THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

93-GT-350

A SINGLE VALVE GAS FUEL FLOW CONTROL FOR GAS TURBINES

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ABSTRACT

A method of mass flow control of fuel gas to a gas turbine has been developed and applied in control retrofits to existing gas turbines. Unlike other gas flow control systems in use on gas turbines this system actually measures the mass flow going into the turbine combustion system and uses this value as the feedback in a control loop to modulate a single throttling control valve.

The system utilizes a common venturi flow element to develop a differential pressure which, along with inlet pressure and temperature, is used to compute the mass flow. Locating this flow element downstream of the control valve where the pressure is low at low flows reduces the usual problem of the wide range of delta-pressure (proportional to the square of the mass flow) to a workable level. This extends the range of this common type of flow measurement system enough that it becomes practical to apply it to the gas fuel flow control loop of a gas turbine.

INTRODUCTION

Figure (1) is a simplified representation of a typical utility or industrial gas turbine. Air is compressed from an inlet pressure roughly equal to atmospheric to a compressor discharge pressure (Pcd), which ranges from only slightly above atmospheric at light-off, to a level at maximum load of as high as 30 atmospheres depending on the compressor design, ambient temperature and turbine firing temperature.

A portion of this air is diverted to combustion zones where fuel (in the present case, a gaseous fuel) is burned, raising the air temperature. Most gas turbines of this type utilize multiple combustion zones in separate "cans" connected with pressure balancing "crossfire" tubes. This heated air is then blended with the rest of the air flow in dilution zones and expanded across a turbine stage back to the atmosphere generating power for both the compressor and the mechanical load.

Gas fuel at pressure (P4) is passed through the fuel nozzles into the combustion zones at combustion pressure (Pcc). The combustion pressure is reduced slightly below the compressor discharge pressure by passage through slots of the combustion liner. The gas nozzles are sized to create a nozzle pressure ratio (P4/Pcc) and gas velocity consistent with the combustor design.

Table (1) lists the calculated fuel gas flows and pressures for a typical medium sized generator drive gas turbine and a typical natural gas fuel.

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
Cincinnati, Ohio — May 24–27, 1993

