ADVANCED CONTROL TECHNIQUE OF CENTRIFUGAL COMPRESSOR FOR COMPLEX GAS COMPRESSION PROCESSES

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ABSTRACT

Centrifugal compressors are widely used in petrochemical industries and natural gas fields. One of the major control problems associated with these compressors is pressure fluctuation. In recent years, various control methods have been designed and applied in the field to reduce pressure fluctuation and improve the overall operating stability.

This paper describes recent advanced complex gas compression control techniques used for centrifugal compressors, and a verification method using a newly developed control system and dynamic simulator.

One of the control techniques used for fuel gas compressors that supply fuel gas to gas turbine combined cycle (GTCC) power plants is described in this paper. Two compressors supply the fuel gas to two independent gas turbines through one common header. Each gas turbine is related to each individual compressor, which has to supply fuel gas for the individual gas turbine according to the load by modulating the control valve opening. The compressor also has to stabilize the common header pressure fluctuation by changing the gas turbine loads. For this process, one control technique is proposed with feed forward (FF) control for the load changing of gas turbine and feedback (FB) control for common header pressure stabilization, as well as application of the above technique in a plant operating in the field. With the FF and FB control, if load shedding of one gas turbine occurs and the gas turbine load suddenly goes down, the other gas turbine can continue operating safely.

The compressor control performance is verified with a compressor dynamic process simulator during the design and manufacturing stages prior to shipping. In particular, in order to ensure smooth startup at site, the control panels are connected to the process simulator and the above control technique is applied. Confirmation that all individual functions are working properly is made before shipping.

INTRODUCTION

In recent years, centrifugal compressors have been used for various gas compression processes. In those processes, a conventional control technique, i.e., the combination of master controller and antisurge controller, has proved very effective.

But other compressor types, like fuel gas compressors, require quick response and reliable controllability. For these compressors, there is a limit to how far the controllability can be tuned using conventional control techniques.

In this paper, an advanced control technique using feed forward control is proposed. This control technique places more emphasis on controllability. First, details of the control technique are introduced. Then the controllability of the control system is checked by studying the dynamic simulation results. After confirmation of results, the control system is applied in the field, where the controllability is verified by studying field test data.

FUEL GAS COMPRESSOR WITH COMMON HEADER

Figure 1 shows a typical integrally geared fuel gas compressor with inlet guide vane (IGV). This compressor has two compression stages, one gear wheel, and one pinion gear. Each gear is mounted on each shaft. The round arrows show the rotating direction of each shaft, and straight arrows show the gas flow direction.

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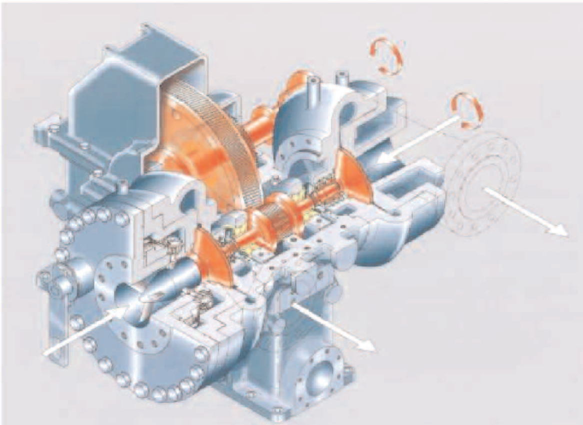


Figure 1. Schematic Diagram for Centrifugal Compressor.

Table 1 shows the fundamental features of the fuel gas compressor. The handling gas is natural gas from the well. The compressor has two stages, and is driven by a three-phase induction motor. The IGV controls the compressor flow. The seal type is a dry gas seal.

Table 1. Fundamental Feature of Compressor.

No.	Item	Specification
1	Application	Fuel Gas Compressor
2	Gas	Natural Gas (MW: 16.6 (design base))
3	Number of Units	3
4	Model No.	3V-2G
5	Feature	Integrally Geared with Inlet Guide Vane
6	Number of Stages	2
7	Shaft Power	1,340 kW
8	Driver	3-Phase Induction Motor 6,600V, 50 Hz, 1500 kW
9	Speed	3,000 rpm (LSS) 14,926 rpm (HSS)
10	Seal Type	Dry Gas Seal

Figure 2 shows the compressor arrangement. The main motor drives the wheel gear. The gear train, consisting of the wheel gear and the pinion gear drives the two-stage compressors. The main motor also drives the lube oil pump mounted on another pinion shaft. The gas enters into the first stage of the compressor through the IGV, is compressed in the first and second stages, and then the discharge gas goes out to the outlet section.

