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# Design and development of a residential gas-fired heat pump

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**Abstract**

Heating, ventilating, and air-conditioning equipment consumes 43% of the total primary energy consumption in U.S. households. Presently, conventional gas furnaces have maximum heating efficiencies of 98%. Electric air conditioners used in association with the furnace for cooling have a minimum seasonal coefficient of performance (SCOP) of 4.1. A residential gas-fired heat pump (RGHP) was developed and tested under standard rating conditions, resulting in a significant increase in heating efficiency of over 40% versus conventional natural gas furnaces. The associated efficiency of the RGHP in cooling mode is comparable in efficiency to a typical electric air conditioner when compared on a primary energy basis. RGHPs also significantly reduce peak electric use during periods of high demand, especially peak summer loads, as well as peak winter loads in regions with widespread use of electric heating.

The RGHP is similar in nature to a conventional heat pump but with two main differences. First, the primary energy savings are higher, based on a site versus source comparison, as the result of using natural gas to supply shaft power to the compressor rather than an electric motor. Second, waste heat is recovered from the engine to supplement space heating and reduce the energy input. It can also be used to provide supplemental water heating. The system utilizes a programmable logic controller that allows variable-speed operation to achieve improved control to meet building loads.

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## 1. Introduction

Residential buildings account for 6,151 TWh or 22% of the total primary energy consumption in the U.S. Of that amount, Heating Ventilation and Air Conditioning (HVAC) systems consume approximately 43% of the total energy used in residential buildings; a total of 2,622 TWh [1]. One of the biggest challenges faced by utilities is how to handle the problem of increasing peak electricity demand. In the U.S., electricity use is increasing faster than natural gas in residential applications. Two of the most important contributors to peak electricity demand are electric heating (winter peak) and cooling (summer peak). In residential buildings, space heating is the

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dominant component of energy consumption, accounting for 28%, followed by space cooling at 15% [1]. Natural gas-fired furnaces and boilers are the most common heating systems. The residential gas heat pump (RGHP), with a cooling Coefficient of Performance (COP) of 1.22, a COP of 1.41, and over 30% primary energy usage reduction, results in a significant increase in efficiency compared to present HVAC systems. In addition, the electric power consumption which is only 1.25 kW, is significantly less than a comparable 4-ton unit and would reduce the peak electricity demand in summer.

Heat pumps based on the vapor compression cycle are widely used to provide heating and cooling to residential homes. One drawback of conventional air-source electric vapor compression heat pumps is that the heating capacity significantly decreases as the ambient temperature decreases. One well-known solution to the capacity problem has been the addition of auxiliary electric heat strips. Unfortunately, the heat strips increase power usage and contribute to peak electricity demand and energy costs. The RGHP uses an internal combustion engine to replace the electric motor to drive the compressor in a conventional vapor compression cycle. In the RGHP, waste heat from the engine is used to maintain capacity, thus eliminating the need for heat strips. In addition, the RGHP uses natural gas, which is less expensive compared to other fuels. Depending on the cost of electricity, the operating cost of the RGHP is usually less than heat pumps driven by a conventional electric motor.

One advantage of using an internal combustion engine in lieu of an electric motor to drive the compressor in a heat pump system is the ability to use the excess heat of combustion generated by the engine. For conventional electric heat pumps, fuel is converted to electrical energy at the power plant and waste heat is discharged to the environment; then electrical energy is converted to mechanical energy by the motor. In this process, energy is converted twice. However, overall efficiency can be higher if the fuel conversion is located at the site where heat is required. Figure 1 shows the theoretical differences in useful work from locating fuel conversion at the site. The excess heat is available for wintertime heat augmentation, thereby reducing or eliminating the need for auxiliary heaters. It has been a common practice with combustion engine heat pump systems to recover the excess heat from the engine by conveying a working fluid (e.g., water and ethylene glycol antifreeze) through the coolant and sometimes the exhaust system such that waste heat from the engine is absorbed by the working fluid. The heated working fluid is then pumped to a heat exchanger or radiator located in the air flow leading to the air-conditioned space. In the RGHP, the heat is added directly to the refrigerant via a heat recovery heat exchanger. The excess heat of combustion generated by the engine could also be used for domestic water heating, thus further reducing energy costs and peak electricity demand for electric water heaters.

