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Experimental performance evaluation of heat pump-based steam supply system

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Experimental performance evaluation of heat pump-based steam supply system

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Abstract. Heat pumps have become increasingly important as a technology to reduce primary energy consumption and greenhouse effect gas emission. They are presently used mainly on residential air-conditioning and domestic hot water and are expected to spread to industrial heating processes. In 2011, Kobe Steel, Ltd. developed and commercialized two heat pump-based steam supply systems; the high efficiency steam supply system with a steam temperature of 120°C (SGH120) and the system which enables a steam temperature of 165°C (SGH165). For promoting the spread of these industrial heat pumps and enhancing the reliability of them, we investigate experimentally steam generation rate, energy efficiency and controlled performance of the SGH165 under various operating conditions on the assumption of actual different industrial processes, and evaluate technical possibilities for better performance.

1. Introduction

Heat pumps contribute to primary energy saving, to greenhouse effect gas abating and to increasing renewable energy use, so they are expected to become more popular and common. In Japanese residential and commercial sector, heat pumps have already penetrated in the form of air-conditioners. In addition, a heat pump water heater (Eco Cute), using carbon dioxide as a refrigerant, was developed and commercialized in 2001 (see [1]), and its cumulative shipments exceeded four million by the end of 2013.

In contrast, in the industrial sector, heat pumps for the heating process are uncommon, for it is difficult to achieve higher output temperature with high efficiency, low initial cost and high reliability. In recent years, Japanese manufactures overcome the difficulty of commercializing heat pumps for industrial use, and installation examples have been reported. Among them, the heat pump system capable of supplying steam with temperatures at 120°C and above is only the Steam Grow Heat pump (SGH), which is commercialized in 2011 (see [2]), by co-development of Kobe Steel, Ltd. and electric companies in Japan. This includes the following characteristics; efficient use of unused thermal energy by waste heat recovery and upgrading, reduction of dissipation heat loss by set close to processes in distributed arrangement. It is hoped that this system will contribute to better energy saving on industrial heat processes, for example distilling, condensation, disinfect and drying, compared to existing inefficient steam supply systems which are usually arranged in the center boiler room and send to each building with dissipating heat from steam piping lines.

In order to promote the spread of industrial heat pumps, it is important to enhance the reliability of them not to mention the high efficiency and the low initial cost. In addition, the performance data

under various operating conditions are required by industrial customers. Moreover, performance evaluations are also important for continuous improvement.

The authors investigate experimentally the energy efficiency and controlled performance of the SGH, and provide the highly reliable evaluation results adapted to the actual state of use. In addition we evaluate technical possibilities for better performance.

2. Outline of Steam Glow Heat Pumps (SGH120 and SGH165)

Figure 1 shows the external appearances of the SGH120 and the SGH165.

The SGH120 is composed of a heat pump unit and a flash tank. The heat pump unit lifts the heat from the heat source water (35-70°C) corresponding to the warm effluent and sends the heat to the pressurized circulating water. In the flash tank, the pressurized water is decompressed and evaporated. Consequently, the flash steam (up to 120°C, 0.1MPaG) is supplied to each process, and the remaining saturated water is back to the heat pump unit.

The SGH165, which is the objective of this study, has a steam compressor in addition to the above system. The steam compressor compresses the saturated steam generated in the flash tank. For preventing the superheat of the discharge steam, water is injected to the compressor. The discharge steam (up to 175°C, 0.8MPaG) is separated from the mist at the drain separator and supplied to each process.

The refrigerant is R245fa for SGH120, and a mixture of R134a and R245fa for SGH165 to achieve a good performance. Because of operating under higher pressure ratio and higher temperature compared to existing heat pumps, a newly-developed screw compressor is equipped and refrigerant mist is sprayed into a motor for cooling down.