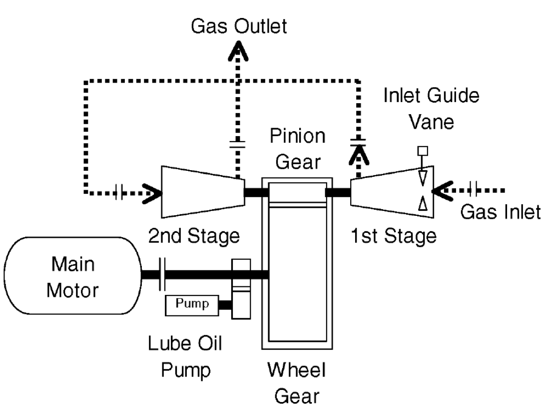


Figure 2. Compressor Arrangement.

Individual fuel gas compressors are usually dedicated to each gas turbine in a GTCC plant so that both can work as a pair. Hence, the control philosophy and the operating sequence are simple. When changing the gas turbine is required, the compressor operation can simply be changed simultaneously by reading the header pressure of the related gas turbines.

However, when the fuel gas is provided to multiple gas turbines through a common header, the compressor control system will be different. Figure 3 shows an example of a fuel gas compressor arrangement for a GTCC plant, with two gas turbines and three fuel gas compressors connected by a common header. The fuel gas compressor flow capacity is the same as that of the gas turbine. One of the fuel gas compressors is a standby unit. When both gas turbines are under operation fed by fuel gas through the common header, a sudden change in operation by one gas turbine will cause a pressure fluctuation in the common header. The above pressure fluctuation will affect the operating performance of the other gas turbines.

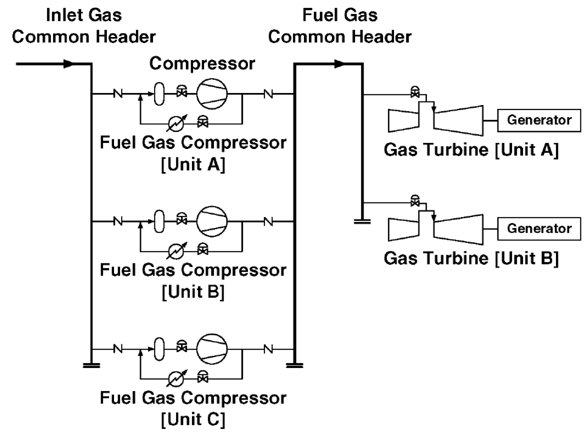


Figure 3. Fuel Gas Compressor Units Arrangement for GTCC Plant Using Common Header.

In this condition, there is a limit to using the conventional control method (like the master controller method for controlling the common header pressure in order to get effective controllability of the common header pressure fluctuation).

COMPRESSOR ADVANCED FLOW CONTROL

Figure 4 shows an example of a performance curve of a compressor driven by a constant speed motor with the individual inlet guide vane opened. The horizontal axis shows the flow rate, and the vertical axis shows the pressure ratio. The IGV opens from 20 percent to 100 percent to supply the required flow in this case. When the IGV position is 20 percent opened (at minimum opening point), the antisurge valve (ASV) will open to reduce the flow of the gas turbine. The pressure ratio of the fuel gas compressor is almost constant in this operating condition. Also control valves control the common header pressure, so the compressor operating point shifts toward the dotted line of the performance curve.

