelling program

Natural, forced and combined convection flow over horizontal tubes was investigated, since that represents practical application in the field and the flow was assumed to be a cross flow. The tubes were also assumed to be straight without any bends. Fluid properties are found in appendix A and the rest of the input parameters for the reference model can be seen in table 1. The diameter and thickness of the pipe is for shallow water depth of 500m.

Table 3: Input parameters for reference model

Inner pipe diameter	0.494 [m]
Thickness	0.0072 [m]
Outer pipe diameter	0.508 [m] (20inch)
Inlet gas temperature	150 [°C]
Outlet gas temperature	120 [°C]
Mass flow of the gas	100 [kg/s]
Mean velocity of the gas	5 [m/s]
Velocity of the sea water	0.2 [m/s]
Sea water temperature	4 [°C]
Surface temperature at inlet	52.3 [°C]
Gas pressure	100 [bar]
Pipe material	Carbon steel

The mean velocity of the gas is calculated with the set mass flow of the gas, the density and diameter as:

$$\dot{m}_{gas} = \rho * \dot{V}_{gas} = \rho * \bar{u}_{gas} * A_c \rightarrow \bar{u}_{gas} = \frac{\dot{m}_{gas}}{\rho\pi\left(\frac{D_i}{2}\right)^2} = 5\ m/s \qquad \textbf{(4)}$$

3.2 Evaluating fluid properties

In the model, both thermodynamic and physical properties of gas were specified, since they do not exist in any equation of state. In appendix A fluid properties are presented. Methane is the product inside the pipe and seawater as the bulk fluid outside.

3.3. Discretization and numerical integration

The model does not use the logarithmic mean temperature difference when calculating the duty as it gives too much error. Instead, numerical integration is employed in order to find the required length. Use of numerical integration requires that the pipe must be divided into discrete elements where the properties are constant. The nodes when dividing the pipe can be placed in the middle or at the end of each segment. This study placed it at the end (see figure 3). Calculations begin in the first element where the inlet temperature is known and continues till the outlet temperature is reached to the desired temperature level.

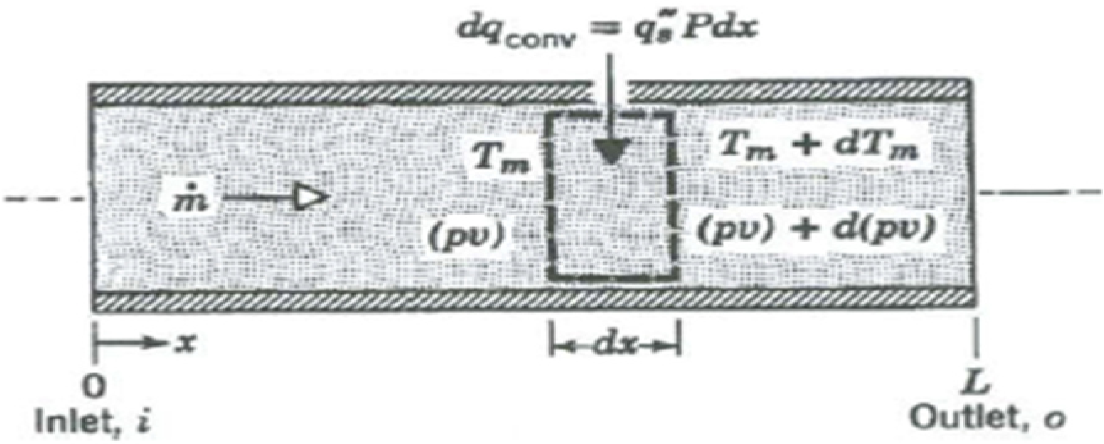

Figure 3: A segment for internal flow (Incropera & Dewitt, 2002)

3.4. Tube side temperature profiles

In order to obtain the temperature inside the pipe, the amount of heat transferred to the water must be known. This can either be achieved by taking the enthalpy at the inlet and outlet to obtain the duty or if the heat transfer occurs without condensation then (5) is appropriate.

$$dQ = -m_{gas} cp_{gas} dT_{gas} \quad (5)$$

Alternatively, the duty can also be expressed with the OHTC:

$$dQ = dA_o U_o (T_{gas} - T_w) \quad (6)$$

Combining (5) and (6), gives the following differential equation for the temperature:

$$\frac{dT_{gas}}{dA_o} = -\frac{U_o (T_{gas} - T_w)}{m_{gas} cp_{gas}} \quad (7)$$

