COMPRESSED-AIR SYSTEMS:

Eliminating the Confusion

Understanding the differences between ACFM, ICFM, FAD, and CAGI-SCFM for improved system design and equipment selection

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The process of designing and selecting equipment for an energy-efficient compressed-air system can be very confusing because of the many ways manufacturers specify system capacity, such as actual cubic feet per minute (ACFM), inlet cubic feet per minute (ICFM), free-air delivery (FAD), and Compressed Air & Gas Institute standard cubic feet per minute (CAGI-SCFM). Added to these are the many different and changing definitions of standard air used to identify SCFM airflows. If the engineer and manufacturer do not use the same method of calculating volume flow, problems with system selection can result. Indeed, the confusion over flows can make comparing bids from different manufacturers difficult and/or result in an incorrect choice of equipment.

The intent of this article is to eliminate the confusion over ACFM, ICFM, FAD, and CAGI-SCFM by providing equations and actual examples of the different flows. The article also will illustrate the significance of mass flows (pounds of dry air per hour), which, unlike volume flows, do not change when pressure, temperature, humidity, or air density change. But first, it will look at existing compressed-air systems in the United States to see if there is room for improvement in the design and selection process.

EXISTING COMPRESSED-AIR SYSTEMS

There are more than 630,000 compressed-air systems in the United States running an average of 5,476 hr and consuming more than 91 billion KWH of electricity per year. With a total installed horsepower of 32.5 million, this equipment consumes $4.2 billion of electricity per year. Compressed-air electricity consumption represents 2.66 percent of the electricity consumed by all sectors in the United States.

Table 1 is a summary of industrial-sector electricity consumption.[1] The industrial sector, which comprises manufacturing and non-manufacturing industries, accounts for 31.7 percent of the electricity consumption in the United States. As can be seen in the table, the manufacturing industry consumes the bulk (85 percent) of the more than 1 trillion KWH of electricity used by the industrial sector. The table includes a summary of the potential energy savings of motor-driven systems in the manufacturing industry, assuming a simple payback of less than three years. The savings for compressed-air systems alone is more than $714 million a year, a reduction of 17 percent. This equates to an annual average savings of $1,133 for each compressed-air system.

It is important to note that compressed-air systems account for 15.8 percent of manufacturing-industry electricity consumption. Compressed air is used to spin tools, drive cylinders and linear activators, atomize paint and other liquids, clamp work in place, clean hard-to-reach areas, drive piston air motors, blow soot, and fluidize fine particles. In addition, it is used in material handling

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and separation, grinding and drilling, pneumatic control, and sandblasting, among other applications.

THE COMPRESSOR

To achieve an energy-efficient compressed-air design and desired air quality, many factors must be considered. The first and perhaps most misunderstood is the compressor. Compressors are devices that raise fluid pressure by more than 5 psig or increase the density from inlet to discharge by more than 7 percent. There are two basic types: positive displacement (reciprocating or rotary) and dynamic (centrifugal or axial). Both are volumetric-flow devices.

Positive-displacement compressors entrap a volume of air and reduce it. Characteristics are constant flow and variable pressure ratio for a given speed. Dynamic compressors, on the other hand, depend on motion to transfer energy. Flow is continuous. The volumetric flow varies

	Kilowatt-hours per year*	Appropriate annual electricity cost
Total electricity consumption		
• Manufacturing	917,834,000,000	$42,220,364,000
• Non-manufacturing	167,563,000,000	$7,707,898,000
Total	**1,085,397,000,000**	**$49,928,262,000**
Manufacturing motor-system energy		
• Fan systems	78,727,000,000	$3,621,442,000
• Pump systems	142,690,000,000	$6,563,740,000
• Compressed-air systems	91,050,000,000	$4,188,300,000
• Other	262,961,000,000	$12,096,206,000
Total	**575,428,000,000**	**$26,469,688,000**
Non-manufacturing motor-system energy	171,677,000,000	$7,897,142,000
Total industrial motor-system energy	747,105,000,000	$34,366,830,000
Potential energy savings, manufacturing		
• Fan systems	4,330,000,000	$199,180,000
• Pump systems	28,681,000,000	$1,319,326,000
• Compressed-air systems	15,524,000,000	$714,104,000
• Motor upgrade	19,799,000,000	$910,754,000
• Motor downsizing	6,786,000,000	$312,156,000
• Motor replacement vs. rewind	4,778,000,000	$219,788,000
• Other	5,259,000,000	$241,914,000
Total	**85,157,000,000**	**$3,917,222,000**

*Assumes $0.046 per KWH

Source: "United States Industrial Electric Motor Systems Market Opportunities Assessment"

TABLE 1. Industrial-sector electricity consumption.

Increase mass-flow output	Decrease mass-flow output
Lower intake-air temperature	Higher intake-air temperature
Higher barometric pressure	Lower barometric pressure
Lower relative humidity	Higher relative humidity
Less pressure drop from inlet to compressor inlet flange	More pressure drop from inlet to compressor inlet flange
Greater air-inlet density	Lower air-inlet density
Lower inlet-specific air volume	Higher inlet-specific air volume

TABLE 2. Changes that increase or decrease compressor output.

inversely with this differential pressure across the compressor.

When used in a tool, compressed air supplies force. Force is equal to mass times acceleration; therefore, the work performed by a tool is dependent on the mass flow of air through the tool. The underlying factor in specifying compressor capacity, then, is the mass flow of air delivered by the compressor. The following equation relates volume flow to mass flow:

$$\begin{aligned}&\mathrm{m}\left(\text{lb wet air per min}\right) = \\ &\mathrm{ACFM}\left(\text{cu ft wet air per min}\right) \times \\ &\text{density}\left(\text{lb wet air per cu ft wet air}\right)\end{aligned}$$

The problem is that compressors are volumetric devices; therefore, their output is influenced by changes in inlet-air density. Air-density (pounds per cubic foot) variation is caused by changes in barometric pressure (and/or gauge pressure), air dry-bulb temperature, and water-vapor content (relative humidity). Table 2 lists some of the changes that increase or decrease compressor output.