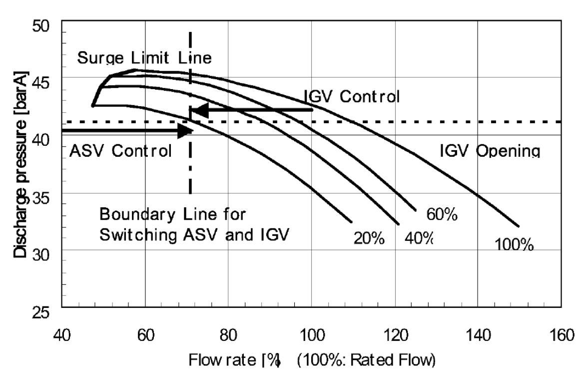
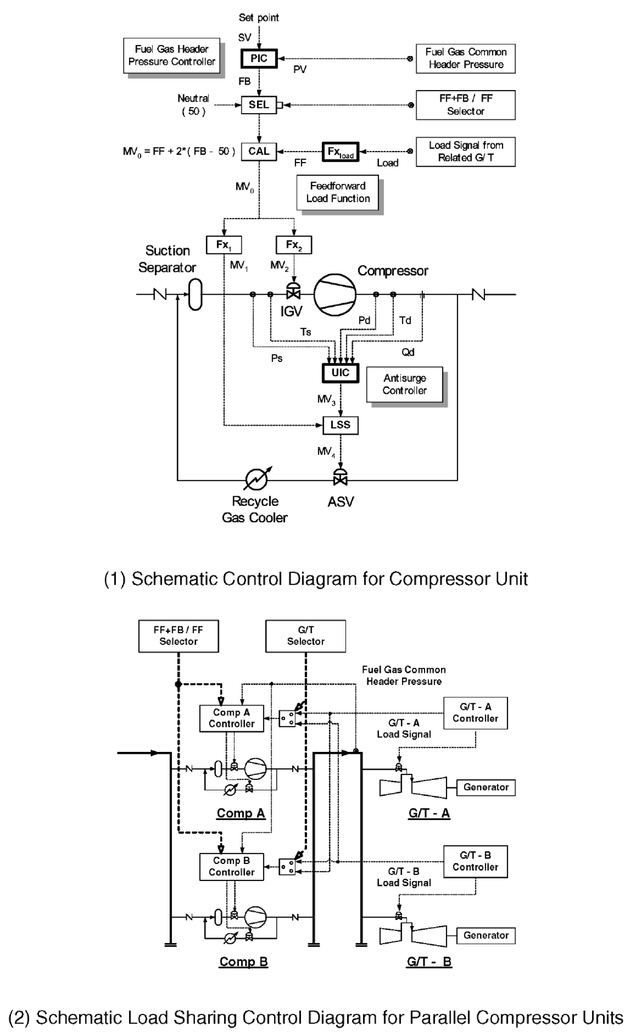


Figure 4. Compressor Performance Curve and ASV/IGV Control Band.

On the performance curve, the IGV and ASV control bands are divided into two by a boundary line, formed by the cross section point of the pressure ratio line (dotted line) and IGV minimum opening line (dotted/dashed line) for ASV control. One band is on the left side of the boundary line and the other band is on the right side.

Therefore, at first, the load signals from the related gas turbines are entered in the function generator called "feedforward load function."

Figure 5 shows the above-proposed control system. Figure 5 (1) shows the schematic control diagram for the compressor unit, and (2) shows the load sharing control diagram.



(1) Schematic Control Diagram for Compressor Unit

(2) Schematic Load Sharing Control Diagram for Parallel Compressor Units

Figure 5. Compressor Units' Load Sharing Control Method by FF+FB/FF Selector.

This control system has two operating modes: FF+FB mode and FF mode. The compressor control panel or distributed control systems (DCS) selects these modes with a selector switch (SEL) mounted on the master control panel or DCS. Then the gas turbine selector (SEL) selects the gas turbine and the compressor unit. The output of SEL gives a FB value for FF+FB mode, and a neutral value (50) for FF mode. The FF+FB mode is controlled by the pressure control value (FB) and feedforward load function (Fx_{load}) output value (FF), whereas the FF mode is controlled directly by FF value. This control system avoids any interference between the compressor units. The manipulating value (MV_0) is calculated (Equation [1] below) in the calculation unit (CAL) as shown in Figure 5 (1) using FB value and FF value.

$$MV_0 = FF + 2 \times (FB - 50) \quad \textbf{(1)}$$

MV_0 is limited to the range of zero percent to 100 percent. If FF value deviates from the actual and design conditions, FB values can cover the full range of FF value according to this formula. The IGV and ASV positions are decided using the split range function generator Fx_1 and Fx_2.

Figure 6 shows an example of feedforward load function. The horizontal axis shows the load signal of the fuel gas flow of the assigned gas turbine. The vertical axis shows the FF signal as a percentage.

