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ABSTRACT

Typical automotive air conditioning refrigeration systems employ thermostatic expansion valve or orifice tube controls in a single stage refrigerant circuit. A new automotive air conditioning compressor has been developed that has a two stage compression process, which makes designing and producing a two stage refrigerant circuit more practical. Since this compressor can function in a single stage circuit and a two stage circuit, this paper uses it when comparing the system efficiencies by simulating both circuits. A model of this new compressor is incorporated in a proprietary computer program that solves the refrigerant loop for a steady state solution under a set of external operating conditions. For a two stage circuit, the results show that the compressor speed and input power can be reduced by 19%, while providing the same cooling capacity as the single stage circuit.

NOMENCLATURE

COP	Coefficient of performance = Q_{evap} / Q_{power} (unitless)
M_{high}	2nd stage compressor mass flow (lb/h)(kg/h)
M_{int}	intermediate pressure mass flow (lb/h)(kg/h)
M_{low}	1st stage compressor mass flow (lb/h)(kg/h)
Q_{comp}	Convective heat transfer from the compressor to ambient (BTU/h)(W)
Q_{cond}	Condenser heat transfer (BTU/h)(W)
$Q_{cooler\ abs}$	liquid intercooler (subcooler) heat transfer to intermediate pressure refrigerant (BTU/h)(W)
$Q_{cooler\ rej}$	liquid intercooler (subcooler) heat transfer from condensed liquid refrigerant (BTU/h)(W)
Q_{evap}	Evaporator heat absorbed (BTU/h)(W)
Q_{power}	Heat equivalent compressor input power (BTU/h)(W)
$Q_{suction}$	Heat absorbed by the low pressure suction line (BTU/h)(W)

INTRODUCTION

Trends in the automotive industry are increasing the importance of improved customer satisfaction in air conditioning system's performance and efficiency. In European markets, sales of new cars equipped with air conditioning are increasing, even though the price of gasoline is 3 to 5 times more costly than it is in the United States.

Other authors have found increased efficiencies of two stage circuits over single stage circuits. Threlkeld (1) gave an example of a two stage circuit with a water intercooler and flash intercooler that resulted in a 22% increase in COP vs. a single stage circuit with ammonia used as the refrigerant. Threlkeld notes that a circuit as shown in figure 1 is suitable for use with refrigerant 12, which was used in automobiles prior to use of refrigerant 134a. Stoecker and Jones (2) explain the advantage of a two stage circuit is to reduce the amount of flash gas developed in the expansion process, and that expanding the intermediate pressure vapor to suction pressure is a wasteful process. The circuit in figure 1 may also be called a liquid subcooler, since it does little interstage cooling for the compressor, but subcools the liquid going to the evaporator expansion valve. The subcooler has the advantage over a flash intercooler vessel of retaining the high pressure in the liquid, which will prevent liquid line flashing if there is a high pressure drop in the liquid line going to the evaporator expansion valve. Shelton and Grossman (3) developed a shortcut for calculations to evaluate a refrigerant circuit and select an efficient refrigerant. One example of a flash intercooling type two stage circuit with presaturators and refrigerant 40 resulted in 8.7% less required power vs. a single stage system. Zubair, Yaqub and Khan (4) found that the arithematic mean of the condenser and evaporator temperatures (AMT) gave a COP closer to the theoretical maximum than the geometric mean of the condenser and evaporator pressures (GMP) for an ideal two stage refrigeration circuit using R134a and a flash intercooler. The same was found for a mechanical subcooling circuit.

There are several ways to construct a two stage circuit, each having its own advantages. This paper compares a single stage refrigerant circuit (figures 2 and 4) to a subcooler type two stage refrigerant circuit (figures 1 and 3). A thermostatic expansion valve (txv) is used as the throttling control in each case, being set to 10°F(5.6°C) superheat. In figure 1, there are two different control points, A and B, for the txv controlling the liquid intercooler. The system for this paper used control point A. Point B would produce cooler second stage compressor discharge temperatures, but may be much more difficult to arrange in a vehicle.

