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Piping System Analysis for High Temperature Heat Delivery from an LWR

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Piping System Analysis for High Temperature Heat Delivery from an LWR

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EXECUTIVE SUMMARY

Background

The United States and countries around the world are seeking to reduce dependence on fossil fuels to achieve climate goals and ensure national energy security. In the United States, industrial process heat based on fossil fuel sources accounts for approximately 30% of all greenhouse gas emissions. Replacing those heat sources with low-carbon “clean” heat is receiving considerable attention. This work focuses on the technical aspects of delivering heat from light water reactors and upgrading that heat to high temperature (350-550°C) using mechanical heat pumps to support a wider array of industrial processes.

This report focuses on determining optimal heat delivery methods and provides lowest cost piping designs that satisfy heat duty requirements and other constraints specific to different applications. Two types of heat loads are considered, including sensible heat loads and uniform temperature loads (such as for phase change processes). Several different heat transfer fluids (HTFs) are evaluated, including air, nitrogen (N_2), helium (He), argon (Ar), carbon dioxide (CO_2) and water/steam. Operational conditions include delivery and return temperatures, pressure, and mass flow rate. As shown in Figure ES 1, the required mass flow rate depends on the heat duty and the HTF. In general, steam/water has the lowest mass flow rate and compressor power requirements.

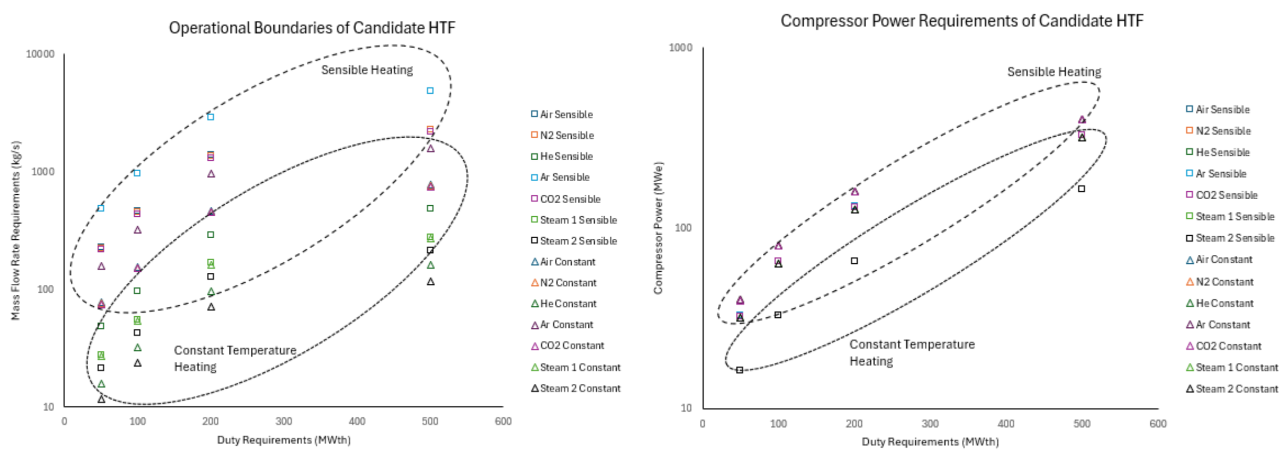

Figure ES 1. Mass flow rate requirements as functions of heat duty for various heat transfer fluids (HTFs), including two vapor compression cycle steam cases. For one case (Steam 1), the steam is condensed at the thermal load and returned to the LWR as cooled liquid water, and for the other case (Steam 2), the water is returned as saturated liquid at the LWR steam generator feedwater inlet temperature.

To deliver the specified heat duty while designing the best cost-effective heat transport piping system, a workflow based on the code standards by the American Society of Mechanical Engineers (ASME), industry-standard-compliant solutions, and cost optimization has been devised and implemented into a Microsoft Excel© file format for pipe design development. The total cost consists of capital cost and operating cost, including costs associated with heat losses. The capital cost is divided into pipe material cost, HTF cost, and insulation material cost. For a specified system lifetime and interest rate, the total capital cost is amortized to provide an annual cost with labor cost included.

The maximum heat delivery distance was determined for the HTFs featured in Figure ES 1 assuming a heat duty of 500 MW, a maximum insulation thickness of 1 m and a maximum temperature loss of 3 °C for fixed gas HTFs and a decrease in steam quality of 0.09 for steam. The total amortized costs listed in descending order were found to be Steam 2 (condensate from industry process returned as cooled water), Steam 1 (condensate from industry process returned as heated, saturated water), CO_2, He, N_2, air, and Ar, as shown in Figure ES 2. This particular order is not necessarily a generalized result but applies for heat delivery at temperatures between 116 °C and 300 °C as well as a delivery pressure of 2.5 bar for single phase gas HTFs and 57 bar – 225 bar for water. For different delivery and return conditions, different HTF rankings are possible. To investigate the sensitivity to heat duty, the same process used to create the data in Figure ES 2(a) was applied to 16 cases different cases. As expected, as heat duty increases, the normalized cost of heat delivery (cost per unit of heat per unit of distance) decreases. For all cases, the dominant cost in optimal heat delivery system is capital cost.

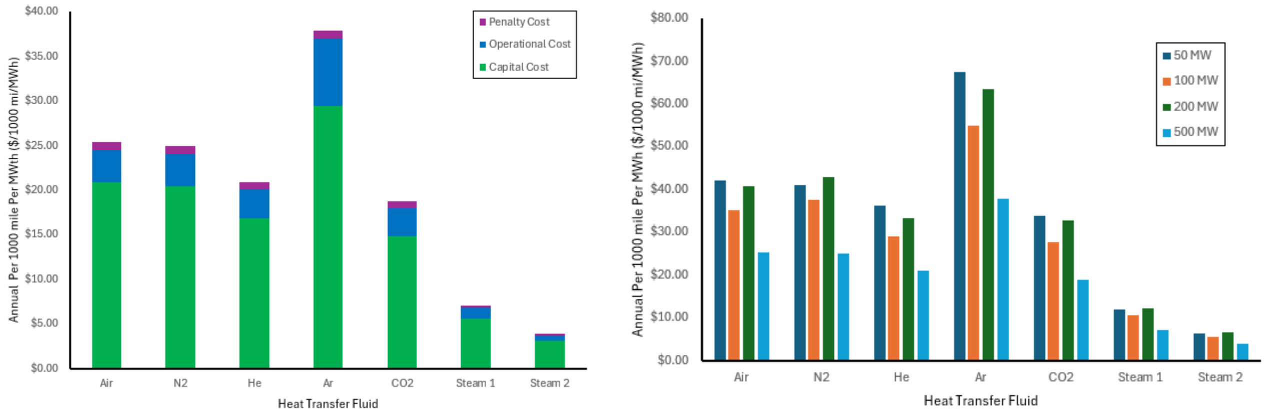

Figure ES 2. Capital cost, operational cost, and penalty cost, which is the cost associated with loss of heat, for a 500 MW heat delivery line for candidate HTFs (left panel); and sensitivity normalized heat delivery cost on heat duty

These results are also compared to those of previous studies of long-distance heat delivery for district heating and power transmission, which are shown in Figure ES 3. For district heating with ~10 km of pipe, the normalized cost has been estimated to be in the range of €2/MWh/km or approximately $3.3/MWh/mile (Figure ES 3, left panel). On a much larger scale, power delivery via oil lines and electric power lines are estimated to be approximately $0.008/MWh/mile for oil and $0.4/MWh/mile for high-voltage electricity (Figure ES 3, right panel). The steam cases in Figure ES 2, with projected costs in the range of $0.02-0.06/MWh/mile compares favorably with these other options.

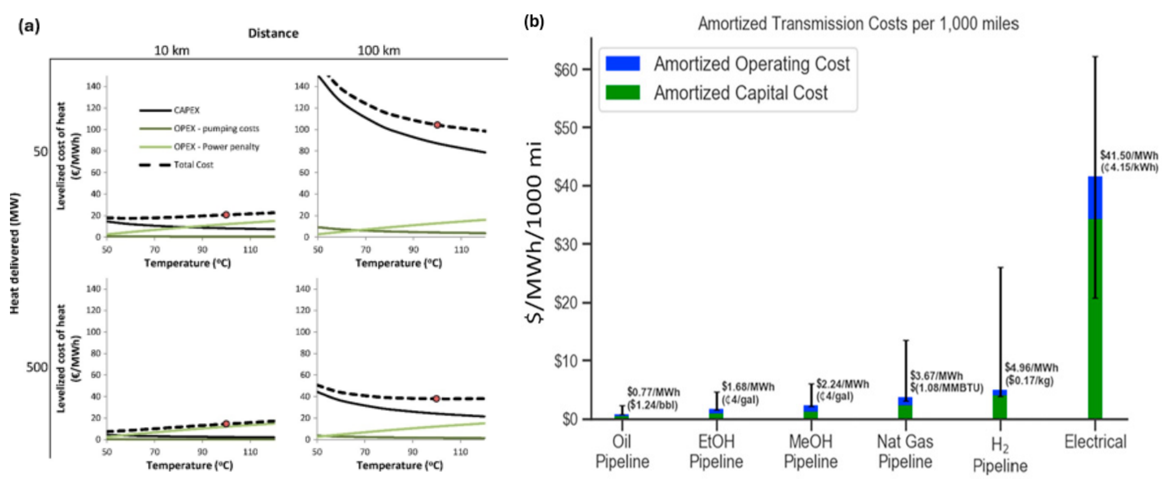

Figure ES 3. Sensitivity of district heating cost for different supply temperatures, distances and heat duties from Kavvadias et al. (left panel); and amortized cost of energy transmission over 1000 miles by different energy carriers from DeSantis et al. (right panel).

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ACRONYMS

ASME	American Society of Mechanical Engineering
CAPEX	Capital expenditures
CHP	Combined heat and power
COP	Coefficient of Performance
HTF	Heat transfer fluid
HVDC	High-voltage direct current
LCOH	Levelized cost of heat
LWR	Light-water reactor
NG	Natural gas
NPP	Nuclear power plant
NPS	Nominal pipe size
OPEX	Operating cost
US	United States
VCC	Vapour compression cycle

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PIPING SYSTEM ANALYSIS FOR HIGH TEMPERATURE HEAT DELIVERY FROM AN LWR

1. INTRODUCTION

The rapid urbanization and growing concerns about climate change have placed energy efficiency and sustainable heating solutions at the forefront of modern infrastructure development. Long-distance heat delivery piping systems are an integral part of this effort offering a means to distribute thermal energy efficiently from a central source to multiple end-users over considerable distances. These systems, which transport hot water, steam, or other means of HTF, have been pivotal in reducing energy waste, lowering carbon emissions, and optimizing energy use across target heat users. Long-distance heat delivery systems can be traced back to the late 19th century, with some of the earliest systems being implemented in European and North American cities [2]. Initially, these systems were steam distributions generated by coal-fired boilers through cast iron pipes to nearby buildings for district heating. As urban areas expanded and the demand for centralized heating systems grew, the need for more efficient and reliable systems became evident. The evolution of these systems was marked by technological advancements, particularly in materials and insulation, which allowed for the extension of these networks over greater distances while minimizing heat losses [3].

The development of pre-insulated pipes in the mid-20th century marked a significant milestone in the evolution of district heating. These pipes, often made of steel or high-density polyethylene and insulated with polyurethane foam, drastically reduced heat loss during transmission, making it feasible to extend district heating networks over longer distances [4, 5]. This innovation was complemented by advancements in heat generation technologies, including the use of CHP plants, which further improved the efficiency of these systems by simultaneously producing electricity and thermal energy [6]. Existing LWRs supply heat for district heating [7,8] or desalination [9,10] and a limited number of cases with delivery of process steam to other industries. Although LWRs without heat pumps or electrical heaters is limited to supplying 300 °C heat, replacing centralized fossil heat generation sources with NPPs provide carbon free heat.

1.1 Principles of Long-Distance Heat Delivery

Long-distance heat delivery systems operate on a simple principle, thermal energy is generated at a central plant and then distributed through a network of insulated pipes to multiple end-users, such as district heating for residential, commercial, and industrial buildings. The two traditional mediums used for heat transport in these systems are hot water and steam [2], each with its advantages and limitations depending on the specific application and distance involved. The following are core aspects of long-distance heat delivery:

- **Heat Generation**: heat can be produced from a variety of sources, including fossil fuels (natural gas, coal, oil), biomass, waste incineration, and renewable energy sources such as geothermal, solar thermal, and industrial waste heat. Considering United States' (U.S.) 2050 carbon neutral goals, the shift towards low-carbon and renewable energy sources has become more pronounced, driven by future environmental regulations and the need to reduce greenhouse gas emissions.

- **Distribution Network**: the heat generated is transported through a network of insulated pipes. For direct heat delivery, large-diameter pipes are used to transport the entire heat to the target heat user. If substations exist, smaller-diameter pipes are used to distribute the heat to each individual target heat user [2].

- **Heat Transfer and Substations**: If more than one heat exist, substations play a crucial role in the efficient operation of heat distribution. They act as intermediaries between the primary and secondary circuits, utilizing heat exchangers to transfer thermal energy from the high-temperature primary circuit to the lower-temperature secondary circuit. This process ensures that the heat is delivered to end-users at the appropriate temperature while maintaining the integrity of the primary circuit [11].

1.2 Challenges and Considerations

Despite the numerous benefits and technological advancements, long-distance heat delivery systems face several challenges that need to be addressed to ensure their continued viability and expansion. These challenges include technical, economic, and environmental factors.

- **Heat Loss and Efficiency**: Even with modern insulation techniques, some heat loss is inevitable during transmission, especially over long distances. Heat loss can reduce the overall efficiency of the system and increase operating costs. The design and material choice for pipes, as well as the proper installation and maintenance of the network, are critical factors in minimizing heat loss.
- **Infrastructure Costs**: The construction of long-distance heat delivery systems requires significant upfront investment. The cost of laying insulated pipes, building piping systems and pipe supports, and integrating with existing infrastructure can be substantial. Financing models, government incentives, and public-private partnerships can play a key role in addressing the financial challenges associated with long-distance heat delivery.
- **Environmental and Social Considerations**: While replacing carbon emission heavy industrial processes with heat delivery offer significant environmental benefits, they are not without their environmental and social impacts. The construction of extensive underground pipe networks can disrupt local ecosystems and communities, necessitating careful planning and stakeholder engagement to mitigate these effects [12]. Public acceptance and community involvement are critical to the successful implementation of long-distance heat delivery.

1.3 Heat Delivery for Heat Augmentation

The focus of the report is to investigate viable options for high temperature industrial process heat delivery systems from LWRs. Since LWR secondary loop temperature supply limit is 300 °C, heat delivery of higher temperatures will require heat augmentation methods. The technology proposed applies vapor compression, Brayton cycle, and split compression recuperated cycles. Two options are considered for the heat sink: sensible heating and uniform temperature heating. Using COP as the main metric to measure thermodynamic performance, each heat pump technology will be evaluated based on set operating conditions. To minimize heat delivery heat loss, heat augmentation mechanisms are placed before the target high temperature industrial process. This will allow long-distance heat delivery temperatures to be below 300 °C and minimizes the temperature gap between ambient conditions.

1.4 Thermal Delivery Design and Technoeconomic Analysis

The objective of this report is to derive optimal designs of the piping and insulation systems. It is expected to achieve heat delivery from the boundary of the LWR to the heat pump before the boundary of high temperature industrial requirements. “Optimal design” refers to an insulated piping system that ensures meeting the desired heat-delivery objective at the lowest cost. In identifying the optimal designs for the heat-delivery scenarios towards the heat pump reference cases, the following considerations were incorporated to suggest a practical, industry-standard-compliant solution suitable for commercial deployment:

1. NPS or custom piping.
2. Design requirements as per ASME B31.1 (power piping), including maximum allowable stress and temperature of the selected pipe material, required thickness of pipes under pressure, and necessary pipe loops to accommodate thermal expansion.
3. Operational practices for high-temperature thermal systems and/or devices (e.g., heat exchanger), such as maximum or minimum velocity limits of the HTF.

