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On heat transfer in screw compressors

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ABSTRACT

Heat transfer between the working fluid and machine parts within a screw compressor does not affect its performance significantly because the thermal energy dissipation is usually less than 1% of the compressor power input. However, it can be detrimental to the machine reliability because the fluid compression creates a non-uniform three dimensional temperature field leading to local distortions, which may be larger than the clearances between the machine parts. This phenomenon is widely known and special control procedures are required to allow for start-up and shut down, as well as for steady running operation. These measures are usually derived only from test-bench data and may result in larger clearances than necessary, thereby reducing the optimum performance.

This paper gives an outline of two methods of computing heat transfer in a screw compressor; namely: by means of a quasi-one dimensional differential model and by three dimensional computational fluid dynamics (CFD). Both methods enable the clearance size for start-up and steady running conditions to be determined. The 3D CFD procedure is more accurate but requires a far longer running time. Two cases are considered: heat transfer in a dry screw compressor where fluid temperatures are high, and an oil-flooded screw compressor where fluid temperatures are relatively low but the convective heat transfer coefficient is substantially higher.

1. Introduction

Screw compressors play a significant role in the process industries for compression of air, gas, refrigerants and other compressible media. The main advantage of screw compressors over other types of positive displacement machine is the pure rotary motion of the moving components. This makes it possible to attain higher rotational speeds and, hence make more compact machines while maintaining high efficiency and delivery rates over a wide range of operating conditions, with less wear and thus a longer service life. Typically, screw compressors are up to five times lighter than reciprocating compressors of the same capacity and their service life is nearly ten times longer. Consequently an increasing proportion of positive displacement compressors sold and currently in operation are of this type. The present world annual production rate of positive displacement compressors is in excess of 200 million units, while approximately 17% of the world's electric power production is required to drive them in industrial, commercial and domestic applications. The majority of these are still reciprocating machines, but many other types, such as screw compressors, play a significant and ever increasing role.

A screw compressor consists essentially of a pair of meshing helical lobed rotors, contained in a casing. Together, these form a series of working chambers, as shown in Fig. 1, viewed from opposite ends and sides of the machine. The dark shaded portions show the enclosed region where the rotors are surrounded by the casing and where compression takes place, while the light shaded areas show the regions of the rotors that are exposed to external pressure. The large light shaded area in Fig. 1(a) corresponds to the low pressure suction port. The small light shaded region between the shaft ends B and D in Fig. 1(b) corresponds to the high pressure discharge port. Admission of the gas to be compressed occurs through the low pressure port which is formed by opening the casing surrounding the top and front faces of the rotors. Exposure of the space between the rotor lobes to the suction port, as their front ends pass across it, allows the gas to fill the passages formed between them and the casing. Further rotation then leads to cut off of the port and progressive reduction in the trapped volume in each passage, until the rear ends of the passages between the rotors are exposed to the high pressure discharge port. The gas then flows out through this at approximately constant pressure. Cross sectional views of a typical screw compressor are presented in Fig. 2.

Machines of this type are normally classified into two main types; namely: (i) oil injected and (ii) oil free, or dry compressors. The bulk of screw compressors manufactured are of the oil injected type. In these, a relatively large mass, but small volume of oil is admitted into the compressor after admission of the air or gas is

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Nomenclature

A	area, constant	R	has constant
a	speed of sound	$\mathbf{S}$	strain tensor
B	constant	T	temperature
C	specific heat, constant	$\boldsymbol{T}$	stress tensor
c	concentration	t	time
D	hydraulic diameter, constant	u, U	internal energy
$\mathbf{f}$	body force	$\boldsymbol{v}$	velocity
h	specific enthalpy, heat transfer coefficient by convection	V	volume
k	kinetic energy of turbulence	w	velocity
L	length	x	spatial coordinate, vapour quality
m	mass	z	real gas factor
Q	heat transferred	γ	isentropic exponent
Nu	Nusselt number	ε	dissipation of turbulent kinetic energy
$\boldsymbol{q}$	diffusion flux	ω	angular velocity
Pr	Prandtl number	ρ	density
Re	Reynolds number	θ	shaft angle
p	pressure		
$\mathbf{s}$	control volume surface		

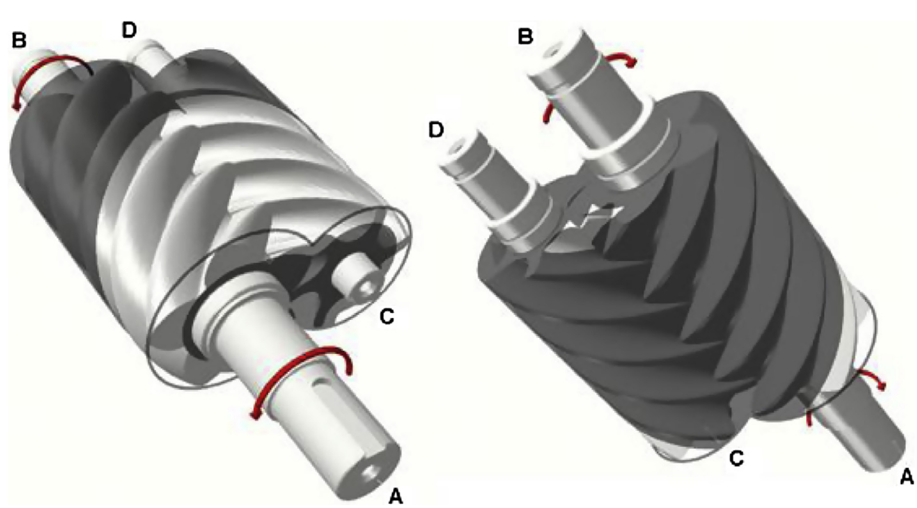
complete, and remains in contact with the gas being compressed as a dispersed liquid, which is then discharged with the gas. After leaving the compressor, the oil is separated from the gas, cooled and then reinjected. The driving force for reinjection is the pressure difference between the discharged gas and that between the meshing rotors, trapped in the compressor immediately after suction is complete. The oil serves three purposes; namely as a lubricant, as a sealant of the clearances between the rotors and between the rotors and the casing, and as a coolant of the gas being compressed. Analysis of oil influence upon the screw compressor process was presented in Stosic et al. (1992) where special emphasis was given to the gas-oil heat transfer. Because of this latter effect, gases can be compressed to pressure ratios of up to about 15:1 in a single stage without an excessive temperature rise.

Dry gas compression in screw machines is limited to a pressure ratio of approximately 3:1. This is because the higher gas temperatures of the uncooled gas cause the rotors and housings to deform. It follows that if the rotors were cooled, higher pressure ratios would be possible. To do this effectively, the principle of heat transfer within these machines needs be properly understood.

2. Quasi one dimensional model

Heat transfer in screw compressors can be evaluated from a general quasi one-dimensional model, based on a simplified description of the thermodynamic and fluid flow processes within the machine due to changes in the working chamber size and shape, as it rotates. These processes are defined by a set of equations which describe the physics of the complete compression process and machine kinematics.

The equation set consists of the equations for the conservation of energy and mass and a number of algebraic equations defining flow phenomena in the suction, compression and discharge processes. This set is complemented with a differential kinematic relationship that describes the instantaneous operating volume and its change with rotation angle or time. In addition, the model accounts for a number of 'real-life' effects which may influence the final performance of a compressor and make the model valid for a wider range of applications. In the past, these equations have often been simplified in order to achieve a more efficient and economical numerical solution. The complete equations, where all the terms



(a) View from front and top (b) View from bottom and rear

Fig. 1. Screw compressor rotors.