

# Freeze Protection in Gasholders

## Problem presented by

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## Executive Summary

In cold weather, the water seals of gasholders need protection from freezing to avoid compromising the seal. These holders have a large reservoir of “tank water” at the base which is below ground. At present freeze-protection is achieved by external heating of the seal water which is in a slotted channel called a cup. Electrical heating or circulation of heated tank water to the cup are examples of systems presently used. The tank water has a large thermal capacity and National Grid wishes to investigate whether circulation of the tank water without external heating could provide sufficient energy input to avoid freezing. Only tanks in which the tank water is below ground are investigated in the report. The soil temperature under the reservoir at depth of 10m and lower is almost constant.

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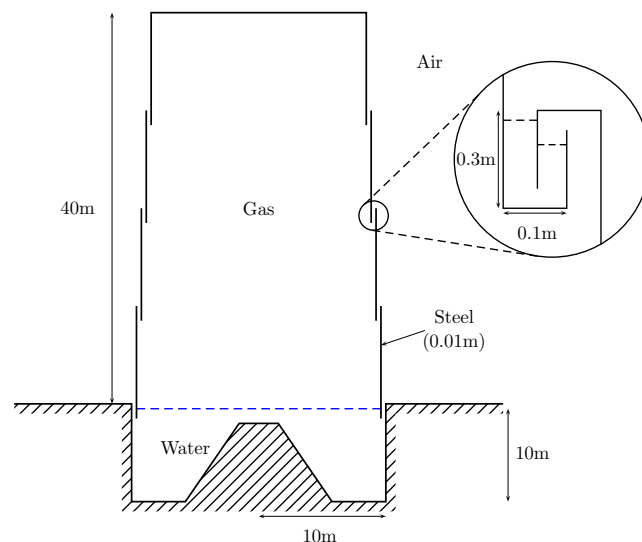
# 1 Introduction

## 1.1 The problem scenario

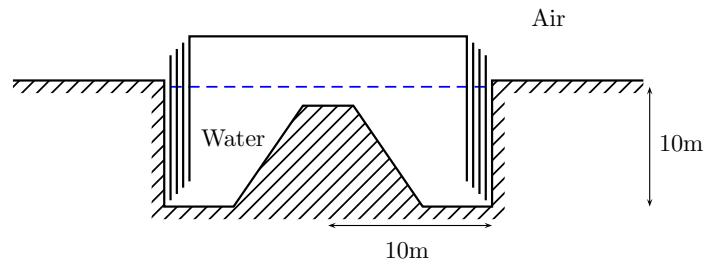
- (1.1.1) In cold weather the water seals of gasholders need protection from freezing to avoid compromising the seal. These holders have a large reservoir of “tank water” at the base which is below ground. At present freeze-protection is achieved by external heating of the seal water which is in a slotted channel called a cup. Electrical heating or circulation of heated tank water to the cup are examples of systems presently used. The tank water has a large thermal capacity and National Grid wishes to investigate whether circulation of the tank water without external heating could provide sufficient energy input to avoid freezing.

## 1.2 The problem-solving approach

- (1.2.1) The thermal capacity of the tank water is the key to determining whether external heating is not needed. There are 3 major components of the system, viz. the tank water, the gas and the circulating water which would act as After initial brainstorming, it was clear that the problem could be treated as 4 major elements, namely the gas, the water delivery hose, the tank water and the water in the cup seal. A standard tank was specified, as shown in Fig 1 (tank full of gas) and Fig 2 (tank empty). The water delivery hose (not shown in Fig 1) lifts water from the tank to each seal. The four problems will be treated in turn.



**Figure 1:** Gasholder geometry.



**Figure 2:** Gasholder configuration when empty.

## 2 The four problems

### 2.1 The gas in the tank

- (2.1.1) The gas in the tank is cooled in inclement weather by convective cooling from the outside of the tank and natural convection in series with this on the inside. A  $10 \text{ Wm}^{-2}\text{K}^{-1}$  heat transfer coefficient inside the tank, and  $100 \text{ Wm}^{-2}\text{K}^{-1}$  outside (slightly increased from the the average expected heat transfer coefficient (see e.g. Incropera and DeWitt) so as to make some allowance for radiative heat loss if skies are clear and dark) are used for illustration.

