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DEVELOPMENT AND OPTIMIZATION OF SCREW ENGINE ROTOR PAIRS ON THE BASIS OF COMPUTER MODELLING

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ABSTRACT

The paper presents some recent results in the development of new lobe profiles for the main and gate rotors of a screw compressor, based on the computer modelling of the compression process. The new profile, featured by slenderer gate lobes, as compared with standard asymmetric SRM profiles, yields larger cross-sectional flow areas, enabling higher delivery rates for the same tip speeds. The profile has also shorter sealing lines resulting in lower leakages. The paper outlines the adopted rationale and method of modelling, compares the shapes of new and conventional profiles and illustrates potential improvements achieved with the new design in application to dry and oil-flooded air compressors as well as to refrigerating screw compressors.

INTRODUCTION

A twin screw engine operates most efficiently when the entire compression or expansion process is confined within the engine built-in volume ratio. However, even if the optimum built-in ratio is maintained, the efficiency depends on the screw configuration and the shape of the rotor lobe profiles. For example, when operating in the compressor mode, the circular symmetric 4/6 profiles yield the maximum adiabatic efficiencies of only about 60%, whereas the asymmetric SRM "C" 4/6 profiles reach more than 70%, and new 5/6 profiles even few percents more.

The SRM "C" profile emerged as the most successful twin screw profile and dominated the world market over the past few decades. In spite of a remarkable success in a variety of applications, the experience over the past years indicated that further improvements are possible, and the search for new profiles has been intensified. Already some manufacturers have substituted the "C" design with new, more efficient profiles. A lack of well defined criteria, as well as a diversity of requirements, which depend on specific application (compressor versus expander, dry versus oil-flooded engine, refrigerants versus common gases, etc.) indicate, however, that a universal screw configuration and lobe profile, which will satisfy all different requirements, may not exist. This emphasizes a need for an engineering tool by which an optimum configuration can be found for each specific application. Unlike in the early days, when the design was based more on the designers experience and intuition, and on laborious trial-and-error tests, the use of mathematical modelling [1,5,6] and computer optimization offers nowadays good prospects for a detailed exploration of the effects of various design parameters and of the engine behavior at different operating conditions [7]. Major problem is still the validation of the model and its ability to mimic the real physical process in sufficient details [8].

This paper discusses in brief some issues concerning the computer-aided generation of the profile geometry, simulation of the thermodynamic and flow processes during the compression (or expansion, as the case may be) and illustrates the model application for the optimization of the screw geometry [2]. A comparison of the two conventional and a new profile shape and their performances, obtained by the simulation package SCORPATH will also be presented.
The computational procedure is split into three phases. The pre-processor generates the lobe profiles and the complete screw rotor pairs from the algebraically or otherwise pre-specified profile curves, or it uses a set of pre-generated points to give a volume function, incorporating all interlobe overlapping and blow hole areas. The main program computes the thermodynamics for the elementary volume for the considered time instant or rotating angle. The integration of the mass and energy equations yields the variation of the pressure and temperature of the gas in function of the rotating angle, from which the integral compressor characteristics are evaluated [6]. The post-processing program supplies the results in graphical form, together with the plot of the rotor pairs at a desired sequence of rotating angles.

Fig 1 Symmetric Circular Profile

Fig 4 'N' 3-5 Profile

Fig 2 SRM 'C' Profile

Fig 5 Optimized 'N' 5-6 Profile

Fig 3 Optimized 'N' 4-5 Profile

Fig 5 Optimized 'N' 5-7 Profile
The computational model employed in the present work consists of a set of conservation equations for mass and energy in differential form which describe the thermodynamic and flow process in a screw engine in terms of the rotational angle (or time). The instantaneous operating volume and its change with the rotation angle are defined by the differential kinematic relationships in a general form applicable to any prespecified screw geometry. The model also incorporates empirical relationships which account for most of the effects encountered in a real engine, such as real properties of the gas-liquid mixture [4], heat transfer between the fluid and compressor rotors and casing, leakages, oil (or other fluid) injection [9]. Prior to the application for the optimization purposes, the model was verified by comparison of computed results with laboratory measurements of the cyclic variation of all important local and bulk properties in a screw compressor [3,10].

