Thermodynamic Modelling of Subsea Heat Exchangers

Kimberley Chieng

Eric May, Zachary Aman
School of Mechanical and Chemical Engineering

Andrew Lee Steere
CEED Client: Woodside Energy Limited

Abstract

The mechanical design of subsea heat exchangers involves consideration of the inlet pressure, seabed temperature, temperature and flow rate of the process fluid, frictional pressure loss through the pipe and, ultimately, operating cost. Subsea process engineering may be enhanced by the development of a thermodynamic model that allows for global optimisation over a range of design parameters. This work introduces a simple model, implemented in Microsoft Excel with Visual Basic for Applications, to explore the behaviour of a gas-dominant subsea heat exchange unit operation. The finite difference approach is employed within the model to estimate heat transferred to the environment over two input parameters: pipe diameter and pipe wall thickness. An initial comparison to OLGA 7.2.3 will be performed. This study also focuses on the sensitivity of heat transfer due to external marine fouling.

1. Introduction

One of the future challenges for the oil and gas industry is the recovery of reserves from remote, deepwater reservoirs. Production in harsh operating conditions may be marginal within the expense and capability of existing technologies. Subsea processing has received attention in recent years, as this may be an enabling technology for deepwater assets. It can offer several advantages for deepwater and ultra-deepwater fields, which tend to have higher water cuts due to the depth (Elde, 2005). Subsea heat exchange is a critical processing step that drives gas-liquid separation prior to re-compression and transport to the surface. An accurate description of the subsea heat exchange process is required to predict separation efficiency. Unintended water in the gas stream may lead to further processing and flow assurance complications.

Five process conditions are required to predict the behaviour and operation of a basic heat exchange unit: (i) inlet pressure, (ii) frictional pressure loss, (iii) ambient fluid temperature, (iv) inlet fluid temperature, and (v) inlet fluid flow rate. These conditions complement an array of system properties required in heat transfer calculations, such as the heat capacity of the process fluids and overall heat transfer coefficient of the pipeline. Fouling effects – such as external marine fouling, internal and external corrosion, and hydrate formation – must also be considered in the heat transfer operation. This project seeks to establish a simple thermodynamic model, which incorporates the above process conditions, system properties, and fouling effects.

Heat exchangers may be categorized according to flow arrangement and type of construction, the most common of which is the shell and tube heat exchanger (Bergman et. al., 2011). Subsea heat exchange systems may further be classified as active or passive. This project focuses on the heat exchange using a passive system, where tube bundles containing warm process fluid are exposed to surrounding seawater. The seawater flows over the tube bundles due to ocean currents, and is assumed to be an infinite thermal reservoir. Process fluid is cooled via natural oceanic convection.

An active cooler makes use of a forcing impeller to push the seawater over the tube bundles, resulting in a greater rate of cooling. However, inclusion of an impeller for active cooling implies the installation of an additional motor to drive. While the greater heat exchange is desirable, this ultimately leads to increases in operating and maintenance costs as well as increased risk in reliability.

Subsea heat exchangers are currently in use only to a limited extent around the world. The Asgard field, operated by Statoil, is expected to come online in 2015. Aker Solutions, who were awarded the contract for Asgard, have developed a subsea cooling module that makes use of an active cooler to reduce the temperature of the process fluid prior to compression.

Transient multiphase flow simulators, including OLGA®, are capable of modeling fluid flow and heat transfer of oil and gas through subsea pipelines. These simulation packages may require extensive computational times, limiting the subsea process engineer's ability to optimise the solution. This work focuses on the development of a simple thermodynamic tool, which may readily identify the optimal process conditions. The objective of this model is to provide a user-friendly interface, combined with a rapid-solution optimisation algorithm for subsea heat exchange unit operations.

2. Methodology

2.1 Finite Difference Approach

The following assumptions were made for this study:

1. The bulk fluid is seawater, and is considered as an infinite reservoir
2. The tube bundle is spaced sufficiently to allow the assumption above
3. The system is operating at steady-state
4. There is no heat generation due to friction in the walls of the heat exchanger tubes
5. Internal fluid properties are homogenized by volume fraction

These assumptions allow for simplification of the thermodynamic model, and provide initial estimates for modelling the base scenario. The effects of fouling by marine growth can then be incorporated into the model.