Compressors installed at higher elevations above sea level (or lower air densities) get less air in each cubic foot of intake air than they would if they were installed at sea level. Table 3 presents performance data on ambient airflow into and out of a rotary screw compressor installed at sea level and at 10,000 ft above sea level. Note that the ACFM-at-the-inlet and FAD flows are only 1.38-percent lower at 10,000 ft, even at a much lower inlet-air density. This illustrates the compressor's ability to hold its intake-volume flow rate. There is, however, a significant change in the mass-flow rate at 10,000 ft: The mass flow into and out of the compressor is 1,798.83 lb of dry air per hour, which is 65.5 percent of the flow at sea level (2,747.60 lb of dry air per hour). Likewise, the CAGI-SCFM at 10,000 ft is 65.5 percent of the CAGI-SCFM at sea level because it is a type of mass-flow term. Note that compressor-discharge ACFM also is significantly lower at 10,000 ft. All of this illustrates that mass flow is related to air density, as well as volume flow, and that the mass flow into a compressor is equal to the mass flow leaving the compressor, provided there is no leakage.

Another loss attributed to humidity is that of water-vapor mass flow, 85 percent or more of which is removed by intercoolers, aftercoolers, and dryer systems as it enters a compressed-air system. The compressor, then, should provide the amount of mass flow required for the worst-case scenario, which is a hot, humid summer day with a low barometric (and/or gauge) pressure. In most cases, the predominant factor influencing compressor output is inlet-air temperature.

The equations for air density are:

Sea level	10,000 ft
P_{bar} = 14.696 psi	P_{bar} = 9.756 psi
P_{gauge} = 100 psig (at compressor discharge)	P_{gauge} = 100 psig (at compressor discharge)
P_{total} = 114.696 psia (at compressor discharge)	P_{total} = 109.756 psia (at compressor discharge)
Ambient air drawn into compressor T_{ob} = 60 F RH = 0 percent Specific volume = 13.1024 cu ft air per lb dry air Density = 0.076322 lb air per cu ft air ACFM = 600 cu ft per min m (lb dry air per hr) = 2,747.60	Ambient air drawn into compressor T_{ob} = 60 F RH = 0 percent Specific volume = 19.7368 cu ft air per lb dry air Density = 0.050667 lb air per cu ft air ACFM = 591.72 m (lb dry air per hr) = 1,798.83
Compressor output Leakage = 0 FAD = 600 cu ft per min (free air delivered at ambient-air properties) CAGI-SCFM = 617.47 m (lb dry air per hr) = 2,747.60 ACFM = 94.6304 cu ft per min (at 100 psig and 180 F)	Compressor output Leakage = 0 FAD = 591.72 (free air delivered at ambient-air properties) CAGI-SCFM = 404.25 m (lb dry air per hr) = 1,798.83 ACFM = 64.7422 cu ft per min (at 100 psig and 180 F)

TABLE 3. Performance data for a rotary screw compressor installed at sea level and at an elevation of 10,000 ft.

$$\text{Density}\left(\text{lb wet air per cu ft wet air}\right) = \frac{\text{MW wet air}\left(\text{lb m per mole}\right) \times P_{total}\left(\text{lb f per sq ft}\right)}{1{,}545.43\left(\text{ft lb f per mole} - {}^{\circ}\text{R}\right) \times \left(t_{db} + 459.67\right){}^{\circ}\text{R}}$$

$$\text{Density}\left(\text{lb wet air per cu ft wet air}\right) = \frac{m\left(\frac{\text{lb wet air}}{\text{hr}}\right)}{\text{ACFM}\left(\frac{\text{cu ft}}{\text{min}}\right) \times 60\left(\frac{\text{min}}{\text{hr}}\right)}$$

When there is a significant drop in inlet-air density (such as with the compressor installed at 10,000 ft in Table 3), the ACFM at the compressor intake usually does not change much. According to the above equations, to compensate for the reduced air density and unchanged inlet ACFM, the inlet mass flow must drop. Because discharge mass flow is equal to inlet mass flow (when there is no leakage), this means that the discharge mass flow must drop as well. This helps to explain why the compressor at 10,000 ft in Table 3 had a significant drop in mass flow (inlet and discharge) and CAGI-SCFM.

PERFORMANCE TERMINOLOGIES

To better understand compressor-system sizing, engineers need to know how compressor manufacturers specify capacity with ACFM, ICFM, FAD, and SCFM performance figures. These performance terminologies have frustrated engineers for many years. This section will provide definitions, examples, and equations intended to eliminate any confusion. A list of nomenclatures that will be used in this discussion is provided in Table 4.

Figure 1 is a piping diagram of a compressor system and aftercooler. Below the diagram are the actual psychrometric air properties and airflows. Above the diagram are the actual conditions converted to the psychrometric properties at the air-intake air properties.

A lot can be learned by following the mass flows in Figure 1. Mass flows, unlike ACFM flows, are not affected by changes in pressure or temperature (or water vapor, in the case of dry-air mass flow). The mass flow of dry air will remain unchanged from inlet to discharge unless there is leakage or loss attributed to the use of pneumatic controls. CAGI-SCFM—what the author prefers to call DSCFM—is a type of dry-air mass flow converted to dry standard cubic feet per minute. DSCFM represents only the dry-air mass flow at the compressor-discharge flange, the discharge mass flow after leakage. In Figure 1, however, DSCFM flows are provided at all points to reinforce that they, like dry-air mass flows, do not change unless there is leakage or loss attributed to pneumatic controls.

