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# Back swept angle performance analysis of centrifugal compressor

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## 1. Introduction

With the development of automotive turbocharger, high efficiency, high pressure ratio and wide operating range are required by centrifugal compressor [1-3]. The compressor performances are important in turbocharging field because they determine the engine air supply. The most important component of centrifugal compressor is impeller [4]. Radial impeller has a wide application in diesel engine turbochargers [5]. With the development of impeller, more and more gasoline engines have installed turbochargers and the engine speed has been reaching more than 5000 rpm. In order to adjust the very wide range of engine speed, the selection of back swept angle has become the most important factor for impeller design.

Liam Barr and Stephen Spence [6] established radial impeller with 25° back swept angel, after the comparing the numerical analysis of 25° back swept blade with radial blade, it was concluded that 25° back swept blade offers significant increases of efficiency while operated at lower velocity ratios. The similar conclusion was drawn by Peng Sen and Yang Ce based on three-dimension Navier-Stokes equations about 30° and 45° angle impeller of a centrifugal compressor [7], it was concluded that an appropriate angle can increase the range of a centrifugal compressor, improve the flow and enhance the isentropic efficiency. Hildebrandt and Genrup [8] gave a numerical investigation about the effect of different back swept impeller of a centrifugal compressor, result shows that the back swept angle provides a uniform flow pattern in term of velocity. The authors found that an appropriate angle will be beneficial to performance of centrifugal compressor. This paper adopts TurboSystem (a set of software applications and software features for designing turbomachinery in the ANSYS Workbench environment) to create the impeller flow field which is impellers with 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15° and 17.5° back swept angles, and adopts CFX (a general purpose Computational Fluid Dynamics software) to simulate every impeller, compares and analyzes the results of flow field on different back swept angles through considering pressure ratio, isentropic efficiency, volume flow rate and Mach number.

## 2. Three-dimensional finite element model of impeller

The TurboSystem consists of Vista CCD, BladeGen and TurboGrid [9-11]. Vista CCD is integrated into ANSYS Workbench so that it may be used to generate an optimized 1D (one-dimension) compressor design before moved rapidly to a full 3D (three-dimension) model and CFD (Computational Fluid Dynamics) analysis. In Vista CCD dialog box, the input data can be specified on the Duty, Aerodynamic Data and Geometry [12]. The data is shown in Table 1; impellers with 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15° and 17.5° back swept angles are obtained by changing the impeller outlet angle.

Table 1

The Vista CCD input data

| Parameters | Value |
|---|---|
| Pressure ratio | 1.8 |
| Mass flow, kg/s | 0.106 |
| Rotational speed, rev/min | 66322 |
| Hub diameter, mm | 5 |
| Shroud diameter, mm | 60 |
| Rake angle, ° | 30 |
| Vane inlet angle, ° | 60 |
| Back swept angle, ° | 0, 2.5, 5, 7.5, 10, 12.5, 15, 17.5 |
| Number of main blade | 9 |
| Number of splitter blade | 9 |

After establishing the 1D model of impeller, we continue to create 3D model of impeller. BladeGen is a geometry creation tool for turbo machinery blade rows and we use it to create the 3D model of centrifugal compressor impeller. After creating model of impeller, we continue to generate the mesh of impeller. The geometry is exported through TurboGrid which is a powerful blade mesh tool. These meshes are used in the workflow to solve complex rotating blade passage problems. In order to save simulation's time, we use single leaf channel flow field [13], the high-quality hexahedral meshes of impeller view is shown in Fig. 1 and the single leaf channel of impellers is shown in Fig. 2.

