Development and implementation of a multilevel high-pressure gas model

Penio Delev Penev

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DEVELOPMENT AND IMPLEMENTATION OF
A MULTILEVEL HIGH-PRESSURE
GAS MODEL

by

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Bachelor of Science
Technical University, Sofia, Bulgaria
1986

A thesis submitted in partial fulfillment
of the requirements for the

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ABSTRACT

Development and Implementation of a Multilevel High-Pressure Gas Model

by

Penio Delev Penev

Dr. Darrell Pepper, Examination Committee Chair
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The development of an accurate hydraulic model of a complex natural gas system is a great challenge for the engineers working to improve an existing system, determining a long-term system expansion or just performing every-day operational studies.

This thesis reviews the fundamental concepts of network analysis, equations, solution techniques, model types, and their applicability.

Further, the process of developing a multilevel model of a high-pressure natural gas system is described. Emphasis is given on the innovative concepts employed in the model development such as GIS integration, use of actual parameters for more accurate modeling of valves, regulators and compressors, derivation of load profiles based on multiple linear regression analyses, and data exchange with external applications.

Model characteristics such as response time and system imbalance are analyzed and quantified. The last part of the thesis describes the implementation of the model.
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CHAPTER 1

INTRODUCTION

The natural gas industry provides the cleanest fossil fuel to millions of customers. With the growth of the industry, the need for adequate and safe transmission and distribution pipeline systems increases. The complexity of those systems is increasing, as well, and it poses new challenges to the engineers responsible for planning, designing, installing, and maintaining natural gas pipeline systems. Increasingly advanced modeling techniques are being developed and employed in the process of analyzing the existing systems and designing new ones. The computational power of the modern computers allows for handling bigger and more complex models of pipe networks.

Various solution techniques are employed for steady-state, time-varying, and unsteady-state modeling. The available commercial modeling applications utilize a number of solution methods, e.g. method of characteristics, finite element method or other methods for solving huge systems of flow equations. Some techniques are applicable only for steady-state analyses, while others can solve also unsteady-state problems. A review of commonly used solution techniques and methods is presented further in this paper.

Some of the commercial modeling software packages provide direct interface to various data bases and systems, e.g. Geographic Information System (GIS), Supervisory Control and Data Acquisition (SCADA) System, and Customer Management System.
Further, even real-time data could be fed to a network simulation application and the model could be tuned to simulate real-time operation of an actual pipeline network.

In some cases, just a steady-state analysis is sufficient to solve a problem while, in other cases, an unsteady-state analysis is required. The determination of which approach to be used is based on a number of factors, such as system configuration, capacitance\(^1\) of the system, experience of the modeler, etc.

Usually, a steady-state analysis is performed on smaller systems with small pipe size and low pressure. Such a system has low capacitance, e.g. low ratio of usable line pack to system capacity. Therefore, no significant transient effects can be expected. Low and medium pressure distribution systems have such characteristics.

On the other hand, high-pressure distribution and transmission systems exhibit characteristics that imply unsteady-state operation. These systems require unsteady-state modeling in order to be analyzed accurately. High operational pressure and large pipe diameter, respectively large capacity, allow for larger flow fluctuations in the pipeline. In this case, the gas flow is far from a steady-state condition and it should be modeled and analyzed as an unsteady-state flow.

The paper reviews techniques employed in solving pipeline network, as well as, derivation of the governing equations used in both steady-state and unsteady-state models. A description of the Method of Characteristics with emphasis on its application to gas pipeline system modeling is provided.

The process of development of a model of an actual multilevel high-pressure feeder system is described. Attention is given to some innovative approaches in developing

\(^1\) The capacitance of the system could be defined as a ratio of the usable line pack to the total volume of the system.
model components such as schematic, load profile, and various Node Connecting Elements (NCE), e.g. regulator stations and compressor stations that are present in the actual system.

Two versions of the model were created, steady-state and unsteady-state. Commercial modeling software, SynerGEE®, was used to conduct the analyses of the system. The model data was prepared in a format appropriate for input to the software.

Some parameters of the model such as system imbalance and system response time were evaluated and presented.

A conclusion is drawn about the applicability of the employed techniques and approaches in the development of the model, and the results of the analyses conducted with the use of the model.

To facilitate the reader, some useful information such as unit conversion factors and glossary of terms commonly used in the natural gas industry is provided in the Appendix.

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2 SynerGEE® is a registered trade mark of Advantica, Inc. Information about the software can be found on Advantica’s web site at http://www.advantica.biz/stoner_software/synergee_gas/.
CHAPTER 2

GAS FLOW NETWORK

Classification of Natural Gas Systems

The natural gas is transported from the well to the consumer through pipeline systems with various pipe diameters and operating pressure.

Based on their operating pressure, gas systems can be classified as high-pressure, medium-pressure, and low-pressure systems. There is no clear-cut definition for what pressure is considered high, medium, or low. Transmission systems that transport gas for hundreds of miles usually operate at pressure of 500 to 1200 psig (3.4 to 8.2 MPa). Medium pressure systems, such as feeder systems that transport the gas from a transmission system to a distribution regulator station, generally operate at pressure from 60 to 720 psig (0.4 to 4.9 MPa). Low-pressure systems are usually distribution systems downstream of a district regulator station and operate at pressures below 60 psig (0.4 MPa).

Based on the service, the gas systems could be classified as gathering, transmission and distribution systems. The gathering systems transport the gas in the gathering field from wells to a central processing plant. There, the gas is separated from particulate matter, water, liquefiable hydrocarbons, sulfur compounds, carbon dioxide and other impurities.
Transmission systems move the gas from the gathering field to the local distribution companies (LDC) or large industrial consumers. The transmission systems are often hundreds of miles long and span across multiple states. The pipe diameter is often larger than 16 inches (400 mm) and interstate transmission pipelines may have diameters of up to 48 inches (1200 mm). A transmission system may include many compressor stations along the mainline. Many transmission systems have laterals that generally have smaller pipe sizes. There may be taps off of the transmission system that supply gas to a downstream distribution system. Transmission systems are operated by interstate or intrastate pipeline companies.

A gas distribution system consists of pipeline network carrying gas to the ultimate consumers from various sources of supply; city gate stations where gas is received from transmission pipelines; storage facilities and supplemental sources, if any. The distribution systems are operated by LDC.

The piping of a distribution system can be classified in five categories according to the Institute of Gas Technology:

1. **Supply mains** receive and carry gas from city gate stations to lower pressure distribution systems. Although, the gas pressure in a supply main is lower than the pressure in a transmission pipeline, it is higher than the pressure in most distribution mains. Supply mains may have a few high-pressure services, such as those serving large industrial customers directly connected to them.

2. **Feeder mains** supply gas from major sources, such as a regulator station fed by a supply main, to distribution mains. Feeder mains may also have services directly connected to them.
3. *Distribution mains* supply gas primarily to residential, commercial, and smaller industrial services.

4. *Service lines* deliver gas from a distribution main in the street to the customer’s meters.

5. *Fuel lines* are customer piping beyond the meter to appliances. They are property and responsibility of the building owner.

Many distribution systems consist of several superimposed networks of mains operated at different pressure levels. Figure 1 is a schematic of a distribution system showing the typical operating pressures of the different segments of the system.

![Figure 1. Schematic diagram of typical natural gas distribution system.](image)

The pipelines are obviously the major component of the natural gas systems. However, there are many other components, such as compressor stations, pressure limiting stations, metering stations, taps, city gates, regulator stations, valves, filters,
storage fields, liquefied natural gas (LNG) facilities, MSA, etc. Many of those facilities include some of the other elements, e.g. a city gate may have a filter, metering equipment, pressure regulating equipment, flow regulating equipment, various types of valves and telemetry that is part of the SCADA system.

Gas Pipeline Networks

The fundamental building blocks of a network are its elements. Elements, also called node-connecting elements (NCE), are the facilities through which fluid flows. Facilities may include pipes, regulators, valves, storage fields, and so on. An element is defined by two nodes representing its endpoints. A part of a gas network is shown on Fig. 2.

![Figure 2. Components of network.](image)

Hydraulic nodes are designated points in a piping system where facilities begin or end. System flows (supplies and demands) and external system pressures are controlled at the nodes. Elevation data, and reference supply fluid properties, including gas specific gravity, heat content, temperature, and compositional data are prescribed at the nodes.
Graphic nodes, unlike hydraulic nodes, have no hydraulic properties associated with them and do not affect the solution. They are used only to define the geometry of the element.

Connectivity refers to the ability to trace the flow of a fluid from one element to another. Two or more elements are connected to each other when they have a common node.

Supply is a flow into the system. The node at which that flow enters the network is called supply node or source. Load is a flow exiting the system.

The equation of the flow in an element can be expressed in terms of flow as a function of the pressure at the two nodes (from-node and to-node) connected by the element, and constants and variables unique to the equation as follows:

$$Q = f(P_1, P_2)$$ \hfill (2.1)

Here $P_1$ is the upstream pressure at the from-node and $P_2$ is the downstream pressure at the to-node.

Kirchhoff's First Law

Kirchhoff's First Law is derived from the Law of Conservation of Energy. Analogically to the electrical currents, the sum of the flows entering the node is equal to the sum of the flows exiting the node. In other words, the flow into or out of a node in a network must sum to zero in order for the mass to be conserved.

The mathematical relationship, for elements adjacent to a node, is expressed as follows:

$$\sum_{i=1}^{n} Q_i = 0$$ \hfill (2.2)
Here $Q_i$ is the flow in the $i$-th element adjacent to the node, a supply, or a load. The number of the flows entering and exiting the node, $n$, is a sum of the number of element flows and the number of supplies and loads. Illustration of Kirchhoff’s First Law is shown in Fig. 3. In this example, there are three elements adjacent to the node with three element flows, $Q_1$, $Q_2$, and $Q_3$. Also, there is a load, $Q_4$, exiting the node.

Figure 3. Kirchhoff’s First Law.

In the sum (2.2), the flows into the node are taken with positive (+) sign and the flows out of the node are taken with negative (−) sign. Kirchhoff’s First Law, applied for node $N$ yields:

$$\sum_{i=1}^{4} Q_i = Q_1 + Q_2 - Q_3 - Q_4 = 0$$

(2.3)

Node Temperature Equation

Node temperature calculation is based on the assumption of ideal mass mixing of all flows, in and out of the node, including any external ones. A node temperature equation is needed when the heat transfer model is not isothermal; for isothermal assumption, this
equation may be ignored [2]. The following equation may be used to calculate the node temperature

\[
T_j = \frac{\sum_{i=1}^{n_i} Q_i T_{io} + F_j T_{ji}}{\sum_{i=1}^{n_i} Q_i + F_j}
\]  

(2.4)

where \( T_j \) is node mixed temperature, \( Q_i \) is the \( i \)-th branch flow, \( T_{io} \) is the \( i \)-th branch exit temperature, \( F_j \) is node net inlet external supply flow, \( T_{ji} \) is node net external supply temperature and \( n_i \) is the number of the inlet flows.

![Figure 4. Node temperature.](image)

Considering Fig. 4 for example, the node temperature is expressed as

\[
T_j = \frac{Q_1 T_{1o} + Q_2 T_{2o} + F_j T_{ji}}{Q_1 + Q_2 + F_j}
\]  

(2.5)

Nodal Approach to Solving a Network

A set of equations representing the connectivity in the network can be established by applying Kirchhoff's First Law for each node in the network. This method is referred to
as the nodal approach to solving a network [1]. The mathematical relationship for the elements adjacent to node $j$ is expressed by the equation:

$$
\sum_{i=1}^{n_j} Q_i + F_j = 0, \ j=1,2,...N
$$

(2.6)

In Eq. (2.4), $j$ represents the $j$-th node, $Q_i$ ($1 \leq i \leq n_j$) are the flows in the elements adjacent to node $j$, $F_j$ is the node flow, $n_j$ is the number of elements connected at node $j$, $N$ is the number of the nodes in the network. Each $Q_i$ is a flow traveling from one node to another through an element $Q_{N_iN_j}$. The sign convention for $Q$ is that a flow is positive when traveling from the from-node to the to-node and negative if the flow is reversed. The $F_j$ represents the flow entering (+) or exiting (-) the system through node $j$.

For example, consider the system shown in Fig. 5.

![Figure 5. Gas network example.](image)

A system of linearly independent equations can be assembled. The system resulting from the network in Fig. 5 is
Each \( Q \) represents a flow equation that is a function of the element properties and node pressures.

Newton-Raphson Method

Steady flow problems are often nonlinear due to the nature of the convective terms or, less often, due to the dependence of the flow properties on the solution, e.g. viscosity on the temperature, typically. Newton’s method in several independent variables is a potentially useful technique for solving steady flow problems because of the rapid convergence when the initial approximation is close to the solution [3].

Newton’s method, also called Newton-Raphson iterative procedure can be utilized to find a solution of the network or solving the system of simultaneous equations like (2.7). The system can be solved for node pressures as functions of externally imposed system flows and used element flow equations.

Newton-Raphson method is a root finding iterative algorithm that uses the first few terms of the Taylor series of a function \( f(x) \) in the vicinity of a suspected root [4].

The Taylor series of \( f(x) \) about point \( x_0 \) is given by

\[
f(x) = f(x_0 + \varepsilon) = f(x_0) + f'(x_0)\varepsilon + \frac{1}{2} f''(x_0)\varepsilon^2 + \ldots \quad (2.8)
\]

Keeping only the terms up to the first order, Eq. (2.8) is rewritten as

\[
f(x_0 + \varepsilon) \approx f(x_0) + f'(x_0)\varepsilon . \quad (2.9)
\]
This expression can be used to estimate the amount of offset $\varepsilon$ needed to get closer to the root starting from an initial guess $x_0$. Setting $f(x_0 + \varepsilon) = 0$ and solving (2.9) for $\varepsilon = \varepsilon_0$ gives

$$
\varepsilon_0 = -\frac{f(x_0)}{f'(x_0)},
$$

(2.10)

which is the first-order adjustment to the root's position. By letting $x_i = x_0 + \varepsilon_0$, calculating a new $\varepsilon_1$, and so on, the process can be repeated until it converges to a root using the adjustment

$$
\varepsilon_k \approx -\frac{f(x_k)}{f'(x_k)},
$$

(2.11)

The algorithm is applied iteratively to obtain

$$
x_{k+1} = x_k - \frac{f(x_k)}{f'(x_k)}
$$

(2.12)

The error $\varepsilon_{k+1}$ after the $(k+1)$st iteration is given by

$$
\varepsilon_{k+1} = -\frac{f^*(x_{k-1})}{2f'(x_{k-1})} \varepsilon_k^2
$$

(2.13)

Therefore, when the method converges, it does so quadratically.

The Newton-Raphson method can be applied for multiple variables on problems requiring solving a system of equations that can be written as

$$
A(X)X = B,
$$

(2.14)

where $X$ is the vector of unknown nodal values, $A$ contains the algebraic coefficients arising from discretisation and, in general, may depend on the solution ($X$) itself. $B$ is made up of algebraic coefficients associated with discretisation and known values of $X$, e.g., given by the boundary conditions.
If (2.14) is written as

\[
R = A(X)X - B = 0, \tag{2.15}
\]

then Newton’s method can be written as

\[
X^{(k+1)} = X^{(k)} - (J^{(k)})^{-1}R^{(k)} \tag{2.16}
\]

or

\[
\Delta X^{(k+1)} = -(J^{(k)})^{-1}R^{(k)}, \tag{2.17}
\]

where \( k \) is the iteration level and \( J^{(k)} \) is the Jacobian. The elements of \( J^{(k)} \) are given by

\[
J^{(k)}_{ij} = \frac{\partial R_{i}}{\partial X^{(k)}_{j}}. \tag{2.18}
\]

Numerical differentiation is used to construct the Jacobian matrix by

\[
\frac{\partial R_{i}^{(k)}}{\partial X^{(k)}_{j}} = \frac{R_{i}^{k}(X_{j}^{k} + \Delta X_{j}^{k}) - R_{i}^{k}(X_{j}^{k})}{\Delta X_{j}^{k}}, \tag{2.19}
\]

where \( \Delta X_{j}^{k} \) is an incremental change of variable \( X_{j} \).

Equation (2.16) or (2.17) can be rewritten in the form

\[
J^{(k)}\Delta X^{(k+1)} = -R^{(k)}. \tag{2.20}
\]

Equation (2.20) is a linear system of equations that is to be solved for the correction vector \( \Delta X^{(k+1)} = X^{(k+1)} - X^{(k)} \) at each stage of the iteration. The method of Gauss-Seidel can be used to solve the system of linear equations.

An attractive feature of Newton’s method is that it demonstrates quadratic convergence if the current iteration \( X^{(k)} \) is sufficiently close to the actual solution \( X_{c} \).

Quadratic convergence implies \([3]\)

\[
\left\| X^{(k+1)} - X_{c} \right\| \approx \left\| X^{(k)} - X_{c} \right\|^2 \tag{2.21}
\]

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The matrix norms in (2.21) are maximum natural norms, e.g.

\[ \|X\| = \max_i \sum_{j=1}^{N} |J_{ij}|. \]  

(2.22)

In order to converge, the method requires initial values that are close to the solution. This is usually not a problem with the pipeline systems. The initial values for the pressures can be based on the operating pressure of the segment of the system. Each pipeline segment has a maximum operating pressure (MOP) and maximum allowable operating pressure (MAOP). The operating pressure is normally above the minimum allowable pressure and below the MOP of the system. MOP can be lower or equal to the MAOP. The MOP is set by operational constrains and the MAOP is imposed by the weakest links in the systems. These usually consist of the pressure ratings of components such as older or thinner-wall pipe, valves, fittings, flanges and other components.

Solution Technique

A flow chart of the solution procedure is presented in Fig. 6.

The first step in the solution process is to develop a node-element schematic of the system. Some simplifying assumptions can be made to reduce the number of elements and nodes. A detailed discussion of the considerations for the schematic preparation is given later on.

The second step requires development of momentum equation for each element and a mass balance equation for each node and this requirement determines the total number of equations that have to be solved. The total number of equations is

\[ N = N_{\text{Nodes}} + N_{\text{Elements}}. \]  

(2.23)
The number of the variables must be equal to the number of linearly independent equations in order to have a unique solution.

The solution method requires a set of known values, such as known nodal pressures or flows. Those values are the boundary conditions for the problem. Also, initial values for the unknown variables are required. As it was mentioned earlier, this requirement usually does not pose a problem for a pipeline system.
A converged solution of the network occurs when the sum of the squares of the mass and energy equations is less than a chosen tolerance value. A practical value for the tolerance is given by [2]

\[ \sum_{i=1}^{N} R_{ij}^2 \leq 10^{-3}. \]  \hspace{1cm} (2.24)

If the above criterion is not satisfied, a new iteration shall resume with the latest corrected set of variables.
CHAPTER 3

PIPE FLOW EQUATIONS

One of the first steps in the network solution technique is establishing a set of equations representing the flow through each element in the network. The equation would depend on the type of the element, i.e. pipe, valve, compressor, or storage. The majority of the node-connecting elements in a gas network are pipes with given internal diameter, length, roughness of the wall, and elevations at the two ends.

Fundamental Gas Flow Equation

If a steady-state solution is sought, the fundamental gas flow equation and variations of it is a prime choice. The fundamental equation presents the steady flow through a pipe as a function of the upstream and downstream pressures. For a given segment of the pipeline, the physical properties of the pipe, i.e. length, internal diameter and roughness of the wall are constants. The equation provides correction for elevation changes and gas compressibility. The fundamental equation is based on the energy conservation equation or Bernoulli’s equation with account for major head loss due to friction, given by Darcy-Weisbach equation.

Consider a segment of a circular pipe with steady flow of gas from point 1 to point 2, as shown in Fig. 7.
Bernoulli’s equation can be written as

$$\frac{P_1}{\rho_1} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_2} + \frac{V_2^2}{2g} + z_2 + h_f$$

(3.1)

where $h_f$ is the major head loss due to friction, given by Darcy-Weisbach equation

$$h_f = \int_0^L f \frac{V(x)^2}{2gd} dx,$$

(3.2)

where $f$ is the friction, $g$ is the acceleration due to gravity and $g = 32.174$ ft/s$^2 = 9.806$ m/s$^2$, $d$ is the internal pipe diameter, $L$ is the length of the pipe, and $V(x)$ is the gas velocity. For a steady, fully developed flow, the average velocity at a given location along the pipe can be expressed as

$$V = \frac{Q_a}{A},$$

(3.3)

where $Q_a$ is the actual volume flow rate and $A$ is the cross-sectional area of the pipe.