P_{bar} (psi) = Barometric pressure.
P_{gauge} (psig) = Air-gauge pressure.
P_{total} (psia) = Total air pressure ($P_{total} = P_{bar} + P_{gauge}$).
t_{db} (F) = Dry-bulb temperature of air.
t_{wb} (F) = Wet-bulb temperature of air.
t_{dp} (F) = Dew-point temperature of air.
RH (%) = Relative humidity.
W (lb W.V. per lb dry air) = Humidity ratio in pounds of water vapor per pound of dry air.
ACFM (actual cu ft per min) = Tested airflow in cubic feet per minute.
SCFM (std cu ft per min) = Standard airflow at a standardized air density of 0.075 lb wet air per standard cubic foot of air. SCFM in this case is a total wet airflow and is calculated by dividing the mass flow of wet air (pounds of wet air per minute) by the density of 0.075. SCFM should not be confused with CAGI-SCFM, which has a standardized air density of 0.07416 lb per cubic foot and is dry standard cubic feet per minute.
DSCFM (dry std cu ft per min) = Dry airflow in dry standard cubic feet per minute. Standard air in this case is air at 14.5 psia, 68 F, and 0-percent relative humidity. It represents the mass flow of dry air only.
CAGI-SCFM (or DSCFM) = The volume of free air in cubic feet per minute at a standardized air property of 14.5 psia, 68 F, and 0-percent relative humidity (air density of 0.07416 lb per dry standard cubic foot). This is calculated by dividing the dry-air mass flow by the standard density of 0.07416. Because the relative humidity is at 0 percent, this is dry standard cubic feet per minute.
ICFM (inlet cu ft per min) = The tested ACFM flow at the inlet flange to the compressor.
FAD (cu ft per min) = Free air delivered. Cubic feet per minute at "free-air" conditions at the compressor-discharge flange. It is the air at the outlet flange of the compressor, but is illustrated at the actual ambient temperature, pressure, and humidity. FAD equals ACFM (air intake ambient) minus leakage ACFM (wet air lost illustrated at ambient-air properties).
V_{actual} (actual cu ft per lb dry air) = Specific volume of air in cubic feet of wet air per pound of dry air.
V_{std} (dry std cu ft per lb dry air) = Specific volume of air in dry standard cubic feet per pound of dry air.
P_{ws} (psi) = ASHRAE term for the pressure of saturated pure water at the air dry-bulb temperature. Some books show it as P_{vpa}, P_v, or P_{sat}. It often is referred to as the saturated vapor pressure of water at the actual inlet temperature.
m (lb dry air per hr) = Mass flow of dry air per hour.
m (lb W.V. per hr) = Mass flow of water vapor per hour.
m (lb total per hr) = Total mass flow of dry air and water vapor per hour.
New CAGI standard air: P_{bar} = 14.5 psi (given). P_{gauge} = 0. P_{total} = 14.5 psia (given). t_{db} = 68 F (given). % RH = 0 percent (given). $MW_{dry\ air}$ = 28.9645 (per ASHRAE). Air density = 0.074162 lb per cu ft (author's calculation). Specific volume (V_{std}) = 13.48389 cu ft per lb dry air (author's calculation).

TABLE 4. Nomenclatures.

It is important to note that the definition of CAGI standard air has changed. The new standard-air properties are listed in Table 4. Figure 1 provides a few equations that can be used to calculate CAGI-SCFM flow.

Compressed-air-industry ICFM was developed to avoid the confusion caused by variable standards. This flow expresses compressor inlet volume in terms of actual inlet pressure, temperature, and humidity. The problem is that ICFM can be calculated at barometric pressure or air-inlet-flange pressure, which is approximately 0.30-psi lower than barometric pressure. Figure 1 calculates ICFM at the inlet-flange pressure, as well as at the inlet-flange dry bulb and humidity, which are the same as the ambient dry bulb and humidity. The ICFM flow is 1,021 cu ft per minute. If we calculate the ICFM at the ambient total pressure of 14.7 psia (instead of 14.39 psia), the flow will be the same as the ACFM at ambient, or 1,000 cu ft per minute. The ACFM value will change as air density changes because of variations in pressure and temperature. The mass flows of dry air and/or water vapor will not change unless there is leakage, use of pneumatic controls, or condensation.

The compressor manufacturer must state the following psychrometric properties of the ICFM flow:

- Barometric pressure (psi).
- Gauge pressure (psig).
- Total pressure (psia).
- Dry-bulb temperature.
- Relative humidity or humidity ratio.
- Air density (pounds of wet air per cubic foot of wet air).
- Specific volume (cubic feet of wet air per pound of dry air).

FAD also is a compressed-air-industry term. It is the total moist airflow (dry air and water vapor) discharged from the compressor. Although FAD airflow is expressed in cubic feet of wet air per minute, it is not the actual ACFM at the compressor discharge flange. In Figure 1, the actual ACFM is 141 cu ft per minute. The FAD airflow is a representation of the actual mass flows of dry air and water vapor, but expressed at the inlet-air psychrometric air properties. In this case, it is very important to understand the psychrometric air properties chosen to represent FAD airflow. In Figure 1, it is the inlet-air conditions shown at the start of the diagram (14.7 psia, 95 F, and 60-percent relative humidity).

The following equation can be used to calculate FAD flow:

$$\text{FAD}\left(\text{cu ft wet air per min}\right) = \frac{\dot{m}\left(\text{lb total per hr}\right)\text{ at compressor discharge flange}}{\text{ambient inlet air density}\left(\text{lb per cu ft}\right)\times 60\left(\text{min per hr}\right)}$$

