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An assessment of roof space solar gains in a temperate maritime climate

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Abstract

New Zealand has a temperate maritime climate. Despite mild winters compared to nations with continental climates, New Zealand houses have been reported to often be at temperatures below internationally recommended levels. Sources of additional heating are therefore of interest to many New Zealand home occupants. Roof space solar gains have been identified as one possible source of heating. This paper investigates the energy gains available in New Zealand homes from ventilation systems drawing air from the roof cavity. Three New Zealand houses were monitored and a computer-based thermal building simulation developed to quantify the heating and cooling energy that might be transferred by home ventilation systems. The computer model simulating the temperature in the roof space and occupied spaces was constructed using MATLAB, and used publicly available weather station data as the inputs. A good match between measured and modelled results was obtained. Small heating and cooling benefits are possible at certain times from pumping roof space air into the living areas of some New Zealand houses. The magnitude of these benefits, however, is not significant compared with the space heating required to maintain reasonable indoor temperatures over the New Zealand winter.

Keywords

Solar gains; home ventilation systems; thermal building simulation; roof space temperatures; indoor temperatures; heat transfer; New Zealand housing; residential space heating; temperate climate

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Nomenclature

A	area (m^2)
c	specific heat capacity (J/kg K)
g	gravitational acceleration (m/s^2)
G	total solar irradiance on a tilted surface (W/m^2)
Gr	Grashof number
h	heat transfer coefficient (W/m^2 K)
H	height (m)
k	thermal conductivity (W/m K)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
P	internal gains (W)
Q	rate of heat transfer (W)
SC	shading coefficient
$SHGC$	solar heat gain coefficient
T	temperature (K)
t	time (s)
U	internal energy (J)
u	air velocity (m/s)
W	width (m)

Greek Symbols

α	absorbance
β	coefficient of thermal expansion (1/K)
ε	emissivity
ν	kinematic viscosity (m^2/s)
σ	Stefan-Boltzman constant ($W/m^2\ K^4$)
ω	humidity ratio ($kg_{water\ vapour}/kg_{dry\ air}$)

Subscripts

a	ambient
b	base of control volume
$cond$	conduction
$conv$	convection
DA	dry air
e	external surface
h	house
i	internal surface
$moist$	moist air
p	constant pressure
r	roof space air

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rad	radiation
ref	constant reference value
s	roof surface
sky	sky
tm	thermal mass
v	constant volume
w	water vapour
wind	window

1. Introduction

New Zealand has a climate that can be broadly classified as temperate [1], due to the moderating influence of close proximity to the ocean for most parts of the country. Despite its temperate maritime climate, Lloyd et al. [2] and French et al. [3] have shown that New Zealand houses are generally very poorly heated and are often well below World Health Organization recommended minimum indoor temperatures. New Zealand home occupants therefore have an interest in additional sources of space heating.

Home ventilation systems that supply air from the roof cavity have become common in New Zealand in the last decade and yet some aspects of their performance are yet to be quantified. One of these is the claim that ventilation provided from a warm roof space contributes to space heating [4]. This project has developed a simple numerical model of the air temperature in a roof space and used this to predict heat transfer into the living zones to give an indication of these energy flows.

To maintain indoor air conditions, a minimum air flow rate equivalent to one complete household air exchange every 2-3 hours has generally been accepted internationally and is supported by the New Zealand Standard NZS 4303 [5]. The New Zealand Building Code [6] is relatively undemanding relating to home ventilation in that it allows for ventilation to be provided entirely by window opening. So long as the area of openable doors and windows exceeds 5% of the floor area then the acceptable solution G4/AS1 for single occupancy homes has been satisfied. There are no current requirements for additional passive or active ventilation in New Zealand homes and there are no requirements for the building to achieve a standard for envelope air-tightness [6].

A trend towards more airtight homes has been identified in earlier studies [7] and this has been explained in terms of the adoption of large sheet internal lining materials and flooring and of tight windows in aluminium frames. Where infiltration may have provided 0.5−1 air changes per hour (ac/h) of background ventilation in houses built before 1960, this has reduced to less than 0.3 ac/h in many recently constructed homes [7]. This has shifted more responsibility for adequate ventilation onto window opening than was the case in older homes. Open windows are often seen as a security risk as well as a source of heat loss and draughts, and many home owners have chosen instead to retrofit supply ventilation systems to help control indoor moisture.

Over recent years, many different mechanical home ventilation systems have appeared on the market in New Zealand as a means to improve the ventilation requirements of homes. As of 2009 it was estimated that around 10% of New Zealand homes had some form of mechanical ventilation system installed [4]. Most of these systems slightly pressurise the house by pumping filtered air down from the roof cavity. The pressurised house attempts to equalise with the ambient conditions by inducing air flow through the natural leaks in the building envelope. These positive pressure systems were initially developed in the 1970s as a means of reducing condensation without affecting the operation of open-flued combustion appliances [4]. The other main types of mechanical ventilation system marketed in New Zealand are balanced pressure heat recovery systems. These systems have two fans, one of which draws in fresh air from outside while the other exhausts an equal volume of indoor air to the outside. These two air streams are kept separate but interact thermally in a heat exchanger to help reduce heating/cooling loads.

The World Health Organization suggests that the internal temperature of a house should be able to be maintained above 16°C for thermal comfort and health reasons [8]. In spite of this, the New Zealand Building Code does not specify that a minimum indoor air temperature must be able to be maintained in buildings other than aged care facilities and early childhood centres [9]. Due to this lack of regulation and other factors, many New Zealand households are under heated for much of the year, thereby exposing residents to the risk of developing health issues. As people become more aware of the health benefits from a warm and dry home, the need for alternative low cost heating methods have increased. The increased availability of technologies such as heat pumps and efficient combustion appliances has helped to reduce the ongoing costs of heating in existing buildings.

Insulation standards for residential buildings have also increased in recent years [3] to the point where a ventilation rate of 0.5 ac/h without heat recovery can make up 20-30% of the total heat loss from the building. This has focussed more attention on ventilation heat losses. In the future, ventilation systems with heat recovery might be more attractive than the simple supply systems that are currently widely used. The energy issue has lead to claims about possible heat gains from roof spaces, and this provided the motivation for this project.

Most existing New Zealand houses have pitched roofs of simple construction and materials, and are relatively unshaded from the sun [3]. These roofs therefore represent a large solar collecting area which can act to heat the roof surface and subsequently the air contained within the roof cavity. The purpose of this paper is, therefore, to investigate the energy/heat balance of New Zealand roof spaces and if this energy can be used to help heat and/or cool the building envelope. The timing of these energy/heat fluxes must also be taken into account to assess the diurnal and annual variability in both roof space solar gains and the building's heating requirements.

