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MECHANICAL FAILURE DIAGNOSIS IN AUTOMOTIVE AIR CONDITIONING SYSTEMS THROUGH THERMAL MEASUREMENTS

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Abstract: The present paper describes a simulation model for the operation of an automotive air conditioning system subjected to typical mechanical failures. A review identified the most common mechanical failures in automotive A/C systems. Simple mathematical models of two of the most common of these failures were developed and introduced in a simulation model of the vapor compression cycle. Simulated components of the cycle, operating trouble-free or under failure, included the compressor and the condenser. The evaporator, thermal expansion valve, filter-dryer, hoses and connections are also sources of failures and mal-functions. The modeling effort was employed to relate temperature and pressure field measurements with typical system failures. Uncertainties of these measurements and their effect in the predictions of the most probable system failures were studied with the simulation model, aiming at the standardization of mobile A/C system diagnosis procedures.

Keywords: diagnosis, automotive air conditioning, standardization.

1. INTRODUCTION

Automotive air conditioning systems have an important role on car safety (defogging) and comfort [1]. They find widespread use, notably in the American and tropical climate markets. Their environmental impact can be reduced by the use of environmentally benign refrigerants, improved energetic efficiency and proper maintenance procedures. Concerning refrigerants, CFC12 has been replaced, since 1995, by HFC134a which, in spite of being ozone-friendly, presents an appreciable global warming impact. Other refrigerants, like CO_2 and refrigerant mixtures, have been considered for replacement in the near future [2, 3].

The predominantly used refrigeration cycle in motor-vehicle air conditioning systems is the engine-driven vapor compression cycle – see, for example, reference [4]. Maintenance is of crucial importance for the proper operation of such systems. Fault and performance diagnosis is mostly done with the help of basic instrumentation, which comprises two pressure manometers, for condensing and evaporating pressures, and two temperature sensors, for ambient and supply air temperatures. The basic combination of these four readings provides the technician with the information for a fault and performance diagnosis. Note that not even ambient air humidity is measured, in spite of its effect on system performance [5].

2. PURPOSE

A mathematical model, based on fundamental mass and energy conservation equations, was developed to simulate the performance of the vapor compression cycle of a HFC134a automotive air conditioning system running on trouble-free and faulty conditions. The four temperature and pressure readings, with their uncertainties, are part of the resulting system of non-linear algebraic equations that describe the operation of the cycle, operating under realistic conditions. A relation between component or system fault and instrumentation readings is to be established, thus providing a simple standardized procedure for automotive air-conditioning system maintenance metrology.

3. METHOD

2.1. System components

The vapor compression cycle of an automotive air conditioning system is composed, figure 1, by the compressor, condenser, expansion device, fans, controls, filter-dryer, hoses and connections. The compressor is of the positive displacement reciprocating multi-piston wobble-plate type. Condenser is, of course, air-cooled and is placed ahead of the engine radiator. The evaporator provides cool air to the cabin. The expansion device can be a thermal expansion valve (TXV) or a fixed area orifice.

2.2. Expected system failures

Typical system failures include [6]: ice formation on evaporator surface, faulty driving clutch, slippery compressor driving belt, obstructed filter-dryer, obstructed expansion device, ice formation on expansion device, defective thermostat control, excess fouling on heat exchangers, air in system, excessive or insufficient refrigerant charge, leaking compressor seals, refrigerant leakage, refrigerant-side fouling, to name but a few. All these failures reflect on the measured values of the condensing and evaporating pressures as well as the temperature of the vehicle cabin, T_{cab}, and of the cold air supplied to it, T_{sup}.

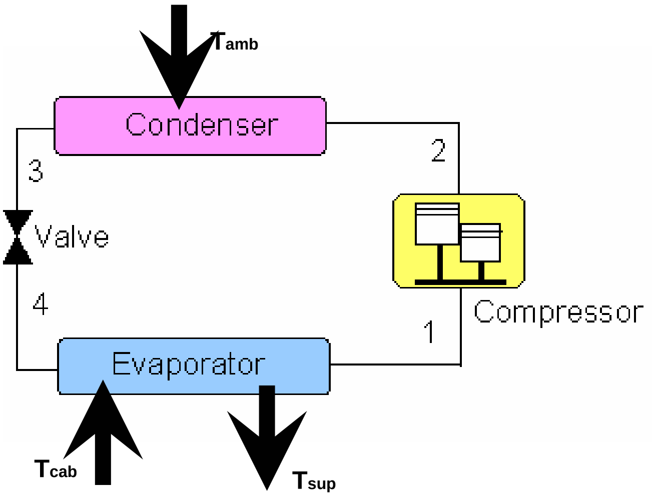

Fig. 1. Vapor compression refrigeration cycle

2.3. Simulation model

Fundamental energy balance conservation and system characteristic equations are applied to each component [7]. Polytropic compression is considered, and the compressor volumetric efficiency is dependent on the clearance volume, internal leakage and valve performance. Both heat exchangers are simulated based on the effectiveness lumped parameter model.