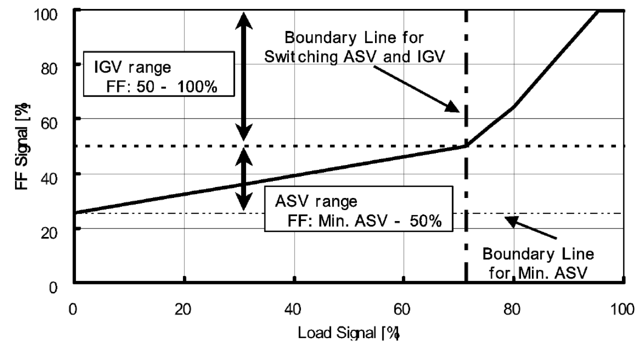
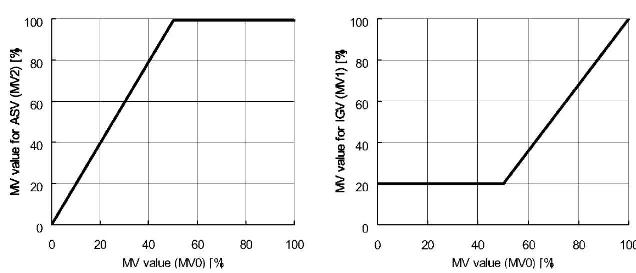


Figure 6. Example of Feedforward Load Function (Fx_{load}).

There are two boundary lines in Figure 6. One is the boundary line (fine double-dotted/dashed line) for the minimum ASV position, and the other is the boundary line (thick dotted/dashed line) for opening/closing the ASV and IGV as shown in Figure 4.

In Figure 6, zero percent load signal means that the gas turbine stops. When the G/T controller gives zero percent load signal, the compressor is in its full recycle operating condition, and the discharge pressure of the compressor should be the same as the set value of the header pressure. The reason for this is that if the gas turbine starts, the compressor has to supply fuel gas to the gas turbine as soon as possible. Therefore, the boundary line for the minimum ASV is used as a limit line to keep the discharge pressure of the compressor above the set value of the header pressure.

Figure 7 shows an example of the split range function for the ASV (Fx_1) and IGV (Fx_2). The horizontal axis shows the output of the calculation (CAL) unit (MV_0) and the vertical axis shows the positions of the ASV (MV_1) and IGV (MV_2).

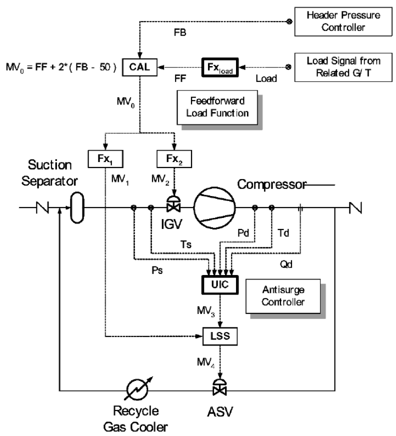


(1) ASV Function (Fx_1) (2) IGV Function (Fx_2)

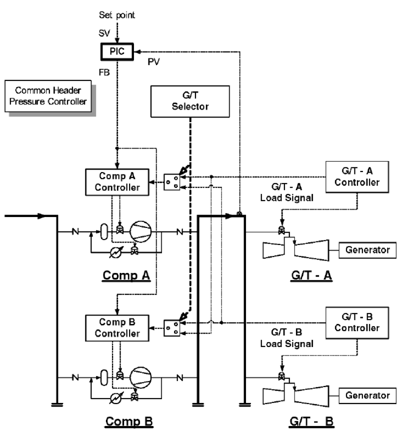
Figure 7. Example of Split Range Function for ASV and IGV.

Thus, using the combination of the feedforward load function, ASV function (Fx_1), and IGV function (Fx_2), the ASV controls the low load from zero percent to the boundary flow, by adjusting the FF signal from minimum ASV position to 50 percent open, and the IGV controls the high load from the boundary flow to maximum flow by adjusting the FF signal from 50 percent to 100 percent.