THE PRE-PROCESSING PHASE: PROFILE GENERATION

In order to provide a basis for reliable comparison of the new profile (hereafter denoted as "N" profile) and to illustrate the model applicability to the profile optimization, the circular symmetric shape, denoted as "A" profile and the SRM "C" profile were also analyzed in parallel and the results of the analysis discussed. Our attention was focused on examining the influence of the shape of the lobe profiles. For that reason, all geometrical parameters except the profile shape were kept constant. In the considered examples the rotor axes distance was assumed 80 mm, the radial clearance (between the rotors themselves and between the rotor tips and housing was 0.2 mm and the front clearance (between the rotor and discharge port section) 0.1 mm. The built-in volume ratio was constant for all profiles analyzed, but according to the type of the compressor, varied from 2 for the dry compressor, up to 5 for the refrigeration compressor. A 4-6 screw pair (four lobes in the main and 6 lobes in the gate rotor) was adopted as the basic configuration, but in addition to this, also 3-5, 5-6 and 5-7 configurations were considered. All configurations mentioned were examined for three different applications: dry and oil flooded air compressors and a refrigeration compressor. Due to the inequality of outer diameters of the main and gate rotors, different rotating speeds were achieved for each configuration for a specific tip speed. Figs 1 - 6 summarize the geometry of all profiles considered.

The computed rotor shapes and the intermeshing of the rotor pairs were visualized by means of a graphic post processor, built in the computer code. Figures 1 to 6 illustrate the considered configurations for different duties. A comparison between the three profiles, the symmetric circular ("A"), the asymmetric SRM ("C") and the new ("N") profile - all in 4-6 arrangement - are presented in Figures 1, 2, and 3 respectively showing clearly notable differences in profile shapes, and, consequently, in the interlobe areas. An experienced profile designer can clearly note the differences between the "N" and "C" profiles. In the "N" profile the forward circle is positioned on the main rotor and excentered both in the radial and circumferential direction as compared with the gate rotor circle in the "C" profile. The backward tip side was profiled utilizing the conventional Lysholm cycloide pair, considered as the well proven an widely accepted method.

Further illustration of the "N" design are supplied in Figures 4 to 6, where the lobe profiles are utilized in the 3-5, 5-6 and 5-7 configuration. It should be noted that all rotors considered were obtained automatically from the computer code by simply specifying the number of lobes in the main and gate rotors, and the lobe curves in a general form.

COMPARISON OF PROFILE PERFORMANCES

The thermodynamic performances of the considered three profiles shapes, A, C and N, for the same numbers of lobes in the main and gate rotor pairs, 4-6, applied to three different types of compressors, a dry, oil-flooded and a refrigerating ones, are illustrated in Figures 7 to 12 in form of the pressure-angle (p-alpha) and temperature-angle (T-alpha) diagrams. For each of the three compressor types, an additional (but different) configuration is also shown. The analysis of the compressor performances for different lobe configurations was performed by maintaining the same operating conditions for all cases, defined by specifying the suction pressure and temperature, and the discharge pressure, as summarized in Figs 7 to 12. The differences between the performances of various profiles are not.
spectacular, as expected, but it is visible that consistent improvements in all respects (lower temperature, lower work requirement) have been achieved, first by the SRM "C" profile as compared with the symmetric "A" design, and later by the new "N" profile, in comparison with both conventional ones. These improvements are further
illustrated in Figures 13, 14 and 15, showing the comparison of the modelled adiabatic efficiencies for the three compressor types considered. These figures indicate that an efficiency improvement has been achieved for the same 4-6 configuration simply by introducing the new lobe shape, but also by changing the number of lobes. For the particular cases here considered it appeared that the optimum for the dry air compressor was the 3-5 configuration, while for the oil-flooded air compressor the optimum is 5-6. Still different, 5-7 configuration enabled the best performances of the refrigerating compressor.

CONCLUSIONS

A lobe shape optimization, performed by means of the computational package SCORPATH, illustrates that improvements in the screw compressor performances are possible as compared with circular symmetric and SRM "C" profiles for the same rotor configurations. A new profile, designed by authors, showed visible improvements in all aspects. Moreover, the optimization showed that for different applications, i.e. compressor types, different configurations (or shapes) may be required to achieve optimum performances. This finding substantiates the use of a general computer code, as a useful tool in the process of development of new, or improvement of existing compressor designs.

The new "N" profile in 4-6 configuration ensures a higher delivery and a better efficiency for the same tip speed, as compared with the circular symmetric or SRM "C" profiles in all applications considered. Still better performances are achieved with other configurations, 3-5, 5-6 and 5-7, but the effects vary from case to case and depend on the compressor type and duty. The final choice will, of course, depend on other factors and a comprehensive cost effectiveness analysis.
Acknowledgement

The initial version of the SCORPATH package (Screw Compressor Optimal Rotor Profiles And THERmodynamic) was developed by the authors at the Mechanical Engineering Department of the University of Sarajevo and it was extensively used for the development and design of a new screw compressor family in the 'Energoinvest' Compressor Factory. The project was also sponsored in part by the Research Fund of Bosnia and Herzegovina.

Literature


