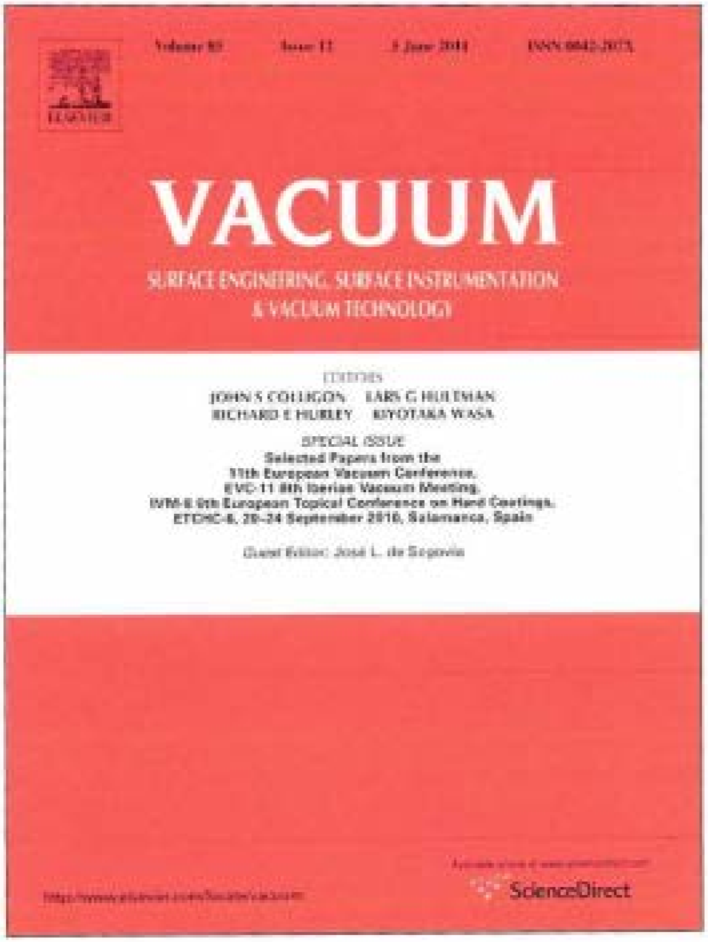
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Optimized rotor pitch distributions for screw spindle vacuum pumps

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A B S T R A C T

Screw spindle vacuum pumps are characterised by a high suction performance and the ability to achieve high pressure ratios. Screw spindle vacuum pumps have varying progressions for the rotor pitch gradient, depending on the manufacturer. From a scientific point of view, the question arises which rotor gradient along the rotors has to be preferred for a particular set of operating conditions with reference to the machine characteristics. To answer this question a simulation of the compression process in the screw spindle vacuum pump is performed. The simulation program is used to calculate an energy-specific optimal rotor pitch applying an evolutionary optimization approach. It turns out that – in contrast to actually available rotor geometries – a continuous increase in rotor pitch from the pressure to the suction side is not ideal. An optimized rotor pitch curve is presented and the underlying physical dependencies are clarified by means of pressure and mass flow diagrams.

1. Introduction

At the chair of fluidics, the design possibilities of screw-type vacuum pumps are surveyed: The research project is aimed at the development of a method for the energetic design of dry-running screw-type vacuum pumps. In practice, the screw-type vacuum pump is characterized, above all, by its high achievable pressure ratio, the high suction capacity and the possibility to be operated in a "dry"-running mode [1] (Table 1).

The first spindle pumps on the market showed a constant pitch and were characterized by an "isochoric" transport phase [2]. From an energetic point of view, however, a compression of the working gas is desirable, so that pumps with internal compression were developed [3].

A possibility to reduce the volume of the working chamber is the modification of the working chamber geometry during the conveying phase. For this purpose, either the pitch along the rotor axis is changed or the ratio of rotor head diameter and root diameter. Both methods result in a variation of the working chamber volume along the rotor axis.

In this regard the question arises of how should the pitch curve of the working chamber volume for a given pressure ratio along the rotor axis be designed to achieve an e.g. energetically favourable operating behaviour. Apart from the pressure ratio of the adjoining working chambers, also the gap mass flow is important for the design of the pitch [4].

2. Multi-criteria optimization

Within the context of this article, the influence of the pitch curve on the energetic parameters of screw spindle vacuum pumps is studied. Studies on the influence of other geometry parameters, as for example of the head diameter, are to be found in the publications [5–7].

