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**A Nightly Conditioning Method to Reduce Parasitic Power
Consumption in Molten-Salt Central-Receiver Solar-Power Plants**

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ABSTRACT

A method to reduce nightly parasitic power consumption in a molten salt central receiver is discussed where salt is drained from the piping and heat tracing is turned off to allow the piping to cool to ambient overnight, then in the morning the pipes are filled while they are cold. Since the piping and areas of the receiver in a molten-nitrate salt central-receiver solar power plant must be electrically heated to maintain their temperatures above the nitrate salt freezing point (430°F, 221°C), considerable energy could be used to maintain such temperatures during nightly shut down and bad weather. Experiments and analyses have been conducted to investigate cold filling receiver panels and piping as a way of reducing parasitic electrical power consumption and increasing the availability of the plant. The two major concerns with cold filling are: 1) how far can the molten salt penetrate cold piping before freezing closed and 2) what thermal stresses develop during the associated thermal shock. Experiments and analysis are discussed.

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A molten nitrate salt central receiver power plant uses concentrated sunlight to produce electricity. A field of heliostats tracks the sun and concentrates sunlight onto a receiver. Molten salt is used as a heat transfer fluid. Salt at 550°F (288°C) is pumped from a tank and enters the receiver. It is heated to 1050°F (565°C) by the concentrated sunlight and then it is stored in another tank. The thermal energy in the molten salt is used to generate steam to power a turbine that makes electricity. The advantages of using molten salt are that the energy can be stored easily, the storage is relatively cheap compared to other methods, and the energy can be dispatched when needed (Chavez, et. al.). Figure 1 is a schematic of a molten nitrate salt central receiver plant. A 10 MW_e solar central receiver power plant - called Solar Two, located in the Mojave desert of California, is being retrofitted with a nitrate salt system. It is scheduled to go on line in 1996.

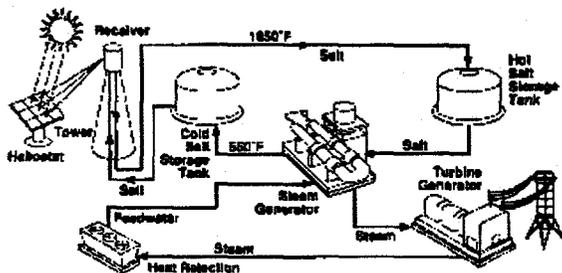


Figure 1. Schematic of molten salt central receiver power plant.

The parasitic electrical power consumption in a molten-salt central-receiver power plant can be a significant fraction of the total power production if it is not properly managed. Good management also involves careful assessment of operating strategies to minimize the parasitics. Since the nitrate salt, which serves as the heat transfer medium between the receiver and the steam generator, has a freezing point of 430°F (221°C) the associated piping, valves, instrumentation, and tanks are kept above this temperature (typically at 550°F, 288°C) to assure that the salt will not freeze. During

inclement weather and during nightly shut down, the plant is not operating, but the pipes are kept hot to maintain the temperature of the salt lines at 550°F (288°C). Three alternative methods to condition the pipes during nightly shut down periods are:

- 1) Keep the large piping - the riser and downcomer - hot (at 550°F) and full of nitrate salt. The pumps are then turned on (bumped) periodically to maintain a hot inventory of salt in the pipes.
- 2) Drain the pipes, turn off the heat trace, and allow the pipes to cool overnight, then a few hours before start up, energize the heat trace to rapidly heat up the pipes.
- 3) Drain the pipes, turn off the heat trace, and allow them to cool overnight, then fill the pipes with the molten salt while the pipes are cold. This method is called cold-filling.

Pump bumping - the first method - has the advantage of using thermal energy to keep the pipes hot, but the salt is distributed over a large surface area. The energy losses grow with time. In addition, drain lines must be kept heat traced to assure that the salt does not freeze past drain valves. Valves have been shown to leak when they are in contact with molten salt (Pacheco, et. al, 1995).

The second option requires that enough heat trace is installed to heat the piping in a reasonable amount of time. It has the advantage that the pipes are heated up prior to their use. A disadvantage of this method is that electricity is used to heat the pipes rather than thermal energy.

