

blissing or curse? Hot Gas Bypass

It's not enough to deliver an HVAC system that meets the unique needs of each application. *The owner also deserves a system that is the most reliable and the least costly to operate.* In specific cases, adding hot gas bypass to the refrigeration system may be necessary in order to achieve all three goals. But more often than not, the responsibility to deliver a reliable, cost-effective system will preclude its addition. Hot gas bypass has the *potential* to modulate capacity or improve reliability, but these benefits will not be realized without *careful evaluation of its appropriateness* and painstaking attention to design, installation, control, and maintenance.

The success of an application with HGBP depends on unflinching diligence at every step. HGBP *can* be a blessing when it is warranted and effectively implemented. In all other cases, HGBP *will* be a curse. To understand why, let's briefly review what hot gas bypass does and how it is applied.

The objective. When diminishing loads force a refrigeration system to operate at unstable conditions, compressor and evaporator capacities balance at ever lower suction pressures and temperatures. Unchecked, the eventual result is coil frosting and compressor flooding.

Hot gas bypass can stabilize the system balance point by diverting hot, high-pressure refrigerant vapor from the discharge line directly to the low-pressure side of the system. This tactic

keeps the compressor more fully loaded while the evaporator satisfies the part-load condition. Also, the diverted vapor raises the suction temperature, which prevents frost from forming.

For decades HGBP has been applied in direct-expansion refrigerant systems to control capacity at low loads. It was probably conceived to correct a job-specific problem but was subsequently, and indiscriminately, added to HVAC systems as a preventive measure.

The curse. True, HGBP has provided frost control and some semblance of capacity control in many applications. But numerous cases exist in which HGBP either fails to safely stabilize the system or, worse still, undermines

HGBP in chilled water applications

Certain process cooling applications may require hot gas bypass to match chiller capacity to an instantaneous load. In such cases, the system is designed to operate continuously and hot gas bypass neutralizes the refrigeration effect.

For comfort cooling, however, the addition of hot gas bypass is seldom necessary. That's because a properly designed chilled water loop can provide an effective buffer between system load and chiller capacity. The key is to establish a loop time that equals the *greater* of two values: either the minimum "compressor off" time for the last stage of cooling, or the minimum loop time permitted by the chiller controller. You can accomplish this by changing the design flow rate or by adding a tank.

The simplicity of this approach will benefit the owner, as will the operating-cost savings. (A chilled water system *without* hot gas bypass requires less power than one that operates the compressors with hot gas bypass.) ■

reliable operation by introducing problems stemming from insufficient oil return and refrigerant logging in the HGBP line. And in *all* cases, hot gas bypass inflates the life-cycle cost of the system:

- **HGBP increases the initial cost.** It requires an additional refrigerant line, which also increases the likelihood of refrigerant leaks and oil/refrigerant logging.
- **HGBP greatly reduces operating efficiency** because the bypassed vapor does no useful cooling.
- **HGBP costs more to operate.** As the load fluctuates, the compressor consumes more energy because it's forced to operate at a compression step that is likely one stage higher than necessary.

The blessing. Despite these shortcomings, hot gas bypass can be appropriate for cooling applications that demand tight, continuous, thermal control—particularly if large amounts of outdoor air, widely varying loads, or excessive compressor on/off delays are involved. Most often these characteristics describe process cooling rather than standard comfort cooling. If applied properly, HGBP can:

- Prevent excessive compressor cycling
- Match system capacity to load
- Allow the system to operate at safe balance points during unsafe loads

Application Considerations

Although it can be applied in refrigerant-to-water systems, comfort cooling applications of hot gas bypass

most often involve refrigerant-to-air evaporators. With countless combinations of equipment and layouts possible, how can you best and most responsibly implement hot gas bypass?