The free parameter for optimization is the pitch or the working chamber volume, C, of the individual rotor stages, i. The working chamber volume of the individual stages is determined by the pitch of the stages based on the constant scoop area and constant design volume (sum of all chamber volumes). Accordingly, the rotor models only differ in the allocation of the chamber volume to the individual stages. But due to the high number of working chambers, a comprehensive variation of the individual pitch elements and the ensuing calculation of the operating parameters are highly time-consuming, since the number of models increases exponentially with the number of stages. However, in order to enable the assessment of the effects of the pitch curves with the highest possible resolution of the segment size, an evolutionary model generation approach was selected.

2.1. Evolutionary model generation

For evolutionary optimization [8], the pitch curve for a defined number of rotor models (start population) is initially determined by

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Table 1
List of symbols.

Symbol	Meaning	Dimension
c_p	Specific heat capacity	$J\ kg^{-1}\ K^{-1}$
$\dot{m}$	Mass flow	$Kg\ s^{-1}$
N	Number of stages	–
n	Rotary speed	min^{-1}
p	Pressure	Pa
P	Internal power	W
R	Specific gas constant	$J\ kg^{-1}\ K^{-1}$
s	Pitch	m
T	Temperature	K
$\dot{V}$	Suction volume flow rate	$m^3\ s^{-1}$
v_i	Volume ratio	–
w	Specific internal work	$J\ kg^{-1}$
Π	Pressure ratio	–

means of random numbers. This is followed by the determination of the suction capacity and the internal compression power using the iterative calculation algorithm described in chapter 2.2 (fitness function). The selection of the population according to the optimization targets occurs subsequently to the calculation of the machine characteristics. From the selected population, the new population is generated by

1. crossover of the models (offspring),
2. mutation of individual pitches based on random numbers (mutation),
3. new models generated by random numbers (immigrants).

The evolutionary methods described provide results after approx. 100 evolution steps both in case of one-dimensional optimization (for example, regarding the specific work) and in combination with the two-dimensional Pareto-front method [8].

2.2. Calculation of machine characteristics (fitness function)

The rotor of a screw spindle for the application as vacuum pump is characterized by a high wrap angle. The high number of working chambers along the rotor axis leads to the assumption that the pressure distribution within the screw-type vacuum pump can be calculated using the conservation of mass principles and thus enables both information about the attainable final pressure and the suction capacity at determined suction pressures.

For the calculation of chamber pressures and mass flows, the closed working chambers of the screw spindle are connected by pump stages. For each pump stage an isotropic compression together with a total re-cooling to T_{LP} inside each chamber is assumed. For details of the model generation and the calculations principles see Ref. [9].

Table 2
Parameters for the calculation of an 8-stage screw-type vacuum pump.

Gas constant R	$287\ J\ kg^{-1}\ K^{-1}$
Specific heat capacity c_p	$1005\ J\ kg^{-1}\ K^{-1}$
Gas temperature $T_{LP} = T_{HP}$	293 K
Total volume of working chambers V_{tot}	$0.0031\ m^3$
Discharge pressure p_{HP}	10^5 Pa (1 bar)
Suction pressure p_{LP}	100 Pa (1 mbar)
Rotational speed n	$5000\ min^{-1}$
Number of stages N	8
Minimal pitch s_{min}	20 mm

3. Optimized pitch curve of an eight-stage screw spindle vacuum pump

The calculation of the optimized pitch curve is based on the boundary conditions set out in Table 2. These parameters will be taken into account by the optimization algorithm.

Fig. 1 shows the result of a Pareto-optimization for an eight-stage screw spindle vacuum pump. The result is a closed Pareto-front plotted against the reciprocal value of the discharge mass flow $\dot{m}_{HD}$ and the internal compression power P_{tot}. All models represent an optimum point for themselves resulting from the combination of discharge mass flow $\dot{m}_{HD}$ and the internal power P_{tot}.

A comparison of the discharged mass flow $\dot{m}_{HD}$, the internal compression power P_{tot} and the specific total work w_{tot} of the three selected models with an isochoric rotor is given in Table 3.

The comparison between model A and the isochoric rotor shows a significant increase in the discharged mass flow $\dot{m}_{HP}$ at almost identical internal power P_{tot}. Due to the increase in the discharge mass flow $\dot{m}_{HP}$, the specific power consumption w_{tot} is reduced by the factor two. Compared to model A models B and C show a decrease in

