WHITEPAPER

SYNERGI™ GAS

Is the gas delivery system you're looking at able to meet your power generation needs: a hydraulic engineer's perspective

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ABSTRACT

The hydraulic characteristics, and system response to demands and supplies, of natural gas pipelines are substantially different from the flow characteristics of electric transmission systems. Pipelines must incorporate consideration of packing and drafting phenomena into their analyses. This is especially true for gas consumed in conjunction with electrical generation requirements, which typically exhibit sudden and substantial demands. These demands can cause pressure problems in the gas pipeline.

The pipeline controller must take this demand nature into account when planning pipeline operations, and should perform careful hydraulic analysis to determine the adequacy of his or her system to meet demands, the impact these demands have on both short- and long-term operations, and the environmental and safety issues involved.

This paper discusses the flow characteristics of natural gas pipelines, the phenomenon of line pack and subsequent draft, and related operational issues. Both steady-state and unsteady-state (transient) flow characteristics are covered, and some typical techniques for dealing with power generation demands are discussed.

INTRODUCTION AND PURPOSE

The natural gas industry has seen, in the past few years, substantial increases in the amount of natural gas being used as fuel for electric power generation. The nature of electrical demand and consumption yields three points that are key to understanding the operating problems caused by natural gas consumption:

- electrical generation usage is characterized by sudden large changes in demand,
- demand changes are usually requested with very short lead times, and
- gas must be delivered at a reasonably high pressure.

Both sides of the issue -- the electric generation industry and the natural gas transmission industry, have room for increased understanding of each other's needs, as well as the phenomenon itself.

The purpose of this tutorial, is to address some of the hydraulic issues involved, from the perspective of natural gas hydraulic analysis. Attendees from the electrical power generation industry will, hopefully, gain insight into the unique operating problems faced by the gas transmission industry. Conversely, attendees from the natural gas transmission industry will, again hopefully, realize benefit from increased insight into steady-state and transient design and operations philosophies.

We will begin with a review of the basic issue involved: steady-state flow. From there, we will deepen our discussion by introducing the idea of transient flow, and the packing and drafting that occurs in a gas pipeline system.

In steady flow conditions, a gas pipeline and an electrical transmission line behave similarly. At any given instant in time, the flow entering the system is equal to the flow leaving, and the amount of product in the line is a constant.

When a flow change is required, however, the response of the two systems differs markedly. Unlike gas pipelines, power transmission lines cannot store significant energy, and thus rapidly settle to a new state, in a few cycles if there is sufficient energy. Depending upon the physics of the system and imposed boundary conditions, however, gas pipelines can take anywhere from a few hours to a few days to settle to a new steady-state. During this settling period the flow rates, pressures, and gas inventory can, and almost certainly will, vary substantially at different points along the pipeline.

Mathematically, the equations describing steady-state and transient flow must be inherently different. First, however, we will limit our discussion to steady-state flow.

STEADY-STATE FLOW FORMULAE

In the natural gas industry there are a number of equations used to describe pressure loss through a pipe. For reasons of practicality, history, experience, etc., different companies will utilize different flow equations. Some of the most common are the General Flow equation, the Panhandle 'A', the Panhandle 'B', and the Weymouth. Without exception, all of these formula have as their common ancestor the *D'arcy-Weisbach* flow equation:

$$\Delta H = f \frac{L}{D} \frac{V^2}{2g}$$

Equation 1

While perfectly valid for all flow conditions, this equation is inappropriate in its this form for compressible flow. If one converts the ΔH term to a pressure differential (ΔP) term, includes the Q=VA relationship, and incorporates the fact that the fluid we are modeling is compressible, one inevitably arrives at what is called the *General Flow*, or sometimes called the *Fundamental Flow*, equation:

$$Q = 77.54 \frac{T_b}{P_b} \left(D^{2.5}\right) e \left[\frac{\left(P_1^2 - P_2^2\right)}{GTLZf} \right]^{0.5}$$

Equation 2

For the sake of simplicity, we have omitted elevation correction factors. Notice that the flow equation contains two terms, *f* and *Z*, the pipeline friction and compressibility factors, which are in turn functions of pipeline flow and pressure respectively. What this means, simply put, is that for frictional effects flow is dependent upon flow -- the pipeline flow equation has flow on both sides of the equals sign, and thus cannot be directly solved. We could attempt to rearrange the equation, putting pressure on the left side of the equation, but then we would have pressure terms (P and Z) on both sides. We can really complicate the problem when we introduce the equation used to determine friction, the *Colebrook-White* equation:

$$\frac{1}{\sqrt{f}} = -0.86\,\mathbf{ln}\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right)$$

Equation 3

The Re term is the Reynold's number, an index of flow (turbulent, transition, laminar), and is:

$$Re = \frac{VD\rho}{\mu}$$

Equation 4

A very cursory examination of the Colebrook-White equation for friction, (Equation 3) shows that this equation must be iterated -- since there is a friction term on both sides of the equation. A reasonable friction approximation is the *Shacham* equation:

$$f = \left\{-2\,\log\left[\frac{\frac{\varepsilon}{D}}{3.7} - \frac{5.02}{Re}\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{14.5}{Re}\right)\right]\right\}^{-2}$$

Equation 5

When we look again at *Equation 2*, the general flow equation, we see that there are 4 distinct variables that interact, the flowrate, Q, the pipeline inside diameter, D, the upstream pressure, P_1, and the downstream pressure, P_2. Most of us have, at some point in our education or work experiences, solved for pressure loss in a liquid pipeline, and have found that there are two absolutely indisputable facts:

- Pressure loss along the horizontal axis of the pipeline is linear.
- An increase X units of pressure on the upstream end of the pipeline yields a corresponding increase of X units of pressure at the downstream end.

Gas pipelines, however, behave quite differently. Let us rearrange the terms of the general flow equation to solve for P_2, given P_1, Q, and D:

$$Q = 77.54 \frac{T_b}{P_b} \left(D^{2.5}\right) e \left[\frac{\left(P_1^2 - P_2^2\right)}{GTLZf} \right]^{0.5}$$

$$P_2 = \left\{ P_1^2 - \left[\frac{Q \bullet P_b}{77.54\left(T_b\right) D^{2.5}} \right]^2 \bullet GTLZf \right\}^{0.5}$$

Equation 6

A brief inspection of this equation yields an immediate observation -- that pressure loss through a gas pipeline cannot be linear. This may seem a trivial observation, but is of vital importance in the gas industry. A plot of pressure with respect to axial distance along the pipeline will follow a concave downward curve. A second observation also comes to mind -- not only is the pressure curve nonlinear, but P_2 is a function of the changing gas properties (specifically, the compressibility factor, Z). Thus, in a gas system, an increase of X units of pressure at the upstream end of the pipeline will yield Y units of pressure at the downstream, and Y>X.

Consider a level 80-mile, 24” nominal diameter pipeline, having a wall thickness of 0.281”. An important consideration in determination of friction is the roughness of the pipeline, which we will assume is 0.0006”, a common value for regularly cleaned transmission pipelines. Pipeline efficiency is assumed to be 0.97. All gas is supplied at a point called SOURCE, and has an inlet pressure of 900 psig. There is a single delivery point, LOAD1, with a demand of 325 mmcfd. We assume the gas has a specific gravity of 0.65, and a flowing temperature of 65 ^{o}FA practical example:

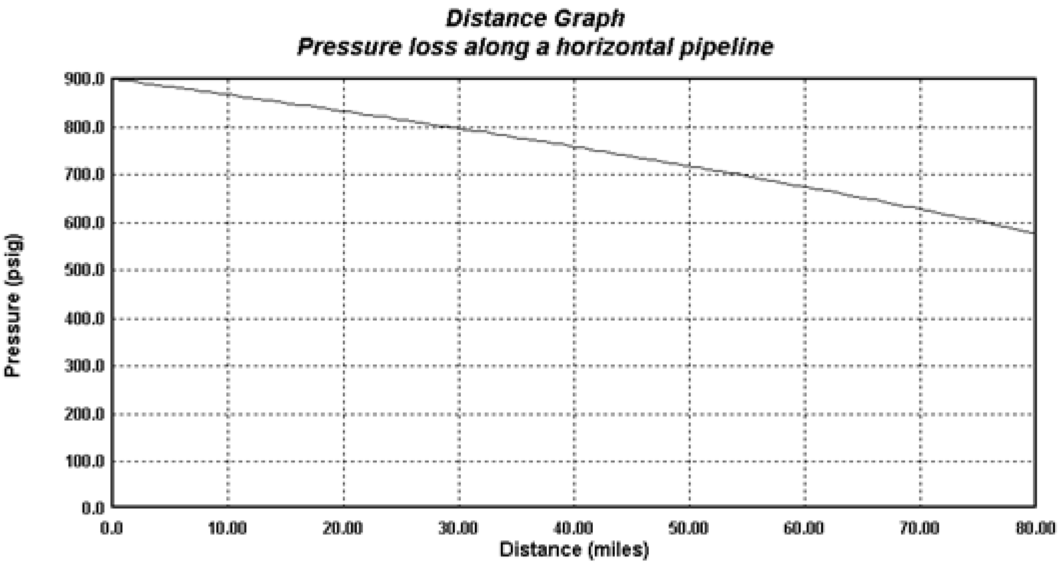


Figure 1 - Steady-state pressure profile -- P1 = 900 psig

The equations we introduced earlier predict a pressure at the delivery of 575.3 psig. The graph shows the non-linear nature of the pressure profile, as discussed earlier. To test our observation regarding pressure decline, we will solve the equations again, this time, however, we will lower the inlet pressure to 850 psig. If our observations are correct, the pressure at the downstream end of the pipeline will be less than (575.3 - 50) 525 psig:

