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Measurement And Application of Performance Characteristics Of A Free Piston Stirling Cooler

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MEASUREMENT AND APPLICATION OF PERFORMANCE CHARACTERISTICS OF A FREE PISTON STIRLING COOLER

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ABSTRACT

Measurements were performed to characterize the performance of a Free Piston Stirling Cooler over a wide range of temperatures and heat lifts. The temperature range investigated was from –120°C to +5°C on the cold head side and from 30 to 60°C at the warm head, while heat lifts from 10 to 100W were evaluated. The publication discusses aspects of the experimental part of the investigation, including a description of a method to quantify thermal losses at the cold end side.

Characteristic maps of measured system efficiency (COP) and input power were drawn as a function of the operating parameters. A regression model was applied to the experimental results, allowing calculation of the Stirling Cooler performance at any operating condition. Subsequently, the regression model has been used for a comparison study between Rankine and Stirling based refrigeration systems. By means of a case study on an upright domestic freezer, it is shown that the obtained Stirling performance characteristics are useful to predict the energy consumption of the final product. The case study also includes those aspects, which have to be taken into account in order to make a proper comparison between a Stirling and a vapor compression based refrigeration system, such as cold/warm side heat exchanger efficiencies. It is concluded that competing efficiency levels on products can be obtained with the Stirling Cooler.

NOMENCLATURE

Q	Heat [W]	T	Temperature [°C]
P	Power [W]	V	Voltage [V]
COP	Coefficient of Performance [-]	I	Current [A]
$\dot{m}$	Massflow [kg/s]	R	Resistance [Ω]
h	Enthalpy [J/kg]	L	Coefficient of induction [H]
UA	Heat transfer coefficient [W/K]		

INTRODUCTION

To properly compare the characteristics of a Stirling based refrigeration system with a Rankine based system (usually applied), first some practical differences between both systems are discussed. Hereafter the measurement system used to characterize the Stirling cooler is explained followed by some of the test results. These test results enable a theoretical comparison of Stirling and Rankine where an example is given for a domestic freezer. Finally the main findings are summarized.

COMPARISON BETWEEN RANKINE AND STIRLING

In the Rankine system refrigerant is transported by a compressor, which pumps the refrigerant from a low to a relatively high pressure. Heat is absorbed in the evaporator and rejected in the condenser. The expansion device reduces the relatively high pressure from the condenser to the relatively low evaporation pressure.

Within the Stirling cooler heat is absorbed at the cold head due to an expansion process and rejected at the warm end due to a compression process. Within an application of this cooling system often additional heat exchangers are necessary to absorb or reject the heat as the surface areas of the heads are limited.

For the compressor applied in a Rankine system the efficiency of the compressor is typically expressed by its coefficient of performance, which is calculated according to the following formula:

(1) $$COP_{rankine} = \frac{Q_{cool}}{P_{input}}$$

The COP of the compressor corresponds with the performance of the compressor at a specific operating point i.e. a specific condensation and evaporation temperature. Globally, different standards apply, e.g. ASHRAE and CECOMAF. These standards only represent the COP at exactly prescribed conditions. In the practical Rankine cycle the COP can be completely different. Therefore, the COP from the catalogue data cannot directly be related to the practical cooling system. For comparison with the Stirling COP it is recommended to calculate the actual cooling capacity on a Rankine system with the following formula (for stationary conditions):

(2) $$Q_{cool} = \dot{m}(h_{evaporator\ exit} - h_{evaporator\ inlet})$$

It needs mentioning that the massflow of the compressor can be derived from catalogue data for a certain condition (evaporation, condensation temperature). However, the enthalpy values, representing the inlet and exit conditions of the evaporator), are different for each application.

