DEVELOPMENT OF A COOLING UNIT FOR SOLAR POWERED TECHNICAL EQUIPMENT AT HIGH AMBIENT TEMPERATURES

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Abstract

Solar powered systems with batteries are often used in remote areas to supply technical infrastructure. Enclosures for technical equipment in hot climates need cooling to prevent overheating of sensitive components and batteries. This paper presents a small scaled cooling unit based on a DC-compressor supplied by photovoltaic electricity. All components are directly energized by a battery with a voltage range from 22 V to 28 V. The paper focuses on measures which were taken to increase the efficiency of the 700 W cooling unit as much as possible. This applies mostly to the sizing of heat exchangers and fans. Moreover, safety measures and control functions are explained in the paper. The cooling unit was tested in laboratory under extreme operation conditions and afterwards integrated into a solar powered energy supply system for remote desert areas.

1. Introduction

It was the task to develop an efficient solution for the cooling of technical devices and a large battery system inside a shelter in a solar powered stand-alone system. The cooling demand of the whole shelter could be reduced by constructional design measures like careful thermal insulation and preventing the enclosure from direct solar radiation. After that a peak cooling demand of about 700 W was still remaining. The best solution to provide this quite low refrigerating capacity was the setup of a compressor based refrigeration cycle.

2. System Design for High Ambient Temperatures

2.1. General

The least possible energy consumption is a main design goal in autonomous systems because of the restricted energy availability. Therefore the efficiency (expressed by the COP - coefficient of performance: cooling power divided by the total energy consumption of the unit) of the integrated cooling unit had to be maximized despite high ambient temperatures.

In general the COP of a given refrigeration unit is as higher as smaller the temperature difference between “cold side” and “warm side” is. In our case the temperature inside the shelter is set to a desired value by the customer and the ambient temperature can’t be influenced. So the temperature difference ΔT between air and the refrigerant inside the heat exchangers remains as an accessible and sensible parameter for the efficiency of the cooling unit. This holds for both, the condenser and the evaporator of the system. Because the cooling of the equipment is done by air cooling and the heat is dissipated to the ambient air, both heat exchangers apply fans for the heat transfer.

2.2. Condenser heat exchanger

To achieve a high COP of the cooling unit, the condenser has to perform the heat transfer to the ambient with the smallest technically justifiable temperature difference ΔT between refrigerant and air. To estimate the required size of an efficient heat exchanger the active area A can be calculated with challenging values for the heat transfer coefficient k (10 W/m²·K; that requires almost no fan) and the desired temperature difference (3 K). This challenging values lead to an oversized heat exchanger. For a condensing power Q_C of 1000 W (required cooling power plus power consumption of the compressor) the chosen parameters result to a required area of about 33 m² (1).

$$
\begin{aligned}
Q_C &= k \times A \times \Delta T \\
A &= \frac{Q_C}{k \times \Delta T} \\
A &= \frac{1000W}{10\frac{W}{m^2 K} \times 3K} \\
A &= 33.3m^2
\end{aligned}
\tag{1}
$$

2.3. Condenser fan

The requirement for large k-values (see equation (1)) is connected with an increased energetic effort for the fan. In small cooling units the energy demand for the fans can easily reach the power consumption of the refrigerant compressor. If ΔT is fixed and the heat which has to be dissipated by the condenser Q_C is known, the required air flow can easily be estimated. The following calculation (2) was used to find the volume flow $\dot{V}$ as parameter for the selection of the fan.

$$
\begin{aligned}
Q_C &= c \times \dot{m} \times \Delta T \\
Q_C &= c \times \rho \times \dot{V} \times \Delta T \\
\dot{V} &= \frac{Q_C}{c \times \rho \times \Delta T} \\
\dot{V} &= \frac{1000W}{1000\, J/kg\,K \times 1{,}2\, kg/m^3 \times 3\,K} \times 3600\, s/h = \underline{1000\, m^3/h}
\end{aligned}
\tag{2}
$$

But the heat transfer within the heat exchanger, its pressure drop, the characteristic line of the fan and its power consumption in dependence of the air flow are in non linear correlation and influence each other. One example for these correlations gives Fig. 1 (measured power consumption of the fan vs. its rotation speed). Rough calculations show, that there is an optimum fan speed with respect to a maximum COP (low power consumption of the fan but a sufficient value of the heat transfer coefficient). Fig. 2 shows calculation results for the condenser fan running with full and with half rotation speed. Finally the optimization of the fan rotation speed was done experimentally.

