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Effective Operation for Heat Cascade System of Absorption Chiller and Desiccant Air Handling System with CHP and Solar Thermal

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Abstract

Temperature and humidity independent control (THIC) of air conditioning system treats sensible and latent heat load separately. One of the representative THIC systems is a combination of desiccant air handling unit (DAHU) and sensible load handling unit which does not require low temperature chilled water for dehumidification. However heat is necessary to regenerate the hygroscopic agent which contains moisture. In many cases heat source such as exhaust heat of combined heat and power (CHP) or solar thermal will be appropriate for that purpose. Heat cascade system including CHP, solar thermal, absorption chiller of gas fired double effect with hot water single effect (ABS-GH) and DAHU is proposed in this study. Temperature of outlet chilled water and inlet cooling water of ABS-GH is important parameter for effective operation of the proposed system. Therefore in this paper, the effect of these parameters on the performance is investigated using static simulation tool. The results show that cascade hot water temperature is lower as chilled water temperature is higher and as cooling temperature is lower. It contributes to the larger use of solar heat. The best performance of ABS-GH is achieved in case with 12°C and 28°C of chilled water and inlet cooling water temperature respectively. However, due to the difference in operation time of CHPs and solar thermal, the best coefficient of whole system performance is achieved 0.74 when chilled water temperature and inlet cooling water temperature are set at 12°C and 30°C respectively.

Keywords - Effective operation; Hot water cascade utilization; Absorption chiller; Combined heat and power; Desiccant air handling unit; Solar thermal system; System simulation

1. Introduction

The climate in Japan is hot and humid in summer therefore sensible and latent heat load should be processed efficiently. Temperature and humidity independent control (THIC) of air conditioning system treats sensible and latent heat separately. One of the representative system for latent heat process in THIC systems is desiccant air handling unit (DAHU); hygroscopic agent removes moisture in air and the dried air is supplied to rooms. The combination of DAHU and sensible heat conditioning unit (SAHU) does not require low chilled water temperature which is used for condensation dehumidification. However, heat is necessary to regenerate the hygroscopic agent which contains moisture. In many cases heat source such as exhaust heat of combined heat and power (CHP) or solar thermal will be appropriate for that purpose. In this study, heat cascade system including CHP, solar thermal, absorption chiller of gas fired double effect with hot water single effect (ABS-GH) and DAHU is proposed for an office building.

Temperature of outlet chilled water and inlet cooling water of ABS-GH is important parameter for effective operation of the proposed system. Therefore, in this study, the effect of these parameters on the performance is investigated for the hottest week with static simulation tool.

2. Air Conditioning System and Heat Source System

The object office building is located in Shizuoka, in Japan. It has 5 stories including exhibition space on the 1st floor and office space from 2nd to 5th floors. The total floor area is 7516m^2. This study focuses on only central air conditioning system for the office space. Figure 1 shows the air conditioning system for the office space. Fresh outdoor air is supplied to a DAHU directly or through earth to air heat exchanger (EAHEX) depending on the outdoor condition. EAHEX means that fresh air flows through cellular mat foundation and it is cooled in summer or heated in winter by heat exchange with soil. The DAHU handles fresh air load and indoor latent heat load. SAHUs at each air conditioning zone handle indoor sensible heat load. For common space such as corridors, there are fan coil units.

The DAHU in this study is composed of pre-cooling coil, dehumidification wheel, sensible heat exchanger, after-cooling coil and regenerating coil in Figure 1. Fresh air cooled by EAHEX is provided to pre-cooling coil to be pre-cooled and dehumidified, and then passes dehumidification wheel, sensible heat exchanger with return air and after-cooling coil. Return air passes sensible heat exchanger, and then it is heated up at regenerating coil heat source of which are solar thermal and heat from CHP. High temperature air will desorb moisture from hygroscopic agent. The high temperature air has limitation that the maximum air temperature depends on the outlet temperature of cascade hot water passed through ABS-GH. Supply air is set at 28°C and 6.25g/kg' (26.7%RH). If the hot water is not enough to handle absolute humidity for set point, the supply air absolute humidity will be raised.