The compression process is assumed to be polytropic, so that:

$$P_{cd}\, v_2^n = P_{ev}\, v_1^n \tag{1}$$

where P_{cd} and P_{ev} are the condensing and evaporating pressures, n, the polytropic exponent, and v_1 and v_2, the specific volumes at compressor suction and delivery, respectively.

The compressor is of the reciprocating positive displacement type. The mass flow rate is, then, given by:

$$\dot{m} = \frac{V_c}{v_1} N \left\{ 1 - r \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\} C_v \tag{2}$$

where $\dot{m}$ is the refrigerant mass flow rate, V_c, the displaced volume, N, the shaft speed, r, the clearance ratio. The term between brackets in equation (2) is the compressor volumetric efficiency due to the re-expansion of the residual gas in the clearance volume and C_v is a volumetric coefficient that takes into account other losses such as from valves and internal gas leakages.

The energy balance over the refrigerant stream in the condenser provides:

$$\dot{Q}_{cd} = \dot{m}\left(h_2 - h_3\right) \tag{3}$$

where $\dot{Q}_{cd}$ is the rate of heat transfer in the condenser and h_2 and h_3 are the specific enthalpies of the refrigerant at the entrance and exit of the condenser, respectively. The heat transfer rate equation, based on the effectiveness method, is:

$$\dot{Q}_{cd} = \varepsilon_{cd}\, \dot{m}_{a,cd}\, c_{p,air} \left(T_{cd} - T_{amb}\right) \tag{4}$$

where ε_{cd} is the effectiveness of the condenser, $\dot{m}_{a,cd}$, the air mass flow rate across the condenser, $c_{p,air}$, the moist air specific heat at constant pressure, T_{cd}, the condensing temperature and T_{amb}, the external ambient temperature. Finally, it is assumed that the refrigerant charge is such that the degree of sub-cooling in the condenser exit is known.

$$\Delta T_{sc} = T_{cd} - T_3 \tag{5}$$

A thermostatic expansion valve is used so that it can be modeled by its function, i.e, constant degree of superheat, ΔT_{sh}, at evaporator exit.

$$T_1 = T_{ev} + \Delta T_{sh} \tag{6}$$

where T_{ev} is the evaporating temperature and T_1, the refrigerant temperature at evaporator exit. With no work done and neglecting the amount of heat transferred from the valve, one has, from the energy balance over the valve:

$$h_3 = h_4 \tag{7}$$

where h_3 and h_4 are the specific enthalpies upstream and downstream the valve.

The energy balances on the refrigerant and air sides of the evaporator are:

$$\dot{Q}_{ev} = \dot{m}\left(h_1 - h_4\right) \tag{8}$$

$$\dot{Q}_{ev} = \dot{m}_{a,ev}\left(h_{cab} - h_{sup}\right) \tag{9}$$

where $\dot{Q}_{ev}$ is the rate of heat transfer, h_1 and h_4, the refrigerant specific enthalpies at evaporator exit and entrance, $\dot{m}_{a,ev}$, the air mass flow rate across the evaporator, and h_{cab} and h_{sup}, the moist air specific enthalpies at return (cabin) and supply (leaving the evaporator) conditions, respectively. The effectiveness of the evaporator, ε_{ev}, is approximated to the following temperature difference relation:

$$\varepsilon_{ev} = \frac{T_{cab} - T_{sup}}{T_{cab} - T_{ev}} \tag{10}$$

Equation (10), involving cabin, evaporating and supply-air temperatures, T_{cab}, T_{ev} and T_{sup}, respectively, assumes, only for effectiveness evaluation purposes, that no latent heat is exchanged in the air side of the evaporator. The energy balance on the air side, equation (9), is correct, nonetheless.