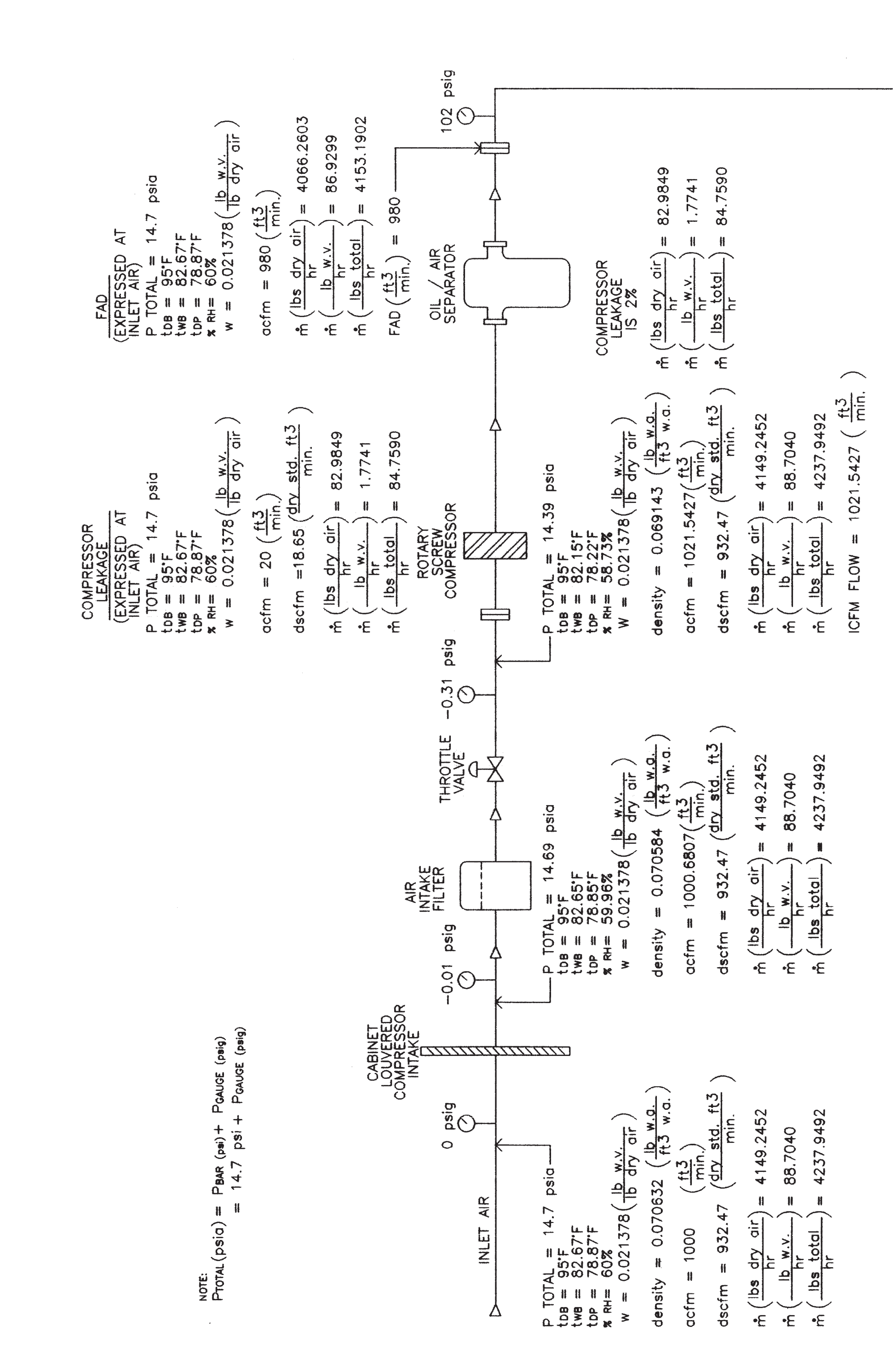
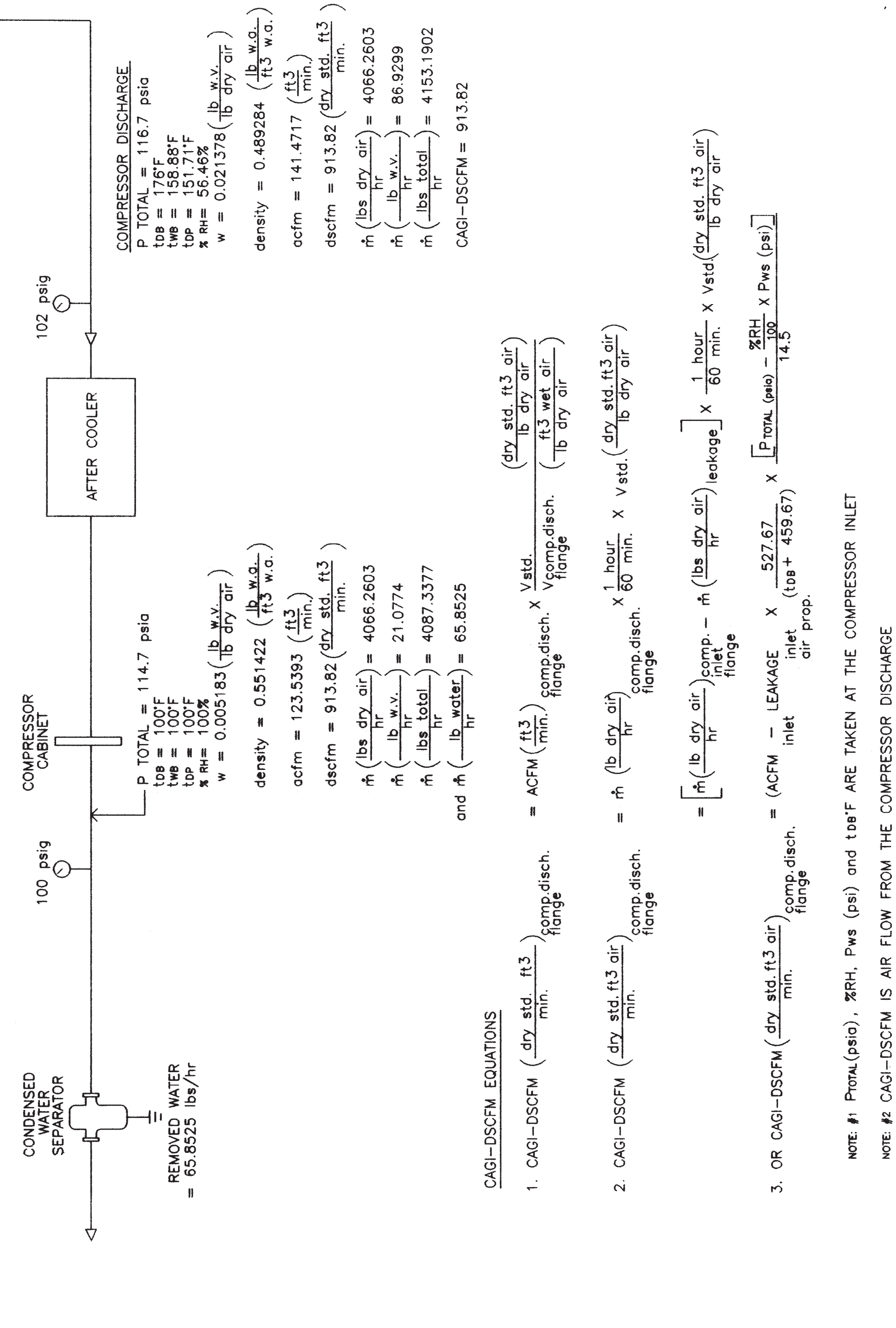


FIGURE 1. ACFM, ICFM, FAD, and DSCFM flows for a compressor system and aftercooler.