Figure 1. Schematic Diagram for a Two Stage Refrigeration Circuit with Liquid Intercooler

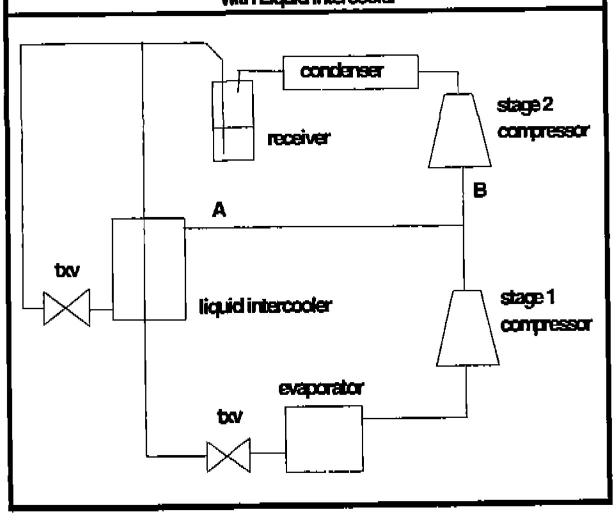


Figure 2. Schematic Diagram for a Single Stage Refrigeration Circuit

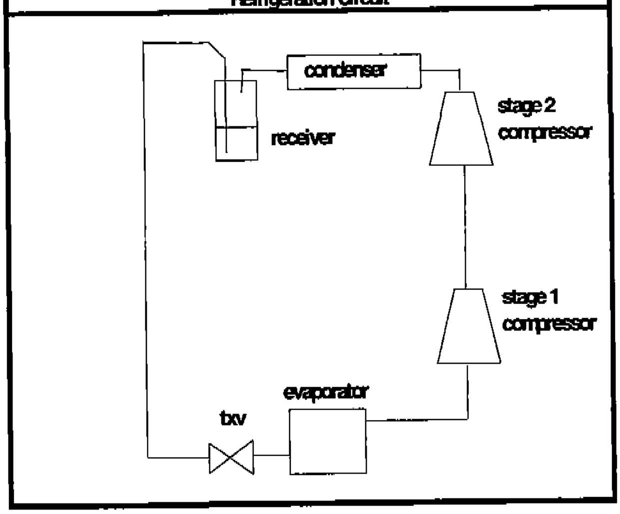


Figure 3. Two Stage Liquid Intercooler Refrigeration Circuit Diagram

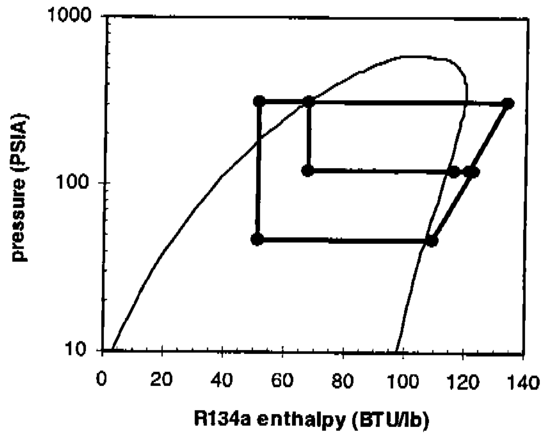
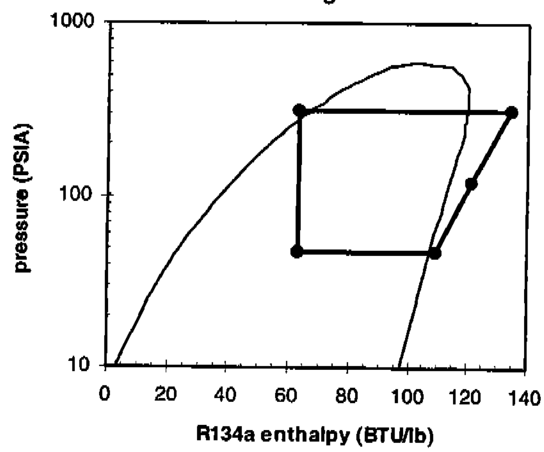


Figure 4. Single Stage Refrigeration Circuit Diagram



With a series of assumptions, one can do a pencil and paper analysis of these systems and compare capacity and COP on a per ton basis. A customer does not know what the a/c system is doing on a per ton basis, but does know if the cabin is cooling faster and if the gasoline mileage changed. The advantage of this program is that it will produce the data that can be used to determine what the customer will see. This program solves the refrigerant loop for a steady state solution under a fixed set of external operating conditions: compressor rpm, ambient temperature around the compressor, condenser airflow, condenser entering airflow temperature, evaporator airflow, and evaporator entering airflow enthalpy.

