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International Compressor Engineering Conference
School of Mechanical Engineering

2002

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EXTERNAL AND INTERNAL CONTROL COMPRESSORS FOR MOBILE AIR-CONDITIONING SYSTEMS

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ABSTRACT

During the last few years, automotive air conditioning suppliers have spent many efforts to reduce the energy consumption of Mobile Air-Conditioning (MAC) systems. External control compressors have been launched on the market and permit to develop smart control systems integrating better in-vehicle temperature control and lower energy consumption of the MAC system.

The paper presents the main differences between internal and external control compressors, based on an experimental study analysing their volumetric, isentropic and global efficiencies referred to the compressor rotating speed and to the compression ratio. For external control compressors, thousands measures are necessary to analyse these efficiencies for different swept volumes. Based on these characteristics, a control algorithm has been developed for the external control compressor taking into account the fast variations of the engine speed of typical urban driving cycles.

A comparison is performed on 2 identical MAC systems installed on a test bench, one using an external control compressor and the other an internal control compressor of the same swept volume. Improvements of energy efficiency based on the control system itself are shown on the typical ECE+EUDC cycle, which is the regulatory cycle used in Europe to measure fuel consumption of cars. Conclusions are drawn on possible energy gains associated to the control.

NOMENCLATURE

C: torque (N.m)
h: enthalpy (kJ/kg)
$\dot{m}$: mass flow rate (kg /s)
N: rotation speed (RPM)
P: power (W)
Q_0: maximum compressor swept volume (m^3)

Subscripts

vol: volumetric
is: isentropic
suc: suction
dis: discharge
mec: mechanical

Greek letters

τ: compression ratio (-)
ρ: density (kg/m^3)

INTRODUCTION

The air conditioning systems are installed in more than 90% of the cars sold in North America and Japan. The same level will be reached in Europe around 2005.

Different studies show that energy consumption of mobile air-conditioning systems is significant. The global impact of fuel consumption due to the AC system needs to be carefully measured. First studies indicate that the additional fuel consumption is in the range of 8% for extra urban conditions and in the range of 15% in the urban ones [BAR98]. A recent study performed by the National Renewable Energy Laboratory [FAR02] estimates the US fuel consumption of MAC systems to be in the range of 7 billions gallons (27 billions liters).

It is a crucial issue to know if energy consumption of MAC systems can be dramatically reduce or not in order to develop low fuel consumption cars, and so the base line of energy consumption of MAC shall be established. This paper uses the European regulatory cycle as a reference for the energy consumption of MAC systems. In parallel, the impact of external control compressor is carefully analyzed.

1. TEST BENCH AND ANALYZED COMPRESSORS

1.1 Test bench

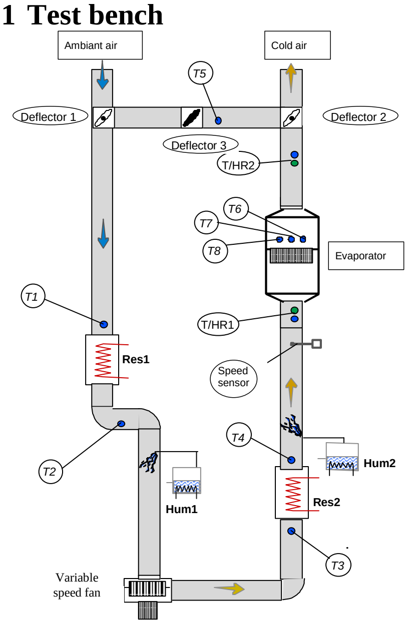

Figure 1 - Evaporator wind tunnel.

The test bench is composed of 2 wind tunnels, one for the evaporator, the other one for the condenser. The evaporator wind tunnel permits to control humidity and temperature from $2g_{water}/kg_{dry\ air}$ to $40g_{water}/kg_{dry\ air}$ and temperatures from 20 to 50°C. The evaporator wind tunnel (see Figure 1) is also designed in order to permit recirculation of air from 0 to 100%. The wind tunnel for the condenser controls the temperature and the air velocity in order to simulate the car speed variations and climatic conditions.

