A theoretical and experimental analysis is carried out of radiative heat transfer in the coaxial geometry of evacuated tubular solar collectors, with emphasis on the operation at high temperatures for industrial process heat. Radiative heat transfer is usually described using the grey body approximation which strictly does not apply to selective absorber coatings. Also, the outer envelope of a tubular collector is partially transmitting in the thermal infrared, especially when constructed of borosilicate glass for durability at high temperatures. We evaluate the likely effect of such transmission by developing analytic expressions for the heat transfer through the outer glass envelope and show the equations of heat transfer no longer follow a simple form so that an effective emissivity for the system as previously used cannot be defined. To test all approximations in practice, an experiment is performed using an evacuated solar collector manufactured in the 1980s by the Nitto Kohki company in Japan using the effective emittance approximation to determine the heat transfer characteristics in a typical operating conditions using net radiative heat flows both inwards and outwards. The effective emittance method in practice enabled a good fit to temperature-time data for cooling and heating of the inner tube. The results confirm that the effective emittance method can be used in practical situations with typical glass wall thickness, even though there is radiative transfer through the outer envelope. The work has verified the stability of the vacuum in this type of evacuated solar collector as the aged tube still functions to a high standard in terms of maintaining a low emissivity and good level of vacuum after approximately 40 years in storage conditions.

Keywords Effective Emittance · Evacuated Tubular Solar Collectors · Aged Study · Coaxial Geometry · Radiative Heat transfer · Cooling Curves

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1 Introduction

The evacuated tube solar collector is now common around the world as a source of heat for domestic [1] and industrial [2] purposes. The design originating from the University of Sydney [3] consists of an all-glass single-ended coaxial design with a sputtered selective surface consisting of a copper base layer and stainless steel-carbon absorbing layer on an inner tube and a vacuum seal on an outer envelope. The tubes are fused together at one end where the interior volume of inner tube is open to the air and both tubes have a hemispherical termination at the other end. This design was manufactured under licence to the University of Sydney in the 1980s by the Nitto Kohki company and subsequently by the Shiroki Company both of Japan. The construction material is borosilicate glass with an outer envelope of wall thickness in the range 1mm-3mm. In operation, the inner tube contains a heat transfer medium such as water [4], air or other fluid, that may fill the inner tube or be contained in metal pipes thermally connected to the interior surface of the inner tube [1][5]. One of the applications of the evacuated tube collector is for producing high temperature fluids such as air [6] and superheated water. Operation at high temperatures increases the amount of heat transmitted through the borosilicate glass transparency windows and requires that this effect be taken into account in heat loss calculations.

A considerable amount of research work has been done on evacuated solar collectors to evaluate their heat extraction efficiency in the context of their intended application using experimental and numerical methods [7][11]. Less has been done however, in the exact analysis of the radiative heat transfer between the inner and outer tubes, where it is normally assumed that an effective emittance can be used to describe radiative transfers between the inner and outer envelopes. It is assumed that the effective emittance takes into account the individual emittances of the two surfaces. Furthermore, while there has been a study of the in-service performance of these tubes, specifically, looking at the degradation of the vacuum due to atmospheric permeation and outgassing of selective coating [12], this was an accelerated aging study rather than a real-time study. The coaxial geometry has also been used to determine a value of the total hemispherical emittance of sputtered copper films [13]. When used for this kind of fundamental measurement, careful account must be made of the outer envelope contributions to the radiative heat transfer.

There are two areas where the conventional approach to calculating radiative heat transfers in the evacuated tubular collector geometry could be deficient. First, the grey body approximation, which allows the radiative heat transfer to be written as a Stefan Law expression consisting of the difference of the fourth powers of the inner and outer tube temperatures, is subject to challenge. The surface coating on the inner tube has an absorbance and therefore an emittance that are spectrally-dependent and therefore the inner tube is far from a grey body, having a strong dependence of the emittance on wavelength in the region near 1 to 5 micrometres where the emittance changes from very high to relatively low. The second area where existing treatments are potentially deficient is associated with the known transparency of the outer envelope to the thermal infrared, which is neglected in previous studies.

