Consider a Non-Spherical Elephant: Computational Fluid Dynamics Simulations of Heat Transfer Coefficients and Drag Verified Using Wind Tunnel Experiments

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ABSTRACT

Animal momentum and heat transfer analysis has historically used direct animal measurements or approximations to calculate drag and heat transfer coefficients. Research can now use modern 3D rendering and computational fluid dynamics software to simulate animal–fluid interactions. Key questions are the level of agreement between simulations and experiments and how superior they are to classical approximations. In this paper we compared experimental and simulated heat transfer and drag calculations on a scale model solid aluminum African elephant casting. We found good agreement between experimental and simulated data and large differences from classical approximations. We used the simulation results to calculate coefficients for heat transfer and drag of the elephant geometry. J. Exp. Zool. 9999A: XX–XX, 2013. © 2013 Wiley Periodicals, Inc.

How to cite this article: Dudley PN, Bonazza R, Porter WP. 2013. Consider a non-spherical elephant: Computational fluid dynamics simulations of heat transfer coefficients and drag verified using wind tunnel experiments. J. Exp. Zool. 9999:1–9.

Traditionally, animal momentum and heat transfer analysis made approximations of animal geometries. The simplest of these approximations is a sphere. In this paper we merged modern 3D digital art and techniques from heat transfer analysis. We compared momentum and heat balance experiments on a solid metal elephant scale model to computational fluid dynamics (CFD) simulations and classical spherical calculations. As this class of CFD software is developed, tested, and optimized for industrial applications, a key question is how closely do simulations and experiments agree when geometries are biological?

Researchers often require the heat transfer coefficient \( h_k \) to aid in calculating an animal’s energy needs, mechanistically model an animal’s distribution, or investigate basic physiological aspects of heat transfer (Porter and Budaraju, 2000; Tieleman et al., 2003; Natori and Porter, 2007). For aquatic and in-flight aerial species the above calculations also require drag (Bullen and McKenzie, 2007). Traditional methods for finding these values fall into three categories. First, direct measurements made using live animals. This method involves training live animals to fly in wind tunnels or swim in flumes to measure drag force and thermal gradients. Tethers or pressure sensors measure the force (Usherwood, 2003; Blake and Chan, 2007) while temperature sensors measure thermal gradients (Tieleman et al., 2003; Marom et al., 2006). There are alternative, noncontact, direct measurement techniques such as visualizing particle flows around the animal and using image capture software to calculate force (Drucker and Lauder, ’99),

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Received 31 December 2012; Accepted 12 March 2013
DOI: 10.1002/jez.1796
Published online XX Month Year in Wiley Online Library (wileyonlinelibrary.com).
using cameras to see reference point movements (Skrovan et al., '99; Tobalske et al., '99) or using thermal cameras to measure temperature profiles (Ward et al., '99). Second, measurements made using dead animals or castings (Dear et al., '97). This method places a dead animal in a moving fluid or drags it through a still fluid to approximate drag (Stephenson et al., '89). Third, models requiring these coefficients will often reduce the animal to a basic shape (sphere or cylinder) (Porter et al., '94) or a combination of basic shapes (Natori and Porter, 2007).

The advantage of using live animals is that it preserves most of the complex, natural environment interactions between the animal and the fluid (air or water). The disadvantages are the handling time, logistics of conducting live animal studies, size limitations on specimens, and inability to separate local fluid interactions. Using dead animals or castings is a simpler experiment, but loses some of the movement causing the most complex fluid–animal interactions. Models using simple shape approximations are quick; however, they lose the complex interactions between the animal and the fluid.

To capture some of the complex fluid interactions lost in dead animal, casting, and modeled measurements, researchers are beginning to use CFD programs. These simulations sometimes use simplified geometries (Sachs, 2007; Hazekamp et al., 2009) or model part of an animal (Pavlov et al., 2007; Weber et al., 2009). Some, however, model a complete animal (Pavlov and Rashad, 2012). To date none of these CFD models simulate terrestrial species and none calculate heat transfer coefficients.