For the Stirling cycle the efficiency of the cooler is also expressed with its COP. This COP is calculated according to the following formula:

(3) $$COP_{Stirling} = \frac{Q}{P_{input}}$$

The COP of the Stirling cooler depends on the cold head temperature, warm head temperature and heat lift. In principle, one can compare the COP of the Rankine compressor with the COP of the Stirling cycle applying formula 2 and 3. However, one should take into account that generally Rankine refrigeration systems are controlled with a thermostat. This thermostat switches the compressor on or off, which differs from the free piston Stirling cooler, which operates in a continuously running mode. The on/off behavior in the Rankine system results in extra losses caused by the following effects:

- Thermodynamic losses; the average condensation temperature is higher and the average evaporation temperature is lower with respect to a continuously running compressor with adjusted capacity.
- Start/Stop losses; at the moment the compressor switches off, vapor from the condenser enters the evaporator. This vapor condenses in the evaporator yielding an extra heat load into the appliance.
- At the start of the compressor the current of the compressor is relatively high with respect to a continuously running compressor.

It needs mentioning that for variable speed compressors these losses are avoided. However, in some cases also these kind of compressors have to operate in an on/off mode, when low cooling capacities are demanded.

For the Stirling cooler the amount of heat from the cold and warm head has to be transported through adequate heat exchangers. This transport yields the following losses:

- Temperature losses at the cold and warm heads caused by the extra heat exchangers. In case of secondary fluids temperature losses of the fluids exist, while for a fin construction with a large surface, the fin efficiency will be lower than unity.
- Pump losses if secondary fluids are used.

It has to be noted that generally the complete construction of a refrigeration system changes, if a Stirling cooler is used instead of a normal Rankine compressor. This makes comparison of both cooling systems not a straightforward task.

THE MEASUREMENT SYSTEM

With respect to the measurements of the Stirling cooler the following details are mentioned (See figure 1).

- The Stirling motor evaluated was of the type M100B, serial number 121.
- The maximum voltage supply of the cooler was 12V (AC), at a frequency of 60 Hz.
- The Stirling cooler was supplied without a (feed back) control unit (for normal use this unit is implemented).
- The warm head of the motor was cooled with water of a controlled temperature, supplied by a pump.
- An electrical heating element was placed on the cold head. The heater was operated by means of a DC Voltage supply allowing to set different heat lifts.

- In order to minimize the heat flow from the ambient to the cold head, the cold head was insulated with Armaflex material.
- The Stirling cooler was put in a climate chamber. Measurements were taken at 25°C ambient temperature.
- Data from the tests was taken after a stable operating period of at least 0.5 hours.

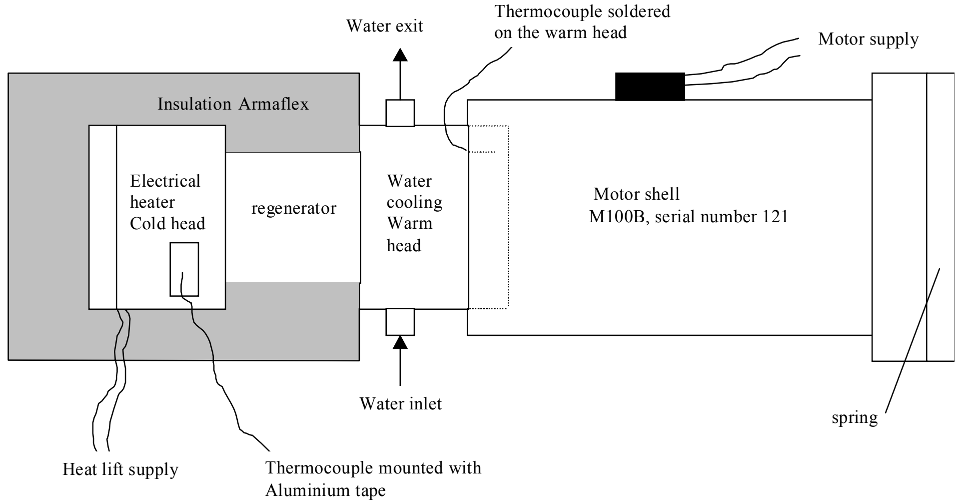

Figure 1; The Stirling cooler

MEASUREMENT PROCEDURE

Before the actual tests were started, the quality of the insulation material (Armaflex) around the cold head and regenerator was measured (See figure 2).

