A Numerical Analysis of a Kinematic Stirling-Cycle Heat Pump for Space Conditioning Applications*

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ABSTRACT

A computer simulation was performed on a kinematic Stirling heat pump (modified from the GPU-3 heat engine mode) using the NASA third-order code. The effects of outdoor air temperature, mean gas pressure, crank speed, dead volume, and working-space isothermalization were investigated. It was found that COP and heat capacity were relatively insensitive to outdoor air temperatures.

INTRODUCTION

Stirling machines in the past were developed mainly as heat engines for power generation and as cryocoolers for ultra low-temperature refrigeration and gas liquefaction. Vapor compression heat pumps, which are by far the most efficient residential space conditioning devices in mild climates, dominate the marketplace.

However, the steady-state performance of a vapor compression heat pump drops rapidly as the outdoor air temperature drops.[1] Duty cycling at mild ambient temperatures, frosting losses of the outdoor coil, and cycle reversals for defrost further deteriorate seasonal performance. Electric resistance heating is therefore needed to meet the heating load at low outdoor temperatures.

An alternate heat pump cycle study[2] was done to evaluate various novel cycles for space conditioning applications that might offer improved performance over the basic vapor compression cycle concept. The Stirling heat pump which offers a number of potential advantages including high efficiency, low sensitivity to heat source temperature, ease of defrosting, and capacity modulation by pressure control, was identified as a promising alternative heat pump concept.

The Stirling machine has long been proven to be cost-effective for cryogenic applications, however its applicability in the residential space conditioning field has only recently been explored. To substantiate the potential advantages, a computer simulation on a kinematic Stirling heat pump for space heating was made. Results from parametric and sensitivity analyses are reported to delineate the effects of various design parameters on performance.

THE COMPUTER MODEL

The third-order computer model used in this study is a modified version of the computer model written by NASA.[3]

In the model, the existing GPU-3 combustion heat absorber was replaced by a configuration similar to the heat rejector design, resulting in a water-to-water heat pump model. With some simple assumptions for water-to-air heat *exchanger performance*, results from the model yielded the performance of an air-to-air Stirling heat pump.

CASE STUDY - THE GENERAL MOTORS GPU-3 ENGINE MODELED AS A HEAT PUMP

Selection of the GPU-3 engine for heat pump modeling was based on the following:

1. It represents a "yardstick" by which to compare other Stirling heat pump machines.

2. Complete geometric specifications are in the public domain.

3. The NASA computer code used for simulation is tailored specifically for the GPU-3 engine and validated by NASA against GPU-3 engine test data.

Figure 1 shows a schematic of the Stirling heat pump. The rhombic drive motion may be seen from the views of the drive given in two positions.

Replacement of the GPU-3 combustion heat absorber[3] with a shell-and-tube heat rejector[3] configuration allowed for an outdoor and indoor heat exchanger to communicate with the heat absorber and the heat rejector of

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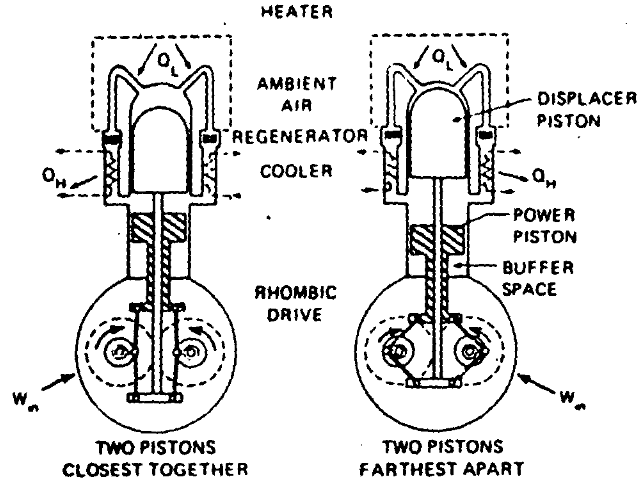

Fig. 1. Schematic of the Stirling-cycle heat pump.

the Stirling heat pump, respectively, by means of a circulating heat transfer fluid. These outdoor and indoor units could be similar to air-refrigerant heat exchangers common to the industry. Figure 2 shows a schematic of a possible air-to-air Stirling heat pump system.

