BSR/ASHRAE Standard 225-2020RA

Public Review Draft

Method for Performance Testing Centrifugal Refrigerant Compressors and Condensing Units

First Reaffirmation Public Review (October 2024)

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ASHRAE, 180 Technology Pkwy NW, Peachtree Corners, GA 30092

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BSR

~~ANSI~~/ASHRAE Standard 225-~~2020~~ (RA 202X)

Method for Performance Testing Centrifugal Refrigerant Compressors and Condensing Units

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NOTE

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(This foreword is not part of this standard. It is merely informative and does not contain requirements necessary for conformance to the standard. It has not been processed according to the ANSI requirements for a standard and may contain material that has not been subject to public review or a consensus process. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

FOREWORD

ANSI/ASHRAE Standard 225 prescribes methods for obtaining performance data relating to centrifugal compressors or compressor units. The intent of this standard is to provide uniform test methods to measure the performance of this equipment by addressing the test and instrumentation requirements, test procedures, data to be recorded, and calculations to generate and confirm valid test results.

The standard has been developed using as a basis ANSI/ASHRAE *Standard 23.1-2019,* Methods for Performance Testing Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Pressures of the Refrigerant, *and ANSI/ASHRAE Standard 23.2-2019,* Methods of Test for Rating the Performance of Positive Displacement Compressors that Operate at Supercritical Pressures of the Refrigerants, *with the following differences:*

*THESE 2 ARE combined into STND 23-2022 TITLE IS underlined

a. *Defines* centrifugal compressor, Mach number, measured property, static property, *and* total property
b. *Allows mass flow measurement using a flowmeter without a simultaneous and independent confirming test of mass flow*
c. *Includes calculations for head factor and flow factor*
d. *Includes requirements to use total enthalpy for head factor and isentropic efficiency calculations*
e. *Excludes volumetric efficiency because it does not apply to centrifugal compressors and condensing units*
f. *Shows calculations for total properties and Mach number (Normative Appendix A)*
g. *Shows calculations for compressor efficiency using total enthalpy (Informative Appendix C)*

Add NXT Pg Paragraph to FWD

1. PURPOSE

This standard provides methods of testing for the performance rating of centrifugal refrigerant compressors and condensing units.

2. SCOPE

This standard applies to centrifugal refrigerant compressors and condensing units using any refrigerants. This standard applies to single and multistage compressors with or without means of intermediate cooling.

3. DEFINITIONS

The following definitions apply to the terms used in this standard.

***accuracy*:** the degree of conformity of an indicated value to the corresponding true value.

bubble-point temperature: a liquid-vapor equilibrium point for a pure liquid or for a multicomponent mixture of miscible, pure component liquids, in the absence of noncondensables, where the temperature of the mixture at a defined pressure is the minimum temperature required for a vapor bubble to form in the liquid.

calorimeter: a thermally insulated apparatus containing a heat exchanger that is used to determine the mass flow rate of a refrigerant by measuring the heat input/output that will result in a corresponding enthalpy change for the refrigerant.

capacity: the rate of heat removal by the refrigerant used in the compressor or condensing unit in a refrigerating system. This rate equals the product of the refrigerant mass flow rate and the difference in the specific enthalpies of the refrigerant vapor at its thermodynamic state entering the compressor or condensing unit and refrigerant liquid at the thermodynamic state entering the expansion device.

centrifugal refrigerant compressor: a machine that increases the pressure of a refrigerant vapor by imparting kinetic energy to a continuous flow of fluid.

FOREWORD

The 2024 edition of Standard 225 updates normative references. This standard was prepared under the auspices of ASHRAE. It may be used, in whole or in part, by an association or government agency with due credit to ASHRAE. Adherence is strictly on a voluntary basis and merely in the interests of obtaining uniform guidelines throughout the industry.

compressor: see *centrifugal compressor*.

compressor or condensing unit efficiency (isentropic efficiency): the ratio of the work absorbed for compressing a unit mass of refrigerant entering the stage of the compressor or condensing unit to the work absorbed for compressing the same unit mass of refrigerant by isentropic compression within the stage.

condenser liquid flow rate: the mass flow rate of liquid through the condensing unit under the conditions specified.

condensing unit: a machine designed to condense refrigerant vapor to a liquid by compressing the vapor and rejecting heat to a cooling medium. A condensing unit consists of a condensing heat exchanger and one or more compressors and motors with ancillaries.

confirming test: an independent and simultaneous test performed to validate the primary test results (compare to *primary test*). Compressor or condensing unit ratings are determined from the primary test results.

cooling liquid flow rate: the total mass flow rate of liquid required for all cooling purposes in a compressor or condensing unit.

dew-point temperature: a vapor-liquid equilibrium point for a pure liquid or for a multicomponent mixture of miscible, pure components, in the absence of noncondensables, where the temperature of the mixture at a defined pressure is the maximum temperature required for a liquid drop to form in the vapor.

economizer: a heat exchanger or flash tank that is used to subcool liquid refrigerant exiting the condenser for vapor injection.

energy efficiency ratio (EER): a dimensional ratio of the cooling capacity (Btu/h) to the power input (W).

error: the difference between the test result and its corresponding true value.

flowmeter: a device employing a detecting element that determines the mass flow rate of a refrigerant in the gaseous or liquid phase within a closed conduit by measuring the corresponding response of the detecting element.

hermetic compressor: a compressor assembly containing a motor within a gas-tight housing that is permanently sealed by welding or brazing with no access for servicing internal parts in the field.

intermediate cooling: a method of using a heat exchanger to (a) cool the compressor mechanism or lubricant or (b) cool the refrigerant to reduce discharge temperature. The heat exchanger component of the intermediate cooling means is integral to the compressor. The intermediate cooling thermal load is not taken into account in the calculations of isentropic efficiency or compressor or condensing unit capacity.

liquid injection: a method of (a) internally cooling the compressor mechanism or lubricant or (b) reducing discharge temperature by introducing saturated or subcooled discharge-side liquid refrigerant into the compressor. Liquid refrigerant injection mass flow rate is not taken into account in the calculations of isentropic efficiency or compressor or condensing unit capacity.

lubricant circulation rate: the ratio, expressed as a percent, of the mass of lubricant circulating through a refrigerant system to the total mass of refrigerant and lubricant flowing through the system at a specified set of operating conditions.

Mach number: the ratio of the speed of a fluid to the speed of sound in the medium.

measured property: the value provided by a gage representing an intrinsic property of the fluid being used, such as temperature or pressure. (See also *static property* and *total property*).

measurement system: the instruments, signal conditioning systems, if any, and data acquisition systems, if any.

motor-compressor: a motor and an *open compressor* mounted onto a common base but not integrated into a gas-tight housing.

multistage compressor: a compressor that has two or more compression chambers connected in series.

open compressor: a refrigerant compressor with a shaft or other moving part extending through its casing to be driven by an external source of mechanical power.

performance factor: the ratio of capacity to power input at specified operating conditions.

primary test: a test performed to determine the ratings of a compressor or condensing unit (compare to *confirming test*).

saturation temperature: the equilibrium temperature of a pure refrigerant or an azeotropic refrigerant in a two-phase mixture of a vapor and liquid at a given absolute pressure.

semihermetic compressor: a motor-compressor assembly contained within a gas-tight housing that is sealed by bolted joints to provide access for servicing internal parts.

single-stage compressor: a compressor that has a single compression chamber or two or more compression chambers that are connected in parallel.

static property: the property of the fluid being measured that would be the result if the measurement device were to be moving with the fluid. (See also *measured property* and *total property*).

subcooling: the difference between the liquid temperature entering the refrigerant control device and the bubble-point temperature at a defined pressure.

suction temperature: the temperature of the refrigerant vapor returning to the compressor or condensing unit.

superheat: the difference between the suction temperature and the dew-point temperature at a defined pressure.

test point: a specific set of test operating conditions and tolerances for recording data.

total property: the property of a flowing fluid that would occur if the fluid's velocity were to be reduced to zero in an isentropic process. (See also *measured property* and *static property*).

true value: the unknown, error-free value of a test result.

uncertainty: a measure of the potential error in a measurement or experimental result that reflects the lack of confidence in the result to a specified level.

unit under test (UUT): a compressor or condensing unit.

vapor injection: a method of (a) increasing evaporator capacity using an *economizer* to subcool liquid refrigerant exiting the condenser and (b) reducing discharge temperature. Refrigerant vapor or wet vapor that exits the economizer enters an intermediate port or an interstage port on the compressor.

4. CLASSIFICATIONS

4.1 Compressor Types. Centrifugal compressors that are within the scope of this standard are classified as one of the following types:

a. Open compressor
b. Hermetic compressor
c. Semihermetic compressor
d. Motor-compressor

Informative Note: A centrifugal refrigerant compressor may include an integral lubricant pump.

4.2 Condensing Unit Types. Centrifugal condensing units that are within the scope of this standard are classified as one of the following types:

a. Liquid-cooled condensing unit
b. Air-cooled condensing unit
c. Evaporatively cooled condensing unit

4.3 Calorimeter Types. Calorimeters that are within the scope of this standard are classified as one of the following types.

a. Evaporator calorimeters:
   1. Secondary refrigerant calorimeter
   2. Secondary fluid calorimeter

3. Primary refrigerant calorimeter

b. Condenser calorimeter

4.4 Flowmeter Types. Flowmeters that are within the scope of this standard are classified as one of the following types:

a. Gaseous refrigerant flowmeter
b. Liquid refrigerant flowmeter

5. REQUIREMENTS

5.1 Test Plan. A test plan shall specify the test points and the calculations to be performed. The test plan shall be one of the following:

a. A document provided by the person or the organization that authorized the tests and calculations to be performed
b. A method-of-test standard
c. A rating standard
d. A regulation or code

5.2 Primary and Confirming Refrigerant Mass Flow Rate Measurements. If the primary test method for mass flow rate measurement is either a gaseous or liquid refrigerant flowmeter, then it is not necessary to provide a simultaneous and independent confirming test of mass flow.

If the primary test method does not meet the previously mentioned criteria, each test data point shall consist of a primary test and a simultaneous, independent confirming test at a specified set of operating conditions.

To be independent,

a. the confirming test measurement systems shall be separate from the corresponding measurement systems in the primary test, and
b. the fluid flows and heat transfers in the confirming test shall neither share nor influence the fluid flows or heat transfers in the primary method.

Compressor or condensing unit ratings shall be determined from refrigerant mass flow rates obtained by the primary method of test. The refrigerant mass flow rates obtained from the primary test are valid only if the corresponding measured refrigerant mass flow rate from the confirming test is within ±3% of the primary test measurement.

Table 1 lists the test method alternatives for measuring refrigerant mass flow rates. The user shall select one of these six test methods to be the primary test method and one of these six test methods to be the confirming test method if required.