The inside heat transfer coefficient is confirmed in the literature; the outer one has various estimates, with a values of  $45 \text{ Wm}^{-2}\text{K}^{-1}$  used in the literature for wind flow past vertical cylinders and  $60 \text{ Wm}^{-2}\text{K}^{-1}$  used in buildings exposed to similar temperature driving forces and windspeeds. Slightly lower values were used in calculations shown to us by National Grid (about  $35 \text{ Wm}^{-2}\text{K}^{-1}$ ), but these seem slight underestimates; experimental validation of the values is desirable.

With the first-quoted values the overall thermal resistance of heat transfer from gas to ambient air through the walls and roof of the gasholder is  $0.11 \text{ m}^2\text{K}/\text{W}$ . A representative temperature driving force is  $13 \text{ K}$  (from a gas temperature of  $276 \text{ K}$  to ambient air at  $263 \text{ K}$ ). This gives a heat loss of  $330 \text{ kW}$ . An estimate of the gas mass is  $8500 \text{ kg}$ . This heat loss from the gas which has low thermal capacity leads to rapid reduction of the gas temperature, on a timescale much shorter than one day.

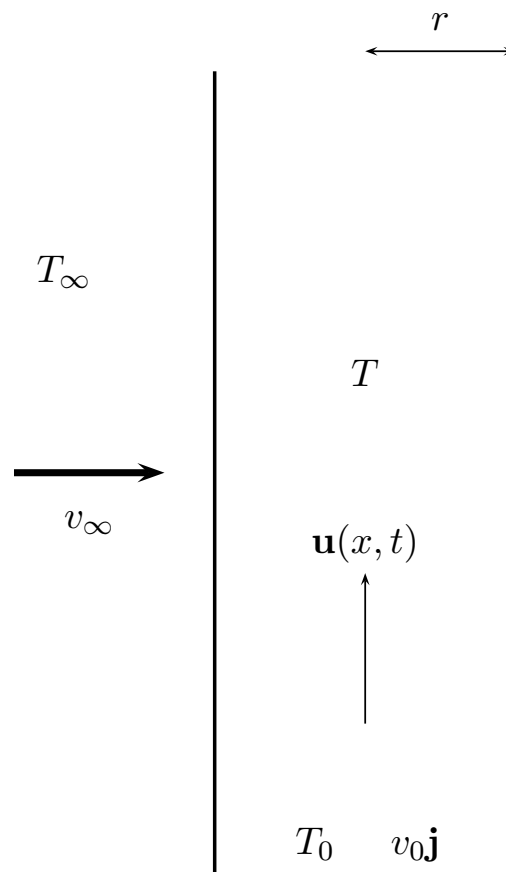
There will of course be some heating of the gas from the tank water with which it is in contact, by natural convection with an estimated heat transfer coefficient of  $10 \text{ Wm}^{-2}\text{K}^{-1}$  akin to underfloor heating. This heat input will be much less than the heat loss as the contact area is less than even the roof area and the heat losses are from both the roof and the walls. Also the temperature driving force for water to gas heating is small initially, so the heat losses predominate, leading to a rapid reduction of the gas temperature. (Later, when the gas is cool, e.g. at a temperature driving force of  $10 \text{ K}$ , heat gain from tank water is  $30 \text{ kW}$ .) The steel wall temper-

ature is also close to ambient. This is also consistent with two operational observations. The first is that in hot weather gas expansion is obvious. The second is that accidental overflow water does indeed freeze in cold weather, indicating that the steel temperature is subzero.

## 2.2 The water delivery hose

(2.2.1) This hose is made of a plastic and is long enough to reach the top cup seal. The outside of the tube is subjected to convective cooling due to wind and the colder ambient air.

For the first analysis, the lowest relevant external air temperature is used and a simplified hose was considered as shown in Figure 3.



**Figure 3:** Schematic of simplified water delivery hose.

An infinite cylinder  $r = a$ , has wind blowing past it with velocity  $v_\infty \bar{\mathbf{j}}$  and

temperature  $T_\infty$  at  $-\infty$  with hose pipe boundary at  $r = a$  and small wall thickness  $d$ .

Inside the hose pipe, water enters at  $z = 0$  with temperature  $T_0$  and velocity  $v_0 \bar{j}$ . In cross section of the pipe we have flow past a cylinder  $\underline{u}(\underline{x}, t)$ ,  $T(\underline{x}, t)$  with the heat equation and continuity of heat flux across the boundaries.

Approximation:

- (a) Assume  $v_\infty$  to be large. This gives that the temperature at the boundary is  $T_\infty$ .
- (b)  $d=0$
- (c) Plug flow  $\underline{u}(\underline{x}, t)$  in pipe is equivalent to  $u\bar{k}$  with  $u=\text{constant}$ .

