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Calculation Model for the Thermal Insulation of Pipes and Tanks

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1. Introduction

Sodium experimental facilities such as STELLA-1 (Sodium Integral Effect Test Loop for Safety Simulation and Assessment) [1] consist of various tanks and pipes at high operating temperatures above 500℃. During the design of experimental facilities, it is essential to create a thermal design to meet the test conditions and to reduce the heat loss from the tanks and pipes. Heat loss from the pipes and equipment has been calculated for the design of experimental systems [2]. In this study, the calculation model to estimate the thickness of insulating materials and the heat loss was applied to the various experimental conditions of STELLA-1. Some parameters dominating the heat transfer were varied for the insulation design sensitivity.

2. Method and Results

2.1 The Calculation Model

The heat balance equations for the cylindrical and plate geometry can be found elsewhere [3]. For cylindrical geometry, heat balance is expressed as:

$$q_{cd} = \frac{T_1 - T_2}{\left[\ln(r_1/r_2)/2\pi k_{ins}L\right]} = q_{cv} = \frac{T_2 - T_\infty}{\left[1/2\pi r_2 Lh\right]} \qquad (1)$$

The overall heat loss from the cylindrical surface is given by

$$q_{loss} = \frac{T_1 - T_\infty}{\left[\ln(r_2/r_1)/2\pi k_{ins}L\right] + \left[1/2\pi r_2 Lh\right]} \qquad (2)$$

For the plate geometry, similar equations are given as

$$q_{cd} = \frac{T_1 - T_2}{\Delta x / k_{ins}A_{pl}} = q_{cv} = \frac{T_2 - T_\infty}{1/A_{pl}h} \qquad (3)$$

and the total heat loss is written as

$$q_{loss} = \frac{T_2 - T_\infty}{\Delta x / k_{ins}A_{pl} + 1/A_{pl}h} \qquad (4)$$

For the given geometry and boundary conditions, we can evaluate the insulation thickness Δx and total heat loss q_{loss} with the given insulating materials if we know the heat transfer coefficients. Usually the outer surface temperature T_2 and room temperature T_∞ are restricted when obtaining the operational conditions.

2.2 The free-convection heat transfer coefficient

The average free-convection heat transfer coefficient can be represented in the following form for a variety of circumstances[3]:

$$\overline{Nu} = C\left(Gr_f \Pr_f\right)^2 \qquad (5)$$

Table1. Constants for use with Eq.(5)for isothermal plate

Heated surfaces	$Gr_f \cdot \Pr$	C	m
Upper surface	2×10^4-8×10^6	0.54	1/4
Upper surface	8×10^6-10^{11}	0.15	1/3
Lower surface	10^5-10^{11}	0.27	1/4

For vertical isothermal surfaces, the free convection heat-transfer coefficients can be expressed as

$$\overline{Nu}^{1/2} = 0.825 + \frac{0.387\, Ra_f^{1/6}}{\left[1 + (0.492/\Pr)^{9/16}\right]^{8/27}}$$

for $10^{-1} < Ra_f < 10^{12}$ (6)

These equations can be applicable for a vertical plate and cylinder when $D_{cyl}/L_{cyl} > 35/Gr^{1/4}$ is satisfied.

The Nusselt number for isothermal horizontal cylinder is given as

$$\overline{Nu} = \left[0.60 + \frac{0.387\, Ra_f^{1/6}}{\left[1 + (0.559/\Pr)^{9/16}\right]^{8/27}}\right]^2$$

for $10^{-5} < Ra_f < 10^{12}$ (7)

Properties for Eqs. (5) to (7) are evaluated at the film temperature.

2.3 Calculation Procedures

At first, we can evaluate the overall heat transfer coefficients from Eqs. (4) to (7) with the given geometry and target value T_2 . For the head part of the tanks, the surface area of oblate spheroid form can be obtained from the mensuration formula and treated as a plate for the calculation of *Nu*. With this heat transfer coefficient, we can find out the Δx meeting the target value T_2 at each surface using Eqs. (1) to (4). If we know the insulation thickness Δx for a given geometry, the total heat loss can be obtained for each surface. After choosing the appropriate Δx, the total heat loss and surface temperature T_2 are re-evaluated to confirm the restricted conditions and experimental conditions.

2.3 Results and Discussions

Table 2 shows the input geometry for the design calculation. The target value T_2 was maintained at 60 ℃ throughout the analysis. The calculation results are shown in Figs. 1 to 2. In Fig. 1(a) and (b) Δx's and q_{loss}'s were nearly proportional to the k_{ins}. The horizontal pipe needs thicker insulation at the same conditions, but the heat loss is smaller than that of the vertical pipe because of different heat transfer coefficients. The h value of the horizontal pipe has 4.17 $w/m^2 \cdot ℃$ which is 8.3% lower than that of the vertical pipe. The higher T_1 requires thicker insulation and looses more heat than the lower T_1.

