Evaluation of Liquid Pressure Amplifier Technology

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ABSTRACT: Liquid pressure amplifiers have been proposed as an energy saving technology for vapor compression refrigeration systems configured with direct-expansion evaporators. The technology utilizes a refrigerant pump in the high pressure liquid line as a means of maintaining a suitable pressure differential across the expansion valve while lowering condensing pressure to achieve the reduction of compressor energy consumption. Applications have been proposed on systems ranging from small unitary air-conditioning to large supermarket and commercial refrigeration systems. This paper clarifies the role of such a device in a vapor compression refrigeration system. Limitations are presented and discussed. Finally, results of detailed analyses are presented to quantify the energy consumption both with and without a liquid pressure amplifier in a unitary air conditioning system. The estimated energy savings associated with the installation of a liquid pressure amplifier are minimal.

Nomenclature

h : enthalpy [kJ/kg]

P : pressure [kPa]

V : volumetric flow [m^3/s]

Greek symbols

η : efficiency

ρ : density [kg/m^3]

1. Introduction

There is a constant quest for methods, equipment, and strategies that can maximize the efficiency of refrigeration systems. Some are focused on improving individual components that comprise a refrigeration system while others are aimed at improving the system as a whole through better integration and operation. Regardless of the approach, some methods are successful at achieving their objective of improving efficiency while others provide marginal or no efficiency benefit.

In 1986, a US patent was issued for the application of a liquid refrigerant pump at a point in the refrigeration system between the condenser (or high pressure receiver if equipped) and the expansion device (US Patent #4,599,873) for the purpose of improving refrigeration system capacity and efficiency.

Figure 1 shows a simple vapor compression refrigeration system with a liquid refrigerant pump analogous to that described in US Patent #4,599,873. The patent stated that efficiency gains were realized by operating the refrigeration system with lower condensing (or "head")

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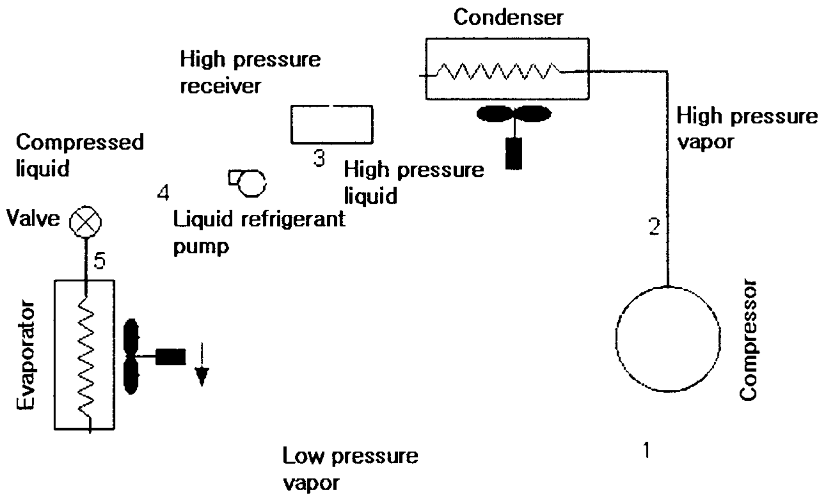

Fig. 1 Vapor compression refrigeration system with pressure amplifier.

pressure while using the smaller liquid refrigerant pump as a device to increase liquid line pressure; thereby, suppressing flash gas. The pressure increases afforded by the pump were reported in the patent to range between 136 kPa and 184 kPa [5 and 12 psig].

This paper discusses the potential for refrigeration system efficiency and capacity improvements through floating head pressure. It also includes an assessment of potentially enabling technologies that include liquid pressure amplifiers and subcoolers. The paper begins by considering the fundamental effects of lowering head pressure on a vapor compression refrigeration system. Barriers to lowering head pressure are discussed followed by an assessment of liquid pressure amplifier technology vs. subcoolers.