In this work, we undertake a theoretical investigation where we develop equations based on the grey body approximation but allowing transparency of the outer envelope. We estimate the effect of the transparency by estimating the error that would arise from the conventional assumption of an effective emittance. Then we test the combined effect of both areas of approximation in a practical situation, by undertaking an experimental study of the heating and cooling curves using an evacuated collector manufactured by the Nitto Kohki company in the 1980’s. The experiment was performed on a tube selected randomly from a batch that has been in long term storage since it was manufactured. The experiment has an additional purpose to verify the long term durability of the evacuated tube concept in terms of the stability of the selective absorber’s low emittance and of the ability of the all-glass envelope to maintain a sufficient level of vacuum to avoid conductive heat transfers. The experiment was performed by allowing a fluid within the tube to heat up or cool due to a difference in ambient temperature and then measuring the temperature-time relationship of the inner and outer surfaces.

2 Theory

2.1 Background

The heat transfers in an evacuated tube collector take place between two cylindrical coaxial bodies, the inner one at temperature $T_1$ and the outer one at temperature $T_2$. It will be assumed that the coaxial cylinders have infinite length along their axis as the diameter is much smaller than the length. The cases where the two bodies are non-transmitting to thermal infrared radiation, has been considered for applications in solar collectors and vacuum glazings [7][13]. The radiative transfer between two coupled, infinitely extended planes with opaque surfaces has been discussed in textbooks such as those of Holman [14] and Howell et al [15]. Applying the grey body approximation in which the total hemispherical absorptivity is assumed not to depend on the spectral distribution of the incoming radiation, the rate of radiative heat energy transfer per unit area per unit time between the two surfaces is given by the Stefan radiation law in
the form

\[ Q_1 = \varepsilon_{\text{eff}} \sigma (T_1^4 - T_2^4), \]  

where \( T_1 \) and \( T_2 \) are the (absolute) temperatures of each surface and \( \sigma \) is the Stefan Boltzmann constant. This equation relies on the assumption that the radiating bodies are ‘fully coupled’ meaning that emitted photons from one surface are always intercepted by the other, and the effective emissivity is given by

\[ \varepsilon_{\text{eff}} = \left( \frac{1}{\varepsilon(T_1)} + \frac{1}{\varepsilon(T_2)} - 1 \right)^{-1}, \]  

where \( \varepsilon_1(T_1) \) and \( \varepsilon_2(T_2) \) are the total hemispherical emissivities of each body at their respective temperatures. In the case where the two bodies are infinite coaxial cylinders, the bodies are not fully coupled and some of the radiation emitted by the outer body does not intercept the inner. The effective emissivity in the coaxial geometry then becomes

\[ \varepsilon_{\text{eff}} = \left( \frac{1}{\varepsilon(T_1)} + \frac{R_1}{R_2} \left( \frac{1}{\varepsilon(T_2)} - 1 \right) \right)^{-1}. \]  

Where \( R_1 \) and \( R_2 \) are the radii of the inner and outer coaxial tubes respectively. Both formulas are derived by considering all possible reflections of emitted photons and summing the contributions of all reflections to infinity. This expression has been used before in the analysis of evacuated solar collectors [7].

Equation (3) applies only to radiative transfer between non-transmitting surfaces. In the case of an evacuated collector, the outer surface may be partially transmitting and as a consequence, the concept of an effective emittance is no longer applicable, and the heat transfer follows a different form as shown below. In this paper, we derive the equations for the two-body coaxial geometry where the outer cylinder is partly absorbing and partly transmitting, for the thermal radiation i.e. an infrared ‘window’. We show that the resulting expression is different to that given in Equations (1) and (3) and involves the transmittance of the outer envelope, which has direct application to evacuated solar collectors.

