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DESIGN AND DEVELOPMENT OF A GAS-ENGINE-DRIVEN HEAT PUMP

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ABSTRACT

Improved air-conditioning technology has the greatest potential impact on the electric industry compared to any other technology that uses electricity particularly during summer peak electric demand. Gas engine-driven units can provide overall peak load reduction and electric grid relief for summer peak demand. Peak-load conditions can lead to high electricity prices, power quality problems, and grid system inefficiencies, and even failures. Improved air-conditioning technology thus has the greatest potential impact on the electric grid compared to other technologies that use electricity. Thermally-activated systems, such as natural gas engine-driven heat pumps, can provide overall peak load reduction and electric grid relief for summer peak demand.

This paper describes the development of an innovative 10 refrigeration ton (RT) natural gas engine-driven heat pump (GHP) for commercial application. The unit was tested at various Air Conditioning and Refrigeration Institute (ARI) heating and cooling conditions in a psychrometric chamber at Oak Ridge National Laboratory. The gas COP at 47ºF rating condition exceeded the goal of 1.6 at both high and intermediate engine speeds. The gas COP in cooling mode also exceeded the goal of 1.2 at 95ºF rating condition. In this study, principles of operation, unit performance and benefits are discussed.

INTRODUCTION

A heat pump utilizing a gas engine is called a gas-engine heat pump (GHP). The GHP can use natural gas as fuel compared to electrical power used by a conventional electric heat pump (EHP).

GHPs are based on well-established engine and compressor technologies and offer significant efficiency, comfort and cost advantage over conventional EHP products. One of the main disadvantages of EHP is that during winter operation, the heating capacity decreases with the ambient temperature. At the same time, building heating loads increase, and under some conditions supplemental heat is required to keep the temperature of the conditioned space at a comfortable level. Typically, auxiliary electric (resistance) heat strips are added which significantly increase the electric power usage and cost during cold winter operations.

In Asia, engine-driven heat pumps have been used to overcome this problem. These heat pumps replace the conventional electric motor with a natural gas or propane-driven engine as the driving source for the compressor [1, 2].

There are several advantages of using an internal combustion (IC) engine in lieu of an electric motor for a heat pump system. These are:

1. The excess heat of combustion generated is available for wintertime heating augmentation, thus reducing or eliminating the need for auxiliary heaters. This energy recovery significantly reduces running costs, while providing stable comfort conditions.
2. Significant electrical power (demand and energy) and cost savings resulting from powering the compressor(s) with an IC engine fueled by natural gas.

While the advantages of using waste heat from a combustion engine are well recognized, the wide range of options for recovery and use of waste heat has required numerous separate components for heat exchange, auxiliary heating, defrosting, and heat rejection to the ambient. The complexity, size and cost of these heat pump systems with effective heat recovery have increased accordingly. Unlike the current GHPs from Asia, this unit is designed for commercial rooftop application for desert Southwest U.S.A. Therefore, the unit has to operate at full capacity at ambient temperature of 110 ^{o}F and designed to adapt with widely used roof curb.

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Additionally, the use of a small IC engine at high ambient temperatures and at increased altitudes is challenging, due to the reduction in output horsepower. Recognizing these challenges, a series of prototype designs were undertaken over the past several years leading toward a final product ready for commercialization.

OPERATING PRINCIPLES

The GHP system is comprised of three major component sections, an engine-compressor section, an indoor section, and an outdoor section all contained within a single package. General system description of the unit is shown in Table 1.

Table 1 General system description

Engine	Water-cooled, 4 cycle, 3 cylinder, 9.5kW rated output
Engine speed	1200 to 2450 rpm
Fuel type	Natural gas or propane
Compressor	Scroll type, 60.5 cc/rev.
Compressor	2280 to 4655 rpm
Refrigerant type	R407C
Design Cooling rating	120,000 Btu/hr (10 ton)
Design Heating rating	43kW (147,000 Btu/hr)
Design cooling gas COP	1.2 at 95°F (35ºC) outdoor temperature
Design heating gas COP	1.6 at 47°F (8.3ºC) outdoor temperature
Electrical power requirement	3 kW
Unit dimension (in)	124 L X 49 H X 65 W
Unit weight	1900 lb (862 kg)
Maintenance interval	10,000 hours

The engine-compressor section shown in Figure 1 is comprised of an IC engine with engine exhaust, waste heat recovery components and multiple belt driven scroll type refrigeration compressors. Engine coolant is pumped through waste heat recovery components and the engine to remove and recover waste heat.

