

OPTIMIZATION OF SCREW COMPRESSOR DESIGN

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ABSTRACT

An optimisation procedure for twin screw compressor design is described, which is based on an analytical algorithm for rotor profile generation combined with a differential model of the fluid flow and thermodynamic processes within the machine. Rotor profiles are generated from a rack which have both stronger female rotor lobes and higher displacement. Compressors thus designed have higher delivery rates and better efficiencies than those using more well known profiles. Some optimization issues of the rotor profile and compressor ports are discussed, using a 5/6-106 mm screw compressor to illustrate the results. It is shown that the optimum rotor profile, compressor speed, oil flow rate and inlet temperature may differ significantly when compressing different gases or vapours.

Key Words: Screw compressor design, rotor lobe optimal profiling, mathematical modelling

INTRODUCTION

Screw compressors are rotary positive displacement machines of simple design which are capable of high speed operation over a wide range of operating pressures and flow rates. Also, they are reliable and compact. Their rotors can now be manufactured with small clearances at an economic cost. Internal leakages have therefore been reduced to a small fraction of their values in earlier designs. As a result, screw compressors are more efficient than most other types of positive displacement machine. Consequently they are widely used to compress air, gases and refrigerants and the majority of all positive displacement compressors now manufactured as well as those currently in operation are of this type.

An outline of the rotors and compressor housing is shown in Fig. 1. The main rotor, on the right, rotates clockwise, while the female or gate rotor, on the left, rotates anticlockwise. Gas enters through the port on the upper face of the rotors and is discharged through the port on their lower face.

Recent advances in mathematical modelling and computer simulation may be used to form a powerful tool for process analysis and design optimisation. Such models have evolved greatly during the past ten years and, as they are better validated, their value as a design tool has increased. Their use has led to a steady evolution in screw rotor profiles and compressor shapes which should continue in future to lead to further improvements in machine performance. Evidence of this may be seen in the publications by *Sauls, 1994* and *Fujiwara and Osada, 1995*. In order to make such computer models more readily accessible to designers and engineers, as well as specialists, the authors have developed a suite of subroutines for the purpose of screw machine design, *Hanjalic and Stosic, 1997*.

The best rotor configuration is almost completely determined by the working fluid and the suction and discharge pressures. Also, there are several criteria for screw profile optimization which are valid irrespective of the machine type and duty. Thus, an efficient screw machine must admit the highest possible fluid flow rates for a given machine rotor size and speed. This implies that the fluid flow cross-sectional area must be as large as possible. In addition, the maximum delivery per unit size or weight of the machine must be accompanied by minimum power utilization for a compressor and maximum power output for an expander. This implies that the efficiency of the energy interchange between the fluid and the machine is a maximum. Accordingly unavoidable losses such as fluid leakage and energy losses must be kept to a minimum. However, increased leakage may be more than compensated by greater bulk fluid flow rates. However, specification of the required compressor delivery rate requires simultaneous optimisation of the rotor size and speed to minimise the compressor weight while maximising its efficiency. Finally, for oil-flooded compressors, the oil injection flow rate, inlet temperature and position needs to be optimised. It follows that a multivariable minimization procedure is needed for screw compressor design with the optimum function criterion comprising a weighted balance between compressor size and efficiency or specific power.



Fig. 1 Screw compressor principal mechanical parts

A box simplex method was used here to find the local minima, which were input to an expanding compressor database. This finally served to estimate a global minimum. The database may be used later in conjunction with other results to accelerate the minimization.

GEOMETRY OF SCREW COMPRESSOR ROTORS

Screw machine rotors have parallel axes and a uniform lead and they are a form of helical gears. The rotors make line contact and the meshing criterion in the transverse plane perpendicular to their axes is the same as that of spur gears. Although spur gear meshing fully defines helical screw rotors, it is more convenient to use the envelope condition for crossed helical gears to get the required meshing condition as described in *Stosic, 1998*. More detailed information on the envelope method applied to gears can be found in *Litvin, 1994*.

To start the procedure of rotor profiling, the profile point coordinates in the transverse plane of one rotor, and their first derivatives, must be known. This profile can be specified on either the main or gate rotors or in sequence on both. Also the primary profile may also be defined as a rack as shown in Fig xx.

