STEAM-GENERATING HEAT PUMPS USING MULTI-STAGE CENTRIFUGAL COMPRESSORS FOR INDUSTRIAL APPLICATIONS

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Abstract: Feasibility of steam-generating heat pumps for industrial applications is investigated. The heat pump system uses water as refrigerant, and it includes a multi-stage centrifugal compressor with water atomization cooling at the intra-stage of the compressor. A component test apparatus is developed to validate design methodology of steam compressor operating under high negative pressures (less than 20kPa absolute) and to check the effectiveness of water atomization cooling. Test results show that compressor performance satisfies the system requirements, and that water atomization cooling can lower steam temperature effectively.

Key Words: ***heat pumps, multi-stage centrifugal compressors, water refrigerant, water atomization cooling, industrial waste heat, efficiency***

1 INTRODUCTION

Reducing carbon dioxide emission associated with fossil fuel combustion is becoming a great matter of worldwide concern from the viewpoint of global warming. In Japan, METI (Ministry of Economy, Trade and Industry) selected heat pumps as one of the innovative technologies for energy savings and has given the priority to their research and development (METI 2005).

Industrial sectors such as food, paper and pulp, chemicals, and so on, consume large amounts of heat energy in their production processes. So far, industrial boilers have been used as a heat source to provide their process heat, even though most of their processes —— water heating, drying, pasteurizing, concentrating and so on —— require only a low temperature steam below 120 °C. It is a complete exergy loss to supply such low temperature steam by the high temperature heat of fossil fuel combustion.

Furthermore, the facilities at these industries are emitting large amounts of waste heat with a temperatures ranging from 40 to 90 °C to their surroundings. It is important to recover such industrial waste heat into their process heat. Heat pumps have great potential for energy savings and emission reductions in industrial sectors.

To date, many types of industrial heat pumps have been proposed for effective use of industrial waste heat (Takagi at al 1986, Endou 1990, Ninomiya et al 2007). These heat pumps, however, have relatively small heat output, and are unsuitable for industries that need large amounts of heat over 1MW. Typical industrial boiler users generally use two or three 2-ton/h-boiler packages, in which heat output per package is about 1.3 MW. Then, we targeted the development of a steam-generating heat pump having a steam output of over 5 ton/h and we focused on multi-stage turbo heat pumps that are appropriate for high discharge temperature as well as large heat output.

The objective of our present research is to develop a steam-generating turbo heat pump, which can recover industrial waste heat into industrially beneficial heat. As the first step in our research, we developed a component test apparatus to check the feasibility of steam generating heat pumps.

2 HEAT HUMP SYSTEMS

2.1 Heat Pump System Examples

We propose steam-generating heat pumps that use water as refrigerant. Water is the ultimate natural refrigerant, because it has no negative impact on the environment. Moreover, water is inexpensive and safe and easy to handle. When water is used as refrigerant, generated steam can be directly delivered to a heat consumption facility in an open cycle, which eliminates a condenser downstream from a compressor. Hence, water as refrigerant is helpful to reduce the initial cost of the heat pump system and to improve system efficiency without heat loss related to the condenser.

Figure 1 shows a motor-driven heat pump, which is the most fundamental system. The system produces 5 ton /h steam at 130 °C by recovering waste heat from 60 °C hot water. It employs a four-stage centrifugal compressor to achieve high discharge temperature. Since steam is apt to be superheated in the compression process, it is important to avoid steam superheating in order to reduce compression work. In the present study, liquid water is sprayed in the intra-stage of the compressor to cool down the superheated steam to nearly the saturated temperature by liquid water evaporation. A simple heat and mass balance calculation shows that this system will achieve a high COP (coefficient of performance) above 1.4 on a primary energy basis.