It is important to note that the selection of HTF can significantly impact the insulated piping system design and its optimization. The following subsections describe the methodology proposed to identify the optimal piping and insulation system designs for the thermal integration scenarios discussed in Section 3, and the optimization results.

2. METHODOLOGY

The piping system analysis, design, and optimization requires an integrated understanding of industry standards, ASME requirements, and operational practices for high-temperature systems. The objective is to find an optimal heat-delivery design that minimizes the capital and lifetime operating costs. The assessment is initiated by selecting the piping material, HTF, heat-delivery distance, and insulation material. Each selected component is chosen to efficiently function under operational conditions defined by the application case and must be well within component operational limitations. To scope the piping analysis, key assumptions are applied, which include: (1) uniform pipe diameter, (2) level piping (no change in elevation), (3) only frictional, form, and velocity change induced pressure losses (ΔP) are considered, (4) capital cost contributions from flanges, fittings, valves, and pumps, along with their varied ratings, are ignored in the piping stress and cost analysis, (5) piping system is above ground level avoiding shear stress from soil, and (6) pipe aging is ignored. For pressure loss, the velocity change can be induced by HTF density changes axially in the piping due to thermal energy loss.

For systems of low thermal energy transport, the density change is assumed negligible with installation of suitable insulation to prevent significant heat losses. However, for higher thermal energy transport systems, the allowable amount of heat loss can result in temperature changes of over 10°C and potentially increase density significantly depending on the HTF. To satisfy all applicable design standards, design requirements, and best practices, the logical flow and structure of the design optimization analysis is paramount. The first design standard or design requirement applied must be carefully selected because one design standard or design requirement could result in multiple constraints to the piping analysis. Figure 1 shows the logical flow and structure of the design optimization analysis proposed in this study, considering the flow of information and the objectives to design low-cost piping systems while meeting the design standards and requirements. The sections below expand on the logical structure described in Figure 4.

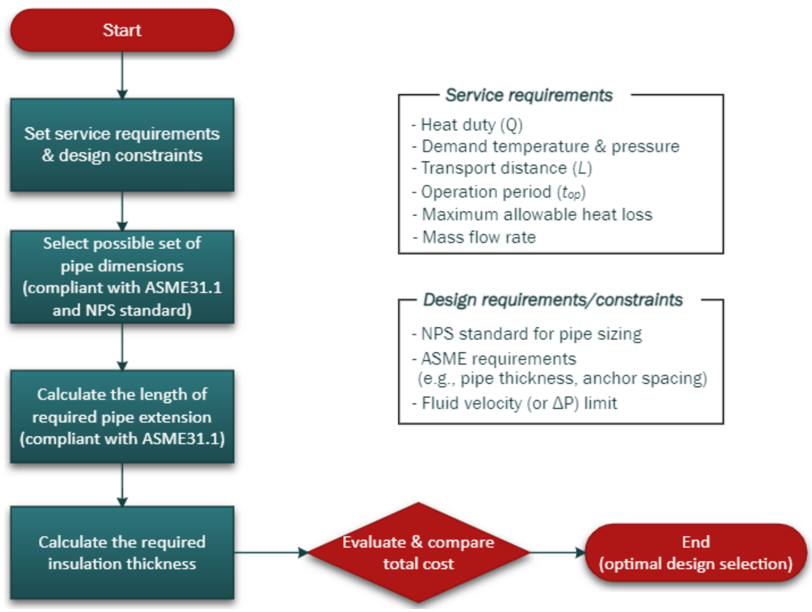


Figure 4. Logical structure of the heat delivery (piping) system analysis for optimization

2.1 Requirements and Design Constraints

The first step is to determine the service requirements and design constraints. This includes the required heat-delivery duty (Q), source and sink demand temperature and pressure, transport distance (L), operation period, maximum allowable heat loss ($Q_{l,max}$), and mass flow rate ($\dot{m}$). These service requirements are also the minimal set of information to bound the heat-delivery design. Absence of this type of information will result in a design analysis of larger degrees of freedom and may yield non-optimized final designs. In the current approach, design optimization is constrained by standard industry pipe sizes, maximum and minimum velocity limits of the HTF, allowable operational temperature range, and heat-delivery system budget limitations. The combination of design standards and constraints will specify the range of applicable pipe outer-diameter sizes to investigate while applying ASME requirements to determine possible pipe dimensions suitable for commercial deployment. The service requirements and design constraints should be defined prior to initiating the thermal delivery design process.

2.2 Possible Set of Pipe Dimensions

The next step is to identify the possible set of NPS and their dimensions to meet the ASME requirement and design constraints for a given service requirement. For the selected pipe material, available NPS and schedules differ for the given operational requirements. ASME B31.1 has categorized and specified the allowable stresses in tables based on material and the operating temperature range. The two major stresses considered are sustained and displacement stresses. Sustained stress ranges are resulting forces from pressure, weight, or any other force applied along the defined piping system. Sustained pressure stresses are either internal or external influences that uniformly distribute along the piping. For internally and/or externally pressurized systems, ASME B31.3 provides the function to estimate the minimum pipe thickness (t_m) required and is shown in Equation (1).

$$t_m = \frac{PD_o}{2(SE \cdot W + Py)} + A \tag{1}$$

where

P	=	applied internal pressure (gauge pressure)
D_0	=	outer pipe diameter
SE	=	maximum allowable stress at operational temperature
W	=	weld strength reduction factor
y	=	coefficient taking into account material properties and operational temperature
A	=	additional thickness to compensate for removed material such as installing mechanical joints and corrosion.

Sustained weight stress or other sustained loads are also called longitudinal stresses, and the sum shall not exceed the basic material allowable stress in hot conditions. The criterion for sustained loads (S_L) is shown in Equation (2).

$$S_L = \sqrt{\left[I_a \left|\frac{PD_O}{4t_n} + \frac{F_a}{A_p}\right| + \frac{\sqrt{(I_i M_{iA})^2 + (I_o M_{oA})^2}}{Z}\right]^2 + \left(\frac{I_t M_{tA}}{Z}\right)^2} \leq S_h \tag{2}$$

where

I_a	=	sustained longitudinal force index
t_n	=	pipe thickness
F_a	=	longitudinal force due to weight and other sustained loads
A_p	=	cross-sectional material area of the pipe
I_i	=	sustained in-plane moment index
M_{iA}	=	in-plane moment
I_o	=	sustained out-of-plane moment index
M_{oA}	=	out-of-plane moment
Z	=	nominal section modulus of pipe
I_t	=	sustained torsional moment index
M_{tA}	=	torsional moment
S_h	=	basic material allowable stress at the maximum temperature expected.

When of sufficient initial magnitude during system startup or extreme displacements, piping system stress ranges caused by thermal expansion and piping displacements, referred to as displacement stress, relax in the maximum stress condition as the result of local yielding or creep. For such non-sustained stresses, the stress is applied and relieved repeatedly creating cyclic stress. Depending on the number of cycles experienced during the lifetime of the piping system, the maximum allowable stress is reduced. This is known as the allowable displacement stress range (S_A). The criterion for displacement stress (S_E) is shown in Equation (3).

$$S_E = \sqrt{\left[\left|\frac{i_a F_C}{A_p}\right| + \frac{\sqrt{(i_i M_{iC})^2 + (i_o M_{oC})^2}}{Z}\right]^2 + \left(\frac{i_t M_{tC}}{Z}\right)^2} \leq S_A \tag{3}$$

where

i_a = axial force stress intensification factor

F_C = axial force range due to reference displacement load range

i_i = in-plane stress intensification factor

M_{iC} = displacement in-plane moment

i_o = out-of-plane stress intensification factor

M_{oC} = displacement out-of-plane moment

i_t = torsional stress intensification factor.

The allowable displacement stress range is calculated using the cyclic stress equation in Equation (4).

$$S_A = f(1.25S_c + 0.25S_h) \tag{4}$$

where

f = the cyclic stress range factor for the total number of equivalent reference displacement stress cycles

S_c = basic material allowable stress at the minimum temperature expected.

When the basic material allowable stress at the maximum temperature is greater than the determined sustained load stress, the difference between both parameters is added in Equation (5).

$$S_A = f(1.25S_c + 1.25S_h - S_L) \tag{5}$$

The cyclic stress range factor depends on the total number of equivalent reference displacement stress range cycles expected during the service life of the piping (N) and is limited to be equal or less than one as shown in Equation (6). The cause for such restraint leads to the concept of reduced allowable stress when the given piping system experiences multiple cycles of loading and unloading over the expected lifetime. Since a piping system cannot extend the allowable stress by having fewer cycles, the maximum allowable stress is when the system is cycle-load free, in other words, the cyclic stress range factor is equal to one.

$$f = \frac{6}{N^{0.2}} \leq 1.0 \tag{6}$$

A minimum value for the cyclic stress factor is 0.15, which results in an allowable displacement stress range for a total number of equivalent reference displacement stress range cycles greater than 10^8 cycles. When considering more than a single displacement stress range, whether from thermal expansion/contraction or other cyclic conditions, each significant stress range shall be computed. For the selected reference displacement stress range, the total number of equivalent reference displacement stress range cycles is in Equation (7).

$$N = N_E + \sum \left(q_i^5 N_i \right), q_i = \frac{S_i}{S_E} \text{ for } i = 1,2, \dots , n \tag{7}$$

where

N_E = number of cycles of the reference displacement stress range

N_i = number of cycles associated with displacement stress range

q_i = ratio of the displacement stress range

S_i = any computed stress range other than the reference displacement stress range.

For available NPS and associated schedules provided by vendors, pipe dimensions can also be determined by knowing the pipe material, pipe diameter, and pipe minimum thickness. It is recommended to cross check if the chosen pipe dimensions comply with the criteria in Equations (2) and (3). Given the operational temperature, pressure, and cyclic behavior, higher pipe schedules may be required.

2.3 Length of Required Pipe Extension

The range of pipe conditions that satisfy the ASME standards in B31.3 means the required extended pipe extension must be determined to accommodate displacement stress ranges for a given displacement length. Piping systems of uniform size have no more than two anchors and no intermediate restraints, and total number of cycles must be less than seven thousand. ASME B31.3 provides the approximate criterion to determine whether the selected pipe extension is within standards and is shown in Equation (8).

$$\begin{cases} (\text{U.S. Customary Units}) \dfrac{D_o Y}{(L-U)^2} \leq 30 \dfrac{S_A}{E_c} \\ (\text{SI Units}) \dfrac{D_o Y}{(L-U)^2} \leq 20800 \dfrac{S_A}{E_c} \end{cases} \tag{8}$$

where

E_c = modulus of elasticity at room temperature

L = developed length of pipe (total length of pipe taken along the pipe axial direction)

U = anchor distance (length of straight line between the anchors, not the axial distance along the pipe).

Using the stated criterion, the required extension length between two anchors can be calculated. As stated in ASME B31.3, all piping shall meet the following requirements with respect to flexibility:

- It shall be the designer's responsibility to perform an analysis unless the system meets one of the following criteria:
    - The piping system duplicates a successfully operating installation or replaces a system with a satisfactory service record.
    - The piping system can be adjudged adequate by comparison with previously analyzed systems.
    - Follows the criteria shown in Equations (2), (3), and (8).
- All systems not meeting the above criteria (or where reasonable doubt exists as to adequate flexibility between the anchor) shall be analyzed by simplified, approximate, or comprehensive methods of analysis that are appropriate for the specific case.

- Approximate or simplified methods may be applied only if they are used for the range of configurations for which their adequate accuracy has been demonstrated.
- Acceptable comprehensive methods of analysis include analytical methods, model tests, and chart methods that provide an evaluation of the forces, moments, and stresses caused by bending and torsion from the simultaneous consideration of terminal and intermediate restraints to thermal expansion of the entire piping system under consideration, and including all external movements transmitted to the piping by its terminal and intermediate attachments. Correction factors shall be applied for the stress intensification of curved pipe and branch connections, as provided by the details of these rules, and may be applied for the increased flexibility of such component parts.

Along with the assumptions previously stated and standards mentioned above, the required extension length starts with determining the allowable maximum anchor distance. As the distance between anchors is increased, the piping is subjected to higher weight loads and, depending on the pipe material, values can vary. Although there is suggested pipe-support spacing for stainless steel and carbon--based metals according to pipe outer diameter and thickness from ASME B31.3, the spacing is general and may not apply to all cases. The following approach that uses Equations (2), (3), and (8) as the thermal delivery piping system designs is a first-of-a-kind design, such that precedent cases of successfully operating systems under the proposed conditions do not exist. First, under the assumptions applied, Equation (2) is simplified due to no out-of-plane, torsional moments, or sustained external longitudinal forces.

$$S_L = I_a \left|\frac{PD_O}{4t_n}\right| + \frac{I_i M_{iA}}{Z} \leq S_h \tag{9}$$

The maximum in-plane moment at the center of the piping assumes a uniform weight load is derived:

$M_{iA} = \frac{wU^2}{8}$ [13] where w is the uniform load in units of weight-force per distance. Considering the weight contributions are from insulation, piping, and HTF, the maximum moment at the center is modified.

$$M_{iA} = \frac{U^2}{8}\left(w_{ins} + w_{pipe} + w_{HTF}\right) \tag{10}$$

where

w_{ins} = insulation weight load

w_{pipe} = pipe weight load

w_{HTF} = HTF weight load.

A visual representation of uniform load can be found in Figure 5. The maximum section modulus is derived to be $Z = \frac{I}{c}$ where I is the moment of inertia and c is the radial distance from the neutral axis (for this case, the center of the pipe). Added with the moment of inertia [13] and radial distance, the maximum section modulus is Equation (11).

$$Z = \frac{\pi}{32D_o}\left(D_o^2 - D_i^2\right) \tag{11}$$

where

D_i = pipe inner diameter.

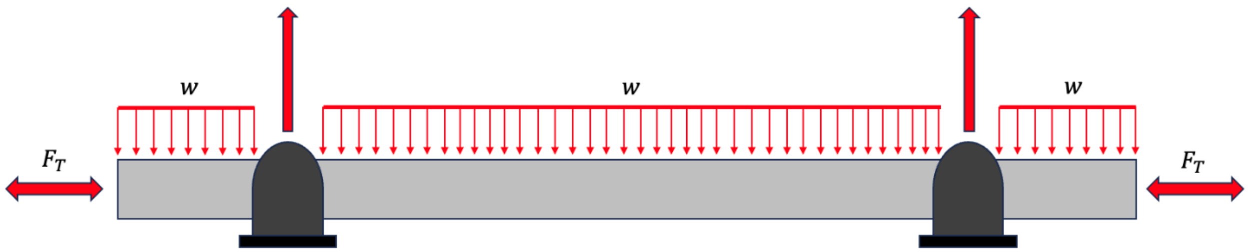


Figure 5. Leveled uniform-sized piping and the resulting uniform weight load.