A helicoid surface and its derivatives for the given rotor profile can be found from the transverse plane rotor coordinates. The envelope meshing condition for screw machine rotors gives the meshing condition either numerically, if the generating curves are given on the compressor rotors, or directly, if the curves are given on the rotor rack. This enables a variety of primary arc curves to be used and basically offers a general procedure. Moreover, numerical derivation of the primary arcs permits such an approach even when only the coordinates of the primary curves are known, without their derivatives.

The sealing line of screw compressor rotors is somewhat similar to a gear contact line. Since there exists a clearance gap between rotors, the sealing line represents the locus of points of the most proximate rotor position. Its coordinates are x_1 , y_1 and z_1 and these are calculated for the same distribution. The most convenient means of obtaining the interlobe clearance gap is to consider it as the shortest distance between two rotor racks of the main and gate rotor sealing points in the cross section normal to the rotor helicoids.

The following are the elements of the rack-generated 'N' profile. 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 main and gate rotors respectively, G-E is a straight line and E-F and E-D are circles on the rack. A full description of the rack generation procedure and rotor geometry is given in *Stosic and Hanjalic, 1997*. Three rotor radii, r_1 - r_3 and the gate rotor addendum r_0 are used as variables for the rotor optimisation.

Full rotor and compressor geometry, like the rotor throughput cross section, rotor displacement, sealing lines and leakage flow cross section, as well as suction and discharge port coordinates are calculated from the rotor transverse plane coordinates and rotor length and lead. They are later used as input parameters for calculation of the screw compressor thermodynamic process. For any variation of input parameters r_0 to r_3 , the primary arcs must be recalculated and a full transformation performed to obtain the current rotor and compressor geometry.

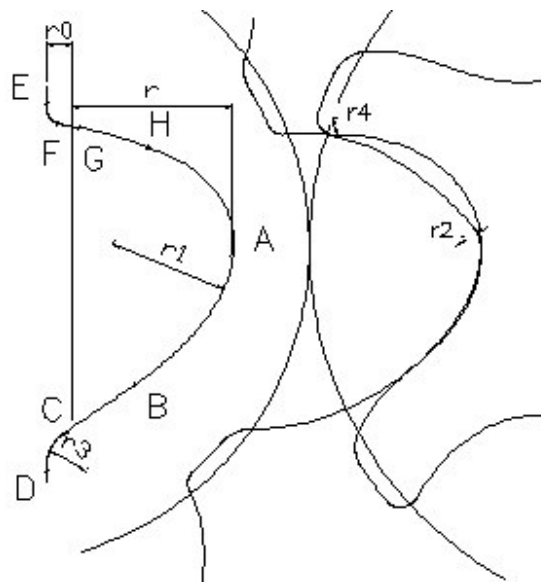


Fig 2. Distribution of generating profile curves on the rack for 'N' rotors

COMPRESSOR THERMODYNAMICS IN OPTIMISATION CALCULATIONS

The algorithm of the thermodynamic and flow processes used is based on a mathematical model comprising a set of equations which describe the physics of all the processes within the screw compressor. The mathematical model describes an instantaneous operating volume, which changes with rotation angle or time, together with the equations of conservation of mass and energy flow through it, and a number of algebraic equations defining phenomena associated with the flow. These are applied to each process that the fluid is subjected to within the machine; namely, suction, compression and discharge. The set of differential equations thus derived cannot be solved analytically in closed form. In the past, various simplifications have been made to the equations in order to expedite their numerical solution. The present model is more comprehensive and it is possible to observe the consequences of neglecting some of the terms in the equations and to determine the validity of such assumptions. This provision gives more generality to the model and makes it suitable for other applications.

A feature of the model is the use energy equation in the form which results in internal energy rather than enthalpy as the derived variable. This was found to be computationally more convenient, especially when evaluating the properties of real fluids because their temperature and pressure calculation is not explicit. However, since the internal energy can be expressed as a function of the temperature and specific volume only, pressure can be calculated subsequently directly. All the remaining thermodynamic and fluid properties within the machine cycle are derived from the internal energy and the volume and the computation is carried out through several cycles until the solution converges. A full description of the model is given in *Hanjalic and Stosic, 1997*.

