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Economics of Internal Pipe Coating

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ABSTRACT

Internal pipe coating technology is available for natural gas pipelines, and can be primarily used to reduce surface roughness, and thus internal friction. This will reduce the pressure drop between compressor stations, and thus allows installing less power and consume less fuel. The potential to lower CAPEX (due to lower compression power requirement) and OPEX (due to lower fuel consumption) are counteracted by the extra cost of the internal coating.

In this paper, a large diameter pipeline case study is used to evaluate the alternatives of (a) coating versus (b) not coating the pipeline, and the results are presented. The impact of coatings on friction factor is based on actual test data. Based on actual cost data from pipeline coating, derived from a large transnational pipeline project, the impact on overall economics is assessed.

The case study will cover a pipeline capacity ramp up curve and the best technical and economical solution with regard to Capital expenditure – CAPEX and Operation expenditure – OPEX and consequently inpact on the transportation cost of service.

INTRODUCTION

With the worldwide demand for gas rising, new pipelines are required to bring gas over longer distances to the market. For long distance pipelines, the transport cost of the gas will make up an increasing portion of the delivery cost to the customer, and can reach 30 to 50% of the total cost at the receiving terminal. This transport cost can be influenced by optimizing the fuel consumption, equipment first cost, equipment operating cost, as well as equipment reliability and availability. The pressure and flow characteristics of pipelines and other factors influence the arrangement of compressors in a station. The question is often about number of units, the spacing of stations, standby requirements or the use of series or parallel arrangements in a station arises, together with type of driver, and type of compressor. When planning a compressor station or, for a new pipeline, a number of stations, considerations include: steady-state and transient capabilities and requirements of the system, growth requirements and capability, availability and total cost of ownership, and delivered cost to shippers and customers. The pipeline hydraulics relate pressure losses to the flow through the pipeline, determine the compressor operating conditions in terms of head and actual flow, and subsequently determine the required power from the driver. Contractual requirements and obligations, such as pressures and volumes at transfer points, have to be met. A key factor in these considerations is the pressure loss in the pipeline for a given flow rate, which, in turn is very much affected by the roughness of pipe.

For a situation where a compressor operates in a system with pipe of the length L_u upstream and a pipe of the length L_d downstream, and further where the pressure at the beginning of the upstream pipe p_u and the end of the downstream pipe p_e are known and constant, we have a simple model of a compressor station operating in a pipeline system .

The pressure gradient in the pipeline can be described by the Fanning equation which can be integrated. With reasonable simplifications, such as assuming the friction factor f to be constant, and a given, constant flow capacity Q_{std} , the pipeline will then impose a pressure p_s at the suction and p_d at the discharge side of the compressor (Eq 1):

$$\frac{p_d}{p_s} = \sqrt{\frac{p_e^2 + \varsigma_e Q_{std}^2}{p_u^2 - \varsigma_u Q_{std}^2}} \qquad (1)$$

with a pressure loss coefficient ζ that incorporates the friction losses, and will vary depending on pipe dimensions (length, diameter), temperature profile, pressure profile, surface roughness and flow velocity profile in the respective pipe sections. In particular, ζ is proportional to the length L of the pipeline section (Mohitpour et al.,2000). Obviously, p_s and p_d are also found as results of more elaborate pipeline hydraulics simulations (Ohanian and Kurz, 2002). Figure 1 outlines the results for $p_u=p_d$, , $p_e=p_s$ and $\zeta_u=\zeta_e$, which is typical for a compressor station in the middle of a pipeline. We find that 'lightly' loaded pipelines (for example pipelines with a relatively small distance between stations) tend to have a flatter pipeline characteristic, while pipelines where the stations are farther apart tend to have a steeper characteristic. This is important for compressor selections, because the compressor map has to cover the entire pipline characteristic.This , again, indicates the crucial impact of the loss factor, which depends on peipline roughness, on the effectiveness of the pipeline operation.

