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OIL FREE TURBO-COMPRESSORS FOR REFRIGERATION APPLICATIONS

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ABSTRACT

Feasibility of replacing oil-lubricated compressing equipment in CO_2 based refrigeration systems with a new generation of oil-free turbo-machinery is assessed. Efficiencies of oil-free turbo-compressors working in its design points were predicted for various operating conditions and sizes. Typical configuration simulated in this study was two-stage compressor with a brushless DC motor placed between two impellers oriented in opposite direction so as to balance axial forces generated on each impeller due to the pressure gradient.

A model for CO_2 radial turbo compressors, including (basic) impeller design and all relevant loss factors of the compressor and motor, has been developed. For each set of conditions machines with different rotational speeds and capacities were simulated. For each rotational speed and operating conditions an optimal capacity exists for which the machine reaches the highest efficiency and this is analogous to specific speed/efficiency relation presented in a number of publications for various turbo-machines. It should however be stressed that in case of oil-free hermetic turbo-compressor an optimal stage specific speed does not always guarantee optimized machine performance. It is dictated by strong motor and thrust bearing sensitivity to shaft speed and it is sometimes advantageous to reduce stage specific speed. As a result, the total machine efficiency can be improved while stage efficiency is reduced. The general trend is however comparable with typical turbo-compressors working in open configuration – the lower the pressure ratio and higher the capacity is, the higher the performance. In the simulated cases it is clearly shown that machines operating with high pressure ratios (exceeding 3) cannot reach high efficiencies with a proposed two stage configuration.

It is concluded that the desirable oil-free operation of a refrigeration system can be achieved without sacrificing reasonably high compression efficiency targets. Usually however, a 4-stage compression is required in order to mitigate otherwise significant motor and gas bearings windage stemming from operation in high density gas and under high rotational speeds. The study proposes placing two two-stage machines in series instead of four impellers on one single shaft. Such configuration allows for better total motor length/diameter ratio (reduced windage losses) and easier startup of the system. Due to the lower work input per impeller, optimal rotational speed is also reduced.

1. INTRODUCTION

Small vending machines, residential heat pump water heaters, commercial and industrial refrigeration, ventilation and air conditioning, small to medium scale power production from waste heat - these are only some of the areas where CO_2 is used as a working fluid heat [Nekså et.al. (2010), "NARECO2" (2009), Chen et. al. (2006), Zhang et. al. (2006)]. Environmentally benign characteristics, low cost and good thermo-physical properties of carbon dioxide account for its increasing popularity. Depending on the type of application, designers of the system must cope with different type of often antagonizing requirements such as: energy efficiency, size limitations, manufacturing, maintenance costs and safety.

Regardless of the application type, oil-free operation is almost always a desired characteristic as it usually results in simplified architecture of the system due to the elimination of oil distributing components, reduced investment and maintenance cost (longer inspection periods) and improved heat exchange conditions (no heat transfer surface contamination with oil). For refrigeration systems, oil-free operation additionally means increased maximum discharge