The modes of heat transfer assumed for this system are one-dimensional steady state conduction through the pipe wall, followed by convective heat transfer between the external pipe wall and bulk fluid. This is shown schematically in Figure 1. The process fluid is assumed to be in thermal equilibrium with the internal pipe wall, while the external pipe wall is assumed to be in thermal equilibrium with the bulk fluid. The heat flux due to conduction can be found by the following equation (Welty, 2007):

$$q_x = \frac{k}{L}(T_{\text{inner}} - T_{\text{outer}}) \tag{1}$$

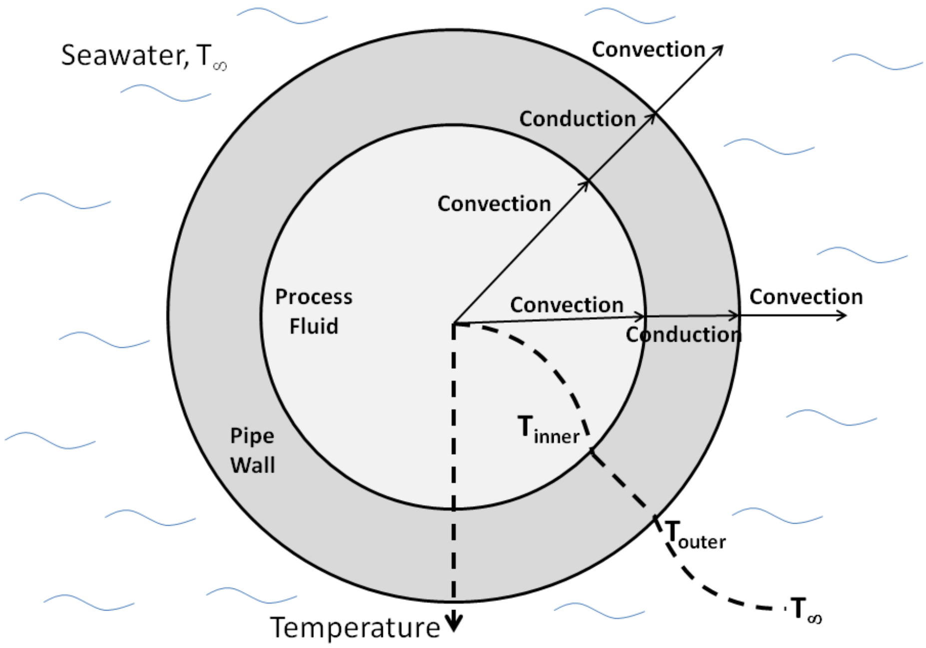

Figure 1 Heat transfer modes assumed for thermodynamic model

where q_x is the heat transfer per unit area, k is the coefficient of thermal conductivity for the pipe wall, L is the length over which thermal energy is transferred, and T_{inner} and T_{outer} are the temperatures at the boundaries of the pipe wall.

Similarly, the heat flux due to convection can be found by (Welty, 2007):

$$q_x = h(T_1 - T_2) \tag{2}$$

where h is the convective heat transfer coefficient, T_1 and T_2 are the temperatures of the two substances between which heat is transferred (e.g. for the second convective case in Figure 1, T_1 is T_{outer} and T_2 is T_∞), with $T_1 - T_2$ being the temperature difference.

These differential equations describe heat flux across radial and axial spatial dimensions as a function of time. The analytical solution becomes increasingly complex, however, when the transient process fluid is taken into account. Heat transfer in the radial direction (i.e., to the seawater) is assumed to operate at steady state. Further, the dependence of fluid temperature on flow length (i.e., changes along the pipe axis) are simplified from their differential form using a finite difference approximation (Morton & Mayers, 2005).