2. Effects of lower head pressure

On a relative basis for a fixed ambient condition and refrigeration load, lowering system head pressure will result in:

- an increase in condenser fan energy consumption
- a large decrease in compressor energy consumption
- a small increase in compressor capacity
- decrease in oil cooling load (applicable to screw compressors)
- lower compressor compression ratio resulting in prolonged compressor life
- the potential for improved system efficiency

As a rule-of-thumb, a compressor will realize approximately 1.3% improvement in efficiency (lower kWe/kWT [BHP/ton]) for each degree Fahrenheit lower in saturated condensing temperature. For example, a compressor operating at −18℃ (210 kPa) saturated suction temperature and 35℃ (1,351 kPa) saturated condensing temperature would require approximately 0.356 kWe per kWT [1.68 BHP for each ton] of refrigeration effect delivered. If the condensing temperature was decreased to 29℃ (1,149 kPa), the efficiency of the compressor improves requiring only 0.310 kWe per kWT of refrigeration [1.46 BHP for each ton]. Thus, a 6℃ decrease on saturated condensing temperature (or 69 kPa decrease in equivalent saturated condensing pressure) leads to a nearly 13% improvement in compressor efficiency. The actual compressor performance enhancement with lower condensing temperatures will depend on the specific compressor technology (reciprocating, screw, etc.) and its individual performance characteristics.

Brownell et al.[1] investigated a new R-22 evaporatively-condensed commercial refrigeration system serving a publicly-owned ice skating rink. As-found, the system operated with head pressures that ranged from 1,517~1,725 kPag [220 to 250 psig] continuously. The authors were able to re-set system controls to allow the system head pressure to float down as low as 1,070 kPa during favorable outdoor air conditions (low wet bulb). Although the lower head pressure increased condenser fan energy consumption, the savings in compressor power more than offset the condenser fan energy increase. Overall, the system realized a 21% reduction in operating costs.

Manske et al.[2] evaluated a separate evaporatively-condensed industrial refrigeration system and identified a novel strategy for optimizing refrigeration system performance by a head pressure re-set strategy based on outdoor air wet bulb temperature. The strategy identified resulted in an estimated 11% reduction in system energy consumption.

Although lowering the head pressure in a refrigeration system is often desirable from an energy perspective, there are practical constraints that limit a particular systems ability to operate at low head pressures. Barriers that commonly stand in the way of successfully lowering system head pressure include:

- presence and selection of thermostatic expansion valves
- sizing of high pressure liquid line
- condenser and compressor selection
- presence of heat recovery systems
- requirements for hot gas defrosting

Most thermostatic expansion valves need at least 517 kPa [75 psig] of pressure differential to function properly. With a given fixed evaporator pressure to meet the air-conditioning or refrigeration needs for a system, that pressure plus the minimum differential across the expansion device establishes a minimum pressure immediately upstream of the expansion device. If the pressure upstream of the expansion device is lower than this minimum, the valve will tend to lose its controllability and hunt. In addition, the evaporator runs the risk of being starved for refrigerant due to a diminished pressure difference with which to move fluid through the expansion valve operating wide open.

3. Determining a systems minimum head pressure

Since no system can carry refrigerant through piping without pressure drop, the operating pressure drop in the high pressure liquid line further increases the minimum condensing pressure for a system. The high pressure liquid line for most commercial and unitary air-conditioning systems is sized for no more than 1.1 ℃ [2°F] of equivalent pressure loss.[3] For a system using R-22, this translates into a liquid line pressure drop of 38 kPa [5.5 psig] at a saturation temperature of 35℃ [95°F]. If the liquid line pressure drop exceeds the available saturation pressure depression provided by subcooling or liquid compression, flash gas will form in the liquid line and the systems refrigeration capacity will decrease.

$$P_{cond,\,min} = P_{evap,\,sat} + \Delta P_{expansion\ valve,\,min} + \Delta P_{high\text{-}pressure\ liquid\ line} \tag{1}$$