temperature (no risk of decomposition of lubricant) allowing for higher pressure ratios per compressor stage, and thus limiting number of necessary components. It also allows entering new areas at temperatures below -40°C or even -56°C [Hafner et. al. (2011)], enabling ultra-low temperature refrigeration technology.
In typical closed circuit processes an oil or lubricant is used to provide lubrication of mechanical moving parts in the compressor. Due to the imperfect sealing of compressor parts, some of this oil is transported with the working fluid into the system and contaminates inner surfaces of tubes, expansion devices and heat exchangers. This oil has to be captured using different types of oil separation and management systems which are usually costly and add complexity to the system. It is therefore desirable to enable an oil free system.
We can distinguish between three main types of compressors: reciprocating, rotary (screw, scroll, swing) and turbo-compressors. Oil-free reciprocating piston compressors are typically of the labyrinth seal type. These are slow running costly compressors with relatively low efficiency. Oil free screw compressors are also costly and the efficiency is relatively low due to excessive leakage losses. Radial oil-free turbo-compressors were proven to work successfully in a wide range of applications [DellaCorte and Bruckner (2011), Walton and Heshmat (1994)].
The selection of the appropriate bearing technology is crucial for proper design of an oil-free compressor. According to Molyneaux and Zanelli (1996), replacing oil or grease in the rolling element of the bearing with refrigerant in liquid phase is possible, however at the expense of the additional losses due to the constant delivery of a bypass refrigerant stream. The life span of such bearing operating at high speeds was not reported. Magnetic bearings are an option provided that operating conditions are stable over a longer period of time. The technology is commercially proven, but adds complexity to the system (auxiliary feedback control, touchdown bearings) and increased investment cost.
Fluid film bearings are the most common candidate for oil-free machinery as they are relatively simple in construction and exhibit successful track record from various fields of application and over a wide range of rotating speeds. Schiffmann (2008) gives an overview of different types of gas lubricated bearings pointing out two especially interesting alternatives, namely foil bearings and herringbone grooved bearings. Both technologies have been successfully applied in many systems and both have their pros and cons. While herringbone groove bearings are easier to manufacture, they display lower misalignment tolerance and load capacities than foil bearings.
Foil bearings, on the other hand, with its greater misalignment tolerance can affect impeller efficiency due to the higher impeller tip clearances required, and they are more difficult to model. The literature on foil bearings seems richer and covers a wide range of applications, from micro-turbines producing barely 100W to a few hundred kW compressors. The foil bearing air cycle machine on the Boeing 747 aircraft has demonstrated an MTBF (mean time between failures) in excess of 100,000 hours. Air cycle machines with foil bearings has served on many other passenger and military aircrafts and their application is a common practice today [Agrawal (1997)].
While substantial amount of information can be found on air oil-free turbo-machinery, the literature on CO_2 oil-free applications is rather scanty. Wright et al. (2010) performed testing and analysis of a supercritical CO_2 Brayton loop with 50kW compressor supported on foil bearings. The study confirms feasibility and performance potential of such cycles and takes notice of significant motor windage in high density gas at high shaft speeds. Performance test of foil journal bearing in various gases conducted by Briggs et. al. (2008) confirms power loss sensitivity to gas density and rotational speeds.
The purpose of this work is to assess the feasibility of replacement of standard oil-lubricated compressors operating in various industrial, household or commercial applications utilizing CO_2 as a working fluid with a new generation of oil-free turbo-machinery.
Due to the closed loop nature of a predominant number of these systems, the study will consider hermetic type of compressors with internal recirculation of gas leaked to the motor cavity through the seals. Foil bearings are selected for this study as a proven technology with much development potential in the future.

2. THEORY AND MODELLING

The intension of this study was to include all possible inefficiencies constituting the final performance of the machine. Not only the aerodynamic performance is predicted, but bearings and motor windage, electrical losses of driver and motor and cooling flow loss are also taken into account. Full assessment of total contribution of various loss mechanisms is necessary to compare new type of compressors with existing commercial solutions. Various relations of main compressor stage dimensions are assumed and kept constant for different machine applications to provide consistent point of reference in assessing different applications' feasibility. Values of non-dimensional relations are chosen based on suggestions found in Whitfield and Baines (1990). It should further be acknowledged that each machine's design is not optimized, but rather based on good engineering practice. Due to the paper volume limitations the design procedure could not be presented here in full detail. The most important assumptions are presented below. The paper in its full scope is expected to be published soon.