Now, we have to the following equation:

$$k\nabla^2 T = \left( \frac{\partial T}{\partial t} + \underline{u} \cdot \nabla T \right) = \left( \frac{\partial T}{\partial t} + u \frac{\partial T}{\partial z} \right).$$

Assume that conduction dominates, i.e.  $\frac{\partial^2 T}{\partial z^2}$  and  $\frac{\partial T}{\partial z}$  can be neglected. So, with  $T = T(r, z, t)$  we can write the above equation as

$$\frac{k}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) = u \frac{\partial T}{\partial z}.$$

Let

$$T = \tilde{T} + R(r)Z(z)$$

with  $T = T_\infty$  as  $z \rightarrow \infty$ , i.e.  $\tilde{T} = T_\infty$  and  $T = T_\infty$  at  $r = a$  for all  $z > 0$ .

Then

$$\frac{1}{r} \frac{\partial}{\partial r} (rR') Z = uk^{-1}RZ'.$$

Separation of variables gives

$$Z = Ae^{-\lambda^2 z}$$

and the above equation can be reduced to the Bessel equation

$$R'' + \frac{1}{r}R' + \lambda^2 uk^{-1}R = 0.$$

Solving this equation, we get

$$R = J_0 \left( \lambda \sqrt{uk^{-1}} r \right).$$

Therefore,

$$T = T_\infty - AJ_0 \left( \lambda \sqrt{uk^{-1}} r \right) e^{-\lambda^2 z}$$

with

$$\lambda = \frac{\delta_0}{\sqrt{uk^{-1}} a}$$

where  $\delta_0$  is the first zero of the Bessel function  $J_0(x)$ .

Now, at  $z = 0$ , we want average temperature equal to  $T_0$  (we ignore entry length problems). Hence,

$$T_\infty \pi a^2 + AI = T_0 \pi a^2$$

where

$$\begin{aligned} I &= \int \int_{r \leq a} r J_0(\lambda \sqrt{uk^{-1}} r) dr d\theta \\ &= \frac{2\pi}{\lambda^2 uk^{-1}} \int_0^{\delta_0} x J_0(x) dx \\ &= \frac{2\pi}{\lambda^2 uk^{-1}} \left[ \left( \frac{x^2}{2} J_0(x) \right)_0^{\delta_0} - \int_0^{\delta_0} -\frac{x^2}{2} J_1'(x) dx \right] \\ &= \frac{2\pi}{\lambda^2 uk^{-1}} \left[ \frac{1}{2} r^2 J_2 \right]_0^{\delta_0} \\ &= \frac{2\pi \delta_0^2 J_2(\delta_0)}{\lambda^2 uk^{-2}} \\ &= \pi a^2 J_2(\delta_0). \end{aligned}$$

Hence

$$A = \frac{(T_0 - T_\infty) \pi a^2}{I} = \frac{(T_0 - T_\infty)}{J_2(\delta_0)}$$

with  $a = 0.05$  m,  $l = 50$  m,  $u = 0.1$  m/sec,  $k = 0.14 \times 10^{-6}$  m<sup>2</sup>/s,  $\delta_0 = 2.4$ . We have

$$\lambda^2 l = \frac{\delta_0^2 l}{uk^{-1}} = 0.0004,$$

so the heat capacity lost as a percentage of the input heat capacity is approximately 0.04%.

An alternative analysis applies Newton's Law of cooling as the boundary condition to take into account the heat transfer between the air, the hose and the water. Hence, we solve the problem described above but now subject to

$$\hat{T}_r = -H\hat{T} \quad \text{on} \quad r = a$$

where  $\hat{T} = T - T_\infty$  and  $H$  is the heat transfer coefficient, rather than imposing  $T = T_\infty$ . The solution is as above but with this new condition essentially leading to an amended decay rate  $\lambda$ . In this arrangement, for the worst case air temperature, an assumed tank water temperature of 7°C, taking an exterior heat transfer coefficient of 60 Wm<sup>-2</sup>K<sup>-1</sup>, yielded a 0.7 degree drop in temperature for a 30m hose.

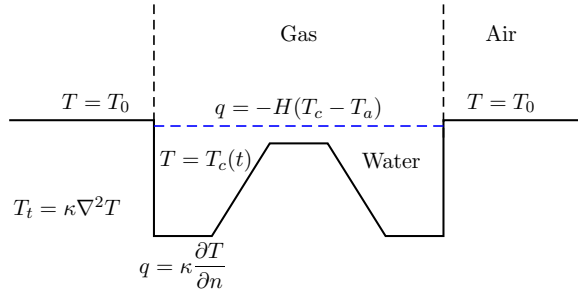
Therefore there is a temperature drop in the feed hoses to the cups but this is no more than 1K.