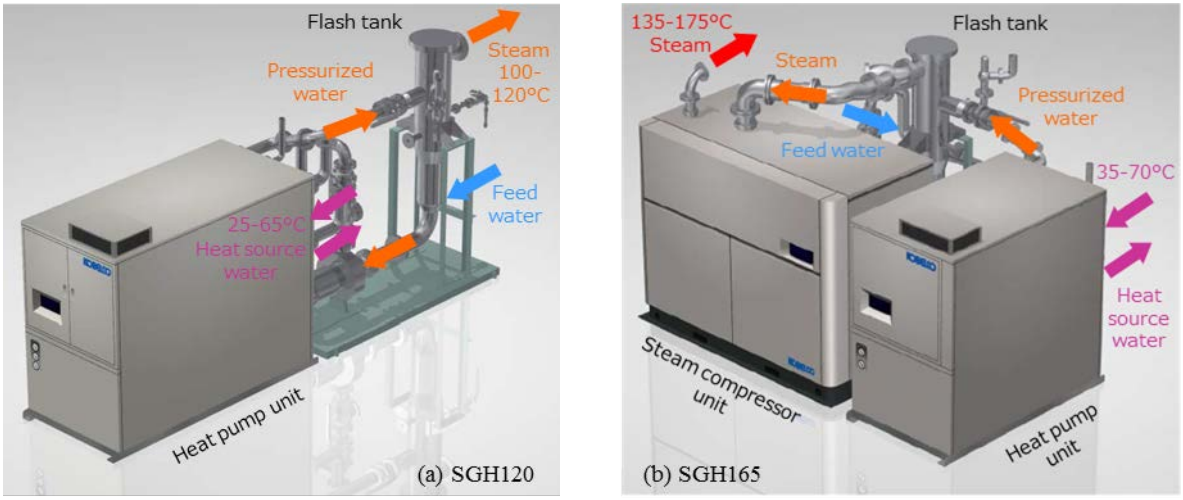

Figure 1. Steam Grow Heat pumps

3. Experimental apparatus and method

3.1. Experimental apparatus

The experimental apparatus, which was previously constructed (see [3]), is shown in figure 2. We can evaluate steam supply heat pump having a rated power output of less than 600kW (up to 200°C and 1 ton/h). The apparatus is composed of four facilities.

The first "brine cooling facility" includes air-cooled brine chillers, a brine tank, brine pumps, three plate-type heat exchangers. The facility is the cold source of the apparatus and supplies brine at 0°C to the following three facilities.

The second "heat source water facility" includes a water tank, a brine cooler, electric heaters, water pumps, and flow-control valves. The heat source water temperature is adjusted by the brine cooler and electric heaters at range of 10-90°C, and the flow rate is adjusted by the valves at range of 2-400m^3/h.

The third "cooling water facility" includes a water tank, a brine cooler, electric heaters, water pumps, flow-control valves and a cooling tower. The temperature and flow rate control methods are the same as for the heat source water facility.

The last "steam cooling facility" includes an pressure-control valve, an air cooler, a water cooler (condenser), two water tank, a brine cooler, an electric heater, a water pump, flow-control valves. The facility is a closed system: the steam generated from the machine to be tested is cooled and condensed, and the water is returned to the machine. The temperature of the feed water is adjusted by the brine cooler and the electric heater at range of 10-95°C. The discharge steam pressure is controlled by the regulation valve, and the pressure and flow rate are adjusted at the range of 0-1.0MPaG and 2-1000kg/h, respectively.

