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Rotor Profile Design for Twin Screw Compressor

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Abstract-Increasing demands for efficient screw compressors requires economic and high efficiency rotor designs of screw compressor.Inorder to design a effective rotor rack has to be generated effectively. Numerical and equation adopted in this paper leads to design a effective rotor profile. The solution obtained depends only on the parameters of the rack, pitch, addendum height, dedendum height, rack co ordinates and meshing condition. A suitable procedure for optimization of the screw compressor shape, size, and dimension is described here, which results in the most appropriate design. Compressors thus designed achieve higher delivery rates and better efficiencies than those using traditional approaches.

Index Terms—Rotor, Rack generation, Screw compressor, Tooth profile

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1 INTRODUCTION

Screw compressor rotors of various profiles can be today conveniently manufactured with small clearances at an economic cost. In a while Litvin [1] generated screw compressor rotors and their tools. Shortly Litvin and Feng [2] used singularity and tooth contact analysis (TCA) to investigate the influence of misalignment on the backlash between the surfaces. Soon after in the year 1987, Rinder [4] proposed a rack-generated rotor profile based on gearing theory. Later Stosic [5] proposed a pair of rack-generated rotors inorder to fill the large gap of years.They stated that high-pressure side of the rack is generated by means of a rotor conjugate action that undercuts an appropriate curve on the rack.

Over the year of 2003 N. Stosic et al., developed a new concept for optimizing screw compressor. They established suitable procedure for optimisation of the screw compressor shape, size, dimension and operating parameters which results in the most appropriate design for a compressor.It is based on a rack generation algorithm for rotor profile combined with a numerical model of the compressor fluid flow and thermodynamic processes. They have shown that the optimum rotor profile, compressor speed, oil flow rate and temperature may significantly di□er when compressing di□erent gases or vapours or if working at the oil-free or oil-flooded mode of operation [7].In a while D. Zaytsev et al.,in the year of 2005 adopteed new techniques for generating rotor profile for screw compreesor.

They adopted a method for generation of the profile of twin screw compressor rotors from a meshing line which was analytically derived. The solution obtained mainly depends only on the distance between the rotor axis, the lobe number of both rotors and the given meshing line description. They carried out this method to obtain optimal profile design[11].Soon after Yu-Ren Wu et al.,in year of 2009 flourished a new concept for generating rotor profile.They replaced the implicit form with explicit equations of the rack with two specific normal-equidistant trochoids for rack-generated rotor profiles.

They implemented parameters which are designed on the rack in order to more instinctively and flexibly adjust each compound curveThey established The parametric study and non-undercut limits are presented for the rotor profile optimization with SUMT (sequential unconstrained minimization technique) method. The performance of the twin-screw compressor depends mainly on the tooth profile of mating rotors[10].

2 MATHEMATICAL MODEL OF RACK GENERATION FOR ROTORS

The fundamental idea is derived from the rack-generated profile .Each compound curve has at least one control parameter on the rack profile, which makes the rack both flexible and instinctively adjustable. The tooth profile of the basic rack depends on the pitch W and the total tooth height (ha + hd). As shown in Fig. 1, the addendum ha and the dedendum hd can be determined by the tooth number of the rotors z1 and z2, the center distance C, and the outer radii of the male and female rotors rp1 and rp2 as given in the following equations[10]:

$r_{p1}=A_c z_1 \ / (z_1+z_2)$ -(1)

$r_{p2}=A_c z_2 \ / (z_1+z_2)$ -(2)

$h_a = r_2-r_{p2}$ -(3)

$h_d = r_1-r_{p1}$ -(4)

$W=2\pi r_{p1} \ / z_1$ -(5)

2.1 Meshing condition for Screw compressor and

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Rack Co-ordinates

$x_2=x_{o1}\cos k\theta-y_{o1}\sin k\theta-c\cos(\theta/i)$ -(6)

$y_2= x_{o1}\sin k\theta+y_{o1}\cos k\theta+c\sin(\theta/i)$ -(7)

$x_{o1}=(r_e-r_1)+r_1\cos t$ -(8)

$y_{o1}=r_1\sin t$ -(9)

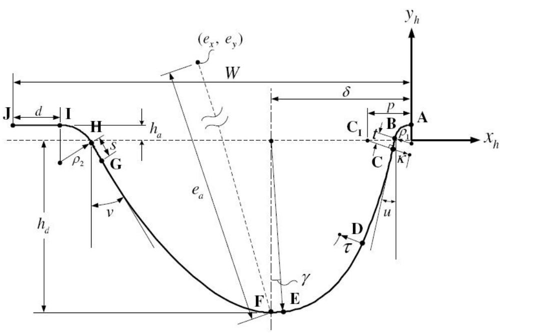

Fig.1 Rack Profile And Design Parameter

$x_r=x_{o1}\cos\theta-y_{o1}\sin\theta$ -(10)

$y_r=x_{o1}\sin\theta+y_{o1}\cos\theta-r_1\theta$ -(11)

$k=1-(1/i)$

$k=1-(1/6.9306)$

$k=1-0.1$

2.2 Calculation of rotor profiles in screw compressor optimization

A procedure to get the required meshing condition as described in [6]. More detailed information on the envelope method applied to gears can be found in [3]. The primary curves are specified on the rack: D–C is a circle with radius r_3 on the rack, C–B is a straight line, B–A is a parabola constrained by radius r_1, A–H–G are trochoids on the rack generated by the small circles of radii r_2 and r_4 from the male and female rotors respectively, G–E is a straight line and E–Fand E–D are circles on the rack. A full description of the rack generation procedure and rotor geometry is given in [10].