Figure 1 Engine/compressor and electrical compartment of GHP

The indoor section is comprised of an indoor heat exchanger containing two interlaced refrigerant circuits, an auxiliary heat circuit, and an air blower driven by a high efficiency multi-speed motor[1].

The outdoor section is comprised of dual outdoor heat exchangers, one for each refrigerant circuit, multiple high efficiency fans driven by high efficiency multi-speed motors, the engine coolant radiator, and various valves and controls.

As shown in Figure 2, the refrigeration system is comprised of two complete heat pump circuits driven by a single natural gas fueled IC engine. The system is designed and controlled such that the system efficiency is maximized by varying the number of compressors running as well as varying the refrigerant flow rate through the circuits to satisfy the heating and cooling load requirement of the conditioned space. The refrigerant flow rate is adjusted by varying the speed of the natural gas fueled IC engine. Part load efficiency is maximized by the interlacing of the refrigerant circuits within the indoor heat exchanger as well as by varying the airflow across the indoor and outdoor heat exchangers to match the ambient conditions and load requirements.

When the GHP is operated in the cooling mode, waste heat is removed from the engine and exhaust by the coolant and is either directed to auxiliary devices for domestic hot water heating, swimming pool heating, or other energy saving devices where the waste heat is utilized. The waste heat can also be directed to the radiator and rejected to the atmosphere.

When the GHP is operated in the heating mode, waste heat is removed from the engine and exhaust by the coolant and is directed to the indoor heat exchanger. All hot engine coolant is directed to the auxiliary heat circuit in the indoor heat exchanger, thus transferring all recoverable waste heat from the engine into the heated space and maximizing the COP of the system. This also allows increasing the supply temperatures above those achieved by EHP at the same outdoor conditions. If freezing or frosting conditions are sensed on the outdoor heat exchanger, one of the circuits reverses the cycle to cooling and defrosts the coil while the other circuit provides heating. The second circuit will reverse to cooling as soon as the first circuit is out of defrosting mode. The cycle is repeated until the frost / freezing condition is abated. The goal of this methodology is to melt the frost build up that occurred during low ambient run while providing warm supply air temperature to the space.

[1] Also, contained within the section are the thermostatic expansion (TXV) devices, check and control valves, and miscellaneous refrigeration and electrical components.

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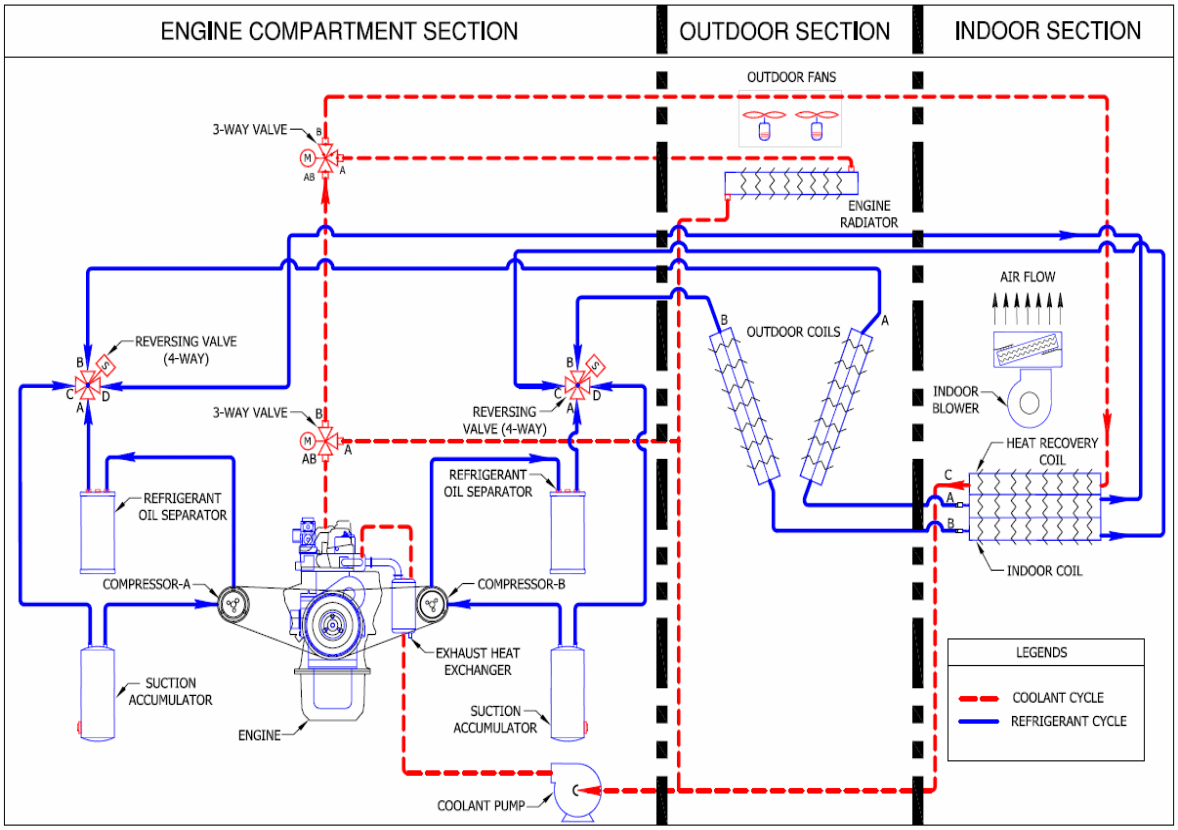