## 2.3 The tank water

### (2.3.1) Tank Temperature

The tank water temperature is crucial to assessing whether external heating may be obviated. For the below-ground tank, the earth will gradually provide heat to replace that lost by circulation. The geometry of the problem is given in Figure 4.



**Figure 4:** Underground tank geometry and model.

The energy required to prevent freezing in the cups must come from the energy stored in the water tank at the bottom if no external heating is going to be provided. The questions we would like to know are how the temperature of the tank water is affected by the circulation through the cups (i.e. how much energy is lost during this process), and whether this will significantly reduce the energy remaining in the tank. We also want to know how the temperature is distributed in the tank, so that the water to pump to the cups may be taken from the optimal location (e.g. is it better to take the water from the bottom of the tank?).

The tank base is typically 10 m underground, and at such depths the earth's temperature varies very little throughout the year. A quick calculation is to solve

$$T_t = \kappa_{rock} T_{zz}$$

for  $z < 0$ , with

$$T = \bar{T}_a + \frac{\Delta T_a}{2} \cos \omega t \quad \text{at } z = 0,$$

$$T_z \rightarrow -\frac{G}{k_{rock}} \quad \text{as } z \rightarrow -\infty,$$

where  $G$  is the geothermal heat flux, which typically has a value between 50 and 100 mW m<sup>-2</sup>.  $\kappa_{rock} \approx 10^{-6}$  m<sup>2</sup>s<sup>-1</sup> is the thermal diffusivity of rock. This gives

$$T = \bar{T}_a - \frac{G}{k} z + \frac{\Delta T_a}{2} \exp \sqrt{\frac{\omega}{2\kappa}} z \cos \left( \omega t + \sqrt{\frac{\omega}{2\kappa}} z \right),$$

and the exponential drop off occurs on a length scale of order  $\sqrt{\frac{2\kappa}{\omega}} \sim 3$  m when the period of the temperature variations is 1 year.  $G/k \sim 0.05$  K m<sup>1</sup> (the thermal conductivity of rock is roughly  $1\text{ W m}^{-1} \text{ K}^{-1}$ ), so the temperature at 10 m depth is almost constant and is approximately given by the average air temperature  $\bar{T}_a$ .

In the UK this temperature may be around 10 K, and our initial thoughts were that the temperature at the bottom of the tank should be at this throughout the year, the temperature of the surrounding earth being assumed not to change considerably. The temperature at the top of the water tank would match the gas temperature above, which is assumed to be comparable with the external air temperature outside. During the winter when this temperature drops to freezing point, a temperature difference of 10 K apparently occurs across the depth of the tank.

Since water expands at temperatures larger than 4 K, the colder water is more dense and should sink - the tank would be unstably stratified. This suggests that convection will occur to mix the tank water. The Rayleigh number, which indicates whether convection will occur is

$$Ra = \frac{g\beta\Delta T d^3}{\nu\kappa} \approx 10^{14},$$

where  $\beta \approx 10^{-4} \text{ K}^{-1}$  is the thermal expansion coefficient of water. This is much much larger than the critical value for convection which is of order  $10^3$ . This suggests that the water in the tank should become well mixed, and therefore of uniform temperature, during the winter when the surface temperature is colder than the bottom temperature of the rock. In the summer when the surface temperature is larger, the water is stably stratified and will maintain a temperature gradient through it's depth.

The very large Rayleigh number caused some intrigue, and led to comparisons with what type of convection, if any, occurs in lakes and ponds during the year. A search of literature on the temperature variation with depth in small lakes led to the conclusion that the same type of mixing during the winter and stable stratification during the summer is commonly found there. A particularly revealing study measured the temperature at different depths in a tarn in the Lake District, which was of similar depth to the gas holder tank and found that the temperature during the winter was uniform with depth, and was somewhere between the rock temperature below and the air temperature above. A completely different scenario applies in summer when the stratification is predominantly stable and the tank is far from well-mixed.

