Dynamic Modeling and Control of the PRICO© LNG process

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1. INTRODUCTION

For transportation of natural gas (NG), pipeline transportation is often used. However, when gas volumes are moderate, and/or transportation distances are large, the capital and operating costs for pipeline transport become prohibitive. In such cases, transport of Liquefied Natural Gas (LNG) in tankers is often the preferred choice for bringing the gas to the market. In the liquefaction process the natural gas is cooled to around -160°C, and this requires significant amounts of energy. It is therefore important that the process can be operated safely, reliably and efficiently. To achieve this, good control is required.

To understand LNG plant dynamics and to design a robust control system for its operation requires a dynamic model of the plant under consideration. Often the liquefaction unit of the plant is the critical unit which requires maximum attention. To develop a dynamic model for a liquefaction unit of an LNG plant is a challenging task and requires time and effort. The academic work on dynamic LNG plant simulation is limited (Hammer, 2004; Zaim, 2002; Melaaen, 1994). A significant part of these works focuses on modeling of a specific LNG plant. Development in process modeling tools such as Process Systems Enterprise's gPROMS has made it easier to develop a dynamic model of typical chemical plants such as an LNG plant. This makes it easy to devote significant time to study control aspects in details of such plants. We aim to do so in our work.

The process considered in this work is a single mixed refrigerant process known as PRICO (poly Refrigerant Integrated Cycle Operations) process. (Stebbing and O'Brien, 1975). The PRICO process has been studied from optimization perspective in several publications. (Zaim, 2004, Lee et al., Del Nogal et al., 2005, Jensen and Skogestad, 2006). These works deal~~s~~ with steady state optimization and there is no literature available on dynamic modeling and control structure design for PRICO process. The focus of current paper is to use the model developed for PRICO process (Singh and Hovd, 2006) for control structure and controller design for the PRICO process. In addition to enabling the use of model based tools for control structure design, this allows testing the effects of common model simplifications, such as assuming constant temperature of the refrigerant at the condenser outlet, or ignoring the flash drum and refrigerant holdup. The effects of these model simplifications for model based control structure development and controller tuning are described in the present work.

2. PROCESS DESCRIPTON

Fig 1 shows the flow sheet of the liquefaction unit of the PRICO process. Some features of the process are removed to make it simple.

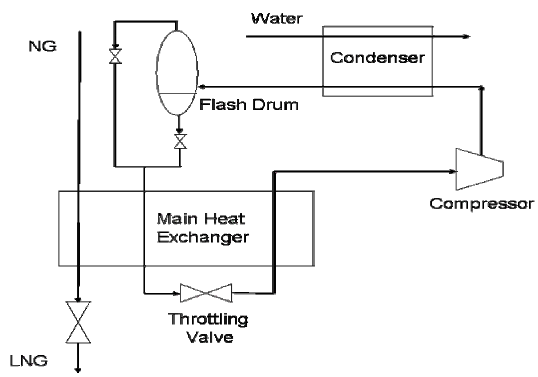

Fig. 1: Flow sheet of liquefaction unit of PRICO process

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 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 sea water cooled condenser before it enters the flash drum. After that it is further cooled in the 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 the natural gas and high pressure hot refrigerant. The refrigerant needs to be superheated (by 5-10 C) before it enters the compressor to avoid damage to the compressor.

3. MODELING

A detailed dynamic model for the plant is developed in gPROMS using Multi-flash for calculation of physical properties for the natural gas and the refrigerant (Singh and Hovd, 2006). The SRK equation of state is used for both refrigerant and natural gas. As evident from Fig. 1, first it is essential to develop model for the main components in the plant flow sheet, namely the heat exchanger, valve, compressor, condenser and flash drum. The model of the heat exchanger and condenser are based on the same principles, the only difference being that in main heat exchanger there is heat exchange between three streams whereas in condenser only two streams exchange heat. Valves are modeled as isenthalpic processes. A brief description of models is given below:

3.1 Main heat exchanger

A one dimensional distributed dynamic mathematical model for a heat exchanger having heat exchange between the three streams is developed using enthalpy and mass balances. Pressure drop in the heat exchanger is neglected. The composition of each stream is assumed to be constant from inlet to outlet. A constant heat transfer coefficient is assumed for each stream. All streams are assumed to exchange heat through one metal wall. The metal wall separating the streams is assumed to have negligible thermal conduction in the axial direction and infinitely fast thermal conduction in the radial direction. A separate energy balance is used for the internal energy of the metal wall. Wall ends are assumed to be adiabatic. The models for both the streams and the wall are one-dimensional.