### 2.2 Effect of transmission through the outer envelope

The expression for \( Q_r \), the net radiation out of the inner tube is derived while accounting for the transmissive outer tube. The grey body approximation is used, meaning the total hemispherical emissivity equals the total hemispherical absorptivity, \( \varepsilon = \alpha \). Referring to Figures 1 and 2, with body 1 being the inner body and body 2 being the outer body, the reflectivities are given by \( \rho_1 = 1 - \varepsilon_1 \) and \( \rho_2 = 1 - \tau_2 - \varepsilon_2 \), where \( \tau_2 \) is the average transmissivity of the outer glass layer. Finally, we define \( f = R_1/R_2 \), where \( R_1 \) and \( R_2 \) are the radii of the inner and outer tubes respectively.

To determine \( Q_r \) we sum contributions from rays which are emitted from either inner tube or outer tube and then undergo an infinite number of reflections. First, considering a ray with intensity \( \sigma A_1 T_1^4 \) emitted from the inner tube (Fig.1) at point \( A \), its intensity will gain a factor of \( \varepsilon_1 \) due to the emission and undergo a reflection at \( B \). At each reflection from the outer tube, some of the radiation is absorbed or transmitted, the rest is reflected. This reflection may either intercept the inner tube with a fraction \( f \) or the outer tube again with fraction \( (1 - f) \). Both these rays will impinge upon the outer tube again upon reflections with decreased intensity. Accounting for an infinite number of reflections, the radiation re-absorbed by the inner tube after being emitted from it is given by:

\[ Q_{R_1} = -\alpha A_1 \sigma (f \rho_2 \varepsilon_1 \rho_1^2 + f \rho_2^2 \varepsilon_1 F_1 + f \rho_2^3 \varepsilon_1 F_1^2 + ...) T_1^4 \]

\[ = -\frac{f \rho_2 \varepsilon_1^2}{1 - \rho_2 F_1} A_1 \sigma T_1^4. \]  

Where a geometric series is used and \( F_1 = 1 - f + f \rho_1 \).
Figure 1: Possible paths of a single ray of radiation initially emitted from the surface of the inner tube at point $A$. Upon reflection from the outer tube at $B$ the ray can either coincide again with the outer tube at $D$ or undergo another reflection at $C$ to later coincide with the outer tube.

The contribution due to radiation initially emitted from the outer tube is now examined. Referring to Figure 2, a ray with intensity of $\sigma A_2 T_2^4$ is emitted from point $A$ and will undergo the same pattern as previously discussed. Hence the total absorbed power into the inner tube by rays emitted from the outer tube is given by:

$$Q_r = - \alpha_2 A_2 \sigma (f \varepsilon_2 + f \rho_2 \varepsilon_1 F_1 + f r_2^2 \varepsilon_1 F_1^2 + ...) T_2^4,$$

where again a geometric series is used. Accounting for the radiation emitted by the tube initially given by $\varepsilon_1 A_1 \sigma T_1^4$, summing the contributions given by Equations (4) and (5), the result is:

$$Q_r = A_1 \sigma \left( \frac{\varepsilon_1 (1 - \rho_2 F_1)}{1 - \rho_2 F_1} - f \rho_2 \varepsilon_1 \right) T_1^4 - A_2 \sigma \left( \frac{\varepsilon_1 \varepsilon_2}{1 - \rho_2 F_1} \right) T_2^4.$$  \hspace{1cm} (6)

Note that this expression cannot be reduced to the form of Equation (1). In the case that $\tau_2 = 1 - \varepsilon_2 - \rho_2 = 0$, however, Equation (3) is obtained. If further $f = 1$, Equation (2) is obtained. Hence the expression matches known limiting cases. Since the average emittance of glass is greater than 0.85 at 300K [16], and the reflectance of glass is less than 0.1 for most wavelengths [17], we can approximate the average transmissivity of the outer layers as zero. The validity of this approximation is further discussed in Section 5. This approximation greatly simplifies the equations and allows the use of the effective emissivity to determine the heat transfer between tubes as assumed previously.
Figure 2: Possible paths of a single ray originating from the outer tube at point A. As in Figure 1, the ray can take various paths within the tube and will undergo, in principle, an infinite number of reflections.