We similarly expect the temperature in the water tank to be somewhere between the temperature of the rock beneath and the temperature of the gas above (which, we believe, closely follows the air temperature). The question then is what this temperature should be, and whether it is almost steady, or varies in time. The details of the convection cells and the

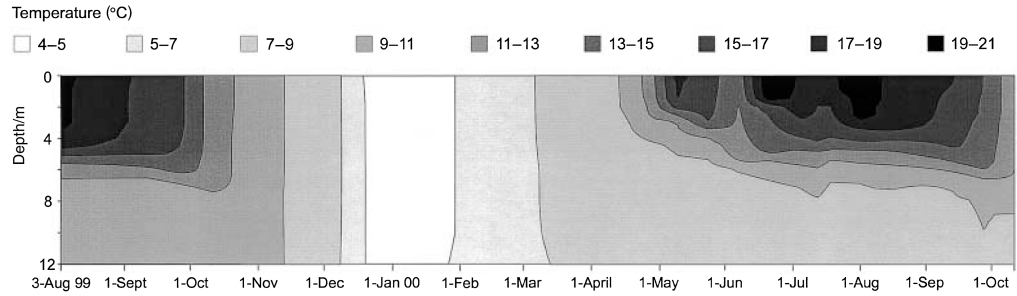


Fig. 2 Contour plots of the temperature (at 2 °C intervals) in Blelham Tarn with depth and time, August 1999–October 2000.

**Figure 5:** Lake temperature profiles and their seasonal variation. Taken from Davis et al (2003).

turbulent boundary layers within the tank, while potentially interesting, are less important than this overall temperature.

Measurements were made at two gas holders in Birmingham and it was found that the temperature in each of the water tanks was almost uniform, decreasing slightly with depth, at around 7 C, while the air temperature outside was 10 C. This agrees with our deduction that the temperature should be similar throughout the tank.

A simple balance between heat coming in from the rock (from below and from the sides of the tank) and heat lost to the gas led to the suggestion that the core temperature of the tank water  $T_c$  could be related to the rock temperature  $T_b$  (assumed constant) and the gas temperature  $T_g$  (close to air temperature) by

$$(T_b - T_c)H_{rock} = (T_c - T_g)H_{gas}$$

where  $H_{rock}$  and  $H_{gas}$  are the overall heat transfer coefficients for transport between the rock and the tank, and the gas and the tank, respectively. Then the core temperature could be calculated from

$$T_c = \frac{T_b H_{rock} + T_a H_{gas}}{H_{rock} + H_{gas}}.$$

The problem here is knowing what these heat transfer coefficients are; it was thought that  $H_{rock}$  might be assumed to be constant and could be somehow measured, and that  $H_{gas}$  might similarly be measured by taking some measurements at different times of the water temperature  $T_c$  and the air temperature  $T_a$ , and that the hoped for linear relationship between the two might be used to recover the constant  $H_{gas}$  from the above formula. This remains a possibility, but there are probably significant problems in assuming that  $H_{rock}$ ,  $T_{rock}$  and  $H_{gas}$  are constant.

It was then realised that the heat transport from the rock into the tank should really be calculated by solving the heat equation in the surrounding

rock. This must be done in a time dependent fashion, as the thermal diffusivity of soil is such that the temperature profile will vary on the annual timescale. It is likely that during the course of the winter the temperature of this rock constantly decreases, not only due to cooling from above, but also by losing heat into the water tank. This transfer of the stored heat in the rock into the water tank is in fact the main source of heating to the water tank, so the calculation of the surrounding thermal field is likely to be quite important in determining whether sufficient extra heat can be provided to compensate for the envisaged heat loss to the pipes and cups.

The surrounding temperature field is found by solving the heat equation

$$T_t = \kappa_{rock} \nabla^2 T,$$

in the region around the tank, subject to boundary conditions

$$T = T_a(t) \quad \text{on the surface (away from the tank),}$$

$$T = T_c(t) \quad \text{at the edge of the tank.}$$

The temperature of the soil will respond very little to diurnal changes in air temperature, so  $T_a$  can be taken to be the average daily temperature. The tank temperature  $T_c$  is unknown and must be found as part of the solution. It is found from knowing the heat flux into the tank, which is

$$Q = \int_{\partial D} k_{rock} \frac{\partial T}{\partial n} ds,$$

where  $\partial\Omega$  is the tank - rock interface, and  $n$  is the normal pointing out of the tank.