The third option allows the maximum energy saving. In the morning or after a long weather or maintenance outage, molten salt is introduced into the pipes while they are below the salt freezing point (temperature at ambient).

Table 1 summarizes the parasitic piping energy losses for the Solar Two receiver during 16 and 30 hour shut downs (Kolb, 1993). The amount of electrical parasitic energy consumed by the pump bumping operation is linearly proportional to the amount of time in

Table 1. Thermal Conditioning Energy for Solar Two Based on a 16 and 30 Hour Shut Down (Kolb, 1993).

Method	Energy Use	
	16 hr	30 hr
Pump Bumping	900 kWh _e	1700 kWh _e
Rapid Heatup to 550°F (288°C)	550	700
Rapid Heatup to 400°F (204°C)	350	500
Cold Fill	180	230

the state (Parasitic Energy = Rate x time). The pump bumping method consumes the parasitic electrical energy at a rate 56 kW_e based on the sum of thermal and electrical parasitics. The rate of consumption is constant because there is a constant heat loss rate. Note that thermal energy used to keep the pipes warm is approximately equal to 33% electrical energy since that is the amount of electrical energy that could have been generated by the Rankine cycle. Thirty-three percent is the approximate Rankine cycle thermodynamic efficiency.

The parasitic energy required to rapidly heat the pipes up is a function of the pipes' temperatures and thus is a function of time. The energy consumed is not a linear function of time but is asymptotic. As the pipe cools, more energy is required to bring its temperature back up to the desired temperature. After approximately 30 hours, the pipes have cooled down to ambient and the electrical energy required to heat them to the desired temperature is a constant.

The energy required to cold fill the pipes is conservatively estimated to be 33% of the energy required to heat the pipes up rapidly to 550°F (288°C). Again the 33% is based on the amount of electrical energy that could have been produced with the thermal energy used to heat up the pipes.

Clearly the cold filling option has the greatest energy savings. But the savings are small relative to the annual energy production for short shut down times. For longer shut down times, the energy savings using cold filling are significantly more than for the pump bumping method. However, there are other advantages that cold filling has over the other methods.

Cold filling has several advantages in the operation of a plant that experiences cyclic operation. After scheduled or unplanned maintenance, it may take several hours to heat up the piping with heat trace - time that could be used to collect and produce energy rather than consume it. If the molten salt can be pumped through part of the system which is below the freezing point, then parasitics could be reduced, and the operation of the plant could be more flexible, increasing the availability.

Hours before morning startup, the heat trace to the receiver headers and jumper tubes (the section of tubing that transitions between the headers and absorber panels) has to be turned on to ensure their temperatures are at the salt temperature. This parasitic power could be reduced if the headers and jumper tube could be filled cold.

Also, the absorber panels do not have heat trace and must be heated with heliostats before they are filled with molten salt. It is difficult

to uniformly preheat the absorber panels with heliostats in the early morning. Some areas will experience much more heating than others due to non-uniform flux profiles from heliostats. This is a particular concern for the east side of an external cylindrical receiver during morning start up. Localized convection will add to the problem. If the receiver can be filled with molten salt when some areas of the receiver are below the salt freezing point, the receiver start up procedure would be much simpler and start up could occur sooner.

There are two major concerns with cold filling components and piping: freezing of the molten salt and transient thermal stresses. This paper describes experiments and analyses performed on cold starting piping and receiver panels with molten salt. The experiments were conducted with a molten salt flow loop and receiver panels where the piping and panels were allowed to cool to ambient before cold filled with molten salt. Thermal and stress analyses were conducted to estimate the stresses that occur during a thermal shock.

THERMAL ANALYSIS OF COLD STARTING PIPING

Assuming that the tube or pipe wall can be approximated as a plane wall, we can use an analytical solution to estimate the transient temperature gradient and heat transfer coefficient. Since the receiver tube and piping have relatively thin walls, the plane wall assumption is a good approximation.