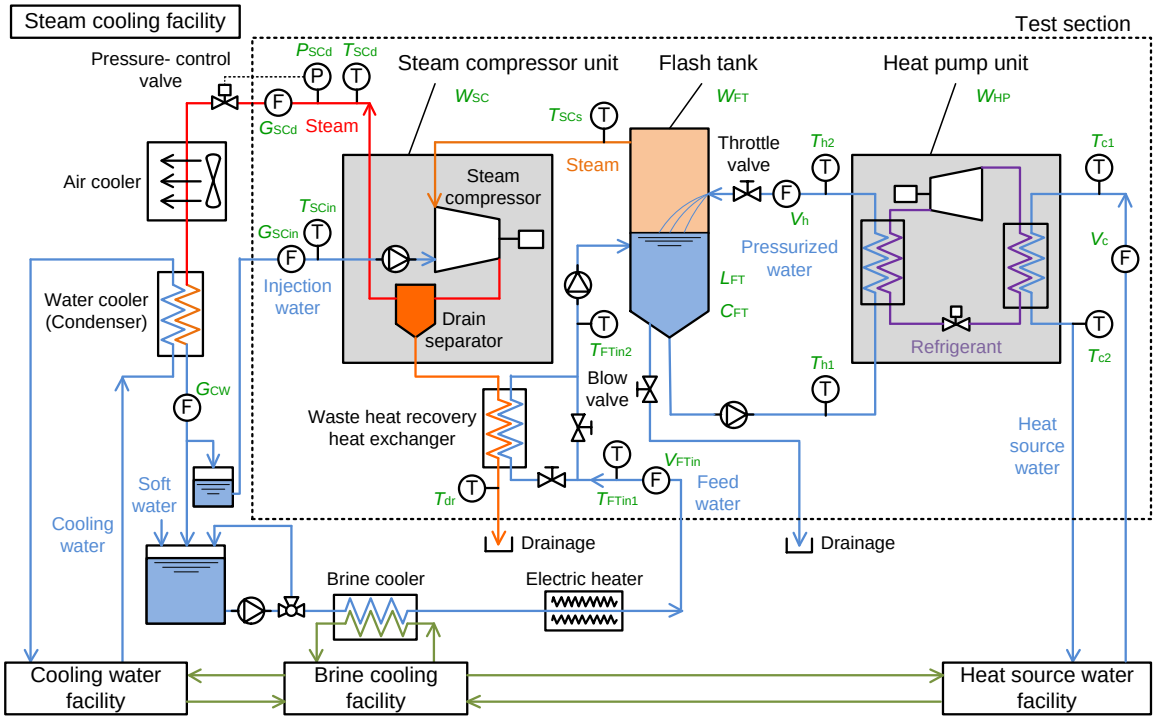

Figure 2. Schematic diagram of experimental apparatus and test section

3.2. Measure points and data acquisition systems

SGH165 needs heat source water, feed water, injection water and electric power, and can supply steam with drainage. As shown in figure 2, we measured each of the inlet-outlet temperatures, the flow rates, the discharge steam pressure, and the consumption powers.

Measuring the temperatures, four-line type high-precision resistance thermometers was used. The sensor was connected to a high-precision temperature logger and has high-accuracy of ±0.005K. The flow rates of the heat source water, pressurized water and feed water were measured by electromagnetic flowmeters with ±0.5% accuracy of reading. The flow rates of injection water and discharge steam were measured by a Coriolis flowmeter with ±0.24% accuracy of reading and a vortex flowmeter with ±1% accuracy of reading, respectively. As well, condensed water was measured by a Coriolis flowmeter with ±0.1% accuracy of reading in order to check validation each other. The electric power consumptions were measured by an electric energy meter with accuracy ±0.06%.

All sensor measurements were collected using a dedicated PC via convenient data acquisition software.

3.3. Performance evaluation indexes

The coefficient of performances (COP) of the heat pump and the system are evaluated using (1) and (2), respectively.

$$COP_{\mathrm{HP}} = \frac{Q_{\mathrm{HP\,h}}}{W_{\mathrm{HP}}} = \frac{\rho_{\mathrm{h2}} c_{p\,\mathrm{h2}} V_{\mathrm{h2}} \left(T_{\mathrm{h2}} - T_{\mathrm{h1}}\right)}{W_{\mathrm{HP}}} \tag{1}$$

$$COP_{\mathrm{Sys}} = \frac{Q_{\mathrm{Sys}}}{W_{\mathrm{HP}} + W_{\mathrm{SC}}} = \frac{G_{\mathrm{SCd}} \left(h_{\mathrm{SCd}} - h_{\mathrm{FTin}}\right)}{W_{\mathrm{HP}} + W_{\mathrm{SC}}} \tag{2}$$

Note that the feed water pump and the circulation pump input powers are considered out of the system. The uncertainties of the heat pump COP and the system COP are up to ±1.0% and ±1.5%, respectively.

4. Experimental results and discussion

4.1. Operating characteristics

Figure 3 shows the operating characteristics of the SGH165 as time-series variations. The heat pump unit starts (see W_{HP}) and heats up the circulating water (see T_{h2}). T_{h2} reaches 100°C and then, the flash steam is generated and supplied to the steam compressor unit (see T_{SCs}). After the suction pressure of the steam compressor increases, the compressor runs to make the pressure constant (see W_{SC}) and generates the steam (see P_{SCd}, T_{SCd} and G_{SCd}). For keeping the flash tank water level constant, the feed water is supplied to the tank (see L_{FT} and G_{FTin}). The water level is measured by the equipped sensor of differential pressure type in the flash tank. In this test condition, the heat exchanger for the waste heat recovery from the steam compressor drainage is used, so the flash tank inlet temperature is increased (see T_{FTin1} and T_{FTin2}). When we assume that the temperature of the drainage into the heat exchanger is the saturated temperature at the steam discharge pressure, the drainage mass flow rate is calculated by the heat balance of the heat exchanger and illustrated (see G_{dr}). About half of the injection water is drained outside the system.

