Application of Screw Compressors

APPLICATION OF SCREW COMPRESSORS IN INDUSTRIAL, PETROCHEMICAL AND RELATED INDUSTRIES - COMPARATIVE EVALUATION OF CHARACTERISTICS OF SCREW AND CENTRIFUGAL COMPRESSORS

By R.P.Paranjpey, Vice President(ACR Operations), Kirloskar Pneumatic Co.Ltd., Pune

Synopsis

This paper deals with functioning of oil injected screw compressors for process cooling application particularly comparing characteristics with centrifugal compressor for similar duties. It throws light on various aspects of selection like like maintanance, reliability, thermodynamic properties and suggests a method of establishing realistic energy consumption parameters for arriving at genuine power consumprtion on annualised basis.

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At a particular point along the length of the casing a discharge port is positioned which will permit the gas to pass out of the compressor.

The amount of internal compression which occurs before release is therefore a characteristic which can be varied by positioning and shaping the discharge port. This feature is known as the 'bust in compression ratio' of the set.

The centrifugal compressor is an aerodynamic machine. It carries out the process of compression by generating kinetic head or 'momentum' in the gas as it passes through the spinning impeller and by 'slowing down', the gas after it leaves the impeller converts the kinetic energy in the gas to pressure energy or in other words 'head'.

Having understood the difference in working of two types, it is worth to go into the short history behind developemnt of screw compressors.

The screw compressor was first invented by Lysholm in Sweden in the middle of 1930's and he at that time was working on the development of a gas turbine and was having troubles with the centrifugal compressor involved. He therefore developed screw compressor as a replacement, the aim being to have a positive displacement compressor thereby eliminating the possibility of surging and being capable of operating at a higher rotational speeds above that possible with reciprocating designs.

Although the design was sucessful, its specific volume, i.e. volume per unit rotation was lower than was needed for the gas turbine and its developemnt was therefore seperated out for futher work as an industrial unit not associated with gas turbine. Initial developement was carried out by a Swedish organisation

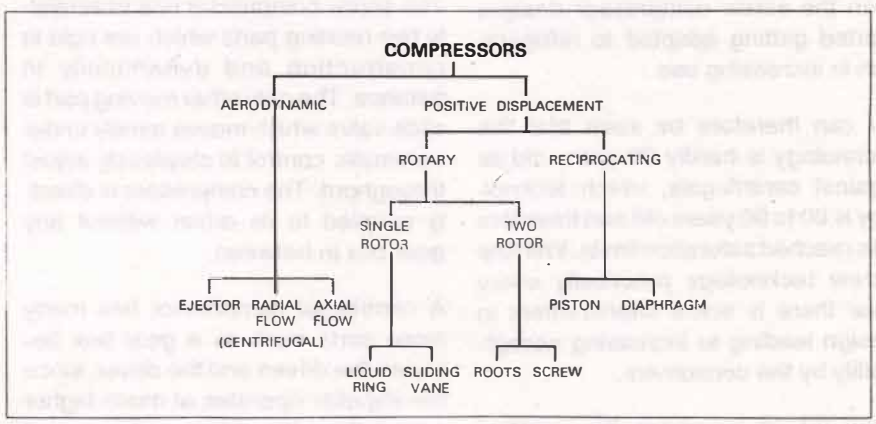

FIG 1

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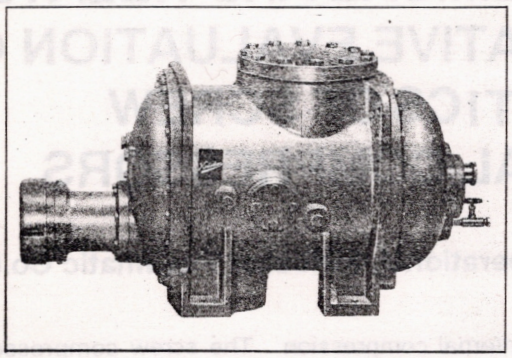

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The capacity control was developed in 1965, rotor designs were modified to unsymetric profile in 1970 and since then the screw compressor designs started getting adapted to refrigeration in increasing use.

As can therefore be seen that this technology is hardly 20 years old as against centrifugals, which technology is 80 to 90 years old and therefore has reached saturation limits. With the screw technology practically every year there is some improvement in design leading to increasing acceptability by the consumers.

Now let us compare the various properties and behaviours of two types of machines under diffrent condition which normally are encountered in the field.

