PSIG 1816

Long-Haul Natural Gas Pipeline Compressor Station Optimization

Vadim Shapiro[1], John Hooker[1], Lawrence Youngblood[1]

[1] Statistics & Control, Inc.

This paper was prepared for presentation at the PSIG Annual Meeting held in Deer Valley, Utah, 15 May – 18 May 2018.

ABSTRACT

Operators of long natural gas pipeline networks with multiple gas compressor stations are looking for ways to optimize operations, reduce fuel consumption and costs, and maximize producer and consumer throughput in real time. This paper introduces a unique method to optimize these complex natural gas pipeline total value streams and achieves the stated objectives by integrating turbomachinery algorithms into a high-fidelity simulator to improve total model and simulation accuracy. Through much trial and error, the authors discovered how to account for the physical network size and how the combination of turbomachinery and gas pipeline algorithms as well as advances to auto-tuning methods may be used to solve the challenge of simulating complex natural gas pipeline networks in real time. Auto-tuning algorithms and the simulator results were validated with real field data from natural gas pipelines to ensure model and simulation accuracy. Technological advances incorporated into the pipeline simulator included a new parsing engine, advanced regulatory control, and incorporating turbomachinery algorithms for mixed mode operations (gas compressors driven by gas and electrical turbines as well as reciprocating engines). This paper describes our method and these advances and demonstrates how they may be applied to all natural gas pipelines to significantly improve operations, reduce costs, and improve the operating envelope.

INTRODUCTION

Natural gas transmission pipelines transport large quantities of gas across long distances and deliver it to major consumers (local distributors, large industrial end users, electrical generation facilities). Natural gas is introduced into a pipeline transmission system at various points, such as LNG terminals, processing facilities near supply fields, and interconnections with other gas pipelines. This gas is transported in high-pressure pipelines and a series of compressor stations. Compressor stations provide the power required to transport the gas in the pipeline from one location to another and usually contain more than one compression unit. A unit is defined as a combination of a compressor and its engine. A gas compressor station may have a diverse combination of units, resulting in a more complex operational envelope not possible to solve using traditional optimization techniques. The methodology presented in this paper is unique in that it was designed to model diverse unit configurations and to generate optimization results based on multiple objective functions.

Compressor station optimization is solved using a common approach of building the operating envelope of each unit. Units are started or shut down or changing the load to determine the optimal load-sharing set points based on comparing the current objective function value with its value computed using the model after the planned load changes. The optimization algorithm has the following components:

- Objective function: Dependence of optimization criteria on the compressor operating modes and unit combinations at the stations
- Dependence equations: Power and material balance pipeline and compressor station models
- Constraints: Ambient conditions, operating limits, unit equipment technical states, fuel supply limits, and pipeline balance models
- Decision points: Station discharge pressure controlled through station pressure master control
- Process set point: Gas consumption
- Optimization conditions: Conditions indicating unit

ordering as well as minimal start-up/shutdown time

This paper discusses modeling and building the objective function for compressor stations and pipelines. At the core of dynamic simulation is a mathematical model describing the static and transient behavior of pipeline components and equipment based on heat and material balances and on the performance characteristics of each system element. An element is a pipe, a valve, a gas compressor, an engine, etc. Dynamic process simulation predicts intermediate process conditions when the pipeline, compressor stations, and units transition from different operating conditions.

TRANSIENT FLOW SEGMENTS

Transient pipe flow simulation is based on the concept that flow changes are a result of pressure waves created by disturbances. A pressure wave travels at sonic velocity, C. During the computation scan time, Δt, the pressure wave propagates a distance L in the pipe. L is calculated using equation 1.

$$L = C \cdot \Delta t \tag{1}$$

Initially, the pipeline is divided into sufficiently small sections, or segments of length L, as illustrated in Figure 1. Segments represent each type of modeled component. For example, compressors, pipeline nodes, and control valves are all modeled as a segment. Each component is described by equations of motion, continuity, energy, and specific state. Pressure, temperature, density, and mass flow are assumed to be the same from the left end of the segment to the right end.