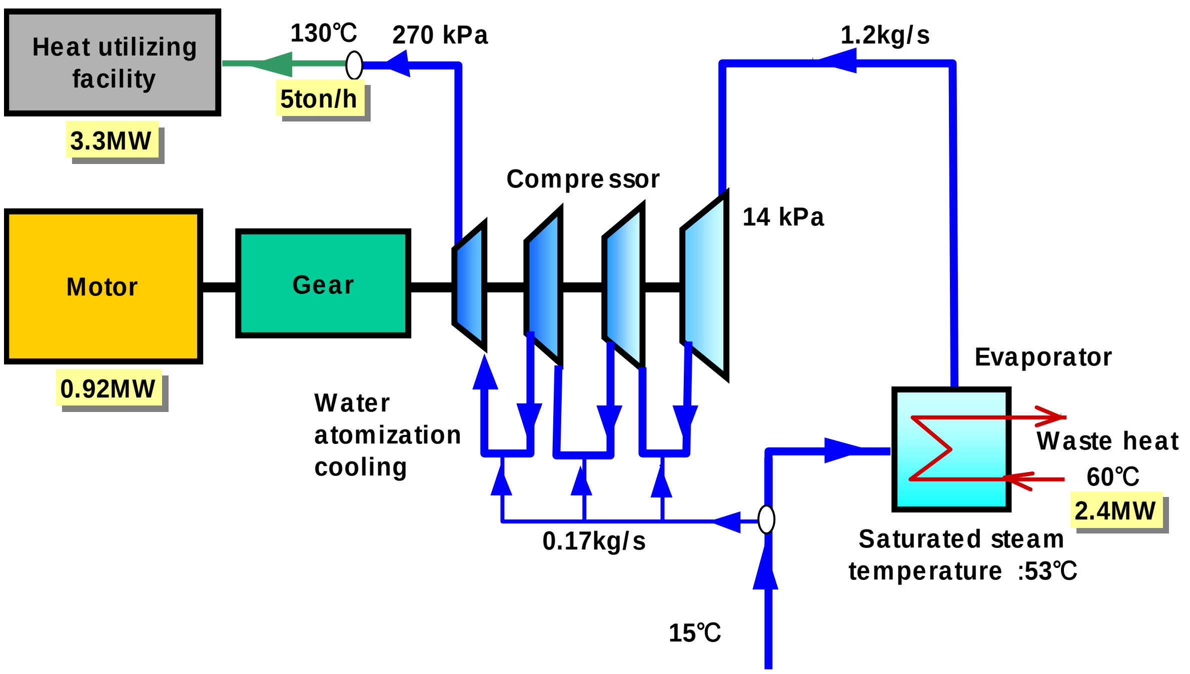

Figure 1: Motor-driven heat pump example (steam output is 5 ton/h at 270 kPa)

Figure 2 shows a steam-turbine-driven heat pump combined with a gas turbine electric generator. The system produces 2.4 MW of electricity as well as 10.7 MW of steam at a saturated steam temperature of 143 °C. In this case, high-pressure steam generated by the heat recovery boiler drives the steam turbine. The exhaust steam from the steam turbine has sufficient temperature to be used at a heat utilizing facility. Accordingly, not only steam from the heat pump, but also steam from the turbine exhaust is available as a heat source at a heat utilizing facility. Hence, this system is suitable for users who need large amounts of steam and electricity. Total thermal efficiency of this system is 120% as combined heat and power.