3.2 Compressor

This model describes the relation between gas mass flow rate and pressure head across the compressor. In this model, infinitely fast dynamics is assumed within the compressor. Negligible hold up and inertia of refrigerant is considered in the compressor. Fan Laws (affinity laws) are used to model speed dependent variations in performance, so that single characteristic curve (head vs. flow) is enough to describe behavior at any speed. The compression process is modeled as polytropic. Constant efficiency is assumed for compressor i.e. efficiency is not assumed to vary with flow rate.

3.3 Flash Drum

It is assumed that the liquid and vapor are at equilibrium at all times and thus there is perfect contact between the vapor and liquid phases. Also it is assumed that there is negligible entrainment of liquid in the vapor stream. The model accounts for the mass balance of each component. The overall energy balance ensures that internal energy is conserved. No heat addition or subtraction is included in the flash calculations, thus resulting in an adiabatic UV flash calculation. This is a standard model from GPROMS model Library.

3.4 Throttling Valve

This valve is assumed to be isenthalpic. Also this model has an equation relating mass flow rate of refrigerant to the valve opening and the pressure difference across valve.

3.5 Mixer

The mixer model mixes the liquid and vapor refrigerant streams coming out of the flash drum before it enters the main heat exchanger. The model uses component mass balance and energy (enthalpy) balance. Pressure of outlet stream from mixer is treated as equal to the average pressure of streams to mixer.

4. CONTROL STUDIES

It is essential to operate plant safely while making sure to produce LNG at desired specifications. Energy optimality of operation is certainly worth studying but we have chosen safety of plant and quality of LNG as main theme for this study. One of the critical aspects for the safety of plant is compressor operation. It is required that refrigerant enters compressor in superheated state as liquid refrigerant would damages the compressor. This makes superheat of refrigerant at compressor suction an obvious choice for controlled variable from safety perspective. Second choice of controlled variable is LNG temperature since we want to make sure plant cools down natural gas to its liquid form to make its transportation possible by ships. Choice for manipulated variables is limited in the plant and we have chosen manipulated variables based on their proximity with controlled variables.

A linear model with these inputs and outputs was obtained using gPROMS. The linear model has 140 states. Control studies on this plant was done in MATLAB™

4.1 Poles and Zeros

It wad observed that model has five poles and five zeros at origin and there was one RHP zero very far into the right half plane. The reason for these five poles and zeros at the origin is that the mass holdup states in the flash drum are constant, thus making these states uncontrollable. The number of poles and zeros matches the number of components in the refrigerant. The locations of these states were found in the state vector and these states and their effect on measurement and other states were removed from model (i.e. the A, B and C matrices of the linearized model were modified accordingly). Comparing step responses of this new system with those of the system having holdup states was compared and it was found that they are identical. This proves that these hold up states do not contribute in input/output behavior of the system.

4.2 Bandwidth Limitations

There is no fundamental limitation on bandwidth in the model, since the remaining RHP zero is very far into the RHP and there is no delay in the system. However, to account for unavoidable un-modeled effects, which are more prevalent at high frequencies, we include a possible delay of up to 60 s. This puts an upper limit on bandwidth of system, which would be:

$\omega_b < .016 \mathrm{rad/s}$

4. 3 Scaling

All the units are SI units. Model has been scaled before carrying out control studies.

