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On performance optimisation for oil-injected screw compressors using different evolutionary algorithms

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Abstract. Energy consumed for pressurizing air makes a significant proportion of total electrical energy consumption worldwide. To reduce the carbon footprint, it is necessary to have air compressors, which can operate efficiently over a large range of pressures and flow including full load and part load conditions. Several studies have been performed in this area including some which monitor the performance of a large number of compressors to develop strategies for their designs.

This paper focuses on the design optimisation of geometrical and oil parameters of oil-injected screw compressors using different evolutionary algorithms such as genetic algorithm (GA), covariance matrix adaptation evolution strategy (CMA-ES), and so on. A comparison of the performance of these algorithms is presented. SCORG and GT-SUITE (commercial software tools for screw compressor thermodynamic simulations and optimisation) are used in the integrated model producing promising results. Feasibility of the optimum outcomes generated by these algorithms is critically evaluated from machine and system design point of view. Finally, in the context of optimisation presented here, the simplex converges fastest as compared to other algorithms. In the future study, the system design limitations are to be incorporated as constraints for the optimisation along with the objective to improve energy efficiency.

Nomenclature

P	pressure	DIOIL	oil port diameter
Q	flow	FIOIL	oil injection angle
SPC	specific power consumption	TOIL	oil injection temperature
ETAK	adiabatic efficiency	POIL	pressure in oil reservoir
ETAV	volumetric efficiency	VISCOIL	oil viscosity
A	flow area	t	time
γ	adiabatic constant	ρ	density
v	fluid velocity	m	mass
ζ	leakage flow resistance coefficient	$\dot{m}$	mass flow rate
P_{hp}	high pressure	P_{hp}	low pressure
h_{in}	fluid enthalpy (inflow)	h_{out}	fluid enthalpy (outflow)

1. Introduction

A screw compressor is a rotary positive displacement machine preferred for relatively low discharge pressure and a wide range of flow rates than reciprocating, centrifugal, and axial compressors. It is a compact, reliable, and cost-effective machine consisting of a pair of helical rotors with four to six bearings enclosed in a casing. It has been categorized as oil-flooded, oil-free, and water-injected screw compressors. Among all these, the oil-flooded holds the majority percentage of share in the market due to fewer mechanical components, better efficiency and the ability to achieve higher pressure in a single stage itself.

Data given by the US EIA (Energy Information Administration), International Energy Outlook 2021, shows that the projected world energy consumption will increase by 50% from 2019 to 2050 [1]. The majority increase is in the energy produced by burning fossil fuels, with the least in renewable energy. The burning of fossil fuels by the industry leads to the production of carbon dioxide, which contributes nearly about (60-70)% to the greenhouse effect. The emission of greenhouse gasses, in turn, leads to global warming. On the other hand, for a screw compressor, statistical data reveals that in a life-cycle cost, the majority cost is undertaken by the operating cost, which is the power cost. Therefore, from an engineering point of view, the aim should be to reduce the compressor power consumption or increase its efficiency. This is only possible by optimising the screw compressor parameters to their maximum with an aim to improve the performance and reduce power consumption, thereby making them more environment friendly.

Referring to the various kinds of literature in this field, an optimisation of a refrigeration twin screw compressor considering the rotor geometrical parameters such as the wrap angle, relative length (length to diameter ratio) and the slide valve parameters using the computer program has been presented by Fleming et al. (1994) [2]. The optimum rotor geometrical parameters are not only important from the compressor efficiency point of view but also for other factors such as female rotor deflection, bearing loads and the rotor contact force. You et al. (1996) talks about the optimisation of lobe combination, relative length and the wrap angle on the factors mentioned above [3]. Stosic et al. (1997) describes the rotor profile generation. It makes use of the conjugacy conditions when solved explicitly and enables a variety of primary arcs to be defined that allows it to generate a higher efficiency profile [4]. A breakthrough in the optimisation of screw machines using evolution strategy (ES) has been carried out by Berlik et al. (2002) [5]. The limitations of using the ES in optimisation of screw machines has been discussed along with how the unsuitability of the penalty function method led to the extended offspring generation scheme. The problem in optimisation is the number of calculations to be performed in order to reach the optimum and whether this optimum is global optimum or not. Some of the common optimisation methods are listed below:

- Steepest descent: It is often known as gradient descent which is a first order iterative optimisation algorithm for obtaining the local minima of a differentiable function
- Newton's method: It is an iterative method for obtaining the extrema (minima or, maxima) of a differentiable function similar to the steepest descent method.
- Davidon-Fletcher-Powell's method: This method is used to obtain the solution to the secant equation closest to the current estimate satisfying the curvature condition. This method requires both the target function and its gradient.
- Random search: It is a family of numerical optimisation methods that do not require the gradient of the function. It can be used for functions that are not continuous or differentiable. It is often known as direct search, derivative-free or, black box method.

- Grid search: This method is much similar to the random search method that allows to select the best parameters from a range of values thus automating the trial and error method.
- Search along coordinate axes: It is a numerical optimisation method that restricts the search direction to the coordinate axes of the input space alone.
- Powell's method: This is used for obtaining the local minima of a real valued function that need not be differentiable. It passes a set of initial search parameters which are simply the normals aligned to the axis.
- Hooke-Jeeves method: It is also known as the pattern search method which keeps the track of the travel direction as the search propagates from point to point. It doesn't need to start over from the origin at each new search point.

Among all the methods, genetic algorithms are widely used which require only the value of the target function and can handle discontinuous functions as well. The only disadvantage is the slow converging nature due to its small seed size. Apart from this box complex method is also convenient which also requires only the function value and not the gradient. The disadvantage of this method is that it is less suitable for discrete parameters. Stosic et al. (2003) utilizes the box complex method for the optimisation of the compressor shape, size, dimension and operating parameters [6].The application of NURBS (Non-uniform rational B-splines) by Hauser et al. (2008) improves optimisation for the complex geometry machines that include profile variation of screw compressors [8]. On the other hand, Herlemann et al. (2014) highlight the use of a chamber model generator and simulation tool for the optimisation of rotor profiles with the same number of lobe combinations to minimize the energy costs that can be validated experimentally [9]. Last but not least, Zhang et al. (2019) show the structural parameter optimisation of the twin-screw expander and the influence of geometrical parameters on the thermal performance of the machine. Considering a single-objective function and using the sequential quadratic programming algorithm for optimisation the influence of leakage on its functional force is also studied [10].

In light of these considerations, in this work, a SCORG [11] model is developed for the oil-flooded screw compressors, which is validated from the experimental testing data performed at City University. This model has been linked to the optimisation tool, i.e., GT-SUITE from Gamma Technologies [12]. Design space exploration has also been used to find sensitive and important design variables. After identifying significant variables, the formal optimisation using different evolutionary algorithms has been carried out. The post optimality analysis has also been performed to determine the feasible outputs. Finally, a comparative analysis to find out the best-suited method for optimisation producing realistic outputs in a short span of time has been presented.

2. Modelling and optimisation

The machine considered in this work is a 22 kW air screw compressor with built-in volume ratio of 4.6 operated at a pressure ratio of 8.5 and a speed of 2000 rpm, manufactured at Kirloskar Pneumatic Company Limited, Pune (India). Since, being an industrial screw compressor, information about the size and profile cannot be disclosed. However, the normalized performance of the given screw compressor block has been shown in the table-3 for visualization of the improvement after optimisation. Experimental testing has been conducted for this compressor, whose data will be used for modeling the screw compressor in the SCORG (Screw Compressor Rotor Grid Generation) software from the PDM Analysis.

The SCORG is an industry-leading grid generation and performance prediction software for positive displacement screw machines. It is used to design, analyze, and optimise screw machines [11]. All the input parameters as per the compressor specifications have been fed into the SCORG, and then using the multi-chamber model in the background, it evaluates the thermodynamic performance, which is in good agreement with the experimental data. For optimisation, the SCORG offers an option of design exploration, which is basically used for observing the effect of parameters, and then the critical parameters can be used for further optimisation in-direct coupling with the GT-SUITE software from GAMMA Technologies [12].