Reliability of operation and simplicity in design

The utmost priority in any industrial process cooling equipment is uninterrupted trouble-free and reliable operation with minimum down time. This requirement though not as critical for comfort airconditioning; it is of vital nature for nuclear power plants, chemical-dyestuff industries, petrochemicals and fisheries; Telephone exchanges, Computer Complexes, Marine airconditioning, mine cooling and other industrial chilled water applications.

Screw compressors stands a clear winner in this respect.

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A centrifugal compressor has many more parts such as a gear box between the driven and the driver, since the impeller operates at much higher speed than the driver and a speed increasing gear box is a must.

Then in order to be competitive powerwise most centrifugal applications have multistages using more than one impeller mounted on one shaft, interstage coolers, movable inlet guide vanes as well as movable discharge diffusers, vane control mechanism with modutrol motors. Since the machine works under vaccum a purge unit using seperate compressor is also necessary.

Thus it can be seen that overall a screw compressor installation has considerably fewer moving components than an equivalent centrifugal compressor plant.

The Mean Time Between Failure of machine (MTBF) is directly related to the number of moving items involved in compressor operation. Less the number of components, the less possibility is to go wrong. The screw compressor meets this requirement ideally. The reliability of the compressor is such that recemmended overhaul period is after 50,000 hours of operation which is equivalent to 6 years and many compressors have clocked upwards of 80,000 hours without major overhauls.

Ability to withstand liquid

As all maintenance plant operators would agree, major problem in any refrigeration plant is liquid slug because of malfunctioning of certain components and parameters.

In case of screw compressor, a slug of liquid will cause no damage to compressor. When liquid refrigerant enters the suction port, a certain amount of liquid separation occurs at the inlet. This means that certain amount of gas is also sucked into compressor together with liquid refrigerant and since the built in volume ratio of refrigeration screw is normally less than 5, the volume of the interlobe space at outlet is large enough to pump out the gas liquid refrigerant mixture without any liquid hammer or without any mechanical stresses or strains.

A centrifugal compressor will suffer severe damage if liquid refrigerant is carried over.

Corrosion

As everyone knows centrifugals operate at negetive pressures i.e. under vaccum and water circulates

through the tubes at higher pressures. In case of leaks the water enters the shell side and with R 11 forms highly corrosive hydrofluoric acid. A centrifugal machine is therefore very prone to such corrosion damages and a machine which has suffered from such leakage can be equated to a person who has suffered a heart attack. It would never be the same with any amount of maintenance.

The screw compressors work with high pressure refrigerants R22, Ammonia and numerous other gases. Hence this possibility does not exist in most of the applications.

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As centrifugal compressors work under vaccum the air leakage together with its moisture in machine is a constant problem and to purge air, a unit is provided which is supposed to purge out non-condensibles. However as has been experienced by every maintenance engineer along with air valuable refrigerant gas also escapes polluting the environment as well as needing frequent topping up of gas.

Having studied the characteristics of both types from maintenance, operation and selection angles we shall now look at the thernodynamic performance of them at various conditions.

Operation with variable speed

It is undisputed fact that maximum efficiency is achieved when the speed is varied proportionately to load requirements. This is not possible with centrifugals as they work on kinetic energy developed and hence have to operate at high speeds through gearboxes. For screw compressor it can develop full discharge pressure regardless of speed and there are therefore no limitations due to pressure conideration. The volume handled and power are directly proportional to speed. The lubricating system being independent is not affected even if the screw compressor speed is reduced. With the advancement of electronic tyrister control speed reducers the systems are bound to become more popular in not too distant future as the cost of the speed reducers, is falling rapidly. The screw compressors can also be driven by variable speed drives such as turbines or diesel engines.

Gases and Refrigerants handled

Screw compressor being a positive displacement can handle almost any gas ranging from lightest Helium to heaviest Propane. Normal refrigerants are Ammonia, R12, R22, Propane and Propylene besides other 22 normally used gases.

Centrifugal compressors tend to use lighter low pressure gases like R11, R12, R21 etc. because of its charecteristics.

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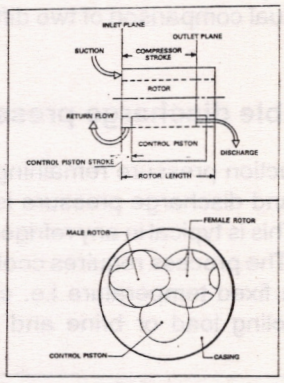

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Under these conditions comparison is simple as the absorbed powers are constant at all times.

