Research on the two-phase flow and separation mechanism in the oil–gas cyclone separator

L Z Wang¹, X Gao¹, J M Feng¹,3 and X Y Peng¹,2

¹ School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an 710049, China
² State Key Laboratory of Multiphase Flow in Power Engineering, Xi’an Jiaotong University, Xi’an 710049, China
E-mail: jmfeng@mail.xjtu.edu.cn

Abstract. The cyclone separator has attracted increasing attention due to its small size, rapid construction and high separation efficiency. This study investigated its gas–liquid two-phase flow and separation characteristics experimentally and numerically. A numerical model of two-phase flow in the cyclone separator was proposed using the Euler–Lagrange method. The distribution of pressure, tangential and axial velocity in the gas-phase flow field was obtained, and the oil droplet movement was traced. Separation efficiency was also studied experimentally, and the diameter distributions of oil droplets at the inlet and the outlet of the separator were measured by a Malvern laser particle size analyser to verify the simulation model. Based on high-speed photography technology, the oil film distribution and flow pattern on the wall of the cyclone separator were visualised. The variation of oil–gas two-phase flow in the cyclone separator was compared under various inlet flow rates. Based on the results, an improved structure was proposed, and the performance of the improved separator was investigated experimentally.

1. Introduction

In oil-injected compressor units, the compressed gas should be separated from the oil to ensure circulation of the lubricating oil and to keep the exhaust gas clean. Improving efficiency at the first separation level effectively reduces the filter load of the second separation, thereby extending the filter’s service life and reducing costs [1]. Improving the efficiency and the structure of the oil–gas separator was therefore the focus of this study [2].

A gravity sedimentation separator can only be used to separate larger oil droplets, those with a diameter of at least 100 um [3]. Inertia collision separators perform poorly when separating oil droplets with a diameter of less than 25 um, so are not used in more demanding environments [4]. In contrast, the oil–gas cyclone has very good performance prospects because of its rapid construction, small size, high efficiency and small resistance loss. Nonetheless, if designed or operated improperly, it will perform poorly. At present, the design and practicality of the oil–gas cyclone separator mainly depends on experience. Given the current lack of a theoretical basis, the two-phase flow process in the separator was studied experimentally and numerically; the separator’s performance under actual operator conditions, and the particle size distribution of import oil droplets, were tested at the same time.
2. Numerical simulation study

2.1. Physical model and simulation method

2.1.1. Physical model. The investigation was carried out in the oil–gas cyclone separator shown in Figure 1, and the geometrical details of the separator are presented in Table 1. The oil–gas mixture entered tangentially through the inlet, and rotated through the inner and outer cylinders. Due to the centrifugal force, droplets moving to the wall were separated, and the remaining droplets followed the discharge gas to escape through the outlet.

![Figure 1. Structure of the oil–gas cyclone separator](image1)

![Figure 2. Mesh of the oil–gas cyclone separator](image2)

Table 1. The structural parameters

| Dimension          | Cyclone height (H) | Central channel height (h) | Cyclone diameter (D) | Central channel diameter (d) | Inlet pipe height (a) | Inlet pipe width (b) | Inlet pipe length (L) |
|--------------------|-------------------|---------------------------|----------------------|-----------------------------|----------------------|---------------------|----------------------|
| Value (mm)         | 600               | 200                       | 200                  | 100                         | 100                  | 300                 | 600                  |

2.1.2. Turbulence model. The gas flow field requires a complex three-dimensional turbulent flow in the separator, involving significant anisotropy, and so the seven-equation RSM (Reynolds Stress Model) was used for the gas-phase flow field turbulence model. The Reynolds number of gas flow field is in the range of 250000 to 350000, so it is turbulent flow completely. Turbulence intensity $I$ and hydrodynamic diameter $D_u$ were used to describe turbulence, and they are given by [5]

$$I \approx 0.16 \left( \text{Re}_{D_u} \right)^{-1/8}$$

(1)

$$D_u = \frac{4A}{P}$$

(2)

where $A$ is the flow area/m², $P$ is the wetted perimeter/m, $\text{Re}_{D_u}$ is the Reynolds number under the hydrodynamic diameter and $\text{Re}_{D_u} = \frac{\rho u D_u}{\mu}$, $u$ is the flow velocity.

For an incompressible fluid, the governing equations for continuity and balance of momentum are given as [6]
\[
\frac{\partial \vec{u}}{\partial t} + \vec{u} \cdot \nabla \vec{u} = -\frac{1}{\rho} \nabla P + \nabla \cdot \left[ \frac{\mu}{\nu} \nabla \vec{u} \right] - \frac{\partial}{\partial x_j} \left( \rho \vec{u} \right)
\]

(3)

where \( \vec{u} \) is the mean velocity, \( x_i \) is the position, \( P \) is the mean pressure, \( \rho \) is the gas density, \( n \) is the gas kinematic velocity and \( R_y = u_i u_j \) is the Reynolds stress tensor. Here, \( u_i = u_i - \bar{u}_i \) is the \( i \) th fluctuating velocity component. The RSM turbulence model provides differential transport equations for evaluating the turbulence stress components:

\[
\frac{\partial}{\partial t} \left( \rho \vec{u}_j \right) + \nabla \cdot \left( \rho \vec{u}_j \vec{u}_i \right) = -\frac{\partial}{\partial x_i} \left( \rho \vec{u}_j \nu \frac{\partial \vec{u}_j}{\partial x_i} \right) - \frac{\partial}{\partial x_i} \left( \rho \vec{u}_j \vec{u}_i \right) - \frac{\partial}{\partial x_i} \left( \rho \vec{u}_j \right) + \nabla \cdot \left( \rho \vec{u}_j \frac{\partial \vec{u}_j}{\partial x_i} \right)
\]

(4)

The turbulence production terms \( P_y \) are defined as

\[
P_y = - \left[ R_y \frac{\partial \vec{u}_j}{\partial x_i} + R_{ij} \frac{\partial \vec{u}_j}{\partial x_i} \right], \quad P = \frac{1}{2} P_y
\]

(5)

where \( P \) is the fluctuating kinetic energy production. \( \nu \) is the turbulence(eddy) kinematic viscosity and \( \delta \) is the empirical constants. The transport equation for the turbulence dissipation rate, \( \epsilon \), is given as

\[
\frac{\partial \epsilon}{\partial t} + \vec{u} \cdot \nabla \epsilon = \nu \frac{\partial \epsilon}{\partial x_i} + \nabla \cdot \left[ \frac{\nu}{\sigma} \frac{\partial \epsilon}{\partial x_i} \right] - \frac{\nu}{\sigma} \left( \frac{\partial \epsilon}{\partial x_j} \right) \left( \frac{\partial \epsilon}{\partial x_j} \right) - \frac{\nu}{\sigma} \left( \frac{\partial \epsilon}{\partial x_j} \right) + \frac{\nu}{\sigma} \left( \frac{\partial \epsilon}{\partial x_j} \right) - \frac{\nu}{\sigma} \left( \frac{\partial \epsilon}{\partial x_j} \right)
\]

(6)

For the oil droplets, the DPM (Discrete Phase Model) was used to track and simulate the oil droplets, and the uncoupled mode was chosen for the calculation. Because the volume fraction of the oil droplets was less than 10\%, so the effect of the oil–gas interaction was ignored. When running the simulation, the gas-phase flow field was calculated first, and then the discrete phase load to simulate the oil droplets trajectories. The oil droplets diameters could be defined directly and they are in the range of \( 10um \) to \( 100um \).

2.1.3. Numerical schemes. The numerical schemes for the gas flow field in this study were drawn from Kaya and Karagoz [7]. They investigated the performance of different discretisation schemes and the suitability of various numerical schemes in highly complex swirling flows in the simulation of cyclone separators. The best settings are summarised in Table 2.

| Numerical setting     | Discrete scheme |
|-----------------------|-----------------|
| Pressure discretisation| PRESTO          |
| Momentum discretisation| QUICK           |
| Turbulent kinetic energy | Second-order upwind |
| Turbulent dissipation rate | Second-order upwind |
| Reynolds stress        | First-order upwind |

2.1.4. Boundary condition. For the gas-phase flow field, the constant velocity inlet and pressure outlet boundary were used, and the wall was regarded as a no-slip boundary. For the oil droplets phase, a
constant inlet velocity at the inlet boundary was also used, and the value was kept the same as that of the gas-phase field. The outlet was defined as ‘escape’, meaning that once the oil droplets came to the outlet, they were free; the wall was defined as a trap boundary, meaning that once the oil droplets came to the wall they were separated from the inlet mixture. The physical properties of the oil in the simulation were the same as those in the lubricating oil in the experiment: the density was 830 kg·m⁻³; the dynamic viscosity was 3.32×10⁻³ kg·m⁻¹·s⁻¹; and the surface tension coefficient was 0.025N·m⁻¹. The Weber number range is 65~650, far more than 1, so the effect of surface tension of the oil droplets can be ignored during calculation.

2.1.5. Grid. The oil gas cyclone separator was divided into the entrance, the upper cylinder, the lower cylinder and the outlet pipe, and then the structured grids were generated and meshed separately, as shown in Figure 2. The tangential velocity, static pressure and total pressure loss were compared when the grid quantities were 101944, 217292 and 397119. The results showed that the difference under the maximum grid quantity and the minimum grid quantity was less than 8%. Considering the complexity of the three-dimensional flow, we chose the grid of 217292 for our calculations.

2.2. Numerical simulation results
The following method was used to simulate the high-efficiency Stairmand cyclone, as the tangential and axial velocity distributions of this system had been measured in a previous study [8]. Figure 3 illustrates the comparison of the simulated and tested tangential and axial velocity distributions. Given the complexity of the turbulence swirling flow in the cyclones, the agreement between the simulation and the experimental data was considered to be acceptable.

Figure 3. Comparison between the simulation results and the experimental results

Figure 4 shows the pressure field distribution on the cross-section of Y=0 at the discharge velocity of 15m·s⁻¹ and discharge pressure of 1atm. It can been seen that both the total pressure field and the static pressure field were symmetrical along the height direction of the separator, and that the symmetry of the static pressure field was better than that of the total pressure. The pressure decreased gradually from the wall to the central axis; both the total pressure and the static pressure were lowest in the central axis, and had their highest values near the wall. The cross-section total rate in the cyclone separator on the cross-section of Y=0 was obtained, and the distribution of tangential velocity and axial velocity are shown in Figure 5 at a discharge pressure of 1atm.
The total velocity and the tangential velocity increased gradually from the wall; after reaching the maximum, they decreased and reached the lowest values at the central axis. Judging from the pressure field distribution in Figure 4, we know that the potential energy of gas continuously converts to kinetic energy. Regarding velocity, the total velocity and the tangential velocity changed very little along the height direction of the separator, and the axial velocity dropped to 2 m s\(^{-1}\) at the bottom of the separator; at the entrance of the internal cylinder, the velocity increased significantly, reaching 8 m s\(^{-1}\). These phenomena show that changes in the entrance area produced stronger airflow in the local vortex.

Figure 6 compares the tangential velocity and the axial velocity on cross-section \(Y=0\) and position \(Z=0.5H\) mm in the oil–gas cyclone separator at inlet velocities of 14 m s\(^{-1}\), 16 m s\(^{-1}\) and 18 m s\(^{-1}\), respectively. The tangential velocity distribution was M-shaped, while the axial velocity distribution was W-shaped. After further analysis, increasing the inlet velocity was found not to change the distribution of the velocity field inside the separator, but it significantly increased the tangential velocity, while the axial velocity remained essentially unchanged.
3. Experimental study

3.1. Experimental system

Figure 7 shows the experimental system. The diameter distribution of the oil droplets was measured using a similar method to that introduced in the literature [9]. A four-way observation tube was installed at the inlet and the outlet of the separator, and high-purity nitrogen was injected between the optical glass and the main flow during the test to ensure the purity of the optical glass, thus ensuring that the Malvern laser particle size analyser was functioning normally.

![Flow chart of the experimental system](image)

Figure 7. Flow chart of the experimental system

3.2. Experimental results

The experiment focused on the distribution and the flow patterns of the oil film on the inner surface of the oil–gas cyclone separator, and the results are shown in Figure 8, when the inlet velocity was 15 m s\(^{-1}\) and the discharge pressure was 1 atm. Figure 9 shows the division of the inner surface according to the characteristics of the oil film flows.
Different parts of the oil film flow pattern were observed by high-speed camera, and Figures 10(a)–(g) show the flow pattern and the seven corresponding areas of A–G, respectively, when the inlet velocity was 15 m·s⁻¹ and the discharge pressure was 1 atm. In region A, the oil film developed gradually to a spreading planar flow from a small stocks flow, meaning that at the inlet of the separator, the oil film increased because the oil droplets constantly bumped into the wall. In region B, groups of intermittent oil droplets moved in with the airflow, with oil droplets from the incoming flow colliding frequently with the oil film on the wall. The oil film in region C was a corrugated flow, reflecting the formation, development and disappearance of the ‘pit’ as oil droplets impacted on the oil film. In region D, the oil film developed gradually into a declining stock flow from the corrugated flow, as the oil in the oil film could not completely cover the entire tube wall as the spreading surface of the oil film continued to expand. In region E, we see that the oil film took on a sloped ‘scales’ flow, with ‘scales’ that were very thick, and corrugated on surface with small bubbles inside, and obviously thicker than the oil films in other areas. Region F featured a small flow of oil film, where oil droplets collided with a dry wall area without oil film flow, meaning transportation of oil in this area was very poor, with no obvious flow channel. In region G, the flow pattern of oil film at the interface of the oil and gas was extremely complex, with strong disturbance airflow, making the interface very unstable, and the lubricating oil which had been separated contained a large amount of bubbles. On the inner wall of the separator, the lubricating oil contained oil that flowed to the bottom along the surface of cylinder body, and oil that sprayed up under the effect of airflow.
Figure 10. The oil film flow patterns of regions A–G

Figure 11 shows the separation efficiency of 70–80% when the inlet velocity was 14.4-18.1 m·s⁻¹ and the discharge pressure was 0.2 MPa. When the inlet velocity was less than 17 m·s⁻¹, the separation efficiency increased with the increase of the inlet flow velocity, and the separation efficiency decreased with the increase of the inlet flow velocity if the inlet velocity was greater than 17 m·s⁻¹. For the simulation results, the separation efficiency at 17 m·s⁻¹ was also slightly higher than that at other velocities. This suggests that there is a critical maximum value for separation efficiency with the change of the inlet velocity, which determines the relationship between the inlet velocity and the separation efficiency. This critical value is the key in the design of the separation, and for the separator studied in this paper, when the inlet velocity is 17 m·s⁻¹. In addition, the measured separation efficiency was lower than the simulated results. This can be explained by the fact that in the numerical simulation the oil level on the bottom was treated as the wall boundary and the trap boundary condition was applied, so that the oil droplets were assumed to be separated if they reached this surface. Due to the effect of the high-speed swirling flow, the oil separated on the baffle might be blown back into the flow field and finally escape through the outlet. Moreover, the effect of the ‘dust ring’ [10] was ignored in the simulation.

When comparing the droplets’ distribution between the inlet and the outlet, Figure 12 shows that the number of droplets with a diameter greater than 45 µm reduced significantly. The increase in the percentage of oil droplets smaller than 20 µm at the outlet implied that droplet breakup happened during the separation process. It was also evident that the percentage of smaller-diameter oil droplets increased at the outlet as the inlet velocity increased. This meant that a higher velocity inside the separator led to easier droplets breakup. The percentage of 10 µm droplets increased in volume from 3% at the inlet to 7.5% at the outlet when the inlet velocity was 15.2 m·s⁻¹. If the inlet velocity reached 18.1 m·s⁻¹, the percentage of 10 µm-diameter droplets would increase to 10% [11]. We conclude that when the inlet velocity reaches a certain critical value, more small oil droplets will be produced in the separator, thereby decreasing separation efficiency. This conclusion is consistent with the ‘the critical velocity’ which is analysed in Figure 11.
4. Improvement of the separator

According to the above findings, two problems exist in conventional oil–gas cyclone separators. First, spraying the separated oil at the bottom of the separator into the gas-phase flow field unfavourably affects separation, and, with the increase of the inlet velocity, the amount of the oil going into the gas-phase flow field will increase significantly. Second, the numerical simulation and experimental results both prove that with an increase of the oil film thickness, more small oil droplets will splash from the oil film because of the impact of the gas flow, thereby decreasing separation efficiency.

Figure 13 shows a proposed solution, with a porous cylinder added to the separator with a diameter 4mm less than that of the original cylinder, providing a 5mm sandwich between the external separator and the porous cylinder. This layer prevents the oil droplets from returning to the mainstream through the round hole into the interlayer; because it is difficult for an oil film to form on the porous surface, it can avoid the oil droplets adhering to the oil film surface, and reduce the possibility of splash. We can see from Figure 13 that the oil film turned to femoral and flowed downward vertically; this means that there is no gas flow in the interlayer space, so the lubricating oil flows naturally under the action of gravity. Looking at the comparison of static pressure loss shown in Figures 14, we can see that there is less pressure loss in the improved structure. So we add a porous cylinder to a real oil–gas cyclone separator and applied it to an oil-injected compressor system for further study. Restricted to the processing conditions, the distance between porous cylinder and external separator is 10 mm. The improved separator is shown in Figure 15 and the geometrical details are presented in Table 3. The Figure 16 shows the comparison results of the separation efficiency, from which we can see that the improved separator is more efficient.

![Figure 13. The distribution of the oil film in the improved separator](image)

![Figure 14. Comparison of the static pressure loss of the original and improved structures](image)

![Figure 15. The improved oil-gas cyclone separator](image)

![Figure 16. Comparison of the separation efficiency of the original and improved structures](image)
Table 3. The structural parameters

| Dimension          | Cyclone height (H) | Central channel height (h) | Cyclone diameter (D) | Central channel diameter (d) | Inlet pipe diameter (a) | Inlet pipe length (L) |
|--------------------|--------------------|----------------------------|----------------------|----------------------------|------------------------|-----------------------|
| Value (mm)         | 550                | 240                        | 240                  | 120                        | 25                     | 100                   |

5. Conclusion
(1) The two-phase flow simulation model in the oil–gas cyclone separator was built using the Euler–Lagrange method. The distribution of the gas flow field and oil droplets trajectories were obtained, and the simulation results were verified by experiments.
(2) The distribution of the oil film on the outside wall of the separator was visualised using high-speed photography technology. According to the oil film distribution and the flow pattern, the wall was divided into seven different regions, A–G, and each of them was described in detail. The fog type annular two-phase flow at the separator inlet was identified as the root cause of the distribution of the oil film and flow pattern.
(3) The results showed that static pressure loss increased with the increase of inlet velocity, while separation efficiency initially increased, and then decreased, with the increase of inlet velocity. There exists a maximum value of separation efficiency affected by inlet velocity; and avoiding the generation of small oil droplets is an effective way to improve separation efficiency.
(4) An improved structure was proposed, which took into account the two main factors that affect the separation efficiency. The experimental results showed that the new structure is feasible and efficient.

References
[1] Hammerl K, Tinder L and Frank M 2000. Modification in the design of the oil injected system for crew compressor. Purdue Compressor Technology Conference. 987-994.
[2] Wiencke B 2011. Fundamental principles for sizing and design of gravity separators for industrial refrigeration. International Journal of Refrigeration. 34 2092-2108.
[3] Liu C S 1994. Development and field test of separator in prying loading type gas collecting device. Natural Gas and Petroleum. 12 36-40.
[4] Wang S Z 2007. Applied exploration of gas-liquid separator and high efficiency separator. Small Nitrogenous Fertilizer. 35 25.
[5] Jing S R and Zhang M Y 2001. Fluid Mechanics ( Xi’an: Xi’an Jiaotong University Press) p189.
[6] Elsayed K and Lacor C 2011. Numerical modeling of the flow field and performance in cyclones of different cone-tip diameters. Computers and Fluids. 51 48-59.
[7] Kaya F and Karagoz I 2008. Performance analysis of numerical schemes in highly swirling turbulence flow in cyclone. Current Science. 94 1273-1278.
[8] Hoekstra A J 2000. Gas flow field and collection efficiency of cyclone separators. (PhD thesis) (Technical University Delft, Stevinweg, Netherlands).
[9] Feng J M, Chang Y F and Peng X Y 2008. Investigation of the oil–gas separation in a horizontal separator for oil-injected compressor units. Journal of Power and Energy. 222 403-411.
[10] Wang B, Xu D L, Chu K W and Yu A B 2006. Numerical study of gas–solid flow in a cyclone separator. Applied MathematicalModelling. 30 1326-1342.
[11] Gao X, Chen J F, Feng J M and Peng X Y 2013. Numerical and experimental investigations of the effects of the breakup of oil droplets on the performance of oil–gas cyclone separators in oil-injected compressor systems. Refrigeration. 36 1894-1904