2.3 Time dependence of temperature including conductive transfers

If the tube is filled with a fixed mass of a fluid, the effective emissivity is determined by considering the total heat lost or gained by the fluid. The heat energy per unit time lost or gained by the fluid is given by:

$$\frac{\partial E}{\partial t} = m_f c_p (T_f) \frac{\partial T_f}{\partial t}$$  \hspace{1cm} (7)

where \(m_f\) is the mass of the contained fluid and \(c_p (T_f)\) is the specific heat capacity at constant pressure (water \[18\] and ethanol \[19\] for this experiment) as a function of temperature. It is assumed that the change in volume of the fluid throughout the experiment is negligible. The total energy lost by the fluid is the energy radiated \(Q_r\), the energy conducted through the insulating cap, \(Q_i\), and the energy conducted through the glass envelope \(Q_g\) (Fig.3), taking the conduction to be linear with respect to the temperature difference an energy balance gives the differential equation:

$$m_f c_p (T_f) \frac{\partial T_f}{\partial t} = \varepsilon_{eff} A_1 \sigma (T_g^4 - T_f^4) + c_1 (T_g - T_f) + c_2 (T_a - T_f).$$  \hspace{1cm} (8)

This equation has three free parameters: \(\varepsilon_{eff}\), \(c_1\) and \(c_2\). The effective emissivity will be the same across all experiments since it depends only on the tube used. The same is true for \(c_1\), which is the conduction between fluid and the outer glass layer. This is a complex heat transfer problem, but is only geometry dependent, so it will also be the same across all experiments. The coefficient \(c_2\) relates heat transfer between the ambient air and the fluid, again this is a complex problem since different convection loops will form depending on the experiment (Fig.4), furthermore, the convection loops depend on the viscosity and density of the fluid used (water or ethanol), hence a different coefficient \(c_2\) is needed depending on both the fluid used, and whether the fluid is being cooled or heated in the experiment.
Figure 3: Heat escapes the water in three main ways: Radiation through the vacuum envelope \((Q_r)\), conduction through the insulating cap \((Q_i)\), and conduction through the glass around the envelope and through residual gas in the vacuum \((Q_g)\). \(T_a\) is the temperature of the ambient air, \(T_f\) is the temperature of the fluid within the tube, and \(T_g\) is the temperature of the outer glass layer.

The above differential equation is fitted to the experimental temperature-time cooling curves through least squares to determine the optimal parameters, with \(\varepsilon_{eff}\) and \(c_1\) being the same across all experiments, and \(c_2\) being different for each case of water, or ethanol filling, heating and cooling (four cases total).

The cooling curves (temperature vs. time) were used to calculate the heat radiated between the selective coating on the inside of the vacuum envelope, and glass on the outside of the envelope. This allows a measurement of the effective emissivity of the system. As well as the heat radiated in the vacuum envelope, there are conductive heat transfers from the water. The main paths of conduction which cannot be neglected are those through the insulating cap, and between the layers of the glass envelope due to possible residual gas (Fig. 3).

The temperatures being measured are the temperature of the fluid, and the temperature of the outside glass surface, we must make the assumption that these measured temperatures are sufficiently close to the temperature of the actual radiating surfaces.

Figure 4: Different convection loops will form within the fluid due to the heat lost through the insulating cap at the top.
3 Experimental Methods

For the experiment, the evacuated tube was set up vertically away from direct sunlight and strong light and heat generating sources and was placed close to the ground. A schematic of the experiment is shown in Figure 5.

![Figure 5: Experimental setup used to obtain cooling curves for the evacuated solar collector](image)

Two versions of the experiment were done; one where the net heat radiation flows from the working fluid to the outside ambient air, the other where the net heat radiation flows in the opposite direction from the ambient air to the working fluid. Water and ethanol were both used as working fluids for additional confirmation and for estimating errors in the experiment.