Finally, thermodynamic properties of humid air and refrigerant were taken from built-in functions of the software employed for the solution. These functions, with corresponding arguments for property determination, are described next.

$$P_{ev} = P_{sat}\left(T_{ev}\right) \tag{11}$$

$$P_{cd} = P_{sat}\left(T_{cd}\right) \tag{12}$$

$$h_1 = h(T_1, P_{ev}) \tag{13}$$

$$v_1 = v(T_1, P_{ev}) \tag{14}$$

$$v_2 = v(s_2, P_{cd}) \tag{15}$$

$$h_2 = h(s_2, P_{cd}) \tag{16}$$

$$h_3 = h(T_3, P_{cd}) \tag{17}$$

$$c_{p,air} = c_p(P_{atm}, T_{amb}, \phi_{amb}) \tag{18}$$

$$h_{cab} = h_{air}(T_{cab}, P_{atm}, \phi_{cab}) \tag{19}$$

$$h_{sup} = h_{air}(T_{sup}, P_{atm}, \phi_{sup}) \tag{20}$$

where P_{sat}, h, v, c_p and h_{air} are property functions and ϕ, the moist air relative humidity.

2.4. Failure and component mal-function simulation

Two of the listed mal-functions were simulated in the present work: dirty condenser and faulty compressor. The translation of the failures into a mathematical describable process is discussed next.

The presence of excessive fouling (bugs, mud, corrosion, debris, etc.) on the air-side of the condenser can be described, in the mathematical model developed above, as a reduced value of the condenser effectiveness, ε_{cd}. Likewise, a faulty compressor, with internal leakage, operates with reduced volumetric efficiency, i.e., reduced values of C_v, thus resulting in lower system efficiency.

2.5. Solution

The resulting non-linear system of algebraic equations was solved with using the EES® software.

3. RESULTS AND CONCLUSIONS

Before being used in the present analysis, the simulation model was validated against existing automotive air-conditioning systems practice. Runs were carried out with typical input data and results and tendencies were compared with those expected from practice, with good agreement.

A typical medium size automotive air conditioning system was simulated. It comprised a fixed angle wobble plate 6-cylinder positive displacement compressor, a thermostatically expansion valve with external bulb, a fan-cooled condenser, a dry-expansion evaporator and a high pressure line filter dryer.

The model was applied to simulate the system operating under different conditions, faulty and trouble-free. A sensitivity analysis was carried out taking into account the measurements uncertainties and the response of the system.

The following input data were applied:
$P_{atm} = 100\,kPa$; $n = 1{,}1$; $r = 0{,}0336$; $V_c = 147\,cm^3$;
$C_v = 0{,}75$; $N = 3000\,rpm$; $\Delta T_{sc} = 2\,^oC$; $\Delta T_{sh} = 15\,^oC$;
$\dot{m}_{a,cd} = 0{,}9\,kg.s^{-1}$; $\varepsilon_{cd} = 0{,}85$; $T_{amb} = 30$ to $50\,^oC$;
$\dot{m}_{a,ev} = 0{,}8\,kg.s^{-1}$; $\varepsilon_{ev} = 0{,}8$; $T_{cab} = 20\,^oC$.

To simulate the condenser with excessive fouling, the condenser effectiveness was reduced from 0,85 to 0,4, for ambient temperatures of 30, 40 and 50 °C. Figures 2 and 3 show how the condensing and evaporating pressures are affected by the condenser effectiveness. As expected, the condensing pressure is the most affected of the two. Also, higher ambient (i.e., external air) temperatures lead to higher condensing and evaporating temperatures. Note that the condensing pressure is affected significantly by the ambient air temperature.

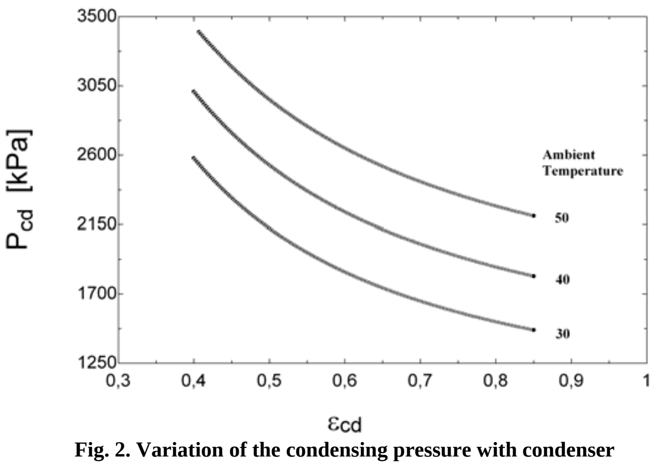

Fig. 2. Variation of the condensing pressure with condenser effectiveness and ambient temperature