These three rotor radii, r_1, male rotor lobe radius, r_2; male rotor tip radius and r_3, rack root radius and the female rotor addendum r_0, as presented in Fig. 1, are used as variables for the rotor optimization[7].

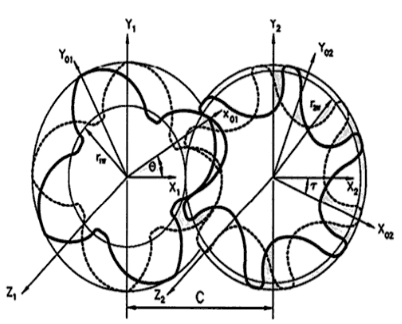

Fig.2 CO-ORDINATE AXIS OF ROTOR PROFILE

3 PARAMETRIC DETAIL DESIGN STUDY

3.1 Design of Rotors in Screw Compressor

(1)To find the center distance between rotors (C)

C= Radius of main rotor pitch circle and gate rotor pitch circle –(Hanjalic and Stotic 1994)

C= (outer diameter of female main/2)+(gate root diameter/2)

C=(98.18/2)+(81.82/2)

C=49.09+40.91

C=90mm

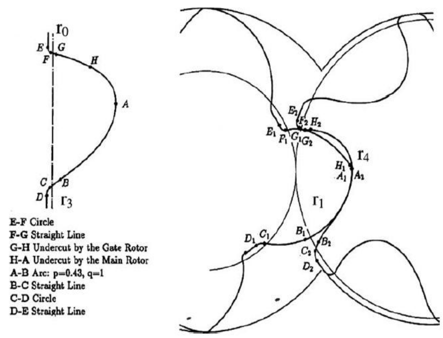

Fig.3 Profiling of Rack

(2)To find the external and internal radius of female rotor (re and ri)

External radius $r_e = r_w+r$

Internal radius $r_i = r_w-r_o$

$r_e =(52.8/2)+(48/2)$

$r_e =26.4+24$

External radius re for female rotor=50.4mm

$r_i = (100.8/2)-(48/2)$

$r_i = 50.4+24$

Internal radius r_i for female rotor= 26.4mm

External radius $r_e = r_w+r$

Internal radius $r_i = r_w-r_o$

$r_e =(81.82/2)+(45.38/2)$

$r_e =40.91+22.69$

External radius re for male rotor=52.8mm

$r_i = (127.2/2)-(45.38/2)$

$r_i = 63.6-22.69$

Internal radius r_i for male rotor= 40.91mm

3.2 Design of Diameter of Pipeline

Quantity of Compressed air flow =200cfm

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$=200*0.02831$
$=5.66 m^3/min$

Working Pressure $=100 psig$
$=100/14.22$
$=7.0323 kg/cm2$

Velocity $=6 m/sec$

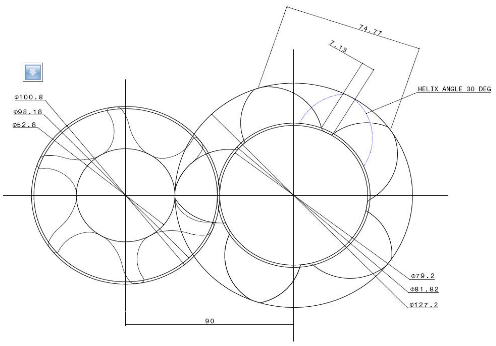

Fig.4 Dimension of Rotors

Apply gas laws. Assume the temperature remains constant

$P_1V_1 = P_2V_2$

$V_2 = P_1V_1/P_2$

Pressure at inlet (P_1) $=1.013 kg/cm^2$

Pressure at outlet (P_2) $=7 kg/cm^2$

Volume at inlet (V_1) $=5.66 m^3/min$

$V_2 = P_1V_1/P_2$

$V_2 =(1.013*5.66)/7$

$V_2 =(0.819\ m^3/min)/60$

$V_2 =0.01365\ m^3/sec$

Quantity of air flow of air flow =Area of pipe line x velocity

$0.01365\ m^3/sec$ = Area of pipe line x 6m/sec

Area of pipe line $=2.275x10^{-3}m^2$

$\pi/4xD_2$ $=2.275x10^{-3}m^2$

Diameter of pipe line = 0.0538m
=2.11"

