SolrHeat Space Heater Thermal Testing (Winter Design Conditions)

M.G. Evans PhD MIEAust
School of Physics and Electronics
The University of New England, Armidale NSW

The SolrHeat space heater is an innovative high-efficiency solar powered heat exchanger designed for space heating in residential dwellings and commercial premises. The heater is a roof-installed ducted heat absorption panel, 1 x 2 m profile, powered by an 80-watt mixed flow fan, producing between 600 and 1000 watts of heated airflow in design (fully installed) winter conditions, the lower value corresponding to light overcast conditions.

Tests were conducted on a prototype in October 2014 and June 2015, to determine:

1. Power rating of the device in warm and cool weather with no building connection
2. Maximum efficiency relative to solar power in ideal conditions

Preliminary tests conducted in 2010 on a full-scale model were numerically modelled using thermodynamic and fluid dynamics theory (Evans, 2010) to determine prospective power and efficiency ratings of a range of design alternatives. Modelling indicated high-efficiency design performance, consistent with the measurement data reported here.

Test Methods

An installation-ready prototype was set up on a ground-level test site with ducting to the 80W inlet fan only. Outlet flows were exhausted directly to atmosphere with no attempt to connect to pressure loss or heat storage conditions in a fully-installed system. The resulting heat flows are ideal, but can be reduced to design conditions.

Table 1: Prototype Specifications

Assembly	Brushed aluminium box with duct fittings, airflow passages, heating plate and shockproof glass cover	2050 x 995 x 152 mm 70 kg total mass
Heating Plate	Thermally-coated extruded aluminium panels with patent fin design for modular assembly	920 x 1973 mm
Glass cover	Toughened low-iron glass	2010 x 953 x 4 mm

The panel was oriented to climatic north (winter solar maximum elevation) and levelled to a 45-degree azimuth (Plate 1). Solar elevation was determined by seasonal data. A cosine to the panel's direct orientation axis was calculated.

An operating maximum airflow was established in the heater at around 11 am on each test day to allow sufficient time for the heater to reach an operating dynamic equilibrium by 12 pm for maximum solar elevation, eastern daylight saving time.

Power and efficiency data were obtained by joint airflow and thermal measurements: air flow rates from the outlet; air temperature measurements; temperature measurements on the heating plate for radiative loss; and temperature gradient measurements at points on the assembly to estimate conductive-convective heat losses.

Air Flow Measurements

Air flows in the outlet duct were measured directly by a calibrated anemometer (Holdpeak model BT-816) arrayed on a radial grid in the outlet duct. The grid provides 17 measurement points to correct for the non-uniform velocity distribution in turbulent airflow. Anemometer measurements were repeated three times across the 17-point grid to test for consistency of the flow distribution and to determine average velocities. An area distribution was estimated by division of the grid into regular area cells.

Volumetric flowrate was determined by summing the product of area and velocity grid values as per the continuity equation $Q = V.A$. Mass flowrate $m. = \rho Q$ was determined by air densities according to temperature as per the ideal gas equation $P = \rho RT$.

The resulting flowrate for the nominal 200 litre / second fan was found to be 140 l/s, indicating a fairly high pressure loss in the heater air flow channels. The channels were designed to operate in a turbulent range to maximise heat exchange, so this pressure loss is not unexpected. Higher losses are expected in ducted airflow through a building connection, suggesting 100 l/s as a feasible target flowrate. This amounts to an estimated 360 cubic metres per hour at 800 watts (heated airflow) for an 80 watt operating input.

Airflow Temperature Measurements

Air temperatures were measured at the fan inlet and duct outlet. Calibrated residential grade thermometers were used, accurate to +/- 0.3 degrees Celsius. Thermometers and duct points were positioned in shaded locations to prevent radiative effects.

In these baseline tests, inflow temperatures were the same as ambient shaded air temperatures. In practice the inflow temperature will be the target minimum indoor temperature of the dwelling or premises, typically 16-20 degrees rising by day.

Test conditions were fine weather, light-moderate, in early October 2014 (6/10, 8/10 and 10/10/14) and cool to cold, fine to light overcast in June 2015 (8/6, 14/6 and 25/6) at Armidale latitude 30.5^{o}S. Maximum solar elevation for these dates (12 pm) varies from 65 to 36 degrees, giving a cosine of 0.94 - 0.99 to the heating plate orientation axis.

Airflow Temperature Results
The typical inflow air temperature during these tests went from 15 to 25 degrees (June to October). The typical heated outflow air temperature went from 25 to 36 degrees.

