SAE International™ SURFACE VEHICLE STANDARD

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Procedure for Measuring System COP [Coefficient of Performance] of a Mobile Air Conditioning System on a Test Bench

RATIONALE

The impact of mobile air conditioning systems on the environment is becoming more important. Fuel used to power the system impacts both fuel economy of the vehicle and tail pipe emissions of carbon dioxide.

1. SCOPE

1.1 This Standard applies to motor driven mobile air conditioning systems consisting of one in-car air coil (evaporator), a compressor, an expansion device, and one under-hood air coil (condenser).

1.1.1 This standard can also be used for measuring systems that use electrically driven compressor if measurement of input power to compressor is carefully considered.

1.1.2 This standard can also be used for measuring systems that include the entire air handling system if air side pressure drop for the entire vehicle system is carefully considered.

1.1.3 This standard can also be used for measuring systems that include the entire front end cooling module if air side pressure drop for the entire vehicle system is carefully considered.

1.1.4 This standard can also be used for measuring systems that include a secondary cooling loop if the power to drive the pumps in this system is carefully considered.

1.2 This Standard specifies procedures, apparatus, and instrumentation that will produce accurate steady state capacity and efficiency data for refrigerant components.

1.3 This Standard does not:

1.3.1 Specify tests for dual evaporator systems,

1.3.2 Specify transient test methods,

1.3.3 Make recommendations for safety

1.3.4 Specify tests for production, specification compliance, or field testing of mobile air conditioning systems.

1.4 Purpose

1.4.1 This Standard provides a method of testing the capacity (performance) and efficiency (COP) of mobile air conditioning refrigerant systems under steady state conditions.

1.4.2 The procedure proposed is designed to give maximum repeatability and minimum error in determining cooling capacity (Q) and efficiency (COP – Coefficient of Performance) of the refrigeration system of the mobile air conditioner. For that reason a "breadboard" type facility is proposed in which components are assembled into a system and exposed to operating conditions in individual, separately controlled chambers. The facility should allow for three methods to determine system performance: refrigerant side, airside, and calorimetric chambers. Each side yields its own energy balance calculation to find the capacity of the system. With the three methods used, instead of two as typically required, redundancy will be maintained even when the evaporator refrigerant exit is two phase. In this case only the airside and chamber balances are available. The design should be made so that two independent methods could provide agreement of less than ±4.0% between two independent balances.

2. REFERENCES

2.1 Applicable Publications

The following publications form a part of this specification to the extent specified herein. Unless otherwise indicated, the latest version of publications shall apply.

2.1.1 ASHRAE Publications

Available from American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329-2305, Tel: 404-636-8400, www.ashrae.org.

[1] ANSI/ASHRAE Standard 41.1-1986 (RA 2001)	Standard Method for Temperature Measurement
[2] ANSI/ASHRAE Standard 41.6-1994 (RA 2001)	Standard Method for Measurement of Moist Air Properties
[3] ASHRAE Standard 41.3-1989	Standard Method for Pressure Measurement
[6] ASHRAE Standard 41.2-1987 (RA 92)	Standard Methods for Laboratory Airflow Measurement
[7] ANSI/ASHRAE Standard 51-1999	Laboratory Methods of Testing Fans for Aerodynamic Performance Rating (AMCA Standard 210-99)
[8] ANSI/ASHRAE Standard 37-2005	Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment
[9] ANSI/ASHRAE Standard 41.4-1996	Standard Method for Measurement of Proportion of Lubricant in Liquid Refrigerant

2.1.2 ASME Publication

Available from American Society of Mechanical Engineers, 22 Law Drive, P.O. Box 2900, Fairfield, NJ 07007-2900, Tel: 973-882-1170, www.asme.org.

[4] Fluid Meters—Their Theory and Application, Sixth Edition, 1971

2.1.3 ARI Publication

Available from Air-Conditioning and Refrigeration Institute, 4100 North Fairfax Drive, Suite 200, Arlington, VA 22203, Tel: 703-524-8800, www.ari.org.

[5] ARI Standard 700-2004 Specifications for Fluorocarbon Refrigerants

2.1.4 NIST Publication

Available from National Institute of Standards and Technology, 100 Bureau Drive, Stop 1070, Gaithersburg, MD 20899-1070, Tel: 301-975-6478, www.nist.gov.

[10] NIST Standard Reference Database 23 NIST Thermodynamic and Transport Properties of Refrigerants and Refrigerants and Refrigerant Mixtures—REFPROP 6

3. DEFINITIONS

3.1 Air, Standard

Dry air having a mass density of 1.204 kg/m^3 (0.075 lb/ft^3).

3.2 Apparatus

As used in this standard, this term refers exclusively to test room facilities and instrumentation.

3.3 Capacity, Total Cooling

The rate, expressed in watts (Btu/hr), at which the equipment removes heat from the air passing through it under specified conditions of operation.

3.4 COP, Coefficient of Performance

A measure of the efficiency of an air conditioning system is defined as the average heat removed through the evaporator divided by the average work input to the compressor. Averaging is done using the balances available and/or as described in 7.4.1. This COP is the highest possible because it does not account for air movement, heat transmission or other losses of similar nature.

3.4.1 Adjustments to this calculation may be necessary for electrically driven compressors, in case the air handling system is used, in the case of front end cooling module is used, and for secondary cooling loop systems.

3.5 Equipment

As used in this standard, this term refers exclusively to the mobile air conditioning equipment to be tested.

3.6 Pressure, Standard Barometric

101.325 kPa (14.696 psi).

3.7 Refrigerant

A refrigerant that changes from the liquid to the vapor state in the process of absorbing heat.