The solution to the energy equation for a plane wall suddenly subjected to a convection boundary condition describes the temperature distribution in the wall as a function of time (Incropera and De Witt, 1985). Its form is:

$$\theta^*(x^*, t^*) = \frac{T(x, t) - T_\infty}{T_i - T_\infty} = \sum_{n=1}^{\infty} C_n \exp(-\lambda_n^2 Fo) \cos(\lambda_n x^*) \quad \text{Eq. 1}$$

where the coefficient C_n :

$$C_n = \frac{4 \sin(\lambda_n)}{2\lambda_n + \sin(2\lambda_n)}, \quad \text{Eq. 2}$$

Fo (the Fourier number) is the nondimensional time and x^* is referenced from the insulated surface:

$$Fo = \frac{\alpha t}{L^2}, \quad x^* = \frac{x}{L}. \quad \text{Eq. 3, 4}$$

The discrete characteristic values (eigenvalues) of λ_n are the positive roots of the transcendental equation:

$$\lambda_n \tan(\lambda_n) = Bi = \frac{hL}{k}. \quad \text{Eq. 5}$$

The length, L , is half the thickness of the plane wall since convection occurs on both faces, but in our case of a pipe wall, it is equal to the wall thickness since one face has convection and the other is insulated. Note the midplane of a plane wall behaves like an insulated surface. The infinite series solution can be approximated by the first term in the series for values of $Fo \geq 0.2$. The solution becomes:

$$\theta^* = C_1 \exp(-\lambda_1^2 Fo) \cos(\lambda_1 x^*) \quad \text{Eq. 6}$$

or

$$\theta^* = \theta_o^* \cos(\lambda_1 x^*) \quad \text{Eq. 7}$$

where θ_o^* is the temperature at the midplane, $x^*=0$, (the insulated boundary, in our case the outside tube wall). The coefficients C_1 and λ_1 are determined from the equations 2 and 5.

Table 2. Penetration distances of molten salt for various pipe diameters, velocities, and salt inlet temperatures for a wall temperature $T_w = 68^\circ\text{F}$ (20°C).

Diameter inches	Flow Velocity m/s	Salt Temp. $^\circ\text{C}$	Penetration Distance, m
0.75	3	288	39
0.75	1	288	17
0.75	1	371	27
1.5	3	288	132
1.5	1	288	58
1.5	1	371	90
6	3	288	1498
6	1	288	657
16	3	288	8340
16	1	288	3660

STRESS ANALYSIS OF THERMAL SHOCK

The stress calculations are important in determining the material behavior in a severe transient condition. For an insulated pipe, we can use the temperature distribution from the thermal analysis to calculate the circumferential, radial, and axial stresses. If the temperature is a function of the radial component only (as in the case of long vertical cylinders or pipes), then each component of stress is (Goodier, 1937, Young, 1989):

$$\sigma_\theta(r) = \frac{E\alpha}{(1-\nu)r^2} \left(\frac{r^2 + r_i^2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} T(r)rdr + \int_{r_i}^r T(r)rdr - T(r)r^2 \right) \quad \text{Eq. 8}$$

$$\sigma_r(r) = \frac{E\alpha}{(1-\nu)r^2} \left(\frac{r^2 - r_i^2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} T(r)rdr - \int_{r_i}^r T(r)rdr \right) \quad \text{Eq. 9}$$

$$\sigma_z(r) = \frac{E\alpha}{(1-\nu)} \left(\frac{2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} T(r)rdr - T(r) \right) \quad \text{Eq. 10}$$

The temperature profile at a given time, Fo , can be found from Equations 1 and 5. The nondimensional length x^* is referenced from the insulated surface (the outside radius) and can be transformed into the nondimensional radial coordinates, $r^* = r/r_o$ and $r_i^* = r_i/r_o$, from:

$$x^* = (1-r^*)/(1-r_i^*) = (1-r^*)/\delta \quad \text{Eq. 11}$$

Carrying out the integration, the three stress components can be expressed in a nondimensional thermal stress format:

$$\sigma^*(r^*) = \frac{\sigma(r)(1-\nu)}{E\alpha(T_i - T_\infty)} \quad \text{Eq. 12}$$