The volume flow rate varies with the pressure and temperature. However, the volume flow rate can be referred to some standard flow rate, based on standard conditions, such as temperature $T_b = 60^\circ F$ and pressure $P_b = 14.7$ psia. Therefore

$$Q_s = \frac{T_b P}{ZTP_b} Q_a$$

(3.4)
The supercompressibility factor $Z$ is a function of temperature and pressure. However, evaluating $Z$ at inlet and outlet conditions just complicates the computation without offering significant benefits. For simplicity, $Z$ is evaluated at some average conditions $P_{av}$ and $T_{av}$. For temperature, an arithmetic average of the flowing temperature is usually used, while the following equation, which accounts for the non-linearity of pressure drop with distance, is generally accepted for determining average pressure [5]:

$$
P_{av} = \frac{2}{3} \left( P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right)
$$

(3.5)

Assuming a steady-state flow, the mass flow entering the pipe and leaving the pipe is preserved. That is $Q_{mass}^{(1)} = Q_{mass}^{(2)}$, and therefore $A_1 V_1 \rho_1 = A_2 V_2 \rho_2$. However, the cross section is constant at any location of the pipe segment. Therefore,

$$
V_1 \rho_1 = V_2 \rho_2
$$

(3.6)

Plugging all of the above relations in (3.1) along with the gas laws equations modified for real gas, solving for $Q_s$ and regrouping yields

$$
Q_s = C_1 \frac{T_b}{P_b} d^{2.5} e \sqrt{ \frac{p_1^2 - p_2^2 - C_2 G (z_2 - z_1) P_{av}^2}{Z_{av} T_{av}} } e^{ - \frac{Z_{av} T_{av} f}{L G Z_{av} T_{av} f} } ,
$$

(3.7)

where

$C_1$ Constant, 77.54 (English units); 0.0011493 (Metric units)

$C_2$ Constant, 0.0375 (English units); 0.06835 (Metric units)

$d$ Internal pipe diameter (inches); (mm)

$e$ Pipe efficiency (dimensionless)

$f$ Darcy-Weisbach friction factor (dimensionless)
$G$  Gas specific gravity (dimensionless); 

\[
G = \frac{\text{molecular weight of gas}}{\text{molecular weight of air}} = \frac{M_w}{29}
\]

$L$  Pipe length (miles); (km)

$P_b$  Pressure base (psia); (kPa)

$P_i$  Inlet pressure (psia); (kPa)

$P_2$  Outlet pressure (psia); (kPa)

$Q_s$  Flow rate (standard cubic feet/day); (standard m³/day)

$T_{av}$  Average flowing temperature (°R); (K)

$T_b$  Temperature base (°R); (K)

$Z_{av}$  Compressibility factor (dimensionless)

$z_1$  Elevation of the inlet (feet); (m)

$z_2$  Elevation of the outlet (feet); (m)

Pipe efficiency, $e$, is introduced for calibration purpose. Since the minor head losses mainly due to fittings, bends and various obstructions are ignored, the efficiency can be used to adjust the model to match the actual flow for a given pipeline segment. There have been arguments that using a correct correlation for the friction factor alleviates the need for providing such an adjustment and that pipe roughness alone is sufficient as an adjustment mechanism. While roughness can account for frictional effects such as bends and fittings, pipes also can have various obstructing materials like condensate accumulations, rust, and sediment that behave more like diameter reductions. For these and other reasons that will become more apparent when the friction factor is discussed, pipe roughness alone is an inadequate compensator [5].
The Friction Factor

There are two friction factor definitions in standard usage: Fanning and Darcy-Weisbach. Since the Darcy-Weisbach factor is simply four times the Fanning factor, it is mostly a matter of personal choice which one is used. In the fundamental flow Eq. (3.7), the friction factor $f$ is Darcy-Weisbach friction factor.

The flow through a pipeline may be classified as laminar, turbulent, or critical flow depending upon the value of the Reynolds number. The Reynolds number depends upon the gas properties, the pipe diameter and the flow velocity and is defined as follows

$$Re = \frac{Vd\rho}{\mu},$$

(3.8)

where

- $Re$ Reynolds number, dimensionless
- $V$ Average gas velocity, (ft/s); (m/s)
- $d$ Pipe inside diameter, (ft); (m)
- $\rho$ Gas density, (lb/ft$^3$); (kg/m$^3$)
- $\mu$ Gas viscosity, lb/ft-s; (Pa.s)

In terms of the more commonly used units in the gas pipeline industry, the following formula for Reynolds number is more appropriate.

$$Re = C \left(\frac{P_b}{T_b}\right) \left(\frac{GQ}{\mu d}\right),$$

(3.9)

where:

- $Re$ Reynolds number, dimensionless
- $C$ Units conversion coefficient, 0.015379 (English units); 49.44 (Metric)
- $d$ Pipe inside diameter, (inches); (mm)
$G$  Gas specific gravity (dimensionless)

$P_b$  Base pressure, (psia); (kPa)

$Q$  Gas flow, (standard ft$^3$/day); (standard m$^3$/day)

$T_b$  Base temperature, (°R); (K)

$\mu$  Gas viscosity, (lb$_{-}$s/ft$^2$); (Pa.s)

This is a better way to see the Reynolds number in a gas industry context, since it
points out that it is essentially proportional to the flow rate. The other parameter in the
friction factor correlation is the pipe roughness. The friction factor is a function of the
Reynolds number and the relative pipe roughness $\frac{\varepsilon}{d}$. This function is usually
presented in the Moody Diagram, shown in Fig. 8.

Figure 8. Moody diagram.

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In the laminar zone, the friction factor is a function of the Reynolds number and can be derived from the definition of Darcy friction factor substituted in the analytical solution of Hagen-Poiseuille flow. The classic relation is

\[ f = \frac{64}{\text{Re}} \]  

(3.10)

On the other end of Moody diagram, the flow is fully turbulent and is a function of the roughness only. The friction factor in the fully turbulent zone is given by the rough pipe law of Nikuradse:

\[ \frac{1}{\sqrt{f}} = 2\log\frac{d}{e} + 1.14 \]  

(3.11)

The Partially Turbulent zone is the part in the middle where the curves exist. Starting on the left side of this zone, the flow is governed by the smooth pipe law of von Karman and Prandtl:

\[ \frac{1}{\sqrt{f}} = -2\log(\text{Re}\sqrt{f}) - 0.8 \]  

(3.12)

How the friction factor varies across this region from the smooth pipe law to the rough pipe law is not completely agreed-upon. Some feel that the straight horizontal lines of the rough pipe law should be extended to the smooth pipe law forming a corner at the intersection. Others feel that nature abhors corners and the Colebrook-White equation, which is nothing more than a combination of the two, is the proper method [5]:

\[ \frac{1}{\sqrt{f}} = -2\log\left(\frac{e}{3.7d} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \]  

(3.13)

In comparison of the two approaches, Schroeder [5] points out several important observations:
1. The Colebrook-White equation always predicts a higher friction factor; hence it is the more conservative approach.

2. The maximum difference is about 17% which translates to a 8.5% difference in the flow rate.

3. This maximum difference occurs at fairly low Reynolds numbers associated with low pressure drop and lessens with increasing pressure drop where it is more important.

4. There is significant scatter to the data.

These equations point out the other reason, mentioned earlier, why the roughness alone is not a sufficient calibration parameter. At lower Reynolds numbers, respectively low flow, the roughness plays a minor role. Even more, the smooth pipe law does not include an effect for roughness.

The Critical zone, where there are no lines, hence no function, poses a challenge. A possible solution is to connect the end of the laminar flow at approximately \( \text{Re} = 2000 \) with the beginning of the smooth pipe law, assumed at \( \text{Re} = 3250 \) with a straight line. This way, the entire diapason of flows, from laminar to fully turbulent can be covered and modeled mathematically. The common approach is to use the Hagen-Poiseuille relationship for laminar flow, connect the end of the laminar flow with a straight line \((f = \text{constant})\) to the beginning of the smooth pipe law, and use either the Colebrook-White equation or a combination of the smooth pipe law and rough pipe law.

The fact that these equations are not explicit, presents a performance problem. Since the friction term appears in both sides of the equations, an iterative method is required for the computation of \( f \). Some explicit equations have been developed to provide a faster
computation method. An example is the equation of Chen, which is the most precise. It is presented as

\[
\frac{1}{\sqrt{f}} = -4 \log \left\{ \frac{\varepsilon}{d} \cdot \frac{5.0452}{3.7065} \cdot \log \left[ \frac{\varepsilon}{d} \cdot \frac{1.1096}{2.8257} + \left( \frac{7.149}{\text{Re}} \right)^{0.8961} \right] \right\} \quad (3.14)
\]

The majority of the pipes in a real natural gas pipeline system would have turbulent flow regime. However, some low-flow pipes may occur and if the Reynolds number is in the range of 2000 to 3250, gradient methods for solving a network using Eq. (3.14) simply may not converge to a unique solution. One solution to this problem is to use the value of \(f\) at the beginning of the turbulent zone as a constant for the flows in the critical zone.

Other Pipe Flow Equations

Due to the complexity introduced by the variable friction factor in the fundamental pipe equation, other equations have been developed that eliminate the friction factor from the calculations. These equations are usually applicable to a certain range of conditions.

More pipe flow equations are presented in the Appendix.
CHAPTER 4

NON-PIPE ELEMENTS

A number of non-pipe elements exist in an actual natural gas system, e.g. various types of valves, regulators, regulator stations, compressors, and compressor stations. Those non-pipe elements are connected to pipes or other non-pipe elements at their end nodes. The flow through such elements is governed by various flow equations specific for the type of the elements.

Valves and Regulators

A regulator is usually defined as an element that has an unknown valve position, with the upstream or downstream pressure held constant at a set value. That is, regulators are pressure-sustaining devices. Depending upon the chosen regulator type, input for these elements may include information such as the maximum Cg value, the regulator's pressure set point, and the node where the set point is located [1]. If some of the equations proposed by the valve manufacturers are used, more valve specific parameters are required. A couple of those equations, i.e. Fisher and Grove valve flow equations are presented in the Appendix.

A valve can be seen as a short pipe. It can be assumed that there is no elevation change. Hence, the elevation correction term in the fundamental Eq. (3.7) disappears and the equation can be rewritten as
where $C'$ is a characteristic constant of the valve.

The above equation could be simplified further for gas by relating the temperature to the pressure through the ideal gas law, and introducing the gas sizing coefficient $C_g$, defined by the following equation,

$$Q_{\text{critical}} = C_g P_1$$

where $Q_{\text{critical}}$ is the critical flow.

Critical flow is reached when the velocity of the flow reaches sonic velocity. The flow and, respectively, the velocity are proportional to the pressure differential across the valve. Increasing the inlet pressure $P_1$ increases the velocity of the gas. When a sonic velocity is reached, any further increase of the inlet pressure has no effect on the velocity. Critical flow through an orifice occurs with natural gas when the ratio of downstream to upstream pressure is equal or less than 0.53 [6].

Equation (4.1) is modified for subcritical conditions to

$$Q = 2C_g \sqrt{(P_1 - P_2)P_1}$$

and for critical conditions to

$$Q = C_g P_1.$$  

Equations (4.3) and (4.4) could be used to model proposed new valves or regulators when the type, size and other manufacturer’s data are still unknown. The flow calculated by these equations is approximate and can significantly deviate from the actual flow of an actual valve or regulator. These apparently simple equations have strict limitations resulting from compressibility effects and critical flow.
Some valve manufacturers have derived their own flow equations for their valves. A comprehensive review of the theoretical and practical basis and the evolution of the Fisher Regulators' equation is presented by Jury [8]. In 1974, Fisher Regulators published the following gas flow equations for their regulators,

\[
Q = \sqrt{\frac{520}{GR}} C_g P_i \sin \left( \frac{59.64}{C_l} \sqrt{\frac{P_1 - P_2}{P_1}} \right) \text{ Radians} \quad \text{subcritical \quad (4.5)}
\]

\[
Q = \sqrt{\frac{520}{GR}} C_g P_i \quad \text{critical \quad (4.6)}
\]

where

- \(Q\) \text{ flow rate (scfh)}
- \(G\) \text{ gas specific gravity (dimensionless)}
- \(T\) \text{ gas temperature at inlet (°R)}
- \(P_i\) \text{ upstream pressure (psia)}
- \(P_2\) \text{ downstream pressure (psia)}
- \(C_g\) \text{ gas sizing coefficient (scfh/psi)}
- \(C_l\) \text{ liquid sizing coefficient (US gpm/psi)}
- \(C_t\) \text{ ratio of the gas sizing coefficient to liquid sizing coefficient, } C_t = C_g / C_l.

Fisher research department had conducted thousands of tests on every different style of valve available including both high and low recovery valves as well as some intermediate ones [8]. When the results of the tests had been normalized in respect to critical flow and plotted, they noted that all the data points in the slopping portion of the flow curve could be closely approximated by the first quarter cycle of a standard sine curve. The sine in Eq. (4.5) is that approximation connecting the low pressure ratio portion of the flow curve, i.e. Eq. (4.3) and the critical flow line, i.e. Eq. (4.4). The
argument of the sine is at most $\pi/2$. At critical conditions, the argument of the sine is $\pi/2$ and Eq. (4.5) becomes Eq. (4.6). Fisher’s equation, i.e. Eq. (4.5) is also known as Universal valve sizing equation for gas, because it is applicable for all types of valves and gases under any pressure conditions. A comparison between the flow curves defined by Fisher and the General valve flow equations is presented in Fig. 9.

![Figure 9. Comparison of the Fisher equation with the General valve flow equation.](image)

The general equation (4.3) and Eq. (4.4) are multiplied by $\sqrt{\Delta P / P_1}$ to make it applicable for any gas and temperature different from the standard temperature of 60°F. Also, Eq. 
(4.3) is multiplied by $\frac{59.64}{C_1} \sqrt{1 - \left(\frac{P_2}{P_1}\right)_{\text{critical}}}$ to eliminate the discontinuity at critical pressure ratio.

In a pipeline gas system, the set pressure is usually downstream of the regulator element. A maximum $C_g$ is specified for each regulator element. The regulator equation is solved for $C_g$. If $C_g < \max C_g$, the regulator is holding the set pressure. If $C_g \geq \max C_g$, then the regulator is fully open. At this condition, the set pressure can not be held and it starts drooping. The outlet pressure is calculated with $\max C_g$ and compared to the set pressure. If the outlet pressure is lower than the downstream set pressure, the regulator is held fully open and the set node pressure is treated as unknown.

If the computed $C_g \leq 0$, the regulator is closed and the set pressure node is treated as node with unknown pressure. On the next iteration the regulator can be returned to regulating regime and the pressure at the set-pressure node can be returned to the specified set pressure if $0 < C_g < \max C_g$.

**General Compressor**

When a new or proposed compressor is modeled, for which full manufacturer’s data are unavailable, the following power/flow equation is typically used [1]

$$\frac{P_{hp}}{Q} = Z \left[ K_1 \left( \frac{P_2}{P_1} \right)^{K_1} - K_2 \right] \frac{T_i}{T_b}$$

where

$P_{hp}$ compressor power
The equation constants may be selected based on correlation to field data or calculated, e.g., for isentropic compression, by integrating the equation for work per unit mass, using the equation of state and introducing compressor efficiency, the following values can be obtained

\[ K_1 = K_2 = C \frac{P_b}{e} \left( \frac{n}{n-1} \right) \frac{Z_1}{Z_b} \]  

(4.8)

\[ K_3 = \frac{n-1}{n} \]  

(4.9)

where

- \( C \) units conversion constant, e.g. \( C = 3.0303 \) for English units
- \( P_b \) base/standard pressure, usually 14.73 psia
- \( Z_1 \) compressibility factor at suction
- \( Z_b \) compressibility factor at standard conditions
- \( e \) compressor efficiency
- \( n \) gas polytropic exponent, 1.3 for natural gas

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\[ Z_1 \] compressibility factor at suction

\[ Z_b \] compressibility factor at standard conditions

\[ e \] compressor efficiency

\[ n \] gas polytropic exponent, 1.3 for natural gas
When an existing compressor is modeled, for which there is not enough manufacturer’s data but ample field data are available, an equation expressing the power/flow rate as a polynomial function of compression ratio can be developed, [1]

\[
\frac{P_{hp}}{Q_s} = (a_0 + a_1 R + a_2 R^2 + a_3 R^3 + a_4 R^4) Z
\]  

(4.10)

where

- \( a_i \)  polynomial coefficients, \( i = 1, 2, 3, 4 \)
- \( R \)  compression ratio
- \( Z \)  compressibility factor

The polynomial coefficients should be developed for wide range of system operations. Otherwise, flows or pressures outside the range for which the data were derived could result in inaccurate power and flow computations.

Equations (4.7) and (4.10) can be used for any type of compressor.

Centrifugal Compressor

Of particular interest is modeling of a centrifugal compressor. The centrifugal compressors provide efficient way of compressing large volumes of gas at high flow rates. They can be found on most of the transmission pipelines.

A performance map of a centrifugal compressor is provided by the manufacturer. A typical performance map is shown in Fig. 10. Curves of adiabatic head are plotted for various actual flows at various speeds. Curves \( S_i \) represent constant speed. The efficiency curves \( e_i \) are superimposed on the map. Under normal operating conditions, the compressor operates inside the map. Optimal operation is achieved if the operating
point is in the “sweet spot” that is inside the highest efficiency curve (inside line $e_1$ in Fig. 10).

Figure 10. Typical performance map of a centrifugal compressor.

With some auxiliary equations, the map describes the operation of a centrifugal compressor. The required auxiliary equations relate the variable adiabatic head $H$, power $P_{hp}$, and actual flow $Q$, to the other compressor variables. [1]

The head equation for centrifugal compressor is

$$H = RT Z_{sv} \left( \frac{n}{n-1} \right) \left[ \left( \frac{P_2}{P_1} \right)^n - 1 \right]$$

(4.11)

where

$H$  adiabatic head (feet); (m)
specific gas constant, \( R = \frac{R^*}{M_w} = \frac{R^*}{M_{w_{air}} G} \), \( R^* \) - universal gas constant

\[
R = \frac{1545}{28.97G} \text{ ft-lb/mol-°R (English)};
\]

\[
R = \frac{8314}{28.97G} \text{ J/kmol-K (SI)}
\]

\( Z_{av} \) compressibility factor (dimensionless)

\( P_1 \) suction pressure (psia); (Pa)

\( P_2 \) discharge pressure (psia); (Pa)

\( T_1 \) gas flowing temperature at suction (°R); (K)

\( n \) polytropic exponent (dimensionless)

The aerodynamic or gas power of the compressor is determined to be [7]:

\[
P_g = \rho_s Q_s H
\]  

(4.12)

The adiabatic efficiency of a compressor does not include the mechanical losses, which typically amount to about 1 to 2\% of the absorbed power [7]. By introducing compressor’s efficiency, \( e_c \), and mechanical efficiency, \( e_m \), in Eq. (4.12), the absorbed power becomes

\[
P_{ap} = \frac{\rho_s Q_s H}{e_c e_m}
\]  

(4.13)

where

\( P_{ap} \) compressor power (hp); (W)

\( \rho_s \) gas density at standard conditions (lb/ft\(^3\)); (kg/m\(^3\))

\( H \) adiabatic head (feet); (m)

\( e_c \) compression efficiency (dimensionless)

\( e_m \) mechanical efficiency (dimensionless)
The actual flow equation for centrifugal compressor is

\[ Q_a = C \frac{P_b}{T_b Z_b} \frac{T_i Z_i}{P_i} Q_s \]  

(4.14)

where

- \( Q_a \) actual flow rate (cfm); (m³/s)
- \( Q_s \) standard flow rate (standard MMcfd); (standard m³/d)
- \( C \) unit conversion factor, 694.4 for English units; 11.574 \times 10^6 for metric units
- \( P_b \) base pressure (psia); (Pa)
- \( T_b \) base temperature (°R); (K)
- \( Z_b \) compressibility factor at base conditions (dimensionless, \( \approx 1 \))
- \( P_1 \) suction pressure (psia); (Pa)
- \( T_1 \) gas flowing temperature at suction (°R); (K)
- \( Z_1 \) compressibility factor at suction (dimensionless)

Turbomachine affinity laws state that

\[ \frac{H}{S^2 D^2} \propto \frac{Q}{SD} \propto e \]  

(4.15)

where

- \( H \) adiabatic head (feet); (m)
- \( S \) impeller speed (rpm); (min⁻¹)
- \( D \) impeller diameter (inch); (mm)
- \( e \) efficiency (dimensionless)

Since the impeller diameter \( D \) is constant for a given unit, these proportions become

\[ \frac{H}{S^2} \propto \frac{Q}{S} \propto e \]  

(4.16)
Expanding these relationships for the two conditions allows determination of the conditions of the second point if the first point is known,

\[
\frac{H_1}{S_1^2} = \frac{H_2}{S_2^2} \propto \frac{Q_1}{S_1} = \frac{Q_2}{S_2} \propto e_1 = e_2
\]  

(4.17)

By utilizing the above relationships, the performance map in Fig. 10 can be reduced to only two curves, shown in Fig. 11. [1]

![Figure 11. Two-curve representation of the performance map.](image)

This approach is employed by SynerGEE® for modeling of a centrifugal compressor by utilizing its actual performance map.