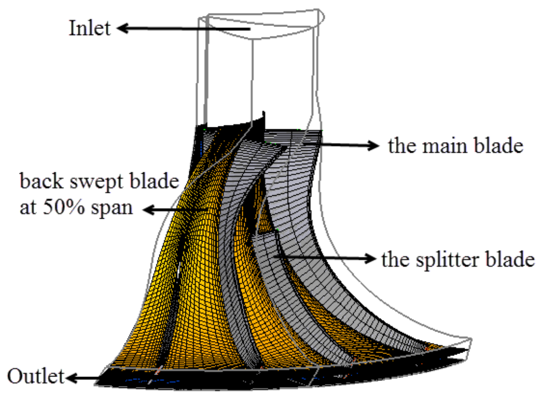


Fig. 1 High-quality hexahedral meshes of impeller

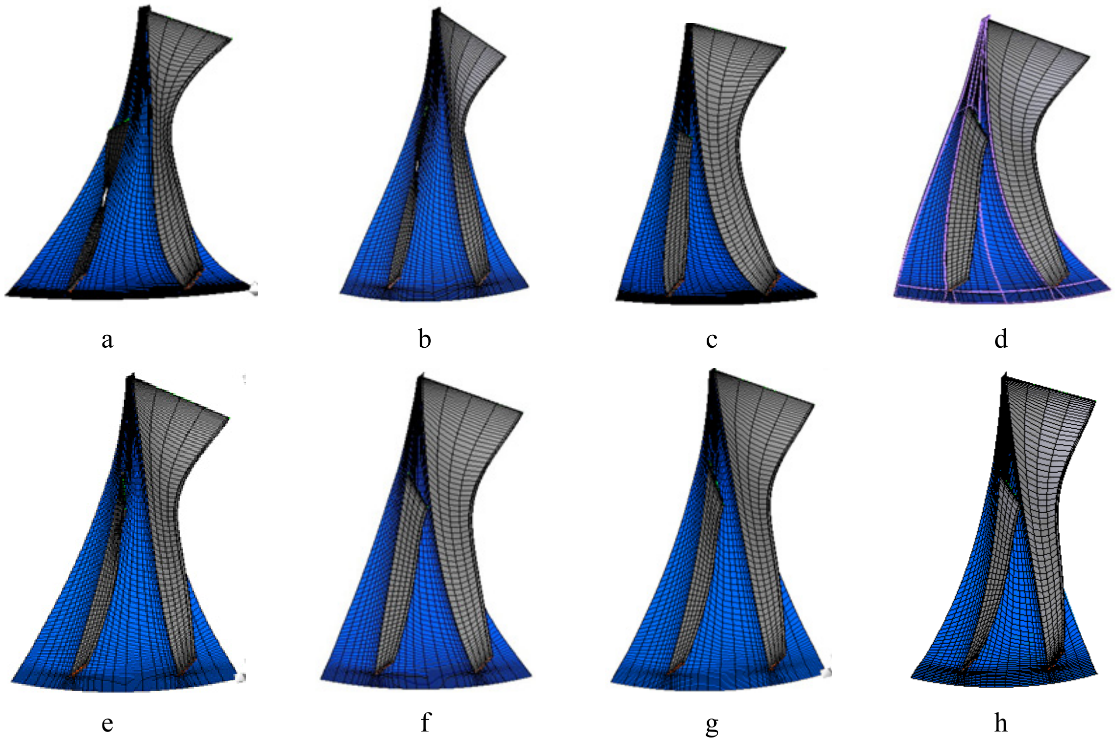


Fig. 2 Single leaf channel of impellers: a) 0°; b) 2.5°; c) 5°; d) 7.5°; e) 10°; f) 12.5°; g) 15°; h) 17.5°

**3 Simulation detail**

The three dimensional numerical simulations to the air compressor are carried out by CFX [14], which adopts a finite element based finite volume to solve the Navier-Stoker equations for a structured grid. The parameters of CFX basic setting are shown in Table 2. The boundary condition consists of Inlet Temperature, Inlet Pressure, Mass Flow Rate and Compressor Speed; the parameters of boundary condition are shown in Table 3.