Number	Manipulated Variable	Units	Controlled Variables	Units
1	Compressor Speed	*Rad/sec*	LNG Temperature	*K*
2	Throttle valve opening	*%*	Superheat at Compressor Suction	ΔK

Following are the scaling matrices used for inputs and controlled variables:

$$D_y = \begin{bmatrix} 2 & 0 \\ 0 & 2 \end{bmatrix}$$

$$D_u = \begin{bmatrix} 15 & 0 \\ 0 & 5 \end{bmatrix}$$

New System would be as follow:

$$\hat{A} = A$$

$$\hat{B} = BD_u$$

$$\hat{C} = D_y^{-1}C$$

$$\hat{D} = D_y^{-1}DD_u$$

4.4 RGA Analysis

RGA analysis indicates using input 1 to control output 1 and using input 2 to control output 2. (Diagonal pairing). Fig 1 shows magnitude of RGA elements.

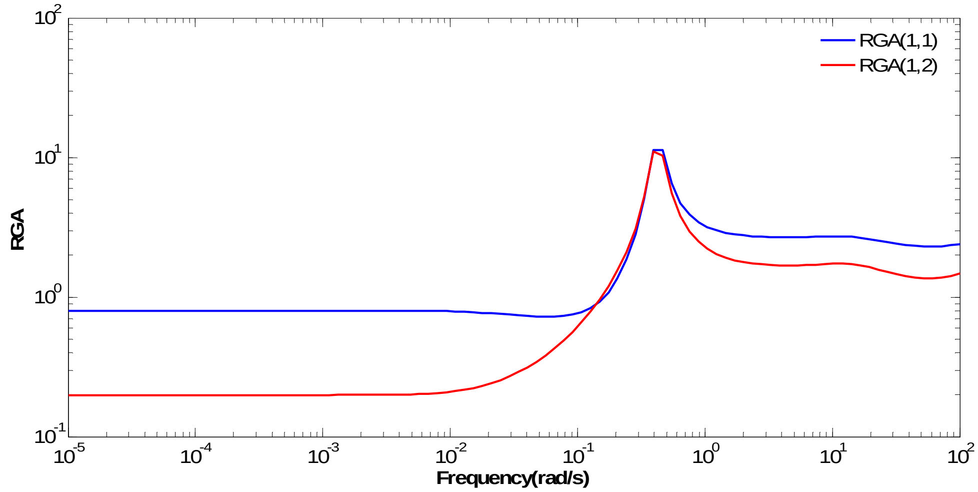

Fig 1: Magnitude of RGA elements

The steady state RGA for the diagonal pairing is 0.80. Figure 1 further indicates modest interactions at low frequencies with the diagonal pairing. At higher frequencies there are more severe interactions, but this is beyond the achievable bandwidth when we account for un-modeled effects.

The pairing based on RGA is contrary to the common practice of pairing variables based on their proximity, which would have led to controlling LNG temperature using throttle valve opening and controlling the superheat using compressor speed.

4.5 Model Reduction

The original linear model was of 140 states which became 135 state models after we removed 5 hold up states. This high number of states is not very practical for controller design, and therefore model reduction is used. Through some trial and error, it was found that most states contribute to input-output behavior mainly at high frequencies, where even the high order model may be inaccurate. The number of states was therefore reduced to only 2 using balanced truncation method. Figure 2 compares the frequency responses of the high order and low order models.