In order to solve (7) numerically, it must be discretised. Since the tube diameter is constant the left side of (7) is expressed as dT_{gas}/dx, where x is the distance along the tube. The gradient of dT_{gas}/dx, was expressed by using a forward difference and the discretisation.

$$\frac{dT_{gas}}{dx} = \frac{T_{i+1} - T_i}{\Delta x} \quad (8)$$

Solving the (8) with respect to T_{i+1}, a forward Euler scheme is obtained:

$$T_{i+1} = T_i - \Delta x \frac{U_o (T_{gas} - T_w) \pi D_o}{m_{gas} cp_{gas}} \quad (9)$$

This Euler method is of first order, which means that the global error is in order of Δx. To achieve accurate result, value of Δx must be small enough to be acceptable. As mentioned earlier, the temperature for the waterside is constant since an open arrangement is considered.

3.5 Wall temperature at inlet and along the pipe

An iteration loop is necessary in order to obtain the inlet wall temperature and hence the outer HTC. The iteration loop is based on a heat balance at the inlet:

$$dA_o U_o (T_{gas,in} - T_w) = dA_o U_{iw} (T_{gas,in} - T_{wall,in}) \quad (10)$$

Where U_o and U_{iw} are functions of the wall temperature. U_{iw} at this instance represent the HTC for the inner wall, and the fluid inside the pipe. The iteration loop is set by changing $T_{wall,in}$ until the equation is satisfied within a given tolerance, refer to figure 4.

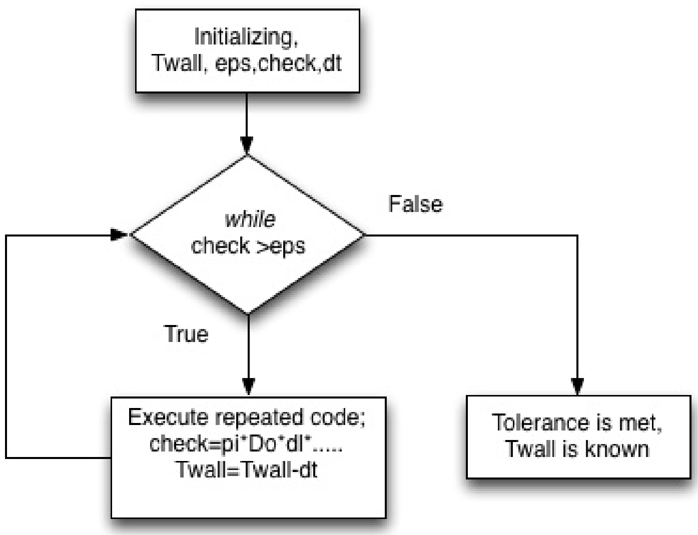

Figure 4: Iteration loop

3.6. Surface temperature
The same heat balance above is used along the whole pipe in order to calculate the surface temperature.

3.7. Variable heat transfer
Cooling spools are not supposed to reduce higher well temperatures below normal operational temperatures. This means that heat transfer must be controlled sensitively to achieve the overall objective of the spool. One approach in controlling heat transfer is to bury some part of the pipe in order to lower the heat transfer. However, if the pipe is buried there will be some heat transfer to the soil and this has to be considered. For a buried pipe the heat transfer is not symmetric so by using conduction shape factor the HTC for a buried pipe can be expressed (Bai, 2001):

$$h_{soil} = \frac{k_{soil}}{\left(\frac{D}{2}\right)\cosh^{-1}\left(\frac{2Z}{D}\right)} \quad \textbf{(11)}$$

Where, k_{soil}: thermal conductivity of soil (W/mK)
D: outside diameter of pipe
Z: distance between top of soil and center of pipe

If Z>D/2, $\cosh^{-1}(x)$ will be replaced by $\ln(x + (x^2 - 1)^{0.5})$, therefore,

$$h_{soil} = \frac{2k_{soil}}{D\ln\left(\frac{2Z+\sqrt{4Z^2-D^2}}{D}\right)} \quad \textbf{(12)}$$

A trenched pipe will experience less heat loss than an exposed pipeline. In order to express a partially buried part of a pipeline, heat transfer is calculated using a weighted average of a buried pipe and an exposed pipe (Bai, 2001) :

$$h_o = (1-f)h_{o,buried} + fh_{o,exposed} \quad \textbf{(13)}$$

Where, f: fraction of outside surface of pipe exposed to the surrounding fluid.