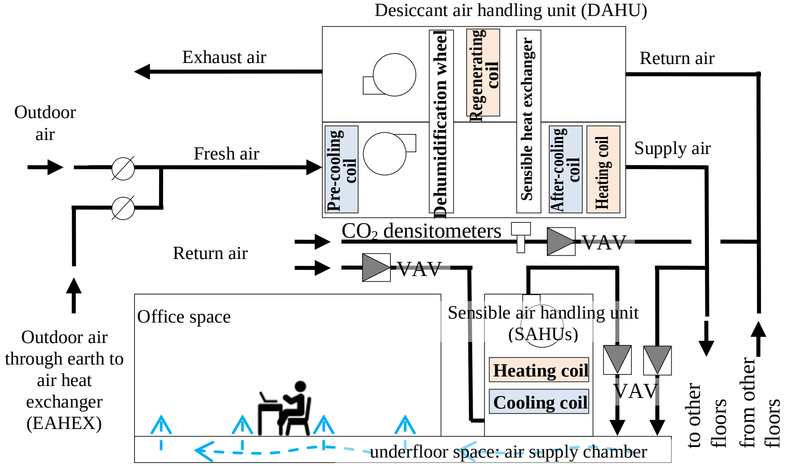


Fig.1 outline of air conditioning system

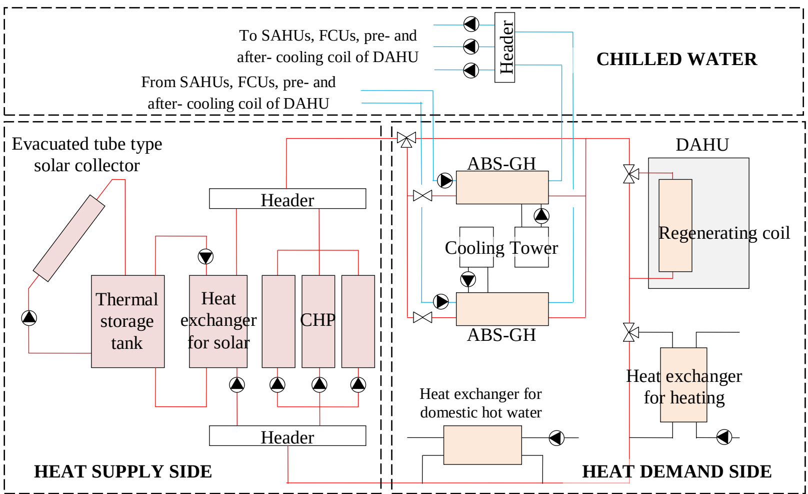


Fig.2 outline of heat source system

Figure 2 shows the heat source system and Table 1 shows the specifications of the main components. In cooling period, two ABS-GH provide chilled water to SAHUs, pre- and after-cooling coil in DAHU, and fan coil units. Three CHPs and solar thermal system are connected in parallel therefore mixed hot water from them is supplied to firstly one of ABS-GH and then regenerating coil of DAHU. This hot water is called as cascade hot water.

The operation time of CHPs is from 8:00 to 17:00. When the temperature in solar thermal storage tank is higher than 85°C, solar heat is utilized instead of one of CHPs, then the pump for solar thermal heat exchanger stops when the temperature in the tank is lower than 75°C. Measurement data of the system such as inlet and outlet temperature of air and water, flow rate, velocity, and energy consumption have been measured and stored with building energy management system (BEMS) since 2013.

Table 1. specifications of main components

Components	Rated Specifications	Number
Combined heat and power (CHP)	power generation: 25 kW, heat generation: 38.4 kW electrical conversion efficiency: 33.5 % thermal efficiency: 51.5 %	3
Solar collector	total area: 83.6 m^2	
Thermal storage tank	2 m^3	1
Heat exchanger for solar (HEX_SOL)	50 kW	1
Absorption chiller of gas fired double effect with hot water single effect (ABS-GH)	cooling (heating) capacity: 281 (186) kW hot water input: 110 kW gas consumption: cooling: 17.2 Nm^3/h(without hot water input) 12.0 Nm^3/h(with hot water input) heating: 17.3 Nm^3/h	2
Cooling tower	521.9 kW	2
Desiccant air handling unit (DAHU)	Regenerating coil: 93 kW pre-cooling coil: 103 kW after-cooling coil: 29 kW	1 1 1

3. System Simulation

The simulation tool used in this study is Life Cycle Management Tool (LCEM tool)[1] which has been developed under supervision of Ministry of Land, Infrastructure, Transport and Tourism. This simulation tool is static simulation with 1 hour step. By connecting all simulation model called "object" representing system components such as CHP and solar thermal collector, whole system model is constructed as shown in Figure 3. Each object is improved for better precision based on the measurement analysis. This revision results in good agreement for not only energy consumption but also air condition in DAHU[2][3].

