Modulating capacity control in the refrigeration cycle

Building Technologies

SIEMENS

Contents

1. Introduction

1.1 General

In air conditioning systems, the cooling load rarely remains constant over long periods, because the internal and external heat and humidity loads are continually changing. Therefore, the demand for cooling capacity varies in terms of time and amount within a broad range.

The refrigeration machine itself is also subject to capacity variations due to changing condensation and vaporization pressures and temperatures. This means that the available refrigeration capacity also varies.

The control task is to adjust the machine's refrigeration capacity as closely as possible to the respective cooling demand. Control normally presents no special difficulties in systems with water chillers: On the consumer side, modulating mixing valves provide exact control of the air coolers. This means that humidity and temperature are continuously controllable. On the machine side, chilled water primary control is, for example, achieved via periodic switching of compressors.

In small-scale air cooling systems, direct expansion coolers are selected for practical reasons, although their control is often problematical. If no special control precautions are taken, the periodic switching of the compressor is transferred to the cooling capacity: Abrupt temperature changes in the supply air and in the room, i.e. considerably reduced comfort, can hardly be avoided.

The range of electronic refrigerant valves allows evaporator output to be controlled continuously and accurately. The control acts on the machine's refrigeration medium. This provides for continuous control of temperature and humidity, even with direct expansion refrigerating machines.

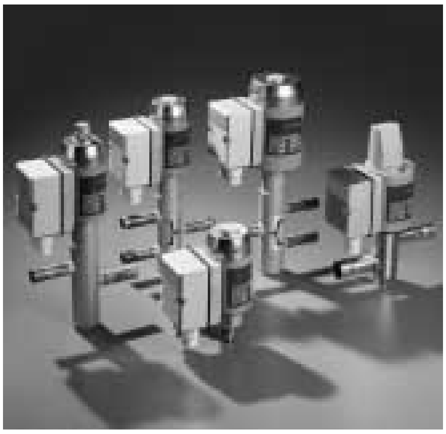

Fig. 1-1
Modulating magnetic valves for halogenated refrigerants

From left to right:
– Suction-throttle control valve (with manual positioning)
– Modulating control valve for condenser control
– Bypass diverting control valve
– Electronic injection valve for safety refrigerants
– Pilot valve (also for ammonia)

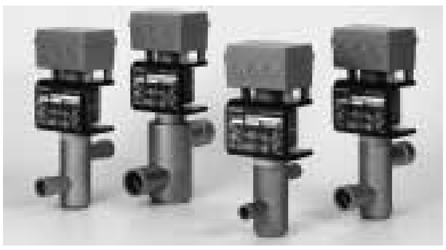

Fig. 1-2
The new range of modulating magnetic valves MVL661… for halogenated refrigerants

One type of valve for three different applications:
– Modulating control valve for condenser control
– Bypass control valve
– Electronic injection valve for safety refrigerants

2. Determining the refrigeration capacity

2.1 Demand

Cooling demand depends on the **internal** and **external** loads of the rooms to be cooled. Internal loads (Q_i) include heat sources such as:

- Occupants
- Machines and equipment
- Lighting

The external loads (Q_a) are comprised of the various heat flows that enter the building in the form of conducted and radiant heat through the exterior walls, roof and windows. These loads primarily depend on the difference between the outdoor temperature (t_{AU}) and the room temperature setpoint t_{RA}.

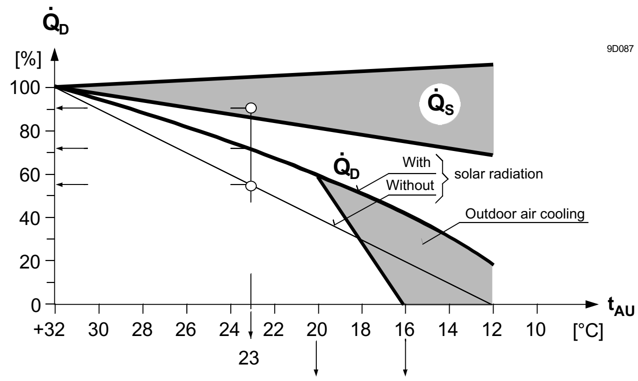

Fig. 2-1 Cooling demand of a building as a function of outdoor temperature

Fig. 2-1 shows an example of the presumable cooling demand Q_D (= Q_i + Q_a) of a building as a function of outdoor temperature t_{AU}. It is assumed that the maximum cooling load Q_D = 100 % occurs at t_{AU} = 32 °C and decreases with falling outdoor temperature. The cooling load also changes according to solar radiation. The cooling load is lower on cloudy days; at t_{AU} = 23 °C, for example, Q_D is approximately 55 % of that with solar radiation (Q_D = 73 %).

If the outdoor temperature falls below the desired room temperature, the outdoor air contributes to the purely mechanical cooling. The demand for mechanical cooling ceases at $t_{AU} < 16$ °C. Conversely, it can be clearly seen that the mechanical cooling demand rises rapidly in the narrow temperature interval between 16…20 °C.