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2. Methodology

2.1. Mathematical model

In order to determine the potential for roof spaces to be used as free heat sources for various locations and climatic conditions, a mathematical model has been developed from first principles to estimate the roof space air temperature of a typical New Zealand house. All required input data were standard weather station variables which are available for many locations around New Zealand [10]. Ambient air temperature, humidity, wind velocity, wind direction, pressure, and global irradiance data are required by the model in order to predict roof space temperatures. These input variables were obtained from the Energy Studies Weather Station at the University of Otago [11] to model the roof space temperatures of several houses in Dunedin, New Zealand. Weather data from other main centres around New Zealand allowed the model to predict the roof space conditions for various house types and locations. Weather data for these locations was obtained from the National Institute of Water and Atmospheric Research National Climate Database [10]. Modelled results are used in this paper to assess the potential of using the roof space as a free heat source and how this may change with location within New Zealand.

A schematic diagram of the modelled system is shown in Figure 1.

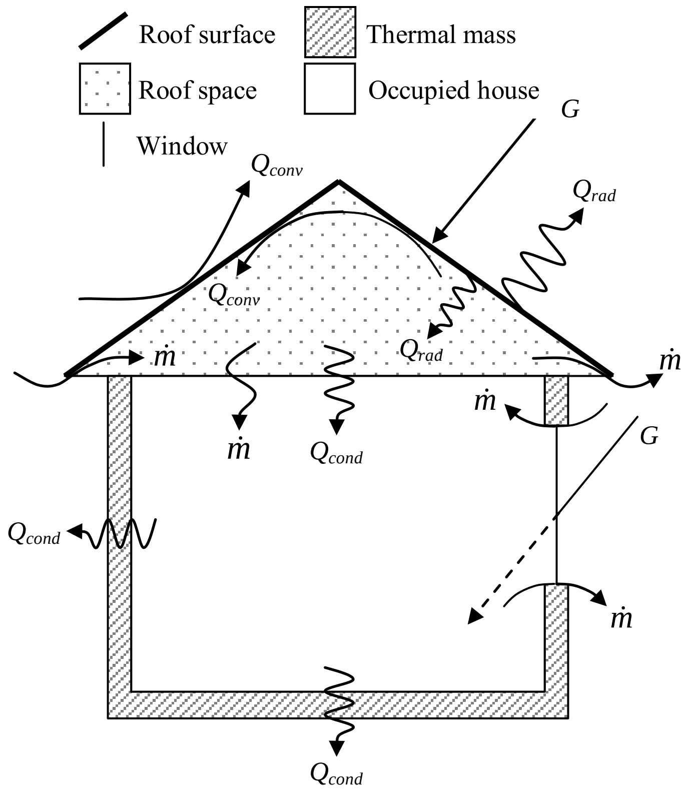

Figure 1. Schematic diagram of the modelled system with energy transfer components shown.

The house shown in Figure 1 is characterised by considering the house to be made up of four control volumes: the roof, the roof space, the occupied house space and a control volume that lumps the thermal mass within the house. The energy balance for each control volume is defined by Eq. (1).

$$\frac{d(mU)}{dt} = \begin{bmatrix} \sum_{in} \dot{m}(c_p \Delta T + \frac{u^2}{2} + gH) - \sum_{out} \dot{m}(c_p \Delta T + \frac{u^2}{2} + gH) \\ + \sum \alpha AG - \sum Q_{conv} - \sum Q_{cond} - \sum Q_{rad} + \sum P \end{bmatrix} \quad (1)$$

Eq. (1) represents a standard energy balance which includes mass (m) and energy (U) transfers and can be applied to any control volume. The change in kinetic and potential energy of the air has been ignored in the current model as all air movement results in the displacement of other air parcels as pressures equalise. Any mass transfers within the control volume of height H (m) in Figure 1 and Eq. (1) are represented by $\dot{m}$, while Q and G indicate direct energy fluxes. Energy fluxes which are associated with mass transfers are represented by the first two summations in Eq. (1), for air of specific heat capacity c_p and temperature T, subject to acceleration due to gravity (g). The remaining summations represent solar gains from a tilted surface of area A (m^2) and absorbance α, convective, conductive, and radiative losses, and any work/power inputs respectively.

The energy balance for the roof leads to Eq. (2).

$$\frac{dT_s}{dt} = \frac{1}{m_s c_{v,s}} \begin{bmatrix} \alpha A_s G - h_{conv,i} A_s (T_s - T_r) - h_{conv,e} A_s (T_s - T_a) \\ - \varepsilon_i \sigma A_s (T_s^4 - T_r^4) - \varepsilon_e \sigma A_s (T_s^4 - T_{sky}^4) \end{bmatrix} \quad (2)$$

The roof is considered to be a single two-dimensional surface of emissivity ε, with a constant mass and a uniform temperature across the entire surface, as schematically shown in Figure 1. This is a reasonable assumption to make given that most New Zealand roof claddings are relatively thin and have high thermal conductivities [3]. It is also assumed that the specific heat capacity of the roof surface is a constant value over the range of temperatures which are observed. In the usual notation, σ denotes the Stefan-Boltzman constant.

The energy balance for the roof space leads to Eq. (3).

$$\frac{dT_r}{dt} = \frac{1}{m_r c_{v,r}} \begin{bmatrix} \dot{m} c_p (T_a - T_r) + h_{conv,i} A_s (T_s - T_r) \\ + \varepsilon_i \sigma A_s (T_s^4 - T_r^4) - h_{cond,b} A_b (T_r - T_h) + P_{fan} \end{bmatrix} \quad (3)$$

As shown in Figure 1, roof space air physically mixes with both the occupied house and the ambient air, while also thermally interacting with the occupied house and the roof surface. Physical mixing occurs due to the lower air-tightness of roof cavities, compared to that of living areas, which encourages natural air infiltration and aids the mixing process of the contained air [12,13]. When a mechanical ventilation unit is present, the operation of the fan can also aid the roof space air mixing process. The roof space air is, therefore, considered as moist air with the same absolute humidity as the ambient air. Measurements in the modelled houses showed that the roof space temperatures were approximately uniform. For these reasons, the air contained within the roof space was considered as a single isothermal control volume, in contact with the roof surface, which simplifies the model significantly. Data presented by Flack [14], Ridouane et al. [15], and Holtzman et al. [16] show that sealed triangular enclosures, exposed to a heat flux between surfaces, are approximately isothermal along the central plane between a relative distance of 10% and 80% of the apex height above the enclosure base. Those papers also show that the temperatures in such isothermal regions are approximately equivalent to the mean temperature within the enclosure. These factors also support the application of a single control volume to model the roof space air temperature, and allow a better correlation with measured data.