The subject of this paper is a newly design pipeline with a length of about 932 miles (1500 km), a pipe diameter of 42 inches (1067 mm) and Maximum Pipeline Operating Pressure (MAOP) of 1423 psig (9.81 MPag) (Figure 2). The attention the authors of this paper were drawn to a discussion where it was claimed that it is more economically feasible to use the pipes without internal coating where as pipeline community's common knowledge is the opposite. Authors of this paper decided to run number of simulations with the different internal roughness of the pipe and then run economical analysis to find out which solution presents the best economical result.

CASE STUDY

This case study is based on a pipeline project that goes from a gas supply receipt point to a interconnection 932 miles (1500 km) away delivering 177, 247, 353 and 529 BSCFY (5, 7, 10 and 15 BSCMY) of natural gas on firm contractual basis. Four pipeline internal roughnesses have been evaluated as described below:

- Alternative I: 395 microinches (10 micrometres)
- Alternative II: 590 microinches (15 micrometres)
- Alternative III: 790 microinches (20 micrometres)
- Alternative IV: 1180 microinches (30 micrometres)

Technical Assumptions

Pipeline	
Nomianl Diameter:	42 inches
Length:	932 miles (1500 km)
Design code:	ANSI B31.8
MAOP:	1423 psig (9.81 MPag)
End pressure:	1000 psig (6.89 Mpag)
Overall heat transfer:	0.29 Btu/h.ft^2.F
Soil temperature:	51.8 F (11 C)
Depth of cover:	3.28 feet (1 meter)
Compressor Station	
Compression ratio:	1.4 ~1.8
Suction/Disch. header press. drop:	18.9 psi (130 KPa)
After cooler disch. temperature:	86 F (30 C)
Site elevation	0 ~492 feet
Site Temperature	54.5 F (12.5 C)
Flow Equation:	Colebrook

THERMOHYDRAULIC SIMULATION

Thermohydraulic simulations were run for all alternatives with capacities and internal roughness as previously defined and the results are presented on table (1).

The simulation was performed using Synergy Gas (REF) steady state anlysis. It solves the steady state, one dimensional mass conservation, momentum conservation and energy conservation equations for each node. The real gas behavior regarding gas density in the hydraulic similutaion is adjusted using a modified Benedict-Webb-Rubin equation, which approximates the Standing-Katz compressibility factor correlation. The friction term, necessary for the closure of the energy and momentum equation is modeled using a friction model, which was, for this study the Colebrook-White friction factor equation with Gas General Flow Equation. The energy equation also allows to consider heat transfer from the pipe to the surrounding environment

Compressor maps for the relationships between flow, head and efficiency, based on actual centrifugal compressor performance, can be implemented into the simulation. The compressor operating point in the head-flow space is determined from pressures and temperatures using the Redlich Kwong equation of state.

The compressor models specifically cover the traditional and active portions of the compressor map from surge to choke. The program has capabilities to recognize surge and choke conditions (recycle flow, generate excess head, etc.) within bounds of available power and speed, but does not include specific performance data beyond surge and choke limits.

This study assumes certain requirements for the compression system. Beyond the quest for higher compressor peak efficiencies, the operating requirements set forth in this study as well as in other references require a compressor capable of operating over a wide operating range at high efficiency.

Wide operating range in a centrifugal compressor can be achieved by a combination of means.

Aerodynamic theory suggests a strong relationship between operating range, efficiency and impeller backsweep. However, there is a practical limit to the amount of backsweep. In particular, increasing backsweep reduces the capability of an impeller of given tip speed to make head.

However, with the capability to use two impellers in a casing, this perceived disadvantage can be eliminated. The operating range is further increased by the use of a vaneless diffuser.

ECONOMIC EVALUATION

A practicle approach when comparing project alternatives is to concentrate on what is different between them and compare their results in terms of Net Present Value – NPV. For this case study internal roughness of 10, 15, 20 and 30 micrometres (microinches) where evaluated and their impact quantified on annual capacity demands of 5, 7, 10 and 15 billion standard cubic meter per year – BSCMY (billion standard cubic feet per year - BSCFY). The alternatives of 10 and 15 micrometres might be associated with internal painting and 20 and 30 micrometres might be associated with pipeline without internal painting.