2.1 Assumptions

- Impeller is open
- Work split between stages is based on equal pressure ratios
- Radial gap in journal bearings = 20µm
- Journal foil bearing load capacity coefficient = 27680 kg/m^3/krpm (1 lb/in^3/krpm)
- Axial foil bearing load capacity coefficient = 830 kg/m^3/krpm (0.03 lb/in^3/krpm)
- Labyrinth seal diameter = $0.35 \cdot D_2$
- Motor electrical efficiency = 0.97
- Driver electrical efficiency = 0.97
- Gas superheat at compressor inlet = 5K
- Inner to outer diameter ratio of thrust bearing = 0.35
- Axial length of impeller = $0.3 \cdot D_2$
- σ_m =18.6 kPa (2.7 psi)
- $D_{1h}/D_{1t} = 0.3$
- $\alpha_{2r} = 75^o$
- β_{1r} (mean) = 45 ^{o}
- $\beta_{2r} = 45$ ^{o}
- $Z_B = 15$
- Clearance = 0.3% of D_2
- $L_{mot}/D_{mot} = 3.9$
- $D_3/D_2 = 1.8$
- Internal cooling ratio = 0.5
- $L_{JB}/D_{JB}=1$
- Foil bearings sizing is based on Dykas et. al. (2009)
- Motor sizing is based on Hanselman (2006)
- Simplified radial force calculation is based on the weight of the rotor.
- Cooling gas undergoes isenthalpic expansion in labyrinth seals and heat exchange process is isobaric
- The motor cavity pressure is equal to compressor inlet pressure
- Mass flows leaking from different stages are proportional to the pressure difference between stage discharge and motor cavity
- Cavity discharge gas temperature equals compressor discharge temperature
- Cooling mass flow is compressed again to compressor discharge pressure with calculated total machine's efficiency

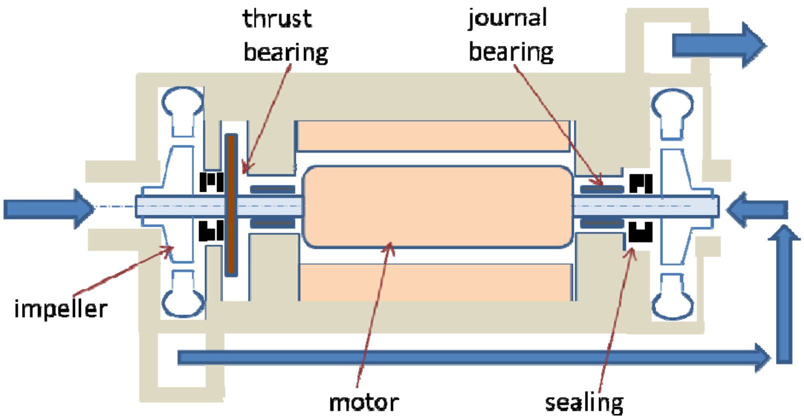

Figure 1. Simulated compressors configuration

2.2 Loss correlations

Table 1. Loss correlations

Loss mechanism	Loss model	Reference
Blade loading loss	$\Delta h_{bl} = 0.05 D_f{}^2 U_2^2$ $D_f = 1 - \frac{W_2}{W_{1t}} + \frac{0.75 \Delta h_{Euler} / U_2^2}{(W_{1s}/W_2)\left[(Z/\pi)(1 - D_{1t}/D_2) + 2D_{1t}/D_2\right]}$ $\Delta h_{Eurler} = C_{\theta 2} U_2 - C_{\theta 1} U_1$	Coppage et. al. (1956)
Impeller skin friction	$\Delta h_{sf} = 2 c_{fi} (L_i / D_{ih}) W_m^2$ $W_m = \frac{C_{1t} + C_2 + W_{1t} + 2W_{1h} + 3W_2}{8}$	Jansen (1967)

	$c_{fi} = 0.3164\,(\mathrm{Re}_i)^{-0.25}$	
Vaneless diffuser loss	$\Delta h_{df} = 2c_{fd}(L_d / D_{dh})C_m^2$ $c_{fd} = k\left(\dfrac{1.8\cdot 10^5}{\mathrm{Re}_d}\right)^{0.2}$ $k = 0.01$	Japikse (1982)
Clearance loss	$\Delta h_{cl} = 0.6\dfrac{\varepsilon}{b_2}C_{\theta 2}\left\{\dfrac{4\pi}{b_2 Z_b}\left[\dfrac{r_{1s}^2 - r_{1h}^2}{(r_2 - r_{1t})(1+\rho_2/\rho_1)}\right]C_{\theta 2}C_{r1}\right\}^{1/2}$	Jansen (1967)
Mixing loss	$\Delta h_{mix} = \dfrac{1}{1+\tan^2\alpha_{2r}}\left(\dfrac{1-\varepsilon_{wake} - b^*}{1-\varepsilon_{wake}}\right)\dfrac{C_2^2}{2}$	Johnston and Dean (1966) Oh et. al. (1997)
Disc friction loss	$\Delta h_{disc} = C_{f\,disc}\dfrac{\bar{\rho} r_2^2 U_2^3}{4\dot{m}}$ $\mathrm{Re}_{disc} = \dfrac{U_2 r_2}{\nu_2} \qquad \bar{\rho} = \dfrac{\rho_1 + \rho_2}{2}$ $\left\{\begin{array}{ll}\dfrac{2.67}{\mathrm{Re}_{disc}^{0.5}}, & \mathrm{Re}_{disc} < 3x10^5 \\ \dfrac{0.0622}{\mathrm{Re}_{disc}^{0.2}}, & \mathrm{Re}_{disc} > 3x10^5\end{array}\right.$	Daily and Nece (1960)
Recirculation loss	$\Delta h_{rc} = 0.02 D_f^2 U_2^2 \tan\alpha_{2r}$	Coppage et. al. (1956)
Volute loss	$\Delta h_{vol} = C_{3r}^2 / 2$	
Motor windage	$P_{mw} = 0.074\rho_{cav} L_{mot}\pi R_{rot}{}^4\omega^3\left(\dfrac{\nu_{cav}}{2\pi R_{rot}^2\omega}\right)^{0.2}$	Saint Raymond et. al. (2008) Etemad et. al. (1992), Allaire et. al. (1998)
Radial bearing loss	$P_{JB} = 2\pi\mu_{cav} U_{JB}^2 r_{JB}\dfrac{L_{JB}}{gap_{JB}}$	Müller and Fréchette (2002)
Axial bearing windage	$P_{AB} = \tau\omega$ $c_M = 0.0622(\mathrm{Re})^{-1/5}, c_M = \dfrac{2\tau}{0.5\rho_{cav}\omega^2 r^5},$ $\mathrm{Re} = \dfrac{r^2\varpi}{\nu}$	Schlichting (1968)

3. RESULTS

Analysis of a lower stage loop in a cascade refrigeration system is presented below. Evaporation and condensation pressures amount to 1 and 3 MPa respectively.

For the same set of operating conditions a number of simulations for different capacities and rotational speeds have been performed. It is clear that in case of a turbo-compressor, for each rotational speed, a specific capacity exists that provide an optimal efficiency. Smaller machines require higher rotational speeds and high speeds involve high windage and cooling losses. It means that where smaller capacities are required, high efficiencies will be difficult to achieve with the proposed two-stage hermetic design (see Fig 2a).

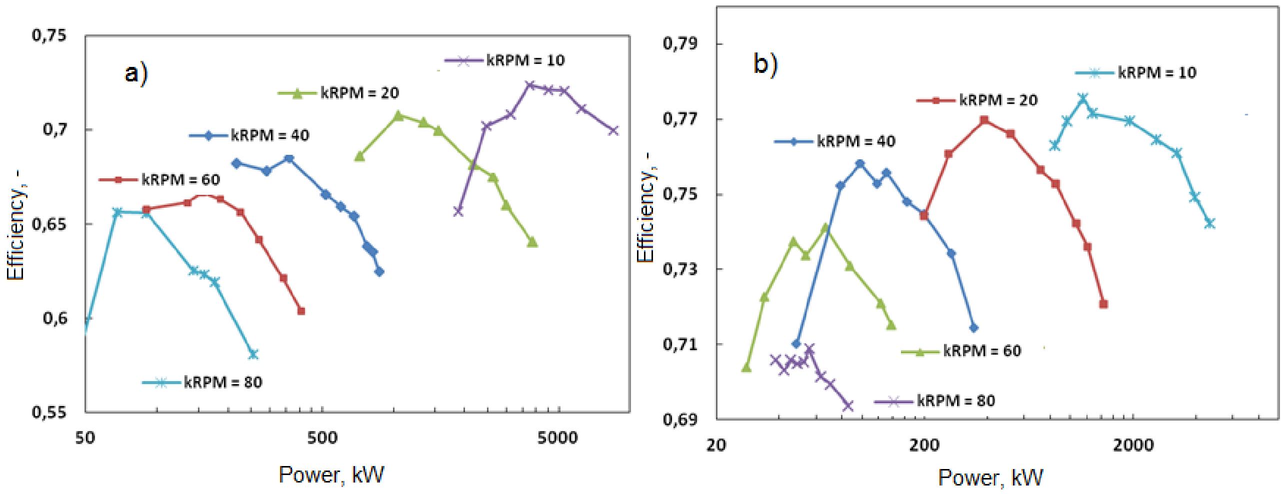

Figure 2. 2-stage (a) and 4-stage (b) compression efficiency for low stage loop in cascade refrigeration system.

If we look at the loss breakdown for a machine in the lower capacity range (**single 2-stage compressor, 80 krpm, 84 kW**), significant windage and associated cooling flow losses are dominant (Fig 3a). The cooling flow involves significant recompression work of CO_2 throttled to low density and then heated in motor cavity.
For bigger compressors (**single 2-stage compressor, 20krpm, 1333 kW**) non-stage losses, such as bearing and motor windage are reduced due to lower rotational speeds (Fig 3b).

One way of increasing the compression efficiency is to increase the number of compression stages. As stated before, it is desirable that the impellers work in pairs in order to reduce the axial forces responsible for additional size and associated windage of the thrust bearing. Therefore we propose to use two compressors in series. This will result in higher total motor length/diameter ratio and lower shaft speeds thus reducing motor windage. Such a configuration allows reaching much better compression efficiencies in wider capacity range, but at the expense of somewhat increased system complexity and cost.
Loss breakdown for 4-stage compression (**2 machines in series, 40 krpm, 98 kW in total**) is presented in (Fig 3c). Thanks to significantly reduced non-stage losses relatively high efficiencies (>70%) are achievable for compressor capacities higher than 40 kW.

