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Modelling and Simulation of Underfloor Heating System Supplied from Heat Pump

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Abstract— **This paper describes thermal capacity and thermal inertia of an underfloor heating system supplied from a heat pump. A MATLAB/SIMULINK based thermal model of the system have been developed and presented with detailed mathematical equations. For this purpose, experimental results and actual measurements are used to model the energy storage and temperatures. The parameters used for the model are temperature, power and energy consumption, time constant and real-time. Hence, this model can be used to find building temperature variations, heat energy production, electrical energy consumption, instantaneous power, coefficients of performance on a real-time scale based on different control strategies used for demand side management. The developed model in SIMULINK includes the effect of thermal storage in the underfloor heating arrangement as well as the thermal mass of the building itself. Heat loss calculations were carried out to develop this model. The paper also employs different control strategies for operating the developed model.**

***Keywords*- Demand-side management (DSM), energy storage, heat pump, modelling and simulation, thermal inertia, thermal model, underfloor heating**

I. INTRODUCTION

An underfloor heating system supplied by a heat pump in the Renewable Energy Laboratory at Queen's University Belfast (QUB) has been used as the experimental set-up. The main reason for choosing an underfloor heating system is that its large heat capacity allows storage of cheaper electricity in the form of thermal energy. It can be charged during different charging regimes, i.e. when demand is low, when it's windy or when real-time price is low. The thermal storage and time constant allows the heat pump to be switched off for a few hours to avoid system peak demand periods, low wind periods or high price periods. The storage can also be used to provide emergency reserve for power supply-demand matching. The room temperature will not change significantly during the floor discharging time.

The advantages of underfloor heating systems supplied by heat pumps as compared to the conventional radiator-based heating system is the large heat emitting area and the low operational water temperature. The benefit is that a large portion of heat is extracted from outside air or ground instead of using a fuel source. Underfloor heating provides a uniform temperature over the floor, in contrast with concentrated heaters. The efficiency of underfloor heating is also better than radiator-based heaters because less heat is rejected from the roof [1]. Underfloor heating requires low temperature water (35 ºC to 45 ºC) as compared to radiators (60 ºC and above). The former is the range where heat pumps have greater efficiency. Hence, underfloor heating with heat pumps is a suitable combination [2].

The comforts of underfloor heating from heat pumps are becoming more popular and the technology is becoming economically viable because of recent development and competition. In addition to a reduction in emissions, heat pumps can be considered as non-essential load that can be switched off during emergency under-frequency conditions [3]. The installation of around 12,000 units so far in the Republic of Ireland, with further installations under consideration, has prompted research work on the use of heat pump storage capability as well as load flexibility provision to facilitate high wind penetration [4]. The storage capabilities of thermal loads, by allowing them to be charged during periods of high wind generation and switched out during lulls, can actively participate in DSM and ultimately be used for managing wind variability. In addition to the support for system operation, DSM can also play an important role for managing wind variability [5-7].

The UK and Ireland climatic conditions are ideal for utilizing space and water heating using heat pumps because of their higher coefficient of performance. The use of heat pumps further saves fuel and reduces undesired emissions. Experimental results reveal that suitable control of flexible loads can be used to provide DSM as well as emergency reserve, which is shown to be faster than spinning reserve from generation [5]. Simulation results suggest that DSM, in the form of heat pumps, can enable the demand shape to be modified, the ramping duty of conventional generation to be reduced and the reserve burden to be improved.

For a typical heat pump, the operation cycle is shown in Fig. 1. The evaporator interacts thermally with a heat source and the two-phase liquid-vapour refrigerant mixture entering the evaporator partially changes phase from liquid to vapour due to the heat absorbed from the source. Then, the refrigerant is further compressed without thermal conduction to saturated vapours, whose temperature is increased at a higher pressure through a compressor. In the next stage, the refrigerant releases heat to the sink at this temperature with the condenser applied which again transforms the refrigerant to the phase of saturated liquid, and the temperature decreases when the refrigerant passes through the expansion valve to the evaporator [2].