Figure 8 shows an example of the compressor's load sharing control system by a master controller for the common header pressure. This alternate control system can be used instead of the above-mentioned system as shown in Figure 5. The difference between the two is that the present control system uses the master controller for the common header pressure instead of the FF+FB/FF selector shown in Figure 8 (2). In this way, the FB value is calculated by the master controller, and is used in the CAL unit of all compressor units shown in Figure 8 (1).



(1) Schematic Control Diagram for Compressor Unit



(2) Schematic Load Sharing Control Diagram for Parallel Compressor Units

Figure 8. Compressor Units' Load Sharing Control Method by Master Controller for Common Header Pressure.

These load sharing control systems are different from common load sharing control system. These load sharing systems concentrate on the controllability rather than the efficiency of the compressors. Fundamentally, the required gas turbine fuel flow modulates the control valves directly using feedforward load function with this control system. The FB value only adjusts the common header pressure to the set point.

SIMULATION

In order to verify the proposed FF+FB/FF selector control system, a dynamic simulation is performed. Some examples of the verification simulation are shown below in G/T parallel operating condition.

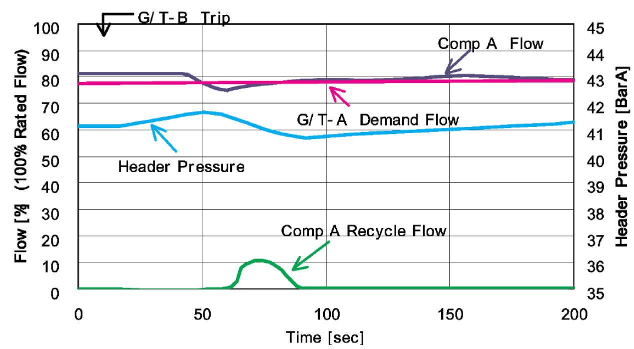
Case 1: G/T B Trip

- Before
  - G/T A at 75 percent load relates to compressor (comp) A at 75 percent load in FF+FB mode
  - G/T B at 55 percent load relates to comp B at 55 percent load in FF mode
- After
  - G/T A keeps the load close to the load condition before G/T B Trip
  - G/T B load suddenly goes down to zero percent at trip and at the same time comp B stops.

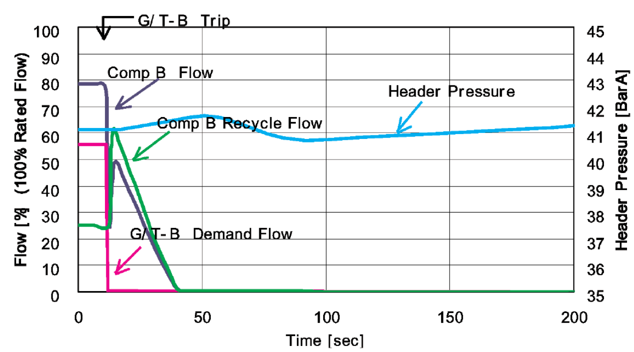
Case 2: G/T A and B Load Shedding at the Same Time

- Before
  - G/T A at 100 percent load relates to comp A in FF+FB mode
  - G/T B at 100 percent load relates to comp B in FF mode
- After
  - G/T A and B load suddenly goes to 30 percent (minimum load) during load shedding

Figure 9 shows the simulation result of Case 1. These graphs show trend data of the G/T demand flow, header pressure, compressor flow, and recycle flow. In this simulation, G/T B trip occurs 10 seconds after simulation start. After G/T B trip, A takes care of the G/T load and adjusts the header pressure with the FF+FB function. Comp B stops and decreases the compressor flow by opening the ASV and coasting down.