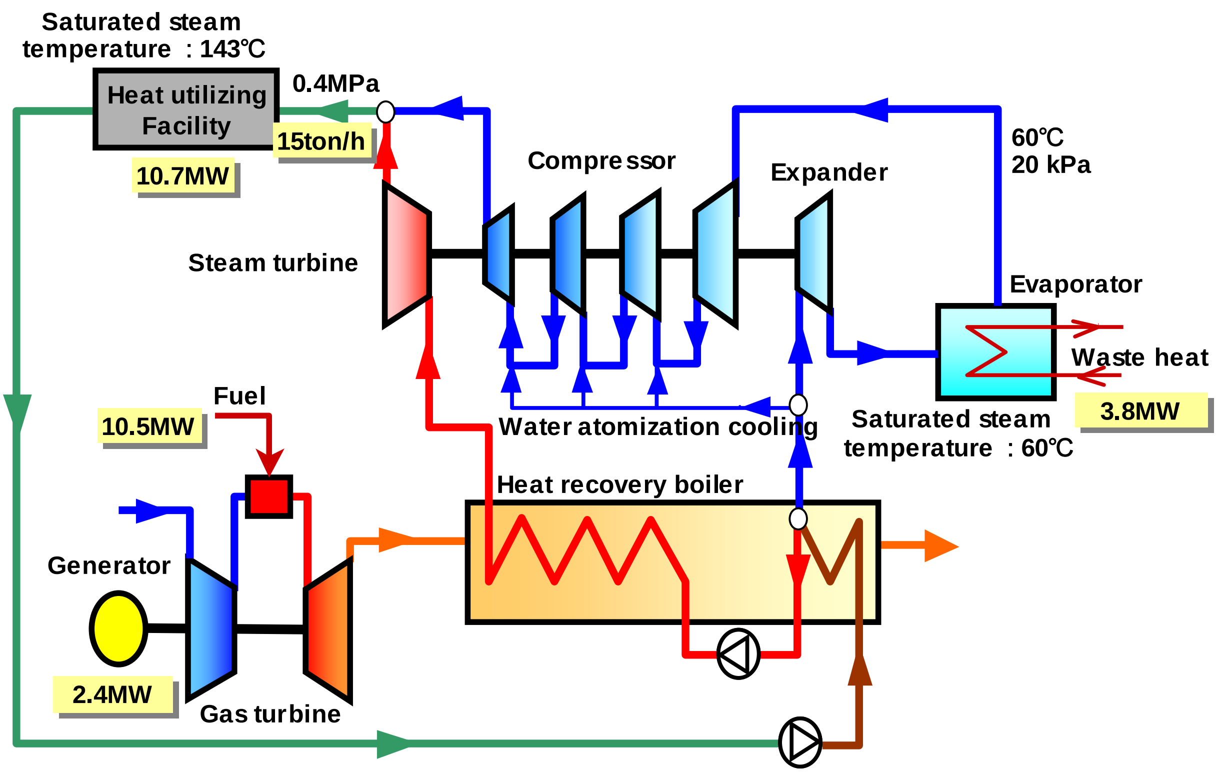

Figure 2: Steam-turbine-driven heat pump example (steam output is 15 ton/h at 400 kPa)

2.2 Technological Problems

Most of the technological problems for the proposed heat pumps are related to the turbo compressor and the use of water as refrigerant.

Water has a lower saturation pressure than typical Freon-based refrigerants. Thus, the internal pressure of a heat recovery heat exchanger (i.e. evaporator) has to be lowered, when water is used as refrigerant. This becomes more important when increasing the heat recovery by the evaporator. For example, when waste heat is recovered from 60 °C hot water (fig, 1), the internal pressure of the evaporator should be lowered below 14 kPa (absolute). Low pressure means low density of compressor suction flow. Hence, a water refrigerant heat pump requires a large volume flow rate for its compressor so that the compressor does not become too large.

In general, there is a trade-off relation for getting a large volume flow rate and a high pressure ratio for turbo-compressors; that is, a larger volume flow rate lowers the stage pressure ratio, which requires more stage numbers to achieve the target pressure ratio for the whole compressor. In addition, a steam compressor needs higher peripheral speed than an air compressor to achieve the same pressure ratio. It is therefore necessary to cope with both getting high-pressure ratio and a large flow rate at the same time in order to realize a compact, small-stage-number, low cost compressor.

Furthermore, low suction density results in low Reynolds number flow, which makes it difficult for a turbo-compressor to achieve high efficiency. Also, regarding the lower internal pressure than ambient pressure, it is important to avoid air leakage through the rotor seal portion into the heat pump system. Both low Reynolds number and air leakage are serious problems related to cycle efficiency.

3 COMPONENT TEST APPARATUS

We developed a test apparatus to check the application feasibility of the main components for a water refrigerant heat pump. The apparatus mainly consisted of an evaporator, a centrifugal compressor, a condenser and an expansion valve.

Although a three- or four-stage centrifugal compressor will be required for the actual heat pump system, for simplicity only a single-stage centrifugal compressor was installed in this test apparatus. Since the lowest pressure stage is the most difficult to design from the technological aspects, the first stage compressor was selected for this feasibility study. Because of this, not only suction pressure, but also discharge pressure of the compressor became lower than the ambient pressure. Hence, the system was composed of a closed loop cycle, not an open loop cycle like the actual product.

3.1 Test System

Figure 3 shows an overall system diagram of the component test apparatus. Hot water as assumed waste heat was supplied into an evaporator, which generated low-pressure steam of 20 kPa and 65 °C. The low-pressure steam was compressed by a single-stage compressor, resulting in relatively higher pressure steam of 40 kPa and 148 °C. The compressed steam was cooled down and condensed in a condenser using cold water as a heat sink. The condensed water was expanded through an expansion valve, and returned to the evaporator.

Auxiliary machines were also installed. To remove non-condensing gases, a deaerator was installed downstream from the condenser. A vacuumed pump was also installed in the main stream line to get lower inside pressure at system start up.

In order to check the effects of water atomization cooling, water spray nozzles were installed upstream and downstream from the compressor. The upstream nozzles are prepared for the second or third stage compressor testing in order to simulate the intra-cooling water spray from the former stage compressor. Injected water droplets cool down the super-heated steam, because of the latent heat of liquid water evaporation. The latent heat of liquid water evaporation is much larger than the heat quantity required for the super-heated steam to be saturated. Hence, just a small amount of liquid water was required to cool down the super-heated steam temperature.

For the purpose of rapid evaporation of water droplet, its diameter should be as small as possible. A high-pressure type nozzle was used as a water atomizer. High-pressure water was impinged to the pointed end of a nozzle target, thus providing fine droplets. Figure 4 shows a water spray plume injected by a single nozzle. Although water droplets were sufficiently fine with a Sauter mean diameter of 20 microns or less, high pressure around 7 MPa was required to keep droplets fine.