Table 2: Airflow Temperature Results

Test Point	Typical Temperature	Observed Range
Inflow (June)	15 degrees	9.5 - 16.2
Heated Outflow (June)	25 degrees	18.2 - 26.7
Inflow (October)	25 degrees	23.8 - 27.0
Heated Outflow (October)	36 degrees	34.4 - 38.2

Heating Plate Temperatures

The plate can become extremely hot, over 110 degrees when not in use.

Plate temperature measurements were obtained by direct metal-metal contact with industrial kitchen-grade (Vaccola) thermometers. These were introduced into the box assembly and positioned high and low on the plate, corresponding to inlet and outlet.

The typical inlet-end temperature of the heating plate went from 23 to 48 degrees (June to October). The typical outlet-end temperature went from 31 to 54 degrees.

Table 3: Heating Plate Temperature Results

Test Point	Typical Temperature	Observed Range
Inflow (June)	23 degrees	14.9 - 25.5
Outflow (June)	31 degrees	18.0 - 34.8
Inflow (October)	48 degrees	45.0 - 51.0
Outflow (October)	54 degrees	51.5 - 57.0

The maximum temperature of 110+ indicates a very high thermal absorptivity-emissivity, rated as $e = 0.97$ in the thermal coating product specifications. This large thermal potential is efficiently extracted into the airflow, resulting in significant cooling of the unit (to 57 max) when in use. This gives low radiative-convective heat losses from the unit in operation.

Heating plate temperatures and the ambient temperature are used to calculate radiative losses from the heating plate as per the Stefan-Boltzmann law, as described in Heat Flow Calculations below.

High non-operating temperatures guided a redesign of the prototype to eliminate all thermally unstable construction materials and maximise operating life.

Thermal Gradient (Heat-Loss) Tests

Thermal losses due to conduction and convection of heat from the glass cover and aluminium panels of the assembly were measured directly by purpose-built gradient thermometers positioned at representative measurement points.

A gradient thermometer is a pair of calibrated domestic grade thermometers assembled one on top of the other (Plate 2). The thermometers are calibrated by selection of pairs that show the "same" temperature (+/- 0.2 degC) after 5 minutes in a common warm, sheltered location. The spacing between bulb heads is measured by micrometer (12.0 mm). Further calibration by correcting for mean difference (error) produces highly precise thermal gradient measurements.

The gradient thermometer assembly is mounted against a warm surface such as glass cover or aluminium panel and both temperatures are recorded after at least five minutes equilibration. The temperature gradient is the difference in temperature between the two thermometers, divided by the bulb spacing. Combined with the known thermal conductivity of air at any temperature, this provides a direct measurement of conductive heat flow rates from the warm surface.

A mathematical correction for an exponential (as opposed to linear) gradient is possible, however the correction is small at the design spacing, less than the experimental uncertainty of Δ +/- 0.4 degC.

Convective heat flow rates are small in light-moderate weather and sheltered locations. Convective losses were conservatively estimated for these tests as equal to conduction, ie a small "wind chill" factor was assumed.

Thermal Gradient Test Results

The results of thermal gradient testing are shown in Table 4:

Table 4: Thermal Gradient (Heat Loss) Test Results in Warm Weather (October)

Test Point (Gradient thermometers with 12 mm spacing)	Typical Temperature Difference	Range DegC	Thermal Gradient over 12mm gap (degC/m)
Glass Cover Near Inflow	0.7 degrees	0.2 - 1.1	61
Glass Cover near Outflow	2.0 degrees	1.2 - 2.4	164
Aluminium Side Walls	2.4 degrees	1.3 - 4.0	196
Aluminium Upper End Wall*	2.6 degrees	2.4 - 3.0	217

* The underside or "floor" and lower end wall of the unit were found to operate at the cool air temperature with no thermal gradient, indicating zero heat loss from those surfaces.

Thermal Gradients: Interpretation
The thermal gradients are indicative of the relative heat loss from elements of area on these measurement points. Although the glass cover tends to be quite cool and shallow-gradient, the larger area of glass results in more heat loss overall from that face.

Compared to thermal output, these gradients reflect only small heat losses, of the order of 0.6% of thermal flowrate in the outlet. These figures are for warm conditions. The corresponding heat flow rates are smaller in cool weather, indicating again that operating design performance is highly efficient cool-running.

Larger thermal losses are due to the heating plate operating as a Stefan-Boltzmann black-body. These radiative losses amount to about 7% of outflow thermal flowrate. Glass and aluminium have low emissivities and at these temperatures their radiative emission would be effectively zero.