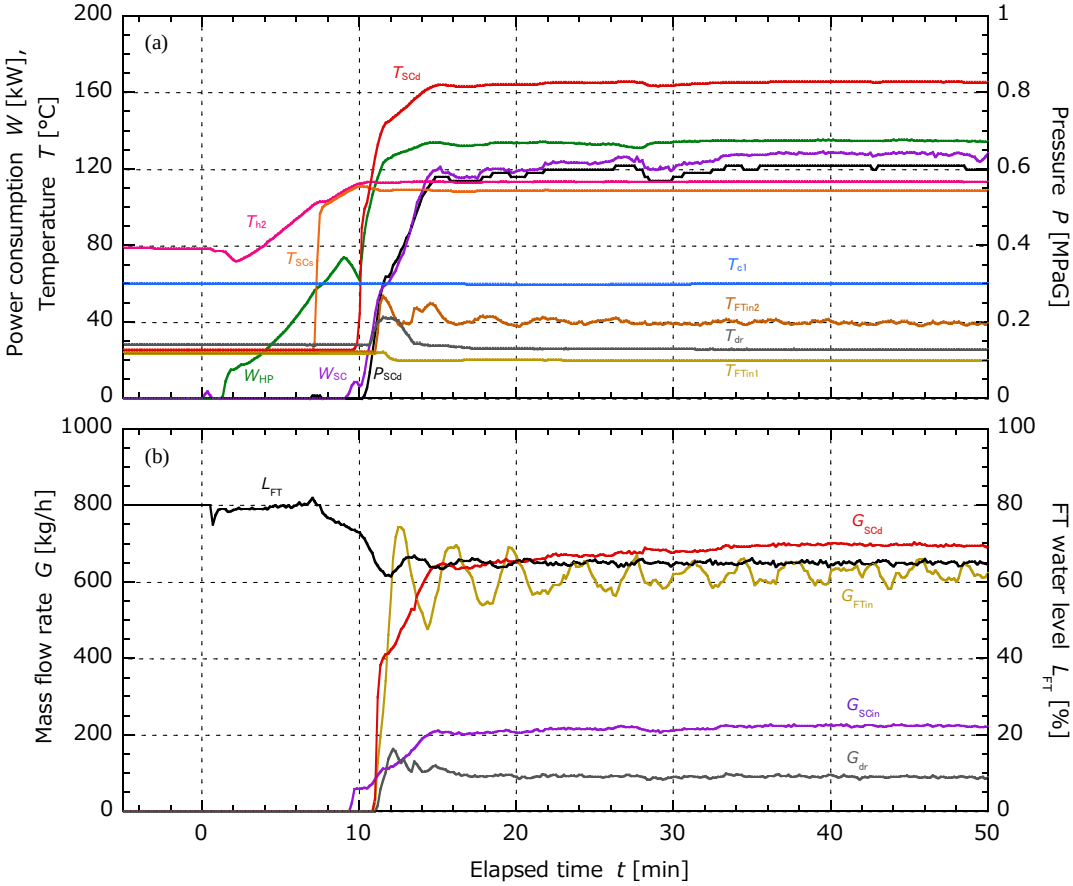

Figure 3. Operating characteristics of the SGH165

4.2. Steady-state characteristics

On the assumption of actual different industrial process temperatures, the steady-state data were obtained under various heat source temperatures, feed water temperatures and steam discharge pressures. This steady-state means meeting the following conditions;

- The system mass balance (MB) and the system heat balance (HB) are kept at or near 1.0. Each balance is calculated by using (3) and (4), respectively, where G_{loss} is the loss from the shaft seal part of the steam compressor and the value is about 30kg/h obtained by a preliminary test.

$$MB_{\mathrm{Sys}} = \frac{G_{\mathrm{SCd}} + G_{\mathrm{dr}} + G_{\mathrm{loss}}}{G_{\mathrm{FTin}} + G_{\mathrm{SCin}}} \tag{3}$$

$$HB_{\mathrm{Sys}} = \frac{Q_{\mathrm{Sys}}}{Q_{\mathrm{HP\,c}} + W_{\mathrm{HP}} + W_{\mathrm{FT}} + W_{\mathrm{SC}}} \tag{4}$$

- The heat source water inlet temperature and the feed water temperature are kept constant.
- The discharge steam pressure and flow rate are kept constant.

