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COMPRESSOR PERFORMANCE AT HIGH SUCTION TEMPERATURES WITH APPLICATION TO SOLAR HEAT PUMPS

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ABSTRACT

As part of the study of Solar Assisted Heat Pump (SAHP) Systems, the performance of the heat pump itself and its components under conditions attendant to series solar input to the evaporator is being investigated at Brookhaven National Laboratory (BNL). Particular emphasis has been placed on the details of the compressor performance, since in order to properly exploit the thermodynamic potential of high solar input temperatures (40 to 100°F), the compressor must operate efficiently over a wide range of (saturated) suction temperatures most of which are well above those for which present compressors are designed. A systematic series of experiments is being conducted at evaporating temperatures in the range from 45 to 100°F using a Solar Heat Pump Simulator and a specially designed Laboratory Model Heat Pump assembled from off-the-shelf components. Two reciprocating compressors have been tested thus far — an open type driven by a 2-speed motor and a hermetic 2-speed, the multi-speed feature providing capacity control, which is a virtual necessity for effective use of solar source. Thorough and highly accurate instrumentation is used in the simulator and in the heat pump refrigeration loop. This paper describes the results to date of the compressor aspects of the solar heat pump experiments at BNL and discusses the general application of heat pumps and their compressors to use with solar input.

INTRODUCTION

One of the most important ways in which actively collected solar energy can be used effectively for space heating is as input to the evaporator of a heat pump. In this configuration solar collection is at relatively low temperatures so that low cost collectors can be used while maintaining good efficiency of collection, the temperature being raised to space heating level by the heat pump cycle. Even though the temperatures involved (40 to 110°F) are low by solar collection standards, they are high by heat pump source standards and can produce very high Coefficients of Performance (COP's), thus being energy conservative, if the theoretical thermodynamic trends dictated by the ideal Vapor Compressor Cycle can be followed. This requires that the heat pump evaporating temperature be sufficiently high to allow for potential efficiencies as high as those achieved by use of a compressor with the lower speed used above 90°F. If the "series" solar heat pump, or SAHP, system is to become viable, heat pumps which exploit the performance potential of high source temperature must be developed, and it must be established whether present heat pump compressors can fill the requirements. BNL as support laboratory in charge of Solar Heat Pumps for the U.S. Department of Energy, Office of Solar Applications - Active Heating and Cooling Systems Branch has been carrying out a systematic in-house experimental study of solar input conditions to residential size heat pumps. A Solar Heat Pump Simulator and specially designed Laboratory Heat Pump, assembled from off-the-shelf components, are being used. Several sequences of performance tests have been carried out thus far with quite encouraging results with regard to achieving high COP and reliable performance of components. [1,2] BNL is also serving as Technical Manager of DOE contracts to develop effective and marketable high source temperature electrically driven heat pumps, [3] with the compressor configuration playing a major role. While the BNL studies have concentrated on reciprocating compressors thus far, the manufacturers' work has included both reciprocating and rotary types.
It should be noted that the technical approach of employing high evaporator temperatures, as available, applies to any vapor compression (VC) heat pump. This includes use of reclaim heat, geothermal heat, and low grade exhaust heat from a heat engine used to drive the compressor. Another long term application may be in conjunction with photovoltaic-powered heat pump concepts wherein hybrid photovoltaic/thermal collectors are used. The approach to the subject work has been to study the internal details of the VC cycle as a basic step, and therefore, the results may be applied rather broadly.

BACKGROUND AND THEORY

It is an inherent property of the vapor compression cycle that the amount of work required in the compression process decreases when the inlet pressure is increased or outlet pressure decreased, these pressures corresponding approximately to the evaporating and condensing temperatures of the working fluid (refrigerant) respectively. In Fig. 1 the results of theoretical cycle calculations for the heating mode compressor COP at a range of high evaporator temperatures for various constant condensing temperatures are shown for the assumed conditions noted. The refrigerant properties for R-12 were taken from [4]. These particular calculations correspond to an open compressor assumed to be operating at low speed with relatively high isentropic efficiencies (linearly decreasing from 85 to 70% based on empirical results). Implicit in the cycle calculations is a condenser effective enough to reject the heat at each suction condition without causing increased effective outlet pressure of the compressor. The non-linearities of the curves in Fig. 1 indicate a more rapid increase of COP with increasing \( T_{\text{evap}} \) but when the power required for pump/fan parasitics is included, they become more nearly linear. [2]