Inserting Equations (10) and (11) into Equation (9) and reorganizing to be explicit about the anchor distance, the ranges of allowable anchor distances are derived.

$$U \leq \sqrt{\frac{\pi\left(D_o^4 - D_i^4\right)}{4D_0 I_i\left(w_{pipe} + w_{HTF} + w_{ins}\right)}\left(S_h - \frac{I_a P D_O}{4t_n}\right)} \tag{12}$$

To abide to flexibility conditions defined in Equation (3), the allowable thermal stress must be ensured under the given operation conditions. The limiting factor that will dictate the maximum allowable displacement stress is the allowable displacement stress range. Acknowledging the only additional reference stress during operations is via thermal expansion and contraction, the displacement stress criterion is simplified.

$$S_E = \frac{i_a F_C}{A_p} = i_a \sigma_T \leq S_A \tag{13}$$

where σ_T is the thermal stress obtained when a known thermal strain (ε_T) and modulus of elasticity at operating temperature (E) is determined ($\sigma_T = E\varepsilon_T$).The thermal strain can be calculated by considering the linear thermal expansion coefficient (α_L) and temperature difference between operational temperature (T_{oper}) and piping installation temperature (T_{init}). Given the parameters provided and Equation (13), the final form of the allowable displacement stress is determined.

$$S_E = i_a E\varepsilon_T = i_a E\frac{U\alpha_L\left(T_{oper} - T_{init}\right)}{U} = i_a E\alpha_L\left(T_{oper} - T_{init}\right) \leq S_A \tag{14}$$

The final form shown in Equation (14) provides insight on how the calculated displacement stress is independent of anchor distance. However, the allowable displacement stress range depends on anchor distance per Equation (5) if the criterion in Equation (2) is followed. Such an arrangement leads to complications. Depending on the modulus of elasticity at operational temperature and temperature difference, no matter how the anchor distance is shortened, criterion from Equation (14) may be impossible to satisfy. One important aspect to consider is the nature of displacement stress. As described in earlier sections, the displacement stress covers non-sustained applied stresses, such as cyclic stresses. During thermal delivery operations, there are anticipated HTF temperature fluctuations depending on dynamic operations and ambient condition changes. Thus, the temperature difference to use is not installation temperature but the minimum temperature ($T_{oper,min}$) anticipated during operations.

$$S_E = i_a E\alpha_L\left(T_{oper} - T_{oper,min}\right) \leq S_A \tag{15}$$

The extended length can now be determined by rearranging the criterion in Equation (8) to be explicit about the extension length.

$$\begin{cases} (\text{U.S. Customary Units}) L \geq \sqrt{\dfrac{D_o Y E_c}{30 S_A}} + U \\ (\text{SI Units}) L \geq \sqrt{\dfrac{D_o Y E_c}{20800 S_A}} + U \end{cases} \tag{16}$$

The developed length of pipe or total pipe length is adapted by using expansion loops, elbows, "Z" bends, or bellows joints, as required. For expansions loops, the extended amount of piping is offset by vertically or horizontally displacing a portion of the piping. Considering how the pipe is offset and returns to the original position as shown in Figure 6, the sum distance displaced is the extended pipe length. In this thermal delivery piping analysis, the expansion loop method was selected as it follows the assumption of no change in elevation. The best practice for expansion loops is to make the expansion loop height (H) half of the total pipe length, and the expansion loop width (W) half of the expansion loop height. The sum of the delivery and return height, and width provides the total pipe length.

$$H = \frac{L}{2}, W = \frac{H}{2} \ where\ L = W + 2H \tag{17}$$

Since the analysis was restricted only between two anchors, the corrected HTF transport distance is the product of anchor numbers and extended lengths. To minimize the corrected HTF transport distance, greater anchor distances and reduced applied sustained stresses will be required.

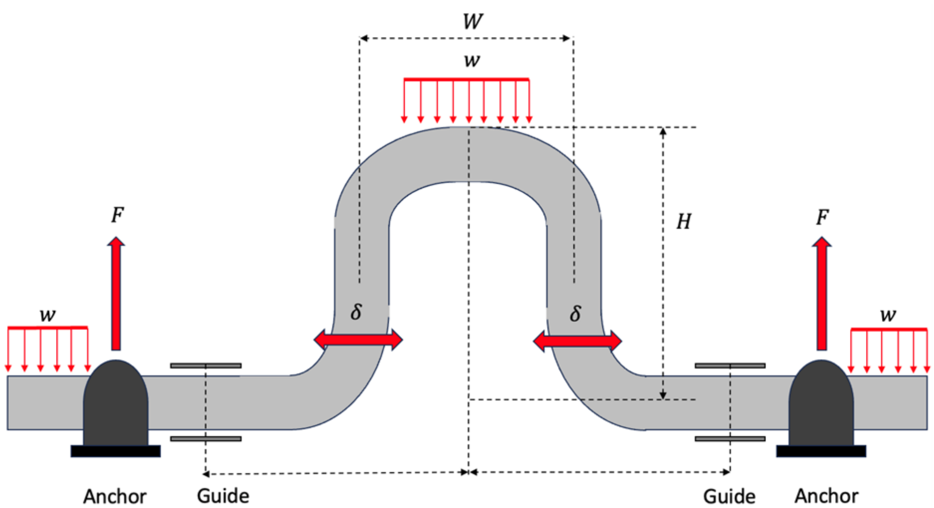
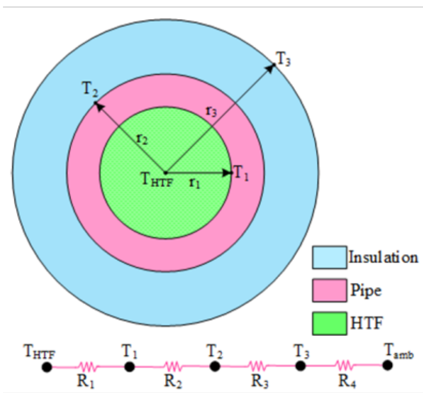


Figure 6.Visual representation of an expansion loop to deal with the extended length required based on ASME standards shown on Equation (16).

Another important feature of determining the treatment to extended pipe lengths is the analysis of pressure loss along the transport distance. Other than the friction pressure loss, the contribution from bends or so-called form loss is intensified when the number of expansion loops installed is increased. Later in Section 4, the effects of pressure loss to equivalent operational cost will be characterized

2.4 Insulation Thickness Determination

After determining the possible set of NPS and their lengths as required by ASME B31.1, the next step is to determine the insulation thickness necessary to achieve the desired heat delivery. Once the insulation thickness is determined, the total heat loss over the heat-delivery distance (*L*), HTF temperature change, pipe-surface temperature, and other design values can be calculated. The insulation thickness can be determined based on one of two methods: (1) the first method involves pre-setting a target heat loss rate, while (2) the second method involves setting the maximum allowable heat loss based on the temperature difference between the heat source and the temperature demand. The first method is to preset the maximum allowable amount of heat loss compared to the amount of required heat duty (Q) and determine the thickness of the insulation material accordingly. The second method is to calculate the maximum allowable heat loss based on the heat-source temperature and the required temperature at delivery and determine the insulation thickness accordingly. Whichever method is used, the thermal resistance (*R*) should first be calculated based on the material properties of the components, (i.e., HTF, pipe, and insulation), as shown in Figure 7. The case study in this section used the second method to calculate the insulation thickness required to meet the desired heat-delivery purpose.



r_1: Inner radius of the piping [m]
r_2: Outer radius of the piping [m]
r_3: Outer radius of the insulation [m]
r_4: Outer radius of the surrounding layer, if any [m]

T_{HTF}: Heat transfer fluid temperature [*K*]
T_{amb}: Ambient temperature [*K*]
T: Temperature [*K*]

Thermal resistance (R)

$R_{total} = R_1 + R_2 + R_3 + R_4$
(where $R_1 = (h_1 A_1)^{-1}$; $R_2 = \frac{\ln(r_2/r_1)}{2\pi L k_p}$; $R_3 = \frac{\ln(r_3/r_2)}{2\pi L k_{ins}}$; $R_4 = \frac{\ln(r_4/r_3)}{2\pi L k_4}$ or $(h_4 A_4)^{-1}$)

Figure 7. Cross-sectional view of insulated piping and thermal resistance (adapted from [14]).

Figure 7 shows the cross-sectional view of the insulated piping structure along with the mathematical equation to calculate the thermal resistance. Given the total heat loss rate Q_{loss} [W], the total heat loss over an infinitesimal transport distance (*dx*) can be expressed as follows:

$$dQ_{loss}(x) = -\dot{m} C_{p,f} dT_x = \frac{\Delta \mathrm{T}}{\mathrm{R}_{total}} = \frac{T_f(\mathrm{x}) - T_{amb}}{\mathrm{R}_{total}} \tag{18}$$

where

$\dot{m}$ = mass flow rate of HTF [kg/s]

$C_{p,f}$ = specific heat of HTF [J/kg-K]

$T_f(x)$ = HTF temperature at the axial location x [*K*]

T_{amb} = ambient temperature [*K*]

R_{total} = total thermal resistance [K/W].

Each component of thermal resistance shown in Figure 7 (R_1, R_2, R_3, and R_4) can also be expressed over an infinitesimal transport distance (dx), leading to the following equation for the total thermal resistance over an infinitesimal transport distance (dx):

$$R_{\text{total}} = \frac{1}{h_1(2\pi r_1)dx} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{\left(2\pi k_{pw}\right)dx} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{(2\pi k_{ins})dx} + \frac{1}{h_4(2\pi r_3)dx} \tag{19}$$

Combining Equations (18) and (19):

$$-\dot{m}C_p\frac{\mathrm{dT(x)}}{dx} = (T_f(\mathrm{x}) - T_{env})/(\frac{1}{h_1(2\pi r_1)} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{\left(2\pi k_{pw}\right)} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{(2\pi k_{ins})} + \frac{1}{h_4(2\pi r_3)}) \tag{20}$$

Defining R_a=$(\frac{1}{h_1(2\pi r_1)} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{\left(2\pi k_{pw}\right)} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{(2\pi k_{ins})} + \frac{1}{h_4(2\pi r_3)})$, Equation (18) can be rewritten as:

$$-\dot{m}C_pR_a\frac{\mathrm{dT}_f(\mathrm{x})}{dx} = (T_f(\mathrm{x}) - T_{env}) \tag{21}$$

Equation (21) has the form of first-order differential equation and has the following analytic solution:

$$T_f(\mathrm{x}) = (T_{src} - T_{amb})\exp\left(-\frac{x}{R_a\dot{m}C_p}\right) + T_{amb} \tag{22}$$

where

x = axial location along the pipe length

T_{src} = heat-source side temperature at $x = 0$. If the heat-source side temperature (T_{src}) and the temperature demand at $x = L$ (T_{demand}) are known, the maximum allowable heat loss over the distance L can be determined using the following equation:

$$Q_{loss,allowable} = -\int_{T_{src}}^{T_{demand}} \dot{m}C_{p,f}\,dT(x) \tag{23}$$

Calculating the average thermal properties of HTF throughout the heat delivery and assuming constant mass flow rate, the above equation can be combined with Equation (18) to write:

$$Q_{loss,allowable} = -\dot{m}C_{p,f}\left[T_f(L) - T_{src}\right] = \frac{T_f(\mathrm{L}) - T_{env}}{\mathrm{R}_{total}} \tag{24}$$

where

$T_f(L)$ = HTF temperature at delivery (i.e., $x = L$).

Once a preset heat loss target is given as a percentage of the desired heat duty (first method) or the maximum allowable heat loss is calculated (second method), Equation (24) can be used to calculate the insulation thickness required.

2.5 Design Optimization

Upon identifying the feasible combinations of standard pipe sizes, required pipe-length extensions, and insulation thicknesses that meet design constraints, ASME requirements, and the given service requirements, the optimal piping design should now be selected from among the possible combinations. An optimal piping system, as previously defined, is one that achieves the desired heat-delivery purpose at the lowest possible cost. Thus, the objective function for optimization is set to minimize the total annual cost (C_{total}) associated with constructing and operating the piping system. The annual total cost comprises the amortized capital costs of the pipe and insulation, along with the operating costs, which can be expressed as the following equation:

$$C_{total}(D_p,\theta) = C_p(D_p,\theta) + C_{HTF}(D_p,\theta) + C_{ins}(D_p,\theta) + C_{\Delta P}(D_p,\theta) + C_{HL}(D_p,\theta) \tag{25}$$

where

C_{total} = annual total cost [$/yr]

C_p = amortized pipe cost [$/yr]

C_{HTF} = amortized HTF cost [$/yr]

C_{ins} = amortized insulation cost [$/yr]

$C_{\Delta P}$ = operating cost due to pumping requirement [$/yr]

C_{HL} = operating cost due to heat loss [$/yr]

D_p = pipe diameter [m]

θ = pipe wall thickness [m].

2.5.1 Amortized Pipe Cost (C_p)

The capital cost of the pipe can be annualized by applying the amortization factor (*AF*), which accounts for the time value of money (i.e., interest). In principle, the *AF* is determined by the capital recovery factor (*CRF*), along with the costs associated with depreciation, property taxes, and income taxes, but the latter three factors are ignored in the current study [15,16]. With this simplification, the amortized pipe cost (C_p) can be calculated as:

$$C_p(D_p,\theta) = Z_p(D_p,\theta) \cdot AF(i,n) \tag{26}$$

where

Z_p = capital cost of the pipe [$]

AF = amortization factor

I = interest rate

n = economic operating period (or lifetime) of a piping system [*yr*].

The *AF* can be calculated using the following equation:

$$AF(i,n) = CRF(i,n) + \zeta \tag{27}$$

where *CRF* denotes the capital recovery factor $\left(= \frac{i(1+i)^n}{(1+i)^n-1}\right)$ at interest rate i over the economic life n of a piping system, and ζ is a coefficient representing a portion of the fixed operation and maintenance cost [15].

2.5.2 Amortized Insulation Cost (C_{ins})

The amortized insulation cost can be calculated in the same way as for pipes:

$$C_{ins} = Z_{ins}\left(D_p, \theta\right) \cdot AF(i, n) \tag{28}$$

where

Z_{ins} = capital cost of the insulation [$].

2.5.3 Amortized HTF Cost (C_{HTF})

The amortized HTF cost can be calculated in the same way as for pipes:

$$C_{HTF} = Z_{HTF}\left(D_p, \theta\right) \cdot AF(i, n) \tag{29}$$

where

Z_{HTF} = capital cost of the HTF [$].

2.5.4 Operating Cost Due to Pumping Requirement ($C_{\Delta P}$) [$/*yr*]

Operating cost is determined by pumping cost to overcome the pressure loss during the flow of HTF as well as the electricity cost:

$$C_{\Delta P} = C_{el} \cdot \dot{W}_P \tag{30}$$

where

$\dot{W}_P$ = pumping power (=$\frac{\dot{m}\Delta P}{\rho\eta_p}$) [W]

η_p = pump efficiency, C_{el} is the specific cost for electricity [$/J].

The pressure loss, required in calculating pumping power and resultant operating cost, consists of two components: (i) frictional loss and (ii) minor (or form) loss. The total pressure loss over the heat-delivery distance L can be calculated using the following equation:

$$\Delta P = \frac{\rho_f \Delta v^2}{2} + \left(f\frac{L}{D_i} + K_{loss}\right)\frac{\rho_f u_f^2}{2} \tag{31}$$

where

Δv^2 = squared velocity difference induced by heat loss

F = friction factor

K_{loss} = minor loss coefficient (= 0.3 for 90-degree flanged elbow)

D_i = pipe inner diameter [m]

L = heat-delivery distance [m].

Once the required number of anchors (N_{anc}) are determined, the number of expansion loops (N_{exp}) can be calculated as $N_{exp} = N_{anc} -$ two-dimensional expansion loop, as illustrated in Figure 6, each loop contains four 90-degree bends. As a result, the minor loss due to these bends over the heat-delivery distance L can be calculated as follows:

$$K_{loss} = 4 \cdot \sum_{i=1}^{N_{exp}} K_{loss,i} \tag{32}$$

where

$K_{loss,I}$ = minor loss coefficient at i_{th} expansion loop.

3. APPLICATION TO REFERENCE CASES

Sections 2.1 and 2.2 confirmed configurations of specific heat removal and supply from a LWR for high temperature industrial requirements. In the proposed thermal power delivery loop, the operating pressure is chosen to achieve efficient heat transfer through phase changes at heat removal and supply ends, delivering the desired heat duty at a specific mass flow rate with minimal pumping power. Choosing a suitable operating pressure will allow steam to be generated on the hot line and condensed liquid water in the cold line of the thermal power delivery loop. The operating pressure constraint for steam based HTF is that the saturation temperature be within the heat pump's hot and cold temperature range. For saturation temperatures below the heat exchanger's cold operating temperatures, the HTF phase is liquid. Alternatively, if saturation temperature rises above the heat pump hot operating temperatures, the HTF phase is gas.