4. RESULTS
The reference model and the input parameters explained in section 3 produced the results contained in table 2 below.

Table 4: Result of the simulation of the reference model

Outputs:	
Length	106[m]
Duty	8.42[MW]
Mean U-value	381[W/m^2 C]
Mean outer HTC	991
Mean inner HTC	712
Mean Prandtl number inside	0.78
Mean Prandtl number outside	5.13
Mean Reynolds number inside	6.87e6
Mean Reynolds number outside	1.24e5

4.1. Length required for different gas temperatures at the outlet
Since the aim of the paper was to investigate the length of pipe that was needed for a certain output temperature, the simulation was performed for a range of output temperatures and obtained corresponding length. In figure 5 output temperature is one of the major factors affecting the length. As contained in table 2 above the length required for an output of 120°C was 106m and by changing that number to 130 °C the length is 68.5m. If lower output temperatures are simulated, the curve will take an exponential shape, since the heat flux is affected by the temperature difference.

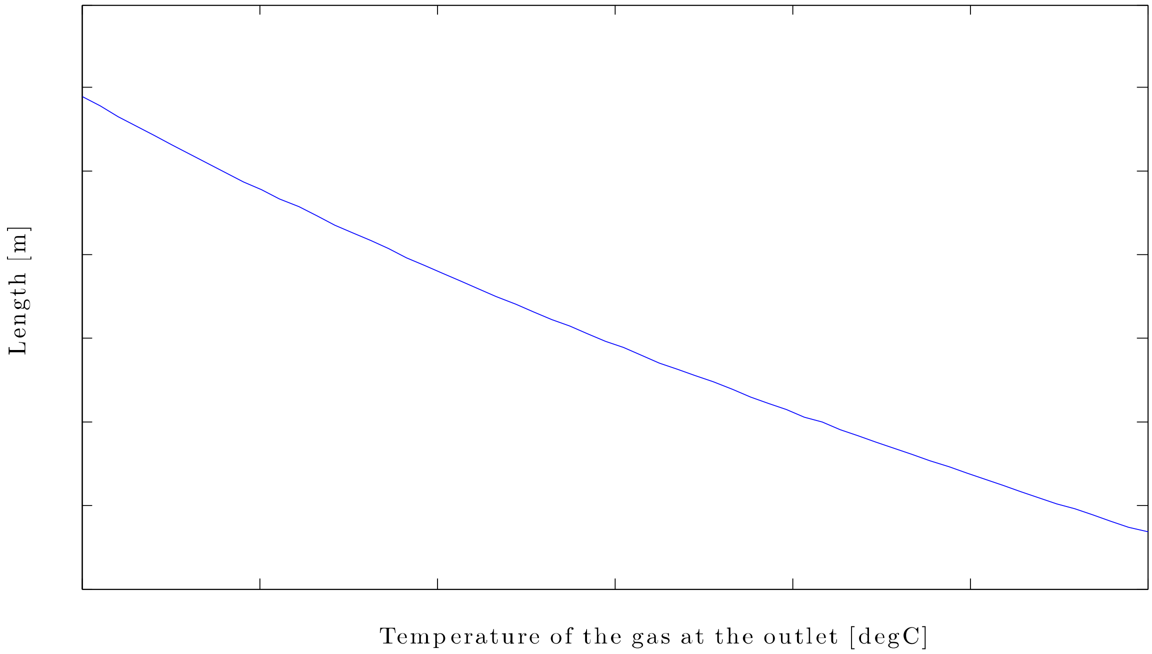

Figure 5: Length required for different gas temperatures at the outlet

4.2. Sensitivity analysis

Sensitivity analyses were conducted by changing one parameter at the time and see how it affects the required length. The model used was the reference model.

4.2.1. Size of the pipe

In this case the thickness of the pipe was set to handle the pressure and with corrosion allowance (CA) of 3mm. The simulation was performed in the same way as earlier but in this case another loop for the 8 different pipe sizes were added. As can be seen in figure 6 the different pipe sizes do not deviate so much from each other with respect to the length required. A clear difference can only be seen for the 8 inch pipe. As the output temperature gets lower also the 10 inch pipe starts to deviate from the others.