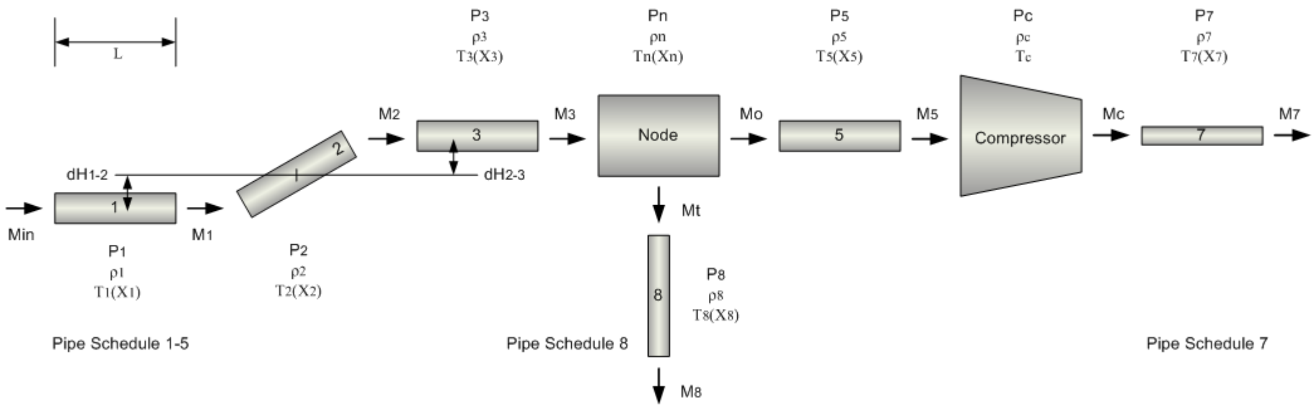

Figure 1 – Dividing a pipeline into segments

BUILDING THE MATHEMATICAL MODEL

The fluid flow mathematical model determines fluid behavior throughout the pipeline by solving a system of nonlinear partial differential equations for four state variables:

- Pressure (P)
- Temperature (T)
- Density (ρ)
- Mass Flow (M)

These equations incorporate the following:

- Derivatives from equations of motion, continuity, energy, and state
- Inertia
- Gas mixture composition expressed in molecular fractions
- Compressibility factor as a function of temperature and pressure
- Equation of state for gas flow related to pressure, temperature, and density
- Fluid properties that change with changing conditions
- Heat exchange with outside environment
- Ambient temperature
- Pipe cross-sectional area, pipe fittings, and pipe resistance configurable for each pipeline segment
- Friction factor as a function of the Reynolds number

Equation 2 expresses the continuity equation, which states that the mass of the control volume remains constant:

$$-\frac{dM}{dx} = A\frac{d\rho}{dt}, \tag{2}$$

where A is the cross-section area. The Navier-Stokes equation that applies Newton’s second law of motion to viscous fluids is given in equation 3.

$$\rho \frac{dw}{dt} = \rho g \cdot \frac{\partial P}{\partial x} + \mu \frac{\partial^2 w}{\partial x^2}, \quad (3)$$

where g is the acceleration due to gravity, μ is the viscosity, and w is the flow velocity. The equation of state given in equation 4 relates pressure, temperature, and density.

$$M = f(P, T, \rho) \quad (4)$$

The equation of energy (equation 5) states that if heat is added to the system or the system does work, the system energy changes according to the First Law of Thermodynamics.

$$q = D\frac{d}{dt}\left(c_v T + \frac{W^2}{2}\right) + \frac{P}{\rho A}\cdot\frac{\partial(Aw)}{\partial w} + \frac{w}{P}\cdot\frac{\partial P}{\partial x}, \quad (5)$$

where D is the diameter and c_v is the specific heat at constant volume.