According to a heat and mass balance calculation, this system will achieve a high cycle COP of 15.7 at the design condition, where the pressure ratio and adiabatic efficiency of the compressor were assumed to be 2.0 and 80 %, respectively. This high cycle COP is due to the small difference of saturated steam temperature between the upstream and the downstream of the compressor. As the temperature difference is increased, the cycle COP is decreased drastically.

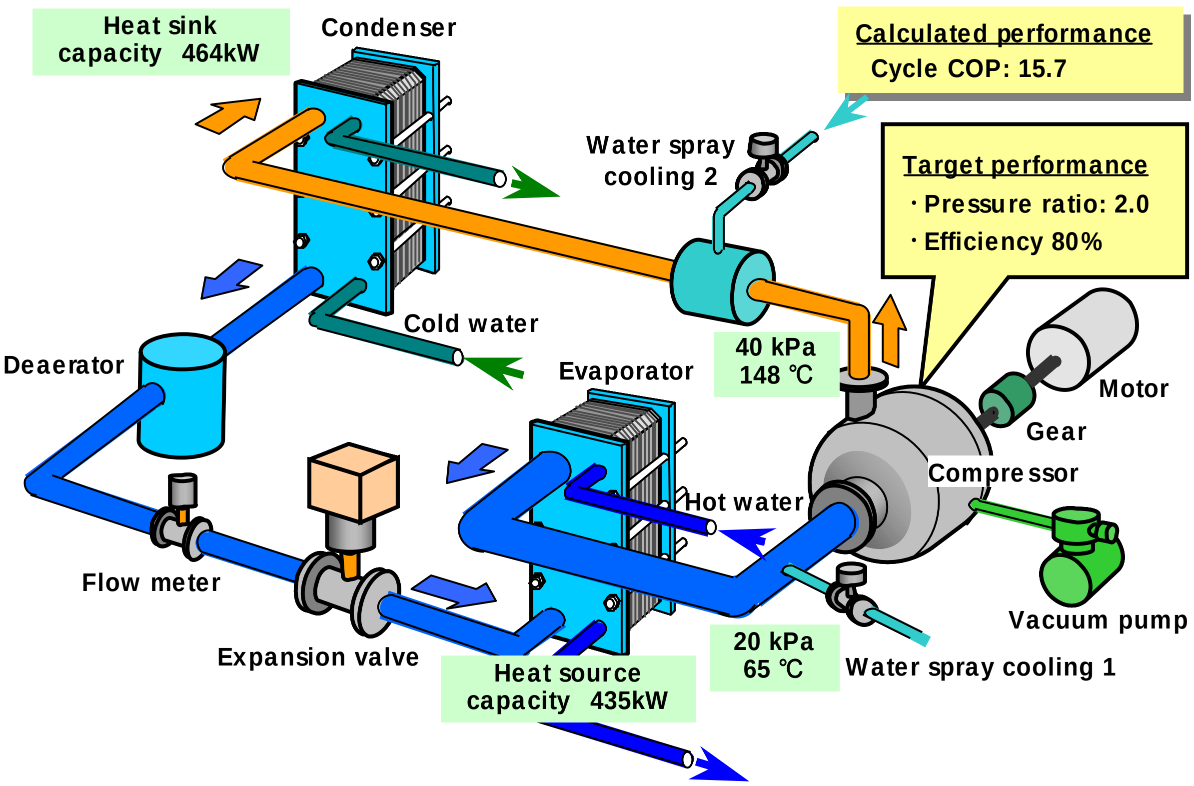

Figure 3: System diagram of a component test apparatus

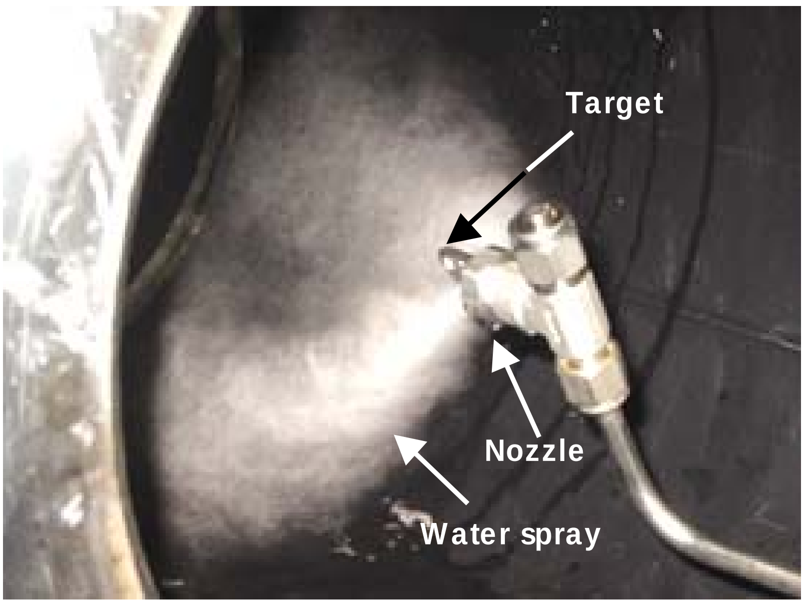

Figure 4: Plume characteristics at operating pressure of 7 MPa

3.2 Heat Exchangers

Plate type heat exchangers were employed as both the evaporator and the condenser for this test apparatus; this selection was based on their compactness. The heat exchange quantity was about 500 kW, which was so large that a boiler and a cooling tower were used as heat source and sink. Hot water heated by the boiler steam was used as a heat source for the evaporator. Cold water from the cooling tower was used as a heat sink for the condenser. The former represents waste heat from a virtual heat utilizing facility, and the latter represents heat consumption there.

In this closed loop apparatus, setting the height of the condenser was important. The inside pressure of this test apparatus was so low that water head of the condenser affected the system pressure balance. Pressure losses of each component including pipes should be predicted accurately. In addition to this, it was important to utilize the water head of the condenser actively. Hence, the condenser was set in an elevated position, as shown in figure 5. Active use of the water head of the condenser is only needed for this low-pressure closed loop test apparatus, but is not needed for actual high-pressure open loop facilities.

