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EVALUATION OF A GROUND SOURCE HEAT PUMP SYSTEM FOR VARYING LOADS AND RETURN WATER TEMPERATURES USING DIFFERENT SYSTEM PERFORMANCE FACTORS

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ABSTRACT

A quasi-steady state mathematical model of a ground source heat pump has been developed. Heating and cooling is delivered to the building using fan coil units, which are either in constant operation (Strategy 1) or switched on/off as water flow to the coil is diverted (Strategy 2). The performance of the system is evaluated using a system performance factor (SPF). It was observed that values for SPF_4 (which includes all components) decreases for decreasing building load. For strategy 2, the energy saving associated with SPF_4 are greater than Strategy 1, but savings are diminished at higher loads. Improved system performance was observed at lower return water temperatures in heating and higher return temperatures in cooling. However, in both cases the capacity of the fan coil units is reduced. The temperature compensation strategy developed resulted in an improved SPF_4 for Strategy 1 and a less significant improvement for Strategy 2.

1. INTRODUCTION

The use of ground source heat pumps (GSHPs) for heating and cooling, particularly when low-carbon electricity is utilised, has generated considerable interest in the reduction of CO2 emissions within the building energy sector. In an effort to further improve the cost effectiveness of GSHP systems, many studies have focused on improvements in efficiency. Considerable advancement in heat pump design has been evident in recent years. Many of the associated improvements can be attributed to incremental development of key system components including: evaporators, condensers, compressors and expansion devices. Although these advances are central to improving heat pump system performance, the issue of better system integration has received less attention to date, but is increasingly being regarded as equally critical in the optimisation of overall heat pump system performance. Central to improved system integration is the use of more sophisticated control algorithms. A key measure in assessing the energy saving potential of different control strategies is the use of a season performance factor, rather than the coefficient of performance (COP) index, to assess system performance over an extended period. Air source heat pumps (ASHPs) are currently the most utilised type of heat pump, particularly for cooling applications, while ground source heat pumps (GSHPs) have predominantly been implemented in space heating applications to date (Urchueguía *et al.,* 2008). The use of GSHPs in warmer climates for heating and cooling application has not received much attention. The energy saving potential of night set-back techniques has been studied by Guo and Nutter (2010) and Tsukamoto *et al.* (2010). Guo and Nutter noted that for buildings with a large thermal mass an optimal night set-back temperature existed that reduced power consumption. Albieri *et al.* (2008) tested on/off compressor control for cooling using building supply, return and floating supply water set-point temperature. For water source heat pumps, Malmberg and Mattsson (2008) examined the effectiveness of free cooling techniques. Variable vs. fixed speed capacity control has also been studied (Da Silva *et al.,* 2010). For GSHP, some studies have looked at variable speed circulation pump control (Karlsson and Fahlen, 2005) and variable speed compressor compared to fixed speed (Fahlén and Karlsson, 2007). For variable speed the efficiency was shown to increase by 4-11% when operating at part load with less significant saving at higher load. The motivation for this study is to develop control strategies aimed at system level integration that considers the performance of all system components, from source to sink for both heating and cooling applications.

2. HEAT PUMP SYSTEM

2.1 Description of Heat Pump System

A modular mathematical model of a GSHP system was developed based on an installed water-to-water reversible ground source heat pump test rig. The heat pump system is installed at the Universidad Politécnica de Valencia (UPV), Spain and has a rated capacity of 18.5 kW (building 35/40°C, borehole 15/10°C) for heating and 15 kW for cooling mode (building 15/10°C, borehole 30/35°C). A schematic represenation of the system is shown in Figure 1. The GSHP system contains a fixed speed compressor, variable speed internal and external circuit circulation pumps and a buffer tank on the internal hydronic circuit. The external circuit contains six boreholes and a series of fan coil units are used on the building side to supply the required heating or cooling capacity. The on/off cycling of the heat pump compressor is controlled by the return water temperature set-point and bandwidth, while the fan coil units bypass valves are controlled by the space temperature set-point. The ground circulation pump is coupled electrically with the heat pump compressor and cycles in tandem with the compressor action.