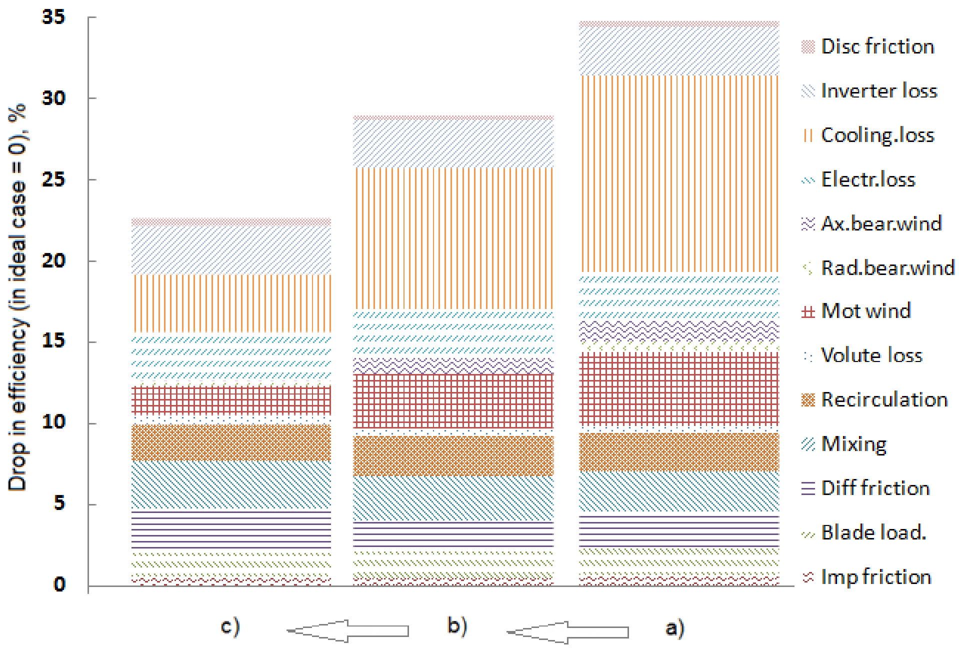

Figure 3. Loss breakdown for selected compressors

As mentioned, there exists an optimal rotational speed for which the compressor of a given capacity and number of stages will achieve an optimal performance. Fig 4 presents expected performances achievable for different operating conditions and compressor configurations (number of stages).

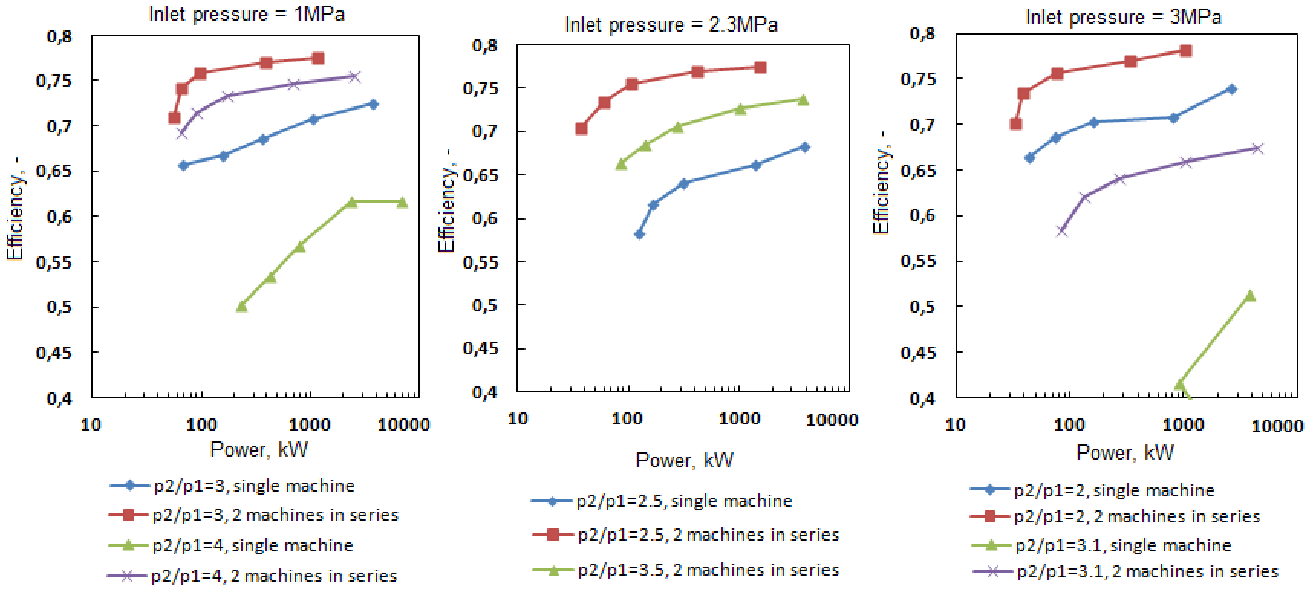

Figure 4. Expected achievable compression efficiency for 2- and 4-stage compression in various operating conditions.