Compressor Model

Building the compressor model begins with manufacturer specifications, including the impeller eye, mass flow, suction pressure, suction temperature, and rotational speed, as well as the compressor map. The compressor map provides a collection of performance curves and characteristics over a range of rotational speeds. For a given speed, a line describes the flow versus polytropic head relationship for any axial or centrifugal compressor. Figure 2 shows an example compressor map.

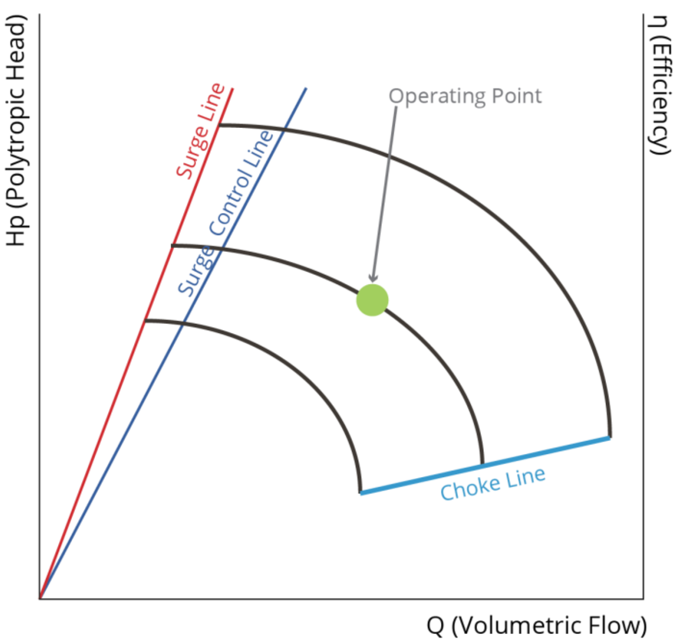

Figure 2 – Example compressor map

Polytropic head cannot be directly measured and is calculated using equation 6.

$$H_p = \frac{R_o \cdot Z_{avg} \cdot T_s}{MW} \cdot \frac{R_c^{\sigma} - 1}{\sigma}, \quad (6)$$

where H_p is the compressor polytropic head, R_0 is the universal gas constant, Z_{avg} is the average gas compressibility factor, T_s is the temperature in the compressor suction, MW is power consumption, R_c is the calculated compression ratio of discharge pressure (P_d) to suction pressure (P_s), and σ is the gas polytropic exponent calculated using equation 7.

$$\sigma = \frac{n-1}{n} = \frac{k-1}{k \cdot \eta}, \quad (7)$$

where n is the gas polytropic volume exponent, k is the gas isentropic exponent, and η is the polytropic efficiency coefficient. The relationship between temperature and the compression ratio for the polytropic processes is shown in equation 8.

$$\frac{T_d}{T_s} = \left(\frac{P_d}{P_s}\right)^{\sigma} \quad (8)$$

Therefore, the polytropic exponent sigma can also be calculated using equation (9).

$$\sigma = \frac{\log\frac{T_d}{T_s}}{\log\frac{P_d}{P_s}} \quad (9)$$

Suction volumetric flow (Q) is calculated using equation (10).

$$Q = A\sqrt{dP \cdot \rho}, \quad (10)$$

where A is a flow measuring device constant coefficient dP is the pressure difference across the compressor flow measuring device, and ρ is the gas density in the compressor suction. Density can be calculated using equation (11).

$$\rho = \frac{MolW \cdot P_s}{R_0 \cdot T_s \cdot Z}, \quad (11)$$

where Z is the gas compressibility factor in the suction and $MolW$ is the gas molecular weight.

COMPRESSOR STATION OPTIMIZATION

Figure 3 shows a simple total value stream with one pipeline system input; the pipeline system branches into two pipelines that each run through two compressor stations and end in two different consumption areas.