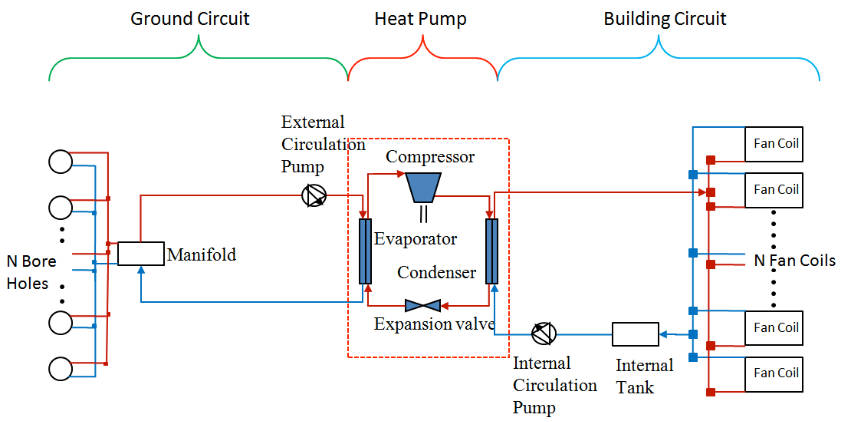

Figure 1: Schematic representation of ground source heat pump system.

2.2 Heat Pump Model Development and Validation

The overall mathematical model was constructed on the basis of extensive experimental data gathered from the test rig. Empirical models were developed for the circulation pumps and fan coil units while the buffer tank model was developed analytically. The heat pump model was constructed using IMST-ART, a program for modelling steady state vapour compression cycles (Corberan *et al.,* 2002). Power consumption and capacity of the heat pump was characterised as a function of circulation pump speeds and return water temperatures using curve fitting methods. This was done for both heating and cooling and the component models were combined in MATLAB to create an overall quasi-steady state system model (MathWorks, 2007). The simulation model was validated against collected data from the test rig in terms of heating capacity, cooling capacity and power consumption. Further details of the model are available in Corberan and Finn (2011).

2.3 System performance factor (SPF) definition

While the COP is a measure of the instantaneous performance of the heat pump, the seasonal performance factor refers to the performance of system components over a heating or cooling season and therefore gives a more realistic indication of overall system performance. Several definitions of seasonal performance factor have been proposed by Nordman *et al.* (2010). For this study, a similar method is used to define system performance factor (SPF) which defines system performance over a specified interval of time, in this case 24 hours. SPF_1 considers the heat pump alone and SPF_2 includes both the heat pump and external circulation pump. SPF_3 also considers the internal circulation pump and SPF_4 includes the fan coils units, as shown in Figure 2. Equations (1)-(4) display these definitions.

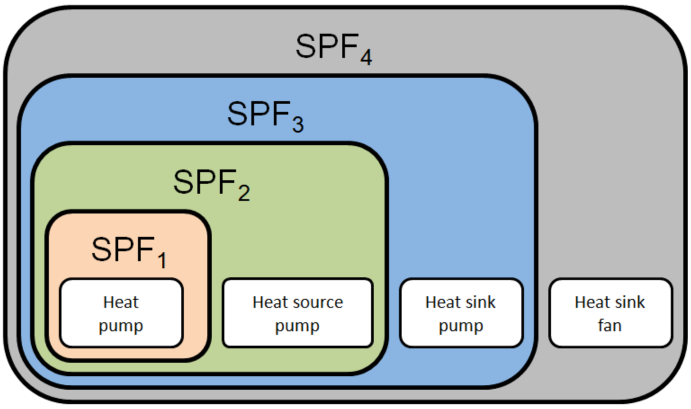

Figure 2: System boundaries for calculations of SPF.

$$SPF_1 = \int_0^{24\ hours} \frac{\dot{Q}_{HP}}{P_{HP}} \qquad (1)$$

$$SPF_2 = \int_0^{24\ hours} \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump}} \qquad (2)$$

$$SPF_3 = \int_0^{24\ hours} \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump} + P_{B\_pump}} \qquad (3)$$

$$SPF_4 = \int_0^{24\ hours} \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump} + P_{B\_pump} + P_{B\_fans}} \qquad (4)$$