Figure 2 Schematic of cooling mode operation of GHP

During both heating and cooling operation, engine coolant is circulated throughout the system by coolant pump. Warm coolant is pumped through the exhaust heat exchanger (shell and tube heat exchanger), where its temperature is raised a few degrees by waste heat recovered from the engine exhaust. The coolant then flows to the water cooled exhaust manifold located on the internal combustion engine, where its temperature is further increased. The coolant then enters the internal combustion engine and removes heat from the engine. This portion of the coolant circuit is where engine waste heat is recovered for efficient use during the heating cycle.

DESIGN CONSIDERATIONS

The design focused on system (IC engine, refrigeration cycle, and controls) durability with an extended maintenance cycle of 10,000 hours or better. Controls had to encompass both IC engine and refrigeration cycle and assure proper system integration (Yokoyama 1992).

The system is designed to output 10 RT at an ambient temperature of 110 ^{o}F and 80% of the total capacity at 120 ^{o}F. The challenge is to maintain refrigerant discharge pressures, engine coolant, and oil temperatures within safe operating conditions as the ambient temperature increases. In order to maximize capacity and maintain engine design operation under high ambient conditions, a high pressure avoidance (HPA) system was implemented. As the refrigerant discharge pressure approaches the maximum allowable operating pressure in one of refrigerant circuits, the system disables the high engine speed signal (engine lowers its speed, thus lowering the refrigerant pressure). The unit will operate in this condition until the refrigerant pressure drops to the desired value, at which time the HPA will enable the high speed signal. The HPA allows the engine to operate at desired design parameters and at higher efficiency for required horsepower. It also protects the compressors and engine internal parts from wear due to higher operating pressures. There are high and low pressure switches in both refrigerant circuits.

Maintaining engine coolant temperature is crucial for the longevity and maintenance interval of the engine. The engine coolant system is designed to maintain an engine temperature of 165°F. This is accomplished with the use of a coolant mixing valve and a coolant system proportional-integral-derivative (PID) controller. The hot engine coolant leaving the engine flows to the engine temperature control valve where a PID control loop in the control system maintains the engine temperature at an efficient temperature by controlling the coolant flow either back to the engine (bypassing the radiator) or to the radiator. Upon initial startup, in order to bring the engine up to the proper operating temperature, all coolant is diverted back to the engine. As the engine temperature approaches the desired operating temperature, the engine temperature control valve starts diverting the coolant to the

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coolant switching valve. During the cooling mode, the switching valve diverts all engine coolant to the radiator where excess engine heat is removed. The coolant temperature is monitored by the coolant system controller (CSC) which receives its temperature signal from a K-type thermocouple installed in the outlet side of the engine head. The CSC sends a 0-10 VDC analog signal to the coolant mixing valve which changes the flow of the coolant from engine-only to full radiator/indoor coil or some setting in between to maintain the 165°F set point. A second valve is installed in the system that is controlled by the heating signal and diverts the coolant between the radiator in cooling mode and the indoor coil during heating mode. The CSC also controls the engine shutdown safety when engine coolant temperatures exceed 180°F. Longer engine life and an extended maintenance cycle of approximately 10,000 hours are accomplished through the addition of an auxiliary oil storage tank and running the engine at coolant temperatures as close to optimal as possible.