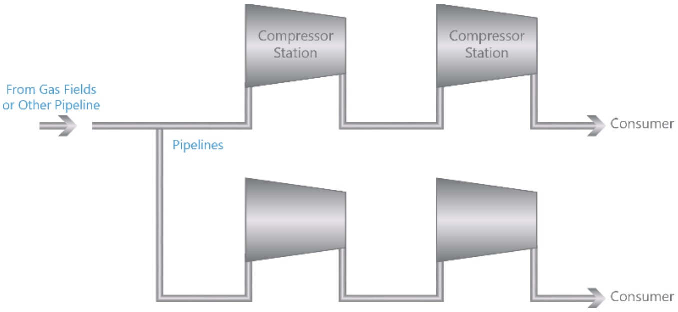

Figure 3 – Example value stream for compressor station optimization

Compressor station optimization begins by determining the required consumer area mass flow or energy, which determines the pipeline and compressor station demand. After mass flow throughput requirements are known, optimization turns to controlling each individual unit within stations as well as controlling each station.

Figure 4 shows an example of how this works. In this example, the compressor station is controlled by the discharge pressure. Optimization works from the Consumer and looks at the Compressor Station 2 discharge pressure. This discharge pressure value becomes the set point for Compressor Station 2. To achieve the discharge pressure set point in each compressor station, optimization manipulates individual compressor speeds (RPM) if the compressors have variable speed drive or determines whether compressors need to be started up or shut down.

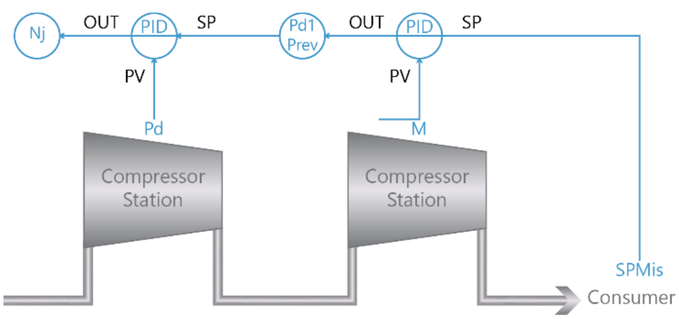

Figure 4 – Controlling compressor stations and compressors

The simulation creates an operating envelope on the compressor map that defines an operating limit low point and an operating limit high point, as shown in Figure 5. The operating limit low point is formed by the speed low limit added to an operating low limit margin. The operating high limit is created by the speed high limit added to an operating high limit margin.

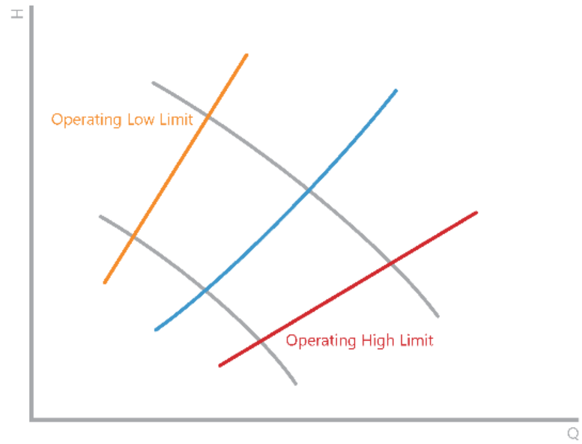

Figure 5 – Compressor map operating envelope

When optimization decides whether a unit needs to be started, it determines if any compressor operating points have crossed the operating high limit, if all compressors have reached their maximum speed, if all engine units are running at their high limit (e.g., temperature high point, vibration high point, noise high point), and if adding a new compressor causes the operating point to cross the operating low limit. Shutting down a compressor is suggested when the operating points of all compressors cross the operating low limit or if turning off a compressor keeps the operating point within the operating high limit.

