Fuel Consumption Minimization under Non-Isothermal Condition at Compressor Stations

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ABSTRACT

Arguably, the natural gas transmission pipeline infrastructure in the U.S. represents one of the largest and most complex mechanical systems in the world. This system delivers about 0.623 tcm (22 tcf) of natural gas per year, and is made up of over 4.828×10^5 km (300,000 miles) of pipe driven by 8,000 engines and 1,000 gas turbines with 2.983×10^5 MW (40 million horsepower) of compression capacity. The system produces over 1.86×10^9 MW-hrs (250 billion hp-hrs) of compression power every year.

One of the goals of operation of this huge system is to find the minimum fuel consumption while maintaining the desired throughput of natural gas.

In this paper, we present a systematic approach for operating the units of a compressor station to meet a specified throughput profile. The first step in developing this approach is the derivation of a numerical method to analyze the flow through the pipeline under transient non-isothermal conditions. We have developed and verified a fully implicit finite difference formulation that provides this analysis capability. Next, the optimization of the compressor stations is formulated as a standard nonlinear programming problem (NLP).

The minimum acceptable throughput is imposed as a constraint. This NLP is solved numerically by a sequential unconstrained minimization technique, using the mathematical model of the system for the required function evaluations. The results show that this approach is very effective in reducing the fuel consumption. An application of this methodology for selecting the number of compressors to be shutdown for most fuel-efficient operation is also presented. Our results further indicate that station level optimization produces results comparable to those obtained by network level optimization. This is very significant because it implies that the optimization can be done locally at the station level, which is computationally much easier.

INTRODUCTION

Mathematical modeling is one of the most cost-effective tools that can be used to aid in design, operation, and optimization studies. The systems under consideration actually operate in an unsteady nature, and although much effort has been and continues to be spent on unsteady mathematical models, many over-simplifications are introduced that bring into question the simulation results.

Several investigators tried to simulate unsteady conditions for pipeline systems and some of them focused in compressor station modeling.

Jenicek and Kralik [1] developed optimized control of a generalized compressor station. The model described algorithm to optimize the operation of the compressor station with a fixed configuration.

Carter [2] presented a hybrid mixed-integer-nonlinear programming method that is capable of efficiently computing exact solutions to a restricted class of compressor models, and attempted to place station optimization in context with regard to simulation.

Boyd et al. [3] considered the fuel cost minimization problem in the steady-state gas pipeline networks by using mathematical model over compressor station.

Carter [4] developed a nonsequential Dynamic Programming (DP) algorithm to handle looped networks when the mass flow rate variables are fixed. The main advantages of DP are that a global optimum is guaranteed to be found and that nonlinearity can be easily handled.

Wu et al [5] presented two-model relaxation, one in the compressor domain and another in the fuel cost function, and derive a lower bounding scheme. The empirical evidence has been presented that showed the effectiveness of the lower bounding scheme.

Siregar et al [6] developed a mathematical model, which in turn solved analytically, and numerically for optimum pipeline diameter and routing.

Cobos-Zaleta and Rios-Mercado [7] used a MINLP model for the problem of minimizing the fuel consumption in a pipeline network. A computational experience was presented by evaluating an outer approximation with equality relaxation and augmented penalty method.

Edgar et al [8] presented a computer simulation to optimize the design of a gas transmission network, which considered the number of compressor stations, the diameter and length of pipeline segments, and the operating conditions of each compressor station. Two solution methods were used.

Many researchers have tried to find a way to optimize the gas pipeline system during transient operation, such as linepack buld up and discharge.

Osiadacz [9] described an algorithm for optimal control of a gas pipeline network based upon hierarchical control and decomposition. He used a simple linear diffusion equation to describe the transient flow through the pipe under isothermal conditions. The constraints considered were on compressor station operation, including minimum and maximum values for flow, pressure and compressor ratio. The constraints on pressures and flows were imposed on delivery and source points.

Vostry et al. [10] showed two different long-term and short-term optimizations. The control of the short-term process was determined by the dynamics of the system, whereas the long-term strategy and decision making depended on steady state conditions only. They presented a new approach to optimize large-scale dynamic networks with hierarchical control by local controllers using a gradient-based optimization method.

Pietsch et al. [11] described a transient optimization that included fuel and energy optimization, survivability under abnormal operational conditions, curtailment management, evaluation of spot market opportunities and optimization of facility expansion or addition designs.

Rachford and Carter [12] presented an algorithm to assist pipeline operations in controlling linepack and fuel consumption so as to enable projected deliveries in a transient condition. They used a rigorous transient model of the nonlinear pipeline hydraulics.

Carter and Rachford [13] explained some control strategies for efficiently operating pipelines through periods of fluctuating loads, which is simply a specific schedule for changing compressor station setpoint values. They represented several possible future scenarios to find an optimal profile for the linepack with uncertain demand.

The work presented in this paper is an important advance over current methods in the accurate simulation of transient non-isothermal behavior in natural gas pipelines, and extends the knowledge found in the literature by demonstrating the impact of varying boundary conditions on compressor station components. In addition, it also shows how this type of detailed simulation can be used for optimizing the operation of a compressor station to minimize fuel consumption while maintaining desired throughput.