Because most engine controls were designed for the automotive industry, care had to be taken with the electric power supply. The power supplied has to be rectified and filtered for use with the engine control systems. The design team abandoned the use of a battery for two reasons: short battery life and the requirement to have a charging circuit. The engine management system (EMS) controls the start sequence of the engine and manages the fuel/air mixer by comparing actual rpm to the desired rpm based on the call for cooling or heating stage. Through a series of relay inputs, the EMS receives its signals from the refrigerant controls and a magnetic pickup on the engine. The fuel/air mixer receives the desired position signal from the EMS and through a Pulse Width Modulation (PWM) signal controls that position. Once a start signal has been received, the EMS will energize the starter relay. When the engine starter relay has de-energized and the engine run relay is energized, the compressor clutches engage. The EMS then maintains the desired engine rpm. The engine ignition system is controlled by the ignition control module (ICM). The ICM receives a signal from the magnetic pickup located on the cam shaft.

As with all heat pump systems, careful consideration had to be taken related to the defrosting of the outdoor coil (evaporator) during heating mode operations. Due to the two independent refrigerant circuits, the defrost cycle is staged by reversing one circuit but not the other. After a set time of 30, 60, or 90 minutes, if the switch remains energized, it will start the defrost sequence. Upon entry into defrost, the engine will lower its speed to 1400 rpm, the appropriate reversing valve will reverse, and the coil will defrost for 10 minutes or until it reaches 51ºF. Once the first coil has completed it's defrost cycle, the second coil will enter into the defrost cycle if necessary. After both coils have defrosted, the engine will return to the operating speed prior to the beginning of the defrost sequence.

To prevent engine vibration from transmitting to the roof, the mounts are isolated from the engine through rubber isolators. The engine isolators are strategically placed between the mounting surface and engine mounting brackets. The current highest vibration level measured at the base of the unit is 0.3 mils pk-pk and 0.04 inch/sec displacement and velocity respectively. This puts the unit's level of vibration at "DESIREABLE" on the vibration severity chart. The other design challenge is the transmission of engine and compressor vibration to refrigeration piping. To minimize the transmission of engine and compressor induced vibration through the system piping, vibration-absorbers are installed in the suction and discharge lines perpendicular to the compressor. Since the vibration is in more than one direction, absorbers are also installed to dampen horizontal movement. The absorbers aid in extending compressor life by relieving stress on the housing and cutting down wear on the bearing.

Proper compressor oil flow is vital in protecting bearings and mechanical seals from failure due to lack of lubrication. The capillary tubes are sized for proper oil flow based on the refrigerant suction and discharge pressure differential.

Roof curb adaptability is another design item requiring design consideration related to the retrofit market. An extensive study on existing rooftop curb configuration for the Southwest United States (U.S.) region was conducted to determine the percentage of each curb orientation. Based on the outcome of the study, the design team decided to go forward with the popular configuration shown in Figure 3.

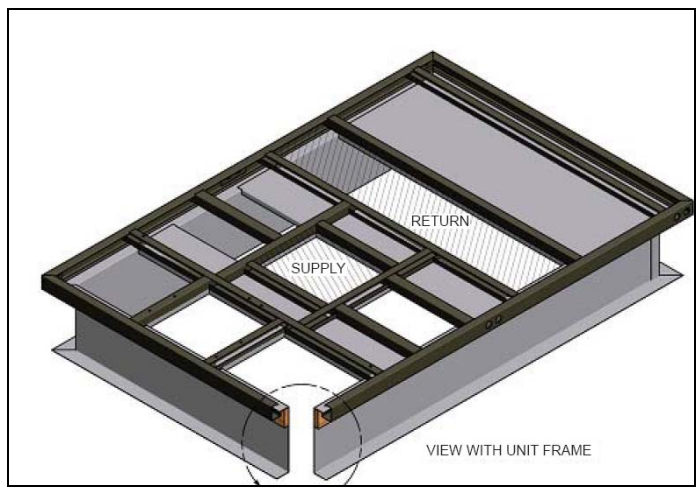

Figure 3 Base frame of 10 RT packaged GHP

Sound level is another design consideration that needed to be addressed. The IC system is much quieter than electric compressors. The operating sound value ranges from 55 to 58 dB (this operating sound value is measured in an anechoic room measured 1m from the unit and 1.5 m above the ground). Therefore, the design team mainly focused on reducing air borne noise from the condenser fan. The sound level is reduced by 2-3dB by having venturi type inlet geometry and selecting high efficiency, fiberglass sickle type fan blades. The current overall sound level of the unit is estimated to fall below

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75dB (the measurement is done in-house using commercially available sound level meter)

Unit dimension was another design constraint in the process of developing the GHP product. As shown in Figure 3 the current overall unit dimensions are comparable to a similar capacity electric heat pump or gas pack.