3. RESULTS

A series of sensitivity studies were conducted to study the effect of the return water temperature set-point and building load on the system performance in both heating and cooling mode. The simulation was carried out under quasi-steady state conditions subject to a constant external ambient temperature for a 24 hour period. The performance of the system is expressed in terms of SPF, as defined in Section 2.3, such that the energy consumption is normalised over a one hour period. External ambient temperatures of 26°C to 32°C in cooling mode and 8°C to 14°C in heating mode were tested. The return water temperatures are varied from 35°C to 45°C and 9°C to 15°C in heating and cooling mode respectively. The room temperature set-point is 22°C in heating and 23°C in cooling, while the room temperature and return water temperature bandwidths are ±1°C in all cases. With regard to fan coil control, two strategies are tested. Strategy 1 assumes the fan coils are in constant operation. Strategy 2 assumes the fan coils are switched off when the water flow is to it is diverted.

As shown in Figure 3, for both heating and cooling, the compressor is observed to be the largest energy consumer. The energy consumption of the compressor decreases for increasing external ambient temperatures in heating mode and decreasing temperatures when cooling. As the external circulation pump on/off control is linked to that of the heat pump a change in heat pump power consumption results in a proportional change in the external pump consumption. In heating mode, a constant water return temperature of 40°C was utilised, while in cooling mode, a constant water return temperature of 12°C was used. For Strategy 1, energy consumption is observed to be constant as the units are in operation all the time. At low building load, this energy consumption constitutes a higher proportion of the total energy consumption of the system. For Strategy 2 it can be seen that the fan coil power consumption increases for increasing building load.

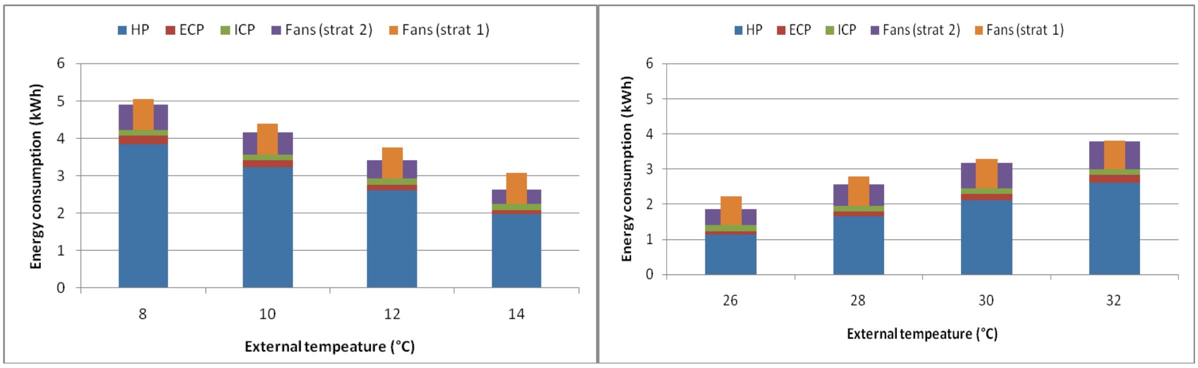

Figure 3: Normalised energy consumption of the heat pump for one hour, internal and external circulation pumps and fan coils in quasi-steady state for different external temperature in (a) heating and (b) cooling mode.

Figure 4 shows SPF for the case studies presented in Figure 3. The values of SPF can be seen to be higher in cooling mode than in heating, which is mainly attributed to the smaller temperature lift associated with the heat pump operation in cooling mode. A constant value of SPF_1 and SPF_2 is observed over the range of external temperatures. This is because an increase in building load and thus required capacity is matched by a proportional increase in power consumption for both the heat pump and external circulation pump, bearing in mind that the water return set-point is unchanged. Considering SPF_3 and SPF_4, for Strategy 1, both SPFs are observed to increase with higher building loads. As mentioned earlier, the internal circulation pump and fan coil energy consumptions constitutes a smaller proportion of the total consumption at higher loads, therefore the SPF is increased. Considering Strategy 2, SPF_4 is higher than for Strategy 1 at lower loads and increases with building load. This is due to the reduced fan coil energy consumption when the system is not heating or cooling. At higher loads the fan coils are in operation the majority of the time so any savings potential is reduced.