The simulation is conducted for continuous representative days which recorded the highest temperature and cooling load in 2015. In cooling season, the heat exchanger for heating and domestic hot water are not under operation. The outdoor boundary conditions including outdoor temperature, relative humidity and solar radiation were provided by measurement data. Figure 4 and 5 show the provided boundary data. During the representative days, as outlet condition through EAHEX is lower than outdoor condition, boundary air condition (temperature, humidity and air volume of supply air and return air) to pre-cooling coil is also given by measurement data. Total cooling load of SAHUs and FCUs is also provided by measurement data. The target room condition is 28°C and 40%RH.

12 simulation cases are conducted in which chilled water temperature of ABS-GH (7°C, 10°C, 12°C) and lower set value (LSV) of inlet cooling water temperature (26°C, 28°C, 30°C, 32°C) are the parameters. Simulation case is named as case_"chilled water temperature"-"LSV of inlet cooling water".

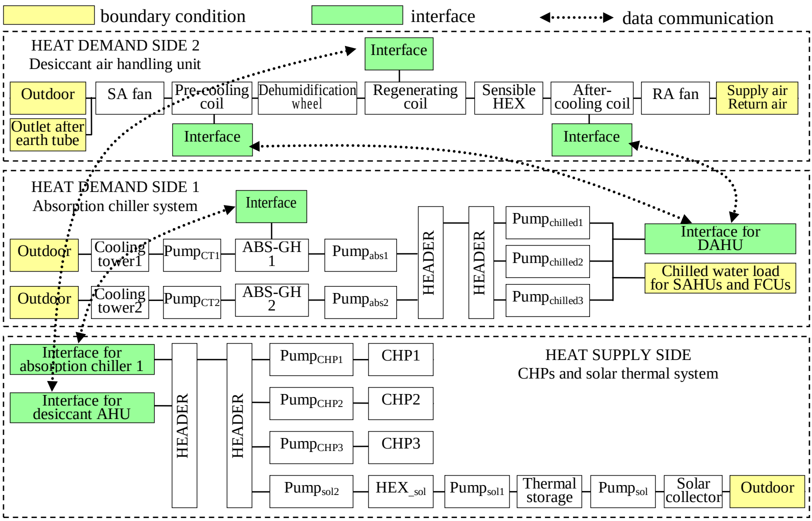


Fig.3 system model of LCEM tool

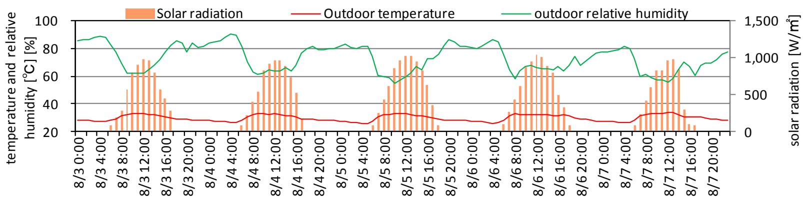


Fig.4 outdoor boundary condition

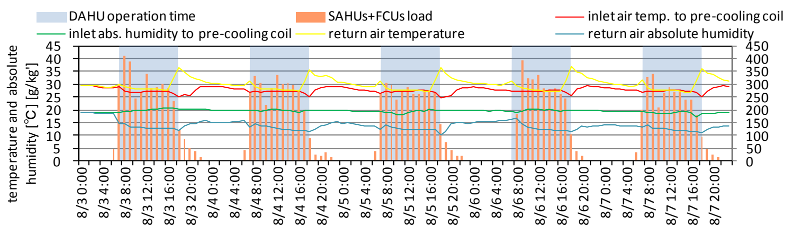


Fig.5 boundary condition for DAHU and chilled water load of SAHUs and FCUs

4. Simulation Results

When chilled water temperature is low at 7°C or when cooling water temperature is high at 32°C, the cascade hot water temperature (inlet hot water temperature) to ABS-GH is higher than that in case of 12°C for chilled water temperature or 28°C for cooling water as shown in Figure 6. The difference in cascade hot water temperature is explained by characteristics of ABS-GH shown in Figure 7. In this simulation, iterative calculation within 1 hour calculation step is conducted in order to keep the heat balance between the amount of heat demand side (ABS-GH and regenerating coil of DAHU) and heat supply side (CHPs and solar thermal system). Consequently, based on the cascade hot water temperature, hot water input to ABS-GH and regenerating coil of DAHU respectively are determined. As higher the temperature of chilled water and cascade hot water is and as lower the temperature of cooling water is, hot water input to ABS-GH increases. It results in the conservation of gas consumption for ABS-GH. When the cascade hot water temperature is low, the use of solar thermal is promoted.