While working within the constraint for operational pressure, the constraint alone is insufficient for selecting a single-pressure value. There remains a variety of applicable pressure ranges that can accommodate the phase at the given heat exchanger operational temperatures. Considering the required heat duty and the threshold for heat loss in the thermal delivery loop, multiple scenarios at different operating pressures were tested to find cost-based optimal configurations. The initial operational temperature and pressure conditions for the LWR thermal delivery piping system are summarized in Table 1 and Table 2.

Table 1. Test matrix of thermal power delivery piping system analysis for sensible heating with demand (load) temperature in the range of 350-550°C. HTF candidates are air, N_2, He, Ar, CO_2, and steam. Two vapor compression cycle steam cases are considered. For one case (Steam 1), the steam is condensed at the thermal load and returned to the LWR as cooled liquid water, and for the other case (Steam 2), the water is returned as saturated liquid at the LWR steam generator feedwater inlet temperature.

	HTF	Fluid T [°C]	Fluid P [kPa]	Fluid Quality	Return T [°C]	Return P [kPa]	Mass Flow Rate [kg/s]			
							50 MWth	100 MWth	200 MWth	500 MWth
Case 1	Air	255.6	250	N/A	182.2	250	229.7	42.4	1378.1	2296.8
Case 2	N_2				182.6		225.1	450.3	1350.8	2251.3
Case 3	He				186.5		48.0	96.1	288.3	480.5
Case 4	Ar				186.4		478.7	957.4	2872.2	4787.2
Case 5	CO_2				175.1		216.5	433.0	1298.7	2164.5
Case 6	Steam 1	272.6	5728.9	1	337.6	14500	27.8	55.6	166.8	278.1
Case 7	Steam 2				272.6	5729.0	21.2	459.4	127.2	212.0

Table 2. Test matrix of thermal power delivery piping system analysis for uniform temperature heating at 550 °C. HTF candidates are the same as in Table 1.

	HTF	Fluid T [°C]	Fluid P [kPa]	Fluid Quality	Return T [°C]	Return P [kPa]	Mass Flow Rate [kg/s]			
							50 MWth	100 MWth	200 MWth	500 MWth
Case 8	Air	255.6	250	N/A	128.7	250	77.5	155.1	465.2	775.4
Case 9	N_2				129.0		75.7	151.4	454.1	756.9
Case 10	He				133.9		16.0	32.0	95.9	159.8
Case 11	Ar				133.7		159.2	318.4	955.1	1591.8
Case 12	CO_2				116.3		74.9	149.9	449.7	749.4
Case 13	Steam 1	272.6	5728.9	1	375.6	22500	27.0	54.1	162.2	270.4
Case 14	Steam 2				272.6	5729.0	11.8	23.6	70.8	118.0

The selection of insulation material is based on material density, operational temperature limits, thermal conductive properties, and cost per kilogram. Candidates for the insulation were mineral wool, aerogel, and polyimide rigid cellular materials. Properties of those materials are listed in Table 3. Mineral wool exhibited the best operating and cost performance for the designated allowable heat loss levels for each scenario in Table 1 and Table 2. Insulation imposes a cost on the entire system through two different mechanisms: the weight contribution to extending nominal pipe length and the required amount of insulation to prevent the heat loss threshold. The former non-linearly and the latter linearly affects the total cost, such that iterations are required in the optimization process. Aerogel performed slightly worse than mineral wool. For the given conditions, mineral wool was the optimal insulation material, but for different cases with higher temperatures and heat duty, the required insulation thickness of mineral wool may be larger than a meter. For these types of cases, aerogel may provide better insulation dimensions as the thermal conductivity at higher temperatures is significantly less than with mineral wool.

For piping material, SA213 stainless steel 316L was chosen. SS 316L is highly corrosion resistant and is used for high-temperature applications, such as high-pressure hydraulic servicing and heat exchangers. Although cast ductile iron and nickel alloy C-276 were candidates for the given operational conditions, further analysis of optimal piping is planned for future activities and will add to the dimensionality of the thermal power delivery piping system design.

Table 3. Key insulation characteristics of selected candidates

	Mineral Wool [17,18,19]	Aerogel [19,20,21]	Polyimide Rigid Cellular [19,21,22]
Insulation Density (kg/m^3)	120–180	30–240	48–128
Insulation Operational Temperature Limits (℃)	<1200	<649	<315
Insulation Thermal Conductivity Coefficient (W/m·K)	0.023–0.1	0.023–0.048	0.023–0.048
Cost per kg ($/kg)	12.5*	266.7*	164.9*

* Unknown uncertainty due to different information provided by vendors. Listed values are representative. Actual values are vendor-specific [14].

Given the thermal power delivery requirements outlined in Table 1 and Table 2, the range of possible default pipe diameters ranged between 0.2 m and 13 m. These pipe dimensions are those that comply with the maximum velocity limit (currently set as 50 m/s for gas and 5 m/s for liquid) of the HTF and the ASME pipe thickness requirements at the operating pressure of the thermal delivery loop. Different pipe diameters and thicknesses require different pipe length extensions as per ASME B31.1 (discussed in Section 2.3), affecting the total capital costs of the pipes. In addition, different pipe diameters will affect the pumping power required to achieve a specific mass flow rate of an HTF. Small-diameter pipes entail large pressure drops, requiring high pumping power and pumping costs. This is the opposite of the capital costs of the pipes. That is, as pipe size increases, pipe capital costs increase, whereas pumping costs decrease.

The calculated results of each cost component (i.e., capital, operating, and heat loss costs as described in Section 2.5) for each possible pipe dimension were combined to compare the annualized total cost, and the optimal case with the lowest total cost was selected among the possible pipe dimensions. The optimal pipe sizes selected on the delivery and return paths are provided in Appendix A. Table 4 and Table 5 are a short list of results initial designs for uniform temperature heating heat delivery at duties 50 MWth and 500 MWth.

Table 4. Initial heat delivery designs and operating conditions duty demand of 50 MWth for uniform temperature heating heat delivery line.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	460	500	530	398	500	1750	2100
Insulation Thickness (m)	0.98	1.09	1.00	1.00	1.01	1.01	1.01
Pipe Inner Diameter (m)	3.10	3.11	2.66	3.70	2.69	0.50	0.35
Pipe Outer Diameter (m)	3.10	3.12	2.67	3.71	2.70	0.53	0.37
Pipe Thickness (m)	0.0038	0.0038	0.0032	0.0045	0.0033	0.015	0.011
Source Velocity (m/s)	6.26	6.27	12.62	6.53	5.26	4.63	4.13
Pressure Loss (bar)	0.0046	0.0050	0.0038	0.0053	0.0061	0.49	0.63
Annual Amortized CAPEX Per Distance	$164.58	$171.07	$128.72	$241.24	$135.03	$106.63	$54.01
Annual Amortized OPEX Per Distance	$31.84	$31.50	$33.08	$61.03	$24.44	$21.09	$10.10
Annual Amortized Penalty Cost Per Distance	$25.73	$23.69	$22.46	$29.10	$22.46	$8.60	$7.27
Annual Corrected Total Cost Per Distance	$222.15	$226.26	$184.27	$331.37	$181.94	$136.32	$71.38

Table 5. Initial heat delivery designs and operating conditions duty demand of 500 MWth for uniform temperature heating heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	1850	1700	2100	1600	2000	8700	11500
Insulation Thickness (m)	1.10	1.04	1.03	1.10	1.03	1.00	1.00
Pipe Inner Diameter (m)	8.78	8.31	7.59	10.43	7.32	1.40	0.96
Pipe Outer Diameter (m)	8.80	8.33	7.61	10.46	7.34	1.48	1.01
Pipe Thickness (m)	0.011	0.010	0.0092	0.013	0.0089	0.042	0.029
Source Velocity (m/s)	7.78	8.77	15.51	8.20	7.11	6.04	5.60
Pressure Loss (bar)	0.014	0.016	0.0096	0.016	0.021	1.89	2.78
Annual Amortized CAPEX Per Distance	$1,263.95	$1,158.76	$988.21	$1,849.69	$944.81	$791.45	$374.58
Annual Amortized OPEX Per Distance	$208.09	$268.00	$197.12	$415.41	$191.79	$150.23	$75.41
Annual Amortized Penalty Cost Per Distance	$56.32	$56.31	$52.57	$65.91	$50.90	$15.78	$12.35
Annual Corrected Total Cost Per Distance	$1,528.36	$1,483.07	$1,237.90	$2,331.01	$1,187.51	$957.45	$462.34

The constraints added to the optimization were to be within 3.4 °C worth of temperature loss and the maximum insulation thickness to be around 1 m. The choice in terms of heat loss being characterized by temperature loss instead of percentages of duty demand was based on the range of duty magnitudes. For example, 2% heat loss of 50 MWth is 1 MWth and 2% loss of 500 MWth is 10 MWth. Although working within 2% heat loss for both cases would seem reasonable, the reflected temperature loss would differ significantly. The goal is to deliver heat as close as possible to the demand temperature minimizing heat loss. Thus, setting a temperature loss limit allows the design optimization to be functioning within a working envelop at all specified duties. Combined with the insulation thickness limit, the maximum travel distance can be determined. The insulation thickness limit is based on availability of insulation specifications. If the heat delivery target temperature drop was within 1 °C, then the simplest solution would be to increase insulation thickness appropriately. However, this can have vast repercussions in terms of commercial product availability as thicker insulation over 1 m or more is potentially limited and custom manufacturing may be required.

The results shown in Table 4 and Table 5 are the reflection of the constraints applied. For 50 MWth, the maximum travel distance for all HTF candidates represent the limitations when abiding to the design constraints. A minimum distance between the nuclear thermal island and industrial processes is assumed to be 500 m, air and Ar are removed as HTF candidates as minimum transport distance is not satisfied. Observing the pipe sizing of the remaining HTFs, other than steam, the optimized outer diameters range is between 2 m and 3 m resulting in pressure losses smaller than 5 kPa. When attempting to apply pipe dimensions to available NPS options, steam cases are the sole choice due to pipe outer diameters being below 0.55 m. For 500 MWth, the suggested pipe designs significantly differ from 50 MWth cases. Regardless of equivalent temperature and pressure conditions, the required mass flow rate has increased by an order of magnitude forcing optimal delivery pipe sizing to 7 m – 10 m for Air, N_2, He, Ar, and CO_2. Although it is possible to reduce pipe sizes to 2 m – 3 m and still be within the gas velocity limit of 50 m/s, the pressure loss would be increased by a factor of 100 inducing significant operational cost that would be less economical than larger piping. Steam cases, as shown in 50 MWth, are the more economical choice with pipe sizing around 1 m.

Other than the superior thermal properties of saturated steam compared to other HTF candidates, the major factor affecting the optimal pipe size is the applied pressure. For other HTFs, the operating pressure is 250 kPa which is significantly lower than steam cases' operating pressure of 5728 kPa. At the significant higher pressures, HTF density is vastly increased reducing the required HTF volume to satisfy the specified mass flow rate. Less required HTF volume leads to smaller optimized piping. The reason for such high pressures for the steam cases relates to the corresponding saturation pressure for the given target delivery temperature. In terms of variation in pressure, it has shown to be relatively cost insensitive at pressure ranges between 200 kPa – 6000 kPa as shown in Figure 8. This is due to the balance between capital and operational cost. For the given delivery temperature, increase in optimized design cost can be observed below 100 kPa and above 6000 kPa. One disadvantage for highly pressurized heat delivery is the increase in pipe thickness. From Table 5, HTFs operating at 250 kPa have a minimum pipe thickness range between 9.237 mm – 12.687 mm and HTFs operating at 5728 kPa have a minimum pipe thickness range between 8.506 mm – 41.571 mm. Although the difference between thickness ranges may not seem significant, this does present challenges when selecting NPS piping. Above NPS 24, available schedule options are limited and do not exceed 17.475 mm. If custom pipe manufacturing is unacceptable, this can present design issues as the selectable pipe size will be constricted to below NPS 24. Such design decisions lead to highly operational cost driven total cost and increase pump requirements to overcome enhanced pressure losses.

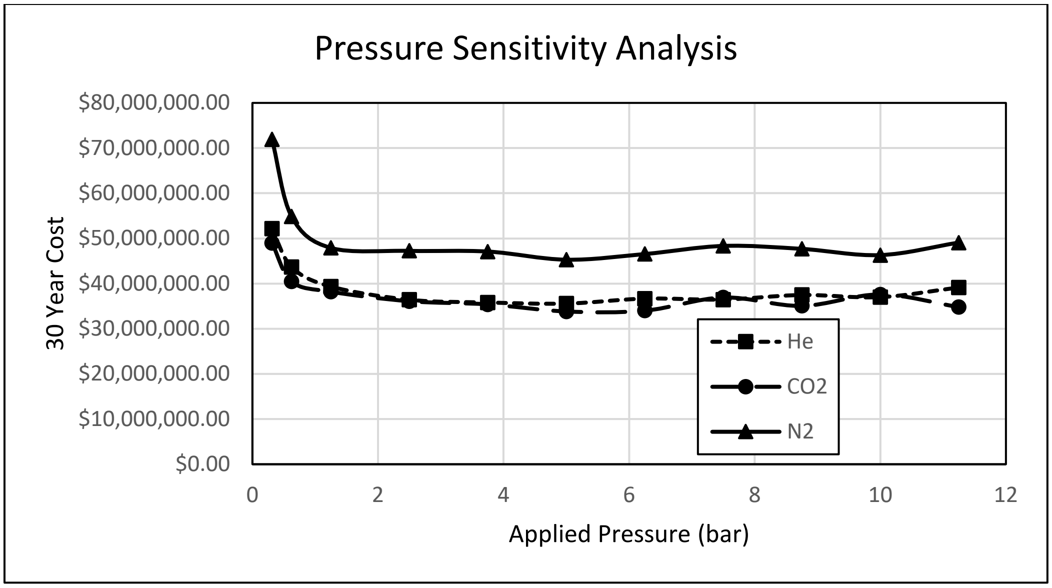


Figure 8. Cost pressure sensitivity for He, CO_2, and N_2 at 30 years' worth of operation.

When downsizing pipe dimensions, extra caution to heat loss is necessary. Recalling Table 4 and Table 5, the maximum travel distance in the 500 MWth case is more than triple of the 50 MWth case. This is the result of thermal resistance change as the HTF volume (pipe inner diameter) and pipe thickness increases. With smaller piping and the limitation of 1 m worth of insulation thickness, the thermal resistance is insulation reliant and cannot suppress heat loss along the axial length of the piping effectively. This issue is the main driver in deciding the optimal number of parallel piping while attempting to satisfy the required mass flow rate.

Applying parallel piping is another way of reducing pipe size. At higher duties as shown in Table 1 and Table 2, the required mass flow rate can increase as high as 4787.155 kg/s. Theoretically, as shown in Table 5, having outer pipe diameters above 7 m will yield the optimized design. At this point, custom manufacturing limitations restrict the possibility of excessively large piping. As the pipe outer is increased, by Equation (1), the minimum pipe thickness will increase linearly assuming the internal pressure, piping material, and operational temperature remains the same. For a minimum transport distance of 500 m defined by NPP probabilistic risk assessments, providing pipe supports and welding equivalent to pressure vessels is extremely labor intensive and may increase the overall total cost of the heat delivery line outside the cost estimate uncertainty of the given analysis. By providing the option to design parallel heat delivery configurations, reduction of each pipe mass flow rate requirements is relaxed.