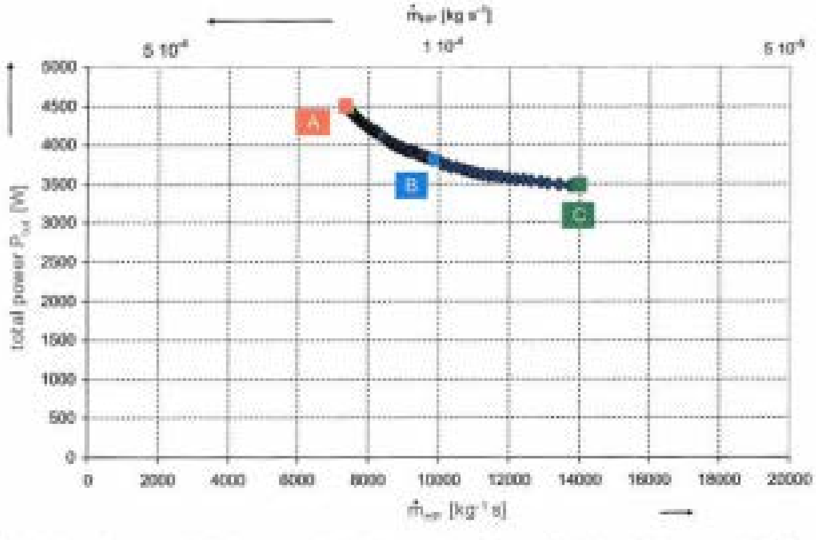

Fig. 1. Optimized total internal power and discharge mass flow shown as Pareto-front after 100 evolution steps with three selected models.

Table 3
Comparison of the integral values for different optimized pitch curves for boundary conditions according to Table 2.

Spindle type	P_{int} [W]	$\dot{m}_{WF}$[kg s^{-1}]	w_{int} [J kg^{-1}]	$\dot{V}_{LP}$ [m^3 s^{-1}]
Constant pitch $s = 30$ mm ($v_{i,tot} = 1$)	4752	0.73×10^{-4}	6.5×10^7	6.1×10^{-2}
Optimized pitch A ($v_{i,tot} = 2.3$)	4486.8	1.36×10^{-4}	3.3×10^7	11×10^{-2}
Optimized pitch B ($v_{i,tot} = 2.9$)	3754.8	0.99×10^{-4}	3.8×10^7	8.3×10^{-2}
Optimized pitch C ($v_{i,tot} = 4.9$)	3487.4	0.71×10^{-4}	4.9×10^7	6.0×10^{-2}

the internal power and the discharge mass flow. Model C, in comparison to the other models, shows the highest specific power and is to be assessed as energetically unfavourable. In practice, however, also the rotor geometry of type C can be interesting as well, since due to the low absolute power consumption it is possible to reduce the thermal stress of the machine.

The comparison of the pitch curves and thus of the volume curves along the stages is illustrated in Fig. 2. An analysis of the calculated stage pressure ratios Π_i, as well as the stage mass flows $\dot{m}_{C,i}$ for the stages $i \in [1, 8]$ shall illustrate the impact of the pitch curve on the thermodynamic properties of the models, Fig. 3.

The optimized models show a maximum stage pressure ratio at the suction side. By comparison, the isochoric rotor model shows the maximum stage pressure ratio Π_i in the medium rotor section. The stage pressure ratios Π_i of the optimized rotors are continuously reduced from the suction to the pressure side. Only above the eighth rotor stage, a slight increase in the pressure stage ratio Π_8 of the optimized rotors is to be noticed. Regarding the optimized rotors, the discharge mass flows $\dot{m}_{C,i}$ of the individual stages increase continuously from the suction to the pressure side. Model C shows the highest discharge mass flows $\dot{m}_{C,i}$ up to the seventh stage, whereas the isochoric reference rotor shows the lowest discharge mass flow $\dot{m}_{C,i}$ up to the seventh stage.

The reasons for this are the low pressure ratios of the isochoric rotor model at the suction side and therefore corresponding low

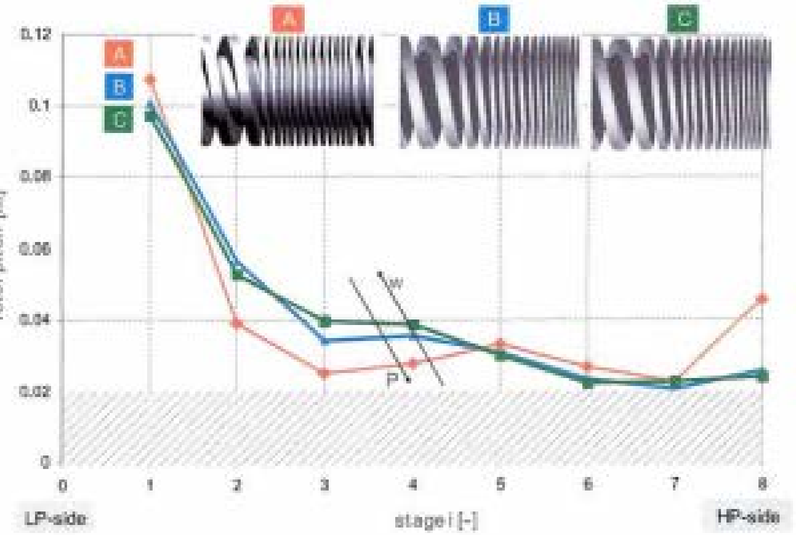

Fig. 2. Comparison of pitch curves for three optimized geometries selected from the Pareto-front, see Fig. 1.