The next step is to build the objective function, which can be done a several ways. The first case simply uses the previous discharge pressure corresponding to the pipeline flow (*Pd Prev*) and Consumer pressure (*Pc*), as shown in Figure 6, to provide suggested flow rates to maintain pressure.

Figure 6 – Objective function case 1

The second case, illustrated in Figure 7, uses compressor station load balancing.

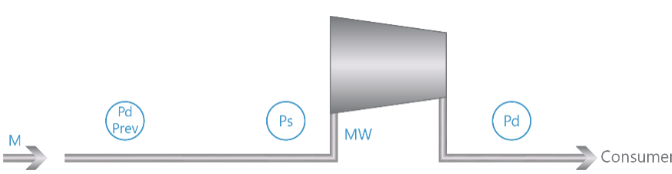

Figure 7 – Objective function case 2

In this second case, the objective function is built such that the mass flow rate (M) equals the Consumer flow rate request. The optimization algorithm steps compressor speed up and down and records the previous discharge pressure and compressor power (MW) consumption. Another objective function is built based on megawatt consumption and compressor stepping. When the lowest megawatt consumption is obtained at a discharge pressure, the optimization algorithms ensure the requested Consumer mass flow rate is achieved.

The third case builds the objective function in the same way as the second case, but it differs in that the algorithm balances load across multiple compressor stations and Consumers, as shown in Figure 8. Instead of looking at the mass flow rates and discharge pressure of one compressor station, the optimization algorithm balances individual units within compressor stations based on the required Consumer flow and discharge pressure.

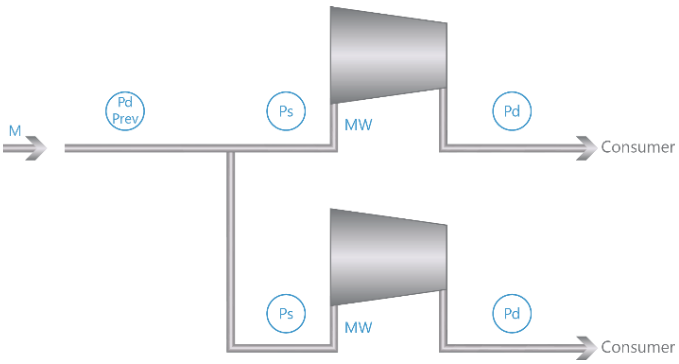

Figure 8 – Objective function case 3

The fourth and final case is similar to the previous two cases except that two pipelines converge into one compressor station instead of branching into multiple compressor stations, as illustrated in Figure 9.

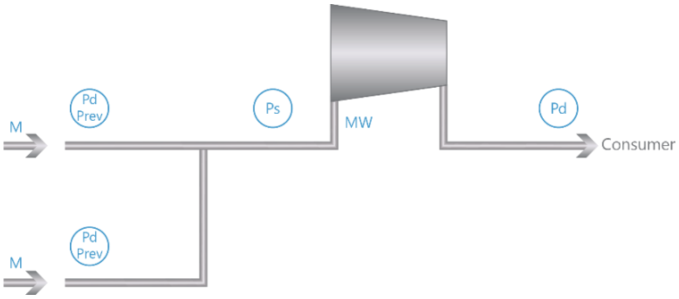

Figure 9 – Objective function case 4

In case 4, the optimization algorithm uses two mass flow set points and balances them across one discharge pressure. The discharge pressures of the two previous pipelines are averaged as the suction pressure entering the compressor station.

Once the objective function is built, the model may be simulated, optimized, and compared to normal operation to view the difference.

CASE STUDY

The modeling and optimization techniques presented were implemented on a 150-km (~93 mi) commercial pipeline network used to transport 180,000 ton/day from two production areas to a complex pipeline network with multiple consumers. A simplified diagram of the total value stream is shown in Figure 10. All 6 compressor stations run in parallel. Stations 1 and 2 have 5 single-stage compressor trains, with each train connected to a gas turbine. Stations 3 and 4 have 3 single-stage compressor trains, while Stations 5 and 6 have two single-stage compressor trains.