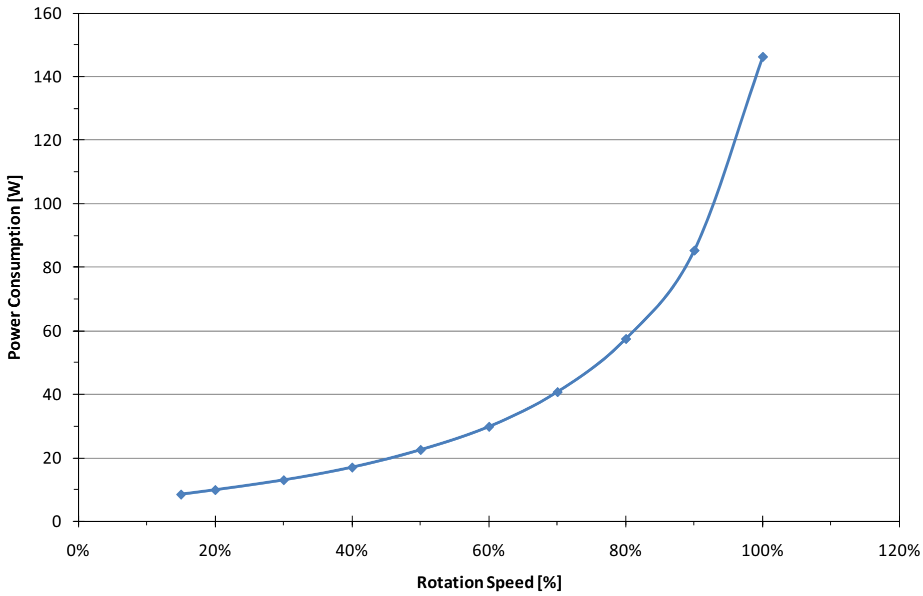

Fig. 1: Measured power consumption of the (axial) condenser fan in dependence on the rotation speed

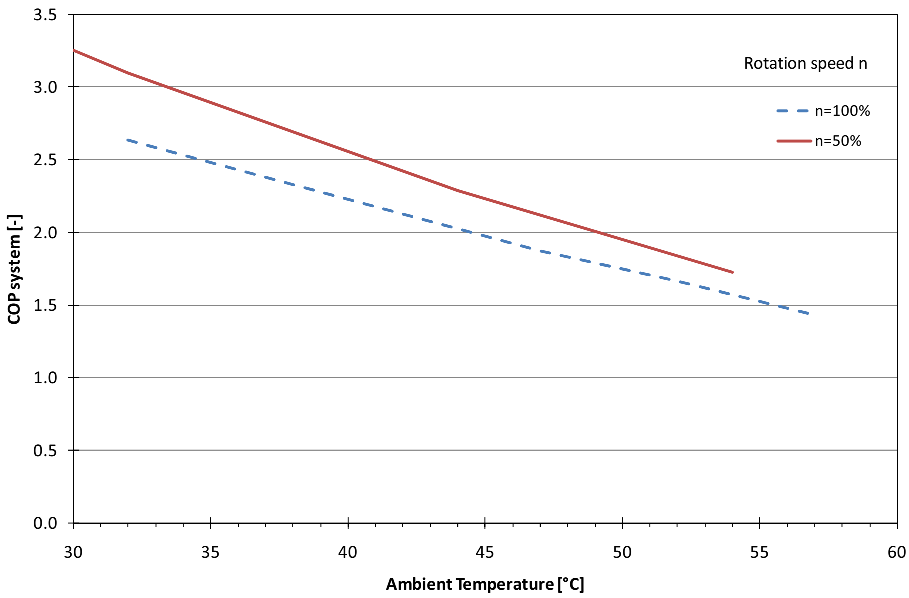

Fig. 2: COP of the cooling unit in dependence on different fan speeds and ambient temperatures

Although an optimum rotation speed was found, three speed steps were defined for the condenser fan to take reserves for heat exchanger fouling due to sand and dust into consideration and to fit the restrictions of the compressor with respect to the condensing temperature (it must be between 30°C and 60°C even at very low and very high ambient temperatures).

Heat exchanger and fan of the evaporator were calculated and chosen under similar optimization aspects but with respect to requirements of the compressor (max. evaporation temperature of +15°C) and the requested shelter temperature (set point between 20°C and 30°C).

3. Safety Measures and Control Functions

3.1. Safety requirements

Some standard safety requirements had to be followed by the system design. To prevent compressor damage the limitation of the minimum suction pressure is necessary. Low suction pressures may occur due to refrigerant losses in case of a leak. Malfunction of the evaporator fan may also cause low suction pressure with icing of the evaporator and a decrease of the heat transfer.
At start up of the cooling unit with high shelter temperatures the evaporation temperature has to be restricted to the allowed 15°C. Otherwise the cooling of the compressor would not be sufficient with the result of a potential defect.
Also the high pressure (discharge) side of the compressor has to be protected against rising of the pressure above the allowed level. Such a pressure increase could result from a failure of the condenser fan or dust at the condenser surface.