The gas temperature at the discharge is given by the equation

\[
T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}}
\]  

(4.18)

where

- \(T_1\) gas flowing temperature at suction (°R); (K)
- \(T_2\) gas flowing temperature at discharge (°R); (K)
- \(P_1\) suction pressure (psia); (kPa)
The fuel used by the compressor has to be considered in the model. The fuel is usually modeled as a known flow taken out of the system at the upstream node of the compressor element. The simplest way to account for compressor fuel is to use a nominal fuel rate in the calculations. The compressor fuel requirement is expressed as a product of the nominal fuel rate and developed power,

\[ Q_F = F_N P_{hp} \]  \hspace{1cm} (4.19)

where

- \( Q_F \) fuel flow rate (standard cfh); (standard m³/h)
- \( F_N \) nominal fuel rate (standard cfh/hp-h); (standard m³/kW-h)
- \( P_{hp} \) developed power by the engine (hp); (kW)

A better way to compute the fuel requirements is to express the fuel flow as a polynomial function of the engine power,

\[ F = c_0 + c_1 P_{hp} + c_2 P_{hp}^2 \]  \hspace{1cm} (4.20)

where

- \( F \) fuel rate (standard cfh/hp-h); (standard m³/kW-h)
- \( P_{hp} \) developed power by the engine (hp); (kW)
- \( c_0, c_1, c_2 \) polynomial coefficients

The polynomial coefficients \( c_0, c_1 \) and \( c_2 \) can be determined by a regression fitting of fuel consumption to engine power. This is done usually on manufacturer’s data for

\[ P_2 \] discharge pressure (psia); (kPa)

\( n \) polytropic exponent (dimensionless)
various engine load, respectively developed power, and fuel consumption or based on field data. The fuel rate $F$ is used in Eq. (4.19) in place of the nominal fuel rate.

Compressor Engine De-rating Factors

As the temperature of the available combustible air rises or falls, its density varies inversely; hence, the power available from an engine also varies inversely with temperature. Likewise, air density varies inversely with elevation. Engines are power rated at a specified air temperature and elevation. Available power therefore varies with actual air temperature and elevation according to the following equation [1]:

$$P_{hp}^{\text{(available)}} = P_{hp}^{\text{(rated)}} \left[ 1 - \frac{F_T (T - T_e)}{100} \right] \left[ 1 - \frac{F_H H}{100000} \right]$$

(4.21)

where

- $P_{hp}^{\text{(available)}}$ available engine power (hp); (kW)
- $P_{hp}^{\text{(rated)}}$ engine rated power (hp); (kW)
- $F_T$ temperature de-rating factor (% per °F); (% per °C)
- $T$ ambient temperature (°F); (°C)
- $T_e$ engine rated air temperature (°F); (°C)
- $F_H$ elevation de-rating factor (% per 1000 ft); (% per 1000 m)
- $H$ elevation (ft); (m)

In the industry, common value for $F_T$ is 0.1% per 1°F (0.18% per 1°C). The theoretical value is 0.2% per 1°F (0.36% per 1°C).

The theoretical value for $F_H$ is 3.6% per 1000 feet (11.8% per 1000 m). Industry commonly used values are:

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Turbocharged engines 2% per 1000 ft (6.5% per 1000 m)

Non-turbocharged engines 3% per 1000 ft (10% per 1000 m).

The manufacturers usually rate the engine power at standard conditions, i.e. ambient temperature of 60°F and sea level elevation. If the engine is rated at a temperature different from the modeled ambient temperature the power should be corrected for temperature. If the engine is not rated for the site, the power should be corrected for elevation. If the engine is rated for the site, no correction of the power is required.
A research initiated in 1981 under the sponsorship of the Gas Research Institute (GRI) in close liaison with the American Gas Association (AGA) Transmission Measurement Committee and data provided by the European Gas Research Group (GERG) led to Compressibility Factors of Natural Gas and Other Related Hydrocarbon Gases publication, also known as AGA Report No. 8. It was published in 1992 and became an industry standard for computing compressibility factors and specific density of natural gas and other hydrocarbon gases. The methodology, the equations of state and the data from AGA Report No. 8 were utilized for compressibility calculations in the developed pipeline network model.

AGA Report No. 8 [9] provides two methods for calculation of compressibility factors:

1. Detail Characterization Method – Requires information about the natural gas composition, i.e. the mole fractions or mole percentages of the components in the natural gas mixture;

2. Gross Characterization Method – Requires characterization information such as the molar ideal gross heating value of the mixture of hydrocarbon components present in the...
natural gas along with the composition of the non-hydrocarbon components in the natural gas mixture.

AGA recommends the use of the Detailed Characterization Method for temperatures from 32°F to 130°F (0°C to 55°C) and pressure up to 1200 psia (8.3 MPa). The gas in most of the natural gas transmission and distribution systems fit in these conditions.

The compressibility factor is defined by the equation

\[ Z = \frac{PV}{nRT} \]  \hspace{1cm} (5.1)

where:

- \( P \) absolute static pressure
- \( V \) gas volume
- \( n \) number of moles of gas
- \( Z \) compressibility factor of gas
- \( R \) gas constant
- \( T \) absolute temperature of gas

The supercompressibility factor \( F_{pv} \) is defined as

\[ F_{pv}^{2} = \frac{Z(\text{standard conditions})}{Z(T, P)} \]  \hspace{1cm} (5.2)

The gas mixture molar mass \( M_r \) is calculated from the composition by

\[ M_r = \sum_{i=1}^{N} x_i M_n \]  \hspace{1cm} (5.3)

where:

- \( x_i \) molar fraction of component \( i \) in gas mixture
\( M_{ri} \) molar mass of \( i \)-th component

\( N \) number of all components in the gas mixture

The mass density \( \rho \) is related to molar density by

\[
\rho = M_r d
\] (5.4)

where

\( M_r \) molar mass (mass per mole)

\( d \) molar density, \( d = n / V \), (moles per unit volume)

Both methods utilize data at defined reference conditions and conversion from different reference conditions is required.

Equations of State

The equation of state for the Detail Characterization Method is a hybrid formulation. It combines features of the viral equation of state (power series in density) for low density conditions and exponential functions for applications at high density conditions (extended Benedict-Webb-Rubin equation). A detailed description of the equation and its performance is given in AGA Report No. 8, Reference 1, Appendix A.7. [9]. The equation of state for the compressibility factor \( Z \) for the Detailed Characterization Method is given in its condensed form by the following equation [9]:

\[
Z = 1 + \frac{DB}{K^2} - D \sum_{n=13}^{18} C_n^0 T^{-\delta_n} + \sum_{n=13}^{58} C_n^1 T^{-\delta_n} (b_n - c_n k_n D^{\delta_n}) D^{\delta_n} \exp(-c_n D^{\delta_n})
\] (5.5)

where

\( B \) second viral coefficient

\( K \) mixture gas parameter

\( D \) reduced density, \( D = K^3 d \)
The equation of state for the Gross Characterization method expresses the compressibility factor $Z$ in terms of the molar density $d$, the mixture second viral coefficient $B_{\text{mix}}$, and the mixture third viral coefficient $C_{\text{mix}}$ as follows:

\[
Z = 1 + B_{\text{mix}}d + C_{\text{mix}}d^2
\]  

(5.6)

\[
B_{\text{mix}} = \sum_{i=1}^{N} \sum_{j=1}^{N} B_{ij} x_i x_j
\]  

(5.7)

\[
C_{\text{mix}} = \sum_{i=1}^{N} \sum_{j=1}^{N} \sum_{k=1}^{N} C_{ijk} x_i x_j x_k
\]  

(5.8)

where

- $B_{ij}$ is the individual components interaction second viral coefficient
- $C_{ijk}$ is the individual components interaction third viral coefficient
- $x_i, x_j, x_k$ are mole fractions of gas components
- $N$ is the number of components in the gas mixture

The components interaction second and third viral coefficients are functions of temperature. For terms involving only nitrogen and carbon dioxide, $B_{ij}$ and $C_{ijk}$ are expressed as:

\[
B_{ij} = b_0 + b_1 T + b_2 T^2
\]  

(5.9)

\[
C_{ijk} = c_0 + c_1 T + c_2 T^2
\]  

(5.10)

For terms involving equivalent hydrocarbons, the interaction viral coefficients are:

\[
B_{\text{CH4-C4H10}} = B_0 + B_1 H_{\text{CH4}} + B_2 H_{\text{C4H10}}
\]  

(5.11)
\[ C_{\text{CH-CH-CH}} = C_0 + C_1 H_{\text{CH}} + C_2 H^2_{\text{CH}} \]  

where, \( B_0, B_1, B_2, C_0, C_1, \) and \( C_2 \) are temperature dependent functions defined as

\[ B_i = b_{i0} + b_{i1} T + b_{i2} T^2, \quad i = 0, 1, 2 \]  

\[ C_i = c_{i0} + c_{i1} T + c_{i2} T^2, \quad i = 0, 1, 2 \]

The constants are provided in AGA Report No. 8. The publication also provides equations for calculating the interaction second and third viral coefficients for terms involving interaction of equivalent hydrocarbons with nitrogen, terms involving interaction of equivalent hydrocarbon with carbon dioxide, and terms involving interaction between nitrogen, carbon dioxide, and equivalent hydrocarbon.

The equation of state for the pressure, using the Gross Characterization Method is obtained by substituting (5.6) into (5.1),

\[ P = dRT(1 + B_{\text{mix}} d + C_{\text{mix}} d^2) \]  

AGA Report No. 8 provides computer codes for computational procedures written in FORTRAN.

Some commercially available computer applications for pipeline modeling allow use of other equations of state, e.g. Redlich-Kwong equation or even a user-defined equation of state. In this work, the AGA Report No. 8 Gross Characterization Method was utilized.
METHOD OF CHARACTERISTICS

Introduction

The method of characteristics has been widely used in one-dimensional unsteady gas dynamics and for steady two-dimensional supersonic inviscid flow. However, the method is rather cumbersome when extended to three or four independent variables, or if internal shocks occur. [3]

The flow in gas pipeline systems is one dimensional. Due to the low density and relatively low velocities, internal shocks are unlikely to occur.

The method of characteristics transforms the partial differential equations of motion and continuity into ordinary differential equations. These are then integrated to obtain a finite difference representation of the variables. The characteristics grid method and, more commonly, the method of specified time intervals provide convenient solution procedures [10].

The method of characteristics is employed by some commercial modeling software to solve unsteady flow pipeline problems. Along with the basic principles, this chapter describes the method as it is implemented in Advantica’s SynerGEE® software.

Continuity Equation

Let us consider a short segment of a pipe with length \( \Delta x \) as shown in Fig. 12.
The mass flow balance for that element is

\[
\dot{M} - \left( \frac{\partial M}{\partial t} + \frac{\partial M}{\partial x} \Delta x \right) = \frac{\partial}{\partial t} (\rho A \Delta x)
\]  

(6.1)

where

- \( \rho \) gas density
- \( \dot{M} \) mass flow rate
- \( A \) pipe cross sectional area.

The cross section area is a constant for a given pipe element. Therefore, Eq. (6.1) simplifies to

\[
\frac{\partial M}{\partial t} + A \frac{\partial \rho}{\partial t} = 0
\]  

(6.2)

The change in density with respect to time could be expressed in terms of change of pressure with respect to time,

\[
\frac{\partial P}{\partial t} = \frac{\partial P}{\partial \rho} \frac{\partial \rho}{\partial t}
\]  

(6.3)

For a general equation of state, the speed of sound \( c \) is given by
Substituting (6.4) into (6.3), solving for $\partial \rho / \partial t$, and substituting it into (6.2), the continuity equation becomes

$$\frac{\partial P}{\partial t} + \frac{c^2}{A} \frac{\partial M}{\partial x} = 0 \quad (6.5)$$

The wave speed $c$ can be calculated by

$$c = \sqrt{kZRT} \quad (6.6)$$

where

$$k \quad \text{adiabatic index,} \quad k = \frac{c_p}{c_v}.$$  

For isothermal calculations, $k$ can be omitted. The isothermal wave speed is given by

$$c = \sqrt{ZRT} \quad \text{(IS units)} \quad (6.7)$$

$$c = \sqrt{g_0ZRT} \quad \text{(English units)} \quad (6.8)$$

By replacing the mass flow rate with the volume flow rate at standard conditions, the continuity equation becomes

$$\frac{\partial P}{\partial t} + \frac{Kc^2}{A} \frac{\partial Q}{\partial x} = 0 \quad (6.9)$$

where $K$ is a conversion factor.

**Equation of Motion**

Let us consider a short segment of a pipe with length $\Delta x$ as shown in Fig. 13, where

$F_R$ \quad \text{resultant force moving the gas}$

$F_f$ \quad \text{friction force}$

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\( F_{P(x)} \) force due to pressure at \( x \)

\( F_{P(x+\Delta x)} \) force due to pressure at \( x+\Delta x \)

\( W \) weight of the gas in the control element

\( W_x \) projection of the weight in \( x \) direction

\( \theta \) angle of pipe inclination.

![Figure 13. Control volume for momentum equation.](image)

The force balance for that element is

\[
F_R = F_{P(x)} - F_{P(x+\Delta x)} + W_x - F_f
\]  
(6.10)

The terms of the Eq. (6.10) are as follows,

\[
F_R = \text{mass \cdot acceleration} = \rho A \Delta x \frac{DV}{Dt} = \rho A \Delta x \left( \frac{\partial V}{\partial t} + V \frac{\partial V}{\partial x} \right)
\]  
(6.11)

\[
F_{P(x)} - F_{P(x+\Delta x)} = AP - \left( AP + A \frac{\partial P}{\partial x} \Delta x \right) = A \frac{\partial P}{\partial x} \Delta x
\]  
(6.12)

\[
W_x = W \sin \theta = \rho A \Delta x \sin \theta
\]  
(6.13)

\[
F_f = \tau_0 \pi D \Delta x = \frac{\rho fV|V|}{2D} \pi D \Delta x
\]  
(6.14)
where

\[ \rho \quad \text{gas density} \]
\[ A \quad \text{pipe cross sectional area} \]
\[ D \quad \text{internal pipe diameter in Eq. (6.13)} \]
\[ V \quad \text{gas flowing velocity} \]
\[ f \quad \text{Darcy-Weisbach friction factor.} \]

The velocity \( V \) can be expressed in terms of mass flow rate. Solving \( \rho AV = \dot{M} \) for velocity, we get

\[ V = \frac{\dot{M}}{\rho A} \quad (6.15) \]

Due to the very short length of the pipe segment, it can be assumed that

\[ V \frac{\partial V}{\partial x} \equiv 0 \quad (6.16) \]

Substituting in (6.11), the resultant moving force term becomes

\[ F_r \equiv \rho A \Delta x \frac{\partial V}{\partial x} \equiv \rho A \Delta x \frac{\partial}{\partial t} \left( \frac{\dot{M}}{\rho A} \right) \equiv \Delta x \frac{\partial \dot{M}}{\partial t} \quad (6.17) \]

Given that

\[ \frac{1}{\rho} = \frac{c^2}{P}, \quad (6.18) \]

the density can be expressed in terms of wave speed \( c \) and absolute pressure \( P \).

Substituting in Eq. (6.13) and (6.14), the gravity term \( W_x \) and the friction term \( F_f \) become

\[ W_x = \frac{P}{c^2} A \Delta x \sin \theta \quad (6.19) \]

and
Substituting Eq. (6.12), (6.17), (6.19) and (6.20) into Eq. (6.10) and rearranging the terms, it becomes

\[
\frac{\partial P}{\partial x} + \frac{1}{A} \frac{\partial M}{\partial t} + \frac{P \sin \theta}{c^2} + \frac{c^2 f \dot{M}}{2DA^2 P} = 0
\]  

(6.21)

If the mass flow term is replaced with flow rate at standard conditions, the equation of motion becomes

\[
\frac{\partial P}{\partial x} + K \frac{\partial Q}{\partial t} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 f \dot{Q}}{2DA^2 P} = 0
\]  

(6.22)

where \(K\) is a conversion coefficient,

\[
K = C \frac{P_b}{T_b} \dot{M}
\]  

(6.23)

where

- \(C\) unit conversion coefficient
- \(P_b\) base pressure
- \(T_b\) base temperature
- \(M\) mass flow rate.

Characteristics Equations

The continuity and momentum equations, Eq. (6.9) and (6.22) form a pair of quasilinear hyperbolic partial differential equations in terms of two dependent variables, flow rate and pressure, and two independent variables, distance along the pipe and time.
The equations are transformed into four ordinary differential equations by the characteristics method [10].

The equations of continuity and motion are identified as $L_1$ and $L_2$, from (6.9) and (6.22):

\[
L_1 = \frac{\partial P}{\partial t} + \frac{Kc^2}{A} \frac{\partial Q}{\partial x} = 0
\]  
(6.24)

\[
L_2 = \frac{\partial P}{\partial x} + \frac{K}{A} \frac{\partial Q}{\partial t} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 f Q |Q|}{2DA^2 P} = 0
\]  
(6.25)

The equations are combined linearly using some unknown multiplier $\lambda$:

\[
L = \lambda L_1 + L_2 = \lambda \left( \frac{\partial P}{\partial t} + \frac{Kc^2}{A} \frac{\partial Q}{\partial x} \right) + \left( \frac{\partial P}{\partial x} + \frac{K}{A} \frac{\partial Q}{\partial t} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 f Q |Q|}{2DA^2 P} \right) = 0
\]  
(6.26)

After rearranging the terms, the equation becomes

\[
\lambda \left( \frac{\partial P}{\partial t} + \frac{1}{\lambda} \frac{\partial P}{\partial x} \right) + K \left( \frac{\partial Q}{\partial t} + \lambda c^2 \frac{\partial Q}{\partial x} \right) + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 f Q |Q|}{2DA^2 P} = 0
\]  
(6.27)

Any two real, distinct values of $\lambda$ will again yield two equations in terms of the two dependent variables $P$ and $Q$ that are in every way the equivalent of Eq. (6.24) and (6.25). Appropriate selection of two particular values of $\lambda$ leads to simplification of Eq. (6.27). In general, both variables $P$ and $Q$ are functions of $x$ and $t$. If the independent variable $x$ is permitted to be a function of the time $t$, then, from calculus, [10]

\[
\frac{dP}{dt} = \frac{\partial P}{\partial x} \frac{dx}{dt} + \frac{\partial P}{\partial t}, \quad \frac{dQ}{dt} = \frac{\partial Q}{\partial x} \frac{dx}{dt} + \frac{\partial Q}{\partial t}
\]  
(6.28)

By examining the terms in the parenthesis in Eq. (6.27) it can be noted that if

\[
\frac{dx}{dt} = \frac{1}{\lambda} = \lambda c^2
\]  
(6.29)

then Eq. (6.27) becomes an ordinary differential equation
The solution of (6.29) yields two particular values of \( \lambda \),

\[
\lambda = \pm \frac{1}{c} \tag{6.31}
\]

By substituting \( \lambda \) back in Eq. (6.29), the relation between \( x \) and \( t \) is given by

\[
\frac{dx}{dt} = \pm c \tag{6.32}
\]

The substitution of these values of \( \lambda \) into Eq. (6.30) leads to two pairs of equations which are grouped and identified as \( C^+ \) and \( C^- \) equations.

\[
C^+ : \begin{cases}
\lambda \frac{dP}{dt} + \frac{K}{A} \frac{dQ}{dt} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 fQ |Q|}{2DA^2 P} = 0 \\
\frac{dx}{dt} = c
\end{cases} \tag{6.33}
\]

\[
C^- : \begin{cases}
\lambda \frac{dP}{dt} + \frac{K}{A} \frac{dQ}{dt} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 fQ |Q|}{2DA^2 P} = 0 \\
\frac{dx}{dt} = -c
\end{cases} \tag{6.35}
\]

It is convenient to visualize the solution as it develops on the independent variable plane, i.e. the \( x-t \) plane. Such visualization is presented in Fig. 14. The dashed lines are the “characteristic” lines, along which Eq. (6.33) and (6.35) are valid. These equations are referred to as compatibility equations, each one being valid only on the appropriate characteristic line.
Finite Difference Equations

The pipeline is divided into an even number of reaches, \( N \), each \( \Delta x \) in length, as it is shown in Fig. 14. A time-step size is computed, \( \Delta t = \Delta x / c \), and Eq. (6.34) is satisfied by a positively sloped diagonal of the grid, presented by the line \( AR \).

If the dependent variables \( P \) and \( Q \) are known at point \( A \), then Eq. (6.33), which is valid along the \( C^+ \) line, can be integrated between the limits \( A \) and \( R \), and thereby be written in terms of unknown variables \( P \) and \( Q \) at point \( R \). Equation (6.36) is satisfied by a negatively sloped diagonal of the grid, shown by \( BR \). Integration of the \( C^- \) compatibility equation along the line \( BR \), with conditions known at \( B \) and unknown at \( R \), leads to a second equation in terms of the same two unknown variables at \( R \). A simultaneous solution yields conditions at the particular time and position in the \( x-t \) plane designated by point \( R \) [10].

Figure 14. Distance-time grid for solving single-pipe problem.
By multiplying Eq. (6.33) and (6.35) by \( c \cdot dt = dx \) and pressure \( P \), the equation, suitable for integration along the \( C^+ \) characteristic line, becomes

\[
\int_{Q_a}^{Q_r} PdP + \frac{cK}{A} \int_{Q_a}^{Q_r} PdQ + \frac{\sin \theta}{c^2} \int_{x_a}^{x_r} P^2 dx + \frac{K^2 c^2 f}{2DA^2} \int_{x_a}^{x_r} QdQdx = 0 \quad (6.37)
\]

After approximate integration by using the trapezoidal rule, the compatibility equation becomes

\[
P_R^2 - P_A^2 + \frac{Kc}{A} (P_R + P_A) (Q_R - Q_A) + \frac{\Delta x \sin \theta}{c^2} (P_R^2 + P_A^2) +
\frac{Kf c^2 \Delta x}{2DA^2} (Q_R |Q_R| + Q_A |Q_A|) = 0
\]

(6.38)

The same steps are performed for the compatibility Eq. (6.35) along the \( C^- \) characteristic line,

\[
P_R^2 - P_R^2 + \frac{Kc}{A} (P_R + P_R) (Q_R - Q_R) + \frac{\Delta x \sin \theta}{c^2} (P_R^2 + P_R^2) +
\frac{Kf c^2 \Delta x}{2DA^2} (Q_R |Q_R| + Q_R |Q_R|) = 0
\]

(6.39)

Both Eq. (6.38) and (6.39) satisfy the steady conditions in the pipe. Under steady conditions, the flow rates are equal at any point along the pipe segment, i.e.,

\[Q_A = Q_B = Q_R = Q\]  \quad (6.40)

By substituting the flow rate terms with \( Q \) in Eq. (6.38) and (6.39) and solving for \( Q \), these equations transform to a form of the general flow Eq. (3.7).