Differential equations of mass, momentum and energy conservation have been employed in the model in function of the compressor shaft rotation angle. Compressor mass flow rate $m=m(\theta)$ and fluid internal energy $u=u(\theta)$ within the control volume with help of the chamber volume $V=V(\theta)$ make a basis for the pressure and temperature calculations. The instantaneous density $\rho=\rho(\theta)$ is obtained from the instantaneous mass m trapped in the control volume and the size of the corresponding instantaneous volume V as $\rho=m/V$. The suction and discharge port flow is defined by the velocity through them and their cross section area. The cross-section area A is obtained from the compressor geometry and it was considered as a periodical function of the angle of rotation θ . Leakage in a screw machine forms a substantial part of the total flow rate and plays an important role because it affects the delivered mass flow rate and hence both the compressor volumetric and adiabatic efficiencies. Injection of oil or other liquids for lubrication, cooling or sealing purposes, modifies the thermodynamic process in a screw compressor substantially. Special effects, such as gas or its condensate mixing and dissolving in or coming out of the injected fluid should be accounted for separately if they are expected to affect the process. In addition to lubrication, the major purpose for injecting oil into a compressor is to cool the gas.

The solution of the equation set in the form of internal energy U and mass m is performed numerically by means of the Runge-Kutta 4th order method, with appropriate initial and boundary conditions. As the initial conditions were arbitrary selected, the convergence of the solution is achieved after the difference between two consecutive compressor cycles is sufficiently small. Numerical solution of the mathematical model of the physical process in the compressor provides a basis for a more exact computation of all desired integral characteristics with a satisfactory degree of accuracy. The most important of these properties are the compressor mass flow rate \dot{m} [kg/s], the indicated power P_{ind} [kW], specific indicated power P_s [kJ/kg], volumetric efficiency η_v , adiabatic efficiency η_a , isothermal efficiency η_t and other efficiencies, and the power utilization coefficient, indicated efficiency η_i .

OPTIMISATION OF THE ROTOR PROFILE AND COMPRESSOR DESIGN

The power and capacity of contemporary computers is only just sufficient to enable a full multivariable optimisation of both the rotor profile and the whole compressor design to be performed simultaneously in one pass and, to the best of the authors' knowledge, this is the first publication of such an attempt.

Screw machines are today used to compress both dry and oil-flooded air, refrigerants and process gases and the requirements for optimum design of their rotors and other elements differ for each application and working fluid. Multivariable optimisation therefore should be employed as the starting point of the design function.

As already stated, there are several criteria for screw profile optimisation, which are valid irrespective of the machine type and duty. Optimisation targets must therefore be set according to the design requirement. Thus, if high efficiency is required, the specific power or adiabatic and volumetric efficiencies will be targets, if the compressor capacity is to be maximized, compressor flow will be the optimisation target.

As an example to show how designs may vary, optimisation studies were carried out on a compressor of given rotor diameter, centre line distance and lobe configuration for dry and oil-flooded air compression and refrigeration duties. Since the isentropic index of compression is much larger for dry air than for oil injected air and refrigerants significant differences were anticipated. Small rotors, 106 mm in diameter with a centre distance of 76 mm and with 5 lobes in the main rotor and 6 lobes in the gate rotor (5/6) were chosen. Different speed ranges were selected to serve as a basis for optimisation. For air, the inlet conditions selected were, $p_0=1$ bar and $T_0=300$ K, to comply with normal air compressor practice, while the discharge pressure was taken as 3 bars for dry air and 8 bars for the oil-flooded compressors respectively. For the refrigeration compressor evaporation and condensation temperatures were taken as 268 and 313 K respectively. In the example presented, the rotor centre distance, as well as the main rotor diameter was kept constant, while the other profile radii, r_0 , r_1 , r_2 , r_3 and r_4 , were allowed to vary. The rotor interlobe clearances were set as input parameters.

A box constrained simplex method was used here to find the local minima. The box method stochastically selects a simplex, which is a matrix of independent variables and calculates the optimisation target. This is later compared with those of previous calculations and then their minimization is performed. One or more optimisation variables may be limited by the calculation results in the constrained Box method. This gives additional flexibility to the compressor optimisation.