For short time scales ($Fo < 0.2$), several terms in the series in Equation 1 must be used to calculate the temperature distribution. The temperature distribution is then used in Equations 8-10 to calculate the stresses. These equations can be used to calculate the transient stresses as a function of the Biot number and the pipe geometry. Figure 2 shows the nondimensional circumferential thermal stresses as functions of the nondimensional radius for several times (Fo) for a specific geometry. The ratio of r_i to r_o was chosen arbitrarily, but the non-dimensional stresses are a weaker function of that ratio than the Biot number. Note how in Figure 2 a skin stress develops at the inner surface. When the pipe is cold relative to the fluid - "up shock", the stresses at the inner surface are compressive and tensile on the outer surface during the thermal shock. When it is hot relative to the fluid - "down shock", the

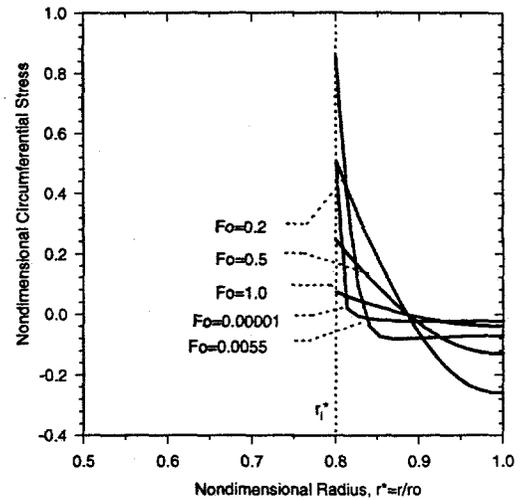


Figure 2. Nondimensional circumferential thermal stresses, σ_θ , in pipe undergoing thermal shock as a function of the non-dimensional radius for several times (Fo) for $r_i/r_o = 0.8$ and $Bi = 100$.

stresses are tensile on the inner surface. The axial nondimensional stresses are not plotted, but are similar to Figure 2. Figure 3 shows the nondimensional thermal (circumferential or axial) stress at the inner surface of the pipe as a function of time (Fo) for several Biot numbers using 30 terms in Equation 1. When the heat transfer coefficient is large relative to the pipe thermal conductivity (large Bi numbers), there will be significant temperature gradients across the pipe wall and large thermal stresses will develop during a thermal shock. At small Biot numbers, conductivity dominates relative to surface heat transfer and there are small thermal gradients across the wall resulting in small thermal stresses. As can be seen, the stresses build with time, reaching a peak, then finally drop as the wall reaches a uniform temperature. Each curve has a maximum thermal stress. The maximum stress along with the time (Fo) when the maximum stress occurs are shown in Figure 4 as a function of the Biot number. Figure 4 is very significant because it enables one to determine the maximum thermal stress a pipe will experience when it is thermally shock using easily determined parameters.

CALCULATIONS OF PENETRATION DISTANCES - TRANSIENT FREEZING IN PIPES

Another issue pertaining to cold filling piping is how far the molten salt can flow through a cold pipe before freezing shut. This length is known as the penetration distance. There are several models which describe transient freezing in pipes, but one model in particular was developed in which data from several experiments and a variety of fluids were correlated to a single equation which describes the penetration distance as a function of the fluid properties, the Reynolds number, the wall temperature, and fluid temperature (Cheng and Baker, 1976). The correlation, Equation 13, describes the axial distance a fluid will flow through a cold pipe whose temperature is held below the fluid's freezing point before the fluid freezes the pipe shut. The outside wall temperature is held constant.

$$\frac{z}{D} = 0.23 Pr^{1/2} Re^{3/4} (\alpha_m / \alpha_s)^{1/9} [h_f / (Cp_s(T_f - T_w))]^{1/3} \times [1 + \gamma Cp_m(T_o - T_f) / h_f] \quad \text{Eq. 13}$$

This correlation should be conservative in our case since the pipes are allowed to heat up and their temperatures are not held at their initial value.