3.2. Implementation

To protect the suction side against low pressures and the discharge side against too high pressures a pressure switch was implemented into the appropriate pipes of the cooling cycle. The upper level of the suction pressure is controlled by a crankcase pressure regulator. With increasing pressure the regulator reduces flow cross section and avoids with that an overload of the compressor motor. Additionally the operation of the fans is monitored by flow switches. To protect the condenser against sand and dust a special sand separation grille is used in front of the condenser. Fig. 3 shows a scheme of the condensing unit with integrated safety measures.

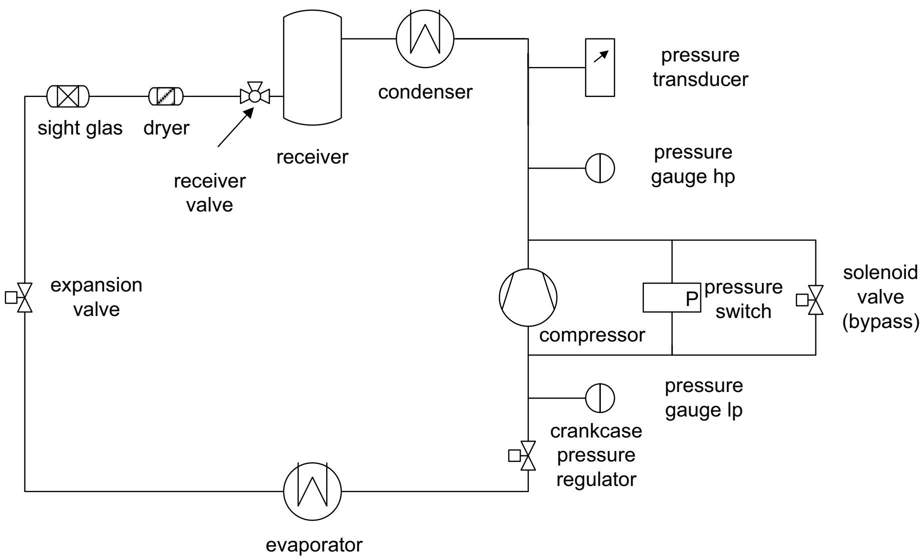

Fig. 3: scheme of the cooling unit with safety measures

4. Design and Test

4.1. Design

The cooling unit was designed as a split unit with separate condensing unit and evaporator. The condensing unit with a footprint of 400 mm x 500 mm was placed outside the shelter and includes the compressor and all necessary parts except of the evaporator with fan and expansion valve. Fig. 4 and Fig. 5 show the condensing unit and denominate the main parts. The evaporator mounted inside the thermal highly insulated shelter is shown in Fig. 6. All electrical components were chosen to fit the voltage range of the 24 V main battery (DC, 22...28V) including the compressor (Danfoss BD350). A compressor bypass was integrated to reduce pressure difference prior and during compressor start up.
The control of the cooling unit is done by the central controller of the solar power supply system. This controller provides all necessary functions like temperature control of the shelter, start up with the bypass valve, monitoring of the cooling system and adjustment of the fan speed. Also the battery monitoring is done by the controller. The large size of the main battery makes it possible to drive the cooling unit in on/off mode according to the shelter temperature. If the state of charge of the main battery is too low, the controller automatically increases the set point of the shelter temperature to the maximum allowable limit to reduce battery load.

Fig. 4: Cooling unit rear view

Fig. 5: Cooling unit front view

Fig. 6: Evaporator inside the shelter

4.2. Laboratory Test

In a first laboratory test the cooling unit was tested in a hot temperature chamber with up to 50°C. The function of the unit was verified under certain extreme conditions with different air temperatures at the condenser and the evaporator. Power consumption, cooling power, temperatures and internal pressures were measured. Pressure switches and regulators were adjusted and the speed steps of the fans were optimized during this test.

4.3. System Test

The final test was done in the target system again at high ambient temperatures. The integration of the control of the cooling unit into the energy management of the energy supply shelter allows the coordination of the cooling in accordance to the energy balance of the system. The test duration was defined to 48 hours. This time is the designed time of operation with sparse sunshine (only little charge from the PV generator). For the test the battery chargers were disconnected completely from the PV generator. The energy supply shelter worked autonomously only supplied by the battery. The ambient temperature was adjusted to a profile of hot days with a daily mean value of 39°C. Fig. 7 shows the measured temperatures (hourly means) during the 48 hours of the load test of the assembled solar energy supply system.