Small spatial and temporal steps are used in the model to support the implementation of finite difference approximations for heat loss within the process fluid; this approach enhances the solution speed for global optimization of the heat exchanger. Figure 2 illustrates the finite difference approach, where each square represents an individual element in the tube wall of the heat exchanger. As the initial pipe diameter and volumetric flowrate of process fluid are provided as input specifications, the linear fluid velocity may be estimated directly; process fluids are assumed to flow at constant velocity, with negligible decreases from frictional pressure drop. Knowledge of fluid velocity allows for coupling between spatial and temporal integration steps; the finite difference heat transfer equation may be reduced to one-dimension (residence time).

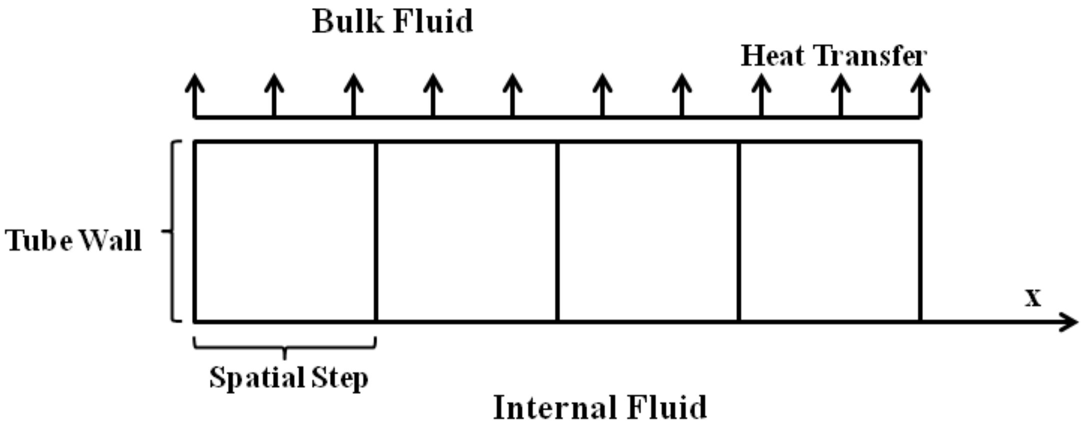

Figure 2 Illustration of the finite difference method

2.2 Thermodynamic Equations

A generic energy balance, over an open control volume, is governed by the following energy balance (Geankoplis, 1993):

$$(heat\ in) - (heat\ out) + (heat\ generated) = accumulation \tag{3}$$

Instead of focusing on one spatial step (i.e., section of the pipeline), the constant velocity assumption discussed above enables a change in reference frame to one "constant volume" packet of fluid (which loses heat as it traverses the pipeline network). With the assumed negligibility of heat generation, this energy balance may be reduced (Geankoplis, 1993):

$$accumulation = (heat\ in) - (heat\ out) \tag{4}$$

$$\frac{dU}{dt} = \dot{m}_{in}h_{in} - \dot{m}_{out}\mathrm{h}_{\mathrm{out}} - Q \tag{5}$$

where Q represents the heat energy lost through the pipe walls, $\frac{dU}{dt}$ represents the change in fluid internal energy per unit time, $\dot{m}$ is mass flow rate, and h is the enthalpy of the fluid.

A mass balance over the same control volume is provided below (Geankoplis, 1993):

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dM}{dt} \tag{6}$$

where $\frac{dM}{dt}$ is the change in mass per unit time.

Assuming steady-state conditions, the mass balance may be simplified:

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m} \tag{7}$$

Substituting equation (7) into equation (5) yields:

$$\frac{dU}{dt} = \dot{m}(h_{in} - h_{out}) - Q = \dot{m}\Delta h - Q \tag{8}$$

where Δh is the change in enthalpy, and

$$Q = h_{eff}A\int \mathrm{T}\,dt \tag{9}$$

where h_{eff} is the overall heat transfer coefficient, A is the surface area over which convection takes place, T is the temperature and $dt \cong \Delta t$ for small Δt.
Therefore, we obtain:

$$\frac{dU}{dt} = \dot{m}\Delta h - h_{eff}A(T_x - T_\infty) \tag{11}$$

The overall thermodynamic equation that will be used to model heat transfer through the subsea cooler is thus found as:

$$\boldsymbol{\Delta U = [\dot{m}\Delta h - h_{eff}A(T_x - T_\infty)]\Delta t} \tag{12}$$