Figure 8 shows the relation between outlet temperature of solar thermal heat exchanger (HEX_SOL) and solar heat supplied to the heat demand side. Figure 9 shows duration curve of the temperature difference between inlet and outlet temperature of HEX_SOL. When the cascade hot water temperature is low in cases such as case_12-28, it enables larger temperature difference between solar thermal storage tank and inlet cascade hot water to HEX_SOL, and then larger amount of solar heat can be supplied. In this simulation, as indicated in section 3, the solar thermal utilization is controlled by only temperature of solar thermal storage tank. Consequently, in case_7-28 and case_7-32, the outlet temperature of HEX_SOL is lower than inlet of HEX_SOL during when temperature in the tank is not enough.

The maximum cumulative amount of solar thermal utilization is 2226 MJ when temperature of chilled water and LSV of cooling water is 12°C, 26°C and 28°C respectively. However, the operation time of solar thermal system in case_12-28 is shorter than other cases.

Absorption chiller system coefficient of performance ($COP_{abs.sys}$) is defined by (1).

$$COP_{abs.sys} = Q_{abs}/(G_{abs}+E_{abs}+E_{pump\_CT}+E_{fan\_CT}+E_{pump\_abs}+E_{pump\_chilled}) \qquad (1)$$

Q_{abs} means chilled water load by ABS-GH; G_{abs} means gas consumed by ABS-GH; E_{abs} means auxiliary power of ABS-GH; E_{pump_CT} and E_{fan_CT} means power consumption of pumps and fans for cooling towers; E_{pump_abs} is power consumption of pumps for chilled water; $E_{pump\_chilled}$ is power consumption of pumps for secondary chilled water to SAHUs, FCUs, and pre- and after cooling coil in DAHU. Gas and electricity consumption is converted to primary energy. Primary energy conversion coefficients for gas

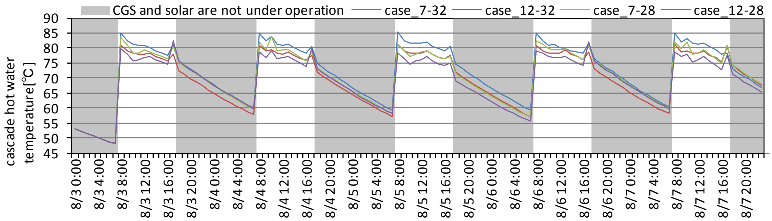
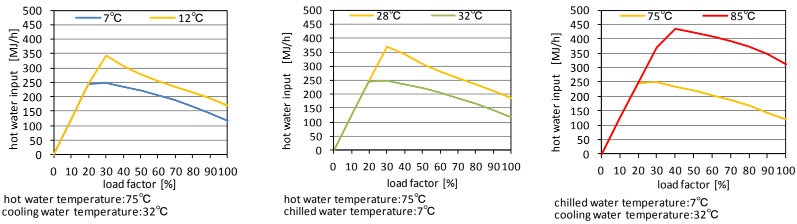


Fig.6 cascade hot water temperature to absorption chiller



(a) chilled water temperature (b) cooling water temperature (c) hot water temperature

Fig.7 characteristics of absorption chiller depending on temperature of (a) chilled water, (b) cooling water, and (c) hot water temperature