Also, we present a systematic approach for operating the units of a compressor station to meet a specified linepack profile. The first step in developing this approach is the derivation of a numerical method for analyzing the flow through the pipeline under transient non-isothermal conditions. This detailed compressor station model can be used to determine power requirements, gas turbine fuel consumption, and the head, isentropic efficiency and speed for each centrifugal compressor with respect to time. We have developed and verified a fully implicit finite difference formulation that provides this analysis capability. In addition, we also show how this type of detailed simulation can be used for optimizing the operation of a compressor station to minimize fuel consumption while maintaining desired throughput and pressure limits along with meeting a specified

NOMENCLATURE

A	Cross- sectional area of pipe (m^2)
$b_1 - b_6$	Coefficients for centrifugal compressor map (-)
C_p	Specific heat at constant pressure (J/kg. K)
D	Pipe diameter (m)
f	Friction factor (-)
g	Gravitational acceleration (m/s^2)
h	Specific enthalpy (J/kg)
$Head$	Isentropic head (kJ/kg)
LHV	Low heating value (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
N	Number of node (-)
N_r	Speed (rpm)
P	Pressure of the gas (Pa)
Q	Capacity (m^3/hr)
R	Specific gas constant (kJ/kg)
t	Time (s)
T	Temperature (K)
v	Velocity of the gas directed along the axis of the pipe (m/s)
V_w	Isentropic wave speed (m/s)
W	Frictional force per unit length of pipe and per unit time (N/m)
x	Distance along the pipe (m)
Z	Compressibility factor (-)
h	Efficiency (-)
g_g	Specific gravity (-)
q	Angle of inclination of pipe to the horizontal (radian)
s	Isentropic exponent (-)
r	Density of the gas (kg/m^3)
Ω	Heat flow (J/ms)

Subscripts

ac	Actual condition
d	Discharge
dr	Driver
f	Fuel
is	Isentropic
$mech$	Mechanical
s	Suction
Sc	Standard condition

GOVERNING EQUATIONS

The non-isothermal compressible flow of natural gas in pipelines is governed by the time-dependent continuity, momentum, and energy equations, and an equation of state for homogeneous, geometrically one-dimensional flow. By solving these equations, the behavior of gas parameters can be obtained along the pipe network.

Chapman and Abbaspour [14,15,16] developed the basic equations for one-dimensional, unsteady, compressible flow that include the effects of wall friction and heat transfer:

Continuity Equation

$$\frac{\partial r}{\partial t}+\frac{\partial}{\partial x}(r\,v)=0 \qquad (1)$$

Momentum Equation

$$r\,\frac{\partial v}{\partial t}+r\,v\frac{\partial v}{\partial x}+\frac{\partial P}{\partial x}=-\frac{w}{A}-r\,g\sin q \qquad (2)$$

Where:

$$w=\frac{f\,r v\left|v\right|}{8}p\,D$$

Conservation of Energy

$$r\,\frac{\partial h}{\partial t}+r v\frac{\partial h}{\partial x}-\frac{\partial P}{\partial t}-v\frac{\partial P}{\partial x}=\frac{\Omega+wv}{A} \qquad (3)$$

The term Ω is heat flow into the pipe per unit length of pipe and per unit time. To obtain the gas enthalpy h in terms of P, r, and T, *Zemansky[17]* described the thermodynamic identity:

$$dh=CpdT+\left\{\frac{T}{r}\left(\frac{\partial r}{\partial T}\right)_P+1\right\}\frac{dP}{r} \qquad (4)$$

Equation of State

$$\frac{P}{r}=ZRT \qquad (5)$$

The gas compressibility factor Z is (*Dranchuck [18]*):

$$Z=1+\left(A_1+\frac{A_2}{Tr}+\frac{A_3}{Tr^3}\right)r\,r+\left(A_4+\frac{A_5}{Tr}\right)r\,r^2+\frac{A_6}{Tr^3}\,r\,r^3 \qquad (6)$$

The resulting set of equations that completely and thoroughly describe transient compressible gas flow is:

$$\left(\frac{\partial P}{\partial t}\right)+v\left(\frac{\partial P}{\partial x}\right)+rV_w^{\,2}\left(\frac{\partial v}{\partial x}\right)=\frac{V_w^{\,2}}{CpT}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]\frac{\Omega+wv}{A} \qquad (7)$$

$$\left(\frac{\partial v}{\partial t}\right)+v\left(\frac{\partial v}{\partial x}\right)+\frac{1}{r}\left(\frac{\partial P}{\partial x}\right)=-\frac{w}{Ar}-g\sin q \qquad (8)$$

$$\left(\frac{\partial T}{\partial t}\right)+v\left(\frac{\partial T}{\partial x}\right)+\frac{V_w^{\,2}}{Cp}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]\left(\frac{\partial v}{\partial x}\right)=$$

$$\frac{V_w^{\,2}}{CpP}\left[1-\frac{P}{Z}\left(\frac{\partial Z}{\partial P}\right)_T\right]\frac{\Omega+wv}{A} \qquad (9)$$

The wave speed V_w is:

$$V_w=\sqrt{\frac{ZRT}{\left\{\left[1-\frac{P}{Z}\left(\frac{\partial Z}{\partial P}\right)_T-\frac{P}{rCpT}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]\right]^2\right\}}} \qquad (10)$$