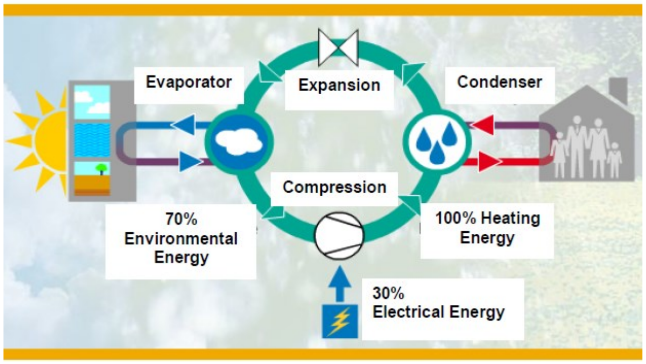

Fig. 1 Heat pump operation cycle [2]

Recent research includes comprehensive reviews of literature related to modelling building energy systems and heat flow [8-10]. This paper is novel in terms of modelling and simulation of a realistic underfloor heating system which is coupled to an air-source heat pump for supplying hot water. The model is based on experimental results, realistic heat loss calculations and embedding into SIMULINK based model for dynamic simulation.

II. Coefficient of Performance

Coefficient of Performance, which is often referred to as COP, is defined as the amount of heat transferred per unit of input work required. Mathematically:

$$COP = \frac{Q_{hot}}{W} \qquad \text{.........................(1)}$$

Q_{hot} is the heat supplied to the hot reservoir and W is the work consumed from the heat pump. For the heat pump, work consumed is in the form of electrical energy.

According to the first law of thermodynamics, in a reversible system we can show that

$$Q_{hot} = Q_{cold} + W$$

and

$$W = Q_{hot} - Q_{cold}$$

Q_{hot} is the heat given off by the hot heat reservoir, Q_{cold} is the heat taken in by the cold heat reservoir and W is the work required to transfer the heat. For heating, equation (1) can be modified to

$$COP_{heating} = \frac{Q_{hot}}{Q_{hot} - Q_{cold}} \qquad \text{................ (2)}$$

For a heat pump operating at maximum theoretical efficiency (Carnot efficiency), it can be shown that

$$\frac{Q_{hot}}{T_{hot}} = \frac{Q_{cold}}{T_{cold}}$$

where T_{hot} and T_{cold} are the absolute temperatures of the hot and cold heat reservoirs respectively. Equation (2) will then become

$$COP_{heating} = \frac{T_{hot}}{T_{hot} - T_{cold}} \qquad \text{.................. (3)}$$

$$COP_{heating} = 1 + \frac{T_{cold}}{T_{hot} - T_{cold}} \qquad \text{...............(4)}$$

Equation (4) demonstrates that COP tends to be higher if the temperature difference between hot and cold regions is lower.

There has been quite a lot of work already done on the COP of various heat pump technologies with ambient temperature variations and the temperature difference of the interior of the buildings over the ambient. Fig. 2 shows COP curves which is the result of an investigation over 100 models of heat pumps [11]. COP variations for the air-source as well as ground-source heat pump reproduced from reference [11] are given below.

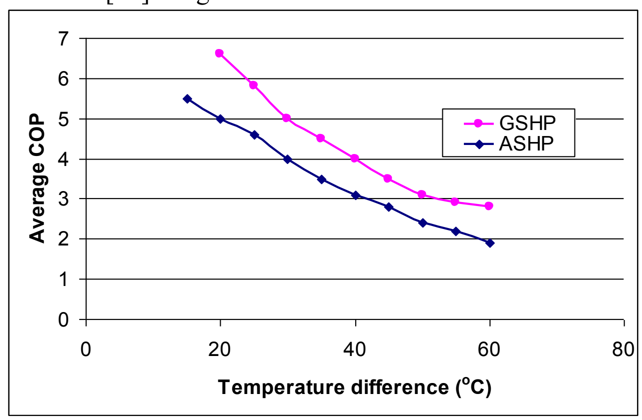

Fig. 2 COP versus ΔT (Reproduced from [11])