Here, we used a commercial CFD package (ANSYS®, Release 14.0) to calculate the heat transfer coefficient and drag for a scale model African elephant aluminum casting, verify the simulation with wind tunnel experiments, and compare our simulation results to the classical spherical approximation. This simulation method keeps much of the complex fluid interactions, has no size limitations, permits analysis of local contributions to drag, and the heat transfer coefficient, does not require a casting, and allows rapid and flexible data collection. If shown accurate, this experimental method has the promise to save researchers a great deal of time and expense.

METHODS

Wind Tunnel Drag

We constructed a solid aluminum elephant in a walking posture from a wax, scale model sculpture using the lost wax process (Porter et al., '73) (Table 1 details casting physical parameters and other experimental parameters). We connected the elephant’s posterior to a sting balance via a 0.27 m long, 0.5 cm diameter metal rod. We placed each of its feet on a ball bearing recessed in a machined cavity in a solid Plexiglas sheet. The Plexiglas sheet had a knife edge at the top of the front end. The feet were flush with the sheet surface but on a movable ball bearing in each of four channels (Fig. 1). We mounted the 4.8 cm thick Plexiglas sheet on four posts bolted to the floor of the wind tunnel. The recirculating wind tunnel had a 0.9 m x 1.2 m x 1.84 m working section (Fig. 1). We measured the drag for set headwind velocities of 5–25 m/sec in 5 m/sec increments. The wind tunnel does not stabilize at the exact set velocities, thus actual recorded velocities are different for set velocities. We averaged the drag force over a 13 sec run for each air speed with measurements taken every 0.125 sec.

Wind Tunnel Heat Transfer Coefficient, $h_c$

We fitted the same aluminum elephant model with four copper–constantan thermocouples on the belly, on the left mid lateral portion of the body, centered behind the left ear and centered on the back (Fig. 2). We used a high intensity heat lamp to heat the

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elephant Casting</td>
<td>$m_e$</td>
<td>3.5</td>
<td>kg</td>
</tr>
<tr>
<td>Mass</td>
<td>$A_f$</td>
<td>0.0147</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Frontal area</td>
<td>$A_s$</td>
<td>0.0326</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Side area</td>
<td>$L$</td>
<td>0.111</td>
<td>m</td>
</tr>
<tr>
<td>Charastic length</td>
<td>$\rho_e$</td>
<td>2,560</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>Density</td>
<td>$c_{pe}$</td>
<td>902</td>
<td>J/kg/K</td>
</tr>
<tr>
<td>Specific heat</td>
<td>$k_e$</td>
<td>202.4</td>
<td>W/m/K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$\rho_a$</td>
<td>1.137–1.161</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>Density</td>
<td>$c_{pa}$</td>
<td>1007.8–1008.0</td>
<td>J/kg/K</td>
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<tr>
<td>Specific heat</td>
<td>$k_a$</td>
<td>0.02583–0.02591</td>
<td>W/m/K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$\mu_a$</td>
<td>$1.821 \times 10^5–1.835 \times 10^5$</td>
<td>kg/m/sec</td>
</tr>
</tbody>
</table>

We set the parameters to match those experienced in the wind tunnel experiments. The ranges are for parameters which changed from run to run owing to the waste heat from the wind tunnel fan.
elephant to approximately 30°C above air temperature. We turned off the lamp before sealing the wind tunnel and beginning the experiment. We waited for the wind tunnel flow to stabilize before taking measurements. We reheated the elephant before the third (15.2 m/sec) run. Given these methods, the initial, elephant temperature was not constant across runs. We measured the transient cooling for each run (Fig. 3). As the velocity of the wind increased, there was an increase in the ambient wind tunnel temperature likely owing to fan waste heat. We measured the temperature change over a 150 sec run for each air speed with measurements taken every 0.125 sec. Using Equation 1, we did a least squares fit using the casting temperature, initial and current air temperature at each time point to find the heat transfer coefficient at each wind speed (Wathen et al., ’71).

\[
\ln \left( \frac{T_e - T_a}{T_i - T_a} \right) = - \frac{h_c A_s}{c_p m_e} t 
\]  

where \( T_e \) is the elephant casting temperature at any instant, \( T_i \) is the initial casting temperature, \( T_a \) is the air temperature, \( h_c \) is the heat transfer coefficient, \( A_s \) is the casting surface area, \( c_p \) is the specific heat of the aluminum casting, \( m_e \) is the elephant casting mass, and \( t \) is the time.