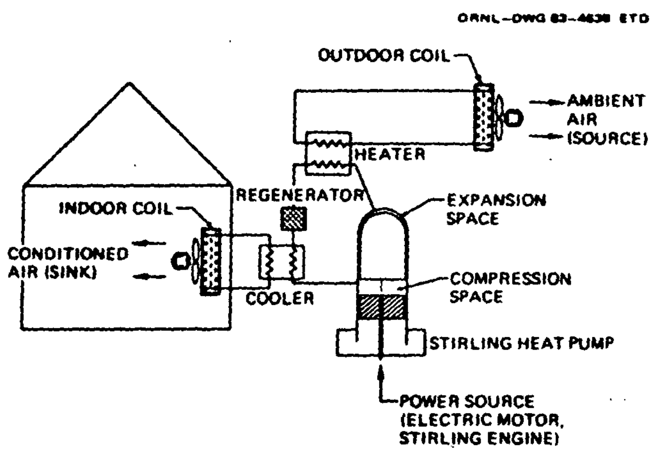

Fig. 2. Stirling-cycle heat pump system integration.

HEAT EXCHANGERS MODELED

The new heat absorber design modeled consisted of 8 car[redacted] 39 small-diameter round tubes in a shell-and-tube configuration. A mixture of water and 50% ethylene glycol was chosen as the circulating fluid because of its good heat transfer properties and low temperature applicability.

Three heat exchanger configurations were simulated: (1) Base Case I, the original heat absorber-regenerator-heat rejector configuration, (2) Base Case II, with the modified heat absorber design and existing regenerator and heat rejector of Base Case I, and heat absorber and ne... Case II.

Dimensions of Base Case I, II, and II heat exchangers are given in Table 2 o reference 4. An 18% increase in gas-sid heat transfer area and a 73% decrease in dea volume is noted for the heat absorber in Bas Case II when compared to Base Case I. I Base Case III, shell-side flow passes fo both the heat absorber and heat rejector ar doubled. Also, the heat rejector hea transfer area and dead volume are increase 50 and 39% respectively, based on result found by performing a sensitivity analysis o heat exchanger tube number and tube length.

In all three cases, no changes to th GPU-3 regenerator were made because it wa noticed that the regenerator had an extremel high effectiveness. Therefore, from a hea transfer point of view the regenerator seeme to be well designed and well matched to th system.

RESULTS FROM HEAT PUMP MODELING

Parametric studies[4] were conducted t determine the effect of changes in mean ga pressure, crank speed, ambient (outdoor) temperature, and working fluid. The effect o changes in heat exchanger (ex-regenerator tube length, tube number, and shell-side flo passes on heat pump performance was als investigated.

In Figures 3 through 6 a temperatur difference* of 16.7K (30 °F) was simulate between the outdoor air temperature and th heat absorber-side fluid inlet temperature i order to initiate the most conservative cas for an outdoor coil.

Since the GPU-3 engine was originall designed for a coolant flow rate of 6.31 10^{-4} m^3 /s (10 gpm) through the shell-and tube heat rejector, it was decided to us this value for the heat absorber also.

EFFECT OF HEAT EXCHANGER DESIGN PARAMETERS

Dimensional changes in heat exchang tube number and tube length provided for t combined effects of changes in total surf area, cross-sectional flow area and volume the heat exchangers. It was found t increasing the number of heat rejector tu by 50% increased COP by about 5.5% decreased heat output by a similar am (5.4%). Increasing heat absorber tubes 50% increased COP by 2.2% but decreased .

*In an outdoor coil of a typical v compression heat pump, the temper difference between the entering air refrigerant can range from 3.3 K (6°. 16.7 K (39°F) (ref. 1).

output by 4.8%. Increasing tube length in the heat rejector by 50% increased COP by 3.1% but decreased heat output by 6.3%. Increasing only the heat absorber tube length did not provide significant improvement in COP.

The higher COP and lower heat output achieved by increasing tube number and tube length was probably due to a lower temperature swing existing in the compression working space as a result of a lower pressure ratio. The lower pressure ratio was attributed to the greater dead volume expected by increasing the size of the heat exchangers. The analysis also showed that the heat rejector was more sensitive to increases in size. This is because as a heat pump the heat rejector delivered a greater amount of heat.

Other parameters including shell-side flow passes and water-glycol flow rates were also investigated.[4] The results showed only a slight change in heat pump performance.

Based on the combined results of the sensitivity analysis, the Base Case III heat exchanger configuration was chosen for the parametric runs.