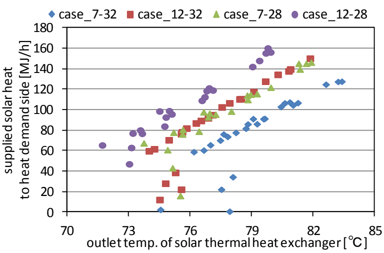


Fig.8 solar heat supplied to heat demand side and the outlet temperature of solar thermal heat exchanger

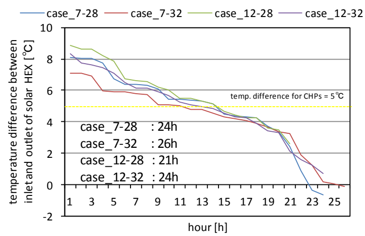


Fig.9 duration curve of temperature difference between inlet and outlet of solar heat exchanger

and electricity are 45MJ/Nm^3 and 9.76MJ/kWh respectively.

Figure 10 shows the $COP_{abs.sys}$, output of chilled water, gas consumption, auxiliary power of ABS-GH, and electricity consumption of chilled water pumps, cooling water pumps and cooling tower fans in all cases. $COP_{abs.sys}$ is affected mainly by gas consumption because gas consumption of ABS-GH accounts approximately 80% of primary energy consumption. The maximum gas consumption saving is up to 4.9GJ. On the other hand, increase of power consumption due to the raise of chilled water temperature and the decrease of LSV of cooling water temperature is only 1.3GJ. Case_12-28 is the best performance and Case_12-26 is slightly lower than Case_12-28 because of larger fan power of cooling towers. Figure 11 shows the frequency distribution of cooling water temperature. As LSV of cooling temperature is set lower, the primary energy consumption of fans for cooling towers increases. The calculated days were so hot that 26°C can be achieved only in the morning and evening during when the chilled water load is low and outdoor condition is not so severe .

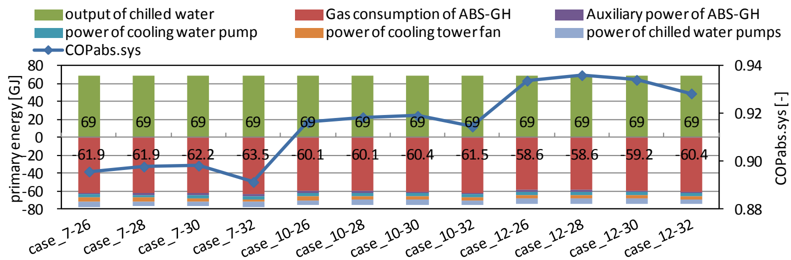


Fig.10 $COP_{abs.sys}$, output of chilled water and each primary energy consumption

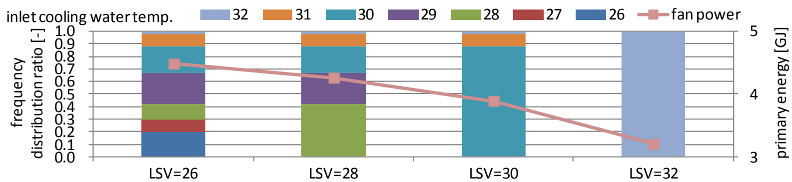


Fig.11 frequency distribution of cooling water temperature

Figure 12 shows the heat balance between heat demand side and heat supply side. The amount of used solar heat is 10% of total supplied heat. 25% of supplied heat is used at regenerating coil of DAHU. There is a small discrepancy in load of regenerating coil because when cascade hot water temperature is low, there is a limitation of temperature for regenerated air therefore the supply air absolute humidity can not be achieved the set point of humidity. However, the difference of absolute humidity between set point and calculated results is only less than 0.3g/kg' so latent load is considered to be processed satisfactorily.