**Simulation**

We used a NextEngine Multidrive 3D laser scanner to scan the 3D elephant model with millimeter precision to create a triangular mesh. We imported the 3D mesh from the scanner into Rhinoceros v.4 software to convert the mesh to a non-uniform rational B-spline (NURBS) format. Computational fluid dynamics meshing requires this format. We imported the 3D NURBS image into ANSYS DesignModeler and enclosed it in a virtual box with a 0.5 m buffer on each side except where the elephant’s feet touched the floor. We imported the box into ANSYS Meshing and meshed the fluid zone with approximately equal parts 4-node tetrahedral and 6-node wedge elements and a few 5-node pyramid elements. We used a 1 cm, 16 layer inflation boundary around the elephant (Fig. 4). The mesh consisted of approximately 1,260,000 elements. We ran simulations for drag and heat transfer with both a headwind and a crosswind from the left at each wind tunnel speed (previous experiments show heat transfer’s low sensitivity to wind.
direction (Wu and Gebremedhin, 2001) and we wanted to verify this result. This combination gave us a total of 20 simulations. For each simulation we set the inlet velocity to the velocity measured in that wind tunnel run, the outlet to zero pressure and the other four boundaries to walls. We set the inlet flow to low turbulence (1%). We set physical conditions (i.e., air temperature, air dynamic viscosity, air density, air thermal conductivity, air specific heat, and casting temperature) to match the values from each wind tunnel experiment. These values changed because of the increasing waste heat from the fans. We ran the steady state simulations in ANSYS Fluent using the $k-\omega$ shear stress transport (SST) and energy transfer model. We calculated drag and the heat transfer coefficient in ANSYS CFD Post.

RESULTS

Figure 5 shows a contour map of the pressure for the fifth, (25.1 m/sec) headwind, drag simulation. The largest contribution to the drag was from the front face owing to pressure drag. There were low pressure sections on the side of the legs and around the edges of the ears.

Figure 6 shows the simulated, experimental, and sphere drag force versus air speed. The headwind simulation and experimental data were in good agreement, both sphere sets had much lower values, and the crosswind drag was the highest. The largest difference was during the second run (10.1 m/sec) while the smallest difference was during the final run (25.1 m/sec). As we were attempting to determine the predictive power of this method without feedback from experimental data, we made no attempt to

Figure 3. The initial elephant, final elephant, and ambient air temperature for the five wind tunnel runs. The slight increase in ambient air temperature was likely due to waste heat from the wind tunnel fan. This temperature increase as well as other air properties (density, viscosity, thermal conductivity, and specific heat) were all matched between each simulation and experimental run. We used a heat lamp to heat the casting while the wind tunnel was shut down. We turned off the heat lamp before sealing the wind tunnel. We heated the model before the 3.6 m/sec run and again after the 15.2 m/sec run. Four thermocouples on the model measured the temperature and the graph presents the averages of these measurements with standard deviations. The low temperature difference between the elephant and ambient air may have affected the accuracy of the heat transfer coefficient calculation from the 15.2 m/sec and 24.9 m/sec runs.