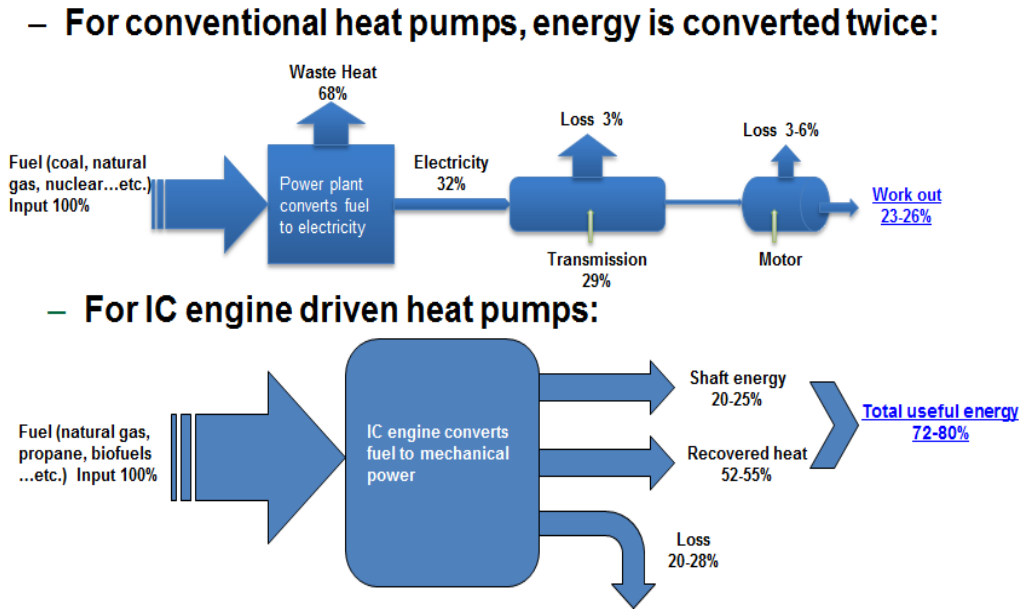



Fig. 1. Fuel conversion efficiency at site versus at the power plant

RGHPs are based on well-established engine and compressor technologies and offer significant efficiency, comfort and cost advantage over conventional electric heat pumps. With conventional electric heat pumps, fuel is converted to electrical energy at power plants and the waste heat is discharged to the environment; then electrical energy is transmitted to the electric heat where it is converted to mechanical energy by the motor. In this process, energy is converted twice. However, energy efficiency can become higher if fuel conversion can be located closer

to where heat released during the energy conversion process can be more efficiently used.

**2. Residential Gas Heat Pump System Design**

The RGHP system, Figure 2, is comprised of the following main components: natural gas engine, variable-speed compressor, coolant pump, thermostat expansion valve, oil separator, outdoor heat exchanger, indoor heat exchanger, radiator, coolant exhaust heat exchanger, and coolant to refrigerant heat exchanger. The performance goals for the RGHP were as follows: Heating (1.4 COP, 20.5 kW capacity) and Cooling (1.3 COP, 14.1 kW (4-ton) capacity).

The RGHP design involves a split system, air-source 4.0-ton heat pump with R410-A as the refrigerant. The unit uses a water-cooled, 4-stroke, single-cylinder, 270 cc, 5.6 kW (7.5 HP) engine. The engine is designed to run for 4,000 continuous hours before maintenance. The engine also has a 40,000-hour life expectancy before a major overhaul. The compressor is a scroll type with volumetric capacity of 60.5 cc/rev. The variable-speed compressor is belt-driven by the engine. The outdoor fan motor is 1/3 horsepower. A two-stage air handler is specified for the indoor unit.