For the example in Figure 1:

$$\text{FAD}\left(\text{cu ft wet air per min}\right) = \frac{4{,}153.1902\text{ lb total per hr}}{0.070632\text{ lb per cu ft}\times 60\left(\text{min per hr}\right)} = 980$$

ACFM is the actual cubic feet per minute of wet airflow (dry air and water vapor) as determined by a Pitot-tube traverse of the duct or pipe. In Figure 1, the two-phase inlet airflow of 1,000



ACFM is made up of 966.77 cu ft per minute of dry air and 33.23 cu ft per minute of water-vapor flow. About 85 percent of the volumetric water-vapor flow will be removed by the aftercooler and refrigerated dryer, which will reduce the amount of compressed air available for use. To convert the ACFM flow to a new set of psychrometric air properties, use the following equation:

$$ACFM_{new} = ACFM_{actual} \times \frac{\text{actual air density}\left(\text{lb per cu ft}\right)}{\text{new air density}\left(\text{lb per cu ft}\right)}$$

Using Figure 1, take the ACFM at the compressor discharge (141.4717 ACFM), and convert it to ACFM at the inlet-air properties:

$$ACFM_{new} = 141.4717 \times \frac{0.489284 \text{ lb per cu ft}}{0.07623 \text{ lb per cu ft}} = 980$$

Air leakage through shafts, seals, and purge systems usually is not discussed by compressor manufacturers. We also may have some air usage for, say, pneumatic controls. This would, of course, reduce the compressor FAD or the CAGI-SCFM at the discharge of the compressor. Figure 1 assumed a loss of 2 percent. The loss is 82.9849 lb of dry air per hour and 1.7741 lb of water vapor per hour, or 20 ACFM at ambient air properties.

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In July, Part 2 of this article will discuss compressed-air systems and refrigerated dryers, including energy flows, dry-air and water-vapor flows, lube-oil circuits, lube oil in compressed air and heat recovery, and heat into plants.

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In May, Part 1 of this article reviewed the compressor-performance variables ACFM (actual cubic feet per minute), ICFM (inlet cubic feet per minute), FAD (free-air delivery), and CAGI-SCFM (Compressed Air & Gas Institute standard cubic feet per minute). Discussion was limited to the compressor and aftercooler portions of compressed-air systems. This month, that discussion expands. Figures 2a and 2b (pages 38 and 40) represent a complete diagram of a compressed-air system and refrigerated dryer, showing all energy, dry-air, and water-vapor flows; the lube-oil circuit; lube oil in the compressed air and heat recovery; and heat into the plant.

The rotary screw compressor in the diagram consumes 80 kW (273,020 Btuh) of electricity at the compressor. The motor radiates 15,016 Btuh of heat into the building, while the belt drive radiates 7,740. The remaining 250,264 Btuh is shaft power delivered to the compressor. The compression of air picks up 70,163 Btuh, while the lube oil picks up 180,101 Btuh. The air-cooled aftercooler in Figure 2a is designed to remove heat from the lube-oil and compressed-air circuits, as well as motor heat and the heat of condensation. In this illustration, the heat released into the building is 256,475 Btuh, which could be used to preheat boiler makeup water or outside makeup air. As a rough rule of thumb, a 50-hp compressor at full load rejects approximately 126,000 Btuh.

The aftercooler removes a significant amount of the water vapor drawn in at the air intake. In this case, the aftercooler removes 51.4 percent of the water that enters the compressor system. This is a significant amount that the refrigerated dryer will not have to remove.

The refrigerated dryer has an air-to-air heat exchanger, or what some call a reheater. The reheater in this case removes 10,909.60 Btuh of heat from the compressed air before it enters the refrigeration section of the dryer. This significantly lessens the burden on the refrigeration section, which removes 10,273.54 Btuh of heat from the compressed air itself. This heat is used to elevate the dry-bulb temperature of the leaving air so that it is not at 100-percent relative humidity. The heat removed by the reheater also condenses 1.7587 lb per hour of water vapor. When the compressed air leaves the refrigerated-dryer section, the dry-bulb temperature is 37 F, with a 37-F dew point (pressure dew point) and 0.000662 lb of water vapor per pound of dry air. To get a feeling for the dryness of this air, air at sea-level pressure would have to be at a dew-point temperature of –7.2 F. To put it another way, if the air were

Photo courtesy of Kaeser Compressors Inc.

A 180-hp screw compressor with variable-frequency control.