3.8 Shall or Shall Not

Is used to indicate a provision that is mandatory if compliance with the standard is claimed.

3.9 Should, Recommended or It Is Recommended

Is used to indicate provisions which are not mandatory but which are desirable as good practice.

4. INSTRUMENTATION

4.1 Temperature Measuring Instruments

4.1.1 All temperature measurements (with the exception of dew point temperature) shall be made in accordance with ANSI/ASHRAE Standard 41.1-1986 (RA 2001).[1]

4.1.2 If used in determining the water vapor content of the air, dew point hygrometers shall be applied as specified in ANSI/ASHRAE Standard 41.6-1994 (RA 2001)[2] and shall be accurate to within ±0.2 °C (0.4 °F).

4.1.3 Inlet air temperature measurements are to be taken upstream of static pressure taps on the inlet duct (if installed). Outlet air temperature measurements shall be taken downstream of the static pressure taps on the outlet. Very often the best practice is to take temperature immediately downstream of the nozzle throat.

4.2 Pressure Measuring Instruments

4.2.1 Pressure measurements shall be made with any device meeting the requirements of ASHRAE 41.3-1989.[3]

4.2.2 The accuracy of pressure measuring instruments shall permit measurement to within ±2.5% of the reading.

4.3 Air Differential Pressure and Airflow Measurements[4]

4.3.1 The static pressure difference across nozzles and velocity pressures at nozzle throats shall be measured with manometers or electronic pressure transducers that have been calibrated against a pressure standard to within ±1.0% of the reading. The resolution of the device shall be equal to or less than 2.0% of the reading.

4.3.2 Areas of nozzles shall be determined by measuring their diameters to within ±0.20% in four places approximately equally spaced around the nozzle in each of two planes through the nozzle throat, one at the outlet and the other in the straight section near the radius.

4.4 Electrical Instruments

4.4.1 Electrical measurements shall be made with indicating or integrating instruments.

4.4.2 Instruments used for measuring the electrical power input to fan motors or other equipment accessories shall be accurate to within ±2.0% of the reading.

4.4.3 Voltages shall be measured at the equipment terminals. Instruments used for measuring voltage shall be accurate to within ±1.0% of the reading.

4.5 Refrigerant Flow Measurement

4.5.1 Refrigerant flow shall be measured with an integrating flow measuring system that is accurate to within ±1.0% of the reading

4.6 Condensate Quantity Measurement

4.6.1 Condensate collection rates shall be determined using a liquid quantity meter that is accurate to within ±1.0% of the reading or other method that provides the same accuracy.

4.7 Speed Measuring Instruments

4.7.1 Speed measurements shall be made with a revolution counter, tachometer, stroboscope, or oscilloscope that is accurate to within ±1.0% of the reading.

4.8 Time and Mass Measurements

4.8.1 Time interval measurements shall be made with an instrument that is accurate to within ±0.2% of reading.

4.8.2 Mass Measurement shall be made with an apparatus that is accurate to within ±1.0% of reading.

4.9 Torque Measurements

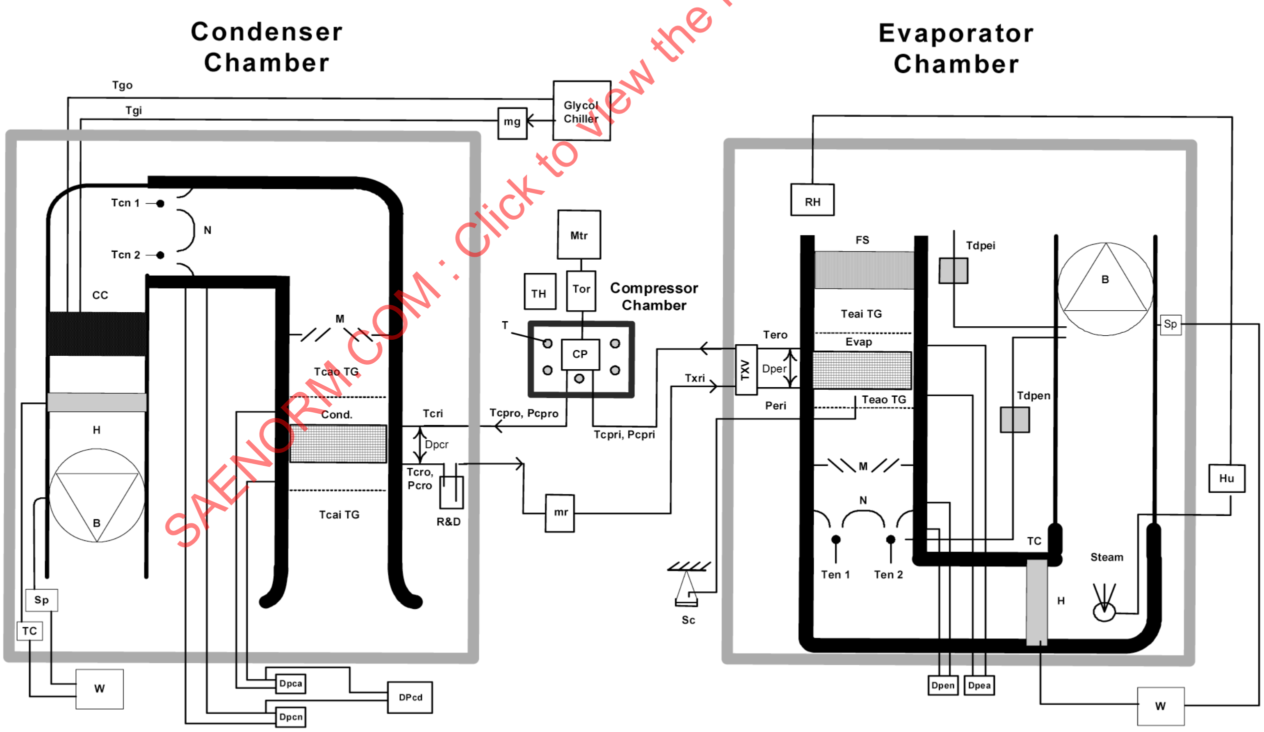
4.9.1 Torque measurements shall be made with an instrument that is accurate to within ±2.0% of reading.