For a R-22 unitary air-conditioning system operating with a typical evaporator pressure of 525 kPag gauge [76 psig], a thermostatic expansion valve with 517 kPag gauge [75 psig] differential and a liquid line sized for 1.1℃ [2°F] equivalent line loss, the minimum condensing pressure would be as follows:

$$P_{cond,\,min} = 525 + 518 + 38 = 1{,}081\ \text{kPag}\ [156.6\ \text{psig}]$$

Unfortunately, many systems operate with head pressures well in excess of the hypothetical example just presented. Brownell et al.[1] investigated a new R-22 commercial refrigeration system serving a skating rink. As-found, the system operated with head pressures that ranged from 1,517~1,725 kPag [220 to 250 psig]. Brownell et al. were able to re-set the system head pressure down to 1,070 kPa [155 psia] and realize a 21% reduction in operating costs without the addition of a device such as a liquid refrigerant pump. If a liquid refrigerant pump was added to this system, the head pressure could have been lowered an additional 34~83 kPa [5~12 psi] further. This would have resulted in a modest amount of incremental compressor power savings; however, the compressor savings is partially offset by the energy required to operate the refrigerant pump itself and additional condenser fans to achieve the lower head pressure operating point.

4. Liquid pressure amplification vs. subcooling

The fundamental principle of lowering or floating head pressure to achieve refrigeration system efficiency improvement is sound. As we have seen, there are limitations in the degree to which head pressure can be lowered in a system. The presence of devices such as a liquid pressure amplifier and subcooling can, potentially, allow a refrigeration system to operate with somewhat lower head pressures than a system absent of a liquid pressure amplifier or subcooler. Confusion has been created by applying unsound principles and methods in the process of quantifying potential efficiency and operating cost savings attributable to the application of these devices themselves. In this section, we discuss the operating principles of both liquid pressure amplifiers and subcoolers in attempts to clear misconceptions with their operation propagated in past publications.

Tomczyk[4] provided a qualitative discussion of liquid pressure amplifiers for air-conditioning and commercial refrigeration applications. Unfortunately, many of the assumptions made by Tomczyk failed to properly portray the functional effects of a liquid refrigerant pump integrated with a vapor compression system. For example, Tomczyk incorrectly states that the liquid refrigerant pump increases the pressure of the refrigerant in the liquid line without increasing its temperature. Although the design of this particular refrigerant pump minimizes the refrigerant temperature rise by separating the electric motor from the refrigerant stream, there will be an enthalpy increase due to the pumping process. Accompanying the increase in enthalpy will be an increase in refrigerant temperature. The following is a simple first law energy balance on the pump relating the work done by the pump on the refrigerant to the change in energy state across the pump.

$$\rho_{liquid} \cdot V \cdot (h_{out} - h_{in})_{pump} = \frac{V \cdot \Delta P_{pump}}{\eta_{pump}} \quad (2)$$

where ρ_{liquid} is the liquid refrigerant density at the condensing pressure, V is the volumetric refrigerant flow through the pump, h_{out} is the refrigerant enthalpy at the pump outlet, h_{in} is the refrigerant enthalpy at the pump inlet, ΔP_{pump} is the pressure increase developed by the pump, and η_{pump} is the pump efficiency. If the electric motor was exposed to the refrigerant stream, the motor efficiency would be included in the denominator along with the pump efficiency. Clearly, the greater the head developed by the pump, the greater the enthalpy increase of the refrigerant. Increasing refrigerant enthalpy across pump leads to decreased refrigerant capacity and diminished ability to avoid flash gas formation with pressure drop in the liquid line. The process line for the liquid refrigerant pump is illustrated on the

