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PARTIAL LOAD OPERATION

The objective of this document is help one to understand how changing load affects system operation and how systems are designed to cope with changing loads. You will also learn how improper capacity control can cause problems in the system.

There is lot of inadvertent confusion and the terms are used unknowingly in a manner, which do not convey right meaning. It is therefore necessary to understand correct meaning of different terminologies and understand differences amongst them.

PART LOAD

The condition in which the load on the system is less than design/maximum load, for which the system is designed.

CAPACITY CONTROL

The ability of a system to directly adjust one or more of its components, in order to adapt to changes in load pattern.

OVERSIZING

That condition in which the system capacity far exceeds design load

PART LOAD IS THE NORM

Systems rarely operate at the peak load for which they are designed.
The designer considers the worst conditions for load calculations, like max. summer conditions of dry bulb or monsoon conditions with high humidity., max occupancy, max lighting load, Load during peak hours of 2PM to 4PM etc.

Further the consultants build safety factors while preparing specifications. Next to it the system designer, while selecting the components takes adequate margins to ensure that his system does not fall short of expectations and continues to deliver expected results even when equipment gets old. All this leads to substantially oversized plant. In addition to this all factors mentioned above do not occur simultaneously in real practice. The general observation therefore is, it operates at design loads less than 50 hours per year, or about 1% of the time. It is therefore necessary to provide some capacity control mechanism to ensure systems operate satisfactorily at all load conditions.

CAPACITY CONTROL STRATEGIES

There are three approaches to capacity control design. These are

1. No Capacity Control
2. One or more capacity control components
3. Control all components

NO CAPACITY CONTROL

1. SIMPLEST
2. LOWEST COST
3. MOST UNSTABLE

CAPACITY CONTROL ALL COMPONENTS

1. MOST COMPLEX
2. HIGHEST COST
3. MOST STABLE

When mechanical refrigeration system is operating properly it will experience load reduction when less heat is available for the system to transfer than at peak conditions. With no capacity control on any component, the system tries to **float** with the load or **rides** with the load. This has however its own limitations.

The system that operates most efficiently, safely, and with the most stability will be the one that does best job of matching system capacity with building load for the broadest possible load range.

This however means increased initial cost and increased complexity, also making it costlier to maintain.

These factors must be considered against the long term cost and comfort visa a vis. Benefits of improved efficiency.

To understand control arrangement, a simple analogy wlth a water tank and float would serve the purpose.

If the water enters the tank at a fixed rate, and pump removes it at the same rate the level stays the same. The water coming in (Load) is equal to water going out (the capacity) .

If the pump removes water faster than it comes in, the water level falls. This is similar to having capacity greater than load. If the pump removes water slower than it enters, the water level rises. This is like having load greater than capacity.

Now let us imagine that the water is heat (a cooling load) and the tank is a space or product to be cooled by a mechanical refrigeration system and the pump with its inlet and outlet piping, representing the compressor with its suction and discharge side.

The level of water in the tank can also represent temperature of the air-conditioned space or refrigeration product to be cooled.

When we say we have designed building / system load, it is like having maximum water flow into the tank.

Design equipment capacity means having the pump & its piping run to its full capacity to remove the water added.

When both flow & pump system would match, the pump will remove water at the same rate at which the water is getting added to the tank & the water level will remain neither constant neither rising nor falling.

This means that in a perfectly sized system, it would run full time at design load & the temperature of space or product would be maintained constant at designed set point at all times.

When cooling load drops below designed level or when load reduces, it is similar to having reduced water supply to the tank. If pump keeps on pumping water to full capacity the level in the tank will drop. Likewise, if there is no capacity control & if the system continues to operate to full capacity even though the load on system is reduced, the net effect would be drop in space/product temperature compared to designed condition.

This would also mean that the saturated suction temperature will fall & the load & capacity will be out of balance, creating uncomfortable conditions in the space & frequent on/off equipment cycling.

The load fluctuation in commercial establishment is more compared to residential systems. As the commercial premises being larger, the equipment is also of bigger capacity & as the commercial loads charge more rapidly & also vary more than residential loads , need for capacity control for system is more in commercial building applications than residential.