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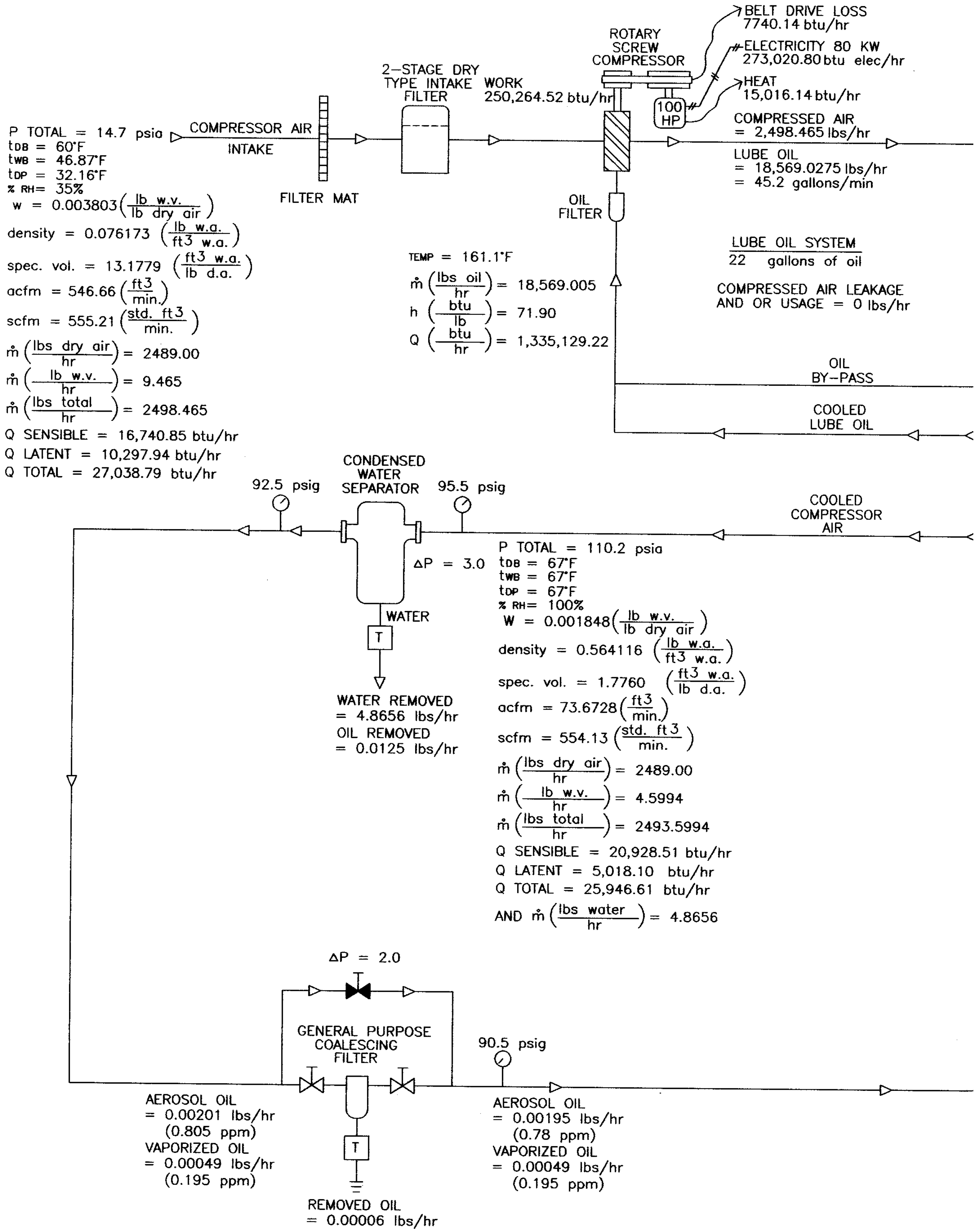
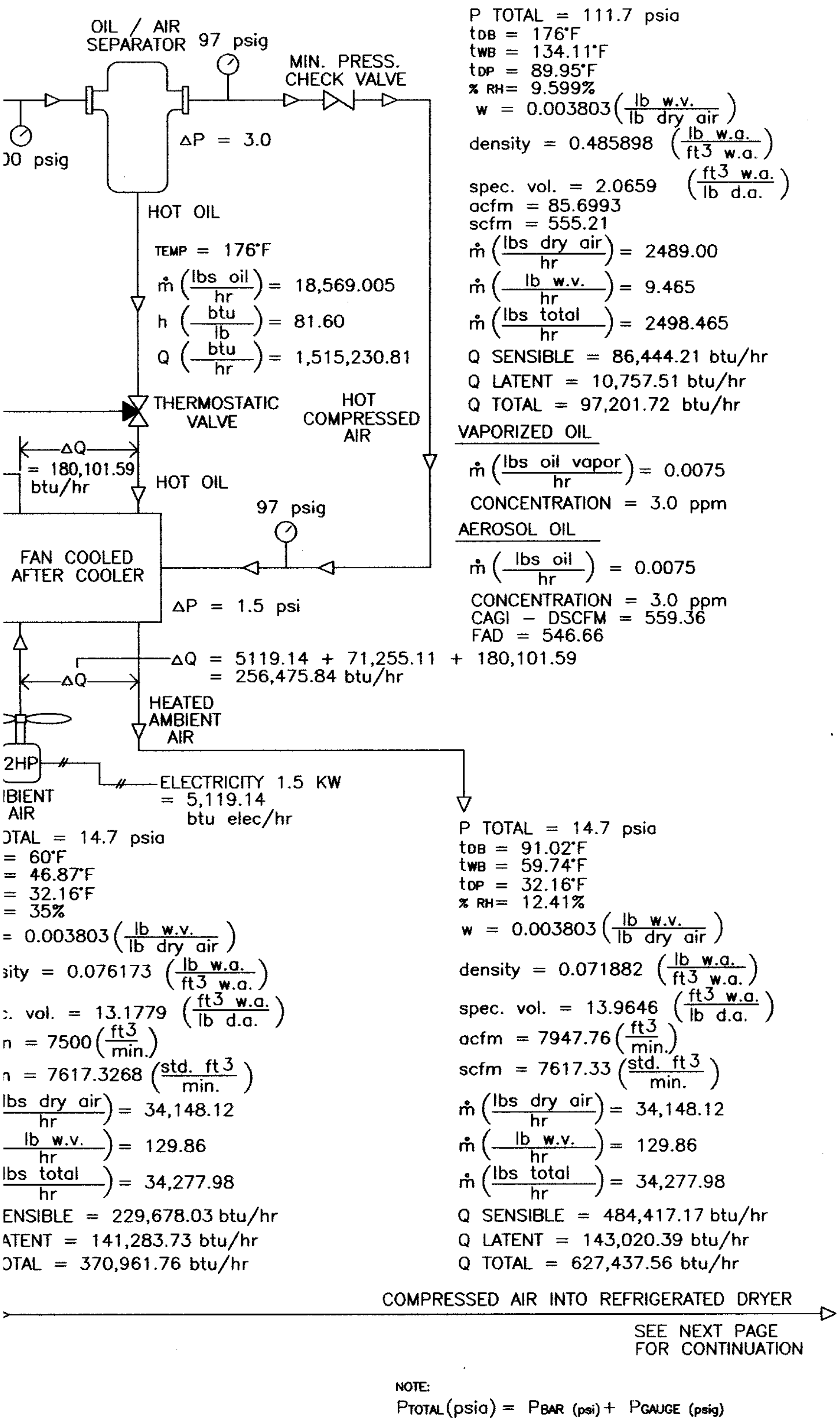


FIGURE 2A. A compressed-air system and refrigerated dryer.



at 60 F, the relative humidity would be at 6.1 percent, which, of course, is very dry. The air-cooled-condenser section of the dryer must remove heat from the refrigerant, which is heat from cooling the compressed air and heat from the refrigerant compressor. The air entering the air-cooled-condenser section then picks up the refrigerant heat and fan heat and discharges the total heat into the building. The total heat removal in this case is 19,497 Btuh, which is dumped into the building. To make use of this waste-heat energy, some plants use water-cooled condenser sections, which can be used to heat process water, or a heat-recovery unit, which can be used to supplement building heating.