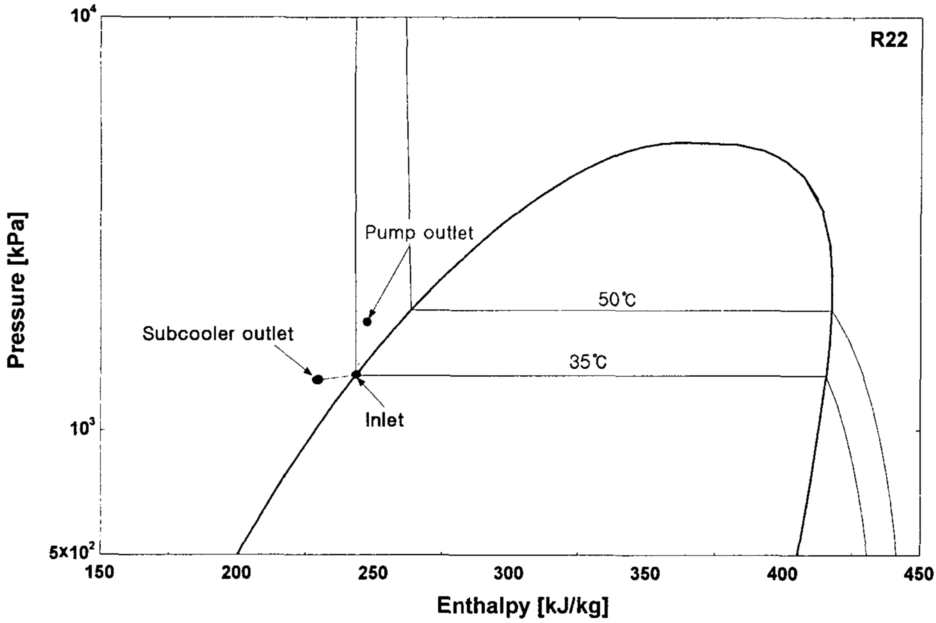

Fig. 2 Pressure-enthalpy diagram for R-22 showing inlet and outlet states for both a liquid refrigerant pump and subcooler.

pressure-enthalpy diagram shown in Fig. 2.

Consider a system operating with R-22 at a nominal saturated condensing temperature of 35℃. Assuming the liquid refrigerant pump developed 138 kPa (the upper range of pressure increase for the pump as-reported by Tomczyk) of head with a pump efficiency of 50%, the enthalpy increase across the pump can be calculated using equation. In this case, the enthalpy increase is 0.239 kJ/kg. The corresponding temperature rise for this enthalpy increase is 0.17 ℃. Clearly, this is not a zero heat gain process. A point also overlooked by MacWhirter.[5]

For comparative purposes, Fig. 2 shows an alternative approach, subcooling, which directly decreases the refrigerant temperature below its saturation temperature. The outlet state of a subcooler will be refrigerant at a lower enthalpy and pressure as compared to the inlet state. A desirable effect of decreased enthalpy across a subcooler is the ability to directly increase the refrigeration effect for the system by feeding refrigerant to loads with decreased enthalpy.

There are a number of alternative methods that can be implemented to achieve refrigerant subcooling including: ambient subcooler, dedicated mechanical subcooler, flash subcooler, and liquid-suction heat exchanger. Cole[6] discusses the positive benefits of subcooling liquid refrigerant and its impact on both capacity and efficiency in refrigeration systems. Klein et al.[7] quantified the performance of liquid-suction heat exchangers on refrigeration systems using a range of working fluid alternatives. The authors found that liquid-suction heat exchangers designed for low pressure loss on the low pressure side are useful for systems using R507A, R134a, R12, R404A, R290, R407C, R600, and R410A refrigerants. The sensitivity of vapor-side pressure drop in the liquid-suction heat exchanger did provide a good match for achieving subcooling in systems using R-22, R32, or R717 refrigerants. FEMP[8] summarized the various types of subcooler options and provided an overview of the benefits of subcooling for increased refrigeration system performance.

A modified version of the ambient subcooler using a small evaporative cooling heat exchanger provided an effective and efficient means of depressing the refrigerants saturation temperature well below the outdoor air dry bulb temperature.

5. System model

A considerable amount of confusion has been propagated with regard to the energy benefits associate with applications that implement the liquid refrigerant pumps applied to increase liquid line pressure. Quite simply, the majority of confusion arises out of taking credit for energy savings where credit is not due.