The heat pump used in this research is an air-source heat pump. Curve fitting has been used to find a polynomial for COP vs Temperature difference as shown in Fig. 3

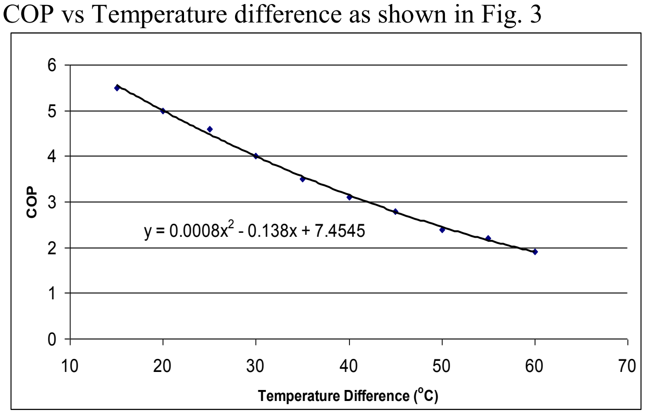

Fig. 3: Relationship between COP and temperature difference

III. Experimental Work

The experimental setup consists of the 10-kW (heat output) air source heat pump coupled with underfloor heating. The room dimensions are approximately 13 × 5 × 4 m, with a concrete floor with underfloor heating covering 12 × 2.5 m. Thermocouples were placed in the floor during construction. LabVIEW has been used to acquire temperature data from thermocouples, in addition to voltage and current from the heat pump. On-line analysis has been performed to calculate electrical power, energy, power factor, etc. and log data for further use. Ambient temperature has been taken from a Weatherlink station installed above the same laboratory.

Fig. 4 shows the temperature variation in the room when the heat pump was turned off. These curves are used to estimate the thermal time constants. Floor temperatures 1 and 2 show the temperature readings taken from thermocouples placed at different positions within the floor.

There is a temperature reduction of only 4°C during a 24-hour time between midnight of Saturday the 17th and Sunday the 18th of January 2009. Time constants for the concrete floor and room temperatures are calculated by assuming an exponential decay for both curves [12].

$$\text{Temperature} = a + b \times e^{(-t/\tau)} \quad \text{..............} \quad (5)$$

a and *b* are constants which depend upon the boundary conditions of the curves, and τ is the time constant. Equation (5) is solved for floor and room temperature curves by applying boundary conditions and selecting sample points of the curves. The resulting equations are as follows:

$$T_{Floor}(t) = 17.4 + 22 \times e^{(-t/13)} \quad \ldots\ldots\ldots\ldots \quad (6)$$

$$T_{Room}(t) = 14.7 + 8 \times e^{(-t/25)} \quad \ldots\ldots\ldots\ldots \quad (7)$$

The time constants for the floor temperature and room temperature are estimated as 13 and 25 hours, respectively. Approximation curves for the floor and room temperatures are shown in Fig. 4 [12].

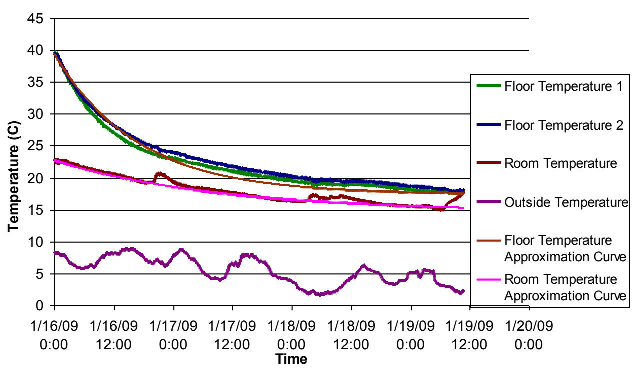

Fig. 4: Decrease in temperature during 4 days of winter [12]

IV. Development of Thermal Model

A thermal model of the system under study has been developed and presented with the default control strategy. This is a two-stage model which includes the thermal storage capacity of the heated floor and the thermal storage capacity of the building itself. Thermal capacities and losses are calculated for the underfloor heating test rig. The symbol P_h has been used for heat transfer instead of Q in order to avoid any confusion, as in electrical systems, Q is used for reactive power. Similarly, P_L has been used to represent heat losses. For heat loss from floor to air P_{Lf} is used and for heat loss from room to outside P_{Lr} has been used.