The compressor is driven by a direct current variable speed electrical motor to simulate any urban or extra urban cycle. The temperatures, the air velocity, the electric motor speed are controlled by a computer, and all the temperatures, pressures, and mass flow rates are recorded on the same computer with a 1-second time step.

The test bench is designed to permit rapid change of heat exchangers, compressors and expansion devices.

The compressor mechanical power consumption P_{mec} is measured using a torquemeter and a speedmeter, and calculated as follows:

$$P_{mec} = \frac{2.\pi.C.N}{60} \quad (1)$$

1.2 Compressor Control System

Two compressors of the same technology, swash plate compressors with 6 pistons and a swept volume of 120 cm^3, and of the same brand name are compared, one using a usual internal control by a mechanical valve and the other one an external control performed by a solenoid valve. The solenoid valve controls the compressor swept volume from 0 to 100%. The tension variation of the solenoid valve permits a fine tuning of the refrigerant mass flow rate according to the cooling needs of the vehicle. This solenoid valve is able to manage the fast variation of the engine rotation speed in order to avoid over or under refrigerating capacity.

1.3 Fast Temperature Cabin Control

As Miyagawa and alii show [MIY98], a number of energy gains are achievable by external control. A first energy gain is to limit reheating. Usual control of air blown temperature in the car is performed by reheating when the air temperature at the evaporator outlet becomes too low. This usual control is set up to limit discomfort due to blown air at too low temperature. External control permits very fast answer of the compressor that can adapt the necessary swept volume whatever the engine speed variations. This quick adaptation of refrigerant mass flow rate limits also the required power of the compressor and so permits a higher energy efficiency as it will be demonstrated.

2. COMPRESSOR CHARACTERIZATION

The same series of tests have been performed for both compressors with the following conditions:

- the compressor rotation speed is varied from 800 to 4400 rpm with a 400 rpm step,
- for each speed step, 3 air flow rates are fixed for the evaporator (200, 350 and 500 m^3/h),
- for each step of air flow rate at the evaporator, and rotation speed of the compressor, 8 air flow rates are fixed on the condenser wind tunnel from 100 to 3500 m^3/h. Each elementary step lasts 40s to reach steady state conditions, which makes 216 tests for each temperature level.

All these tests have been performed at 4 air temperatures levels : 25°C, 35°C, 45°C and a combination 25°C at the evaporator and 40°C at the condenser (to simulate a re-circulation mode). 864 elementary tests have been performed for the internal control compressor.

For the external control compressor, tests shall be performed from 2V to 10V with 1V step, making 6912 elementary tests. All those tests permit the mapping of volumetric, isentropic and global efficiencies as defined in the following equations.

- Volumetric efficiency $$\eta_{Vol} = \frac{\dot{m}}{Q_0.\rho.\frac{N}{60}} \qquad (2)$$
- Isentropic efficiency $$\eta_{is} = \frac{h_{is} - h_{Suc}}{h_{Dis} - h_{Suc}} \qquad (3)$$
- Mechanical efficiency $$\eta_{Mec} = \frac{\dot{m}.(h_{Dis} - h_{Suc})}{2.\pi.C} \qquad (4)$$

All efficiencies are dependent on the rotation speed and the compression ratio and can be written as follows

$$\eta = (a.\tau + b).(c.N + d) = a_1.\tau.N + a_2.\tau + a_3.N + a_4 \qquad (5)$$

2.1 Volumetric Efficiency Comparisons

As it can be seen on figures 2 and 3 (external control compressor) and figure 4 (internal control compressor), the volumetric efficiencies are low for both compressors. They are strongly dependent on the compression ratio and the rotation speed, the lower the compression ratio, the lower the efficiency and the higher the rotation speed, the lower the efficiency. Moreover, for the external control compressor, the lower the tension, the lower the efficiency. Those mappings indicate that the compressor designers limit the volumetric efficiency at high rotation speed in order to avoid too high refrigerating capacity.

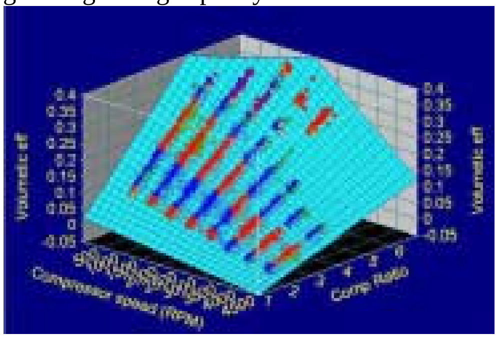

Figure 2 – Volumetric efficiency variation (4V).