4. CONCLUSIONS

For the simulated cases it is clearly visible that machines with high pressure ratios (exceeding 3) cannot reach high efficiencies with the proposed two stage configuration. Increased unbalanced axial forces resulting in bigger thrust bearing as well as higher motor windage and losses due to cooling have a detrimental impact on the machine's performance. A solution to overcome this can be to put two machines in series, especially when capacities are at the lower range of what is applicable for centrifugal compressor applications. Placing compressors in series instead of increasing the number of stages of a single machine has an important advantage. It increases the specific speed, reduces the axial forces and makes the total motor diameter/length ratio low. It should also provide easier operation during start-up modes. The inhibiting factor for such a configuration can be the additional cost.
As mentioned before, achievable efficiencies are calculated based on fixed set of assumptions. There seem to be ways of performance improvement, such as balancing of axial forces by shifting some amount of work between the two stages or manipulating the diameter of the labyrinth seal. Constant development of foil bearing technology allows us presuming that higher load capabilities, and thereby smaller bearings, could be applied.
In some cases inter-stage cooling also could be a viable option. From the modeling approach it is vital to note that calculation of axial forces as well as windage on the thrust bearing is a subject to a significant uncertainty, and simple 1D prediction may not be sufficient to approximate real physical behavior with good accuracy. Typically, to obtain reliable data, full-scale tests are indispensable [Wright et. al. (2010), Rababa (2010), Noall and Batton (2011)].
At a first glance, introducing oil-free hermetic turbo-compressors into commercial or industrial refrigeration applications seems feasible, still maintaining reasonable efficiencies, however, at the expense of more compression stages, particularly in lower range of capacities. Whether such configuration may serve successfully in applications where operating conditions change significantly is yet to be answered in more detailed studies that are planned to be performed in the near future.
For the investigated application with a suction pressure of 1MPa and a pressure ratio of 3 a single two stage compressor seems viable if the cooling capacity is in the range of 1.3 MW or higher. Efficiencies close to 70% may be expected. Considering two two-stage compressors in series for the same application will increase the obtainable compressor efficiencies, thus feasible capacity range may be lowered to approximately 130 kW.

NOMENCLATURE

b	blade height (m)	β_{b2}	discharge blade angle(rad)	cl	clearance
b*	ratio of vaneless diffuser inlet width to impeller exit width (-)	ε	clearance (m)	d	diffuser passage
C	absolute velocity (m/s)	ε_{wake}	wake fraction of blade-to-blade space (-)	df	diffuser friction
c_f	skin friction coefficient (-)	ε	clearance (m)	h	hub/hydraulic
c_M	Dimensionless torque coefficient (-)	ε_{wake}	wake fraction of blade-to-blade space (-)	i	Impeller channel
D	diameter (m)	ω	angular velocity (rad/s)	JB	journal bearing
D_f	diffusion factor (-)	W	relative velocity (m/s)	LS	labirynth seal
h	specific entalphy (kJ/kg)	τ	torque (N·m)	m	mean
L	length (m)	μ	viscosity (Pa·s)	mix	mixing
p	pressure (Pa)	ρ	density (kg/m^3)	mot	motor's rotor
P	Power (W)	σ_m	the motor air gap shear stress (Pa)	mw	motor windage
r	radius (m)	**Subscripts**		r	meridional direction (angle) / meridional component
Re	Reynolds number (-)	1	impeller inlet	sf	skin friction
T	temperature (K)	2	impeller discharge	t	shroud
U	impeller blade tip speed (m/s)	3	diffuser discharge	θ	circumferential direction (angle) / circumferential component
Z_B	number of blades (-)	AB	axial bearing		
α	absolute flow angle (rad)	bl	blade loading		
β	relative flow angle (rad)	cav	motor cavity		

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