The system is controlled by a programmable logic controller (PLC) that controls the compressor and fan speeds and allows the HVAC system and building demand to be monitored. From predetermined settings stored in the PLC's memory, the PLC also determines how waste heat is allocated based on the space heating requirement. If no space heating is required, the waste heat is directed to the radiator. It can also be used to supplement domestic water heating.



Fig. 2. RGHP unit

The engine coolant circuit, Fig. 3, is equipped with two diverting valves (DV1, DV2.) The first one is at the outlet of the coolant from the engine block. When the engine is cold, the valve diverts all coolant flow to a bypass path, where no heat is extracted from the coolant. This helps increase the engine temperature to its desired operating temperature. When the engine temperature reaches the desired operating temperature, the valve diverts the flow to the second diverting valve. When the coolant temperature is below its set point, the second diverting valve diverts all coolant flow to a refrigerant-to-coolant heat exchanger. When the coolant temperature exceeds the set point of the valve, it diverts all coolant flow to a domestic water-to-coolant heat exchanger, if it exists, then to the radiator. The radiator fan is activated by a temperature switch that senses the coolant temperature. During cooling operation,

the hot refrigerant from the compressor flows through the refrigerant-to-coolant heat exchanger. Since it is hot, it does not remove enough heat from the coolant. The second diverting valve directs all flow to the domestic hot water heat exchanger, if it exists, then to the radiator. During heating operation, the cold refrigerant from the outdoor coil flows through the refrigerant-to-coolant heat exchanger. It gains heat from the coolant. Thus, it boosts the heating capacity of the GHP.