The thermal mass of the floor is calculated as:

$$\text{Volume of the floor} = \text{length} \times \text{width} \times \text{height} = 12 \times 2.4 \times 0.075 = 2.16\ \text{m}^3$$

$$\text{Density of concrete} = \rho = 2300\ kg/\ m^3$$

$$\text{mass} = M = 2.16 \times 2300 = 4968\ kg$$

$$\text{specific heat of concrete} = C_p = 880\ J/kg.K$$

$$\text{heat transfer} = \Delta P_h = MC_p\Delta T = 4968 \times 880 \times \Delta T = 4371840\Delta T$$

$$\Delta T = \Delta P_h / 4371840 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

The model subtracts heat losses from the heat supplied via a feedback loop as shown in Appendix A. So ΔP_h comes out to be the net heat supplied to the thermal store which is equal to the heat supplied from heat pump (P_{hp}) minus the heat rejected to the room from the floor. ΔT is the temperature rise above the reference temperature. Reference temperature for floor temperature is room temperature while the reference temperature for room temperature is the ambient temperature. The above equation has been integrated over time to get absolute temperatures represented in ºC [13]. So, ΔP_h can further be represented as:

$$\Delta P_h = P_{hp} - P_{Lf}$$

The loss component for this storage has been considered from the floor to inside air which can be calculated as under [14]:

$$\text{heat loss (floor to inside air)} = P_{Lf} = hA(T_s - T_a) \quad \ldots\ldots \ (9)$$

h is the heat transfer coefficient and *A* is the surface area. T_s and T_a are the surface and air temperatures respectively.

$$A = \text{length} \times \text{width} = 12 \times 2.5 = 30\ \text{m}^2$$

The heat transfer coefficient depends upon the environmental conditions and the temperature difference of surface and air. A simplified equation for free convection with a heated plate facing upward is employed in the thermal model [14]:

$$h = 1.52(T_s - T_a)^{1/3} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

Equation (9) will now become

$$P_{Lf} = 1.52(T_s - T_a)^{1/3} \times 30 \times (T_s - T_a) = 45.6 \times (T_s - T_a)^{4/3} \quad \ldots\ldots\ldots (11)$$

Equation (11) gives heat transferred from the floor to the inside air.

The structure of the room also has inherent heat storage. A 224 m² house in Northern Ireland has a heat loss of approximately 353 W.K⁻¹ [15]. The area of the room under study is 68 m². The heat loss for this room is approximately 102 W.K⁻¹. The heat loss has been calculated for this room using the basic formula:

$$P_L = U \times Area \times (T_{in} - T_{out}) = U \times A \times \Delta T \quad \ldots\ldots\ldots \ (12)$$

U values of walls, windows and roof are 0.27, 1.9 and 0.16 and the areas are 33.26 m², 43 m² and 68.33 m² respectively [16]. So, the calculated heat loss is

$$P_{Lr} = (0.27{\times}33.26 + 1.9{\times}43 + 0.16{\times}68.33)\ \Delta T$$

$$P_{Lr} = 101.62\ \Delta T \ldots\ldots\ldots\ldots\ldots\ldots\ldots (13)$$

Heat losses are taken as 120 W.K⁻¹ which includes heat losses to the uncovered floor and ventilation.