The occupied rooms of the house are also considered as a single control volume, and the energy balance of this leads to Eq. (4), where *SHGC* is the solar heat gain coefficient.

$$\frac{dT_h}{dt} = \frac{1}{m_h c_{v,h}} \begin{bmatrix} A_{wind} G \times SHGC \times SC + \sum \dot{m} c_p (T - T_{ref}) + h_{cond,b} A_b (T_r - T_h) \\ - \sum h_{cond} A (T_h - T_a) - h_{cond,tm} A_{tm} (T_h - T_{tm}) + P_h \end{bmatrix} \tag{4}$$

The mean internal house temperatures can thus be approximated and compared to the modelled roof space air temperatures to assess if any heating/cooling benefits are possible. Due to the natural air infiltration and mixing of the occupied areas of the house, the contained air is assumed to have the same absolute humidity as the ambient air. The model is not particularly sensitive to humidity variations, so this assumption significantly simplifies the overall model without noticeably affecting its accuracy. A thermal mass component is also present in the occupied rooms in the form of furniture and the building components. Due to the large number of thermal objects and their extremely variable physical and thermal properties, it is assumed that they can all be approximated by a single mass which is evenly distributed within the occupied house. The energy balance for the thermal mass control volume leads to Eq. (5) below.

$$\frac{dT_{tm}}{dt} = \frac{h_{cond,tm} A_{tm} (T_h - T_{tm})}{m_{tm} c_{tm}} \tag{5}$$

The exposed surface area, mass, specific heat capacity, and heat transfer coefficient of the thermal mass are subsequently estimated for each house to match the measured data.

The effective black body sky temperature, T_{sky}, is calculated from equations presented in Al-Hinai et al. [17]. The temperature of the roof surface is then calculated from Eq. (2) and is controlled by solar gains, as well as convective and radiative heat transfers from both the internal and external surface of the roofing material. The temperature of the roof space air is calculated from Eq. (3) and is primarily controlled by the mass flow rate of air through the control volume and the amount of heat transferred from the roof surface. The temperature of the occupied house is calculated from Eq. (4), which is also dependent on the mass flow rate of air through the control volume, as well as the conductive losses and the internal and solar gains. The modelled thermal mass component is included in the model through Eq. (5), which acts to dampen the internal temperature swings of the occupied rooms.

The developed model is primarily controlled by the incident solar radiation, as well as the ambient air temperature and the air infiltration rate into the roof cavity. Solar irradiance is generally measured on a horizontal surface at weather stations and is adjusted for the tilted surfaces of house roofs following the methods described in Duffie and Beckman [18]. Assuming a homogeneous sky, the instantaneous clearness index was evaluated for every weather station measurement and a piecewise correlation, developed by Reindl et al. [19], was used to estimate the diffuse fraction of the incident solar radiation. The empirical correlation from Reindl et al. [19] was used as it is based on data from several sites around the world and recent studies have shown it produces smaller errors than other simple correlations [20]. Combining diffuse and direct components allows the total solar radiation incident on a combination of tilted surfaces to be calculated.

Similar to Bassett [21], it is assumed that the main source of air infiltration through the roof space control volume occurs around the perimeter of the roof between the base of the roof cladding material and the supporting framework. The open area is therefore related to the profile of the roofing material and the wind direction incident on the building. The effective open area is used together with the wind velocity data to calculate the natural air infiltration rate, based on natural ventilation equations presented by Moss [22]. Modelling air infiltration rates can be particularly complicated; this therefore represents one of the largest uncertainties in the current study. The infiltration rate used in the model is the larger of the natural infiltration rate, and that which would be induced by the home ventilation system under calm conditions. Induced ventilation rates are a function of the fan flow rate and the volume of the roof cavity.

The specific heat capacity of dry air and water vapour at constant pressure is derived from correlations provided by Wexler et al. [23]. Assuming that moist air behaves as a mixture of ideal gases, the specific heat capacity of moist air at constant pressure was calculated on a dry air basis by Eq. (6). It is assumed that the specific heat capacity of air at constant volume can be approximated as a constant value over the range of temperatures observed. The humidity ratio ($kg_{\text{water vapour}}/kg_{\text{dry air}}$) is denoted by ω.

$$c_{p,moist} = c_{p,DA} + \omega \times c_{p,w} \tag{6}$$

Individual energy fluxes from each of the modelled control volumes are determined by heat transfer coefficients specific to each energy transfer term. It is assumed that natural convection occurs between the roof space air and the internal boundary of the roof surface, as this surface is sheltered from the ambient conditions. The natural convection heat transfer coefficient ($h_{conv,i}$) is defined by Eq. (7) - (10) which have been modified from Ridouane and Campo [24] for the current study. The thermal conductivity (k) of the roof space air is used.

$$h_{conv,i} = Nu \times {k_r}/{H} \tag{7}$$

$$Nu = 4.63 \qquad \text{(hot surface)} \tag{8}$$

$$Nu = 0.286(H/W)^{-0.286} Gr^{0.25} \quad \text{(cold surface)} \tag{9}$$

$$Gr = \frac{g\beta_r |T_s - T_r| H^3}{\nu_r^2} \tag{10}$$

To calculate the Nusselt number (Nu), Eq. (8) is applied when the roof surface is hotter than the roof space air, whereas Eq. (9) is applied when the roof surface is colder than the roof space air. This difference occurs due to the natural buoyancy of different temperature fluids, which is the driving force behind natural convection. Eq. (9) is a correlation from [24] based on original experimental work done by Flack and others [14, 25] on natural convection in triangular enclosures and takes into account the enclosure's aspect ratio (H/W). Although Eq. (9) was developed for Grashof Numbers (Gr) in the range of 2.9×10^6 to 9×10^6, it has been shown by Anderson et al. [26] that it is also accurate to within ±5% for Grashof Numbers in the range of 10^7 to 10^9. It was therefore assumed that this correlation is a suitable generalised representation of natural convection heat transfer in attic shaped enclosures [26]. In Eq. (10), the coefficient of thermal expansion (β) and kinematic viscosity (ν) of the roof space air are used.

The external surface of the roof is exposed to the ambient conditions and is therefore subject to forced convection heat transfer. Forced convection heat transfer coefficients are different for each roof surface as they are a function of wind speed and direction, as shown by Eq. (11) and (12) for windward and leeward surfaces respectively [27]. A surface is defined as windward if the angle of incidence between the wind direction and the normal to the surface is less than ±90° and leeward for all other directions.

$$h_{c,e} = 7.4 + 4.0u_a \quad \text{(windward)} \tag{11}$$

$$h_{c,e} = 4.2 + 3.5u_a \quad \text{(leeward)} \tag{12}$$

Eq. (11) and (12) were used to calculate the forced convection heat transfer coefficients as they take into account the wind direction, unlike many other simple correlations, and are often just as effective at fitting empirical data as more complex power relations [27].

Conduction heat transfer coefficients (h_{cond}) are a function of the level of insulation installed in individual houses and the temperature difference across that insulation. The building materials and air boundary layers must also be taken into account and values for these have been obtained from Holger [28].