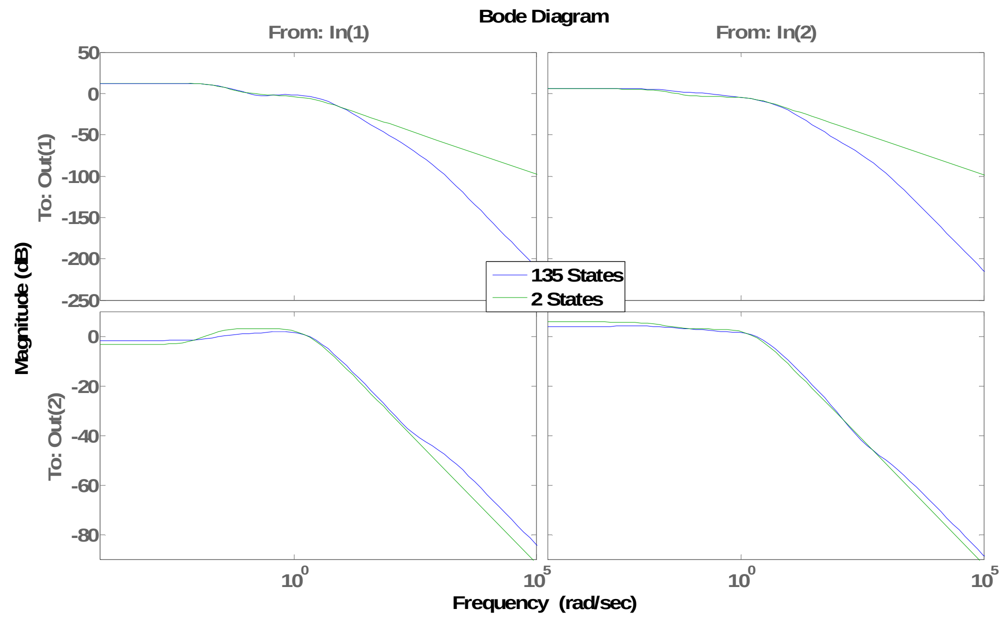

Fig.2: Comparison of 135 states model with 2 states model

5 CONTROLLER DESIGN

SIMC rules (Skogestad, 2003) have been used for controller tuning. Half rule (Skogestad, 2003) has been used to convert 'two state' model to a first order model. Using these rules following are the tuning parameters:

- For PI Controller which controls LNG Temperature using compressor speed

$$K_c = -1e-3 \qquad \tau = .54 \tag{1}$$

- For PI Controller which controls Superheat at Compressor Suction using valve opening

$$K_c = -2.25e-3 \qquad \tau = .54 \tag{2}$$

5.1 Performance of controller designed above

A disturbance of 5 C in the inlet temperature of the Natural Gas is introduced (at 3000 s) and the performance of controllers is observed.

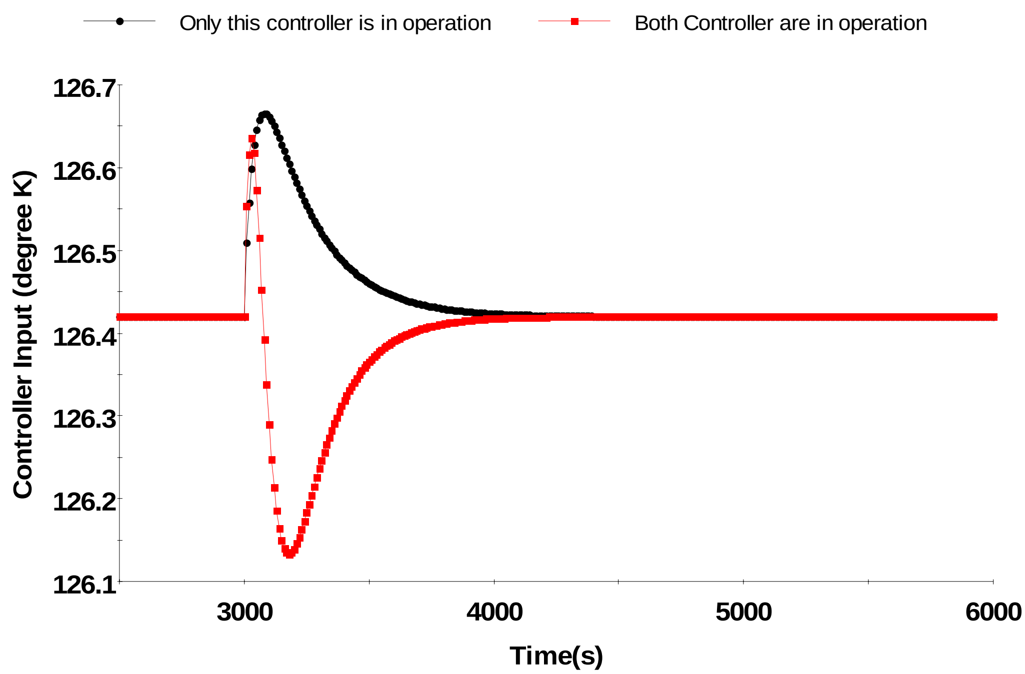

Fig.3: Controller 1 Input (Controlling LNG Temperature by Compressor Speed)