To simulate the compressor station, the following equations are used to describe the performance of a centrifugal compressor. Compressor head is determined by:

$$Head=0.28704\frac{T_sZ_s}{s\,g_g}\left(\left(\frac{P_d}{P_s}\right)^{s}-1\right) \qquad (11)$$

and the relationship between the flow rate for standard conditions and the actual mass flow rate is:

$$Q_{sc}=\frac{\dot{m}_{ac}R}{97.67\times10^{-3}} \qquad (12)$$

The power required by the compressor for these conditions is:

$$Power=\frac{Head\times\dot{m}_{ac}}{h_{is}h_{mech}} \qquad (13)$$

For the purpose of inputting centrifugal compressor characteristics into a pipeline simulation model, it is suggested that the entire head versus capacity map be digitized and stored as a table. However, a simplified but still accurate representation of the head versus capacity curve can be obtained through the use of the normalized characteristics. Three normalized parameters are necessary to describe a compressor map, $\frac{Head}{N_r^{\,2}}$, $\frac{Q_{ac}}{N_r}$, h_{is}. Using standard polynomial curve-fit procedures for each centrifugal compressor, the relationship between these parameters is:

$$\frac{Head}{N_r^{\,2}} = b_1 + b_2\left(\frac{Q_{ac}}{N_r}\right) + b_3\left(\frac{Q_{ac}}{N_r}\right)^2 \qquad (14)$$

and:

$$h_{is} = b_4 + b_5\left(\frac{Q_{ac}}{N_r}\right) + b_6\left(\frac{Q_{ac}}{N_r}\right)^2 \qquad (15)$$

Where the coefficients $b_1, b_2, b_3, b_4, b_5, b_6$ make Eqs. 14 and 15 fully characterize the specific centrifugal compressor map. With the coefficients for Eqs. 14 and 15 stored in the computer, knowing the isentropic head and inlet volumetric flow allows computation of compressor speed and isentropic efficiency.
The fuel consumption for the compressor driver is currently obtained by:

$$\dot{m}_f = \frac{Power}{LHV \times h_{dr}} \qquad (16)$$

The gas discharge temperature is obtained by:

$$T_d = T_s + \frac{T_s}{h_{is}/100}\left[\left(\frac{P_d}{P_s}\right)^s - 1\right] \qquad (17)$$

and the mass balance for suction and discharge of the compressor is:

$$\dot{m}_{ac,s} = \dot{m}_{ac,d} + \dot{m}_f \qquad (18)$$

The fully implicit method consists of transforming Eqs. 6, 7, and 8 from partial differential equations to algebraic equations by using finite difference approximations for the partial derivatives. These equations are nonlinear and the Newton-Raphson method is applied to solve these equations for the compressible, non-isothermal transient flows through a pipe. Quasi-steady flow can be assumed at each time step of the numerical solution for the centrifugal and reciprocating compressor equations.

Formulation of the Optimization Problem

In order to optimize the operation of a pipeline network, we first formulate the problem at hand in the format of a standard nonlinear programming problem (NLP). This standard form is developed as:
Find the values of the design variables:

$$[b_1, b_2, \ldots, b_r]^T$$

to minimize an objective function:

$$f(b)$$

Subject to the constraints:

$$h_j(b) = 0\ ,\ j = 1, \ldots, m$$

and $$g_j(b) \le 0\ ,\ j = m+1, \ldots, n$$

The formulation of the network operation problem in the standard NLP form must be done carefully, making sure that the NLP formulation captures all the relevant aspects of the associated network problem.

Let have the following assumptions:

N Number of compressor stations in the pipeline network
NC_j Number of compressors in station j
n_{ik} Speed of compressor k in station i.
$nmin_{ik}$ Minimum speed of compressor k in station i
$nmax_{ik}$ Maximum speed of compressor k in station i
m_{fi} Fuel consumption rate of station i
m_i Mass flow rate at station i and and let the specified
$mmin_i$ Minimum allowable mass flow rate at station

Then, the set of design variables is defined by

$$\{n_{ik}\},\quad i = 1,\ldots,N;\quad k = 1, \ldots NC_i$$

while the objective function is given by

$$f = \Sigma(m_{fi})\ ,\ i = 1,\ldots,N$$

and the constraints are

$$nmin_{ik} \le n_{ik} \le nmax_{ik},\quad i = 1,\ldots,N;\quad k = 1, \ldots NC_i$$

$$mmin \le m_i\ ,\quad i = 1,\ldots,N.$$

Solution of the Optimization Problem

Once the network operation problem has been formulated as an optimization problem as outlined above, it can be solved using any of a variety of available methods. In this work, we used the sequential unconstrained minimization technique (SUMT) with an exterior penalty function. A directed grid search method was used for the unconstrained minimization that is required by the SUMT approach.

RESULT AND DISCUSSION

In this section we consider two parts, first a baseline simulation that uses the model to demonstrate linepack, and then second optimization.

Simulation

The purpose of this initial simulation is to show the behavior of pipeline parameters before reaching a steady sate condition. Figure 1 shows the schematic of a compressor station that explains the boundary conditions and geometry of the compressor station. Note that the number of compressors could be different. To know the dynamic response of the compressor station's parameters, an example is provided.