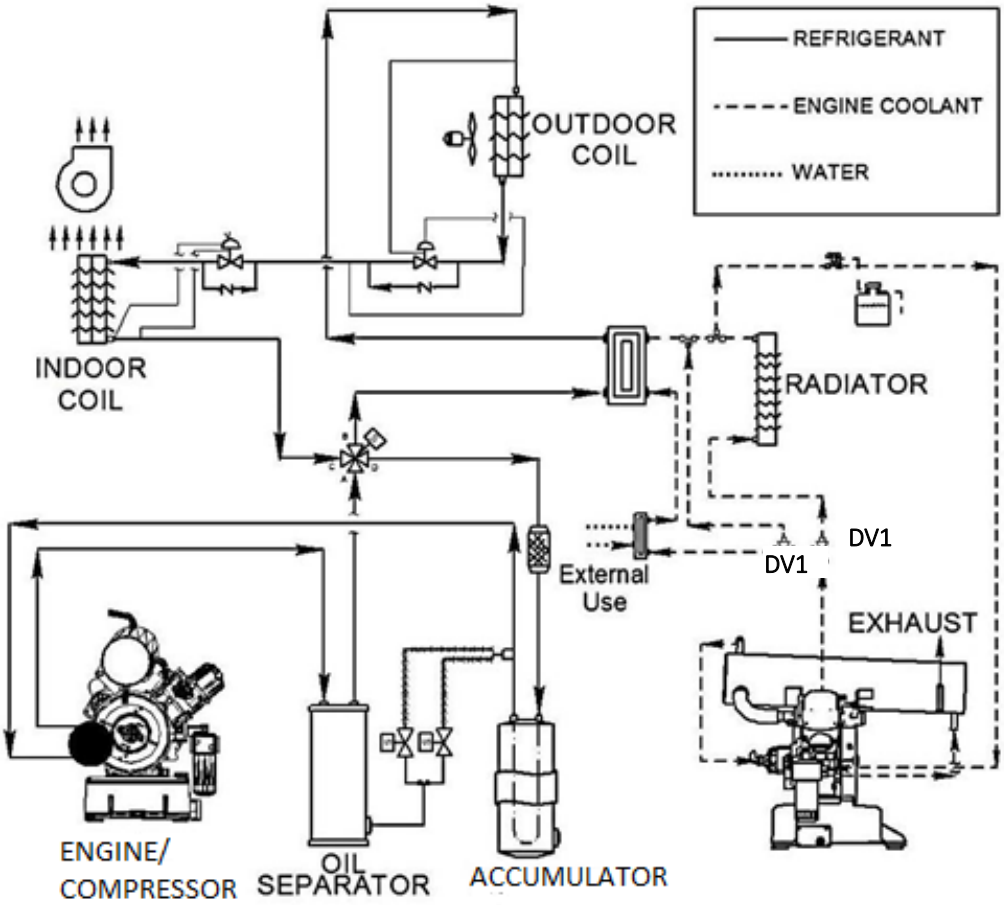



Fig. 3 RGHP cooling cycle schematic

### 3. Experimental Evaluation

The RGHP was installed in an environmental chamber and tested over a wide range of ambient conditions including the operating conditions for standard rating and performance tests [2] [3] [4]. The chamber consists of two rooms; outdoor and indoor. The outdoor (condensing) unit of the RGHP was installed in the outdoor room while the air handler was installed in the indoor room. The unit was then operated over a wide range of ambient conditions and all data points needed to calculate capacity and COP were measured and recorded. The evaluations were conducted at various engine speeds to evaluate performance (capacity and COP) to determine the optimal speed at which the unit achieved the target capacity without a significant drop in COP.

To reduce the noise on the measurement, the natural gas consumption was measured by a diaphragm type gas meter with pulse output. The refrigerant flow rate was measured by a Coriolis mass flow meter. Coolant flow was also measured with a Coriolis mass flow meter. Humidity transmitters were used to measure relative humidity. Averaging thermistors were used for the dry bulb measurements while a sampling array was used to obtain an average dew point temperature of the supply and return air. Supply air flow rates were measured using

a multi-point, self-averaging Pitot traverse station with an integral air straightener/equalizer honeycomb cell. This arrangement allows the capability of continuously measuring fan discharges or ducted airflow.

The air handler was operated on high-stage when in heating mode. In cooling mode, the air handler was operated on low-stage when running the engine at low to intermediate speeds (1200−2400 rpm) and on high-stage when running the engine at high speeds (above 2400 rpm.)

Table 1. Operating test conditions

| Test | INDOOR UNIT | | | OUTDOOR UNIT | | |
|---|---|---|---|---|---|---|
| | Air Entering | | | Air Entering | | |
| | DB °F (°C) | DP °F (°C) | WB °F (°C) | DB °F (°C) | DP °F (°C) | WB °F (°C) |
| **COOLING TESTS** | | | | | | |
| Standard Rating Conditions - "A" Cooling Steady State[a] | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 95 (35) | 66.5 (19.2) | 75 (23.9) [b] |
| "B" Cooling Steady State[a] | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 82 (27.8) | 55 (12.8) | 65 (18.3) [b] |
| "C" Cooling Steady State - Dry Coil[a] | 80 (26.7) | 36.8 (2.7) | 57 (13.9) [c] | 82 (27.8) | 55 (12.8) | 65 (18.3) [b] |
| Low Temperature Operation Cooling[a] | 67 (19.4) | 49.8 (9.9) | 57 (13.9) | 67 (19.4) | 49.8 (9.9) | 57 (13.9) [b] |
| Maximum Operating Cooling Conditions[a] | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 115 (46.1) | 55 (12.8) | 75 (23.9) [b] |
| High Ambient Temperature | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 110 (43.3) | 58.2 (14.6) | 75 (23.9) [b] |
| Higher Ambient Temperature | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 120 (48.9) | 51.3 (10.7) | 75 (23.9) [b] |
| Highest Ambient Temperature | 80 (26.7) | 60.2 (15.7) | 67 (19.4) | 125 (51.7) | 47.1 (8.4) | 75 (23.9) [b] |
| **HEATING TESTS** | | | | | | |
| Standard Rating Conditions - High Temperature Heating Steady State[a] | 70 (21.1) | 53.5 (11.9) | 60 (15.6) (max) | 47 (8.3) | 38.7 (3.7) | 43 (6.1) |
| High Temperature Heating Cyclic[a] | 70 (21.1) | 53.5 (11.9) | 60 (15.6) (max) | 47 (8.3) | 38.7 (3.7) | 43 (6.1) |
| High Temperature Heating Steady State[a] | 70 (21.1) | 53.5 (11.9) | 60 (15.6) (max) | 62 (16.7) | 52.7 (11.5) | 56.5 (13.6) |
| Low Temperature Heating Steady State[a] | 70 (21.1) | 53.5 (11.9) | 60 (15.6) (max) | 17 (-8.3) | 9.4 (-12.6) | 15 (-9.4) |
| Maximum Operating Conditions[a] | 80 (26.7) | | | 75 (23.9) | 59.5 (15.3) | 65 (18.3) |