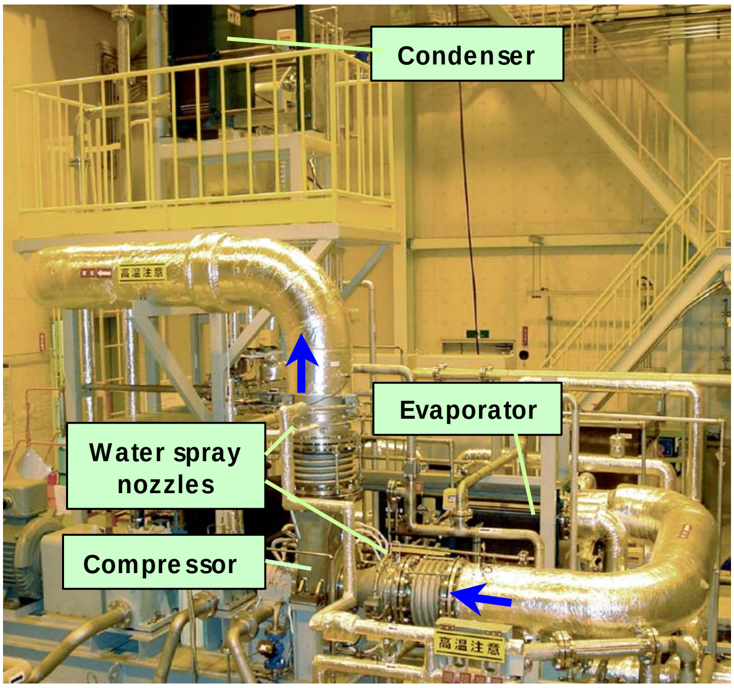

Figure 5: Component test apparatus

3.3 Single-Stage Centrifugal Compressor

Target pressure ratio and adiabatic efficiency for the model compressor are 2.0 and 80%, respectively. This model compressor was scaled down in size, so that these performance targets were sufficiently high enough for the multi-stage compressor system to realize the system COP above 1.4 on a primary energy basis. Design rotational speed of the compressor is 49,500 rpm. Impeller tip speed is over 500 m/s. At the inlet conditions of 20 kPa and 65 °C, mass flow rate of the compressor is 0.186 kg/s. Then, specific speed of the compressor, N_s, is about 370, where specific speed is given by the next formula:

$$N_s = N \times \frac{\sqrt{Q}}{H_{ad}{}^{3/4}},$$

N: rotational speed [rpm], Q: volume flow rate [m^3/min], H_{ad}: adiabatic head [m].

For centrifugal compressors, there is a trade-off between high specific speed and high impeller tip speed from the standpoint of material strength. When a specific speed is high, blade height at the impeller exit becomes large, so that it becomes more difficult to achieve high impeller tip speed because of an increase of centrifugal stress. Figure 6 shows calculated results of flow pattern and centrifugal stress obtained using a computational fluid dynamics analysis and a finite element method. In order to maximize performance levels while avoiding local high stress above the strength limit of the material, aerodynamic performance and structural stress were iteratively analyzed, thus we optimized blade and disc shape of impeller in the present study.

