17th European Symposium on Computer Aided Process Engineering – ESCAPE17
V. Plesu and P.S. Agachi (Editors)

Model Requirement for Control Design of an LNG Process

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Abstract

Systematic control structure design requires the use of a model of the plant. Developing rigorous plant models based on physical and chemical principles is often a laborious and time consuming task. This paper studies the effect of simplifications of the heat exchanger model on the resulting control structure design for the PRICO LNG process. It is found that significant model simplifications may be introduced without compromising the control structure design.

Keywords

Liquefied Natural Gas, Dynamic Modeling, Plate-fin Heat Exchanger, Thermal Conductivity, Control Structure

1. Introduction

A systematic approach to control system development is to develop a dynamical model for the plant based on the laws of physics. This model then can be used to understand plant dynamics and to develop a robust control system for plant so that plant can be reliably operated close to its optimal operating point. However, physical modeling requires time and expertise, and is thus a costly endeavor. It is clear that adding more complexity to the model makes it more accurate but requires more effort and time. Therefore it is important to know the required level of model complexity in order to achieve the design of a reliable and efficient control system. This work is focused on addressing this question for modeling of liquefaction unit of LNG plants.

2. Process Description

This work studies a common LNG process, known as the PRICO (Poly Refrigerant Integrated Cycle Operations) process (Fig.1). This is a relatively simple LNG process, but has all the main components that are present in more complex LNG plant designs. We have chosen gPROMS for the entire modeling and Multi-flash as a physical property package for calculation of physical properties.
In the PRICO process, natural gas enters the heat exchanger with a pressure of around 60 bars and temperature of about 12 C. Natural gas is composed of methane, ethane, propane, n-butane and nitrogen. A mix-refrigerant having the same components cools the natural gas in the heat exchanger. When leaving the heat exchanger, the temperature of the natural gas has been reduced to around -155 C. The temperature is further lowered to around -163 C when pressure is lowered to near atmospheric. After compression, the mixed-refrigerant is cooled in a sea-water cooled condenser before it enters the flash drum. After that it is cooled with natural gas in main heat exchanger. The high pressure (~ 30 Bar) sub cooled refrigerant is throttled in a valve to produce a low temperature two-phase mixture which is vaporized in the main heat exchanger to cool natural gas and high pressure hot refrigerant. The refrigerant needs to be superheated (by 5-10 C) before it enters the compressor. Details of the process can be found out in [1].

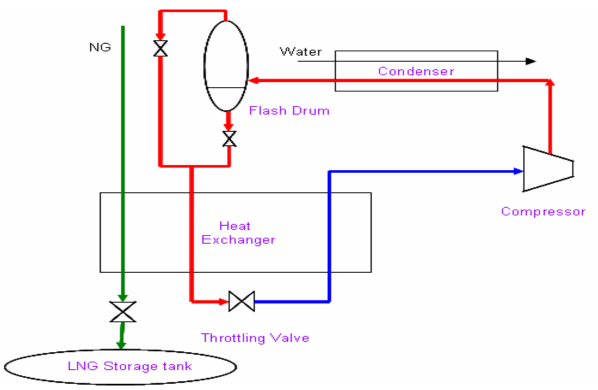

Fig. 1: PRICO Process

3. Modeling

In order to develop a complete dynamic model of the PRICO plant, it is essential to first build model for the individual components of the PRICO process, i.e., the heat exchanger, valve, compressor, flash drum and condenser. We have used standard model for these units from gPROMS model library except for the heat exchanger. Singh and Hovd has studied dynamic modeling of the PRICO process[2] and have suggested few model simplifications for the overall modeling of PRICO process such as neglecting flash drum and assuming a fixed refrigerant temperature at the condenser outlet, since it was found that these assumptions don't affect control structure design of the plant[3]. However, the heat exchanger model which has been used in above work [2] & [3] by Singh and Hovd, is based on certain assumptions and in this work we demonstrate that some of these assumptions don't affect control structure design for the PRICO process. We study the effect of following assumptions in the heat exchanger model on control structure design for the plant:

1) Assumption of negligible heat transfer through conduction along the longitudinal direction of the metal wall separating the cold and hot streams [4]
2) Assumption that all three streams in the PRICO Heat exchanger exchange heat through a common wall
3) Assumption of conserving enthalpy instead of internal energy

Following are the governing equations for energy balance for each stream and wall:

Energy balance for metal wall separating natural gas and cold refrigerant stream:

$$\rho C_p \frac{\partial T_w(x)}{\partial t} = k(\frac{\partial^2 T_w(x)}{\partial x^2}) - \frac{U_c(T_w(x)-T_c(x))}{t} - \frac{2U_h(T_w(x)-T_h(x))}{t} \qquad x=(0:L) \qquad (1)$$

Energy balance for metal wall separating cold and hot refrigerant stream:

$$\rho C_p \frac{\partial T_w(x)}{\partial t} = k(\frac{\partial^2 T_{w1}(x)}{\partial x^2}) - \frac{U_c(T_{w1}(x)-T_c(x))}{t} - \frac{2U_{h1}(T_{w1}(x)-T_{h1}(x))}{t} \qquad x=(0:L) \qquad (2)$$

Energy balance for natural gas stream:

$$\frac{-\partial E_h(x)}{\partial x} + 2U_h * w * (T_w(x)-T_h(x)) = \frac{du_{v,h}(x)}{dt} * A_{f,h} \qquad x=(0:L] \qquad (3)$$

Energy balance for hot refrigerant stream:

$$\frac{-\partial E_{h1}(x)}{\partial x} + 2U_{h1} * w * (T_{w1}(x)-T_{h1}(x)) = \frac{du_{v,h1}(x)}{dt} * A_{f,h1} \qquad x=(0:L] \qquad (4)$$

Energy balance for cold stream:

$$\frac{-\partial E_c(x)}{\partial x} + U_c * w * (T_w(x) - T_c(x)) + U_c * w * (T_{w1}(x) - T_c(x)) = \frac{du_{v,c}(x)}{dt} * A_{f,c} \qquad x = [0:L] \tag{5}$$

Boundary Conditions:

$$\frac{\partial T_w(0)}{\partial x} = 0 \quad AND \quad \frac{\partial T_w(L)}{\partial x} = 0$$
$$\frac{\partial T_{w1}(0)}{\partial x} = 0 \quad AND \quad \frac{\partial T_{w1}(L)}{\partial x} = 0 \tag{6}$$

Table 1 explains the meaning of the symbols used above:

Table 1

Symbol	**Variable(typical magnitude)**
i	Subscript, *i= c, h and h1*(*c*- cold refrigerant, *h*- natural gas and *h1*- hot refrigerant)
ρ	Mass density of metal wall (8005 kg/m^3)
C_p	Specific Heat Capacity of metal wall (480 J/kg K)
x	Axial direction of metal wall
L	Heat exchanger Length(4~5 m)
$T_w(x)$ *and* $T_{w1}(x)$	Metal wall temperatures
U_i	Heat Transfer coefficient for stream(~1000 W/m^2K)
$E_i(x)$	Energy flow rate
w	Width of heat exchanger
k	Thermal conductivity of metal wall(~16 W/m K)
$T_i(x)$	Stream temperature
$A_{f,i}$	Frontal flow area of stream
$u_{v,i}$	Volume specific enthalpy of stream
m_i	Mass flow rate of stream
t	Wall thickness(2~4 mm)

We refer to the work of Singh and Hovd [3] and replace the flash drum by assuming a fixed high side pressure and assume a fixed temperature for refrigerant at condenser outlet. With these simplifications we develop dynamic model for the PRICO plant by connecting sub model as per flow sheet shown in Fig.1. We study the behavior of this dynamic model for the below mentioned four cases which enlist the details of the heat exchanger model used in plant model:

Case 1) Base Case

- Heat transfer through conduction along longitudinal direction of metal wall is neglected

- Enthalpy is conserved
- All three streams exchanging heat are separated by walls.