**Boundary Conditions**

At either end of a single pipe only one of the compatibility equations is available in the two variables. For the upstream end, \( x = 0 \) in Fig. 14, the Eq. (6.39) holds along the
characteristic line, and for the downstream boundary, \( x = L \) in Fig. 15, the Eq. (6.38) is valid along the \( C^- \) characteristic.

Figure 15. Boundary characteristics.

An auxiliary equation is needed that specifies \( Q_r \) and \( P_r \), or some relation between them. That is, the auxiliary equation must convey information on the behavior of the boundary to the pipeline. This may be just the end condition of the pipeline, or it may be a different element or facility attached to the end of the pipe. Each boundary condition is solved independently of the other boundary, and independently of the interior point calculations [10].

The Alpha Multiplier

For a pipeline system with a number of different-length pipe elements, a base length is specified. That is usually the length of the shortest pipe element.

The calculation of transient flows and pressures in a gas network by the method of characteristics has a time interval restricted to the shortest pipe length, divided by the
isothermal wave speed, and a pipe length equal to a multiple of the base length. This methodology leads to the following restriction [1]:

\[
\frac{\text{Length of every pipe in the system}}{\text{Base length specified for the system}} = \text{Integer}
\]

This restriction is almost impossible to satisfy on models with a large number of pipe elements. This restriction can be avoided by utilizing a factor referred to as \textit{alpha} (\( \alpha \)).

If we consider the momentum equation, in gas systems, the inertial term is small relative to the other terms. This means that, if the wave speed is introduced to the inertia term, a significant change in wave speed may have little computational effect on the model. Alpha is a multiplier of the inertial term that directly adjusts the wave speed. The momentum Eq. (6.22) becomes

\[
\frac{\partial P}{\partial x} + \frac{K \alpha^2}{A} \frac{\partial Q}{\partial t} + \frac{P \sin \theta}{c^2} + \frac{K^2 c^2 f Q |Q|}{2DA^2 P} = 0
\]  

(6.41)

The inclusion of \( \alpha \) yields a direct modification to the \( \lambda \) multiplier, giving the result

\[
\frac{dx}{dt} = \frac{1}{\lambda} = \frac{\lambda c^2}{\alpha^2} \quad \Rightarrow \quad \frac{dx}{dt} = \frac{1}{\lambda} = \pm \frac{c}{\alpha}
\]  

(6.42)

Wave speed adjustments, controlled by \( \alpha \), result in every pipe in the system having a slightly different wave speed.

Since the reciprocal of the wave speed is the slope of the characteristic lines on the \( x-t \) planes, alpha is a direct multiplier to those slopes. This provides a mechanism that can be used to ease the restriction that every pipe must be an integer number of base lengths.
CHAPTER 7

IMPLEMENTATION OF THE MODEL

The objective of the development of a model of a real multilevel natural gas pipeline system was to create a model that realistically represents the system and can be utilized for operational studies and long-range planning studies. A high-pressure feeder system was modeled, including all facilities downstream of the city gates, where the gas is taken from the interstate transmission pipelines, to the district regulating stations that feed local medium and low pressure distribution systems. The two types of analysis impose different requirements on the model. There are also different limitations dictated by the modeling software. In order to satisfy the requirements specific to each of the two different types of analysis, two versions of the model were created:

1. Steady-state model that would be utilized mainly for operational studies and to provide initial conditions for the unsteady-state analysis. Operational steady-state studies are conducted for various conditions, ranging from zero-HDD to peak-hour and peak-day conditions.

2. Unsteady-state model that would be used for long range planning studies or, if an analysis of the change of the operational parameters during a long period, i.e. a few days, is required. Planning studies of the system are conducted typically for peak conditions.
Many stages of the development process are similar for both models. However, the specifics of each model will be noted along with the description of the steps of the development process. Figure 16 shows the structure of the model.

![Diagram of model structure](image)

Figure 16. Typical components and features of gas network model.

The schematic and the load are the two major components of the steady-state model. The unsteady-state model includes also some additional parameters and functionality. SynerGEE® is a commercial software package that was utilized in the analysis. More details about the model components are given later.

**Schematic**

The schematic is a geographical presentation of the system, the connectivity of its elements and includes all significant elements’ parameters.
Figure 17. Schematic diagram of actual multilevel gas network.

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For the development of a model of an existing system, an extract of the schematic from Geographic Information system (GIS) can be utilized. In the development of the model of the high-pressure feeder system, a selective extract from the Company's Electronic Mapping and Records System (EMRS) was utilized. In this case, selective extract means that only elements of the system rated and marked as "transmission" or "feeder" were extracted. The information system provides data about many physical and operational parameters of the gas network. That data was extracted and manipulated in a way to comply with the requirements for input to the modeling software. The schematic of the system is shown in Fig. 17. ³

The following elements' physical parameters, required for the model, were extracted with the schematic from GIS:

1) Pipeline – nominal diameter, wall thickness, MAOP, MOP, length, and material, i.e. steel, polyethylene, or PVC.

2) Valves – type and size.

3) Regulators – type, size, and set pressure.

4) Compressors – type.

There are some physical parameters of the elements that are not stored in the GIS, e.g. pipe roughness or maximum valve constant. Default values were assigned for those parameters after the extraction. For example, the default roughness of the pipe elements is based on the material indicated in the EMRS and the default maximum valve constant is based on the type and size of the valve. More information, which is not available from

³ Due to security concerns and confidentiality of the actual system information, all references to locations and facility names have been generalized and substituted with fictitious names.
EMRS, is required for accurate modeling of the regulator stations and compressor stations, e.g. configuration of the station and operational limits. That information was gathered from other sources like maintenance reports, construction drawings and manufacturer's datasheets. A detailed discussion about how the regulator stations and compressor stations were modeled is provided later in this work.

Each element and node has a unique name (up to 8 characters) that was preserved during the extraction and used as an identifier in the model. Keeping the original element and node names allows for easier update of the model in the future. Another reason for preserving the original node names is that each customer account is assigned to a node and it is easier to determine where a particular load should be applied into the model. However, since the model would not include the distribution systems, an area of influence (AOI) had to be developed to relate the individual loads on the distribution systems to the district regulator stations. A detailed discussion on the AOI is provided further in this work.

As it was mentioned earlier, due to difference in the modeling techniques and different requirements and restrictions, two versions of the model were required – a steady-state model and an unsteady-state model. The schematic of the steady-state model includes all of the pipes that are upstream of the district regulator stations. That means that even short pipes like stubs, drop sections and inlet piping to the regulator stations are included in the model. The presence of some of those elements, i.e. the stubs, does not contribute to the accuracy of the model but rather makes it more computationally intensive. At the same time the inlet and outlet piping of the regulator stations may be a cause for a significant pressure drop. Sometimes, it is a relatively long pipe with a small
diameter. That is why it is a good idea to eliminate the stubs, which just clutter the model, but keep the inlet and outlet piping. While it is practical to include the regulator station inlet and outlet piping in the steady-state model, it is almost impossible to have it in the unsteady-state model due to a restriction on the shortest length of the pipe elements that is determined by the base length.

Figure 18. Example of district and feeder regulator stations.

To eliminate the insignificant pipe elements for a steady-state model and to comply with the minimum length requirement for an unsteady-state model, a reduction of the
model schematic is required. Let us consider a portion of the system shown on Fig. 18. The high-pressure model would include only elements located upstream of the district regulator stations.

First, let us concentrate on the reduction process of the schematic for a steady-state model. The outlet piping between NODE7 and NODE8, and the 4-inch polyethylene distribution main would not appear in the model. The district regulator station would be represented as a node in the steady-state model, NODE6. An area of influence would be used to determine the distribution load that has to be applied at NODE6, in the steady-state model. The dead-end stub between NODE9 and NODE10 could be eliminated without any effect on the model because there is no load at NODE10, and therefore, there is no flow leaving the system at NODE9. All other pipe elements and the feeder regulator station would be included in the steady-state model. If two pipe elements with different physical properties have to be joined together in preparation of a steady-state model, the principle of equivalent lengths should be applied to determine the physical properties of the resulting pipe element. The feeder regulator station would be modeled between NODE2 and NODE3 with a set pressure of 300 psig at NODE3.

For the unsteady-state model, even further reduction of the schematic is required. All inlet and outlet piping would be eliminated. NODE4 and NODE5 would be converged in a single node, NODE4. This is required in order to eliminate the short pipe between the original NODE4 and NODE5 that would be shorter than the base length. NODE9 would be eliminated by joining the upstream and downstream pipe elements. If two pipe elements with different physical properties are joined together in preparation of an unsteady-state model, the principle of equivalent volume should be applied to determine
the physical properties of the resulting pipe element. The feeder regulator station would be connected directly from NODE1 to NODE4 and the pressure would be set to 300 psig at NODE4. The load from the original NODE6 would be moved to NODE4. All of the eliminated pipe elements have a relatively small size and are relatively short, and therefore, there is not significant volume, respectively capacitance, that could cause any significant transient effects to occur. The unsteady-state modeling is naturally much more computationally intensive than the steady-state modeling due to the different nature of the solved problems and the difference between the employed solution methods.

The reduced schematics of the steady-state and the unsteady-state versions of the model are shown on Fig. 19.

Figure 19. Schematic reduction for a) steady-state model and b) unsteady-state model.

The modeling software, i.e. SynerGEE®, provides model reduction functionality. However, it is important to preserve those nodes in the model that are required for maintaining the adequate representation of the system, e.g. those nodes where the load will be applied, or “tee” nodes like NODE1, NODE4, and NODE5 in Fig 18. If a given
node is included in a partial list of nodes, SynerGEE® can preserve it during a model reduction. A Visual Basic program, Inlet Piping Reduction Utility (IPRU), was developed to analyze the connectivity of the system and to create a list of significant nodes that have to be preserved. That list could be included in the problem definition file, which serves as input to the modeling software. The IPRU code is presented in the Appendix.

The originally extracted model consisted of more than 1,500 nodes and more than 1,700 node connecting elements (NCE). Through the reduction process the steady-state model was reduced to 980 nodes and 1046 NCEs, and the unsteady-state model was reduced to 521 nodes and 570 NCEs.

Area of Influence

The load in this model is applied at distribution regulator stations (DRS) and at nodes that represent the location of large customers on the high-pressure system. The allocation of the load on the distribution systems that are not modeled is determined by developing an area-of-influence map. An example AOI map is shown in Fig. 20.

The objective of creation of AOI is to determine how the load in a unit area is supplied and through which district regulator stations. In the example, a unit area is a section, denoted in Fig. 20 as SEC1 and SEC2. Usually the local distribution company (LDC) maintains records that have reference of each customer’s account to the section number or some other means of geographical grid, e.g. an atlas number or a tile with a constant size. Since there is a database with the accounts data and the corresponding section numbers available, it is easy to calculate the load for each section.
The areas of influence are determined by reviewing the distribution-system map and by exercising best judgment about where the “no-flow” point on each distribution main might be. The boundaries of the influence of each district regulator station are drawn on the map and the percentage of the load in each section is estimated for each influenced DRS. The sum of these percentages in each section must be 100%. For example, in SEC1, it is estimated that DR1 feeds 30% of the load in SEC1, DR2 feeds 60% of the load in SEC1, and DR3 feeds 10% of the load in SEC1. Note that the last is just the area in SEC1 that is influenced by DR3 and DR3 is located in the next section, SEC2 where it feeds 30% of the load in SEC2.

Let us assume, for example, that the estimated total peak-hour load in SEC1 is 20 mcfh and in SEC2 – 30 mcfh. Therefore, the load allocation by the area of influence, as shown in Fig. 20 is as follows:
The final allocation of the load is

- DR1 6 mcfh
- DR2 12 mcfh
- DR3 11 mcfh (2 mcfh in SEC1 and 9 mcfh in SEC2)
- DR4 15 mcfh
- DR5 6 mcfh
- Total 50 mcfh

The development of an area of influence is a time and effort consuming process.

Sometimes, a model of the distribution systems may have been created earlier and the load allocation at the district regulating stations can be determined from that model.

Once the AOI is created and the load allocation is calculated, a comparison of the load allocated at each DRS with the capacity of that DRS is required. The development of the AOI is based on many assumptions and judgments. Therefore, some district regulating stations may be overloaded. A comparison of the physical capacity of DRS with the load would show whether the station is overloaded or not. If there are indications that a regulator station is overloaded, an adjustment of its area of influence is required. Such adjustment would affect the adjacent areas as well. After all affected areas have been adjusted and the load allocation is recalculated, a new comparison of
load and capacity is performed. These iterations are repeated until all DRS have assigned load that does not exceed their physical capacity.

In the actual process of the development of AOI, a map of the entire gas distribution system was utilized and the analysis included approximately 300 district regulating stations. To determine the load by section, more than 400,000 actual customers accounts were analyzed. Individual regression equation for each customer account was derived expressing the gas usage as a function of heating degree day (HDD)\(^4\). The regression analysis was based on a whole heating season, October 1\(^{st}\) to March 31\(^{st}\). Each individual load was projected to the design HDD. Then the individual loads were summed together for each section and, based on the AOI, the load allocation was calculated.

Load and Load Profile

An important component of a model is the loading of that model. The load is actually a part of the boundary conditions imposed on the model. Two different approaches were used for loading the model subject to this work:

1) *Peak-hour load* is applied to the steady-state version of the model. The peak-hour load was determined by a regression analysis performed on actual gas usage data. A regression equation expressing the load as a linear function of HDD was derived. This equation is later used to load the model for any HDD that might be of interest. Planning studies are performed typically on a model with a load corresponding to a design-day HDD. The design HDD is determined by the gas distribution company’s policy for what HDD the system must deliver the contractual volumes at contractual pressure to the

\[\text{HDD} = 65^\circ F - \frac{\text{maximum daily temperature} + \text{minimum daily temperature}}{2}.\]

---

\(^4\) HDD is calculated as 65°F - (maximum daily temperature + minimum daily temperature)/2.

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customers. The design-day HDD could be the coldest day on record for a given area, the
coldest day for a given number of years, or determined by some other criteria. Designing
the system to survive a design-day event without outages is a sound industry practice
guaranteeing that the local distribution company (LDC) can provide reliable service to
the customers, bound by its contractual obligations.

2) An *intraday load profile* and a *daily volume* are used with the unsteady-state
version of the model. The actual gas usage by certain types of customers is not constant
throughout the day. However, there is a pattern in those variations. For example, the
residential customers use most natural gas in the morning when the heating equipment in
the house is working harder to warm the house after a cold night. Also most warm water
is used for showering in the morning hours. Some commercial customers, e.g. offices use
more gas in the beginning of the work day for space heating. The increased gas usage is
most pronounced between 7am and 9am and that is the time when the morning peak
occurs. Later, the space heating usage decreases due to increase of the ambient
temperature. At the end of a work day, when people come home and some gas is used for
warm water, cooking and space heating, another peak of the gas usage occurs. The
evening peak is not as well pronounced as the morning peak. It spans over longer period,
usually between 5pm and 11pm, and it is lower than the morning peak. While the
absolute values of the flow rate and the daily volume change with the weather, the pattern
remains and the ratio of the hourly gas usage to the average-hour gas usage remains
almost the same for a given group of customers.

Figure 21 represents the relationship between actual HDD and actual daily gas usage
for a group of customers, mostly residential and some small commercial customers,
measured at a pressure regulating station supplying gas to an area with mostly residential customers.

![Gas Usage Plot](image)

**Figure 21.** Linear approximation of gas usage.

The straight line is a representation of the best linear approximation of gas usage, based on least-squares fit of the data points. The equation of the line is

$$Q_{day}(HDD) = A \cdot HDD + B$$  \hspace{1cm} (7.1)

where the linear regression coefficients $A$ and $B$ are determined by least-squares fitting of the data points $(HDD, Q_{day})$. The constant $B$ represents the base load, i.e. the gas usage that is independent of the weather. The total load is a sum of the base load $B$ and heat-sensitive load, i.e. $A \cdot (HDD)$.

The data points used for Fig. 21 consist of values of HDD and daily volume $Q_{day}$. If a least-squares fitting is performed for a given time of the day, e.g. 8 o’clock in the morning, a similar relationship with the HDD can be derived,

$$Q(t, HDD) = A(t) \cdot HDD + B(t)$$  \hspace{1cm} (7.2)
where the regression coefficients $A(t)$ and $B(t)$ are functions of the time of the day. If there is flow rate data available and a least-squares fitting is performed for a given time point, the coefficients in Eq. (7.2) could be determined for that time point. The linear regression equation, in this case, is

$$Q_t = A_t \cdot HDD + B_t$$  \hspace{1cm} (7.3)

where $A_t$ and $B_t$ are the regression coefficients at time $t$. For example the 8am regression equation would be

$$Q_8 = A_8 \cdot HDD + B_8$$  \hspace{1cm} (7.4)

![Figure 22. Intraday profile for various HDD.](image)

If the day is divided into 24 even intervals between 25 time points, i.e. $t = 0, 1, \ldots, 24$, then 25 equations like Eq. (7.3) can be derived for each hour of the day. The number of unique equations will be 24, because the last equation ($t = 24$) will be the same as the first.
one \((t = 0)\). These equations can be used to estimate the flow rate at a given moment as a function of the HDD. Equation (7.2) represents a surface like the one presented in Fig. 22. The lines for constant HDD on that surface represent intraday load profiles.

In planning studies, typically the load profile for a design-day HDD is used. It is practical to normalize the intraday profile by dividing the flow rate to the average flow rate. The average flow rate can be obtained by dividing the volume by the time, for which that volume was delivered,

\[
Q_{av}(HDD) = \frac{\int_{t_0}^{t_{end}} Q(t)_{HDD} \, dt}{t_{end} - t_0} \tag{7.5}
\]

Where

- \(Q_{av}(HDD)\) average flow rate
- \(Q(t)_{HDD}\) flow rate as a function of time at a given HDD
- \(t_0, t_{end}\) starting and ending times

The integral in Eq. (7.5) yields the volume delivered for a period from \(t_0\) to \(t_{end}\). Due to a frequent need for integration, a user defined function, INTEGRAL, for MS Excel was developed. The VBA code of the function is presented in the Appendix. By dividing the instantaneous flow rate by the average flow rate, the profile is not only normalized but also non-dimensionalized,

\[
\bar{q}(t)_{HDD} = \frac{Q(t)_{HDD}}{Q_{av}(HDD)} \tag{7.6}
\]

Where

- \(\bar{q}(t)_{HDD}\) normalized, dimensionless flow at a given HDD
Figure 23 presents the normalized dimensionless flow as a function of time and HDD.

A few observations can be made. The shape of the profile changes with the HDD. The profile has more pronounced peak at lower HDD; the profile is "peakier". With increase of the HDD, corresponding to colder weather, the profile becomes flatter. The ratio of peak-hour flow to daily flow also changes with HDD. Figure 24 shows that relationship.
Figure 24. Peak-hour gas usage as percentage of daily flow.

An integration of any of the intraday profiles in Fig. 23, a constant-HDD line, would yield unity (1.0). Therefore, if the normalized non-dimensional profile is multiplied by the daily volume, each point of the profile would represent the instantaneous flow rate at that time. This property of the profile is utilized by SynerGEE® to account for load change at a given node. The daily volume and the load profile are provided as input to the software. The flow rate is calculated as a product of the daily volume and the dimensionless profile at each time step.

It is practical to use the same load profile for the same type of customers. In the unsteady-state model, subject to this work, a single profile for the mixture of residential and commercial customers was developed and applied at all district regulator stations.
Separate profiles were utilized for specific large industrial customers, such as power plants.

The load in the steady-state version of the model corresponds to the peak load determined by the profile used in the unsteady-state model. This guarantees coherency of the two versions of the model.

A planning study of the high-pressure feeder system is typically performed with a design-day load. An intraday load profile for the corresponding HDD is utilized. A design-day intraday dimensionless load profile is shown in Fig. 25.

Figure 25. Normalized intraday load profile.

If a longer simulation, e.g. five days at various HDD, is required, it is the best to combine the profiles at the required HDD into a single profile. This way, each-day load would be most adequately represented. Creating a single 5-day profile has certain
advantages, i.e. only one profile is provided to the software instead of five, the control of the simulation is simplified because there is no need to scale the profile for each day, and finally, a smooth transition between the days can be achieved. If a one-day profile is used in a multi-day simulation and it is manipulated every day of the simulation, there would be a "jump" in the load at the beginning of each day. The transition from one day to another could be made smoother by averaging the profile values for a couple of hours before midnight and a couple of hours after midnight. A five-day "smoothed" profile is shown on Fig. 26.

![Five-day "smoothed" load profile.](image)

Figure 26. Five-day "smoothed" load profile.

