Design of Resonating, Oil-Free Linear Compressors for Five-Stage Cascade System with New Refrigerants

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ABSTRACT

Due to its reliability and high efficiency compared to other systems, Vapor Compression Refrigeration (VCR) systems are most famous systems being used for refrigeration purposes. To make this system more efficient, eco-friendly, durable and compact, a linear compressor seems to be advantageous over the conventional compressor. The linear motor has been proved to be electrically more efficient than that of a rotary induction motor. Along with this, absence of motion conversion mechanism makes linear compressor drive mechanically more efficient and durable. Also, oil-less operation is possible in this case which have its own benefits.

A cascade refrigeration system is one of the ways to reach low temperatures. It consists of number of VCR stages cascaded together and each run by its own individual compressor. In this study, a cascade system with total cooling capacity of 20 W at the temperature of 78.8 K had been taken into consideration. This cascade system consisted five stages with refrigerant R1270, R290, R1150, R50 and R729 in series. The first loop of the cascade system (with R1270) was designed such that it could be used as dual purpose, either as one of the loops in a cascade system or individually for air conditioning purpose. The aim of this study was the design of linear compressors for each stage with resonating and oil-free operation.

The variation of gas spring stiffness had been observed and with some allowance on natural frequency, design has been done for resonate operation. For oil-free operation, enthalpy leakage rate due to transfer of refrigerant past the radial clearance between piston and cylinder has been analyzed and its effect on overall system performance was determined. By putting a maximum limit on drop in COP of the system, maximum radial clearance that could be provided for oil-free operation had been found for the five stages.

INTRODUCTION

A major part of energy ever produced is consumed either for cooling or heating [1]. Though there are many methods available for cold production, vapor compression refrigeration is very famous among them due to its high COP [2]. The vapor compression system is essentially comprised of a compressor, condenser, expansion valve and evaporator arranged in circuit. But it is the compressor where actual power is consumed [3]. Therefore its efficiency and durability is very important.

The linear compressor is driven by a linear motor. the linear motor exerts force to the compressor in desired linear fashion. As there is no need for a motion conversion mechanism, more mechanical

