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# Evaluation of a Steam Pipeline

T. K. Stovall



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## ABSTRACT

This evaluation investigates the possibility of supplying steam to an industrial park from a nuclear power plant located up to 16 km away. The design steam load was estimated to be ~454 kg/s at a delivery pressure of 2.75 MPa. While the chief focus of this evaluation is on technical feasibility, the general methodology for calculating pressure drops and energy losses in steam pipes as well as a rough estimation of pipeline costs are included.


## 1. INTRODUCTION

The industrial sector of the United States consumed 9.1 EJ ($8.6 \times 10^{15}$ Btu) of natural gas and 8.0 EJ ($7.6 \times 10^{15}$ Btu) of petroleum during 1979.[1] This study examines the possibility of substituting coal or nuclear power for these prime fuels by means of a steam line.

While coal and nuclear energy sources could be placed at individual industrial sites, there are several potential advantages to central-station energy plants, including improvements in economics, safety, and environmental protection.

Electrical transmission is the most commonly used transport mechanism for central-station energy. The chief disadvantage to this option is relatively low overall efficiency, which leads to high costs. Direct steam transmission is occasionally used when the industrial user is sited adjacent to the power plant.

This evaluation investigates the possibility of supplying steam to an industrial park from a power plant located up to 16 km (10 miles) away. The steam load was estimated to be ~454 kg/s ($3.6 \times 10^6$ lb/h) at a delivery pressure of 2.75 MPa (400 psi).

The results presented in Sect. 3 were derived for this specific case. The general methodology described in Sect. 2, however, is applicable for most steam-pipe installations.

# 2. METHODOLOGY

## 2.1 Approach

The total length of the steam pipe was divided into many small segments (Fig. 1). Pressure losses from friction and heat losses caused by conduction were calculated for each segment, beginning with the segment closest to the user. Steam properties, including pressure, temperature, enthalpy, viscosity, and specific volume, were identified at the ends [points (1) and (2) in Fig. 1] of each segment. The steam would be delivered to the user's distribution point at 2.75 MPa (400 psi) and 260°C (500°F).

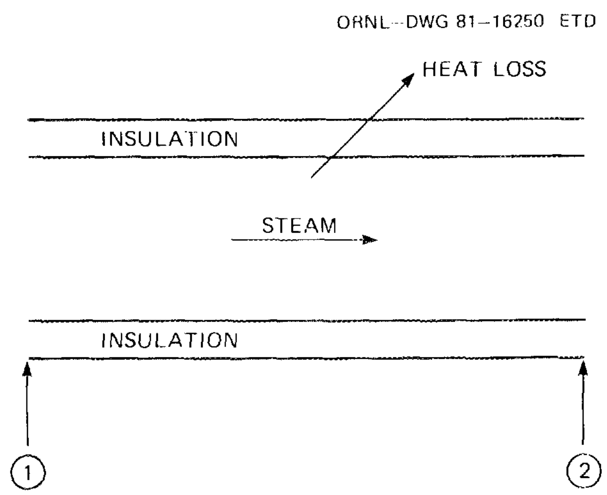


Fig. 1. Steam pipe segment.

## 2.2 Pressure Losses

The equation used[2] to calculate pressure loss is completely general and holds for both compressible and incompressible flow in pipes of constant cross section under these conditions: (1) the function $T = F(x)$ can be assigned and (2) $dp/dx$ is negative at every point along the pipe, where $T$ is the absolute temperature and $dp/dx$ is the fluid-friction pressure gradient. Thus, the pressure loss can be calculated as follows:

$$p_1 - p_2 = 2\frac{G^2}{2gc} v_1(v_R - 1) + f \frac{L}{D} \frac{G^2}{2gc} v_1 \phi (v_R + 1) ,$$

where

$p_1$ and $p_2$ = pressure at locations 1 and 2,

$G$ = mass velocity = $\frac{V}{v}$ = constant,

$g_c$ = conversion constant = 1 kg·m/N·s² = 32.17 ft·$lb_m/lb_f \cdot s^2$ ,

$v$ = specific volume of fluid,

$v_R$ = $v_2/v_1$,

$\phi$ = averaging factor (most engineering problems are concerned with the case where v is almost linear in T and $\phi \simeq 1/2$),

$f$ = friction factor (i.e., number of velocity heads lost in a length of pipe equal to diameter),

$L$ = pipe length,

$D$ = pipe inside diameter,

$V$ = velocity,

$T$ = temperature.

Pipe elbows and fittings cause pressure drops that are frequently estimated by using empirical correlations of test data in the form of equivalent pipe lengths. Since the number and nature of such fittings are not known in this study, an approximation of their pressure losses was made. An equivalent pipe-length addition equal to 40% of the actual pipe length was included in all pressure-drop calculations. This equivalency factor is likely to vary from 20 to 60%.