Since the mass flow rate is a function of HTF density, pipe inner area, and HTF velocity, the pipe diameter is not simply reduced by dividing by the number of parallel piping. Given the heat loss at the operational pressure conditions, the average density from the NPP to the industrial process changes. The pipe inner area and HTF velocity are inversely proportional which will be dependent on the selected pipe dimensions. The pipe dimension is further dependent on the pipe internal pressure and selected pipe outer diameter. With all the components considered and using the estimated cost to find the optimized pipe dimension and design configuration, the pipe dimension reduction per number of parallel piping is parabolic where at the local minima, the outmost optimized design exists. For example, Figure 9 shows a special case application where the heat delivery temperature is 611 °C, operating pressure is 74 bar, and the transport distance is 700 m. Before the number of parallel piping of 10, the heat transport total cost is operating cost dominant. Comparing the cost difference between the highest and lowest cost estimate, the cost can be reduced by 80% by increasing the number of parallel piping. The high operating cost is attributed to the exceedingly high pipe internal HTF velocity. Although the set limit is 50 m/s, the effect of increased velocity is squared when determining the pressure loss. Higher levels of pressure loss will proportionally increase the pump operating cost.

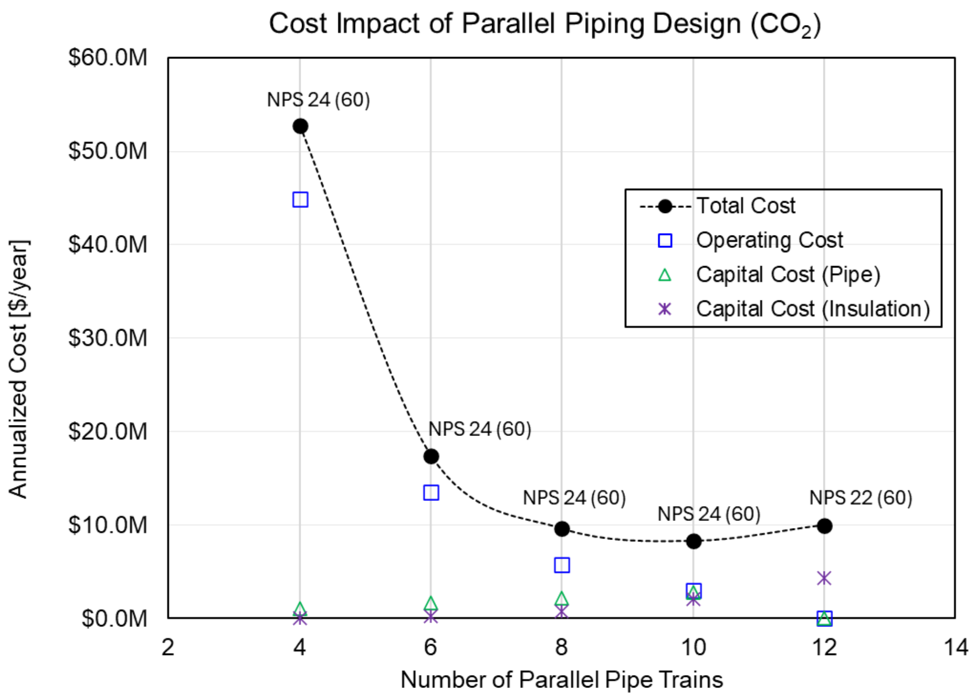


Figure 9. Example application case for heat delivery temperature of 611 °C using CO2 in parallel piping configurations.

With all under consideration, it is the suggestion of the ASME B31.1 informed piping analysis for both sensible heating and uniform temperature heating heat delivery cases to transport the required duty in a highly pressurized environment with parallel piping when necessary. When applied to the same uniform temperature heating 500 MWth case shown in Table 5, pipe dimensions were significantly reduced and the equivalent estimated cost was matched as shown in Table 6.

Appendix A summarizes the design characteristics, operating constraints, and cost details of the optimal piping system selected for a thermal delivery loop for the cases presented in Table 1 and Table 2. It includes key information such as pipe size and thickness, as well as the number of anchors to meet ASME loading requirements, total loop length required to meet ASME flexibility requirements, insulation thickness, and the estimated cost details. Overall, the HTF option with delivery saturated steam and returning saturated liquid (Steam 2) shows the best cost performance for all heating modes. However, it is important to note that these are preliminary evaluation results assuming a very simplified thermal delivery loop design and will require refinement such as adding compression cost and pipe size dependent labor cost.

Table 6. Custom heat delivery designs and operating conditions duty demand of 500 MWth for uniform temperature heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	550	600	630	500	600	1500	2000
Insulation Thickness (m)	1.05	1.02	1.05	1.02	0.94	1.02	1.01
Pipe Inner Diameter (m)	0.87	0.78	0.77	1.02	0.72	0.72	0.66
Pipe Outer Diameter (m)	0.92	0.82	0.81	1.07	0.75	0.76	0.70
Pipe Thickness (m)	0.022	0.019	0.019	0.026	0.018	0.021	0.020
Source Velocity (m/s)	8.04	8.52	15.38	8.68	7.23	7.58	5.93
Pressure Loss (bar)	0.33	0.42	0.22	0.43	0.52	0.82	0.73
Annual Amortized CAPEX Per Distance	$1,309.07	$1,280.92	$1,059.17	$1,848.34	$949.94	$640.35	$358.39
Annual Amortized OPEX Per Distance	$226.66	$225.60	$203.13	$475.36	$204.19	$134.30	$58.57
Annual Amortized Penalty Cost Per Distance	$52.67	$60.23	$49.09	$59.03	$50.19	$31.05	$19.86
Annual Corrected Total Cost Per Distance	$1,588.39	$1,566.74	$1,311.39	$2,382.72	$1,204.31	$805.70	$436.82
Number of Parallel Piping	5	6	5	5	5	3	2

4. TECHNOECONOMIC ANALYSIS COMPARISON TO INDUSTRY TRANSPORT CASES

For NPP heat delivery, the minimum transport distance is dictated by the minimum buffer distance between NPP and industrial process (500 m as stated in Section 3). The maximum travel distance depends on the heat delivery requirements (duty, source/sink demand temperature and pressure), constraints (operation period, maximum allowable heat loss), HTF type, and materials used for piping and structures. For the given application cases, the range of maximum delivery distance dictated by requirements and constraints are from 500 m to 5400 m. For heat delivery cases observed world-wide, heat delivery distances as long as 140 km for a total of 2.2 GW worth of district heating exists at Sizewell, London [23]. Other locations such as Czech, Sweden, Russia, and Denmark have similar transmission pipe systems for district heating. Table 7 shows the duty and transport distance for district heating. As the required duty increases, the transport length increases as well due to possible enlargements in pipe size to accommodate the increased mass flow rate and minimize friction loss. This is a trend partially exhibited in the optimized results for application references cases in Section 3 when increasing the required duty from 50 MW to 200 MW. For district heating, delivery temperature ranges from 50 °C to 110 °C and return temperatures are around 50 °C [23]. The relatively lower temperature compared to the higher application reference case temperatures allow the heat loss to be at a minimum, an indication of smaller temperature difference between ambient conditions.

Table 7. World-wide cases of heat transmission pipelines [24].

Location	Country	Duty (MW)	Transport Length (km)	HTF	Ref.
Lindesberg	Sweden	25	18	Water	[25]
Helsinki	Finland	490	20	Water	[23]
Tilburg	Netherlands	170	25	Steam/Water	[23]
Viborg	Denmark	58	12	Water	[26]
Oradea	Romania	546	86.3	Water	[27]
Akranes	Iceland	60	62	Water	[28]
Aachen	Germany	85	20	Water	[29]
Sankt Pölten	Austria	50	31	Steam/Water	[30]
Kozani	Greece	137	16.5	Water	[31]

To express the cost uniformly, effective cost per duty per applied transport distance or levelized cost of heat (LCOH) is used as expressed in Equation (33) and Equation (34).

$$C_{eff} = \frac{Amortized\ Total\ Cost\ (\$)}{Delivered\ Heat\ Duty\ (kWh) \cdot Transport\ Distance} \tag{33}$$

$$C_{LCOH} = \frac{Amortized\ Total\ Cost\ (\$)}{Delivered\ Heat\ Duty\ (MWh)} \tag{34}$$

For Kavvadias et al., the LCOH was used extensively to test the sensitivity against insulation thickness, transport distance, delivery temperature, return temperature, and alternative heat supply technologies. The temperature dependence for delivery and return temperature are shown in Figure 10 and Figure 11. Red dots mark a reference case in [24].

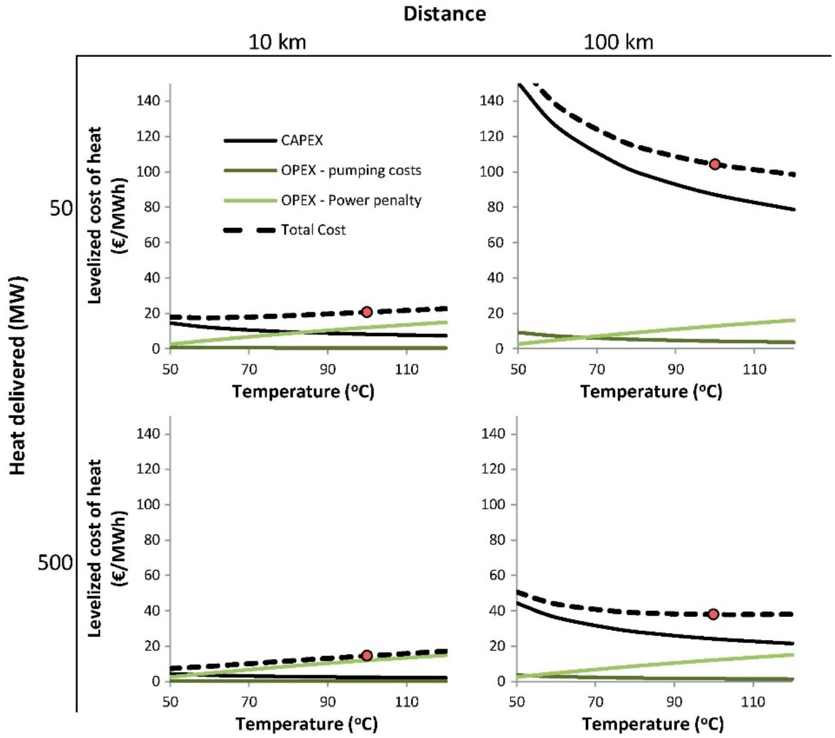


Figure 10. Sensitivity of LCOH for different supply temperatures, distances and amount of heat [24].

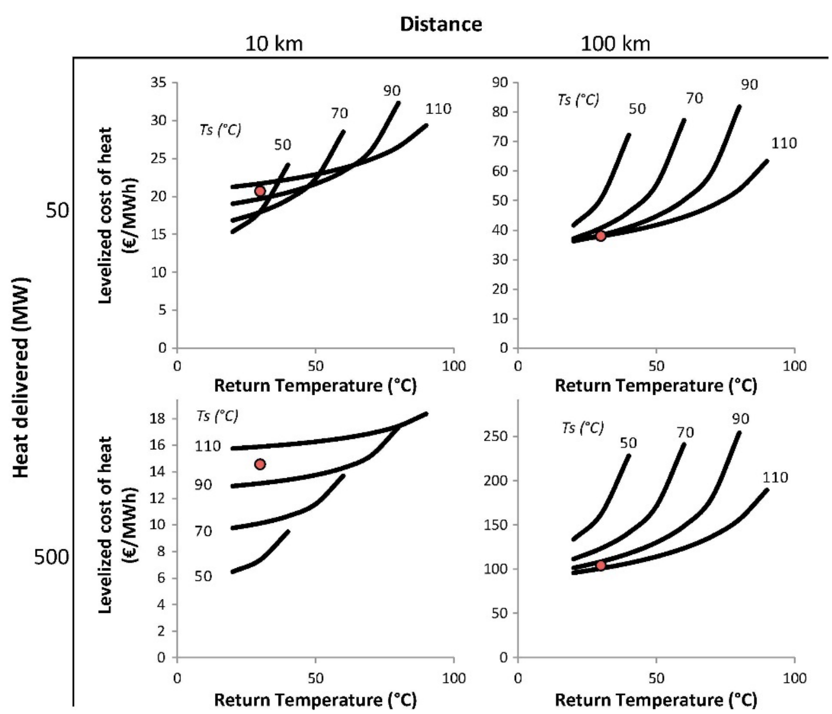


Figure 11. Direct return temperature on costs of delivered heat for different amount of heat delivered, transmission distances and supply temperatures [24].

For the short distance supply temperature sensitivity, the reduction of capital and pumping costs, and addition of penalty costs were observed. For capital cost, this is contrary to the trends discovered by the workflow defined in Figure 4. Although as stated by Kavvadias et al. the dominance of capital cost does indeed increase at longer distances as the required amount of piping escalates, elevation of supply temperature increases capital cost according to the findings under this report. It has been identified that the main cause of this trend is due to the inevitable increase of pipe size as the HTF density is reduced at higher temperatures inducing enhanced flow rates. There are two paths to increase flow rate; one is to increase velocity and the other is to increase the pipe inner cross section. Optimization results have shown when the flexibility of pipe size selection is available, the most economic configuration is when capital cost is slightly dominant than operational cost. The deviation between both analyses is possibly rooted in the theory applied to the developed tools. For Kavvadias et al., the object function of the code is the maximum distance achievable by constraining the problem to find the point of zero net present value. This includes considering the problem specification variables, design variables, physical variables, pumping needs based on friction loss, heat loss, and the economic model as a cost estimator. For the code written following workflow in Figure 4, this includes ASME B31.1, ASME standards for power piping. From Section 2.2 Section 2.3, code standards such as minimum pipe thickness, maximum anchor distance, and mechanical/thermal stress induced pipe extension are defined and is essential to ensure the prevention of power piping failure. Following the standards in Section 2.2 Section 2.3 will increase the overall axial pipe length as pipe thermal expansion strain is distributed on heat expansion loop bends (if there is an expansion joint available for the pipe size and thermal stress, overall axial pipe length is equal to the line distance between the heat source and heat sink). The pipe extension will cause higher capital cost more than the influence to pumping cost regardless of increased friction and form loss.

The pumping cost trend matches the conclusions of this report. As the optimized pipe selection is enlarged due to lower HTF density, the pipe inner cross section area is larger than colder supply temperatures and for a given mass flow rate requirement, the velocity is minimized. Since the friction loss and form loss is proportional to velocity squared, the reduction of velocity is squared and reduces pressure loss by the same magnitude, thus providing lower pumping costs. The power penalty as well is in good agreement with the current report analysis. The higher the supply temperature, the larger the gap between the supply temperature and ambient temperature (if underground, the ambient temperature is replaced with the soil temperature) leading to increased heat loss and higher penalties. For return temperature sensitivities, the fundamental trend of minimized LCOH at lower temperatures is common to the report conclusions. Analogous to the power penalty trend, lower temperatures will minimize heat loss and optimized pipe size selection.

In Table 8 and Table 9, the LCOH of the delivery and return application reference cases from 50 MWth to 500 MWth are provided. Judging from the magnitude difference between values reported from Kavvadias et al. (Figure 8 and Figure 9, output optimized pipe selection from the report are significantly higher. There are three major possibilities that contribute to the given differences. First, although the application reference case duty spans from 50 MWth to 500 MWth, the denominator value used for Equation (34) does not apply the duty span directly. The application reference cases are meant for heat delivery from the NPP to the heat pump near the target industry process. The duty values reported in Table 2 are the combined heat and power (CHP) demand. Added with the heat delivery and compressor power, the sum spans from 50 MWth to 500 MWth. Therefore, the denominator value used for Equation (34) is the duty minus the corresponding compressor power.

$$C_{eff} = \frac{Amortized\ Total\ Cost\ (\$)}{Delivered\ Heat\ Duty\ (MWh) - Compressor\ Power\ (MWh)} \tag{35}$$

Table 8. Deliver LCOH for all application reference cases based on maximum feasible transport distance and without distance normalization, the magnitudes are not comparable. Not normalized per unit length to facilitate comparison with [24].