Case 2) In this case we include heat transfer through conduction along longitudinal direction of metal wall and keep other assumptions same as in base case.

Case 3) In this case internal energy is conserved instead of enthalpy keeping other assumptions same as in base case.

Case 4) In this case we assume that all streams exchange heat via common wall and other assumptions remain same as in base case.

4. Control Structure Design

LNG temperature and Superheat at compressor suction are chosen as controlled variable. LNG temperature has been chosen based on quality concern of the product and superheat at compressor suction has been chosen based on safety concern of the compressor. Available manipulated variables are the compressor speed and the throttle valve opening. With these variables we linearize the model in the gPROMS for above mentioned cases and use Relative Gain Array (RGA) analysis [5] to select pairing of Manipulated and Controlled Variables. Also we investigate fundamental limitation on bandwidth for above cases.

5. Results and Discussion

RGA analysis:

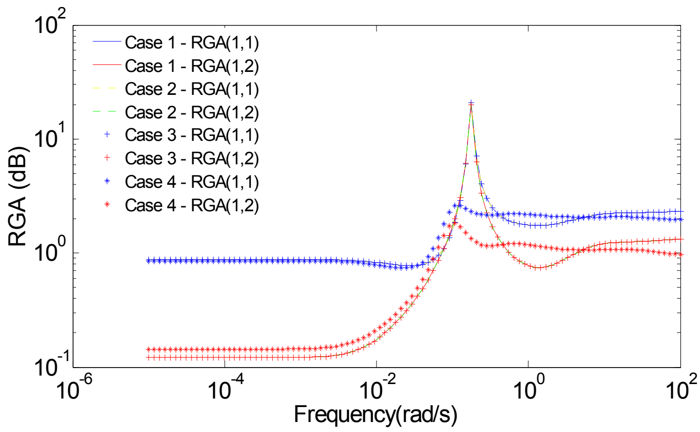

Fig 2: Frequency response of RGA elements (Magnitude)

Fig. 2 shows the variation of magnitude of RGA elements with frequency for all four cases. It is clear from this figure that for case 1,2 and 3, magnitudes of RGA elements are visible indistinguishable at low frequencies. Case 4 differs slightly from these cases; however, all cases give same pairing of manipulated and controlled variables. Steady state RGA value for diagonal pairing is 0.87 for case 1, 2 and 3 whereas it is 0.85 for case 4. At higher frequency there is some variation in the RGA for these cases but these frequencies are not significant for feedback control since delay due to neglected dynamics (assumed of order of 60 seconds) posed an upper limit on bandwidth which happens be lower than the frequency at which RGA plot shows a peak for all cases and differentiate among cases. Therefore, we can conclude that control structure design remains unaffected by these assumptions and model simplification of Heat exchanger model. Also for all cases, it has been found out that for every case there is only one right half plane (RHP) zero and no RHP pole. But this RHP zero in all cases, is very far into right (~e8), hence this is not relevant from bandwidth limitation point of view.

6. Conclusion

Through this work we have demonstrated that neglecting heat transfer through conduction along longitudinal direction of metal wall doesn't not affect control structure design and fundamental limitation on bandwidth for PRICO process. Similarly it has been demonstrated that assuming enthalpy as a conserved property rather than internal energy also doesn't affect control structure design and fundamental limitation on bandwidth for PRICO process. Also simplifying model by assuming that all streams exchanges heat via a common wall does not change control structure design and fundamental limitation on bandwidth for PRICO plant.

Acknowledgement

We would like to thank Prof. Costas Pantelides at Process System Enterprise, London, UK and John C. Morud, SINTEF, Trondheim, Norway for their useful inputs on modeling. We greatly appreciate Norwegian University of Science and Technology, Trondheim for funding this work.

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