Cryocoolers 21, edited by R.G. Ross, Jr., J.R. Raab and S.D. Miller

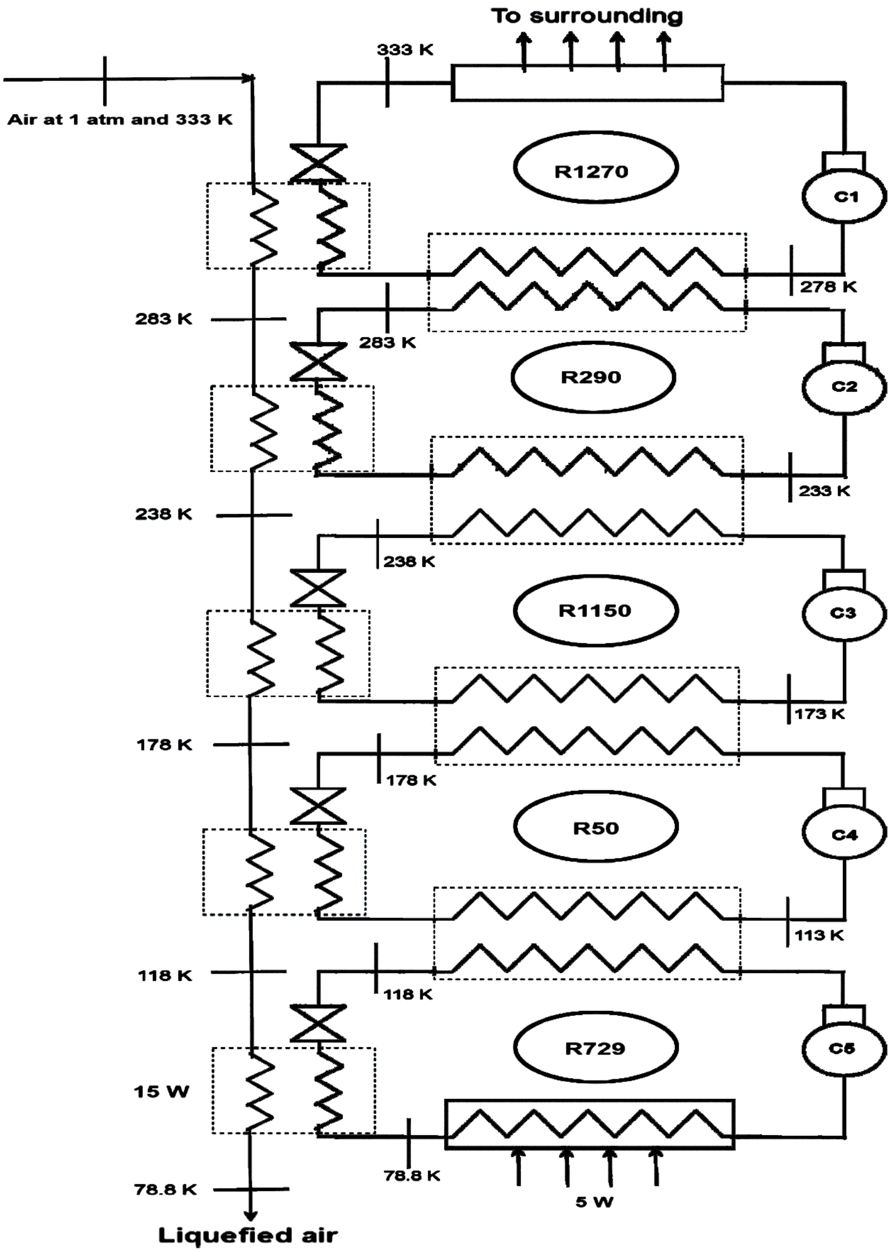

Figure 1. Cascade system.

efficiency can be expected from the linear compressor. Also, linear motors are proved to be more electrically efficient than that of rotary induction motor [4]. One more benefit of using a linear compressor is oil-less operation is possible which would obviate all the problems and constraints associated due to presence of oil in VCR system.

Studies have found that performance of linear compressors is at peak when operated at resonance but decreases considerably for off resonance operation [5]. Therefore, to observe high overall COP of VCR systems with linear compressors, they should be operated at resonance frequency. The oil-less operation has its own benefits and this combined resonating and oil-free operation of

Table 1. System requirement for various stages of cascade system.

Stage	Refrigerant	Condenser temperature (K)	Evaporative temperature (K)	Refrigerating effect (W)	Mass flow rate of refrigerant (g/s)
1	R1270	313	278	390	1.4070
2	R290	283	233	249	0.7070
3	R1150	238	173	131	0.4358
4	R50	178	113	48	0.1684
5	R729	118	79	20	0.1606

linear compressor is good not only in the sense of efficiency but also in terms of durability and can bring compactness in heat exchangers and overall system size.

CASCADE SYSTEM

This work was primarily concerned about liquefaction of air and hence the minimum temperature to be attained was 78.8 K. This could be achieved by a five stage cascade system as shown in Figure 1.

One application had been chosen for this study which was, normal cooling load of 5 W at 78.8 K, with liquefaction of air with capacity of 15 W at 78.8 K

So a total load of 20 W at air liquefaction temperature of 78.8 K had been chosen to be a requirement for cascade system and design of linear compressors for each stage of the VCR system for resonating and oil-free operation was the objective of this study. For no unhealthy emission from the system, all the VCR stages had to run on eco-friendly refrigerants. In this sense, propylene (R1270), propane (R290), ethylene (R1150), methane (R50) and finally dry air (R729) had been chosen to be used in sequence. Air at normal atmospheric pressure and temperature of 40 °C would be taken and then cooled in the evaporators of successive stages of cascade system. Temperature differences of 5 °C had been assumed to be there for heat transfer and as operating temperature were very low; it was assumed that 20% refrigerating effect had been lost for non ideal insulation. With all these requirements and assumptions, the refrigerating effect needed at various stages, their working temperature range and thus required mass flow of refrigerant came out to be as shown in Table 1.

The compression process had been assumed to be isentropic. The actual compressor operates at a lower volumetric efficiency than the theoretical value (which depends on clearance volume and pressure ratio employed) and it was found that theoretical calculated efficiency is 15 to 25% more than the actual one, so accordingly, after calculation, requirements came out to be as shown in Table 2. The choice of stroke length depends on many factors like speed of compressor, pressure ratios etc. and owing to use of linear compressor with resonating and oil-free operation, some more constraints had come out as follows, stroke to diameter ratio, gas spring stiffness, maximum radial clearance and maximum stroke length of linear motor

The overall isentropic efficiency decreases with an increase in stroke to diameter ratio. Also piston drift increased with the stroke to diameter ratio of the piston [5].Therefore, this ratio should be low to realize good performance. There was another problem of decreasing this ratio. In resonating operation,

Table 2. System requirement for various stages of cascade system.

Stage	Evaporative pressure (MPa)	Condenser pressure (MPa)	Clearance space provided (%)	Theoretical Volumetric efficiency (%)	Actual Volumetric efficiency	Swept Volume (cm^3)
1	0.676	1.713	5	93.3	77.75	2.51
2	0.111	0.636	5	80.3	66.92	8.03
3	0.126	1.729	5	65.6	54.64	5.99
4	0.119	3.131	4	57.1	47.55	3.51
5	0.103	1.838	4	72.1	60.07	1.19

Table 3. Chosen stroke lengths and bore sizes for compressors.

Stage	Refrigerant	Stroke length (mm)	Bore size (mm)	Stroke to diameter ratio
1	R1270	14	15.2	0.921
2	R290	14	27	0.518
3	R1150	18	20.6	0.873
4	R50	16	16.7	0.958
5	R729	10	12.5	0.800

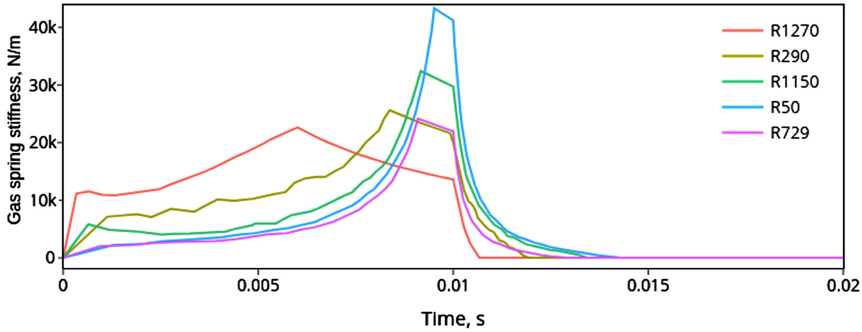

Figure 2. Variation of gas spring stiffness during the compression cycle.

gas spring stiffness is important factor to be considered. It is seen that its magnitude is highly depended on bore size. With these guidelines and after some trials, choices mentioned in Table 3 seemed to be good.

RESONATING OPERATION

When the natural frequency of the system matches with the frequency of excitation force, resonance is said to occur. The frequency of the excitation force would be equal to that of input AC (=50 Hz). The natural frequency of the system depends on total spring stiffness and moving mass of assembly. Adjusting these two parameters, the natural frequency of the system can be made equal to the excitation frequency (or resonance frequency). This strategy had been adopted in this study to achieve resonance. The magnitude of gas spring stiffness is given by,

$$k_{gas} = \frac{\text{Force}}{\text{Displacementfrommeanposition}} = \frac{\Delta p.A}{x} = \frac{\Delta p.A}{\Delta V/_{A}} = \frac{\Delta p.A^2}{\Delta V} = \frac{\Delta p.A^2}{V_1 - V} \quad (1)$$

The variation of gas spring stiffness in the compressor for different cascade stages was observed to be as shown in Figure 2. Its values at the end compression and at the end of discharge are given in Table 4.

Table 4. Gas spring stiffness at the end of compression and at the end of delivery.

Stage	Refrigerant	Gas spring stiffness at the end of compression (N/mm)	Gas spring stiffness at the end of discharge (N/mm)
1	R1270	22.6318	13.6042
2	R290	25.6295	21.4613
3	R1150	32.4374	29.7407
4	R50	43.3200	41.2129
5	R729	24.1583	21.9758

Table 5. Maximum variation of natural frequency for different stages.

Stage	Refrigerant	Spring stiffness (N/mm)	Moving mass (Grams)	Minimum natural frequency (Hz)	Maximum natural frequency (Hz)	Maximum variation in natural frequency (Hz)
1	R1270	47	565	45.885	55.85	9.965
2	R290	50	600	45.926	56.48	10.555
3	R1150	58	730	44.843	56.00	11.152
4	R50	50	700	42.519	58.09	15.569
5	R729	53	630	46.144	55.68	9.532

The total spring stiffness thus equals,

$$k_{total} = k_{gas} + k_{spring} \tag{2}$$

and natural frequency of the system is given by,

$$(fn) = \frac{1}{2\pi}\sqrt{\frac{k_{total}}{m}} = \frac{1}{2\pi}\sqrt{\frac{k_{gas} + k_{spring}}{m}} \tag{3}$$

In the above formula, spring stiffness and moving mass would be constant once fixed, however gas spring stiffness varies during the cycle. So, for fixed moving mass and mechanical spring stiffness, natural frequency of the compressor system would also vary. But this variation can be controlled by choosing sufficiently high spring stiffness, so that effect of gas spring stiffness on the natural frequency can be brought under control. With this strategy, an allowance of 9 to 16 Hz had been given on natural frequency of each compressor system in the cascade system. The efficiency/frequency curves are fairly flat at the resonate frequency, it is not necessary to be very precise in choosing the operating frequency, particularly for low pressure ratio operation [6]. Therefore, due to slight off-resonance operation, there would be minimal drop in performance system. However, this is good for the spring structure life to obviate surging. One constraint on choosing of mechanical spring stiffness and moving mass was that they should not cross practical achievable limits in attempt to bring up resonating operation.

After some trials, results came as shown in Table 5. And with these values of mechanical spring stiffness and moving mass, the natural frequency of the linear compressors was observed to be as shown in Figure 3.

OIL-FREE OPERATION

Oil-free operation is possible in a linear compressor and this has many benefits. One of which is an increase in effectiveness of the heat exchangers [5]. Deposition of oil on heat exchanger surfaces interfere with its performance. Also, for low temperature application, oil-free operation is preferable.

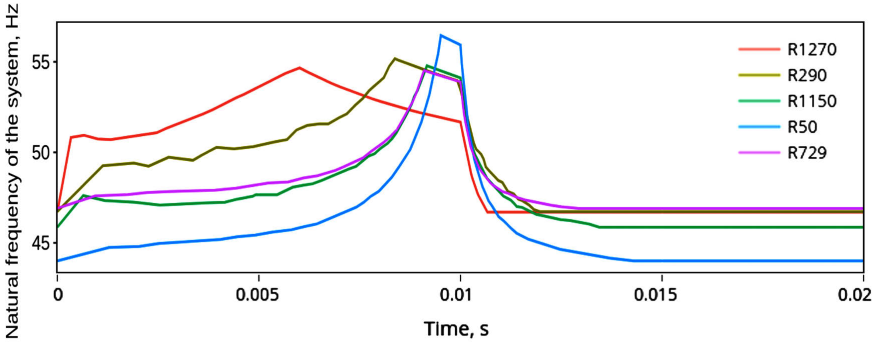

Figure 3. Variation of natural frequency of the system during cycle.

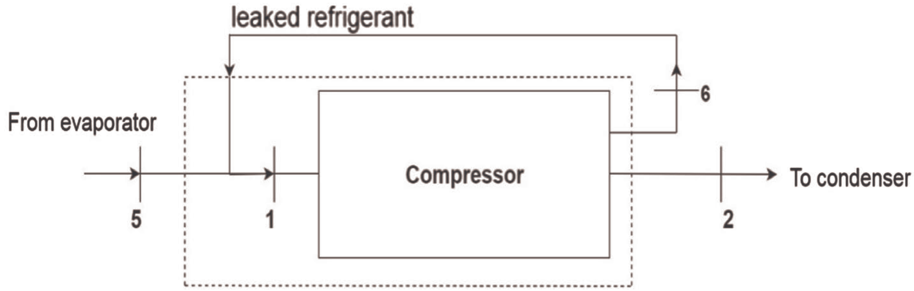

Figure 4. Block diagram of oil-free compressor system.

The oil-free operation can be achieved by providing sufficient clearance between the piston and cylinder and allowing a small amount of compressed refrigerant to flow past the radial clearance. This portion of refrigerant would be very small (< 1%) compared to main portion. It would provide a lubricating action. But strength would not be expected to be as good as that of oil, therefore, proper support for guiding of motion of piston is necessary. For that linear ball bearings or simply flexure bearings can be provided, preferably at two locations. The ideal path of the piston is that its axis coincides with that of cylinder.

The compressed refrigerant flow past the narrow annular space between piston and cylinder would be expanded to the evaporative pressure. This can be considered similar to throttling process where enthalpy is conserved. This expanded refrigerant would then be circulated back to the inlet manifold where it would mix with vapors coming from the evaporator as shown in Figure 4.

The velocity field in this portion is obtained by using Navier-Stoke's equation putting appropriate boundary conditions as follow,

$$V_z = \frac{1}{4\mu}\frac{\partial p}{\partial z}\left[(r^2 - r_2{}^2) + \frac{(r_1{}^2 - r_2{}^2)\ln\left(\frac{r_2}{r}\right)}{\ln n}\right] + \frac{V_p}{\ln n}\ln\left(\frac{r_2}{r}\right) \tag{4}$$

The total discharge through this annular space would be,

$$\dot{Q}_{leak} = \frac{\Pi}{2\mu}\frac{\partial p}{\partial z}(n^2 - 1)r_1{}^4\left[\frac{(n^2-1)}{4}\left(\frac{1}{\ln n} - 1\right) - \frac{1}{2}\right] + 2\Pi\, V_p\, r_1{}^2\left[\left(\frac{(n^2-1)}{4\ln n}\right) - \frac{1}{2}\right] \tag{5}$$

The variation of effective enthalpy leakage with radial clearance ratio n was observed to be as shown in Figure 5. The enthalpy leakage through annular space affects the VCR system performance.

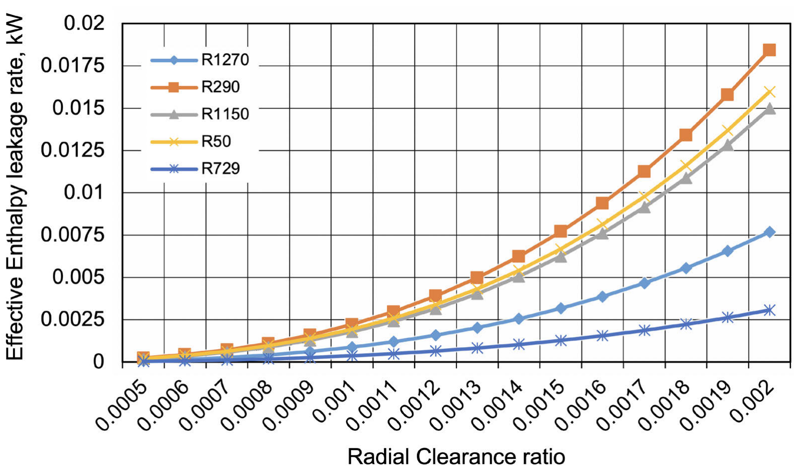

Figure 5. Variation of effective enthalpy leakage rate with radial clearance ratio.