EFFECT OF HEAT EXCHANGER CONFIGURATION

Stirling Heat Pump performance data are presented in Figure 3 for three categories of heat exchanger design (Base Cases I, II, and III).

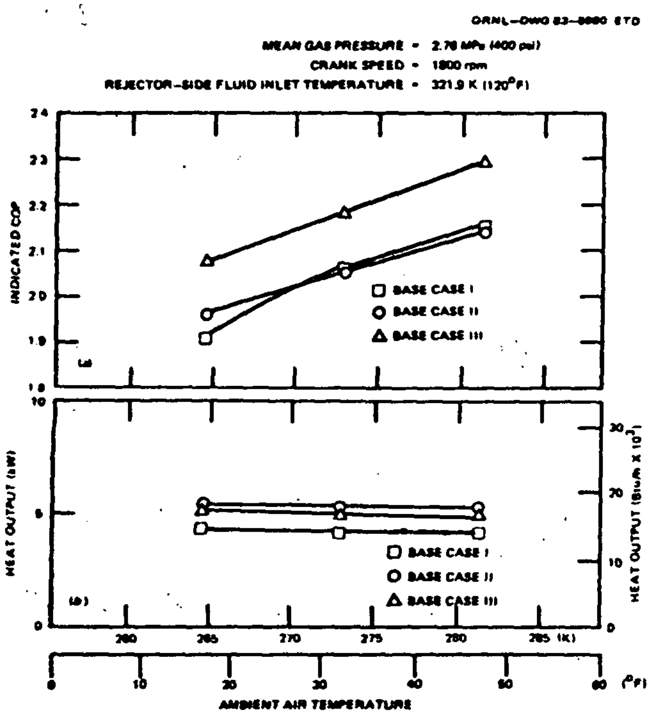

Fig. 3. Performance for three heat exchanger configurations using hydrogen: (a) COP as function of ambient air temperature and (b) heat output as function of ambient air temperature.

Indicated COP (ex-mechanical and parasitic losses) for Base Cases I and II appear to be similar until ambient air temperatures fall below 269.7 K (26°F). The heat output for Base Case II averages about 25% higher than Base Case I regardless of ambient air temperature.

Values of COP, using Base Case II configuration were about 9% higher than those obtained with Base Case I, however, heat output was slightly below that for Base Case I. For Base Case III, the COP ranged from 2.08 to 2.3, at the ARI ratings points of 264.7 and 281.3 K (17° and 47 °F), respectively, while heat output remained fairly constant at about 5 kW (17,064 Btu/h).

The analysis shows that the 50% increase in heat rejector dead volume for Base Case III configuration tends to lower the pressure ratio in the working spaces resulting in a lower power input and a higher COP. Increasing the heat transfer area in the heat rejector increases the heat output rate.

EFFECT OF WORKING FLUID AND SOURCE TEMPERATURE

Figure 4 shows that indicated COP values increased with increasing ambient temperatures, but heat output values remained fairly constant for the range of temperatures covered. Also, COP is greatly increased by

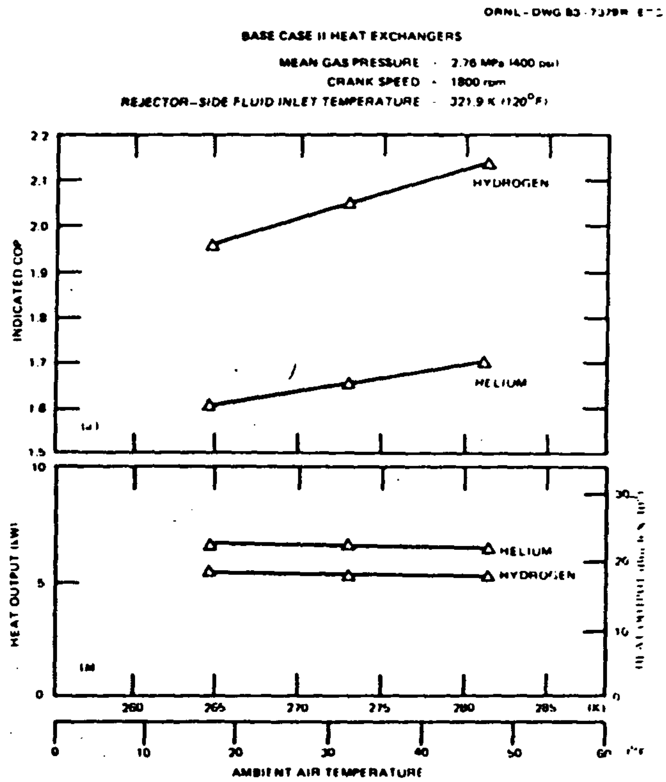

Fig. 4. Stirling-cycle heat pump performance for helium and hydrogen: (a) COP as function of ambient air temperature and (b) heat output as function of ambient air temperature.