Table 2

The parameters of CFX basic setting

| Parameter settings | Value |
|---|---|
| Fluid region | Turbo mode |
| Machine type | Centrifugal compressor |
| Rotating axis | Z |
| Analysis type | Steady state |
| Turbulence model | k-Epsilon |

Table 3

Data of centrifugal compressor in different engine speed conditions

| Engine speed, rpm | Mass flow, kg/s | Compressor speed, rpm | Inlet temperature, K | Inlet pressure, Bar |
|---|---|---|---|---|
| 500 | 0.00893 | 39023 | 299.65 | 0.999 |
| 1000 | 0.0193 | 43820 | 299.43 | 0.998 |
| 1500 | 0.0320 | 48544 | 299.19 | 0.997 |
| 2000 | 0.0460 | 52503 | 298.97 | 0.995 |
| 2500 | 0.0645 | 57367 | 298.67 | 0.991 |
| 3000 | 0.0822 | 61434 | 298.44 | 0.987 |
| 3500 | 0.106 | 66322 | 297.93 | 0.978 |
| 4000 | 0.139 | 72767 | 297.18 | 0.966 |
| 4500 | 0.185 | 79977 | 296.08 | 0.947 |

**4. Effects of back swept angle impeller**

4.1. Performance analysis of compressor impeller

Numerical simulation of compressor impeller pressure ratio with different back swept angles (0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15°, 17.5°) can be gotten and the 0° back swept impeller is radial impeller [15].

Fig. 3 details the variation of pressure ratio of different back swept angles. From 500 rpm to 1500 rpm, all the curves of pressure ratio are increasing with engine speed; from 1500 rpm to 2500 rpm, all the curves of pressure ratio are decreasing with engine speed; more than 2500 rpm, all the curves of pressure ratio are increasing with engine speed. Back swept impellers deliver decrease in pressure ratio while operated from 500 rpm to 4500 rpm when compared to the radial impeller.

Fig. 4 details the variation of isentropic efficiency of different back swept angles. From 500 rpm to 1000 rpm, all the curves of isentropic efficiency are increasing with engine speed; from 1000 rpm to 3000 rpm, all the curves of isentropic efficiency are decreasing with engine speed; more than 3000 rpm, all the curves of isentropic efficiency are decreasing with engine speed. At 1000 rpm of the engine speed, the isentropic efficiency of 15° back swept impeller is more efficient than the other impellers (more than the radial impeller 0.78%); at 2000 rpm of the engine speed, the isentropic efficiency of 2.5° back swept impeller is more efficient than the other impellers (more than the radial impeller 2.96%).

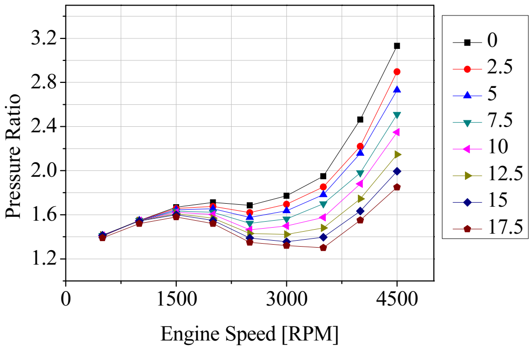


Fig. 3 Line plot of pressure ratio and engine speed for 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15°, 17.5° back swept angles

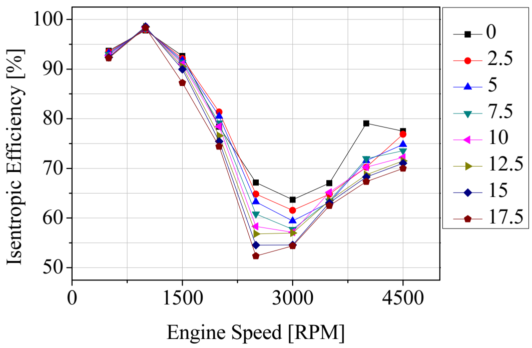


Fig. 4 Line plot of isentropic efficiency and engine speed for 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15°, 17.5° back swept angles

The parameters of volume flow rate and engine speed for 0°, 5°, 10° and 15° back swept angles are shown in Table 4.

From 500 rpm to 1500 rpm, the back swept angle impellers deliver an increase in volume flow rate when compared to the radial impeller.