The incoming heat in the room has been taken from the heat loss from floor to room air. Again the similar methodology has been used to subtract heat losses using a feedback loop. The thermal mass of the structure also stores heat energy. The thermal capacity of the structure itself is calculated from the experimental measurements of time constant. The time constant, from experimental results, is approximately 25 hours for the room temperature. The heat storage capacity will be τ times the rate of heat loss as given in [15]

$$\tau = \frac{MC_p\Delta T}{KA\Delta T} \quad \text{..............................} \quad (14)$$

$$Capacity = MC_p = (25 \times 3600) \times 120 = 10{,}800{,}000\ J/K \quad \text{..................}(15)$$

$$\Delta T = (P_{Lf} - P_{Lr})/10{,}800{,}000 \quad \text{.............} \quad (16)$$

This change in temperature has been integrated over time to get the absolute value of room temperature in ºC. Actually, this equation gives temperature results in Kelvin but 1º change in Kelvin is equal to a similar change in ºC. Moreover, the reference temperature is given in ºC. The equations calculate change in temperature above the reference value. So, the resulting unit can be taken as ºC. The block diagram for the developed model is given in Fig 5.

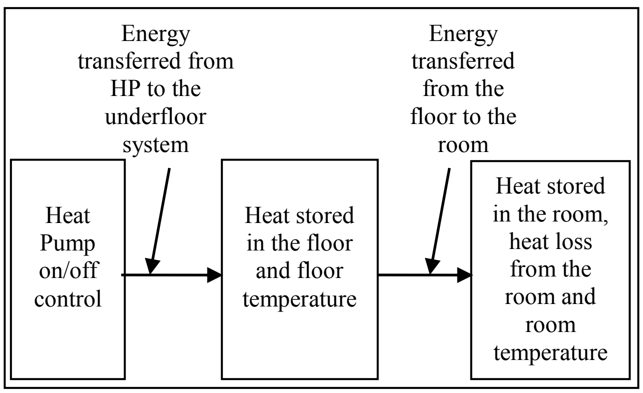

Fig. 5 Block diagram of the thermal model

V. MATLAB SIMULINK BASED MODEL IMPLEMENTATION

The MATLAB (Simulink) models are given in Appendix A and B, where all parameters are given. In addition to the block diagram, the Simulink model includes some additional blocks for input, output and display of data during simulation. The developed model has been used for calculation of building temperatures under different control strategies.

VI. SIMULATION RESULTS

The default control strategy is based on the building thermostat which keeps the heat pump running in the dead band of ±0.5 ºC of the set-point temperature. Normally, the set-point is 20 ºC. The thermal energy storage inside the floor causes a significant delay in achieving the room temperature as it also depends on the stored energy. The stored energy keeps heating up the room even after the heat pump is off. On some occasions, when the floor temperature is low, it takes some time to raise the temperature to a level which can supply enough heat energy to the room to overcome the thermal losses and raise the temperature to the required level. If the set-point is 20 ºC, with a dead band of ±0.5 ºC, the actual variations in the room temperature are in the range of 19 ºC to 21.5 ºC. To find out the range of temperature variations, the set-point has been set to 19 ºC and the variations obtained are from 18.5 ºC to 20.7 ºC. The on-off status shows the flexibility of this system for load deferment and to participate in demand-side management and providing reserve on demand. This flexibility could be very useful for matching supply and demand. Fig. 6 shows the change in house temperature with the default control strategy based on the thermostat.