The side benefit of pipeline internal painting related to atmospheric corrosion protection while in storage prior to assembling and burying has not being quantified since this is close related to the quality of handling and storage of the pipes by Constructors and therefore varies with their expertise and procedures.

Another point of interest while evaluating pipeline internal coating is a commom practice of adopting load factor for pipeline design and simultaneous intall standby compressor units for the compressor station. This practice is not economically optimal. The optimal economic results can be achived by incorporating Monte Carlo simulation and failure analysis in the feasibility study as proposed by Santos (2009).

Technical Assumptions

- Two sizes of compressor sets selected according to the power requirement:
  - 16000 ISO hp
  - 10000 ISO hp
- Fuel consumption based on compressor and driver performance maps.
- One standby compressor unit for each compressor station.

Economic Assumptions

• Construction schedule:	2 years
• Compressor units Capex	
– (1) x 10000 ISO hp:	14.5 MMUS$
– (1) x 16000 ISO hp:	17.7 MMUS$
• Internal coating:	30 US$/pipeline ton
• O&M C. Sta. (without Fuel):	5% of C.Sta. Capex
• Depreciation:	20 years
• Economic life:	20 years
• Taxes:	40%
• Fuel price:	170 US$/MMSCM 4.75 US$/MMBTU
• Discount rate:	12% a year

ECONOMIC ANALYSIS

The purpose of the economic analysis is to quantify the influence of the internal roughness of the gas pipeline on the cost of service. This evaluation takes into account the technical and economical assumptions defined previously.

For pipeline operation conditions with low capacity (well under pipeline design capacity) changes in internal roughness did not presented significant operation cost reduction.

The economical benefit is identified for higer pipeline capacity – at or close to optimum design capacity – as shown in Table 1 and Figure 3 and 4. The effect of lowering cost of service is a consequence of lower fuel consumption and/or lower Capex for the pipeline project.

CONCLUSIONS

Pipeline internal painting is feasible and economically attractive even if side benefits of atmospheric corrosion protection are not accounted and also adopting conservative values for internal roughness of 10 micrometres (assuming this value after aging of 20 years of operation).

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ABOUT THE AUTHORS

Matt Lubomirsky, is a Consulting Engineer, Systems Analysis at Solar Turbines Incorporated, in San Diego, California. He is responsible for predicting gas turbines and compressors performance, for conduction of applications studies that involve pipeline and compressor stations modeling. Matt Lubomirsky attended Leningrad Institute of technology in Saint Petersburg, Russia where he received Master Degree in Mechanical Engineering. He has authored numerous publications about turbomachinery and pipeline related topics.

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Zhan Kulzhanov, the author, Deputy Director of Technical Development Department for Intergas Central Asia in Astana, Republic of Kazakhstan. Zhan has graduated Aviation Institute in Kazan, Russia with specialty in Jet Engines. He has Master Degree in Mechanical Engineering. Zhan's responsibilities include economic and technical evaluation of the pipeline projects. He was involved in Trans Asian pipeline project that delivers gas from Central Asian republics to Eastern part of China. Zhan has also previous experience in ICA pipelines Operations and Maintenance Department.

TABLES

Table 1 – Pipeline Alternatives I, II, III and IV – Thermohydraulic Results

Alternative	Pipeline Capacity, BSCFY (BSCMY)	Internal Roughness, 10^{-6} inches (10^{-6} meter)	Station	Station Flow, MMSCFD (MMSCMD)	Compression Ratio (Pd/Ps)	Required Compressor Units Quantity	Required Power hp	Fuel required MMSCFD	Comp. Unit size hp ISO
I	177 (5)	395 (10)	CS1A	535 (15.15)	2.72	1+1 series	29201	5.9046	16000
		590 (15)					29203	5.9046	
		790 (20)					29204	5.9046	
		1180 (30)					29207	5.9046	
II	247 (7)	395 (10)	CS1A	749 (15.15)	2.72	2+2 series	41774	9.1571	16000
		590 (15)		749 (15.15)	2.72		41780	9.1606	
		790 (20)		749 (15.15)	2.73		41785	9.1606	
		1180 (30)		749 (15.15)	2.73		41792	9.1606	
III	353 (10)	395 (10)	CS1A	1033 (29.25)	2.73	2+2 series	56463	11.5055	16000
			CS4	940 (26.63)	1.62	3	22945	5.6963	10000
							79407	17.2018	
		590 (15)	CS1A	1033 (29.62)	2.73	2+2 series	56502	11.5091	16000
			CS4	940 (26.63)	1.69	3	25268	6.0741	10000
							81770	17.5832	
		790 (20)	CS1A	1034 (29.27)	2.73	2+2 series	56530	11.5161	16000
			CS4	940 (26.63)	1.76	3	27270	6.3213	10000
							83801	17.8374	
		1180 (30)	CS1A	1032 (29.22)	2.73	2+2 series	56458	11.5055	16000
			CS3	941 (26.66)	1.39	2	14605	3.6586	10000
			ICS	360 (10.20)	1.25	1	4204	1.0842	10000
							75266	16.2483	
IV	530 (15)	395 (10)	CS1A	1056 (29.90)	2.73	2+2 series	57841	11.7209	16000
			CS2	1578 (44.68)	1.38	3	24666	5.9611	10000
			CS3	1485 (42.05)	1.43	3	25616	6.0988	10000
			CS4	1479 (41.88)	1.41	3	24941	6.0000	10000
			ICS	895 (25.35)	1.36	2	14188	3.5774	10000
							147252	33.3582	
		590 (15)	CS1A	1057 (29.94)	2.73	2+2 series	57928	11.7351	16000
			CS2	1579 (44.71)	1.41	3	26567	6.2330	10000
			CS3	1486 (42.07)	1.47	3	27810	6.3884	10000
			CS4	1452 (41.11)	1.45	3	26923	6.2789	10000
			ICS	895 (25.35)	1.41	2	15826	3.8740	10000
							155053	34.5095	
		790 (20)	CS1A	1058 (29.96)	2.73	2+2 series	57994	11.7457	16000
			CS2	1580 (44.73)	1.44	3	28193	6.4379	10000
			CS3	1487 (42.11)	1.51	3	29712	6.5932	10000
			CS4	1479 (41.90)	1.49	3	28635	6.4838	10000
			ICS	895 (25.35)	1.46	2	17288	4.0965	10000
							161823	35.3570	
		1180 (30)	CS1A	1063 (30.11)	2.73	2+2 series	58328	11.7986	16000
			CS2	1583 (44.84)	1.50	4	31177	7.6668	10000
			CS3	1488 (42.15)	1.58	4	33568	8.0517	10000
			CS4	1481 (41.93)	1.55	4	31997	7.8081	10000
			ICS	895 (25.35)	1.55	3	20282	5.2195	10000
							175352	40.5448	

FIGURES

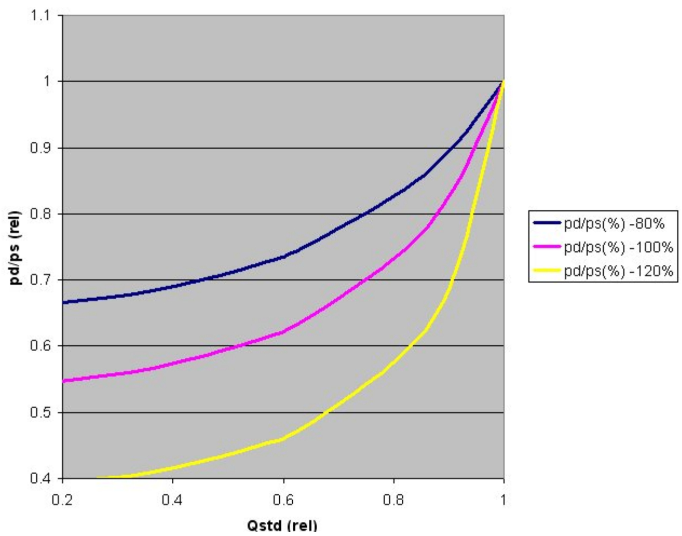

Figure 1 - Pipeline characteristics for different pipeline resistance for nominal loss coefficient, 80% of nominal loss coefficient and 120% of nominal loss coefficient. (Lubomirsky et al, 2010).

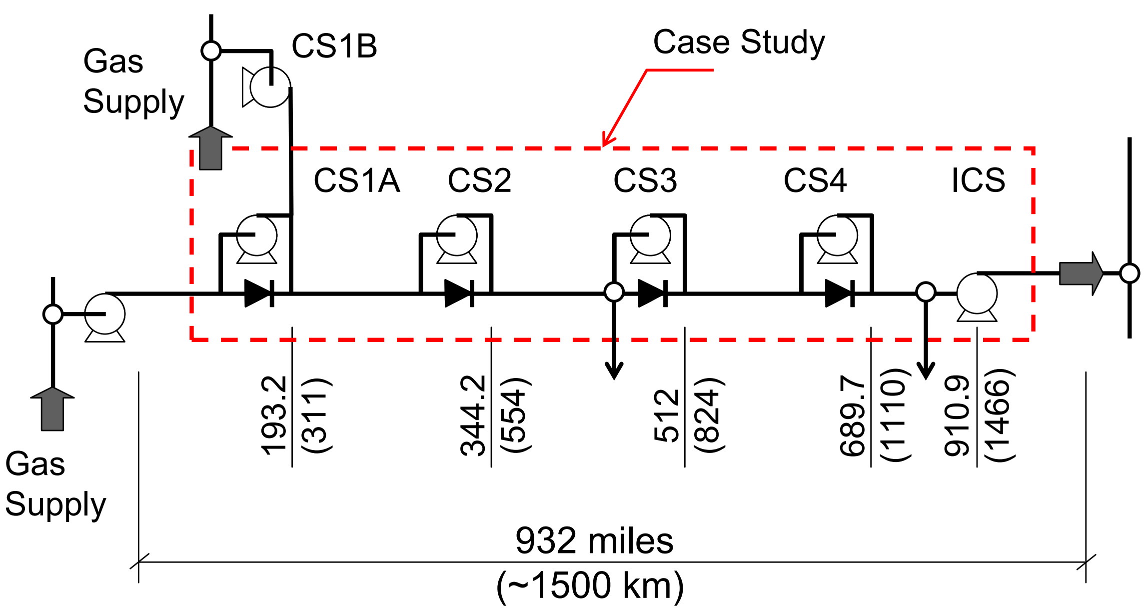

Figure 2 – Case Study – Gas Pipeline Configuration

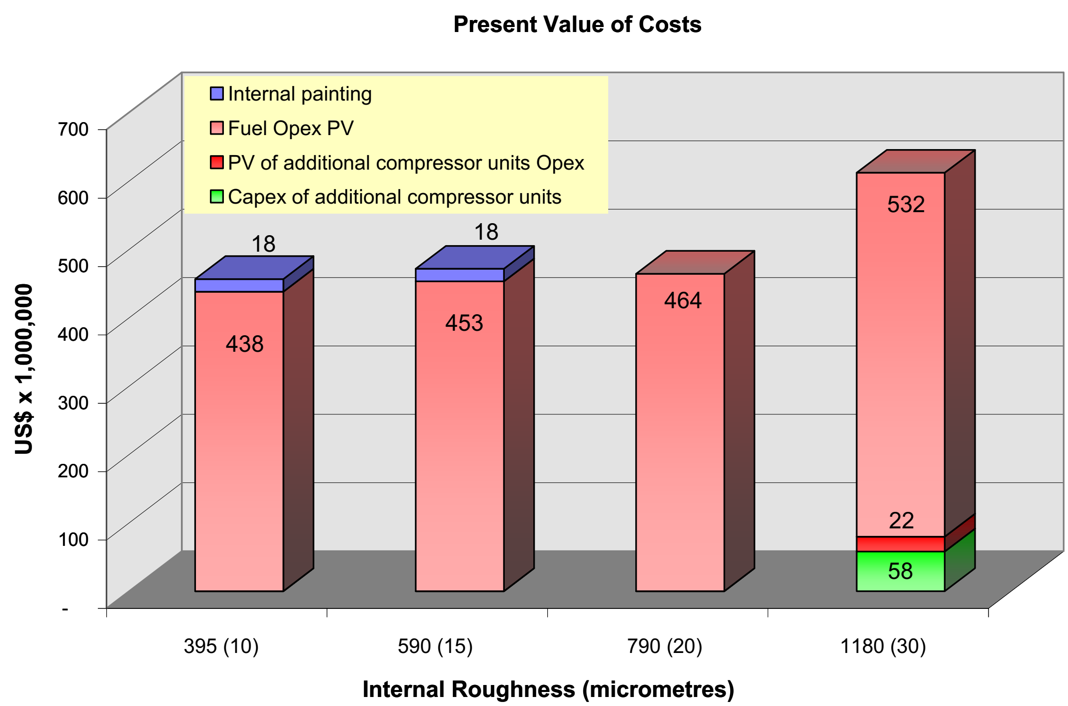