The temperature in the water is therefore governed by the equation

$$\rho c V_{tank} \frac{dT_c}{dt} = Q - L,$$

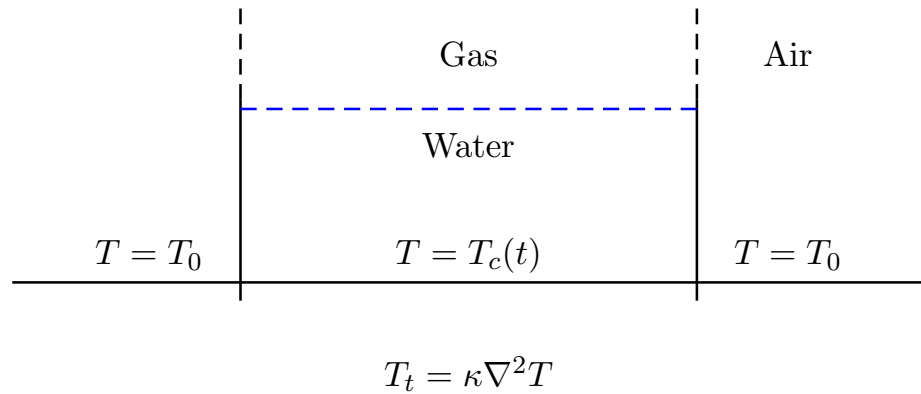
in which  $L$  is now the lumped heat loss. If no antifreeze pumping were done, this would simply be the heat loss to the gas, which could be written as  $L_{gas} = H_{gas}(T_c - T_g)$ . However, when water is being pumped to the cups, this also takes away heat from the tank and this must be included in  $L$ .

The problem of finding the heat flux from the earth and the tank temperature is therefore time dependent, and is coupled to the problem of heat loss from the cups and pumping equipment. This will require a numerical solution, which has not yet been done.

The fact that the tank temperature is uniform (the water is well mixed) in winter indicates that the exact location of the inlet for the antifreeze pumps is not important. It is however suggested that the pump intake be placed not immediately next to the tank sides or bottom, where there

may be boundary layers whose structure has not been examined, and in which the temperature may vary. Since the pipe cannot intrude through the sides of the tank (because the steel cylinders must be able to retract), it was suggested that it should be placed along the bottom of the tank, and the intake drawn from somewhere in the middle of the tank, perhaps a metre or two above the bottom.

- (2.3.2) For an above-ground tank a similar problem needs solving, as shown in figure 6. This problem was not studied further, but it was noted that, unless  $T_c = T_0$ , the net heat flux into or out of the tank must be infinite in this idealised model; moreover, if a dimensionless heat transfer coefficient is introduced into the boundary condition, the heat flux will tend to infinity as the logarithm of this coefficient.

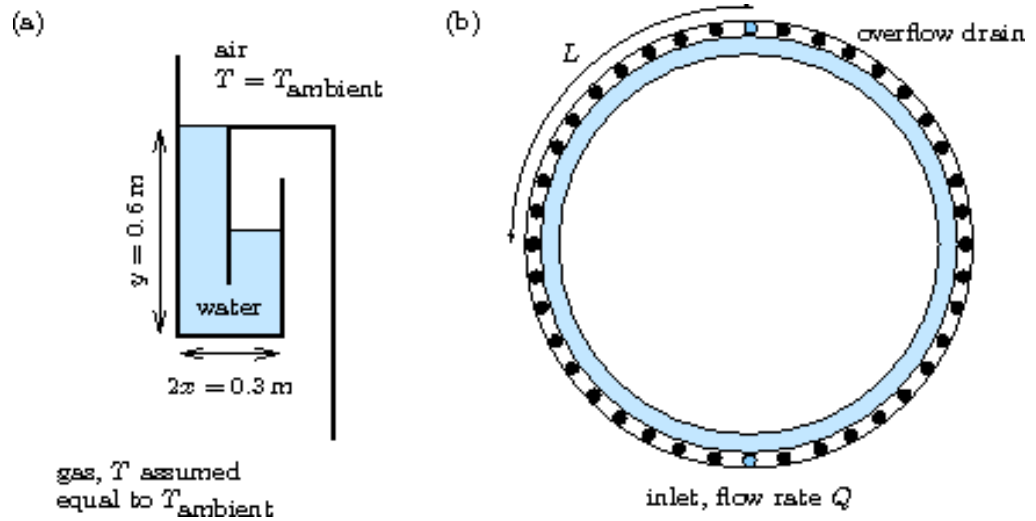


**Figure 6:** Above-ground tank geometry and model.

## 2.4 The water in the cup seal

- (2.4.1) The “standard” cup we consider is shown in Figure 7. The circumference of the cup is  $4L = 60$  m, the depth is  $y = 0.6$  m, and width is  $2x = 0.3$  m. The cross-sectional area is denoted  $A$ .