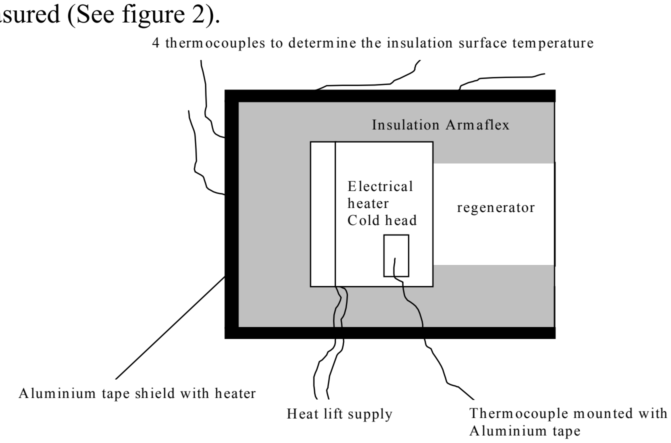

Figure 2; Cold head insulation covered with Aluminium tape

The insulation was covered with an aluminium shield. Through this shield an electrical heater was lead, which was only used to determine the insulation quality. On the aluminium shield 4 thermocouples were mounted. With these thermocouples the representative insulation surface temperature can be calculated. To obtain the heat transfer coefficient of the insulation, two measurements were performed at the same cold and warm head temperature:

Test 1: Without using the heating wire on top of the insulation a heat supply $Q_{DC\ heater1}$ was applied. This test yielded an insulation surface temperature T_1. For this test the following equation is valid:

(4) $$Q_1 = Q_{DC\ heater1} + UA_{insulation}(T_1 - T_{cold\ head})$$

Test 2: Using the heating wire and applying a heat $Q_{DC\ heater2}$ such that $T_{cold\ head}$ is the same as in test 1. This test yielded an insulation surface temperature T_2. For this test the following equation is valid:

(5) $Q_2 = Q_{DC\ heater2} + UA_{insulation}(T_2 - T_{cold\ head})$

Since the same cold head temperature (as well as the same warm head temperature and AC cooler supply) was applied in both tests, Q_2 is equal to Q_1. Now the heat transfer coefficient of the insulation can be calculated according the following formula:

(6) $UA_{insulation} = \dfrac{Q_{DCheater1} - Q_{DCheater2}}{T_2 - T_1}$

The total heat lift measured can be calculated with the following formula:

(7) $Q_{heat\ lift} = Q_{resistance} + UA_{insulation}\ (T_{outside\ insulation} - T_{cold\ head})$

Tests were performed at the conditions described in table 1.

Condition	Normal tests	Low temperature tests
Warm head temperature [°C]	30 – 45 – 60	30
Cold head temperature [°C]	Between –60 and +5	Between -120 and -60
Nominal heat lift [W]	10 –30 – 50 – 100	20 - 40
Ambient temperature [°C]	25	25
Cooler position	Horizontal	Horizontal

Table 1; Test conditions

TEST RESULTS

For each test point the motor efficiency of the cooler can be calculated according to the following formula:

(8) $\eta = \dfrac{P_{input} - P_{loss}}{P_{input}}$ in which: $P_{loss} = \dfrac{V_{rms}^{\ 2}}{R_s} + I_{rms}^{\ 2} R_{dc}$ and $R_s = \dfrac{(\omega L)^2}{(R_{ac} - R_{dc})}$

The heat transfer coefficient of the insulation around the cold head and the regenerator is 0.025 W/K (based on the cold head temperature and the outer insulation surface temperature). This coefficient is measured with an accuracy of +/- 0.005W/K.

In order to predict the performance of the Stirling cooler a regression model was applied on the test results. The following regression function, which calculates the input power of the cooler, was used:

(9) $$P_{input} = a_0 T_c^3 + a_1 T_c^2 + a_2 T_c + a_3 T_w^3 + a_4 T_w^2 + a_5 T_w + a_6 Q^3 + a_7 Q^2 + a_8 Q + a_9 (T_w - T_c) Q + a_{10} (T_w - T_c)^2 Q + a_{11} (T_w - T_c) Q^2 + a_{12}$$

Note that the heat lift is an input parameter of this equation. With this equation and optimized coefficients an average error (with respect to the measured points) on the input power of 2.3% is found between –60 and +10°C cold head temperature. With equation 3, one can obtain the COP of the Stirling cooler now.