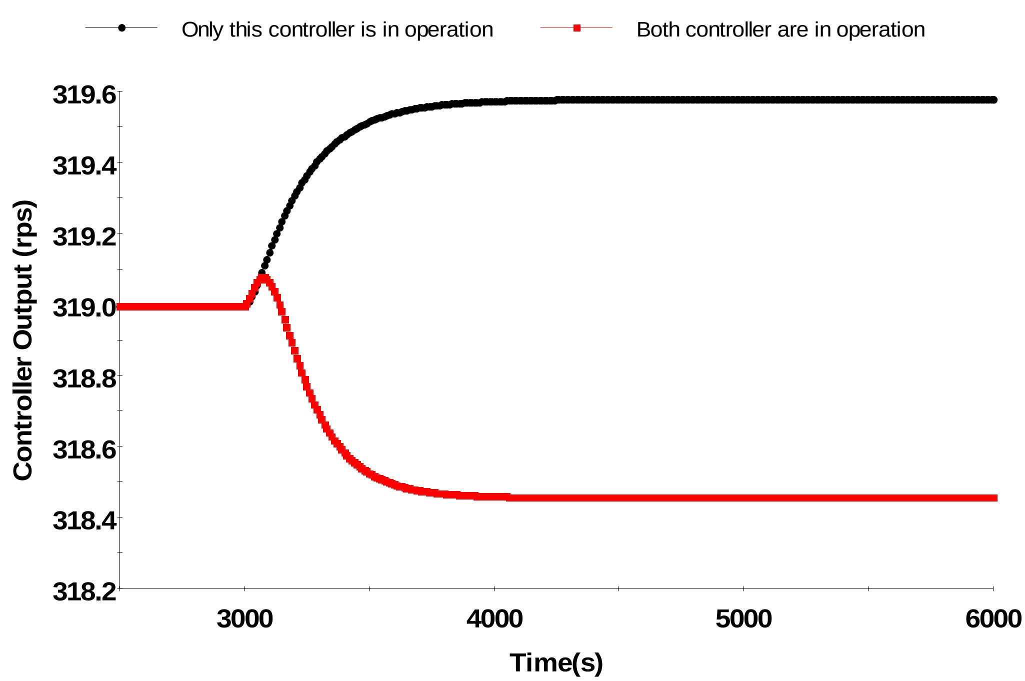

Fig.4: Controller 1 Output (Controlling LNG Temperature by Compressor Speed)