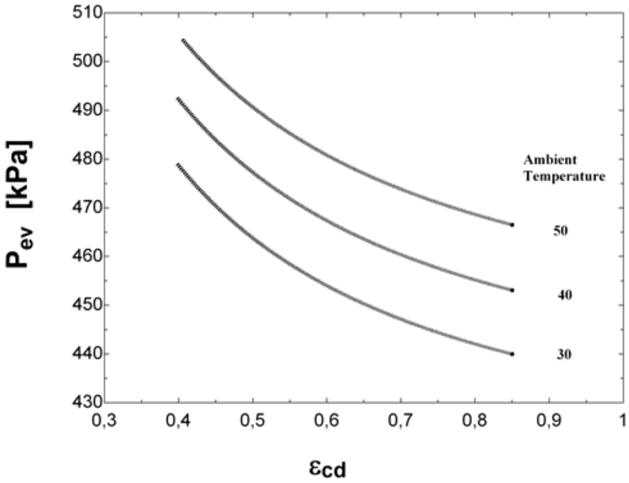

Fig. 3. Variation of the evaporating pressure with condenser effectiveness and ambient temperature

Likewise, the compressor volumetric coefficient, C_v, was reduced from 0,75 to 0,15, for ambient temperatures of 30, 40 and 50 ^oC, with the latter value indicating severe internal leakage. Figures 4 and 5 show the variation of the condensing and evaporating pressures with the compressor volumetric efficiency. The expected trends were observed, namely, that the system high pressure decreases and the low pressure increases, both approaching a middle value.

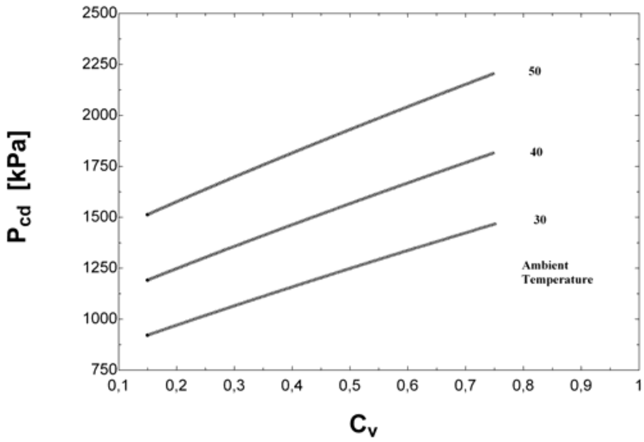

Fig. 4. Variation of the condensing pressure with compressor volumetric coefficient and ambient temperature

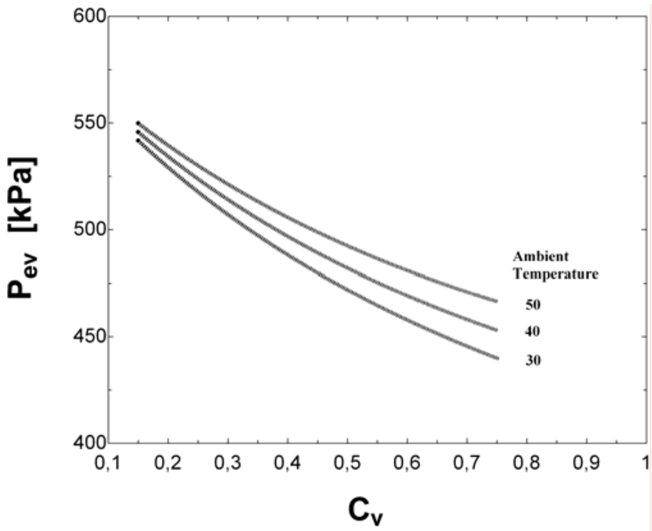

Fig. 5. Variation of the evaporating pressure with compressor volumetric coefficient and ambient temperature

In practice, technicians employ manometers of the bourbon type with uncertainties of approximately 2%. On top of that, figures 2 to 5 have shown that the ambient temperature has a significant role in altering both measured high and low pressures. Observation of figures 2 to 5, though, generated for a typical automotive air conditioning system, leads to the conclusion that severe penalties on either the condenser or compressor may still be detected in spite of the instrumentation uncertainty and of the ambient temperature influence.

On the other hand, subtle damages (small reductions on condenser effectiveness or on compressor volumetric efficiency) may pass unnoticed. In such cases, additional temperature measurements may prove useful.

4. CLOSURE

The instrumentation used in the diagnosis process of automotive air conditioning systems is often limited by a number of factors, including accessibility, time, budget, and others. A simulation model such as the one here described can provide important clues to the understanding of the system behavior, even if measured through limited instrumentation or under the influence of uncontrolled factors, such as the ambient temperature. The methodology here presented can become a useful tool for the diagnosis of automotive air conditioning systems.

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