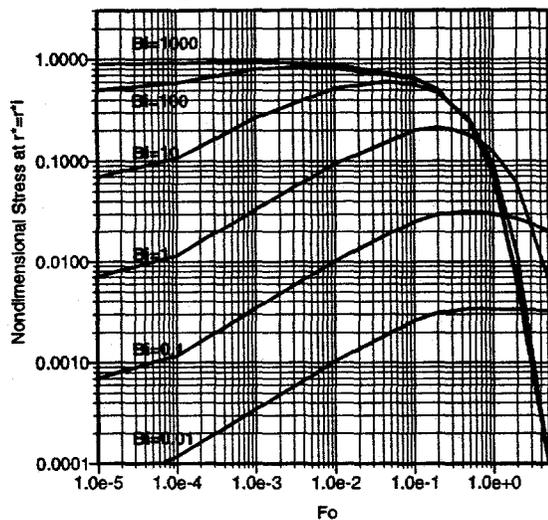


Figure 3. Nondimensional thermal (circumferential or axial) stress at the inner surface of the pipe undergoing thermal shock as a function of time (Fo) for several Biot numbers for $r_i/r_o=0.8$.

The penetration distances were calculated with molten salt properties for several pipe diameters and flow velocities. These results are shown in Table 2. For large diameter piping, such as used with the riser or down comer in the Solar Two central receiver power plant, we could theoretically flow through hundreds or thousands of feet of piping. In a commercial scale plant (100 MWe), we may be able to flow through *miles* of cold piping.

For the Solar Two receiver designed by Rockwell, the correlation predicts the receiver panels should be preheated above 200°F (93°C) when the headers and jumper tubes are heated to 550°F (288°C) to avoid freezing during the flood fill process at design flow rate. The panels should be preheated at least to 390°F (199°C) with ambient temperature jumper tubes and headers.

EXPERIMENTAL SETUP

We conducted cold fill experiments on two molten-salt receiver panels that we removed from a salt-in-tube receiver and on a section of piping in a molten-salt loop. Each panel consists of two serpentine-flow passes. Each pass has six 1 inch (2.5 cm) OD 304 stainless steel tubes with 0.065 inch (1.65 mm) thick walls. The two passes are connected to a common 6 inch (15 cm) diameter manifold (schedule 80 piping) at the top of the panel. Each panel vent connects to a common 1 inch vent line. The experiment is located at the base of the Solar Tower at the National Solar Thermal Test Facility at Sandia National Laboratories in Albuquerque, NM.

In this flow loop, salt is pumped from the salt sump and can either return to the sump or can be diverted up the riser. At the top of the riser is the pressurized accumulator (surge) tank. The salt flows through the down comer and can either be diverted to the panel or back to the sump. The outlet of the panel returns to the sump. The pump can flow salt at 100 gallons per minute (380 liters/min) through 2 inch (5.1 cm) schedule 40 stainless steel piping.

RESULTS OF COLD FILL PANEL AND PIPING TESTS

We conducted tests where we varied the initial panel temperature to determine whether salt could flow through all four passes of the panel before freezing. The flow velocity was approximately the same for each test. The purposes of these tests were to 1) determine if salt flow could be established in "cold" manifolds, panels, and piping, 2) measure the thermal responses of the tubes and manifolds undergoing thermal shock, and 3) estimate the corresponding stresses in the materials.

We found we were able to consistently flow through ambient temperature manifolds and panels without blocking tubes with frozen salt. In our set up, we were able to fill the panels only in a serpentine fashion. To prevent entrapment of air, we had to fill the panel slowly (~2 ft/s, 0.6 m/s). The total length of tubing is about 60 feet (18 m). The correlation predicts the fluid should freeze in about 50 feet (15 m). This means we were probably close to freezing shut. The correlation may be conservative relative to an insulated pipe since the insulated pipe has a finite heat capacitance.

Figure 5 shows the temperature response of the outside surface of the receiver tubes and upper manifold as they are filled with 550°F (288°C) salt. The receiver tubes were initially at 50°F (10°C). The