This pipe-length correction factor is for pressure drops only and does not include the increased pipe length necessary for expansion formations; these should be counted as part of the pipe length. Reference 3 gives pipeline expansion factors as a function of temperature. The factor for 260°C (500°F) is ~1.5, which would indicate a pipe length of ~24 km (15 miles) for a transmission distance of 16 km (10 miles). This expansion factor is based on right-angle U-loops and could be lower for alternative expansion formations.

Friction factors are determined empirically. The friction factor used in this report is equal to the fluid-friction loss in units of velocity heads per diameter length of pipe. To calculate this factor, a Fanning friction factor correlation for steam (Ref. 4) was multiplied by 4:

$$f = 4(0.0027) \times \left(1 + \frac{3.6}{D}\right) ,$$

where D is the pipe inside diameter in feet.

## 2.3 Heat Losses

A solution for steady-state heat conduction from an underground insulated pipe installed horizontally at a finite depth in homogeneous soil is found in Ref. 5. In the case of metallic pipes, the terms involving the heat transfer coefficient of the fluid and the thermal conductivity of the pipe wall are customarily ignored because of their low numerical values.

With these terms removed, the heat loss is calculated as follows:

$$Q = K_p(T_f - T_g),$$

and

$$\frac{1}{K_p} = \frac{1}{2\pi} \left\{\left(\frac{1}{K_i}\right) \ln\left(\frac{r'}{r}\right) + \left(\frac{1}{K_s}\right) \ln\left[\frac{d}{r'} + \sqrt{\left(\frac{d}{r'}\right)^2 - 1}\right]\right\} ,$$

where

- $Q$ = heat loss from a unit length of pipe,
- $K_p$ = pipe heat transfer factor,
- $T_f$ = temperature of the pipe fluid,
- $T_g$ = undisturbed average earth temperature,
- $K_i$ = thermal conductivity of pipe insulation,
- ln = natural logarithm,
- $r$ = pipe outside radius,
- $r'$ = pipe outside radius, including insulation ($r' = r + t$),
- $t$ = pipe insulation thickness,

$K_s$ = thermal conductivity of soil,

d = depth of the pipe measured from the ground surface to the centerline of the pipe.

$T_g$ was assumed to equal 16°C (60°F), $K_s$ to equal 0.35 W/m·°C (0.2 Btu/ft·h·°F), and d to equal 1.5 m (5 ft).

## 2.4 Steam Properties

The 1967 American Society of Mechanical Engineers formulations and iterative procedures for the calculation of the steam properties[6,7] were adapted by D. W. Altom and used at the Oak Ridge National Laboratory.

## 2.5 Cost

The costs were estimated using a methodology described in Refs. 8 and 9. This approach calculates the installed cost as a function of pipe weight, diameter, and insulation volume:

$$C = A_1 W + A_2 D^{0.48} + A_3 + A_4 Vi \ ,$$

where

C = pipe cost,

$A_1$ = pipe cost per unit weight,

W = pipe weight,

$A_2$ = installation cost,

D = pipe outside diameter,

$A_3$ = right-of-way cost,

$A_4$ = installed insulation cost on a volume basis,

Vi = insulation volume.

This equation is valid for pipe diameters >41 cm (16 in.). For smaller diameters, the installation cost is directly proportional to diameter, and the exponential installation term must be replaced by the linear term $A_5 D$.

The pipe cost factors are in 1981 dollars. The estimate assumed $A_1$ to equal \$0.82/kg (\$740/ton) (Ref. 10), $A_2$ to equal \$185/$m^{1.48}$ (\$51,160/$in.^{0.48}$/mile), $A_3$ to equal \$6,800/km (\$11,000/mile), and $A_5$ to equal \$295/$m^2$ (\$12,080/in./mile). A 30% contingency allowance was added to cover such costs as engineering, site-specific details, and steam traps. The insulation cost varies with the type of insulation.

This pipe weight is calculated as a function of pipe thickness, diameter, and length:

$$W = \pi t(D + t)L\rho$$

where

$W$ = pipe weight,
$\pi$ = 3.14159,
$t$ = pipe-wall thickness,
$D$ = pipe inside diameter,
$L$ = pipe length.

The pipe thickness is calculated from Part 2 of the Code for Power Piping (ANSI B31.1.0-1967) (Ref. 4):

$$t = \frac{PD'}{2(SE + Py)} + A ,$$

where

$t$ = minimum pipe-wall thickness (in.),
$P$ = maximum internal service pressure (psig),
$D'$ = outer diameter of the pipe (in.),
$SE$ = maximum allowable stress in material caused by internal pressure and joint efficiency at the design temperature (psi) = 16,000 psi for carbon steel at temperature below 600°F (Ref. 3).
$y$ = a coefficient = 0.4° below 482°C (900°F),
$A$ = allowance for threading, mechanical strength, and corrosion = 0.065 in. for plain-end steel pipes 1.25 in. and larger.