GT-SUITE is a transient multi-physics tool used to pre-process, build, simulate, and post-process the data using the GT-POST interface. It is uniquely positioned to model the positive displacement pumps and compressors for several applications. This work focuses on the design space exploration and optimisation using the design optimiser available in the GT-ISE build interface.

2.1. Thermodynamic multi-chamber model

SCORG thermodynamic module performs thermodynamic calculation based on the multi-chamber model using equations of conservation of mass and internal energy applied to control volumes. The boundary conditions consist of suction gas pressure and temperature, the discharge gas pressure and oil injection parameters along-with geometry values of the chamber volume, port and leakage areas. Based on the multi-chamber model, the integral parameters such as indicated power, leakage flows, mass flow rate, volumetric and adiabatic efficiencies are computed [11].

Depending on the rotor position, the working chamber formed between the meshing rotor pair and the housing changes the shape and size with time. This chamber is itself connected to the suction and discharge chambers through flow areas which vary with time both in shape and size. The schematic view of a screw machine is shown in figure-1 [11].

According to the mass conservation law, the mass variation in the control volume is given as:

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{1}$$

The mass inflow into the control volume consists of the suction flow when the chamber is connected to this port, the flow of injected fluid and the leakage flow which enters the control volume.

$$\dot{m}_{in} = \dot{m}_{suc} + \dot{m}_{inj} + \dot{m}_{lg} \tag{2}$$

The mass flow of the leakage flow through the clearance gap is calculated by simplified continuity equation assuming constant temperature which is given as [7]:

$$\dot{m}_{lg} = \rho \dot{V}_{cg} = \rho w_{cg} A_{cg} = A_{cg} \sqrt{\frac{\gamma (p_{hp}^2 - p_{lp}^2)}{a^2(\zeta + \log p_{hp}/p_{lp})}} [kg/s] \tag{3}$$

The mass outflow from the control volume consists of the discharge flow and the leakage flows.

$$\dot{m}_{out} = \dot{m}_{dis} + \dot{m}_{ll} \tag{4}$$

The above mentioned mass flow rates satisfy the continuity equation.

$$\dot{m} = \rho * v * A \tag{5}$$

The conservation of energy within the control volume, neglecting the kinetic energy of the fluid (as the velocity of the fluid within a screw machine is relatively low) is given as:

$$\frac{dU}{dt} = \dot{m}_{in} h_{in} - \dot{m}_{out} h_{out} + \dot{Q} - p\frac{dV}{dt} \tag{6}$$