2.3 Thermodynamic Model

Figure 3 illustrates the algorithm followed for the construction of the thermodynamic model. The model has been built to ultimately provide a solution quantified by cost. The mechanical design will satisfy internal and external pressure containment requirements due to process fluid pressure and hydrostatic pressure respectively. This influences material selection, which has an impact on the effective heat transfer coefficient, resulting in a change in effective tube length. As an additional simplification, Duplex Stainless Steel is the only material modelled in this project. The solution will satisfy the dimensional constraints imposed by the client in order to preserve practicality. It must also satisfy an outlet pressure constraint as this is what will ultimately drive production volume and recovery rate.

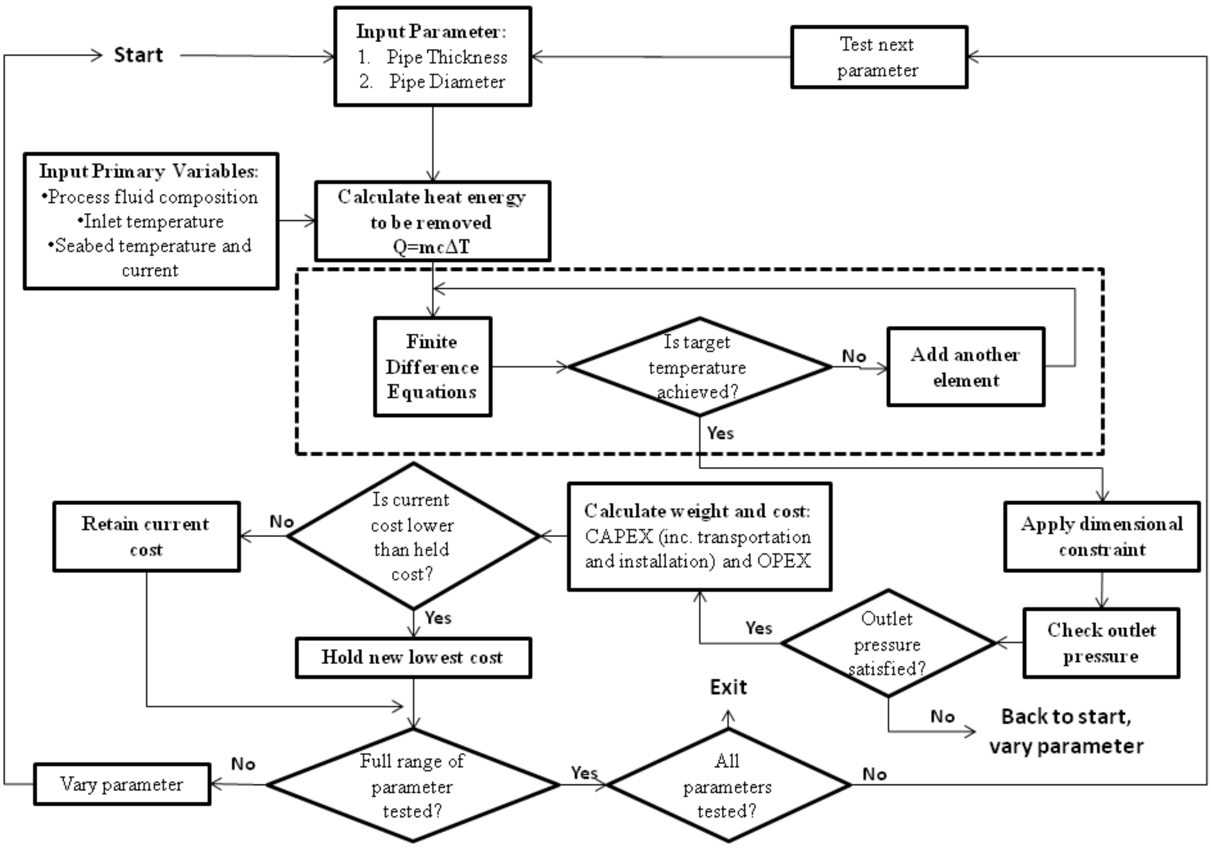

Figure 3 Algorithm flow for thermodynamic model

3. Results and Discussion

The base scenario has been modelled in Microsoft Excel and convergence obtained for a single set of operating variables. With input parameters of (i) Pipe Diameter = 0.05m, (ii) Pipe Thickness = 0.005m, (iii) Ambient Temperature = 280K, (iv) Inlet Temperature = 330K and (v) Desired Outlet Temperature = 300K, the effective length is calculated as 305 metres.