As shown in figure 4, the steam discharge mass flow rate and the system COP increase with the heat source inlet temperature under the condition that the steam discharge pressure and the feed water are constant. Especially, the influence of heat source temperature on the steam discharge rate is marked. Figure 5 shows the results related to steam discharge pressures. The steam discharge rate increases with the pressure, but the system COP decreases. Figure 6 shows the results related to feed water temperatures. The steam discharge rate increases with the temperature, but the system COP slightly decreases. This figure also illustrates the case without the waste heat recovery heat exchanger

(HRHX). The steam discharge rate slightly increases in the case of having the exchanger, especially when the feed water temperature is low, but the change of the system COP is negligible little.

We can conclude the following. It is most important to secure the higher heat source temperature, which depends on the warm effluent temperature. The waste heat recovery heat exchanger is valid for slightly increasing of the steam discharge rate when the feed water temperature is low, but there is little change in the system COP.

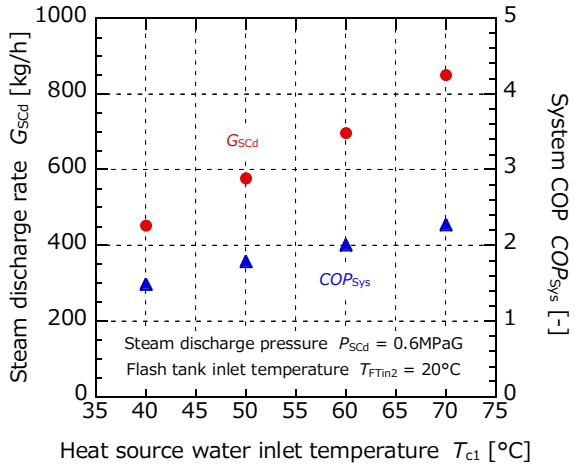

Figure 4. Influence of heat source temperature on steam discharge rate and system COP

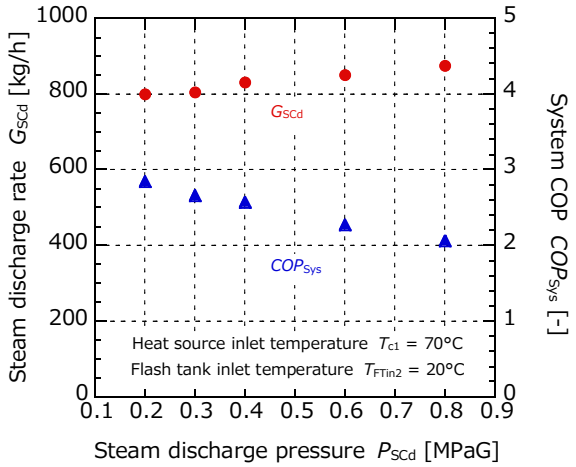

Figure 5. Influence of steam discharge pressure on steam discharge rate and system COP