Take the ice rink that Brownell et al. evaluated. The original condition of the system found head pressures ranging from 1,517 to 1,725 kPag [220 to 250 psig] throughout the year. Brownell et al. were able to re-set the head pressure down to 1,070 kPa [155 psia] and achieve reliable operation without a pump. If a liquid refrigerant pump was added, the system head pressure could be lowered further. A conservative estimate of the additional reduction in head pressure would be based on the head the pump was able to develop. If the pump could develop 83 kPa, the new head pressure set point would be 986 kPa. The system would then realize an additional 3% savings (this is a conservative estimate not including the additional energy for operating the pump). Marketing materials for this product would take credit for the entire reduction in head pressure from the as-found condition to 986 kPa with an estimated savings on the order of 25% – not the 3% that is rightly attributable to the pump itself.

To accurately estimate savings associated with floating head pressure in systems both with and without a liquid refrigerant pump in the high pressure liquid line, a first-principles model of a unitary air-cooled vapor compression-based air conditioning system was developed. The air conditioning system assumes the evaporator serves a single zone with a fixed outdoor air fraction. A bin analysis is performed to estimate the annual energy consumption of the system. A diversified load profile is used to generate the air conditioning load for each bin of ambient condition.[9] The system utilizes an air-side dry bulb economizer. Table 1 lists relevant parameters and their nominal values for the air conditioning system used in the simulations.

Table 1 Air conditioning system parameters

Parameter	Nominal value
Capacity, kW_T [tons]	77 [22]
Supply air temperature, ℃ [°F]	12.2 [54]
Space set point temperature, ℃ [°F]	23.3 [74]
Return air temperature, ℃ [°F]	23.9 [75]
Economizer change temperature, ℃ [°F]	17.8 [64]
Suction line loss, ℃ [°F]	1.1 [2]
Discharge line loss, ℃ [°F]	1.1 [2]
Subcooling, ℃ [°F]	1.7 [3]
Supply air flow rate, l/s [CFM]	5,899 [12,500]
Outdoor air flow rate, l/s [CFM]	472 [1,000]
Supply fan power, kW_e	13.4
Liquid refrigerant pump pressure, kPa [psi]	83 [12]
Liquid refrigerant pump efficiency	0.5

6. Results

This section describes the results obtained from the model with the primary goal to better understand the potential for liquid pressure amplifiers to achieve energy savings when implemented. Three separate US city locations were included in the simulation to obtain a representative sample of varying climate times. The locations included: Madison, WI (upper Midwest), Washington, DC (mid-eastern costal location) and Cocoa Beach, FL (southern costal location). Figure 3 shows the hours of occurrence over a range of dry bulb temperatures for each of the locations. Cocoa Beach is a warm climate dominated by a significant number of hours where air conditioning is required. Madison and Washington climates are quite different with a bimodal distribution in hours of occurrence at moderately high and lower temperatures.

Figure 4 shows a cooling load distribution expressed as a fraction of the design cooling load as a function of the bin temperature for the Cocoa Beach location. When the outdoor air dry bulb temperature drops below the economizer change over temperature, the cooling load on the air conditioner drops to zero and the entire facility cooling needs can be met solely by air-side economizer operation. Interesting, the conditions where a liquid pressure amplifier can provide the greatest benefit (low condensing pressures) is exactly the conditions where the cooling loads are reduced and the need for mechanical cooling is diminished.

Table 2 shows estimated annual energy consumption for cooling mode operation of a unitary air conditioner operating both with and without a liquid refrigerant pump for the three locations identified previously. Annual energy consumption figures are provided over a range of minimum saturated condensing temperatures that would be programmed into a systems's controls based on the expansion valve selection. In each case, the system with a pump is allowed to operate at a minimum condensing temperature of 2.8℃ below that without a pump.