When performing the experiment with the heated working fluid, the tube was filled with boiling water to preheat the tube. Preheating the tube means less heat in of the working fluid will be lost to the mass of the glass, and the cooling curves will be more representative of a radiation curve. After preheating, the tube was emptied and filled with the working fluid at temperature, then sealed with an insulating cap made from EVA foam. Two K-Type thermocouples were placed within the tube to measure the internal water temperature, and two were placed on the outside surface of the glass to measure the glass temperature. A single K-Type thermocouple was also used to measure the ambient temperature of the air surrounding the tube. Measurements were read every 5 seconds by a data logger, using MAX31855 temperature sensors, which provide an accuracy of up to 0.25°C. The experiment is run over at least a 72 hour period, to ensure that thermal equilibrium has been reached with the ambient temperature. All data analysis and calculation using MATLAB.

In order to let the net heat flow in the opposite direction for the second experiment, a copper cylindrical copper tube was placed around the collector and sealed from both ends using densely packed open cell foam. The copper tube was then heated with a ribbon heater and after being left to may result in different background radiation entering the tube through the outer envelope. This result was deemed negligible after the cooling experiment was conducted with the copper tube around it, and the same results were obtained.

The cooling experiments were done from a temperature just below the boiling point of the fluid to ambient temperature of the room, i.e. heated fluid was placed inside the tube and allowed to cool. For the heating experiments, chilled fluid above the freezing temperature was filled in the tube and allowed to heat to a temperature below its boiling point. That is, the copper heating tube was set to heat the outer glass layer to a temperature a small amount below the boiling point of the fluid. It is important that no phase change of the fluid occurs during the experiment.

4 Results

When fitting the experimental measurements, the initial sections of data corresponding to the start of the experiment, and the end section corresponding to the system reaching equilibrium were removed. This avoids errors that may arise due to initial turbulence in the fluid when preparing the experiment, and environmental noise that may arise when the system is in thermal equilibrium. The fitting parameters used are presented in Table 1.
Table 1: Fitted parameters in Equation (8) to the measured data using least squares

|                 | Water - cooling | Water - heating | Ethanol - cooling | Ethanol - heating |
|-----------------|-----------------|-----------------|-------------------|-------------------|
| $\varepsilon_{e,f}$ | $(7.11 \pm 0.03) \times 10^{-3}$ | 0.00            | $(5.27 \pm 0.01) \times 10^{-3}$ | $(5.30 \pm 0.02) \times 10^{-3}$ |
| $c_1$          | 0.00            | 0.00            | 0.00              | 0.00              |
| $c_2$          | $(1.29 \pm 0.01) \times 10^{-3}$ | $(1.29 \pm 0.01) \times 10^{-3}$ | $(3.77 \pm 0.02) \times 10^{-3}$ |

Knowledge of the parameters in Table 1 allows one to determine the transient behaviour of the temperature of the fluid given the initial temperature and the boundary condition, in this case, the boundary condition is the outer glass temperature. The results for solving Equation (8) numerically with the fitted parameters for each experiment case are shown in Figure 6. In the heating experiments (Figures 6 (b) and (d)) the glass temperature exhibits oscillations, particularly with in the ethanol experiment, a result of the controller used to maintain the outer glass envelope at the elevated target temperature.

![Figure 6](image-url)

Figure 6: Measured results compared with numerical solution to Equation (8) using fitted parameters in Table 1. (a) and (b) are the cooling and heating experiments respectively with water, (c) and (d) are the cooling and heating experiments respectively with ethanol.

The Root Mean Squared (RMS) deviation of the fitted curve from the measured data from curves (a), (b), (c), and (d) are 0.0997, 0.243, 0.0789, and 0.398 respectively. Since Equation (8) is linear in the parameters $\varepsilon_{e,f}$, $c_1$, and $c_2$, the coefficient of determination $R^2$ can be used as a goodness of fit. The coefficient of determination for the fits (a), (b), (c),
and (d) are $0.9999, 0.9998, 0.9999, 0.9992$ respectively. The total hemispherical emissivity of the selective coating can be estimated given the effective emissivity and the total hemispherical emissivity of glass. Using $\varepsilon_2 = 0.88 \pm 0.02$ for borosilicate glass at 300K [16], $r_1/r_2 = 0.86 \pm 0.02$ for the evacuated tube used in the experiment, and Equation (3), we find the emissivity of the selective coating on the inner tube to be $\varepsilon_1 = 0.072 \pm 0.001$.