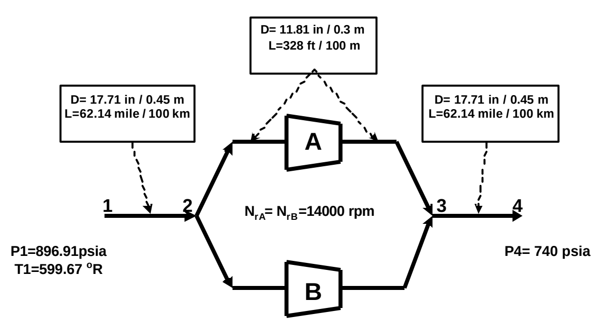
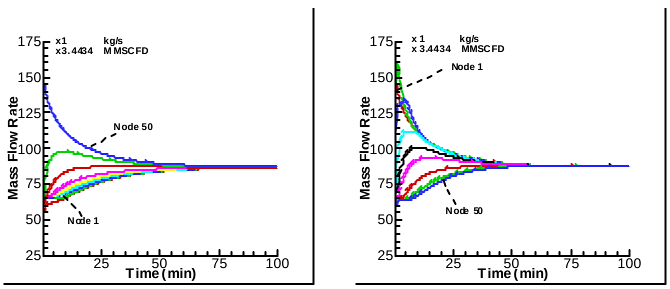


Figure 1 – Schematic of compressor station

The compressor station is located between two long pipes (100 km/ 62.14 mile).

A constant pressure boundary condition was applied for the head of the inlet pipe that enters the compressor station, and the end of the outlet pipe that exit the compressor station, as shown in Figure 1. Another boundary condition for this simulation is constant speed (14000 rpm) for each compressor. The panhandle equation is used to initialize the mass flow rate and pressure drop in pipe to start the simulations.

Figure 2 shows the variation of mass flow rate of the inlet pipe to compressor station for different points along the pipe. As shown in this figure, between 50 and 75 minutes is required for the mass flow rate to become uniform throughout the pipeline segment because of conservation of mass. At this time the flow is considered to be at steady state. In the same manner, we can explain the results for the outlet pipe as shown in Fig. 2-b. Figure 3 shows the fuel consumption of the compressor station with respect to time that is calculated by using the difference between the mass flow rate at inlet and outlet of compressor station.



(a)　　(b)

Fig. 2 Mass flow rate for inlet pipe to (a) and outlet from compressor station (b)

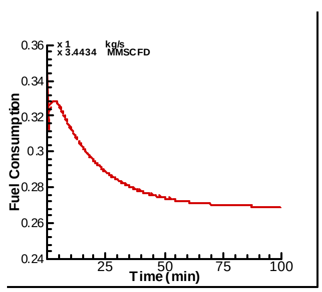
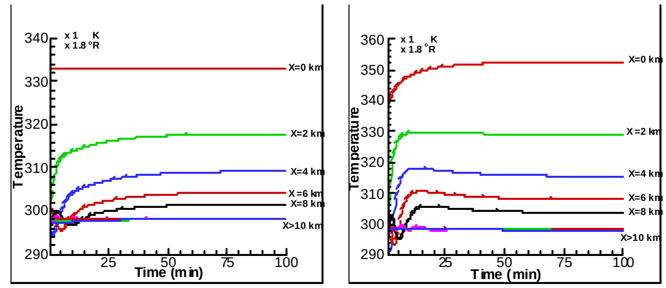


Fig. 3 Fuel consumption at compressor station

Figure 4 illustrates the temperature distribution within the inlet pipe to, and the outlet pipe from, the compressor station. Because of conservation of energy and heat transfer between the pipe and environment, the values of temperature at each node are different as shown in these figures. But the interesting thing is that the temperature of gas after about 10 km of pipe length will asymptotically approach the surrounding temperature. Control of exhaust temperature at the outlet of the compressor is an important goal of operation. By applying the non-isothermal pipe model, the station discharge can be determined and cooling system can be appropriately sized



(a)　　(b)

Fig. 4 Temperature distribution for inlet pipe to (a) and outlet from compressor station (b)

Figure 5 represents the behavior of the centrifugal compressor parameters during the transient condition. Because the compressor map is for steady-state conditions, then with constant compressor speed as a constraint, the operating point on the compressor map will change until the steady state condition (point B) is reached. This point is exactly for 14000 rpm.

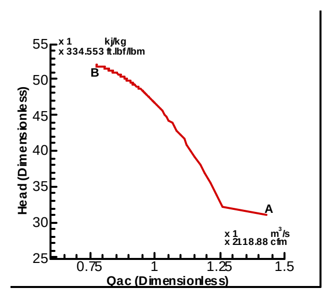


Fig. 5 Variation of head with respect to flow rate

Optimization

The optimization examples presented in this paper have been carefully selected to illustrate specific points. A summary of the examples is given in Table 1. The first example, we consider a compressor station with three dissimilar compressors; this is a common situation in practice and the optimum operating condition is difficult to find by other means. The second example considers two compressor stations in series to compare the results obtained by optimizing both stations simultaneously ("network level optimization") to the results obtained by optimizing each station separately ("station level optimization"). Finally the third example is about linepack problem. Each example is discussed briefly below:

Table 1- Different case for compressor station optimization

Case 1	Compressor station with three different compressors
Case 2	Two compressor stations in series, with each compressor station as in case 2
Case 3	Linepack problem

Example 1:

The system considered here is a single compressor station with three dissimilar compressors. Thus, the station configuration is just one of the two clusters shown in Figure 6. In this case, the three units are not identical and they each have different compressor maps. This is more realistic, since few compressor stations have all their compressors identical. This also makes it more difficult to find the optimum operating configuration since we now have no notion of symmetry. The compressor speed limits for this case are given in Table 2, and the goal of the optimization is to minimize the total fuel consumption while maintaining a station throughput of 170 kg/s (585.38 MMSCFD). The results obtained by optimization are shown in Table 3. It is seen that the optimal solution in this case gives us three different speeds for the three compressors (12650, 11650, and 10650 rpm), and the final mass flow rate is close to its minimum allowable value at 170.7 kg/s (585.62 MMSCFD).

The fuel consumption in this case is reduced from 42.86×10^{-3} kg/s (147.59×10^{-3} MMSCFD) to 39.39×10^{-3} kg/s (135.65×10^{-3} MMSCFD). As seen in Table 3, the efficiency of the first unit actually drops from 79.75% to 79.60% at the optimal solution, while the efficiency of the other two units does not change much. The outlet temperature is seen to decrease from 342.67 K to 340 K.

Example 2:

In the previous example, we have only considered the problem of optimizing the compressors in one compressor station. In order to optimize the operation of a pipeline network we must ideally perform "network level optimization" of compressor speeds, i.e., the speeds of all the compressors in all the compressor stations must be optimized simultaneously. This is numerically very difficult and computationally very expensive. On the other hand, this task will be considerably simplified if we can obtain high quality solutions through "station level optimization," i.e., by optimizing the speeds of compressors in each station independently. In order to compare network level optimization with station level optimization, we now consider a small network consisting of two compressor stations, each of which is identical to the compressor station in Example 1. Thus, the solution obtained in Example 1 gives us the optimal speeds obtained by station level optimization for this problem. The compressor speed limits for this case, as shown in Table 4, are identical to the limits in Example 1. We now apply numerical optimization to find the optimal speeds of all six compressors simultaneously, i.e., we apply network level optimization to this network. The goal of the optimization is to minimize the total fuel consumption in both stations combined while maintaining a line throughput of 170 kg/s (585.38 MMSCFD). The results obtained by optimization are shown in Table 5. It is seen that the optimal compressor speeds obtained in this case are very close to those obtained by station level optimization in Example 1.

The fuel consumption in this case is reduced from 85.51×10^{-3} kg/s (294.44×10^{-3} MMSCFD) to 77.76×10^{-3} kg/s (267.76×10^{-3} MMSCFD). These are almost exactly double the values obtained for a single station in Example 1. We may therefore conclude that in this example, station level optimization is a viable alternative to network level optimization. This is a very important and encouraging result in terms of the feasibility of optimizing compressor speeds in large networks using the methods developed in this paper.

Table 2- The input data for optimization case 1

	N_{rA}	N_{rB}	N_{rC}
Initial Value	13000 rpm	12000 rpm	11000 rpm
Max. Value	15000 rpm	15000 rpm	15000 rpm
Min. Value	10000 rpm	10000 rpm	10000 rpm
Minimum Mass flow rate	170 kg/s – 585.38 MMSCFD		

Table 3- Final result for speed and fuel consumption for optimization case 1

	Initial	Final
N_{rA} (rpm)	13000	12650
N_{rB} (rpm)	12000	11650
N_{rC} (rpm)	11000	10650
Fuel Consumption (kg/s-MMSCFD) $\times10^3$	42.86 – 147.59	39.39 - 135.65
Mass flow rate (kg/s- MMSCFD)	173.12- 596.13	170.07- 585.62
h_{isA}	79.75	79.60
h_{isB}	79.35	79.35
h_{isC}	76.98	76.97
Discharge Temp. T_3 (K)	342.67	340

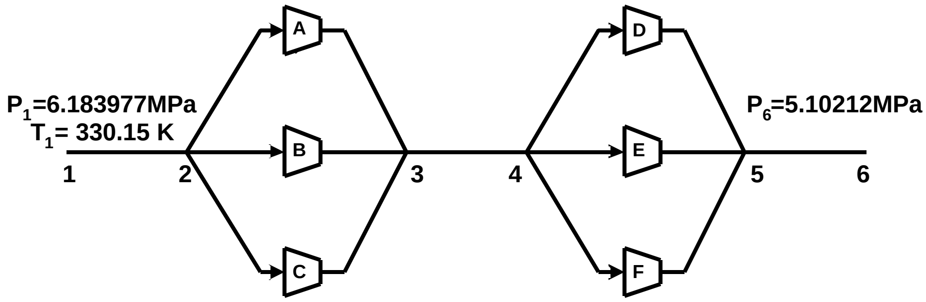


Fig. 6 Two compressor stations case 1 and 2

Table 4- The input data for optimization case 2

	N_{rA}	N_{rB}	N_{rC}	N_{rD}	N_{rE}	N_{rF}
Initial Value	13000 rpm	12000 rpm	11000 rpm	13000 rpm	12000 rpm	11000 rpm
Max. Value	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm
Min. Value	10000 rpm	10000 rpm	10000 rpm	10000 rpm	10000 rpm	10000 rpm
Min. Mass flow rate	170 kg/s – 585.38 MMSCFD					