The surface absorbance and emissivity was considered constant over the long wave lengths, which predominantly control the heat balance of the surface. These values were obtained from available literature for the relevant internal and external surfaces of the cladding material [18, 22, 29].

Several studies have shown that New Zealand houses are generally very poorly heated, although this is highly variable and depends on both the heating and ventilating habits of the occupants. Additional heating and/or ventilating is, therefore, not included in the current model although several major internal gains are taken into account. The roof space has some internal gains which are derived from the operation of any electrical equipment operating the roof cavity. The electrical power drawn by the mechanical ventilation system is a function of the fan speed and can be obtained for different operating conditions. Internal gains have also been included in the living zones by using a similar methodology as the Building Research Association of New Zealand (BRANZ) Annual Loss Factor (ALF) model [30]. Background internal gains have been quantified as 100 W per hot water cylinder, 5.3 W/m^2 of floor area for appliance losses and 60 W/resident.

A volumetric flow rate of 0.2 m^3/s was defined for the theoretical transfer of air from the roof space to the house to assess any potential heating/cooling benefits. This flow rate was chosen as it is close to the maximum flow rate of the most widely installed mechanical ventilation systems in New Zealand houses [4]. Based on this flow rate and average air properties, an initial approximation suggests that approximately 0.25 kW of heating or cooling could be obtained from a 1 °C roof-space/indoor temperature difference. Any heating benefit should therefore nearly always exceed the power drawn by the fan to circulate the air.

Two situations were examined to quantify the available energy in roof spaces and how this varies with location:

1. The flow rate is maintained for periods when the roof space air temperature is above or below a specified reference temperature (T_{ref}), which represents heating and cooling loads respectively. This situation assumes that the thermal energy of the roof space air is transferred to a constant temperature sink and that all of this energy contributes to space conditioning. This therefore represents the maximum possible heating and cooling energy which can be obtained from the roof space air.
2. For heating and cooling loads respectively, the flow rate is maintained for periods when the roof space air temperature is greater/less than the occupied house temperature and the house is below/above the desired reference temperature. This situation represents a more realistic scenario, whereby the thermal energy in the roof space air cannot be stored, and its heating/cooling potential is only taken advantage of during times when the occupied house is below/above the desired temperature respectively.

Both of these ventilator control functions allow us to quantify the maximum benefits that are possible from a mechanical ventilation system that is specifically designed to maximise the thermal benefits of roof space air. This is not the case with existing mechanical ventilation systems, as their main function is to improve ventilation and moisture control, which means that they may still transfer air during periods when it is not desired for thermal comfort reasons. The actual performance of existing mechanical ventilation systems will be less than the values calculated in this research, and will be a function of how the individual units are controlled.

MATLAB, a high-level technical computing language, was used to implement and run this model. The system of differential equations (2) – (5) were solved simultaneously using MATLAB's built in differential equation solver, ODE45, which solves non-stiff differential equations by a medium order method, based on the explicit Dormand-Prince pair Runge-Kutta formula [31]. From these results, the amount of heat which is transferred from the roof surface to the roof space air via natural convection and radiation is determined. As an initial condition, the roof surface and roof space air temperatures were set equivalent to the ambient air temperature. This is often relatively accurate as each model run began at midnight on the first day, which allowed several hours for the control volumes to equilibrate with the ambient conditions without any solar gains. The initial temperature of the occupied house and thermal mass were set to either the ambient air temperature or a 15 °C reference temperature, whichever was higher. This reference temperature was chosen because Lloyd et al. [2] found 15 °C to be the average annual internal temperature of 100 Dunedin houses. All days are simulated individually with only the first day requiring initial values, as the following days are initialised by the final values of the previous day. In some situations, the initial conditions may not be applicable to the first day and the model will therefore require a spin-up period before the results gain accuracy.

2.2. Experimental measurements

Temperature and humidity data were recorded for periods between December 2007 and July 2009. These data were used in model testing, as outlined in Section 3 (Model validation) of this paper. Three measured houses were identified by the three letter international code of the city they were located in (e.g. Dunedin, DUD) followed by a single numerical reference number. One of the houses (DUD1) had a mechanical ventilation system installed and operating for most of the monitored period, while a neighbouring house (DUD2) had only passive ventilation. DUD1 and DUD2 were located approximately 150 metres from the weather station. Preliminary temperature data from DUD1 and DUD2 has been previously presented as a conference paper by Smith et al. [32], where the houses were labelled as M1 and P1 respectively. A third house (DUD3), also without a mechanical ventilation system, was located approximately 2 km from the weather station and was monitored independently. Measured data from these houses was then used to test the model, which is based on parameters specific to each house. The temperature and humidity was recorded in houses DUD1 and DUD2 at specific locations in the roof space, hallway, and living room, using two different measurement devices. Thermochron iButtons (DS1921G) were used to measure and log the temperature at the installed location with a set sampling interval of between five and thirty minutes. These iButtons have a temperature resolution of 0.5 °C and an accuracy of ±1 °C, as specified by the manufacturer [33]. Onset HOBO data loggers (H08 series) had a similar sampling frequency to the iButtons and were used to record both temperature and humidity. HOBOs have a temperature accuracy of ±0.7 °C and a relative humidity accuracy of ±5% [34]. All sensor outputs were checked against each other, as well as a Testo 625 hygrometer, an Extech 446580 hygro-thermometer data logger, and a whirling hygrometer. Tests were done in a controlled climate room both before and after the in situ measurements were taken. Any faulty sensors which tracked outside of their accuracy limits were replaced and their recorded measurements were not included in the final dataset. A HOBO and iButton were installed next to each other at each location to provide redundancy and allow sensors to be cross checked with each other. The hallways and living rooms of both houses had sensors installed at three different heights, which were approximately 0.4 m, 1.6 m, and 2.4 m above the floor. Roof space sensors were paired up where possible and installed in accessible places at various heights between 6% and 83% of the roof cavity height above the installed insulation. Thermochron iButtons were used to record temperature for a single location in the roof space, hallway, bedroom, and outside of DUD3. Because these measurements were recorded by a third party, and no post-manufacturing calibration was undertaken on the sensors, the dataset from DUD3 has only been used for periods when no other data is available.