HTF	Uniform Temperature LCOH [$/MWh]				Sensible Heating LCOH [$/MWh]			
	50 MWth	100 MWth	200 MWth	500 MWth	50 MWth	100 MWth	200 MWth	500 MWth
Air	23,044	29,764	25,637	15,880	65,514	47,659	39,313	22,289
N_2	22,470	33,613	26,521	16,901	30,049	31,621	31,975	19,387
He	66,792	26,940	20,794	14,786	53,732	27,036	26,308	16,027
Ar	101,025	43,750	36,693	21,938	48,578	46,640	52,270	32,921
CO_2	60,528	27,725	21,875	12,592	53,110	41,594	33,577	18,544
Steam 1	23,368	34,713	37,720	12,160	37,523	68,462	22,905	6,684
Steam 2	14,365	21,364	41,184	8,752	32,376	47,533	90,620	9,469

Table 9. Return LCOH for all application reference cases. Recall, each case has a designated maximum transport distance and without distance normalization, the magnitudes are not comparable.

HTF	Uniform Temperature LCOH [$/MWh]				Sensible Heating LCOH [$/MWh]			
	50 MWth	100 MWth	200 MWth	500 MWth	50 MWth	100 MWth	200 MWth	500 MWth
Air	15,372	18,995	16,585	9,314	51,249	36,496	29,651	16,163
N_2	15,751	19,784	16,300	10,273	24,846	22,444	24,572	14,518
He	43,009	17,250	14,199	9,100	20,831	19,705	21,409	13,143
Ar	26,265	27,101	22,908	13,784	38,556	33,393	39,605	24,586
CO_2	34,591	15,878	11,895	7,566	39,489	31,831	25,465	12,782
Steam 1	23,136	33,946	42,058	14,729	46,313	62,815	34,025	10,549
Steam 2	3,277	4,311	6,460	1,621	6,565	8,879	15,150	1,772

With the reduced heat delivery, the LCOH is effectively increased. Corresponding compressor power for uniform temperature and sensible heating can be found in Table 10. The second reason, is the same for different trends observed for the capital LCOH supply temperature sensitivity, following ASME B31.1 standards adds extra constraints that increase the required amount of piping, HTF, insulation, number of anchors, and number of expansion loops. As a result, the LCOH shown in Table 8 and Table 9. For higher transport temperatures, pipe enlargement triggered mechanical stress and operating temperature induced thermal stress will significantly increase LCOH. Considering the operational temperature difference between district heating and industry process heat delivery, it is possible to yield the observed magnitude deviation.

Table 10. Corresponding compressor power for the given duty levels.

HTF	Compressor Power (Uniform Temperature/Sensible Heating) [MW]			
	50 MWth	100 MWth	200 MWth	500 MWth
Air	10.08/17.35	20.17/34.69	40.33/69.38	100.83/173.46
N_2	10.09/17.34	20.18/34.68	40.36/69.35	100.90/173.38
He	10.10/17.24	20.20/34.48	40.40/68.97	101.00/172.41
Ar	10.12/17.27	20.23/34.53	40.47/69.06	101.16/172.66
CO_2	10.32/17.58	20.64/35.17	41.28/70.34	103.19/175.84
Steam 1	18.23/33.69	36.45/67.39	72.90/134.77	182.25/336.93
Steam 2	18.23/33.69	36.45/67.39	72.90/134.77	182.25/336.93

The third is the missing exercise of delivery distance normalization. For both Kavvadias et al. and this report, the optimization process has been modified to output options for maximized transport distance. Without this consideration, fair comparisons of LCOH are difficult to discern. After normalizing by the maximum extended pipe length, the application reference case uniform temperature delivery and return the distance normalized LCOH results are shown in Table 11. Dividing the distance by either 10 km or 100 km and converting euros to United States (U.S.) dollars for the supply and return LCOH in Figure 8 and Figure 9, the range of LCOH values span from $0.66 \cdot 10^{-3}$ \$/kWh/km to $2.73 \cdot 10^{-3}$ \$/kWh/km. The magnitude gap remains and indicates district heating cases under 110 °C will vastly more economic than the application reference cases.

Comparing the uniform temperature heating application reference cases from Table 8 and Table 11, shifts in cost ranking are observed. For regular LCOH values in Table 8, the overall cost performance ranking in best order is two-way saturated gas/liquid (Steam 2), He, CO_2, one-way saturated gas (Steam 1), N_2, air, and Ar. For normalized LCOH values in Table 11, the overall cost performance ranking in best order is Steam 2, Steam 1, CO_2, He, N_2, air, and Ar. Due to the superior performance of saturated steam cases, the maximum achievable distance while limiting insulation thickness within 1 m and restraining heat loss to 3 °C is the longest. Also, the applied normalization has provided a common evaluation metric that has stabilized the ranking order. Before the normalization, depending on the required duty, the rank order between CO_2, He, and Steam 1 were various. However, from Table 11, the ranking order remains consistent for all required duties. The data provided in Appendix A follows the observed fundamental ranking as well. The evaluation metric in Appendix A is amortized cost per maximum transport distance and thus yields the same ranking.

Table 11. Application reference case uniform temperature heating maximum distance normalized total LCOH.

HTF	Normalized Cost (Delivery / Return) [$/kWh/km]			
	50 MWth	100 MWth	200 MWth	500 MWth
Air	26.12/20.25	21.86/16.57	25.27/19.53	15.75/11.15
N_2	25.50/20.75	23.27/16.44	26.59/19.56	15.53/11.28
He	22.46/17.13	18.09/13.71	20.62/16.61	12.98/9.51
Ar	41.96/33.66	34.10/25.33	39.46/29.52	23.55/17.72
CO_2	21.05/14.69	17.26/12.05	20.30/13.54	11.67/8.53
Steam 1	7.48/6.95	6.57/6.04	7.61/7.64	4.42/4.70
Steam 2	3.92/0.95	3.34/0.71	4.13/0.68	2.40/0.44

Other than district heating, different forms of transport piping cases exist. For energy transport piping, the application includes natural gas (NG), oil, and H_2. For these cases, the objective is to deliver energy in terms of mass for long-distances in place of tankers, cargo vessels, railcars, ships, etc. [32]. The mass transport piping system is designed to deliver energy resources to power utilities and industry infrastructures. From DeSantis et al., the amortized cost in $/MWh per 1,000 mi were provided for high-voltage direct current (HVDC), crude oil, methanol (MeOH), ethanol (EtOH), NG, and H_2. The cost range in this report is a combination of operation cost and capital cost, similar to Kavvadias et al.. Capital cost is broken down to material cost, labor cost, right of way cost, miscellaneous cost, substation cost, pump station cost, and compressor station cost. The operation cost consists of pump and compressor electrical costs. The cost summary is provided in Table 12.

Conducting the same cost comparison with the report values in Table 11 and multiplying the values by the km-to-mile conversion constant of 1.609, the magnitude of $/MWh/1000 mi are in the same range. For Steam 2 application reference cases, the cost is within the range of liquid and gas pipeline minimum and maximum values. For Ar application reference cases except for 50 MWth, the cost values are below HVDC. This implication is significant as the current analysis is the alternative to sole electric power commodity transmission from NPP. From Table 11 and Appendix A, Ar will constantly underperform other HTF for all application reference cases and is due to the relatively low volumetric heat capacity at delivery and return operational temperatures. If Ar application reference cases can outperform electrical transmission, the option of heat commodity rather than sole electric transmission is beneficial from the long-distance energy transmission standpoint. In current U.S. Department of Energy national programs (Light Water Reactor Sustainability Program, Integrated Energy Systems Program, etc.), CHP options and variable loads of thermal energy storage are being investigated, designed, and analyzed for NPP baseline operation flexibility and decarbonization of industrial processes. Depending on NPP type, sitting location, target industrial process, and market feedback, the ratio of optimized heat and power supply will dictate the design requirements and constraints for the heat delivery system and hence will affect the final optimized design. Acknowledging during what heat delivery conditions will outperform power transmission will become a key decision factor on whether long-distance heat delivery for industrial processes is economically advantageous or not.

The same can be said for other forms of energy transport. For example, H_2 transportation is a developing technology dealing with hydrogen leakage and combustion at compressed conditions [33,34].

If H_2 delivery underperforms heat transportation for high temperature steam electrolysis, it would be recommended to choose heat transportation instead.

Table 12. Summary data for comparing energy transmission costs in \$/mile, \$/MWh, and \$/mile/MW [32].

Parameter	Electrical	Liquid Pipeline			Gas Pipeline	
Energy carrier	HVDC	Crude Oil	MeOH	EtOH	NG	H_2
Total flow (kg/s)	6,000*	1,969	1,863	1,859	368.9	69.54
Delivered power (MW_{LHV})	2,656**	91,941	37,435	50,116	17,391	8,360
Power loss in transmission	12.9%	0.78%	2.02%	1.51%	2.67%	1.94%
Capital Cost (\$/mile/MW)	\$1,502	\$16	\$51	\$38	\$97	\$166
Amortized cost (\$/MWh/1000 miles)	\$41.5	\$0.77	\$2.2	\$1.7	\$3.7	\$5.0

* Units are Amps. **Units are Mwe.

5. CONCLUSION

To provide a structured method to select the best heat delivery piping system for heat pumps or directly to industry processes at least 500 m away from NPPs, the analysis focused on optimal designs of the piping, insulation, and heat expansion systems. In identifying the optimal designs for the heat-delivery scenarios towards the heat pump reference cases, pipe sizing, insulation thickness, design requirements such as maximum allowable stress and required thickness of pipes under delivery pressures, necessary pipe loops to accommodate thermal expansion, and operational practices for high-temperature thermal systems such as HTF maximum velocity limits were considered. The set objective was to find an optimal heat-delivery design that minimizes the capital, lifetime operating, and penalty costs. The assessment was initiated by selecting stainless steel or cast ductile iron piping material, HTF candidates as shown in Table 1 and Table 2, and insulation materials from Table 3. Each selected component is chosen to efficiently function under operational conditions defined by the application reference cases in Table 1 and Table 2 for sensible and uniform temperature heating heat pumps. Following the workflow in Figure 4 and assuming uniform pipe diameter, level piping (no change in elevation), friction and form loss induced pressure only, and ignoring capital cost contributions from flanges, fittings, valves, and pumps, the following are the findings of the heat delivery design:

- Optimal heat delivery designs were found using stainless steel 316L piping and mineral wool insulation. Aerogel was also nearly as good as mineral wool. For future references, if the weight of insulation starts to affect maximum anchor distance, it is suggested to use aerogel as the thermal conductivity is significantly smaller than mineral wool and less insulation material will be required to acceptable heat loss levels.
- Insulation imposes a cost on the entire system through two different mechanisms: the weight contribution to extending nominal pipe length and the required amount of insulation to prevent the heat loss threshold. The former non-linearly and the latter linearly affects the total cost, such that iterations are required in the optimization process.
- For the given application cases, the range of maximum delivery distance dictated by requirements and constraints are from 500 m to 5400 m.
- As applied mass flow rate is increased due to higher heat supply demand, the capital cost drives the cost-based optimization via larger piping sizes.

- When NPS size piping is required, the tendency for operation cost driven optimization increases. Due to NPS size and schedule limitations, NPS 24 and NPS 48 are typical selected pipe sizes. NPS 24 is the last size with various pipe thickness and NPS 48 is the largest piping available.
- To avoid large pipe sizes, distributing the required mass flow rate to more than 1 parallel heat delivery pipe is recommended. Generally, the total cost is reduced as the number of parallel piping increases. At a certain parallel piping number threshold, the total will increase again. The threshold point is application case dependent.
- Another mechanism to avoid large pipe sizes is to compress the HTF to higher levels. The HTF density will increase requiring less volumetric flow rate and smaller pipe sizes.
- At mildly compressed piping systems (10 bar – 60 bar), heat delivery cost is insensitive to pressure. Below or above the mild compression state will increase the heat delivery cost.
- The application reference case HTF ranking in best performance order is Steam 2, Steam 1, CO2, He, N2, air, and Ar.
- Out of the application reference case options provided, the best cost performance is given for the uniform temperature heating for 500 MWth.

To compare calculated cost values with other studies developing delivery pipe optimization tools and investigating various application cases, results from district heating and energy transmission were selected. For the district heating comparison, the district heating cost performance were magnitudes better than the given heat pump heat delivery application reference case cost optimized designs. This attributed to lower supply temperature, lower user demand temperature, and no considerations for ASME B31.1 standards. If ASME code was applied, pipe extension lengths may have increased pipe and insulation capital cost higher to levels observed in this report. For energy transmission comparison, the cost performance of the heat pump heat delivery application reference cases was within the minimum and maximum $/MWth/1000-mi energy transmission values. Interestingly, most HTF candidates outperformed power transmission cost and suggest high benefits in terms of long-distance heat delivery compared to long-distance power transmission.

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Appendix A

Optimized Custom Heat Delivery Design and Effective Cost

Delivery Uniform Temperature Heating

Table 13. Custom heat delivery designs and operating conditions duty demand of 50 MWth for uniform temperature heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	500	500	1650	1300	1600	1750	2100
Insulation Thickness (m)	1.05	1.10	1.03	1.02	1.03	1.01	1.01
Pipe Inner Diameter (m)	0.55	0.52	0.75	0.97	0.69	0.50	0.35
Pipe Outer Diameter (m)	0.58	0.55	0.78	1.02	0.72	0.53	0.37
Pipe Thickness (m)	0.014	0.013	0.017	0.024	0.017	0.015	0.011
Source Velocity (m/s)	5.07	5.68	9.06	4.78	3.94	4.63	4.13
Total Heat Loss (kW)	152.70	145.64	305.40	291.28	283.60	283.96	281.76
Anchor Spacing (m)	22.73	21.75	28.05	32.57	26.28	21.70	16.68
Pipe Unit Length (m)	40.11	38.32	50.49	60.26	47.18	38.66	29.12
Number of Expansion Loops Required	22	23	59	40	61	81	126
Corrected Pipe Length (m)	882.41	881.28	2973.96	2407.59	2875.45	3124.16	3667.83
Pressure Loss (bar)	0.17	0.22	0.19	0.36	0.44	0.49	0.63
Required Pump Head (m)	54.37	71.21	487.09	80.33	86.30	168.82	217.38
Required Pump Power (kW)	20.68	26.44	76.35	125.44	63.44	44.78	25.16

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Pump Volume Flow Rate (m^3/s)	2.39	2.42	3.93	3.53	1.45	0.92	0.40
Annual Amortized CAPEX Per Distance	$212.45	$197.48	$179.33	$336.36	$175.41	$106.63	$54.01
Annual Amortized OPEX Per Distance	$34.48	$44.15	$37.78	$76.68	$32.47	$21.09	$10.10
Annual Amortized Penalty Cost Per Distance	$16.37	$15.64	$9.72	$11.45	$9.33	$8.60	$7.27
Annual Corrected Total Cost Per Distance	$263.31	$257.27	$226.83	$424.49	$217.21	$136.32	$71.38
Number of Parallel Piping	2	2	1	1	1	1	1

Table 14. Custom heat delivery designs and operating conditions duty demand of 100 MWth for uniform temperature heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	760	800	830	700	830	2900	3600
Insulation Thickness (m)	0.92	1.07	0.98	0.97	1.05	1.01	1.01
Pipe Inner Diameter (m)	0.70	0.74	0.70	0.87	0.62	0.68	0.47
Pipe Outer Diameter (m)	0.73	0.78	0.73	0.91	0.65	0.72	0.50
Pipe Thickness (m)	0.018	0.019	0.014	0.022	0.015	0.020	0.014
Source Velocity (m/s)	6.24	5.68	11.48	6.00	4.89	5.12	4.56
Total Heat Loss (kW)	288.86	290.63	303.96	298.87	295.18	564.46	566.27
Anchor Spacing (m)	27.22	27.64	26.91	30.60	24.39	26.37	20.85
Pipe Unit Length (m)	48.69	49.87	48.18	55.95	43.48	48.07	36.99
Number of Expansion Loops Required	28	29	31	23	37	110	173