Water-vapor removal in figures 2a and 2b occurs at the aftercooler, the reheater, and the refrigeration section of the dryer system. The amount of water vapor removed by each section is listed in Table 5. Water vapor entered the compressor at a rate of 9.465 lb per hour. After all removal, the rate was 1.6468 lb per hour, a reduction of 82.6 percent.

The figures also illustrate the amount of oil released into the compressed air. This oil exists as vaporized oil and aerosol (colloidal particles) oil. The total amount released into the compressed air after the air/oil separator is 0.0150 lb per hour, or 6.0 ppm (mass). The oil in this example is a mineral oil. Excessive foaming in the compressor lube-oil circuit will cause an increase in lubricant carryover in the compressed-air stream. The annual oil loss, assuming 8,400-hr-per-year operation, is 18.4 gal. The oil-holding capacity of this compressor is 22 gal, so it will be necessary for maintenance crews to replace the lost oil throughout the year. The oil remaining in the air after the high-efficiency coalescing filter is 0.00003 lb per hour; therefore, 99.8 percent (0.01497 lb per hour) was removed by the entire system. The coalescing filters remove the aerosol oil in the compressed air. Vaporized oil can be removed by dropping the temperature of the compressed air, which condenses the oil, or by using a vapor-removal filter. One type of vapor-removal filter is the activated-carbon cartridge filter bound in a nonwoven polyester substrate.

The compressed-air system removes a total of 7.8182 lb of water per hour and

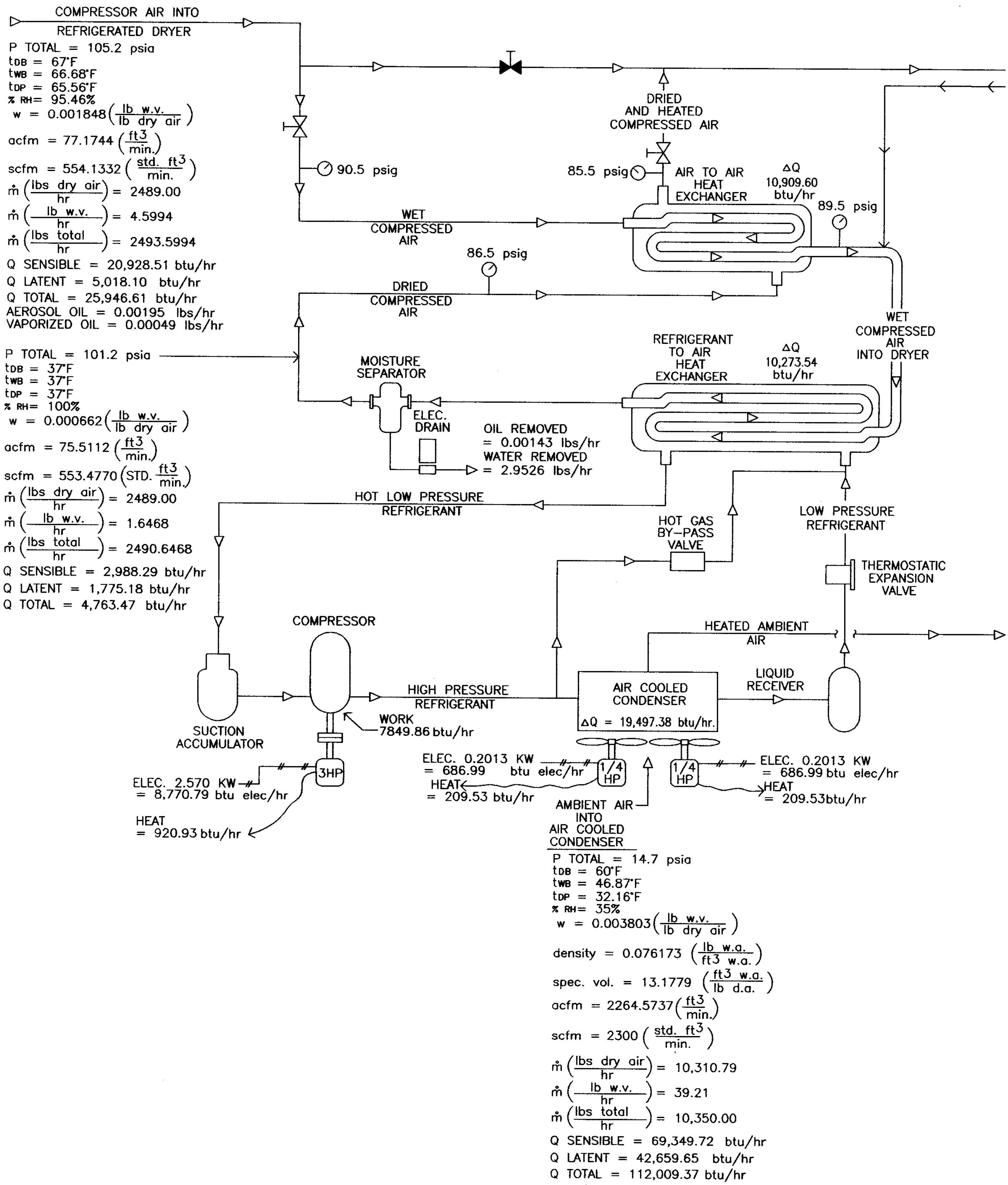
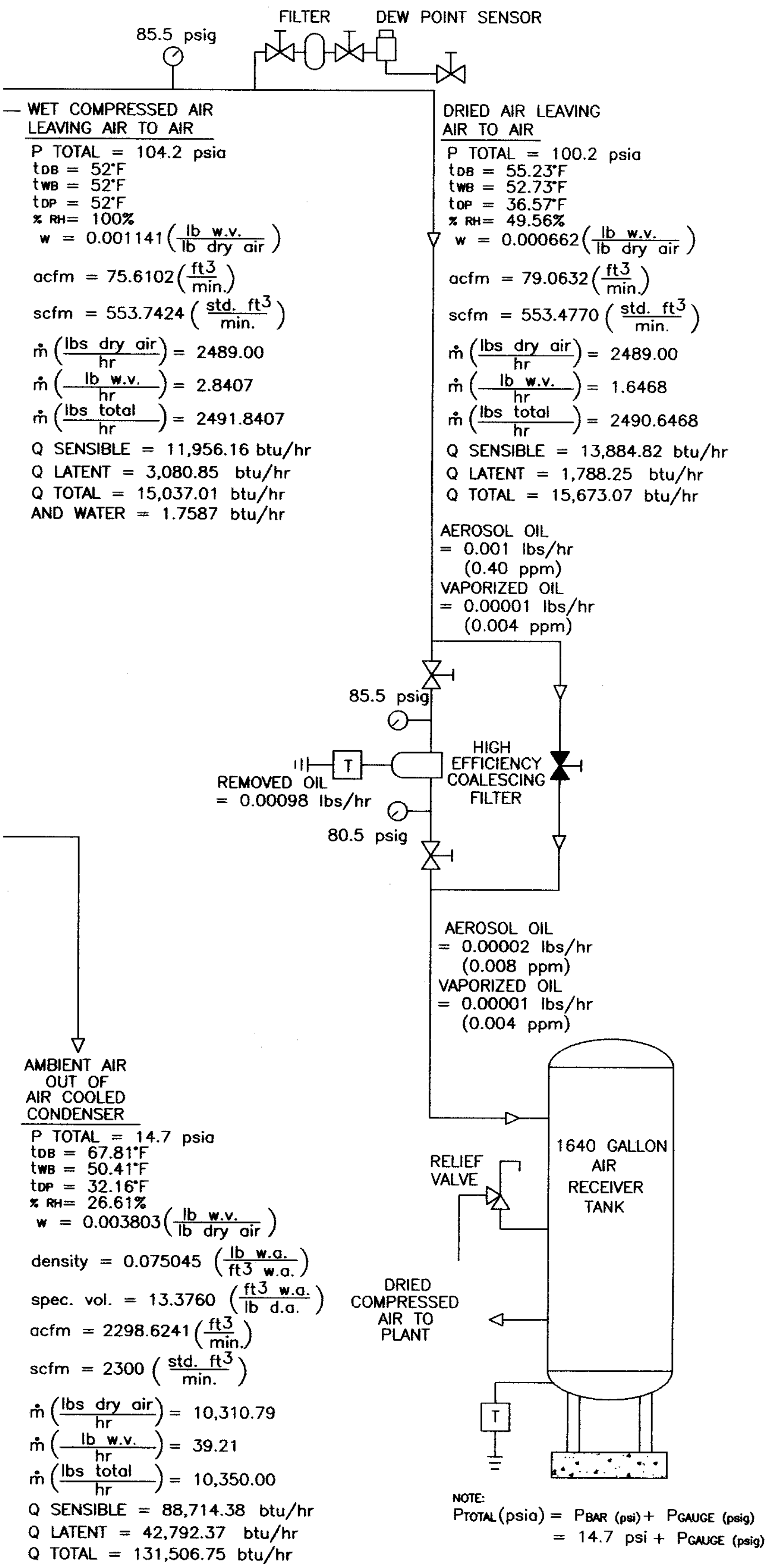