(1) Comp A Trend Data

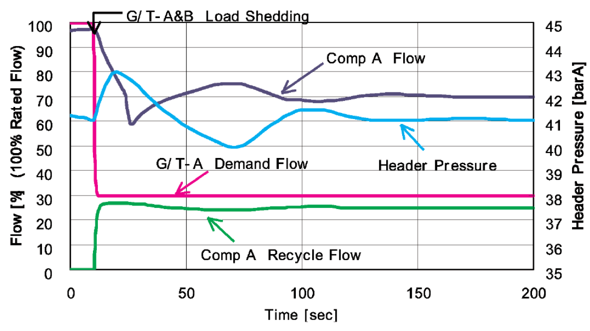


(2) Comp B Trend Data

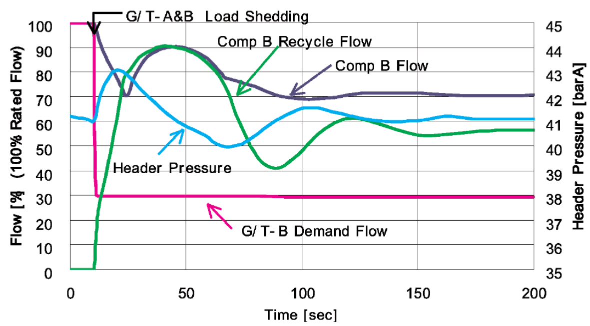
Figure 9. Simulation Result for Case 1 (G/T B Trip).

From the result of this G/T B trip simulation, it is found that the common header pressure fluctuation is within a small range (+0.66/−0.29 bar [+9.57/−4.21 psi] from SV), and the pressure became stable in three minutes.

Figure 10 shows the simulation result of Case 2. In this simulation, G/T A and B load shedding occurs 10 seconds after simulation start. Comp A and B load goes down to 30 percent with feed forward load function following G/T load, and MV_0 becomes small. The ASV opens to increase the flow, so comp A and B recycle flow increases and comp A adjusts the header pressure with the FF+FB function.



(1) Comp A Trend Data



(2) Comp B Trend Data

Figure 10. Simulation Result for Case 2 (G/T A and B Load Shedding at the Same Time).

From the result of this G/T A and B load shedding simulation, the common header pressure fluctuation is found to be within 3 bar (43.5 psi) (+1.86/−1.10 bar [+26.98/−15.95 psi] from SV), and the pressure became stable in two minutes in spite of the large gas turbine (G/T) load change from 100 percent to 30 percent in a relatively short time. Table 2 shows a summary of these simulation results.

Table 2. Simulation Result.

Case	G/T Condition		Load (%) G/T	Load (%) Comp.	Header Press. Fluctuation (Bar)
1	A	Run	75	75	+ 0.66 - 0.29
	B	Trip	55→0	55→0	
2	A	L/S	100→30	100→30	+ 1.86 - 1.10
	B	L/S	100→30	100→30	

VERIFICATION OF CONTROL SYSTEM USING DYNAMIC SIMULATOR

The FF+FB/FF selection control system is applied to the operating compressor units. The control system is installed in the programmable logic controller (PLC). Before this control system is applied to the actual compressor, the verification test using the dynamic simulator is performed in order to verify the control function and sequence during the factory acceptance test (FAT).

Figure 11 shows the system configuration of the test condition. The control system is programmed using a PLC program loader, and is downloaded to the PLC located in the control panel. The operating condition can be confirmed in the operating panel. The simulation model of the gas turbines, compressors, and other auxiliaries is installed in the process simulator that is the same as the dynamic simulator used in the previous "SIMULATION" section, and run in real time. The switching panel initiates the start permissive condition, trip cause, etc. The PLC, process simulator, and switch panel are interconnected using hard wire cables.

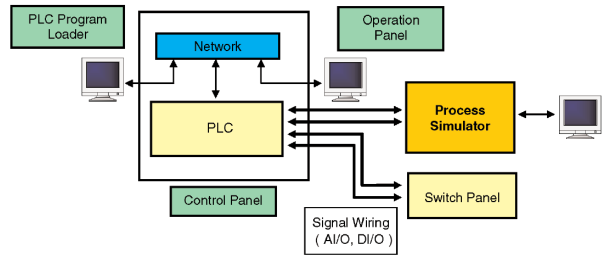


Figure 11. Schematic System Configuration Diagram of Simulation Test Using Dynamic Simulator.

Figure 12 shows the simulation test condition using the dynamic simulator to verify the PLC program. In this verification test, the detail sequence and actual control functions can be verified. Table 3 shows examples of the test item using the dynamic simulator.