### 5 Discussion

The derivation of Equation (6) shows that not all problems relating to radiation heat transfer between two bodies can be related in the form of Equation (1), but is true when the transmittance of the glass outer layer is zero. The experiments conducted here show that making the approximation that the transmittance of the outer glass layer is zero is appropriate, given the good fit achieved using the effective emissivity equations. This may not always be true, however. Nonetheless, in the case of predicting the performance of these tubes, the approximation is appropriate, and most studies tend to use these equations. It is important to understand the underlying reality. An interesting point to note about the differences between Equations (1) and (6) is that there is no longer symmetry in the absolute value of the heat under the exchange of $T_1$ and $T_2$. That is, the magnitude of the heat transferred will depend on which surface is hotter. As a result, the transient behaviour of the tube depends on if it is being heated or cooled. In the practical operating regime for these tubes, which is approximately between the ambient air temperature and boiling point of water, this is not an issue. However, in cases where $T_2 \approx 0$ is no longer a good approximation, the heating and cooling characteristics will no longer be symmetrical. This effect comes about simply due to the transmitting properties of the outer glass layer and the grey body approximation.

The uncertainty on all the fitting parameters is small, and the RMS deviation is also smaller than 0.1 in all cases. The RMS deviation is smaller and $R^2$ is higher in both heating experiments. The instability of the outer glass temperature due to the heating controller in the heating experiments will cause such errors, as well as the asymmetry predicted by Equation (6) not being accounted for in Equation (8). Nonetheless, the resulting deviation is still on the order of the accuracy of the temperature sensors and the coefficient of determination is still very high and is hence acceptable.

Interestingly, from Table 1, the fitting process found that $c_1 = 0$. This being the coefficient associated with conductive heat transfer through the glass indicates that a negligible amount of heat energy was conducted through the glass during the life of the experiment. Using each term in Equation (8) and the fitted parameters in Table 1, the heat transferred per unit time throughout the experiment for each mode of heat transfer is shown in Figure 7. In all cases, especially far away from equilibrium, the total heat transfer is dominated by radiative heat losses.

The fact that $c_1 = 0$, and radiation dominated the heat transfer indicates that very little heat was lost through the outer envelope. If there was degradation of the vacuum, there would be a larger heat loss term proportional to $(T_g - T_f)$. These results indicate that little vacuum degradation has occurred and the tube is quite stable. This minor degradation of the vacuum is consistent with the results obtained from the accelerated aging study by S.P. Chow et al. [12].

The measured emissivity of the tube tested in this work when it was brand new is unknown. However, it is of the Nitto Kohki type which was mass produced in the 1980s. The emissivity of the selective surface for these tubes when new was approximately 0.05 at 100°C [20,21]. The emissivity decreases with temperature, so it is expected that the average emissivity over the tested range is lower when the tube is new. S.P. Chow reports an emissivity of 0.05 ± 0.006 before the aging process, and an emissivity of 0.065 ± 0.002 after aging at 430°C for approximately 10 000 hours. These emissivities apply at 100°C. The emissivity of the surface calculated in this paper was 0.072±0.001, which is higher than the emissivity after the accelerated aging, an expected result since the tube used here is approximately 40 years old. The emissivity calculated here is an average total hemispherical emissivity over a temperature range of approximately 20°C to 70°C. Hence for this particular tube the emissivity at 100°C is expected to be higher than the quoted value. Nonetheless, the results demonstrate the excellent ability for these tubes to maintain a good vacuum and a still functional low emissivity coating. Figure 7 demonstrates that the majority of the heat lost by the tube is through radiation, and some through the insulating cap. In a practical situation, this tube is still effective. The Nitto Kohki tube used here is fitted with a getter which will help in removing small amounts of air.
Figure 7: Different heat loss components for each experiment over time. (a), and (b) correspond to the cooling and heating experiments respectively with water, (c) and (d) correspond to the cooling and heating experiments respectively with ethanol.