In the following table the performance of the Stirling cooler is given for two standard operating points.

Cold head temperature [°C]	0	0
Warm head temperature [°C]	35	30
Heat lift [W]	100	33
COP [-]	2.27	2.90

Table 2; Performance at standard operation points

In figure 3 the COP values measured are drawn combined with the regression lines. The lines of 100W heat lift do not cover the complete cold head temperature field because of cooler restrictions. Namely, at 30°C warm head temperature the maximum input voltage (12V) is reached at approx. -25°C cold head temperature.

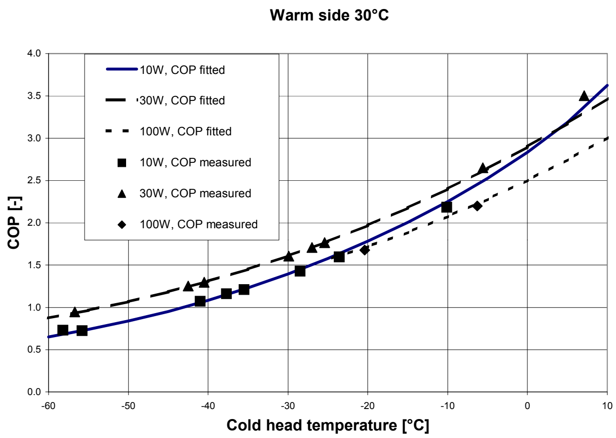

Figure 3; COP values measured and fitted at 10, 30 and 50 Watts heat lift at 30°C warm head temperature

In figure 4 the COP is given versus the heat lift. From this figure it can be concluded that:

- At very small heat lifts (<20W), a reduction in COP can be noticed.
- The effect of the heat lift on the COP is relatively small; e.g. if the heat lift increases from 20 to 80W at –20°C cold head temperature, the COP decreases only from 1.55 to 1.40.

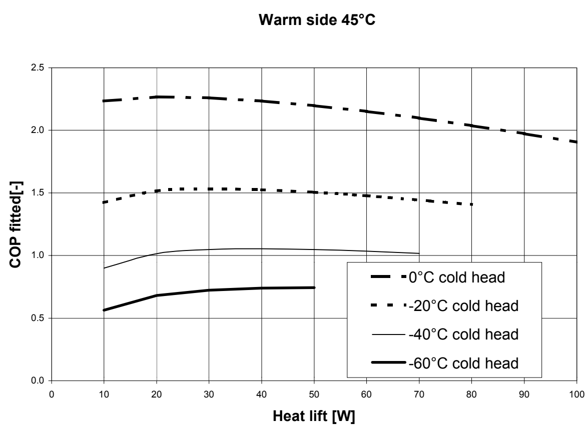

Figure 4; COP values versus the heat lift for different cold head temperatures at 45°C warm head temperature

If the ratio between the COP versus the COP_{carnot} is plotted (See figure 5), one can observe a flat profile over the entire cold head temperature range. Only at extreme high or low heat lifts the ratio reduces.

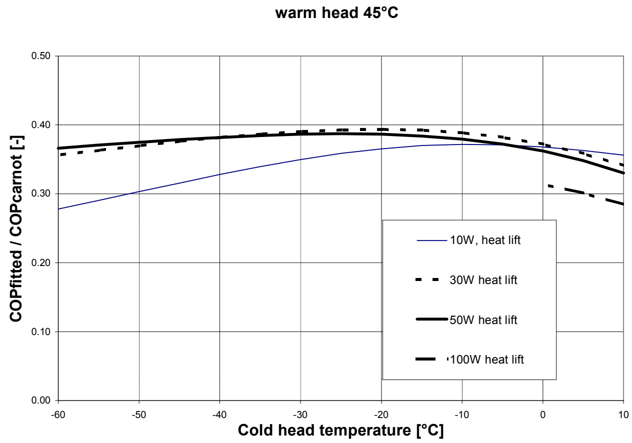

Figure 5; COP divided by COP_{carnot} versus cold head temperature

In figure 6 the motor efficiency is given versus the input power of the cooler a various cold and warm head temperatures. Generally efficiencies between 0.80 and 0.85 are calculated for the specific cooler.