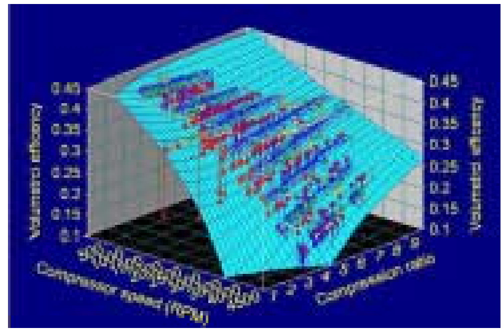

Figure 3 – Volumetric efficiency variation (8V).

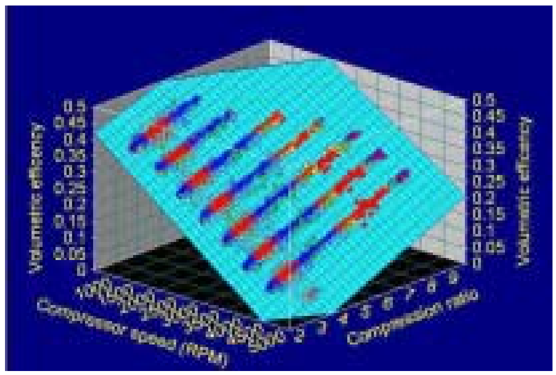

Figure 4 - Volumetric efficiency variation (IC).

2.2 Isentropic Efficiency Comparisons

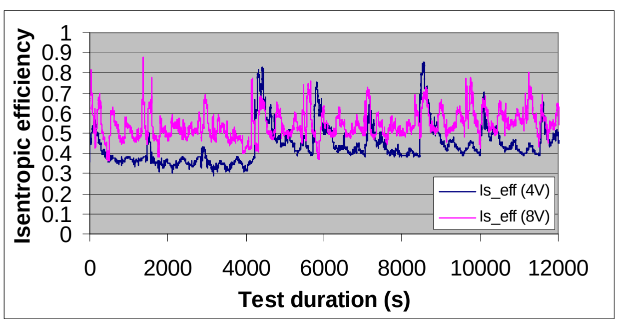

Figure 5 – Is. efficiency of external control compressor.

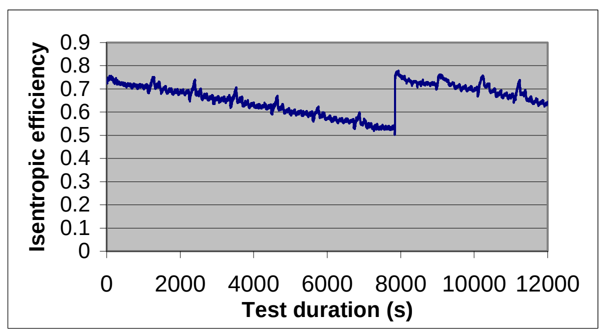

Figure 6 – Is. efficiency of internal control compressor.

Figure 5 and 6 present series of tests where rotation speed is varied for a given level of air temperature and then the air temperature is changed as it can be seen on the step of Figure 6. For external control compressor, as indicated in Figure 5, tests are performed for different tensions of the solenoid valve. Analysis of the tests indicate that the lower the tension, the lower the isentropic efficiency. A detailed analysis of Figure 6 indicates that isentropic efficiency varies from 0.72 to 0.52 depending on the speed, which makes a variation of more than 38% when the engine speed increases. For the external control compressor, variations are slightly lower, from 0.7 to 0.45.

2.3 Mechanical Efficiency Comparison

Figures 7 and 8 indicate that

- for a 8V tension of the solenoid valve, the mechanical efficiency of the external control compressor is nearly identical to the internal control one,
- both mechanical efficiencies are significantly low compared to stationary AC compressors where efficiencies vary from 0.7 to 0.9,
- for low tension of the solenoid valve, the mechanical efficiency of the external control compressor is lower.