The above results rest on the assumption that transmitted radiation through the outer envelope can be safely neglected. This assumption only holds if a negligible amount of the emitted radiation from the inner tube is transmitted through the outer tube, likewise only if a negligible amount of radiation from external sources enters. In the case of the experiment, the tube was shielded from the sunlight or any other sources, however the surrounding air is a radiation source that can be approximated as a blackbody at room temperature. Looking at the fraction of blackbody radiation transmitted can serve as an upper limit. Figure 8 shows a normalized blackbody distribution, along with the transmittance of borosilicate glass. The transmittance used here is for thinner samples than those found in the tubes, however this serves as a good qualitative indicator. In the case of the thicker glass, transmittance will be even less due to greater absorption of light passing through.

At lower temperatures, near 300K, the peak of the blackbody curve lies near minima of the transmittance, i.e. smaller fractions of energy are transmitted through the transmitting layer. Furthermore, at these lower temperatures, the total power is relatively small compared to other energy transfers within the system. Hence the results above are well described using the effective emittance approximation. As the temperature increases, the peak of the blackbody distribution nears a transmitting window at the lower end of the near-infrared region, and the total power being radiated is larger. Here transmittance effects become less negligible. This has practical applications where the temperature of the inner tube is high, as in the delivery of process heat. In this case, the transmitting layer should be accounted for using Equation (6).
Figure 8: Blackbody distribution superimposed with spectral hemispherical transmittance at 300K (a) and 600K (b), dashed lines indicate the relative transmitted power of the distribution

It is worth noting the type of aging which is being studied here. The tube used in this experiment had been in long term storage and was not subject to the expected operating conditions. In reality, a tube in use will be exposed to sunlight and will undergo daily thermal cycles. Thermal cycling can damage the thin film surface due to differences between the thermal expansion of the glass and sputtered materials, exposure to sunlight will result in photon assisted decomposition of the selective surface.

From the results in the paper, some comments can be made on the practical aspects of the more fundamental physics involved in radiation heat transfer. The validity of Kirchhoff’s law has long been the subject of debate. Kirchhoff’s law relates the emitting and absorbing abilities of a body. Simply, Kirchhoff’s can be stated by saying that the total hemispherical absorptivity of a material equals its total hemispherical emissivity, i.e. $\varepsilon = \alpha$. The debate is generally when this is true, and if it is true at all. The aim of this paper is not to go into the details of this issue; however, we have taken Kirchhoff’s law to be true in the assumption that all surfaces are diffuse grey bodies. This means it was assumed that the emissivity was independent of direction and wavelength. The unique properties of the selective coating mean it is far from a grey emitter, however the results have shown that using this assumption can provide excellent predictions for the radiated heat in the evacuated collector in this temperature regime. Practically, the temperatures tested are the expected operating temperatures of these devices.

6 Conclusions

The evacuated solar collector is an important form of renewable heat generation system and understanding its performance and properties in terms of radiative heat transfer is crucial. This is especially true in the long term. We derived an expression for the radiative heat transfer, showing that the equations closer to reality are more complex when the transmission of the outer layer is accounted for. As a result, there may be an asymmetry in the cooling/ heating of the tube. For practical purposes, it is sufficient to neglect the transmissivity of the layer and use the effective emissivity for calculating heat transfer. Using this approximation, the effective emissivity of the tube system and the emissivity of the coating was calculated. The experiment was found to adhere well to the theory excellently and hence produce an accurate result. Furthermore, the results showed that the tube maintained a stable vacuum after approximately 40 years of storage, and that the selective coating had undergone some degradation, but was still at a functional level. The amount of degradation found was slightly higher than what was obtained in previous accelerated aging studies. The evacuated tubes manufactured by the Nitto Khoki company were found to have good storage life with regards to their vacuum stability and selective coating.
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