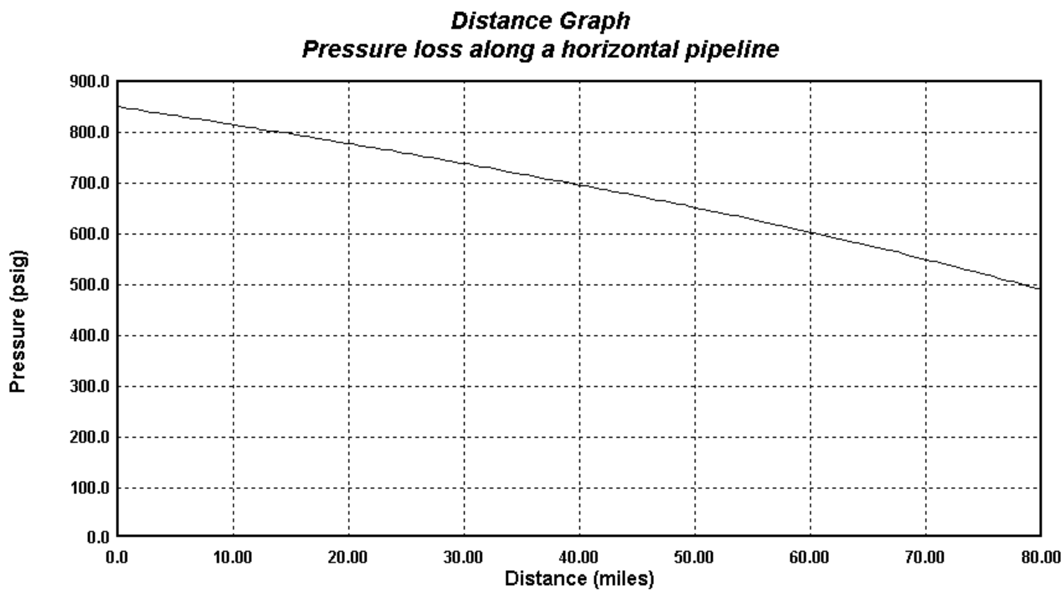


Figure 1 - Steady-state pressure profile -- P1 = 850 psig

Indeed, this is the case, the computed pressure is 488.2 psig. It is also of interest to note that in both Figure 1 and Figure 2 the graph does concave downward, as discussed. Further, it is apparent that the slope of Figure 2 is 'steeper' than that of Figure 1.

This is another hydraulic characteristic of natural gas pipelines that operators and designers must acknowledge -- the pressure loss along a pipeline behaves differently as inlet pressure is lower, becoming increasingly critical. In the first case, the overall pressure loss was (900 - 575.3) 324.7 psig. In the second case the pressure loss is (850 - 488.2) 361.8 psig. Let us further compare pressure losses at mileposts along the pipeline.

	Pressures			
Milepost	P1 = 900 psig	*ΔP/10 miles*	P1 = 850 psig	*ΔP/10 miles*
10	866.9	*33.1*	814.5	*35.5*
20	832.2	*34.7*	777.2	*37.3*
30	795.8	*36.4*	737.7	*39.5*
40	757.5	*38.3*	695.7	*42.0*
50	716.7	*40.8*	650.6	*45.1*
60	673.2	*43.5*	601.8	*48.8*
70	626.4	*46.8*	548.2	*53.6*
80	575.3	*51.1*	488.2	*60.0*

Table 1 - Pressure along a horizontal pipeline

The increasing pressure loss per section of pipeline, a function of the compressibility of the fluid and the flowing velocity, which in turn affects, as we have seen earlier, the friction calculations, has a direct impact on the linepack volume within the system. The linepack of a gas pipeline is best thought of as the capacitance of the system. As we move on to transient considerations consideration of this capacitance becomes of supreme importance.

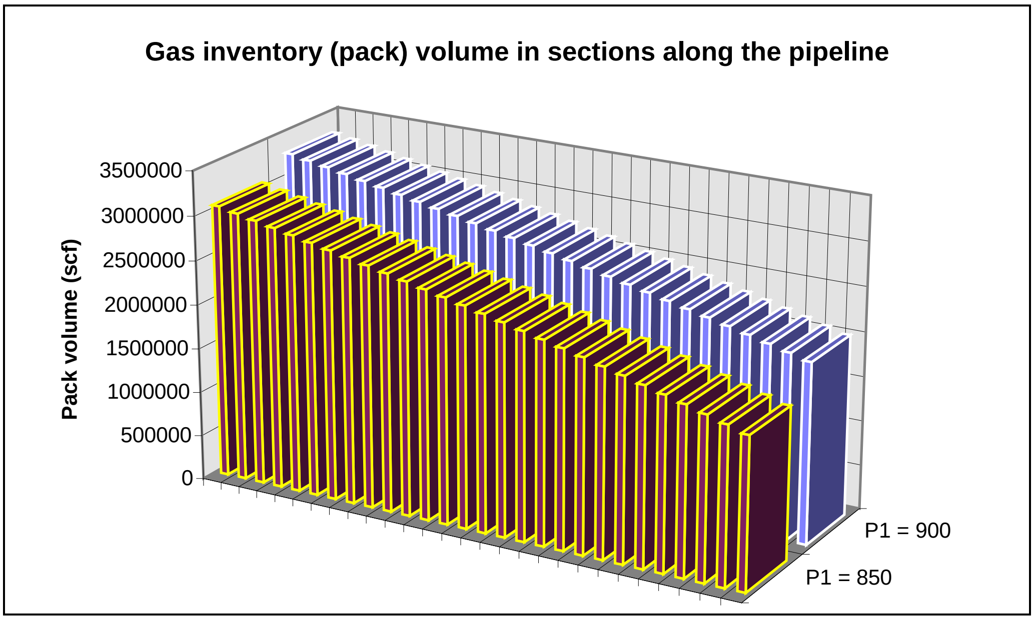


Figure 3 - Gas volume in along the pipeline, under different initial boundaries

Figure 3 shows the volume of gas in the pipeline. The pipeline has, for the purposes of this graphic, been segregated into several discreet sections, and the volume of gas in each section is displayed. As can be seen, there exists a distinct volume of gas in each segment, which is a function of the average pressure in each section, the flow through the section (constant for each section in this steady-state analysis), and the physics of the gas itself.

This is not all useable inventory. In other words, there may or may not be substantial capacitance within the pipeline, despite the volume stored in what we are here referring to as the 'static inventory'. The gas pipeline operator must be aware of the operational/capacitance characteristics of the pipeline -- there are not fixed and general rules, each pipeline has unique characteristics.

INDUCING AND ANALYZING TRANSIENT BEHAVIOR

To get an initial sense of the concept of transients, we will assume that the customer at the end of the pipeline increases load from 325 mmcfd to 455 mmcfd very rapidly -- over a 10 minute period, hold this increased flow for 4 hours, then return to the original flow value.

First, we model the pipeline in steady-state, placing a load of 455 mmcfd on the end of the pipeline. Analysis tells us that the pressure at the end of the line will be -14.73 psig, absolute 0. If we revise our equation, setting 0 psig at the delivery point and solving for the system flow, we find that the maximum steady-state flow that can be maintained is 421 mmcfd -- and that results in 0 gauge pressure at the end of the pipeline. Thus, steady-state analysis tells us that this additional load is not feasible -- the system cannot support this increased load.

So, even before we commence the transient analysis, we know that any load increase of this magnitude will necessarily be short-lived. Our pipeline simply cannot sustain long-term steady flow of this magnitude. However, can it support this load for a 4-hour period?

When a transient analysis is made, we find the following result:

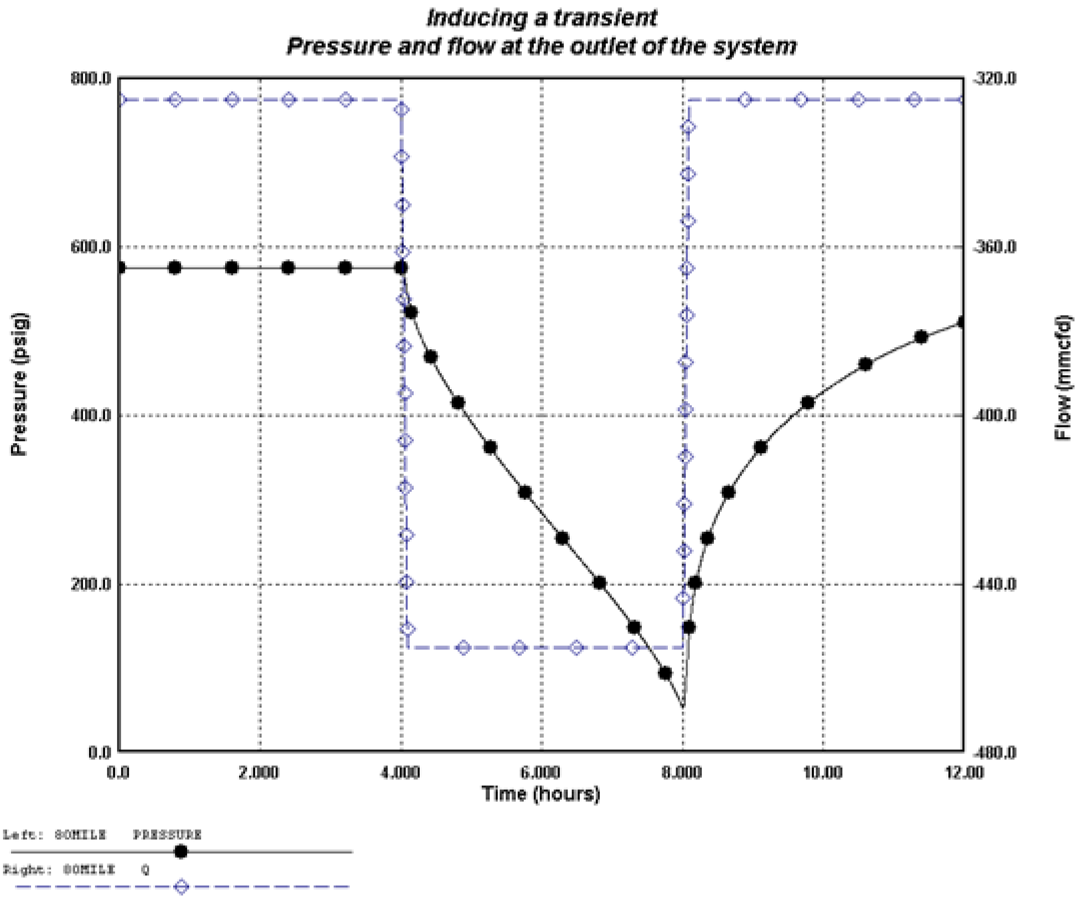


Figure 4 - Pressure (left axis) and Flow (right axis) at the end of the pipeline

Transient analysis of the system shows that we can support the load (Figure 4). The minimum pressure that is seen on the system is approximately 75 psig. Obviously, there is very little ‘leeway’ in our timing. The slope of the pressure decline curve is very steep, the system would have run out of pressure quite soon after 8 hours into the simulation.