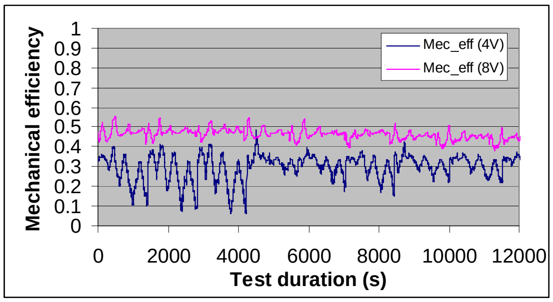

Figure 7 – Mech. efficiency of ext. control compressor.

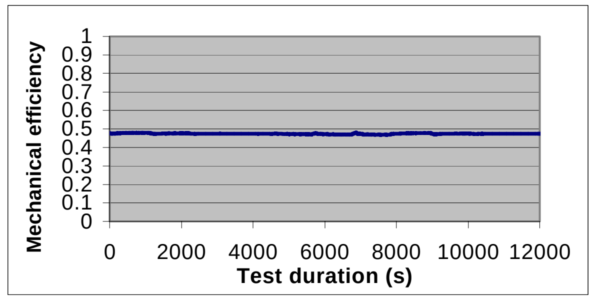

Figure 8 – Mech. efficiency of int. control compressor.

Those tests confirm that both compressors present the same level of efficiencies confirming that the mechanical design is identical. Low tension of the external control compressor implies strictly speaking lower efficiencies. Possible energy gains due to external compressor are not related to the mechanical performances.

3. REGULATORY CYCLE

A number of regulatory cycles exists world-wide (in the US SFTP US 06 driving cycle, in Japan (10.15 mode, 11 mode), and in Europe where the typical regulatory cycle for measurement of fuel consumption of vehicles is the MVEG-B as presented in Figure 9.

The European MVEG regulatory cycle is composed of 4 successive urban cycles (200sec each) with fast engine speed variations and 1 extra urban cycle (380sec), the total duration is about 20min. As indicated on figures 9 and 10, those quick speed variations imply fast variations of the compressor rotation speed. The typical variations are of more than 250% compared to the reference line, making the same variations of the refrigerant mass flow rates whatever the cooling needs. Those constraints indicate clearly that when the thermal loads are in the medium or in the low range, the AC system is significantly oversized and explain why reheating is the traditional answer to refrigerating overcapacity.

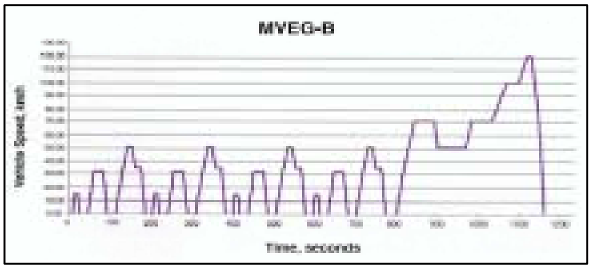

Figure 9 – MVEG cycle fuel consumption.

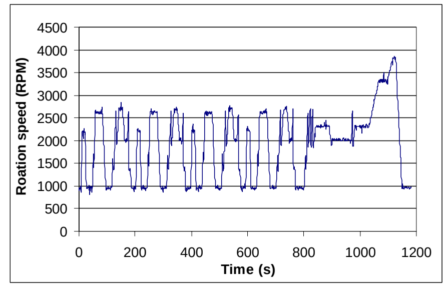

Figure 10 – Evolution of the rotation speed.

4. CONTROL SYSTEM SPECIFICATIONS

The control system shall take into account the following input parameters:

- outside air temperature $T_{ambient\_air}$ and humidity ($RH_{air\_inlet\_evap}$),
- evaporator inlet air temperature $T_{air\_inlet\_evap}$ (which is different from outside air temperature when recirculation is set up),
- car speed and engine rotation speed (engine speed directly impacts on the belt driven compressor and car speed implies different air flow rates on the condenser and partly on the evaporator),
- air mass flow rate blown in the cabin (Q_{air_evap}); this airflow rate is defined by temperature and humidity conditions in the vehicle.