Each cup is served by two inlets of volume flow rate  $Q$ . At present a single pump of flow rate  $0.01 \text{ m}^3 \text{ s}^{-1}$  is used to supply all inlets (two on every cup of the gasholder). We assume three cups for our standard tank so  $6Q = 0.01 \text{ m}^3 \text{ s}^{-1}$ . In principle more pumps could be installed and for comparison purposes we also consider the case where each inlet has a dedicated pump so  $Q = 0.01 \text{ m}^3 \text{ s}^{-1}$ . Each cup has approximately 50 equally spaced overflow drains. For simplicity we assume a spacing of  $\Delta x = 1$  m.



**Figure 7:** The standard cup; cross-section (left) and plan (right).

The inflow temperature of the water to the cup is assumed to be  $T_{\text{in}} = 5^\circ\text{C}$  and the “worst case” ambient temperature is  $T_{\text{ambient}} = -10^\circ\text{C}$ .

The heat loss coefficient per unit length of the cup is estimated based on a coefficient of  $H_s = 10 \text{ W m}^{-2} \text{ }^\circ\text{K}^{-1}$  for the sections of the cup exposed to the gas either directly or through the steel and  $H_a = 60 \text{ W m}^{-2} \text{ }^\circ\text{K}^{-1}$  for the water exposed to the air. We have ignored conduction through the steel. This gives a heat loss per unit length of  $H = (2y + 3x)H_s + xH_a$ . For the standard cup we find  $H \approx 26 \text{ W m}^{-1} \text{ }^\circ\text{K}^{-1}$  (consistent with the report provided by National Grid).

(2.4.2) As a first estimate we have a heat flux into a cup  $\rho Q c_p (T_{\text{in}} - T_{\text{out}})$  and a rate of heat loss of  $2LH(T_{\text{in}} - T_{\text{ambient}})$ . For the present supply rate to the inlets, we have a temperature drop of approximately  $1.7^\circ\text{C}$ . With the installation of additional pumps, the temperature drop would be approximately  $0.3^\circ\text{C}$ .

(2.4.3) The fact that drainage from the cups is distributed increases the temperature drop. We assume that inlets are directed vertically downwards (so there is no net supply of momentum to the water in the cup) that all the water in the cup rapidly begins to flow and is reasonably well mixed.<sup>1</sup>

Over a quarter circumference, conservation of mass implies

$$\rho u A = \frac{1}{2} Q - \sum_i Q_i \Theta(x - x_i),$$

where  $u$  is the average velocity,  $\Theta$  is the step function,  $x$  is the distance from the inlet and  $x_i$  is the location of the  $i^{\text{th}}$  drainage point. As a first

<sup>1</sup>The Richardson number of the flow is of the order of  $10^{-2}$  and so we may ignore the effects of temperature-induced density differences in the water. The Reynolds number is of the order  $10^3$ .

approximation we assume that the outflow is evenly distributed between the drains (a hydraulic model suggests this may be correct),  $Q_i = Q/N$ , where  $N$  is the number of drains.<sup>2</sup>

Conservation of heat gives

$$\frac{d}{dx}(\rho u A c_p T) = \frac{1}{2} Q c_p T \delta(x) - \sum_i c_p T Q_i \delta(x - x_i) - H(T - T_{\text{ambient}}),$$

where  $\delta(x)$  is the delta function. This reduces to

$$\rho u A c_p \frac{dT}{dx} = -H(T - T_{\text{ambient}}).$$

Thus the temperature in the section between the  $I^{\text{th}}$  and  $I + 1^{\text{st}}$  drains is

$$T - T_{\text{ambient}} = (T_{\text{in}} - T_{\text{ambient}}) \exp \left[ -\frac{H}{Q c_p} \sum_{i=0}^{I-1} \frac{\Delta x}{1/2 - i/N} - \frac{H}{Q c_p} \frac{x - x_I}{1/2 - I/N} \right].$$

Temperature profiles are plotted in figure 8, assuming that the stagnation point where the flow from two inlets converges occurs at an overflow. For a dedicated pump for each inlet, the temperature loss is less than 1 °C. At the present supply rate to each inlet, the temperature loss is 6 °C for our “standard” cup parameters and the water in the final section falls below 0 °C.