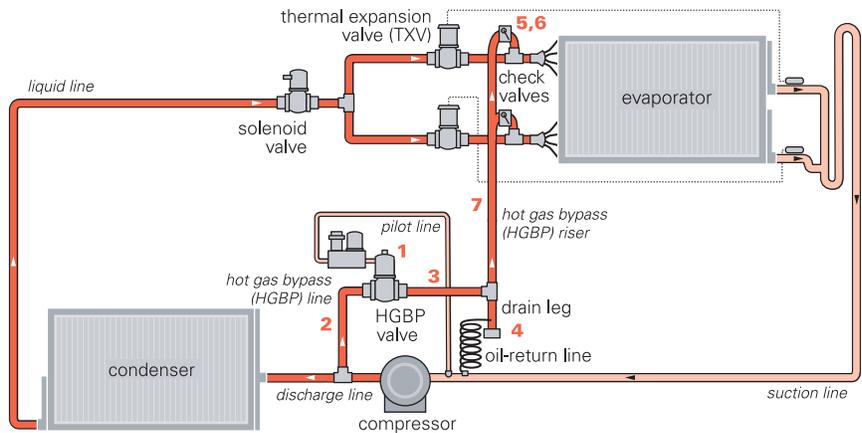
Apply HGBP as a last resort. Only use hot gas bypass if all other design options fail to meet the demands of the application. To make this determination, it is critical to ...

Understand the year-round loads and requirements of the system. “Most comfort cooling applications can operate between steps of loading without loss of temperature or humidity control.” Not only does this statement hold true between the third and fourth steps of cooling, when the outdoor latent load is high, but it also holds true for the step between off and first-stage cooling when the outdoor latent load is comparatively low.

Choose the right arrangement. Hot gas bypass is arranged in one of two ways (see the comparison in Table 1, p. 4); both require special care during design and installation.¹ *HGBP to the evaporator inlet* delivers hot refrigerant vapor between the expansion valve and the distributor (Figure 1). During HGBP operation, the expansion valve meters enough liquid refrigerant to both desuperheat the bypassed vapor and satisfy the evaporator load. The resulting refrigerant flow rate is sufficient to carry oil through the coil and suction line.

HGBP to the suction line bypasses both the condenser and the evaporator, diverting hot vapor from the compressor discharge directly to the suction line (Figure 2). A liquid-injection valve meters liquid refrigerant into the stream of bypassed vapor, cooling it

Figure 1. Hot gas bypassed to evaporator inlet (preferred arrangement for HGBP)



Keys to successful implementation:

- 1** Position the HGBP valve *above* the discharge line, near the compressor. If the system includes pump-down, provide a means to shut off refrigerant flow.
- 2** Pitch the line *upstream* of the HGBP valve to drain oil back into the discharge line.
- 3** Pitch the line *downstream* of the HGBP valve toward the evaporator, away from the valve.
- 4** If the HGBP line includes a riser, regardless of height, provide a drain leg of the same

diameter as the riser. Add an oil-return line 1 in. (25 mm) from the bottom of the trap; use tubing that is 1/8 in. (6 mm) and at least 5 ft (300 mm) long. Precharge the trap with oil.

- 5** Divert hot gas to each active distributor at the expected operating points for hot gas bypass.
- 6** If the HGBP line feeds multiple distributors, provide a check valve for each distributor.
- 7** Insulate the *entire* length of the HGBP line.

enough to prevent the compressor motor from overheating.

Do not oversize the system. An oversized system quickly meets the sensible load without satisfying the latent load. Adding HGBP may mitigate this error ... but at the expense of unnecessarily high power bills.

Once again, understanding the year-round loads and properly sizing the system to match them is paramount to providing thermal comfort (controlling temperature and humidity) and eliminating the need for HGBP.

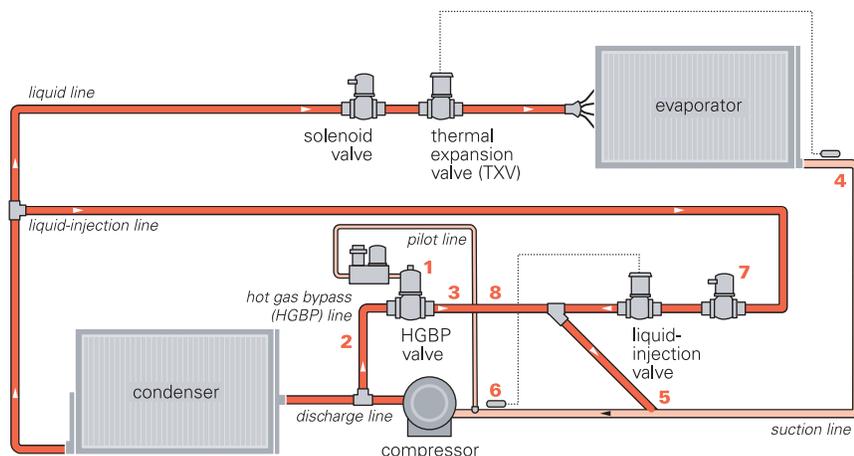
Select equipment with multiple-step refrigerant circuits. The logic for avoiding system oversizing also supports the choice of multiple-step refrigerant circuit(s) whenever possible. The capacity of an unloading refrigerant circuit will attempt to match a reduced load at better balance points.

Select an evaporator coil that can maintain a high suction temperature to permit the system to aggressively stage down before a frost condition develops. In all cases, *the design should maximize the suction temperature while maintaining the desired conditions.* For comfort cooling applications, the *minimum* saturated suction temperature at design should be 43°F to 45°F for variable-volume air distribution (VAV), or 40°F to 43°F for constant-volume air distribution (CV).

Coils with intertwined circuits tend to reduce the risk of coil frosting because they use more of the available fin surface (than face-split coils) at part-load conditions. This allows the system to operate at higher suctions and more reliable balance points.

¹ For more information, see *Hot Gas Bypass Control* (Trane publication AM-CON10) and the 1998 ASHRAE Handbook—Refrigeration.

Figure 2. Hot gas bypassed to suction line



Keys to successful implementation:

- 1** Position the HGBP valve *above* the discharge line, near the compressor. If the system includes pump-down, provide a means to shut off refrigerant flow.
- 2** Pitch the line *upstream* of the HGBP valve to drain oil back into the discharge line.
- 3** Pitch the line *downstream* of the HGBP valve toward the suction line, away from the valve.
- 4** Assure that the evaporator and suction line freely drain to the suction-line HGBP connection.

- 5** Site the suction-line HGBP connection upstream of the pilot-line tap for the HGBP valve and at least 5 ft (1.5 m) upstream from the compressor inlet. Angle the connection into the suction flow.
- 6** Attach the remote bulb for the liquid-injection valve to the suction line, downstream of the HGBP connection.
- 7** Provide a solenoid valve upstream of the liquid-injection valve. Synchronize the operation of the HGBP and liquid-line solenoid valves.
- 8** Insulate the *entire* length of the HGBP line.

Target the cooler, drier part of the thermal comfort envelope.

Compressor cycling usually isn't an issue for comfort-cooling VAV or CV applications. But this strategy provides more latitude for the space condition to "float" during compressor staging without sacrificing thermal comfort.

Comply with local codes. ASHRAE/IESNA Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, precludes the use of hot gas bypass for HVAC systems larger than 7.5 tons unless they are designed with multiple steps of unloading or continuous capacity modulation. The maximum amount of HGBP allowed by the standard varies with system size: 25 percent of total capacity for systems larger than 20 tons, 50 percent for systems ranging from 7.5 to 20 tons.²

Line Design Considerations

Make gravity work for you. A typical HGBP valve can infinitely vary the rate of hot gas flow. The resulting refrigerant velocity can become so low that it "traps" oil in the HGBP line. Gravity then becomes the sole means for returning oil to the compressor. It is therefore critical to design the refrigerant piping system so that all oil (and condensed refrigerant) drains freely out of the HGBP line.

Suction-line lift and the design of the evaporator coil particularly limit the use of HGBP to the suction-line. Because

² Standard 90.1 (which excludes process applications and low-rise residential buildings) grants an exception to systems smaller than 7.5 tons. Be sure to consult the standard for specific requirements.

this arrangement bypasses the evaporator and part of the suction line, the refrigerant flow rate upstream of the HGBP point of entry may become too low to move oil. To overcome this limitation, restrict the maximum amount of HGBP to a quantity that maintains sufficient refrigerant velocity for oil entrainment.

Note: If this design cannot be accomplished, it is critical to assure that oil drains freely from both the evaporator and the suction line to the inlet of the HGBP line. Failure to do so will starve the compressor of oil and shorten the life of the compressor.