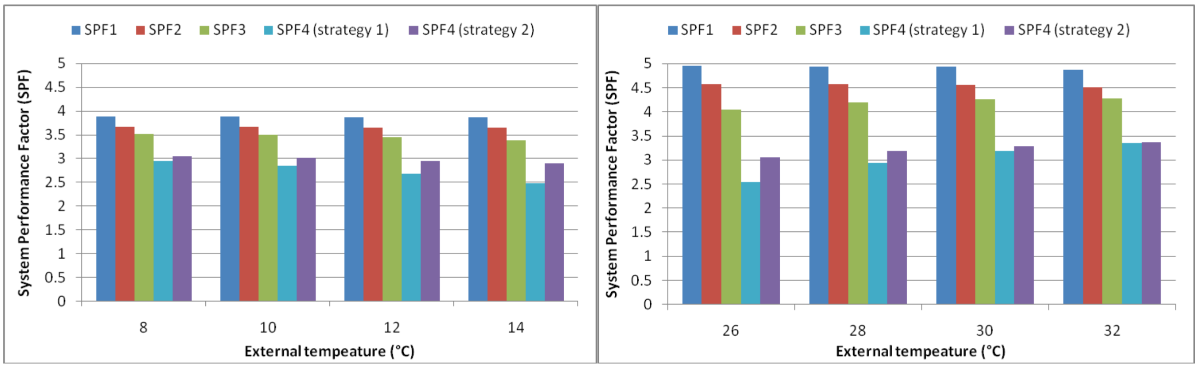

Figure 4: Quasi-steady state results for SPF_1, SPF_2, SPF_3 and SPF_4 for strategy 1 and 2 in (a) heating and (b) cooling mode for different external temperature set-points.

The variation in SPF_4 for Strategies 1 and 2 for varying external air temperatures and return water temperatures are shown in Figure 5 for heating and Figure 6 for cooling. As the return water temperature decreases in heating and increases in cooling, the temperature lift across the heat pump is reduced. In heating mode, this results in a slight increase in capacity and more significant decrease in heat pump power consumption. The reduced temperature lift across the compressor results in a lowered condenser saturation temperature. However, the lowered condenser saturation pressure does not affect the refrigerant mass flow-rate through the condenser, but does result in a small capacity increase. Therefore, there is little effect on the heating capacity of the system. In cooling mode, a reduced heat pump temperature lift results in an increased evaporator saturation pressure. This results in an increased refrigerant mass-flow which increases the cooling capacity of the system. The power consumption is also reduced in this case. Because of this, a greater change in SPF is observed per degree change in return water temperature in cooling mode than in heating.

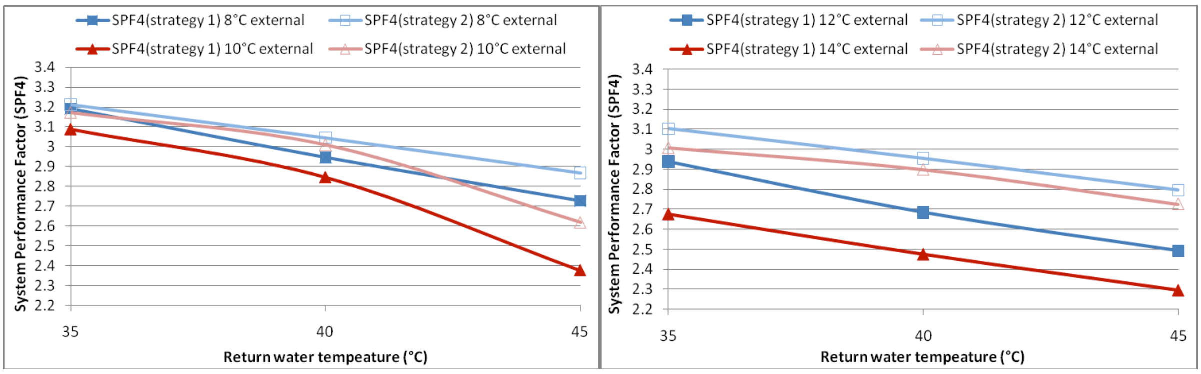

Figure 5: Quasi-steady state results for SPF_4 for Strategies 1 and 2 in heating mode for external temperatures of (a) 8°C and 10°C and (b) 12°C and 14°C and return water temperature set-points.