Table 4

Parameters of volume flow rate and engine speed with different back swept angles

| Engine Speed, rpm | Volume Flow Rate, $m^3/s$ | | | |
|---|---|---|---|---|
| | 0° | 5° | 10° | 15° |
| 500 | 0.0417 | 0.0418 | 0.0420 | 0.0421 |
| 1000 | 0.0936 | 0.0945 | 0.0953 | 0.0960 |
| 1500 | 0.1509 | 0.1518 | 0.1528 | 0.1534 |
| 2000 | 0.1916 | 0.1901 | 0.1916 | 0.1913 |
| 2500 | 0.2031 | 0.2018 | 0.2002 | 0.1982 |
| 3000 | 0.2055 | 0.2038 | 0.2020 | 0.2000 |
| 3500 | 0.2077 | 0.2059 | 0.2040 | 0.2020 |
| 4000 | 0.2098 | 0.2085 | 0.2067 | 0.2046 |
| 4500 | 0.2134 | 0.2116 | 0.2098 | 0.2078 |

### 4.2. Comparison and analysis of flow field

Compressible flows $u$ can be characterized by the value of the Mach number $M$ [16]:

$$M \equiv u / c , \tag{1}$$

where $c$ is the speed of sound in the gas.

When the Mach number is less than 1.0, the flow speed is called as subsonic. At Mach numbers much less than 1.0, compressibility effects are negligible and the pressure variation caused by gas density can safely be ignored in flow modeling. When the Mach number approaches 1.0, compressibility effects become important. When the Mach number exceeds 1.0, the flow speed is called as supersonic. The shock wave is one of several different ways that supersonic flow gas can be compressed.

To each back swept angle, results are considered from the boundary condition of 3500 rpm engine speed. Fig. 5 shows contour of Mach number at different back swept angles. In Fig. 5, $\theta$ (radian) is used as ordinate, and $M$ (meter) is used as abscissa.

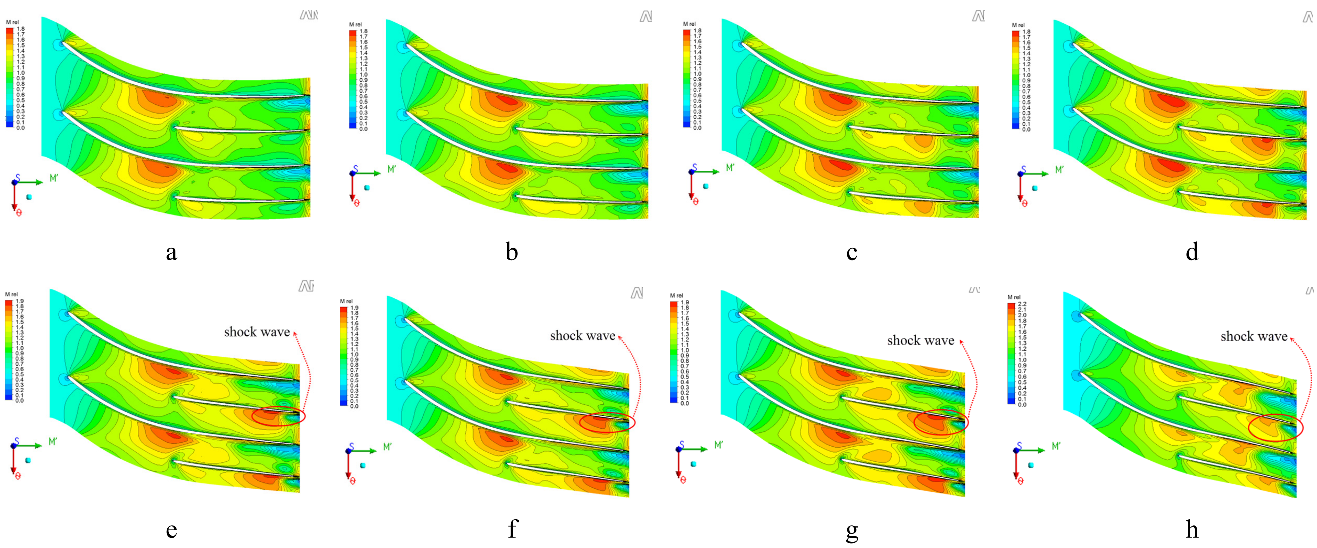


Fig. 5 Mach number at different back swept angles: a) Mach number of 0° back swept angle; b) Mach number of 2.5° back swept angle; c) Mach number of 5° back swept angle; d) Mach number of 7.5° back swept angle; e) Mach number of 10° back swept angle; f) Mach number of 12.5° back swept angle; g) Mach number of 15° back swept angle; h) Mach number of 17.5° back swept angle

The shock wave is formed when fluid speed change faster than sound speed [17]. The result shows that shock waves are formed on splitter blade root when the back swept angle is increased. With the increasing of back swept angle, the scope of shock wave is widened. Shock wave significantly decreases the velocity at blade outlet. Shock wave can cause a loss of total pressure.