4.10 Refrigerant mass composition measurements for refrigerant mixtures shall be made using an instrument that is in accordance with ARI Standard 700-2004.[5]

5. TEST FACILITY AND MEASURING APPARATUS

5.1 Test Facility (Figure 1)

Recommended configurations for the test facility are provided below. In all cases, suitable means for determining the dry-bulb temperature and water vapor content of the air entering and leaving the heat exchangers shall be provided.



B - Blower, **CP** - Compressor, **CC** - Cooling Coil, **CH** - Glycol Chiller, **Cond** - Condenser, **Dp** - Differential Pressure Transducer, **Evap** - Evaporator, **FS** Flow Straightener, **H** - Heater, **Hu** - Humidifier, **mg** - Glycol Mass Flow Meter, **mr** - Refrigerant mass Flow Meter, **Mtr** - Motor, **N** - Nozzle, **P** - Pressure Transducer, **RH** - Relative Humidity Probe, **Sc** - Condensate Scale, **Sp** - Speed Controller and Tachometer, **T** - Thermocouple, **TC** - Temperature Controll **TG** - Thermocouple Grid, **Tor** - Torque Transducer, **W** - Watt Transducer, **TXV** - Thermal Expansion Valve
Indices : **a** - air, **c** - condenser, **cp** - comp ressor, **e** - evaporator, **g** - glycol, **I** - inlet, **n** - nozzle, **o** - outlet, **r** - refrigerant,

FIGURE 1 - SAMPLE TEST FACILITY

5.2 Test Room Requirements

5.2.1 Typically, two test rooms are required; an in car room and an under-hood room. If desired, a third room may be used for the compressor to simulate under-hood conditions. That room can be located either outside or in the under-hood room.

5.2.2 An in car condition test room is required. This may be any room or space in which the desired test conditions can be maintained within the tolerances given in Table 1.

5.2.3 An under-hood test room is required. This may be any room or space in which the desired test conditions can be maintained within the tolerances given in Table 1. A condenser wind tunnel may be substituted for the under-hood test room provided it is capable of maintaining the conditions given in Table 1.

TABLE 1 - PERMISSIBLE VARIATIONS IN TEST CONDITIONS

Controlled Test Parameter	Operating Tolerance(1) ±	Condition Tolerance(2) ±
Evaporator inlet air dry-bulb temperature (°C)	0.5	0.3
Evaporator inlet air humidity (%)	3	1
Evaporator air flow rate (kg/min)	0.3	0.1
Targeted air temp. downstream of evaporator (°C)	0.5	0.3
Condenser inlet air dry-bulb temperature (°C)	0.5	0.3
Condenser face air velocity (m/s)	0.18	0.06
Compressor speed (rpm)	2	1

1. Operating Tolerance is the maximum permissible range of any measurement.
2. Condition Tolerance is the maximum permissible variation of the average value of the measurement from the specified test condition.

5.3 Wind Tunnels for Heat Exchangers

5.3.1 To produce repeatable and comparable results, it is recommended to control the inlet air temperature and velocity profile across the face of the test heat exchangers.

5.3.2 Below is an example of one setup that has been used to accomplish this end.

5.3.2.1 Each test heat exchanger shall be mounted in a wind tunnel inside their respective test room. The wind tunnel may be constructed from any suitable duct material. The illustration of a duct sections for the test heat exchanger in a wind tunnel are shown in Figure 2.

5.3.2.2 Air enters over the flow straightener (1) and inlet thermocouple grid (2) to the test heat exchanger (evaporator, condenser, or gas cooler). At the exit of the test heat exchanger is the outlet thermocouple grid (3) before the flow measuring devices – nozzles (4). After the nozzles, average air stream temperature is measured by a thermocouple or other appropriate thermometer (5). Reconditioning of the air (cooling or heating) is done in the conditioning test facility heat exchanger (6). For control purposes, an electrical heater is recommended. The test facility Blower (7) provides air circulation for the test heat exchanger.

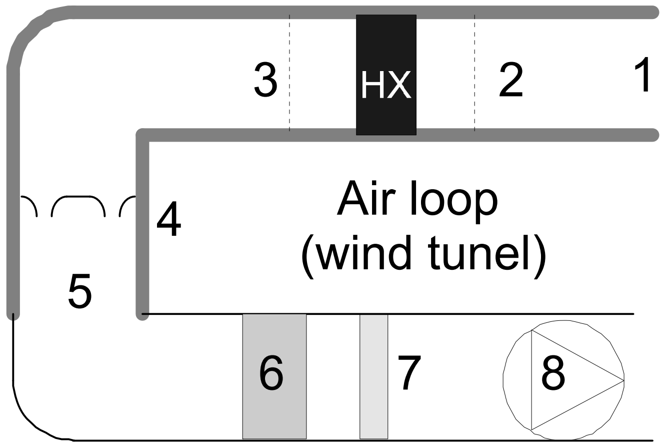


FIGURE 2 - POSSIBLE AIR LOOP

5.4 Airflow Measuring Apparatus

5.4.1 A suitable means of measuring heat exchanger air flow shall be provided. One such method is given in ANSI/ASHRAE Standard 41.2-1987 (RA 92).[6] As shown in Figure 4, the nozzle airflow measuring apparatus consists of a receiving chamber and a discharge chamber separated by a partition in which one or more nozzles are located.