Independent of which HGBP arrangement is applied, the HGBP line must connect to the *top* of both the discharge line and the evaporator inlet (or suction line). Otherwise, a mixture of oil and refrigerant will pour into the HGBP line, starving the compressor of oil and causing a liquid slug when the HGBP valve opens.

The refrigerant that collects in the HGBP line also reduces the refrigerant charge available to the system, which may lower the suction pressure. Ironically, the HGBP valve may respond by opening and allowing bypass flow of hot gas. The repetitive, unstable operation and slugging that result make it critical to prevent this condition.

Select the appropriate pipe diameter. With HGBP lines, typically smaller is better to promote oil movement; however, a design that allows gravity to purge any liquid from the HGBP line lessens this requirement. Size the line for a pressure drop of 6 psid to 20 psid.

Keep the HGBP line short. Refrigerant vapor will condense in, and fill any portion of, the HGBP line that is colder than the saturated temperature of the suction or discharge gas. Incidences (and consequences) of

refrigerant condensation will increase with the length of the line, despite insulation.

Oil return also can be a problem in properly designed HGBP lines because the HGBP valve can reduce the flow of bypassed gas to nearly zero—well below the minimum flow rate needed to carry oil back to the compressor. Gravity will be very slow to bring the oil that deposits along the length of the HGBP line back to the compressor. The longer the HGBP line, the more oil it will collect. It is imperative to keep the oil charge in the system, not in the HGBP line.

Select an appropriate HGBP valve.

There are two basic types of HGBP valves: proportional and electronic. A *proportional*, pressure-actuated HGBP valve opens a set amount based on the difference between the valve’s suction-pressure setpoint and the actual suction pressure. Generally, valves of this type require a pressure difference of 6 psig to open fully. Although this characteristic makes it difficult to maintain a specific suction pressure, a proportional HGBP valve can satisfactorily maintain a suction-pressure range that is equivalent to ± 4°F (saturated) suction temperature.

By contrast, an *electronic* HGBP valve can regulate bypass flow to maintain the desired suction pressure or discharge air temperature. If used in conjunction with a supply-air-temperature control, optimize system efficiency by coordinating the operation of the HGBP valve and compressors.

Note: If the refrigeration system includes pump-down, make sure that the HGBP system includes a means to prevent refrigerant flow during the pump-down cycle.

Add head-pressure control. Diverting hot gas from the discharge line and around the condenser reduces the head pressure of the air conditioning system. As head pressure decreases, operation may become unstable — and unreliable. To avoid this risk, include a method for head pressure control in every HGBP application.

A Final Caveat

Risk of oil and refrigerant logging is inherent to hot gas bypass. Do not assume that adding hot gas bypass to the job specification will improve reliability or enhance capacity modulation. Neither benefit of HGBP will be realized without meticulous attention to the desired effect, component selection, and implementation. Anything less will

compromise system performance ... and curse the owner with excessive maintenance and power bills for years to come. That said, if HGBP best addresses the demands of a particular application then specify it. If implemented responsibly, HGBP will be a blessing. ■

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Table 1. Comparison of common HGBP arrangements

Evaporator inlet (preferred; Figure 1, p. 2)	Suction line (Figure 2, p. 3)
<p>Advantages</p> <ul style="list-style-type: none"> ■ Merging the bypassed load with the system load in the evaporator helps the expansion valve maintain superheat control ■ Sustains a higher gas velocity in the evaporator and suction line at low loads, enabling more reliable oil return 	<p>Advantages</p> <ul style="list-style-type: none"> ■ Less expensive to install because it requires less piping, particularly if the condensing unit and evaporator are far apart
<p>Disadvantages</p> <ul style="list-style-type: none"> ■ Installation cost is directly proportional to the distance between the condensing unit and evaporator 	<p>Disadvantages</p> <ul style="list-style-type: none"> ■ Requires an additional liquid-line solenoid valve and expansion valve ■ At low evaporator loads, refrigerant-gas velocity may not be sufficient for adequate oil movement ■ Challenges the designer to slope the suction line in two directions: toward the compressor during the on cycle, toward the evaporator during the off cycle



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