It can be found that the performance of screw and centrifugal compressors are comparable and can be seen from the actual comparison of two different types.

Variable discharge pressure

The suction pressure remaining constant and discharge pressure is variable. This is typical in any refrigeration plant. The process requires cooling at certain fixed temperature i.e. chemical cooling load or brine and water

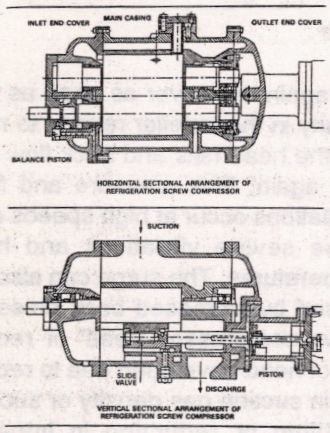

FIG 4

chilling duty and the suction duty is therefore constant and automatically controlled to match design operating temperatures.

The condenser design and selection although based on worst set of conditions foreseen to ensure that the plant is capable of operating in such conditions i.e.allowing for the hottest time of years and highest ambient humidity. In practice this condition occurs for less than 5% of the total time cycle and the condensing systems experience a wide range of conditions from the heat of the day to the cold of the night and from summer to winter. In screw compressor the pressure developed at the discharge is that pressure necessary to move the gas out of the compressor and no more. The power absorbed therefore is proportional to the discharge pressure.

In case of centrifugal compressors the discharge pressure capability is directly related to the diameter of impeller, its speed of rotation and the density of gas it is handling.

As there is a clear gas connection through the compressor from discharge, to the suction through impeller, changes in discharge conditions have a direct effect on the suction volume. The power absorbed does not fall in line with reduction in discharge pressure and the power increases with reduction in discharge pressure. The volume also tends to increase and therefore to limit suction volume to desired level the vanes close. The relative flat power characteristic results from compressor generating a discharge pressure or head to suit design conditions which (at other duties) is in excess of requirement and the extra power is therefore wasted.

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This is an area which is always ignored in comparative assessment but has a surprisingly significant effect on actual absorbed power. As one knows, fouling factors are allowances built into heat exchanger designs so that when fouling occurs the heat exchangers will operate at its design heat transfer rate. Significantly from the time of installation upto fouled conditions, screw compressors would perform much better than centrifugal as the areas available for the given duty being higher it would tend to close the gap between discharge pressure and suction pressure reducing compression ratio and there would be direct power saving as machines would then operate at part load leading to direct power saving. Centrifugal machine would continue to operate at a designed discharge pressure nullifying the advantages which otherwise it would have derived if it was a positive displacement machine.

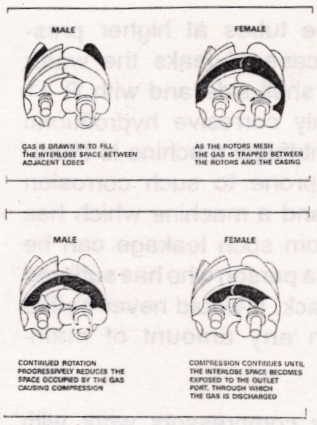


FIG 5

Power consumption and suggested method of plant assessment

No matter how reliable any machine might be, it is of little consequence if it does not operate efficiently. From the foregoing paras it can be seen that two types of plants may have virtually identical absorbed powers at one condition and quite different ones when there conditions alter in actual practice.

The present system of comparing is various acceptable bids are compared on power absorbed. A value is given to each KW of power used, based on cost per KWH over the number of years operation, each bid is compared against the lower power absorbed bid and the Kilowatt power difference multiplied by the value given to each KWH is added to the bid price. The totals thus obtained are then used for bid selection.

The entire method is based on Design point selection only and therefore the validuty of such results has a very serious flow as it does not reflect the total energy consumed by the machine over the entire yearly operating cycle.

It is natural that while sizing the plant and its drives it is necessary to rate them both to be capable of coping with the most arduous conditions. However the plant will run at lesser condi-

tions for majority of its operating life. Hence it has now become necessary to use more sophisticated method to obtain more realistic and genuine operating costs.

