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ABSTRACT

This investigation studied the influence of cylinder wall ports on the performance of a refrigerant compressor by means of computer simulation and experiment. It proved that wall ports in the cylinder liner are useful in increasing the cooling capacity and COP of a refrigerant compressor when they are correctly designed. To give full play to the charging effect of the wall ports, valve lift and spring characteristics should be adjusted properly in coordination with the wall port design. With the increase of compression ratio, wall ports' benefit becomes still more remarkable. So, provision of wall ports are more desirable in case of compressor working under lower evaporating temperature.

INTRODUCTION

The idea of using proper cylinder wall ports around the cylinder liner at a position near the B.D.C. of the piston stroke for compressor performance improvement was patented in the late 30s [1], but had not been put into practice until 70s when S.A. Parker had made an attempt to test it on an air-conditioning compressor, resulting in a volumetric efficiency increase less than 1% [2]. Later, D.Squarer and others tried by means of compressor computer modeling to evaluate the possible performance improvement of a heat pump compressor through several potential compressor design modifications, including a provision of cylinder wall inlet ports. He concluded that a marked improvement in efficiency was clearly evident only at a temperature below -29°C [3], which has not been ascertained by experiment.

With the purpose of clarifying the actual influence of wall ports on the compressor performance at different test conditions, displaying the changes in cylinder processes and valve behaviour produced by the disposal of wall inlet ports and searching for optimal correlation between port size, location and valve design parameters, computer modeling technique combined with testing on an actual compressor has been used successfully and achieved good results. Theoretical analysis and experiment desmonstrate that a noticeable gain in refrigerating capacity of compressor accompanied with some increase of COP can be expected only when the suction valve design parameters are properly determined to adapt the design of wall ports.

The investigation was conducted with a domestic-made six cylinder compressor for R-12 having a rated capacity of 73.26 Kw under standard condition (evap. temp. -15°C, cond. temp. 30°C, return gas temp. 15°C, subcooling 5°C) with 100 mm bore diameter, 70 mm stroke and 1440 rpm. The compressor valve system is of ring-plate type (Fig. 1) with intake holes uniformly disposed on the periphery around the upper flange of the cylinder liner. The determination of design parameters like valve lift and spring characteristics have been elaborately studied and checked up by experiment so that the compressor was featured with satisfactory characteristics [4].

COMPRESSOR PROCESSES COMPUTER MODELING

The simulation model describing the processes taking place in the cylinder is based on the First Law of thermodynamics applied to well defined control volumes within the compressor. It contains all necessary details to permit true simulation of the internal compressor modification related specifically to wall port and valve design, and is easily manipulatable, rapid and inexpensive to run.

The present model comprises a set of coupled differential and algebraic equations, such as energy conservation equation, mass flow equation, valve dynamic equation and refrigerant equation etc. [5] which were solved digitally by iteration with a step size of one crank angle degree. In the model the gas force coefficient was assumed to be independent of valve lift and the valve flow coefficient was represented by a bi-linear characteristic [6]. The wall port flow coefficient was considered as an empirical constant which had been generally employed in the scavenging port calculation in two-stroke internal combustion engine [7].

Computer modeling was carried out to study the effect of the following design changes:

1) Change of wall inlet port number, size and location.
2) Change of suction valve lift.
3) Change of valve spring characteristic.

Change of wall inlet port size, number and location.

The effect of wall ports communicating with the surrounding suction chamber consists in that they allow an additional flow area for suction vapour when uncovered by the piston top edge and hence, a larger quantity of refrigerant can be drawn into the cylinder if the timing and port size are correctly designed. Otherwise, an adverse effect occurs.

In compressor simulation, two sets of port data were calculated. One was 50 smaller holes and the other 20 larger holes. The 50 holes had a maximun openning area 18% larger than that of the latter. Fig. 2 and 3 give the simulation results which show the cooling capacity QO and the indicated power consumption POWI vs. the location of the wall ports H (distance from the top edge of the piston to the top of inlet port expressed in percentage of the piston stroke, Fig. 1) under standard test condition and low temperature condition (evap. temp. -25°C,cond. temp. 30°C, return gas temp 5°C, subcooling 5°C) respectively with different number of holes.

One can see clearly from the plots that each curve has a maximized capacity at a port location near H= (96-97)% which is almost the same value as indicated in reference [2]. However, it must be noticed that the port location H in reference [2] was defined as the distance from the top of ring at T.D.C. to top of wall ports in percentage of piston stroke. The figures show that the compressor with 20 holes in the cylinder liner produces 1.2% and 1.9% more cold under standard condition and low temperature condition respectively than without wall port provision, which are higher than the 0.86% capacity gain under high temperature condition (evap. temp. 7.2°C,cond. temp. 51.6°C, return gas temp, 18.3°C) [2]. Simulation also shows that the port location is optimal when a period of back flow from cylinder to the suction chamber lasts some (10-15) crank angle degrees before the complete closure of wall ports (see Table 1). This results in a higher cylinder pressure over the suction pressure at the moment when the piston has just fully covered the wall ports. But, moving the wall ports up further toward the T.D.C. as shown in Fig. 2 and 3 causes excessive refrigerant backflow and therefore, capacity reduction.

Fig. 2 and 3 indicate also that cylinder with 50 holes produces more cooling capacity and has a higherCOPI than that with 20 holes.

Change of Suction Valve Lift

The wall ports function cooperatively with the suction valve during the late period of the suction process. Table 1 lists the cylinder charge through wall ports per cycle. In spite of only a few percent of the total cylinder charge it is an important and innegligible part of the charging effect.

In spite of the elaborate valve system design and satisfactory operation of the original compressor, it is apparent that the valve behavior has to be changed with the provision of wall ports. So, there exists the need for better coordination between port and valve design.

We tried to decrease the suction valve lift. This was only because in valve system shown in Fig.1, the size of the cylinder clearance volume is closely related with the suction valve lift. Suction valve lift depression reduces the clearance volume but at the meanwhile it decreases the suction flow area as well. The adverse effect of the latter can be compensated for with the presence of wall ports. Computer simulation results for port location H = 96.5% are plot in Fig. 4. Just as what we have expected with the reduction of suction valve lift beginning from 2 mm, the capacity keeps on increasing until a 1.5 mm lift where a maximum value is reached. Reduction of suction valve lift to 1.5 mm ensures a capacity increase of 1% and 5.5% under standard and low temperature conditions respectively in relation to suction valve lift 2 mm unchanged, i.e. 2.3% and 7.5% in comparison with the case without wall ports (see Table 1). The COPI decreases under standard condition and increases under low temperature condition as the suction valve lift is reduced to 1.5 mm (Fig.4).

It is also shown in Table 1 that the cylinder charge through wall ports has increased from 5.7% to 7.7% of total cylinder charge at standard condition and from 4.9% to 9.1% at low temperature condition.

Change of Valve Spring Characteristics

Three valve systems with three different spring characteristics—two springs with constant stiffness 1.47 and 2.06 KN/m, the third one which we actually tested and had a variable stiffness proportional to the spring deflection—have been simulated. Fig.5 shows one of those simulation results under standard condition with optimal port location H =96.5% and reduced suction valve lift 1.5 mm. It can be seen from the plot that compressor using springs with constant stiffness 1.47 KN/m has the highest output and COPI while springs with constant stiffness 2.06 KN/m give the smallest capacity and middle COPI.

EXPERIMENTAL VERIFICATION OF THE COMPRESSOR MODELING AND RESULTS ANALYSIS

Experimental verification has been carried out on a hot gas bypass loop test stand. During the test period only two of the six cylinders were actually working and the other four cylinders were unloaded by removing away all their valve elements and isolating them from the working ones. That is the reason why we have taken account of compressor power consumption and COP by indicated parameters in our investigation. Cylinder liners only with 20 larger holes located at H = 99%, 97.69, 96.5% and 95.33% respectively were tested under both standard and low temperature conditions. In addition to the original suction valve lift 2 mm, the optimal valve 1.5 mm determined in simulation analysis has also been checked up by experiment while the existing spring with variable stiffness remained unchanged through out the entire test period.

Simultaneous recording of cylinder pressure, suction and discharge pressures, motion of the valve plate and B.D.C. were done in the experiment. For determining the valve lift inductive valve plate displacement transducers have been used. The cylinder pressure transducer, suction and discharge pressure transducer were of strain gauge type. A typical pressure and valve motion vs. crank angle oscilloscope trace is illustrated in Fig.6 which shows fair match of the theoretical and experimental results. The valve motions, however, are less steep than the computed ones. This might be considered as a result of the non-linear spring characteristics.

Actual cooling capacity, POWI and COPI of the compressor with 2 mm suction valve lift at different test conditions and their comparison with simulation results (solid curves) have been plot on relevant diagrams by △ and ⊙ (Fig.4 and 7).

The accuracy of compressor simulation is within acceptable limits. Accuracy for predicting cooling capacity is within 1% over the entire test range. The worst error for predicting POWI is about 4% occuring at low temperature condition.

Analysis on the recorded pressure indicating diagrams of the suction process (Fig. 8) reveals the actual influence of the wall ports and suction valve lift depression on the thermal process occuring in the suction period. Optimal arrangment of the wall ports at H = 96.5% with the original suction valve lift allows more captured cylinder charge described by the higher cylinder pressure at the closing moment of the wall ports and the suction valve lift reduction brings about a noticeable cylinder pressure depression oweing to the reduced suction flow area from the beginning of the suction process until the openning of the wall. Once the ports open, the increased pressure difference between the cylinder and suction chamber pushes the refrigerant rushing into the cylinder and causing a more steep pressure rise. This raises further the cylinder pressure at the closing moment of the ports and therefore, increases the cylinder charge although there exists a longer duration of backflow.

CONCLUSION

The results of this investigation can be summarized as follows.

1) The agreement of the theoretical and experimental results proves the compressor computer model a successful one for wall port design.

2) Cylinder wall ports are useful in increasing the cooling capacity and COP of a refrigerant compressor if their size and location are reasonabley determined.

3) In order to give full play to the charging effect of the cylinder wall ports the valve lift and spring characteristics should be adjusted properly to coordinate the port design.

4) With the increase of compression ratio wall ports' benefit becomes still more remarkable.

5) In our experimental investigation, with 20 holes at H = 96.5% the compressor performance improvement against the unmodified one are listed below.

Running condition	Standard		Low temp.	
Suct. valv. lift mm	2	1.5	2	1.5
Capacity gain %	2	4.5	3	7
COPI increase %	2	0	3	6

6) More effective wall port flow area at an optimal H, if possible, is desirable for better improvement.

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TABLE 1

COMPRESSOR SIMULATION RESULTS
(20 holes, valve springs with variable stiffness, 2 working cylinders)

Running condition	Standard condition				Low temp. condition			
H %	100	96.5	93	96.5	100	96.5	94	96.5
Suct. valve lift, mm	2	2	2	1.5	2	2	2	1.5
Ref. capacity Kw	26.6	27.0	26.8	27.2	15.4	15.7	15.6	16.5
Vol. efficiency	.75	.76	.754	.777	.713	.724	.72	.74
Total cyl. charge $\times 10^{-2}$ Kg/cycle	.132	.134	.133	.137	.085	.087	.086	.092
Wall port charge $\times 10^{-2}$ Kg/cycle	0	.077	.057	.106	0	.043	.07	.084
Indicated power Kw	7.60	7.63	7.50	7.94	5.93	5.99	5.91	6.16
COPI Kw/Kw	3.51	3.54	3.57	3.43	3.07	3.04	3.07	3.12
Suction valve:								
open °	38	38	38	36	48	48	48	47
close °	202	200	197	189	200	195	189	204
Discharge valve:								
open °	305	305	305	305	317	317	317	317
close °	5	5	5	4	5	6	5	6
Max. impact vel. m/s								
Suct. valve:								
toward seat	1.69	1.69	1.64	1.61	1.48	1.57	1.57	1.52
toward stop	3.56	3.99	3.60	2.97	2.97	3.26	3.01	2.51
Disch. valve:								
toward seat	1.35	1.35	1.59	1.38	1.34	1.69	1.53	1.56
toward stop	5.01	5.25	4.43	4.96	4.87	5.20	4.74	4.23
Wall inlet ports								
open angle °		157	147	157		157	149	157
close angle °		203	213	203		203	211	203
back flow °		188-203	180-213	188-203		190-203	183-211	202-203

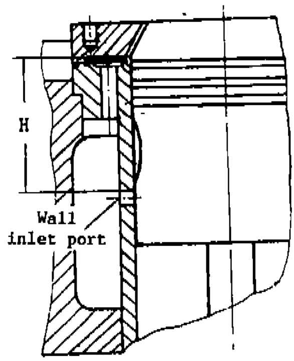

Fig 1. Section view of the suction valve and wall inlet ports of the compressor

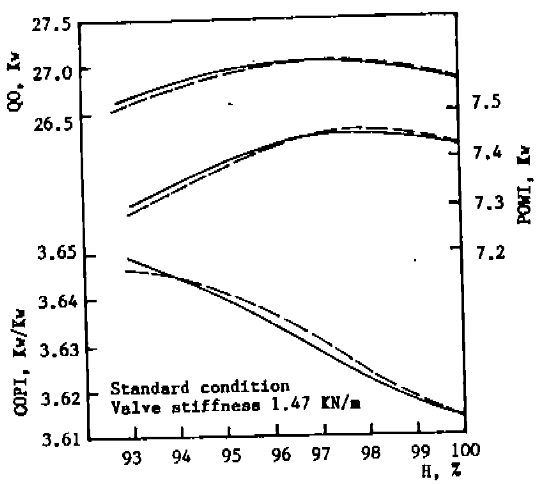

Fig 2. Effect of wallport area and location on QO, POWI and COPI
——— 20 holes ------ 50 holes

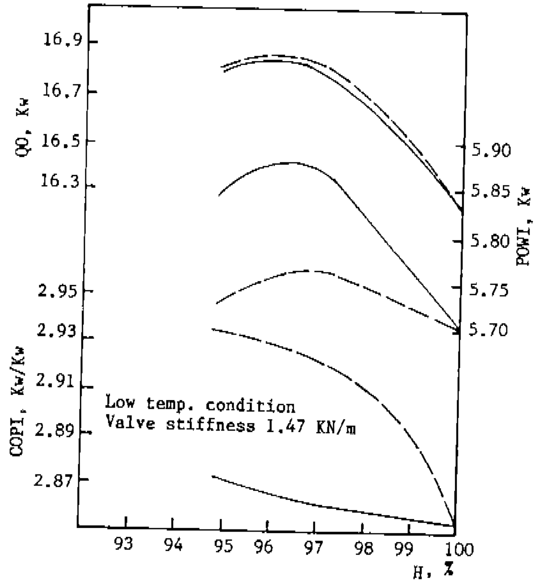

Fig 3. Effect of wall port area and location on QO, POWI and COPI

——— 20 holes   ------ 50 holes

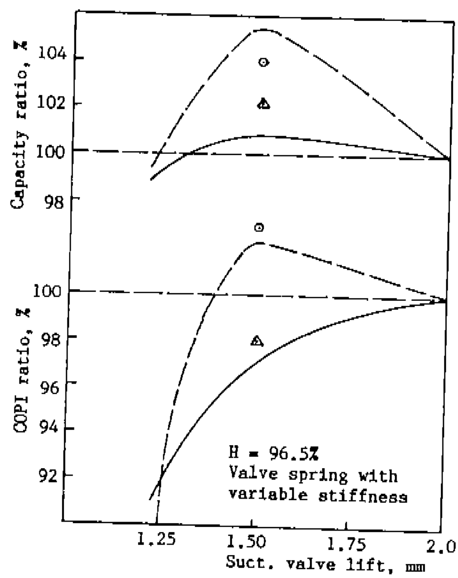

Fig 4. Effect of suction valve lift reduction on capacity ratio and COPI ratio

——— Standard condition ----- Low temp. condition

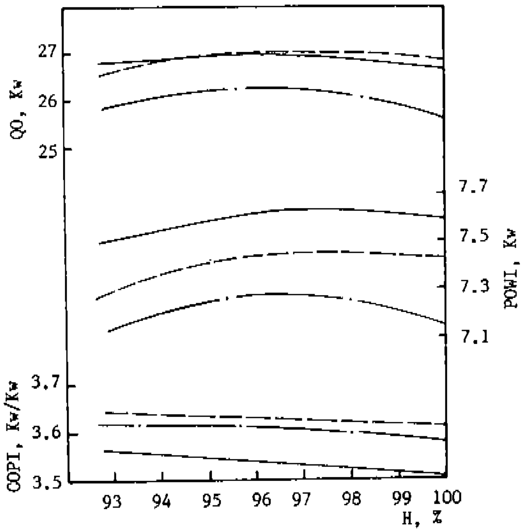

Fig 5. Effect of valve spring characteristics change on QO, POWI and COPI at standard condition

——— variable stiffness - - - - spring constant 1.47 KN/m

—- - — spring contant 2.06 KN/m

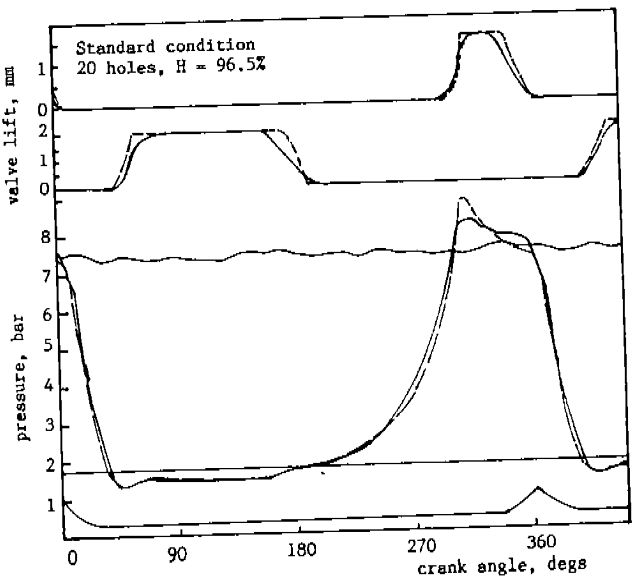

Fig 6. Pressures and valve plate motions vs. crank angle diagrams at standard condition

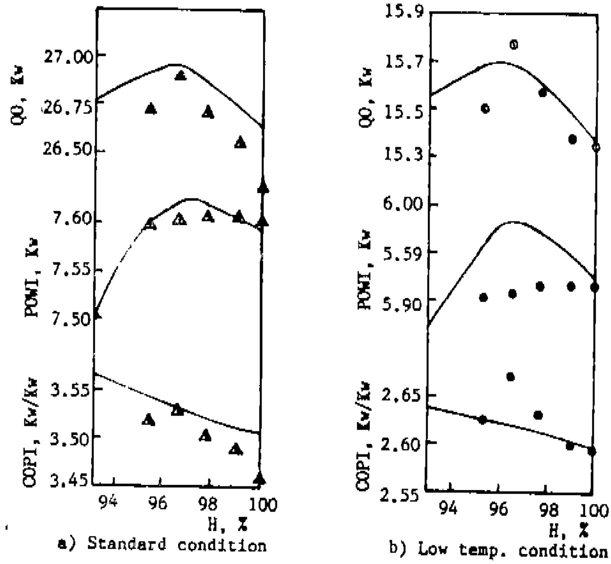

Fig 7. Actual QO, POWI COPI of the compressor plot against the simulation results

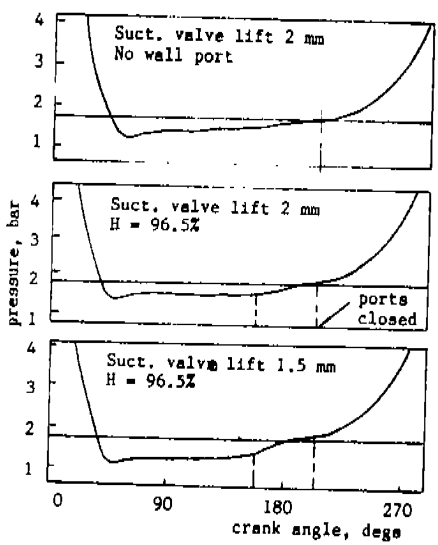

Fig 8. recorded cylinder and suction chamber pressure vs. crank angle diagrams of the suction process at standard condition