Figure 4 illustrates the convergence towards a solution for a single step in the thermodynamic model. The results obtained from the model are yet to be validated. The next step in this study is to perform a global optimisation across all input parameters and to verify the results against previously established models.

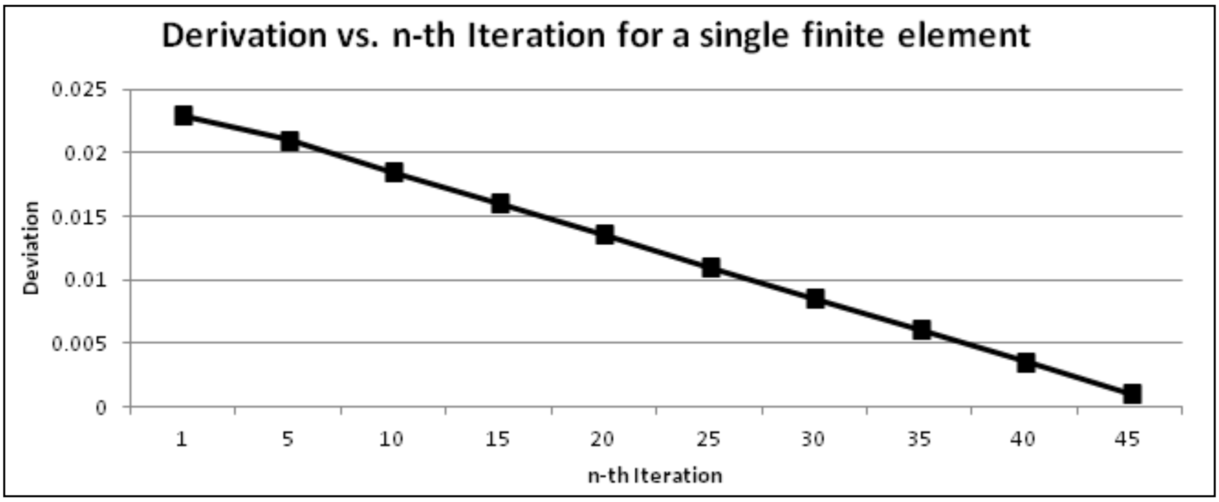

Figure 4 Graph of deviation vs. n-th iteration within a single finite element to converge towards a solution

4. Conclusions and Future Work

This study provides a starting point for subsea heat exchanger design. However, in order to simplify the project, sensitivities such as erosion, corrosion and hydrate formation have been excluded. In practice, oil and gas wells may produce sand which could lead to erosion of internal pipe walls. The effects of erosion and the formation of hydrates on heat exchanger efficiency may be incorporated into the model, and may form the basis of future studies. Detailed consideration of external corrosion and the impact of corrosion prevention mechanisms may also be analysed. There is also scope for CFD modelling of the effects of seabed currents on heat transfer.

5. References

Bergman et. al., 2011. *Fundamentals of Heat and mass Transfer*. Seventh ed. United States of America: John Wiley & Sons, Inc..

Elde, J., 2005. Advantages of Multiphase Boosting. *Exploration and Production: The Oil and Gas Review*.

Geankoplis, C. J., 1993. *Transport Processes and Unit Operations*. 3rd ed. s.l.:Prentice Hall PTR.

Morton, K. & Mayers, D., 2005. *Numerical Solution of Partial Differential Equations, An Introduction*. s.l.:Cambridge University Press.

Welty, J. e. a., 2007. *Fundamentals of Momentum, Heat, and Mass Transfer*. 5th ed. s.l.:John Wiley and Sons.