A typical comparison gives following results:

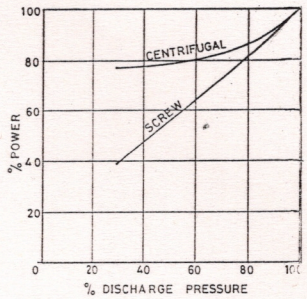
We are furnishing herewith an example which shows power consumption of the compressors at different condenser inlet water temperatures. This is for a 300 ton package chillers producing standard chilled water at 7°C.:

Condenser Water inlet Temperature	Power Consmp. of Screw compr. KW/TR Shaft Power	Power Consmp. of Centrifugal Compr. KW/TR Shaft Power	Duration of operation in a year
32 DEG C	0.79	0.803	400
30 DEG C	0.741	0.746	400
27.5 DEG C	0.691	0.708	1600
25 DEG C	0.645	0.656	2140
22.5 DEG C	0.598	0.634	1060
20.5 DEG C	0.558	0.596	1600
17.5 DEG C	0.521	0.566	400
15 DEG C	0.482	0.544	400
Total KWH/Ton	5024	5226	8000

Total power consumption per ton for screw package chiller in Kwhr over a period of one year = 5024 Kwhr.

Total power consumption per ton for centrifugal package chiller in Kwhr over a period of one year = 5226 Kwhr.

Total saving per ton over a period of one year @ Rs.1.50 per Kwhr power charges will be Rs.303/-. For this example using 300 tons capacity, power saving using screw compressor will be Rs.90,900/-. Additionally since the maintenance cost of the screw compressors are much lower, the total operating cost will be much lower.



Comparison Of Power and Discharge Pressure Characteristics at Constant Speed With Fixed Suction Pressure.

FIG 6

In a non coastal city for 95% of the total operating time, the condenser water temperature will be in the range of 15 deg C to 25 deg C whereas only for 5% of the operating time the water temperatures will be around 30 deg C. This clearly shows considerable power saving by screw compressor than centrifugal at varying condensing temperatures.

Environmental safety

Presently the most important aspect worrying all manufacturers of refrigeration equipment and system designers is effect of refrigerants on environmental pollution. With the 1987 Montreal Protocol most of the countries have imposed heavy restrictions on use of R12 and R12 and by 1992 these refrigerants are to be phased out partially and by 1998 totally. As the centrifugal has to necessarily depend on these two R11 and R12 refrigerants and cannot used R22 for the size of the projects normally encountered, new designs are under development. The new refrigerants HCFC 123 and HCFC 134 as substitutes although have been developed, the availability of these refrigerants is extremely limited even for the developed countries. Only one manufacturer in the world has started commercial production on a small scale. The compatible lubricating oils are also under development stage, the after-effect of these refrigerants are yet to be realised although lab tests prove these to be safe, and with these restricting factors coupled with high cost of refrigerant ($2^1/_2$ - 3times) and that of oils many leading companies like carrier. Trane, Hitachi are having renewed interest in screw compressors and would be introducing these in world market in 1991/92. M/s. Hitachi Japan have already developed chilled water package using screw compressors and marketing these in Middle East markets.

India is fortunate in this respect since this technology already has been assimilated by Indian manufacturers and more than 60 to 70 installations are already using indigenously produced screw compressors with environmentally friendly refrigerants.

Based on the various factors discussed above, I now leave it to consultants, areas and the august gathering to take into consideration all above aspects while evaluating their requirements. ■

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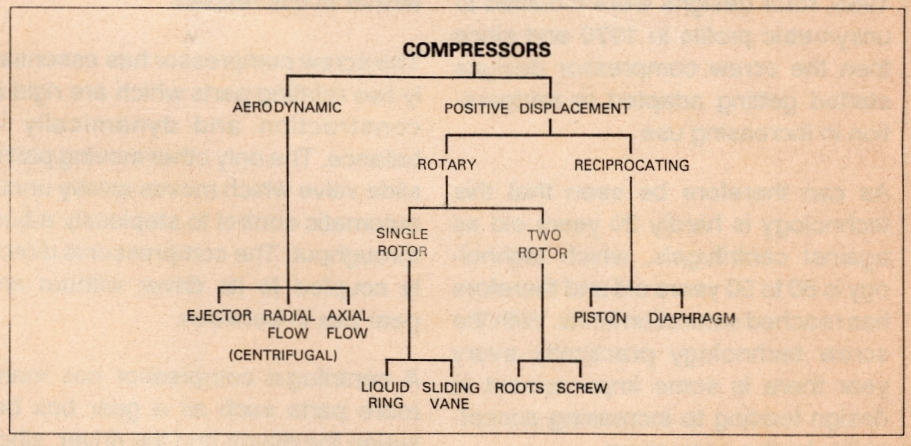