3. Model validation

For validation purposes, the model was run for a representative period in summer, autumn, winter, and spring for each of the three monitored houses in Dunedin, New Zealand. Whenever possible, periods when the houses were unoccupied, or had low occupancy, were used. The output data from the model were then compared with temperature data from the same period. Because the measured temperature data were not continuous for each house, the location with the most comprehensive record was used. Results are presented in Figures 2-5 for a five day period in each of the respective seasons. Three iButtons returned useable data for each of the hallway and lounge of DUD1 for the periods shown in Figures 2 and 3. Three HOBOs returned useable data for each of the hallway and lounge of DUD1 for the period shown in Figure 2, and one HOBO returned useable data for the hallway and for the lounge of DUD1 for the period shown in Figure 3. Six iButtons and two HOBOs returned useable data for the roof space for the period shown in Figure 2. Five iButtons and one HOBO returned useable data for the period shown in Figure 3. Due to the limitations on setting iButtons to start logging at a future date, only HOBO data was available for DUD1 for the period shown in Figure 4, with one HOBO data set from each of the roof space and hallway, and two from the lounge. One iButton data set was available from DUD3 from each of the roof space and bedroom for the period shown in Figure 5.

The occupied house temperatures are much more difficult to model accurately as they depend largely on the behaviour of the occupants. A background rate of internal gains was included in the model. Because all of the monitored houses were occupied throughout the majority of the study, activities such as heating or increasing ventilation rates by opening windows/doors would occur frequently. These events were not directly monitored. However, as stated above, whenever possible, validation periods were when the houses were likely to be unoccupied or lightly occupied. Where temperature excursions associated with occupant heating were identified in the data, another period of time was examined within the lightly occupied period.

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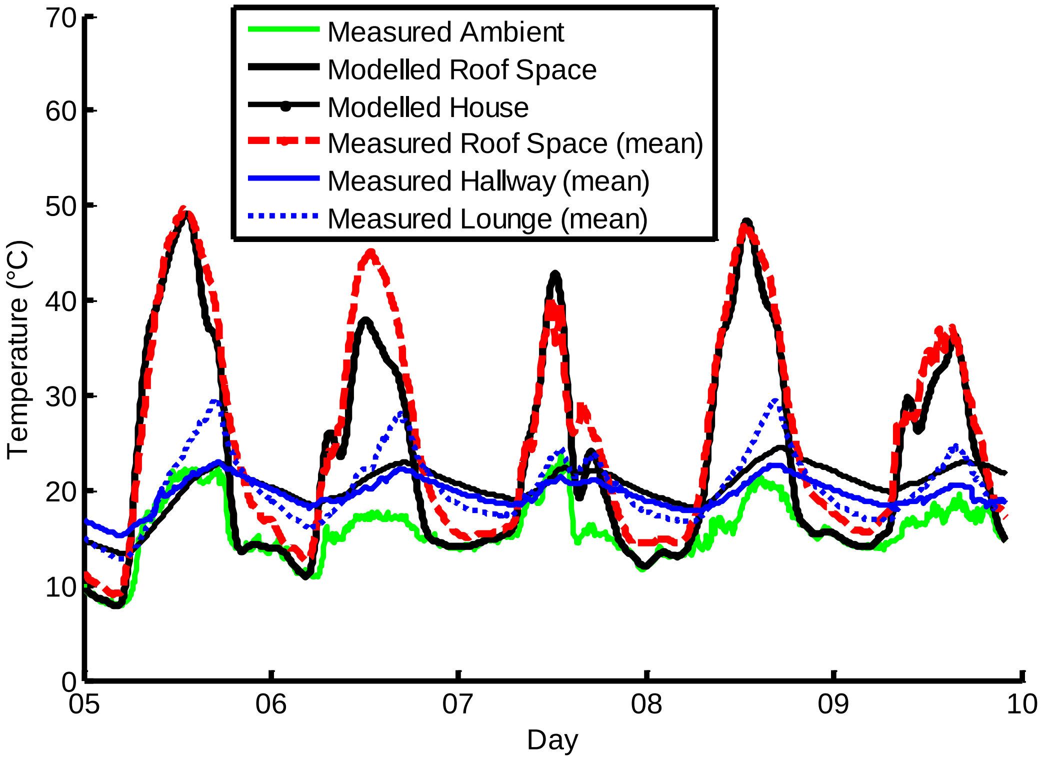

Figure 2. Model validation for the 5th – 9th of January 2008, representing a typical summer period for house DUD1. Measured temperature data from within the roof space (mean of eight sensors), from the hallway and lounge of the house (mean of six sensors in each case), and from a nearby weather station are presented, along with the modelled temperature from within the roof space and the house.

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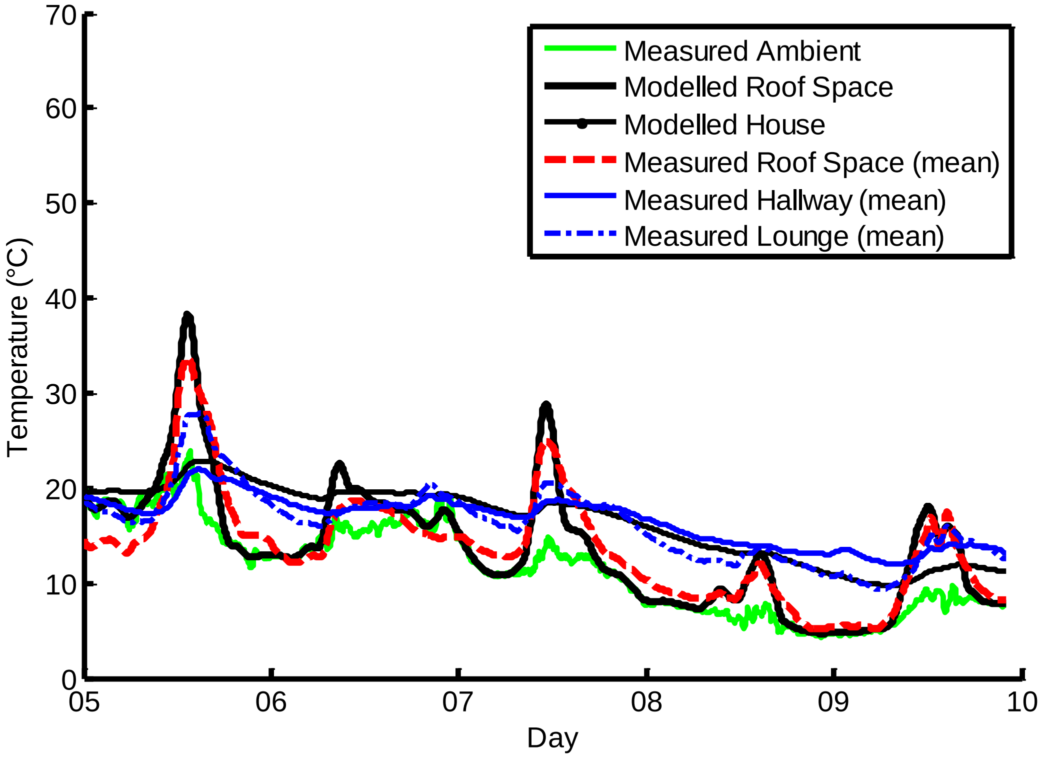

Figure 3. Model validation for the 5th – 9th of April 2009, representing a typical autumn period for house DUD1. Measured temperature data from within the roof space (mean of six sensors), from the hallway and lounge of the house (mean of four sensors in each case), and from a nearby weather station are presented, along with the modelled temperature from within the roof space and the house.