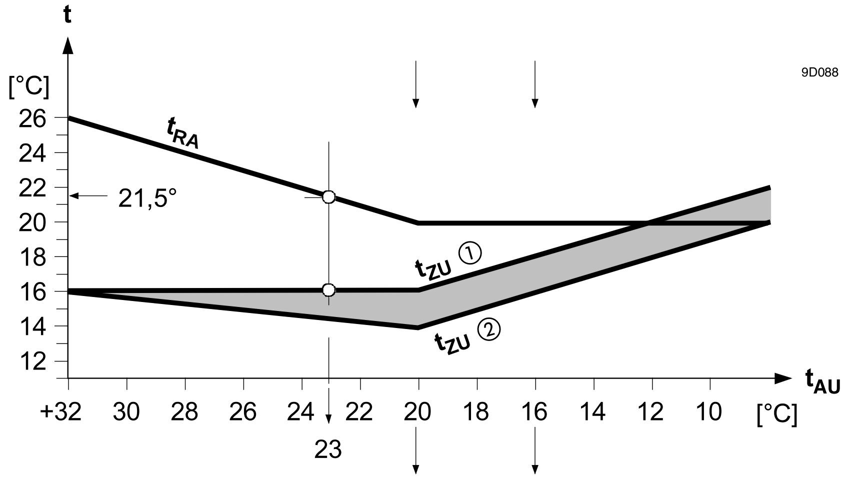

Fig. 2-2 Progression of the room temperature as a function of outdoor temperature
1) Without sun
2) With sun

Fig. 2-2 describes the presumable progression of the room temperature t_{RA} as a function of the outdoor temperature t_{AU}. The room temperature is raised on hot days for reasons of comfort. The cooling load accumulating in the room is dissipated by cooled supply air. In order to cool the room to a given temperature, the supply air would have to be cooled to 16 °C, for example, if the outdoor temperature were 32 °C. On sunny days, the supply air temperature falls (lower characteristic) in order to overcome the increased cooling load (! depending on the ventilation circuit type, temperatures below 16…18 °C may not be permissible, because they can give rise to drafts).

2.2 Supply

The refrigeration machine is designed for a given point, which depends on the calculated cooling load and the outdoor temperature on which it is based.

The capacity of the refrigeration machine varies greatly according to the effects of the weather. Changing temperatures in particular give rise to considerable output fluctuations at the evaporator and condenser.

Fig. 2-1 shows the range of possible supply curves Q_s of an uncontrolled refrigeration system. The supplied and actually required refrigeration capacity only coincide at the system's design point. As the outdoor temperature falls, there is an increasing excess of output with respect to the actual cooling demand. At an outdoor temperature of 23 °C without solar radiation, there is a cooling demand Q_D of 55 % compared to a capacity of about 90 % of the refrigeration machine. The refrigeration machine is greatly oversized at this point.

A temperature frequency diagram, which shows the number of hours per year during which a given outdoor temperature interval occurs (Fig. 2-3, example of Zurich), makes it possible to estimate how often the refrigeration machine will operate in certain partial load ranges. It can be seen that temperatures of 16…20 °C occur most frequently. Within this range, the cooling load rises rapidly from 0 to 60 %. This situation highlights the importance of capacity control.

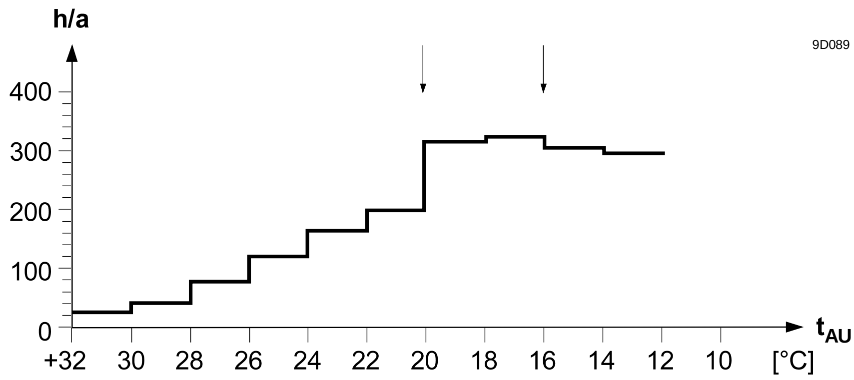

Fig. 2-3 Temperature frequency diagram (example of Zurich)

3. Matching refrigeration capacity to demand

3.1 Cylic switching of the compressor (two-position control)

Capacity control is necessary in order to adjust the refrigeration capacity to the demand. This can basically be achieved in the following two ways:

- Cyclic switching of the compressor (-> 3.1)
- Modulating control by influencing the flow of refrigerant (-> 3.2)

This capacity control possibility gives rise to the following difficulties:

- Comfort problems (-> 3.1.1)
- Mechanical problems (-> 3.1.2)

3.1.1 Comfort problems with ON/OFF control

The limitation of ON/OFF control to two operating states gives rise to major, sustained oscillations of the supply air temperature and, therefore, of the room air temperature. In terms of comfort, temperature deviations of more than ±1.5 K are perceived as unpleasant [Lit. 2]. In order to avoid drafts, the supply air temperature must not fall below a given minimum. The SWKI guidelines [Lit. 3] recommend maximum differences between room temperatures and inlet temperatures that should not be exceeded, for example:

- 26 °C room temperature = 16 °C inlet temperature
- 22 °C room temperature = 14 °C inlet temperature

The following system example (Fig. 3-1) explains the interaction of switch-on frequency, room air temperature and supply air temperature with ON/OFF control.