Let us now see what effect his load increase has had on other sections of the pipeline. Below is shown a graph of pressure at 20, 40, 60, and 80 miles along our pipeline. For the next several graphs we have separated our pipeline into four sections:

Section	From	To
1	Origin	Milepost 20
2	Milepost 20	Milepost 40
3	Milepost 40	Milepost 60
4	Milepost 60	Milepost 80

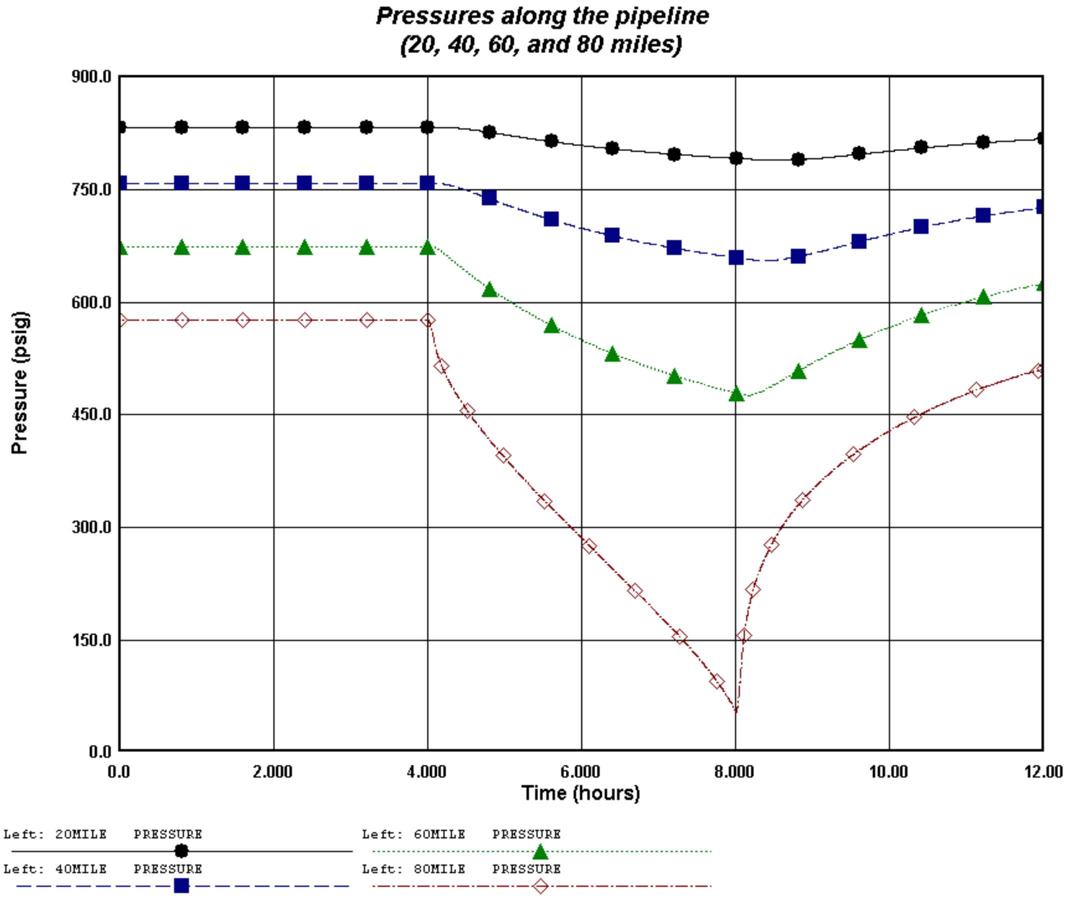


Figure 5 - Pressures at various points along the pipeline

Note that as we move upstream, the pressure decline effect is less pronounced. This is very common in transient analyses. Another common feature is flow variance along the pipeline. Recall that in a steady-state analysis we consider that the overall sum of the flows entering a system to equal those leaving ($\Sigma Q=0$). This is not true in transient analysis, as shown below:

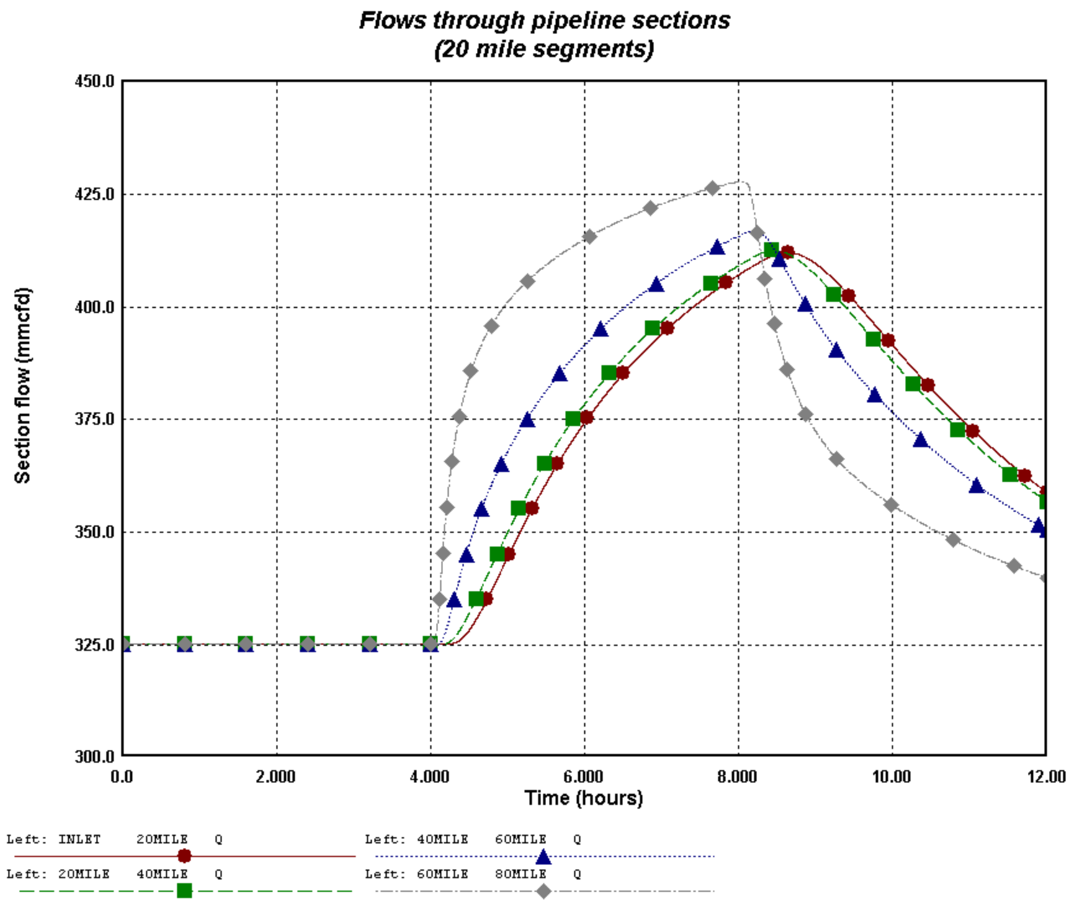


Figure 6 - Flows along the pipeline, shown in 20-mile segments

Our flow increased at 4 hours into the simulation, and the flow in section 4 of the pipeline (from 60 to 80 miles) began increasing almost immediately, then asymptotically began approaching the full load of 455 mmcfd. The next section, section 3, of the pipeline began to see flow increase some time later, followed by the section 2, then section 1. The flow changes in each section seem very evenly distributed.

There are a few very interesting facts to study regarding this graph. First, note that there is a distinct time lag between the response times of each segment. While this spacing seems very consistent at the beginning of response, the shapes of the response curves are different, as is their spacing. Further, note that the peak flow reached in each segment differs, and in fact none of the segments ever reach the demanded flow of 455 mmcfd.

Similarly, after the flow returns to its base value at 8 hours of the simulation, there are similar response differences along the pipeline sections as they recover. These are classic examples of pipeline drafting (withdrawing from pack), and repacking, and are the issues with which the pipeline operations staff must deal daily.

While it seems trivial to discuss, it is of vital importance to the understanding of natural gas pipeline operations. The pipeline segments closer to the load exhibit substantial flow changes, with their associated increased velocities and frictional losses, than do the upstream pipes. After the load ‘shuts-off’, all sections begin returning to their original, steady-state, conditions. Note that in our model, even though the load ends at 8 hours into the simulation, all of the sections of the system take substantially longer to apparently return to their initial conditions. In fact, further simulation shows that the system does not fully return to its stable, steady conditions until approximately 18.25 hours into the simulation, over 10 hours after the load returned to normal.