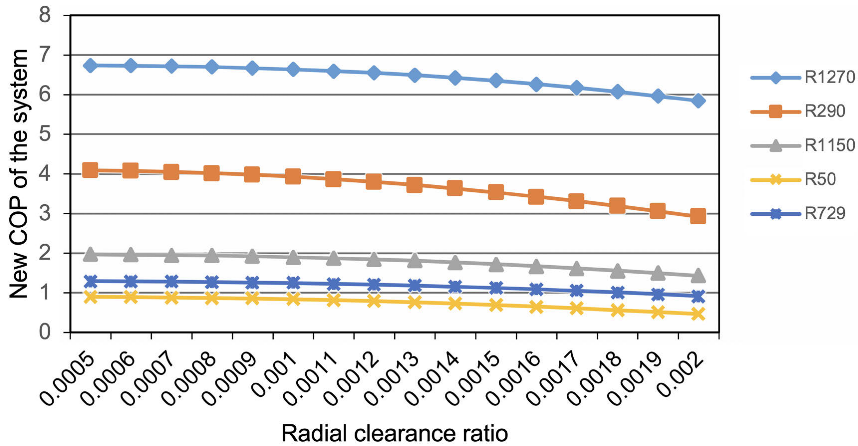

Figure 6. Effect of enthalpy leakage on new COP of the system.

It was observed that new COP of the VCR system with oil-free operation of compressor was,

$$(COP)_{New} = \frac{R.E. - (H_{leak})}{W + (H_{leak})} \tag{6}$$

The new COP of the VCR stages with radial clearance ratios was observed to be as shown in Figure 6. The percentage drop in COP compared to that without leakage would be as shown in Figure 7. The overall COP of the system, if work input is considered as electrical energy would be,

$$\begin{aligned}(COP)_{overall} &= \frac{R.E.}{Work\ input} \\ &= \frac{R.E.}{Work\ done\ on\ gas} \cdot \frac{Work\ done\ on\ gas}{work\ on\ Piston} \cdot \frac{Wrk\ on\ piston}{Electrical\ input\ energy} \\ &= (COP)_{system} \cdot \eta_{mech} \cdot \eta_{motor}\end{aligned} \tag{7}$$

compared to a conventional compressor run by a rotary induction motor. So, to make a VCR system with a linear compressor competitive compared to that with a conventional compressor, its overall COP should be more or at least equal to that of conventional one. The strategy adopted in this was that a drop in the COP of the system is compensated by high electrical motor efficiency of linear compressor. K. Park et al. [4] showed that for the same output of 100 %, a linear motor and conventional rotary induction motor required 104.53% and 116.93% input power respectively.

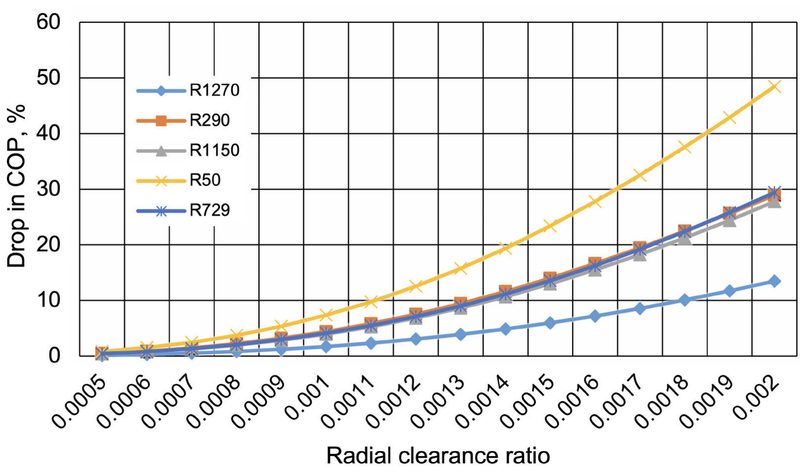

Figure 7. % drop in COP of new oil-less system compared to that with no leakage.