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Corrected Pipe Length (m)	1361.34	1444.62	1489.46	1283.04	1606.03	5286.43	6392.16
Pressure Loss (bar)	0.32	0.26	0.15	0.33	0.41	0.77	1.01
Required Pump Head (m)	100.45	85.38	411.45	73.41	81.10	266.18	346.95
Required Pump Power (kW)	76.41	63.39	64.50	114.63	59.62	141.21	80.31
Pump Volume Flow Rate (m^3/s)	4.78	4.84	8.83	7.06	2.91	1.84	0.80
Annual Amortized CAPEX Per Distance	$338.21	$385.93	$282.31	$536.38	$284.24	$189.94	$94.95
Annual Amortized OPEX Per Distance	$82.60	$64.58	$63.73	$131.48	$54.64	$39.31	$18.49
Annual Amortized Penalty Cost Per Distance	$20.07	$19.03	$19.31	$22.04	$17.39	$10.10	$8.38
Annual Corrected Total Cost Per Distance	$440.89	$469.54	$365.35	$689.90	$356.27	$239.35	$121.83
Number of Parallel Piping	2	2	2	2	2	1	1

Table 15. Custom heat delivery designs and operating conditions duty demand of 200 MWth for uniform temperature heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	560	550	560	500	600	2700	5400
Insulation Thickness (m)	0.99	0.99	0.97	0.98	1.02	0.96	1.02
Pipe Inner Diameter (m)	0.75	0.79	0.75	0.93	0.68	0.74	0.77
Pipe Outer Diameter (m)	0.79	0.83	0.78	0.98	0.71	0.78	0.81
Pipe Thickness (m)	0.019	0.020	0.015	0.023	0.017	0.022	0.023
Source Velocity (m/s)	8.05	7.46	15.19	7.76	6.02	6.51	5.24

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Total Heat Loss (kW)	430.57	433.40	427.93	449.88	425.01	1137.76	1134.83
Anchor Spacing (m)	28.26	29.05	28.05	31.93	26.15	27.87	28.40
Pipe Unit Length (m)	50.99	52.60	50.47	58.80	46.90	51.12	52.35
Number of Expansion Loops Required	20	19	20	16	23	97	191
Corrected Pipe Length (m)	1014.55	997.58	1008.27	929.94	1077.40	4954.77	9974.12
Pressure Loss (bar)	0.37	0.29	0.16	0.37	0.38	1.07	1.36
Required Pump Head (m)	116.17	94.51	452.22	83.32	74.76	369.42	470.18
Required Pump Power (kW)	132.54	105.26	106.33	195.17	82.44	293.98	326.52
Pump Volume Flow Rate (m^3/s)	14.34	14.51	26.50	21.17	8.72	5.51	2.40
Annual Amortized CAPEX Per Distance	$786.70	$876.62	$637.82	$1,241.99	$688.07	$445.95	$242.07
Annual Amortized OPEX Per Distance	$192.26	$155.28	$155.20	$308.86	$112.61	$87.32	$48.18
Annual Amortized Penalty Cost Per Distance	$40.15	$41.10	$40.15	$45.77	$37.32	$21.72	$10.76
Annual Corrected Total Cost Per Distance	$1,019.10	$1,073.00	$833.17	$1,596.62	$838.00	$554.99	$301.01
Number of Parallel Piping	4	4	4	4	4	2	1

Table 16. Custom heat delivery designs and operating conditions duty demand of 500 MWth for uniform temperature heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	550	600	630	500	600	1500	2000
Insulation Thickness (m)	1.05	1.02	1.05	1.02	0.94	1.02	1.01
Pipe Inner Diameter (m)	0.87	0.78	0.77	1.02	0.72	0.719	0.66
Pipe Outer Diameter (m)	0.92	0.82	0.81	1.07	0.75	0.76	0.70
Pipe Thickness (m)	0.022	0.020	0.019	0.026	0.018	0.021	0.020
Source Velocity (m/s)	8.04	8.52	15.38	8.68	7.23	7.58	5.93
Total Heat Loss (kW)	561.14	692.91	590.90	581.12	572.30	902.74	766.37
Anchor Spacing (m)	30.66	28.71	28.77	33.50	27.29	27.30	25.88
Pipe Unit Length (m)	56.10	51.97	51.89	62.26	49.05	50.04	47.05
Number of Expansion Loops Required	18	21	22	15	22	55	78
Corrected Pipe Length (m)	1008.03	1088.44	1138.70	931.40	1078.87	2750.60	3651.59
Pressure Loss (bar)	0.33	0.42	0.22	0.43	0.52	0.82	0.73
Required Pump Head (m)	102.05	134.83	501.32	96.33	101.81	283.89	251.13
Required Pump Power (kW)	155.25	166.85	157.17	300.84	149.69	251.01	145.33
Pump Volume Flow Rate (m^3/s)	23.90	24.15	35.42	35.29	14.53	9.20	4.01
Annual Amortized CAPEX Per Distance	$1,309.07	$1,280.92	$1,059.17	$1,848.34	$949.94	$640.35	$358.39

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized OPEX Per Distance	$226.66	$225.60	$203.13	$475.36	$204.19	$134.30	$58.57
Annual Amortized Penalty Cost Per Distance	$52.67	$60.23	$49.09	$59.03	$50.19	$31.05	$19.86
Annual Corrected Total Cost Per Distance	$1,588.39	$1,566.74	$1,311.39	$2,382.72	$1,204.31	$805.70	$436.82
Number of Parallel Piping	5	6	5	5	5	3	2

Return Uniform Temperature Heating

Table 17. Custom heat delivery designs and operating conditions duty demand of 50 MWth for uniform temperature heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	500	500	500	500	500	1750	2100
Insulation Thickness (m)	1.02	1.03	1.02	1.03	1.05	1.14	0.32
Pipe Inner Diameter (m)	0.79	0.78	0.73	1.00	0.63	0.19	0.14
Pipe Outer Diameter (m)	0.82	0.81	0.76	1.04	0.66	0.25	0.14
Pipe Thickness (m)	0.016	0.016	0.013	0.020	0.012	0.032	0.0040
Source Velocity (m/s)	3.75	3.83	7.31	3.44	3.17	3.086	1.07
Total Heat Loss (kW)	37.18	37.26	123.35	45.85	89.21	290.30	286.75
Anchor Spacing (m)	31.34	31.31	30.14	35.86	26.70	11.32	8.80
Pipe Unit Length (m)	47.55	47.49	45.80	55.89	39.29	21.51	14.40
Number of Expansion Loops Required	16	16	55	14	60	155	239
Corrected Pipe Length (m)	759.28	758.95	2511.20	780.38	2355.45	3329.90	3439.30
Pressure Loss (bar)	0.084	0.084	0.15	0.082	0.39	5.07	2.51
Required Pump Head (m)	19.90	20.80	284.66	14.17	53.00	164.41	33.31
Required Pump Power (kW)	15.14	15.44	44.62	22.13	38.96	43.61	3.86
Pump Volume Flow Rate (m^3/s)	1.81	1.84	3.04	2.70	0.99	0.086	0.015
Annual Amortized CAPEX Per Distance	$170.15	$174.82	$142.18	$293.20	$123.61	$99.10	$7.83

Annual Amortized OPEX Per Distance	\$29.34	\$29.94	\$26.15	\$41.73	\$24.34	\$19.27	\$1.65
Annual Amortized Penalty Cost Per Distance	\$4.63	\$4.64	\$4.65	\$5.56	\$3.58	\$8.25	\$7.89
Annual Corrected Total Cost Per Distance	\$204.13	\$209.41	\$172.98	\$340.49	\$151.53	\$126.63	\$17.37
Number of Parallel Piping	1	1	1	1	1	1	1

Table 18. Custom heat delivery designs and operating conditions duty demand of 100 MWth for uniform temperature heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	760	800	830	700	900	2900	3600
Insulation Thickness (m)	1.03	0.99	0.99	0.65	1.01	1.03	0.35
Pipe Inner Diameter (m)	0.71	0.68	0.68	0.83	0.57	0.26	0.19
Pipe Outer Diameter (m)	0.73	0.71	0.71	0.86	0.59	0.34	0.20
Pipe Thickness (m)	0.014	0.014	0.011	0.017	0.011	0.044	0.006
Source Velocity (m/s)	4.62	5.10	9.32	4.98	3.90	3.41	1.15
Total Heat Loss (kW)	105.34	110.89	121.10	149.57	96.64	579.91	562.84
Anchor Spacing (m)	29.24	28.66	28.64	33.39	25.06	14.03	10.59
Pipe Unit Length (m)	44.09	43.08	43.39	51.01	36.65	27.19	17.77
Number of Expansion Loops Required	26	28	29	21	36	207	340
Corrected Pipe Length (m)	1146.09	1203.69	1257.85	1069.90	1317.24	5623.73	6042.75
Pressure Loss (bar)	0.21	0.26	0.11	0.26	0.36	7.85	3.79

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Required Pump Head (m)	49.35	64.71	244.71	45.29	48.19	254.39	50.33
Required Pump Power (kW)	37.54	48.05	38.36	70.73	35.43	134.96	11.65
Pump Volume Flow Rate (m^3/s)	3.62	3.67	6.83	5.39	1.98	0.17	0.031
Annual Amortized CAPEX Per Distance	$277.32	$264.22	$223.02	$401.98	$202.24	$174.95	$14.35
Annual Amortized OPEX Per Distance	$48.20	$58.75	$44.88	$97.28	$39.58	$35.32	$2.84
Annual Amortized Penalty Cost Per Distance	$8.70	$8.72	$9.11	$13.23	$6.94	$9.76	$8.81
Annual Corrected Total Cost Per Distance	$334.22	$331.69	$277.01	$512.49	$248.76	$220.02	$26.00
Number of Parallel Piping	2	2	2	2	2	1	1

Table 19. Custom heat delivery designs and operating conditions duty demand of 200 MWth for uniform temperature heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	560	550	560	500	600	2700	5400
Insulation Thickness (m)	1.02	1.02	1.00	0.93	0.93	1.05	0.36
Pipe Inner Diameter (m)	0.78	0.76	0.78	0.89	0.60	0.41	0.30
Pipe Outer Diameter (m)	0.81	0.79	0.80	0.93	0.62	0.55	0.32
Pipe Thickness (m)	0.016	0.015	0.013	0.018	0.012	0.071	0.0089
Source Velocity (m/s)	5.73	6.14	10.86	6.45	5.35	3.92	1.32

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Total Heat Loss (kW)	165.60	160.01	175.42	181.16	137.76	704.14	1137.51
Anchor Spacing (m)	31.16	30.62	31.15	33.88	26.23	18.52	13.97
Pipe Unit Length (m)	47.23	46.36	47.54	52.29	38.35	37.75	24.47
Number of Expansion Loops Required	18	18	18	15	23	146	387
Corrected Pipe Length (m)	849.39	833.32	855.02	776.11	878.78	5506.41	9462.94
Pressure Loss (bar)	0.22	0.24	0.092	0.31	0.43	6.72	5.08
Required Pump Head (m)	51.31	58.98	199.96	52.71	57.25	219.05	67.59
Required Pump Power (kW)	58.54	65.69	47.02	123.45	63.14	348.63	46.94
Pump Volume Flow Rate (m^3/s)	10.85	11.00	20.50	16.17	5.93	0.52	0.092
Annual Amortized CAPEX Per Distance	$667.59	$655.29	$570.54	$938.18	$438.10	$451.55	$36.84
Annual Amortized OPEX Per Distance	$101.42	$116.01	$80.92	$234.09	$105.73	$93.18	$7.30
Annual Amortized Penalty Cost Per Distance	$18.45	$18.17	$19.41	$22.08	$14.83	$12.10	$5.63
Annual Corrected Total Cost Per Distance	$787.45	$789.47	$670.87	$1,194.35	$558.67	$556.82	$49.77
Number of Parallel Piping	4	4	4	4	4	1	1

Table 20. Custom heat delivery designs and operating conditions duty demand of 500 MWth for uniform temperature heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	550	600	630	500	630	1500	2000
Insulation Thickness (m)	0.93	1.01	0.99	0.92	1.01	0.99	1.00
Pipe Inner Diameter (m)	0.81	0.79	0.72	0.98	0.69	0.51	0.38
Pipe Outer Diameter (m)	0.85	0.82	0.74	1.02	0.72	0.68	0.40
Pipe Thickness (m)	0.016	0.016	0.014	0.020	0.014	0.087	0.011
Source Velocity (m/s)	6.97	7.45	13.70	7.11	5.26	4.27	1.38
Total Heat Loss (kW)	220.16	226.13	236.34	242.18	180.52	461.56	291.66
Anchor Spacing (m)	32.42	31.60	30.20	35.87	28.63	20.85	14.42
Pipe Unit Length (m)	49.20	47.97	45.75	55.73	42.30	43.53	26.40
Number of Expansion Loops Required	17	19	21	14	21	72	139
Corrected Pipe Length (m)	835.30	911.04	956.55	778.09	887.05	3132.79	3664.76
Pressure Loss (bar)	0.30	0.37	0.21	0.34	0.37	3.80	1.88
Required Pump Head (m)	70.87	90.50	368.93	58.91	49.38	124.29	25.07
Required Pump Power (kW)	107.81	134.40	115.66	183.98	72.61	329.70	29.01
Pump Volume Flow Rate (m^3/s)	18.08	18.33	27.43	26.96	9.89	0.87	0.16
Annual Amortized CAPEX Per Distance	$909.38	$897.18	$759.46	$1,414.70	$740.45	$688.06	$61.41

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized OPEX Per Distance	$189.94	$217.10	$177.95	$347.98	$120.46	$154.88	$11.65
Annual Amortized Penalty Cost Per Distance	$24.94	$23.48	$23.38	$29.45	$19.25	$13.94	$7.53
Annual Corrected Total Cost Per Distance	$1,124.25	$1,137.77	$960.78	$1,792.12	$880.16	$856.88	$80.59
Number of Parallel Piping	5	5	5	5	5	1	1

Delivery Sensible Heating

Table 21. Custom heat delivery designs and operating conditions duty demand of 50 MWth for sensible heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	1000	500	530	500	1100	5150	5500
Insulation Thickness (m)	0.98	1.00	0.94	0.94	0.96	1.01	1.03
Pipe Inner Diameter (m)	0.85	0.77	0.67	0.81	0.70	0.50	0.45
Pipe Outer Diameter (m)	0.89	0.80	0.70	0.86	0.74	0.53	0.48
Pipe Thickness (m)	0.021	0.013	0.013	0.020	0.018	0.015	0.014
Source Velocity (m/s)	6.28	7.27	12.57	6.82	5.41	4.77	4.48
Total Heat Loss (kW)	417.05	289.12	570.83	312.10	410.67	835.08	833.20
Anchor Spacing (m)	30.38	27.86	26.29	29.57	26.92	21.70	20.08
Pipe Unit Length (m)	55.38	50.54	46.86	53.73	48.35	38.66	35.57
Number of Expansion Loops Required	33	18	40	17	41	238	274
Corrected Pipe Length (m)	1824.83	908.24	1872.73	910.65	1978.65	9187.67	9745.06
Pressure Loss (bar)	0.37	0.19	0.23	0.31	0.55	1.49	1.52
Required Pump Head (m)	116.16	86.13	631.33	70.23	107.69	511.98	523.06
Required Pump Power (kW)	130.86	63.41	99.19	109.93	114.33	139.70	108.80
Pump Volume Flow Rate (m^3/s)	7.08	10.21	13.24	10.61	4.20	0.94	0.72
Annual Amortized CAPEX Per Distance	$495.57	$440.76	$387.89	$710.95	$367.31	$106.63	$87.42