5.3 Intermediate Cooling and Liquid Injection. If intermediate cooling or liquid injection is included in the unit under test (UUT):

a. Intermediate cooling shall be performed according to the manufacturer's instructions with respect to the parameters needed to operate the selected method of intermediate cooling. Figure 1 shows the cycle schematic and pressure-enthalpy diagram for a two-stage compressor with intermediate cooling. Figure 2 shows the cycle schematic and pressure-enthalpy diagram for compressors in series with intermediate cooling. Use Equation 3 or 4 to calculate the isentropic efficiency for the compressors shown in Figures 1 and 2.
b. Liquid injection shall be performed according to the manufacturer's instructions with respect to pressure, temperature, quality, and refrigerant mass flow rate at the injection location.
c. Liquid injection mass flow rates shall be measured at each test point in accordance with Section 5.2.

Informative Note: For example, where testing compressors that use liquid injection for cooling and include a test method that measures total refrigerant mass flow rate on the discharge side of the UUT, the liquid injection mass flow rate must be measured by one of the methods listed in Table 1 and then subtracted from the total refrigerant mass flow rate to determine the refrigerant mass flow rate entering the UUT. The resulting refrigerant mass flow rate is used in the calculations in Section 5.8.

5.4 Input Power. In the primary test method, the total input power in W (hp), to the UUT shall be measured at each test point in accordance with ASHRAE Standard 41.11 ~~[3]~~.

Should be cited as #1 not 3

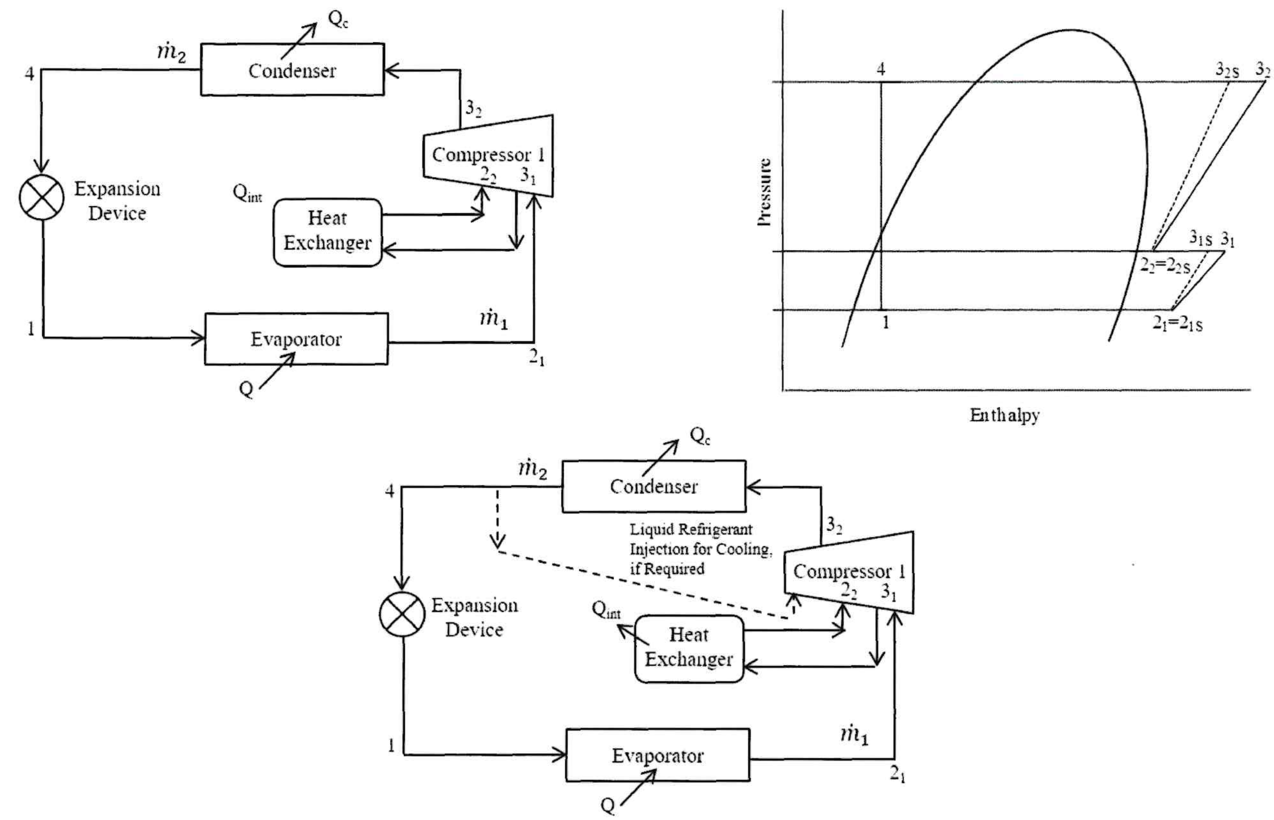


Figure 1 Cycle schematic and pressure-enthalpy diagram for a two-stage compressor with intermediate cooling.

5.5 Measurement Uncertainty. The uncertainty in each refrigerant mass flow rate measurement and power input measurement shall be estimated at each test point using the methods prescribed in ASHRAE Standard 41.9 [1] or ASHRAE Standard 41.10 [2] at 95% probability unless otherwise specified by the test plan. Alternatively, the worst-case uncertainty for all test points shall be estimated and reported for every test point.

5.6 Refrigerant Data. The primary source of refrigerant properties shall be an industry recognized source such as NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (REFPROP) [4]. The source of refrigerant properties shall be stated in the test report.

5.7 Refrigerant Numbers. The ASHRAE refrigerant number [5] for the refrigerant used during these tests shall be stated in the test report.

5.8 Calculations

5.8.1 Compressor Head Factor and Isentropic Efficiency. The head factor and isentropic efficiency calculations in this section apply to compressors that are tested separately and to compressors that are an integral part of condensing units.

5.8.1.1 Head Factor and Isentropic Efficiency for a Single-Stage Compressor. Figure 3 shows the cycle schematic and pressure-enthalpy diagram for a single-stage compressor. This cycle includes liquid injection if required for cooling.

The head factor and isentropic efficiency for a single-stage compressor as described in Figure 3 shall be calculated using Equations 1 and 3 for SI units or using Equations 2 and 4 for I-P units.

Head factor and isentropic efficiency shall be calculated using the enthalpy calculated from the measured properties as well as the total properties. If the calculated Mach number at the compressor inlet or outlet is equal to or below 0.13, it shall be assumed that the difference between measured and total properties is negligible, and measured properties shall be used to calculate head factor and isentropic efficiency. For UUTs with an inlet or outlet Mach number

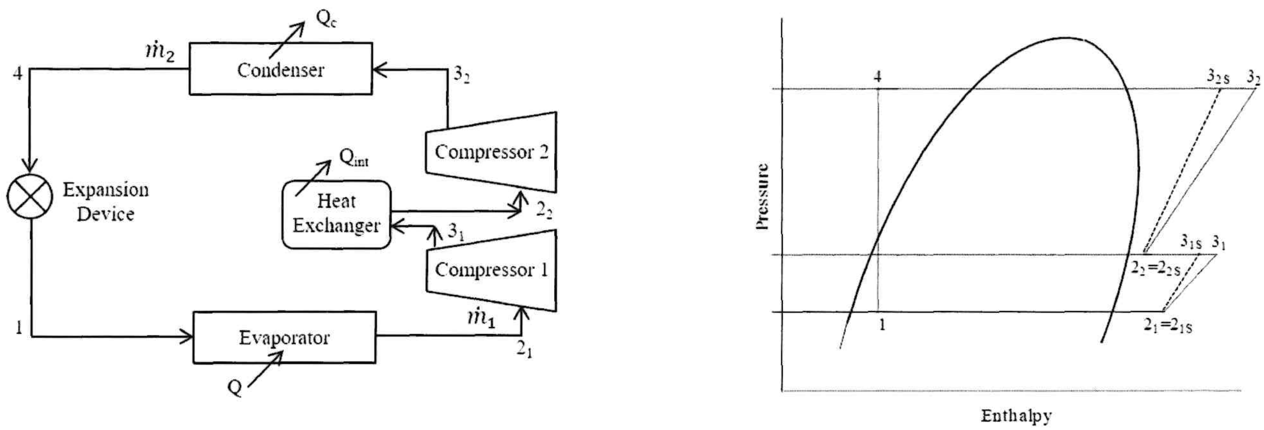


Figure 2 Cycle schematic and pressure-enthalpy diagram for compressors in series with intermediate cooling.

Table 1 Alternative Test Methods for Measuring Refrigerant Mass Flow Rates when the Primary Mass Flow Measurement Does not Use a Flowmeter

List of Test Methods (Column A)	Measurement Standard	Primary Test Method (Column B)	Confirming Test Method (Column C)
Secondary refrigerant calorimeter	ASHRAE Standard 41.9[1] [(a)]	Select any test method listed in Column A.	Select any test method from Column A, provided that the primary and confirming test methods are independent and performed simultaneously.
Secondary fluid calorimeter			
Primary refrigerant calorimeter			
Condenser calorimeter			
Gaseous refrigerant flowmeter	ASHRAE Standard 41.10[2]		
Liquid refrigerant flowmeter			

a. When using a secondary fluid calorimeter per ASHRAE Standard 41.9[1], it shall be acceptable to circulate the secondary fluid through either the outer circuit or the inner circuit.

above 0.13, total properties shall be used to calculate head factor and isentropic efficiency. The procedure for calculating the total properties is provided in Normative Appendix A.

$$\text{HF} = \frac{(h_{3_S} - h_{2_S})}{a_{2_S}^2} \times 1000 \qquad (1)$$

$$\text{HF} = \frac{(h_{3_S} - h_{2_S})}{a_{2_S}^2} \times 25,037 \qquad (2)$$

$$\eta = \frac{[\dot{m}_1(h_{3_S} - h_{2_S})]}{P} \times 100 \qquad (3)$$

$$\eta = \frac{[\dot{m}_1(h_{3_S} - h_{2_S})]}{P} \times 0.02931 \qquad (4)$$

where

HF = head factor

a_{2_S} = refrigerant sonic velocity entering the compressor, m/s (ft/s)

η = isentropic efficiency for a single-stage compressor, %

$\dot{m}_1$ = refrigerant mass flow rate entering the compressor, kg/s (lb_m/h)

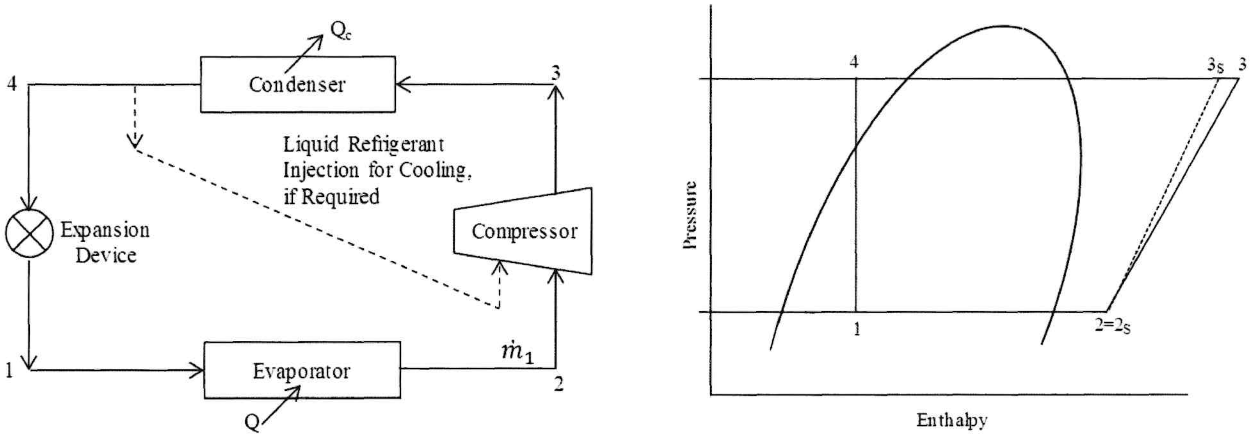


Figure 3 Cycle schematic and pressure-enthalpy diagram for a single-stage compressor.

h_{2_S} = specific enthalpy of refrigerant vapor at suction pressure and temperature entering the compressor, kJ/kg (Btu/lb$_m$)

h_{3_S} = specific enthalpy of refrigerant vapor at discharge pressure following an isentropic compression of the refrigerant from compressor suction pressure and temperature, kJ/kg (Btu/lb$_m$)

P = total power input to the UUT, kW (kW)

5.8.1.2 Head Factor and Isentropic Efficiency for a Multistage Compressor. The head factor and isentropic efficiency for a multistage compressor shall be calculated using Equations 5 and 7 for SI units or using Equations 6 and 8 for I-P units.