The control parameter is the blowing temperature in the in-vehicle, and the output of the control system permitting to control the air blowing temperatures are:

- the tension of the solenoid valve of the external control compressor,
- the speed of the electrical motor of the condenser fan.

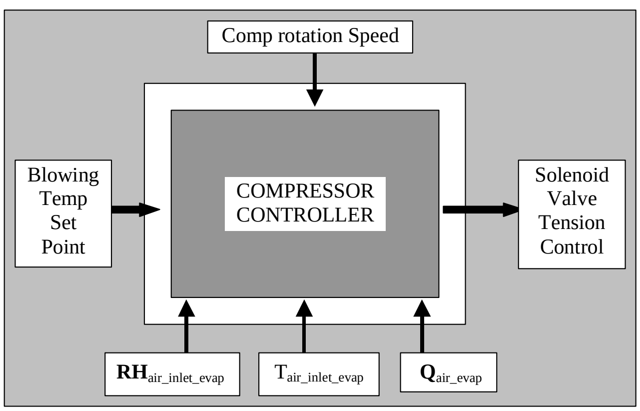

Figure 11 –Solenoid valve control algorithm.

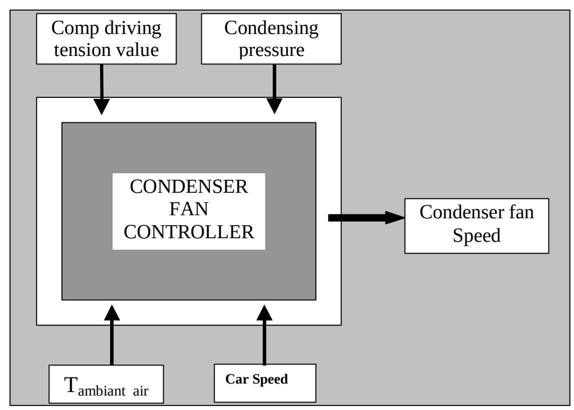

Figure 12 – Condenser fan control algorithm.

The algorithms shall send orders to solenoid valve (Figure 11) and fan motor (Figure 12) every second in order to quickly adapt the refrigerant mass flow rate and the airflow rate on the condenser to the:

- driving conditions (car speed and engine rpm)
- cooling needs of the in-vehicle (related to $T_{air\_inlet\_evap}$ and Q_{air_evap}).

The control algorithms are based on a model of the AC system integrating evaporator and condenser geometries in order to determine what are the needed refrigerant mass flow rates adapted to the in-vehicle cooling needs. The model integrates also the compressor characteristics (volumetric and isentropic efficiencies).

5. TESTING CONDITIONS AND COMPARISON OF INTERNAL AND EXTERNAL CONTROL COMPRESSOR AC LOOPS

5.1 Test Conditions

To compare the performances of internal and external control compressors on MVEG cycle, 4 tests have been performed on the same AC loop using successively each compressor technology with two outside air temperatures (30 and 35°C) and two airflow rates at the evaporator (175 and 420m^3/h) with adapted set point temperatures (in-vehicle air blown) as indicated in Table 1.

Table 1 – Test conditions.

Tests	Air temperature (°C)	Airflow rate (m^3/h)	Set point temperature (°C)
1	30	175	7
2	30	420	12
3	35	175	8
4	35	420	14

Mechanical energy consumption is measured every second by torquemeter and speedmeter. The refrigerating capacity is measured twice

- one by air flow measurement inlet and outlet temperatures and hygrometries, and
- by pressure, temperature and mass flow rate of the refrigerant (R-134a),

in order to verify the measurement.

5.2 Test Results

Instantaneous refrigerating capacity and mechanical power are integrated along the complete MVEG cycle in order to evaluate the average refrigerating capacity on the cycle and the average mechanical power consumption, permitting to calculate the average COP of the two AC loops (internal and external control) for the 4 testing conditions mentioned previously.

As indicated in table 2 the air blowing temperature is maintained in the same range and figure 13 indicates that external control compressor permits better air blown temperature control with much smaller variation. One of the consequence is that the average evaporating temperature is slightly higher compared to the one of internal control compressor, and by the way the COP is improved. Secondly, it can be verified that the cooling capacity of the external control compressor AC loop is equal or slightly lower compared to the internal control compressor AC loop. The mechanical power of the external control compressor AC loop is significantly and systematically lower.