For an optimal (lowest) cost estimate, the minimum pipe-wall thickness and pipe weight were calculated for each pipe segment. This pipe cost represents an ideal cost based on optimal pipe-wall sizing. A more conservative pipe cost estimate was made by assuming that the pipe-wall thickness is constant over the entire length and is based on the maximum steam pressure anticipated. These costs do not include the condensate return line and are for the installed steam pipe only.

## 2.6 Sensitivity Analyses

Several parameters were examined to determine their impact on required steam conditions and costs.

The pipe segment length was varied from 10 to 3050 m (33 to 10,000 ft) to determine the maximum length that could be used without distorting the accuracy of the final results. A maximum segment length of 150 m (500 ft) was chosen.

The total pipe length varied from 8 to 24 km (5 to 15 miles), the pipe inside diameter from 0.9 to 1.27 m (35 to 50 in.), the steam flow from 125 to 500 kg/s (1 to 4 million lb/h), the insulation thickness from 0.08 to 0.3 m (3 to 12 in.), and the insulation conductivity from 0.061 to 0.095 W/m·°C (0.035 to 0.055 Btu/h·°F·ft).

# 3. RESULTS

## 3.1 Pipe Diameter

The pipe inside diameter varied from 0.5 to 1.27 m (20 to 50 in.) with a steam flow of 454 kg/s (3.6 x $10^6$ lb/h). Pipe diameters less than ~1 m (40 in.) showed very high pressure drops. A source pressure greater than 6.9 MPa (1000 psi) over a pipe length of 8.8 km (5.5 miles) would be required for a 1-m (40-in.) pipe.

Figure 2 and Table 1 show the required source pressure vs pipe length as a function of pipe inside diameter. The maximum steam pressure available from a pressurized-water reactor (PWR) after a reboiler is ~4.5 MPa (650 psi). Only the 1.27-m (50-in.) pipe is able to deliver the required steam from a PWR over a 19.3-km (12-mile) pipe length.

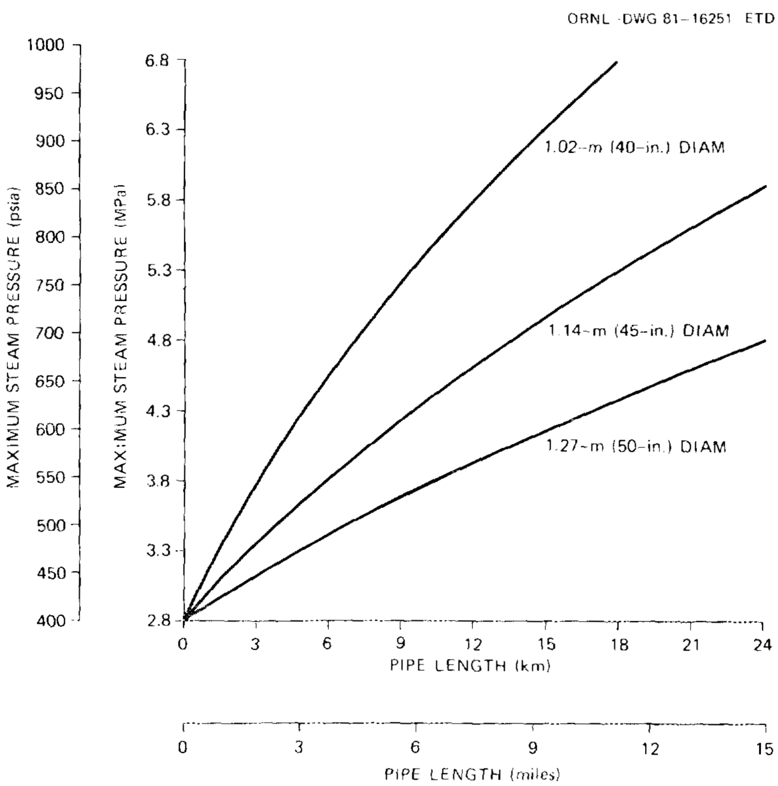


Fig. 2. Maximum steam pressure vs length for a steam flow of 454 kg/s (3,600,000 lb/h) and a delivered steam pressure of 2.75 MPa (400 psia).