5.4.2 The nozzles used to measure air flow shall comply with ANSI/ASHRAE Standard 41.2-1987 (RA 92)[6] or suitable alternative (ASME, ISO, JIS, etc.). The discharge coefficient shall be calculated and evaluated as given in ANSI/ASHRAE Standard 51-1999[7] or ASME Fluid Meters.[4]

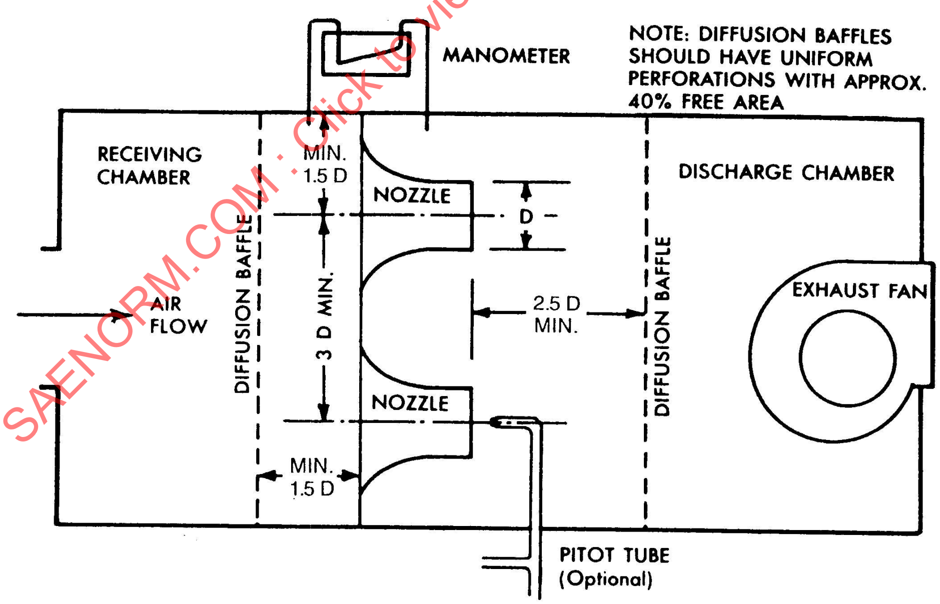


FIGURE 3 - NOZZLE AIRFLOW MEASURING APPARATUS (ANSI/ASHRAE STANDARD 37-2005[8])
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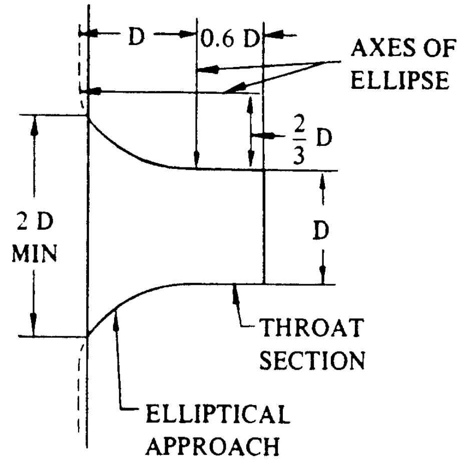

FIGURE 4 - FLOW NOZZLE CONSTRUCTION (ANSI/ASHRAE STANDARD 37-2005)
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5.4.3 The velocity profile when measured at not less than nine (9) grid points should be within ±10%. Additional means shall be used when needed to improve uniformity of the air velocity profile. Consult ANSI/ASHRAE Standard 41.2-1987 (RA 92)[6] for methods of flow straightening to achieve the desired velocity profile.

5.4.4 Temperature measurements shall be made at not less than the same nine (9) locations as in 5.4.3. at the center of equal segments of cross-sectional area throughout the duct. Inlet air temperature should be uniform within ±0.5 °C. Consult ANSI/ASHRAE Standard 41.1-1986 (RA 2001)[1] for methods of mixing the air stream to achieve the desired uniformity of temperature.

5.5 Refrigerant Loop (Figure 5)

5.5.1 The refrigerant loop should mimic as much as possible conditions encountered in the reference vehicle environment. The components used in the vehicle should be used in the test loop if possible. Alteration of the piping (lines, hoses, and fittings) from the real application should be fully documented.

5.5.1.1 Suction line pressure drop should be ±10% of the reference vehicle application

5.5.1.2 If an Internal Heat Exchanger [IHX] or suction line heat exchanger is used in the system, it should be located in the same ambient environment as the compressor.

5.5.1.3 Relative height of the components should be located within ±25 mm of the reference vehicle system. If no reference vehicle is defined, the bottom of compressor should be located at bottom of condenser and bottom of evaporator shall be located at top of condenser. [Figure 5a]

5.5.1.4 System refrigerant charge amount should not be greater than 125% of the reference vehicle system.

5.5.1.5 Oil amount should be revised based on charge amount to approximate the same ratio of refrigerant to oil ratio as the reference vehicle system.

5.5.1.6 Oil in circulation shall meet the compressor manufacturer's requirements.

5.5.2 Instrumentation shall be installed so as to minimize their effect on system operation or performance.

Add nozzle temperatures, additional instrumentation, show 2nd view that illustrates height of components

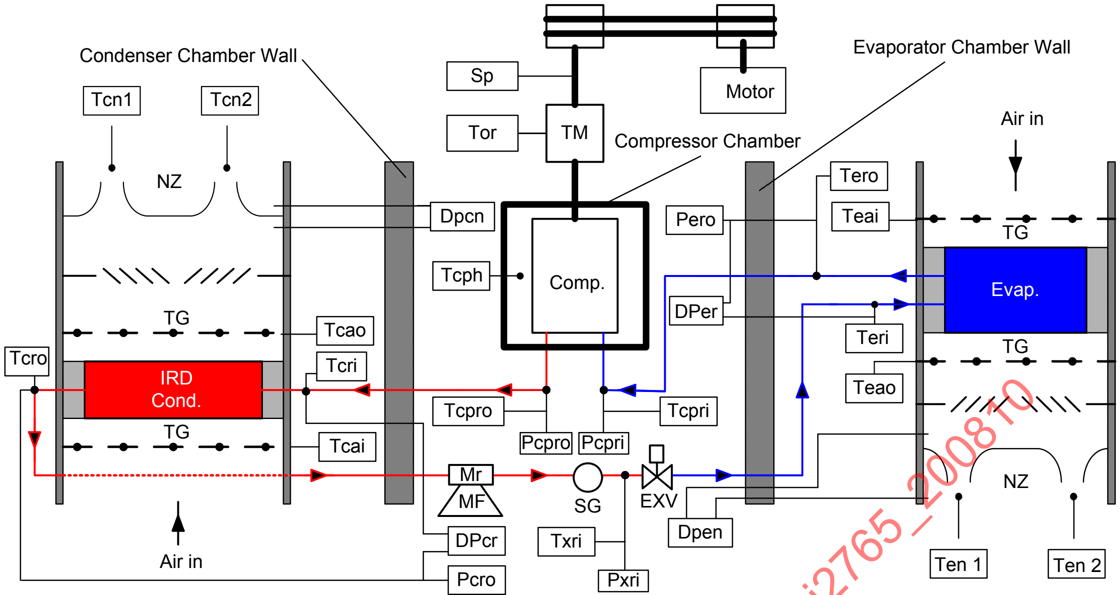


FIGURE 5 - TYPICAL REFRIGERANT LOOP

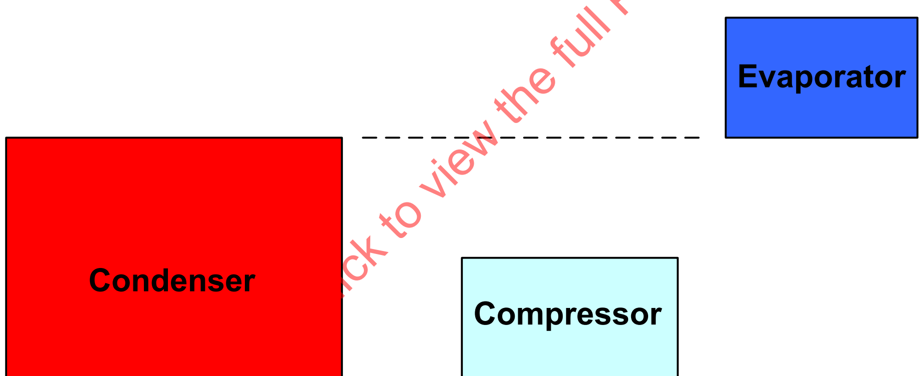


FIGURE 5A - ILLUSTRATION OF COMPONENT HEIGHTS.

5.5.3 Compressor Test Stand and Chamber

5.5.3.1 The compressor is placed inside a third chamber that provides and controls specified conditions. The temperature should be maintained within ±2 °C.

5.5.3.2 The torque meter should be located between the compressor and the clutch in order to measure compressor power at the shaft.

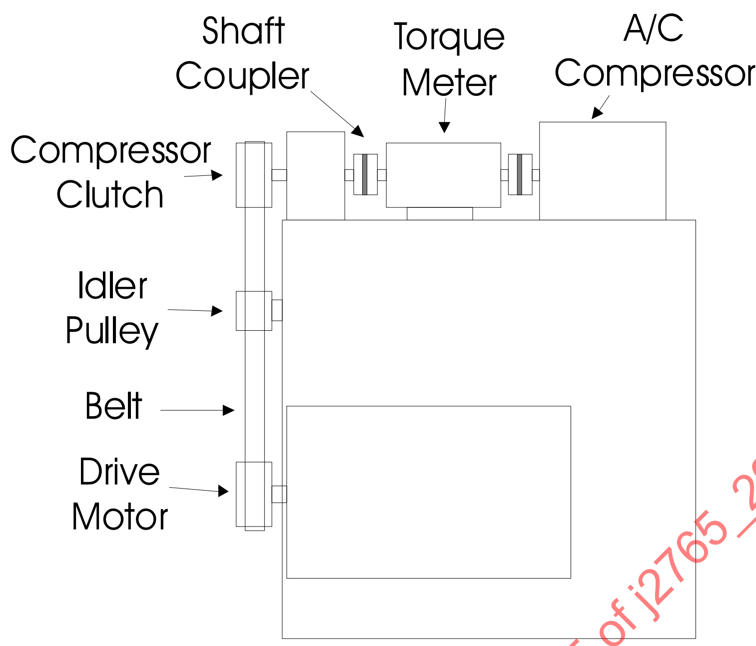


FIGURE 5B - TORQUE METER LOCATION

6. TEST CONDITIONS

6.1 To allow for variation of specified compressor speeds for different drive ratios or compressor types RPM of compressor should be adjusted as necessary based to match the system capacity at L35 test conditions.

6.2 The test matrix is shown in Table 2. The conditions cover a significant range of typical vehicle air conditioning operation.

6.3 Performance can be determined with the system operating at full capacity or at a reduced capacity to limit evaporator air outlet temperature (measured at the outlet of the airflow nozzles). Limiting evaporator air outlet temperature allows comparisons at approximately equal capacity, allows simulation of smart variable temperature control, and can provide evaporator icing prevention. Depending on the compressor, capacity can be controlled by varying displacement, cycling rates/temperature swings, etc. Precisely achieving a target temperature can sometimes be difficult due to resolution in the controls. Specification of the target temperature should be consistent across systems to be compared.

TABLE 2 - TEST MATRIX

Test Name	Ambient Temp. (°C)	Compressor Speed (rpm)	Condenser Temp. (°C)	Condenser Face Velocity (m/s)	Evaporator Temp. (°C)	Evaporator Humidity (%)	Evaporator Mass Flow (kg/min)	Evaporator Simulated Air Selection	Target Air Temp. Downstream of Evap. (°C)
I70	45	900	70	1.5	35	25	9.0	RECIRC	3
I60	45	900	60	1.5	35	25	9.0	RECIRC	3
I45	45	900	45	1.5	35	25	9.0	RECIRC	3
L45	45	1800	45	2.0	35	25	9.0	RECIRC	3
M45	45	2500	45	3.0	35	25	9.0	RECIRC	3
H45	45	4000	45	4.0	35	25	9.0	RECIRC	3
I50a	35	900	50	1.5	35	40	9.0	OSA	3
I35a	35	900	35	1.5	35	40	9.0	OSA	3
L35a	35	1800	35	2.0	35	40	9.0	OSA	3
M35a	35	2500	35	3.0	35	40	9.0	OSA	3
H35a	35	4000	35	4.0	35	40	9.0	OSA	3
I40a	25	900	40	1.5	25	80	6.5	OSA	3/10
I25a	25	900	25	1.5	25	80	6.5	OSA	3/10
L25a	25	1800	25	2.0	25	80	6.5	OSA	3/10
M25a	25	2500	25	3.0	25	80	6.5	OSA	3/10
H25a	25	4000	25	4.0	25	80	6.5	OSA	3/10
I40c	25	900	40	1.5	25	50	6.5	OSA	3/10
I25c	25	900	25	1.5	25	50	6.5	OSA	3/10
L25c	25	1800	25	2.0	25	50	6.5	OSA	3/10
M25c	25	2500	25	3.0	25	50	6.5	OSA	3/10
H25c	25	4000	25	4.0	25	50	6.5	OSA	3/10
I30	15	900	30	1.5	15	80	6.5	OSA	3/10
I15	15	900	15	1.5	15	80	6.5	OSA	3/10
L15	15	1800	15	2.0	15	80	6.5	OSA	3/10
M15	15	2500	15	3.0	15	80	6.5	OSA	3/10
H15	15	4000	15	4.0	15	80	6.5	OSA	3/10

6.4 System Setup Tests

6.4.1 A typical Refrigerant charge determination procedure is shown below. Other procedures may be used based on user preferences.

6.4.1.1 A charge determination shall be performed at the following conditions:

6.4.1.1.1 The charging condition should be: compressor speed 2000 rpm, full displacement,

6.4.1.1.2 Compressor chamber temperature 70 °C;

6.4.1.1.3 Air temperature inlet to the gas cooler/condenser 40 °C,

6.4.1.1.4 Face air velocity through the gas cooler/condenser 3.7 m/s

6.4.1.1.5 Air temperature inlet to evaporator 40 °C, 6.4.1.1.6 Face air velocity through the evaporator 2.5 m/s

6.4.1.1.7 Relative humidity at evaporator inlet of 40%.

6.4.1.1.8 Other conditions in addition to this are permitted.

6.4.1.1.9 Compressor RPM may be adjusted based on equalizing capacity.

6.4.1.2 The system shall be operated for a minimum of ten (10) minutes at the above conditions before starting the charge determination. This is to insure stable operation during the charge determination.

6.4.1.3 Select the initial amount of refrigerant based on system size. In no case should be system be operated with less than 1/2 the expected total charge.

6.4.1.4 After allowing system to stabilize, record data for ten (10) minutes. Average the data to obtain average readings.

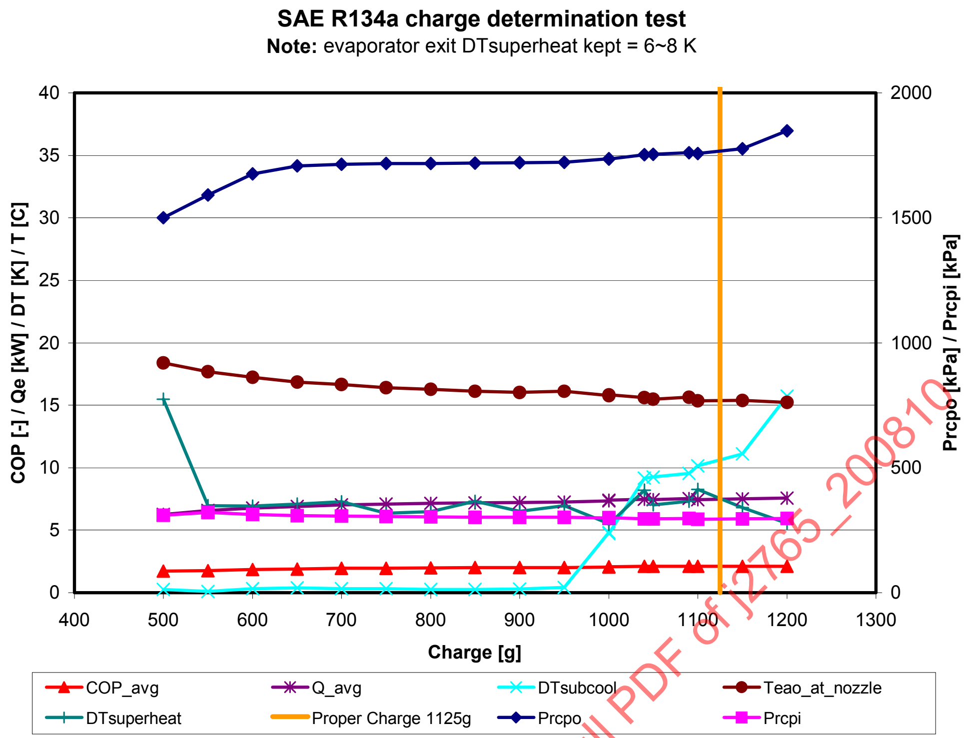
6.4.1.5 Add refrigerant in increments of no more than 50 g.

6.4.1.6 The system shall be allowed to stabilize for a minimum of ten (10) minutes after adding charge and before recording data.

6.4.1.7 Figure 8 shows a typical charge determination result for a non-subcooled condenser R134a system. The recommended criteria for R134a is: [other fluids may have different criteria]

6.4.1.7.1 Minimum of 5 K subcool

6.4.1.7.2 Compressor out pressure not to exceed 1800 kPa



Legend: COP_avg – Coefficient Of Performance (average), Q_avg – Refrigeration capacity at evaporator (average), Teao - Temperature of evaporator air out measured at nozzle, DTsubcool – Condenser outlet subcool, DTsuperheat – Evaporator outlet superheat, Prcpo – Compressor outlet (discharge) pressure, Prcpi – Compressor inlet (suction) pressure

FIGURE 6 - SAMPLE CHARGE DETERMINATION

6.4.2 Oil in Circulation Rate

6.4.2.1 Using test conditions as specified in Table 2 at 45 °C and 900 RPM and 45 °C and 2500 RPM, a sample of the circulating refrigerant and oil mixture should be taken from the equipment and the percentage of oil measured in accordance with ANSI/ASHRAE Standard 41.4-1996.[9] [Except sample amount is 30 grams.] The total indicated flow rate should then be corrected for the amount of circulating oil. Adjust oil amount to obtain compressor supplier recommended oil in circulation. [Other methods are acceptable based on user preferences]

6.4.3 Temperature and Air Velocity Uniformity

6.4.3.1 To provide equal conditions for all systems to be tested and insure reproducibility of the test results, uniformity of air velocity and temperature at the inlet to the heat exchangers must be verified prior to starting the refrigeration system performance test. Each test should be performed for both heat exchangers (condenser/gas cooler and evaporator) and over a relevant range of air velocities and inlet air temperatures.

6.4.4 Air Side Pressure Drop [Dry]

6.4.4.1 At room temperature, change airflow rate to cover the range of operation in Table 2 and record the pressure drop from inlet to outlet of each heat exchanger with the compressor turned off.

6.4.5 Suction Line Pressure Drop

6.4.5.1 Check refrigerant pressure drop from evaporator out to compressor inlet and verify that the requirements in 5.5.1.1 are met.

6.4.6 Refrigerant Control Strategies

6.4.6.1 Verify that all control strategies that would normally be used in the refrigerant system are duplicated per the reference vehicle that is being simulated. [For example: superheat control, compressor out pressure control]

6.5 Operating Procedure

6.5.1 The facility operating procedures for the steady state tests procedure elements are:

6.5.1.1 Pre-set test condition parameters.

6.5.1.2 Warm up or cool down the test chambers and adjust the parameters as needed until a stable condition is reached.

6.5.2 Establish and maintain stable conditions for at least twenty [20] minutes prior to taking data. Steady state conditions are characterized by no more than ±0.2 °C variation of heat exchanger inlet air and ±0.1 °C for dew point temperatures recorded at the beginning and after ten (10) minutes. Trend in temperatures shall not be downward or upward by greater than 0.2 °C.

6.5.3 Once stable conditions are achieved, data shall be collected for a minimum of 10 additional minutes with a minimum frequency of 0.1 Hz. [In the case of cycling system operation, measurements should be taken with a minimum frequency of 10 Hz for compressor torque and speed. Torque will be time averaged.]

7. CALCULATIONS

7.1 Evaporator Capacity shall be measured using three independent methods [See Figure 7]

7.1.1 Air Capacity = dry air mass flow rate * air enthalpy difference – condensate enthalpy (small): Additional factors should be added to compensate for any heat loss in the duct from the heat exchanger outlet to the nozzle outlet.

$$Qindoor_{air} = Ma_{indoor,dry} \cdot (heai - hean) + Qleak_{duct} - Ma_{indoor,dry} \cdot (Wei - Wen) \cdot \mathbf{h}(\text{'Steam}_{NBS}\text{'}, T = Ten, x = 0) \qquad \text{(Eq. 1)}$$

7.1.2 Refrigerant capacity shall be calculated using the following:

Refrigerant Capacity = refrigerant mass flow rate * refrigerant – oil enthalpy difference. In the case refrigerant oil enthalpies are not available, a simplified approach can be taken:

$$Qindoor_{ref} = Mr*(1-xoil)*(hero-heri)+Mr*xoil*(Cpoil(Tero-Teri_{sat}) + \text{mixing energy}) \qquad \text{(Eq. 2)}$$

Where approximately Cpoil = 2.05 kJ/kg °C and mixing energy for a refrigerant oil combination = $0.00113*(Tero^2-Teri_{sat}^2)$