- Room temperature setpoint t_{RA} = 21.5 °C at a room temperature t_{AU} without solar radiation = 23 °C (as per Fig. 2-2). Average required supply air temperature t_{ZU} = 16 °C (as per Fig. 2-2)
- Cooling demand Q_D 55 % (as per Fig. 2-1)
- Supplied cooling capacity Q_S approx. 90 % (as per Fig. 2-1)
- Minimum supply air temperature = 14.5 °C (as per SWKI)
- Switching differential x_D = 1 K

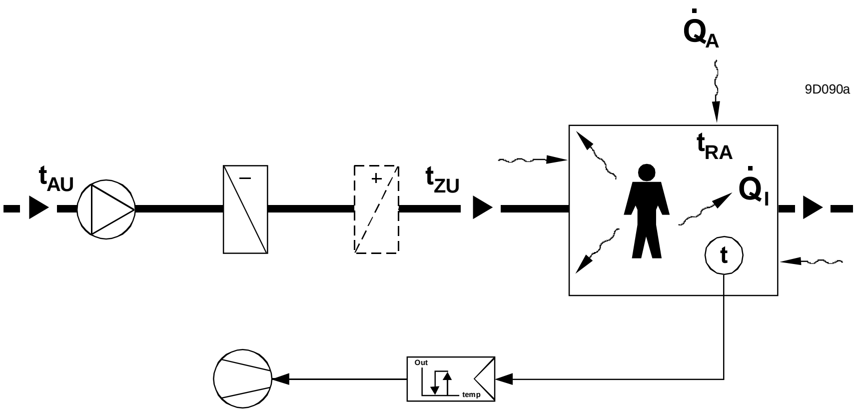

Fig. 3-1 Partial air conditioning system with two-position control

Measurements were performed on a system of this kind. The results are shown in Fig. 3-2.

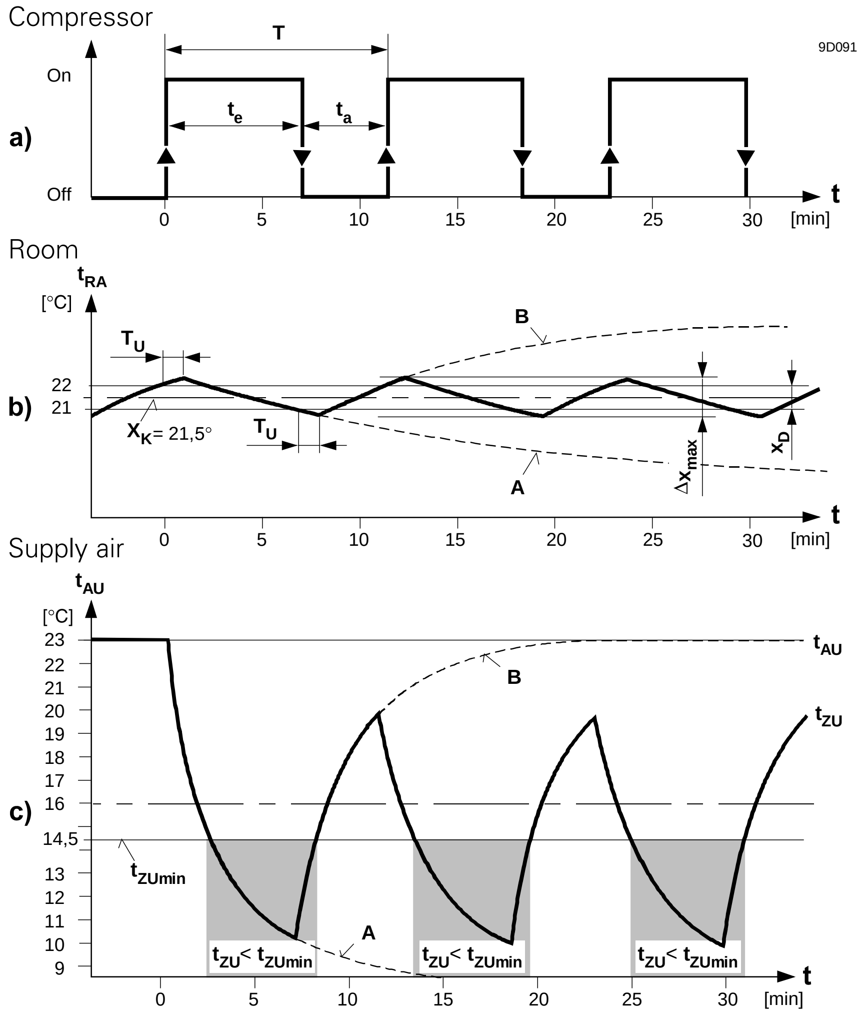

Fig. 3-2 Dynamic response of the two-position control of a partial air conditioning system (measurement results)

Fig. 3-2a Compressor switching diagram T = period
t_e = ON time
t_a = OFF time

Fig. 3-2b Room air temperature progression
A = cooling curve
B = reheating curve

Fig. 3-2c Supply air temperature progression
A = cooling curve
B = reheating curve

The comfort criterion that no greater deviation than ± 1.5 K should occur between the room temperature and the setpoint is definitely met. Fig. 3-2b shows a maximum temperature oscillation Δx_{max} of 1.6 K. However, the fact that the delay time T_u is passed twice gives rise to a peak-to-valley value that is considerably greater than that of the switching differential alone. On the other hand, Fig. 3-2c shows that the supply air temperature is well below the recommended minimum value of 14.5 °C for about 5 minutes of each 11-minute switching cycle. The supply air temperature then falls to 10.2 °C. As will be shown in the following, this disadvantage cannot simply be corrected by increasing the switching frequency.

3.1.2 Mechanical problems

- An excessively high switching frequency causes excessive **motor heating**
- During the startup phase, the **oil pressure** is low; bearing lubrication is not optimal; high switching frequencies reduce the life of the respective parts
- **Oil return** in intermittent mode: More oil enters the refrigerant cycle during startup than during continuous operation; frequent switching does not provide sufficient oil return

These problems can be partially avoided via cylinder cut-off or by dividing the output amongst several compressors. For economic reasons, however, such solutions are restricted to systems with large-scale compressors.

The switching differential does not solve the problems.

Analytical relationships:
The compressor running time (= ON time) t_e depends on the control factor E:

$$E = \frac{\text{Cooling demand } \dot{Q}_D}{\text{Refrigeration supply } Q_S} = \frac{\text{Compressor running time } t_e}{\text{Period length T}}$$

For the compressor running time, this means that:

$$t_e = \frac{\dot{Q}_D}{\dot{Q}_S} \cdot T$$

There are **approximation formulae** for determining the period T and the temperature oscillation Δx_{max} [Lit. 4, 6]:

$$\text{Period } T \approx 4 \cdot (T_u + \frac{T_g \cdot x_D}{x_h})$$

Temperature oscillation $\Delta x_{max} \approx S \cdot X_h + x_D$ (in the room)

x_D = switching differential (adjustable on the controller)
X_h = maximum possible cooling of the room
S = degree of difficulty = $\frac{T_u}{T_s}$
T_u = delay time of the controlled system
T_g = balancing time of the controlled system

The switching differential is the only adjustable variable. It has a direct impact on the compressor running time t_e, the period length T and, in particular, on the maximum temperature oscillation Δx_{max} in the room and in the partial controlled system of the supply air.
The degree of difficulty S is determined by the system, so it cannot be influenced without structural changes. The degree of difficulty S is a measure of the controllability of controlled systems.

According to experience, a value of < 0.3 for the degree of difficulty S should be targeted for room controlled systems with ON/OFF control [Lit. 5]. In multiple storage element systems, S is the quotient of the delay time T_u and the balancing time T_g; in single storage element systems, it is the quotient of the dead time T_t and the time constant T.

Examples with figures:

The system data as per Fig. 3-1 and the measurement results as per Fig. 3-2 give rise to the following values:

Period $T \approx 4 \cdot (1 + \frac{11 \cdot 1}{6}) = 11.3$ min

Amplitude $\Delta x_{max} \approx 4 \cdot \frac{1}{11} \cdot 6 + 1 = 1.54$ K

Compressor running time $t_e = \frac{55}{90} \cdot 11.3 = 6.9$ min

If the switching differential x_D is now increased to 2 K in order to prevent excessive switching, the values are as follows:

$T \approx 4 \cdot (1 + \frac{11 \cdot 2}{6}) = 18.6$ min

$\Delta x_{max} \approx \frac{1}{11} \cdot 6 + 2 = 2.54$ K

This new period length corresponds to an ON time t_E of

$\frac{55}{90} \cdot 18.6 = 11.4$ min

With this ON time, the supply air temperature falls to just below 9 °C (see Fig. 3-2c, supply air cooling curve). The price of a longer compressor running time is considerably poorer compliance with the comfort criteria in the supply air and in the room.

3.2 Modulating capacity control

In contrast to two-position control, modulating control varies the refrigerant flow and, therefore, the thermal states of the circulating refrigerant. This gives rise to the following possibilities:

- Adjustment of the vaporization temperature according to demand. The cooling of the supply air is continuous, which means the temperature progression in the supply air and in the room also becomes continuous. This means that the periodic temperature fluctuations caused by switching control no longer occur
- The continuous temperature progression guarantees better air mixing in the conditioned space, because this type of control enables a supply air temperature to be achieved that is suited to the air routing and air outlets
- Supply air temperature low limit control and summer compensation, which can be easily achieved with water chillers, can also be applied to direct-expansion refrigeration machines
- The actual power demand is determined in a central control system, taking into account the various optimization criteria. Not only temperature and humidity but also air quality, room occupancy and especially time schedules can influence refrigeration capacity according to demand

3.2.1 Control loop selection

In order to be able to fully utilize the advantages of modulating capacity control, the actual control task and requirements must be clearly defined.

The controlled parameters room temperature and possibly room humidity result from the system's design. Additionally, the control task definition may have to be extended to include limits to be complied with by the supply air, a shift of room temperature and humidity according to outdoor temperature etc.

In order for the control system to meet the requirements regarding thermal comfort and accurate, continuous compliance with setpoints, the measuring location of the control parameters and the appropriate control loops and equipment must also be determined.

3.2.2 Measuring location

The question of the measuring location is significant, because different temperature sensor locations give rise to different characteristics for the room temperature/humidity controlled system. If the measuring location is in the room, it is normally desirable for the sensor to be located in one of the secondary airflows induced by the air blown into the room. This ensures that the average room temperature and not the supply air temperature is measured with a sufficiently small delay, which provides for rapid control action. Rooms have zones with zero airflow; placement of the sensor there will cause even the best control equipment to fail. A defined and, therefore, reliable measuring location is in most cases the exhaust air. This usually provides the most reliable air state measurements, which ensures that the control equipment works well. The prerequisite also in this case is that the air routing provides proper air mixing and purging of the conditioned space [Lit. 7].

3.3 Capacity control in various systems

3.3.1 Room temperature control

The sensor measures the room temperature t_{RA}. If the room temperature rises above the setpoint X_K, the controller first enables the refrigeration machine, then continuously raises the evaporator capacity with increasing deviation.

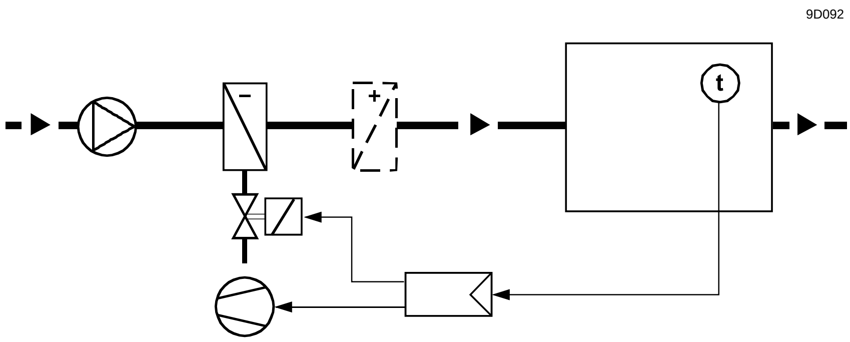

Fig. 3-3 Principle of room temperature control

The data from the example in Fig. 3-1 (two-position control) gives rise to the dynamic response as shown in Fig. 3-4 below.

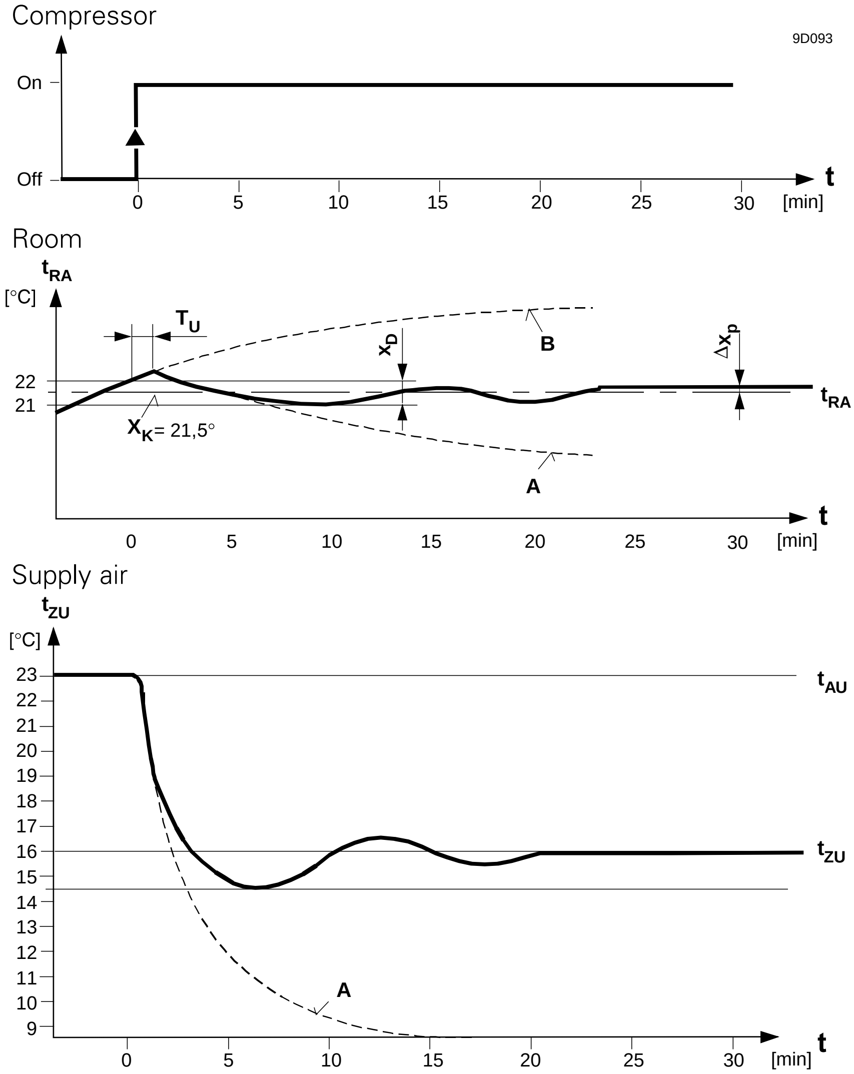

Fig. 3-4 Dynamic response of modulating control

The compressor starts as soon as the switch-on point of 22 °C is reached. The delay time T_u resulting from the room allows the room temperature to overshoot slightly along the heating curve until the continuously cooler supply air t_{ZU} makes the room temperature t_{RA} fall again. The control action on the refrigeration system rapidly separates the room temperature progression from the cooling curve A.

The transient response specific to the P-controller is completed after about 20 minutes. The room temperature t_R now corresponds to the setpoint X_K plus the control offset Δx_P.
The control offset is small in room temperature control loops, because stable control is achieved with narrow proportional bands. In order to prevent the compressor from switching off prematurely during the transient response, the switching differential x_D of the compressor is based on the greatest negative amplitude half.

In order to preserve the balance between heat supply and dissipation and to maintain the setpoint, the supply air must in this example be cooled from 23 °C to 16 °C. When the compressor starts, the temperature downstream of the evaporator falls, thus overshooting the theoretically necessary supply air temperature during the transient response.

3.3.2 Supply air temperature low limit control

This refers to the maintenance of a minimum supply air temperature for comfort reasons, even if lower temperatures are necessary for cooling according to demand. Therefore, the maintenance of the setpoint temperature in the room is superseded by the limit control of the supply air temperature. In this operating case, the total output of the refrigeration machine is deliberately not fully utilized.

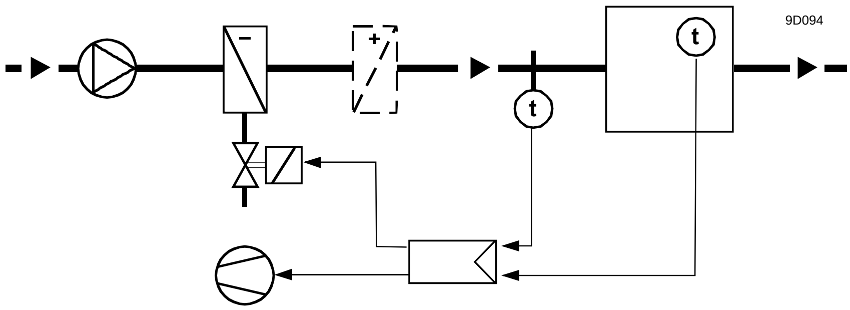

Fig. 3-5 Principle of supply air limit control

The supply air low limit temperature is based on the airflow routing and is selected such that drafts do not occur in the conditioned space. In the example as per Fig. 3-4, the minimum supply air temperature $t_{ZU\ min}$ was assumed to be 14.5 °C. Fig. 3-5 shows a control loop that influences the output of the continuously controllable refrigeration machine according to the room temperature. The limiting sensor monitors the minimum permissible temperature in the supply air duct in the sense of an auxiliary control loop. If this temperature falls below a given value, the limiting controller intervenes and reduces the cooling capacity to the acceptable limit value. In this operating case, the limiting control loop has priority over the room control loop.

3.3.3 Room/supply air cascade control

If interference variables occur in the supply air duct, e.g. position change of the outdoor air damper or compressor output level change, cascade control prevents disturbances in the room, especially in controlled systems with long delay times, because an auxiliary control loop compensates for the disturbance variables in the supply air before they can have an effect on the room.

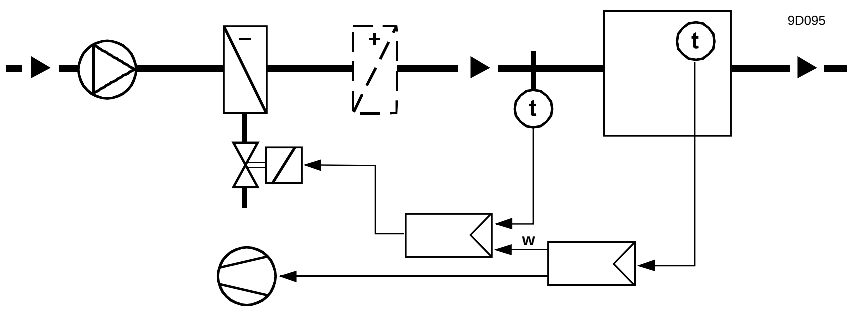

Fig. 3-6 Principle of room/supply air cascade control

In room/supply air temperature cascade control, the controller output variable of the room temperature controller (primary controller) acts as the reference variable w of the supply air controller (auxiliary controller). If the room temperature deviates from the setpoint X_K, the control valve is not adjusted directly but only the setpoint w of the auxiliary controller (see Fig. 3-7).

Cascade control usually also offers the additional possibility of influencing the supply air temperature via low and high limit control of the room inlet temperatures. This enables drafts due to excessively cold supply air or stratification due to excessively warm supply air to be prevented by relatively simple means. Typical places where cascade control is applied are shops, lecture theatres, restaurants, conference rooms and general rooms whose size and design gives rise to a long delay time although they are conditioned by a supply air system with disturbance variables and short delay times.

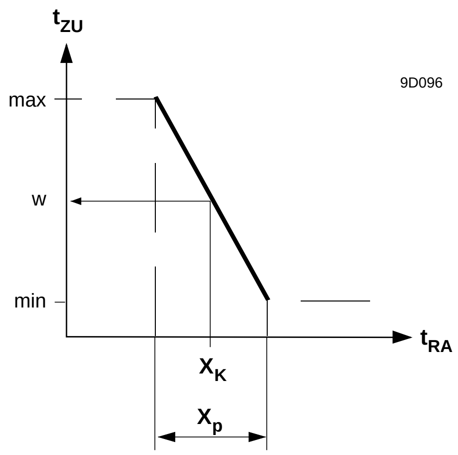

Fig. 3-7 Progression of compensating variable

4. Control action in the refrigeration cycle

4.1 Suction-throttle control

The compressor delivers a quantity of gaseous refrigerant that is determined by the displacement. This volume remains approximately constant. However, the mass flow can vary due to pressure changes on the intake side. The lower the intake pressure falls below the vaporization pressure, the more the suction gas concentration is reduced, and the mass flow decreases [Lit. 8].

Two methods of suction-throttle control are known:
- Simple suction-throttle control, and
- Suction-throttle control with hot-gas bypass

4.1.1 Simple suction-throttle control

Fig. 4-1 shows the principle of simple suction-throttle control. A control valve between the evaporator and compressor throttles the gas flow depending on the temperature, humidity and other control criteria. This throttling increases the pressure gradient between the evaporator and compressor:

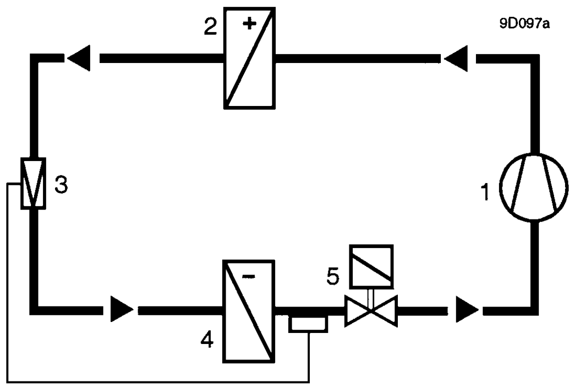

Fig. 4-1 The suction gas throttle valve in the refrigerant cycle (simple suction gas control)
1 Compressor
2 Condenser
3 Expansion valve
4 Evaporator
5 Suction-throttle valve

The pressure and, therefore, the vaporization temperature, rises in the evaporator, and it falls between the control valve and compressor. The temperature difference between the air to be cooled and the cooler decreases. As a consequence, the evaporator's output falls.

The superheating of the refrigerant decreases with rising vaporization temperature. Therefore, the expansion valve reduces the flow volume until the superheating of the suction gas reaches its original value. At the same time, the compressor draws in less refrigerant vapor due to the lower gas density downstream of the control valve – the circulating refrigerant volume decreases, and with it the refrigeration capacity.

Simple suction-throttle control (Fig. 4-1) provides for continuous throttling of the refrigeration capacity down to:
- approx. 40 % in semihermetic and fully hermetic compressors
- approx. 15 % in open compressors.

These lower limits are determined by the minimum refrigerant throughput that must be guaranteed for the cooling of the compressor. The exact limits are determined by the compressor used in each case.

In the case of simple suction-throttle control, this minimum throughput is assured by a bypass (not shown in Fig. 4-1) across the suction-throttle valve. This type of capacity control is applied:

- in case of high permanent loads, especially internal heat gain from machines, e.g. in computer rooms, railway signal boxes, etc.
- in systems with permanently high recirculated air quantity
- in cases where the refrigeration machine is not activated until a given cooling load is exceeded.

General: in systems with a gentle cooling demand progression (Q_D in Fig. 2-1). Therefore, a certain cooling load is already present at the switch-on point.

Simple suction-throttle control can be realized without the additional expense of auxiliary equipment. The minimum gas flow for compressor cooling, however, requires careful adjustment of the gas bypass at the control valve.

4.1.2 Suction-throttle control with hot gas bypass

In suction-throttle control with hot gas bypass (Fig. 4-2), compressor cooling is provided by a substitute gas that does not circulate via the evaporator circuit. This provides for modulating control of the evaporator output over the entire range from 100 to 0 %.

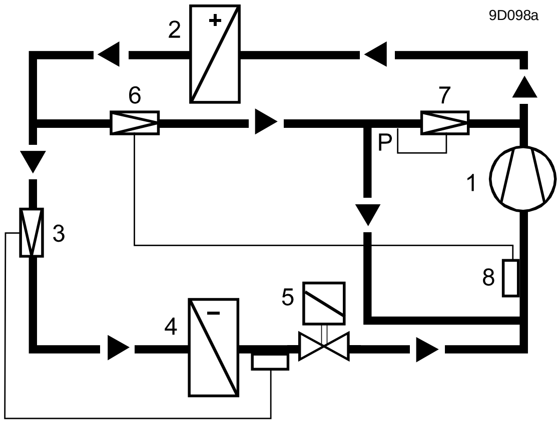

Fig. 4-2 Suction gas control with hot gas output controller and post-injection valve (principle)

1 Compressor	5 Suction-throttle valve
2 Condenser	6 Post-injection valve
3 Expansion valve	7 Automatic hot gas bypass valve
4 Evaporator	8 Sensor

The throttling via the control valve (5) reduces the pressure in the intake pipe below the value set on the automatic capacity control valve (7). This bypass valve opens, supplying a given quantity of hot gas to the compressor. The hot gas causes the temperature in the intake pipe to rise. In order to avoid excessive heating of the suction gas and, therefore, of the compressor, the suction gas is cooled via a post-injection valve (6). The sensor (8) acquires the temperature in the intake pipe and opens the post-injection valve (6) as the suction gas temperature rises. Due to the evaporation of the liquid refrigerant, the temperature in the intake pipe falls to the desired operating value.
In large-scale systems with a compressor power of 15 kW or more, it is often economical to provide for cylinder cut-off or to divide the load between two compressors.
Suction-throttle gas valves can control evaporator outputs of up to 80 kW per valve. For outputs > 80 kW, electronic output control is provided by a pilot valve in combination with a main valve.

4.2 Hot-gas control

A refrigeration cycle can also be controlled by dividing and diverting the hot-gas flow. This is achieved either via a direct bypass, i.e. a hot gas bypass between the discharge and intake side of the compressor, or via an indirect hot-gas bypass to the evaporator.

4.2.1 Direct hot-gas bypass control

A modulating magnetic valve (Fig. 4-2, No. 5) in the bypass pipe varies the volume of refrigerant flowing through the evaporator. The smaller volume reduces the refrigeration capacity via the rising pressure in the evaporator because of the increased vaporization temperature.

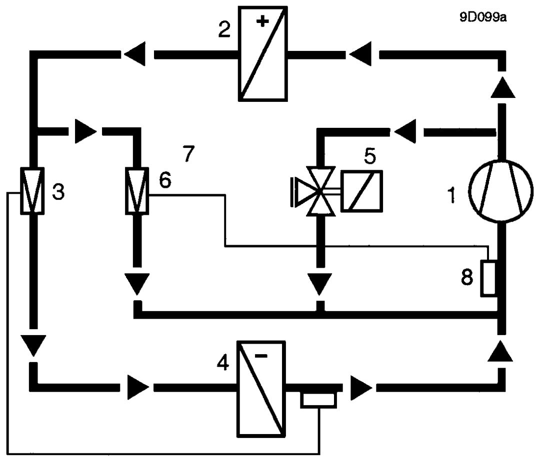

Fig. 4-3 Direct hot gas bypass control:
1 Compressor
2 Condenser
3 Expansion valve
4 Evaporator
5 Bypass control valve
6 Post-injection valve
7 Sensor

If full cooling capacity is required, the hot-gas bypass control valve (5) and the automatic, thermostatic post-injection valve (6) are closed. The compressor transports the total gas volume around the original cycle through the condenser, expansion valve and evaporator. If the demand falls, the control valve (5) opens continuously, drawing off a portion of the compressed gas and mixing it with the suction gas.