Table 1. Maximum steam pressure for a steam flow of 454 kg/s (3.6 x $10^6$ lb/h) and a delivered steam pressure of 2.75 MPa (400 psia)

| Inside diameter [cm (in.)] | Pipe length [km (miles)] | Pressure [MPa (psi)] |
|---|---|---|
| 51 (20) | 0.1 (0.07) | 4.56 (662) |
| | 0.2 (0.14) | 5.72 (830) |
| | 0.3 (0.2) | 6.66 (967) |
| 64 (25) | 0.1 (0.07) | 3.31 (481) |
| | 0.5 (0.34) | 4.92 (714) |
| | 1.1 (0.68) | 6.35 (922) |
| 76 (30) | 0.1 (0.07) | 2.96 (430) |
| | 1.1 (0.68) | 4.39 (638) |
| | 2.2 (1.36) | 5.55 (806) |
| | 3.3 (2.03) | 6.51 (945) |
| 89 (35) | 0.1 (0.07) | 2.84 (413) |
| | 2.2 (1.36) | 4.17 (606) |
| | 5.5 (3.39) | 5.66 (822) |
| | 8.2 (5.08) | 6.65 (966) |
| 102 (40) | 0.1 (0.07) | 2.80 (406) |
| | 5.5 (3.39) | 4.39 (638) |
| | 10.9 (6.78) | 5.56 (808) |
| | 16.4 (10.17) | 6.53 (948) |
| 114 (45) | 0.1 (0.07) | 2.78 (403) |
| | 5.5 (3.39) | 3.71 (538) |
| | 10.9 (6.78) | 4.45 (646) |
| | 16.4 (10.17) | 5.09 (739) |
| | 19.1 (11.86) | 5.38 (781) |
| | 21.7 (13.50) | 5.65 (821) |
| | 24.1 (15.00) | 5.89 (855) |
| 127 (50) | 0.1 (0.07) | 2.77 (402) |
| | 5.5 (3.39) | 3.32 (482) |
| | 10.9 (6.78) | 3.80 (552) |
| | 16.4 (10.17) | 4.23 (614) |
| | 19.1 (11.86) | 4.43 (643) |
| | 21.7 (13.50) | 4.61 (670) |
| | 24.1 (15.00) | 4.77 (693) |

Note that the actual pipe length must include all necessary expansion formations. Therefore, the pipe length may be much greater than the delivery distance (Sect. 2.2 and Ref. 3).

Pipe cost as a function of diameter is given in Fig. 3 and Table 2 for three pipe lengths. Two cost lines, a low optimum case and a more conservative one, are given for each length. The differences between these two estimates are discussed in Sect. 2.5. These costs include 15 cm (6 in.) of cellular glass insulation. The installed cost for a 1.27-m (50-in.) pipeline 24 km (15 miles) long with 15 cm (6 in.) of insulation would be approximately $53 million.

Table 2. Installed pipe cost

| Pipe diameter [cm (in.)] | Maximum pressure [MPa (psi)] | Pipe length [km (miles)] | Pipe cost ($10^6$ $) | |
|---|---|---|---|---|
| | | | Low | Conservative |
| 89 (35) | 6.6 (959) | 8 (5) | 11.6 | 13.0 |
| 102 (40) | 5.0 (724) | 8 (5) | 12.5 | 13.6 |
| | 6.5 (941) | 16 (10) | 27.0 | 30.7 |
| 114 (45) | 4.1 (592) | 8 (5) | 13.6 | 14.5 |
| | 5.1 (735) | 16 (10) | 28.9 | 31.8 |
| | 5.9 (855) | 24 (15) | 45.4 | 51.3 |
| 127 (50) | 3.6 (516) | 8 (5) | 15.0 | 15.6 |
| | 4.2 (611) | 16 (10) | 31.2 | 33.6 |
| | 4.8 (693) | 24 (15) | 48.5 | 53.3 |

## 3.2 Steam Flow

The steam flow was varied from 125 to 500 kg/s (1 to 4 million lb/h). Figures 4 through 6 and Table 3 show the effect of pipe flow on steam source pressure for three pipe diameters: 1, 1.14, and 1.27 m (40, 45, and 50 in.). The pressure is less sensitive to flow variations in the larger diameter pipes.