[a] Operating Conditions for Standard Rating and Performance Tests
[b] Wet bulb temperature (WB) condition is not required
[c] Wet bulb sufficiently low that no condensate forms on evaporator
Note: DB is the dry-bulb temperature and DP is the dew-point temperature

Thermocouples, tachometer, hygrometers and pressure transducers along with the flow rate measuring devices were used to monitor the RGHP via a PC-based data acquisition system. Sensors for these measurements and their accuracies are shown in Table 2. The required accuracy of the test instrumentation is in accordance with ASHRAE and/or ASME documents [5,6].

Table 2. Instrumentation accuracy

| Measurement | Sensor | Range | Accuracy |
|---|---|---|---|
| Temperature | Thermistor | -55−150°C | ±0.2°C<br>0 to 70°C |
| Pressure | Transducer | 0−3,447 kPa | ±1% of full scale |
| Air flow | Pitot tube array | 0−125 $m^3$/min | ±2% |

| | | | |
|---|---|---|---|
| Coolant flow | Coriolis mass flow sensor | 0−3,402 kg/h | ±0.1% |
| Natural Gas Flow | Coriolis mass flow sensor | 0− 9 kg/h | ±0.1% |
| Refrigerant flow | Coriolis mass flow sensor | 0−907 kg/h | ±0.1% |
| Dew-Point Temperature | Chilled Mirror | -40− 60°C | ±0.2°F |
| Rotational speed | Portable tachometer | 0− 5000 rpm | ±0.1% |
| Electric power | Watt transducer | 0− 5 kW | ±0.5% of full scale |

## 4. Results

The overall performance including capacity and COP for the RGHP was evaluated over a range of operating speeds and at several cooling and heating conditions. For this evaluation, capacity was calculated by multiplying the mass flow rate of the refrigerant times the enthalpy difference across the heat exchanger (evaporator or condenser). The COP is Gas COP and calculated by dividing the capacity by the fuel input. Following are the results from all the tests.

### *4.1 Cooling tests*

For the 35ºC standard rating condition, several tests were run to assess the performance over a range of speeds to determine the optimal speed (Table 3). In a production RGHP, the optimal speed would be used as part of the control strategy to control the unit for peak performance. From Table 3, the unit achieves better COPs at lower speeds. However, the capacity is too low. The goal for the RGHP was to achieve 14.1 kW (4 tons) with a COP of 1.3. Interpolating the results at the standard rating condition of 35ºC, a speed of 2516 rpm was selected to achieve a capacity of 14.1 kW at a COP of 1.22, which is slightly below the goal of 1.3.

Since this is a variable speed unit, the speed could be increased if more capacity was required as the building loads change over time due to increased internal loads (miscellaneous plug loads, lighting, appliances) or degradation in the insulation. Although the loss in COP would be up to 14%, the capacity could be increased 20%.

The RGHP was also tested at higher ambient temperatures to determine the degradation in performance, especially at extreme conditions (greater than 48.9°C). For these tests, the engine speed was maintained at approximately 3200 rpm to achieve approximately 14.1 kW (4 tons) at higher ambient temperatures. However, the engine speed was reduced above a temperature of 46.1°C by the PLC as a protective feature to prevent the unit from shutting off due to high pressure.