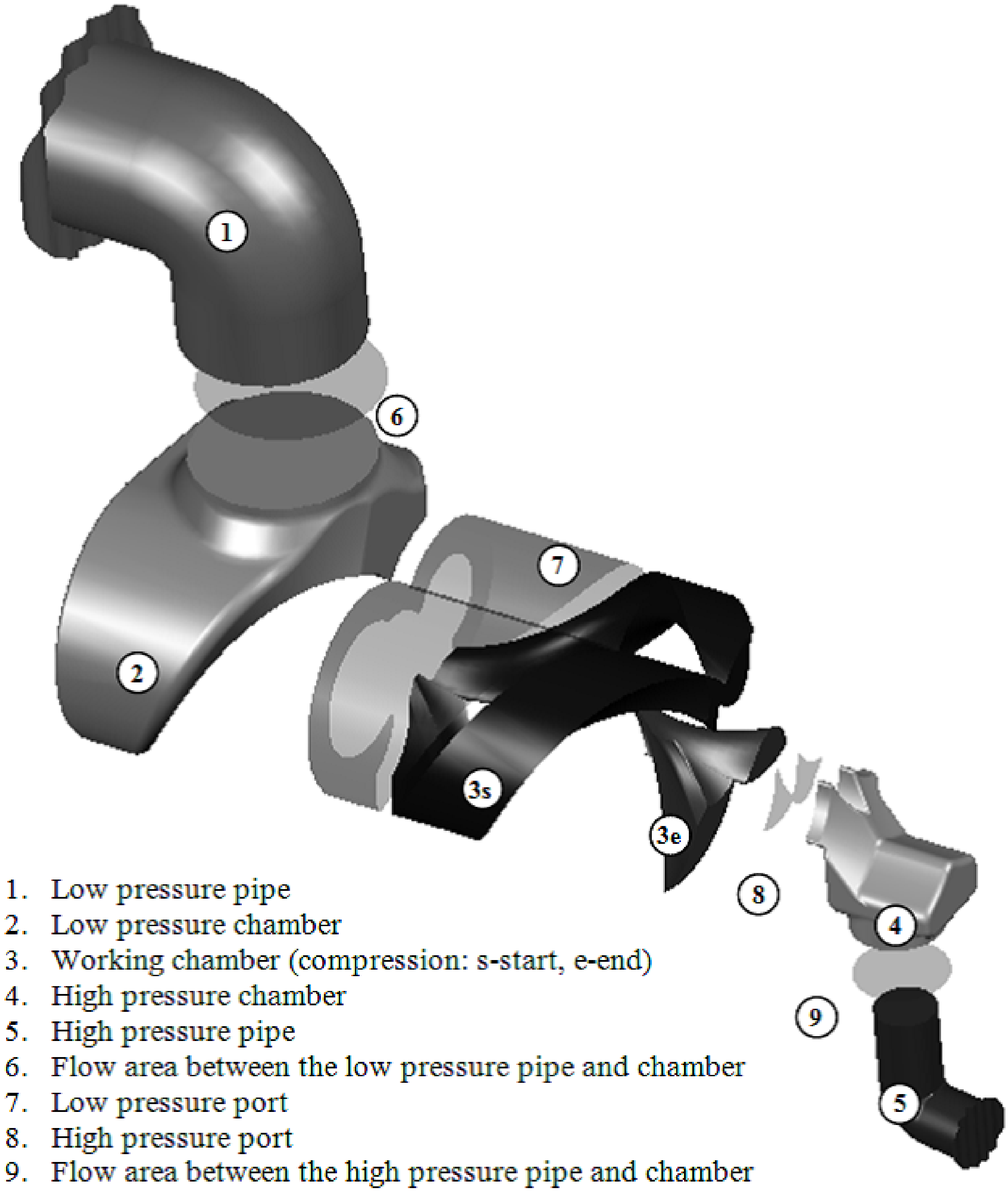

Figure 1. Configuration of a screw machine

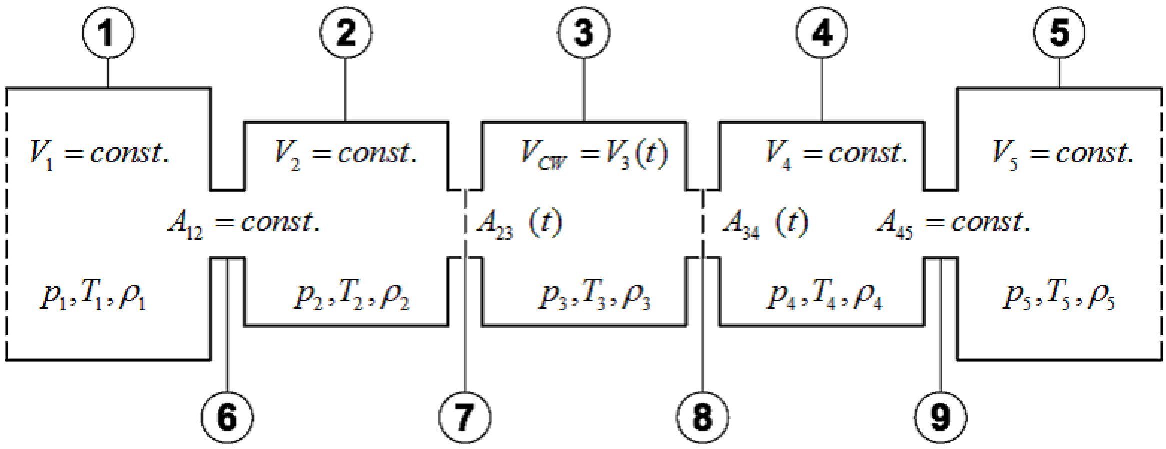

Figure 2. Schematic view of a screw machine chamber configuration

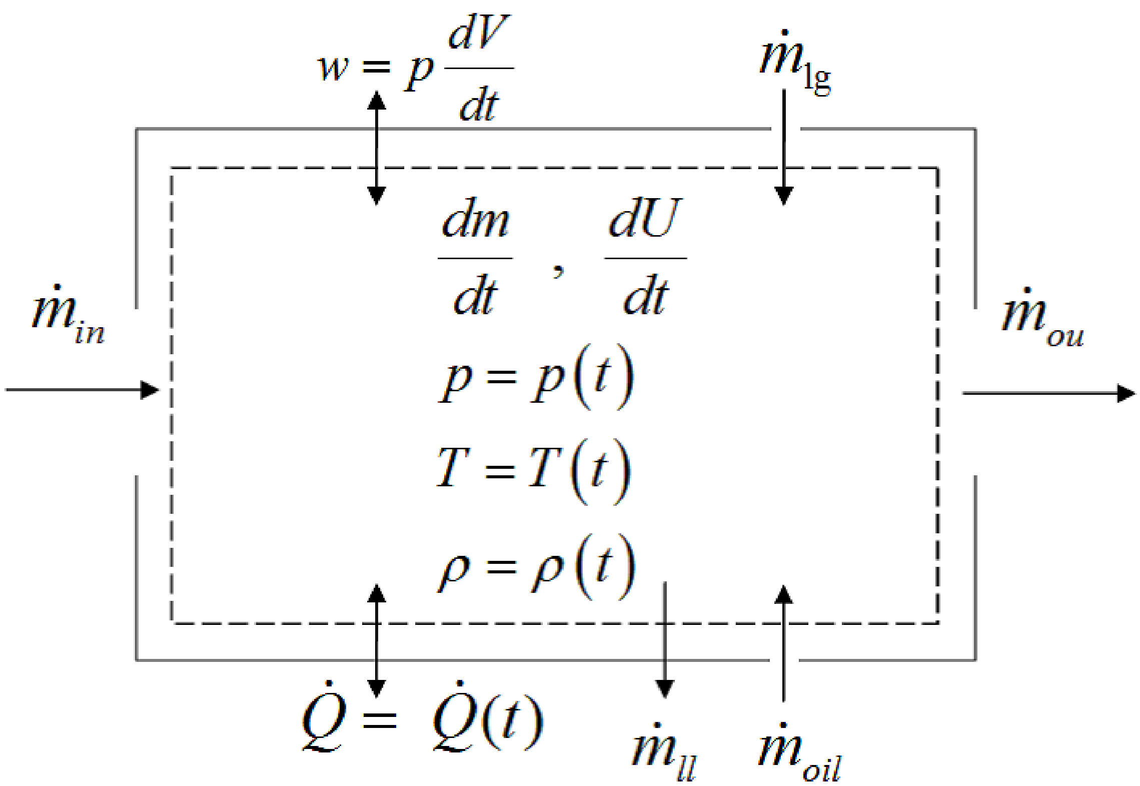

Figure 3. Control volume of the main chamber for thermodynamic analysis

Where,
$\dot{m}_{in}$ = Mass flow rate (inflow)
$\dot{m}_{out}$ = Mass flow rate (outflow)
$\dot{m}_{suc}$ = Mass flow rate (suction)
$\dot{m}_{dis}$ = Mass flow rate (discharge)
$\dot{m}_{inj}$ = Mass flow rate (injected fluid flow)
$\dot{m}_{lg}$ = Mass flow rate (leakage flow entering the control volume)
$\dot{m}_{ll}$ = Mass flow rate (leakage flow leaving from the chamber)
$\dot{V}_{cg}$ = Volume flow rate (clearance gap)
$\dot{Q}$ = Heat transfer through the control volume boundary
$p\frac{dV}{dt}$ = Thermodynamic work
On solving the above differential equation the mass and internal energy within the chamber are obtained. Further using the equation of state for an ideal gas all other thermodynamic properties can be calculated.

2.2. Optimisation algorithms
The GT-ISE build interface offers different types of evolutionary algorithms for optimisation, which are described in the below section:

- Simplex Search: The Nelder-Mead simplex algorithm is a deterministic local search algorithm used to obtain the maxima or minima of the given objective function in a multidimensional space. For example: when working with two influencing variables it simply involves comparing the function values at the three vertices of a triangle. If more variables are taken into account then corresponding geometric shape is considered and thus it is n-dimensional simplex. When using this algorithm, the factor resolution must be 3% or less. This method is only partially parallel capable.
- Genetic Algorithm: An evolutionary global search algorithm is recommended for medium to high complexity problems. The algorithm calculates the fitness value followed by selection, crossover, and mutation.
- Accelerated Genetic Algorithm: It is the modified version of the genetic algorithm with a faster convergence rate. It finds the better solutions in fewer total design iterations.
- Covariance Matrix Adaptation Evolution Strategy (CMA-ES): CMA-ES is also an evolutionary algorithm as the genetic algorithm. In this evolution, strategies are stochastic, derivative-free for numerical optimisation of non-linear or non-convex continuous optimisation problems.

The GT-SUITE design optimiser interface also offers other two optimisation algorithms, such as the Brent and discrete grid methods. A discrete grid is a bisection search algorithm that iteratively reduces the search space by a factor of 2 until convergence is reached. On the other hand, the Brent algorithm is based on the root-finding algorithm. Both are not recommended as the simplex algorithm is expected to perform better for most problems. Therefore, here in this work, for a comparative analysis of optimisation, only the above four listed algorithms are used as they perform better and are recommended by the Gamma Technologies [12].

2.3. Design approach
GT-SUITE design optimiser interface allows two different design approaches for multi-objective optimisation, which are as follows:

- Pareto-design: This approach is used to maximize, minimize or target an output response and study its competing objectives. It gives many optimal designs considered equally optimal and not a single best design.

- Weighted-sum approach: This approach combines the multiple objectives into a single-objective function by assigning a weight function to each output. An example of how this work is shown below:

Let,
P_1= First objective function which is to be maximized
P_2= Second objective function which is to be minimized
The combined single-objective function as per weighted-sum approach is given as:

$$F = \frac{P_1}{P_{norm1}} - \frac{P_2}{P_{norm2}} \tag{7}$$

where, P_{norm1} and P_{norm2} are the normalized value for each output function so that each output function is of the same order of magnitude when combined into a single-objective function. If the values are not normalized, the single-objective function will be dominated by the output function with a larger magnitude of weight function.

In this work, the weighted-sum design approach is preferred for multi-objective optimisation with equal weights so that each output function is given equal importance which is better for comparison of different algorithms.

2.4. Input and output functions

In an air screw compressor, as the pressure rises, the air temperature in the compression chamber also increases. This can be better understood with an example: in an oil-free twin screw air compressor operating at a pressure ratio of 6 and running at 6000 rpm (approx.) will result in an average discharge temperature of more than 320 °C. This temperature can be substantially reduced by injecting oil. That is why oil plays an important role in cooling the compressor chamber. Apart from this, oil also serves in improving the volumetric efficiency by sealing the leakage gaps and providing lubrication to both the rotors as well as the bearings [13].

According to the thumb rule used in any screw compressor industry, the oil contribution by mass to cooling, sealing and lubrication should be in the proportion of 100:10:1, respectively. Injecting oil has many advantages but it should be injected in the right quantity otherwise injecting more or less amount of oil would result in additional power losses. In the figure-4 it shows the oil injection parameters along with the expected effects. The box colour depicts the influence of parameters on the compressor performance. The darker the color intensity the higher is the influence on the performance [13].