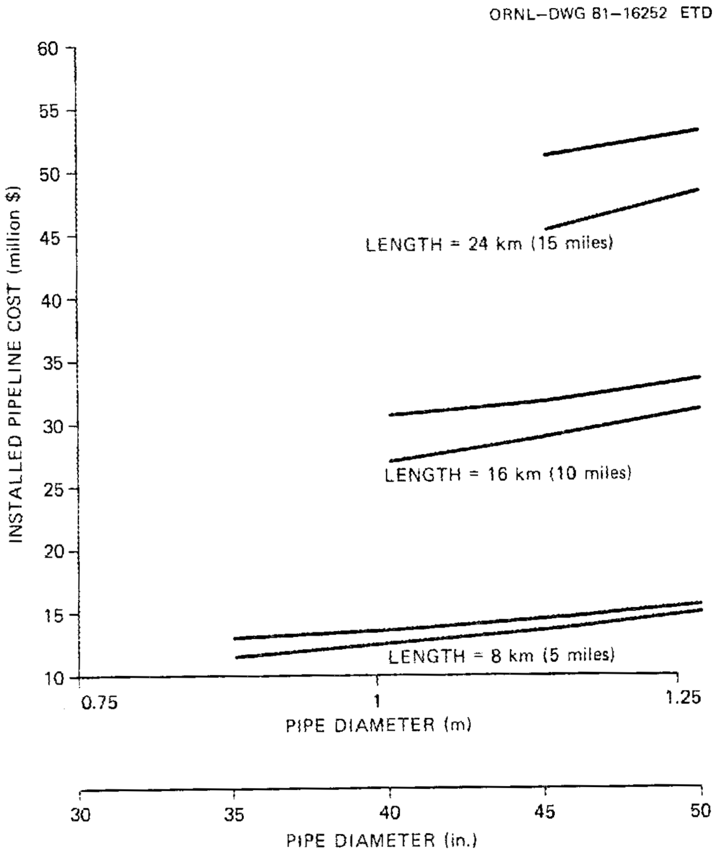


Fig. 3. Installed pipeline cost.

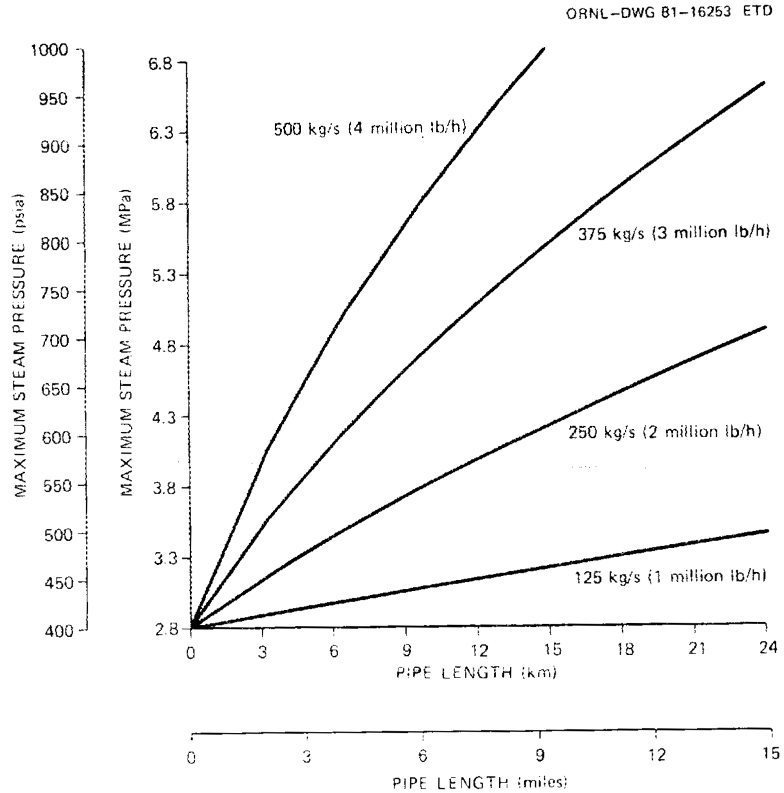


Fig. 4. Maximum steam pressure vs pipe length for a pipe diameter of 1.0 m (40 in.) and a delivered steam pressure of 2.75 MPa (400 psia).

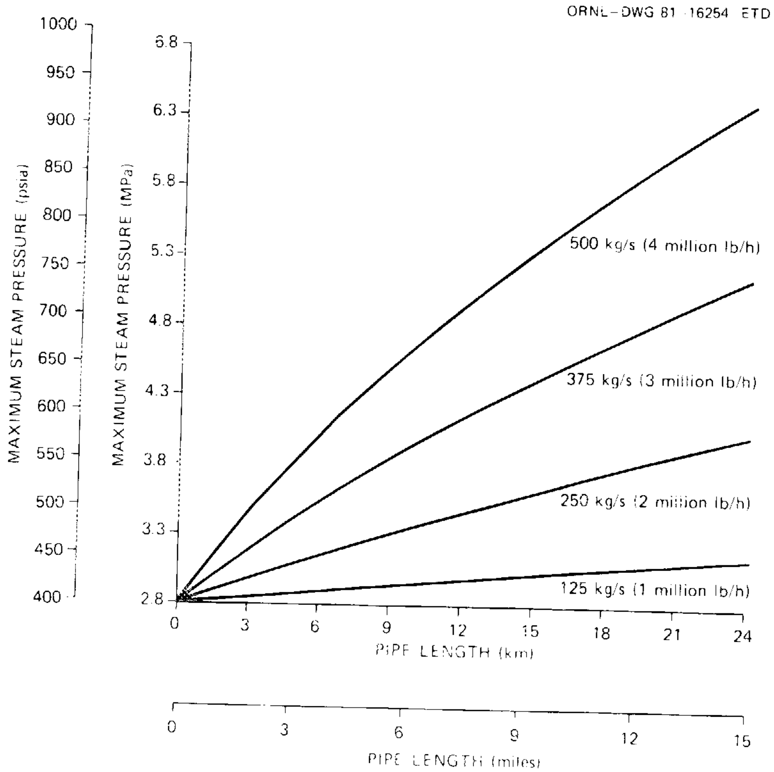


Fig. 5. Maximum steam pressure vs pipe length for a pipe diameter of 1.1 m (45 in.) and a delivered steam pressure of 2.75 MPa (400 psia).