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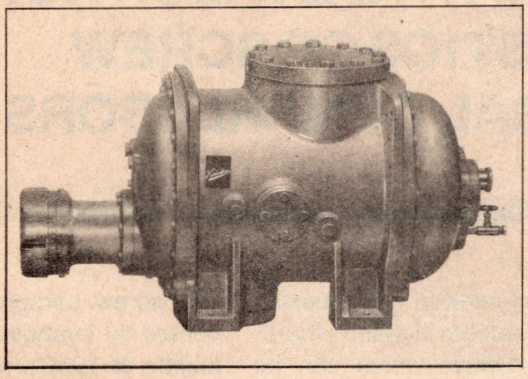

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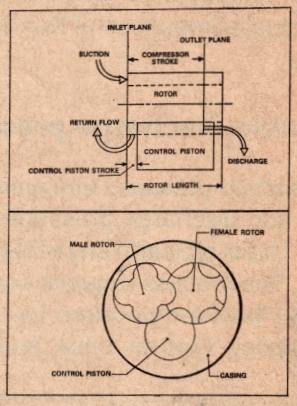

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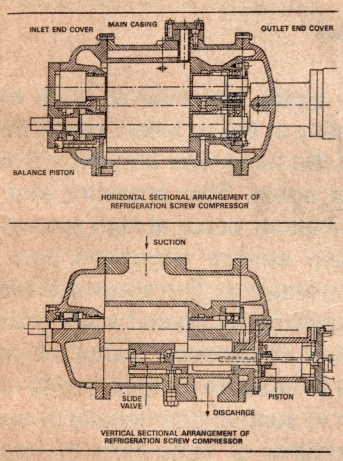

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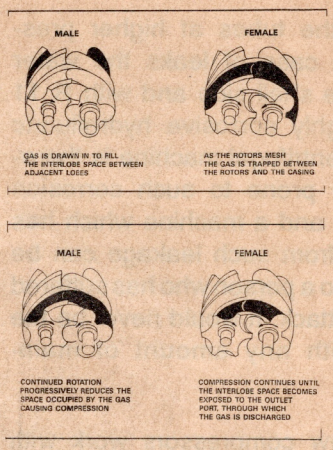


FIG 5

Power consumption and suggested method of plant assessment

No matter how reliable any machine might be, it is of little consequence if it does not operate efficiently. From the foregoing paras it can be seen that two types of plants may have virtually identical absorbed powers at one condition and quite different ones when there conditions alter in actual practice.

The present system of comparing is various acceptable bids are compared on power absorbed. A value is given to each KW of power used, based on cost per KWH over the number of years operation, each bid is compared against the lower power absorbed bid and the Kilowatt power difference multiplied by the value given to each KWH is added to the bid price. The totals thus obtained are then used for bid selection.

The entire method is based on Design point selection only and therefore the validuty of such results has a very serious flow as it does not reflect the total energy consumed by the machine over the entire yearly operating cycle.

It is natural that while sizing the plant and its drives it is necessary to rate them both to be capable of coping with the most arduous conditions. However the plant will run at lesser condi-

tions for majority of its operating life. Hence it has now become necessary to use more sophisticated method to obtain more realistic and genuine operating costs.

A typical comparison gives following results:

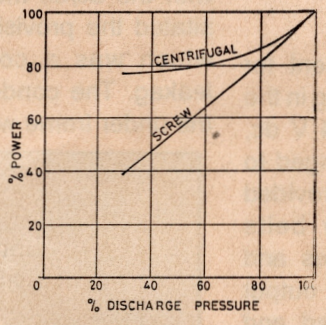
We are furnishing herewith an example which shows power consumption of the compressors at different condenser inlet water temperatures. This is for a 300 ton package chillers producing standard chilled water at 7°C.:

Condenser Water inlet Temperature	Power Consmp. of Screw compr. KW/TR Shaft Power	Power Consmp. of Centrifugal Compr. KW/TR Shaft Power	Duration of operation in a year
32 DEG C	0.79	0.803	400
30 DEG C	0.741	0.746	400
27.5 DEG C	0.691	0.708	1600
25 DEG C	0.645	0.656	2140
22.5 DEG C	0.598	0.634	1060
20.5 DEG C	0.558	0.596	1600
17.5 DEG C	0.521	0.566	400
15 DEG C	0.482	0.544	400
Total KWH/Ton	5024	5226	8000

Total power consumption per ton for screw package chiller in Kwhr over a period of one year = 5024 Kwhr.

Total power consumption per ton for centrifugal package chiller in Kwhr over a period of one year = 5226 Kwhr.