The benefit of increasing \( T_{\text{evap}} \) to reduce required compression work per pound of refrigerant may be demonstrated by the relation for isentropic compressor work:

\[
\Delta h_{1s} = \frac{\gamma}{\gamma-1} \frac{p_1^2}{p_1^{\gamma-1}} \left( \frac{1}{\gamma} - 1 \right)
\]

where 1 and 2 denote compressor inlet and outlet conditions respectively. Eq. 1 is an approximation here because refrigerant vapor does not behave as a thermally (or calorically) perfect gas near saturation, but is a good enough one to demonstrate the effect of pressure levels. With this in mind, if we further use:

\[
\frac{\gamma}{\gamma-1} \frac{p_1}{p_2} = c T_1
\]

it can be seen that the term inside the bracket of Eq. 1 dominates since the percent change in absolute temperature is small. Raising \( p_1 \) which corresponds closely to the saturation pressure for \( T_{\text{evap}} \) substantially decreases the required work if \( p_2 \) does not rise attendantly.

In the range of saturated suction temperatures considered here there is an important effect in the trend of compressor power requirement, \( P_c \). That is, the effect of increased mass flow, i.e.:

\[
\dot{m} = \rho_1 V N
\]

as suction density, \( \rho_1 \), increases which causes the power input to rise, becomes countered by the decrease in work required per pound of refrigerant, dictated principally by \( \Delta h_{1s} \) from Eq. 1 (modified by the isentropic and mechanical efficiencies) to the extent that \( P_c \) required levels off and then decreases with increasing evaporator temperature. This effect begins to occur in the range of 55 to 60°F for 4-pole speeds or less.

The heating capacities corresponding to the system assumed in Fig. 1 are shown in Fig. 2 (the compressor corresponding to 3 tons of cooling at 45, 105°F at 1800 RPM). The calculations of capacity assumed a linearly increasing volumetric efficiency, \( \eta_v \), with increasing \( T_{\text{evap}} \). The use of R-22, more typical in current heat pumps, would provide increased capacity, but, with solar assist, low source temperatures are not encountered and R-12 could be satisfactory. The capacity levels indicated in Fig. 2 demonstrate that (1) compressor sizing for a SAHP can be quite different than for a conventional heat pump and (2) some form of capacity control would, indeed, be important. Further factors pertaining to the above are the fact that the source (and evaporating) temperatures will not be at the highest temperature end very often because of solar
Fig. 2 Heating capacity at high source temperature

collection limitations and by the potential use of
a back-up heat source such as ambient air, the
earth, or ground water when solar supply is insuf­
ficient which would require evaporating temperatures
below the (nominal) 35 to 40°F minimum considered
for solar usage. This will be discussed further in
a later section.

**BNL EXPERIMENTS**

**Solar Heat Pump Simulator**

The simulator is liquid-to-liquid and consists of
a simulated solar source subsystem and heating load
subsystem, as described in [1]. Both are straight­
forward and are designed to provide tight control
of heat pump entering temperatures and water flow
rate. The source subsystem uses electric resistance
heaters to replace the heat withdrawn in the evap­
orator thus simulating solar input. Control set­
tings are manual at this time, with automatic ca­
pability to be incorporated in the future. Capa­
bility for air-cooled condensing is presently being
installed.

**Laboratory Heat Pump**

A liquid-to-liquid heat pump, complete except for
controls, was assembled using off-the-shelf compo­
ments. A schematic is shown in Fig. 3. Two dif­
ferent commercially available compressors have been
tested in this heat pump thus far:

(a) **Open-type reciprocating compressor (Duham­**
    *Bush BF-42).** This is a 2-cylinder unit with oil
    pump designed for military field use with belt
    drive, and has a nominal cooling capacity of 3 tons
    at 1750 RPM using R-12. For the experiments the
    belt drive was removed and the compressor was di­
    rectly coupled to a 2-speed (4-pole, 8-pole) 3-
    phase motor and driven at speeds of 870 and 1750
    RPM. Most data were obtained at the low speed in
    a power range where the nominal motor efficiency
    is 76% (no performance curves were run on the spe­
cific motor, however). The compressor cylinders

(b) **Two-speed hermetic reciprocating compressor (Len­**
    *nox L7A).** This is a 2-cylinder unit with rated
    capacity of 4 tons of cooling with R-22 at high
    speed of 3450 RPM, the low speed being 1725 RPM.
    The total displacement is 4.83 in.³. Although de­
    signed for R-22, the compressor was here again
tested with R-12.