7.1.3 Chamber capacity = electric heat + steam supplied energy ± heat gain/loss through the chamber walls (small).

$$Qch = W + Qsteam + Qtrans \qquad \text{(Eq. 3)}$$

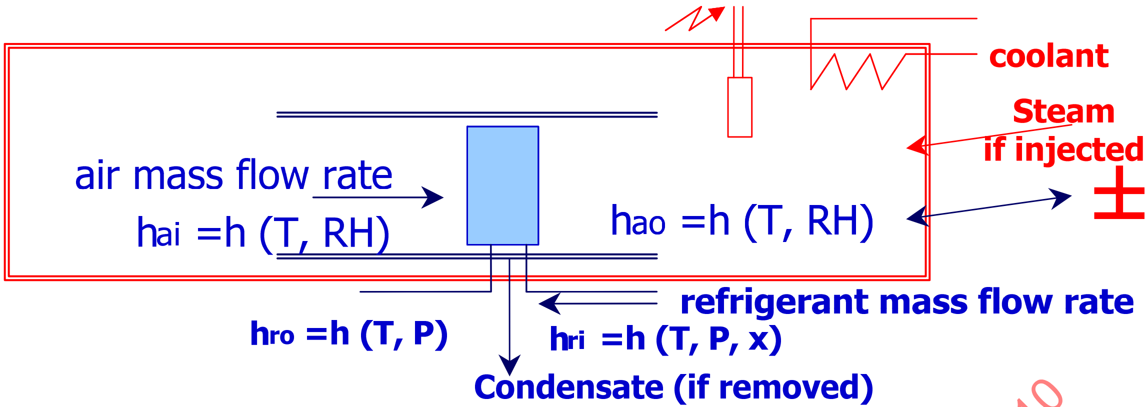


FIGURE 7 - CAPACITY MEASUREMENT SCHEMATIC FOR ALL THREE METHODS

7.1.4 Properties of refrigerants shall be obtained from *2001 ASHRAE Handbook-Fundamentals*[10] or from an established refrigerant property database.[10]

7.1.5 Air enthalpies shall be determined in accordance with ANSI/ASHRAE Standard 41.6-1994 (RA 2001)[2] or other established database.

7.2 Reported Capacity Calculation

7.2.1 The reported capacity is used to calculate COP. At least two of the independent capacities measurements shall be within ±4.0% and the reported capacity shall be the average of these two values.

7.3 Compressor Power Calculation

7.3.1 Compressor power is calculated as given below:

$$W_{comp} = rpm \cdot \frac{2\pi}{60} \cdot T \qquad \text{(Eq. 4)}$$

7.3.2 In the case of cycling compressors, the average compressor power is calculated by integrating the power over complete cycles during the stabilized ten minute measurement.

7.3.3 For variable compressors, the average power over the stabilized ten minute measurement is used.

7.4 Refrigerant System Coefficient of Performance (COP) Calculation

7.4.1 Coefficient of performance (see also 3.4) shall be calculated based on average value of heat removed $Qindoor_{avg}$ as follows:

$$COP_{indoor,avg} = \frac{Qindoor_{avg}}{W_{comp}} \qquad \text{(Eq. 5)}$$

Averaging Q indoor shall be done using all available balances, not less than two.

7.4.2 Condenser/gas cooler fan and evaporator blower power is not included in this COP calculation. This calculation shall be modified if configurations described in 1.1 are being evaluated.

8. DATA TO BE RECORDED

8.1 System Description

8.1.1 The system and test conditions should be described, as far as possible, specifying:

8.1.1.1 Size and weight of components.

8.1.1.2 Air pressure drop through the evaporator and condenser/gas cooler.

8.1.1.3 Uniformity of airflow velocity and inlet air temperature to the evaporator and condenser.

8.1.1.4 Line lengths (should be similar in length and diameters to real system).

8.1.1.5 Charge determination criteria.

8.1.1.6 Oil type, charge, and concentration in circulation.

8.1.1.7 Expansion device.

8.1.2 Control of evaporator air outlet temperature should be described. Some systems use variable displacement compressors whose capacity is controlled to maintain the desired air outlet temperature (if the system could reach that temperature), some are clutch cycling, some have other ways of reducing capacity. All systems typically exhibit some oscillation at very low loads and these should be reported.

9. SYMBOLS USED IN EQUATIONS

A	nozzle throat area (m^2)
C	nozzle discharge coefficient (dimensionless)
E_i	power input, evaporator side (W)
h_{a1}	enthalpy, air entering evaporator (J/kg dry air)
h_{a2}	enthalpy, air leaving evaporator side (J/kg dry air)
h_{r1}	enthalpy, refrigerant entering evaporator (J/kg)
h_{r2}	enthalpy, refrigerant leaving evaporator (J/kg)
P_{comp}	power, compressor (W)
P_v	pressure drop across the nozzle (Pa)
q_{tci}	total cooling capacity, evaporator (W)
Q_s	air flow, standard air (m^3/s)
Q_v	airflow rate (m^3/s)
rpm	revolutions per minute
T	torque, compressor (N-m)
v_n	specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature and barometric pressure at the nozzle exit, and vapor content evaluated at the leaving conditions (m^3/kg dry air)
v'_n	specific volume of air at nozzle (m^3/kg)
v_{nsp}	specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature at the nozzle exit and vapor content evaluated at the leaving conditions but at standard barometric pressure (m^3/kg dry air)
W_n	humidity ratio at the nozzle (kg water vapor/kg dry air)
w_{ro}	mass flow rate, refrigerant oil mixture (kg/s)
x	mass ratio, refrigerant to refrigerant oil mixture (dimensionless)
ρ	density of refrigerant (kg/m^3)