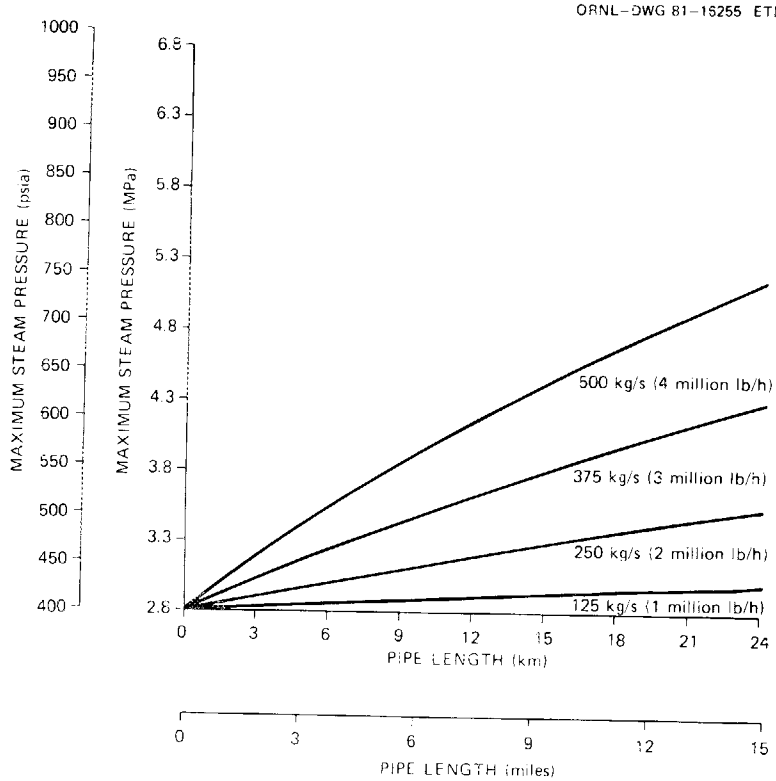


Fig. 6. Maximum steam pressure vs pipe length for a pipe diameter of 1.27 m (50 in.) and a delivered steam pressure of 2.75 MPa (400 psia).

Table 3. Maximum steam pressure for varying steam flows

| Pipe diameter [cm (in.)] | Steam flow [kg/s ($10^6$ lb/h)] | Steam pressure [MPa (psi)] for pipe lengths [km (miles)] | | | | |
|---|---|---|---|---|---|---|
| | | 0.1 (0.1) | 4.4 (2.7) | 8.7 (5.4) | 13.1 (8.1) | 17.5 (10.8) |
| 102 (40) | 126 (1) | 2.75 (400) | 2.89 (419) | 3.00 (436) | 3.13 (454) | 3.24 (471) |
| | 252 (2) | 2.77 (402) | 3.24 (470) | 3.66 (531) | 4.03 (585) | 4.38 (636) |
| | 378 (3) | 2.78 (404) | 3.75 (545) | 4.54 (659) | 5.20 (755) | 5.79 (841) |
| | 504 (4) | 2.81 (408) | 4.38 (636) | 5.54 (804) | 6.49 (943) | |
| 114 (45) | 126 (1) | 2.71 (400) | 2.82 (410) | 2.89 (419) | 2.95 (429) | 3.02 (438) |
| | 252 (2) | 2.76 (401) | 3.02 (438) | 3.26 (473) | 3.48 (506) | 3.69 (536) |
| | 378 (3) | 2.77 (402) | 3.31 (481) | 3.79 (551) | 4.21 (612) | 4.60 (668) |
| | 504 (4) | 2.78 (404) | 3.69 (536) | 4.44 (644) | 5.07 (736) | 5.63 (818) |
| 127 (50) | 126 (1) | 2.75 (400) | 2.80 (406) | 2.83 (411) | 2.87 (417) | 2.91 (422) |
| | 252 (2) | 2.76 (401) | 2.91 (422) | 3.04 (442) | 3.18 (462) | 3.31 (481) |
| | 378 (3) | 2.76 (401) | 3.08 (447) | 3.37 (490) | 3.65 (530) | 3.90 (567) |
| | 504 (4) | 2.77 (402) | 3.31 (481) | 3.79 (550) | 4.21 (612) | 4.60 (668) |

Based on Reynolds numbers, the steam flow was fully turbulent in each pipe segment for all flow rates and pipe diameters considered. The lowest Reynolds number was ~6.6 million, which corresponds to the lowest flow (125 kg/s) in the largest pipe diameter (1.27 m) considered. The lowest steam velocity in that case was 7.74 m/s (25.4 ft/s).