The above profile was compiled from four profiles for 28.5, 43, 43, 38, 25.5 HDD. It was used for all residential loads in the unsteady-state version of the model.
Regulator Stations

The actual district regulator stations and the pressure limiting stations are usually more complex than just a single regulator.

Most often the district regulator stations are designed as a single-stage regulator station with two regulators in series. The upstream regulator is usually set as a monitor and the downstream regulator is set as a worker. The worker is set to regulate the pressure to the MOP of the downstream system. The monitor is set usually just slightly higher than the MAOP of the downstream system. Because the worker’s set pressure is lower than the monitor’s set pressure, the monitor remains fully open under normal operating conditions. In case of a failure of the worker, the downstream pressure would increase and then the monitor will start regulating the pressure to its set pressure. The monitor is a mean of an overpressure protection in such a configuration.

If the pressure difference between the upstream and downstream systems is very large, the regulator station is designed as a two-stage regulator station. The maximum pressure differential across each stage of a regulator station is determined by the MAOP of the upstream and downstream systems, manufacturer’s specified maximum pressure differential, and the applicable standards, i.e. company design standards and safety regulations. The first stage is usually made of a single regulator, primary worker that is set to a lower pressure than the upstream pressure and higher than the MAOP of the downstream system. Its role is to provide adequate inlet pressure for the second stage. The second stage is designed usually as a worker and monitor set.

Most of the regulator stations are single-run regulator stations. Sometimes it is not practical to design a regulator station as a single-run configuration. If the maximum flow
through the station is very high and chances are that it will occur very rarely and most of
the time the flow will be significantly lower than the maximum, then it is more practical
to design the station as a dual-run station with two parallel worker-monitor runs. An
illustration of various configurations of an actual regulator stations is shown in Fig. 27.

![Diagram of regulator station configurations]

Figure 27. Basic regulator station configurations.

In a dual-run configuration the set pressure for each run is different. The primary run
that would be operating most of the time is set to a higher pressure than the secondary
run. The regulators of the secondary run are closed due to a higher downstream pressure
than their set pressure. When the flow through the primary run reaches the maximum,
i.e. choking or sonic conditions, the regulators on the primary run are fully open and they
can not hold the set pressure. The downstream pressure starts drooping and when it
reaches the set pressure of the secondary run, the worker regulator on the secondary run
starts regulating.

Besides the configurations shown in Fig. 27, two other unique configurations were
found on the system. There was a large pressure limiting station (PLS) built with two
parallel runs, feeding the same downstream system, but supplied from two different upstream systems. Each run had three large regulators in series designated as primary monitor, worker and secondary monitor. Each run could be set to either downstream or upstream pressure control, meaning that the set pressure could be set at the outlet or at the inlet of the station. All six regulators of the station were actuated large-size ball valves. Also, each run could be switched to flow control mode. Each run of that PLS was modeled as separate regulator station. In that case, it was easier to simulate the various operating regimes by just changing the controlling parameters.

Also, there was a regulator station consisting of a single actuated ball valve and a relief valve as an overpressure protection devise.

The behavior of a complex regulator station could be better simulated by modeling the station with its actual configuration and various set pressures. The commercial modeling software, i.e. SynerGEE® allows input of the configuration and the individual parameters of each regulator. Only the feeder regulator stations, i.e. pressure limiting stations and district regulator stations between the various levels of the high-pressure system were modeled. A catalog of over 50 stations was created. Operational and manufacturer's information about more than 200 individual regulators of 16 different types and sizes was gathered and manipulated to prepare the input data for the model. Various regulator flow equations used in the model are presented in the Appendix.
Compressor Station

Large compressor stations are typically comprised of more than one compressor unit. Various compressor station configurations can be utilized, depending on the application and the regimes of operation. The two basic configurations, series and parallel, are shown in Fig. 28.

![Diagram of compressors in series and parallel]

Figure 28. Basic compressor station configurations.

The decision which configuration should be utilized is based on two main factors - flow through the station and compression ratio. Each compressor unit has physical limits such as maximum engine power, maximum flow rate, and maximum head that translates to maximum compression ratio at given pressure conditions.

A configuration of two compressor units in series is used when relatively low flow has to be compressed to higher compression ratio. In that case, the flow through the individual compressors is the same, but the station compression ratio is higher than the individual units' compression ratio. The following relations are valid

---

5 The term *compressor unit* is usually used for a compressor set consisting of a driver (engine) and a compressor. Large centrifugal compressors are typically driven by gas-turbine engines.
\[ Q = Q_1 = Q_2 \]  
(7.7)

\[ R = \frac{P_{\text{DISP}}}{P_{\text{SUC}}} = R_1 R_2 \]  
(7.8)

When higher flow rate is required at lower compression ratios, a parallel configuration is utilized. In that case, the total flow rate will be equal to the sum of the flow rates of each individual compressor. The station's compression ratio will be the same as the compression ratio of each individual compressor due to common suction and discharge pressures. The following relations hold

\[ Q = Q_1 + Q_2 \]  
(7.9)

\[ R = R_1 = R_2 \]  
(7.10)

There is one compressor station present in the system that was modeled. The compressor station consists of two centrifugal compressors. The two compressors are connected to two separate systems operated at different pressures. Even if the two compressor units are not designed to work together, the modeling software still provides certain advantages for modeling those compressor units as single-unit compressor stations. The advantages include more precise control of the operation during the simulation, utilization of actual performance maps of the compressors, account for pressure drop due to station piping, simulation of intermediate and after cooling, and more detailed reporting of the operational parameters. These advantages justified the modeling of the compressors as compressor station elements.

More realistic simulation of the operation of the compressor stations is achieved by modeling the compressors as compressor stations, which requires more detailed input of the operational parameters of the individual compressors and the compressor station. The
compressors were modeled with their full performance maps as described in Chapter 4. Manufacturer’s data was obtained and used for both compressors.

An important parameter of a compressor model is the fuel usage. Natural gas from the pipeline is used as fuel for the engines driving the compressors. The fuel is typically calculated and applied as a load at the upstream node of the compressor element. Doing so, it is guaranteed that the fuel is included in the flow upstream of the compressors.

As it was outlined in Chapter 4, there are two basic approaches, nominal fuel rate (NFR) and polynomial approximation of the fuel rate as a function of the engine load, respectively developed net power. The NFR method estimates the fuel rate accurately at conditions that are close to full load, i.e. the engine net power is close to its rated power. At lower load, the error could be significant. It was known from actual historical data that the compressors operate at various regimes, i.e. a wide range of flow. Therefore, the net output power developed by the engines varies widely.

For existing compressors, there is usually plenty of field data or manufacturer’s data that could be used to derive the polynomial function, i.e. Eq. (4.19). Manufacturer’s data about the actual compressors is presented in Table 1.

<table>
<thead>
<tr>
<th>LOAD [%]</th>
<th>NET OUTPUT POWER [HP]</th>
<th>FUEL FLOW [MMBtu/hr]</th>
<th>FUEL FLOW [scf/hr]</th>
<th>FUEL RATE [scf/HP.day]</th>
<th>FUEL USED [scfd]</th>
</tr>
</thead>
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<td>586</td>
<td>17.02</td>
<td>17020</td>
<td>697.065</td>
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<td>20</td>
<td>781</td>
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<td>18780</td>
<td>577.106</td>
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<tr>
<td>40</td>
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<tr>
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<td>2344</td>
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<td>37.16</td>
<td>37160</td>
<td>228.326</td>
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</tr>
</tbody>
</table>

Table 1. Compressor fuel consumption data.
For the actual modeled compressors, operational data for various engine load was obtained from the compressor manufacturer. The data was manipulated and the fuel rate was converted from thermal units to volumetric units. The thermal energy content of the gas was assumed to be 1000 BTU/scf.

MS Excel® was used to fit the data by LINEST worksheet function and calculate the polynomial coefficients of Eq. (4.19). LINEST is a least-square linear fitting function. In order to use it, the power and the square of the power were treated as independent variables, and the used fuel as a function of these variables. The results of the regression are as follows:

\[ C_0 = 307696.9076, \quad C_1 = 187.745822, \quad C_2 = -0.010221, \]
\[ r^2 = 0.99794, \text{ Coefficient of determination}^6 \]
\[ SE_y = 725.968, \text{ Standard error of the } y \text{-estimate.} \]

Equation (4.19) becomes

\[ F = 307696.9076 + 187.745822 \cdot P_{hp} - 0.010221 \cdot P_{hp}^2 \]  

where

\[ F \quad \text{fuel rate (standard cfh/hp-day)} \]
\[ P_{hp} \quad \text{developed power by the engine (hp)} \]

The coefficient of determination indicates a very good correlation between the engine power and the fuel rate. Therefore, Eq. (7.11) is an adequate representation of the fuel rate as a function of the net output power developed by the compressor engine.

---

\(^6\) The coefficient of determination is an indicator of how well the equation resulting from the regression analysis explains the relationship among the variables.
Figure 29 shows the comparison between the polynomial and the nominal fuel rate approximations.

The difference between the two approximations is noticeable. The polynomial approximation given by Eq. (7.11) is very close to the actual fuel rate. The NFR line is very close to the actual curve and the polynomial curve at high values of the net output power. The net fuel rate and the polynomial approximations show huge divergence at the low end of the power range.
Validation of The Model

After the model was assembled, it was necessary to validate it. A validation of the model serves two main purposes: 1) Adjusting some physical properties like pipe roughness and efficiency to make the model behave as close as possible to the actual pipeline system. 2) Evaluating the accuracy of the model and its reliability.

Many of the physical parameters are originally set to some average or default values. For example, the default pipe roughness was set to the average value for a new steel pipe of 0.0015 inches (0.038 mm) and efficiency of 100%. These two parameters appear in the fundamental pipe flow equation. A match of an actual pipe flow and inlet and outlet pressures, measured in the field, can be achieved by an adjustment of the efficiency and roughness values. Another reason for adjusting these values is that various pipe segments were installed at different times and the age of those segments may be a significant factor in changing the actual pipe roughness. The gas contains some hard particles and liquid drops that act like abrasive agents. The older pipes become smoother than the newer pipes due to the scrapping of the internal pipe surface by the impurities in the gas. Therefore, the assumed roughness of the pipe has to be adjusted to represent realistically the field condition of the pipe.

Besides the roughness of the internal surface of the pipe, the efficiency of the pipe may change with the aging of the pipe segments. There are locations where a build up of materials may occur and the effective diameter of the pipe, i.e. its open cross sectional area, will be reduced. Also, the local losses due to fittings were neglected when the model was put together. The pipe efficiency can be adjusted to account for local losses...
and reduction of the pipe cross sectional area. With aging, the pipe efficiency decreases due to the increase of local losses.

The actual system consists of two main parts – transmission system and a high-pressure feeder loop. These systems were validated separately, due to expected differences in their operational behavior. The transmission system is approximately 100 miles long. It includes two main parallel large-diameter pipelines with loops at some locations. The transmission lines transport gas from two separate taps on two different interstate pipelines to a large pressure limiting station. Due to the large diameter of the pipe, long distance and high operating pressure, the transmission system has large volume and, respectively high capacitance. That system is relatively old and the roughness of its pipes was expected to be low. Also, there are many elevation changes and drop sections where the probability of accumulation of liquids and build ups is high. That was an indication for expecting lower flow rates due to higher local losses. Therefore, the efficiency of the pipes had to be adjusted.

Plenty of reliable measurement data from the Company’s SCADA system was available for various points along the transmission pipelines. The validation was performed for steady-state operation. An unsteady-state validation is much more complex and would require much more data to be analyzed and more parameters to be monitored and compared with the field data. The average efficiency of the transmission pipelines was determined to be 97%. The average effective roughness of the pipes was estimated at 0.0006 inches (0.015 mm). These values were confirmed later by the real-time pipeline simulator.
The validation of the high-pressure feeder loop was performed for steady-state operation as well. That system is extensively looped, as can be seen in Fig. 17. It operates at relatively low pressure, below 300 psig (2 MPa), the pipe sizes are smaller and the distances between the connections are short. The capacitance of the system is relatively small. Therefore, that system is most likely to operate at steady-state conditions. The primary use of the developed model would be for planning studies. Planning analysis is conducted usually for peak-hour conditions. This is why peak-hour operational data was extracted from the SCADA system and compared with the model data.

The coldest winter day during the last heating season was selected as a reference. The peak-hour pressures and flows at the city gates/taps and the pressure limiting stations were matched with the actually measured values from the SCADA. The load at various locations on the model, i.e. check points, was set equal to the actual peak-hour load. The resultant pressures at those locations on the model were compared with actual data from Company’s SCADA system. Those check points were selected among large customers that were represented as separate nodes in the model. Such locations usually have reliable flow and pressure measurement facilities. The number of the check points was 32 and they were spread as evenly as possible over the entire feeder loop. There was at least one point on each separate system operating at different pressure. The average efficiency and roughness of the pipes in the feeder loop system were adjusted to produce best match of the pressure and the flow predicted by the model with the actual pressure and flow at the check points. The best match was achieved with efficiency of 96% and effective roughness of the pipes on various systems from 0.0005 to 0.0007 inches (0.013
to 0.018 mm). The maximum deviation from the actual pressure at the check points was in the range of ± 3%.

The efficiency and the roughness values that were determined for various subsystems in the steady-state version of the model were applied in the unsteady state version as well.
CHAPTER 8

ANALYSIS OF THE RESULTS

The developed model was used to analyze the multilevel feeder system. The primary goal was to determine what improvements on the system are required to sustain a three-day peak event without outages. The planning horizon was 10 years. Due to population growth in the service area, it was expected that some major system reinforcements would be required.

Along with the planning study, some parameters of the system were evaluated, such as change of the imbalance between demand and supply, system response time and dampening of the load profile caused by mixing large number of customers, including large industrial customers. Those parameters were quantified and taken into consideration for further analyses.

Imbalance

Total system imbalance between demand and supply was defined as the difference of system’s total load, i.e. flow exiting the system, and total supply, i.e. flow entering the system. In the same manner, imbalance was defined for the transmission system and for the feeder loop system, individually.

The imbalance equals zero under steady-state conditions, i.e. the flow entering the system is equal to the flow leaving the system.
Under unsteady-state conditions, the imbalance changes during the simulation. A change in the load causes change in the supply. However, it takes time to propagate the change from the load locations to the supply sources. Also, portion of the load is temporarily supplied from the available line pack in the system. When the load increases rapidly, part of the line pack is converted to flow leaving the system. Therefore, the pressure at the load locations decreases. The utilization of the available line pack for peak shaving is very common in the real operation of the pipeline systems. However, this requires enough useful line pack. Otherwise, the system could be brought to a lower-than-the-acceptable minimum pressure. Temporarily lowering the system line pack might be acceptable, if it could be recovered later. Generally, the imbalance, as it was defined above, is proportional to the change of the line pack, i.e. negative imbalance means that the systems line pack is reduced, because part of it is used to compliment the supply, and vise versa, positive imbalance means that the system is being packed, i.e. the line pack is increasing.

The imbalances in the transmission system, the feeder loop system, and the total imbalance in the system as a whole were compared. The imbalance was expressed as a percentage of the load on each of the systems. The comparison is presented in Fig. 30. The transmission system exhibits the greatest imbalance, while the imbalance in the feeder loop system is much smaller. The transmission system is made of long large-diameter pipes that operate at higher pressure than the feeder loop does. This means that the transmission system has much greater line pack that could be utilized. Transient events such as drafting and packing can naturally occur in that system. The capacitance, i.e. the ratio of the steady-state flow to the line pack in the transmission system is much
higher than the capacitance of the feeder loop. These differences explain the difference in the specific imbalance between the two main components of the actual system. The transmission system operates under unsteady-state conditions.

![Figure 30. Normalized system imbalance.](image)

On the other side, the feeder loop operates under conditions that are close to steady state due to its low capacitance. The imbalance of the entire system is determined mostly by the imbalance of the transmission system.

Figure 31 illustrates the integral imbalance of the entire system. It represents the accumulated imbalance since the beginning of the simulation. The simulation starts with a fully balanced system, i.e. the imbalance is equal to zero. Then, it is drafted down until the first-day peak is reached at 8 am, due to increase of the load. Later, the system is
recovered and even packed in the afternoon. Next three days of the simulation are peak days. The system is drafted down and does not have a chance to fully recover during that period. It takes approximately 12 hours of the fifth day to reach a full recovery. Later, the system is even packed.

The accumulated imbalance is an indicator of how well packed the system is. It could be used to evaluate the amount of line pack with respect to some reference conditions at a given time. For example, Fig. 31 shows that the system had about 6000 mcf less line pack at the 56th hour of the simulation than it had at the beginning of the simulation, i.e. more gas exited the system than entered it.

Figure 31. Integral system imbalance.
Response Time

The response time of a system is the time it takes to the system to respond to a certain event. In this study, the response time was measured as a lag between two identical phases of the load and supply curves. Figure 32 illustrates the concept.

![Figure 32. System response time.](image)

There is almost no lag between the load and supply curves of the feeder loop. That system responds almost immediately and the supply is almost equal to the demand at all times. In other words, the flow entering the system is equivalent to the flow exiting the system. The very short response time means that the system could be considered a “steady-state” system.

Unlike the feeder system, the transmission system experienced 1:00 to 1:50 hours lag between identical phases of the load and supply curves. The average response time of that system is about 1:40 hrs. The transmission system is an “unsteady-state” system, i.e., as opposed to a “steady state” system, the flow entering the system is not equal to the flow exiting the system. There is more room for some transient effects to occur on such a system.
Dampening Effect

The load profile was applied at the district regulator stations. Most of the district regulator stations are fed off the feeder loop. Figure 33 represents a comparison between the load profile, the profile of the total load on the feeder loop, and the profile of the supply to the feeder loop.

Figure 33. Load profile dampening effect.

All of the three curves are normalized and dimensionless, i.e. the instantaneous flow rates were divided by the corresponding daily average flow rates. It is visible that the resulting load and supply curves have lower peaks and higher deeps, i.e. they are flatter than the original load profile. The amplitude of the profile is reduced by approximately 6%. The effect is mostly due to mixing weather-sensitive loads with firm loads between

95
the load locations and the supply sources. On more “transient” systems, the available line pack could contribute to the dampening effect.

There is usually reliable flow metering equipment present at the city gates and the large PLS. At the same time, there is rarely any flow measurement present at the district regulator stations. This justifies the development of a load profile utilizing flow measurement data at the city gates and/or pressure limiting stations and applying it at the district regulator stations. In such a case, the dampening effect should be taken into account.

The load profile should be scaled before applying it at the DRS to account for the dampening effect. The scaling factor could be calculated as a ratio of the total load (including both heat-sensitive and not heat-sensitive loads) to the heat-sensitive load only.

Facility Requirements

The main goal of modeling the high-pressure system was to determine what system reinforcements are required and, when they are required, to keep the system capable of sustaining a design-day event in the future. A 10-year planning horizon was assumed.

The load volume was determined upon a long-range growth forecast and its distribution was based on current development data for the first three years and estimated distribution of the available land, after that.

Certain assumptions were made and a set of criteria was established. Among the criteria are minimum acceptable pressures on the feeder loop, minimum contractual pressures that have to be maintained during operation, minimum supply pressures and maximum supply rates at some of the city gates, MAOP of the individual systems, and
maximum discharge pressure at the compressors and set pressures at the feeder regulator stations.

The model was loaded for each year with the forecasted load and load distribution. A steady-state balance was sought initially. Then, key pressure and flow data from the balanced solution were used as initial conditions for the unsteady-state simulation.

Due to the increasing load each year, some of the criteria were being violated. That prompted certain system reinforcements. The required facilities were incorporated in the model and a balanced solution of the steady-state model was sought. The process continued until a balanced solution of the network was found and all of the criteria were met. An unsteady-state simulation was run then to test the system with the increased load for each year under unsteady-state conditions. If any of the criteria were violated, the improvements determined by the steady state solution were incorporated in the model and the simulation was repeated until all of the criteria were met. These steps were repeated for each year.

Sometimes alternative reinforcements had to be evaluated and the more economical ones were carried over to the following years.

The steady-state mode was instrumental in revealing problems such as long and small-size inlet piping to a district regulator station, or a too small regulator that needs to be replaced with a larger one to increase the capacity of the station.

Upon completion of the analysis, a list of required facilities by year was compiled, based on the analysis' results. Examples of required facilities include looping of existing pipeline segments, upgrade of city gates, MAOP upgrades, installations of new feeder regulator stations or upgrades of existing stations.
CHAPTER 9

CONCLUSIONS

In the development of a model of a real natural gas transmission or distribution system, attention was given to many factors, such as gas properties, pipeline properties, type of the system (transmission, feeder, or distribution), geographical configuration of the pipe network, other elements (valves, regulators, compressors, storage fields) and their parameters, type of customers (residential, commercial, or industrial), climate, operational parameters (flow, pressure, gas temperature, ambient temperature), contractual parameters (daily quantities, hourly flow variations, minimum delivery pressure). An accurate model should account for all of these factors.