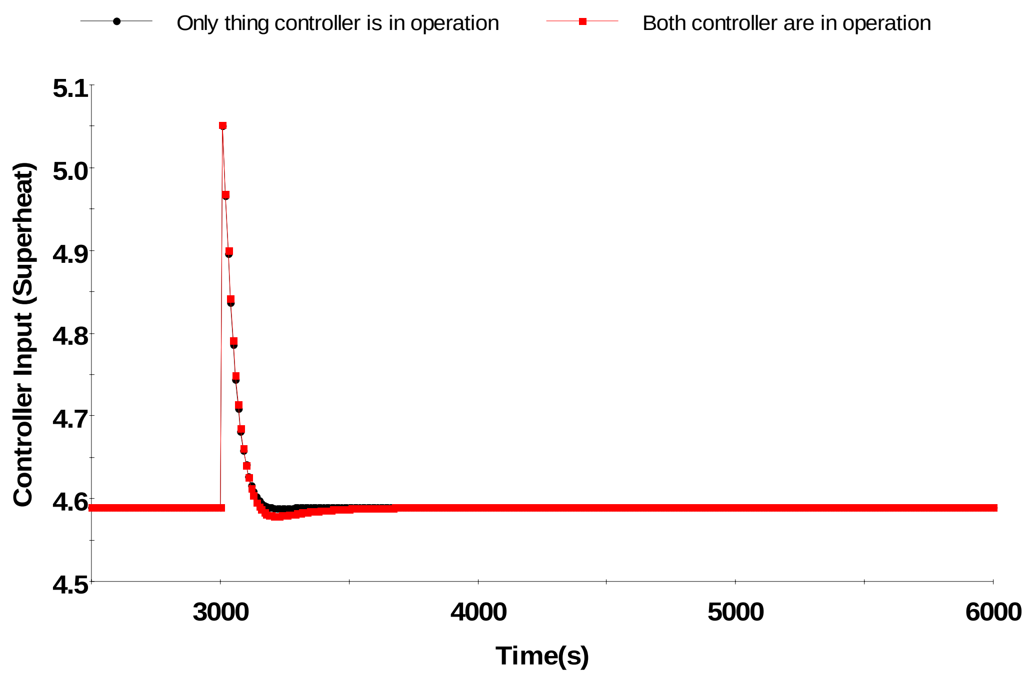

Fig.5: Controller 2 Input (Controlling Superheat at compressor suction by throttling valve opening)

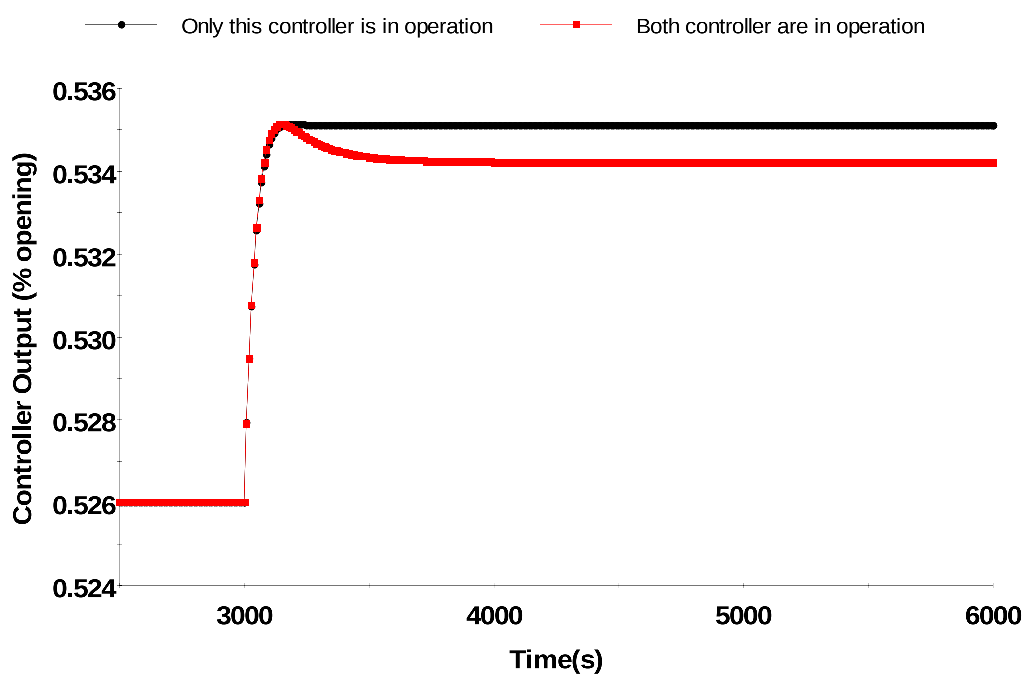

Fig.6: Controller 2 Output (Controlling Superheat at compressor suction by throttling valve opening)

5.4 PI Controller design for opposite pairing of input/output

We also design the PI controller for the opposite pairing of inputs and outputs, which is to control the LNG temperature using the throttle valve opening and to control the superheat at the compressor suction using the compressor speed.