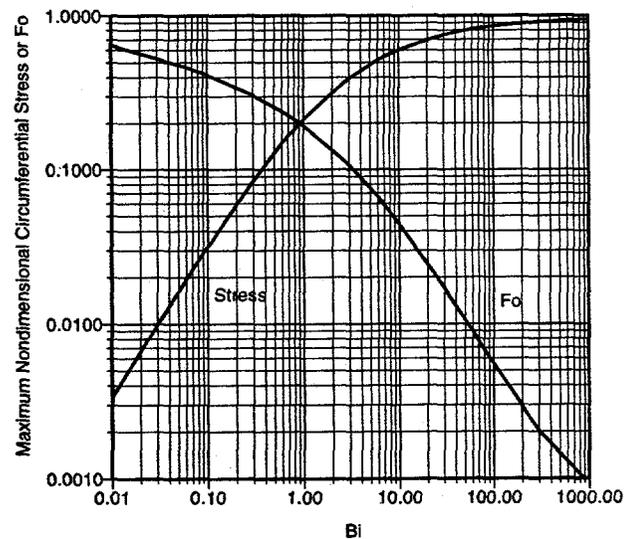


Figure 4. The maximum nondimensional thermal stress, σ' , and time (Fo) when maximum stress occurs as a function of the Biot number for a pipe undergoing thermal shock for $r_i/r_o=0.8$. These are the maxima of Figure 3. header was hotter than the panels, initially, since an adjacent heat trace zone conducted heat to it. The header and first pass receiver tubes experienced the greatest thermal shock. As the salt continued through the other passes, the temperature of the initial slug of salt decreases resulting in the deposition of a frozen layer of salt on the tube-wall which reduces the shock then, melts away. This can be inferred from the change in slope of the fourth pass tube temperature and the upper header temperature. A frozen layer develops which limits the rate at which the temperature can rise and provides some thermal capacitance.

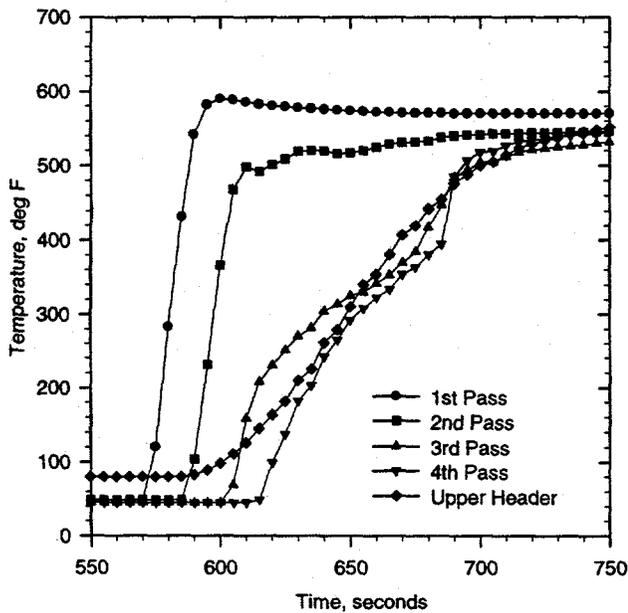


Figure 5. Temperature response of the outside surface of cold receiver tubes and upper header as they are filled with 550°F (288°C) salt.

In addition to cold filling the panels and manifolds, we conducted similar tests on a section of piping. We turned off the heat trace to a section of piping and let it cool to ambient, then initiated salt flow to determine its thermal response and estimated heat transfer coefficients and stresses. We measured the thermal response of an insulated 40 foot (12 m) long, 2 inch (5.1 cm) diameter 316 SS, schedule 40 pipe undergoing thermal shock. The piping was part of the riser. We turned off the heat trace to allow it to cool to ambient. When the piping was cold (at ambient) we pumped salt through it at approximately 9.5 ft/s (2.9 m/s) and measured the temperature at the outside of the pipe. Figure 6 is a plot of the outside wall temperature as a function of time.

In the thermal shock experiment on the panel, manifolds, and piping, we measured the outside wall temperatures. From that data, we calculated the average Biot number and stresses developed in the wall of the pipe or tube as it rapidly heated up.

The solid line in Figure 6 is a fit of the data to the model for a constant heat transfer coefficient - adjusting for the starting time of the transient since the heat transfer coefficient is not constant initially. In each case the stresses were lower than the endurance

Table 3. Maximum Velocities For Cold Fill Where Maximum Thermal Stresses are Below Endurance Limit of the Material $T_{wall} = 25^{\circ}\text{C}$ and $T_{salt} = 288^{\circ}\text{C}$.