Figure 4. (A) The meshing of the fluid space around the elephant. Meshing was approximately equal parts 4-node tetrahedral and 6-node wedge elements with a few 5-node pyramid elements with an inflation layer in the boundary region around the elephant. (B) The mesh size was sensitive to curvature to achieve a high quality mesh around regions such as the elephant's ear. This figure shows a close up of the inflation layer near the elephants ear. The inflation region had 16 layers and a maximum height of 1 cm.
fine tune constants (viscosity or density of air) in this simulation to better match the data. Using Equation 2 with $A$ defined as the projected frontal area, the drag coefficients for a headwind and crosswind calculated from the simulation were 0.73 ± 0.02 and 0.80 ± 0.04, respectively. These drag coefficients mean that, depending on its angle to the wind, an elephant would have between 34% and 74% more drag than the spherical approximation would assume. For $3 \times 10^4 \leq Re \leq 1.8 \times 10^5$, the drag coefficient for a sphere is 0.469 ± 0.003 (Haider and Levenspiel, '89).

$$C_D = \frac{F}{\frac{1}{2} \rho A_f U^2} \text{Pa}$$

(2)

$C_D$ is the drag coefficient, $F$ is the drag force, $A_f$ is the projected frontal or side area, $U$ is the air speed, and $\rho$ (rho): air density.

Figure 5 shows a contour map of the pressure on the elephant in a 25.1 m/sec headwind. There was high pressure on the front face and a region of low pressure behind the ears. Both scale bars are 0.10 m long.

Figure 6. Experimental versus simulated drag data. Poor agreement at the lower speeds was likely due to low precision of the sting balance at low loads. The drag coefficient from the simulated data was 0.73 ± 0.02 for a headwind and 0.80 ± 0.04 for a crosswind. We used the front and side projected area in the headwind and crosswind calculations, respectively.

Figure 7 shows a contour map of the aluminum elephant heat flux for the fifth, (24.9 m/sec), headwind, heat transfer run. The highest heat transfer was in the trunk and tusk tips, followed by the legs and ear edges. Figure 8 shows the same run with streamlines on the ears and left foreleg as well as a turbulence field. The figure shows the high velocity around the ear tips and the turbulence in the low heat transfer regions behind the ears and legs. The streamlines show that there was air flow off the chest which passed behind the ear. Figure 9 shows the experimental and simulated data for the heat transfer coefficient. There was good agreement at three velocities but the experimental data was lower than the calculations for the third (15.2 m/sec) and final (24.9 m/sec) runs. In these the sphere matched better. This simulation data allowed us to calculate the regression coefficients, $a$ and $b$, in the heat transfer coefficient equation (Equation 3). For the headwind $b$ was 0.723 ± 0.004 and $a$ was 0.101 ± 0.005, while for the crosswind $b$ was 0.71 ± 0.03 and $a$ was 0.11 ± 0.04. For a sphere $b$ is 0.6 and $a$ is 0.37 (Mcadams, '54). As expected, these coefficients

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and Figure 9 show little difference between crosswind and headwind heat transfer coefficients.

\[ h_c = \frac{k_a}{L} \left( \frac{\rho_a U}{\mu_a} \right)^\beta \]

(3)

\( h_c \) is the heat transfer coefficient, \( k_a \) is the thermal conductivity of the air, \( L \) is the characteristic length (cube root of volume), \( \rho_a \) (rho) is the air density, \( U \) is the air speed, and \( \mu_a \) (mu) is the dynamic viscosity of the air.

The Nusselt–Reynolds correlation makes the heat transfer coefficient equation more compact, \( Nu = aRe^b \), where \( Nu = h_c L/k \) (Bird et al., 2002).

Figure 10 shows the percentage difference between the heat transfer coefficient for a 3.5 m tall elephant and a sphere of equivalent volume. At wind speeds as low as 2 m/sec the sphere was already off by 15%. Towards higher wind speeds which elephants may experience (Garstang et al., ’95), the difference reached 33%.