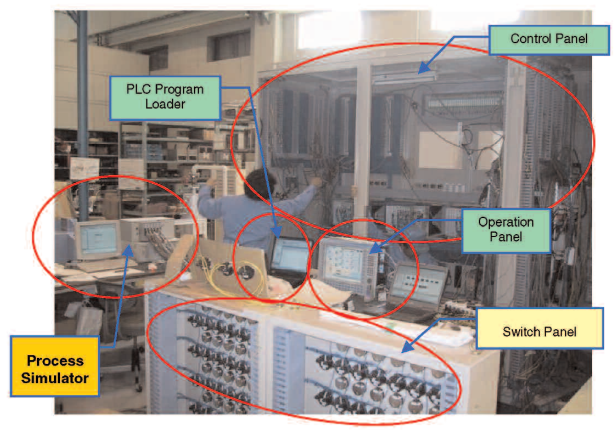


Figure 12. Simulation Test Condition Using Dynamic Simulator to Verify PLC Program.

Table 3. Examples of Simulation Test Item.

No.	Item	Confirmation
1	Start-up Primary (FF+FB) Unit	- Start-up sequence
2	Start-up Secondary (FF) Unit	- Start-up sequence
3	Comp. Unit Switch Over FF+FB and FF	- Switch over sequence - Header Pressure fluctuation
4	Comp. Unit Stop	- Switch over sequence when FF+FB Unit stop - G/T run back - Header Pressure fluctuation
5	Comp. Unit Trip	- Switch over sequence when FF+FB Unit stop - Header Pressure fluctuation
6	G/T Load Shedding	- Load Shedding sequence - Header Pressure fluctuation
7	G/T Trip	- G/T Trip sequence - Header Pressure fluctuation
8	Antisurge Control	- Antisurge controller function
9	Inlet Temp. Change	- Header Pressure control performance by G/T load swing
10	Inlet Press. Change	- Header Pressure control performance by G/T load swing

An example of the run back test shown in Table 3, Item No. 5, is described as follows.

Case 3: G/T - A and B Run Back by Comp Trip

- Before
  - G/T A at 75 percent load relates to comp A at 75 percent load in FF+FB mode
  - G/T B at 75 percent load relates to comp B at 75 percent load in FF mode

- After
  - G/T A and B load suddenly goes down to 30 percent (minimum load) by run back when comp A trips
  - Compressor B supplies the total fuel to gas turbine A and B

Figure 13 shows the result of the run back test. This graph shows trend data of the G/T A and B demand flow, header pressure, and compressor A and B flow. In this simulation test, comp A trip occurs 60 seconds after simulation test start. After comp A trips, comp A flow goes down quickly. G/T A and B load goes down to 30 percent load. Comp B flow goes down at once, then goes up to recover the decreasing common header pressure.

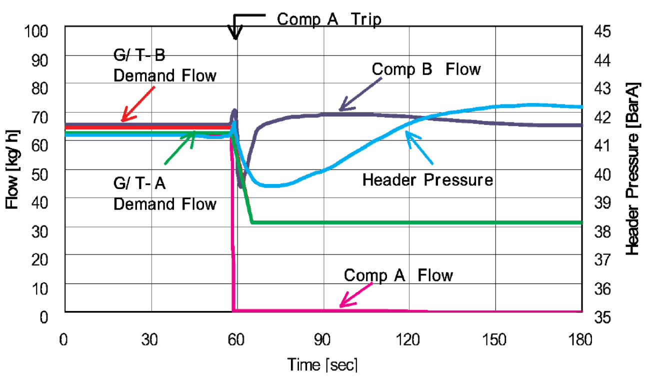


Figure 13. Example of Simulation Test Result (Run Back Test).

From this result of the simulation test, the common header pressure fluctuation is found to be within 3 bar (43.5 psi) (+1.06/−1.80 bar [+15.37/−26.11 psi] from SV), and the pressure became stable in two minutes. Table 4 shows a summary of this simulation result.

Table 4. Simulation Test Result for Run Back Test.

Test Item	Header Press. Fluctuation (Bar)
Run Back Test	+ 1.06 / - 1.80

ACTUAL CONTROL PERFORMANCE

In the following paragraph, the application of the FF+FB/FF selector control system in a plant operating in the field is described.

Figure 14 shows the load shedding test result. This graph shows the trend data of the comp A flow, G/T A demand flow, and header pressure. In this test, G/T A relates to comp A, and comp A is in FF+FB mode. G/T B and comp B are stopped. After G/T A load shedding occurred, G/T Flow decreases. At the same time, comp A flow is decreased by feedforward load function. Header pressure at first goes up then comes down.