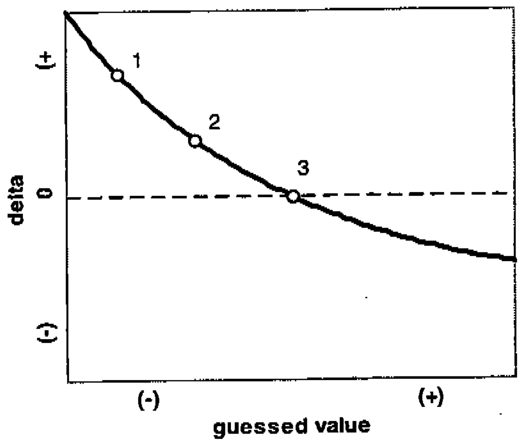
SIMULATION DETAILS

The compressor model is neither adiabatic nor isentropic. The volumetric and isentropic efficiencies of each stage are interpolated on a map as a function of rpm and pressure ratio. Convective heat loss from the compressor to the engine compartment ambient air is calculated. The condenser is a geometrically driven model that solves for the refrigerant state in each increment through the flow path. The evaporator model is not geometrically driven, but is based on the NTU-effectiveness method, as described in Kays and London (5). The liquid intercooler is modeled similar to the evaporator, being NTU-effectiveness based with a simple geometry driving the calculated effectiveness. The expansion valves are modeled as constant superheat to constant specific volume curves. The condenser, evaporator and compressor models were based on calorimeter data.

The single stage circuit solves for the intermediate stage conditions inside the compressor, as the first stage feeds directly into the second stage with no other connections. In the two stage circuit, the intermediate pressure is part of the solution, but is not an input, or a controlled condition. The two stages of the compressor are fixed in displacement and are forced to run at the same shaft speed. Zubair, Yaqub and Khan (4) show an example of a different two stage circuit where the compressors are independent such that the intermediate pressure can be a controlled condition.

The solution to any refrigerant system is a computationally intensive one. To solve for the steady state conditions, the refrigerant circuit was divided into its components and broken into 6 layers of iterations. The outermost layer of iteration starts with an assumed value for the condenser exit pressure, P_{guess}, then calculates the rest of the system, arriving back at the condenser exit. The calculated value, P_{calc}, is compared to the guessed value to determine the difference, Δ.

Figure 5. Iterative Solution Method



$$\Delta = P_{guess} - P_{calc} \tag{1}$$

A new P_{guess} is derived by taking a step towards the solution, where the step is a function of Δ in terms of magnitude and direction. P_{calc} calculated again (figure 5 point 2). This is repeated, until the solution is found (figure 5 point 3). The solution is converged when Δ is less than the convergence limit. Care was taken to select a tight enough limit to give a consistent solution, yet not require double precision variables. Values are typically 0.001 or less. Using this method, the solution is typically found in 3-5 iterations, minimizing the CPU time required. System results are checked for accuracy with not only the internal convergence checks on iteration variables, but also in these overall system equations must be satisfied:

$$M_{low} + M_{int} = M_{high} \tag{2}$$

$$Q_{evap} + Q_{suction} + Q_{power} + Q_{cooler\ abs} = Q_{cond} + Q_{comp} + Q_{cooler\ rej} \tag{3}$$

RESULTS

The two systems in figures 1 and 2 were compared at one steady state point. The heat loads at this point were determined from a wind tunnel test of a mid-sized late model sedan at the 10 minute point in a cooling procedure that starts cooling the car's cabin after it has been thermal soaked with simulated sunshine at 110°F(43.3°C) ambient. Both systems were run under the same external conditions, with three different specified levels of condenser exit subcooling. The base point for comparison was the single stage system running at 10°F(5.6°C) subcooling. The two stage circuit averaged 6.4% higher COP, 10.1% higher capacity, and 3.5% higher input power when compared to the single stage circuit (figures 6,7 &8).

To provide an easier comparison, the compressor rpm in the two stage circuit was lowered until the capacities matched at 20°F(11.1°C) subcooling. This is a conservative approach, as the compressor rpm could be decreased even more, if the comparison was done at 0° subcooling. Not all system manufacturers charge a system to the same amount of subcooling. At the lower compressor rpm, the two stage circuit averaged 24.8% higher COP, 1.7% higher capacity, and 18.5% lower input power when compared to the single stage circuit (figures 6,7 &8).