Now using half and SIMC rules, tuning parameters for these PI Controllers would be:

Controller for controlling LNG temperature by throttle valve opening

$$\begin{aligned} K_c &= -.165 \\ \tau &= 50 \end{aligned} \quad (3)$$

Controller for controlling Superheat at compressor suction by compressor speed

$$\begin{aligned} K_c &= 9.4e-4 \\ \tau &= 0.54 \end{aligned} \quad (4)$$

A disturbance of 5 C in inlet temperature of Natural Gas is introduced (at 3000 s) and performance of controllers is observed (in GPROMS).

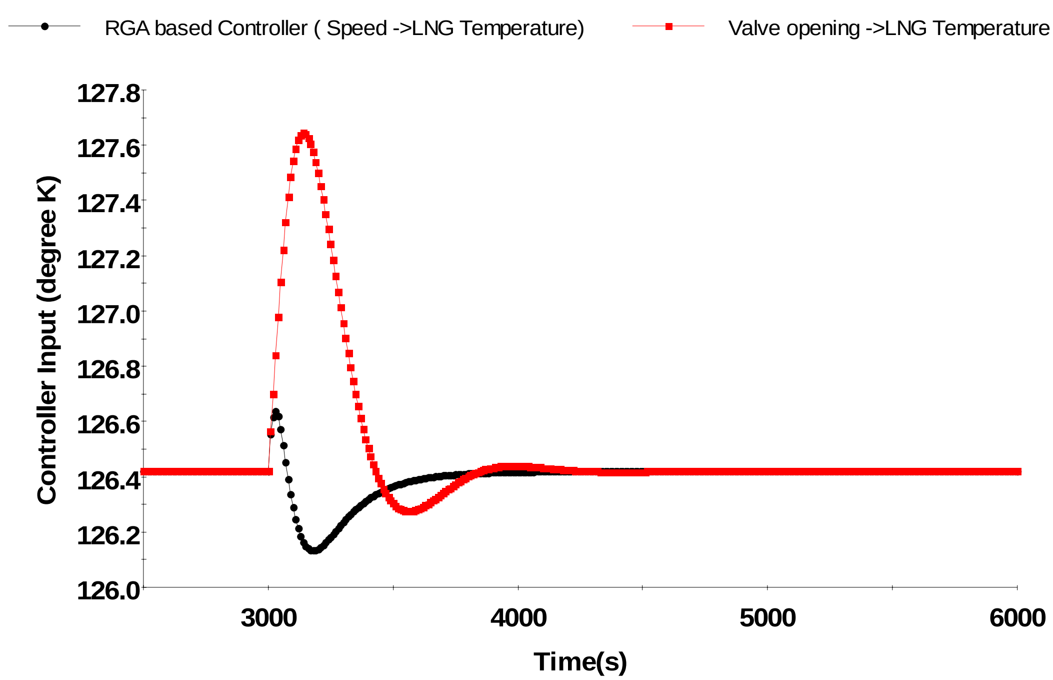

Fig. 7: Comparison of performance of controllers designed based on RGA based pairing (diagonal pairing) and off diagonal pairing to control LNG temperature