Heat Flow Calculations

The space heater operates in a state of dynamic equilibrium possibly quite far from thermal equilibrium. There are temperature differences throughout the unit, and emission-absorption processes are not in equilibrium due to heat exchange and extraction.

Direct measurement of insolation by a pyranometer or photometric device is of limited use as the heating plate absorbs differentially over the range of wavelengths of incident light.

The testing method measures solar heat input directly by instrumenting the unit as a bulk pyranometer. That is, measurements are made that enable calculation of all heat outputs, which in dynamic equilibrium must equate to total solar input. The sum of heated air outflow plus radiative, conductive and convective losses are used as an measurement of specific insolation as the cosine of direct sunlight at effective wavelengths.

Thermal Outflow

Heat outflow in the heated air is measured by a temperature difference and mass flow rate. The accepted heat capacity of dry air is 1005 J/kgK, so as the mass flowrate in kg/s and temperature difference in Kelvin (equivalent to Celcius in temperature differences) can be measured, the heat flow rate in J/s can be estimated:

$$Q = \rho q C_p \Delta T \quad \text{J/s = Watt}$$

where ρ = density in kg/m³ ~ 1.0 kg/m³ at 1000 m altitude
q = volumetric flowrate m³/s = 0.14 m³/s
C_p = specific heat capacity = 1005 J/kgK
ΔT = Inflow-Outflow Temperature difference K ~ 6 to 10 K

In addition the density depends on pressure and temperature:

$$\rho = P/R_s T$$

where P = atmospheric pressure in Pa = 95,000 Pa at 1000 m
R_s = specific heat capacity of air = 287 J/kgK
T = Temperature (Kelvin) ~ 15 to 25 deg or 298 to 308 K

Mean pressure at 1000 m altitude and 30.5 S latitude is around 95,000 Pa. Dry air at 36 degC in these conditions thus has a density of around 1.02 kg/m³

$$\rho = 1.0 \text{ kg/m}^3$$

For the measured October heating of 25 to 36 = 11 K and volumetric flowrate of 140 l/s = 0.14 m³/s there is a heat flow rate of around 1500 Watts:

$$Q = \rho q C_p \Delta T = 1.0(0.14)1005(11) = 1547.7 \text{ W}$$

Winter conditions with no house connection were tested in June 2015. In fine weather, temperature changes similar to October (15 to 25 = 10 degrees warming) were observed, possibly due to the improved cosine of orientation. The similar degree of warming indicates a similar power rating, 1400 W in ideal (not connected) conditions.

Light overcast, cold weather was also observed (test date 25/6/15). These minimum operating conditions provided a still-significant warming from 12.1 to 18.2 degrees. Combined with an estimated 100 l/s volume flowrate due to house connection pressure losses, this gives a low-range design heat rating:

$$\Delta T = 18 - 12 = 6 \text{ degC}$$

$$Q = \rho q C_p \Delta T = 1.0(0.10)1005(6) = 600 \text{ Watt}$$

A 6 degree, 600 Watt heater on cold, dry winter days for only 80 Watt fanpower input would be a significant contribution to residential or commercial heating budgets.

The figure of 600 W is a power rating in expected optimum conditions, that is light overcast in mid-winter. In fine winter weather, the heater would provide up to 10 degrees warming at 1000 W output. The values of 600-1000 W are a design range for the heater power rating.

In heavy overcast conditions or at night the heating unit is inoperable.

Thermal Losses

Radiative and conductive loss rates are based on textbook thermodynamic models:

Radiative Loss

Radiative power is a function of the 4th power of absolute temperature in the radiating surface compared to the same in ambient temperature. The surface is characterised by an emissivity which is not equal to absorptivity if the system is not in thermal equilibrium. For the thermal surface coating used in the SolrHeat heating plate, an absorptivity of 0.97 is coupled with an emissivity of 0.35 which is assumed to operate in non-equilibrium:

$$Q_{rad} = \sigma e A(T_p^4 - T_a^4)$$

where σ = Stefan-Boltzmann constant = 5.67x10^{-8} J/m²K^4
e = Emissivity = 0.35
A = Surface Area m²
T_p = Plate Temperature K
T_a = Ambient Temperature K

Substituting the values of 49 degC plate temperature (322 K) and 25 degC ambient (298 K) we have a maximum (October) radiative loss of around 100 watts:

$$Q_{rad} = \sigma eA(T_p^4 - T_a^4) = 5.67\times10^{-8}(0.35)1.815(322^4 - 298^4) = 103.2 \text{ J/s}$$