For all return water temperatures, a greater increase in SPF_4 (Strategy 2) is observed for low building loads when compared to SPF_4 (Strategy 1). However at higher building loads, the relative SPF improvement is reduced for decreasing return water temperatures in heating and increasing temperatures in cooling. This is because a lower return water temperature in heating and higher temperature in cooling, while improving the heat pump COP, will reduce the temperature difference across the fan coil units and thus the space heat transfer capacity. This will increase the operational time of the units and thus power consumption for Strategy 2. Due to this, a less substantial change in SPF_4 for Strategy 2 is observed when compared to Strategy 1 for a change return water temperatures.

At an external temperature of 32°C in cooling, the system is operating above capacity. Therefore, to improve the SPF value, a lower return water temperature would be desired in heating mode and a higher return water temperature in cooling. However as mentioned, this results in a reduced heat transfer across the fan coil units and thus compromises delivered system capacity. In cooling mode, at an external temperature of 30°C, the fan coil units do not supply sufficient load for a return temperature of 15°C despite the increased cooling capacity of the heat pump at this higher temperature. At return temperatures of 9°C and 12°C, sufficient load is supplied to the building

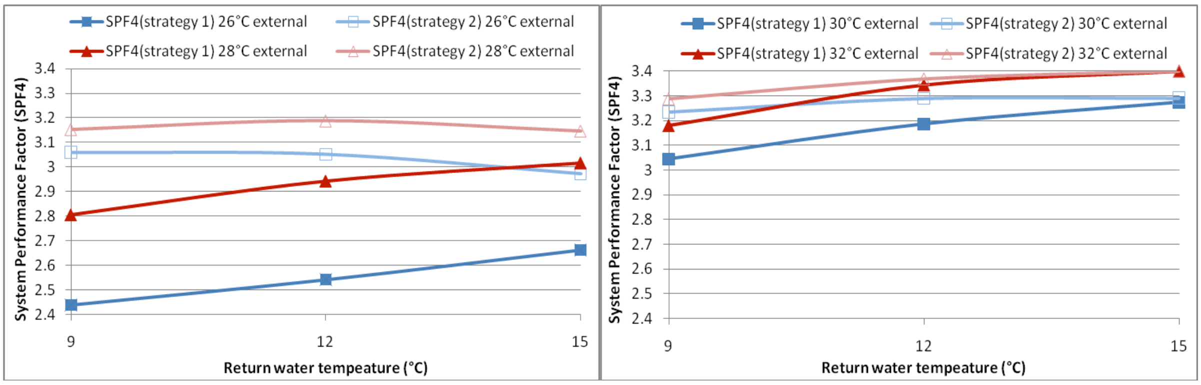

Figure 6: Quasi-steady state results for SPF_4 for strategy 1 and 2 in cooling mode for external temperatures of (a) 26°C and 28°C and (b) 30°C and 32°C and return water temperature set-points.

Using the results shown in Figure 6, a strategy was developed in cooling mode where the return water temperature is controlled using the external ambient temperature to maximise the value of SPF_4 while maintaining the space temperature within the specified bandwidth limits. The return water temperature is set to 15°C for external temperatures below 29°C, 12°C for temperatures between 29°C and 31°C and 9°C for external temperatures above 31°C. These strategies were tested for a 24 hour diurnal profile with a varying external temperature profile, such that the system is in operation between 7am and 10pm. Figure 7 shows the return water, supply water, external air and space air temperature (of a single zone) profiles using the return temperature compensation strategy.

The SPF_4 values for Strategy 1 and 2 are shown in Figure 8 for temperature compensation and for a fixed return water temperature of 9°C. For Strategy 1, an increase in SPF_4 is observed for the temperature compensation algorithm when compared to a return temperature of 9°C. This is because the return water is set to 12°C and 15°C degrees with reduced building load. At these temperatures the SPF_4 is increased by 4.6% from 2.4 to 2.51. For Strategy 2 a less significant increase in SPF_4 is observed. This is due to the reduced SPF_4 for varying return water temperatures particularly at high load.