The input and output variables need to be specified to optimise the problems. The input parameters considered in this work for both multi-objective and single-objective optimisation along with the effects are listed below:

- Oil port diameter (mm): It determines the flow rate of oil. The heat exchange between gas-oil depends on the mass flow rate of the oil injected. Increased oil mass flow rate can reduce discharge temperature and increase the compressor's volumetric efficiency and adiabatic efficiency. However, excess oil can lead to power losses during injection and transport through compression chambers as its density is several magnitudes greater than the gas.
- Oil injection angle (°): If the oil injection port is positioned so that the oil injection temperature is close to the gas temperature, there will be low convection between gas and oil. If the oil injection port is positioned close to the discharge port, then the residence time for oil will be low, and heat transfer between air and oil will not largely benefit the compression chamber. Therefore, optimal positioning of oil injection will help in controlling the compression chamber temperature.
- Oil injection temperature (°C): A low oil injection temperature is preferred for better heat exchange but cannot be achieved without the high load on the cooling system.

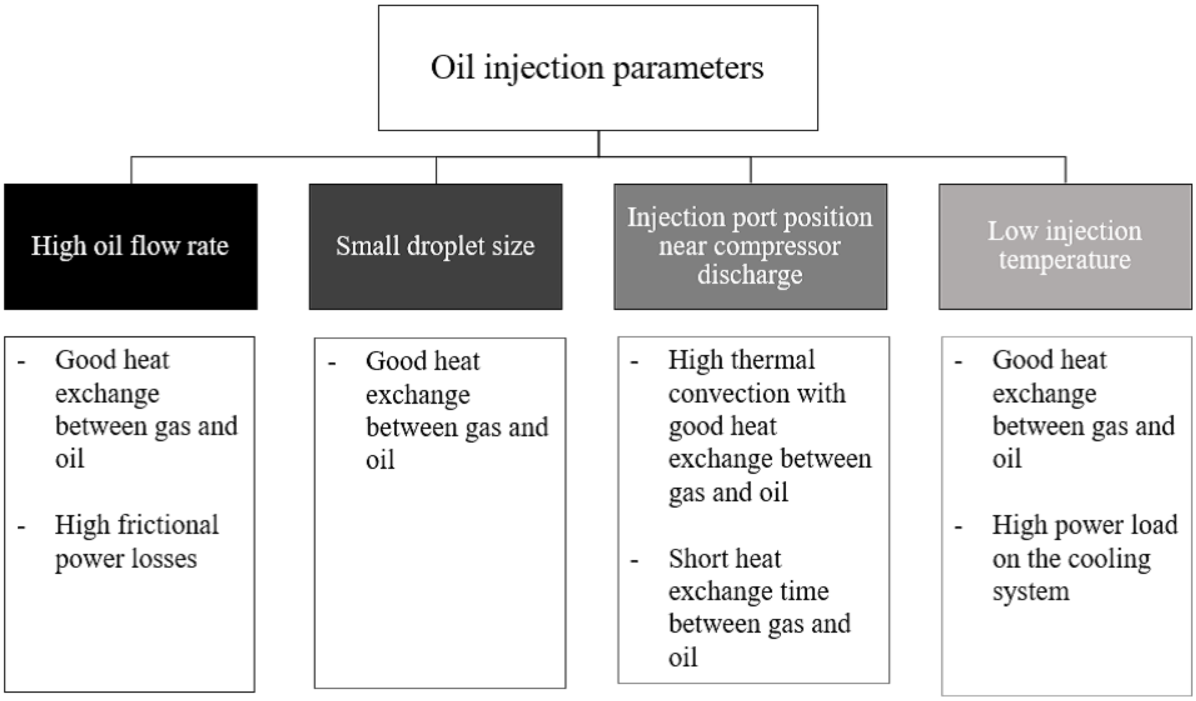

Figure 4. Various oil-injection parameters with their expected effects

- Pressure in the oil reservoir (bar): It affects the oil droplet size. Higher pressure means small oil droplet size which means a high surface area for exchanging heat with the hot gas, ultimately resulting in lower temperature in the compression chamber
- Oil viscosity (m^2/s): If the oil viscosity is low then the compressor experiences metal to metal contact leading to component failures whereas on the other hand if the oil viscosity is too high the compressor will operate less efficiently thus generating excessive heat and more power losses.

The above listed input parameters affect the temperature in the compressor chamber which directly affects the performance of the compressor. Therefore, for this performance optimisation oil parameters are taken into consideration as the temperature in the chamber is directly dependent upon the interaction between gas and oil.
The output function for multi-objective optimisation are as follows:

- Flow (m^3/min) : Mass flow rate is the mass of gas delivered per unit time. It is obtained from the mass of the gas in the working chamber just before opening the discharge port multiplied by the rotational speed and number of cycles per revolution of the male rotor.

$$\dot{m} = mz_1 \frac{n}{60} [kg/sec] \tag{8}$$

The volume flow rate is always normalised to the suction conditions.

$$\dot{V} = \frac{60\dot{m}}{\rho_o} [m^3/min] \tag{9}$$

- Specific power consumption ($kW/m^3/min$) : Specific power is power required to deliver a unit quantity of mass of the gas . It is calculated as the ratio of the compressor power and volume flow rate.

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$$SPC = \frac{P}{\dot{V}}[kW/m^3/min] \tag{10}$$

- Volumetric efficiency (%) : Volumetric efficiency is the ratio of the volume flow rate and the theoretical capacity of the machine.

$$\eta_v = \frac{\dot{V}}{\dot{V}_t} * 100[\%] \tag{11}$$

- Adiabatic efficiency (%) : Adiabatic efficiency is the comparison of actual indicated power and theoretical adiabatic power for the compression.

$$\eta_{ad} = \frac{P_{ad}}{P} * 100[\%] \tag{12}$$

The output function for single-objective optimisation is specific power consumption of the oil-injected twin screw air compressor.

The range of input parameters as per general operating conditions of a typical screw compressor are listed below in the table-1.

Table 1. Range of input parameters

Input parameters	Range
DIOIL	(1-12) mm
FIOIL	(35-130) °
TOIL	(35-80) °C
POIL	(3.5-8) bar
VISCOIL	(1-8)*E-06 m^2/s

2.5. SCORG_GT-SUITE Integration model

The SCORG model for the 22 kW oil-injected twin screw air compressor is prepared using the geometrical and design input parameters which is validated from the experimental data.

The GT-SUITE model has been set up in the GT-ISE interface using the SCORG batch file for the 22 kW machine. Once the model has been linked to the SCORG thermodynamics, the design optimiser in the GT-SUITE can be activated. The simulation is made to run iteratively in the design optimiser with the parameters values changing each time until the stopping criteria is satisfied. The results can be viewed and post analysis can be carried out in the GT-POST interface. The flow chart showing the complete process of SCORG and GT-SUITE integration is shown in figure-5.

3. Results

3.1. Single-objective optimisation

The simulation results from the GT-SUITE_SCORG model are presented in this section. In the single-objective optimisation using the different evolutionary algorithms, a comparative analysis has been presented. An important thing to note is that all the different algorithms

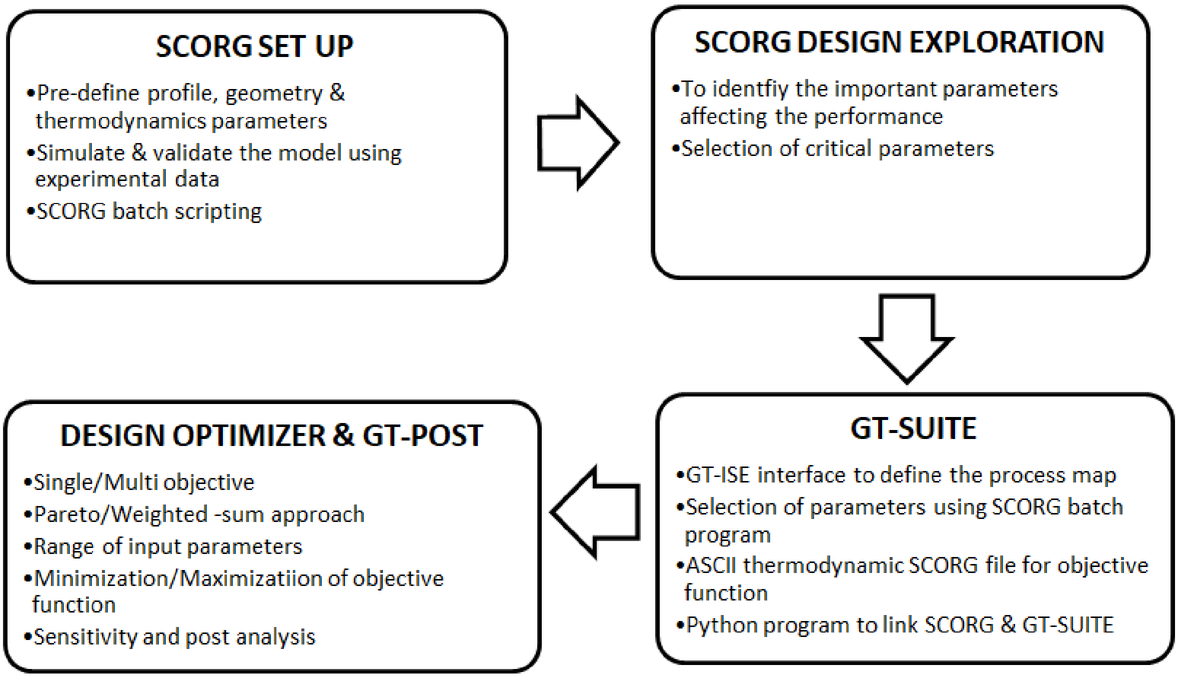

Figure 5. SCORG_GT-SUITE process map

for optimisation are simulated independently with the same input range for variables and the same objective function. For a single objective optimisation the specific power consumption is considered. The input parameters are the same for both the multi-objective and single-objective optimisation so that a proper conclusion can be obtained and the results are validated.

From figure-6 it can be seen that the change in specific power consumption for GA is around 10.8 % , for simplex is 10.5%, for CMA-ES is 9.6% and for accelerated GA is 9.2% respectively.

The time elapsed for each search algorithm is shown in figure-7. Simplex method takes the least time whereas the GA takes the highest time for single objective optimisation.

3.2. Multi-objective optimisation

In this section, a comparative analysis using the different evolutionary algorithms are presented for each objective function. Figure-8 shows the percentage change in specific power consumption for different algorithms with GA showing the highest variation of 10.8% and the accelerated GA with the lowest of 9.2% which is validated from single-objective optimisation.

It is also evident from figure-9 and 10 that the change in flow and the volumetric efficiency is maximum for GA and the least for accelerated GA. Apart from the thermodynamics performance analysis a comparative study of the elapsed time is also presented in figure-11. Here, it can be seen that the simplex search algorithm takes the least time for multi-objective optimisation and the GA takes the highest time for predicting the optimum output. This depends on the step-length and the seed size taken during the optimisation.

The seed size taken here for optimisation using the evolutionary algorithms for both the single-objective and multi-objective are listed below in the table-2.

The fitness function determines how fit an individual is (the ability of an individual to compete with other individuals). It gives a fitness score to each individual. The probability that an

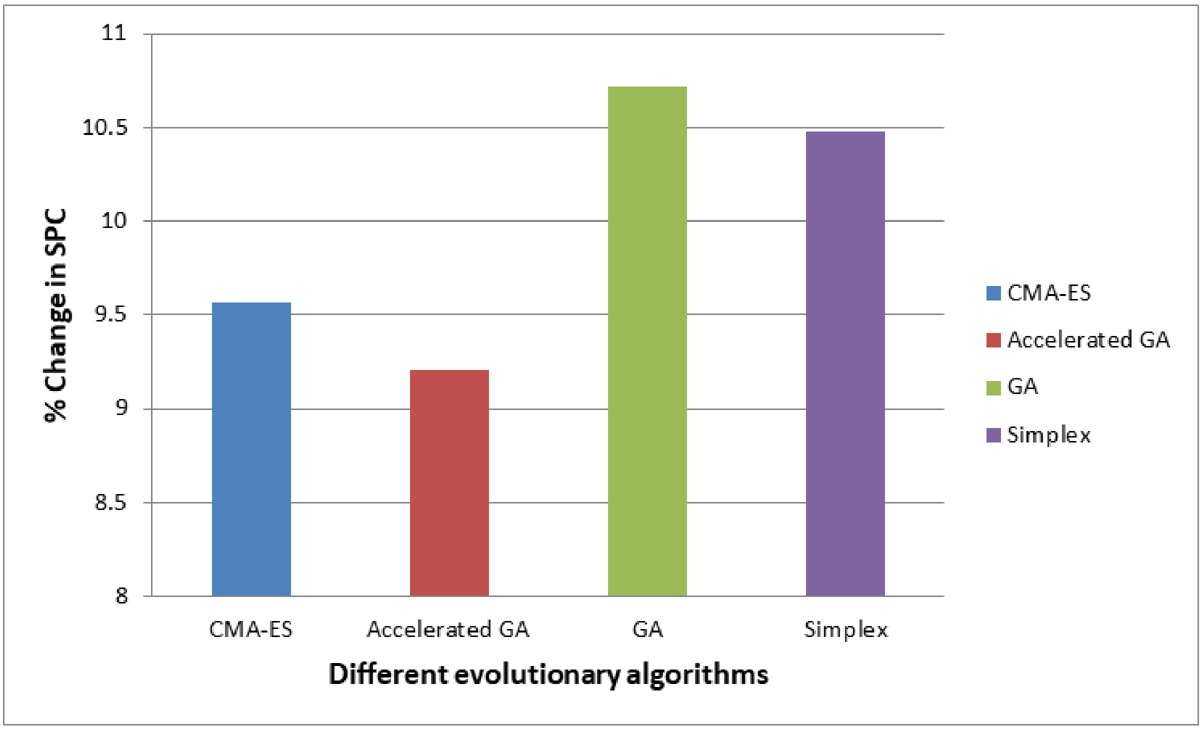

Figure 6. % Change in SPC (Single-objective optimisation)