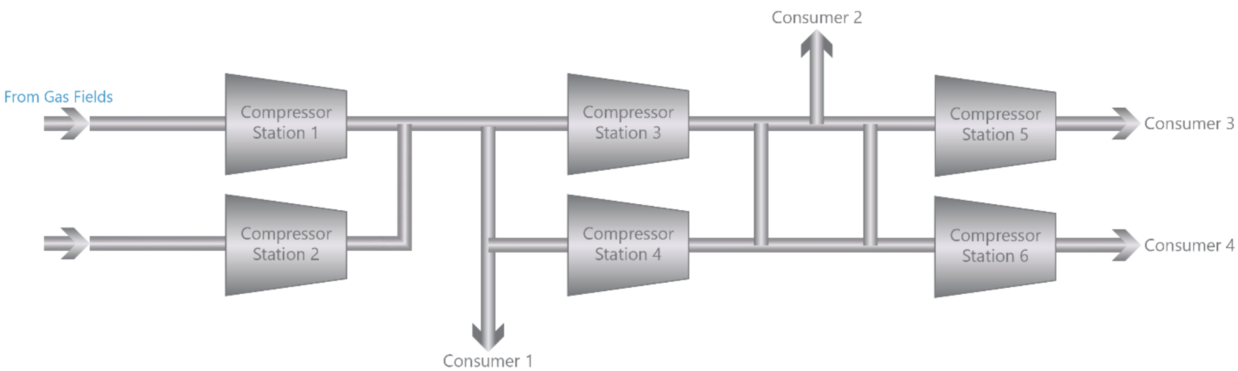

Figure 10 – Pipeline system model

Problem Statement

In this pipeline system, the two parallel pipeline systems had different pressures, causing one compressor station to receive a larger amount of fluid. The end user wanted to balance and distribute the flow coming out of compressor stations at similar points in the pipeline more evenly. The four Consumers also had different pressure or flow set points, requiring the model to consider the pressure set points and flow requests for each Consumer.

Initial Data

For modeling, simulation, and optimization, the following initial data was collected:

- Compressor inputs (manufacturer compressor map, flow, suction pressure, temperature, and impeller diameter)
- Gas properties (refer to Table 1)
- Gas turbine inputs (from manufacturer datasheets)
- System flow and pressure data (refer to Table 2)
- Piping geometry and elevation changes (P&IDs with pipe diameters, pipeline elevation changes, ambient temperature, and pipeline material)

Table 1 – Gas properties

Mole Fraction	Formula	Name	Cp/Cv	Cv
0.989	CH_4	Methane	1.31	1.7
0.006	CO_2	Carbon Dioxide	1.32	0.655
0.005	N_2	Nitrogen	1.4	0.743

Table 2 – System flow and pressure data

Model Area	Input Pressure (PSIG)	Output Pressure (PSIG)	Flow (mscf/hr)	Power (MW)
Station 1	101	435	1713	1.4
Station 2	72.5	363	1271	1.1
Consumer 1		406	194	
Station 3	94.3	421	1501	1.4
Station 4	71.1	370	1289	1
Consumer 2		123	35.3	
Station 5	101	406	1487	1.2
Station 6	62.4	348	1268	1
Consumer 3		102	1466	
Consumer 4		92.8	1289	

With this information, the baseline simulation was created to match the initial field results.

Results

The goals were to maintain pipeline pressure and flow rate while reducing compressor power consumption from 1% to 3%. After simulating the model and running the optimization algorithms, the flow and pressure data results shown in Table 3 were suggested.