Head factor and isentropic efficiency shall be calculated using the enthalpy calculated from the measured properties as well as the total properties. If the calculated Mach number at the compressor inlet or outlet is equal to or below 0.13, it shall be assumed that the difference between measured and total properties is negligible, and measured properties shall be used to calculate head factor and isentropic efficiency. For UUTs with an inlet or outlet Mach number above 0.13, total properties shall be used to calculate head factor and isentropic efficiency. The procedure for calculating the total properties is provided in Normative Appendix A.

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 1000 \tag{5}$$

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 25,037 \tag{6}$$

$$\eta = \frac{\left[\sum_{i=1}^{\mathrm{NS}} \dot{m}_i (h_{3_{iS}} - h_{2_{iS}})\right]}{P} \times 100 \tag{7}$$

$$\eta = \frac{\left[\sum_{i=1}^{\mathrm{NS}} \dot{m}_i (h_{3_{iS}} - h_{2_{iS}})\right]}{P} \times 0.02931 \tag{8}$$

where

HF_i = head factor for compressor stage i

$a_{2_{1S}}$ = refrigerant sonic velocity entering the compressor, m/s (ft/s)

η = isentropic efficiency, % (%), for a multistage compressor

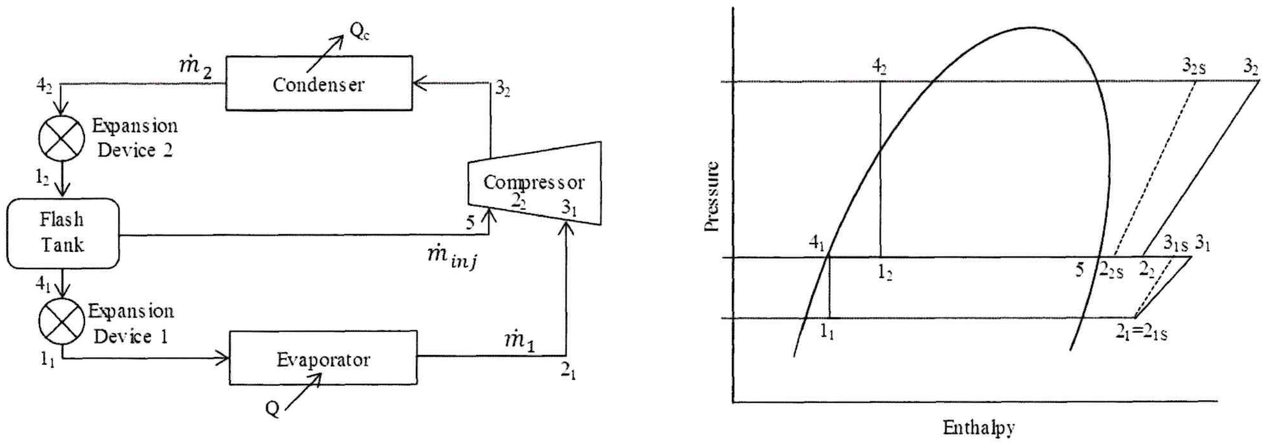


Figure 4 Cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a flash tank economizer.

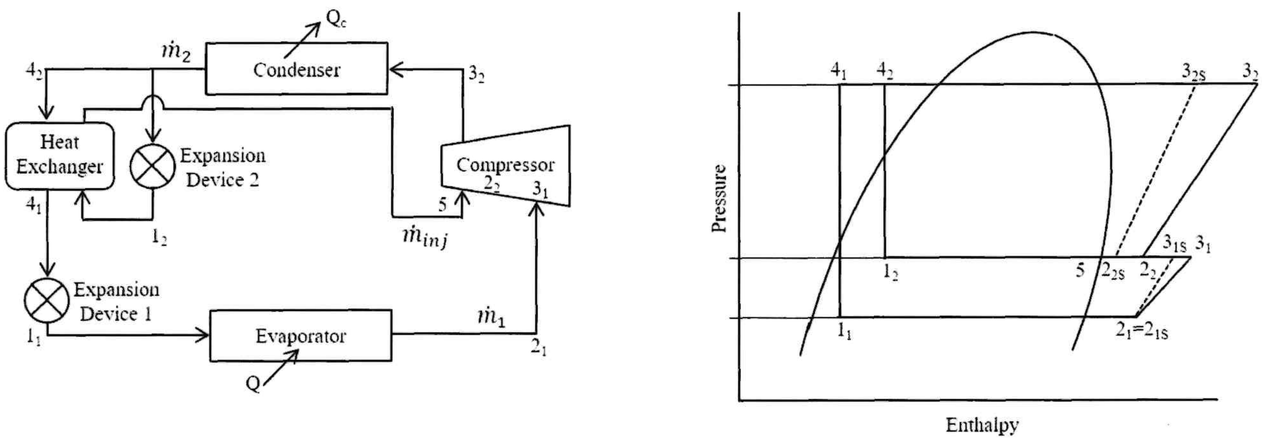


Figure 5 Cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted upstream of the economizer.

NS = number of compressor stages

$\dot{m}_i$ = refrigerant mass flow rate entering the compressor stage i, kg/s (lb_m/h)

$h_{2_{iS}}$ = specific enthalpy of refrigerant vapor at suction pressure and temperature entering the compressor stage i, kJ/kg (Btu/lb_m)

$h_{3_{iS}}$ = specific enthalpy of refrigerant vapor at the discharge pressure for stage i following an isentropic compression of the refrigerant from compressor stage suction pressure and temperature, kJ/kg (Btu/lb_m)

P = total power input to the UUT, kW (kW)

5.8.1.3 Head Factor and Isentropic Efficiency for a Two-Stage Compressor with Vapor Injection. Figure 4 shows the cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a flash tank economizer. Figure 5 shows the cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted upstream of the economizer. Figure 6 shows the cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted downstream of the economizer.

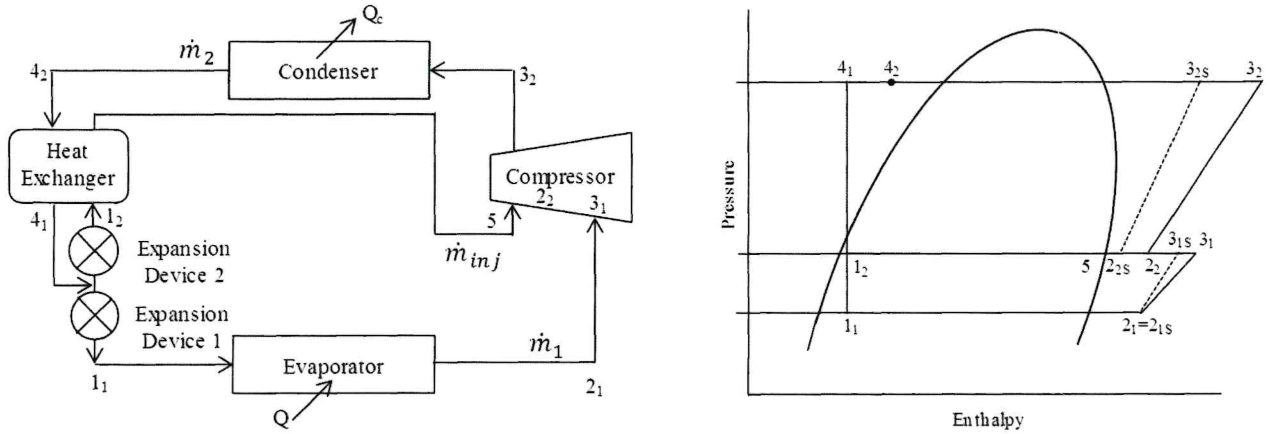


Figure 6 Cycle schematic and pressure-enthalpy diagram for a two-stage compressor with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted downstream of the economizer.

The head factor and isentropic efficiency for a two-stage compressor with vapor injection using an economizer shall be calculated using Equations 9 and 11 for SI units or using Equations 10 and 12 for I-P units.

Head factor and isentropic efficiency shall be calculated using the enthalpy calculated from the measured properties as well as the total properties. If the calculated Mach number at the compressor inlet or outlet is equal to or below 0.13, it shall be assumed that the difference between measured and total properties is negligible, and measured properties shall be used to calculate head factor and isentropic efficiency. For UUTs with an inlet or outlet Mach number above 0.13, total properties shall be used to calculate head factor and isentropic efficiency. The procedure for calculating the total properties is provided in Normative Appendix A.

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 1000 \tag{9}$$

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 25,037 \tag{10}$$

$$\eta = \frac{[\dot{m}_1(h_{3_{1S}} - h_{2_{1S}}) + \dot{m}_2(h_{3_{2S}} - h_{2_{2S}})]}{P} \times 100 \tag{11}$$

$$\eta = \frac{[\dot{m}_1(h_{3_{1S}} - h_{2_{1S}}) + \dot{m}_2(h_{3_{2S}} - h_{2_{2S}})]}{P} \times 0.02931 \tag{12}$$

where

HF_i	=	head factor for compressor stage i
$a_{2_{1S}}$	=	refrigerant sonic velocity entering the first compressor stage, m/s (ft/s)
η	=	isentropic efficiency, % (%), for a two-stage compressor with vapor injection
$\dot{m}_1$	=	refrigerant mass flow rate entering the compressor, kg/s (lb_m/h)
$\dot{m}_2$	=	refrigerant mass flow rate after mixing the injection flow and inlet flow, kg/s (lb_m/h)
$h_{2_{1S}}$	=	specific enthalpy of refrigerant vapor at suction pressure and temperature entering the compressor, kJ/kg (Btu/lb_m)
$h_{2_{2S}}$	=	specific enthalpy of refrigerant vapor after mixing the intermediate pressure flow at State Point 5 with the flow at State Point 3_{1S} shall be calculated using Equation 13, kJ/kg (Btu/lb_m):

$$h_{2_{2S}} = \frac{(\dot{m}_1 h_{3_{1S}} + \dot{m}_{inj} h_5)}{\dot{m}_2} \tag{13}$$

$h_{3_{1S}}$ = specific enthalpy of refrigerant vapor at intermediate pressure following an isentropic compression of the refrigerant from compressor suction pressure and temperature, kJ/kg (Btu/lb$_m$)

$h_{3_{2S}}$ = specific enthalpy of refrigerant vapor at compressor discharge pressure following an isentropic compression of the refrigerant from State Point 2_{2S}, kJ/kg (Btu/lb$_m$)

h_5 = specific enthalpy of refrigerant injected into compressor at intermediate pressure shall be calculated based on pressure p_5 and temperature T_5 measured within 50 mm (2 in.) of the inlet to the compressor except where otherwise specified by the test plan in Section 5.1, kJ/kg (Btu/lb$_m$).

$\dot{m}_{inj}$ = refrigerant mass flow rate injected into compressor at intermediate pressure, kg/s (lb$_m$/h):

a. For a flash tank economizer as shown in Figure 4, $\dot{m}_{inj}$ shall be calculated using Equation 14 or 15.
b. For a heat exchanger economizer as shown in Figure 5, where liquid refrigerant is extracted upstream of the economizer, $\dot{m}_{inj}$ shall be measured in the liquid line at the inlet of Expansion Device 2 or calculated using Equation 16.
c. For a heat exchanger economizer as shown in Figure 6, where liquid refrigerant is extracted downstream of the economizer, $\dot{m}_{inj}$ shall be measured in the liquid line at the inlet of Expansion Device 2 or calculated using Equation 17.

$$\dot{m}_{inj} = \frac{\dot{m}_1 \times x_{vapour}}{(1 - x_{vapour})} \tag{14}$$

$$\dot{m}_{inj} = \dot{m}_2 - \dot{m}_1 \tag{15}$$

$$\dot{m}_{inj} = \frac{\dot{m}_1(h_{4_2} - h_{4_1})}{(h_5 - h_{4_2})} \tag{16}$$

$$\dot{m}_{inj} = \frac{\dot{m}_2(h_{4_2} - h_{4_1})}{(h_5 - h_{4_1})} \tag{17}$$

where

h_{4_2} = specific enthalpy of liquid refrigerant entering economizer, kJ/kg (Btu/lb$_m$)
h_{4_1} = specific enthalpy of liquid refrigerant leaving economizer, kJ/kg (Btu/lb$_m$)
x_{vapor} = quality of refrigerant in economizer based on h_{4_2} and pressure p_5, %
P = total power input to the UUT, kW

5.8.1.4 Head Factor and Isentropic Efficiency for Compressors Connected in Series with Vapor Injection. Figure 7 shows the cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a flash tank economizer. Figure 8 shows the cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted upstream of the economizer. Figure 9 shows the cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted downstream of the economizer.

The head factor and isentropic efficiency for compressors connected in series with vapor injection using an economizer shall be calculated using Equations 18 and 20 for SI units or using Equations 19 and 21 for I-P units.

Head factor and isentropic efficiency shall be calculated using the enthalpy calculated from the measured properties as well as the total properties. If the calculated Mach number at the compressor inlet or outlet is equal to or below 0.13, it shall be assumed that the difference

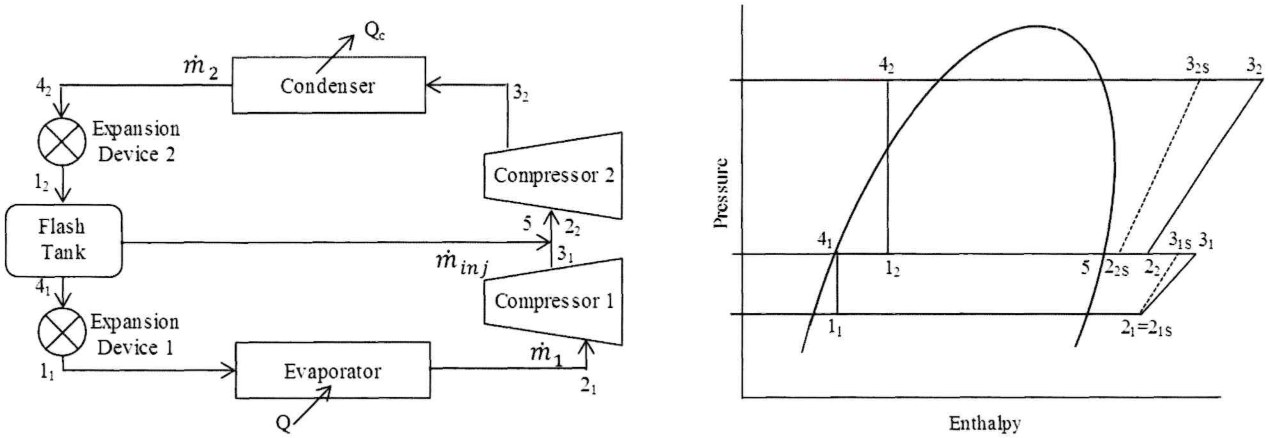


Figure 7 Cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a flash tank economizer.

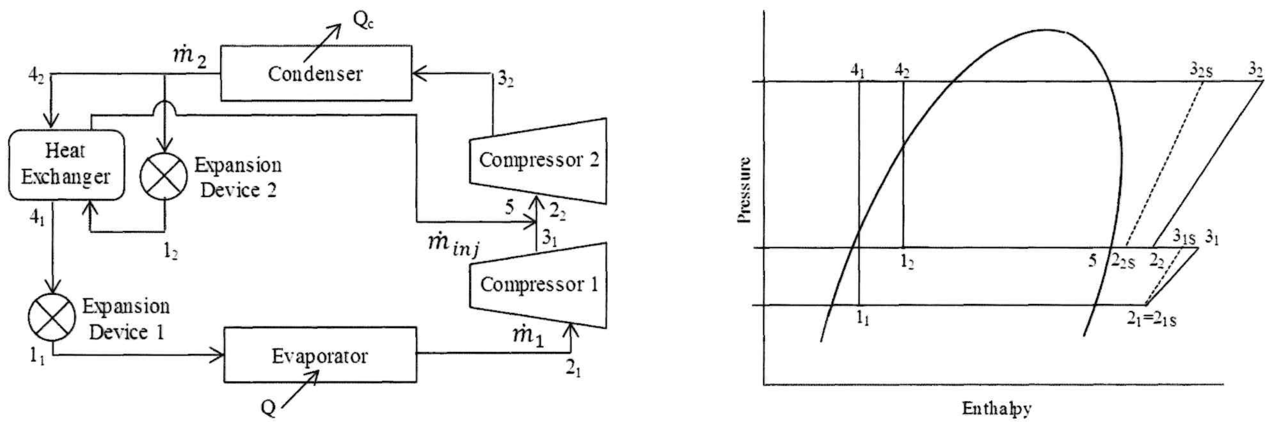


Figure 8 Cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted upstream of the economizer.

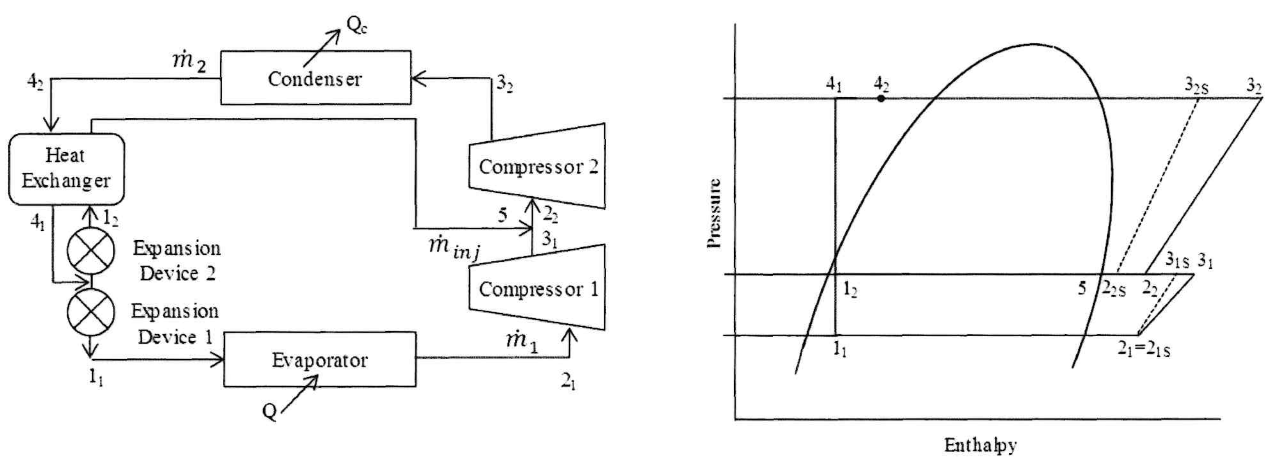


Figure 9 Cycle schematic and pressure-enthalpy diagram for compressors connected in series with vapor injection using a heat exchanger economizer and liquid refrigerant that is extracted downstream of the economizer.

between measured and total properties is negligible and measured properties shall be used to calculate head factor and isentropic efficiency. For UUTs with an inlet or outlet Mach number above 0.13, total properties shall be used to calculate head factor and isentropic efficiency. The procedure for calculating the total properties is provided in Normative Appendix A.

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 1000 \tag{18}$$

$$\mathrm{HF}_i = \frac{(h_{3_{iS}} - h_{2_{iS}})}{a^2_{2_{1S}}} \times 25,037 \tag{19}$$

$$\eta = \frac{[\dot{m}_1(h_{3_{1S}} - h_{2_{1S}}) + \dot{m}_2(h_{3_{2S}} - h_{2_{2S}})]}{P} \times 100 \tag{20}$$

$$\eta = \frac{[\dot{m}_1(h_{3_{1S}} - h_{2_{1S}}) + \dot{m}_2(h_{3_{2S}} - h_{2_{2S}})]}{P} \times 0.02931 \tag{21}$$

where

HF_i = head factor for compressor stage i

$a_{2_{1S}}$ = refrigerant sonic velocity entering the first compressor stage, m/s (ft/s)

η = isentropic efficiency, % (%), for compressors connected in series with vapor injection

$\dot{m}_1$ = refrigerant mass flow rate entering the first compressor, kg/s ($\mathrm{lb_m}$/h)

$\dot{m}_2$ = refrigerant mass flow rate after mixing the injection flow rate and inlet flow to the first compressor, kg/s ($\mathrm{lb_m}$/h)

$h_{2_{1S}}$ = specific enthalpy of refrigerant vapor entering the first compressor, kJ/kg (Btu/$\mathrm{lb_m}$)

$h_{2_{2S}}$ = specific enthalpy of refrigerant vapor after mixing the intermediate pressure flow at State Point 5 with the flow at State Point 3_{1S} shall be calculated using Equation 22, kJ/kg (Btu/$\mathrm{lb_m}$):

$$h_{2_{2S}} = \frac{(\dot{m}_1 h_{3_{1S}} + \dot{m}_{inj} h_5)}{\dot{m}_2} \tag{22}$$

$h_{3_{1S}}$ = specific enthalpy of refrigerant vapor at intermediate pressure following an isentropic compression of the refrigerant from compressor suction pressure and temperature, kJ/kg (Btu/$\mathrm{lb_m}$)

$h_{3_{2S}}$ = specific enthalpy of refrigerant vapor at compressor discharge pressure following an isentropic compression of the refrigerant from State Point 2_{2S}, kJ/kg (Btu/$\mathrm{lb_m}$)

h_5 = specific enthalpy of refrigerant injected into compressor at intermediate pressure shall be calculated based on pressure p_5 and temperature T_5 measured within 50 mm (2 in.) of the inlet to the compressor, except where otherwise specified by the test plan in Section 5.1, kJ/kg (Btu/$\mathrm{lb_m}$).

$\dot{m}_{inj}$ = refrigerant mass flow rate injected into compressor at intermediate pressure, kg/s ($\mathrm{lb_m}$/h)

a. For a flash tank economizer as shown in Figure 7, $\dot{m}_{inj}$ shall be calculated using Equation 23 or 24.
b. For a heat exchanger economizer as shown in Figure 8, where liquid refrigerant is extracted upstream of the economizer, $\dot{m}_{inj}$ shall be measured in the liquid line at the inlet of Expansion Device 2 or calculated using Equation 25.
c. For a heat exchanger economizer as shown in Figure 9, where liquid refrigerant is extracted downstream of the economizer, $\dot{m}_{inj}$ shall be measured in the liquid line at the inlet of Expansion Device 2 or calculated using Equation 26.

$$\dot{m}_{inj} = \frac{\dot{m}_1(x_{vapour})}{(1 - x_{vapour})} \tag{23}$$

$$\dot{m}_{inj} = \dot{m}_2 - \dot{m}_1 \tag{24}$$

$$\dot{m}_{inj} = \frac{\dot{m}_1(h_{4_2} - h_{4_1})}{(h_5 - h_{4_2})} \tag{25}$$

$$\dot{m}_{inj} = \frac{\dot{m}_2(h_{4_2} - h_{4_1})}{(h_5 - h_{4_1})} \tag{26}$$

where

h_{4_2} = specific enthalpy of liquid refrigerant entering economizer, kJ/kg (Btu/lb$_m$)

h_{4_1} = specific enthalpy of liquid refrigerant leaving economizer, kJ/kg (Btu/lb$_m$)

x_{vapor} = quality of refrigerant in economizer based on h_{4_2} and pressure p_5, %

P = total power input to the UUT, kW

5.8.2 Capacity. The capacity of a UUT, if required by the test plan in Section 5.1, shall be calculated as described in Section 5.8.2.

5.8.2.1 The capacity of a UUT without vapor injection shall be calculated using Equation 27.

$$Q = \dot{m}_1(h_2 - h_1) \tag{27}$$

where

Q = capacity of a UUT at the specified operating conditions, kW (Btu/h)

$\dot{m}_1$ = refrigerant mass flow rate entering the evaporator, kg/s (lb$_m$/h)

h_1 = specific enthalpy of refrigerant entering the evaporator, kJ/kg (Btu/lb$_m$)

h_2 = specific enthalpy of refrigerant entering the compressor from the evaporator, kJ/kg (Btu/lb$_m$)

5.8.2.2 The capacity of a UUT with vapor injection shall be calculated using Equation 28.

$$Q = \dot{m}_1(h_{2_1} - h_{1_1}) \tag{28}$$

where

Q = capacity of a UUT at the specified operating conditions, kW (Btu/h)

$\dot{m}_1$ = refrigerant mass flow rate, entering the evaporator kg/s (lb$_m$/h)

h_{1_1} = specific enthalpy of refrigerant entering the evaporator, kJ/kg (Btu/lb$_m$)

h_{2_1} = specific enthalpy of refrigerant entering the compressor from the evaporator, kJ/kg (Btu/lb$_m$)

5.8.3 Performance Factor. The performance factor shall be calculated based on the capacity and power input at specified operating conditions. The alternative forms of the performance factor are the coefficient of performance (COP), as defined in Equation 29; the energy efficiency ratio (EER), as defined in Equation 30; and ratio of power input to capacity, as defined in Equation 31.

$$\text{COP} = \frac{\text{Capacity, W}}{\text{Power Input, W}} \tag{29}$$

$$\text{EER} = \frac{\text{Capacity, Btu/h}}{\text{Power Input, W}} \tag{30}$$

$$\frac{\text{bhp}}{\text{ton}} = \frac{\text{Power Input, bhp}}{\text{Capacity, ton}} \tag{31}$$

5.8.4 Flow Factor. Flow factor, if required in the test plan, shall be calculated using Equation 32 for SI units or Equation 33 for I-P units. For multistage compressors, the flow factor shall be calculated for the first stage only.

$$FF = \frac{\dot{m}}{\rho a} \tag{32}$$

$$FF = \frac{\dot{m}}{3600 \rho a} \tag{33}$$

where

FF = flow factor, m^2 (ft^2)

$\dot{m}$ = refrigerant mass flow rate entering the compressor, kg/s (lb_m/h)

a = refrigerant sonic velocity entering the compressor, m/s (ft/s)

ρ = refrigerant density entering the compressor, kg/m^3 (lb/ft^3)

6. INSTRUMENTS

6.1 Instruments and data acquisition systems shall be selected to meet the measurement system accuracy specified in the test plan.

6.2 Measurements from the instruments shall be traceable to primary or secondary standards calibrated by the National Institute of Standards and Technology (NIST) or to the Bureau International des Poids et Mesures (BIPM) if a National Metrology Institute (NMI) other than NIST is used. In either case, the indicated corrections shall be applied to meet the uncertainty stated in subsequent sections. Instruments shall be recalibrated on regular intervals that do not exceed the intervals prescribed by the instrument manufacturer, and calibration records shall be maintained. Instruments shall be installed in accordance with the instrument manufacturer's requirements, or the manufacturer's accuracy does not apply.

6.3 Instruments shall be installed and applied in accordance with the following:

a. Refrigerant mass flow rate: ASHRAE Standard 41.9[1] if a calorimeter method is a selected primary or secondary test method
b. Refrigerant mass flow rate: ASHRAE Standard 41.10[2] if a flowmeter method is a selected primary or secondary test method
c. Input power: ASHRAE Standard 41.11[3]

Informative Note: Measurement instrument test condition tolerances are included in ASHRAE Standards 41.9[1] and 41.10[2].

6.4 Flowmeter Installation and Accuracy. If ASHRAE Standard 41.10[2] is a selected primary or secondary test method, the flowmeter measurement system accuracy shall be within ±1.0% of the quantity measured unless otherwise specified in the test plan in Section 5.1.

6.5 Input Power Measurement Accuracy. Input power measurement system accuracy shall be within ±1.0% of the quantity measured unless otherwise specified in the test plan in Section 5.1.

7. COMPRESSOR TEST REPORT

7.1 Test Identification

a. Date, place, time, and duration of test
b. Operator's name

7.2 Unit Under Test Description

a. Unit under test (UUT) model number and serial number
b. Refrigerant identification in accordance with ASHRAE Standard 34[5]
c. Source of refrigerant thermodynamic property data
d. Lubricant identification

7.3 Primary Method Equipment Description

a. Identify the calorimeter or flowmeter test method selected.
b. Test apparatus description, model number, serial number, and date of calibration

7.4 Confirming Method Equipment Description

a. Identify the calorimeter or flowmeter test method selected.
b. Test apparatus description, model number, serial number, and date of calibration

7.5 Test Conditions and Limits

7.5.1 Where power input is determined by electrical power measurement, set and maintain the voltage for each phase at the motor terminal within ±1% of the voltage specified in the test plan in Section 5.1.

7.5.2 Where power input is determined by shaft power measurement, set and maintain the shaft speed within ±1% of the speed specified in the test plan.

7.5.3 Where power input is determined by the load setting on a variable-speed compressor or on a pulse-width modulated compressor, set and maintain the load within ±1% of the load specified in the test plan.

7.5.4 Set and maintain the compressor ambient air temperature within ±4°C (±7°F) of the value specified in the test plan.

7.5.5 Unless otherwise specified in the test plan or required for ambient air temperature control in Section 7.5.4, airflow from a fan shall not be directed onto the compressor.

Informative Note: Air circulation specifications in the test plan may include volumetric airflow rate, air velocity, temperature, or airflow orientation with respect to the compressor.

7.5.6 Set and maintain the compressor suction pressure within ±1% of the absolute compressor suction pressure specified in the test plan.

7.5.7 Set and maintain the suction superheat within ±1 K (±1.8°R) of the superheat specified in the test plan.

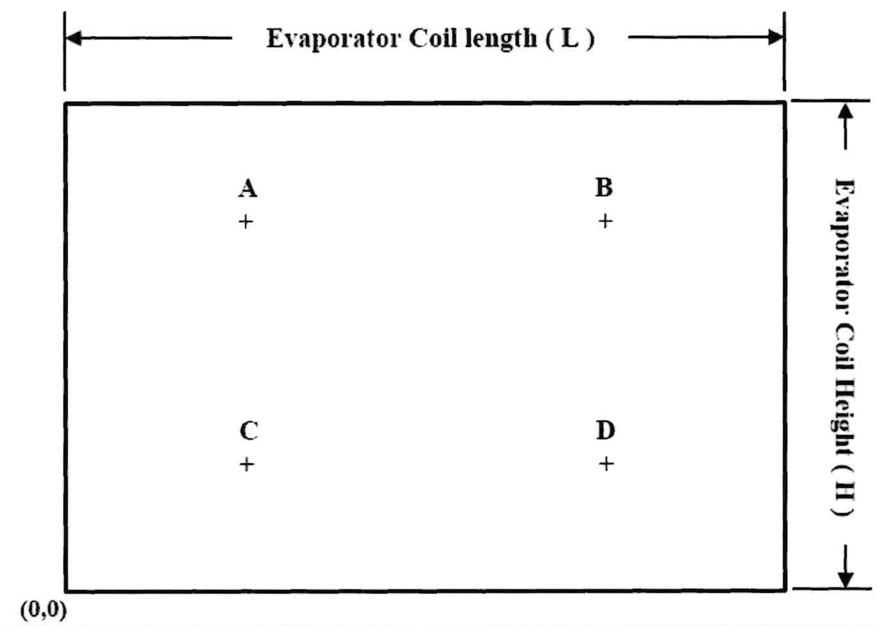
7.5.8 Vapor or liquid injection shall be performed according to the manufacturer's instructions with respect to pressure, temperature, quality, and refrigerant mass flow rate at the injection location.

7.5.9 Set and maintain the compressor discharge pressure within ±1% of the absolute compressor discharge pressure specified in the test plan.

7.5.10 The UUT manufacturer's requirements for the compressor break in procedure shall be performed prior to any test data recording unless otherwise specified in the test plan.

7.6 Measured Compressor Test Results

a. Ambient air temperature, °C (°F)
b. Specifics regarding ambient circulation over the UUT if required by the test plan
c. Barometric pressure if a pressure sensing device is referenced to atmospheric pressure
d. Electrical data if required by the test plan:
   1. Voltage, V
   2. Frequency, Hz
   3. Current, A:
      i. Single phase current, A
      ii. Three phase current, A, for all 3 legs
   4. Power factor
e. Compressor speed if required by the test plan:
   1. Shaft rotational speed, rev/s (rpm)
f. Suction pressure, kPa (psia)
g. Suction temperature, °C (°F)
h. Discharge pressure, kPa (psia)
i. Discharge temperature, °C (°F)
j. Refrigerant temperature, °C (°F), entering the calorimeter or flowmeter for the primary test method
k. Refrigerant pressure, kPa (psia), entering the calorimeter or flowmeter for the primary test method
l. Refrigerant temperature, °C (°F), leaving the calorimeter or flowmeter for the primary test method
m. Refrigerant pressure, kPa (psia), leaving the calorimeter or flowmeter for the primary test method



Dry bulb temperature sensor A coordinates: ([0.25×L ± 3 mm (0.1 in)], [0.75×H ± 3 mm (0.1)(in)])
Dry bulb temperature sensor B coordinates: ([0.75×L ± 3 mm (0.1 in)], [0.75×H ± 3 mm (0.1)(in)])
Dry bulb temperature sensor C coordinates: ([0.25×L ± 3 mm (0.1 in)], [0.25×H ± 3 mm (0.1)(in)])
Dry bulb temperature sensor D coordinates: ([0.75×L ± 3 mm (0.1 in)], [0.25×H ± 3 mm (0.1)(in)])

Figure 10 Temperature sensor locations.

9.5.9 Set and maintain the compressor discharge pressure within ±1% of the absolute compressor discharge pressure specified in the test plan.

9.5.10 The UUT manufacturer's requirements for the compressor break in procedure shall be performed prior to any test data recording unless otherwise specified in the test plan.

9.6 Measured Condensing Unit Test Results

a. Ambient temperature, °C (°F)
b. Specifics regarding ambient airflow circulation over the UUT if required by the test plan
c. Barometric pressure if a pressure sensing device is referenced to atmospheric pressure
d. Electrical data if required by the test plan:
   1. Voltage, V
   2. Frequency, Hz
   3. Current, A:
      i. Single phase current, A
      ii. Three phase current, A, for all 3 legs
   4. Power factor
e. Compressor speed if required by the test plan:
   1. Shaft rotational speed, rev/s (rpm)
f. Suction pressure, kPa (psia)
g. Suction temperature, °C (°F)
h. Discharge pressure, kPa (psia)
i. Discharge temperature, °C (°F)

j. Refrigerant temperature, °C (°F), entering the calorimeter or flowmeter for the primary test method
k. Refrigerant pressure, kPa (psia), entering the calorimeter or flowmeter for the primary test method
l. Refrigerant temperature, °C (°F), leaving the calorimeter or flowmeter for the primary test method
m. Refrigerant pressure, kPa (psia), leaving the calorimeter or flowmeter for the primary test method
n. Refrigerant temperature, °C (°F), entering the calorimeter or flowmeter for the confirming test method
o. Refrigerant pressure, kPa (psia), entering the calorimeter or flowmeter for the confirming test method
p. Refrigerant temperature, °C (°F), leaving the calorimeter or flowmeter for the confirming test method
q. Refrigerant pressure, kPa (psia), leaving the calorimeter or flowmeter for the confirming test method
r. Refrigerant temperature, °C (°F), entering the injection points if vapor injection is included
s. Refrigerant pressure, kPa (psia), entering the injection points if vapor injection is included
t. Lubricant circulation rate, % by mass, for the primary test
u. Refrigerant mass flow rates:
   1. Refrigerant mass flow rate, kg/s (lbm/h), for the primary test
   2. Refrigerant mass flow rate, kg/s (lbm/h), for the confirming test in accordance with Section 5.2
   3. If supercritical fluid or vapor injection is not included, or if injection is included and the injected refrigerant mass flow rate is calculated using Equation 14, 15, 16, 17, 23, 24, 25, or 26, then either of the following may be selected for the primary and confirming tests:
      i. Total refrigerant mass flow rate entering the compressor, kg/s (lbm/h)
      ii. Total refrigerant mass flow rate leaving the compressor, kg/s (lbm/h)
   4. If supercritical fluid or vapor injection is included and the injected refrigerant mass flow rate is not calculated using Equation 14, 15, 16, 17, 23, 24, 25, or 26, then select one of the methods below:
      i. Total refrigerant mass flow rate kg/s (lbm/h), measured at each of the following locations, entering the compressor, leaving the compressor, and entering the injection points
      ii. Total refrigerant mass flow rate kg/s (lbm/h), measured at any two locations, with confirming measurements at both locations
v. Compressor torque, N·m ($ft \cdot lb_f$), if required by the test plan
w. Power input to condensing unit including ancillaries, W (hp)
x. Average condenser inlet dry-bulb air temperature, °C (°F), if the condensing unit is air-cooled or evaporatively cooled
y. Average condenser inlet wet-bulb air temperature, °C (°F) if the condensing unit is evaporatively cooled
z. Liquid-cooled condenser inlet temperature if the condensing unit is liquid cooled
aa. Liquid-cooled condenser outlet temperature if the condensing unit is liquid cooled

9.7 Calculated Compressor Test Results

a. Uncertainty in refrigerant mass flow rate, kg/s (lb_m /h)
b. Uncertainty in power input, W (hp)
c. Flow factor, m^2 (ft^2)
d. Head factor, (kJ/kg)/(kJ/kg) [(Btu/lbm)/(Btu/lbm)]
e. Compressor isentropic efficiency, %
f. Performance Factor, not less than one of the following if required by the test plan:
   1. Coefficient of Performance, W/W
   2. Energy Efficiency Ratio, (Btu/W·h)
   3. Ratio of power input to capacity, bhp/ton
g. Capacity, kW (Btu/h), if required by the test plan

10. REFERENCES

2021 1. ASHRAE. ~~2018~~. ANSI/ASHRAE Standard 41.9, *Calorimeter Test Methods for Mass Flow Measurements of Refrigerants*. Atlanta: ASHRAE.

2020 2. ASHRAE. ~~2013~~. ANSI/ASHRAE Standard 41.10, *Flowmeter Test Methods for Mass Flow Measurement of Refrigerants*. Atlanta: ASHRAE.

2020 3. ASHRAE. ~~2014~~. ANSI/ASHRAE Standard 41.11, *Standard Methods for Power Measurement*. Atlanta: ASHRAE.

4. NIST. 2019. NIST *Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (REFPROP)*, NIST Standard Reference Database 23, Version 10, National Institute of Standards and Technology, Gaithersburg, MD.

5. ASHRAE. 2019. ANSI/ASHRAE Standard 34, *Designation and Safety Classification of Refrigerants*. Atlanta: ASHRAE.

2021 6. ASHRAE. ~~2014~~. ANSI/ASHRAE Standard 41.6, *Standard Method for Humidity Measurement*. Atlanta: ASHRAE.

7. ASME. 2014. ASME Standard PTC 10, *Performance Test Code on Compressors and Exhausters*. New York, NY: The American Society of Mechanical Engineers.

Informative Notes:

1. Reference 1 is only required if a selected primary or secondary test method is a calorimeter method.
2. Reference 2 is only required if a selected primary or secondary test method is a flowmeter method.
3. Reference 6 is only required if the condensing unit is evaporatively cooled.

(This is a normative appendix and is part of the standard.)

NORMATIVE APPENDIX A
CALCULATION OF TOTAL PROPERTIES AT COMPRESSOR INLETS AND OUTLETS

A1. TOTAL PROPERTIES

Refrigerant compressors generally do not operate with ideal gases; therefore, the techniques used for ideal gazes will not provide useful results. A method for determining the real gas total properties for locations where velocities are high is presented below.

When total temperature and total pressure probes are used in the measurement of the inlet and outlet flow from the compressor, the total enthalpy may be calculated directly from the total measurements taken.

The concept of a recovery factor is used to calculate the static and total enthalpy of the flow in the gas stream. The recovery factor is defined by Equation A-1.

$$r_f = \frac{h_{measured} - h_{static}}{h_{total} - h_{static}} \tag{A-1}$$

For temperature measurement devices, this standard will utilize a recovery factor, r_f of 0.65 as shown in ASME PTC 10[7]. This recovery factor represents the relationship between static, measured, and total enthalpy for the measurement device. The recovery factor for the condenser inlet enthalpy will be different (generally higher) than that for the measurement device.

The static temperature will be calculated using an iterative process. To start the process, it is usually assumed that $T_{static} = T_{measured}$ for iteration 1. The value of static temperature is then determined using the following steps:

1. Compute measured enthalpy $h_{measured}$ from P_{static} and $T_{measured}$.
2. Determine ρ_{static} from P_{static} and T_{static}.
3. Compute the velocity using Equation A-2.

$$V = \frac{60w}{\rho_{static} A} \tag{A-2}$$

4. Compute kinetic energy using Equation A-3.

$$k_e = \frac{V^2}{2g_c J} \tag{A-3}$$

5. Compute the static enthalpy using Equation A-4.

$$h_{static} = h_{measured} - 0.65K_e \tag{A-4}$$

6. Compute the resulting static temperature T'_{static} using P_{static} and h_{static}.

Steps 2 through 6 are repeated until T'_{static} and T_{static} are within an acceptable tolerance (a typical number would be 0.28 K [0.05°R]). Use T'_{static} as the value of T_{static} for the succeeding iteration.

After the static temperature algorithm has converged to the correct static temperature, the total enthalpy is determined from $h_{total} = h_{static} + K_e$.

Then the entropy s is calculated using P_{static} and T_{static} as inputs. Since the difference from total to static conditions are based on an adiabatic and reversible change in speed, the entropies for the static and total properties are the same.

The total pressure is calculated from the total enthalpy h_{total} and entropy s.

The total temperature T_{total} is calculated from the total pressure and total enthalpy.