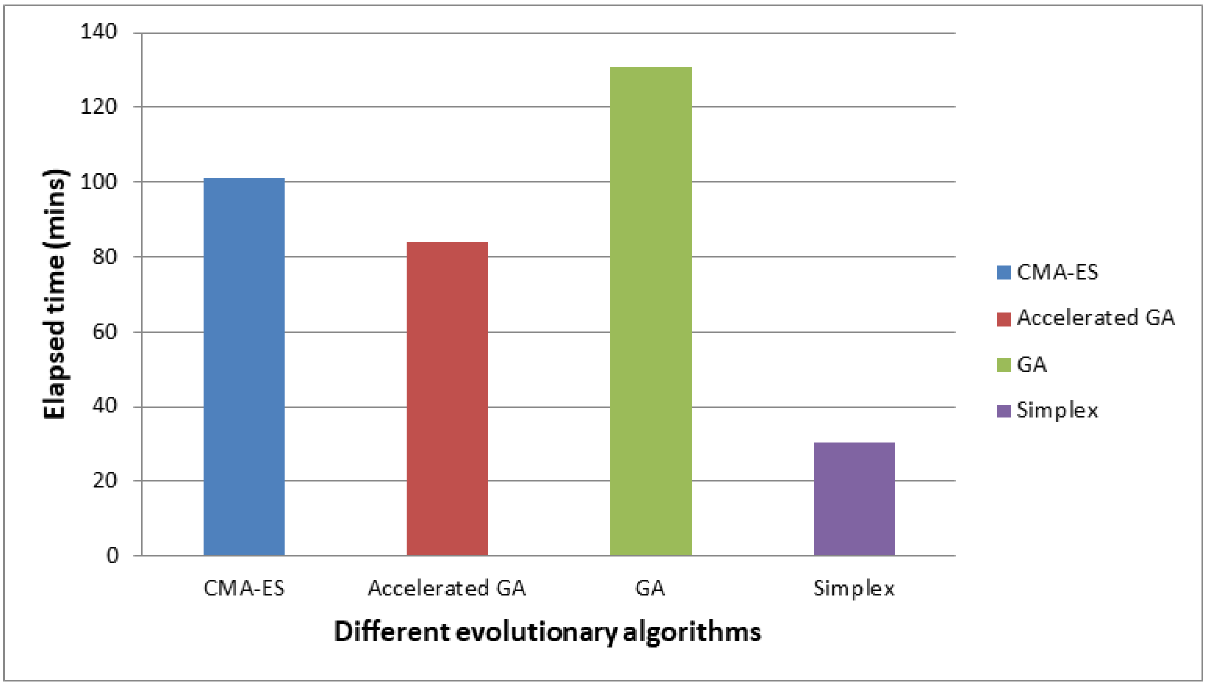

Figure 7. Time elapsed for single-objective optimisation

individual will be selected for reproduction is based on its fitness score. Functions with higher fitness value are recommended. Also, the fitness value is inversely proportional to the seed size.

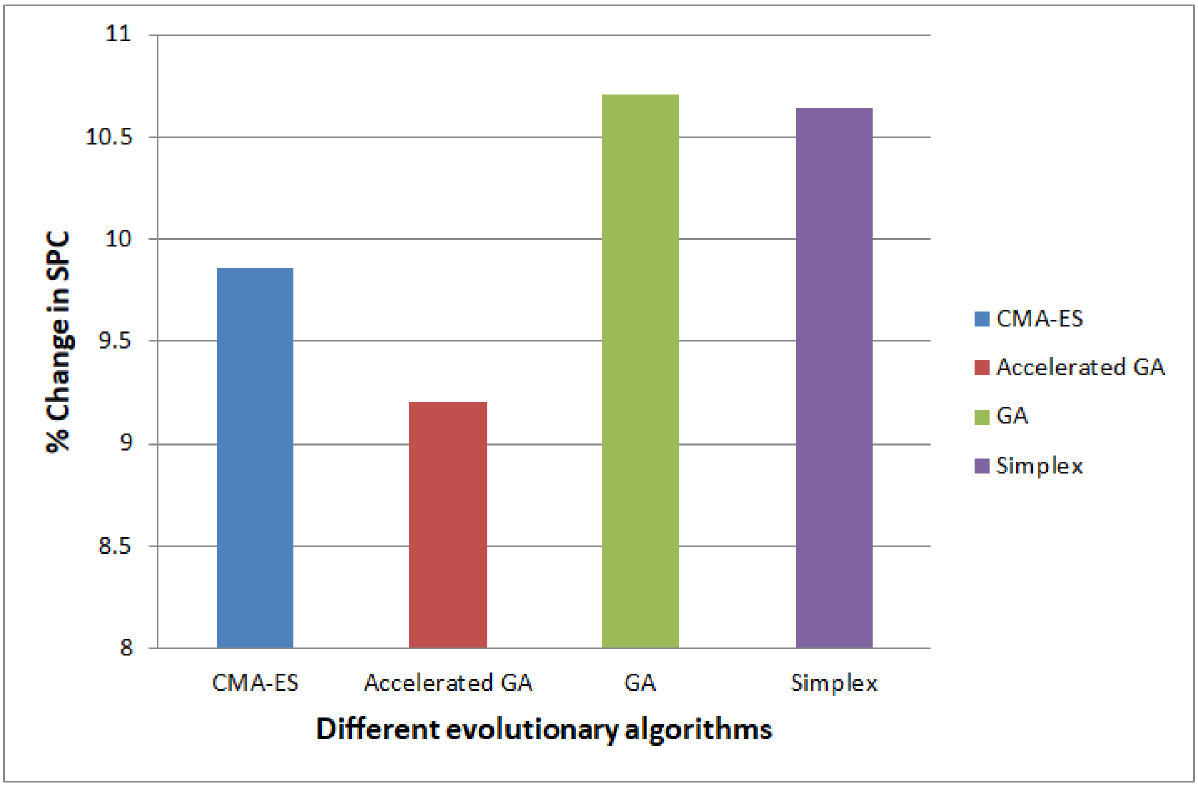

Figure 8. % Change in SPC (Multi-objective optimisation)

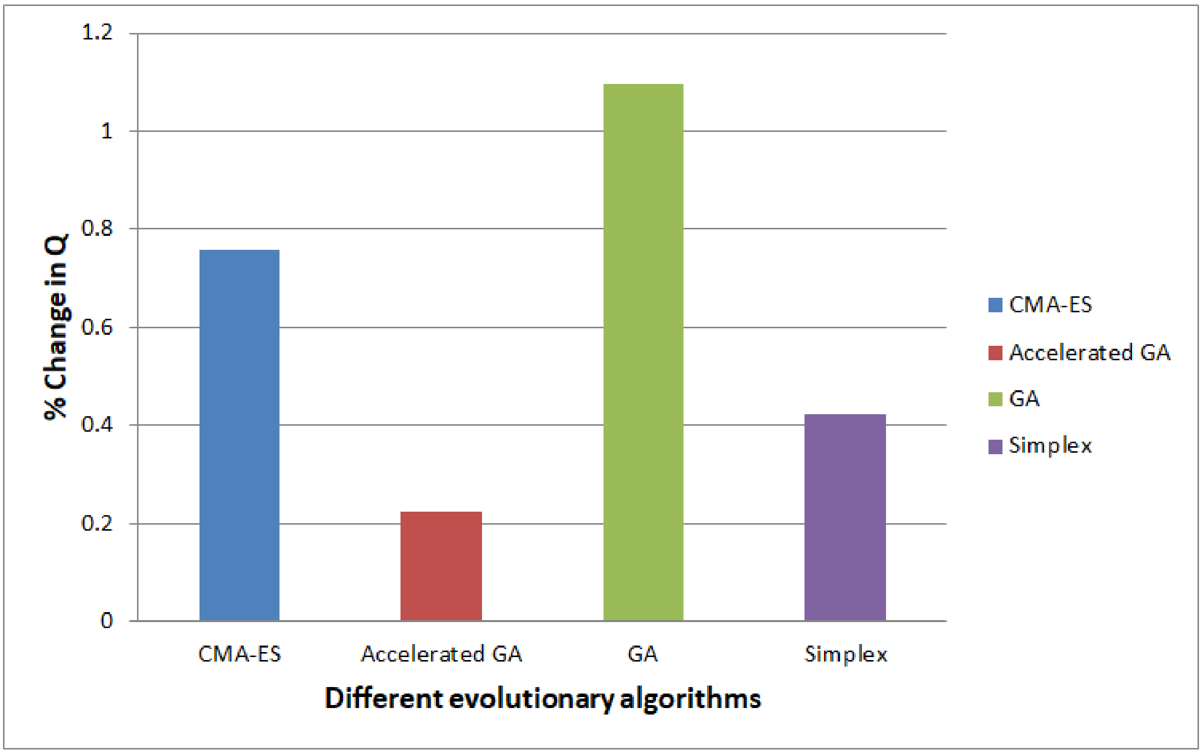

Figure 9. % Change in Q (Multi-objective optimisation)

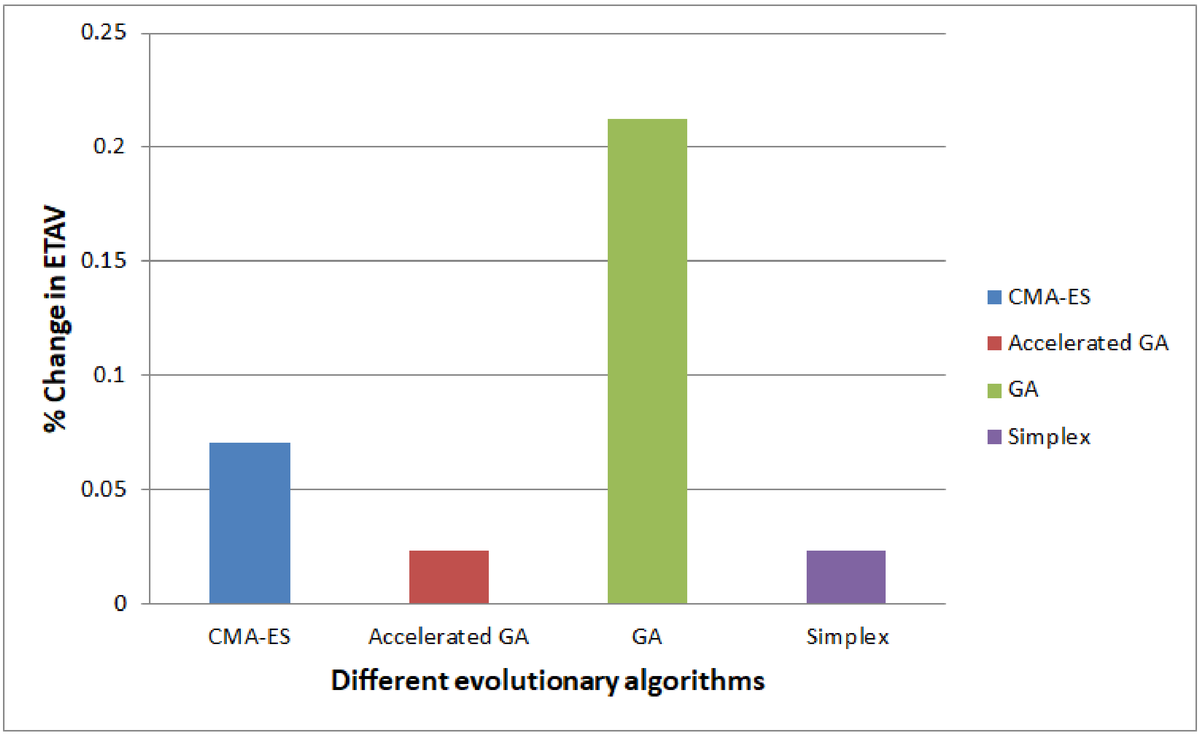

Figure 10. % Change in ETAV (Multi-objective optimisation)

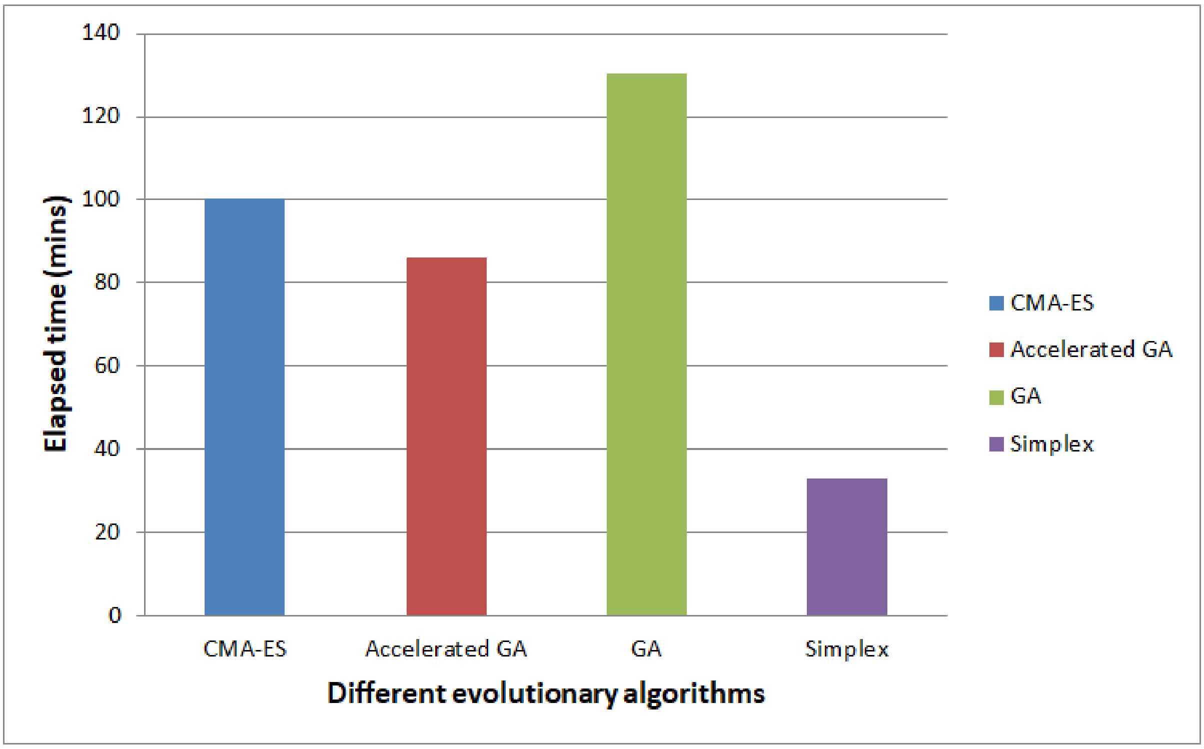

Figure 11. Time elapsed for multi-objective optimisation

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Table 2. Seed size for optimisation

Search algorithm	Seed size
GA	0.033005270
CMA-ES	0.10439938
Accelerated GA	0.24159176

Therefore, the one with the smallest seed size will have the highest fitness value. In this case, the GA will have the highest fitness value followed by the CMA-ES and then the accelerated GA.

Table 3. Tabular column showing normalised performance map

$MODEL$	SPC	Q	$ETAV$
KPCL MODEL	1.120	0.989	0.998
CMA-ES	1.009	0.997	0.999
ACC. GA	1.017	0.991	0.998
GA	1.000	1.000	1.000
SIMPLEX	1.001	0.993	0.998

The normalized performance map after the optimisation is shown in table-3 in comparison to the original KPCL model of screw compressor. The specific power consumption is normalized with the one having the minimum value and the remaining two i.e flow and volumetric efficiency are normalized with the one having the maximum value.

On account of the optimisation performed here, the optimum parameters obtained from any of the evolutionary algorithms shows that they are almost in the same range such as optimum oil injection angle being in the range of (40-65) °, the oil injection temperature in the range of (30-40) °C, the oil port diameter in the range of (6-7) mm, pressure in the oil reservoir in the range of (5-6.5) bar and the oil viscosity in the range of (1-2)*E-06 m^2/s. This signifies that injecting oil at a lower temperature with lower oil viscosity and injecting earlier will result in improvement of performance by more than 5% as it will lead to near isothermal efficiency which is indeed a good choice of compressor. Thus from the screw compressor block point of view these optimum parameters are the best suited ones.

However, from the system design point of view there are other important conditions which need to be incorporated as constraints to the optimisation algorithms for reliable performance of the compressor.

For better understanding a sensitivity analysis has been shown in figure-12 to understand the effect of parameters on the objective functions. The analysis clearly shows that the major parameters which are contributing to the significant amount of change in the performance of the 22 kW twin screw air compressor are the oil viscosity followed by the oil injection temperature. To achieve an improvement in specific power the oil viscosity needs to be drastically reduced. But, there is a limitation on the oil viscosity to keep the bearings operating smoothly, provide an feasible bearing life and form a film for bearing lubrication.