using hydrogen as the working gas; however, a decrease in heat output is noted. At a mean gas pressure of 2.76 MPa (400 psi) and speed of 1800 rpm, COP values increased as much as 25% while heat output decreased by about 24% for hydrogen. The increase in COP with hydrogen is mainly attributed to the lower flow losses (low density) through the heat exchangers. The decrease in heat output with hydrogen is due to lower values for heat transfer coefficients between the heat exchanger wall and gas. Hydrogen has the highest thermal conductivity, lowest density, and a low specific heat on a volume basis. Helium has a lower volumetric specific heat than hydrogen and a comparable thermal conductivity, but a density almost twice that of hydrogen.

For ratios of specific heats (k) close to 1, the heat addition and heat rejection processes with variable volumes become nearly isothermal ($PV^k = C$, a constant, defines an isothermal process when $k = 1$). The factor, k, is also an indication why hydrogen ($k = 1.40$) achieved higher COP values than helium ($k = 1.66$) with adiabatic variable volumes.

PARAMETRIC ANALYSIS WITH MODIFIED HEAT EXCHANGERS

Performance data from a parametric analysis are presented in Figures 5 and 6

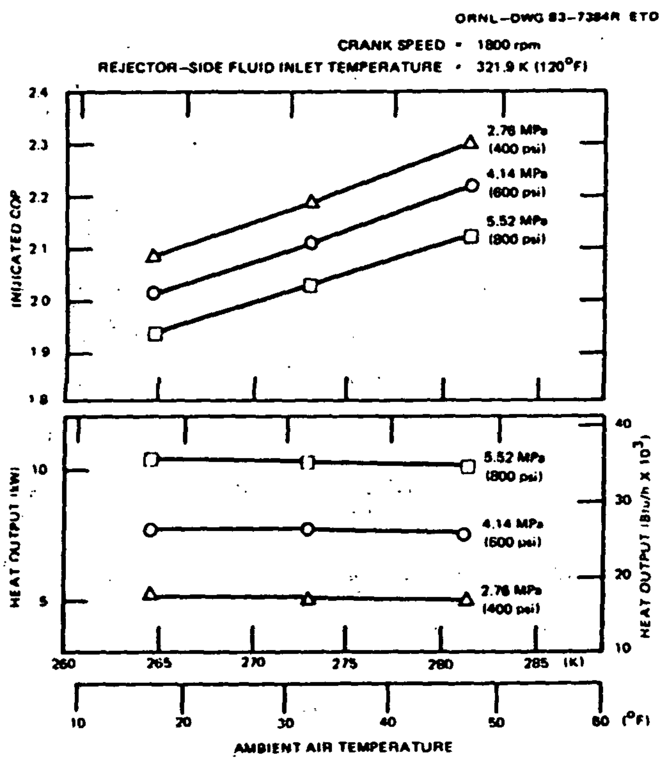

Fig. 5. Performance with modified heat exchangers as function of ambient air temperature and mean gas pressure at crank speed of 1800 rpm.

using the Base Case III configuration heat exchangers and hydrogen as the working fluid.

In Figure 5, for a fixed pressure, indicated COP decrease slightly with decreasing ambient air temperature. However, unlike vapor compression heat pumps, COP for a Stirling heat pump drops only moderately as ambient air temperature drops. Also, in contrast to vapor compression systems, the Stirling heat output remained unsusceptible to changes in ambient air temperature. The low performance sensitivity to outdoor temperature suggests the possible application of Stirling heat pumps in cold climate.

Figure 5 also shows that heating COP decreases and heat output increases with increasing pressure levels. The COP decreases because of the higher temperature difference between the compression space and expansion space at the higher pressures. The increase in heat output with increasing gas pressure is due to the larger mass of working gas in the cycle which must alternately be heated and cooled.

The best range in COP varied from 2.0 to 2.3 for ambient air temperatures of 264.7 to 281.3 K (17°to 47°F) and a mean gas pressure of 2.76 MPa (400 psi) and a crank speed of 1800 rpm. At these operating conditions, the heat output remained relatively constant at 5 kW (17,064 Btu/h).