Table 5- Final result for speed and fuel consumption for optimization case 2

	Initial	Final
N_{rA} (rpm)	13000	12600
N_{rB} (rpm)	12000	11600
N_{rC} (rpm)	11000	10600
N_{rD} (rpm)	13000	12600
N_{rE} (rpm)	12000	11650
N_{rF} (rpm)	11000	10650
Fuel Consumption (kg/s-MMSCFD) $\times 10^3$	85.51– 294.44	77.76 – 267.76
Mass flow rate (kg/s- MMSCFD)	173.41 - 597.14	170.01 – 585.42
h_{isA}	79.69	79.39
h_{isB}	79.35	79.34
h_{isC}	76.95	76.9
h_{isD}	79.76	79.67
h_{isE}	79.35	79.35
h_{isF}	76.99	76.96
Discharge Temp. T_3 (K)	342.44	339.11
Discharge Temp. T_5 (K)	342.67	339.83

Example 3:
Linepack simulation

Figure 7 shows the schematic of two sequential compressor stations. Each station contains three identical centrifugal compressors. A pipe with 100 km length and 0.6 m diameter connects two compressor stations, and a pipe with the same geometry transports gas to compressor station *A* and the other pipe delivers the gas from compressor station *B*. The pipeline delivers 200 kg/s (~ 690 MMSCFD) of gas at the end point (point 6), and we assume that the end delivery rate of this system is constant. At point 5 gas can also be diverted to side delivery points at flow rates that may fluctuate with time.

Suppose the consumer at point 5 tells the dispatch control that they need 10% of throughput (20 kg/s, ~60 MMSCFD) for five hours. Therefore the dispatch control needs to pack the gas before the consumer starts to use 10% of the gas, since the dispatch control has to maintain the delivery at point 6.

The dispatch control starts to pack the gas 450 min before the consumer request time. They increase the mass flow rate at point 1 to 205 kg/s (~ 706 MMSCFD) for 1200 min. During the first 450 min, the gas is packed throughout the pipeline.

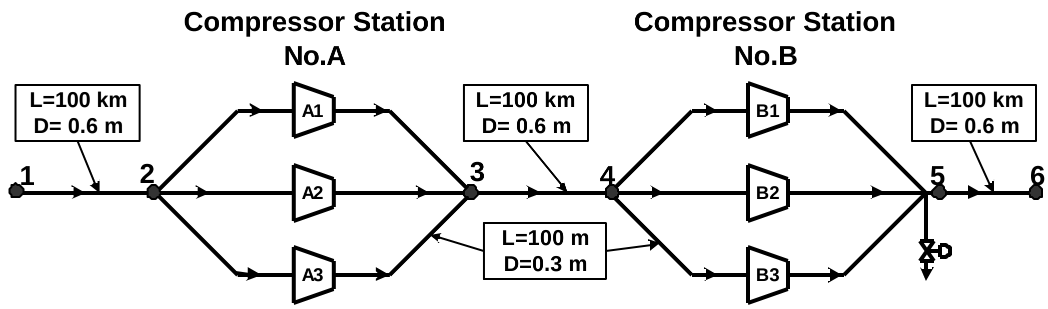


Fig. 7 Schematic of two compressor stations for linepack problem

Then the valve opens at point 5 and 10% of throughput is delivered to the consumer for 300 min. Following completion of the delivery to point 5, the flow at point 1 is maintained at 205 kg/s for an additional 450 minutes to return pipeline pressures and flow rates to stable conditions.

We consider two different operating conditions, first using all compressors operating at maximum speed (15000 rpm), and second using the compressor operating at 12000 rpm. Figure 8 compares the variation of mass flow rate at different points in the system for each compressor speed.

The mass flow rate at point 4 at the time that the valve is opened and gas deliver to the consumer, will increase to satisfy the mass flow rate for the consumer, so the mass flow rate in the exit pipe from compressor station B decrease.

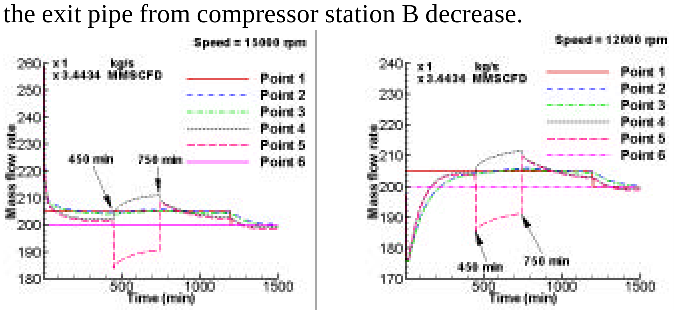


Fig. 8 Mass flow rate at different point of system with respect to operating time

Figure 9 represents the variation in system pressure. As shown in these figures, to maintain the linepack, if we use the maximum operation speed the maximum discharge pressures occur at the exit of each compressor station. But if the compressors operate at lower speed, the maximum pressure occurs at the source area and controls the pressure of the line and linepack.