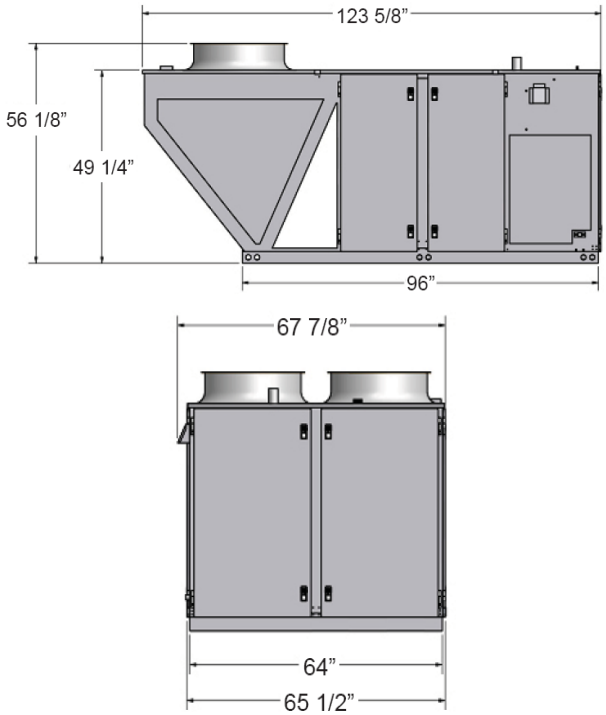

Figure 4 Unit dimension of GHP

The GHPs being tested utilize a small (less than one liter) lean burn internal combustion engine that is fueled by natural gas. Currently these units are exempt from emissions regulations due to their size and horsepower output. The EPA has new regulations that will take effect starting January of 2008 that would apply to the GHP. The engine manufacture has done the prescribed tests and the results indicate that the engine meets the new regulations. The manufacturer will be certifying the engines as meeting the EPA requirements starting in 2008. Future and more stringent EPA requirements are being contemplated by the EPA and other agencies, particularly in California, that would require the GHP engine to be fitted with a type of post combustion device to mitigate emissions. Research is currently underway to develop a solution that would allow the GHP engine to comply with these new regulations. The downsizing and application of well-established automotive post combustion technology is being evaluated along with other potential solutions.

Other design challenges undertaken over the past several years include cost and emissions. The GHP has to be cost competitive in order to be considered as a viable option. The initial cost of the GHP inherently will be higher than an electric heat pump. The goal is to limit the incremental cost difference to less than five years payback period. Preliminary market introduction simple payback period calculations show a 2 to 5 year payback period for Southern Nevada and Arizona based on current natural gas and electric utilities rates (in regions with high demand charges the paybacks are much shorter). As shown in Figure 5, GHP operating savings are highly dependent on the natural gas and electric rates.

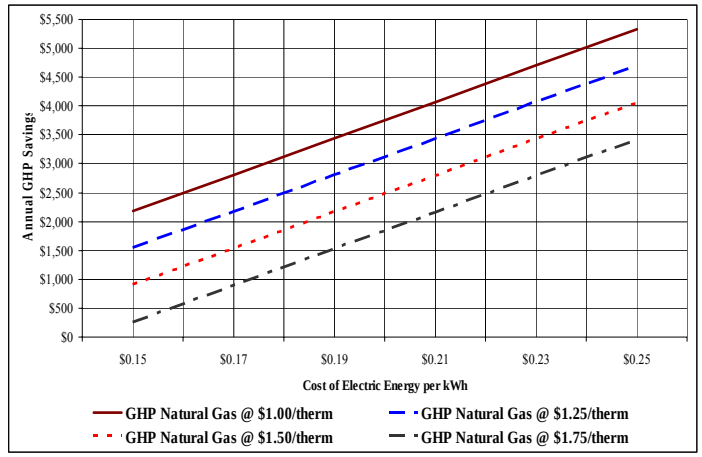

Figure 5 Annual GHP operating savings as functions of natural gas and electric rate for Las Vegas, NV (based on 1773 and 1642 cooling and heating hours)

LABORATORY EVALUATIONS

The GHP unit was installed in the outdoor room of an environmental chamber with supply/return air from the smaller indoor room. The unit was then operated over a wide range of ambient conditions including the operating conditions for standard rating and performance tests [3,4]. Table 2 shows these operating conditions. The evaluations were conducted at high (2250 rpm) and intermediate (1900 rpm) engine speeds to determine the effects on GHP performance when the system is controlled to match building loads.