(3)To find the Angular parameter (τ):

$\tau=\theta/i$ -Stotic 1994

θ-helix angle

i-Pressure ratio

$i=P_2/P_1$

$i=(7/1.01)$

$i=6.930$

$\theta=300$

$\tau=\theta/i$

$\tau=(30/6.930)$

$\tau=4.329$

(4)Calculated pitch and total height:

$r_{p1}=A_cz_1/(z_1+z_2)$

$r_{p2}=A_cz_2/(z_1+z_2)$

$h_a = r_2-r_{p2}$

$h_d = r_1-r_{p1}$

$W=2\pi r_{p1}/z_1$

W-pitch.

h_a, h_d –addendum and dedendum height.

r_{p1},r_{p2}-pitch radii of female and male rotor.

$h_a = 51.8mm$

$h_d = 86.3mm$

Total height = 138.1mm

$W = ((2*3.14)/5)*r_{p1}$

Pitch (W) = 51.374mm

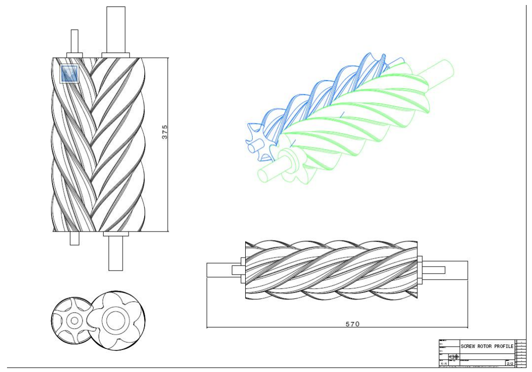

Fig.5 Rotor design

4 FACTORS AFFECTING COMPRESSOR EFFICIENCIES

- the blowhole area,
- the length of contact line,
- the volumetric efficiency
- the isentropic indicated efficiency

the related formulation and mathematical expression can be seen in the published book [9].

Only parameters ρ, u, t, κ and τ in the high-pressure side of the rotor can directly affect both the tip of the sealing line and the blowhole size within the housing cusp on the compression side, and the parameter has the greatest sensitivity on the blowhole area and volumetric efficiency because it primarily changes the cross-sectional volume of two rotors [10]. The fluid leakages and energy losses must be kept to a minimum to obtain the maximum volumetric and isentropic indicated efficiencies.

5 LIST OF SYMBOLS

TABLE1

Symbol	Quantity

r_w	Pitch radius of female and male rotor
r	Rotor radius of female and male rotor
r_o	Root radius of female and male rotor

6 CONCULSION

Optimisation of screw compressor geometry has been performed to establish the most efficient rotor design. This paper numerical solution and calculation enables optimum screw compressor flow power and compressor efficiencies. Each segment on the rack is given at least one instinctive adjustable parameter for modifying the generated rotor profile. As shown by the numerical formulation the proposed design method is able to improve the performance of twin-screw compressor. Rack-generated profiles of rotors which is used in the paper explains optimisation may permit both better delivery and higher efficiency.

REFERENCES

[1] F.L. Litvin, Teoria zubchanih zaceplenii (Theory of Gearing), Nauka, Moscow, 1956.

[2] F.L. Litvin, P.H. Feng, Computerized design, generation, and simulation of meshing of rotors of screw compressor, Journal of Mechanism and Machine Theory 32 (2) (1997) 137–160.

[3] F.L. Litvin ,Nauka, Moscow, Third ed.,1968; also Gear Geometry and Applied Theory, Prentice-Hill, Englewood Cliffs, NJ, 1994.

[4] L. Rinder, Screw rotor profile and method for generating, U.S. Patent No. 4,643,654, 1987.

[5] N. Stosic, Plural screw positive displacement machines, U.S. Patent No. 6,296,461 B1, 2001.

[6] N. Stosic, On gearing of helical screw compressor rotors, Proceedings of IMechEng, Journal of Mechanical Engineering Science 212 (1998) 587.

[7] N. Stosic, I.K. Smith, A. Kovacevic, Optimisation of screw compressors, Applied Thermal Engineering 23 (2003) 1177–1195.

[8] N. Stosic, K. Hanjalic, Development and optimisation of screw machines with a simulation model, Part I: Profile generation, ASME Transactions, Journal of Fluids Engineering 119 (1997) 659.

[9] Z.W. Xing, Screw Compressors: Theory, Design and Application, China Machine Press, Beijing, 2000 (in Chinese).

[10] Yu-Ren Wu, Zhang-Hua Fong, Optimization design of an explicitly defined rack for the generation of rotors for twin-screw compressors, Mechanism and Machine Theory 44 (2009) 66–82.

[11] D. Zaytsev, C.A. Infante Ferreira, Profile generation method for twin screw compressor rotors based on the meshing line, International Journal of Refrigeration 28 (2005) 744–755.