Figure 3 – Present Value of OPEX and CAPEX

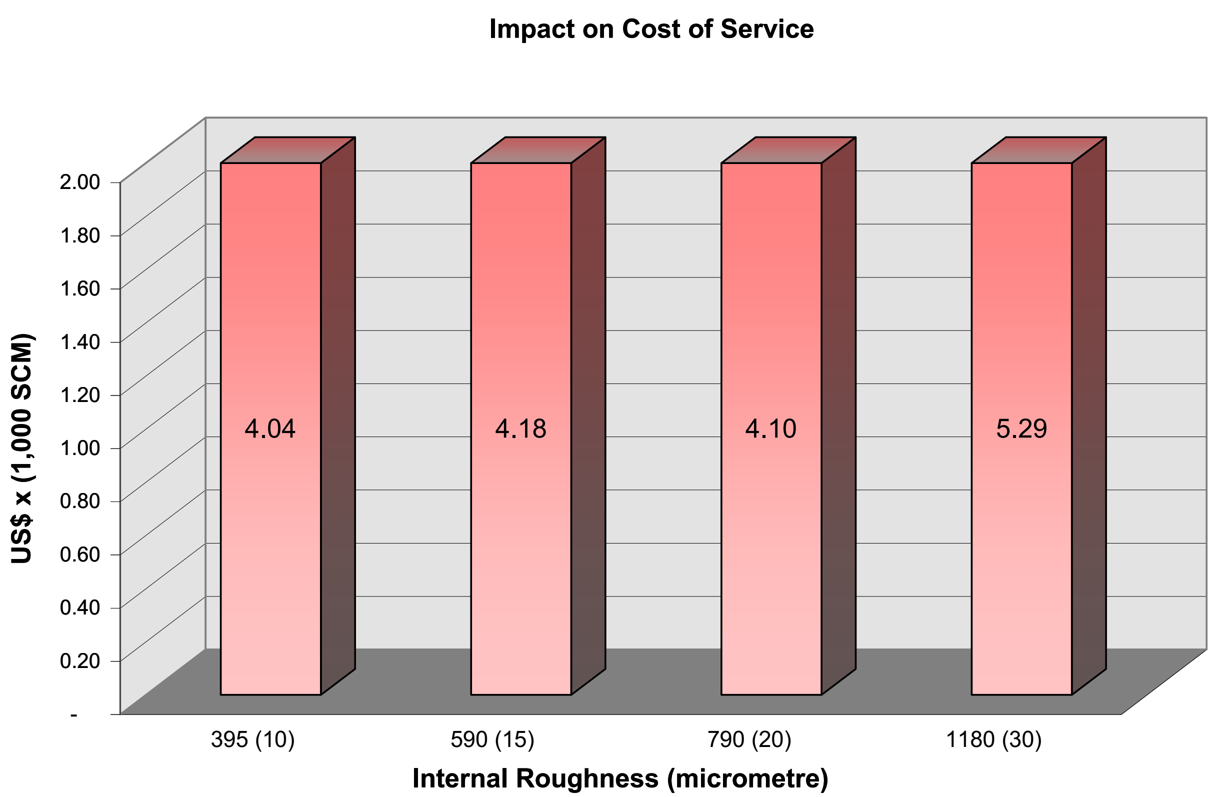

Figure 4 – Capacity Frequency and Availability