A GAS CONTROLLER OPERATING PROBLEM

We will modify our initial conditions somewhat. Let us assume that we have the same basic 80-mile long, 24-inch diameter pipeline. We will assume that our basic inlet pressure is 750 psig, and the system maximum allowable operating pressure is 900 psig. The controller can, based upon system operating parameters, increase the pressure at the inlet at his or her discretion, in order to satisfy downstream demands.

The power plant at the end of the pipeline has a base load of 300 mmcfd. At steady-state conditions, the pressure at the terminus of the pipeline is 261.8 psig, determined from Equation 2, presented earlier in this paper.

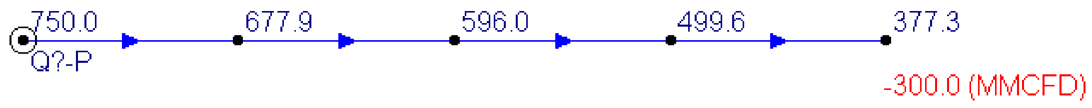


Figure 7 - Gas Controller Problem -- Steady-State Conditions

Let us presume that the power plant consists of 4 turbine generation units, each of which consumes gas at a rate 100 mmcfd at peak load. The minimum operating pressure of the plant is 250 psig. Three of the units operate continuously, the fourth is used for peak-shaving, and runs only intermittently.

Can our pipeline provide continuous service to all four units? Application of Equation 2 shows that the answer is both yes and no. If we maintain an inlet of 750 psig, the answer is no:

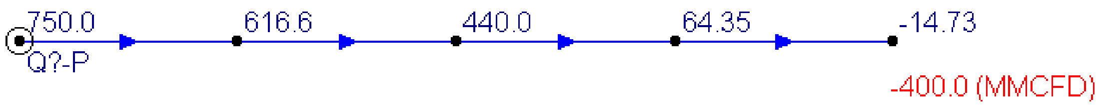


Figure 2 - Gas Controller Problem -- Full Turbine Load with 750 psig Inlet

If, however, the inlet pressure is raised to 900 psig, we see that steady-state computations yield:

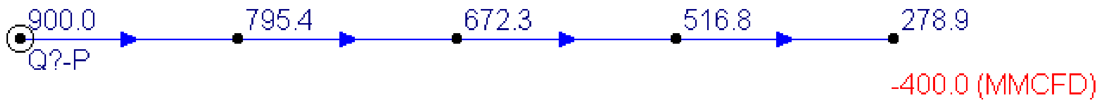


Figure 3 - Gas Controller Problem -- Full Turbine Load with 900 psig Inlet

Clearly we can provide the full 400 mmcfd to this plant by simply increasing the inlet pressure to 900 psig. A fine idea, this is not particularly attractive to the pipeline company. To increase the pressure at the source from 750 psig to 900 psig requires the addition of some energy -- typically by starting a compressor, or a series of compressors. There are two highly negative implications to running compression:

- it burns fuel, which adds to the pipeline operating costs, and
- it increases the NO_x emissions, which are pretty tightly regulated in most states.

We know from previous examples that the pipeline operator cannot simply 'crank up the compressor' and expect that the increased pressure will be felt downstream immediately -- it takes time to pack the system. Given the wavespeed of natural gas, about 1200 feet per second, we can anticipate that the amount of time required for the downstream end to possibly see any positive effects of compression is at least (80 miles x 5280 feet/mile)/1200 = 5.86 minutes, and any pipeline operator will tell you that it will be considerably longer -- pipelines do not pack at the wavespeed.

What the operator must do is determine, or know, the hydraulic characteristics of his or her pipeline. For example, consider the following plot -- a transient analysis has been run on the pipeline, showing the pressure response of the system, given our initial 750 psig inlet pressure, and assuming the turbine load starts one hour into the simulation. We know, from earlier calculations, that the system cannot provide continuous steady-state operation without increasing the inlet pressure, but want to consider the characteristics:

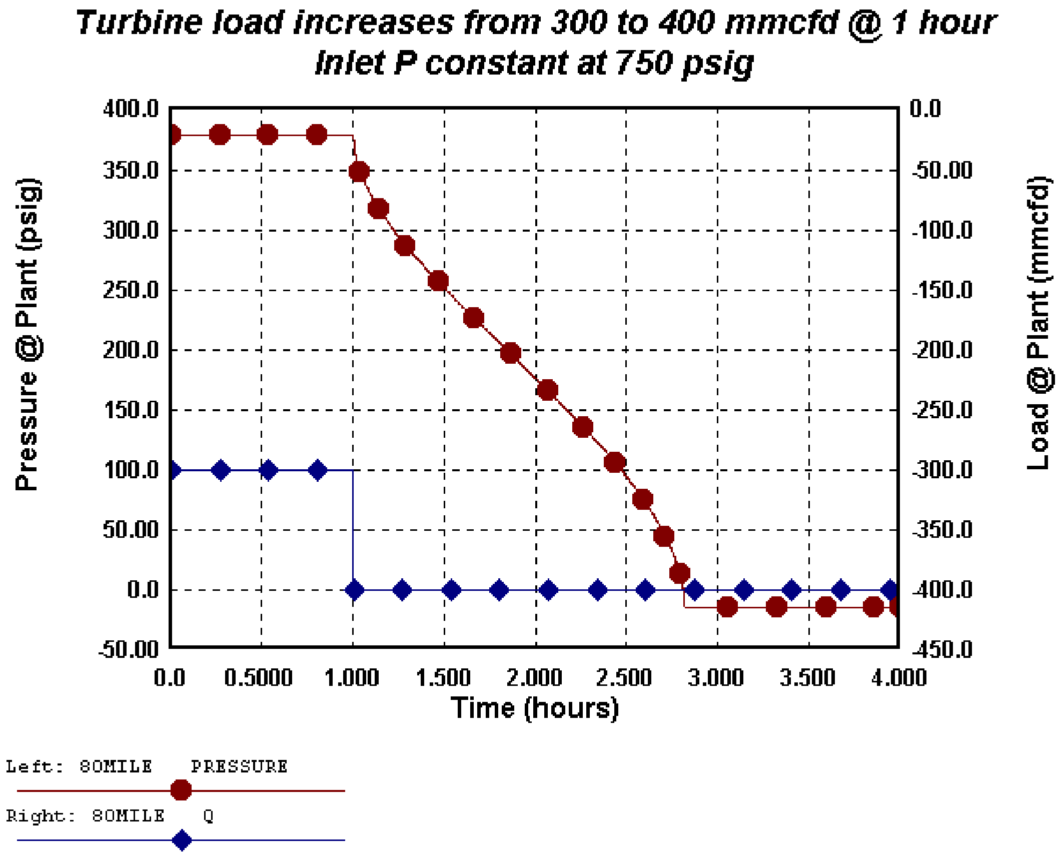


Figure 10 - Gas Controller Problem -- Transient Analysis of Full Turbine Load with 750 psig Inlet

A few very important factors in our pipeline are immediately apparent. Even though the load increases at t=1 hour, the plant does not see a complete loss of pressure until approximately 2.8 hours. If the minimum plant operating pressure is 250 psig, we see that we can feed this plant for approximately 1/2 hour from the time the load increases before this pressure minimum is reached. There is clearly substantial line pack in this pipeline, evidenced by the 1.8 hour time difference between load increase and complete zero pressure. Equally clearly, this load can be supported by line pack, but only for a short time.

Let us now add to the problem somewhat. We will assume that the power plant's fourth turbine will come on at 6:00 am (6 hours into the simulation, assuming that the simulation starts at midnight), will be on for 2 hours, then comes back on at 6:00 p.m., and remains on for 3 hours.

Given this scheduling, at what times do we need to run our compressors to adequately support the load?

Planning in an off-line mode requires a number of scenario analyses. While going through the entire process is far beyond the scope of this paper, we will show some of the interim results. Since there is some pack capability in the system, we will simply turn the compressor on concurrently with the load increases:

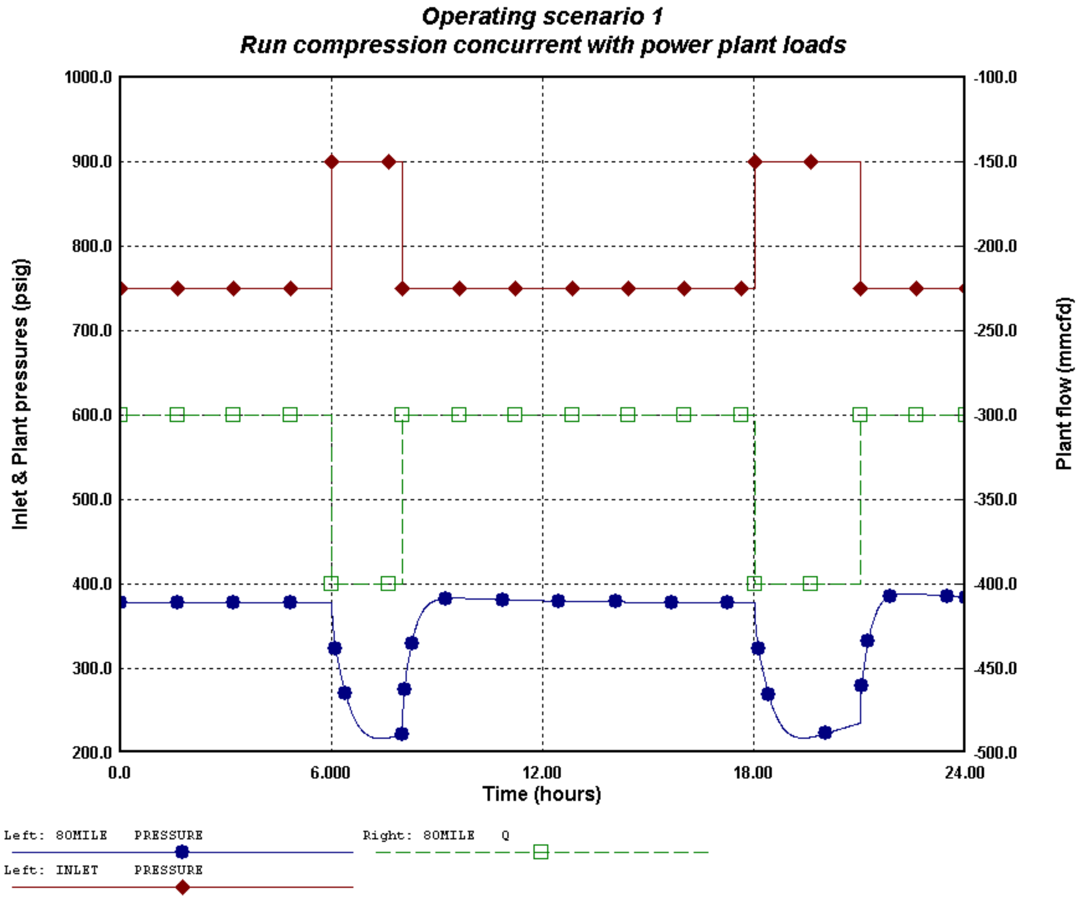


Figure 4 - Gas Controller Problem -- Operating Compressors with Power Plant Flows

We see from this graph that the system is very close to adequate operation simply by running the compressors in conjunction with the load increases. Now, we try starting the compressors a bit earlier in the morning, say at 5.5 hours into the simulation –

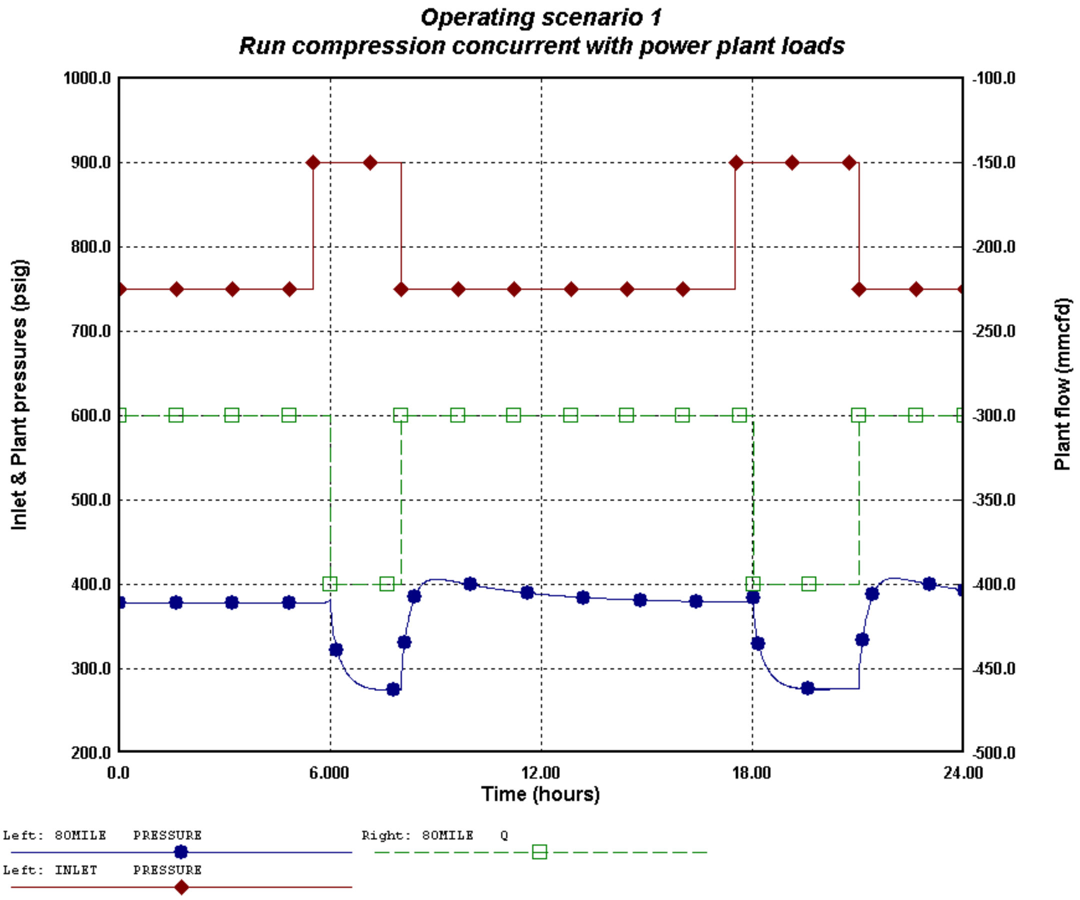


Figure 5 - Attempt a different compressor start time

We have found, therefore, a reasonable scenario for a 1-day case. To verify operation, the pipeline operator will typically test on a 48, 72, or even 96-hour basis. Even though our system pressures do not go below minimum, it is important that we consider the overall impact this load will have on our system pack volume. System pack is a variable commodity. Loads such as these will impact the pack, and in many cases we find that a continuous operation will show acceptable pressures, however, there will be a clear decline in system pack.

To coin a phrase, when this occurs our pipeline is '*playing a losing game*'. While our system pressures may be adequate on days 1, 2, and even 3 or beyond, an overall downward trend in pack volume results in steady system pressure losses. At some point in time we will find that when our fourth turbine begins loading, it will trip due to low pressure, an event neither the pipeline operator nor the electric power utility enjoys. Our pack for this scenario is shown below:

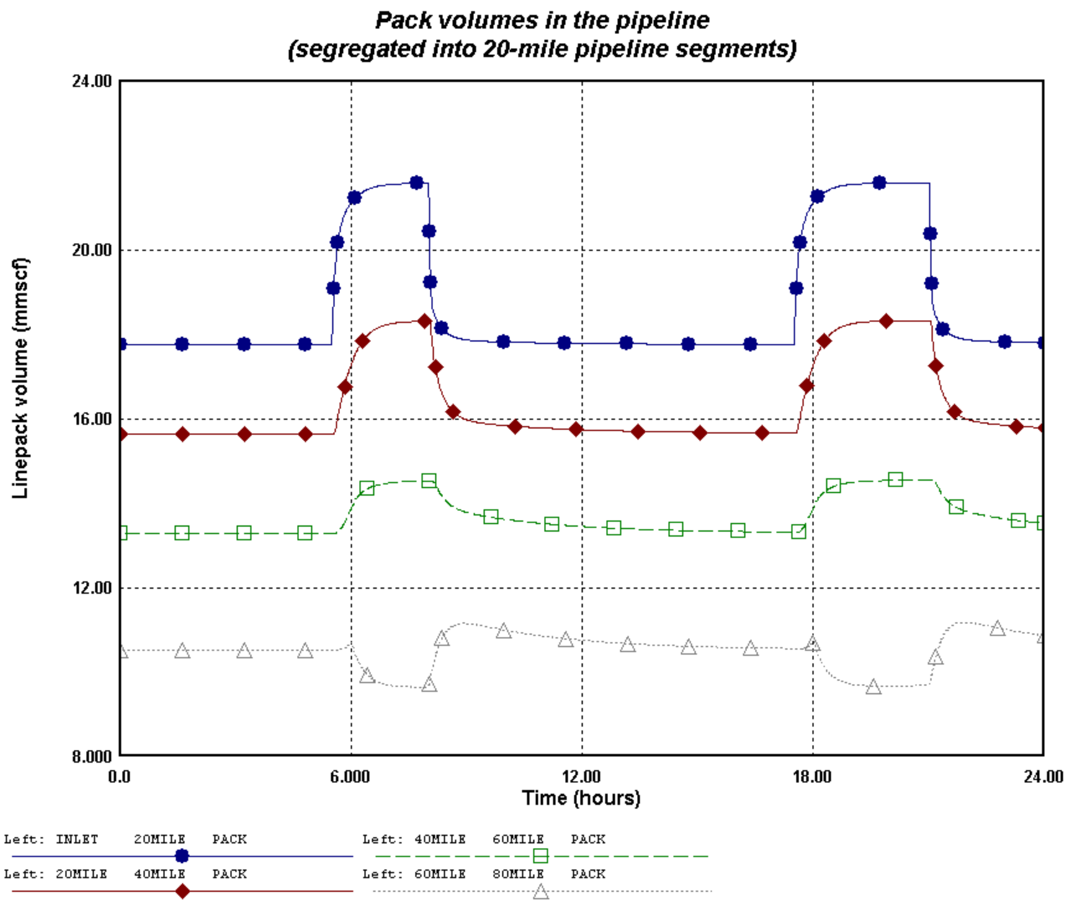


Figure 6 - System pack volume response along the pipeline

We see from this graph that the pack in each of our 20-mile pipeline segments returns to at least its original value at the end of the simulation. This implies that, over a long period of time, we will continuously increase the system static volume, and our electrical utility customer will experience no pressure problems. In fact, at some point in the future, we will have to guard against overpressuring the pipeline.

From this exercise our hypothetical pipeline operator has learned that the power plant must provide at least sufficient advance notice of intent to start the fourth turbine to allow the pipeline's compressors to start and "come up to full speed" at least 30 minutes before the fourth generator starts. Any less time would result in a low-pressure trip of the power turbine. This phenomenon is the reason why gas supply contracts typically include notification time clauses.