The natural gas, refrigerant, and coolant flow rates were monitored by Coriolis mass flow sensors. Temperature probes, rotational speed measuring device, and pressure transducers along with the mass flow sensors were used to monitor the GHP via a web-based data acquisition system (DAS). The dry-bulb and dew-point temperatures of the supply and return air were monitored by averaging thermistors and chilled mirrors respectively. 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. Sensors used for these measurements and associated accuracies are shown in Table 3. The required accuracy of the test instrumentation is in accordance with documents [5-10]. The DAS calculated and displayed important parameters such as the flow rates, capacities, and COP.

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More than one hundred cooling and heating tests were conducted on the GHP unit at various controlled ambient conditions in the environmental chamber. These included cooling tests up to 120ºF (48.9ºC) and heating tests down to 17ºF (-8.3ºC). The cooling mode was conducted in the range of 67ºF (19.4ºC) to 120ºF (48.9ºC) at an intermediate speed of 1900 rpm and 67ºF (19.4ºC) to 120ºF (48.9ºC) at a high speed of 2250 rpm. The heating mode was conducted in the range of 17ºF (-8.3ºC) to 62ºF (16.7ºC) at an intermediate speed 1900 rpm and 17ºF (-8.3ºC) to 47ºF (8.3ºC) at a high speed of 2250 rpm.

Table 2 Operating conditions for evaluation of GHP #16

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 [3, 4,7]
[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

Table 3 Major test instrumentation and measurement accuracies

Measurement	Sensor	Range	Accuracy
Temperature	Thermistor	-67 to 302°F (-55 to 150°C)	±0.4°F (±0.2°C) (32 to 158°F or 0 to 70°C)
Pressure	Transducer	0 to 500 psia (0 to 3,447 kPa)	±1% of full scale
Air flow	Pitot tube array	0 to 4,400 cfm(0 to 125 m^3/min)	±2%
Coolant flow	Coriolis mass flow sensor	0 to 7,500 lb/hr (0 to 3,402 kg/hr)	±0.1%
Natural Gas Flow	Coriolis mass flow sensor	0 to 20 lb/hr (0 to 9 kg/h)	±0.1%
Refrigerant flow	Coriolis mass flow sensor	0 to 2,000 lb/hr (0 to 907 kg/h)	±0.1%
Dew-Point Temperature	Chilled Mirror	-40 to 140ºF (-40 to 60°C)	±0.2°F
Rotational speed	Portable tachometer	0 to 5000 rpm	±0.1%
Electric power	Watt transducer	0 to 5 kW	±0.5% of full scale

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GHP Performance

Evaluation of the gas engine-driven heat pump unit in heating mode was completed over a wide range of conditions (engine speeds, outdoor and indoor temperatures, and humidity). Figure 6 and 7 show the performance at high and intermediate engine speeds. Gas heating COP of 1.6 with capacity of 160,000 Btu/h (46.9 kW) at ARI steady-state rating condition of 47ºF outdoor was achieved at high engine speed.

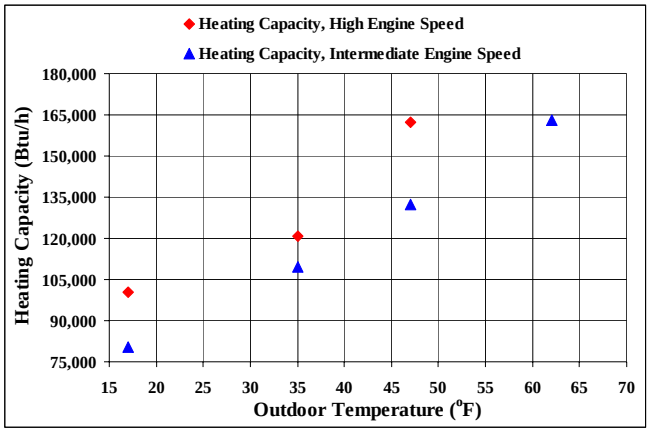

Figure 6 Heating capacity based on ARI indoor and outdoor inlet air temperatures

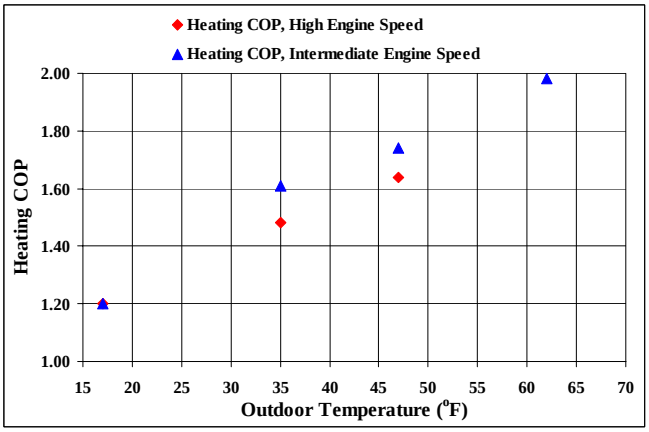

Figure 7 Heating coefficient of performance based on ARI indoor and outdoor inlet air temperatures