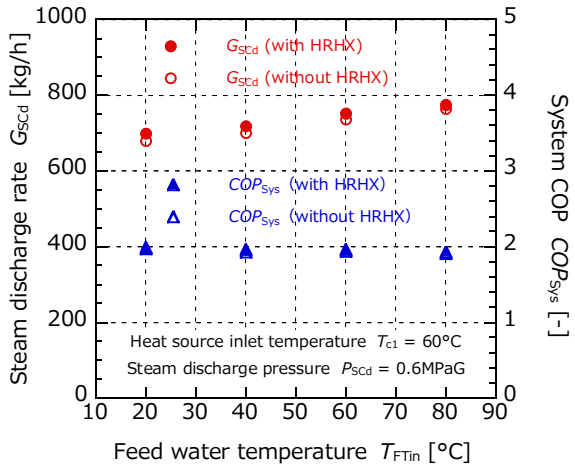

Figure 6. Influence of feed water temperature and effect of waste heat recovery

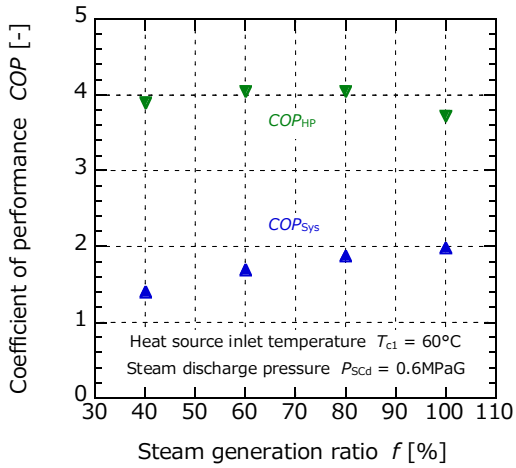

Figure 7. Part-load performance

4.3. Part-load performance

Figure 7 shows the part-load performance. The data were obtained by narrowing the steam pressure-control valve. The minimum of the steam generation rate was restricted by the refrigerant compressor rotating speed of 1500rpm. The system COP decreases with decreasing the steam generation rate. On the other hand, the heat pump COP has maximum at the range of the steam generation rate of 60-80%.

We can conclude that the part-load performance has a large influence on the system performance, so it is recommended to operate the system at the rated power.

4.4. Influence of condensed water blow

As generating the steam, the water in the flash tank becomes condensed and the electric conductivity becomes more. For continuous operation, it is required to blow the condensed water.

Figure 8 shows the time-series change. In this condition, the bowing is operated with a period of 24 minutes and the blowing time is 6 minutes. During the blowing, the net steam discharge rate is decreased by about 5%. In contrast, as shown in figure 9, the change of the system COP is negligible little, so the symbol of the COP (without blowing) is covered by that of the COP (with blowing).

We can conclude that the condensed water blow has a small influence on the system performance.

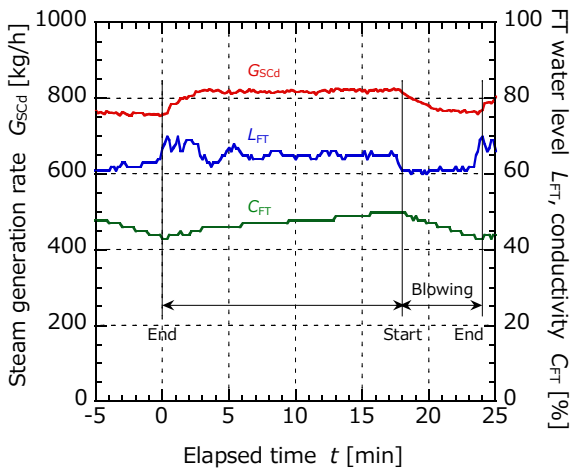

Figure 8. Time-series of condensed water blow

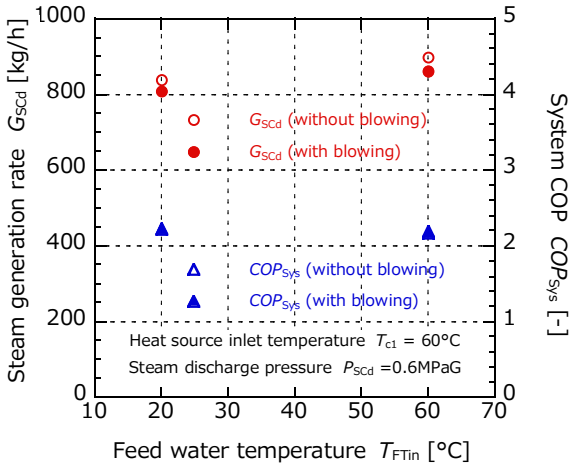

Figure 9. Influence of condensed water blow

4.5. Load-following capability

In actual plants, steam demand fluctuates, so the steam supply system is required to follow the fluctuation.

Figure 10 shows the load-following capability when the steam supply rapidly decreased by 20%. As soon as the steam discharge rate decreases, the steam compressor instantly loads down (see W_{SC}) and makes it possible to remain the discharge pressure constant (see P_{SCd}). On the other hand, the heat pump loads up (see W_{HP}) and the pressurized water temperature increases (see T_{h2}). When T_{h2} reaches a temperature, the heat pump loads down, and the steam compressor loads up. The result is that the steam pressure and flow rate are kept constant while the heat pump and steam compressor loads slowly fluctuate with opposite-phase.

Figure 11 shows the load-following capability when the steam supply rapidly decreased by 40%. As soon as the steam discharge rate decreases, the steam compressor instantly loads down (see W_{SC}), but the steam discharge pressure goes disturbed in some degree (see P_{SCd}).