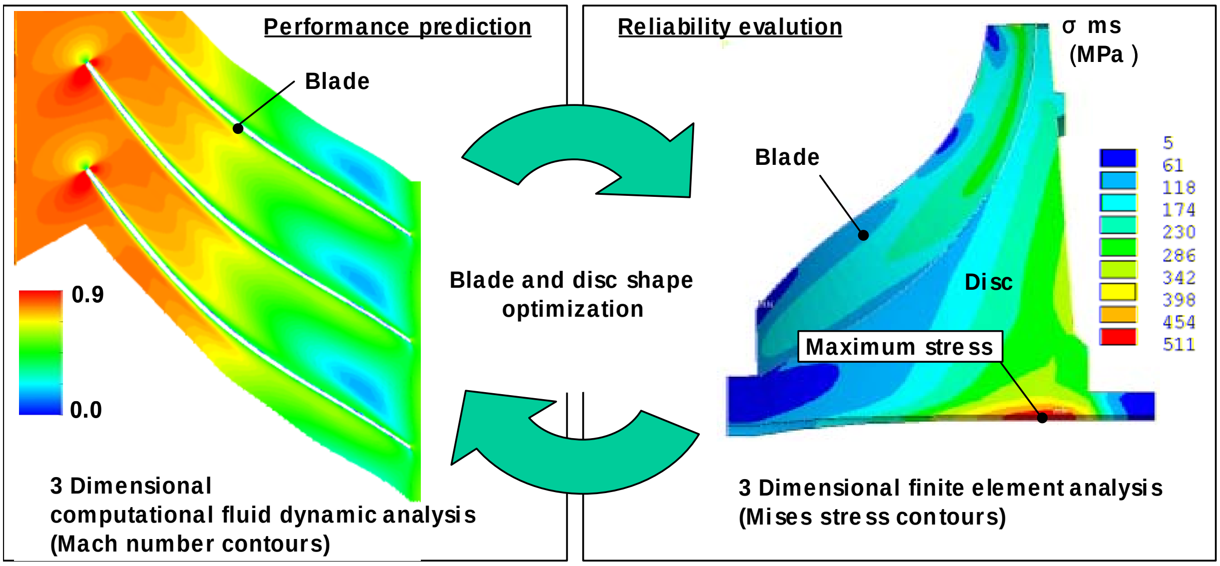

Figure 6: Calculated results of flow pattern and centrifugal stress

For this model compressor, the machine Reynolds number, which is the Reynolds number based on an impeller diameter as a length, is about 1.1 million. This value is the same as the machine Reynolds number for the air compressor, whose impeller size is about 30 mm in diameter. We do not have much experience in designing such a small sized of compressor impeller. Hence, it was important for us to validate our prediction tool of compressor performance.

Figure 7 shows a cross-sectional view of the model compressor. The compressor impeller overhangs from a compressor rotor, which is supported by two oil bearings. Because of the negative pressure of the compressor inside, the ambient air can flow into it through the rotor-sealing portion. In order to prevent this, higher-pressure steam than the inside pressure is supplied to the inner side of sealing portion and the air is removed from the outer side of it. Thus, the higher-pressure steam purges the ambient air outside, and most of the leakage fluid becomes steam.

For the purpose of achieving high efficiency and a wide operating range, a low solidity airfoil (LSA) diffuser was employed for this model compressor. The LSA diffuser has a tandem design consisting of the inner LSA diffuser and the outer LSA diffuser. The outer LSA diffuser is thick enough for bolts to go through and acts as a strut to support the compressor casings.