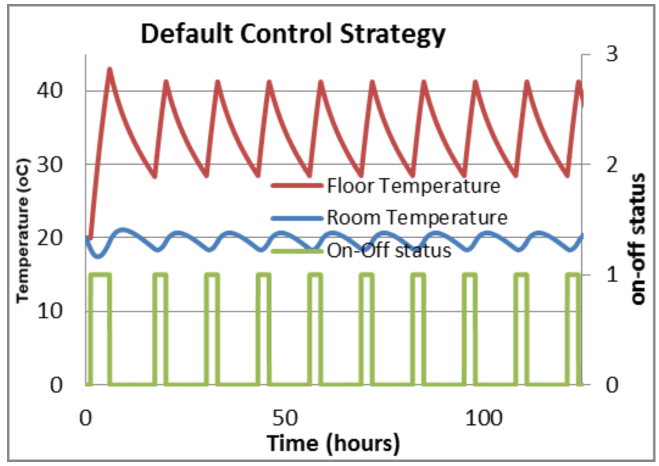

Fig. 6 Temperature variations for 5 days for the default control strategy

The average temperature is higher than the set-point temperature because it takes less time to charge the floor than it's discharging but this statement is true in most of the cases except for extremely cold periods when it takes more time to charge the floor than discharging. For default control strategy, this happens below -10 ºC.

In order to see the operation of heat pump model in extremely cold weather, say below -10 ºC, the COP is approximately 2.0 as given in Fig. 3. The temperature variations are given in Fig. 7.

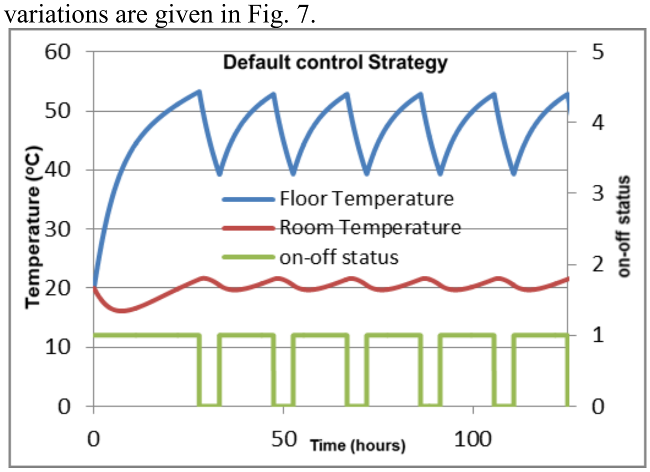

Fig. 7 Temperature variations at -10 ºC ambient temperature

The set-point temperature is also required to be increased as mentioned before that it will take longer to charge the floor than its discharging and the average temperature will be less than the set-point. For this case, the set-point has been taken as 21 ºC.

Now, how would the temperatures vary if the heat pump load is set to follow some specific tariff, say Economy 7, which provides 7 hours of cheap electricity at night. The Economy 7 tariff is offered by the distribution network operators by supplying cheap electricity between 12:00 am and 7:00 am. Appendix B shows a change in the control strategy in order to operate on the Economy 7 tariff.

When the initial temperature is set to 20 ºC, the ambient temperature was varied from 10 ºC to 2 ºC and the room temperature varies between 18.5 ºC to 23 ºC when the temperature was above 5 ºC . For the lower values of ambient temperature, i.e. 2 ºC, the temperature variations are between 17.5 ºC to 22.4 ºC. The lower temperature is a bit of problem. That is why it is suggested to operate this heating system on Economy 7 tariff above 5 ºC of ambient temperature.

The performance of heat pump with underfloor heating system affects severely for ambient temperatures below 0 ºC. Fig. 8 shows the temperature variations when ambient temperature was considered as -10 ºC (ambient). The average COP is approximately 2.0.