Table 3 – Optimized system flow and pressure set point suggestions

Model Area	Input Pressure (PSIG)	Output Pressure (PSIG)	Flow (mscf/hr)	Power (MW)
Station 1	97.9	449	1501	1.2
Station 2	90.6	443	1483	1.1
Consumer 1		406	194	
Station 3	92.1	428	1416	1.25
Station 4	85.6	413	1374	1.1
Consumer 2		123	35.3	
Station 5	87.0	428	1481	1.2
Station 6	71.1	363	1414	1
Consumer 3		102	1466	
Consumer 4		92.8	1289	

Comparing the suggestions with the original data in Table 2, we can see that Compressor Station 1 had the greatest efficiency and power saving gains. For Compressor Station 1, the third objective function case was used to balance Compressor Stations 1 and 2 and also distribute pipeline flow more evenly.

Figure 11 takes the data for Compressor Station 1 and shows a power over flow curve with pressure. This chart shows operators the current compressor station operating point in the operating envelope as well as optimization suggested set points to move the operating point.

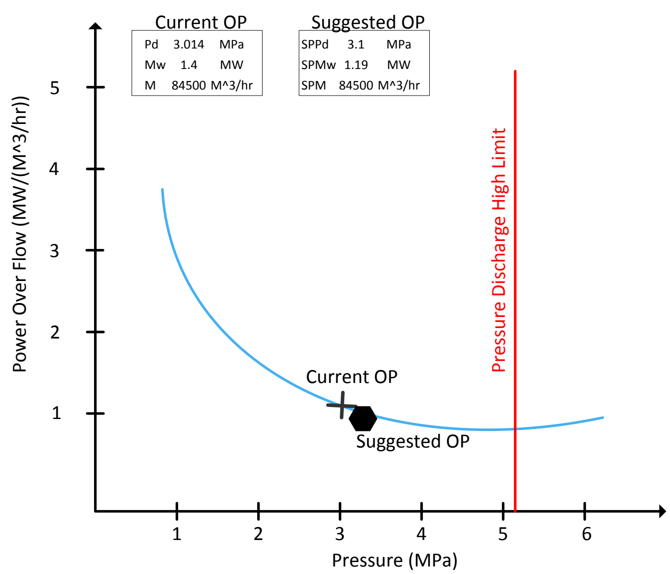

Figure 11 – Power over flow curve for Compressor Station 1

By optimizing load sharing and load balancing in Compressor Station 1, the compressors run more efficiently. The optimization suggested a slightly higher discharge pressure set point, resulting in a lower power consumption set point (represented by the Suggested OP hexagon).

CONCLUSIONS

Based on the case study results, the described modeling and simulation techniques can suggest real-time set points to help balance the compressor load given desired discharge pressure and mass flow rate targets. Variable-speed compressors offered a wider operating range than simply starting up and shutting down on/off for compressors, also providing another savings opportunity. Future work includes extending the power over flow curve for the entire pipeline instead of on a station-by-station basis to help maximize efficiency across the total pipeline value stream.

AUTHOR BIOGRAPHY

Vadim Shapiro is co-founder and President of Statistics & Control, Inc. He manages all aspects of engineering, including development of the company's *OptiRamp*® Advanced Process Control software. Vadim has over 25 years' leadership experience in systems and software engineering, project management, and product development in the field of turbo machinery control, advanced process control, and power management systems. He holds six patents in the areas of turbomachinery control and advanced process control, with several applications pending.

John Hooker is a systems engineer and the *OptiRamp* group manager at Statistics & Control, Inc. He has a Bachelor of Science in Aerospace Engineering from Iowa State University. Over the past 7.5 years, John has participated in several global projects in the oil & gas and power generation sectors and has implemented numerous applications for control solutions, equipment monitoring, design testing, data analysis, and brownfield optimization.

Lawrence Youngblood, PE, is an experienced engineer with over 27 years in the global oil & gas industry and a proven track record managing small, medium, and large technical teams. He is recognized for analyzing problems, developing and simplifying procedures, and finding innovative process control solutions (APC) for optimizing equipment and process streams within oil, gas, and chemical plants.