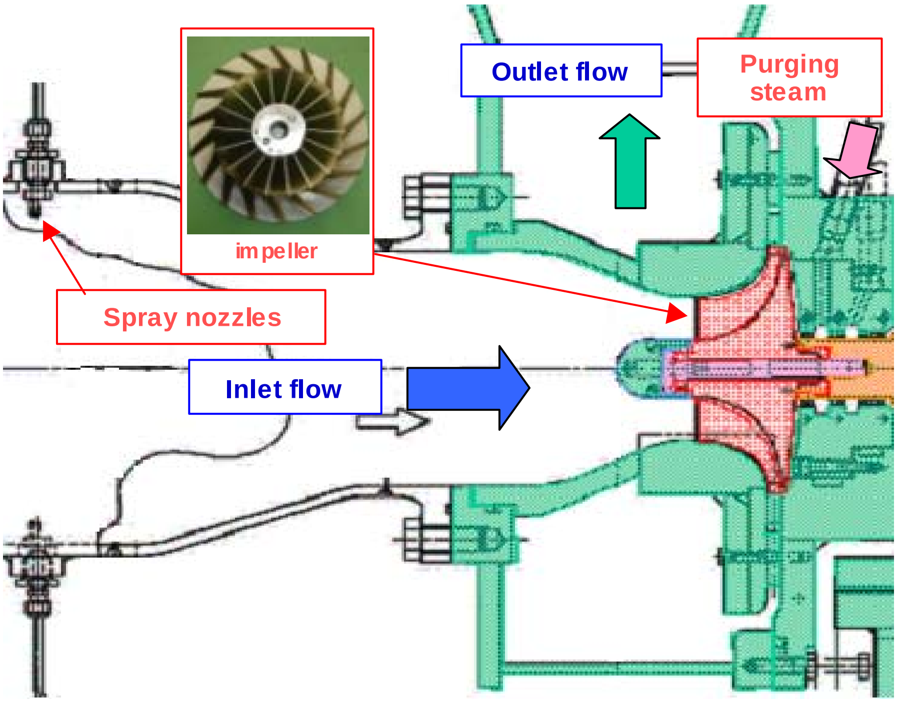

Figure 7: Cross-sectional view of the model compressor

4 RESULTS AND DISCUSSION

4.1 Compressor Performance

Figure 8 shows performance test results of the model compressor. It could achieve a higher-pressure ratio and a higher adiabatic efficiency than their design target values. In particular, adiabatic efficiency of the model compressor was satisfactorily high, due to the contribution of the tandem LSA diffuser. We concluded that the present level of the machine Reynolds number is within the range that our compressor design methodology is applied to.

In order to design a good performance multi-stage compressor, it is important to predict each compressor stage performance accurately. We have an in-house analysis tool, PRECENT, to predict centrifugal compressor performance. In figure 8, predicted performance curves by PRECENT at planning are marked as prediction (plan). Prediction of the pressure ratio agreed well with test data, while prediction of adiabatic efficiency did not agree well.

To get better prediction of compressor performance, some empirical parameters, such as slip factor of the impeller and loss coefficient of the diffuser, were modified to correlate well with test data. These results are marked as prediction (tuned) in figure 8. Owing to this tuning of some empirical parameters, we concluded PRECENT can predict the model compressor performance well and it is a good analysis tool within the present range of Reynolds numbers. We will be able to design a good performance multi-stage compressor using this analysis tool.

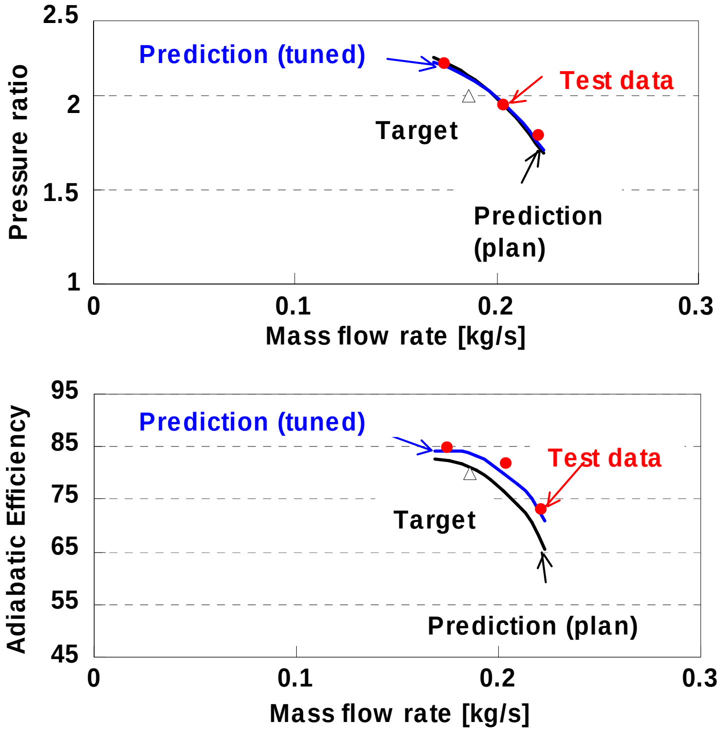

Figure 8: Stage characteristics of the model compressor

4.2 Water Atomization Cooling

To verify the cooling performance of the water spray, liquid water was injected into super heated steam at the compressor outlet. The injected water quantity was 6 wt%, which was equivalent to the saturated quantity for the super heated steam from the compressor outlet. To know the liquid water evaporation quantity, steam flow temperatures (1), (3) and pipe surface temperature (2) were measured by thermocouples, as shown in figure 7. Supplied pressure of the injected liquid water was 7 MPa, which was high enough to provide fine droplets.