### 3.3 Insulation Type and Thickness

Several insulation manufacturers were interviewed[11] to determine an appropriate range of insulation thermal conductivities and installed costs for this study. Three types of insulation were chosen for consideration; their thermal conductivities and costs are shown in Table 4. Most of the cost quotations were for specified insulation thicknesses on given pipe sizes and were converted to a price per unit volume basis and averaged to give a representative cost for each type of material.

Table 4. Insulation thermal conductivities and costs

| Insulation type | Thermal conductivity[a] [W/m·°C (Btu/h·ft·°F)] | Cost [\$/m³ (\$/ft³)] |
|---|---|---|
| Cellular glass | 0.061 (0.035) | 990.00 (28.00) |
| Calcium silicate | 0.087 (0.050) | 1590.00 (45.00) |
| Inorganic granular | 0.095 (0.055) | 134.00 (3.80) |

[a]Approximate for 260°C (500°F).

Figure 7 and Table 5 show the variation of thermal loss with insulation thickness for a 1.25-m-diam (50-in.) pipe 24 km long with a steam flow of 454 kg/s ($3.6 \times 10^6$ lb/h). Figure 8 shows the variation in installed pipe cost for the same cases. The choice of insulation would depend on a combination of capital cost and heat loss, as well as projected energy costs.

Consider, for example, this case: (1) the lost energy is valued at approximately \$5.7/GJ (\$6/$10^6$ Btu), (2) the pipeline operates at a design flow of 454 kg/s ($3.6 \times 10^6$ lb/h), and (3) the pipeline operates 70% of

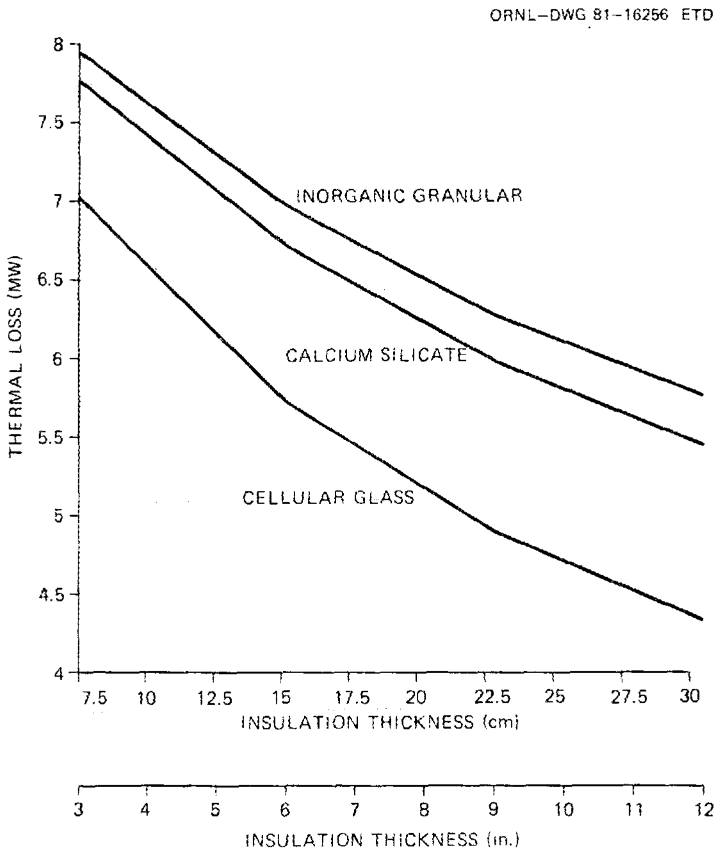


Fig. 7. Thermal loss vs insulation thickness for a 24-km (15-mile) pipeline.