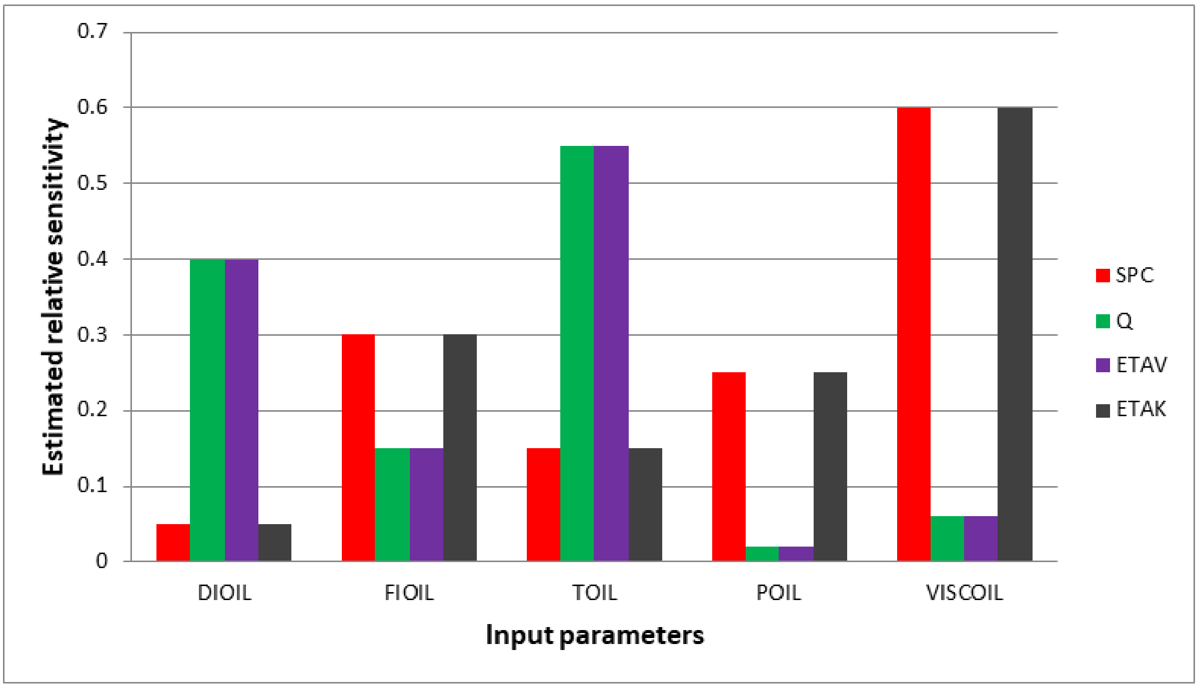

Figure 12. Sensitivity comparison for output vs. input

According to the bearings manufacturer for the bearings to work properly the viscosity ratio i.e., the kappa value must be greater than or equal to one. Kappa is the ratio of the actual oil viscosity to the minimum viscosity required to support a dynamic load within a bearing to provide full film lubrication. The kappa value for the current KPCL model is already at the verge of limit provided by the manufacturer thus using an oil with lower viscosity is not recommended from bearings point of view.

Lastly in this optimisation context a lower oil injection temperature would be desirable from the compressor block point of view but considering the system design point of view a lower oil injecting temperature signifies the temperature difference between the inlet and outlet of the compressor would increase. The desirable discharge temperature from the compressor is in the range of (80-90)°C. Thus, as the temperature difference increases the heat load on the cooling unit also increases which is an important consideration from system point of view. Therefore, the optimum oil injecting temperature should be in the range of (40-60)°C.

4. Conclusions

In the present paper, the optimisation of a 22 kW oil-injected twin screw air compressor using the different evolutionary algorithms has been carried out. Firstly, the SCORG model is set up in good agreement with the experimental results. The batch file of this model is linked to the GT-SUITE model of the screw compressor. Secondly, a GT-Process Map for the model is generated with the GT-ISE interface, which uses the SCORG thermodynamic calculation in the background for performance optimisation. Finally, the design optimiser is executed from the GT-ISE interface by providing the limits for the variation of the input parameters and defining the search algorithm. The input parameters considered in this case are the oil parameters as the oil plays a vital role in the oil-injected twin screw compressor performance. The output functions are the integral parameters used for the performance comparison of these machines such as specific power consumption, adiabatic efficiency, volumetric efficiency, and flow. Different

search algorithms have been made to run one by one independently for optimisation.
A comparative analysis has been carried out for both the single as well as the multi-objective optimisation. After the analysis, it is observed that the simplex algorithm converges the fastest and provides almost the same range of optimum parameters as other evolutionary algorithms. However, GA is an evolutionary global search algorithm recommended for all problems having medium to high complexity. The seed size is the smallest for the GA, thus having the highest fitness value.
In the future scope it is recommended that since the optimum oil parameters obtained in this context are good from the screw compressor block point of view but from the system design point of view other design and manufacturing constraints need to be incorporated to the algorithms for a feasible performance. Therefore, these conditions have to be applied for the best screw compressor optimisation algorithms, and then the simulation needs to be performed, which needs to be validated experimentally.

References

[1] *International Energy Outlook- US Energy Information Administration, 2021* https://www.eia.gov/outlooks/ieo/ (accessed Mar. 15, 2022)
[2] Fleming J S, Tang Y and Anderson H 1994 *Optimisation Techniques Applied to the Design of a Refrigeration TiN Screw Compressor* (United States, Purdue University) In International Compressor Engineering Conference paper 1049
[3] You C X, Tang Y and Fleming J S 1996 *Optimum Rotor Geometrical Parameters in Refrigeration Helical Twin Screw Compressors* (United States: Purdue University) In International Compressor Engineering Conference paper 1074
[4] Stosic N and Hanjalic K 1997 *Development and optimisation of Screw Machines With a Simulation Model - Part I: Profile Generation* (London: City University) ASME Transactions Journal of Fluids Engineering Vol 119 p 659
[5] Berlik S and Helpertz M 2002 *Optimising Screw Machines using an Evolution Strategy with Extended Creation Scheme* (Germany: University of Dortmund) In IEEE International Conference on Systems, Man and Cybernetics Vol 5, pp 6-pp
[6] Stosic N, Smith I K and Kovacevic A 2003 *Optimisation of Screw Compressors* (London: City University) Journal of Applied Thermal Engineering 23 pp 1177-1195
[7] Stosic N, Smith I and Kovacevic A 2005 *Screw compressors: mathematical modelling and performance calculation* (London: City University) Springer Science & Business Media
[8] Hauser J, Bruemmer A and Kauder K 2008 *Rotor profile generation and optimisation of screw machines using NURBS* (United States: Purdue University) In International Compressor Engineering Conference, paper 1861
[9] Herlemann S, Hauser J and Henning N 2014 *Optimisation of rotor profiles for energy efficiency by using chamber-based screw model* (Germany: University of Dortmund) In International conference on Screw machines pp 211-224
[10] Zhang C, Gao J, Fu J, Li Y and Lin J 2019 *Research on structural parameter optimisation and thermal performance of twin-screw expander* (China: Harbin Institute) International Journal of Low-Carbon Technologies 14(3) pp 386-393
[11] PDM Analysis Ltd. 2014 *SCORG - screw compressor rotor grid generation* (London: City University) Software Package for Design and Analysis of Positive Displacement Machines Retrieved from https://pdmanalysis.co.uk/scorg/
[12] Gamma Technologies LLC 2022 *GT-SUITE* (Illinois: USA) Multi-Physics Software for Design Optimisation Component Analysis and Root Cause Investigation Retrieved from https://www.gtisoft.com/
[13] Basha N 2021 *Numerical analysis of oil injection in twin-screw compressors* (Doctoral dissertation, City, University of London).

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