The Mach number of the fluid entering or leaving the compressor shall be calculated by dividing the velocity determined in Step 3 by the acoustic velocity as shown in Equation A-5.

$$M = \frac{V}{a} \tag{A-5}$$

A2. SYMBOLS

r_f = total temperature recovery factor
h = enthalpy
a = acoustic velocity
M = Mach number
P = pressure
S = entropy
T = temperature
T' = estimated temperature
V = velocity at the pressure measurement location
w = mass flow rate through the compressor (stage)
ρ = density
A = area at the pressure measurement location
k_e = kinetic energy
g_c = gravitational constant
J = units conversion

A3. SUBSCRIPTS

total = property at total conditions
static = property at static conditions
measured = value of the property measured during test

(This appendix is not part of this standard. It is merely informative and does not contain requirements necessary for conformance to the standard. It has not been processed according to the ANSI requirements for a standard and may contain material that has not been subject to public review or a consensus process. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX B
BIBLIOGRAPHY

2019 ANSI/ASHRAE Standard 15-~~2013;~~ *Safety Standard for Refrigeration Systems and Addenda*. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA

(This appendix is not part of this standard. It is merely informative and does not contain requirements necessary for conformance to the standard. It has not been processed according to the ANSI requirements for a standard and may contain material that has not been subject to public review or a consensus process. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX C
CALCULATION OF COMPRESSOR EFFICIENCY

C1. TOTAL ENTHALPY

This standard explicitly requires the calculation of efficiency and head factor using both measured and total properties. For this standard, the internal workings of the compressor are considered to be within the control volume, which defines the boundary of the compressor. The flow streams in and out of the compressor then must enter through the inlet and outlet flanges of the compressor. The energy equation for this scenario would be written as shown in Equation C-1.

$$h_{static,in} + \frac{v_{in}^2}{2g_c} + \gamma_{in} + W = h_{static,out} + \frac{v_{out}^2}{2g_c} + \gamma_{out} \qquad \text{(C-1)}$$

For most compressors, the elevation difference between the inlet and outlet is insignificant, therefore $\gamma_{in} = \gamma_{out}$. Rearranging Equation C-1, the result is Equation C-2.

$$W = \left(h_{static,out} + \frac{v_{out}^2}{2g_c}\right) - \left(h_{static,in} + \frac{v_{in}^2}{2g_c}\right) \qquad \text{(C-2)}$$

By definition, the term

$$\left(h_{static,in} + \frac{v_{in}^2}{2g_c}\right)$$

is the total enthalpy of the compressor inlet $h_{t,in}$. The term

$$\left(h_{static,out} + \frac{v_{out}^2}{2g_c}\right)$$

is likewise the total enthalpy at the outlet of the compressor $h_{t,out}$. Using these terms, Equation C-2 becomes Equation C-3.

$$W = h_{total,out} - h_{total,in} = h_{out} - h_{in} \qquad \text{(C-3)}$$

This is the form of the energy equation that is used to derive the equations for efficiency and head factor. For compressors with low entrance and exit velocities, the kinetic energy terms $(v_{out}^2/2g_c)$ and $(v_{in}^2/2g_c)$ will be negligible and are ignored. For some centrifugal compressors, entrance and/or exit speeds will be such that *Mach numbers* will be of the order of 0.2 where these terms can easily result in a ±3% change in calculated efficiency.

The amount of the kinetic energy recovered as pressure in the condenser will depend on the design of the compressor to condenser flow path.

C2. SYMBOLS

h = enthalpy of the gas entering or exiting the compressor

v = velocity of the gas entering or exiting the compressor

g_c = gravitational constant

W = work into the compressor

γ = elevation head

C3. SUBSCRIPTS

in	=	property at the compressor inlet flange
out	=	property at the compressor exit flange
static	=	a static property, the value that would be measured by an instrument moving with the fluid
total	=	a total property, the value that would be measured if the fluid velocity were to be reduced to zero in an isentropic process

(This appendix is not part of this standard. It is merely informative and does not contain requirements necessary for conformance to the standard. It has not been processed according to the ANSI requirements for a standard and may contain material that has not been subject to public review or a consensus process. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX D
THERMODYAMIC STATE POINTS

D1. THERMODYNAMIC STATE POINTS

Figure D-1 shows a system with a single-stage compressor. If required for cooling, the cycle may include liquid injection. Heat transfer into the evaporator Q transforms the refrigerant into a superheated vapor between State Points 1 and 2. The compressor increases the refrigerant pressure between State Points 2 and 3. State Point 3_S illustrates the compressor discharge condition that would correspond to an isentropic compression process. Liquid refrigerant injection may be included, but it will not impact the calculation of isentropic efficiency. Heat transfer out from the condenser Q_c transforms the refrigerant into a subcooled liquid at State Point 4. The remaining step in the cycle between State Points 4 and 1 is an isenthalpic pressure decrease through an expansion device to transform the refrigerant to the conditions at the evaporator inlet.

Figure D-2 shows a system with two compressors connected in series and a flash tank economizer. Heat transferred into the evaporator Q transforms the refrigerant into a superheated vapor between State Points 1_1 and 2_1. State 2_{1S} represents the inlet conditions for an isentropic compression process and is identical to State Point 2_1. The first compressor increases the refrigerant pressure between State Points 2_1 and 3_1. State Point 3_{1S} illustrates the compressor discharge condition that would correspond to an isentropic compression process for the first compressor. The intermediate pressure refrigerant leaving the economizer as saturated vapor at thermodynamic State Point 5 with a flow rate of $\dot{m}_{inj}$ combines with the first compressor discharge flow $\dot{m}_1$ to enter the second compressor with a flow rate of $\dot{m}_2$ at State Point 2_2. The second compressor increases the refrigerant pressure between State Points 2_2 and 3_2. State Point 2_{2S} illustrates the thermodynamic state that results from the intermediate pressure flow from State Point 5 mixing with the refrigerant at State 3_{1S}. State Point 3_{2S} illustrates the second compressor discharge condition that would correspond to an isentropic compression process for the second compressor from the suction gas State 2_{2S}. Heat transfer out from the condenser Q_c transforms the refrigerant into a liquid at State Point 4_2. The refrigerant expands isenthalpically through an expansion device to the intermediate pressure State Point 1_2, where it separates in a flash tank to a saturated vapor at State Point 5 and a saturated liquid at State Point 4_1. The intermediate pressure saturated vapor flow $\dot{m}_{inj}$ combines with the first compressor discharge flow $\dot{m}_1$ while the saturated liquid flow $\dot{m}_1$ expands isenthalpically through a metering device to State Point 1_1 at the evaporator inlet.

Figure D-3 shows a system with two compressors connected in series and a heat exchanger economizer. Heat transferred into the evaporator Q transforms the refrigerant into a superheated vapor between State Points 1_1 and 2_1, where 2_{1S} is identical to 2_1. The first compressor increases the refrigerant pressure between State Points 2_1 and 3_1. State Point 3_{1S} illustrates the compressor discharge condition that would correspond to an isentropic compression process for the first compressor. The intermediate pressure refrigerant leaving the economizer as saturated vapor at thermodynamic State Point 5 with a flow rate of $\dot{m}_{inj}$ combines with the first compressor discharge flow $\dot{m}_1$ to enter the second compressor with a flow rate of $\dot{m}_2$ at State Point 2_2. The second compressor increases the refrigerant pressure between State Points 2_2 and 3_2. State Point 2_{2S} illustrates the thermodynamic state that results from the intermediate pressure flow from State Point 5 mixing with the refrigerant at State 3_{1S}. State Point 3_{2S} illustrates the second compressor discharge condition that would correspond to an isentropic compression process for the second compressor from suction gas State 2_{2S}. Heat transfer out from the condenser Q_c transforms the refrigerant into a liquid at State Point 4_2. One part of the refrigerant expands isenthalpically through an expansion device to the intermediate pressure State Point 1_2, passing into a heat exchanger and taking the heat from the remaining liquid refrigerant that flows through the same heat exchanger. The intermediate pressure vapor flow $\dot{m}_{inj}$ at State Point 5 combines with the first compressor discharge flow $\dot{m}_1$ while subcooled liquid flow $\dot{m}_1$ expands isenthalpically through a metering device to State Point 1_1 at the evaporator inlet.

Figure D-4 shows a system with vapor injection and heat exchanger economization. Heat transferred into the evaporator Q transforms the refrigerant into a superheated vapor between State Points 1_1 and 2_1, where 2_{1S} is identical to 2_1. The initial compression increases the refrigerant pressure between State Points 2_1 and 3_1. State Point 3_{1S} illustrates the compression condition that would correspond to an isentropic compression process to the intermediate pressure. The intermediate pressure refrigerant leaving the economizer as saturated vapor at State Point 5 with a flow rate of $\dot{m}_{inj}$ combines with the intermediate compression condition $\dot{m}_1$ to continue compression with a flow rate of $\dot{m}_2$ from State Point 2_2. The postinjection compression increases the refrigerant pressure between State Points 2_2 and 3_2. State Point 2_{2S} illustrates the thermodynamic state that results from the intermediate pressure flow from State Point 5 mixing with the refrigerant at State 3_{2S}. State Point 3_{2S} illustrates the compressor discharge condition that would correspond to an isentropic compression process for the postinjection vapor from state 2_{2S}. Heat transfer out from the condenser (Q_c) transforms the refrigerant into a liquid at State Point 4_2. One part of the refrigerant expands isenthalpically through a metering device to the intermediate pressure State Point 1_2, passing into a heat exchanger and taking the heat from the remaining liquid refrigerant that flows through the same heat exchanger. The intermediate pressure vapor flow $\dot{m}_{inj}$ at State Point 5 combines with the first compressor discharge flow $\dot{m}_1$, while subcooled liquid flow $\dot{m}_1$ expands isenthalpically through a metering device to State Point 1_1 at the evaporator inlet.

Figure D-5 shows a system with vapor injection and a flash tank economizer. Heat transferred into the evaporator Q transforms the refrigerant into a superheated vapor between State Points 1_1 and 2_1, where 2_{1S} is identical to 2_1. The initial compression process increases the refrigerant pressure between State Points 2_1 and 3_1. State Point 3_{1S} illustrates the compression condition that would correspond to an isentropic compression process to the intermediate pressure. The intermediate pressure refrigerant leaving the economizer as saturated vapor at thermodynamic State Point 5 with a flow rate of $\dot{m}_{inj}$ combines with the intermediate compression condition $\dot{m}_1$ to continue compression with a flow rate of $\dot{m}_2$ from State Point 2_2. The postinjection compression increases the refrigerant pressure between State Points 2_2 and 3_2. State Point 2_{2S} illustrates the thermodynamic state that results from the intermediate pressure flow from State Point 5 mixing with the refrigerant at State 3_{1S}. State Point 3_{2S} illustrates the compressor discharge condition that would correspond to an isentropic compression process from State Point 2_{2S}. Heat transfer out from the condenser Q_c transforms the refrigerant into a liquid at State Point 4_2. The refrigerant expands isenthalpically through an expansion device to the intermediate pressure State Point 1_2, where it separates in a flash tank to a saturated vapor at State Point 5 and a saturated liquid at State Point 4_1. The intermediate pressure saturated vapor flow $\dot{m}_{inj}$ combines with the flow in the compressor $\dot{m}_1$ while the saturated liquid flow $\dot{m}_1$ expands isenthalpically through a metering device to State Point 1_1 at the evaporator inlet.

