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Operation and performance of screw machines with high built-in volume ratio

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Operation and performance of screw machines with high built-in volume ratio

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Abstract. A performance calculation of a screw compressor with increased built-in volume ratio was performed in this paper to establish how increased built-in volume ratio influences its efficiency. It is known that screw compressors have limited built-in ratio which is determined by their standard discharge port size and position. However, if the discharge port is reduced beyond its cusp position, the screw machine built-in volume is increased. In such a case, influence of the oil volume in the air-oil mixture of oil-flooded compressors increases the machine built-in volume further. The performance improvement achieved if the built-in volume ratio is doubled in comparison with the standard port during the machine operation at high pressure ratio of more than 20, is up to 26% for the specific power and adiabatic efficiency. This confirms superiority of the reduced size high pressure port for compressors which operate at high pressure ratio.

1. Background

It is well known that efficiency of screw machines is highly dependent upon the good matching of their built-in volume ratio and volume ratio of the fluid being compressed. This is especially significant for the high fluid pressure ratio which can not be matched by a standard machine built-in volume ratio.

The screw compressor is a positive displacement rotary machine. It consists essentially of a pair of meshing helical lobed rotors, which rotate within a fixed casing that totally encloses them, as shown in Fig. 1. The space between any two successive lobes of each rotor and its surrounding casing forms a separate working chamber of fixed cross sectional area. The length of this chamber varies as rotation proceeds due to displacement of the line of contact between the two rotors. It is a maximum when the entire length between the lobes is unobstructed by meshing contact with the other rotor. It has a minimum value of zero when there is full meshing contact with the second rotor at the end face. The two meshing rotors effectively form a pair of helical gear wheels with their lobes acting as teeth.

As shown in right top side of the figure, gas or vapour enters from the front and on top, through an opening, mainly in the front plane of the casing which forms the low pressure or inlet port. It thus



fills the spaces between the lobes, starting from the ends corresponding to A and C in the lightly shaded area. As may be seen, the trapped volume in each chamber increases as rotation proceeds and the contact line between the rotors recedes. At the point where the maximum volume is filled, the inlet port terminates and rotation proceeds without any further fluid admission in the region corresponding to the darkly shaded area.

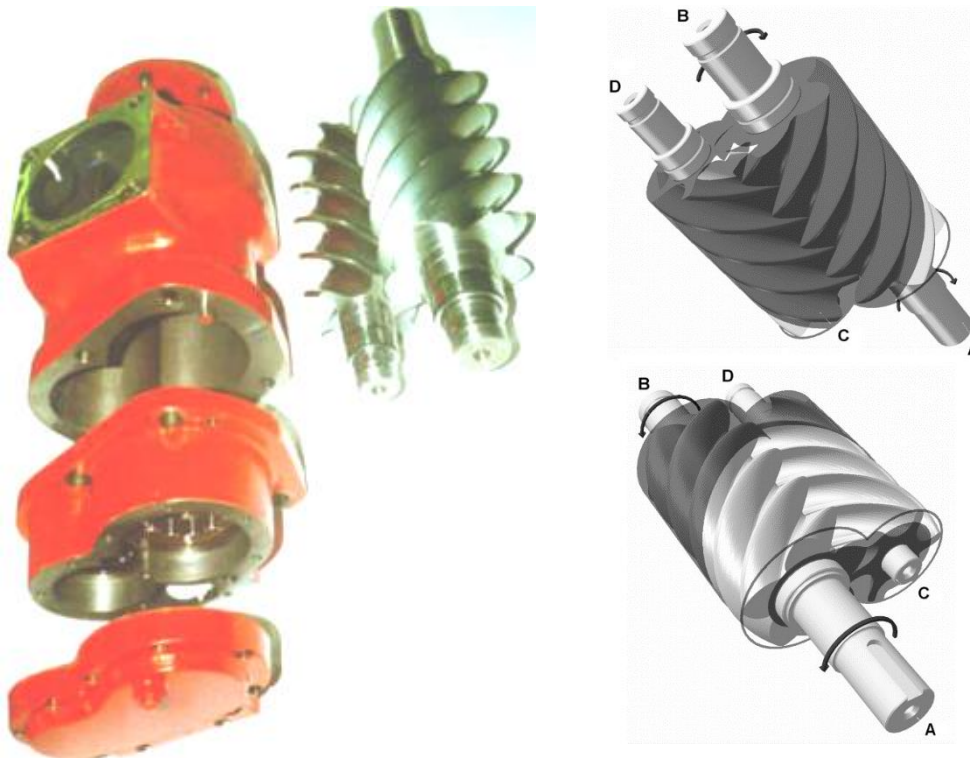


Figure 1. Screw compressor principal mechanical parts

Viewed from the bottom, it may be seen that the darkly shaded area begins, from the end corresponding to A and C, at the point where the male and female rotor lobes start to reengage on the underside. Thus, from that position, further rotation reduces the volume of gas or vapour trapped between the lobes and the casing. This causes the pressure to rise. At the position where the trapped volume is sufficiently reduced to achieve the required pressure rise, the ends of the rotors corresponding to D and B are exposed to an opening on the underside of the casing, which forms the high pressure or discharge port. This corresponds to the lightly shaded area at the rear end. Further rotation reduces the trapped volume causing the fluid to flow out through the high pressure port at approximately constant pressure. This continues until the trapped volume is reduced to virtually zero and all the gas trapped between the lobes at the end of the suction process, is expelled. The process is then repeated for each chamber. Thus there is a succession of suction, compression and discharge processes achieved in each rotation, dependent on the number of lobes in the male and female rotors.

2. Influence of built-in volume ratio upon screw compressor performance

A ratio between the compressor volume at the moment when the suction port is closed and the volume when the discharge port becomes open is built-in volume ratio.

A mismatch of the built-in volume ratio and actual compressor pressure ratio causes either overcompression or undercompression, both of which cause higher compressor indicator losses and consequently higher specific power and lower adiabatic efficiency. Only a matched built-in volume and

pressure ratio give the best compressor performance. This value of the built-in volume ratio is at the same time the optimal one. A p-V diagram of a screw compressor is presented in Figure 2. P_1 and P_2 are compressor suction and discharge pressures, V_2 is the built-in volume matching the given pressure ratio, while V_{22} represents a late discharge port opening and V_{21} a too early discharge port opening, which both increase compressor indicated power if compared with the matching built-in volume ratio.

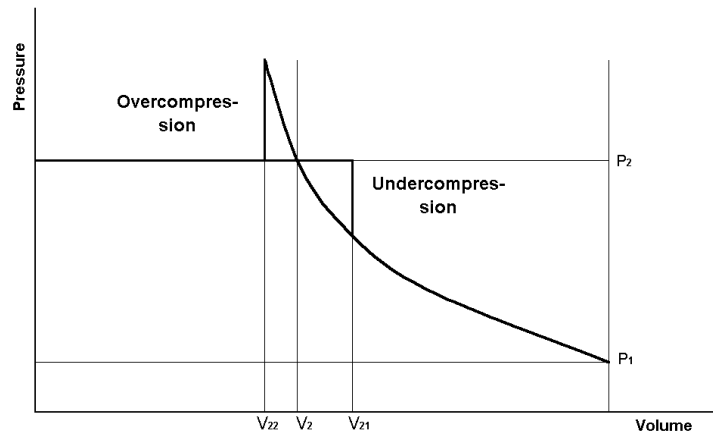


Figure 2. Mismatch in the built-in volume ratio: too large, overcompression, too small, undercompression

A qualitative diagram in Figure 3 represents a variation of indicator losses in function of the compressor built-in volume ratio.

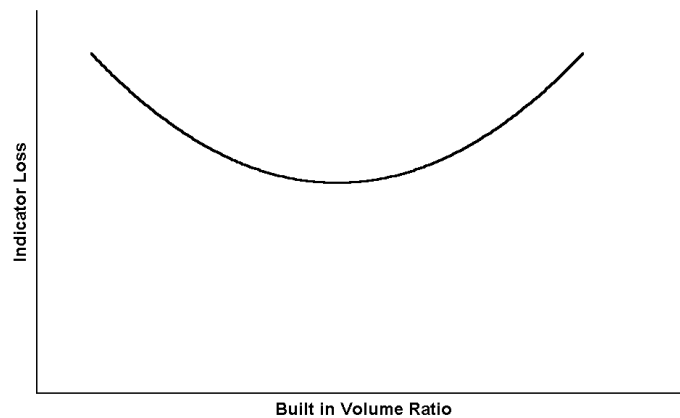


Figure 3. Influence of the compressor built-in volume ratio upon the compressor indicator efficiency

3. Discharge ports for increased built-in volume ratio

There is no doubt that increase of a screw compressor built-in volume ratio will be beneficial for compressors operating under high pressure ratio. A new approach in shaping and sizing of the high pressure port is presented here which doubles the machine built-in volume ratio which improves the machine specific power and adiabatic efficiency for up to 26% for high machine pressure ratios compared with the built-in volume ratio achieved by use of standard high pressure ports. Such discharge ports are obtained by the procedure presented in Figure 4.

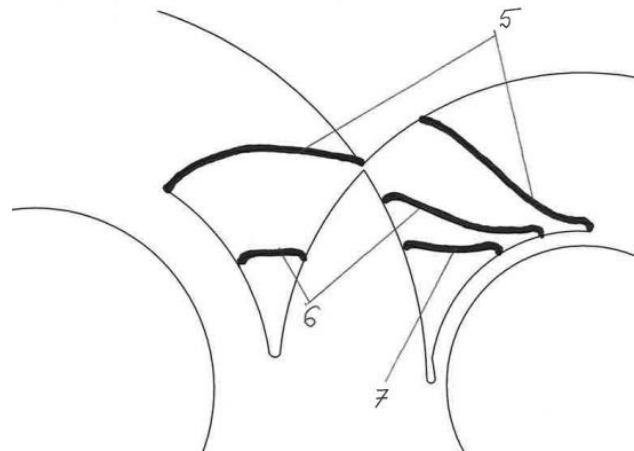


Figure 4. Position of edges coinciding with rotor lobes in high pressure ports: 5, position located at bore cusp which coincides with a limit in common compressor designs achieving a built-in volume ratio $\varepsilon=4$, 6, position for smaller port achieving $\varepsilon=6$, 7 position for even smaller port achieving $\varepsilon=8$

The opening edge of the discharge port is formed by the rotor contours in different positions presented in Figure 4. Label 5 denotes the rotor edge represents the male and female rotor positions coinciding with the housing bore cusp, considered as the smallest possible port in usual compressor design, label 6 denotes the male and gate rotor positions for the reduced area of the high pressure port for a higher built-in volume ratio of 6 and label 7 denotes the main and gate rotor positions for the further reduced area of the high pressure port for the even higher built-in volume ratio 8. The rotor edge position of the male rotor in the last case is below the position where the high pressure can be open and thus this part of the port does not exist.

The positions of the port edges coincide with the lobe edges and are achieved through the simple rotor rotation around their respective centres.

4. Performance calculation for various built-in volume ratios

A compressor with 132 mm male rotor diameter with the four lobes male and five lobes female rotor is used as a vehicle to demonstrate the improvements achieved by increasing of the compressor built-in volume ratio in the high pressure ratio application.

The SCORPATH (Screw Compressor Optimal Rotor Profiles And THERmodynamics), proprietary design program suite was used to design rotor profiles and other geometric parameters and to calculate compressor performance for two different built-in volume ratios, 4 and 8.

The inlet pressure and inlet temperature for both cases was 1 bar and 20°C, while the outlet pressures were 8 bar and 20 for oil-flooded compression of air as the working fluid.

In all cases, the rotors were designed as described in [1], while the compressor performance was calculated according to modelling principles given in [2].

5. Presentation of results and discussion

Standard discharge port, which is the smallest when the port shape is determined by the point where rotor tips meet the housing cusp, is presented in Figure 5 and denoted as 1 together with the working volume in immediate vicinity of the high pressure port, denoted 2. The volume ratio achieved with the standard discharge port is in this case $\varepsilon=v_1/v_2=4$ where V_1 is machine swept volume and V_2 is the working chamber volume at cut off of the discharge port.

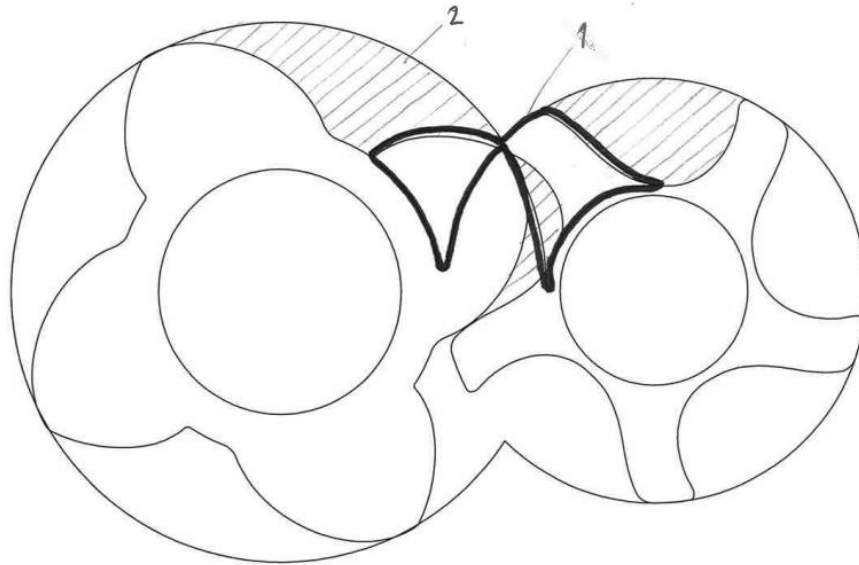


Figure 5. Standard high pressure port, 1 and volume in immediate vicinity of the high pressure port, 2

An extract of the performance calculation with this high volume ratio is presented in Table 1, where w is rotor tip speed, Q is flow through machine, η_v volumetric efficiency P is power and P_{spec} specific power and η_{ad} is adiabatic efficiency.

A reduced size discharge port is presented in Figure 6 and denoted as 3 together with the working volume in immediate vicinity of the high pressure port, denoted 4. The volume ratio achieved with the reduced high pressure port is $\varepsilon = V_1/V_2 = 8$ where V_1 is swept volume at the suction port cut off and V_2 is volume at cut off of the discharge port.

Table 1. Performance calculation for the built-in volume ratio $\varepsilon=4$

		SUCTION PRESSURE							1.00 BAR
		RESERVOIR PRESSURE							8.00 BAR
w m/s	rpm	Q m ³ /min	η_v	P kW	P_{SPEC} kW/m ³ /min	η_{ad}			
20.000	2624.247	4.104	.717	29.499	7.188	.659			
25.000	3280.306	5.450	.761	37.363	6.856	.690			
30.000	3936.365	6.790	.790	45.652	6.723	.704			
35.000	4592.423	8.130	.811	54.390	6.690	.708			
40.000	5248.481	9.471	.827	63.553	6.710	.705			
		SUCTION PRESSURE							1.00 BAR
		RESERVOIR PRESSURE							20.00 BAR
w m/s	rpm	Q m ³ /min	η_v	P kW	P_{SPEC} kW/m ³ /min	η_{ad}			
20.000	2624.247	3.286	.574	59.412	18.078	.437			
25.000	3280.306	4.585	.640	73.729	16.080	.491			
30.000	3936.365	5.884	.685	88.595	15.057	.524			
35.000	4592.423	7.201	.718	104.098	14.457	.546			
40.000	5248.481	8.520	.744	120.237	14.112	.560			

It can be seen from Figures 4 and 5 that a substantial reduction of the high pressure volume was achieved by the discharge port of the reduced size. The ratio between the volumes denoted by 2 and 4 is approximately doubled. Since the built-in volume ratio of the machine which standard high pressure port is presented in Figure 4 is $\varepsilon=4$, the built-in volume ratio of the machine with the reduced size high pressure port is approximately $\varepsilon=8$.

It is necessary to notice that the substantial reduction of the discharge port size of approximately 6 times only doubled the built-in volume ratio from $\varepsilon=4$ to 8. Thus, it is necessary to check air velocities through the discharge port to avoid high flow losses caused by this substantial reduction of the flow cross-section area.

Certainly, since the volume of injected oil compares well to the air volume at high pressures achieved in such a compressor, it will contribute to the additional reduction of the air volume immediately before the discharge port opening and as consequence will additionally increase the compressor built-in volume ratio. This fact has been taken into account and the oil volume influence upon the screw compressor process at high pressure was introduced into the mathematical model.

An extract of the performance calculation with this high volume ratio is presented in Table 2, where w is rotor tip speed, Q is flow through machine, η_v volumetric efficiency, P power and P_{spec} specific power and η_{ad} is adiabatic efficiency.

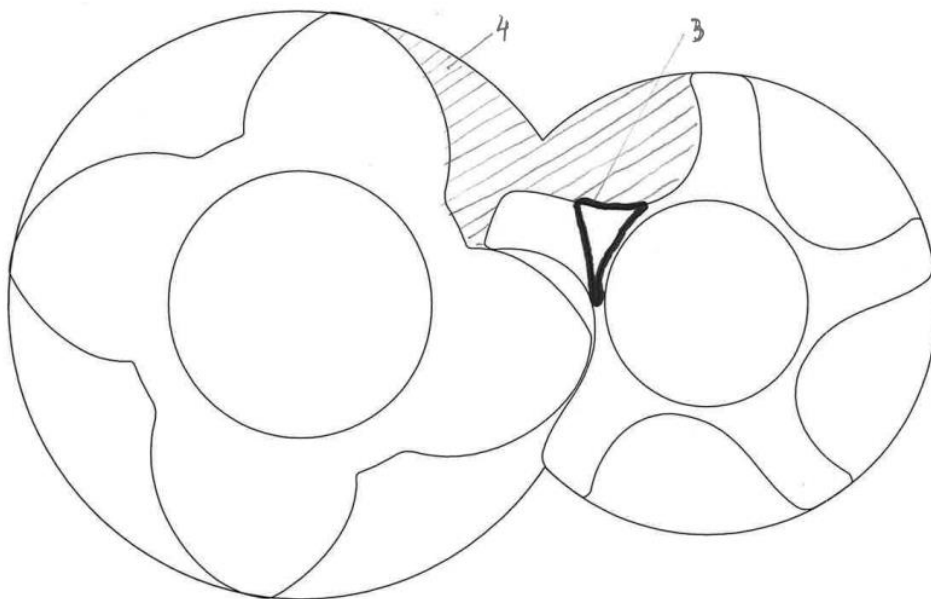


Figure 6. Reduced size high pressure port, 3 and volume in immediate vicinity of the high pressure port, 4

Comparison of flow and specific power of the both, common and high built-in volume screw compressors are presented respectively in Figures 7 and 8 for the 1-8 bar compression and in Figures 9 and 10 for the 1-20 bar compression.

Table 2. Performance calculation with the built-in volume ratio $\epsilon=8$

		SUCTION PRESSURE			1.00 BAR			
		RESERVOIR PRESSURE			8.00 BAR			
w m/s	rpm	Q m ³ /min	η_v	P kW	P _{SPEC} kW/m ³ /min	η_{ad}		
20.000	2624.247	3.976	.694	31.661	7.963	.594		
25.000	3280.306	5.320	.743	40.096	7.537	.628		
30.000	3936.365	6.661	.775	49.081	7.368	.642		
35.000	4592.423	7.999	.798	58.653	7.332	.646		
40.000	5248.481	9.339	.815	68.772	7.364	.643		

		SUCTION PRESSURE			1.00 BAR			
		RESERVOIR PRESSURE			20.00 BAR			
w m/s	rpm	Q m ³ /min	η_v	P kW	P _{SPEC} kW/m ³ /min	η_{ad}		
20.000	2624.247	3.381	.590	48.792	14.432	.547		
25.000	3280.306	4.711	.658	59.630	12.659	.624		
30.000	3936.365	6.036	.703	71.256	11.806	.669		
35.000	4592.423	7.368	.735	83.734	11.364	.695		
40.000	5248.481	8.695	.759	96.906	11.145	.708		

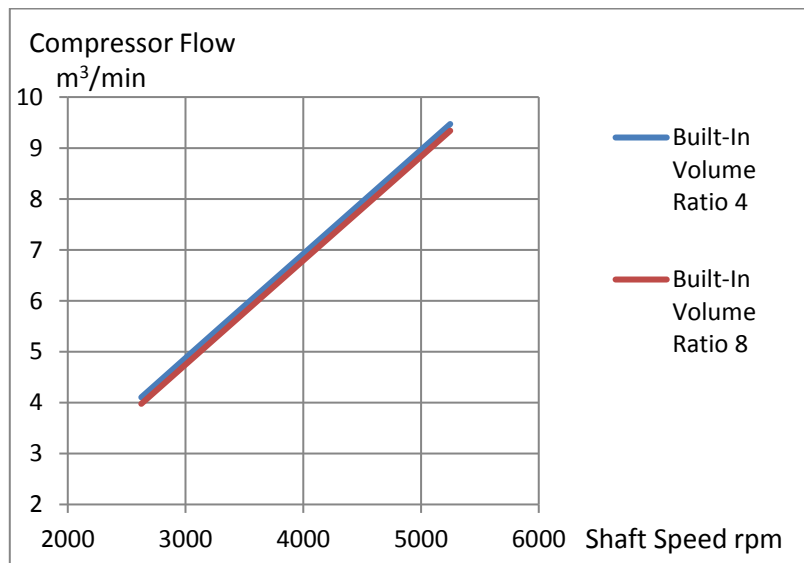


Figure 7. Comparison of flow between compressors of different built-in volume ratio, $\epsilon=4$ and 8 at pressure ratio $p_2/p_1=8$

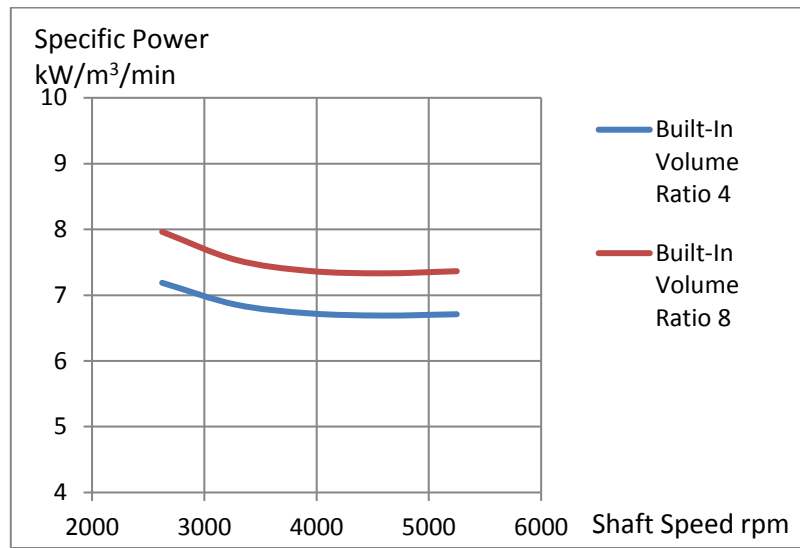


Figure 8. Comparison of specific power between compressors of different built-in volume ratio, $\epsilon=4$ and 8 at pressure ratio $p_2/p_1=8$

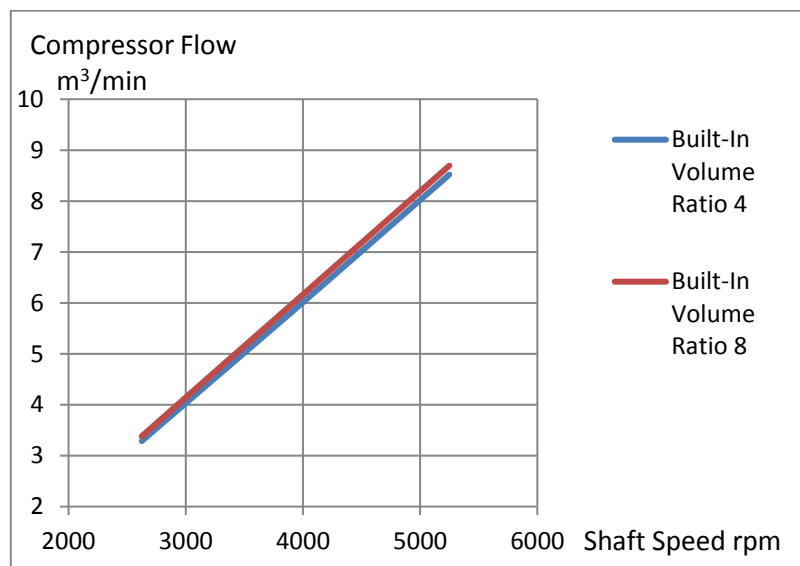


Figure 9. Comparison of flow between compressors of different built-in volume ratio, $\epsilon=4$ and 8 at pressure ratio $p_2/p_1=20$

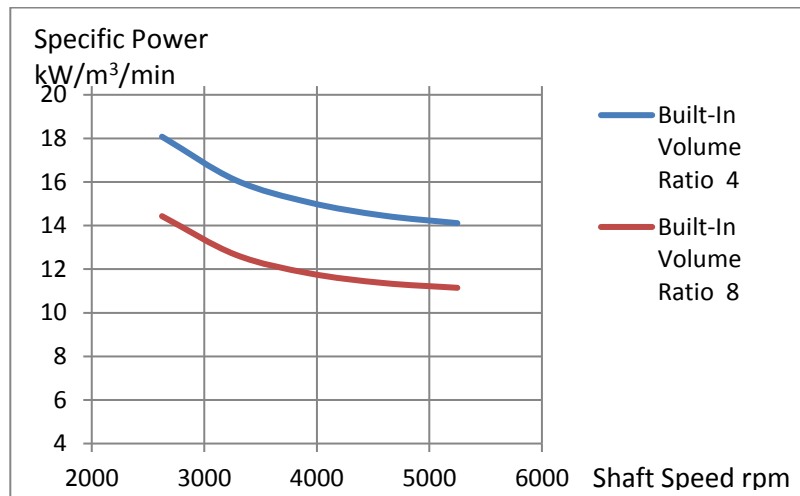


Figure 10. Comparison of specific power between compressors of different built-in volume ratio, $\epsilon=4$ and 8 at pressure ratio $p_2/p_1=20$

As it can be seen from Tables 1 and 2, as well as in Figures 9 and 10, the performance improvement achieved with the discharge port of the reduced size, when the built in volume ratio is doubled in comparison with the standard discharge port during the machine operation at high pressure ratio of $p_2/p_1=20$, where p_1 is low pressure and p_2 is machine high pressure, is up to 26% for the specific power and adiabatic efficiency. This confirms superiority of the reduced size high pressure port at high pressure ratio.

As expected the low pressure ratio compression in the high built-in volume ratio compressor suffered and its performance is lower than performance of the low built-in volume compressor. This can be seen in Figures 7 and 8.

Conclusion

It is confirmed that performance improvement of a screw compressor up to 26% has been achieved if a reduced size discharge port is applied instead of the common port in high pressure ratio application at pressure ratio $p_2/p_1=20$. This increased the machine built-in volume ratio from $\epsilon=4$ to 8.

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- [2] Stosic N, Smith I K, Kovacevic A and Mujic E, 2011, Review of Mathematical Models in Performance Calculation of Screw Compressors, *International Journal of Fluid Machinery and Systems*, DOI: 10.5293/IJFMS.2011.4.2.200 Vol.4, No.2, April-June, 2011, ISSN (Online): 1882-9554