1. Let us look at the size of plant in more details. The residential units are maximum up to 2.1/2 Ton capacity. It is one thing to cycle compressor of this size on / off & It is quite another thing to cycle a 25 Ton commercial compressor. The electricity board would penalize frequent starts & stops of such large loads.

   The part load operation with capacity control steps therefore becomes preferred option. The 2.1/2 ton increment in residential buildings is not large but 25 ton on / off increment is certainly very large to be accepted.

2. The loads vary more rapidly for commercial buildings. Internal loads can change quickly in response to opening / closing hours, matching power use & occupancy changes within the building.

   The system must respond to this in timely manner.

3. Thirdly the internal loads make up much larger portion of the total cooling load on commercial jobs than on residential premises.

   At design conditions typical residence has only 20% internal load while a single floor commercial office building has about 50% internal load. A restaurant may have 70% internal load while a core office area with no walls exposed may have 90% of the internal load or more.

   With 50% or more internal load, the commercial job experiences much larger building load & thereby system load swings, much more than residential building.

   In many cases the system capacity cannot simply float with the load & cycle often while maintaining safe & stable & efficient operating conditions.

   Since the components of refrigeration system are interconnected, the rate of refrigerant flow through the compressor must be same as through the condenser or evaporator & meeting device.

   The rate of evaporation in the evaporator has to be same as the rate of condensation in the condenser, when in equilibrium.

   In fact, the capacity of all components is the same at any given time and the total system operates at the net effect of all these components. Capacity is not determined by any one component but the change in any component affects the system balance capacity.

   The system capacity will stabilize as per the capacity of weakest component of the system. It is the combined effect that determines the capacity of system.

   It is again similar to our pump system analogy. The flow through the pump equals the flow, through the inlet & outlet piping. A change in any of the three components – inlet pipe, pumps or discharge pipe will affect the performance of the entire system.

   The capacity of evaporator / metering device combination increases as the saturated suction temperature falls, whereas the condenser & compressor combination capacity decreases when suction pressure falls. The intersection of these two capacity lines is the balance point or the point at which system stabilizes.

The evaporator capacity increases as the saturated suction temperature falls because lower saturated suction temperature means lower saturated evaporator temperature.

The lower saturated evaporator temperature produces lower coil surface temperature which increases the cooling effect on the air flowing over the coil,. due to larger temperature gradient

If entering temperature is the same for two evaporator coils, the surface temperature of the coil will be lower when the saturated evaporator temperature is lower. A cooler surface will absorb heat faster from air than warmer coil, all other things being equal.

Let us say the evaporator is operating at 36.5^{0}F saturated evaporator temperature has an effective surface temperature of 50^0 F. The evaporator operating at 46.5^0 F saturated evaporator temperature has an effective surface temperature of 56^0 F.

The greater the temperature difference between the coil surface & entering air, more rapidly the coil will absorb heat.

In other words a coil having 50^0 F surface temperature will pick up heat faster from incoming air than coil with surface temperature of 56^0 F. Therefore evaporator cooling capacity is highest at the lowest saturated evaporator (suction) temperature.

The condensing unit capacity pattern is just the opposite of the evaporator. The capacity at 38^{0}F saturated suction temperature is lower than that at 47^0 F. Therefore we should expect higher compressor & condensing unit capacity at higher suction temperatures. This also means power consumption of compressor will be higher at higher suction temperature although specific power consumption reduces at higher suction temperature thereby improving system efficiency.

The highest capacity & best efficiency occurs at higher suction temperature because the specific volume of the refrigerant gas decreases or density increases at higher saturation temperatures.
Like all gases, refrigerant occupies less space per pound as pressure surrounding it increases.

Say if refrigerant R22 is occupying 0.782 cu. Ft at 54.9 psig (30^0 F saturation suction temperature), the same refrigerant would occupy 0.556 Cu. ft at 84 psig (50^0 F sat. suction temperature) which illustrates that refrigerant occupies nearly 30% less space at 50^0 F compared to 30^0 F temperature.

At constant RPM, a refrigerant compressor pumps a constant volume per unit time. Let us take an example where a compressor is of 10 Cu. Ft. per minute capacity displacement & it would pump 10 Cu. Ft./ minute, regardless of the condition of the refrigerant gas entering its suction.

However, since the specific volume of the refrigerant gas varies with saturation temperature, the mass(pounds) of refrigerant pumped by the compressor also varies. In the example mentioned earlier the compressor pumping 10 Cu. ft at 30^0 F sat. suction temperature is pumping about 12.8 pounds per minute around the system (10 cfm ÷ 0.782 cu.ft / lb).

At 50^0 F sat. suction temperature it pumps about 18 pounds per minute (10 cfm ÷0.556 Cu. Ft / lb).

As the mass at 50^0 F is higher, the capacity & the power consumption are also higher.

Even though the system components are always is in balance with one another, that does not necessarily mean that the balance point occurs at an acceptable capacity at all times. When the system is having problems, it balances at an insufficient capacity & usually at high side & low side conditions that are too high or too low, causing unsafe or unstable system operation.

Fig – 25, This slide shows the system will balance at point 1 when it is 95^0 F outdoors & everything is working properly. The balance capacity is more than adequate. R-22 balance point saturation suction temperature is 40^0 F & saturated condensing temperature is 120^0 F.

If the condenser coil gets dirty or its air flow is reduced the balance point will be at 2. Now the capacity is too low to satisfy the building cooling load. The saturated suction temperature 50^0 F is high which causes higher compressor power draw. The saturated condensing temperature at 145 F is also high and the system must be repaired to bring about a balance at acceptable conditions.

Building load changes caused by solar, lighting & people loads can cause capacity imbalance. Any time that a system starts or experiences a load change, the system capacity will not be equal to the load and the system will try to restore equilibrium. It takes sometime for the system to settle down, once a change is made. During this period, the system operation will be unstable as high side & low side conditions are changing.

The systems therefore, even in good condition would always spend some of its operating life at a balance point where system capacity is not in equilibrium with building load.

It is therefore advisable to take readings of pressure/temperature once the system reaches steady state or equilibrium. The readings taken in a hurry before the system settles down can lead to wrong diagnosis.

PART LOAD CAUSED BY SERVICE PROBLEMS:

The part load operation due to load changes has been studied earlier, but the system in need of repair may also operate at part load because it is not capable of moving heat effectively.

Again going back to our tank & pump analogy, if we close the valve partially in the inlet pipe it is like reducing the evaporator capacity. If the load remains constant & the evaporator capacity falls, a higher than normal space temperature will result, as it happens with water level, which will steadily rise if inlet pipe to pump has restriction.

The evaporator capacity may fall due to many reasons, such as dirty filters, low evaporator air flow, dirty coil, damaged coil tubes or fins, restricted refrigerant liquid flow or low system charge. On the equilibrium diagram the point will shift to left indicating reduced suction pressure, lower compressor power draw & higher than required space temperature.

Similarly, if the pipe leaving the pump is smaller or has restriction it would not pump out water at the desired rate & it would be lower than the rate at which water is being added to tank resulting in water level rise.

This is equivalent to condenser capacity being inadequate. If the load remains constant but condenser capacity falls due to service problem, the space temperature will climb.

The reasons for reduction in condenser capacity can be many such as a dirty coil, corroded fins & tubes, damaged fins or tubes, restricted air flow, re circulation of condenser air, low system charge & non condensable gases in refrigerant.

On the balance diagram, the performance of the condensing unit shifts to left. The equilibrium reaches at higher suction pressure. The system capacity is reduced. Head pressure & compressor power draw will be high as also space temperature.

CAPACITY CONTROL METHODS :

Up to 10 Ton capacity systems, normally no capacity control strategy is adopted & the units simply turn on or off.

Taking the pump & tank analogy, if we provide a float switch that makes or breaks pump motor circuit then this can be compared to mechanical refrigeration system with only on – off control.

The float switch closes the circuit as the float rises to a predetermined level & opens the circuit when it falls to a predetermined level. The pump & piping attached to it deliver either full capacity or none.

The float switch is similar to thermostat used in mechanical refrigeration system. As capacity exceeds load, the space temperature drops & shuts off the system at a preset temperature level. When the system is off, the temperature rises until a preset temperature is reached, then the thermostat turns the system on. None of the components are having capacity control device.

FLOATING WITH THE LOAD

The system with no capacity control devices can still adjust itself somewhat to a change in the cooling load so capacity matches with load.