While our power plant customer may well be quite pleased to learn that they must provide only 30 minutes notice, this is a rather unrealistic example. The inlet pressure to the system would be, in a real pipeline, most likely controlled by a compressor station. Compressors are very capital- and maintenance-intensive units. Generally, they cannot be cold-started immediately, and require (depending upon the type of compressor) up to a one hour lead time for warming-up and loading.

Further, maintenance costs are very high, and compressors generally incur maintenance charges directly proportional to the number of hours they run. It logically follows, then, that starting and stopping a compressor, as we did in this example, would greatly benefit the pipeline company, resulting in reduced maintenance costs. This may be logical, but it is not true. Statistically, simply starting a compressor is roughly equivalent to 4 to 8 hours of operating time. Thus, for cost effective operation, pipeline operators are loathe to frequently start and stop their machines.

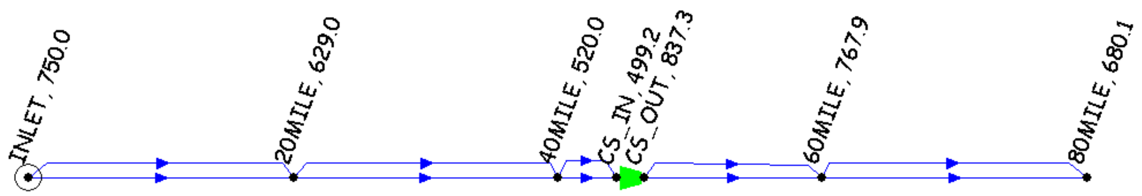
A MORE REALISTIC EXAMPLE

The previous example problem may have served to illustrate, for those fundamentally unfamiliar with the operation of natural gas pipelines, some of the operational problems that the operators, or dispatchers, face each day. To those who do, in fact, deal with these problems on a fairly regular basis, the previous example was almost trivially easy. The model was kept simple for the sake of clarity and understanding.

It is extremely unlikely to find an 80-mile x 24" transmission pipeline devoted solely to providing gas to one customer. It is highly more probable that there are other users, Local Distribution Companies (LDCs), for example, that also are served by the pipeline.

Further, it is also highly likely that in a real pipeline system there is at *least* one compressor station that serves as a delivery resource. In transmission pipelines typical compressor stations support several machines of different capabilities.

In this section, we have expanded our simplistic model by adding loads along the pipeline, a mid-pipeline compressor station, and pipeline looping, as shown below:



At each of what we earlier referred to as the '20-mile' points we have added different loading conditions 'on top of' the power plant loads we considered earlier:

Milepost	**Loading conditions**
20 miles	An industrial load, with 'ramping' loads varying from 100 to 175 mmcfd during the day. This would be similar to any number of industrial manufacturing applications.

Milepost	Loading conditions
40 miles	A residential, LDC, type loading. Residential loading profiles are typically diurnal, with loads peaking at about 6:30 in the morning, with a secondary peak in late afternoon. The profile used is a very typical residential profile, with a 5% peak hour factor. The average hourly flow is 75 mmcfd.
45 miles	A 12,500 hp compressor station is added, with a maximum discharge pressure of 900 psig, the pipeline MAOP (Maximum Allowable Operating Pressure).
60 miles	A widely-varying industrial load profile, with step-changes in the loads, varying from a low of 30 mmcfd up to 225 mmcfd. Typical of any industrial concern that has large gas 'heats' -- glass plants, asphalt plants, fertilizer plants, brick plants, etc.
80 miles	In addition to our base 3-turbine load and 4th turbine peaking, we have added 125 mmcfd of residential load and 90 mmcfd of industrial load.

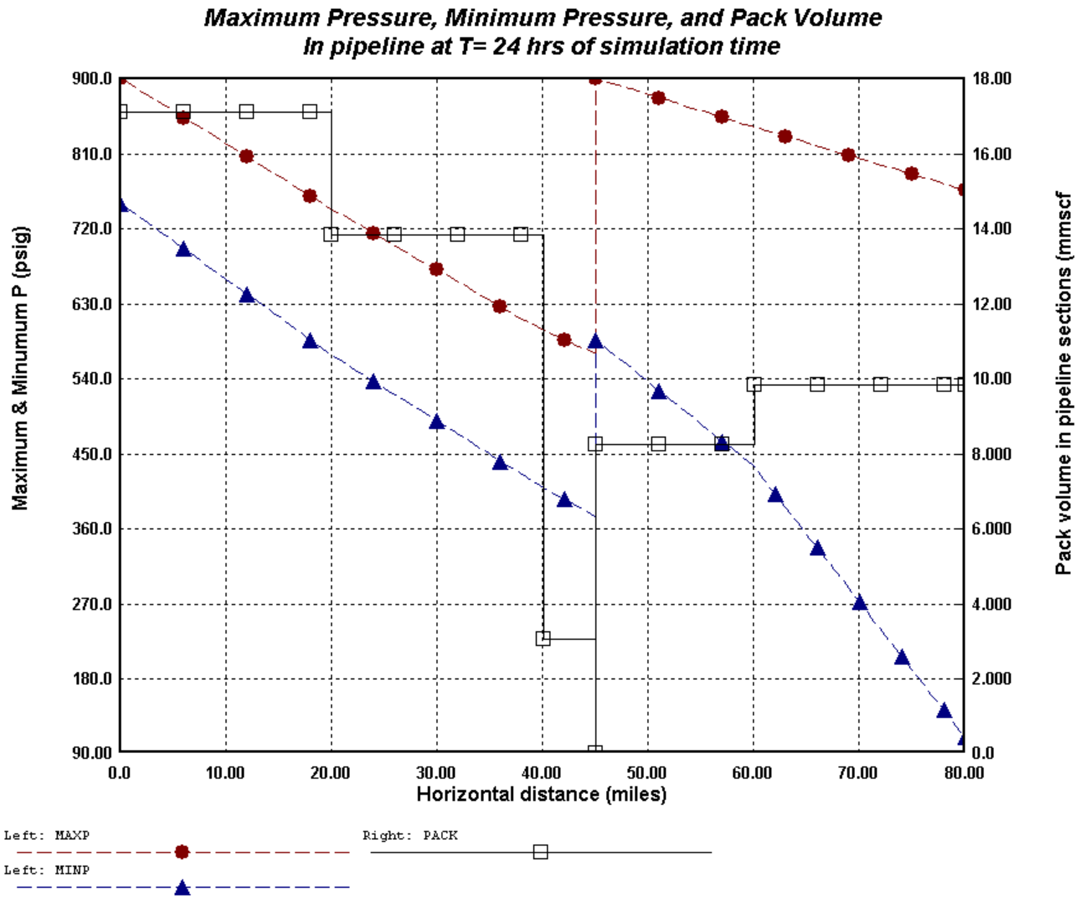


Figure 7 - Maximum and minimum pressures and pack volume for the more complex problem

The effects of these load changes on the model are viewed in the graph shown above (Figure 14), which is a distance graph showing the maximum and minimum pressures seen throughout the pipeline during the simulation, and the one shown below, which is the graph of pressure and flow at the power plant. Our flow and pressure patterns are now just a bit more complex:

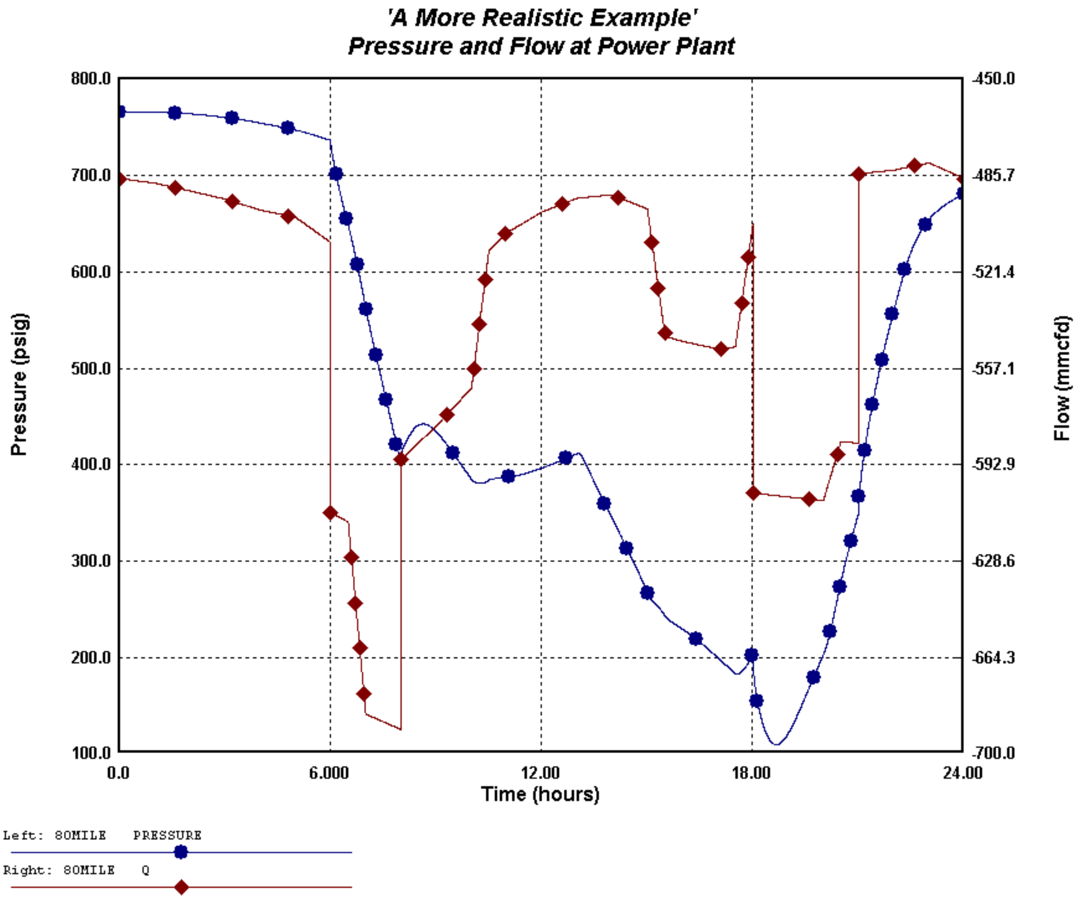


Figure 8 - Complex load interaction and resultant pressures at power plant

Both graphs show that our operator is faced with a problem. If the power plant requires a minimum of 250 psig to operate efficiently, we will lose at least a portion of the system load in the evening. We are still, however, maintaining the 750 psig at the initial source to the system, and increasing it to 900 psig concurrent with the operation of the fourth turbine at the power plant.