Evaluation of the gas engine-driven heat pump unit in cooling mode was completed over a wide range of conditions (engine speeds, outdoor and indoor temperatures, and humidity). Figure 8 and 9 shows the cooling performance at high and intermediate engine speeds. As shown in Figures 8 and 9, gas cooling Coefficient of Performance (COP) of 1.2 at ARI steady-state rating condition of 95ºF outdoor (80ºF dry-bulb/60.2ºF dew-point temperatures for indoor) with capacity of 120,000 Btu/h (10.0 RT) was achieved at high engine speed.

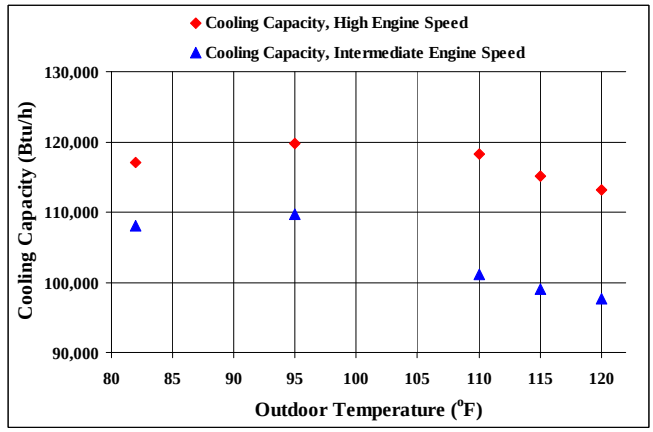

Figure 8 Cooling capacity based on ARI indoor and outdoor inlet air temperatures

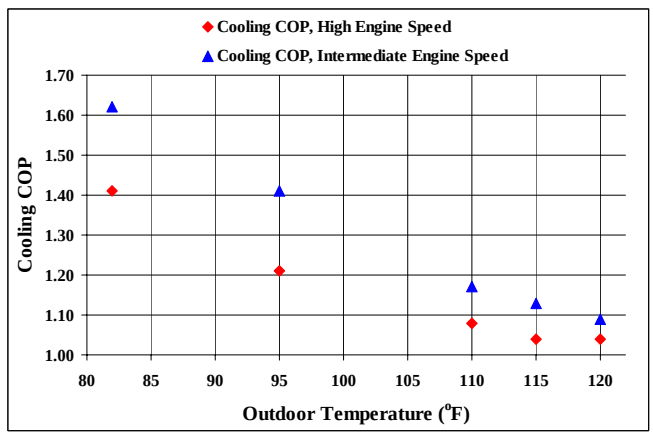

Figure 9 Cooling coefficient of performance based on ARI indoor and outdoor inlet air temperatures

The 2006 Energy Information Adiministration [11] energy review diagram of energy flow from the primary energy source to the electricity delivered to the consumer at the end use level showed 32 % efficiency including power generation, transmission and distribution [11]. A breakdown of the EIA's diagram shows that the "Energy Consumed to Generate Electricity" in 2006 was 41,27 Quadrillion BTU, the "Net Generation of Electricity" was 13.83 Quadrillion BTU and the "End Use" after 9 % T&D losses was 13.03 Quadrillion BTU; yielding the end use level efficiency of 13.03/41.27 = 0.3157. Therefore, 1 unit of primary energy is converted into 0.32 unit of electricity ready to operate electric cooling device.

The minimum cooling energy efficiency ratio (EER) required of a unitary heat pump in the range of 5.42 to 11.25 ton capacity is 11.6 EER for the Southwest region of the United States. In terms of cooling coefficient of performance (COP) with a conversion factor of 3.413 between the COP and the energy efficiency ratio (EER),

$$COP = 11.6/3.413 = 3.4$$

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Or

$$\eta_{cooling} = 0.32 \times 3.4 = 1.09 \qquad \text{(electric heat pump)}$$

The following shows laboratory testing of cooling performance of a GHP at high speed for comparing resource COP:

- Electrical Parasitic of GHP ≈ 1,500 W (excluding the indoor blower)
- Gas COP of GHP at 95ºF (35ºC) outdoor cooling condition ≈ 1.22
- Capacity at 95ºF (35ºC) outdoor cooling condition ≈ 118,322 Btu/h (34.7 kW)
- Primary Energy use for 1,500 W of electricity ≈ 1,500/0.32*3.412 = 15,994 Btu/h (4.7 kW)
- Resource COP for GHP at 95ºF (35ºC) outdoor cooling condition = 1.05

The comparison to an electric chiller indicates a 3.7% decrease in resource COP values in cooling mode for the GHP. However, the main advantage of a GHP over a conventional electric driven heat pump is the ability to reduce the peak demand.