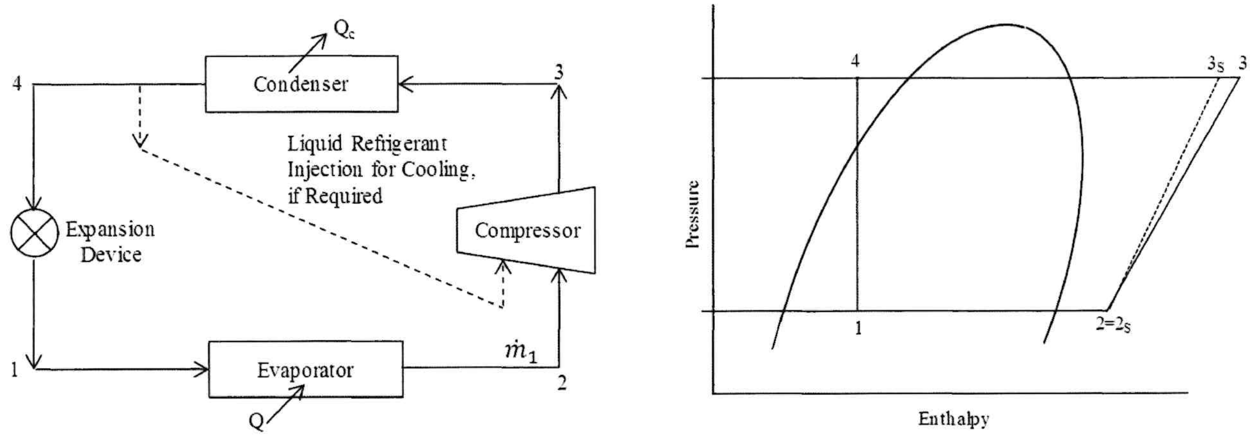


Figure D-1 Cycle schematic and pressure-enthalpy diagram for a single-stage compressor.

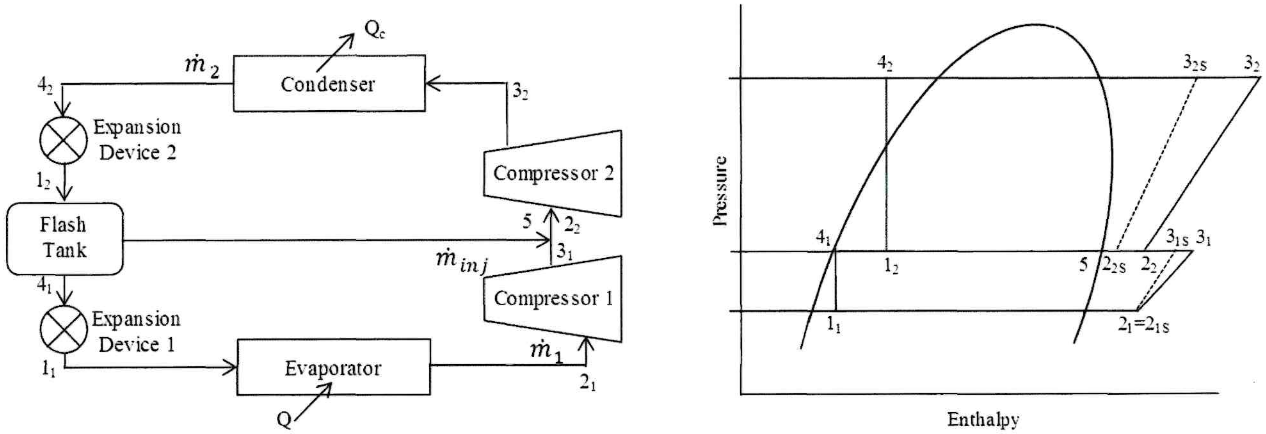


Figure D-2 Cycle schematic and pressure-enthalpy diagram for two compressors in series with a flash tank economizer.

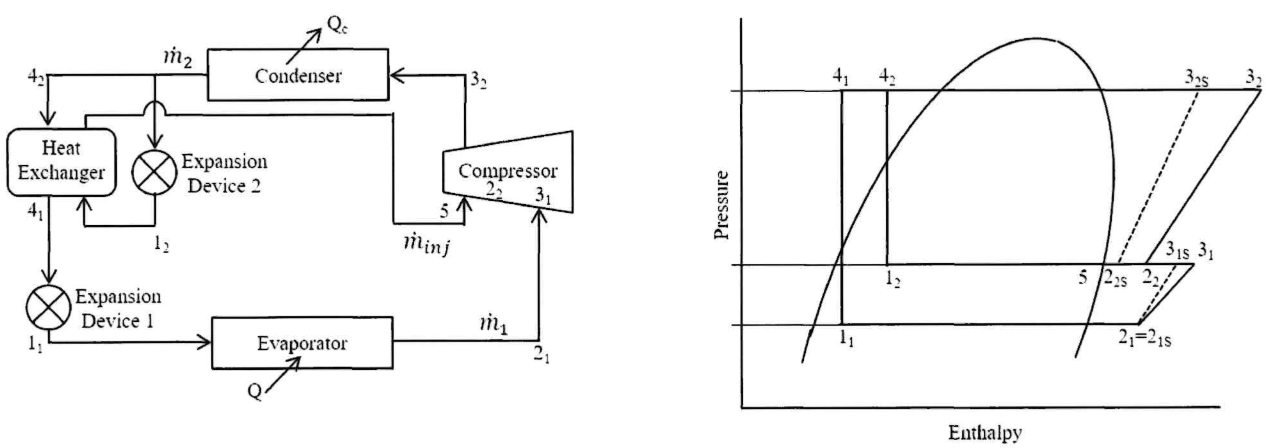


Figure D-3 Cycle schematic and pressure-enthalpy diagram for two compressors connected in series with a heat exchanger economizer.

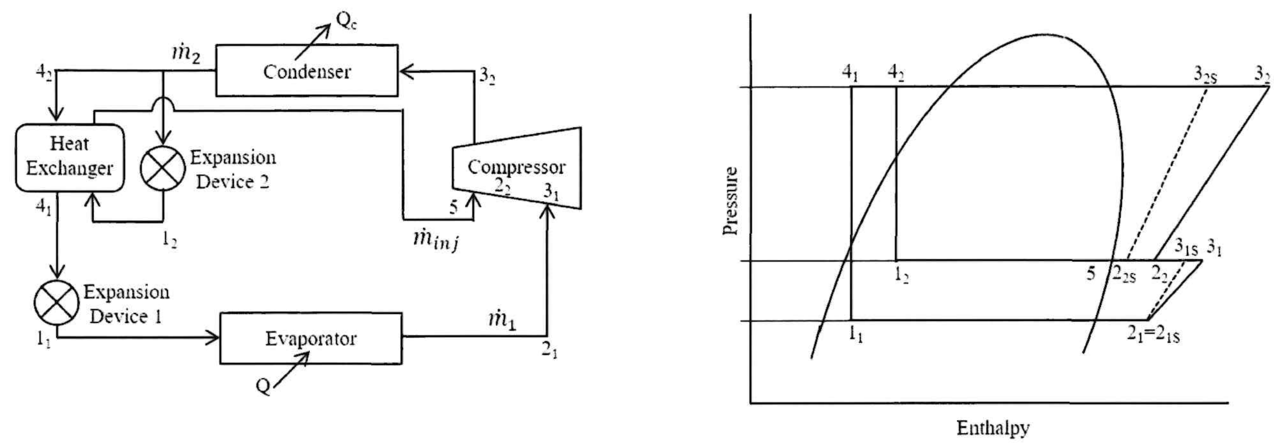


Figure D-4 Cycle schematic and pressure-enthalpy diagram for one compressor with vapor injection and a heat exchanger economizer.

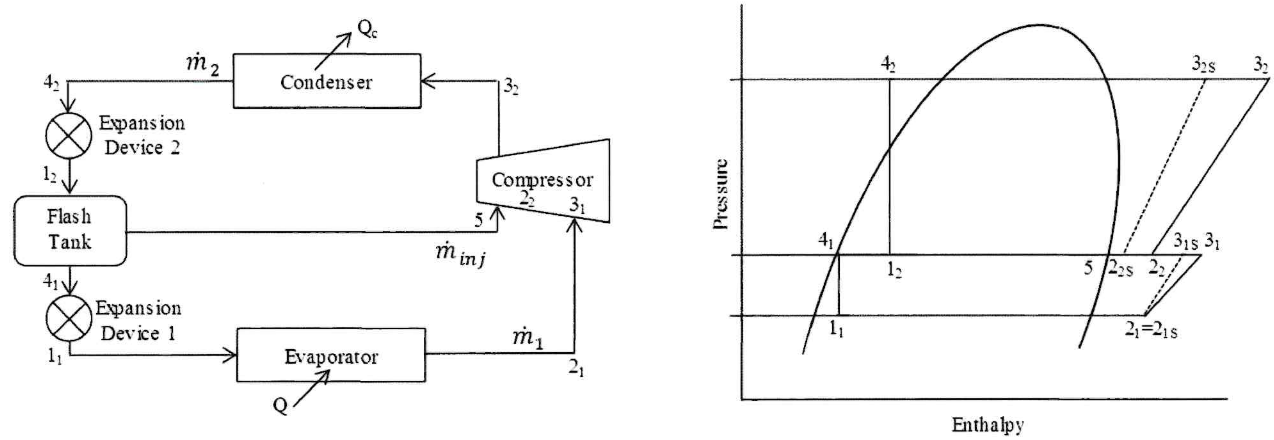


Figure D-5 Cycle schematic and pressure-enthalpy diagram for one compressor with vapor injection and a flash tank economizer.

POLICY STATEMENT DEFINING ASHRAE'S CONCERN FOR THE ENVIRONMENTAL IMPACT OF ITS ACTIVITIES

ASHRAE is concerned with the impact of its members' activities on both the indoor and outdoor environment. ASHRAE's members will strive to minimize any possible deleterious effect on the indoor and outdoor environment of the systems and components in their responsibility while maximizing the beneficial effects these systems provide, consistent with accepted Standards and the practical state of the art.

ASHRAE's short-range goal is to ensure that the systems and components within its scope do not impact the indoor and outdoor environment to a greater extent than specified by the Standards and Guidelines as established by itself and other responsible bodies.

As an ongoing goal, ASHRAE will, through its Standards Committee and extensive Technical Committee structure, continue to generate up-to-date Standards and Guidelines where appropriate and adopt, recommend, and promote those new and revised Standards developed by other responsible organizations.

Through its *Handbook*, appropriate chapters will contain up-to-date Standards and design considerations as the material is systematically revised.

ASHRAE will take the lead with respect to dissemination of environmental information of its primary interest and will seek out and disseminate information from other responsible organizations that is pertinent, as guides to updating Standards and Guidelines.

The effects of the design and selection of equipment and systems will be considered within the scope of the system's intended use and expected misuse. The disposal of hazardous materials, if any, will also be considered.

ASHRAE's primary concern for environmental impact will be at the site where equipment within ASHRAE's scope operates. However, energy source selection and the possible environmental impact due to the energy source and energy transportation will be considered where possible. Recommendations concerning energy source selection should be made by its members.

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Founded in 1894, ASHRAE is a global professional society committed to serve humanity by advancing the arts and sciences of heating, ventilation, air conditioning, refrigeration, and their allied fields.

As an industry leader in research, standards writing, publishing, certification, and continuing education, ASHRAE and its members are dedicated to promoting a healthy and sustainable built environment for all, through strategic partnerships with organizations in the HVAC&R community and across related industries.

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