FIGURE 2B. A compressed-air system and refrigerated dryer (continued).



0.01497 lb of oil per hour. In some cases, the water and oil are collected in a central collection-tank system and sent to an oil/water separator. The oil in this condensate is 1,991 ppm (mass), which is too high a concentration for some wastewater regulatory agencies. The required maximum concentration discharged to a wastewater-treatment plant usually is 5 to 100 ppm (mass). Condensate-separation systems are designed to remove oil from water to meet wastewater-code requirements.

Designing an energy-efficient compressed-air system with adequate compressor capacity is not easy. It requires the elimination of excessive pressure losses on inlet and discharge piping, filter systems, coolers, and dryer systems. It also requires the removal of contaminants, such as water, oil, pipe scale and rust, and air-intake particulate. To select a compressor, an engineer first must identify the SCFM usages and convert that required flow to a mass-flow number. This eliminates confusion over tool-usage SCFM and CAGI-SCFM. The engineer then must identify the worst-case air-intake psychrometric properties to ensure adequate capacity on the worst-case day. Having identified the required compressor output, the engineer can review bids, comparing units based on efficiency, internal losses, power requirements, total installed cost, and lifetime maintenance costs.

To ensure energy efficiency, pressure drops must be kept low. Excessive pressure losses from undersized or dirty piping, filter systems, aftercoolers, and dryers entail more brake horsepower at the compressor and higher annual electricity consumption. An excessive pressure drop of 1 psi for a 1,000-CAGI-SCFM compressor system will cost more than $630 in added electricity consumption a year. Therefore:

- Piping should be designed for 0.2 to 0.3 psi per 100 ft of pipe (or about 50 ft per second).
- Filter systems should be carefully selected, and all systems should be properly maintained to reduce the demand for electricity.
- Regulators should be properly selected for systems that do not require full pressure.
- Inappropriate uses of compressed air

	Water vapor removed	Percent removed
Aftercooler	4.8656 lb per hour	62.2
Reheater	1.7587 lb per hour	22.5
Refrigerated-dryer section	1.1939 lb per hour	15.3
Total	7.8182 lb per hour	100

TABLE 5. Water-vapor removal in the compressed-air system of figures 2a and 2b.

should be eliminated.

• Water-cooled compressors, water-cooled intercoolers and aftercoolers, and oil coolers should be carefully selected.

• High pressure for compressed-air tools, such as paint guns, grinders, and sandblasters, should be avoided because increased pressure above design pressure only causes these tools to use more air. Compressed-air control systems help control pressure to plus or minus 2 psi.

• If the compressor building is of the high-temperature, high-humidity variety, the intake air should come from an outside-air intake, which will reduce the compressor brake horsepower and dryer load, as well as the electricity consumption of the compressor and dryer system.

CONCLUSION

The design steps taken during system planning have direct effects on a system's overall operation and maintenance; therefore, it is important that an experienced engineer and an experienced compressor manufacturer be involved. This will eliminate excessive operating costs and operational problems.

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