Table 2 – Test results.

		Compressor control		
		Internal	External	Gain (%)
Test no. 1	Mechanical power (kW)	1.83	1.38	25
	Cooling capacity (kW)	2.15	2.00	-7
	COP	1.17	1.45	23
	Blowing temperature (°C)	6.5	7.09	
Test no. 2	Mechanical power (kW)	2.95	1.88	36
	Cooling capacity (kW)	3.2	3.05	-5
	COP	1.08	1.62	50
	Blowing temperature (°C)	11.5	12.07	
Test no. 3	Mechanical power (kW)	2.12	1.58	25
	Cooling capacity (kW)	2.3	2.2	-4
	COP	1.08	1.39	28
	Blowing temperature (°C)	7.8	8.08	
Test no. 4	Mechanical power (kW)	3.3	2.11	36
	Cooling capacity (kW)	3.23	3.24	0
	COP	0.98	1.54	57
	Blowing temperature (°C)	14.15	14.11	

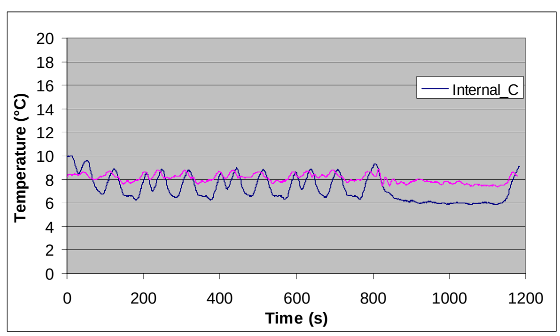

Figure 13 – Blowing air temperature variation.

Figure 14 indicates that when the engine speed variation is high, the internal compressor requires a much higher torque compared to the external control compressor AC loop. Taking into account that both compressors are identical in terms of mechanical performances, the energy gain is clearly associated to the refrigerant mass flow rate management.

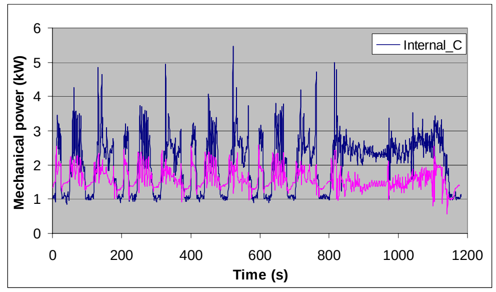

Figure 14 – Mechanical power variation.

It can be noticed that for high air flow rates on the evaporator, the energy gains are significantly higher (36% mechanical power reduction for $420m^3/h$ compared to 25% obtained for $175m^3/h$).

CONCLUSIONS

A first series of conclusions addresses the necessary reference for measurement of energy consumption of MAC systems.

Due to the targets of limitation of fuel consumption and limitation of CO_2 emissions of cars, a number of improvements are required for mobile air-conditioning systems. Regulatory cycles for fuel consumption of cars shall be used in order to test the dynamic behavior of the AC loop. Other issues shall be addressed specially to take into account progresses that are actually made to limit heat loads in the vehicle. Independently of this global analysis of energy consumption due to mobile air-conditioning system, it is possible to analyze the energy consumption of the AC loop itself for different thermal loads and for typical driving conditions. The results that have been presented indicate that tests performed under steady state conditions permit only to know volumetric, isentropic and global efficiencies of compressor but, do not permit to forecast energy consumption of AC loops under realistic conditions. Dynamic driving conditions shall be used to analyze and compare energy performances of AC loops taking into account those driving conditions. Energy gains due to smart control of compressors permit to reach significant progress in terms of energy efficiency.

Tests on the European MVEG cycle show that for external control compressors, energy efficiency improvements are significant :

- COP can be increased between 30 and 60% depending on the test conditions,
- the mechanical power can be lowered by 25 to 40%.

Nevertheless, volumetric and mechanical efficiencies are low whatever the type of compressors, and a large potential of improvements exists.

ACKNOWLEDGEMENTS

The work has been supported by the French Agency for Environment and Energy Management (ADEME) and the Fiat Research Center (CRF).

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