As mentioned earlier, the waste heat during cooling operation could be used for auxiliary water heating instead of being exhausted to the atmosphere through the radiator. From Table 3, the waste heat recovery efficiency was higher than 50% across all the tests. Using the previously calculated engine speed of 2516 rpm and for the 35ºC standard rating condition, 6 kW of heat would be added to the water, which raises the overall efficiency of the RGHP to 1.74. At higher ambient temperatures, the input to water heating increases up to an ambient temperature of 48.9ºC; where it begins to tail off due to the engine speed reduction. The average daily input required for a residential water heater is approximately 11.7 kWh, so the unit would only have to run approximately 2 hours to satisfy the water heating load at the 35ºF rating condition.

Table 3. Cooling performance with and without auxiliary water heating

| Engine Speed, rpm | Outdoor Air Temperature, °C | Cooling Capacity, kW | Water Heating Capacity, kW | Fuel Consumption, kW | Gas COP | Gas COP with Water Heating | Waste Heat Recovered, % |
|---|---|---|---|---|---|---|---|
| 1273 | 35.0 | 7.38 | 2.67 | 4.92 | 1.50 | 2.04 | 54.2 |
| 1661 | 35.0 | 9.65 | 3.79 | 7.06 | 1.37 | 1.90 | 53.6 |
| 2409 | 35.0 | 13.59 | 5.54 | 10.78 | 1.26 | 1.77 | 51.3 |
| 2818 | 35.0 | 15.40 | 7.16 | 13.77 | 1.12 | 1.64 | 52.0 |
| 3210 | 35.0 | 16.86 | 8.52 | 16.17 | 1.04 | 1.57 | 52.7 |
| 2248 | 40.6 | 11.48 | 5.01 | 10.50 | 1.09 | 1.57 | 47.7 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 3179 | 40.6 | 15.23 | 8.54 | 16.73 | 0.91 | 1.42 | 51.0 |
| 3188 | 46.1 | 13.91 | 9.96 | 18.24 | 0.76 | 1.31 | 54.6 |
| 3047 | 48.9 | 12.64 | 9.88 | 18.24 | 0.69 | 1.23 | 54.1 |
| 2652 | 51.7 | 10.68 | 8.56 | 16.06 | 0.67 | 1.20 | 53.3 |
| 2244 | 54.4 | 8.54 | 6.92 | 13.31 | 0.64 | 1.16 | 52.0 |

Figure 4 is a graphical representation of the results to show the impact that higher ambient temperatures have on the RGHP performance, especially the capacity.

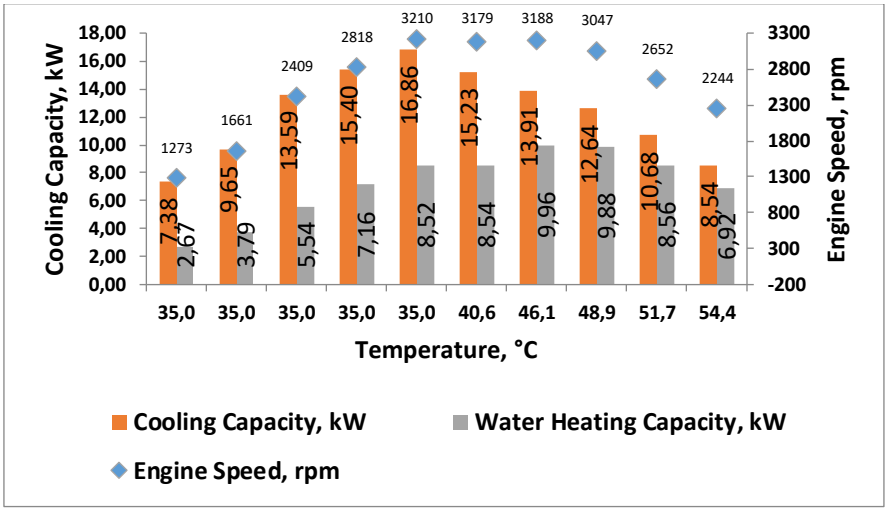



Fig. 4. Cooling performance with and without auxiliary water heating

Figure 5 represents the fraction of fuel input converted to useful energy over the range of outdoor ambient cooling temperatures. Three values are given: 1) shaft energy; 2) recovered waste heat for water heating; and 3) total energy. From Fig. 5, energy input to heating water stays fairly constant across the range of temperatures. However, shaft power, which is the energy to drive the compressor, drops off significantly as outdoor temperature increases. This is mainly due to the fact that as outdoor temperature increases; the air becomes less dense which reduces the efficiency since it affects the air/fuel ratio.