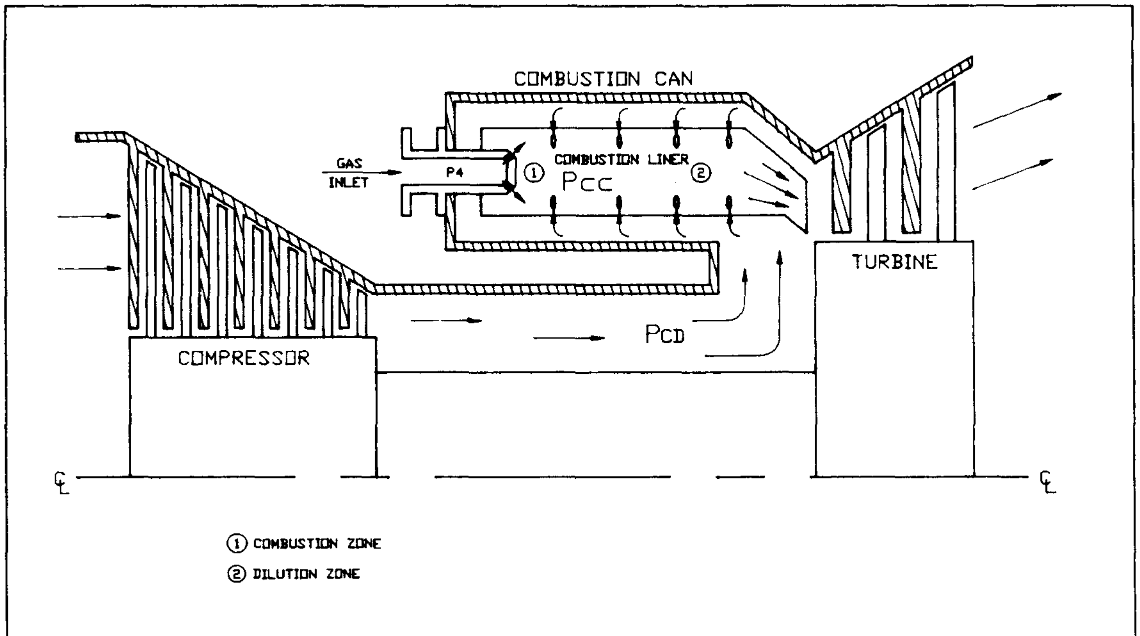

FIGURE 1

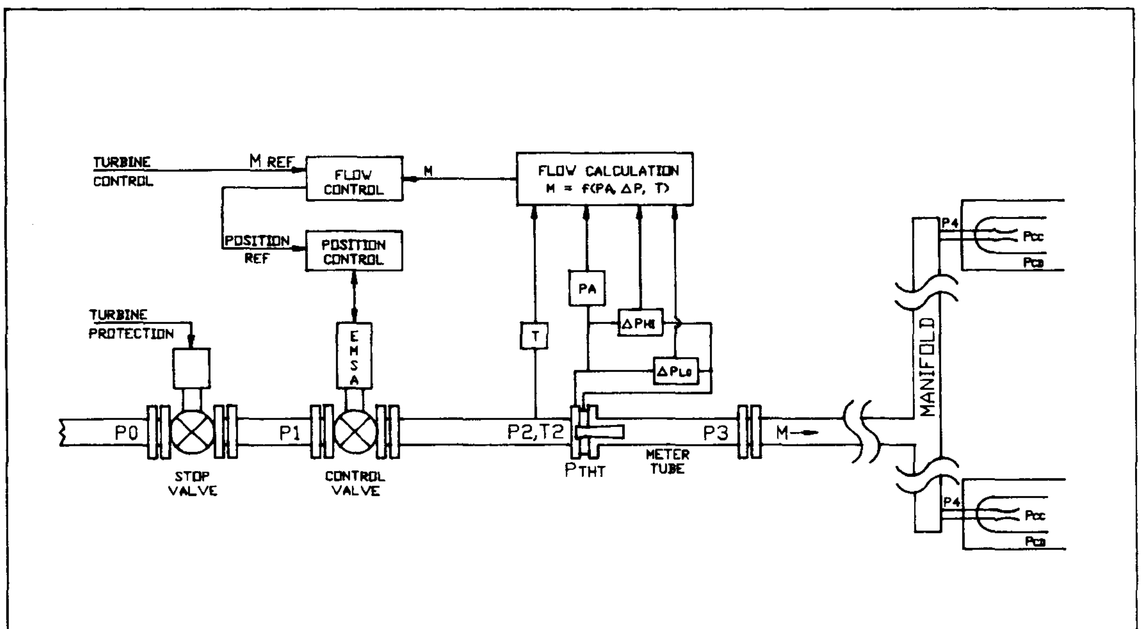

FIGURE 2

TABLE 1

Fuel - Natural Gas
- Lower heating value -20000 Btu/lb
- Specific Gravity (Air=1) -0.59 (Mw=17.09)
- Temperature (T2) -520 degR
- Specific heat ratio (K) -1.29

Flow tube (Reference 2)
Badger Meter Lo-Loss $_{Reg}$ Modified Venturi
Din - 3.07 inches
Dtht - 1.75 inches
Cflo - .8279
Loss Coeff - .0335

Control valve - Fisher 3" ET equal % trim; 1.5 inch lift
Stop valve - Fisher 3" ETR quick opening trim
Supply press (P0) - 215 psia (constant)

Turbine Operating Condition	Gas mass flow - M (lbs/sec)	Pcd (psia)	Pcc (psia)	P4 (psia)	P3 (psia)	Ptht (psia)	P2 (psia)	delta press ("H_2O)	P1 (psia)	Valve lift (%)
light-off	.25	18.00	17.37	18.03	18.34	17.78	18.36	16.19	215.0	7.0
part speed	.60	50.00	48.25	49.61	50.19	49.01	50.23	33.73	215.0	18.0
full speed	1.0	98.50	95.05	96.97	97.72	96.05	97.77	47.70	215.0	28.5
1/2 load	3.0	121.9	117.6	131.8	136.0	124.1	136.4	340.9	213.2	53.8
rated load	3.85	129.8	125.2	147.3	153.2	135.0	153.9	522.3	212.1	60.5
max load	5.0	150.0	144.8	177.2	185.2	158.8	186.1	755.3	210.1	73.0

EQUATION 1

$$M = Cflo*Atht*Y*\left[\frac{2*RHO2*(P2-Ptht)}{1-B^4}\right]^{1/2}$$

Where:
RHO2 = P2*Mw/(R*T2)
Weight flow (lbs weight/second) = M*G (English System)

and

$$Y = \left[\frac{r^{2/k}*K*(1-r^{(k-1)/k})*(1-B^4)}{(k-1)*(1-r)*(1-B^4*r^{2/k})}\right]^{1/2}$$