Let us assume that on a commercial comfort job the design load occurs on a 95^0 F & produces evaporator air entering wet bulb temperature of 67^0 F.

As the load reduces (wet bulb & enthalpy line coincide, hence for understanding load we would use wet bulb), the wet bulb of the air entering evaporator decreases, the capacity to absorb heat in evaporator is reduced to 64^0 F wb, moving the line to the left .

The condensing unit capacity line shifts to the right because the refrigerant within the tube is at the same temperature, whereas the ambient temperature has dropped to 85^0 F (load reduction due to ambient temperature drop). Hence the temperature difference has increased making condensing unit operate more efficiently giving higher output.

As the load is reduced, less refrigerant boils off whereas compressor continues to pump design volume, this leads to drop in suction pressure. The equilibrium balance system capacity almost remains same, however at lower suction pressure.

If the capacity still continues to be greater than load, the suction pressure will drop further & then the system will cycle off.

Thus it would cycle on / off depending upon thermostat set point. In other words, it floats with the load.

This type of on / off control is common on small capacity systems like window A.C, packaged terminal units, splits up to 5 Tons, household refrigerators & freezers etc.

The difference between on / off set point is called differential. A smaller differential will produce more frequent cycling, shorter running time & closer control on temperature set point. (Similar to water level).

A larger differential will produce less frequent system cycling & longer running time for each cycle with poorer temperature control on set point.

If the system is cycling excessively or has too short a running time, the service technicians can sometimes improve operation by increasing differential settings if available on the control thermostat.

The on/off control does not satisfy for wide load fluctuations & it has practical limits to how far the load can vary while maintaining safe & stable part load operation with reasonable efficiency.

As discussed earlier, the suction temperature falls rapidly in response to load reduction. Low suction temperature means system efficiency is reduced & in extreme cases lower suction temperature may be harmful. This is one of the reasons why capacity control devices are used to maintain saturated suction temperature within a reasonable range during load reduction.

When the system capacity is too large compared to load, the suction pressure tends to reduce quickly. Once the suction temperature drops to about 28^0 F, ice will begin to form outside of evaporator coil. If it builds up, air flow will be restricted & airside cooling capacity to the space will drop. The system will keep operating as it attempts to meet the cooling load. Ice will continue to build, dropping the air side system capacity still further. The evaporator capacity also drops off because the ice acts as an insulator, slowing down heat flow into the coil. Liquid refrigerant is likely to slug the compressor if the ice buildup becomes severe. The compressor will become noisy, vibrate excessively, overheat & consume extra energy. If the buildup continues the compressor may be cycled off by low pressure controls. Worse, it may possibly be damaged or even fail.

CAPACITY CONTROLLED METERING DEVICE

A normally used & most popular, inexpensive & reliable device for capacity control is a metering device called a thermostatic expansion valve or TXV. It is a modulating flow control valve which meters liquid refrigerant into evaporator in response to the superheat of the refrigerant gas leaving the evaporator. It allows refrigerant to enter the evaporator at the same rate at which it evaporates.

Modulating means it is gradual & smooth change, rather than in steps, in response to pressure or temperature signals from a sensing device.

Using a TXV is like placing a modulating control valve on the inlet pipe feeding water pump of an on/off controlled system. The flow sensor signals the valve to open or close, depending upon the flow rate of water entering the tank. As long as water level is maintained between the 'on' & 'off' float valve levels, the pump continues to run.

Similarly, the TXV under normal conditions allows the system capacity to balance with the load & to remain in operation until the temperature of the space drops enough to cycle the compressor & system off.

A fixed metering device like capillary tube or orifice tube & on –off control are standard on residential systems & heat pumps.

The TXV is offered as an option to the fixed metering device where the customer wants improved efficiency.

When properly sized & installed, most TXV's will provide stable control down to about 40 – 50 % of design capacity. When load drops below these limits, the valve will hunt, producing unstable system operation & endangering the compressor. The liquid feed to a TXV should be shut off before it enters its unstable range of low load operation.

An oversized valve will hunt, not stabilizing with load, but alternating from overfeeding to underfeeding the evaporator.

Most valves over sizing normally happen because the entire system is oversized even though the valve is properly matched to the system.

MULTIPLE COMPRESSORS:

Many times compressor is used as capacity control. One approach is to use two or more compressors, cycling them on & off to match the load.

This approach provides more stable operation on part load than a single cycling compressor.

It is as good as providing two half sized pumps in the tank analogy. This provides at a slightly greater cost, a closer match of water output with input, than does cycling on – off one single speed pump. Installing 2 small compressors will cost less than one single variable speed compressor or pump that modulates pump speed to match water input.

The compressor starting & stopping can be reset on suction pressure when suction pressure of 1st compressor reduces blow set point this compressor will be turned off. if the system load still continues to fall the second compressor would modulate(float) to match with the load till such time the load becomes too low even for one compressor & then second pressostat activates & turns off the second compressor.

The new reciprocating package chillers manufactured by leading Companies world over use multiple compressors in the same package & each is programmed to turn on / off in sequence, to synchronize with the load demand. This way the power saving effected is much better than using single compressor with step capacity control of unloading cylinders.

This control strategy also supplies some cooling even if one compressor fails.

Looking at the slide 52, one can notice that if the load is reduced (represented by wb. reduction from (67^0 F to 62^0 F) the point 1 will shift to point 2 reducing suction pressure/load, thereby matching with revised load.

If however, 2 compressors were in the system the line would shift to left & point 3 will be the equilibrium point, which indicates higher suction & reduced capacity. Higher suction means better efficiency.

TWO SPEED COMPRESSORS

In two speed compressors, compressor speed is changed in response to a thermostat or pressostat. The balance point would be similar to the one for a two compressor application.

UNLOADERS

Unloading reciprocating compressors is a popular method on many commercial & high capacity equipment. It is achieved by preventing one or more cylinders from pumping refrigerant into the system. This type of control helps in reducing capacity & power.

The number of steps depends on compressor design & number of cylinders.

In larger tonnage compressors unloading provides more & smaller capacity steps at part load than is possible with only multiple compressors.

If multiple compressors to meet total load are used, the compressor cycling is often combined with cylinder unloading to achieve desired steps of capacity control.

The ideal situation for matching load with capacity is variable speed compressor rather than incremental or stepwise techniques. The compressor would then operate continuously, safely & efficiently at virtually any system load. Space temperature would remain constant because equipment capacity would match space load all the time. Electronic controls technology has made such motor control feasible for most types of compressors.

The centrifugal compressors use inlet guide vane control for modulating capacity & screw compressors use slide valve at the inlet to modulate capacity.

EVAPORATOR SECTIONS OR SPLITS

As discussed earlier other components of the refrigeration system can also be provided with capacity control arrangement.

The popular method is to provide multiple section, or split circuit. Such type of coil uses multiple metering devices for each evaporator section or split. As cooling load drops below a certain level, the refrigerant feed to one or more evaporator sections or splits is stopped. This allows the portion of the evaporator still in operation to function more efficiently safely & smoothly at reduced load than the entire coil surface could.

It also allows the metering device, distributor & nozzle, which feeds the evaporator section or split, to stay within safe operating limits.

Suction pressure or air temperature is most often used to activate or deactivate the liquid line solenoid valve which feeds each split or section.

Now let us take an example in which air at 62^0 F wet bulb is entering the evaporator & cooling load reduces. If the evaporator has two splits & both remain active, the system will balance say at point 1 (slide 57). This capacity is actually a bit higher than actual load. Therefore, the suction temperature will drop off significantly in an attempt to balance capacity with load. If the suction temperature drops for enough, the coil will frost & will endanger the compressor. If the space temperature / suction pressure falls enough to cycle for some period causing poor efficiency, unstable operation & possibly occupant discomfort will result.

If one of the splits is shut off, the evaporator capacity line will shift to the left. This is as good as installing a smaller evaporator in the system. If the condensing unit capacity is not controlled, the balance point will be at point 2. Although it is having lower suction & thereby less efficiency, it still provides safer, more efficient & more stable operation.

Point 3 shows the result of using a capacity controlled condensing unit along with split evaporator. With condensing unit operating at 50% of capacity, its performance curve shifts to the left, intersecting the evaporator line at point 3.

As the capacity in this case is too low compared to requirement, the condensing unit will return to 100% capacity in an attempt to equalize capacity with load.

However, at lower loads, the benefit of a capacity controlled condensing unit is obvious. It allows a greater capacity drop off with higher saturated suction temperature.