The optimisation criterion was the lowest compressor specific power. As a result, three distinctively different rotor profiles were calculated, one for oil-free compression and the other two for oil-flooded air and refrigeration compression. They are presented in Figs. 4-6. Although the profiles look alike, there is a substantial difference between their geometry which is given in the following table as well as further results of the compressor optimization.

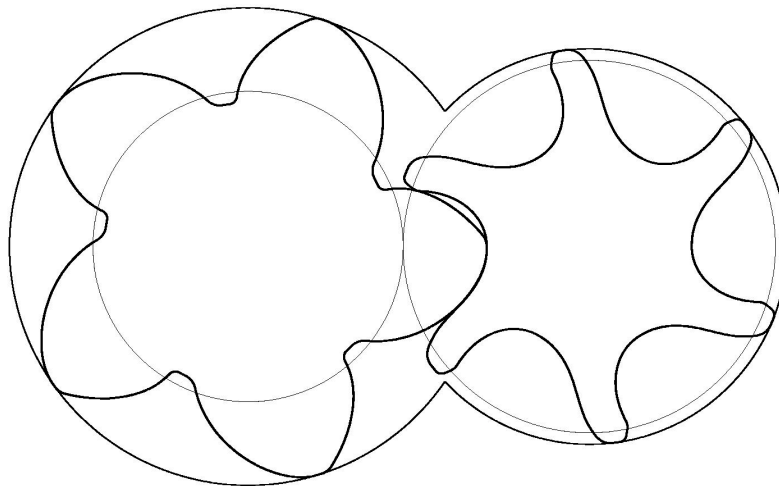


Fig. 3 Rotor profile optimized for an oil-free air compressor

The optimisation results, after being input to an expandable compressor database, finally served to estimate a global minimum. The database may be used later in conjunction with other results to accelerate the minimization. As is the case of any result of multivariable optimization, the calculated screw compressor profile and compressor design parameters must be considered with the extreme caution. This is because multivariable optimisation usually finds only local minima, which may not necessarily be globally the best optimisation result. Therefore, extensive calculations should be carried out before a final decision on the compressor design is made.