The heat exchangers are both shell-and-tube type
of nominal 5 ton size, thus, large relative to the
compressor. The evaporator is a typical chiller
with refrigerant flow through inner-finned tubes
and the water passing through the baffled shell.
The condenser cooling water flows through externally-finned tubes and the refrigerant passes over them in the shell.

The expansion devices are externally balanced thermal expansion valves. A 3-ton and 5-ton valve are mounted in parallel, with shut-off valves allowing selection of one or the other. A manual fine-control valve installed in parallel can be used to complement the selected TXV.

Other elements include a receiver, an accumulator, and a subcooler, all having by-passes to allow operation with or without them. Three sight glasses are used. The configuration is one-way with no reversing valve. Either heating or cooling data are obtained by the same procedure, with the water flow being interpreted as reversed in an actual application.

Instrumentation and Data Acquisition

Heating capacity is determined by measurement of condenser water flow rate with a high precision float-type flowmeter (+0.5% instantaneous) and temperature rise of the water with platinum resistance thermometers (RTD's) matched to linearized bridge amplifiers to give 0.1°F accuracy on temperature difference. A 5T transducer (0.1°F) supplies redundancy. Motor KWe input is read on a 3-phase or 1-phase wattmeter, as appropriate (+1%) and the ratio of heat out/work in gives COP. Heat input to the evaporator is measured with a similar flowmeter and RTD's. Within the refrigerant loop, 6 temperatures are measured by RTD's (+0.1°F) and 5 pressures are measured with strain gauge transducers, at locations indicated in Fig. 3. These include compressor inlet and outlet conditions to allow computation of compressor efficiencies. Refrigerant flow rate is measured by a positive displacement flowmeter in the liquid line. Bourdon gauges and precision thermometers are also used in key locations for rapid visual observation and redundancy.

All analog data signals are recorded on a datalogger and on strip chart recorders when continuous traces are desired. On-line data reduction is available via PDP-11 computer.

RESULTS

Open-type Compressor

Results of a baseline series of steady-state performance tests showed that for the low speed a monotonically increasing compressor COP following the general trend dictated by the Carnot and Ideal Vapor Cycles was obtained in the range from 45 to 98°F evaporating temperature. No detrimental effects to the compressor were noticeable after approximately 200 hours of run time at the high evaporating temperatures. Representative data for heating COP are shown in Fig. 4 for a condensing temperature held constant at 120°F. Maximum compressor COP was 9.7 at T\text{evap} = 98°F. Several high speed data points are also included. The symbols in Fig. 4 indicate which expansion device configuration were used - above approximately 85°F evaporating it was necessary to switch to the 5-ton TXV to keep superheat down and performance up, and above 95°F the manual by-pass was added. The level of superheat entering the compressor varied from about 8°F at the low end to approximately 15°F at the high end. The performance with the modified inlet reed valves (solid symbols) indicates a slight, but not significant, improvement in performance. Upon disassembly, however, it was found that an attachment pin on one of the modified valves was improperly installed and inhibited valve deflection somewhat, so that performance in this configuration was probably impaired. Inspection of both sets of valves after testing found them in "as new" condition. During testing, oil level was monitored via a sight glass in the compressor crankcase, and no oil management problems were observed. In Fig. 5 the measured heating capacities corresponding to the COP data of Fig. 4 are shown.

In Fig. 6 are shown the compressor isentropic efficiencies calculated from enthalpies from measured state properties as:

\[ \eta_{is} = \frac{h_{2is} - h_1}{h_2 - h_1} \]  

where \( h_1 = f(p_1, T_1) \), \( h_2 = f(p_2, T_2) \) and \( h_{2is} = f(s = s_1, p_2) \).
Fig. 5 Measured heating capacities with open type compressor

Fig. 6 Isentropic efficiency of open compressor

Note that pressures and temperatures were measured in the lines immediately before and after the compressor, not in the compressor itself. The trends are as might be expected for an open compressor and the relatively mild fall-off is encouraging for high source temperature operation, being attributable largely to the low speed.

In Fig. 7 are shown the calculated volumetric efficiencies, determined as the ratio of actual system flow rate as indicated by the refrigerant flowmeter to the theoretical compressor pumping rate as determined from Eq. 3. As can be seen they increase linearly with $T_{evap}$, and reach very high values.