What happens if we specify that the additional compressor(s) needed to create this pressure increase run continuously, holding a full 900 psig MAOP at the source?

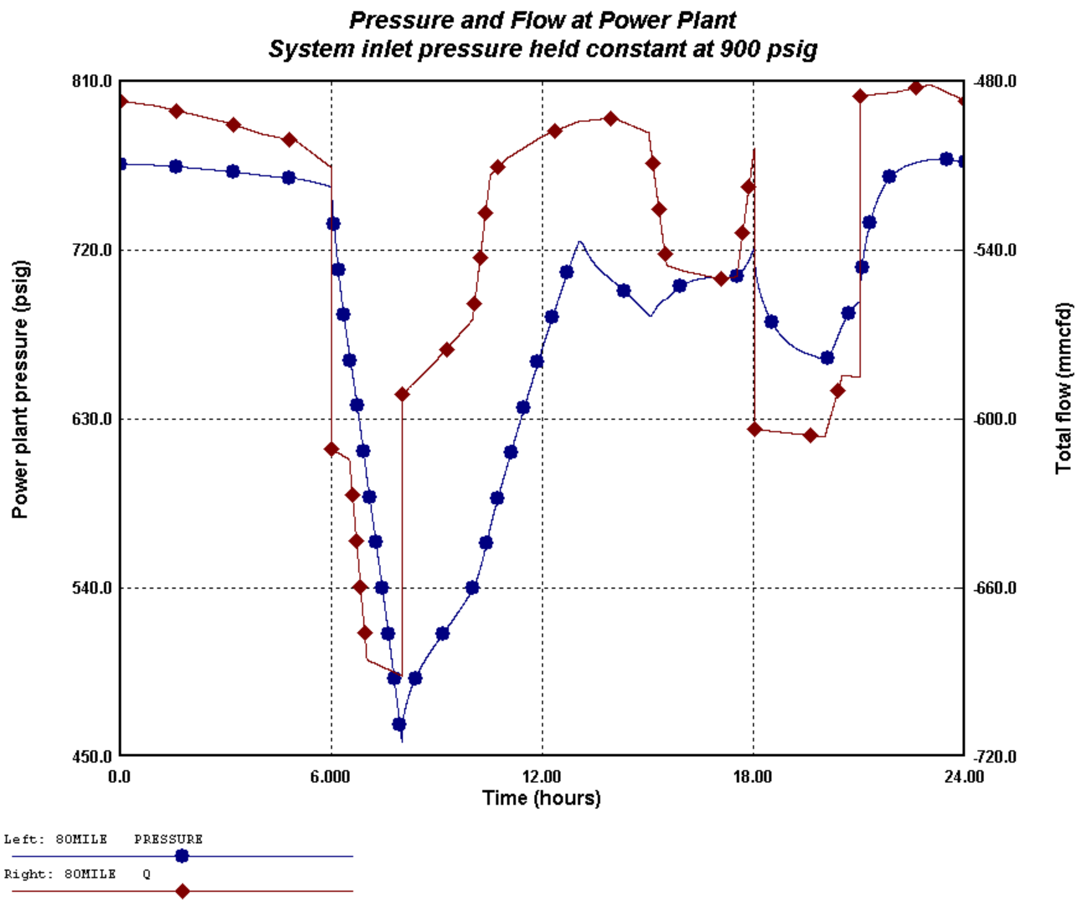


Figure 9 - Pressures and flow changes due to increases source pressure

In fact, it made a tremendous difference. Why? The main reason is due to operation of the compressor at mile post 50. In our original scenario, the compressor reached a maximum power limitation much earlier in the run:

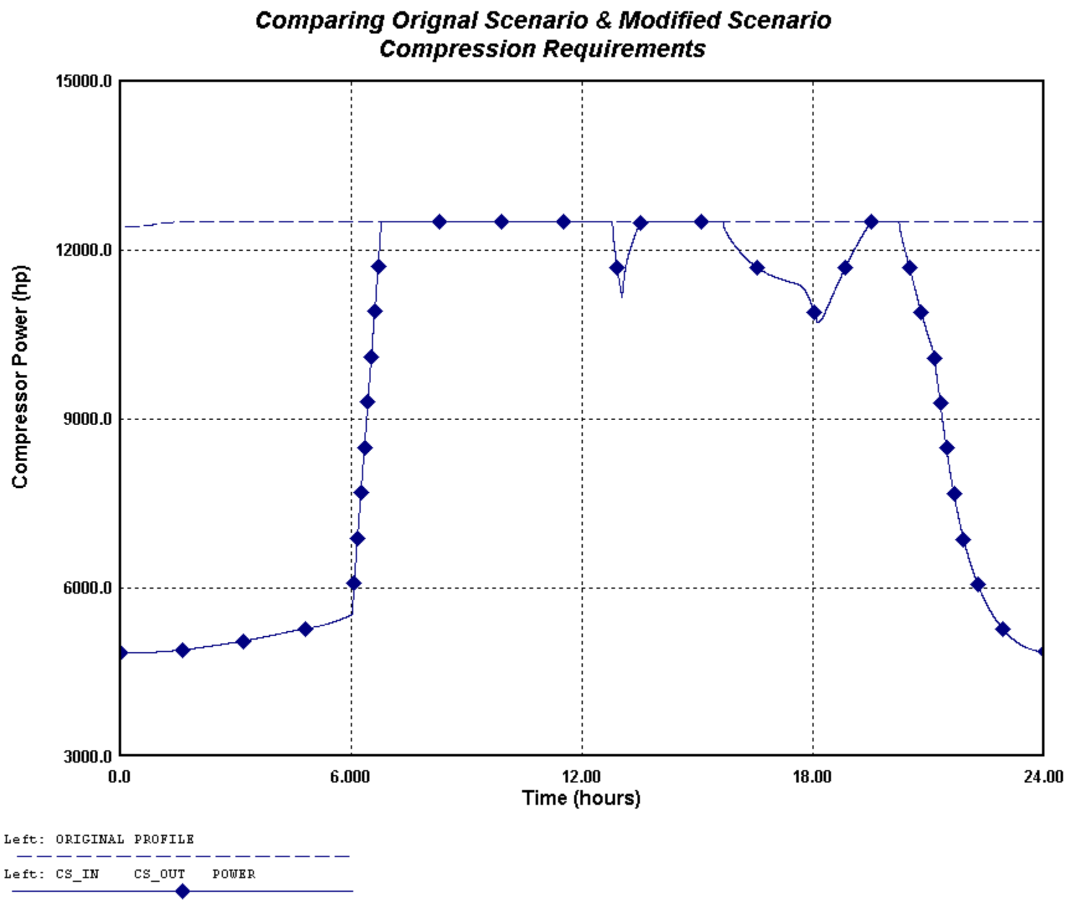


Figure 10 - Original scenario compressor operation

In our modified scenario, however, we began the source compressor at time 0, which in turn served to increase the linepack upstream of the compressor, thus lowering the compression ratio required, and, in turn, reducing the period of maximum power. At any time, there are several constraints imposed upon a gas compressor unit -- maximum discharge pressure, minimum suction pressure, maximum power, minimum power, flow capacity, maximum compression ratio, minimum compression ratio, etc. Further, for the most common type of transmission system compressor, the centrifugal compressor unit, power is not a simple limit. The available compressor power is reduced, or de-rated, by the ambient temperature. In addition, this de-rated power is further modified by the fact that for a centrifugal compressor available power, for a given temperature, is a function of flow *and* ratio. Thus, as the flow rate changes the available power similarly changes.

Natural gas pipeline operations and calculations must incorporate these limitations and reductions into their operational plans. A simple computation of horsepower is seldom, in and of itself, sufficient.

In our modified scenario, we increased the pack volume upstream of the 50-mile compressor. To the unindoctrinated, this may seem irrelevant. This increase, however, serves to increase the suction pressure to the compressor unit, thus lowering the required compression ratio (P_2/P_1). Even though compressor power availability is a rather complex function, there is a direct correlation between ratio and power -- the higher the ratio the higher the power required.

When a compressor reaches its power limit, something has to give. That something, for the majority of cases, is the discharge pressure. Thus, in our original scenario, the midpoint compressor hit its power maximum early in the modeled period, and the suction pressure dropped. When this happened, the pack in the pipeline downstream of the was reduced, because of lower pressures, and as the load increased there was simply not enough pack in this section of the pipeline to sustain operations. At a lower pressure there are fewer molecules of gas in a given cubic foot (due to density changes), thus, the increasing flow demands more rapidly ‘drain’ the pipeline.

As you can see from the graph below, the linepack differences are substantial. Our natural gas pipeline operator will be much more satisfied with the modified operating conditions. Note that the issue we discussed earlier, the return of linepack to its original value at the end of one load period (day) is satisfied in the modified scenario, as opposed to the original operating plan.