Pipe Diameter	Schedule	Material	Maximum Vel., m/s
6 inch	80	Stainless 316	0.9
6 inch	80	Carbon	3.7
6 inch	40	Stainless 316	1.5
6 inch	40	Carbon	6.3
6 inch	10	Stainless 316	3.8
16 inch	80	Carbon	1.9
16 inch	40	Carbon	3.7
16 inch	10	Carbon	12.2
16 inch	10	Stainless 316	5.7

limit of the material. It is likely these stresses are conservative since the heat transfer coefficient is not constant with time but grows to the equilibrium value.

For heat transfer in fully developed pipe flow when applied to freezing with turbulent flow, the following correlation has been suggested to estimate heat transfer coefficients between the fluid and the frozen layer (Mujumdar and Mashelkar, 1984):

$$Nu = 0.0155 Re^{0.83} Pr^{0.5} (R_o/R)^{0.83} \quad \text{Eq. 14}$$

where R_o is the inner pipe radius and R is the radial coordinate of the frozen layer. This correlation is applicable beyond the thermal entrance length (approximately 10 tube diameters) and gives a conservative estimate of the heat transfer to the pipe since the frozen salt layer will act as an insulator.

Using Equation 14, a conservative estimate of the maximum circumferential stresses at the inside surface of a pipe or tube can be calculated as a function of velocity. There is a maximum velocity at which the stresses are below the endurance limit of the material. These velocities are listed in Table 3 for several pipe schedules and materials proposed for handling molten salt. Carbon steel is able to handle thermal stresses better than stainless steel due to the fact that carbon steel has a much higher thermal conductivity and lower coefficient of thermal expansion, even though its endurance limit is lower.

Even though the stresses in the walls of piping or tubes are low when thermally shocked, high stresses could develop where there is a sudden change in wall thickness or abrupt changes in contour resulting in stress concentrations. It should be noted that this analysis applies to vertical runs of piping or tubes where the temperature gradient is a function of the radial component only. In horizontal pipes, the leading edge of the fluid could have a sloped profile resulting in a circumferential temperature gradient and a different stress distribution. Most of the piping in a molten-salt central-receiver solar power plant, though, is in vertical runs.

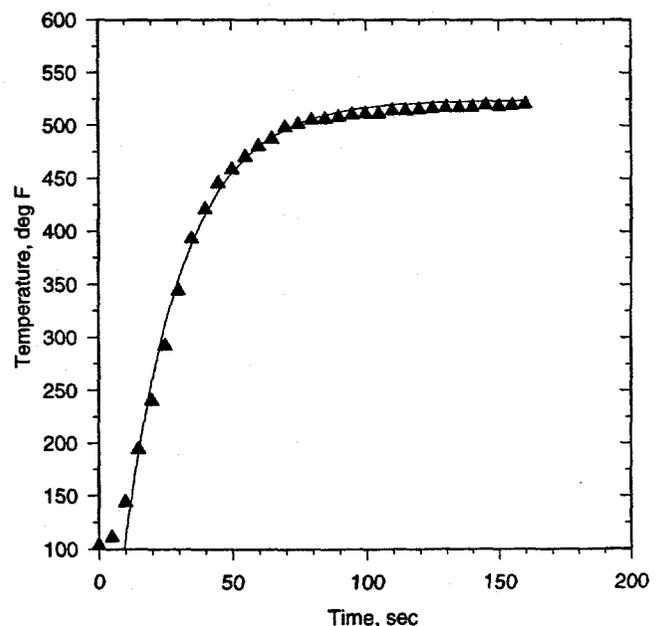


Figure 6. Outside wall temperature as a function of time for a 2 inch schedule 40 pipe undergoing thermal shock. The symbols are actual data points. The solid line is a fit of the data for $Bi = 0.444$.

SUMMARY OF COLD FILL TESTS

The following conclusions can be made about the cold filling. Cold filling should have the lowest parasitic power consumption of the methods discussed for thermal conditioning the pipes in a molten salt solar power plant.

Cold filling piping, receiver panels and/or manifolds in a molten-salt central-receiver solar power plant is feasible. The molten salt will not freeze in large pipes such as used in the riser and down comer provided the velocity is high enough. In normal operation, the panels are preheated with heliostats so it would not be necessary to cold fill the panels. As a minimum, our results show that the entire panel does not have to be above the salt freezing temperature before salt flow is established.