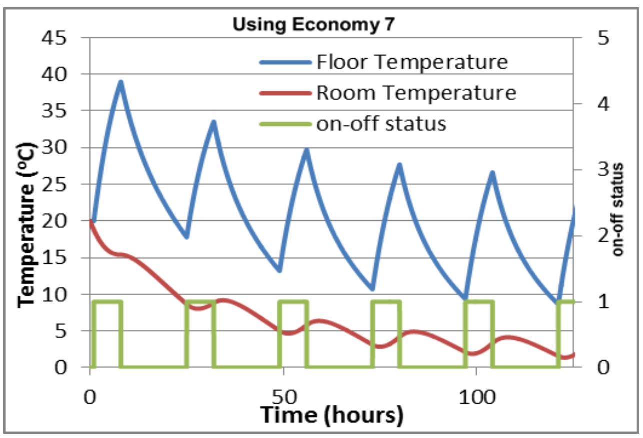

Fig. 8 Performance at -10 ºC and Economy 7 tariff

The thermal model developed can be used to schedule heat loads to meet the heat demand determined from the generation scheduling process. The inside temperature of a house can also be determined from the model. The future work will focus to integrate this approach at a large scale and to use it for supporting more wind into the network. This work would also be useful to find electrical energy consumption for assessing the impact on distribution networks. Also, the effect of temperature variations on instantaneous and average coefficient of performance will be focused in the future.

VII. CONCLUSION

This paper describes modelling and simulation of underfloor heating system supplied from a heat pump. The renewable energy laboratory at QUB has been used for this purpose. The actual dimensions of the lab and ratings of the heat pump and underfloor system are used to model the system and implement in MATLAB SIMULINK for analysis. Then default control strategy, which aims to maintain the room temperature within comfortable range regardless of price or time of use, and Economy 7 tariff are used to test the model. The model can be used to find building temperatures and to include these type of loads for demand side management and to support wind variability.

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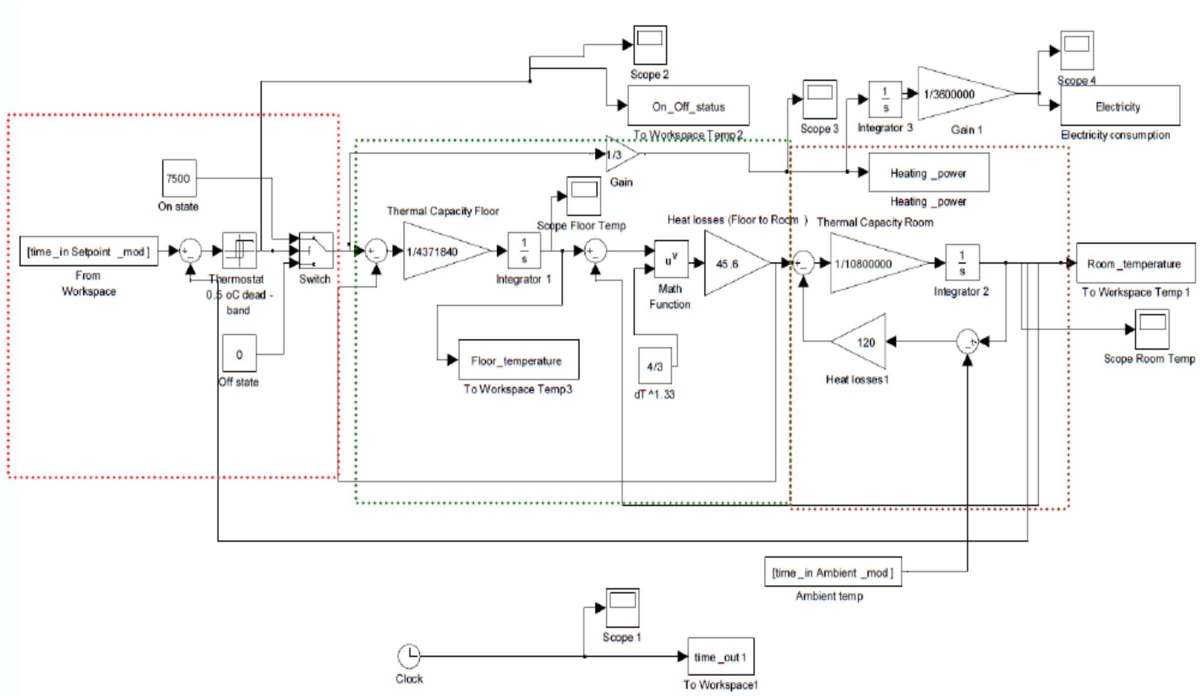
Appendix A. Simulink model with default control strategy



Appendix B. Changes in the control strategy to operate on Economy 7 tariff