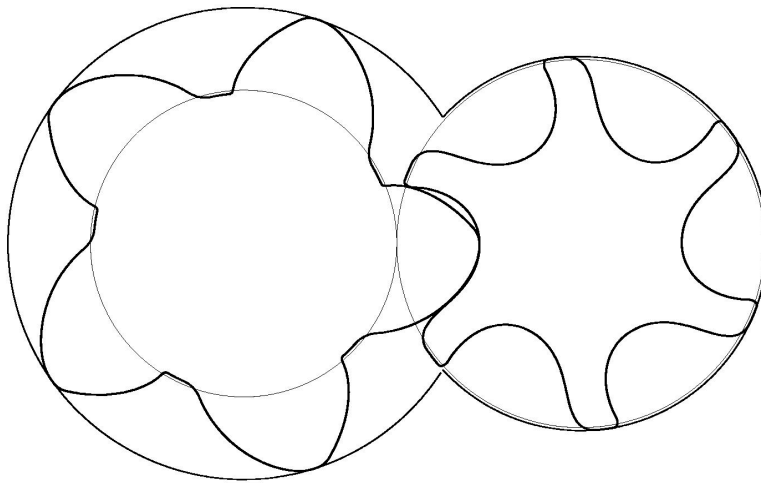


Fig. 4 Rotor profile optimized for an oil-flooded air compressor

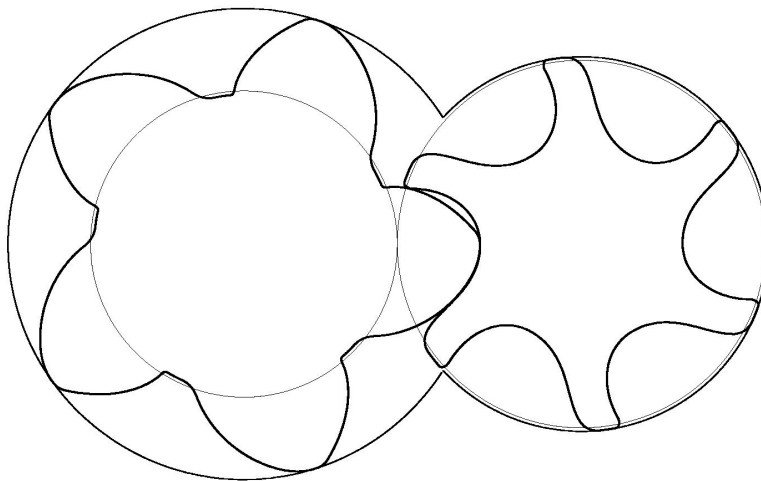


Fig. 5 Rotor profile optimized for a refrigeration compressor duty

	DryAir	Oil-Flooded Air	Refrig
r_0 [mm]	2.62	0.74	0.83
r_1 [mm]	19.9	17.8	19.3
r_2 [mm]	6.9	5.3	4.5
r_3 [mm]	11.2	5.5	5.2
Built-in volume ratio	1.83	4.1	3.7
Rotor speed [rpm]	7560	3690	3570
Oil flow [lit/min]	-	12	8
Injection position [°]	-	65	61
Oil temperature [°]	-	33	32

The dry air compressor was chosen for further analysis. This is because the compression process within it is close to that of an ideal gas compressed adiabatically in which γ , the isentropic exponent, has the relatively large value of approximately 1.4.

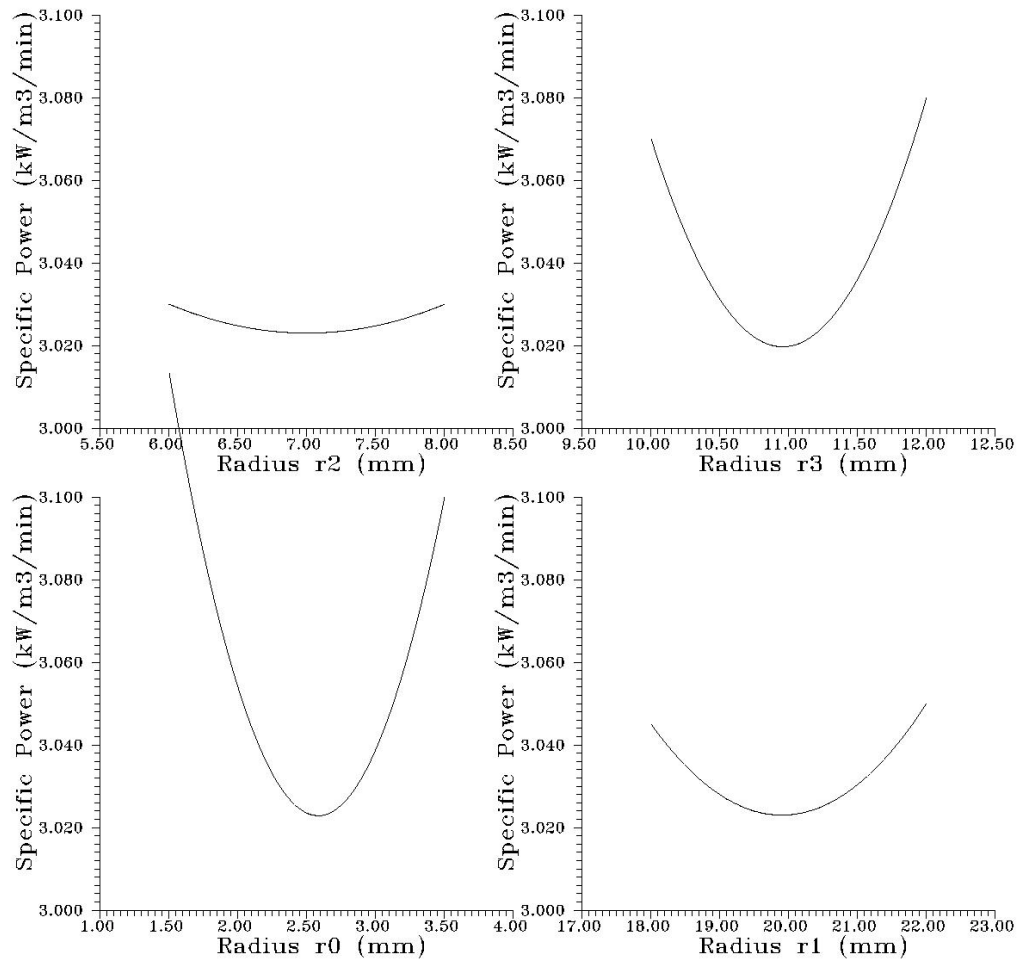


Fig. 6 Variation of the compressor specific power as function of the compressor rotor parameters

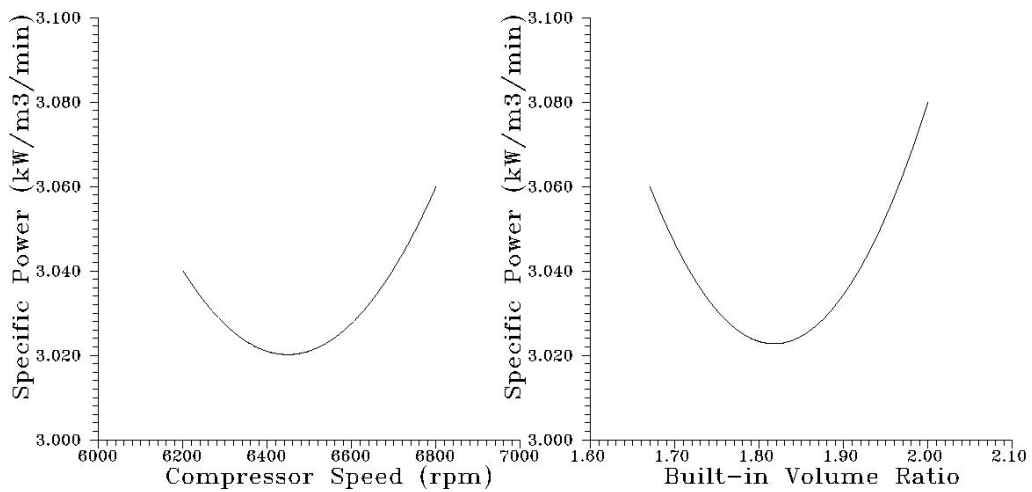


Fig. 7 Specific power as function of the compressor built-in volume and speed

As an example of how the optimisation variables influence the compressor specific power, the radii r_0 - r_3 are considered. The influence of the gate rotor tip addendum r_0 , and the gate rotor radius r_3 are presented in Fig. 6, as

well as the main rotor radii r_1 and r_2 . In Fig. 7, the influence of the compressor built-in volume ratio, as well as compressor speed is presented. If the gate rotor addendum is analysed in detail, it can be concluded that, the size of the rotor blow-hole area is proportional to the addendum. Therefore r_0 should be made as small as possible in order to minimise the blow-hole. It would therefore appear that ideally, r_0 should be equal to zero or even be 'negative'. However, reduction in r_0 also leads to a decrease the fluid flow cross-sectional area and hence a reduction in the flow rate and the volumetric efficiency. It follows that there is a lower limit to the value of r_0 to obtain the best result. More details of single variable optimisation of screw compressor rotors can be found in *Hanjalic and Stosic, 1994*

CONCLUSIONS

A full multivariable optimisation of screw compressor geometry and operating conditions has been performed to establish the most efficient compressor design for any given duty. This has been achieved with a computer package, developed by the authors, which provides the general specification of the lobe segments in terms of several key parameters and which can generate various lobe shapes. Computation of the instantaneous cross-sectional area and working volume could thereby be calculated repetitively in terms of the rotation angle. A mathematical model of the thermodynamic and fluid flow process is contained in the package, as well as models of associated processes encountered in real machines, such as variable fluid leakages, oil flooding or other fluid injection, heat losses to the surrounding, friction losses and other effects. All these are expressed in differential form in terms of an increment of the rotation angle. Numerical solution of these equations enables the screw compressor flow, power and specific power and compressor efficiencies to be calculated.

A rack generated profile in 5/6 configuration rotors of 106 mm was used as an example to show how optimisation may permit both better delivery and higher efficiency for the same tip speed. Several rotor geometrical parameters, namely the main and gate tip radii, as well as the compressor built-in ratio and compressor speed and oil flow and temperature and injection position are used as optimisation variables and applied to the multivariable optimisation of the machine geometry and its working parameters for a defined optimisation target. In the case of the example given, this was minimum compressor specific power. It has thereby been shown that for each application, a different rotor design is required to achieve optimum performance.

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