DISCUSSION

With a few exceptions, we had good agreement between experiment and simulation. At low air speed, the experimental drag force did not follow the expected curve. Reduced precision of the sting balance at low forces may have caused this difference. There was the possibility that a transition to turbulence reduces the drag between the first (5.2 m/sec) and second (10.1 m/sec) runs. While the Reynolds numbers for the first and second runs were in the range of turbulent transitions (\( Re = 100,000 \) (Kreith, ’68)), the simulation was accounting for turbulence and should capture any transitions, therefore we think a transition is unlikely. In the higher numbers we could possibly achieve better agreement with slight changes to either the air density or viscosity; however, we did not attempt these changes as our goal is to test the predictive power of the simulation without reference to a model.

The heat transfer coefficient was in good agreement with the experimental data. The model’s low initial temperature at the third (15.2 m/sec) and final (24.9 m/sec) air speed (Fig. 2) may have caused the low heat transfer coefficient. We also think that these experimental values are incorrect as they were lower than a sphere of the same volume. This lower value was unreasonable given the more elaborate surface of the elephant. At temperatures that close to ambient, the model may have not had a high enough thermal gradient for an accurate heat transfer coefficient measurement. Nevertheless, the trend between experimental and simulated data is consistent.

We need to improve the simulation in several ways to successfully model an actual animal. First, and most easily, the simulation will need a surface roughness parameter input to Fluent. This parameter allows Fluent to account for the additional turbulence effects from surface roughness. Roughness data is accessible on preserved animal skins. Second, we will need to
Figure 10. The percentage difference between the heat transfer coefficient for a 3.5 m tall elephant and a sphere of equivalent volume. At wind speeds less than 2 m/sec the sphere was off by as much as 15%. Towards higher wind speeds elephants may experience, the difference reached 33%. This graph is the absolute percentage difference, thus below 0.5 m/sec the sphere cools faster than the elephant, while the elephant cools faster at all wind speeds above 0.5 m/sec.

Figure 9. The experimental versus simulated heat transfer coefficient, $h_c$, data. There was good agreement for three values when there was a high temperature difference between the model and the ambient temperature (Fig. 3), but when the elephant was cooler there was not as good agreement. The heat transfer coefficient regression coefficients (Equation 3) from the simulated data were $b = 0.705 \pm 0.013$ and $a = 0.118 \pm 0.018$ for a head wind and $b = 0.71 \pm 0.03$ and $a = 0.11 \pm 0.04$ for a crosswind.

Figure 8. A contour map of the heat flux for an elephant in a 24.9 m/sec headwind with streamlines around the ears and left foreleg. Regions of higher turbulence were in blue. There were regions of higher turbulence behind the legs and ears where there is low heat transfer.
divide the animal model into different thermal conductivity regions. The need for this division is evident from the large heat loss from the elephant tusk, where an actual elephant tusk will have much less heat transfer. Conversely, elephants can control the perfused blood to their pinna resulting in higher or lower heat transfer coefficients than the animal body (Wright, '84; Williams, '90). The final and most difficult addition to the simulation is movement. Considering the elephant, Fluent could simulate the additional fluid movement from thermoregulatory ear flapping which can substantially increase heat loss [Phillips and Heath, '92]. The ability to incorporate movement could also allow calculations on thrust and lift from aquatic and flying animals.

Researchers need biophysical parameters to compose models of animals in their environment. Given these parameters and the abiotic conditions of a habitat, researchers can construct models to predicted animal persistence, behavior, fitness, etc. These models are important guides for conservation [Porter et al., 2006; Kearney et al., 2008] as well as agriculture practices [Parsons et al., 2001; Souto et al., 2011]. This paper’s initial verification of Fluent’s ability to correctly simulate complicated biological geometries is a first step towards using the thermal and momentum capabilities of ANSYS Fluent (or equivalent software) to model actual animals. Researchers could use these models to extract important physical coefficients without the need for live specimens. This method promises to save researchers time and expense as well as serve as a highly adaptable tool to calculate many biophysical parameters of organisms.

ACKNOWLEDGMENTS

We would like to thank Pobtsuas Xiong in the University of Wisconsin Digital Media Center for help in scanning the casting. This work was supported by NSF grant IOS–1116343 to W.P.P. and by a Dept. of Zoology summer research support grant to P.N.D.

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