Conductive and Convective Loss

Conductive loss is dependent on the thermal conductivity of the fluid, in this case air at warmed surface temperatures, and the temperature gradient. Temperature gradient data are provided above. These are applied to variable surface areas on the different faces of glass and aluminium:

$$Q_{cond} = kA\Delta T/\Delta y$$

where k = Thermal conductivity of warm dry air = 0.025 J/mK
A = Surface Area m²
$\Delta T/\Delta y$ = temperature gradient

Given a total glass and aluminium surface area of 2.7 m² (not counting the cool floor and lower wall) and temperature gradients averaging 150 K/m (area-averaged), this gives a conductive loss of around 10 watts total:

$$Q_{cond} = kA\Delta T/\Delta y = 0.025(2.7)150 = 10.1 \text{ J/s}$$

Convective loss is assumed to be approximately the same as conductive loss and is in fact limited by the conductive rate so this is reasonable:

$$Q_{conv} = Q_{cond} = 10 \text{ J/s}$$

Total Heating Rate

The combined heat outflow, radiative, conductive and convective loss must add to the total area insolation at effective wavelengths and the cosine of the normal angle of incidence:

$$Q_{ins} = Q + Q_{rad} + Q_{cond} + Q_{conv} = 1550 + 100 + 10 + 10 = 1670 \text{ Watts}$$

This is average effective solar input over the 1.8 square meter heating plate, or 930 W/m². This is a typical estimate of insolation at this altitude and latitude. It is not an unreasonable value considering direct pyranometer measurements that were also taken.

June insolation measured by combined heat flow method averaged 850 W/m² (836 - 950). The 1500 W performance of the heating unit in optimum June conditions was only slightly less than the October average, the difference being directly proportional to insolation.

Heating Efficiency

The efficiency of the heating unit can be assessed as the heating output per total inputs including fan power, which comes to about 89%:

efficiency = 1550/(1670 + 80) = 88.6%

Detailed differential modelling resolves this figure to about **85%**, up to **88%** with forecast 800 Watt output in Winter.

Actual winter output in light overcast, cold conditions was 940 watts, with 960 Watt total heat flow as before. Again counting the 80 Watt fan power, design winter efficiency is:

Optimum efficiency = 940/(960 + 80) = 90.5%

Given that the output power is driven by a free resource (when available) from the sun, it is tempting to calculate efficiency as output per electrical input: the heat output is more than 10 times larger than the fan electrical input, which is a very promising exchange rate.

Conclusion

The tested SolrHeat space heater is a high-efficiency, high-output solar heater with a small rooftop footprint (< 2 square metres in plan) and large flow rate. Tests compare well with theoretical models, with agreement on heat flow and efficiency calculations compared to actual measurements. Forecast high-efficiency and excellent heat rating in winter ideal conditions has been confirmed by direct tests in June 2015.

Design-condition tests with a fully installed building connection are proposed for winter 2016.

Plate 1: SolrHeat space heater with thermal instrumentation.

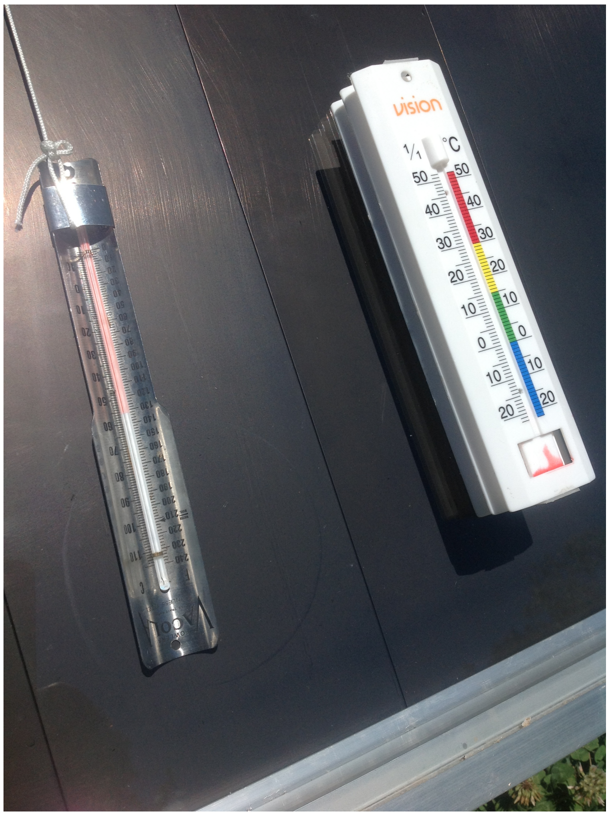

Plate 2: Vaccola and gradient thermometers