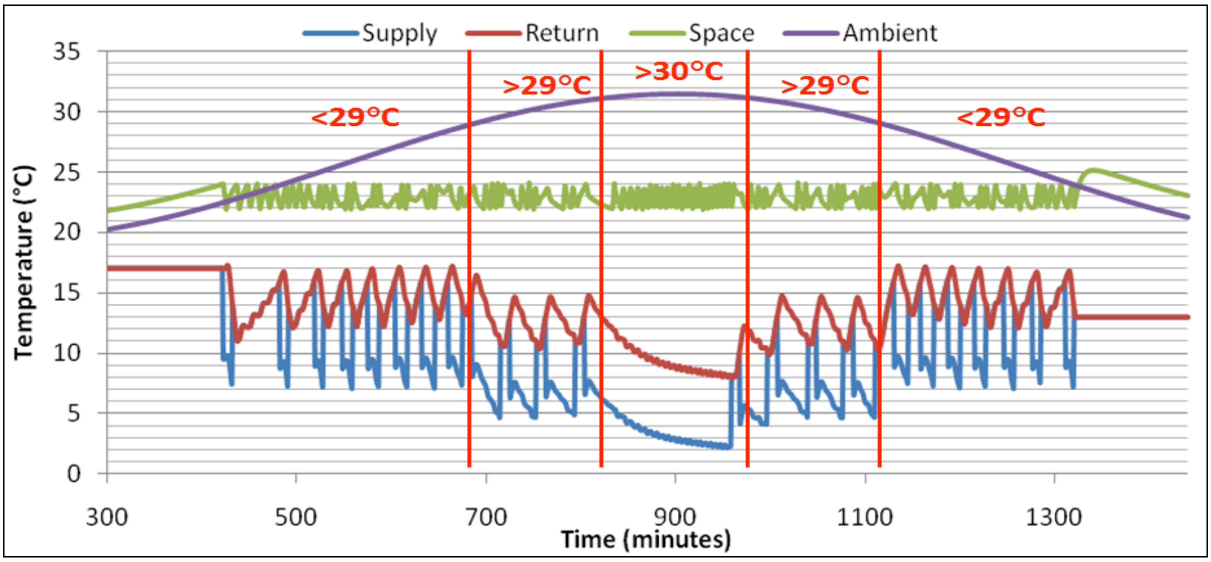

Figure 7: Daily temperature profile of return water, supply water, external and space air using the temperature compensation strategy from 5am to 12am.

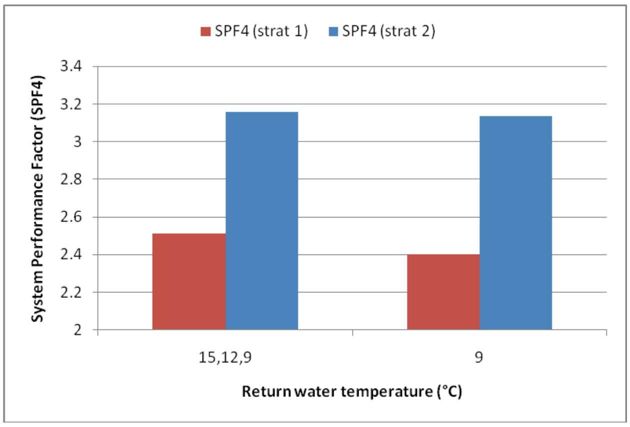

Figure 8: SPF_4 for Strategies 1 and 2 for the temperatures compensation strategy and a fixed return water temperature of 9°C.

4. CONCULSIONS

By considering the power consumption of all the units in a GSHP system, lower values of system performance factor (SPF_4) is observed than when considering the heat pump and external circulation pump (SPF_2) alone. Values of SPF_4 for were shown to decrease for reduced building load for both Strategy 1 and 2. However, at an increased building load, possible savings due to Strategy 2 were diminished. Better system performance was observed at lower return water temperatures in heating and higher temperatures in cooling; however the capacity of the fan coil units is reduced. The temperature compensation strategy developed resulted in an improved SPF_4 for Strategy 1 and a less significant improvement for Strategy 2. If the study was performed of a low load factor, the SPF_4 values would be improved.

5. NOMENCLATURE

P	power consumption [W]
$\dot{Q}$	heating/cooling energy output [W]

Subscripts

B	Building	NFC	No fan coils
S	Source	HP	Heat pump

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