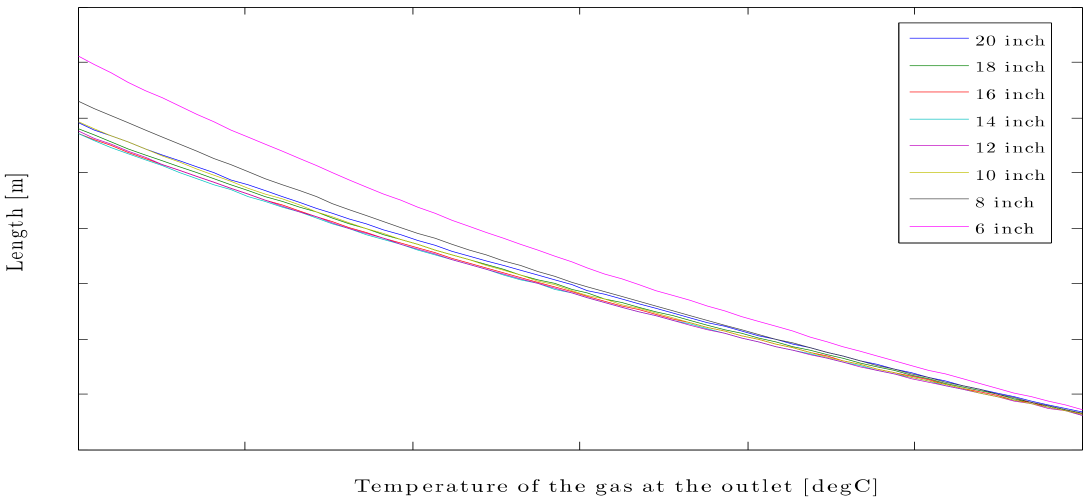

Figure 6: Comparison of how different pipe sizes affect the length.

4.2.2 Thickness of the pipe (CA and manufacturer tolerance)

As mentioned above the thickness of the pipe is calculated to handle the internal and external pressure with CA and manufacturer tolerance is added. In this case pipe diameter was fixed and the thickness was changed from the minimum CA of 3mm to 10mm. The result of the simulation can be seen in figure 7 where the conclusion can be made that the thickness doesn't make any major difference to the length. Instead the choice of thickness should be based on issues related to corrosion and manufacturer's tolerance rather than heat transfer. The reason why the thickness doesn't affect the heat transfer so much is because a pipe material is same for CA and manufacturers tolerance. In addition, cooling spools does not require massive coating in order not to inhibit heat transfer.

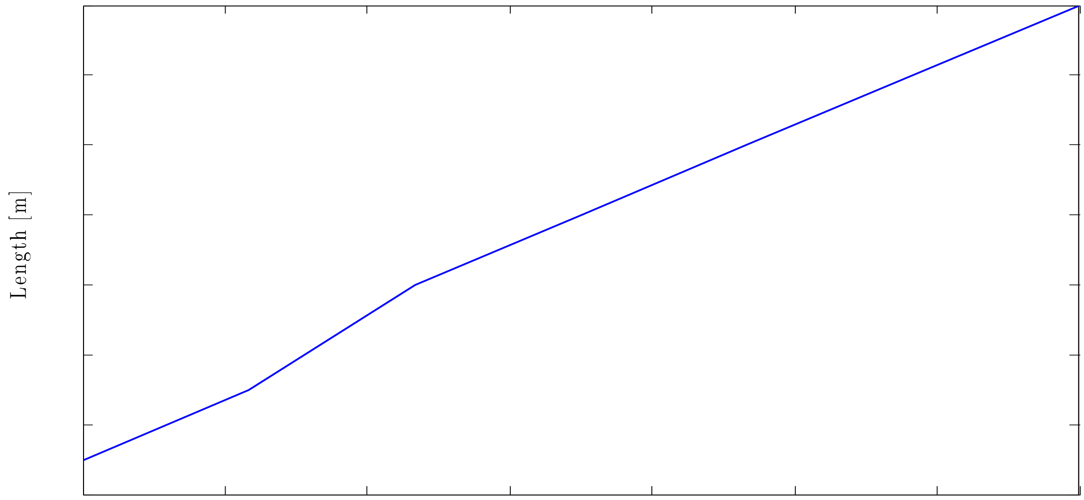

Figure 7: Sensitivity analysis of how the thickness affects the length

4.2.3. Seawater velocity

Another parameter discussed in section 2 is seawater velocity. The seawater conditions at the seabed are often uncertain and needs to be analysed carefully. As mentioned earlier the natural convection will be present at seawater velocity of 0.1m/s. This can also be seen in figure 8 (a) & (b), where the beginning of the lines is horizontal. Also, above 0.1 m/s, forced convection takes over (figure a) and the U-value increases as the velocity increases.