Input from various systems was used in the process of the development of the model. The schematic of the pipeline network along with some attributes representing physical or operational parameters, such as pipe diameter, length, roughness, regulator station configuration, regulator set pressure, customer location and meter capacity can be extracted from Geographic Information System (GIS). The elevations of the nodes in the network were determined accurately by utilization of a Digital Elevation Model (DEM). Data from Supervisory Control and Data Acquisition (SCADA) system was collected and analyzed to derive load profiles and to validate the model. Data from customers billing system was used to determine the load for specific categories of customers.
There are two major types of gas pipeline network models, based on the type of the analysis they are used for. A planning study of the system requires a model that can be modified to include future pipeline elements, as well as other elements as valves, regulators, and compressors. Determination of the design parameters of those elements is usually the main goal of a planning study. Often, in a planning model, the initial conditions and the boundary conditions reflect the “worst-case” scenario, i.e. maximum load and minimum supply pressures. The second type is operational model that is used to analyze “what-if”, i.e. operational scenarios. An operational model is more detailed and represents an existing system as realistically as possible. The initial conditions and the boundary conditions reflect the actual operational conditions, i.e. actual load, actual pressure, actual configuration of pressure controlling elements and compressors and actual compressor performance. Often an operational model is used to analyze a line break or outage required for maintenance.

The developed model was initially used in planning studies to determine what facility improvements are required to keep the system capable of sustaining a design-day event. Later the steady-state version of the model was used for operational studies, i.e. “what if” scenarios were explored.

The model was validated and a great confidence in the reliability of the model was gained.

The evaluation of the behavior of the various parts of the system helped to better understand how these individual systems work together. It also facilitated the determination of what approach to be used for future analyses of those systems. For example, the feeder loop can be confidently modeled as a steady-state system, while the
transmission system most likely would be modeled with the unsteady-state version of the model.

The facility requirements determined as a result of the planning studies with the model are used in the long-range capital investment plan.

The significance of modeling complex natural gas systems is proven, whether it is an every-day operational analysis or a long-range planning analysis. An adequate model is instrumental in a multimillion-dollar operational or planning decision making process. An optimization of the system’s operation, based on an accurate model, could substantially increase the efficiency of the operation and lower the operational cost.
APPENDIX

Pipe Flow Equations

The “practical” flow equations are discussed by Schroeder in “A Tutorial on Pipe Flow Equations” [5] presented during the 2001 PSIG Conference in Salt Lake City, UT.

*Spitzglass Equation:*

\[
Q_s = 1172d^3e \sqrt{\frac{(P_1 - P_2) P_{av}}{LG(d + 3.6 + 0.03d^2)}} \quad \text{Low pressure (inches H}_2\text{O)}
\]

\[
Q_s = 1128d^3e \sqrt{\frac{P_1^2 - P_2^2}{LG(d + 3.6 + 0.03d^2)}} \quad \text{High pressure (psia)}
\]

The equivalent friction factor for this equation is

\[
f = \frac{4}{354} \left(1 + \frac{3.6}{d} + 0.03d\right).
\]

The Spitzglass equation allows the friction to vary with the diameter of the pipe. The friction factor decreases with the increase of the diameter, assuming constant roughness. This is valid, with the Spitzglass equation for diameter up to 10.95 inches. For larger diameters, the friction would start increasing instead of decreasing.

*Weymouth Equation:*

\[
Q_s = 433.49 \frac{T_b}{P_b} d^8e \sqrt{\frac{P_1^2 - P_2^2 - 0.0375G(z_2 - z_1)P_{av}^2}{Z_{av} T_{av} L GT_{av} Z_{av}}}. 
\]
The equivalent friction factor is

\[ f = \frac{4}{(11.18d^4)^2} \]

and it decreases consistently with increasing of the pipe diameter. With proper tuning of the efficiency, the Weymouth equation could work well for high flow pipelines.

Panhandle A Equation:

\[ Q_s = 435.87 \left( \frac{T_h}{P_h} \right)^{1.0788} d^{2.6182} e \left( \frac{P_1^2 - P_2^2 - 0.0375 G(z_2 - z_1) P_{av}^2}{Z_{av} T_{av} Z'_{av} T'_{av}} \right)^{0.5394} \]  

(English)

\[ Q_s = 0.0045965 \left( \frac{T_h}{P_h} \right)^{1.0788} d^{2.6182} e \left( \frac{P_1^2 - P_2^2 - 0.0683 G(z_2 - z_1) P_{av}^2}{Z_{av} T_{av} Z'_{av} T'_{av}} \right)^{0.5394} \]  

(Metric)

The equivalent friction factor is

\[ f = \frac{4}{(6.78 \text{Re}^{0.07305})^2} \cdot \]

The Panhandle A equation was developed and it is appropriate for relatively low Re.
Panhandle B Equation:

$$Q_s = 737 \left( \frac{T_b}{P_b} \right)^{1.02} d^{2.55} e \left( \frac{P_1^2 - P_2^2 - 0.0375G(z_2 - z_1)P_{av}^2}{Z_{av}T_{av}} \right)^{0.51}$$  \hspace{1cm} \text{(English)}$$

$$Q_s = 0.010019 \left( \frac{T_b}{P_b} \right)^{1.02} d^{2.55} e \left( \frac{P_1^2 - P_2^2 - 0.06835G(z_2 - z_1)P_{av}^2}{Z_{av}T_{av}} \right)^{0.51}$$  \hspace{1cm} \text{(Metric)}$$

The equivalent friction factor is

$$f = \frac{4}{\left(16.49 \text{Re}^{0.01961}\right)^2}.$$  \hspace{1cm} (1.1)

The Panhandle B equation is more applicable for higher Reynolds numbers than the Panhandle A equation.

The IGT Equation:

$$Q_s = 92.66 \frac{T_b}{P_b} G^{\frac{1}{2}} \mu^{\frac{1}{2}} \left( \frac{P_1^2 - P_2^2 - 0.0375G(z_2 - z_1)P_{av}^2}{Z_{av}T_{av}} \right)^{\frac{1}{3}}$$  \hspace{1cm} \text{(English)}$$

$$Q_s = 0.0012753 \frac{T_b}{P_b} G^{\frac{1}{2}} \mu^{\frac{1}{2}} \left( \frac{P_1^2 - P_2^2 - 0.06835G(z_2 - z_1)P_{av}^2}{Z_{av}T_{av}} \right)^{\frac{1}{3}}$$  \hspace{1cm} \text{(Metric)}$$

The equivalent friction factor is

$$f = \frac{4}{\left(4.619 \text{Re}^{0.1}\right)^2}.$$  \hspace{1cm} (1.1)
All of these flow dependent equations share one common attribute: at low flow they are conservative and under-predict flow, while at high flow, overly optimistic and over-predict flow. The main difference is where low and high flow are defined [5].

Regulator Equations

General Regulator

\[ Q = 2C_g \sqrt{(P_1 - P_2)P_2}, \quad \text{(subcritical)} \]

\[ Q = C_g P_1, \quad \text{(critical)} \]

Fisher Regulator [8]

\[ Q = \frac{520}{GT} C_g P_1 \sin \left( \frac{59.64}{C_1} \sqrt{\frac{P_1 - P_2}{P_1}} \right) \text{Radians} \quad \text{(subcritical)} \]

\[ Q = \frac{520}{GT} C_g P_1 \quad \text{(critical)} \]

where

\[ Q \quad \text{flow rate (scfh)} \]

\[ G \quad \text{gas specific gravity (dimensionless)} \]

\[ T \quad \text{gas temperature at inlet (°R)} \]

\[ P_1 \quad \text{upstream pressure (psia)} \]

\[ P_2 \quad \text{downstream pressure (psia)} \]

\[ C_g \quad \text{gas sizing coefficient (scfh/psi)} \]
Grove Regulator

\[ Q = 59.6 C Y_1 P_1 \sqrt{\frac{\Delta P}{P G Z}}, \quad \frac{\Delta P}{P_1} < X_i F_k \]  

\[ Q_c = C_1 C P \sqrt{\frac{F_k}{G Z}}, \quad \frac{\Delta P}{P_1} \geq X_i F_k \]  

where

\[
C = \begin{cases} 
0 & \text{when } \Delta P \leq 0.8 E \\
C_p \left( \frac{\Delta P - 0.8 E}{1.2 E} \right) & \text{when } 0.8 < \Delta P < 2 E \\
C_p & \text{when } \Delta P \geq 2 E 
\end{cases}
\]

\[
Y_i = \begin{cases} 
1 & \text{when } \frac{\Delta P}{P_1} < 0.02 \\
1 - \left( \frac{0.333 \Delta P}{X_i P F_k} \right) & \text{when } 0.02 < \frac{\Delta P}{P_1} < X_i F_k \\
0.667 & \text{when } \frac{\Delta P}{P_1} \geq X_i F_k 
\end{cases}
\]

- \( Q \) gas flow rate (scfh)
- \( Q_c \) critical gas flow rate (scfh)
- \( G \) gas specific gravity (dimensionless)
- \( P_1 \) inlet pressure (psia)
- \( P_2 \) outlet pressure (psia)
- \( \Delta P \) pressure differential (psid)
- \( C \) partially open capacity factor
- \( C_p \) wide open capacity factor
- \( C_1 \) critical flow capacity factor
- \( F_k \) specific heat ratio factor
$E$  
Tube expansion factor

$X_t$  
Critical pressure drop ratio

$Y_1$  
Expansion coefficient

$Z$  
Compressibility factor

$C_p, C_1, E, X_t$ are provided by the manufacturer for various sizes and pressure ranges.

**Units Conversion Factors**

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<td>2.237 mi/minute</td>
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<tr>
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<tr>
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<td>1/6 sq yd</td>
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<tr>
<td>square meters</td>
<td>2.471 x 10⁴ ac</td>
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<td>Temp (Degs C) + 32</td>
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<tr>
<td>Temp (Degs F) + 460</td>
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<tr>
<td>Temp (Degs F) - 32</td>
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<td>10^1</td>
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<tr>
<td>Therms Per Hour</td>
<td>10^1</td>
</tr>
<tr>
<td>Therms</td>
<td>10^1 (Btu per cf)^1</td>
</tr>
<tr>
<td>Therms Per Hour</td>
<td>10^1 (Btu per cf)^1</td>
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<tr>
<td>Therms Per Hour</td>
<td>10^1 (Btu per cf)^1</td>
</tr>
<tr>
<td>Thousand Cubic Feet</td>
<td>10^1 (Btu per cf)^1</td>
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<tr>
<td>Thousand Cubic Feet per Hour</td>
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<td>Tons (Long)</td>
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<td>Tons (Long)</td>
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<tr>
<td>Tons (Short) per sq in</td>
<td>2000</td>
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**Glossary of Gas Industry Terms**

**Absolute Pressure**  
Gauge pressure plus barometric pressure. Absolute pressure can be zero only in a perfect vacuum.

**Absolute Viscosity**  
The measure of a fluid's tendency to resist flow, without regard to its density. By definition, the product of a fluid's kinematic viscosity times its density.

**Absolute Zero**  
The zero point on the absolute temperature scale. It is equal to -273.16 degrees C, or 0 degrees K (Kelvin), or -459.69 degrees F, or 0 degrees R (Rankine).

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**Adiabatic**  A term indicating that no heat is lost or gained by a material being subjected to a thermodynamic process. An adiabatic process is one in which there is no exchange of heat with the surroundings.

**After-Cooling**  The process of cooling a compressed air or gas immediately after compression.

**American Gas Association (AGA)**  Trade group representing natural gas distributors and pipelines.

**Base Conditions**  The ANSI Z132 has established 60°F and 14.73 psia as the base temperature and pressure to which all volumes are commonly referred.

**Base Load**  As applied to gas, a given consumption of gas remaining fairly constant over a period of time, usually not temperature-sensitive.

**Base Pressure**  The pressure used as a standard in determining gas volume. Volumes are measured at operating pressures and then corrected to base pressure volume. Base pressure is normally defined in any gas measurement contract. The standard value for natural gas in the United States is 14.73 psia, established by the American National Standards Institute as standard Z-132.1 in 1969.

**Btu per Cubic Foot**  A measure of the heat available or released when one cubic foot of gas is burned.

**Btu, Dry**  Heating value contained in cubic foot of natural gas measured and calculated free of moisture content. Contractually, dry may be defined as less than or equal to seven pounds of water per Mcf.

**Btu, Saturated**  The number of Btus contained in a cubic foot of natural gas fully saturated with water under actual delivery pressure, temperature and gravity conditions. See BTU, DRY.

**Capacity, Effective**  The maximum load which a machine, apparatus, device, plant, or system is capable of carrying under existing service conditions.

**Capacity, Heat**  See HEAT CAPACITY.

**Capacity, Installed**  The maximum load for which a machine, apparatus, device, plant, or system is designed or constructed, not limited by existing service conditions.

**Capacity, Peaking**  The capacity of facilities or equipment normally used to supply incremental gas under extreme demand conditions; generally available for a limited number of days at maximum rate.

**Chill Factor**  The temperature (at zero wind velocity) which would produce the same chilling effect as a particular combination of temperature and wind velocity. See also WIND CHILL FACTOR.

**City Gate Station**  Point at which a distribution gas company receives gas from a pipeline company. See GATE STATION.

**Compressibility**  The property of a material which permits it to decrease in volume when subjected to an increase in pressure. In gas-measurement usage, the compressibility factor "Z" is the deviation from the ideal Boyle and Charles' law behavior. See SUPERCOMPRESSIBILITY FACTOR.

**Compressibility Factor**  See SUPERCOMPRESSIBILITY FACTOR.

**Compression**  The action on a material which decreases its volume as the pressure to which it is subjected increases.

**Compression Cycles**  Adiabatic (isentropic) compression takes place when there is not heat added to or removed from the system. Compression follows the formula $p_1V_1^k = p_2V_2^k$, where exponent $k$ is the ratio of the specific heat capacities. Although an adiabatic cycle is never totally obtained in practice, it is approached typically with most positive-displacement machines and is generally the base to which they are referred. Isothermal compression takes place when the temperature is kept constant as the pressure increases, requiring continuous removal of heat generated during compression. Compression follows the formula $p_1V_1 = p_2V_2$. However, in practice it is never possible to remove the heat of compression as rapidly.
as it is generated. Polytropic compression is a compromise between the two basic processes, the adiabatic and the isothermal. It is primarily applicable to dynamic continuous-flow machines such as centrifugal or axial compressors. Compression follows the formula \( p_1 V_1^n = p_2 V_2^n \), where exponent \( n \) is experimentally determined for a particular type of machine. It may be lower or higher than the exponent \( k \) used in adiabatic cycle calculations.

**Compression Efficiency** The ratio of the theoretical work requirement (using a stated process) to the actual work required to compress a given quantity of gas. It accounts for the gas friction losses, internal leakage and other variations from the idealized thermodynamic process.

**Compression Ratio** The relationship of absolute outlet pressure at a compressor to absolute inlet pressure.

**Compressor** A mechanical device for increasing the pressure of a gas.

**Compressor Fuel** Natural gas consumed by the engines in a compressor station, reported as a percentage of the gas transported through the station. Thus, a transporter whose gas passes through one or more compressor stations will be entitled to take delivery of less than 100 percent of the gas introduced into the pipeline network.

**Compressor Station** Any permanent combination of facilities which supplies the energy to move gas at increased pressure from fields, in transmission lines, or into storage.

**Compressor Stations** Locations along the interstate pipeline at which large (thousands of horsepower) natural gas-powered engines increase the pressure of the market natural gas stream flowing through the station by compression.

**Consumption** The quantity of natural gas used by ultimate consumers.

**Content of Fuel** The heat value per unit of fuel expressed in Btu as determined from tests of fuel samples. Examples: Btu per pound of coal, per gallon of oil, per cubic foot of gas.

**Contract Demand (CD)** The amount of the system's capacity to deliver gas which a natural gas pipeline or distributor agrees to reserve for a particular customer and for which the customer agrees to pay a demand charge as specified in the applicable tariff. Also, the daily quantity of gas which a supplier agrees to furnish and for which the buyer agrees to pay, under a specific contract.

**Contract Pressure** The maximum or minimum required operating pressure at a receipt or delivery point as specified in the Service Agreement.

**Core Customers** Residential and small commercial customers who must rely on the traditional distributor bundled service of sales and transportation. Compare NON-CORE CUSTOMERS.

**Cross Over** Piping used to connect two or more pipelines.

**Cubic Foot** The most common unit of measurement of gas volume. It is the amount of gas required to fill a volume of one cubic foot under stated conditions of temperature, pressure, and water vapor.

**Cubic Foot Metered** The quantity of gas that occupies one cubic foot under pressure and temperature conditions in the meter.

**Cubic Foot, Standard** That quantity of gas which under a pressure of 14.73 psia and at a temperature of 60°F occupies a volume of one cubic foot without adjustment for water vapor content.

**Customer** An individual, firm, or organization which purchases service at one location under one rate classification, contract, or rate schedule. If service is supplied at more than one location or under more than one rate schedule, each location and rate schedule shall be counted as a separate customer unless the consumption at the several locations is combined before billing and billed on one rate schedule.

**Customer Density** Number of customers in a given unit or area or on a given length of distribution line.
Degree Day, Cooling: A measure of the need for air conditioning (cooling) based on temperature and humidity. Although cooling degree days are published for many weather stations, a specific procedure has not been generally accepted.

Degree Day, Heating: A measure of the coldness of the weather experienced, based on the extent to which the daily mean temperature falls below a reference temperature, usually 65 degrees F. For example, on a day when the mean outdoor temperature is 40 degrees F, the HDD is 25.

Dekatherm: A unit of heating value equivalent to 10 therms or 1,000,000 Btu's.

Delivery Point: Point at which gas leaves a transporter's system completing a sale or transportation service transaction between the pipeline company and a sale or transportation service customer.

Demand: The rate at which gas is delivered to or by a system, part of a system, or a piece of equipment, expressed in cubic feet or therms or multiples thereof, for a designated period of time called the demand interval. Compare LOAD.

Demand, Coincident: The sum of two or more demands which occur in the same demand interval.

Demand, Contract: The daily quantity of gas which the supplier agrees to furnish and for which the buyer agrees to pay, under a specific contract.

Demand, Integrated: The demand averaged over a specified period, usually determined by an integrating demand meter or by the integration of a load curve. It is the average of the instantaneous demands during a specified demand interval.

Demand, Maximum: The greatest of all the demands under consideration occurring during a specified period of time.

Demand, Minimum: The smallest of all the demands under consideration occurring during a specified period of time.

Demand, Non-Coincident: The sum of two or more individual maximum demands, regardless of time of occurrence, within a specified period.

Design Day: A 24-hour period of demand which is used as a basis for planning gas capacity requirements.


Design Load: The maximum average rate of gas use by a group of appliances or customers over a specified short time period, usually 15 to 30 minutes.

Design Pressure: The maximum operating pressure permitted by various codes, as determined by the design procedures applicable to the material and location involved.

Distribution: The act or process of distributing gas from the city gas or plant that portion of utility plant used for the purpose of delivering gas from the city gate or plant to the consumers, or to expenses relating to the operating and maintenance of distribution plant.

Distribution Company: Gas Company which obtains the major portion of its gas operating revenues from the operation of a retail gas distribution system, and which operates no transmission system other than incidental connections within its own system or to the system of another company. For purposes of A.G.A. statistics, a distribution company obtains at least 90 percent of its gas operating revenues from sales to ultimate customers, and classifies at least 90 percent of mains (other than service pipe) as distribution. Compare INTEGRATED COMPANY; TRANSMISSION COMPANY, GAS.

Distribution System, Gas: See SYSTEM TYPE.

Downstream: Any point in the direction of flow of a liquid or gas from the reference point. Compare UPSTREAM.

Downstream Pipeline: The pipeline receiving natural gas at a pipeline interconnect point.
Efficiency  Relating to heat, a percentage indicating the available Btu input to combustion equipment that is converted to useful purposes.

End-User  An entity which is the ultimate consumer for natural gas. An end-user purchases the gas for consumption but not for resale purposes.

Equivalent Length of Pipe  The resistance of pipe valves, controls, and fittings to gas flow expressed as equivalent length of pipe or pipe of other sizes, for convenience in calculating pipe diameters.

Federal Energy Regulatory Commission (FERC)  An agency of the government of the United States created by an Act of Congress, the Department of Energy Organization Act, in 1977. This Act transferred to the FERC most of the former Federal Power Commission's interstate regulatory functions over the electric power and natural gas industries. The Act also transferred from the Interstate Commerce Commission the authority to set oil pipeline transportation rates and to set the value of oil pipelines for ratemaking purposes. In 1978, Congress passed the Natural Energy Act, broadening the FERC's jurisdiction and regulatory functions. The FERC now also regulates producer sales of natural gas in intrastate commerce. The FERC establishes uniform ceiling prices for each of several categories of natural gas, and these prices apply to all sales on a nationwide basis.

Feed Points  Connections between gas feeder lines and distribution networks.

Feeder (Main)  A gas main or supply line that delivers gas from a city gate station or other source of supply to the distribution networks.

Gas Day  A period of twenty-four (24) consecutive hours commencing at a specified hour on a given calendar day and ending at the same specified hour on the next succeeding calendar day.

Gas Impurities  Undesirable matter in gas, such as dust, excessive water vapor, hydrogen sulphide, tar, and ammonia.

Gas Research Institute (GRI)  An organization sponsored by a number of U.S. gas companies to investigate new sources of supply and new uses (applications) for natural gas.

Gas Turbine  A prime mover in which gas, under pressure or formed by combustion, is directed against a series of turbine blades; the energy in the expanding gas is converted into mechanical energy supplying power at the shaft.