## 5. Conclusions

The present study was undertaken to analyze the performance of centrifugal compressor by using Turbo-System and CFX. We have simulated the flow about compressor impeller at different back swept angles, there are five conclusions:

1. Compared to the radial impeller, the back swept angle impellers deliver decrease in pressure ratio, that is to say, back swept impellers don't amplify pressure ratio of centrifugal compressor.

2. Compared to the efficiency of radial impeller, the back swept impellers offer an increase in isentropic efficiency while operated from 1000 rpm to 2000 rpm. Draw a conclusion, that is, back swept angle impeller can improve the centrifugal compressor isentropic efficiency less than 2000 rpm engine speed.

3. Compared to the radial impeller, the back swept angle impellers deliver increase about volume flow rate while operated less than 1500 rpm engine speed.

4. At 3500 rpm engine speed, shock waves are formed on splitter blade root when the back swept angle is increased. The total pressure of compressor is influenced by the shock wave. Shock wave can cause a loss of total pressure.

5. As shown in Table 4 and Fig. 5, we find an appropriate back swept angle can improve the volume flow rate of blade and enhance the outlet velocity of compressor.

## References


1. **Nili-Ahmadabadi, M.; Poursadegh, F.** 2013. Centrifugal compressor shape modification using a proposed inverse design method, Journal of Mechanical Science and Technology 27(3): 713-720. http://dx.doi.org/10.1007/s12206-013-0120-0.
2. **Korakianitis, T.; Hamakhan, I.A.; Rezaienia, M.A.** 2012. Design of high-efficiency turbomachinery blades for energy conversion devices with the three-dimensional prescribed surface curvature distribution blade design (CIRCLE) method, Applied Energy 89(1): 215-227. http://dx.doi.org/10.1016/j.apenergy.2011.07.004.
3. **Marsan, A.; Trébinjac, I.; Coste, S.** 2013. Temporal behaviour of a corner separation in a radial vaned diffuser of a centrifugal compressor operating near surge, Journal of Thermal Science 22(6): 555-564. http://dx.doi.org/10.1007/s11630-013-0662-6.
4. **Heinze, K.; Schrape, S.; Voigt, M.** 2012. Probabilistic endurance level analyses of compressor blade, CEAS Aeronautical Journal 3(1): 55-65. http://dx.doi.org/10.1007/s13272-012-0043-y.
5. **Oyelami, A.T.; Adejuyigbe, S.B.; Waheed, M.A.** 2012. Analysis of radial-flow impellers of different configurations, The Pacific Journal of Science and Technology 13(1): 24-33.
6. **Liam, B.; Stephen, S.; Paul, E.** 2009. Design and Analysis of a Radial Turbine with Back Swept Blading, Fluid Machinery and Fluid Mechanics, 115-121. http://dx.doi.org/10.1007/978-3-540-89749-1_15.
7. **Peng, S.; Yang, C.; Ma, C.C.; Wang, H.** 2005. Influence of front lean angle on centrifugal compressor performance, Journal of Tsinghua University (Science and Technology) 45(2): 250-253.
8. **Hildebrandt, A.; Genrup, M.** 2007. Numerical investigation of the effect of different back sweep angle and exducer width on the impeller outlet flow pattern of a centrifugal compressor with vaneless diffuser, Journal of turbomachinery 129(2): 421-433. http://dx.doi.org/10.1115/1.2447873.
9. **Wang, Y.; Luo, Z.** 2011. Computational investigation on centrifugal compressor performance at various altitudes, Advanced Materials Research 230: 1123-1128. http://dx.doi.org/10.4028/www.scientific.net/AMR.230-232.1123.
10. **Obrovsky, J.; Krausová, H.** 2013. Development of high specific speed Francis turbine for low head Hpp, Engineering Mechanics 20(2): 139-148.
11. **Benini, E.; Biollo, R.; Ponza, R.** 2011. Efficiency enhancement in transonic compressor rotor blades using synthetic jets: A numerical investigation, Applied Energy 88(3): 953-962. http://dx.doi.org/10.1016/j.apenergy.2010.08.006.
12. **Chen, S.; Yang, C.; Yang, C.M.** 2012. Investigation of geometrical parameters influence to the stress and aerodynamic performance of centrifugal impeller, Fluid Machinery 40(3): 21-26.
13. **Luo, Q.G.; Si, D.Y.; Ran, Z.G.** 2013. Modal characteristics of compressor blades considering fluid-solid interaction, Applied Mechanics and Materials 37(6): 407-410. http://dx.doi.org/10.4028/www.scientific.net/AMM.376.407.
14. **Dong, S.J.; Qi, B.; Wang, J.** 2013. Optimization of loss models for centrifugal compressor performance prediction based on numerical analysis results, Applied Mechanics and Materials 300: 225-231. http://dx.doi.org/10.4028/www.scientific.net/AMM.300-301.225.
15. **Chunxi, L.; Ling, W.S.; Yakui, J.** 2011. The performance of a centrifugal fan with enlarged impeller, Energy Conversion and Management 52(8): 2902-2910. http://dx.doi.org/10.1016/j.enconman.2011.02.026.
16. **Muhammad, N.L.; Seong, S.K.** 2012. Numerical investigation of the effect of inlet skew angle on the performance of mechanical vapor compressor, Desalination 284: 66-76. http://dx.doi.org/10.1016/j.desal.2011.08.038.
17. **Settles, Gary S.** 2006. High-speed Imaging of Shock Wave, Explosions and Gunshots, American Scientist 94(1): 22-31. http://dx.doi.org/10.1511/2006.57.986.