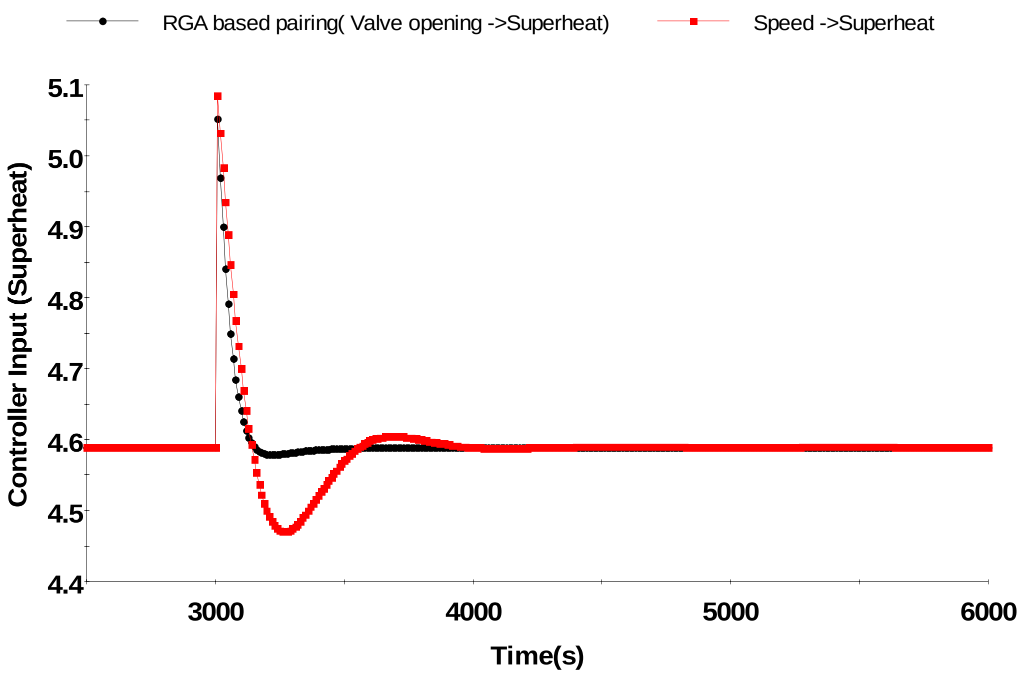

Fig. 8: Comparison of performance of controllers designed based on RGA based pairing (diagonal pairing) and off diagonal pairing to control Superheat at Compressor suction.

6 EFFECT OF MODEL SIMPLIFICATION ON CONTROL STRUCTURE DESIGN

We investigate effect of two model simplifications on control structure design. First is the assumption of constant temperature of the refrigerant at the condenser outlet. We carry out an RGA analysis for following pair of manipulated and controlled variables:

Manipulated variable	Controlled variable
Compressor Speed	LNG Temperature
Throttle valve opening	Superheat
Water mass flow rate in Condenser	Temperature of refrigerant at condenser outlet

RGA analysis indicates that water mass flow rate should be used to control temperature of refrigerant at condenser outlet, as may be expected from physical understanding. It is reasonable to require controlling the refrigerant temperature at the condenser outlet since there could be significant fluctuations in inlet temperature of the water in the condenser. But at the same time generally there is no limit on mass flow rate of water, so it is a reasonable assumption to assume a constant temperature for refrigerant at condenser outlet (Hammer, 2004).

Proceeding with a model which has a constant temperature of refrigerant at the condenser outlet and we find that the open loop plant behaves in the same way as it does when we don't have this temperature fixed, and RGA analysis gives the same input/output pairing as before. Steady state RGA's are quite close for the two cases. Thus we conclude that assumption of a constant temperature for the refrigerant at the condenser outlet is a valid assumption and doesn't change control structure design.

Second is the need for flash drum in PRICO cycle from modeling perspective. Certainly there is a need to have flash drum in the cycle since it acts as a hold up for refrigerant.However, from the modeling point of view, inclusion of a flash drum model makes the model more complex, and thus it is of interest to see how control structure design is affected when we don't have a flash drum model in the cycle and we assume a fixed high side pressure. With this new model we observe that still RGA analysis gives us same pairing for inputs/outputs as we had when flash drum was included in the model. This confirms that for control structure design it is reasonable assumption to omit the flash drum from the PRICO cycle model and instead assume a fixed high side pressure.

7. CONCLUSION

Controllability analysis shows that the pairing of the two main control loops in the PRICO process should be reversed, when compared with established industrial practice which is done on the basis of proximity of controlled and manipulated variables. That is, compressor speed should be used to control LNG temperature and throttle valve opening should be used to control the degree of superheat at the compressor suction. This conclusion is verified by controller tuning and dynamic simulation.

The effects of two model simplifications were studied and it was found out the control structure design is neither affected by omitting the flash drum from the model nor by assuming a constant temperature for the refrigerant at the condenser outlet.

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