Figure 6. Compressor Power vs. Condenser Subcooling

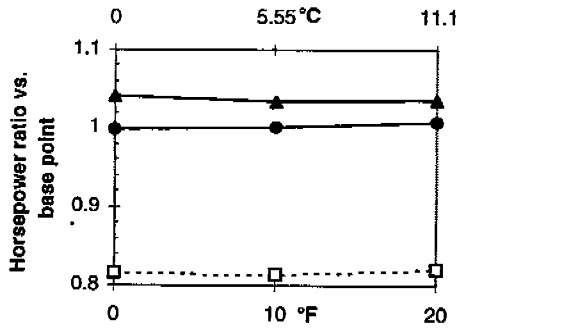


Figure 7. COP vs. Condenser Subcooling

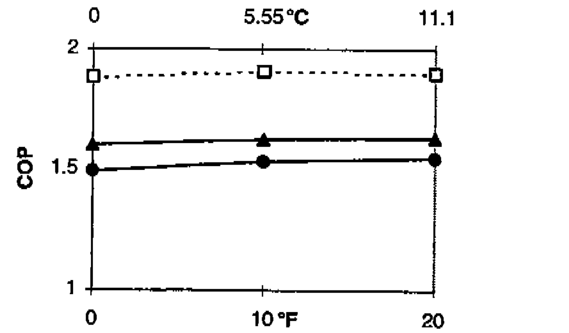


Figure 8. Evaporator Capacity vs. Condenser Subcooling

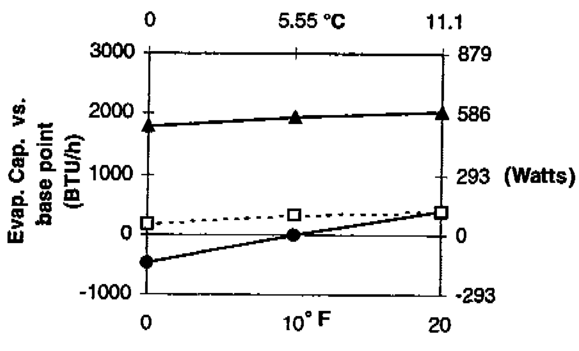


Figure 9. Discharge Pressure vs. Condenser Subcooling

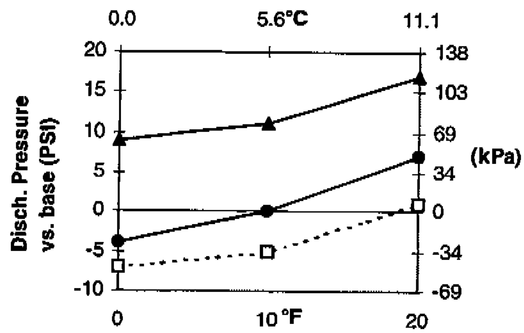


Figure 9 shows the trend of increased compressor discharge pressure associated with higher subcooling levels, and the lower pressures that could be gained by running at the lower compressor rpm. Figure 10 shows the subcooling level of the refrigerant entering the evaporator expansion valve. The model has no heat transfer or pressure drop included in the liquid line or the receiver. The intercooling (subcooling) heat exchanger was given a geometry that would yield an effectiveness of nearly 85%.

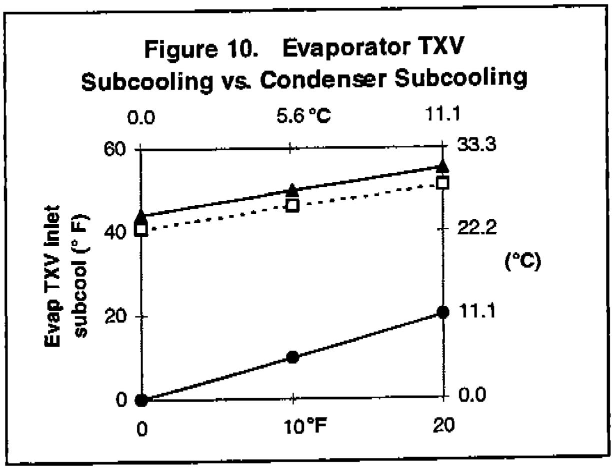
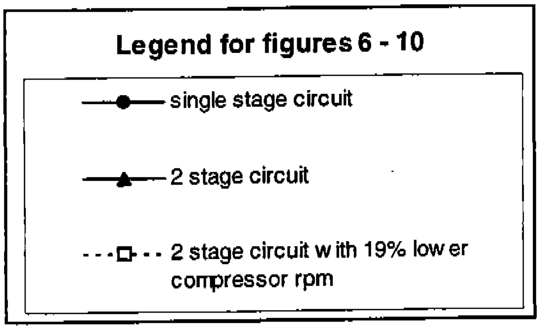


Figure 10. Evaporator TXV Subcooling vs. Condenser Subcooling



CONCLUSIONS

The liquid intercooler two stage refrigerant circuit has shown an increase in cooling capacity with the existing condenser, evaporator and compressor components, but does this at the cost of higher input power requirements and higher discharge pressures. By lowering the compressor rpm to match cooling capacity, input power can be reduced and compressor durability can be increased. These conclusions will be confirmed with system test data in a controlled labatory environment in the near future.

ACKNOWLEDGEMENTS

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