Figure 6 shows clearly that heat pump COP decreases and heat output increases as crank speed increases for a fixed ambient air temperature and mean gas pressure. At higher speeds, flow losses through the heat exchangers increase. Also, as speed increases, compression-space gas temperature increases while expansion-space gas temperature decreases. As a result of these effects, COP decreases and heat output increases.

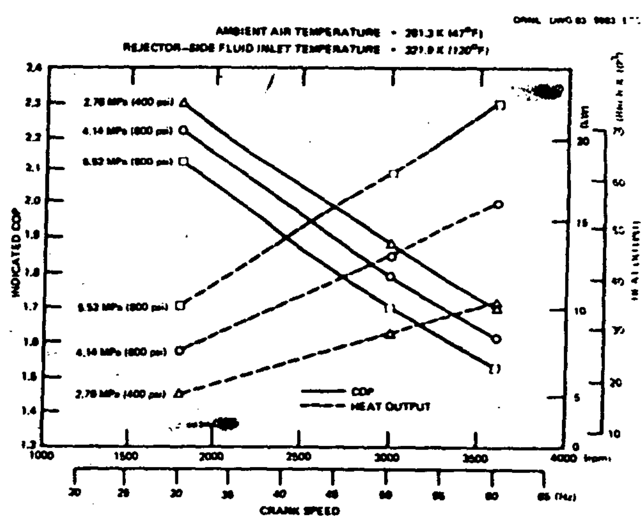

Fig. 6. Performance with modified heat exchangers as function of crank speed and mean gas pressure.

EFFECT OF TEMPERATURE DIFFERENCE BETWEEN OUTDOOR AIR AND ABSORBER-SIDE HEAT TRANSFER FLUID INLET TEMPERATURE

Most simulation results presented earlier were obtained by choosing a very conservative temperature difference [16.7 K (30 °F)] between (1) the outdoor air temperature and heat absorber-side water-glycol inlet temperature and (2) the heat rejector-side water-glycol inlet temperature and indoor air exit temperature. Consequently, a higher temperature difference existed between the hot and cold sides, resulting in lower COP.

Figure 7 illustrates that decreasing the temperature difference between air and fluid by half increases the COP up to 9.7%, yet the heat output remains practically unchanged at about 5 kW (17,064 Btu/h).

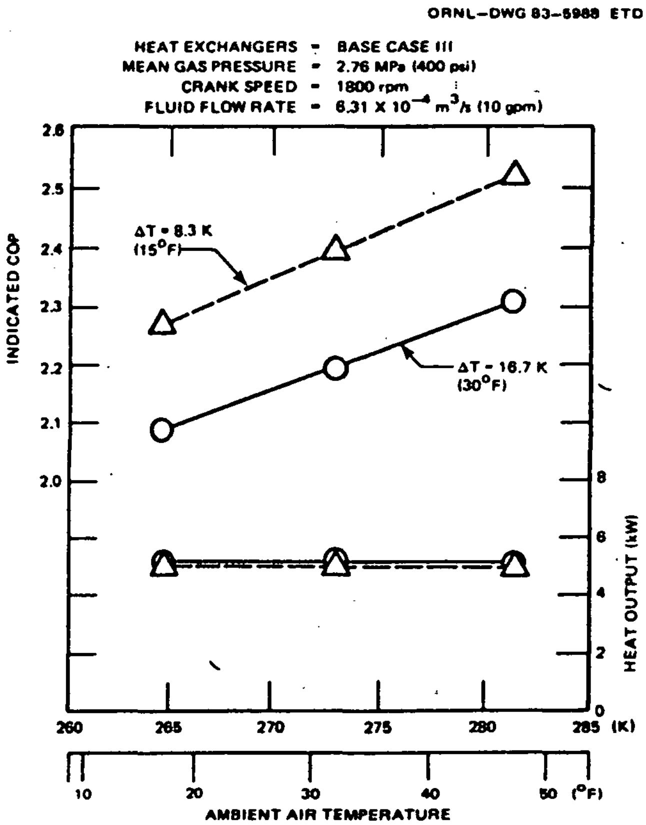

Fig. 7. Sensitivity of ambient air temperature to heat absorber-side heat transfer fluid inlet temperature.