- (2.4.4) The assumption that stagnation points occur at drains in the example above means that they are not a significant problem in that particular case. However it is more likely that a stagnation point develops between two drains and that an entire section becomes stagnant (and hence cools much more substantially). Noise in the system may be sufficient to eliminate these regions, however a more effective method might be to angle all inlets in the same azimuthal direction to generate a strong swirl. Alternatively, varying the supply rate to the individual inlets in time would move stagnation points and potentially avoid freezing. Varying the ratio of the volume fluxes of the two inlets by approximately 15% would move the stagnation point between adjacent overflows. Increasing this to 30% would move it between three neighbouring overflows.

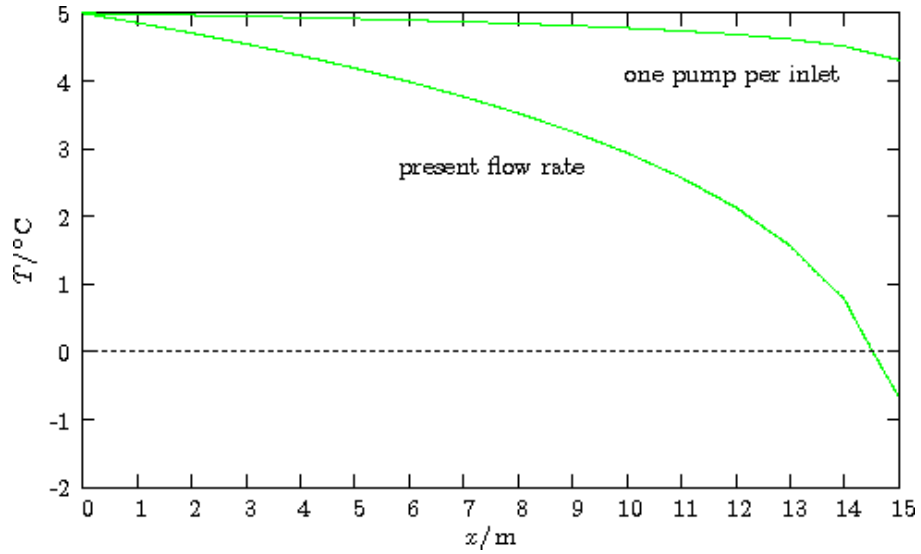
## 3 Discussion and Conclusions

### 3.1 Discussion

- (3.1.1) The possibility of avoiding freezing depends on there being sufficient enthalpy in the tank water (and the soil) and on these elements having a

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<sup>2</sup>Despite the drains being relatively dense compared to the inlets, approximating them as continuous drainage gives a poor approximation close to the stagnation point and we retain the discrete description.



**Figure 8:** The temperature profile along the cup for a each inlet being supplied by a single pump and each pump supplying six inlets.

large thermal capacity. This seems to be the case, but tank water temperature will be affected by long pumping hours and by multiple holder lifts in a cold spell. The tank water temperature and its stability is crucial to the feasibility of operating without external heating. If a cold spell follows normal weather, there is several days' worth of heat which can be circulated.

- (3.1.2) It is also of crucial importance to keep the water circulating in the cup and avoid either stagnation points of flow “short-circuiting” rapidly to the lower cup.
- (3.1.3) The withdrawal point for tank water needs to be well away from thermal boundary layers in the tank. Otherwise, there is a risk that the hose inlet temperature is too low.
- (3.1.4) For 60m tanks the timescales will be approx 3 times greater than those for a 20m tank as cup and tank surface heat losses are (approximately) proportional to  $R$  whereas tank water thermal capacity is proportional to  $R^2$ .
- (3.1.5) Radiation has not been systematically included in the above analysis. On a clear cold night, the effective sky temperature for black body heat interchange could be 210K. This would lead to a radiant heat loss from the surface of the cup water of 6.4kW, if this had a direct view of the sky.
- (3.1.6) Aboveground tanks are not closely connected to the thermal capacity of the soil. The tank water temperature would be expected to vary more



with the weather and there is very little possible heat input from the soil. Without an external heat source, these tanks would be much more vulnerable.

### 3.2 Points for further experimental consideration

- (3.2.1) In situ assessment of tank temperature is important; measurements of the tank temperature should be taken over a long time when deciding the heating strategy for the gasholders.
- (3.2.2) Multiple hose inlets to the cup will allow most uniform temperature scenarios. Also, they will provide most flexibility to pump tank water where it is most needed, i.e. to the windward side. It will also avoid short circuiting of warmer water into downcomers. Another strategy may be to alter the flowrate in a cyclical way to ensure cold and near stagnant spots are not fixed, or to help establish a circulating flow around the entire cup.

## References

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