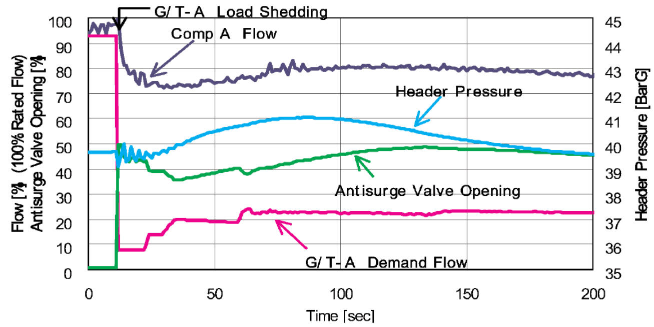


Figure 14. Actual Plant Load Shedding Test Result.

From this result of the actual test, the common header pressure fluctuation is found to be within 2 bar (29 psi) (+1.34/−0.61 bar [+19.44/−8.85 psi] from SV), and the pressure became stable in three minutes. Table 5 shows a summary of this test result.

Table 5. Load Shedding Test Result.

Test Item	Header Press. Fluctuation (Bar)
Load Shedding Test	+ 1.34 /- 0.61

Furthermore the feedforward load function was verified in the field. Figure 15 shows the verification result of the function. This graph shows the original design function line, modified function line, and actual operating points. The modified function line is based on the actual operating points, using a linear equation.

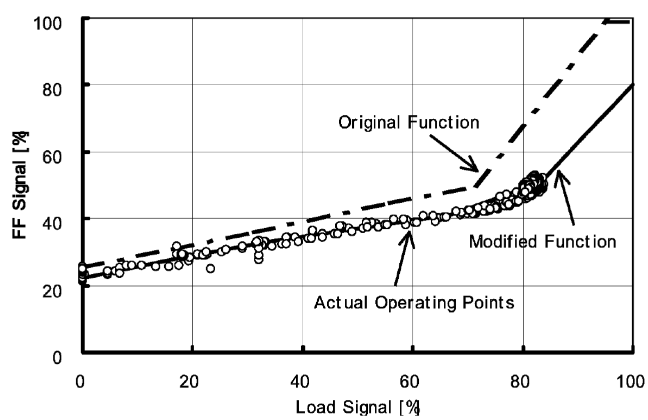


Figure 15. Feedforward Load Function and Actual Operating Points.

The difference between the original and modified function was the molecular weight of the gas. The original function is set by the compressor performance curve based on the design base molecular weight, 16.6. As for the actual gas, methane was less and ethane was more than the design base gas composition, and the actual gas molecular weight was 17.9 (about 108 percent from design). Because the actual gas base molecular weight became heavier than design base, the compressor performance has become better. Therefore the inclination of the function became small, and the IGV control band became narrow for the feedforward load function.

If the gas composition and molecular weight change, the feedforward function will actually shift. But in this control system, the FB signal covers the deviation from the design. It is possible to correspond by this control system if the molecular weight change is within 30 percent. This modified function is applied to the actual compressor controller.

CONCLUSION

In this paper, a compressor load sharing control system with a FF+FB/FF selector has been proposed. This control system is mainly provided for a GTCC plant fuel gas compressor. The key factor in this control system is the controllability of the compressor. This control system uses FF+FB mode and FF mode for each compressor unit. The gas turbine fuel flow is input as the FF signal and the control valves are modulated by the FF signal. The FB signal comes from the header pressure controller, and adjusts the control valves.

The functions of the control system are verified using a dynamic simulator. From the result of the simulation, it was confirmed that this control system has good controllability. Furthermore, this control system was verified using a simulation test, so it is concluded that the above control system shows good controllability in a plant operating in the field.

NOMENCLATURE

ASV	= Antisurge valve
CAL	= Calculation unit
Comp	= Compressor
FB	= Feedback
FF	= Feed forward
Fx	= Function generator
G/T	= Gas turbine
GTCC	= Gas turbine combined cycle
IGV	= Inlet guide vane
L/S	= Load shedding
LSS	= Low signal selector
MV	= Manipulating value
Pd	= Discharge pressure
PIC	= Pressure indicating controller
PLC	= Programmable logic controller
Ps	= Suction pressure
PV	= Process value
Qd	= Discharge flow
SEL	= Selector
SV	= Set value
Td	= Discharge temperature
Ts	= Suction temperature
UIC	= Antisurge controller

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