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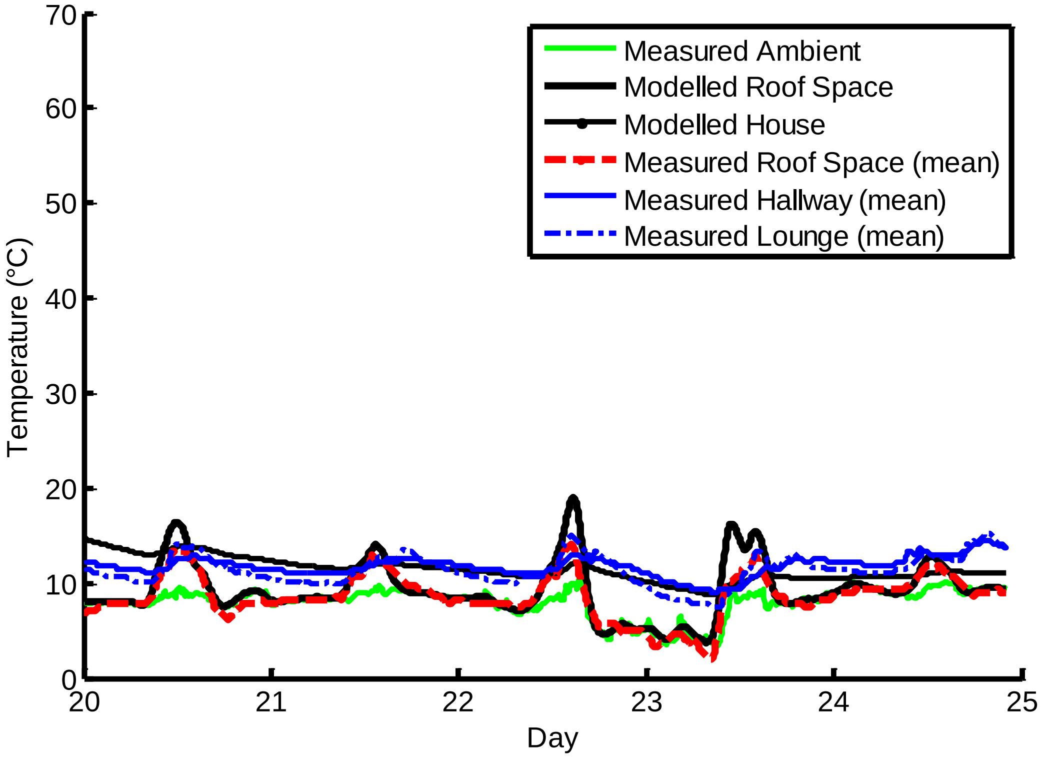

Figure 4. Model validation for the 20th – 24th of June 2009, representing a typical winter period for house DUD1. Measured temperature data from within the roof space (one sensor), from the hallway and lounge of the house (one sensor and mean of two sensors, respectively), and from a nearby weather station are presented, along with the modelled temperature from within the roof space and the house.

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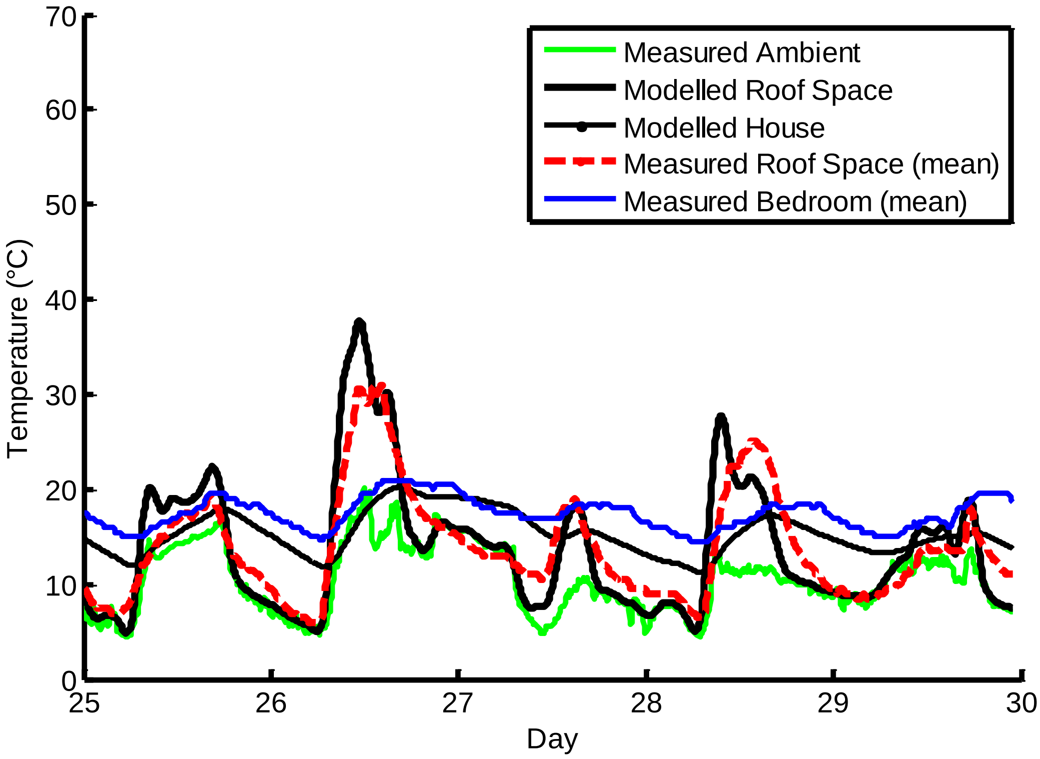

Figure 5. Model validation for the 25th – 29th of September 2008, representing a typical spring period for house DUD3. Measured temperature data from within the roof space and from the bedroom of the house (one sensor each), and from a nearby weather station (approximately 2 km away) are presented, along with the modelled temperature from within the roof space and the house.

The model is able to predict roof space temperatures relatively accurately for DUD1 during summer, autumn, and winter periods, as shown in Figures 2-4. Modelled roof space temperatures for DUD3 during spring periods (Figure 5) also follow the measured data relatively closely. One exception to this is the 26th September 2008. It is unclear exactly what has caused this difference, and why it only occurs on some days. Since model outputs are very sensitive to incident solar radiation, it is possible that this behaviour may be due to variations in local factors such as shading which has not been taken into account in the present model.