Fig. 7 Volumetric efficiency of open compressor

at the suction conditions explored here. Very good agreement was obtained between the flow rate indicated by the flowmeter and that calculated from the ratio of heat out divided by enthalpy change per pound of refrigerant passing through the condenser. An important feature of the data is that unloading of the motor was obtained at high evaporating temperatures. Power input was quite flat, from 1.78 to 1.85 kW, over the range from 50 to 80°F evaporating temperature, then gradually fell off to 1.55 at 98°F with the best expansion valve configuration.

Two-speed Hermetic Compressor

A series of tests similar to those described above was conducted using the 2-speed hermetic compressor in place of the open compressor, the remainder of the system being the same. The maximum evaporating temperature in this case was limited to 85°F, levels greater than this probably not being necessary for practical space heating. The heating COP results, shown in Fig. 8, demonstrated trends quite similar to those with the open type compressor, with each compressor at low speed - the low speed of the Lennox compressor being 4-pole (1725 RPM) rather than 8-pole, however. The importance here is that...
the results correspond to a production hermetic machine which is a component that could readily be introduced into a practical SAHP, and the speed is at a practical level for sufficient capacity.

The motor power is shown in Fig. 9 and shows that at low speed unloading was obtained above approximately T_{evap} = 63°F with 115°F condensing. High speed power continued to increase in the range tested, to T_{evap} = 70°F.

![Figure 9: Motor input power versus evaporating temperature for 2-speed hermetic compressor](image)

Fig. 9 Motor input power versus evaporating temperature for 2-speed hermetic compressor

**DISCUSSION, APPLICATION TO SOLAR HEAT PUMP SYSTEMS**

The ability to achieve very high COP's in the SAHP can be significant for energy conservation. The results reported herein and those of DOE contractors have shown the performance potential of the heat pump portion of the system. In this latter work results with small rolling piston compressors have shown particularly good performance at the low pressure ratios encountered [5] and in larger sizes the helical screw compressor can be very effective with solar source [6].

For the SAHP to become practical and marketable, however, it must become cost-competitive; and at this time it may be said that high COP's are a necessary, but not sufficient conditions for this to occur. The fact that solar supply often becomes depleted in winter except for prohibitively large collector areas, requires some form of back-up or auxiliary heat supply. If electric resistance heat is used for this task, as has been assumed in most simulations, the poor energy efficiency (COP = 1) dilutes severely the gains of high COP to the extent that the Seasonal Performance Factor (SPF) does not reflect enough energy savings for cost-competitiveness with conventional systems. Thus, a back-up with higher COP is required and since a heat pump is already present in the system, a source processed through it is a logical candidate. Possibilities include earth-coupling, air source (three-coil), ground water as available, and swimming pools. Earth-coupling appears to be the leading candidate and an extensive DOE-sponsored program is underway to develop its viability for a wide geographical area. [7] The effect of adding earth-coupling to a SAHP could affect the compressor requirements by presenting a wider range of evaporating temperatures on the low end, particularly if anti-freeze is used in an earth coil. Additionally, in the cooling season, with heat being rejected to the earth, the condensing temperatures and pressures would often be significantly lower than those for an ambient air cooled system. It is highly possible in fact, that earth coupled systems without solar energy will prove more competitive than SAHP's (at least sooner) and heat pumps and their compressors may need to be tuned to this application in the near future. High temperature solar input still provides the unique opportunity for very high SPF's, however, and the SAHP system potential must be explored fully. Also of importance for back-up is ambient air, providing a "dual-source" system with two evaporators. This system also would present a wide range of conditions to the compressor, wider than the earth-coupled SAHP, and include the necessity for defrost provision (solar heat being a possibility here). The dual source system has potential for high performance but presents complex design problems, including refrigeration management with two evaporators and optimization of the many possible control modes.

The need for compressor capacity control is important for any of these systems, and it is for energy efficient conventional heat pumps. There are, of course, satisfactory methods other than the speed variation treated in the reported work, including two compressors, twinned compressors in a single shell, and blocked-suction cylinder unloading [8]; but at the high suction temperatures of interest with solar input speed reduction affords the best opportunity to maintain high cylinder efficiencies and obtain motor unloading. Continuously variable speed could eventually prove important with solar heat pumps, particularly if photovoltaic electric drive becomes practical.

In summary, the introduction of solar energy into heat pump systems presents some new technical problems for compressor design and application engineers and this paper has attempted to introduce some of the considerations.

**REFERENCES**