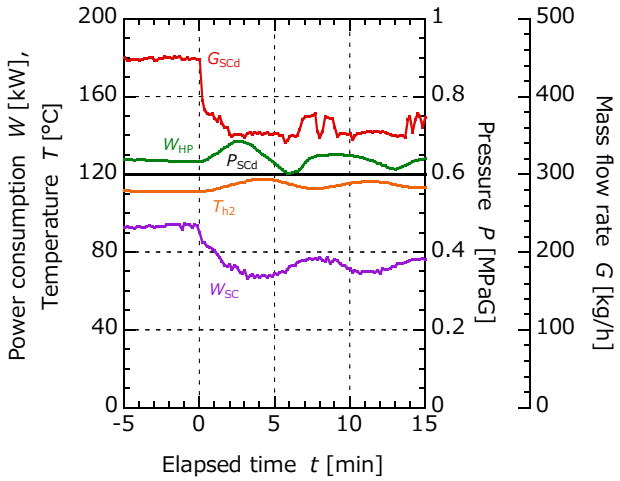

Figure 10. Load following capability (steam supply rapidly decreased by 20%)

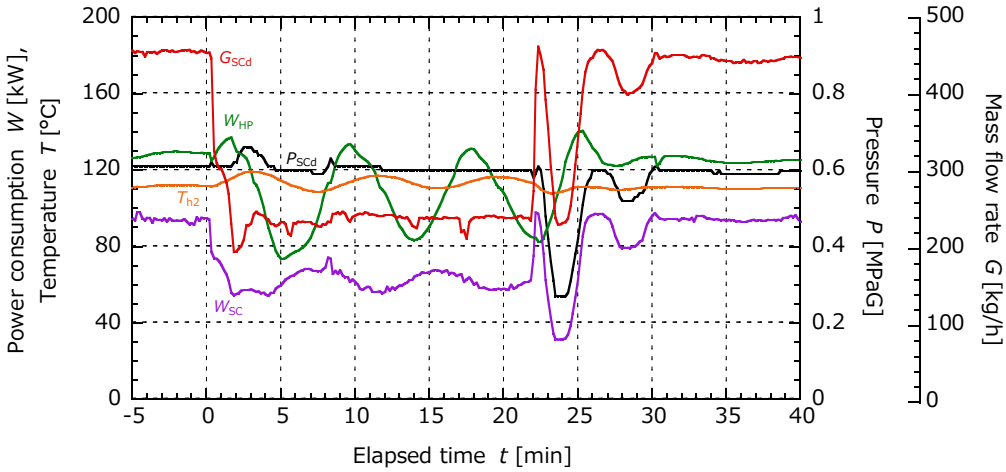

Figure 11. Load following capability (steam supply rapidly decreased by 40%)

4.6. Discussion for better performance of heat pump-based steam supply system

The current state of the system COP is insufficient for use in wide range of heat source water temperatures. As shown in section 4.2, the system COP is influenced by heat source water temperature more than by feed water temperature. As we estimate effects of exchanging the heat from the steam compressor drainage to the heat source water not to the feed water, the heat source water inlet temperature will increase by about 0.2K, so there will be little hope of the system COP enhancement. For enhancement, it is necessary to drastically enhance the heat pump COP. It is desirable to develop the refrigerant suited to high temperature output as an alternative to R245fa.

Regarding load-following capability, the responsivity of the steam compressor is so good, and the system load-following capability will be expected to be better by tuning up the PID controller. For enhancing the reliability, it is necessary to confirm the load-following capabilities related to heat source temperatures and flow rates. This will be the subject of future investigations.

5. Conclusion

For promoting the spread of these industrial heat pumps and enhancing the reliability of them, we investigate experimentally steam generation rate, energy efficiency and controlled performance of the

SGH165 under various operating conditions on the assumption of actual different industrial processes, and then we can conclude the following.

For high efficiency, it is most important to secure the heat source temperature as highly as possible, although the temperature depends on the warm effluent temperature and has a limit. The waste heat recovery heat exchanger is valid for slightly increasing of the steam discharge rate when the feed water temperature is low, but there is little change in the system COP. The condensed water blow has a small influence on the system performance. The SGH165 can uneventfully follow the load down when the steam supply rapidly decreased by 20%.

We will examine the possibility of enhancing the heat pump COP and will confirm the load-following capabilities related to heat source temperatures and flow rates.

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