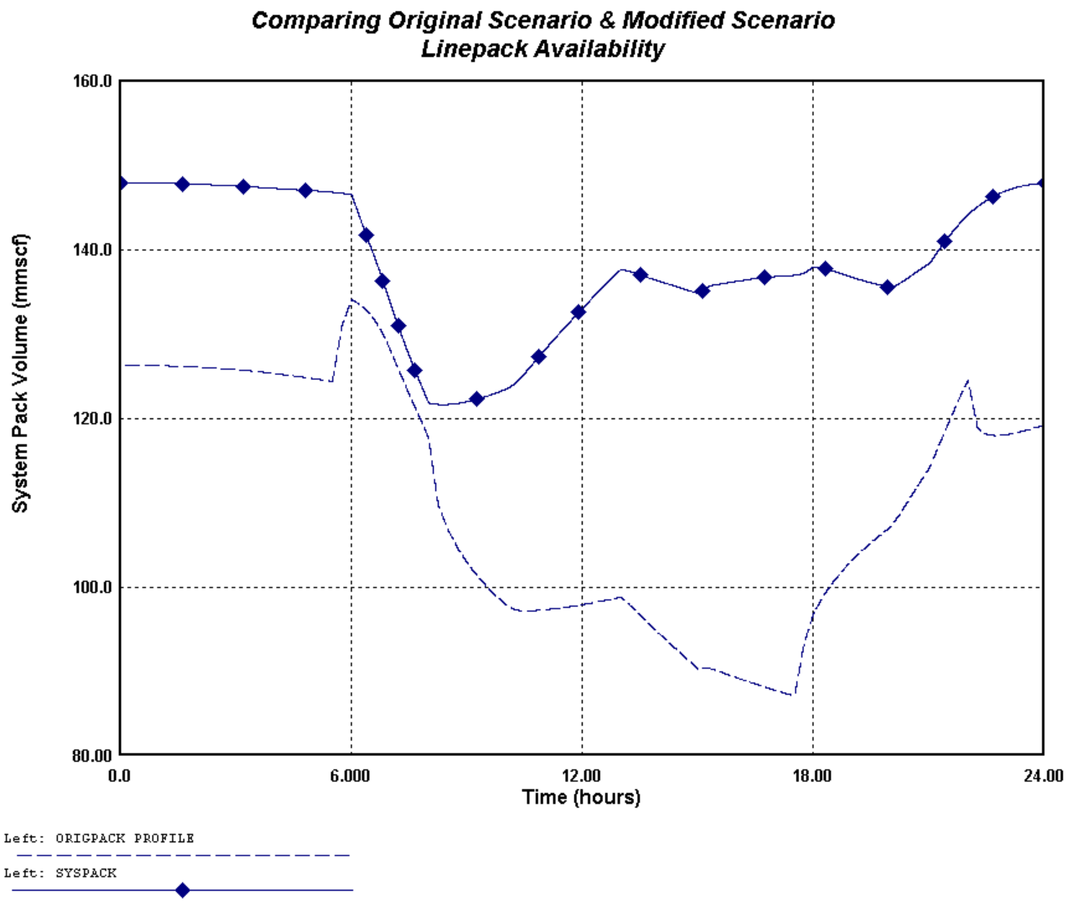


Figure 11 - Comparisons of linepack in the original and modified operating scenarios

CONCLUSIONS

This second example provides, hopefully, some insight into the task facing the transmission system controller. There are a number of computational aids to the controller, including off-line modeling tools (as were used in this paper), on-line simulation tools, telemetry, and the like. The controller, however, is ultimately responsible for handling the many different loads with different characteristics, the multiple source streams, each with its own contract maximum and minimum pressures, compression equipment with its unique idiosyncrasies, underground storage supply/injection -- and emergency situations. Consider the graph shown below. Now, imagine that this graph is being drawn dynamically on your computer screen, with pressure changing with time, and that you are the pipeline controller. At around 6 hours into your day the pressure suddenly begins to drop, very, very fast -- losing about 300 psig in approximately an hour. If you were not intimately familiar with the fact that there is an electric generation station that fires off at around 6 hours, and your system's response to it, and if you did not have the capability to, just in case, shut down that plant feed, you could indeed be a very worried controller.

Automated tools assist the controllers with scheduling, contract maintenance, regional supply issues, compressor operational constraints and availability, emission controls for exhaust gases, but as you can see from the graph, there is still a great deal of knowledge, and a sense of the system, involved.

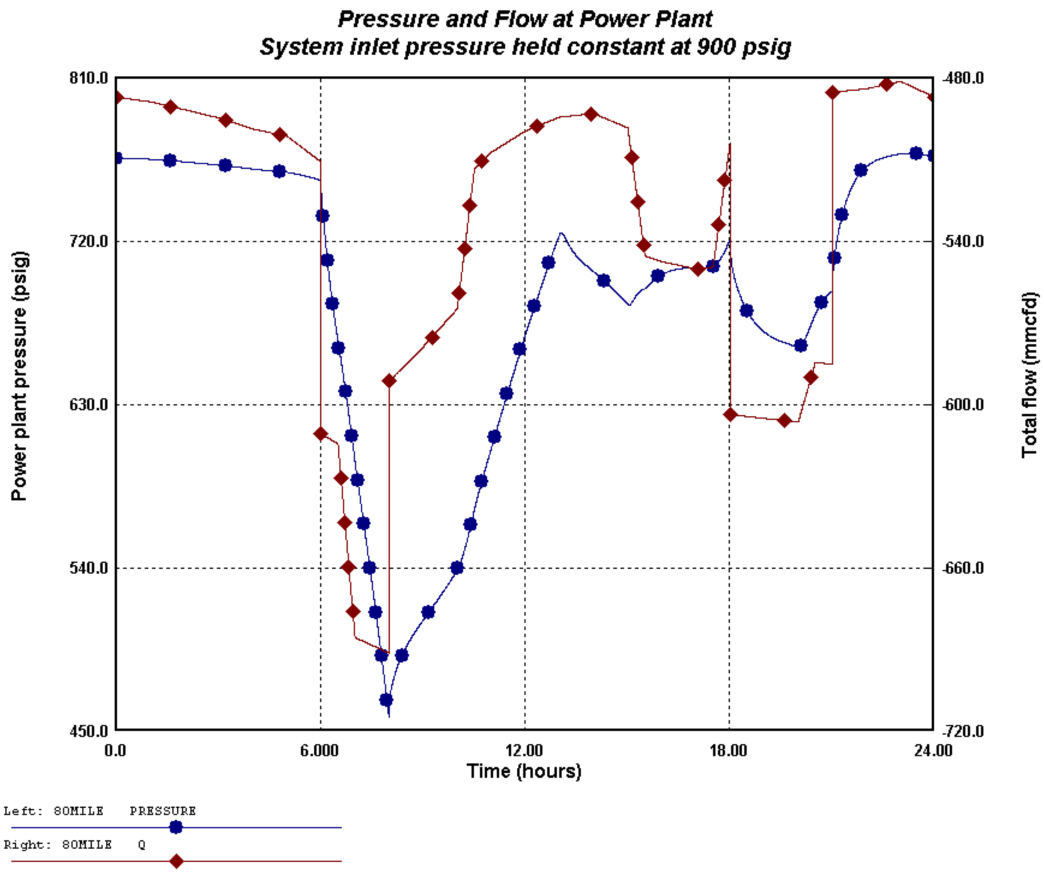


Figure 12 - Does the pressure drop so quickly due to a load, or a problem?

Successful gas control means meeting all these boundary conditions, while at the same time maintaining appropriate ‘cushions’ to meet unexpected loads or equipment problems, operating safely, and attempting to minimize waste. The advent of large loads, such as those for electric power generation which come and go quickly, increases the difficulty of operations, as we have discussed in this paper.

Making this more complex is the simple fact that many of the gas pipelines were designed before this type of loading became prevalent. Thus, demand characteristics that were not even contemplated during the design and construction are now placed on pipelines. It is in the best interest of gas consumers, especially power plant users, to recognize some of these fundamentals. Doing so provides a technical basis for recognizing the value of as much advance notice as possible of operating changes, making the planning and scheduling of gas services much more stable. Ultimately the resulting cooperation will make the gas transportation service, on which both gas and electric consumers depend, substantially more secure.

ACKNOWLEDGEMENTS

Rachford, H. H., Stoner Associates, Inc. “Tutorial on Transients in Gas Pipelines”

Schroeder, D. W., Stoner Associates, Inc. “The Steady-State Assumption”

Richwine, T. E., and Schroeder, D. W., Stoner Associates, Inc. “Transient Flow in Gas Pipelines, A Tutorial”

APPENDIX - EQUATION ABBREVIATIONS USED IN THIS PAPER

Equation 1:

ΔH	head loss due to friction	feet
f	friction factor	dimensionless
L	pipeline length	feet
D	pipeline inside diameter	feet
V	fluid flowing velocity	feet/second
g	gravitational constant	feet/second^2

Equations 2 and 6:

Q	pipeline flow rate	scfd
T_b	gas base temperature	^oR
P_b	gas base pressure	psia
D	pipeline diameter	inches
e	pipeline efficiency factor	dimensionless
P_1	upstream pressure	psia
P_2	downstream pressure	psia
G	gas specific gravity with respect to air (1.0)	dimensionless
T	gas flowing temperature	^oR
L	pipeline length	miles
Z	gas compressibility factor	dimensionless
f	pipeline friction factor	dimensionless
77.54	constant used for units correction	n/a

Equations 3 and 5:

f	pipeline friction factor	dimensionless
ε	pipeline roughness factor	inches
D	pipeline diameter	inches
Re	Reynold's number	miles

Equation 4:

V	gas flowing velocity	feet/second
D	pipeline diameter	feet
ρ	gas density	pounds/ft^3
μ	absolute (dynamic) viscosity	pound-sec/ft^2

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