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized OPEX Per Distance	$105.54	$102.74	$77.95	$177.66	$85.04	$22.38	$16.43
Annual Amortized Penalty Cost Per Distance	$21.62	$30.12	$28.84	$32.42	$19.64	$8.60	$8.09
Annual Corrected Total Cost Per Distance	$622.73	$573.63	$494.68	$921.03	$471.98	$137.60	$111.94
Number of Parallel Piping	2	3	3	3	2	1	1

Table 22. Custom heat delivery designs and operating conditions duty demand of 100 MWth for sensible heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	850	550	580	500	1000	8000	9100
Insulation Thickness (m)	0.95	1.10	1.04	1.07	1.03	1.08	1.00
Pipe Inner Diameter (m)	0.91	0.81	0.80	1.02	0.76	0.54	0.61
Pipe Outer Diameter (m)	0.95	0.85	0.83	1.07	0.79	0.57	0.65
Pipe Thickness (m)	0.023	0.020	0.016	0.026	0.019	0.016	0.018
Source Velocity (m/s)	7.37	7.09	13.25	6.53	6.27	8.36	4.96
Total Heat Loss (kW)	566.51	416.18	446.70	450.29	564.58	1631.43	1661.78
Anchor Spacing (m)	31.64	29.08	29.05	33.36	27.89	22.42	24.69
Pipe Unit Length (m)	57.98	52.89	52.66	62.06	50.48	40.25	44.60
Number of Expansion Loops Required	27	19	20	15	36	446	369
Corrected Pipe Length (m)	1561.13	1002.45	1052.32	930.53	1813.31	17948.87	16448.23

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Pressure Loss (bar)	0.41	0.26	0.12	0.24	0.63	8.22	2.40
Required Pump Head (m)	128.57	85.25	341.52	55.04	124.45	2834.20	822.79
Required Pump Power (kW)	193.13	94.14	80.49	129.24	176.18	1546.65	342.30
Pump Volume Flow Rate (m^3/s)	14.16	14.39	26.56	21.23	8.39	1.88	1.43
Annual Amortized CAPEX Per Distance	$842.69	$916.35	$733.20	$1,480.59	$634.24	$121.61	$154.55
Annual Amortized OPEX Per Distance	$182.06	$138.20	$112.56	$204.40	$142.98	$126.81	$30.63
Annual Amortized Penalty Cost Per Distance	$34.33	$39.28	$40.16	$45.78	$29.46	$8.60	$9.56
Annual Corrected Total Cost Per Distance	$1,059.08	$1,093.83	$885.92	$1,730.77	$806.69	$257.02	$194.73
Number of Parallel Piping	3	4	4	4	3	1	1

Table 23. Custom heat delivery designs and operating conditions duty demand of 200 MWth for sensible heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	610	500	500	500	700	3000	13150
Insulation Thickness (m)	0.95	0.92	0.93	1.09	1.00	0.97	0.99
Pipe Inner Diameter (m)	0.95	0.88	0.79	1.09	0.81	0.73	0.99
Pipe Outer Diameter (m)	1	0.93	0.83	1.14	0.85	0.77	1.04
Pipe Thickness (m)	0.024	0.022	0.020	0.027	0.020	0.022	0.029

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Source Velocity (m/s)	9.97	10.10	18.79	9.79	8.20	6.80	5.69
Total Heat Loss (kW)	846.90	773.68	713.75	832.94	840.41	1255.31	3325.85
Anchor Spacing (m)	32.60	31.28	29.64	34.64	29.19	27.72	32.95
Pipe Unit Length (m)	60.03	57.15	53.42	64.88	53.11	50.82	62.20
Number of Expansion Loops Required	19	16	17	15	24	109	400
Corrected Pipe Length (m)	1131.12	913.93	904.36	953.59	1274.08	5517.16	24849.14
Pressure Loss (bar)	0.52	0.44	0.25	0.53	0.71	1.30	3.20
Required Pump Head (m)	162.26	142.04	570.99	119.91	139.24	451.41	1104.23
Required Pump Power (kW)	365.60	268.89	230.68	482.69	295.66	369.51	1378.15
Pump Volume Flow Rate (m^3/s)	42.47	43.16	63.91	63.66	25.18	5.67	4.31
Annual Amortized CAPEX Per Distance	$1,864.96	$1,913.29	$1,556.16	$2,958.01	$1,449.72	$439.41	$397.20
Annual Amortized OPEX Per Distance	$475.67	$432.99	$375.40	$744.94	$341.52	$98.57	$81.62
Annual Amortized Penalty Cost Per Distance	$70.84	$80.09	$74.67	$82.64	$62.41	$21.53	$12.66
Annual Corrected Total Cost Per Distance	$2,411.46	$2,426.37	$2,006.22	$3,785.59	$1,853.64	$559.50	$491.48
Number of Parallel Piping	6	7	7	7	6	2	1

Table 24. Custom heat delivery designs and operating conditions duty demand of 500 MWth for sensible heating delivery heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	550	500	500	500	620	1500	2500
Insulation Thickness (m)	1.07	1.04	0.96	1.04	0.99	1.03	1.01
Pipe Inner Diameter (m)	1.07	0.94	0.87	1.32	0.90	0.72	0.82
Pipe Outer Diameter (m)	1.12	0.99	0.91	1.38	0.94	0.76	0.86
Pipe Thickness (m)	0.027	0.024	0.022	0.033	0.022	0.022	0.024
Source Velocity (m/s)	9.90	11.52	20.04	9.78	8.34	7.77	6.94
Total Heat Loss (kW)	1026.90	986.70	962.83	1109.24	1067.76	897.40	1093.44
Anchor Spacing (m)	34.53	32.17	31.33	38.65	31.05	27.29	29.46
Pipe Unit Length (m)	64.43	59.26	57.01	73.77	57.03	50.04	54.61
Number of Expansion Loops Required	16	16	16	13	20	55	85
Corrected Pipe Length (m)	1028.41	933.47	910.85	956.51	1139.51	2751.30	4637.51
Pressure Loss (bar)	0.42	0.56	0.27	0.45	0.60	0.86	1.05
Required Pump Head (m)	132.64	180.77	598.78	101.83	118.27	298.12	362.34
Required Pump Power (kW)	373.57	443.58	313.59	597.79	313.93	271.14	376.86
Pump Volume Flow Rate (m^3/s)	70.79	71.94	106.50	106.14	41.96	9.46	7.21
Annual Amortized CAPEX Per Distance	$3,130.35	$2,801.48	$2,426.99	$4,912.94	$2,367.49	$642.67	$546.01

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized OPEX Per Distance	$534.59	$699.33	$506.67	$919.76	$405.44	$145.03	$119.59
Annual Amortized Penalty Cost Per Distance	$94.47	$100.00	$100.01	$109.71	$88.65	$30.86	$22.31
Annual Corrected Total Cost Per Distance	$3,759.41	$3,600.82	$3,033.67	$5,942.42	$2,861.58	$818.56	$687.91
Number of Parallel Piping	8	9	9	8	8	3	2

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Table 25. Custom heat delivery designs and operating conditions duty demand of 50 MWth for sensible heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	1000	500	530	500	1100	5150	5500
Insulation Thickness (m)	0.99	1.00	0.95	1.05	1.04	1.30	0.35
Pipe Inner Diameter (m)	0.86	0.61	0.64	0.81	0.68	0.30	0.17
Pipe Outer Diameter (m)	0.90	0.64	0.67	0.84	0.71	0.36	0.19
Pipe Thickness (m)	0.019	0.016	0.011	0.018	0.015	0.028	0.0052
Source Velocity (m/s)	5.22	5.95	11.91	6.05	4.70	0.64	1.17
Total Heat Loss (kW)	261.12	159.51	181.40	189.62	229.37	825.40	832.72
Anchor Spacing (m)	32.31	26.30	26.79	30.42	27.52	14.27	10.21
Pipe Unit Length (m)	53.58	42.49	43.68	50.62	44.60	26.98	17.05
Number of Expansion Loops Required	31	20	20	17	40	361	539
Corrected Pipe Length (m)	1659.30	823.82	867.68	843.47	1783.06	9738.87	9189.07
Pressure Loss (bar)	0.28	0.27	0.12	0.28	0.48	22.33	6.20
Required Pump Head (m)	74.35	64.24	281.46	54.10	77.82	637.34	82.05
Required Pump Power (kW)	83.76	47.29	44.22	84.69	82.62	173.90	17.07
Pump Volume Flow Rate (m^3/s)	6.09	5.18	11.58	9.20	3.45	0.078	0.028

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized CAPEX Per Distance	$446.57	$420.11	$319.13	$620.21	$309.06	$125.93	$12.76
Annual Amortized OPEX Per Distance	$74.29	$84.48	$75.00	$147.76	$68.19	$26.28	$2.73
Annual Amortized Penalty Cost Per Distance	$14.89	$18.32	$19.78	$21.27	$12.17	$8.02	$8.57
Annual Corrected Total Cost Per Distance	$535.74	$522.91	$413.91	$789.24	$389.43	$160.22	$24.07
Number of Parallel Piping	2	9	3	3	2	1	1

Table 26. Custom heat delivery designs and operating conditions duty demand of 100 MWth for sensible heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	850	550	550	500	1000	8000	9100
Insulation Thickness (m)	1.08	1.00	0.95	0.97	1.03	1.02	0.36
Pipe Inner Diameter (m)	0.90	0.73	0.73	0.93	0.76	0.40	0.24
Pipe Outer Diameter (m)	0.94	0.77	0.76	0.97	0.79	0.48	0.25
Pipe Thickness (m)	0.020	0.016	0.013	0.020	0.016	0.037	0.0071
Source Velocity (m/s)	6.44	7.35	13.70	6.80	5.08	0.69	1.24
Total Heat Loss (kW)	326.15	258.20	287.10	287.57	334.82	1654.39	1661.19
Anchor Spacing (m)	32.79	29.20	29.28	33.47	29.47	17.78	12.31
Pipe Unit Length (m)	54.64	47.86	48.15	56.25	48.10	34.21	21.12

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Number of Expansion Loops Required	26	19	20	15	34	450	739
Corrected Pipe Length (m)	1418.08	904.80	957.23	841.63	1633.30	15392.10	15607.47
Pressure Loss (bar)	0.35	0.32	0.15	0.30	0.47	31.30	8.90
Required Pump Head (m)	93.23	90.78	361.24	59.35	75.75	893.46	117.84
Required Pump Power (kW)	140.04	100.25	85.13	139.36	107.23	474.00	49.03
Pump Volume Flow Rate (m^3/s)	12.18	12.40	23.15	18.40	6.90	0.15	0.055
Annual Amortized CAPEX Per Distance	$725.72	$670.10	$550.56	$1,094.06	$569.35	$219.51	$23.64
Annual Amortized OPEX Per Distance	$145.34	$163.05	$130.89	$243.69	$96.62	$45.32	$4.62
Annual Amortized Penalty Cost Per Distance	$21.76	$27.00	$28.38	$32.33	$19.39	$10.17	$10.07
Annual Corrected Total Cost Per Distance	$892.81	$860.15	$709.83	$1,370.08	$685.37	$275.00	$38.34
Number of Parallel Piping	3	3	7	4	3	1	1

Table 27. Custom heat delivery designs and operating conditions duty demand of 200 MWth for sensible heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	610	500	500	500	700	3000	13150
Insulation Thickness (m)	0.98	1.04	1.06	1.00	1.05	1.03	0.36
Pipe Inner Diameter (m)	0.94	0.86	0.81	1.06	0.81	0.68	0.39
Pipe Outer Diameter (m)	0.98	0.90	0.85	1.11	0.84	0.81	0.41
Pipe Thickness (m)	0.021	0.019	0.018	0.023	0.017	0.064	0.012
Source Velocity (m/s)	8.81	9.21	15.42	8.89	6.79	0.74	1.42
Total Heat Loss (kW)	509.65	447.76	435.58	545.04	481.37	868.90	3316.60
Anchor Spacing (m)	34.03	32.04	31.36	36.14	30.48	23.93	16.05
Pipe Unit Length (m)	56.81	53.20	51.97	61.47	49.98	48.77	28.83
Number of Expansion Loops Required	18	16	16	14	23	126	820
Corrected Pipe Length (m)	1019.99	838.56	829.79	854.58	1148.40	6130.26	23628.79
Pressure Loss (bar)	0.44	0.41	0.18	0.47	0.55	9.55	11.81
Required Pump Head (m)	118.82	115.88	354.75	91.71	89.96	273.35	156.84
Required Pump Power (kW)	267.72	219.36	143.32	369.16	191.02	447.51	195.74
Pump Volume Flow Rate (m^3/s)	36.53	37.19	55.73	55.18	20.69	0.47	0.17
Annual Amortized CAPEX Per Distance	$1,583.39	$1,596.69	$1,475.50	$2,504.63	$1,275.22	$627.17	$60.94

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Annual Amortized OPEX Per Distance	$386.27	$384.98	$254.19	$635.74	$244.79	$107.43	$12.19
Annual Amortized Penalty Cost Per Distance	$47.27	$50.52	$49.66	$60.34	$39.66	$13.41	$13.28
Annual Corrected Total Cost Per Distance	$2,016.93	$2,032.18	$1,779.36	$3,200.70	$1,559.66	$748.01	$86.41
Number of Parallel Piping	6	7	7	7	6	1	1

Table 28. Custom heat delivery designs and operating conditions duty demand of 500 MWth for sensible heating return heat delivery.

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Maximum Distance (m)	550	500	500	500	500	1500	2500
Insulation Thickness (m)	1.04	1.01	1.03	1.03	1.00	1.00	1.01
Pipe Inner Diameter (m)	1.00	0.92	0.90	1.23	0.77	0.62	0.49
Pipe Outer Diameter (m)	1.04	0.96	0.94	1.29	0.80	0.74	0.52
Pipe Thickness (m)	0.022	0.020	0.020	0.027	0.017	0.058	0.015
Source Velocity (m/s)	9.78	10.39	16.38	9.66	8.32	0.73	1.50
Total Heat Loss (kW)	620.28	610.24	607.35	682.57	637.82	830.46	419.07
Anchor Spacing (m)	35.06	33.54	33.47	39.22	29.75	22.83	17.01
Pipe Unit Length (m)	58.87	55.96	55.85	67.62	48.54	46.04	31.78

Parameter	Air	N_2	He	Ar	CO_2	Steam 1	Steam 2
Number of Expansion Loops Required	16	15	15	13	21	66	147
Corrected Pipe Length (m)	931.05	836.21	835.67	869.17	1014.67	3031.95	4671.07
Pressure Loss (bar)	0.47	0.49	0.19	0.50	0.76	5.04	2.23
Required Pump Head (m)	128.14	136.95	368.46	98.45	122.97	144.32	29.77
Required Pump Power (kW)	360.91	336.05	192.97	577.92	290.12	196.89	61.93
Pump Volume Flow Rate (m^3/s)	60.88	61.97	92.87	92.00	34.46	0.78	0.28
Annual Amortized CAPEX Per Distance	$2,377.62	$2,349.60	$2,303.10	$3,830.98	$1,734.73	$1,050.72	$99.84
Annual Amortized OPEX Per Distance	$570.48	$591.43	$339.83	$978.54	$420.80	$95.57	$19.51
Annual Amortized Penalty Cost Per Distance	$63.03	$69.04	$68.76	$74.30	$59.47	$25.91	$8.49
Annual Corrected Total Cost Per Distance	$3,011.13	$3,010.08	$2,711.69	$4,883.81	$2,215.00	$1,172.20	$127.84
Number of Parallel Piping	9	9	9	9	9	2	1