NOMENCLATURE

		English	S.I.
Atht	- Flow element throat area	ft^2	m^2
B	- Flow element Beta ratio (Dtht/Din)	dimensionless	
Cflo	- Flow element discharge coefficient	dimensionless	
Din	- Flow element inlet diameter	ft	m
Dtht	- Flow element throat diameter	ft	m
G	- Gravitational constant	ft/sec^2	**
K	- Gas specific heat ratio (Cp/Cv)	dimensionless	
M	- Gas mass flow	**	kg/sec
Mw	- Gas molecular weight	dimensionless	
P0	- Gas line supply pressure *	lb/ft^2	n/m^2
P1	- Gas control valve inlet pressure *	lb/ft^2	n/m^2
P2	- Flow element inlet pressure *	lb/ft^2	n/m^2
P3	- Flow element discharge pressure *	lb/ft^2	n/m^2
P4	- Fuel nozzle inlet pressure *	lb/ft^2	n/m^2
Ptht	- Flow element throat pressure *	lb/ft^2	n/m^2
Pcc	- Gas turbine combustor pressure *	lb/ft^2	n/m^2
Pcd	- Gas turbine comp. disch. pressure *	lb/ft^2	n/m^2
R	- Gas constant	**	**
r	- Flow element pressure ratio (Ptht/P2)	dimensionless	
RH02	- Flow element inlet density	**	kg/m^3
T2	- Flow element inlet temperature	degR	degK
Y	- Flow element expansion factor	dimensionless	

* All pressure units are absolute.

** English units for "mass" flow (M) are pounds weight per second requiring use of the gravitational acceleration constant (G) (32.17 ft/sec^2) in English calculations. The gas constant (R) is 49720 lbf*ft/(slugmole)*degR in the English system and 8315 $kg*m^2/sec^2$*(kg-mole)*degK in the S.I. system. The inlet density (RH02) in the English system is $slugs/ft^3$.

MEASUREMENT AND CONTROL SYSTEM

As is evident from the values in Table (1) the fuel gas pressure (P3), at the point of delivery to the gas distribution manifold has a range of approximately 10/1 (185/18.3) for a mass flow range of 20/1 (5.0/.25). As will be described next it is this strong relationship between the fuel mass flow and the delivery pressure that extends the practical range of the flow measurement system.

Equation (1) (Reference 1) is the basic expression for the mass flow of a compressible fluid through a flow restriction.

From an examination of Equation (1) it can be seen that, with a constant value of (RH02) (essentially a constant value of (P2) if (T2) is invariant), the value of (P2-Ptht) will be proportional to (M) squared. This square function is the fact of fluid flow that usually limits the application of a flow restriction type flow meter to a mass flow range of considerably less than that required in the operation of a gas turbine. In the example this flow range is 20/1 which would result in a 400/1 range of the primary measurement for the flow calculation, the venturi delta-pressure. This would greatly exceed the practical range of commercial differential pressure transmitters.

If however, the flow restriction is placed in a location where the inlet pressure (P2), is reduced substantially at low mass flow, the reduction of (RHO2) will alleviate this range problem. This is exactly the case in the present application. Figure (2) is a schematic representation of the fuel control and delivery system. Table (1) includes the pressures for a particular venturi. Typical control and stop valves are also included and the per-cent lift of the control valve is listed for each flow.

With the venturi section located downstream of the throttling valve the 20/1 range of the mass flow (M), is accompanied by a 10/1 reduction of (P2). This results in roughly only a 46/1 range in the venturi delta-pressure, well within the capacity of two staged commercial differential pressure transmitters.

The micro-processor based flow control is a conventional digital P.I.D. loop with the gains adjusted to provide a stable, responsive control. The delta-pressure and absolute pressure transmitters are typical industrial models with a 200 milli-second time constant. The flow loop applies an analog position reference to a position/velocity servo-amplifier which operates a brushless D.C. servo-motor connected to the control valve stem by a re-circulating ball screw. This flow-position control is operated at an approximately 50Hz rate giving a more than adequate response to the fuel flow changes demanded by the turbine control. Field experience has shown adequate dynamic response to such transients as full load rejection and transfers between gas and distillate fuel where the gas flow is reduced to zero at full gas turbine load.

SUMMARY

By taking advantage of the reduction in inlet density (pressure) at low mass flows to extend the range of a common venturi type mass flow meter, a method of controlling the mass flow of fuel gas to a gas turbine has been accomplished. The control consists of a single modulating valve and a flow meter system with no moving parts. A patent (#5,146,941) for this system has been issued by the U.S. Patent and Trademark Office.

Other turbine fuel gas flow controls either use two modulating valves (a pressure control valve followed by a linear valve which approximates flow control by position control) or a simpler system in which a single valve is modulated by the outer loop turbine controls (acceleration, speed, temperature, load, etc.) without operating a flow loop.

The system described in this paper achieves the stability and response advantages of having a flow control loop operating inside the outer turbine control loops while maintaining the simplicity of a single modulating valve. The fact that the system provides a true mass flow control allows for accurate matching of gas and liquid fuels as well as supplying a mass flow signal for use in emissions control and performance data gathering systems.

At the time this paper was written this system had been put in operation in three installations. One case was the addition of gas fuel to a unit originally equipped for distillate only operation. A micro-processor was added to operate the new gas flow control and to interface with the existing analog turbine governor and relay type sequencing system. The other two applications were in conjunction with turbine control conversions where the original electro-hydro-mechanical turbine governor was replaced with a digital control which included the gas flow measurement and control software.

REFERENCES

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