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## IŠCENTRINIO KOMPRESORIAUS MENČIŲ ATLENKIMO KAMPO TINKAMUMO ANALIZĖ

R e z i u m ė

Įvertinta atlenkimo kampo įtaka kompresoriaus tinkamumui panaudojant radialinę atlenktą atgal sparnuotę, kuri buvo suprojektuotą programine įranga TurboSystem. 3D sparnuočių su 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15° ir 17.5° atlenktais kampais projektavimas atliktas keičiant sparnuotės nutekėjimo kampą. Sparnuočių vidinio srauto lauko modeliavimui ir išcentrinio kompresoriaus reakcijai nustatyti, kurios parametrai priklauso nuo slėgio dydžio, izotropinio efektyvumo ir srauto tekėjimo greičio panaudota CXF (skaitinė skysčių dinamika) programinė įranga. Šie parametrai yra išanalizuoti ir palyginti. Skaitinio skysčių dinamikos modeliavimo rezultatai rodo, kad tinkamas atlenkimo kampas gali pagerinti šiuos parametrus.

Reliatyvūs Macho skaičiaus pasiskirstymai yra vizualizuoti, išanalizuotos priežastys, kurios sukelia bendrus slėgio nuostolius. Smūgio banga gali sukelti bendro slėgio praradimą. Šie tyrimai yra pagrindas sparnuotės metodiškam optimizavimui.

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## BACK SWEPT ANGLE PERFORMANCE ANALYSIS OF CENTRIFUGAL COMPRESSOR

S u m m a r y

The effects of back swept angle about compressor performance are investigated by using a radial back swept impeller designed on TurboSystem. 3D impellers Design with 0°, 2.5°, 5°, 7.5°, 10°, 12.5°, 15° and 17.5° back swept angles are obtained by changing the impeller outlet angle. CFX is used to simulate internal flow field of these impellers and generate the centrifugal compressor report that parameters consist of pressure ratio, isentropic efficiency and volume flow rate. These parameters are analyzed and compared. The results of the CFD indicate that an appropriate back swept angle can improve these parameters. The relative Mach number distributions are visualized and the reason that causes total pressure loses are analyzed. Shock wave can cause a loss of total pressure. These studies are the foundation of multidisciplinary optimization of impeller.

**Keywords:** centrifugal compressor, back sweep angle, TurboSystem, CFX.

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