Another exception to the relatively accurate roof space temperature predictions (not shown in the figures in this paper) occurs when the roof space temperatures are below the ambient air temperature. This situation arises due to the radiative cooling from the roof surface on clear calm nights [35, 36]. As shown in Eq. (2), the radiative losses for the external surface of the roof are effectively bound by T_{sky}, the sky temperature. Because the model only uses ambient humidity and temperature data to calculate the effective sky temperature, factors such as the sky clearness are not taken into account, and cannot be included without continuous specialised measurements. This may therefore alter the radiative losses under certain conditions; however, a sensitivity analysis on the model showed that the roof temperature is more dependent on the external forced convection than the effective sky temperature. The difference between measurements and model output for roof space temperatures below ambient are therefore due to the convective heat transfer between the ambient air and the roof surface, which prevents the radiative cooling of the roof below the ambient temperature. Potential parameters such as the exact direction and degree of shelter the house may experience compared to the weather station will affect variables such as the wind direction and velocity, and subsequently the convective heat transfer coefficient. Various relevant convective heat transfer coefficient correlations, which have been reviewed by Palyvos [27], were applied to the current model but did not improve the match between model output and measured temperature values. Model results may therefore slightly over predict the roof space air temperatures, and subsequently the amount of energy available for heating, on clear calm nights.

The initial temperature is more important in living rooms because of the large amount of thermal mass located in these occupied zones of the house. As shown in Figures 3-5, the spin-up effect is more obvious when modelling the occupied house zones than the roof space air temperatures. This is partly due to a representative initial value being chosen to initiate the model, and partly due to dampening of temperature fluctuations by the thermal mass.

DUD1 has been modelled in Figures 2-4, which has more accurate and comprehensive measured data, and is also located much closer to the weather station where the model input data was obtained. DUD3 however, is located much further from the weather station, so there may be some discrepancies between the model input variables and the ambient conditions experienced by the house. For these reasons, as well as the uncertainty in the sensor accuracy at DUD3, model outputs for DUD1 and DUD2 are considered more representative than those for DUD3.

Table 1 shows the absolute maximum difference, as well as the root mean squared difference (RMSD) and the mean difference (MD) between the modelled and measured roof space temperatures presented in Figures 2-5. Table 2 provides a similar statistical analysis between the modelled and measured occupied house air temperatures. Modelled results were subtracted from the measured data; therefore, negative MD values represent periods when the model slightly over predicts the measured data and positive MD values represent periods of under prediction.

Table 1. Statistical performance of the developed model to predict roof space air temperatures.

	Abs. max. difference (°C)	RMSD (°C)	MD (°C)
Figure 2	9.9	3.3	2.2
Figure 3	5.4	2.1	-0.12
Figure 4	6.1	1.4	-0.68
Figure 5	11	3.1	0.14

Table 1 indicates the model's ability to predict roof space temperatures with a mean accuracy of between ±1.4 °C and ±3.3 °C for DUD1. The model also performs relatively well for DUD3 (Figure 4), with a maximum difference between the measured and predicted values of 11 °C and a RMSD of 3.1 °C. The mean difference between the measured and predicted temperatures all range between −0.68 °C and 2.2 °C, where two are positive and two negative, which shows that on average the model will both over predict and under predict temperatures by approximately the same amount.

Table 2. Statistical performance of model to predict air temperatures in the occupied house.

	Abs. max. difference (°C)	RMSD (°C)	MD (°C)
Figure 2	4.4	1.4	-0.75
Figure 3	3.0	1.5	0.11
Figure 4	3.4	1.3	0.30
Figure 5	5.5	2.4	2.1

Table 2 shows that the model can be used to predict the indoor air temperature of the living zones of DUD1 with a mean accuracy of between ±1.3 °C and ±1.5 °C. The mean difference between the measured and predicted house temperatures (DUD1 and DUD3) range between −0.75 °C and 2.1 °C, which shows that the model is generally slightly under predicting these temperatures. Any calculated heating benefits from the roof space air will subsequently be potentially greater than the actual amount of energy which could be extracted in the real house. Table 2 shows that the model consistently under estimates the internal house temperatures in Figure 5. This under estimation suggests that the occupants may have been using additional heating during the September period.

4. Results

The model was run for four main centres around New Zealand to investigate how roof space air temperatures vary with location and hence their potential to be used as free heat sources. The same house parameters, based on DUD1, were modelled for Auckland (AKL), Wellington (WLG), Christchurch (CHC), and Dunedin (DUD), to provide a fair comparison between locations. The location of the modelled house was set to the weather station location for each centre to ensure accurate input data [10]. The University of Otago Energy Studies weather station data was retained for model runs of houses in Dunedin [11].

The model was run in each of these locations for a ten day period in 2008 from the 10th to the 19th of January, April, July, and October to represent summer, autumn, winter, and spring conditions respectively. Results from these model runs are presented in Tables 3-5. All energy quantities are presented in terms of a continuous energy rate in kilowatts, instead of an absolute value in Joules, to place any benefits in context with traditional heating/cooling methods. Heating and cooling benefits are obtained by physically displacing cooler or warmer air respectively with filtered roof space air which is closer to the desired temperature of the house.

Table 3. Average (*a*) heating and (*b*) cooling rate benefits in kW (kilowatts) to a constant 18 °C heat sink from the 10th – 19th of January, April, July, and October in 2008.

a

	January	April	July	October
DUD	1.6	0.41	0.06	0.79
CHC	2.0	0.47	0.15	1.2
WLG	0.70	0.45	0.08	0.47
AKL	2.2	0.61	0.14	0.86

b

	January	April	July	October
DUD	0.49	1.3	2.1	1.3
CHC	0.27	0.93	2.1	1.1
WLG	0.25	0.78	1.5	0.99
AKL	0.13	0.49	1.4	0.84

Table 3 represents the energy benefits which are possible if the flow of roof space air was changed to a temperature of 18 °C and the change in internal energy of the air could be stored for when it was desired. Although this situation is unrealistic with existing mechanical ventilation systems, it does represent an upper limit to the amount of energy available in roof space air when transferred at 0.2 m^3/s. Technologies such as hot water heat pumps located in the roof cavity could be one potential method of converging the air temperature to a constant value and using a medium such as water to store the energy until it is required.

With the exception of WLG, a general trend seen in Table 3 in that the heating potential of roof spaces increases while the cooling potential decreases the further north the house is located. This is because results presented in Table 3 are based on a constant 18 °C sink for the roof space heat and so the only variables effectively taken into account are the ambient weather conditions.