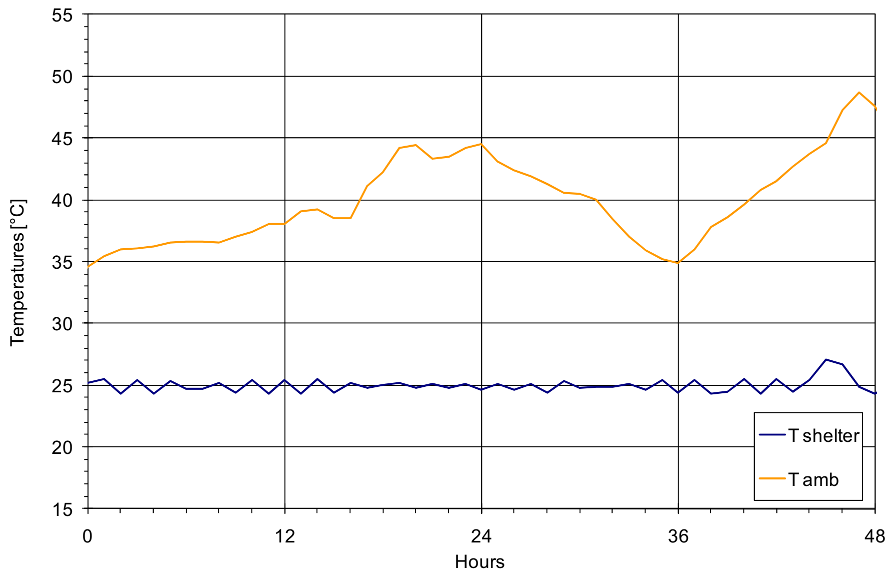

Fig. 7: Ambient and shelter temperatures for two days (hourly values)

The set point of the shelter temperature was 25 °C with an upper and lower temperature hysteresis. The duty cycles of the cooling unit are visible in the one minute resolution diagram in Fig. 8. Depending on the ambient temperature and the internal thermal load the duty time of the cooling unit was from 20 minutes to 50 minutes per hour. The differing mean values of the shelter temperatures in Fig. 7 and Fig. 8 result from different measuring systems located at different places inside of the shelter.

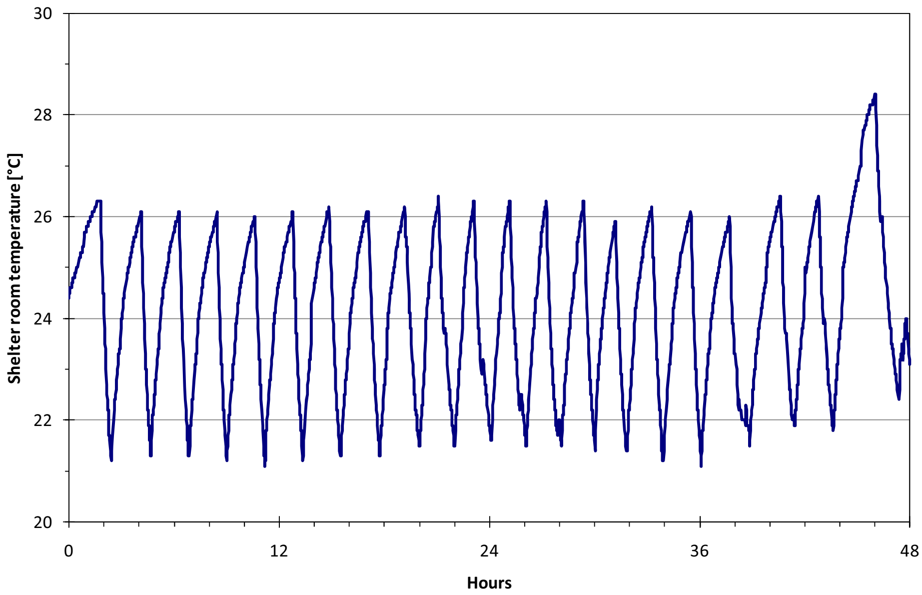

Fig. 8: Temperature inside of the shelter for the same period in minute resolution

5. Conclusions and Remarks

Compressor refrigerating systems which are energized by PV-modules are a preferable choice for small scale solar cooling applications (up to several Kilowatts cooling power). They can be designed according to various needs and set up with industrial components. Some special measures have to be taken to ensure an efficient and reliable operation.

The presented small cooling unit is powered by a large battery which is charged by the photovoltaic generator. That makes the system quite simple and easy to control. If the battery bank should be small for technical or economical reasons, the cooling unit can be driven by a variable frequency drive according to the currently available solar energy. In these cases a cold storage should be integrated into the system [1, 2]. A further reduction of the electronic system can be achieved when the DC input of the variable frequency drive is directly coupled to the PV generator and the battery is only used for auxiliary components [3].

References

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