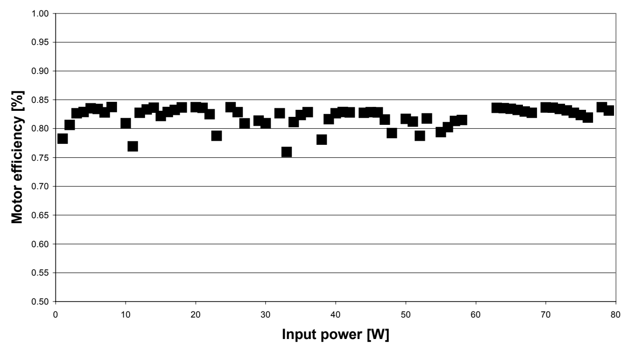

Figure 6; Motor efficiency versus the input power

With the same cooler also very low temperatures can be achieved as can be seen in figure 7. At 40 W heat lift a temperature of –92°C can be obtained at a COP value of 0.44, while at 20W heat lift this temperature can be –115°C. For this condition the COP of the cooler drops to 0.20.

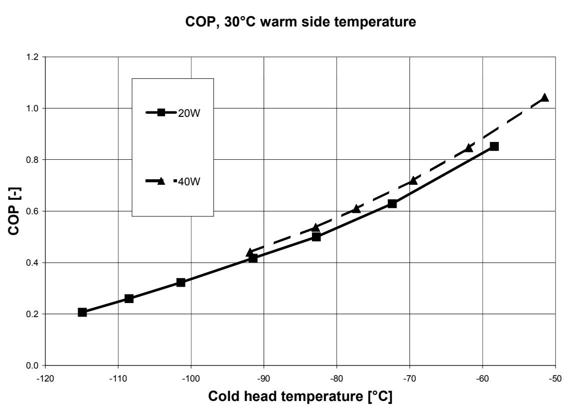

Figure 7: Low temperature runs

CALCULATION OF THE ENERGY CONSUMPTION

In this chapter an example is given how to determine theoretically the efficiency of a Stirling based system using the experimentally obtained data map. Suppose a domestic upright freezer (360 Litres) with a heat load factor of the insulation of 1.30 W/K and an average freezer temperature of –20°C. The maximum ambient temperature for this product is 43 °C (tropical category).

For the Rankine system the following specifications are assumed:

- Compressor to be applied: Variable speed compressor, which has a cooling capacity of 187 W and a COP of 1.69 at 3200 RPM at ASHRAE conditions. Note that this compressor has an efficiency, which is very high compared to average compressors applied in domestic appliances.
- The cooling capacity increase due to the capillary suction tube heat exchanger is 15%.
- The condensation takes place in the complete condenser (no superheating and subcooling).
- The evaporation takes place in the complete evaporator (no superheating).
- The refrigerant applied is R600a (isobutane).

For the Stirling system the following specifications are assumed:

- The Stirling cooler to be applied is the M100B cooler.
- The energy consumption of the electronic control is not taken into account.

To evaluate the difference between both systems the electrical energy consumption of each system is calculated at various ambient temperatures. For the Rankine process it is assumed that the evaporator thermal conductance is 7 W/K and for the condenser 10 W/K. Based on the heat load of the appliance, the evaporation and condensation temperatures can be calculated (using a simple linear model). From a compressor map the operating speed is determined, which matches the desired heat load. The required compressor input power is obtained from the same compressor map. The result is shown in figure 8 over the ambient temperatures.