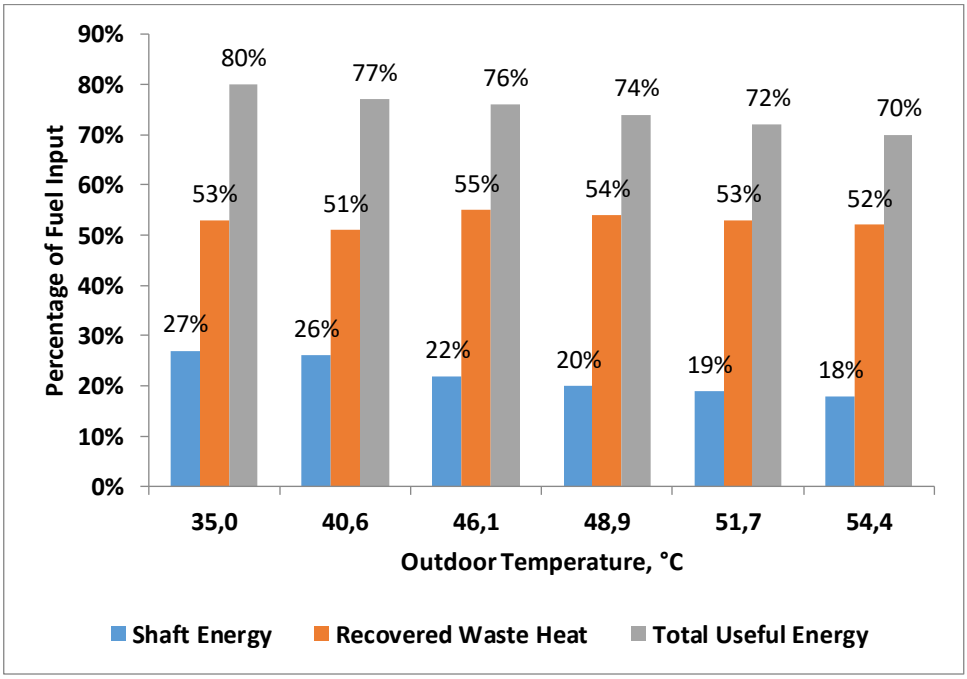


Fig. 5. Fraction of fuel input converted to useful energy at multiple loads

### *4.2 Heating tests*

Numerous tests were run at standard outdoor temperatures of -8.3°C and 8.3°C, along with an intermediate temperature of 1.7°C to assess the performance over a range of speeds and determine the optimal speeds for operation (Fig. 6). The RGHP was run at a speed of approximately 3400 rpm to achieve the maximum capacity at the lower ambient temperatures (-8.3°C and 1.7°C). At 8.3°C, a range of speeds from 2463−3248 rpm was evaluated to determine the speed at which the goals for capacity (20.5 kW) and COP (1.4) could be achieved. By interpolating the results at the standard rating condition of 8.3°C, it was determined that both goals could be reached at an engine speed of 2976 rpm, yielding 20.5 kW with a COP of 1.41.

Figure 6 also shows that the waste heat recovered from the engine is a significant contributor to the overall capacity, especially at the lower ambient temperatures. At -8.3°C, the waste heat contributed 39% of the overall capacity, while at 1.7°C, waste heat accounted for 31%. At all the 8.3°C test points, waste heat was in the 28−29% range.