Total saving per ton over a period of one year @ Rs.1.50 per Kwhr power charges will be Rs.303/-. For this example using 300 tons capacity, power saving using screw compressor will be Rs.90,900/-. Additionally since the maintenance cost of the screw compressors are much lower, the total operating cost will be much lower.



Comparison Of Power and Discharge Press Characteristics at Constant Speed With Fixed Suction Pressure.

FIG 6

In a non coastal city for 95% of the total operating time, the condenser water temperature will be in the range of 15 deg C to 25 deg C whereas only for 5% of the operating time the water temperatures will be around 30 deg C. This clearly shows considerable power saving by screw compressor than centrifugal at varying condensing temperatures.

Environmental safety

Presently the most important aspect worrying all manufacturers of refrigeration equipment and system designers is effect of refrigerants on environmental pollution. With the 1987 Montreal Protocol most of the countries have imposed heavy restrictions on use of R12 and R12 and by 1992 these refrigerants are to be phased out partially and by 1998 totally. As the centrifugal has to necessarily depend on these two R11 and R12 refrigerants and cannot used R22 for the size of the projects normally encountered, new designs are under development. The new refrigerants HCFC 123 and HCFC 134 as substitutes although have been developed, the availability of these refrigerants is extremely limited even for the developed countries. Only one manufacturer in the world has started commercial production on a small scale. The compatible lubricating oils are also under development stage, the after-effect of these refrigerants are yet to be realised although lab tests prove these to be safe, and with these restricting factors coupled with high cost of refrigerant ($2^1/_2$ - 3times) and that of oils many leading companies like carrier. Trane, Hitachi are having renewed interest in screw compressors and would be introducing these in world market in 1991/92. M/s. Hitachi Japan have already developed chilled water package using screw compressors and marketing these in Middle East markets.

India is fortunate in this respect since this technology already has been assimilated by Indian manufacturers and more than 60 to 70 installations are already using indigenously produced screw compressors with environmentally friendly refrigerants.

Based on the various factors discussed above, I now leave it to consultants, areas and the august gathering to take into consideration all above aspects while evaluating their requirements. ■

Airconditioning Systems For Rail Coaches In India

Vincent D'Silva

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The Indian Railways are the second largest Railway System in the world, operating nearly 8000 passenger services a day (including suburban services) to move about twelve million people over a route length of approximately 70,000 Kms. Apart from the responsibility of ensuring the safe and speedy transportation of passengers, the railways have to provide for the convenience and comfort of tourists and long distance passengers. As an effort in this direction, airconditioned coaches were introduced on the Indian Railways in the 1950's. Since then the airconditioned coaches have been greatly patronised by the travelling public especially due to the tropical weather conditions prevailing in India.

Power Availability

The power supply arrangement for train lighting and airconditioning in the older trains was initially at 24 V dc, which was subsequently changed to 110 V dc. The coaches were provided with alternators driven by belts mounted with pulleys on axles and alternator shaft. The 3-phase output of the alternator was rectified and regulated. The battery provided in parallel with the rectifier regular output fed the load while the coaches were stationary. On the new trains introduced in the last few years, power cars have been provided which have diesel generating sets whose output is 415 V, 3-phase. The power supply from these power cars is comparitively more reliable and it also facilitates the use of 3-phase induction motors for different requirements.

The Old Design

In the older AC rail coaches, the compressor and condenser were mounted in the underframe of the coach, and the evaporator coils were provided at both ends of the coach between the ceiling and the false roof. Due to this arrangement, the system required the laying of copper pipelines for the refrigerant from the underframe to the roof of the coach as well as to the control panel located inside the coach. The joints in these pipelines as well as the bends were a perpetual source of gas leakage. Cracks also developed in the pipes due to the vibrations experienced continuously on the rolling stock.

The open type compressors, coupled with the ac/dc motors used, necessitated the provision of a shaft seal which was a constant source of leakag. The condensers mounted in the under frame were exposed to accumulation of dust, resulting in clogging of fins and damage by flying ballasts during the running of the coaches at high speeds. The accumulation of dust between the fins of the condenser resulted in the decrease of the heat rejection capacity of the condenser, requiring the provision of a higher size of the condenser. During extreme summer conditions, it was sometimes necessary to spray water on the condenser for its cooling to reduce the discharge pressure.

The capacity of the airconditioning plant was also limited on account of the availability of space in the underframe, since a battery of very high capacity apart from the alternator and rectifier-cum-regulator was required to be accommodated in the underframe itself.