

A/C Cycle Power Recovery Method

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ABSTRACT

The conventional A/C cycle develops high pressure refrigerant that is throttled (via a small opening) to low pressure and temperature for cooling an evaporator's airflow. This paper explores the conceptual and practical development of deriving useful work from that pressure potential by replacing the throttling device with an expander/motor. One key technical challenge is to maintain a large pressure drop across the expander/motor. This translates into a strict control requirement of the location of the refrigerant flow's phase change within the motor. The cycle's COP benefits from recovered work and the evaporator's refrigerant energy content reduction while delivering conventional cooling performance.

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INTRODUCTION

Although the concept of recovering power from an A/C cycle is not new, the practical obstacles to significant power recovery achievement from a two phase refrigerant flow (i.e. R1234yf, R134a, R12...) are not obvious and have been met by unique methods disclosed in this paper. The theoretical potential of the concept is explained followed by a demonstration of a practical application that depends on direct measurement of an expander/motor shaft power (which avoids measurement accuracy risk from refrigerant properties and mass flow). Throughout the development, conventional A/C cooling performance must be served.

The conventional A/C cycle develops high pressure and temperature refrigerant in the condenser that is throttled to low pressure and temperature for cooling air within an evaporator. This throttling process involves little more than passing the flow through a small orifice interface (or expansion valve). At this interface of high and low pressure, liquid refrigerant expands to form vapor. This mechanical action provides an opportunity for a motor/expander to extract work, thereby recovering a portion of the energy spent to compress the refrigerant. Further, refrigerant energy reduction prior to evaporator entry is desirable. Figure 1.0 shows the basic structure of this study that replaces the throttling device with an expander/motor.

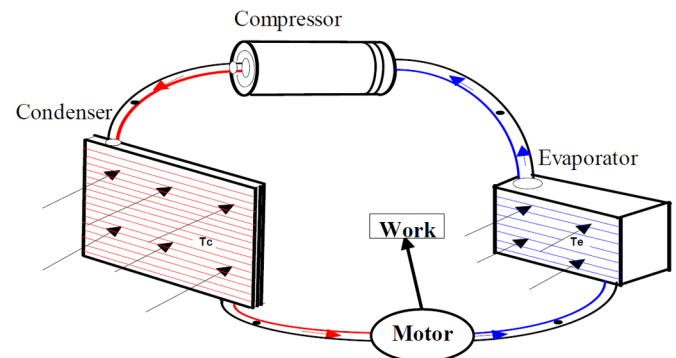


Figure 1.0. Expander/Motor in the A/C Circuit

THEORETICAL REVIEW OF POTENTIAL BENEFIT

Conventionally, liquid phase refrigerant exits from the condenser and passes through an expansion valve without energy gain or loss as it drops to a low pressure en route to the evaporator. This pressure reduction causes some of the liquid to change phase (cavitate) into a gas. In this way, the energy (enthalpy) is divided into an energetic cool gas and a cool liquid. This gas creation represents a volumetric expansion ratio of nearly 70:1 relative to the liquid (R134a) [1]. Thus, it is believed that this action can be used to extract mechanical work with a theoretical limit dictated by an isentropic expansion assumption. This assumption is used in the following developments:

In the case of Chart 1.0, a conventional cycle (containing an orifice) has refrigerant exiting the condenser as a saturated liquid (no sub-cooling). A second cycle is overlaid on the first that contains an expander/motor (dashed line). The “width” of the evaporator's refrigerant specific enthalpy change (refrigeration effect) is kept constant for each cycle. Since the expander motor removes energy from the refrigerant, the condenser-exit refrigerant must have higher specific enthalpy content. This suggests that the second cycle's relative refrigerant content is reduced.

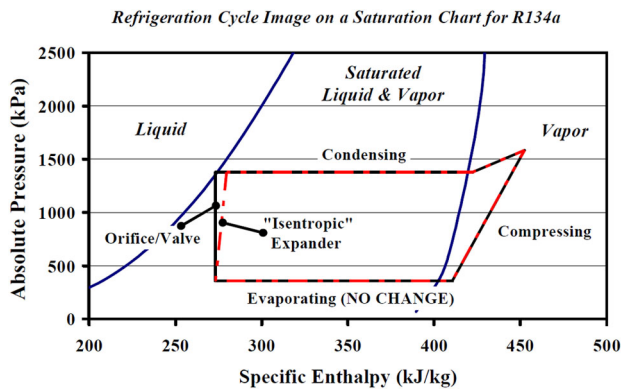


Chart 1.0. A/C Cycle overlay of Expansion Valve and Expander/Motor

If one assumes an ideal expander/motor, a refrigerant charge reduction shown in Chart 2.0 results in a potential compression power recovery of 15%, while keeping the same refrigeration effect and A/C cooling function.