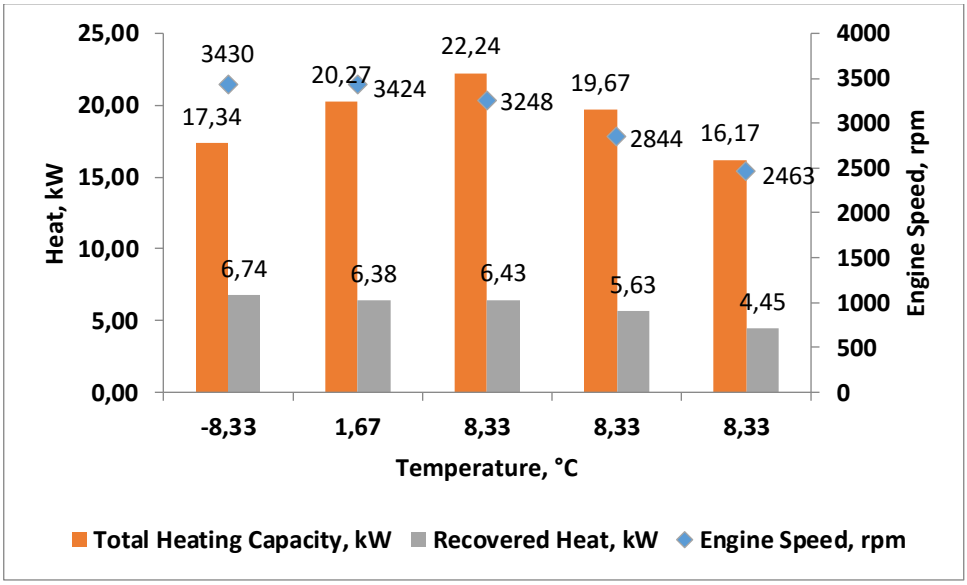



Fig. 6. Heating performance as a function of ambient temperature

Figure 7 shows the variation in Gas COP with outdoor temperature. At the 8.3°C rating point, the COP is 1.46, which is approximately 50% higher than that of a natural gas furnace. At -8.3°C, the COP is still higher than a natural gas furnace, which has a maximum value of 98% for the highest efficiency furnaces. Figure 7 reinforces the contribution of the waste heat to COP. From Fig. 7, waste heat recovery consistently adds approximately 37 points to the overall efficiency.

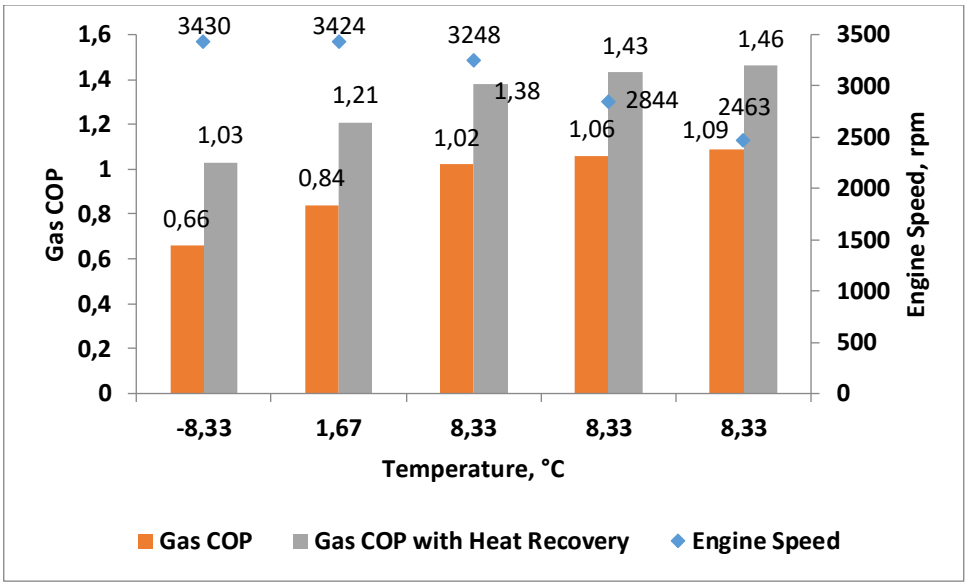



Fig. 7. Heating COP as a function of ambient temperature

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