Ninety five percent of all buildings use electricity for cooling. As a result, many electric utilities experience peak summer loads that strain their ability to meet the required capacity levels. In order to meet the temporary peak cooling demand, utilities are faced with either high-priced solutions, such as the construction of new power plants, or promoting demand-side management programs. At the present, many of the nation's natural gas facilities are underutilized during the summer months. One solution to the utilities' problems would be to utilize gas-driven cooling systems to displace electricity during the summer months. This solution would reduce peak electricity demand and improve the utilization of existing natural gas facilities.

Comparing a 10-ton electric rooftop unit to the GHP unit, the peak power demand electrical savings is approximately 84%. This is based on the equivalent electric heat pump [12] unit using 9.2 kW at the 95ºF (35ºC) cooling condition and the GHP only drawing 1.5 kW.

This reduction in electrical generation capacity would also have other benefits, such as lower NO_X and CO_2 emissions and reduced water consumption. GHP emission levels are much lower than coal-fired and even gas-fired plants producing electricity since the GHP is more efficient than a gas-fired plant at converting the gas to cooling.

Central station power plants also require large quantities of water for cooling. In arid regions, such as Nevada and Arizona, reducing electrical generation capacity during the summer months through the use of GHP units could reap huge savings in water that could be used for other purposes. At present, central stations in that area use approximately 0.294 to 0.515 gallons of water per kWh (1.1 to 1.9 L/kWh) [13]. At a savings rate of 7.7 kW (9.2 – 1.5 kW), the GHP would save 2.2 to 4 gallons of water per hour (2.2 to 15.1 L/h) of operation compared to electric equipment.

In the heating mode of operation, recovered heat from the engine was used to supplement the vapor compression cycle during heating. The following shows typical values for comparing resource COP of an electric driven unit (Heating Seasonal Performance Factor or HSPF of 12 or COP = 12/3.412 = 3.52) with the GHP in heating mode:

- Electrical Parasitic of GHP ≈ 1,500 W (excluding the indoor blower)
- Gas COP of GHP at 47ºF (8.3ºC) outdoor heating condition ≈ 1.64
- Capacity at 47ºF (8.3ºC) outdoor heating condition ≈ 157,995 Btu/h (46.3 kW)
- Primary Energy use for 1,500 W of electricity ≈ 1,500/0.32*3.412 = 15,994 Btu/h (4.7 kW)
- Resource COP for GHP at 47ºF (8.3ºC) heating condition = 1.41
- Resource COP for electric chiller at 47ºF (8.3ºC) heating condition = 3.52 * 0.32 = 1.13
- Results show that the use of the engine waste heat results in 25% greater operating efficiency in heating mode compared to electric driven units.

CONCLUSIONS

This paper demonstrates the design challenges that have been overcome. Laboratory results have shown that the unit operates at a high COP, reduces peak power, and provides enhanced comfort. The unit operates at a lower noise level than conventional electric rooftop units and has been designed to adapt to any roof curb configuration. The GHP system holds considerable promise in being one of the solutions to the current issue of over-taxed electric systems during summer peak conditions. The fact that the GHP system permanently reduces the electric peak power demand compared to a comparable sized electric unit by over 80%, provides a significant opportunity to rethink how commercial building use energy to provide building conditioning. By capturing waste heat, the GHP system provides many of the same benefits provided by a Combined Heat and Power (CHP) system. The efficient use of clean burning natural gas at the site combined with utilizing the waste heat for other work (such as domestic water heating) can increase overall efficiency compared to the current strategy in the United States where natural gas is used to generate electricity at a central power plant and the waste heat goes with the vented to the atmosphere.

Work is continuing on optimizing the overall performance of the system based on the feedbacks from the laboratory and field tests. Some of the proposed revision in the next generation GHP include: change type of refrigerant from R407C to R410A, reconfigure the current piping from dual to single circuit, and switch from relay logic type control system to programmable logic controller (PLC). The proposed

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changes will allow better controllability of the system, reduce weight by 20%, and reduce overall cost significantly.

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