CONDENSER CAPACITY CONTROL

It is more commonly called as head pressure control. It reduces the condensing heat rejecting capacity as outdoor temperature falls thereby holding the head pressure at a minimum level for proper TXV feed control.

The methods of head pressure control include condenser fan cycling, condenser fan speed control, condenser air dampers & refrigerant flood back control. The part load control of head pressure is normally accomplished by high side refrigerant pressure or temperature sensing device.

Head pressure control is, as a rule provided on all commercial refrigeration applications installed in cold climates. This ensures proper pressure differential available for metering devices to function efficiently.

HOT GAS BYPASS

Hot gas bypass is a capacity control strategy applied to refrigerant piping system.

As cooling load drops, the compressors will be unloaded & cycled, the TXV's modulated & evaporator splits deactivated as necessary. Hot gas bypass will be activated on a lowest step of cooling capacity, just before the mechanical refrigeration system is cycled off.

It ensures minimum saturated suction temperature regardless of the load & capacity step.

Equilibrium of capacity with load is established in an artificial way. This adds stability to system operation & insures a low side pressure & saturation temperature which is in a safe range at low load.

As the hot gas bypass valve opens, hot gas from the compressor discharge is mixed with low temperature, low pressure mixture of liquid & gas coming from metering device. This mixture entering the evaporator adds artificial cooling load to evaporator from the inside because of the introduction of higher temperature discharge gas coming from the compressor.

Some of the gas being pumped out of the compressor is bypassing the condenser & metering device and entering the evaporator directly.
Hot gas bypass does not lead to energy efficiency, however when properly applied is an effective safety device for stable operation of system at low loads.

It is also essential where the applications prohibit on / off cycling & require compressor to work continuously without tripping at all load conditions. Such applications are Bus A.C. / Refrigeration working with separate diesel engine, special units running on Gen. sets etc., where high starting current can cause malfunctioning of electronics if it is done too often.

Also if the loads are too low, as a rule & in oversized plants, if installed can create havoc, as hot gas circuit then is constantly on, increasing not only energy bills but making the compressor operate continuously with high temperatures, thus reducing compressor's life.

MYTH ABOUT OVERSIZING EQUIPMENT AND SYTEM

The very best product & the most careful design will not produce stable, efficient & safe part load operation if the system is severely oversized.

Equipment over sizing is common design error which causes problems for building occupants & service technicians.

As a thumb rule system should never be more than 20% oversized. It is however, common to find equipment that is 50% - 100% oversized.

At part load, it will load & unload or cycle excessively. Space humidity is likely to climb up because of reduced running time. The space conditions may be difficult to maintain, causing objectionable temperature swings. Installation operating & service costs will tend to be high & the equipment life expectancy will be reduced.

It is generally agreed that maximum equipment life occurs with continuous, stable equipment operation. In fact the testing of equipment is done with rapid cycling & de-stabilizing the conditions to test the components after manufacture. To duplicate these worst conditions in the field is asking for trouble.

Let us take an example to prove how over sizing a plant proves detrimental.

It involves a small office building with an actual design load of 30 tons. The system was sized for a 60 Ton load. It has two, 30 Ton condensing units each with 3 steps of capacity control. Coil used in air handler is of 60 Ton capacity.

As can be seen system has total 6 small capacity control steps ideal for 60 Ton load. In reality, however, second condensing unit never ran due to maximum design load being 30 Ton. As a result, 3 of the capacity steps were never available.

The actual steps available therefore were 100%, 67% & 33%on a single compressor, each step being of 10 Ton. Since the building often operated below 50% of the actual 30 Ton design load (below 15 Ton), the steps were in reality larger compared to the existing load, rather than small as desired.

The system operated poorly from the 67% step downward. Two of the three steps would not equalize capacity with load at acceptable conditions. Because of low suction pressure, hot gas was activated at the 67% (20 Ton) capacity step rather than at the lowest step. The 33% (10 Ton) step never operated & energy costs at reduced load were high. Customer had to pay higher energy bills, in spite of installing energy saving variable air volume system.

A smaller cooling coil & two 15 Ton condensing units were recommended as a solution.

Like many oversized jobs it was overpriced, costly to operate & produced discomfort to the occupants.