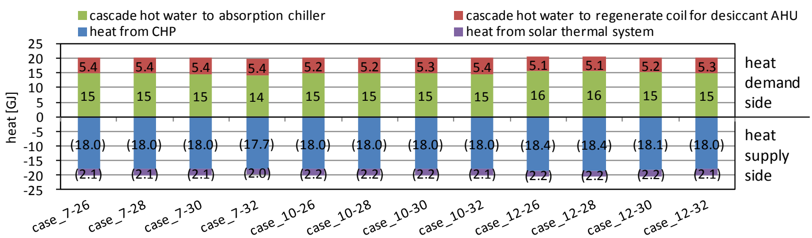


Fig.12 heat balance between heat supply side and heat demand side

The whole system coefficient of performance (COP_{WS}) is defined by (2).

$$COP_{WS} = (Q_{AHU+FCU} + Q_{DES}) / (G_{CHP} + G_{ABS} + E_{total}) \qquad (2)$$

$Q_{AHU+FCU}$ and Q_{DES} means processed air load at SAHUs, FCUs and DAHU. G means primary based gas consumption of CHPs and ABS-GH respectively. E means total electrical primary energy consumption. Gas and electricity is converted to primary energy. G_{CHP} is defined by (3).

$$G_{CHP} = G_{CHP\_total} \times \eta_{CHP\_T} / (\eta_{CHP\_T} + \eta_{CHP\_E} \times F_{p\_E}/3.6) \quad (3)$$

G_{CHP_total} is total gas consumption; η_{CHP_T} means thermal efficiency of the CHP; η_{CHP_E} means electrical conversion efficiency; F_{p_E} means electrical primary energy conversion coefficient.

Figure 14 shows COP_{ws} with processed air load and energy consumption in all cases. The best COP_{ws} is achieved 0.741 of case_12-30. $COP_{abs.sys}$ of case_12-28 is the best performance, however, as shown in Figure 10, the operation time of solar thermal is slightly shorter which means longer operation time of CHPs and larger gas consumption of CHPs than case_12-30. This slight amount of difference affects the whole system performance. However, as shown in Figure 13, sensitivity due to temperature change of chilled water and LSV of cooling water of ABS-GH is little.

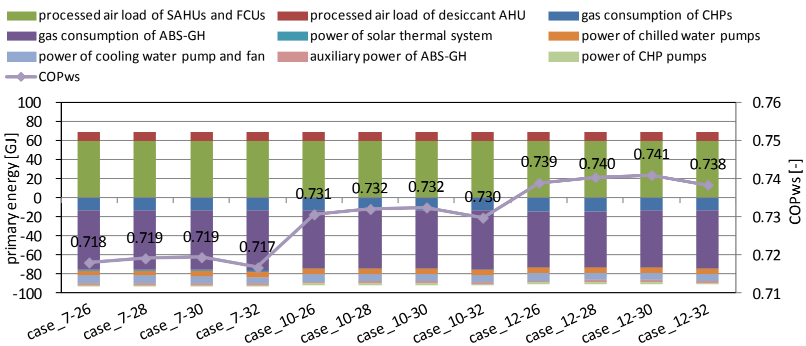


Fig. 13 COP_{ws}, processed air load and each energy primary energy consumption

5. Conclusion

In this study, heat cascade system including combined heat and power (CHP), solar thermal, absorption chiller of gas fired double effect with hot water single effect (ABS-GH) and desiccant air handling unit (DAHU) is proposed for an office building. In order to improve the whole system performance, the static simulation representing the whole system was conducted for the hottest days in 2015. Simulation parameters are chilled water temperature and lower set value of cooling water temperature. The simulation results conclude as follows;

- To counterbalance the heat between heat demand side and heat supply side, the change of chilled water temperature and lower set value of cooling water can affect the cascade hot water temperature. As chilled water temperature is raised and cooling water temperature is lowered, cascade hot water temperature gets lower.
- When cascade hot water temperature become low, more solar heat can be used to heat demand side. When return temperature of cascade hot water become high, consideration is necessary to prevent the radiation of cascade hot water from solar thermal heat exchanger.
- The best coefficient of performance for ABS-GH system is achieved in case with 12°C of chilled water temperature and 28°C lower set value of cooling water temperature.
- Although the best coefficient of performance for ABS-GH system is the case with 12°C of chilled water temperature and 28°C of cooling water temperature, the best whole system coefficient of performance is achieved 0.741 with 12°C of chilled water and 30°C of lower set value for cooling water because of larger gas consumption of CHP due to shorter operation time for solar thermal. However, the sensitivity due to set points for ABS-GH is small.

This study suggests that there is other potential to improve whole system performance with control strategy for not only cascade system but also for air conditioning system. Operation control strategy for CHPs taking cooling load into account should be investigated. Furthermore, set points for DAHU and SAHU should be optimized considering chilled water temperature.

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