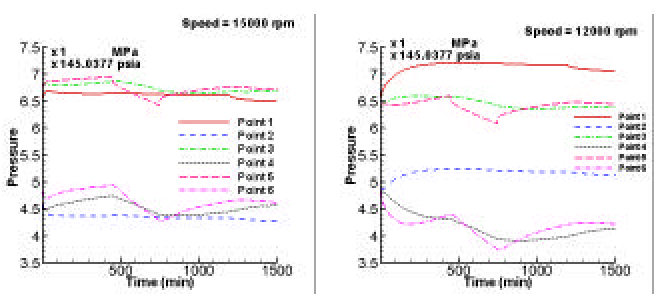


Fig. 9 Variation of system pressure

One limiting factor of compressor station operation is the station discharge temperature. The gas temperatures at point 3 and 5 are shown in Figure 10. Under the high-speed condition, the exit temperatures are about 20 K higher when compared to low speed condition. The temperature decrease at 450 min shown in this figure for point 5 is duo to the valve opening. This temperature decrease results from the pressure decrease shown in Figure 9.

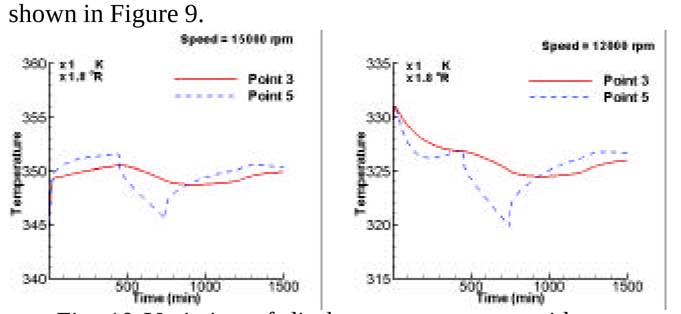


Fig. 10 Variation of discharge temperature with respect to operating time

Figure 11 shows the pressure ratio with respect to time for the high- and low-speed cases. As shown in this figure the maximum pressure ratio occurs at maximum speed.

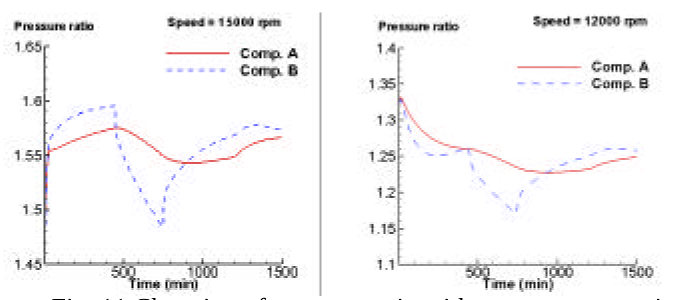


Fig. 11 Changing of pressure ratio with respect to operating time for each compressor inside the compressor station

Because the compressor station *B* is the closest one to the consumer, then we can see the significant jump in parameters at this compressor station. It is clearly shown in Figure 11 for pressure ratio.

The relationship between head and pressure ratio is almost linear as shown in Eq. (11), if we consider isentropic exponent as one. Therefore, the variation in head is the same as the pressure ratio as shown in Figure 12.

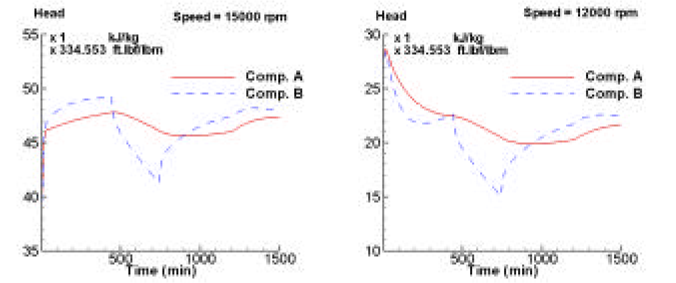


Fig. 12 Variation of head with respect to operating time for each compressor inside the compressor station

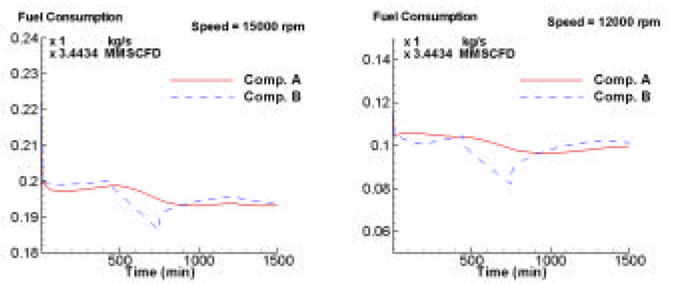


Fig. 13 Variation of fuel consumption with respect to operating time for each compressor inside the compressor station

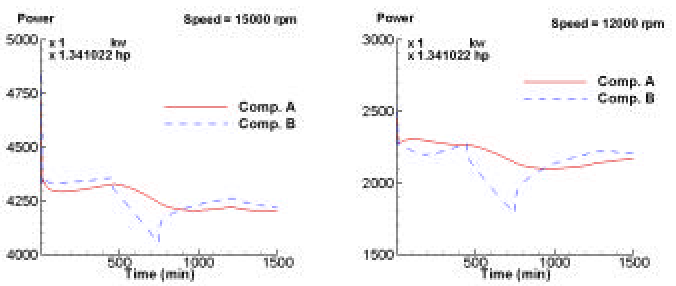


Fig. 14 Variation of Power with respect to operating time for each compressor inside the compressor station

In the same manner, we have a linear relationship between head and power (Eq. 13), and fuel consumption and power (Eq. 16). Therefore the same variation can be applied for these parameters as shown in Figure 13 and 14.