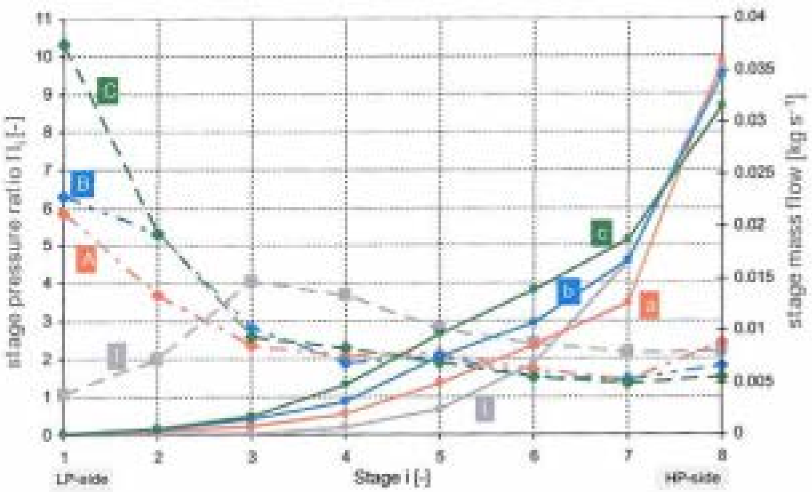

Fig. 3. Stage pressure ratios Πi and mass flows $\dot{m}_{C,i}$ along the rotor stages for: A: Stage pressure ratios model A; B: Stage pressure ratios model B; C: Stage pressure ratios model C; I: Stage pressure ratios isochoric rotor; a: Mass flows model A; b: Mass flows model B; c: Mass flows model C; i: Mass flows isochoric rotor.

gap mass flows $\dot{m}_{G,i}$. The discharge mass flow in the last stage $\dot{m}_{C,8}$ at the pressure side reaches a value of approx. 0.035 kg/s. In relation to a medium discharge mass flow of the screw spindle vacuum pump set out in Table 3, a value of $\frac{\dot{m}_{C,8}}{\dot{m}_{HP}} = 350$ results. This high ratio shows that the working chamber near the discharge side is mainly filled by returning gap mass flows, $\dot{m}_{G,i}$.

The optimization of the specific total work w_{tot} and the total internal power P_{tot} of the rotor results from the opposite development of the stage pressure ratio Π_i and the discharge mass flow $\dot{m}_{C,i}$ of the stages. Decisive for the determination of the absolute power P_{tot} of the models is the product of the discharge mass flow $\dot{m}_{C,i}$ of the stage and the internal work w_i of the stage resulting from the stage pressure ratio Π_i. The total pressure ratio, $\Pi_{tot} = \Pi_1 \cdot \Pi_2 \cdots \Pi_8 = \frac{p_{HP}}{p_{LP}}$, predetermined for the optimization, can be distributed to the individual stages indirectly by their pitch. The gap mass flow and thus also the discharge mass flow of the stages depend on the stage pressure ratio. Further influencing factors for the gap mass flow are the gap inlet pressure as well as the gap area, G, which change according to the geometry of the stages.

It is the objective of the energetic optimization to design the distribution of the total pressure ratio Π_{tot} to the stages and the multiplication by the discharged stage mass flow $\dot{m}_{C,i}$ in a way that the total amount of the absolute powers P_i of all stages is minimized. The optimized rotor models fulfil these requirements due to a shift of the maximum stage pressure ratio Π_i to the suction side. Thus, the specific internal work w_i reaches maximum values at the suction side. The discharge mass flow $\dot{m}_{C,i}$ in the stages adjacent to the suction side increases due to higher gas density, but remains still low compared to the pressure side. Due to the shift of the high stage pressure ratios Π_i to the suction side, both the specific internal work w_i, and the gap mass flows $\dot{m}_{G,i}$ decrease at the pressure side. Both influences reduce the internal power P_i in the stages at the pressure side.

4. Summary and outlook

A method for the energetic optimization of a screw spindle vacuum pump is presented which, based on an evolutionary calculation algorithm, accordingly changes the pitch of a rotor at constant design volume.

The calculation results for an optimized eight-stage screw spindle vacuum pump are presented by means of a Pareto-front and individual models are compared to an isochoric rotor.

The comparison to the isochoric rotor shows an increase in the discharge mass flow and a reduction of the internal compression work in the optimized models. Finally, the pitch curves of the optimized rotor models are discussed and the stage pressure ratios as well as the internal specific work per stage are compared to the isochoric rotor model.

Acknowledgement

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