In order to prevent excessive heating of the suction gas in the partial load range, as in the case of suction-throttle control, cool refrigerant is mixed in via the post-injection valve (6) according to the suction gas temperature.

This in turn raises the vaporization temperature, causing the refrigeration capacity to fall. The superheating of the suction gas upstream of the compressor is monitored and controlled by the expansion valve. Therefore, the expansion valve must be capable of controlling the refrigerant flow between 100 and 20 %. Hot-gas valves can control evaporator outputs of up to 100 kW.

The capacity range of the evaporator can be varied between 100 and 0 % with direct hot-gas bypass control. The field of applications includes comfort and process air conditioning systems where major load changes must be accommodated.

4.2.2 Indirect hot-gas bypass control

This refers to a circuit with a controllable bypass from the high-pressure side to the low-pressure side (see Fig. 4-3) with injection between the expansion valve and evaporator.

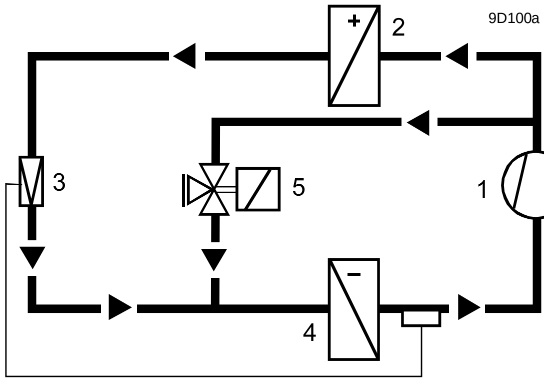

Fig. 4-4 Indirect hot-gas bypass control:
1 Compressor
2 Condenser
3 Expansion valve
4 Evaporator
5 Bypass control valve

The control valve (5) in the bypass remains closed in case of high refrigeration demand. The refrigeration system supplies its full output in this case. If the demand falls, the controller continuously opens the hot-gas bypass valve. Hot-gas now flows through the bypass to the evaporator inlet (4). There, it is mixed with the refrigerant flowing from the expansion valve and cooled. Therefore, the mixture already partially evaporates in the evaporator supply pipe. This in turn raises the vaporization temperature, causing the refrigeration output to fall. The superheating of the suction gas upstream of the compressor is monitored and controlled by the expansion valve. Therefore, the expansion valve must be capable of controlling the supply of refrigerant to the evaporator between 100 % and the minimum load. Hot-gas valves can control evaporator outputs of up to 100 kW.

With correct system design, a reduction of the evaporator capacity to 0 % is possible. Due to its simplicity and reliability, this type of control is popular in small-scale refrigeration systems.

In all hot-gas control systems, the compressor drive power remains approximately constant in the partial load range due to the increased circulation rate of the refrigerant.

4.3 Selection of control action

The appropriate type of output control – suction throttle or hot-gas bypass depends on various factors.

Suction-throttle control:

- In systems where the evaporator is located a distance away from the compressor
- In open and semihermetic compressors that permit low intake pressures and vaporization temperatures (to > –15 °C)
- In case of relatively high-powered compressors (from approx. 15 kW refrigerating capacity)

Indirect hot-gas bypass control:

- In systems where the evaporator is located close to the compressor, e.g. in compact systems
- With (hermetic) compressors with relatively high minimum permissible vaporization temperatures (to > –15 °C)
- In relatively small systems

Direct hot-gas bypass control:

- In systems where the evaporator is located a distance away from or at a higher level than the compressor (if the evaporator is at the same level as or at a lower level than the compressor, a bypass pipe must be provided to ensure oil return to the compressor) [Lit. 1]
- With (hermetic) compressors with relatively high minimum permissible vaporization temperatures (to > –15 °C)
- In small systems (compact devices)

The control performance is very similar with all three methods.

5. Summary

5.1 Brief description

Balanced temperatures in the supply air and in the room provide comfort. Modulating control of refrigeration capacity gives rise to comfort due to balanced supply air temperatures that correspond to actual demand. The cooling capacity is varied infinitely by modulating control valves directly in the circulating refrigerant vapor flow.
Hermetic, modulating magnetic valves provide for very precise, continuous metering of the refrigeration capacity to be controlled. The electronic control, high permissible operating pressures and differential pressures of modulating magnetic valves allow refrigeration systems to be sized and built according to this concept. This equips direct-expansion refrigeration machines with the same advantageous operating behavior as water chillers.

The efficiency of the overall system is improved by embedding the refrigeration machine control in the air conditioning system control: Operating conflicts, such as simultaneous heating and cooling, are prevented, and standby losses are considerably reduced.

6. Literature

6.1 Sources

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Siemens Switzerland Ltd
Building Technologies Group
International Headquarters
Gubelstrasse 22
CH-6301 Zug
Tel +41 41 724 24 24
Fax +41 41 724 35 22

Siemens Building Technologies Ltd
Hawthorne Road
Staines
Middlesex TW18 3AY
United Kingdom
Tel +44 1784 46 16 16
Fax +44 1784 46 46 46

Siemens Ltd
Building Technologies
Units 1006-10
10/F, China Resources Building
26 Harbour Road
HK-Wanchai
Tel +852 2870 7888
Fax +852 2870 7880

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