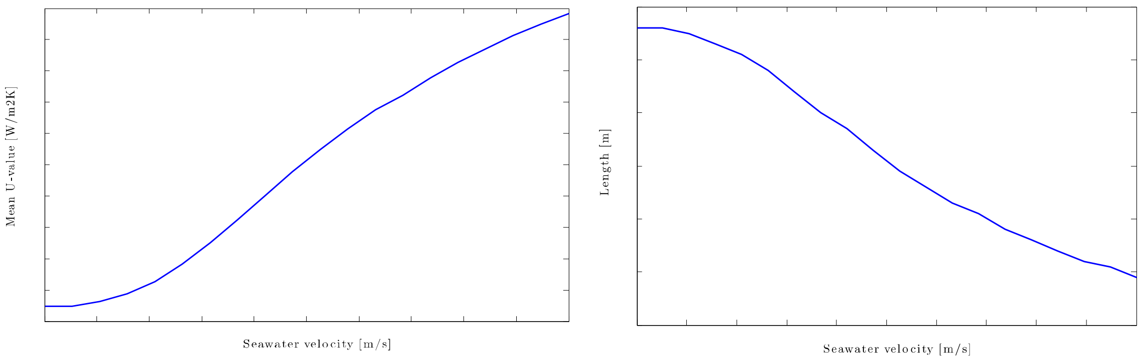

Figure 8: Sensitivity analysis of how the thickness affects the length (a) (b)

4.2.4. Buried and partially buried pipe

Some part of the pipe was buried in order to lower the heat transfer. Since the model was developed to calculate the length by integrating the pipe, therefore it was not possible to do minor engineering changes to the pipe. Instead a sensitivity analysis was performed by observing how the length would be affected by partially burying pipe. Figure 9 indicates how the length is changed for the reference model for different fractions of pipe exposed to the water. For example a 100% exposure means a fully exposed pipe and 0 % means a fully buried pipe. The length did not change significantly as expected, if the pipe is buried up to 40%. After 40% the length increases exponentially.

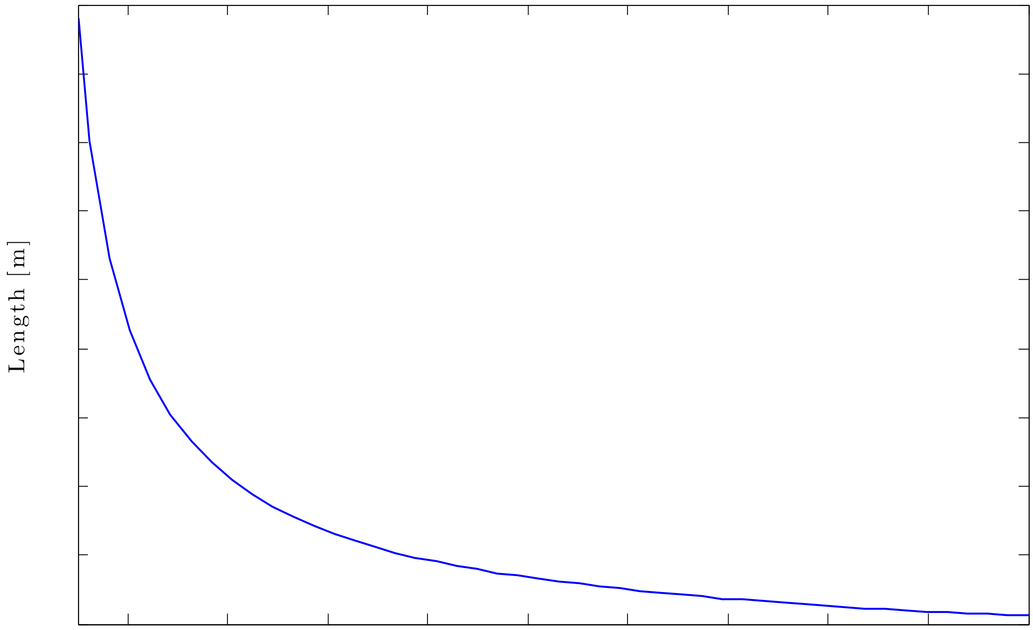

Figure 9: Length needed for different fractions of buried pipe.