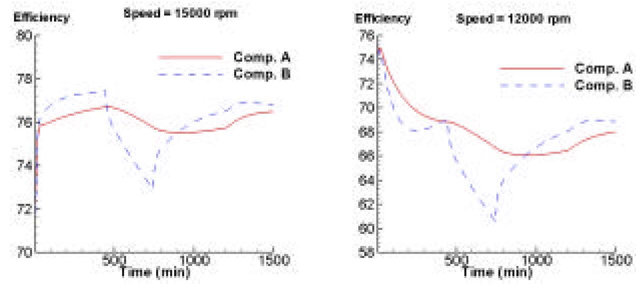


Fig. 15 Variation of isentropic efficiency with respect to operating time for each compressor inside the compressor station

Figure 15 shows the variation in isentropic compressor efficiency with respect to time. As shown in this figure, although the fuel consumption and power for the low-speed case are less than for the high-speed condition, the isentropic efficiency is less for the low-speed case when compared to that of the high-speed case.

Linepack optimization

The example pipeline system that was used in the simulation presented for linepack was also used as the test problem for optimization. The desired linepack profile is also the same as in the simulation example, but now we try to find values for the compressor speeds to minimize the average fuel consumption rate over the entire operational window of 1500 minutes summed over all six units.

Since there are six compressors, we have six design variables as shown in Table 6. The limits on the speeds of each compressor are also shown. In addition, we also impose a constraint on the maximum pressure in line, which cannot exceed the allowable maximum of 7.2 Mpa.

The results of the optimization are summarized in Table 7. It is seen that the optimum speeds selected for the compressors are neither at the maximum nor at the minimum allowable speed, but at an intermediate value. The desired linepack profile is achieved without violating the maximum pressure constraint. Most importantly, the average fuel consumption rate over the interval is reduced from 1.173 kg/s (4.0401 MMSCFD) at the initial design to 0.6095 kg/s (2.099 MMSCFD) at the optimum, which is a savings of almost 50%.

Table 6- The input data for optimization

	N_{rA1}	N_{rA2}	N_{rA3}	N_{rB1}	N_{rB2}	N_{rB3}
Initial Value	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm
Max. Value	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm	15000 rpm
Min. Value	10000 rpm	10000 rpm	10000 rpm	10000 rpm	10000 rpm	10000 rpm
Max. Pressure at pipeline	7.2 MPa – 1044.27 psia					

Table 7- Final result for speed, fuel consumption, temperature, isentropic efficiency and pressure

	Initial	Final
N_{rA1} (rpm)	15000	12048.44
N_{rA2} (rpm)	15000	12050
N_{rA3} (rpm)	15000	12050
N_{rB1} (rpm)	15000	12075
N_{rB2} (rpm)	15000	12100
N_{rB3} (rpm)	15000	12100
Ave. Fuel Consumption (kg/s- MMSCFD) *	1.173– 4.0401	0.6095– 2.099
h_{isA1} **	76.49	67.97
h_{isA2} **	76.49	67.97
h_{isA3} **	76.49	67.97
h_{isB1} **	76.77	69.52
h_{isB2} **	76.77	69.42
h_{isB3} **	76.77	69.42
Discharge Temp. (^{o}R) Point 1 **	629.9	587.1
Discharge Temp. (^{o}R) Point 3 **	630.6	589.5
Discharge Pressure (psia) Point 1 **	943.04	1022.92
Discharge Pressure (psia) Point 3 **	969.44	925.11
Discharge Pressure (psia) Point 5 **	974.03	939.52

* Average fuel consumption calculated for 25 hr of operation
** These values are calculated at the end of operation time (25 hr)

CONCLUSION

This study used a fully implicit finite difference method to analyze transient and non-isothermal flow within a pipe and a quasi-steady flow assumed at each time step of the numerical solution for centrifugal compressor equations to simulate compressor stations under non-isothermal conditions. The results show that:

- The simulation approach that is developed here is adequate for supporting numerical optimization.
- Numerical optimization is an effective tool for optimizing compressor speeds, and can yield significant reductions in fuel consumption. This, in turn, will increase throughout.
- The optimization can be extended beyond the compressor station level to the network level where the benefits will be even greater.
- Using this simulation as a basis, we are able to optimize the operation of compressor units along the pipeline to achieve the desired linepack profile with minimum fuel consumption. It is seen that the fuel savings can be quite substantial, particularly if large networks are considered. Thus, the simulation and optimization methods developed in this paper have the potential to produce great operational benefits in practice.
- This approach provides a broad foundation on which we can build more complete compressor station models including features such as scrubbers, coolers, etc.

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