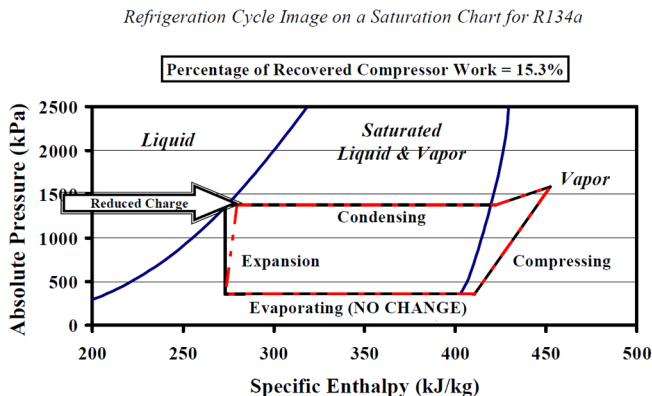


Chart 2.0. Expander System Enables a Reduction of Refrigerant

Power recovery is largely derived from the vapor phase refrigerant. A low charge condition contains resident high pressure vapor as it exits the condenser. This combined with the liquid-to-vapor transition potential (cavitation) augments the power recovery. Chart 3.0 provides an A/C cycle image of a low charge condition whose compression power recovery percentage is increased due to this effect.

Table 1.0 contains a summary of the important performance parameters of the two approaches. This includes

an expanded refrigeration effect with an enhanced COP with the expander/motor.

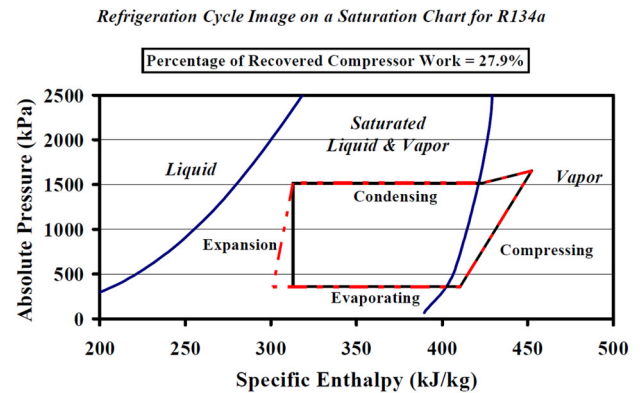


Chart 3.0. Reduced Refrigerant Charge Cycle

Table 1.0. Reduced Refrigerant Charge System

	Expansion Valve System	Expander/Motor System
Specific Refrigeration Effect (kJ/kg)	97.7	109.4
Specific Compression Work (kJ/kg)	41.9	41.9
Specific Recovered Work (kJ/kg)	0.0	11.7
Net Specific Work (kJ/kg)	41.9	30.2
Coefficient of Performance (COP)	2.3	3.6

When the cycle of Chart 3.0 is exposed to a lower ambient temperature, a reduction in condensing pressure occurs. Thus, the pressure differential across the expander is lowered and the relative vapor mass that participates in the expansion is reduced. Chart 4.0 shows an example of how a lower percentage of energy recovery with ~800 kPa condensing pressure when compared to the cycle of Chart 3.0 having a condensing pressure of ~1500 kPa.