Table 4. Average (*a*) heating and (*b*) cooling rate benefits in kW (kilowatts) to occupied rooms with a desired internal temperature of 18 °C. Model has been run from the 10th – 19th of January, April, July, and October in 2008.

a

	January	April	July	October
DUD	0.07	0.24	0.26	0.52
CHC	0.01	0.19	0.34	0.30
WLG	0.02	0.06	0.18	0.20
AKL	0	0.09	0.31	0.19

b

	January	April	July	October
DUD	0.92	0.19	0	0.23
CHC	1.0	0.33	0	0.67
WLG	0.63	0.44	0	0.40
AKL	0.93	0.60	0	0.55

The magnitude of the effective continuous heating gains for occupied rooms with a set point temperature of 18 °C is quantified in Table 4(*a*). As an example, pumping 0.2 m^3/s of warmer roof space air into the living areas of the modelled Dunedin house provides energy gains equivalent to a 0.5 kW heater running continuously over the ten day period in October. This can be compared to the 0.2 kW of continuous heating benefits which are possible for the modelled house in Auckland over the same period. Table 4(*a*) shows that in general, most of the heating benefits occur in winter (July) and spring (October) when the roof space air is warmer than the occupied room air, but the living areas are still colder than the desired temperature for thermal comfort.

The opposite effect is observed for the cooling potential of roof space air as shown by Table 4(*b*). Table 4(*b*) quantifies the amount of excess heat which can effectively be continuously displaced from the house by pumping 0.2 m^3/s of cooler roof space air into the living areas.

Table 5. Average continuous (*a*) heating and (*b*) cooling benefits in kilowatts to occupied rooms with a desired internal temperature of 20 °C (and 16 °C). Model has been run from the 10^{th} – 19^{th} of January, April, July, and October in 2008.

a

	January	April	July	October
DUD	0.16 (0.01)	0.41 (0.06)	0.26 (0.25)	0.77 (0.16)
CHC	0.06 (0)	0.35 (0.07)	0.34 (0.31)	0.59 (0.10)
WLG	0.07 (0)	0.21 (0.01)	0.18 (0.14)	0.33 (0.07)
AKL	0.01 (0)	0.25 (0.03)	0.31 (0.25)	0.44 (0.05)

b

	January	April	July	October
DUD	0.84 (0.98)	0.01 (0.52)	0 (0)	0.02 (0.64)
CHC	0.97 (1.0)	0.09 (0.63)	0 (0.03)	0.36 (0.99)
WLG	0.58 (0.69)	0.14 (0.64)	0 (0.03)	0.12 (0.61)
AKL	0.92 (0.93)	0.22 (0.67)	0 (0.06)	0.20 (0.78)

Table 5 provides an example of how the potential heating and cooling energy benefits change depending on the desired internal temperature as well as the location of the modelled house. Values are presented for an internal set temperature of 20 °C first, and 16 °C in brackets. It is assumed that there is no additional form of heating or cooling, except for the air transferred from the roof cavity. The general trend visible from Table 5(*a*) is that potential heating benefits are increased with a higher desired internal temperature. Table 5(*b*) shows the opposite effect whereby the potential cooling benefits are increased when the internal set temperature is reduced. These trends occur because the occupied house temperature spends a greater proportion of the time below or above the desired temperature for heating and cooling respectively when the set temperature is increased. The difference in energy available to the house is therefore strongly related to the amount of time that air is being pumped down from the roof space into the living areas.

5. Discussion and conclusions

The modelled results for a Dunedin house (DUD1) show that the roof surface is a very good solar collector and is able to heat the roof space air above the ambient temperature when solar radiation is incident upon it. However, the high thermal conductivity and lack of insulation which allows the roof space air to reach such high temperatures also means that this air cools extremely quickly when the incident solar radiation is no longer available. This situation occurs at night when the sun is below the horizon, but also when cloud cover is present, or when there are local features such as neighbouring buildings and trees that may shade the roof surface. The effects of this are enhanced during winter months when the days are shorter and the sun is lower in the sky which creates longer and more intrusive shadows.

During periods when heating is required, the roof space air may spend a large proportion of the time below the internal temperature of the house. This condition is prevalent during winter periods (Figure 4), but also frequently occurs in autumn and spring periods as shown in Figures 3 and 5 respectively. Any air transferred during these times will subsequently displace warmer air from the house and result in a net cooling effect. This is opposite to the desired effect and would result in an additional heating load being required for the house during these periods. If the mechanical ventilation system is controlled to maximise heating and cooling potential however, up to 0.5 kW of continuous heating and 0.9 kW of continuous cooling is possible for any ten day period in the Dunedin house with a desired internal temperature of 18 °C (Table 4).

As shown by Table 5(*a*), the ability of the roof space air to be used as a heat source for the occupied rooms in the house depends on the actual and desired internal house temperatures. Windows in the building envelope allow solar heat gains to the occupied rooms, which mean that internal house temperatures also tend to oscillate on a diurnal cycle. The larger thermal mass and better insulation of the occupied rooms, compared to the roof cavity, results in a smaller and less responsive temperature variation when subject to the same changing ambient conditions. In general, the higher the desired internal temperature of the house, the greater the potential gains from the roof cavity air as the occupied zones of the house will spend a greater proportion of the time below the desired temperature.

Based on the modelled results in Tables 4 and 5, the potential exists that some small heat gains may be possible by pumping the air down from the roof cavity into the occupied rooms during certain periods. However, as shown in Figures 2-5, the majority of the times that any heat benefits may be possible, the occupied rooms are already above the desired temperature for comfort. Any additional heating during these times would therefore be unnecessary, and in some cases undesirable. The same applies for any cooling benefit which may be possible form transferring roof space air into occupied rooms. This occurs as cooling is usually only possible during times when the temperature of the occupied rooms is below the desired level.

If a method of energy storage was employed to absorb any potential gains and release them when desired, Table 3 shows that much larger benefits are possible than just using the air directly. An example of such a storage process could be a hot water heat pump with the evaporator located in the roof cavity, with the heated water contained in a well insulated cylinder and stored for later use.

It has been shown that some small heating and cooling benefits are possible at certain times by pumping air from the roof space to the living areas of the house for most regions in New Zealand. The magnitude of these benefits however, is not significant enough to change the internal temperature of the house to any noticeable extent. The current model has also been optimised to maximise heating and cooling benefits to the living areas of the house by completely shutting down when the thermal conditions are not satisfactory. It is known that this does not occur in practice as most mechanical ventilation systems are designed to maximise indoor air quality, while reducing undesired thermal losses/gains is of secondary importance. Therefore, based on the presented model outputs, it is suggested that the roof space air should not be used directly for heating or cooling purposes in existing typical New Zealand houses. Subsequently, existing *positive pressure*[1] mechanical ventilation systems should not be promoted and marketed on their heating and/or cooling potential.

The health and comfort benefits of using roof space air as a means of household humidity control have not been investigated in the current research. Other potential concerns with interchanging roof and living space air, such as fire/smoke hazards and New Zealand building code requirements, have also not been taken into consideration in the current research. These factors should all be fully investigated in future research before determining the suitability of roof space mechanical ventilation systems to New Zealand conditions.

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[1] This term has been added to clarify this sentence that appears in the journal version of the paper.

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