Results from the stress analysis showed that the stresses in the piping, header and receiver tubes were below the endurance limit during a thermal shock as long as a maximum velocity was not exceeded. Analysis should be done for a particular design of the tube-to-header junction, transitions in piping cross section, and expansion loops to make sure there are not any localized stress concentrations and to estimate the life based on fatigue of these areas.

The best combination of reduced parasitics and increased availability might be partial preheating (e.g., preheating to 300°F).

We recommend that even if the piping is cold filled, valves, flanges, and instrumentation should be kept near the salt temperature (i.e., heat traced) to minimize reliability issues that could arise if these components were thermally stressed.

Delicate components such as valves and flanges should not be thermally shocked whenever possible. Their complicated geometry could cause severe stress concentrations and premature failure or salt leaks.

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NOMENCLATURE

Bi = Biot number
Cp_s = specific heat of solid
Cp_m = specific heat of liquid
D = diameter of pipe
E = modulus of elasticity
Fo = Fourier number
h = heat transfer coefficient
h_f = heat of fusion
k = thermal conductivity of pipe (Eq. 5)
L = wall thickness
Nu = Nusselt number
Pr = Prandtl number
r = radial coordinate of pipe
r_i = inner radius of pipe

r_o = outer radius of pipe
r* = nondimensional pipe radius
R = radial coordinate of inner radius of pipe
R_o = radial coordinate of frozen layer
Re = Reynolds number
T = temperature
T_f = freezing point
T_i = initial wall temperature
T_o = inlet liquid temperature
T_w = wall temperature
T_∞ = fluid temperature
x* = nondimensional distance from insulated surface
z = distance to freeze closed
α = thermal diffusivity (Eq. 3) or coefficient of thermal expansion (Eq. 8-10, 12)
α_m = thermal diffusivity of liquid
α_s = thermal diffusivity of solid
δ = 1 - r_i* = nondimensional wall thickness
λ_n = characteristic values of transient conduction equation
γ = parameter measuring the relative importance of sensible to latent heat, assumed to be 0.7 (water)
θ* = nondimensional temperature
θ_o* = nondimensional temperature at the insulated surface
σ_θ = circumferential stress
σ_r = radial stress
σ_z = axial stress
σ* = nondimensional thermal stress
ν = Poisson's ratio

REFERENCES

- Bergan, N. E. *Testing of the Molten Salt Electric Experiment: Solar Central Receiver in an External Configuration*, Sandia National Laboratories report SAND86-8010, October 1986.
- Boyer, H. E. and Gail, T. L., editors, *Metals Handbook Desk Edition*, American Society for Metals, 1985.
- Burr, A. H., *Mechanical Analysis and Design*, Elsevier Science Publishing Co., Inc, 1982.
- Chavez, J. Kolb, G., Tyner, C., *Today's Solar Power Towers*, Sandia National Laboratories brochure, SAND91-2018, January 1995.
- Cheung, F. B. and Baker, L. Jr, "Transient Freezing of Liquids in Tube Flow", *Nuclear Science and Engineering*, Vol. 60, pp. 1-9, 1976.
- Goodier, J. N., "Thermal Stress", *ASME J. Appl. Mech.*, vol.4, no. 1, March 1937.
- Incorpera, F. P. and De Witt, D. P., *Fundamentals of Heat and Mass Transfer*, second edition, John Wiley & Sons, pp 181-191, 1985.
- Mujumdar, A. S. and Mashelkar, R. A., editors, *Advances in Transport Processes*, Vol III, Wiley Eastern Limited, pp. 35-117, 1984.
- Kolb, G., Sandia memo to Irving Katter, *Overnight Thermal Conditioning of Salt Pipes at Solar Two*, October 12, 1993.
- Pacheco, J., Ralph, M., Chavez, J., Dunkin, S., Rush E., Ghanbari, C., Matthews, M., *Results of Molten Salt Panel and Component Experiments for Solar Central Receiver Power Plants: Cold Fill, Freeze/Thaw, Thermal Cycling and Shock, and Instrumentation Tests*, Sandia National Laboratories report, SAND94-2525, January 1995.
- Young, W. C., *Roark's Formulas for Stress and Strain*, sixth edition, McGraw-Hill, pp. 722-724, 1989.