EFFECTS OF ISOTHERMALIZATION

The theoretical loss of performance when the variable volume chambers are adiabatic rather than isothermal is more significant in a Stirling heat pump, because of its operation across a small temperature difference (cold to hot). Benson[5] has shown that ring-filled "Thermizers" heat exchangers can adequately isothermalize the periodic compression and expansion process; the degree of isothermalization being dependent on frequency, mean gas temperature, and pressure. Benson demonstrated that high NTU values can be obtained for a closed chamber compression/expansion process utilizing the Thermizer concept. From Figure 7 of reference 6, NTU values for the process with thermizers can be calculated to be about 80 as compared to 0.07 without Thermizers, implying approximately a 1140 increase in NTU when the Thermizers are evaluated as heat exchangers.

To determine the effect of isothermalization, computer runs were made by increasing the calculated gas-side heat transfer conductance by 1000 in both the expansion and compression space. Figure 8 shows that indicated COP improved notably. For a fixed

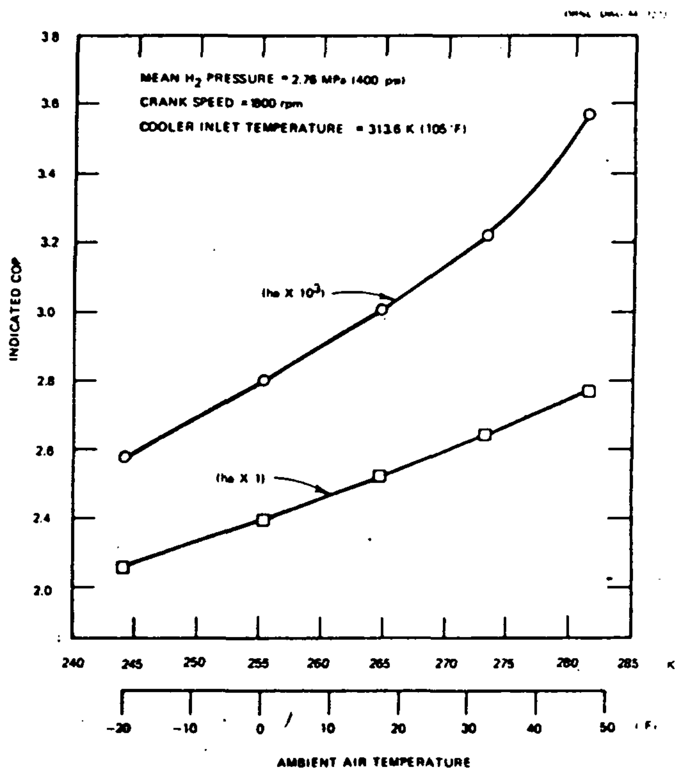

Fig. 8. Effects of isothermalizing the expansion and compression spaces of the modeled Stirling heat pump.

operating condition, the indicated COP increased 19 to 28% above the adiabatic case for the range of outdoor air temperatures indicated. These results are encouraging, but it must be pointed out that these isothermalization predictions are based on [illegible] high values for gas-side heat transfer conductances which are unlikely to be ever obtained in reality. The simulated COP improvement over the adiabatic cycle may or may not be achievable, therefore data in Figure 8 should be used with caution. It was the intent here, to determine only what effect, if any, heat transfer enhancement had on Stirling heat pump performance.

SUMMARY

The main results of this analysis on a kinematic Stirling engine mode[illegible] operate as a heat pump (heating mode), may be summarized as follows:

1. The indicated COP ranged from 2.14 to 2.53 for outdoor air temperatures between 255.2 K ($0\,^{0}F$) and 281.3 K ($47\,^{0}F$) respectively. Heat capacity ranged from 5.1 kW (17,400 Btu/h) to 4.9 kW (16,724 Btu/h). The above performance was obtained using hydrogen at a mean gas pressure of 2.76 MPa (400 psi) and a crank speed of 1800 rpm for an assumed temperature difference of 8.3 K ($15\,^{0}F$) between outdoor air and heat absorber-side fluid inlet temperature.
2. The COP and heating capacity were relatively insensitive to outdoor air temperature.
3. The COP decreased while heat capacity increased with increasing mean gas pressure and/or crank speed.
4. For the same operating conditions, the COP predicted with the hydrogen working fluid was 25% greater than with helium, yet the heat capacity decreased by 24%.
5. Changes in heat exchanger dead volume, which affects gas compression ratio and flow friction, had a significant effect on COP and heat capacity for a given set of operating conditions.
6. Isothermalization of expansion and compression space improved COP considerably for a given set of operating conditions.

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