Table2. Geometry of Pipes and Tanks for Sample Calculation (unit : mm)

Pipe and Tank \ Geometry	Diameter	Length	Minor axes for head
10SCH10 pipe(SC10)	273.0	1000	
4SCH10 pipe(SC4)	114.3	1000	
Horizontal tank(HT)	2800	4800	700
Vertical tank(VT)	1100	1100	275

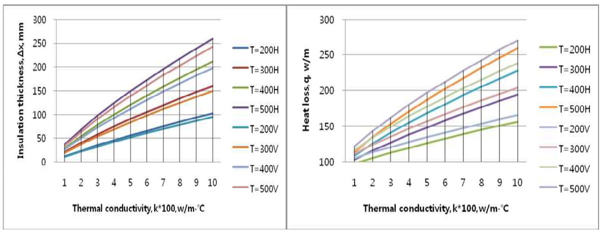

Fig1. Insulation thickness and heat loss of horizontal and vertical 10SCH40 pipes at 35℃ room temperature.

In Fig. 2(a) and (b), a large D_{cyl} pipe requires thicker insulation than a small D_{cyl} pipe at the same T_2, and a low T_∞ has greater heat loss at the same conditions.

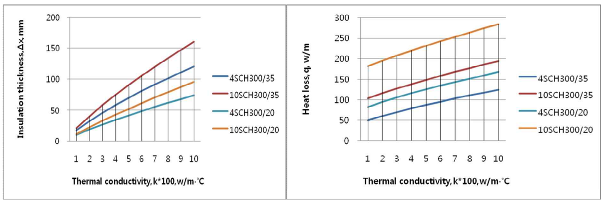

Fig2. Calculation results of the 2 different size horizontal pipes at 2 different room temperatures.

Table3.Calculated insulation thickness and heat loss for HT at T_1=300℃and T_2=60℃

Values	Δx(mm)		q_{loss}(kw)	
k_{ins}(w/m·℃) \ Room temp (℃)	T_∞=35	T_∞=20	T_∞=35	T_∞=20
0.03	77.1	41.1	5.9	10.8
0.05	127.0	68.0	6.1	11.0
0.08	202.3	107.5	6.8	11.5

Table4. Calculated insulation thickness of each face and heat loss for VT at T_1=300℃.and T_2=60℃

Case \ Values		Insulation thickness Δx(mm)			q_{loss}(W)
Room temp.(℃)	k_{ins} (w/m · ℃)	cyl.	up-head	down-head	total
$T\infty$=35	0.05	110.9	96.4	343.9	660.6
	0.08	170.2	154.3	564.0	883.6
$T\infty$=20	0.05	61.0	50.6	185.7	1303.6
	0.08	94.5	81.0	287.4	1235.5

The calculated results of HT show the similar trends as the results of the pipe in Table 3. In Table 4. each face of VT needs a different insulation thickness to meet the requirements of T_2. Using the results of the insulation thickness of the cylindrical part, the down face of VT has a higher T_2 as shown in Table5.

Table5. Surface temperatures and heat loss with the insulation thickness of the cylindrical part

Values			Temp. T_2(℃)			q_{loss}(W)
T_1/T_∞ (℃)	k_{ins} (w/m.℃)	Part Δx(mm)	Cyl.	Up	Down	Total
300/35	0.05	110.9	60.0	57.0	99.7	713.8
	0.08	170.2	60.0	57.9	103.0	1001.9
300/20	0.05	61.0	60.0	54.0	114.2	1392.8
	0.08	94.5	60.0	55.0	114.2	1315.5

3. Conclusion

The insulation thickness and heat loss of various experimental conditions were estimated. The thermal conductivity of insulating materials and the room temperature were dominant parameters for insulation design. It was confirmed that the natural convection heat transfer correlation of the horizontal cylinder can be simply applied to a thermal insulation design with some margin of heat loss compensation except the downward face of the tanks. For the downward face, a more rigorous model should be selected and checked to meet the object conditions.

Nomenclature

- T_1 pipe or tank surface temperature
- T_2 surface temperature of insulating material
- T_∞ ambient room temperature
- L length or characteristic length
- D_{cyl} diameter of cylindrical tank
- q_{cd}, q_{cv} heat transferred by conduction, convection
- q_{loss} heat loss from the insulating material surface
- k_{ins} thermal conductivity of insulating material
- h heat transfer coefficient outside the insulator
- Δx insulation thickness
- A_{pl} surface area
- r_1 outer radius of pipe or tank
- r_2 outer radius of insulating material (=r_1+Δx)
- Ra_f, Gr_f Rayleigh and Grashof number respectively
- Pr, Nu Prandtl and Nusselt number respectively

REFERENCES

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