Gas, Natural  A naturally occurring mixture of hydrocarbon and nonhydrocarbon gases found in porous geologic formations beneath the earth's surface, often in association with petroleum. The principal constituent is methane. 1. Dry. Gas whose water content has been reduced by a dehydration process. Gas containing little or no hydrocarbons commercially recoverable as liquid product. Specified small quantities of liquids are permitted by varying statutory definitions in certain states. 2. Liquefied (LNG). See LIQUEFIED NATURAL GAS. 3. Sour. Gas found in its natural state, containing such amounts of compounds of sulfur as to make it impractical to use, without purifying, because of its corrosive effect on piping and equipment. 4. Sweet. Gas found in its natural state, containing such small amounts of compounds of sulfur that it can be used without purifying, with no deleterious effect on piping and equipment. 5. Wet. Wet natural gas is unprocessed natural gas or partially processed natural gas produced from strata containing condensable hydrocarbons. The term is subject to varying legal definitions as specified by certain state statutes. (The usual maximum allowable is 7lbs./MMcf water content and .02 gallons/Mcf of Natural Gasoline.)

Gate Station  Generally a location at which gas changes ownership, from one party to another, neither of which is the ultimate consumer. It should be noted, however, that the gas may change from one system to another at this point without changing ownership. Also referred to as city gate station, town border station, or delivery point.
Head  The differential or pressure, usually expressed in terms of the height of a liquid column that the pressure will support. Also, the differential across a primary measuring device in feet of flowing fluid.

Header  A pipe or fitting that interconnects a number of branch pipes.

Heating Value  The amount of heat produced by the complete combustion of a unit quantity of fuel. The gross of higher heating value is that which is obtained when all of the products of combustion are cooled to the temperature existing before combustion, the water vapor formed during combustion is condensed, and all the necessary corrections have been made. The net or lower heating value is obtained by subtracting the latent heat of vaporization of the water vapor, formed by the combustion of the hydrogen in the fuel, from the gross or higher heating value.

High Pressure Distribution System  See SYSTEM TYPE.

Horsepower (hp)  A unit of power; equivalent to 33,000 ft-lb per minute, or 550 ft-lb per second (mechanical horsepower), or 0.746 kilowatts.

Horsepower Hour  The equivalent of one horsepower expended for one hour. One horsepower hour equals 1,979,980 foot-pounds.

Horsepower, Compressor  The horsepower rating on the name plate.

Hourly Peak  The maximum demand for gas from a transmission or distribution system in a one hour period of time.

Hydrocarbon  A chemical compound composed solely of carbon and hydrogen. The compounds having a small number of carbon and hydrogen atoms in their molecules are usually gaseous; those with a larger number of atoms are liquid, and the compounds with the largest number of atoms are solid.

Ideal Gas Law  The ideal gas law is the combination of the volume, temperature, and pressure relationships of Boyle's and Charles' laws resulting in the relationship PV=RT. Real gases deviate by varying amounts from the ideal gas law. See SUPERCOMPRESSIBILITY FACTOR.

Inch of Water  A pressure unit representing the pressure required to support a column of water one inch high. Usually reported as inches W.C. (water column) at a specified temperature; 27.707 inches of water (at 60o and standard gravity of 32.174 ft/sec^2) is equal to a gauge pressure of one pound per square inch.

Interconnection, System  A connection between two utility systems permitting the transfer of gas in either direction.

Interstate  With respect to natural gas companies, the transporting and sale of gas for resale across state lines.

Intrastate  With respect to natural gas companies, the transporting and sale of gas for resale within the boundaries of a state.

Joule-Thomson Effect  The cooling which occurs when a compressed gas is allowed to expand in such a way that no external work is done. The effect is approximately 7 degrees Fahrenheit per 100 psi for natural gas. See LAWS.

Joule-Thomson Expansion  The throttling effect produced when expanding a gas or vapor from a high pressure to a lower pressure with a corresponding drop in temperature.

Kinetic Energy  Energy possessed by a body due to its own motion.
Lateral  A pipe in a gas distribution or transmission system which branches away from the central and primary part of the system.

Line Pack  Natural gas occupying all pressurized sections of the pipeline network. Introduction of new gas at a receipt point "packs" or adds pressure to the line. Removal of gas at a delivery point lowers the pressure (unpacks the line).

Line Pack, Gas Delivered From  That quantity of gas delivered to the markets supplied by the net change in pressure in the regular system of mains, transmission, and/or distribution. For example, the change in the content of a pipeline brought about by the deviation from steady flow condition.

Line Packing  Increasing the amount of gas in a line section by increasing pressure to meet a heavy demand, usually of short duration.

Liquefied Natural Gas (LNG)  Natural gas which has been liquefied by reducing its temperature to minus 260 degrees Fahrenheit at atmospheric pressure. It remains a liquid at -116 degrees Fahrenheit and 673 psig. In volume, it occupies 1/600 of that of the vapor at standard conditions.

Liquefied Petroleum Gas (LPG)  A gas containing certain specific hydrocarbons which are gaseous under normal atmospheric conditions, but can be liquefied under moderate pressure at normal temperatures. Propane and butane are the principal examples.

Load  The amount of gas delivered or required at any specified point or points on a system; load originates primarily at the gas consuming equipment of the customers. Also, to load a pressure regulator is to set the regulator to maintain a given pressure as the rate of gas flow through the regulator varies. Compare DEMAND.

Load Center  A point at which the load of a given area is assumed to be concentrated.

Load Curve  A graph in which the load of a gas system or segment of a system is plotted against intervals of time.

Load Density  The concentration of gas load for a given area expressed as gas volume per unit of time and per unit of area.

Load Factor  The ratio of the average requirement to the maximum requirements for the same time period, as one day, one hour, etc.

Load Profile  Pattern of a customer's gas usage, hour to hour, day to day, or month to month.

Load, Base  See BASE LOAD.

Load, Connected  The sum of the capacities or ratings of the gas-consuming apparatus connected to a supplying system or any part of the system under consideration.

Load, Net  The active requirement for gas at a particular time. Compare LOAD, CONNECTED.

Local Distribution Company (LDC)  See DISTRIBUTION COMPANY, GAS.

Looping  A paralleling of an existing pipeline by another line over the whole length or any part of it to increase capacity.

Low Pressure Distribution System  See SYSTEM TYPE.

M

Main  A distribution line that serves as a common source of supply for more than one service line.

Main Extension  The addition of pipe to an existing main to serve new customers.

Main System  See SYSTEM TYPE.

Mains, Distribution  Pipes transporting gas within service areas to the point of connection with the service pipe.
Mains, Field and Gathering  See SYSTEM TYPE.

Mains, Gas  Pipes used to carry gas from one point to another. As contrasted with service pipes, they carry gas in large volume for general or collective use.

Mains, Transmission  See SYSTEM TYPE.

Maximum Actual Operating Pressure  See PRESSURE, MAXIMUM ACTUAL OPERATING.

Maximum Allowable Operating Pressure  See PRESSURE, MAXIMUM ALLOWABLE OPERATING.

Maximum Daily Quantity  The greatest quantity of gas to be received and/or delivered in a twenty-four hour period by the transporting pipeline on behalf of the shipper under terms defined in a contract. See MDQ.

Maximum Working Pressure  The maximum actual operating pressure existing in a piping system during a normal annual operating cycle or the maximum pressure for safe operation of a system.

Mcf  The quantity of natural gas occupying a volume of one thousand cubic feet at a temperature of sixty degrees Fahrenheit and at a pressure of fourteen and seventy-three hundredths pounds per square inch absolute.

MDQ  The term MDQ refers to maximum daily quantity of gas which a buyer, seller, or transporter is obligated to receive or deliver at each receipt or delivery point or in the aggregate as specified in an agreement.

Measuring and Regulating Station  Facilities installed at a given location for measuring and regulating the flow of gas in connection with distribution system operations other than the measurement of gas deliveries to customers.

Meter Set (Meter Installation)  The meter and appurtenances thereto, including the meter, meter bar, and connected pipe and fittings. Also called METER SET ASSEMBLY.

Meter, Gas  An instrument for measuring and indicating or recording the volume of gas that has passed through it.

MMBtu  A thermal unit of energy equal to 1,000,000 Btus, that is, the equivalent of 1,000 cubic feet of gas having a heating content of 1,000 Btus per cubic foot, as provided by contract measurement terms. See DEKATHERM.

MMcf  A million cubic feet. See CUBIC FOOT.

Molecular Weight  The sum of the atomic masses of the elements forming the molecule. In high polymers the molecular weights vary so widely they must be expressed as averages.

Monitoring Regulator  A pressure regulator set in series with a control pressure regulator for the purpose of automatically taking over the control of the pressure downstream in case that pressure tends to exceed a set maximum.

N  

Network  A system of transmission or distribution lines so cross-connected and operated as to permit multiple supply to any principal point on it.

O  

Off-Peak  The period during a day, week, month, or year when the load being delivered by a gas system is not at or near the maximum volume delivered by that system for the corresponding period of time.

P  

Pack  See LINE PACK.

Panhandle Formula  A formula for calculating gas flow in large diameter pipelines, particularly at relatively high pressures and velocities. Compare WEYMOUTH FORMULA.
**Partial Looping** A method for increasing carrying capacity of a pipeline by constructing a series of pipe sections parallel to the main pipeline for a portion of the distance between compressor or pump stations and connecting them to the main pipeline at the beginning and end of each segment. This reduces pressure drop in the portions of the pipeline that are "looped" (i.e., with parallel sections), allowing an increased pressure drop in the unlooped sections and, thus, an increased flow rate. Over time, a series of partial loops may be constructed resulting in a complete, second, parallel pipeline. At which time the pipeline will be totally looped.

**Peak Day** The one day (24 hours) of maximum system deliveries of gas during a year. Peak day data is used to, among other things, determine the allocation of certain costs between classes of service. The Commission sometimes required allocation based on an average of three continuous days of maximum deliveries (i.e., three day peak). See also DESIGN DAY.

**Peak Day Design** See DESIGN DAY.

**Peak Hour** The one-hour period of greatest total gas sendout or use.

**Peak Load** The maximum load consumed or produced by a unit or group of units in a stated period of time.

**Pipe** See PIPING.

**Pipeline** All parts of those physical facilities through which gas is moved in transportation, including pipe, valves, and other appurtenances attached to pipe, compressor units, metering stations, regulator stations, delivery stations, holders, and fabricated assemblies. See SYSTEM TYPE.

**Pipeline Capacity** The maximum quantity of gas that can be moved through a pipeline system at any given time based on existing service conditions such as available horsepower, pipeline diameter(s), maintenance schedules, regional demand for natural gas, etc.

**Pipeline Fuel** Natural gas consumed in the operation of a natural gas pipeline, primarily in compressors.

**Piping** A conduit for fluids and gases consisting of pipe or tubing with all necessary valves and fittings.

a. **Pipe.** Refers to rigid conduit of iron, steel, copper, plastic, or brass.
b. **Tubing.** Refers to a semi-rigid conduit of steel, copper, plastic, brass, or aluminum.

**Potential Energy** Stored energy. Energy possessing the power of doing work but not actually performing such work.

**Pressure** When expressed with reference to pipe, the force per unit area exerted by the medium in the pipe.

**Pressure Base** The standard pressure used in determining a gas volume, expressed in terms of pounds of pressure per square inch, usually 14.73 psia.

**Pressure Control** Maintenance of pressure, in all or part of a system, at a predetermined level or within a selected range.

**Pressure Differential** Difference in pressure between any two points in a continuous system. Compare PRESSURE DROP.

**Pressure Drop** The loss in static pressure of the fluid (air, gas, or water) due to friction or obstruction in pipe, valves, fittings, regulators, burners, appliances, and breeching. See PRESSURE LOSSES.

**Pressure Gauge** See GAUGE, PRESSURE.

**Pressure Limiting Station** Equipment installed for the purpose of preventing the pressure on a pipeline or distribution system from exceeding some maximum pressure as determined by one or more regulating codes by controlling or restricting the flow of gas when abnormal conditions develop. See PRESSURE RELIEF STATION and PRESSURE REGULATING STATION.

**Pressure Losses** Losses in static or velocity pressure in a piping system due to friction, eddies, leaks, or improper piping design. See PRESSURE DROP.
Pressure Rating  The estimated maximum pressure that the medium in the pipe can exert continuously with a high degree of certainty that failure of the pipe will not occur.

Pressure Regulating Station  Equipment installed for the purpose of automatically reducing and regulating the pressure in the downstream pipeline or main to which it is connected. Included are piping auxiliary devices such as valves, control instruments, control lines, the enclosures, and ventilating equipment. See PRESSURE LIMITING STATION and PRESSURE RELIEF STATION.

Pressure Regulator  See REGULATOR, PRESSURE.

Pressure Relief Station  Equipment installed for the purpose of preventing the pressure on a pipeline or distribution system to which it is connected from exceeding the maximum allowable operating pressure by venting gas to the atmosphere whenever the pressure exceeds this valve.

Pressure, Absolute (PSIA)  Pressure in excess of a perfect vacuum. Absolute pressure is obtained by algebraically adding gauge pressure to atmosphere pressure. Pressures reported in "Atmospheres" are understood to be absolute. Absolute pressure must be used in equations of state and in all gas-law calculations. Gauge pressures below atmospheric pressure are called "vacuum."

Pressure, Atmospheric  The pressure due to the weight of the atmosphere (air and water vapor) on the earth's surface. The average atmospheric pressure at sea level (for scientific purposes) has been defined at 14.696 pounds per square inch absolute.

Pressure, Gauge (PSIG)  Pounds per square inch above atmospheric pressure.

Pressure, Maximum Actual Operating  The maximum pressure that occurs during normal operations over a one-year period.

Pressure, Maximum Allowable Operating  The maximum operating pressure at which a system or a device may be operated as determined by regulating codes. MAOP

Pressure, Suction  The inlet pressure to a compressor, pump, or fan.

Pressure, Total  The sum of the static pressure and the pressure due to the velocity motion.

Psi  Pounds per square inch.

Public Utility  A business organization performing a service relating to or affecting all of the people within a specified area, usually under provisions of a franchise, charter or "certificate", and subject to special governmental regulations. See SERVICE AREA.

Quad  An energy quantity of one quadrillion Btu, which is approximately the energy equivalent contained in one trillion cubic feet of natural gas.

R

Rankine Scale of Temperature  The absolute Fahrenheit scale. Degrees F + 459.67 = degrees R. (The factor is usually rounded to 460 for commercial usage).

Regulator, Monitoring  A pressure regulator set in series with a control pressure regulator for the purpose of automatically taking over, in an emergency, the control of the pressure downstream of the station in case that pressure tends to exceed a set maximum.

Regulator, Pressure  A device that maintains the pressure in a fluid flow line, less than its inlet pressure within a constant band of pressures, regardless of the rate of flow in the line or the change in upstream pressure.

Regulator, Relief Pressure  A device for the purpose of relieving pressures in excess of a predetermined pressure.

Regulator, Service Pressure  A device designed to reduce and limit the gas pressure at the customer's meter.
Run An assembly of more than one piece of pipe; a portion of a fitting having its end in line or nearly so, as distinct from the branch or side opening, as of a tee.

S

Schematic An outline, systematic arrangement, diagram, scheme, or plan. An orderly combination of events, persons, or things according to a definite plan. A diagram showing the relative position and/or function of different components or elements of an object or system.

Service (Service Line, Service Pipe) The pipe which carries gas from the main to the customer’s meter.

Service Area A geographic area where a utility provides service, usually under provisions of a franchise, charter or certificate, and subject to special government regulations. See PUBLIC UTILITY.

Service Pipe See SERVICE.

Service Pressure, Standard The gas pressure that a utility undertakes to maintain on its domestic customers' meters. (Sometimes called the normal utilization pressure).

Specific Gravity The ratio of the density of a substance to the density of a reference substance, both at specified physical conditions. As applied to gas, air is the reference substance. Two specific gravity definitions are recognized in gas measurement: 1. Real Specific Gravity. The density ratio between a gas and air determined by measurement at the same temperature and pressure. 2. Ideal Specific Gravity. The ratio of the molecular weight of a gas to the molecular weight of air. (Mol. wt. of air = 28.9644).

Specific Heat The heat required to raise a unit mass of a substance through a degree of temperature difference. Also, the ratio of the thermal capacity of a substance to that of water. The specific heat of fluids varies with temperature and pressure.

Specific Weight The weight of a unit volume, usually expressed as pounds weight per cubic foot.

Station, Pressure Regulating See PRESSURE REGULATING STATION.

Supercompressibility Factor A factor used to account for the following effect: Boyle's law for gases states that the specific weight of a gas is directly proportional to the absolute pressure, the temperature remaining constant. All gases deviate from this law by varying amounts, and within the range of conditions ordinarily encountered in the natural gas industry, the actual specific weight under the higher pressure is usually greater than the theoretical. The factor used to reflect this deviation from the ideal gas law in gas measurement with an orifice meter is called the "supercompressibility factor Fpv". The factor is used to calculate actual volumes from volumes at standard temperatures and pressures from actual volumes. The factor is of increasing importance at high pressures and low temperatures.

System Type - Distribution Generally mains, services, and equipment which carry or control the supply of gas from the point of local supply to and including the sales meters. The system operates at various pressures as indicated below. a. High Pressure. A system which operates at a pressure higher than the standard service pressure delivered to the customer; thus, a pressure regulator is required on each service to control pressure delivered to the customer. Sometimes this is referred to as medium pressure. b. Low Pressure or Utilization Pressure. A system in which the gas pressure in the mains and service lines is substantially the same as that delivered to the customers' appliances; ordinarily a pressure regulator is not required on individual service lines.

System Type - Field and Gathering A network of pipelines (mains) transporting natural gas from individual wells to compressor station, processing point, or main trunk pipeline.

System Type - Main The network of distribution piping to which customers' service lines are attached. Generally, large pipes are laid in principal streets with smaller laterals extending along side streets and connected at their ends to form a grid; sometimes laterals are brought to dead ends. Compare with DISTRIBUTION, this section.

System Type - Transmission Pipelines (mains) installed for the purpose of transmitting gas from a source or sources of supply to one or more distribution centers, to one or more large volume customers, or a pipeline installed to interconnect sources of supply. In typical cases, transmission lines differ from gas mains in that they operate at higher pressures, are longer, and the distance between connections is greater.
Temperature Scale, Absolute (Kelvin)  A temperature scale independent of the thermometric properties of the working substance. For convenience, the absolute (Kelvin) degree is identified with the Celsius degree. The absolute zero in the Kelvin scale is minus 273.160 Celsius (°C).

Temperature, Ambient  The temperature of the air, atmosphere or other fluid that completely surrounds the apparatus, equipment or the workpiece under consideration. For devices which do not generate heat, this temperature is the same as the temperature of the medium at the point of device location when the device is not present. For devices which do generate heat, this temperature is the temperature of the medium surrounding the device when the device is present and generating heat. Allowable ambient-temperature limits are based on the assumption that the device in question is not exposed to significant radiant-energy sources such as sunlight or heated surfaces.

Temperature, Effective  An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation. See CHILL FACTOR.

Temperature, Ground  In the gas industry, the temperature of the earth at pipe depth.

Therm  A unit of heating value equivalent to 100,000 British thermal units (Btu).

Transmission Company, Gas  A company which obtains at least 90% of its gas operating revenues from sales for resale and/or transportation of gas for others and/or main line sales to industrial customers and classifies at least 90% of its mains (other than service pipe) as field and gathering, storage, and/or transmission.

Transmission System  See SYSTEM TYPE.

Tubing  See PIPING.

Upstream  From a reference point, any point located nearer the origin of flow, that is, before the reference point is reached.

Upstream Pipeline  The first pipeline to transport natural gas en route to an inter-connect point for delivery to another pipeline. See DOWNSTREAM PIPELINE.

Utility, Gas  A company that is primarily a distributor of natural gas to ultimate customers in a given geographic area.

Valve  A mechanical device for controlling the flow of fluids and gases; types such as gate, ball, globe, needle, and plug valves are used.

Valve, Ball  A valve in which a pierced sphere rotates within the valve body to control the flow of fluids. The sphere may be trunnion mounted or free.

Valve, Butterfly  A throttling valve made up of a disc that rotates on an axis within the valve body, thereby varying the cross-section that is open to fluid or gas passage.

Valve, Check  A valve built to pass a fluid in one direction but to close automatically when the fluid tries to flow in the opposite direction. Compare VALVE, BACK PRESSURE.

Valve, Gate  A full-opening valve controlled by a vertical movement of a single or pair of solid discs perpendicular to the direction of flow. There are several other types such as wedge, slab, expanding gate, etc.

Valve, Globe  A valve equipped with an orifice and a stem attached to a plug and matching circular seat. Shut-off is obtained by direct contact of the plug and the seat. Body of valve is normally spherical.
**Valve, Needle** A small valve that is opened and closed to permit or restrict fluid or gas flow by the movement of a pointed plug or needle in an orifice or tapered orifice in the valve body.

**Valve, Plug** Metal valve in which a pierced plug rotates in a tapered or cylindrical body to control flow through the valve.