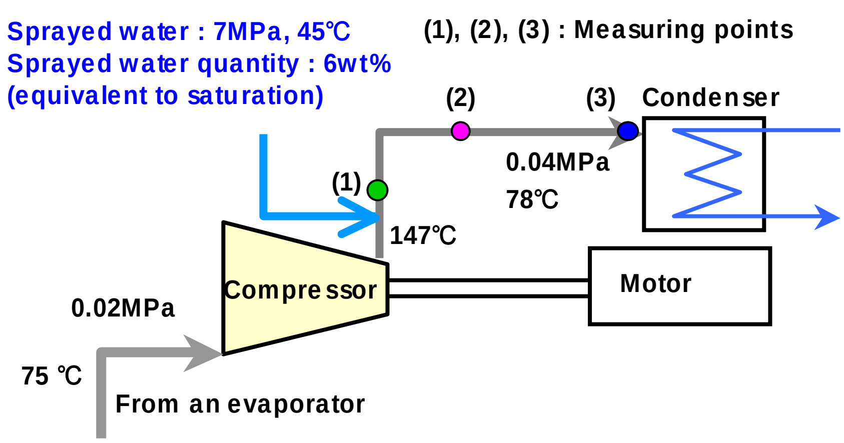

Figure 9: Temperature measuring point for performance evaluation of water atomization cooling

Figure 10 shows time history of measurement temperatures during the water atomization cooling. Temperature of thermocouple (1) indicated a rapid drop to the saturated temperature immediately after the spray started. This means that the thermocouple (1) was wetted with water spray. This is reasonable because it was located just downstream from the spray nozzles. In contrast to this rapid change of temperature (1), temperatures (2) and (3) changed very slowly. This is because those temperatures were strongly affected by the heat capacity of the pipe between the compressor outlet and the condenser inlet. In spite of the slow change of those temperatures, they finally reached the saturation temperature in 30 minutes. We concluded shows that the injected liquid water can evaporate completely and cool down the super heated steam up to saturation.

It took 30 min for the system to reach a heat equivalent condition. Once the pipe is heated by the super heated steam, it is difficult to cool it down because of its heat capacity. This effect is noticeable because the mainstream density is very low. For a heat pump system using a very-low-density working medium, the pipe heat capacity has strong influence on the total system heat balance. This finding is important for knowing the transient system heat balance and determining the timing of spray start and stop.

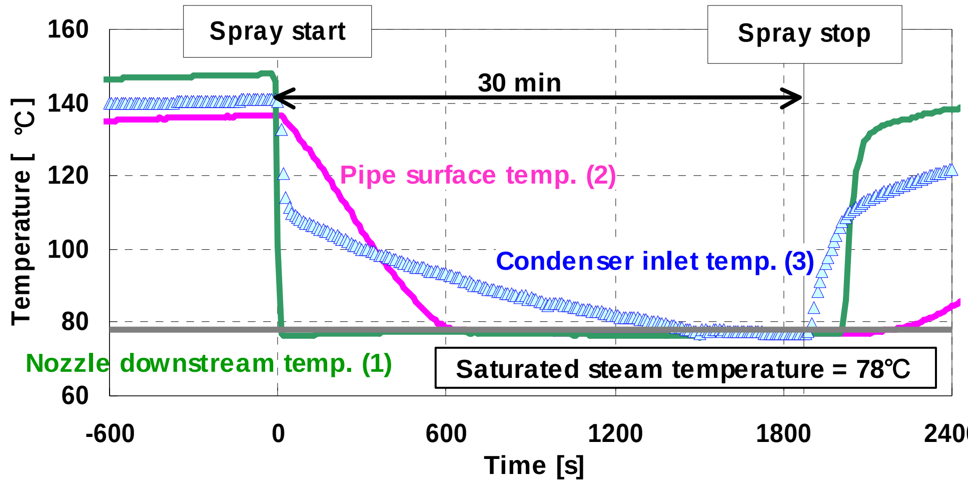

Figure 10: Time history of temperature variations caused by water atomization cooling

5 CONCLUSIONS

A simple heat and mass balance calculation showed that our proposed water refrigerant heat pump would be able to be possible to achieve a high COP above 1.4 on a primary energy basis in some favorable waste heat conditions. Water refrigerant has a great potential for reducing heat pump cost and of improving the system efficiency as a condenser is not needed at compressor discharge.

Feasibility tests of the main components for the steam-generating heat pump were conducted using a newly developed test apparatus. Compressor performance satisfied the multi-stage compressor system requirements. It could achieve a high adiabatic efficiency of over 81% with a pressure ratio of 2.1. The design methodology of the steam compressor was confirmed to be valid within the present range of low Reynolds numbers.

Test results showed that water atomization cooling could cool down super heated steam to saturated steam condition within a moderate convection time of water droplets. It could reduce compression work effectively and improve the system coefficient of performance.

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