Table 6. Maximum radial clearance for compressors for oil-free operation.

Stage	Refrigerant	Maximum radial clearance ratio	Maximum radial clearance (μm)	% drop in COP
1	R1270	0.0018	13.7	10.070
2	R290	0.0013	17.6	9.410
3	R1150	0.0014	14.4	10.679
4	R50	0.0011	10.0	12.468
5	R729	0.0014	10.0	16.254

With this guess, it was found that 11.5% drop in COP of the system could be tolerated for oil-free operation of the linear compressor. With this, following results (Table 6) have been obtained for maximum radial clearance in a different compressor of cascade system that could be provided so that overall COP can be ensured.

There is usually a minimum 10 μm radial clearances provided for oil-free operation [5]. For the first four stages, it came out to be more than this minimum limit. Therefore, any value between this minimum and maximum limit can suit. For the last stage, the pressure ratio was high and its bore size was also low, so maximum radial clearance came out to be less than the minimum limit 10 μm. It could have been increased to the value 10 μm by increasing the cylinder length, thus reducing pressure gradient, but it would have lead to other difficulties in design. It was a matter to choose between efficiency and practicability. Therefore, a minimum radial clearance of 10 μm can be taken for this stage with chance of slight loss in performance.

CONCLUSION

The aim of this study was to design linear compressors with resonating and oil-free operation, for 5 stage cascade system with special application of air liquefaction of total capacity of 20 W at the temperature of 78.8 K. Normal atmospheric air would be cooled down with a sequence of cascade stages. The first stage of cascade system was chosen such that it could serve dual purpose, as one of the stage in cascade system and individually in air conditioning application. Refrigerant used were R1270, R290, R1150, R50 and finally R729 in a sequence.

For resonating operation, the strategy was to tune the natural frequency of system with frequency of input A.C. (=50 Hz). Gas spring stiffness seemed to vary a lot during the compression cycle, making it difficult to achieve perfect resonance operation throughout the cycle. The variation of this natural frequency had been checked and brought down within some limit by employing suitable high magnitude of mechanical spring stiffness. The allowance of 9-16 Hz had been provided to achieve this with reasonable mechanical spring stiffness and corresponding moving mass. Moving mass would be comprised of mass of piston, armature, flange etc. and one third mass of the resonating springs. In the case of VCR stages of R50 and R729, owing to high pressure ratio, achieving resonance became difficult even with high mechanical spring stiffness and moving mass. For achieving resonating operation with close tolerance, option available is to either increase the mechanical spring stiffness (and corresponding moving mass) or insert one more intermediate stage in between.

Oil-free operation is possible with linear compressors offering its own benefits. A very small amount of compressed refrigerant was purposely allowed to flow past the radial clearance between piston and cylinder. This expanded refrigerant then recirculated back to inlet manifold to get mixed with incoming vapors from evaporator. The VCR system performance is affected by this. For analysis, equation for discharge through this annular space had been derived and with knowledge of variation of density and specific enthalpy, mass leakage flux and enthalpy leakage flux due to refrigerant leakage had been calculated with radial clearance ratio. The COP of VCR system with oil-free operation had been obtained and compared with COP without leakage, percentage drop in COP had been obtained. Owing to high electrical efficiency of linear motor compared to that of a conventional one, some drop in COP could be tolerated. With this, maximum radial clearance that may be provided for making this VCR system with oil-free operation competitive to that of conventional one, had been obtained for first three stage of cascade system. For last two stages of cascade system minimum radial clearance of 10 um had been taken with corresponding further drop in COP due to manufacturing considerations.

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