5. DISCUSSION

The major factor that affects spool length is the desired output temperature of the gas. With an input of temperature of 150 °C and an output of 140 °C the required length to achieve the desired cooling is 33.5m of pipe. According to the inputs design parameters, subsea pipelines were supposed handling fluid at a temperature of 120°C. The simulation was required to estimate the length of spool required to deliver gas at 120°C to pipelines. 106m of spool length was the required length to perform this task. Due to the large difference between the water temperature and gas temperature the relation of output temperature and length are almost linear. However, as the difference gets lower the shape is more of an exponential form. Assuming, a linear correlation exists between the calculated values for each segment of the pipe, a mean value for how the length has changed with the output temperature is calculated at $4.36m/°C$ with a standard deviation of 0.74. This means that roughly 4.36m of pipe is needed for every lower degree at the outlet with conditions according to reference model. This value of course cannot be seen as a general one, but only to give an indication of the amount of temperature affecting length per design parameters.

The other major parameter that affects the length of spool is the U-value. The U-value consists of three major parts, the inner HTC, the outer HTC and the resistance in the pipe material. The model did not consider corrosion resistance coating and marine fouling. In the case of linear spool with the set inputs, the outer HTC is the higher with a deceasing shape along the pipe. This is due to lower wall temperature along the spool resulting in lower natural convection.

Pipe sizes from 8inch and above will not have any significant impact on the length. The choice of diameter can instead be based on other factors rather than the heat transfer.

6. CONCLUSION

The design envelope for linear cooling spool analysis should involve the important factors affecting heat transfer. These factors include the outlet temperature, density of the tube fluid, seawater velocity, mass flow and amount of pipe exposed to the water. Also good measurements of the seawater conditions are necessary to obtain accurate value of U. Factors that do not affect the length significantly are the pipe diameter and pipe thickness. These parameters can instead be designed with respect to other issues like corrosion. The direction of seawater flow must also be analysed in future work beside the seawater velocity. This might also have impact on heat transfer as this was not considered by this study.

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APPENDICES

Nomenclature

Gr	Grashof number
Nu	Nusselt number
Re	Reynolds number
Pr	Prandlt number
$\dot{m}$	Mass flow rate
ρ	Density
$\dot{V}$	Volume flow rate
A	Surface temperature
$\bar{u}$	Mean velocity of gas
cp	Heat capacity

Methane properties

CH4 properties			
Density [kg/m3]			
	80°C	60°C	40°C
100 bar	58,83	63,98	70,42
60 bar	34,58	37,28	40,55
30 bar	16,89	18,05	19,42
Cp [J/kg-C]			
	80°C	60°C	40°C
100 bar	2848,4	2866,8	2921
60 bar	2672,9	2651	2645,8
30 bar	2537,5	2488	2444,4
Viscosity [cP]			
	80°C	60°C	40°C
100 bar	0,0149	0,01451	0,01417
60 bar	0,01393	0,01341	0,0129
30 bar	0,01334	0,01277	0,01218
Thermal conductivity [W/m-C]			
	80°C	60°C	40°C
100 bar	0,05164	0,04948	0,04764
60 bar	0,04821	0,04567	0,04332
30 bar	0,04568	0,04294	0,04033

Water properties

Water properties											
Density [kg/m3]											
	100°C	90°C	80°C	70°C	60°C	50°C	40°C	30°C	20°C	10°C	0°C
100 bar	962.9	969.8	976.2	982	987.5	992.3	996.5	1000	1002.7	1004.4	1004.8
Cp [J/kg-C]											
	100°C	90°C	80°C	70°C	60°C	50°C	40°C	30°C	20°C	10°C	0°C
100 bar	4195	4184	4174	4167	4161	4157	4155	4154	4155	4160	4172
Dynamic viscosity 1e-3*[Pa*s]											
	100°C	90°C	80°C	70°C	60°C	50°C	40°C	30°C	20°C	10°C	0°C
1 bar	1.78	1.31	1	0.798	0.653	0.547	0.467	0.404	0.355	0.314	0.28
Thermal conductivity [W/m-C]											
	100°C	90°C	80°C	70°C	60°C	50°C	40°C	30°C	20°C	10°C	0°C
1 bar	0.6809	0.6761	0.67	0.663	0.654	0.644	0.632	0.619	0.605	0.588	0.57
Expansion coefficient [1/K]											
	100°C	90°C	80°C	70°C	60°C	50°C	40°C	30°C	20°C	10°C	0°C
1 bar	0.752	0.665	0.643	0.585	0.523	0.457	0.385	0.303	0.207	0.088	-0.07