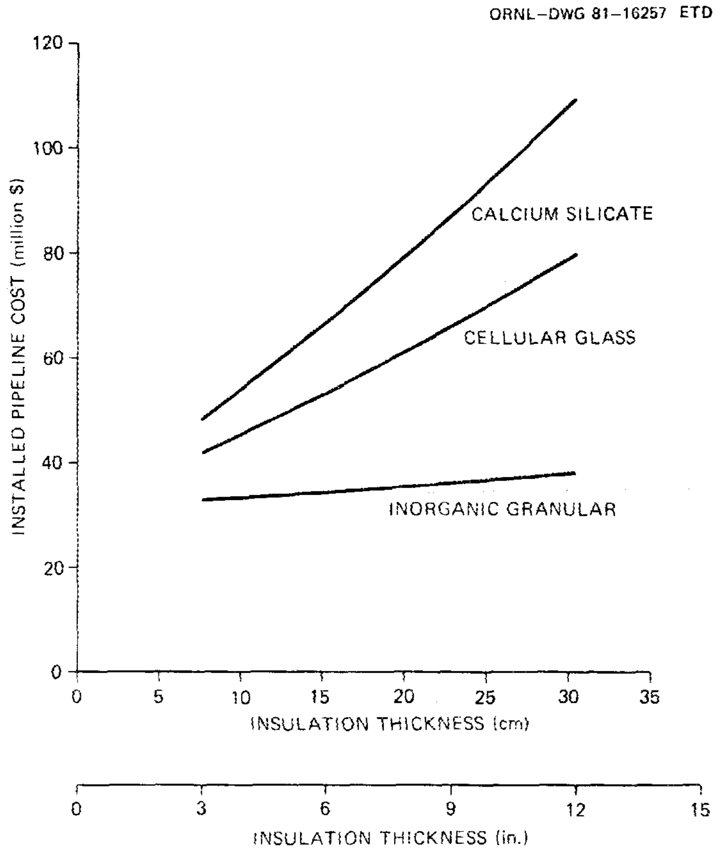


Fig. 8. Installed pipeline cost for varying insulation types and thicknesses on a 1.25-m (50-in.) pipe 24 km (15 miles) long.

Table 5. Thermal loss and installed pipe costs for varying insulation types and thicknesses for a 1.27-m (50-in.) pipe 24 km (15 miles) long with a 454 kg/s ($3.6 \times 10^6$ lb/h) steam flow

| Insulation type | Insulation thickness [cm (in.)] | Thermal loss [MW ($10^6$ Btu/h)] | Pipe cost ($10^6$ \$) |
|---|---|---|---|
| Cellular glass | 7.6 (3) | 7.0 (24.7) | 41.74 |
| | 15.2 (6) | 5.7 (19.3) | 53.30 |
| | 22.9 (9) | 4.9 (16.6) | 66.00 |
| | 30.5 (12) | 4.3 (14.6) | 79.84 |
| Calcium silicate | 7.6 (3) | 7.8 (26.2) | 48.09 |
| | 15.2 (6) | 6.7 (22.7) | 66.68 |
| | 22.9 (9) | 6.0 (20.2) | 87.10 |
| | 30.5 (12) | 5.5 (18.4) | 109.33 |
| Inorganic granular | 7.6 (3) | 7.9 (26.9) | 32.72 |
| | 15.2 (6) | 7.0 (23.5) | 34.28 |
| | 22.9 (9) | 6.3 (21.2) | 36.00 |
| | 30.5 (12) | 5.8 (19.5) | 37.88 |

the time. In going from 7.5 to 15 cm (3 to 6 in.) of cellular glass insulation, the annual energy loss drops from 15.5 to $12.6 \times 10^{13}$ J (14.72 to $11.96 \times 10^{10}$ Btu). At \$5.7/GJ, the annual saving is \$165,000. The increased cost, however, is \$11.6 million. This is a very low savings for such a large expenditure; the simple payback period (with no cost escalations) is ~70 years.

For the lower-cost inorganic granular material, the same comparison shows an annual energy savings of $2.14 \times 10^{13}$ J, an annual cost savings of approximately \$122,000, and an increased initial capital cost of approximately \$1.6 million. This alternative has a payback period of ~13 years.

The expected range of flow rates must also be considered when choosing an insulation type and thickness. The rate of heat loss is a function of steam temperature and insulation thickness. The absolute energy loss will be the same for a very low steam flow rate as for a full design flow rate, if the steam temperature stays the same. Since the energy loss at low flow rates comes from a smaller quantity of steam, the relative energy loss per unit of mass is much greater. These higher energy losses can lead to condensation in the pipeline. Therefore, if variable steam flow rates are anticipated, more insulation may be necessary than would be chosen on an economic basis.

## 4. CONCLUSIONS

This evaluation considers the pipeline flow of steam for industrial use. Pipe lengths up to 24 km (15 miles) were examined and appear to be feasible. This pipe length corresponds to a transport distance of ~16 km (10 miles).

The report indicates that such a pipeline would cost approximately $53 million. If the pipeline, operating at a capacity factor of 0.70 with a fixed-charge rate of 17%, delivers $2.33 \times 10^{16}$ J ($2.21 \times 10^{13}$ Btu) in one year, the transport cost of the steam would be approximately $0.38/GJ ($0.41/$10^6$ Btu). This cost would be increased by the addition of a condensate return line but would still be within a reasonable range.

The costs presented here are for feasibility purposes only. The site-specific costs will vary widely from one region to another.

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