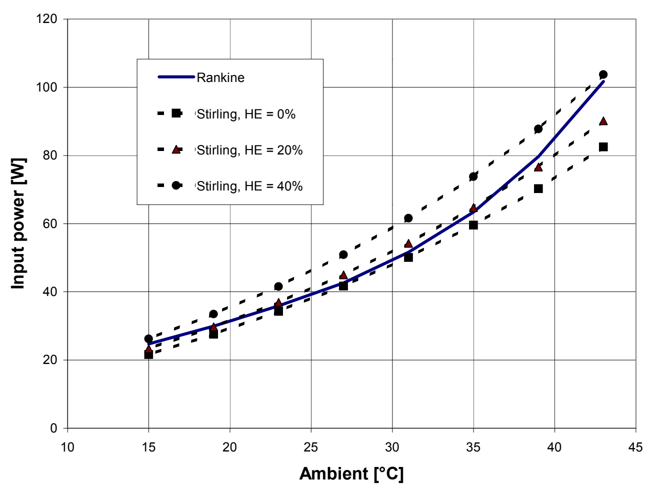

Figure 8: Estimated input power of the Rankine system with a high efficient variable speed compressor and the Stirling system (with the same heat exchanger conductance as for the Rankine, 20% lower and 40% lower respectively)

The variable speed compressor can not be operated under a certain minimum speed (due to minimum lubrication levels). In this example the compressor had to be operated in on/off mode an ambient temperature of 28°C or lower. For these conditions it is assumed that the compressor operates at the minimum speed here. During this mode the temperature differences at the condenser/evaporator are increased proportionally with the running time percentage. Note that at 43°C ambient temperature the compressor is running at a speed of 3850 RPM, which is near its maximum of 4000 RPM. This indicates that the variable speed compressor is properly selected for a tropical climate class appliance.

For the Stirling based system, a similar procedure is applied where instead of the compressor map, the Stirling performance map described earlier is applied. There is no minimum capacity to the Stirling system so it also

operates in continuous mode at low ambient temperature. For the thermal conductances it is assumed that these are equal, 20 % and 40 %, respectively lower than those of the Rankine based system. The thermal conductance on both the warm and cold side of the Stirling based system, is likely to be lower than those for the Rankine since an additional heat transfer mechanism (e.g. secondary fluids) may be required.
It can be seen that at the same conductance the Stirling system has an equal or lower energy consumption than the Rankine based system over the ambient temperature range. Obviously at reduced conductances the energy consumption of the Stirling based system increases. It is of interest to see that the Stirling performs relatively better at the high ambient temperatures. This is due to the fact that the process outperforms the Rankine process at large temperature lifts. Also at low ambient temperatures the Stirling performs relatively better. This is due to the linear motor concept allowing the possibility to operate at very low capacities, whereas the variable speed reciprocating compressor obtains a reduced efficiency due to friction losses and due to the fact that on/off cycling is required here.

CONCLUSIONS

By means of a characteristic map of the performance of a Stirling cooler, a comparison with the Rankine system can be made. To obtain such a map experimental tests on the Stirling cooler were performed in combination with a regression analysis. From these tests it is concluded that:

- At very small heat lifts a relatively small reduction in COP is noticed.
- Motor efficiencies between 0.80 and 0.85 were found for the cooler.
- At cold head temperatures of -92°C, COP values of 0.44 can still be obtained with a heat lift of 40 W.

A comparison between a Stirling system and a Rankine system is not a straightforward task. Heat exchangers efficiency differences between both systems have to be taken into account to determine the actual temperature of the cold en warm head of the Stirling cooler or the evaporation and condensation temperatures of the Rankine system. In addition on/off losses of the Rankine sytems deteriorates this system substantially. A comparison was made between a high efficient variable speed compresser and the Stirling cooler evaluated on a domestic freezer. It was concluded that at high ambient temperatures the performance of the Stirling cooler can be better at high heat lifts. Also a low ambients the Stirling can perform better. This is due to the linear motor concept allowing the possibility to operate at very low capacities, whereas the variable speed reciprocating compressor obtains a reduced efficiency due to friction losses and due to the fact that cycling is required for the Rankine system.

ACKNOWLEDGEMENTS

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