**Valve, Relief** An automatic valve designed to discharge when a preset pressure and/or temperature condition is reached. 1. **Pressure Relief Valve.** An automatic valve which opens and closes a relief vent, depending on whether the pressure is above or below a predetermined value. 2. **Temperature Relief Valve.**
   a. **Fusible Type.** A valve which opens and keeps open a relief vent by the melting or softening of a fusible element at a predetermined temperature.  b. **Manual Reset Type.** A valve which automatically opens a relief vent at a predetermined temperature and which must be manually returned to the closed position.
   c. **Reseating or Self-Closing Type.** An automatic valve which opens and closes a relief vent when the temperature reaches a predetermined value.  d. **Vacuum Relief Valve.** An automatic valve which opens or closes a vent for relieving a vacuum, depending on whether the vacuum is above or below a predetermined value. Frequently used in a hot water supply system.

**Viscosity** In general, resistance to flow; that property of semi-fluids and gases by virtue of which they resist an instantaneous change of shape or arrangement of molecules.

**Volume, Specific** The volume of a unit weight of a substance at specific temperature and pressure conditions.

**W**

**Weight, Specific** Weight per unit volume of a substance.

**Weymouth Formula** A formula for calculating gas flow in large diameter pipelines. Compare PANHANDLE FORMULA.

**Wind Chill Factor** The equivalent temperature resulting from the combined effect of wind and temperature. For example: At 10 degrees Fahrenheit above 0 with a 20-mile per hour wind, the effect is the same as 24 degrees Fahrenheit below 0 without wind. See also CHILL FACTOR.

**Z**

**Zero, Absolute** See ABSOLUTE ZERO.

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**Supplemental Computer Codes**

**Inlet Pipeline Reduction Utility (IPRU)**

```vbp
Attribute VB_Name = "PipeReduction"
Public DimN As Integer
Public i, k, l, m, N As Integer
Public Buffer, WholeLine() As String 'Buffer variables where lines are read
Public TempIndex(3) As Integer 'Temporary index -> 1: NCE index
                                2: FromNode index
                                3: ToNode index
Public dd As Integer 'Number of deleted NCEs
Public nn As Integer 'Number of deleted NODEs
Public ConnectCount As Integer

Dim ProgressStatus, ProgressTotal As Double
```

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'Elements
Public NumberOfNCE As Integer
Public FID(), FromNode(), ToNode(), ElemType() As String
Public ElementLine() As String
Public ElStatO, Parl(), Par2(), Par3(), Par4() As String

'Nodes
Public NumberOfNODE As Integer
Public Node(), NodeLine() As String

'Partials
Public PART As Boolean
Public NumberOfPartReg, NumberOfPartTee As Integer
Public PartialReg, ParRegNode() As String
Public PartialTee, ParTeeNode() As String

'Descriptions
Public NumberOfDescrElem, NumberOfDescrNode As Integer
Public DescrElem(), DescriptE() As String
Public DescrNode(), DescriptN() As String

Public WillPrintNCE(), WillPrintNODE() As Boolean

Sub PipeReduceMain()
' Redimension the arrays **************************************
ReDim WholeLine(DimN)

'Elements
ReDim FID(DimN), FromNode(DimN), ToNode(DimN), ElemType(DimN)
ReDim ElementLine(DimN)

'Nodes
ReDim Node(DimN), NodeLine(DimN)

'Partials
ReDim ParRegNode(DimN)
ReDim ParTeeNode(DimN)

'Descriptions
ReDim DescrElem(DimN), DescriptE(DimN)
ReDim DescrNode(DimN), DescriptN(DimN)

ReDim WillPrintNCE(DimN), WillPrintNODE(DimN)

'**************************************************************************
i = 1
k = 1
PART = False
'****** Read from File ****************************************************
Open fMainForm.InputTB.Text For Input As #1 ' Open file for input.
ProgressStatus = 0
ProgressTotal = FileLen(fMainForm.InputTB.Text) / 50
Do While Not EOF(1) ' Loop until end of file.
    Line Input #1, Buffer ' Read data
If Mid(Buffer, 1, 3) = "NCE" Then
    fMainForm.StatusLbl.Caption = "Reading NCE Chapter"
    fMainForm.StatusLbl.Refresh
    Call ReadNCE
End If
If Mid(Buffer, 1, 4) = "NODE" Then
    fMainForm.StatusLbl.Caption = "Reading NODE Chapter"
    fMainForm.StatusLbl.Refresh
    Call ReadNODE
End If
If Mid(Buffer, 1, 4) = "PART" Then
    PART = True
    fMainForm.StatusLbl.Caption = "Reading PARTIAL Chapter"
    fMainForm.StatusLbl.Refresh
    Call ReadPARTIAL
End If
If Mid(Buffer, 1, 4) = "DESC" Then
    fMainForm.StatusLbl.Caption = "Reading DESCRIPTION Chapter"
    fMainForm.StatusLbl.Refresh
    Call ReadDESCRIPTION
End If
Call ReadLine
Loop
Close #I
fMainForm.StatusLbl.Caption = "Creating TEEs Partial ..."
fMainForm.StatusLbl.Refresh
Call CreateTEESPartial
fMainForm.StatusLbl.Caption = "Updating DESCRIPTION Chapter ...
Call SwitchDescription
'* Write to File *********************************************
Open fMainForm.OutputTB.Text For Output As #2 ' Open file for output.
Open fMainForm.ReportTB.Text For Output As #3
ProgressStatus = 0
ProgressTotal = NumberOfNCE + NumberOfNODE + NumberOfPartReg + 
NumberofPartTee + NumberOfDescrElem + NumberOfDescrNode + 100
Width #2, 80
Call PrintTitleReport
k = 1
Do Until Mid(WholeLine(k), 1, 3) = "NCE"
    Call PrintLine
Loop
fMainForm.StatusLbl.Caption = "Saving NCE Chapter"
    fMainForm.StatusLbl.Refresh
Call PrintNCE
fMainForm.StatusLbl.Caption = "Saving NODE Chapter"
    fMainForm.StatusLbl.Refresh
Call PrintNODE
'Call PrintPARTIAL 'Prints RegNodes
fMainForm.StatusLbl.Caption = "Saving PARTIAL Chapter"
fMainForm.StatusLbl.Refresh
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Call PrintPARTIAL2 'Prints DescrNodes as RegNodes
fMainForm.StatusLbl.Caption = "Saving DESCRIPTION Chapter"
fMainForm.StatusLbl.Refresh
Call PrintDESCRIPT
Close #2 ' Close file.
Call PrintBottomReport
Close #3
End Sub

Sub PrintTitleReport()
Print #3, WholeLine(2)
Print #3, WholeLine(3)
Print #3, Now
Print #3,
Print #3, "Input File: ", fMainForm.InputTB.Text
Print #3, "Output File: ", fMainForm.OutputTB.Text
Print #3,

Print #3,
End Sub

Sub PrintBottomReport()

Print #3,

Print #3, "Number of NCEs in input file: ", NumberOfNCE
Print #3, "Number of NCEs deleted: ", dd
Print #3, "Number of NCEs in output file: ", NumberOfNCE - dd
Print #3,
Print #3, "Number of NODEs in input file: ", NumberOfNODE
Print #3, "Number of NODEs deleted: ", nn
Print #3, "Number of NODEs in output file: ", NumberOfNODE - nn
'Print #3,
'Print #3, "Progress Bar Status variable", ProgressStatus,
FileLen(fMainForm.InputTB.Text)
End Sub

Sub CreateTeesPartial()
  ProgressStatus = 0
  ProgressTotal = CDbl(NumberOfNODE)
  NumberOfPartTee = 0
  For i = 1 To NumberOfNODE
    ConnectCount = 0
    For k = 1 To NumberOfNCE
      If Node(i) = FromNode(k) Or Node(i) = ToNode(k) Then
        ConnectCount = ConnectCount + 1
      End If
    Next k
    If ConnectCount >= 3 Then
      NumberOfPartTee = NumberOfPartTee + 1
      ParTeeNode(NumberOfPartTee) = Node(i)
    End If
    Call UpdateProgressBar
  Next i
End Sub
Sub SwitchDescription()
' Finds Reg Nodes on just one pipe and if the other end is a Tee Node
' the element is deleted and the description is moved to the Tee node

    ProgressStatus = 0
    ProgressTotal = NumberOfPartReg
    For i = 1 To NumberOfPartReg
        If CountElementsAtNode(ParRegNode(i)) = 1 Then
            If ParRegNode(i) = FromNode(TempIndex(1)) Then
                For k = 1 To NumberOfPartTee
                    If ToNode(TempIndex(1)) = ParTeeNode(k) Then
                        If TeeInDescrip(ParTeeNode(k)) = True Then Exit For
                        WillPrintNCE(TempIndex(1)) = False
                        WillPrintNODE(TempIndex(2)) = False
                        For l = 1 To NumberOfDescrNode
                            If Node(TempIndex(2)) = DescrNode(l) Then
                                DescrNode(l) = Node(TempIndex(3))
                                Exit For ' 1
                        Next l
                        Exit For ' k
                    End If
                Next k
            Else 'If ParRegNode(i) = FromNode(TempIndex(1))
                For k = 1 To NumberOfPartTee
                    If FromNode(TempIndex(1)) = ParTeeNode(k) Then
                        If TeeInDescrip(ParTeeNode(k)) = True Then Exit For
                        WillPrintNCE(TempIndex(1)) = False
                        WillPrintNODE(TempIndex(3)) = False
                        For l = 1 To NumberOfDescrNode
                            If Node(TempIndex(3)) = DescrNode(l) Then
                                DescrNode(l) = Node(TempIndex(2))
                                Exit For ' 1
                        Next l
                        Exit For ' k
                    End If
                Next k
            End If 'If ParRegNode(i) = FromNode(TempIndex(1))
        End If
    Next i
    Call UpdateProgressBar
End Sub

Function CountElementsAtNode(CountNode As String) As Integer
' Counts the number of elements connected to the CountNode and stores
' the index of the last element in TempIndex(I).
' Returns the number as integer.

    CountElementsAtNode = 0
    For ii = 1 To NumberOfNCE
        If CountNode = FromNode(ii) Or CountNode = ToNode(ii) Then
            CountElementsAtNode = CountElementsAtNode + 1
        End If
    Next ii
End Function
TempIndex(1) = ii
End If
Next ii
Call FindNode
End Function

Function TeeInDescr(NodeD As String) As Boolean
TeeInDescr = False
For iii = 1 To NumberOfDescrNode
    If NodeD = DescrNode(iii) Then
        TeeInDescr = True
        Exit For
    End If
Next iii
End Function

Sub FindNode()
' Finds the FromNode and ToNode of an NCE in the NODE chapter
'******************************************************************************
For kk = 1 To NumberOfNODE
    If FromNode(Templndex(1)) = Node(kk) Then Templndex(2) = kk
    If ToNode(Templndex(1)) = Node(kk) Then Templndex(3) = kk
Next kk
End Sub

Sub ReadNCE()
'******************************************************************************
' Read NCE Chapter
Call ReadLine
i = 1
    Line Input #1, Buffer ' Read data - next line
    Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
        If Mid(Buffer, 1, 2) = "* " Or Mid(Buffer, 1, 2) = " &" Then
            Line Input #1, Buffer ' Read data - next line
            GoTo 1000
        End If
    FID(i) = Mid(Buffer, 1, 8)
    FromNode(i) = Mid(Buffer, 10, 8)
    ToNode(i) = Mid(Buffer, 19, 8)
    ElemType(i) = Mid(Buffer, 28, 2)
    ElementLine(i) = Mid(Buffer, 30, Len(Buffer) - 30)
    WillPrintNCE(i) = True
Next i
Call UpdateProgressBar
1000: Loop
    NumberOfNCE = i - 1
    i = 1
End Sub

Sub ReadNODE()
'******************************************************************************
' Read NODE Chapter
Call ReadLine
```vbnet
i = 1
Line Input #1, Buffer ' Read data - next line
Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
    If Mid(Buffer, 1, 2) = "* " Then
        Line Input #1, Buffer ' Read data - next line
        GoTo 1000
    End If
    Node(i) = Mid(Buffer, 2, 8)
    NodeLine(i) = Mid(Buffer, 9, Len(Buffer) - 9)
    WillPrintNODE(i) = True
Line Input #1, Buffer ' Read data - next line
    i = i + 1
    Call UpdateProgressBar
Loop
NumberOfNODE = i - 1
i = 1
End Sub

Sub ReadPARTIAL()
    '* ***** Read PARTIAL chapter **********
    Call ReadLine
    i = 0
    '* Read Regulator Nodes in PARTIAL Chapter **********
    Line Input #1, Buffer ' Read data - next line
    Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
        If Mid(Buffer, 1, 8) = "RegNodes"
            Then PartialRegs = "RegNodes"
        ParRegNode(i) = Mid(Buffer, 1, 8)
        Line Input #1, Buffer ' Read data - next line
        i = i + 1
        Call UpdateProgressBar
    Loop
    NumberOfPartReg = i - 1
    Call ReadLine
    i = 0
    '* Read Tee Nodes in PARTIAL Chapter **********
    Line Input #1, Buffer ' Read data - next line
    Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
        If Mid(Buffer, 1, 4) = "TEEs" Then PartialTees = "TEEs"
        ParTeeNode(i) = Mid(Buffer, 1, 8)
        Line Input #1, Buffer ' Read data - next line
        i = i + 1
    Loop
    NumberOfPartTee = i - 1
    i = 1
End Sub

Sub ReadDESCRIPTION()
    '* Read DESCRIPTION chapter **********
    Call ReadLine
    i = 1
```

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********* Read Element Descriptions in DESCRIPTION Chapter **********
Line Input #1, Buffer ' Read data - next line
Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
    DescrElem(i) = Mid(Buffer, 1, 18)
    DescriptE(i) = Mid(Buffer, 19, Len(Buffer) - 19)
    Line Input #1, Buffer ' Read data - next line
    i = i + 1
Call UpdateProgressBar
Loop
NumberOfDescrElem = i - 1

Call ReadLine
i = 1

********* Read Tee Nodes in PARTIAL Chapter **********
Line Input #1, Buffer ' Read data - next line
Do While Not (Mid(Buffer, 1, 4) = "ZZZZ")
    DescrNode(i) = Mid(Buffer, 1, 8)
    DescriptN(i) = Mid(Buffer, 9, Len(Buffer) - 9)
    Line Input #1, Buffer ' Read data - next line
    i = i + 1
Loop
NumberOfDescrNode = i - 1
i = 1
End Sub
Sub ReadLine()
' Read Any Line
' Read Any Line
WholeLine(k) = Buffer
k = k + I
Call UpdateProgressBar
End Sub
Sub PrintLine()
If k = 4 Then
    Print #2, "Main to regulator reduction run on "; Format(Date, "dd-MMM-yy"); " at "; Time
Else
    Print #2, WholeLine(k)
End If
k = k + I
Call UpdateProgressBar
End Sub
Sub PrintNCE()
dd = 0
Print #3, "Elements deleted from NCE chapter:")
Print #3,
Call PrintLine
For i = 1 To NumberOfNCE
    If WillPrintNCE(i) = True Then
        Print #2, FID(i); Tab(10); FromNode(i); Tab(19); _
            ToNode(i); Tab(28); ElemType(i); Tab(30); ElementLine(i)
    Else
...
Sub PrintNODE()
    nn = 0
    Print #3, "Nodes deleted from NODE chapter:"
    Call PrintLine
    For i = 1 To NumberOfNODE
        If WillPrintNODE(i) = True Then
            Print #2, Tab(2); Node(i); Tab(10); NodeLine(i)
        Else
            Print #2, "* "; Tab(4); Node(i); Tab(12); NodeLine(i)
            Print #3, Tab(2); Node(i); Tab(10); NodeLine(i)
            nn = nn + 1
        End If
        Call UpdateProgressBar
    Next i
    Call PrintLine
End Sub

Sub PrintPARTIALO()
    Call PrintLine
    Print #2, PartialRegs
    For i = 1 To NumberOfPartReg
        Print #2, ParRegNode(i)
    Next i
    Call PrintLine
    Print #2, PartialTees
    For i = 1 To NumberOfPartTee
        Print #2, ParTeeNode(i)
    Next i
    Call PrintLine
End Sub

Sub PrintPARTIAL1()
    Call PrintLine
    Print #2, PartialRegs
    For i = 1 To NumberOfDescrNode
        Print #2, DescrNode(i)
    Next i
    Call PrintLine
    Print #2, PartialTees
    For i = 1 To NumberOfPartTee
        Print #2, ParTeeNode(i)
    Next i
End Sub
Sub PrintPARTIAL2()
 If PART Then
   Call PrintLine
   Print #2, PartialRegs
   For i = 1 To NumberOfDescrNode
     Print #2, DescrNode(i)
     Call UpdateProgressBar
   Next i
   Call PrintLine
   Print #2, PartialTees
   For i = 1 To NumberOfPartTee
     Print #2, ParTeeNode(i)
     Call UpdateProgressBar
   Next i
   Call PrintLine
 Else
   Print #2, "PARTIAL"
   Print #2, "RegNodes"
   For i = 1 To NumberOfDescrNode
     Print #2, DescrNode(i)
   Next i
   Print #2, "ZZZZ"
   Print #2, "TEEs"
   For i = 1 To NumberOfPartTee
     Print #2, ParTeeNode(i)
   Next i
   Print #2, "ZZZZ"
 End If
End Sub

Sub PrintDESCRIPT()
 Call PrintLine
 For i = 1 To NumberOfDescrElem
   Print #2, DescrElem(i); Tab(19); DescriptE(i)
   Call UpdateProgressBar
 Next i
 Call PrintLine
 For i = 1 To NumberOfDescrNode
   Print #2, DescrNode(i); Tab(9); DescriptN(i)
   Call UpdateProgressBar
 Next i
 Call PrintLine
End Sub

Sub UpdateProgressBar()
  ProgressStatus = ProgressStatus + 1
  valueprocent = ProgressStatus * fMainForm.ProgressBar1.Max / (ProgressTotal)
  If valueprocent < fMainForm.ProgressBar1.Max Then
    fMainForm.ProgressBar1.Value = valueprocent
End Sub
Else
    PrTotal = PrTotal * 1.2
End If
End Sub

Function INTEGRAL

Function INTEGRAL(y_Range, Optional x_Range)
' MS Visual Basic for Applications
'
' The function integrates values in a row-range or a column-range by
' the trapezoidal rule.
'
' y_Range is a column or a row range with the y-values and x_range is
' an optional column or a row range with the x-values.
'
' If no x_range is specified, the limits of integration are assumed to
' be 0 to 1 with evenly spaced n-1 intervals (n = number of y-values).
'
' If an x-range is provided, the x-values and y-values are treated as
' pairs. The limits of the integration are determined to be from the
' first x-value to the last x-value.

Dim Ry As Integer 'Number of rows in the Y range
Dim Cy As Integer 'Number of columns in the Y range
Dim Rx As Integer 'Number of rows in the X range
Dim Cx As Integer 'Number of columns in the X range
Dim i As Integer 'Index
Dim R As Integer 'Row number in the range
Dim C As Integer 'Column number in the range

INTEGRAL = 0
Ry = y_Range.Rows.Count
Cy = y_Range.Columns.Count
R = Ry
C = Cy

If Not IsMissing(x_Range) Then
    Rx = x_Range.Rows.Count
    Cx = x_Range.Columns.Count
    If Ry <> Rx Or Cy <> Cx Then MsgBox _
        "X and Y ranges have different size!", _
        vbCrLf, "Integral Function Error"
    If Ry <> 1 And Cy <> 1 Then MsgBox _
        "Not a column or a row range!", vbCrLf, _
        "Integral Function Error"
    If R = 1 Then 'Row range
        For i = 1 To C - 1
            INTEGRAL = INTEGRAL + ((y_Range.Cells(1, i).Value + _
                y_Range.Cells(1, i + 1).Value) / 2) * _
                (x_Range.Cells(1, i + 1).Value - _
                x_Range.Cells(1, i).Value)
        Next i
    Else
        For i = 1 To Ry - 1
            INTEGRAL = INTEGRAL + ((y_Range.Cells(i, 1).Value + _
                y_Range.Cells(i, C).Value) / 2) * _
                (x_Range.Cells(i, C - 1).Value - _
                x_Range.Cells(i, 1).Value)
        Next i
        INTEGRAL = INTEGRAL + (y_Range.Cells(Ry, C).Value + _
            y_Range.Cells(Ry, 1).Value) / 2 * _
            (x_Range.Cells(Ry, C - 1).Value - _
            x_Range.Cells(Ry, 1).Value)
    End If
End If
End Function
End If
If C = 1 Then 'Column range
    For i = 1 To R - 1
        INTEGRAL = INTEGRAL + ((y_Range.Cells(i, 1).Value +
            y_Range.Cells(i + 1, 1).Value) / 2) * 
            (x_Range.Cells(i + 1, 1).Value - 
            x_Range.Cells(i, 1).Value)
    Next i
End If
Else ' X range not specified
    If Ry <> 1 And Cy <> 1 Then MsgBox
        "Not a column or a row range!", vbExclamation,
        "Integral Function Error"
    If R = 1 Then 'Row range
        For i = 1 To C - 1
            INTEGRAL = INTEGRAL + (y_Range.Cells(1, i).Value +
                y_Range.Cells(1, i + 1).Value) / 2
        Next i
        INTEGRAL = INTEGRAL / (C - 1)
    End If
    If C = 1 Then 'Column range
        For i = 1 To R - 1
            INTEGRAL = INTEGRAL + (y_Range.Cells(i, 1).Value +
                y_Range.Cells(i + 1, 1).Value) / 2
        Next i
        INTEGRAL = INTEGRAL / (R - 1)
    End If
End If
End If
End Function
BIBLIOGRAPHY


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