The incremental savings associated with the presence of the liquid refrigerant pump are noted for that operating state. For a system that fixes the minimum condensing temperature equally for both cases with and without the

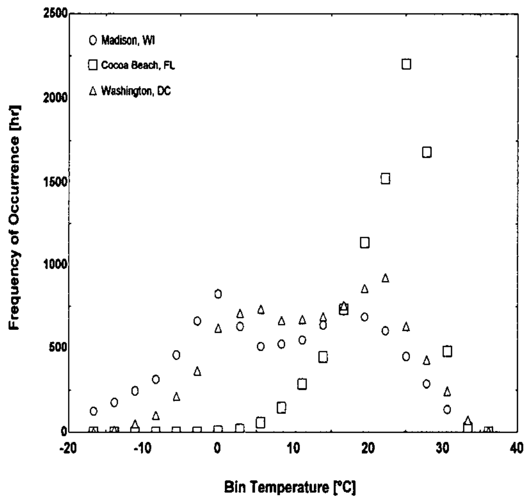

Fig. 3 Frequency distribution of dry bulb temperatures for locations included in analysis.

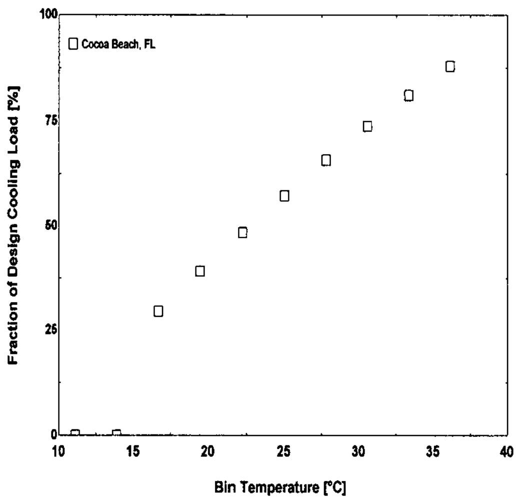

Fig. 4 Load distribution for Cocoa Beach, FL location.

Table 2 Annual air conditioning energy consumption with and without a LPA

Location	Minimum saturated condensing temperature ℃ [°F]	Annual air-conditioning energy consumption [kWh]		
		No LPA	LPA	Difference [%]
Madison, WI	41 [105]	98,082	98,180	0.1
	38 [100]	96,812	97,116	0.3
	35 [95]	95,958	96,052	0.1
	32 [90]	95,466	95,560	0.1
	29 [85]	95,466	95,325	−0.1
Washington, DC	41 [105]	106,749	106,928	0.2
	38 [100]	105,142	105,275	0.1
	35 [95]	104,101	104,232	0.1
	32 [90]	103,527	103,671	0.1
	29 [85]	103,537	103,427	−0.1
Cocoa Beach, FL	41 [105]	142,357	142,647	0.2
	38 [100]	139,915	140,209	0.2
	35 [95]	138,527	138,820	0.2
	32 [90]	137,891	138,183	0.2
	29 [85]	137,891	137,945	0

pump, there is an annual performance penalty associated with the additional energy required to operate the pump. In general, the liquid refrigerant pump offers an opportunity to reduce the annual energy consumption by only fractions of a percent. This opportunity is afforded by the additional allowable reduction in minimum condensing temperature while suppressing the formation of flash gas at the expansion valve inlet. Unfortunately, the outdoor air conditions that gives the operation of an air conditioning system the greatest potential (low ambient temperatures) is also coincident with low demands for air conditioning. The small of savings potential associated with the installation of a liquid refrigerant pump is clearly not sufficient to justify the additional cost, complexity, and operational risk.

7. Conclusions

The ability to lower the condensing pressure of a vapor compression refrigeration system during favorable outdoor air conditions can significantly reduce the system's energy consumption and improve its coefficient of performance. All vapor compression refrigeration systems have a lower limit in their ability to "float" the condensing pressure. A system equipped with a liquid refrigerant pump in the high pressure liquid line can achieve stable operation at slightly lower saturated condensing temperatures; however, the benefit of this lower condensing temperature operating condition is diminished by a consequent lower demand for air conditioning. For the cases considered in this paper, the addition of a liquid refrigerant pump yielded only a 0.1% reduction in annual energy consumption. Such a small savings in system energy consumption does not warrant the installation of such devices.

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