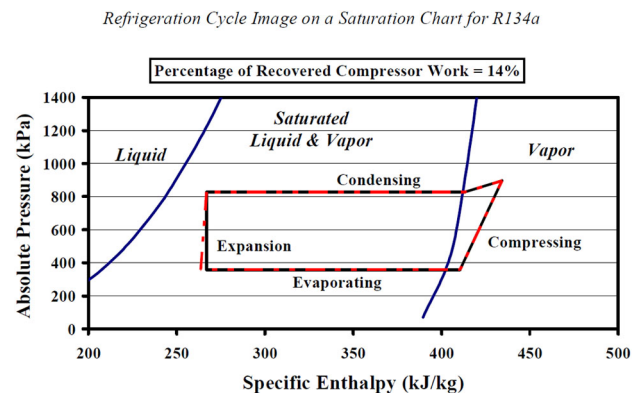


Chart 4.0. Reduced Refrigerant Charge & Low Heat Load

PHASE CHANGE BEHAVIOR

Although the physical location of the expander may be fixed, the “location” of the expander in the A/C cycle is not. [Chart 5.0](#) illustrates this characteristic. The true cycle “location” depends on the relative ability of the compressor to empty the low-side when compared to the motor's ability process refrigerant. If the compressor is aggressively emptying the low-side, the cavitation may occur prior to the motor inlet (post-cavitation position). If the compressor is having difficulty clearing the low-side, the high pressure refrigerant passes through the motor and a pre-cavitation motor position exists. In either of these cases (1 and 3 in [Chart 5.0](#)), the motor does not provide the intended function and cannot experience the “explosion” of liquid into vapor during cavitation - foregoing the opportunity for significant power recovery.

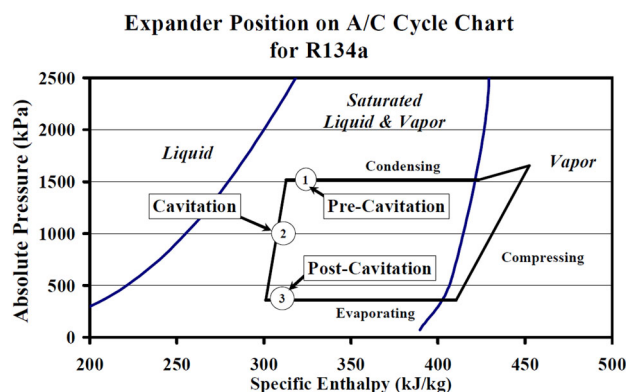


Chart 5.0. Expander Location Issue

This behavior was dramatically demonstrated with a simple vane type motor that was slowly turning in response to a low compressor speed while the motor was “flooded” with predominantly saturated liquid (position 1). The compressor speed was significantly increased, which rapidly reduced the motor's saturated liquid content and temperature (position 3). In that state, the motor continued to turn very slowly. When the compressor was allowed to return its slower speed, the high cavitation region met the motor (position 2 in [Chart 5.0](#) and [Figure 2.0](#)) as the “liquid region” migrated toward the motor. For a several seconds, the motor's very high rotational speed, noise and power persisted until the liquid again filled the motor - restoring its original equilibrium state.

Refrigerant Phase Transition Image

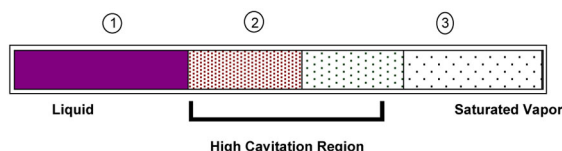


Figure 2.0. Phase Change Image

REFRIGERANT FLOW CONTROL

An initial goal for high efficiency is to match the volumetric change of the expansion chamber to the expected volumetric change of the refrigerant as it moves from high to low pressure. Thus, an ideal match of pressure drop can be achieved. In practice, a fixed expander inlet volume must receive a precise amount of high pressure refrigerant with a design intent density at a rate that corresponds to the expander's processing rate (speed). Flow metering devices tend to have a significant pressure drop and a response time that will limit the expander's speed and ultimately mass flow. The device's pressure drop represents a lost opportunity to recover work and compromises the planned refrigerant density by a phase change. In an automotive application, the mass flow demand will vary, so the expander speed is required to vary according to the need. This translates into an expander loading control mechanism with complex requirements.

For the above reasons, a prototype was investigated that did not contain flow metering device. The volumetric ratio was selected to allow for a positive net pressure at the end of the “stroke” relative to the target evaporator pressure. The intent is to avoid suction losses particularly when higher expander speeds result in under filling of the inlet volume. Unfortunately, the control of the expander's rotational speed was not effective in regulating mass flow or controlling the “position” of the expander in the cycle. This was due to internal expander blow-by leakage, which mimics the conventional orifice arrangement. This characteristic undermines attempts to control the evaporator at an intended temperature and pressure to support the conventional A/C cooling function.

From a power recovery perspective: If the expander speed is high relative to the compressor's ability to reduce the low side pressure, the expander acts as a pump removing refrigerant from the high-side followed by over-expansion en route to the low-side (consuming power). If the expander's rotational speed is slow, relative to the compressor being able to pump down the low-side, the expander can easily be positioned in the post-cavitation position of [Chart 5.0](#). For these reasons, another means to control the refrigerant flow was deemed necessary; and led to the structure of [Figure 3.0](#) [2]. As shown, a low pressure drop valve is inserted at the inlet of the expander/motor: