DESIGN AND IMPLEMENTATION OF A PILOT TEST FOR AN INDUSTRIAL PROTOTYPE OF HARMONIC DRIVE USING A FPGA REAL-TIME SYSTEM.

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Abstract. *Harmonic transmission or Harmonic Drive (HD) is a gear based on the principle of transmissibility of mechanical waves. Its development started in 1955 by Walton Musser, he got the first patent on the device. Actually there are four manufacturers of these transmissions in the world and there is little information about its design. This planetary mechanism has a flexible link in its kinematic chain. The principal attributes of this mechanism are an excellent positioning accuracy, high torque capacity, high single-stage reduction ratio and zero backlash. These characteristics allowed several specific applications in the aerospace and aviation industry, as well as in the robotics industry.*

The Universidad Nacional de Colombia built an industrial prototype. To validate the design assumptions, our goal was to estimate the transmission error. However, the used instrumentation required special techniques to make the measurements of the output angular velocity of the transmission; and this task was fulfilled whit a PAC (Programmable Automation Controller), which included an FPGA (Field Programmable Gate Array) as an I/O method and an operating system that allow make real time control.

The article shows the research process followed on the instrumentation of the HD, its implementation, the waveform of kinematic transmission error, analysis of results and a new model for predicting the error. This paper aims to characterize the industrial prototype, more specifically, to use this knowledge to design the next prototype more effectively.

Keywords*: Planetary gear, Harmonic Drive, Instrumentation, Kinematic error.*

1. INTRODUCTION

Harmonic Drive (HD), also named strain-wave gearing, is a gear transmission based on the principle of transmissibility of mechanical waves and this concept was developed by Walton Musser during the mid-1950 (Musser, 1955). This planetary mechanism has a flexible link in its kinematic chain. The principal attributes of this mechanism are an excellent positioning accuracy, small volume, high torque capacity, high single-stage reduction ratio, and high efficiency and nearly zero backlash. It has been used widely in precision pointing and torque conversion purposes with space and weighs constraints. These characteristics had allowed betters applications in the aviation industry, aerospace and robotics industry.

The Universidad Nacional de Colombia built an industrial prototype (Santamaría and Herrera, 2001) with the idea of understanding this type of technology. Actually there are four manufacturers of these transmissions in the world and there is little information about its design. For the many favorable features they have, there are at least two unwanted characteristics, specifically soft torsional stiffness and gear error. The goal of this study is to characterize the behavior in the industrial prototype using techniques of real time processing especially the measure of kinematic error.

2. MODEL OF TRANSMISSION

The principal components of HD are: the wave-generator, the circular spline and the flexible spline (Fig. 1). The wave-generator typically is an elliptically shaped steel core surrounded by a flexible race bearing. The circular spline is a rigid steel ring with teeth machined into the inner circumference. The flexible spline (flexspline) is a thin-walled flexible cup having less teeth (usually tow fewer) on its outer edge than on the inner edge of the circular spline. Upon assembly, the wave-generator is inserted into the flexspline which assumes an elliptical shape at that end. The other end, however, is circular in shape and is attached to the output shaft. The circular spline teeth then mesh with the flexspline teeth at the major axis of the ellipse defined by the wave-generator.

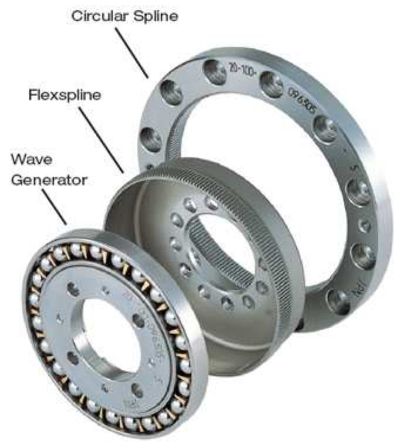

Figure 1. Components of a typical Harmonic Drive

It has been shown in the literature that nonlinear characteristics including transmission error, flexibility, and hysteresis are intrinsic in harmonic drives (Hsia, 1988). The strain-wave gearing is a special flexible transmission mechanism. HD employs a constant deflection wave along a non-rigid gear to let gradual engagement of gear teeth. Because of this unusual gear-tooth meshing action, harmonic drives can bring very high reduction ratios in a very small package. Their typical construction with meshing at two diametrically opposite ends gives them many useful properties. The transmission is designed such that several teeth are engaged at any given time making backlash virtually zero. But since a link in the kinematic chain is a non-rigid gear it reduces stiffness, and brings a transmission error into the system.

Every harmonic drive is distinguished by his transmission ratio, N, whish describes its position, velocity and torque behavior. Specifically, given a know rotation on two of the three Harmonic-Drive ports as well as a value for N, the ideal rotation of the third harmonic-drive port can be predicted by the equation (Tuttle, 1992):

$$\theta_{wg} = (N+1)\theta_{cs} - N\theta_{fs} \tag{1}$$

Where θ_{wg} is the rotation of the wave-generator, θ_{cs} is the rotation of the circular spline, and θ_{fs} is the rotation of the flexspline. All tree rotations in this equation are defined in the same frame of reference; similarly, the derivative of this relationship yields a similar velocity constraint:

$$\omega_{wg} = (N+1)\omega_{cs} - N\omega_{fs}, \tag{2}$$

Where ω_{wg}, ω_{cs} and ω_{fs} represent the angular velocities of the three harmonic-drive components relative to the same velocity reference.

3. MODEL OF KINEMATIC ERROR

In applications requiring high positional accuracy, the inherent kinematic errors manifested by harmonic drives expose the deficiencies of an ideal transmission model. The kinematic errors, θ_{err}, is typically measured by subtracting the rotation at the output from the input rotation scaled by the ideal ratio for the given transmission configuration (Tuttle and Seering, 1993):

$$\theta_{err} = \frac{\theta_{in}}{N} - \theta_{out} \tag{3}$$

Typically the input port is the wave generator, the output port is the flexspline and the circular spline is constrained. Making use of the kinematic model, Hsia in 1988 could consider how structural errors vary as functions of the geometric parameters of the mechanism, specifically the relationship between the major and minor axis of the ellipse. From this analysis, it shows that there are inherent positioning errors associated with the harmonic gear drives irrespective of manufacturing and assembly errors. These errors should be of particular concern when this mechanism is used in precision pointing applications.

Then in 1991 Nye and Kraml consider the gear error causes. They found that these errors are attributed, mainly, to: tooth placement errors on the flexspline, tooth placement errors on the circular spline and lack of concentricity between flexspline and circular spline. Nye and Kraml has demonstrated that typical kinematic-error signatures, vary periodically at once and twice the rotational frequency of the wave-generator and subsequent harmonics. Based on this result, the harmonic-drive kinematic error θ_{err} was modeled as sum of sinusoidal functions. For most cases, the first three sinusoidal terms were adequate to describe the kinematic-error profile (Hidaka et al, 1989):

$$\theta_{err} = A_1 \sin(\theta_{wg} + \phi_1) + A_2 \sin(2\theta_{wg} + \phi_2) + A_3 \sin(4\theta_{wg} + \phi_3) \quad (4)$$

Where θ_{err} , is the wave-generator angle, and A_n and ϕ_n are the measured amplitudes and phases of the kinematic error components at each frequency. A Fourier series is proposed for a model:

$$\theta_{err} = \sum A_k \sin\left(2k\theta + \varphi_k\right) \quad (5)$$

Here θ_{err} is transmission error, A_k .is amplitude of respective harmonic function, θ is input shaft position, and φ_k is phase lag of respective harmonic function. When the gear is in operation, a transmission error produces a speed ripple as a derivative of the transmission error. Frequencies of the speed ripple become 2nd multiples of the input shaft turning frequency.

Kennedy and Desai (2003) say that kinematic error has a significant effect on the torque transmission characteristics of HD, they found that compensating for coulomb friction using the torque required to maintain slow velocity eliminated almost all the effects of kinematic error. Calvente in 2006 gave a characterization of the kinematic error using FEA and found that the simulation performed without restrictions on the wave generator show how its center moves during operation of the HD resulting in the appearance of vibrations that are transmitted on the output shaft.

In order to reduce the speed ripple Godler et al (1994) proposed an active repetitive control and proved that the Fourier series is an appropriate mathematical model to represent the transmission error of the strain wave gearing. Later Taghirad and Belanger (1996) development a series of models with increasing complexity to describe the harmonic drive behavior. Their most complex model involved kinematic error, nonlinear stiffness, and gear-tooth interface with frictional losses. In 1997 Taghirad and Belanger proposed a robust torque control; its objective is well-suited to the general H_∞, problem. Miyazaki and Ohishi in 1999 treated the angular transmission error as harmonic function of motor angular (6) and propose the model showed in figure 2.

$$\theta_{out} = \frac{1}{N}\theta_{in} + A\sin\left(2\theta_{in}\right) \quad (6)$$

The complete model of the transmission and the power unit is:

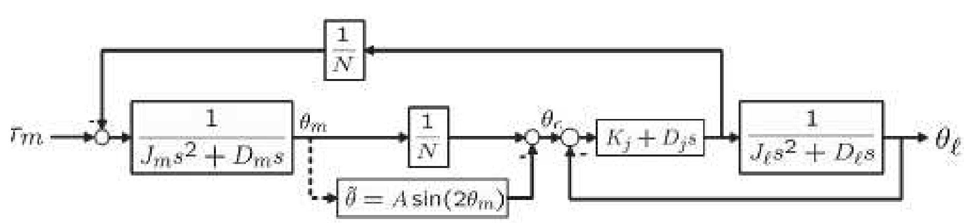

Figure 2. Non-linear model f HD into a motor model

Ghorbel et al (2001) decomposed the kinematics error. They present a more precise characterization and propose that the kinematic error is mostly dominated by two major components. The first component, $\tilde{\theta}_p$, is a basic component that is ‘‘pure’’ kinematic error resulting from the kinematic structure of the harmonic drive. The second component,

$\tilde{\theta}_x$, is mostly due to the stiffness properties of the drive. Consequently, the expression of the kinematic error introduced in (3) could actually be decomposed into

$$\theta_{err} = \tilde{\theta}_p + \tilde{\theta}_s \tag{7}$$

The flexibility induced component of the kinematic error:

$$\tilde{\theta}_p = \frac{a_0}{2} + \sum_{n=1}^{k} \left[a_n \cos(n\theta_{in}) + b_n \sin(n\theta_{in}) \right] \tag{8}$$

Where

$$a_n = \frac{1}{\pi} \int_0^{2\pi} \tilde{\theta}(\theta_{in}) \cos(n\theta_{in}) d\theta_{in} \tag{9}$$

$$b_n = \frac{1}{\pi} \int_0^{2\pi} \tilde{\theta}(\theta_{in}) \sin(n\theta_{in}) d\theta_{in} \tag{10}$$

The flexibility-induced component in the kinematic error is mostly the source of the high-frequency components of kinematic error reported by Ghorbel et al (2001).

Harmonic Drive should have a robust speed control algorithm to suppress vibration phenomena. When the frequency of the angular transmission error coincides with the resonant frequency of control system, the joint actuator having planetary gear may generate large torsional vibration torque. In this sense Gandhi and Ghorbel (1999) use a Closed-loop compensation of kinematic error. Miyazaki and Ohishi (2002) introduce a robust speed control. Lu and Lin propose a Disturbance-observer-based adaptive feedforward control when the control scheme consists of the internal model control (IMC) and an adaptive feedforward cancellation (AFC) based on a disturbance observer (DOB). Han et al (2008) reduce de velocity ripple using peak filter with acceleration feedback.

4. DESCRIPTION OF EXPERIMENT

The goal of this study is to characterize the kinematic error presented in the industrial prototype. This harmonic drive has a wave generator planetary-type, tow rollers instead of an elliptical bearing, and the flexspline use a dynamic circular spline coupled to output shaft, instead of a flexspline cup type (Santamaria and Herrera, 2001). It has 56.5 of transmission ratio. We can see the image of industrial prototype of harmonic drive (Fig. 3).

Figure 3. Industrial prototype of harmonic drive builds by Universidad Nacional de Colombia

High resolution encoders are used to measure the angular positions of input and output shaft. The input encoder has a resolution of 0.075 degrees and the output encoder has a resolution of 0.015 degrees. The motor is driven by a Variable Speed Drive communicated with a Programmable Automation Controller (PAC) it combines an embedded real-time processor; running a real-time operating system (RTOS);and a Field Programmable Gate Array (FPGA). This system is a high-performance, reconfigurable chip. It is programmed with LabVIEW. Then the LabVIEW code is converted in VHDL code to configure the FPGA chip. The FPGA allows simultaneously encoder readings at high speed (the maximum frequency of signal variation of the encoder is 5 MHz.). Meanwhile the RTOS performs data processing like calculations of fast Fourier transform. In addition, this hardware has two main benefits: 1) the adequacy of the encoder signal is less vulnerable to electromagnetic noise due to the card that is used allows differential signal acquisition. 2) The implementation of the algorithms is done in a deterministic environment with precision time.

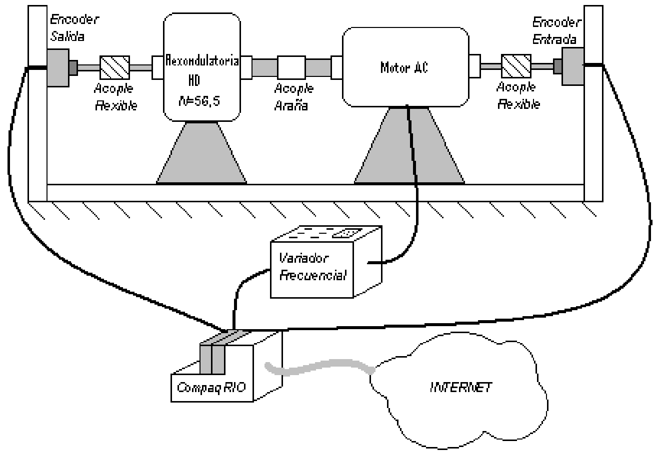

Figure 4. Scheme of Flexondulatoria (H. D.) testing apparatus

PAC communicates via Ethernet with a computer which holds data from an experiment for later analysis on multiple conditions of use of the transmission (Fig. 4). In the experiment the software delivers: error range, mean error, standard deviation, variance, input speed in Hertz (also in RPM), number of peaks found in the spectrum, location and value of the peaks found.

The kinematic error is calculated using the algorithm made by Tuttle (1992):

1. By sending a constant signal to the variable speed drive, rotate the harmonic drive at the slowest velocity at which resonance vibrations is minimal.
2. Collect input and output position information from the encoders at equally spaced time intervals over a given motion range.
3. Calculate the position error, θ_{err} , using the equation (3), the result is expressed in degrees.
4. Plot the resulting position-error signal versus the number of input, or wave-generator, revolutions to analyze the shape of the error signature.
5. Linearly interpolate the time-based position-error signal to produce data points that are equally spaced by input rotation.
6. Take a Fast Fourier Transform of the interpolated data vector to identify the frequency components of the error signature in terms of input revolutions.

5. RESULTS OF THE KINEMATIC TRANSMISSION ERROR

In order to measure the position error, the engine must have a constant speed, and then wait for the input shaft reaches certain angular position to start the measurement. All samples are taken from the same start angle until complete a lap on the output shaft. The experiment was repeated several times to ensure repeatability of samples and reduce the effect of measurement errors. Figure 5 illustrates the position-error waveforms observed and Fig. 6 show the FFT result.

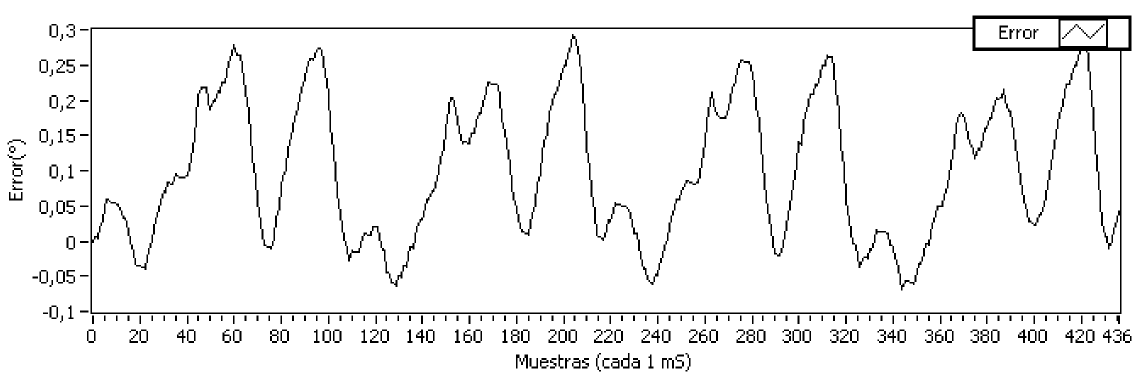

Figure 5. Position-error waveform at two input rotation (counterclockwise)

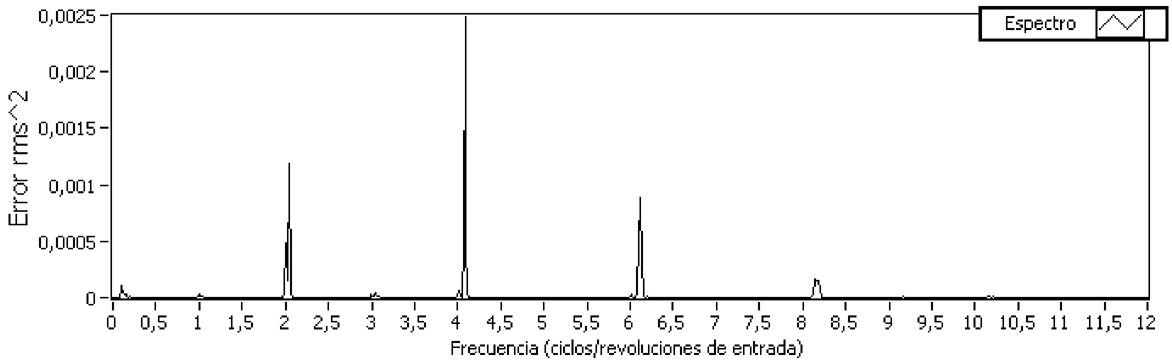

Figure 6. FFT of position error in cycles per input revolution (counterclockwise)

Figure 5 shows a periodic signal for two full rotations of the input port, the peak to peak value of this signal is 0.35 degrees. To use this type of transmission for applications with higher accuracy is necessary to include a system that reduces the kinematic error. This figure was taken when the output shaft was spinning counter-clockwise. Figure 7 shows that the harmonics of the signal are close to the values proposed by Hidaka *et al.* (equation 5). Note that the peaks present in the spectrum are close to orders pairs: 2, 4, 6 and 8 but not exactly in that place, per example the highest peak is located beyond the eighth order.

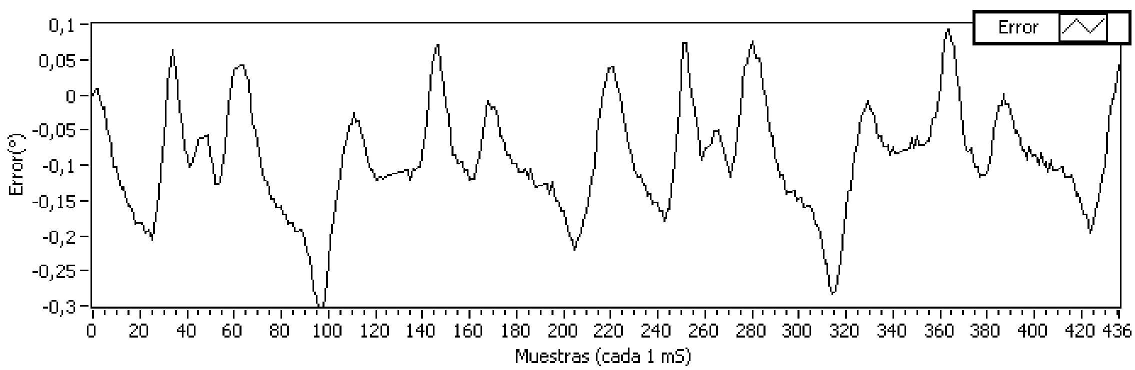

Figure 7. Position-error waveform at two input rotation at clockwise

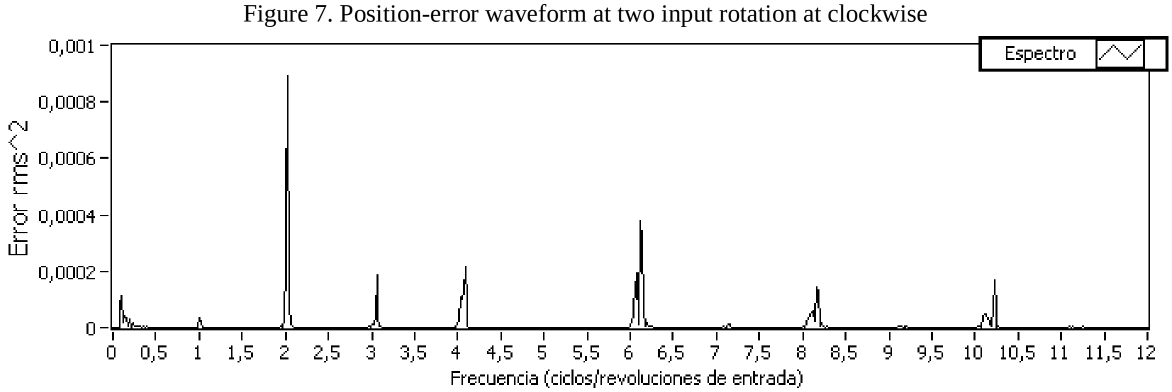

Figure 8. FFT of position error in cycles per input revolution at clockwise

From figures 6 and 8 can be seen that the direction of rotation affects the harmonics present in the position error. In the position error waveform there are more harmonics (7) at clockwise that harmonics (4) at counterclockwise. After performing a significant number of samples we found that in all cases the harmonic frequencies were higher than those pred7icted for Hidaka *et al.* In this model the source of failure is the loss of concentricity between the circular spline and the flexspline. With the change or rotation the amplitude of the signal changes dramatically, for example counterclockwise the maximum peak is 0.0025 while clockwise the maximum value is 0.0009. In both graphs the harmonics moves to increasing frequency as can be seen near 8X. The spectrum of kinematic error signals present peak at 3X in both directions. We also see a small peak in 1X (rotation speed). The difference in the spectra depending on the direction of rotation can be explained by problems of ensemble, but the shifts in the harmonics are not explained by literature.

Several experiments have lent further insight into the repeatability and variation of typical gear-error waveforms. The most important conclusion is that the harmonics are not presented precisely in the frequency predicted by the equation 5. It is necessary to propose a new model to justify the observed behavior. One possibility being studied currently assessing whether the disruption of transmission error moves with the rotation of the wave generator and this would explain why the peaks in frequency are moved towards higher values.

To justify the shift in frequency peak we consider a point moving with the flexspline (moving with the output port), as would happen with a crack in this element. The flexspline is a thin wall element that supports large stresses due to loading and deformation forced by the wave generator so the flexspline is critical in the design of HD.

The proposed model for the error is:

$$\theta_{err} = \sum_{k=1}^{n} A_k \sin\left(2k\theta_{in}\left(\frac{N+1}{N}\right)+\varphi_k\right) \tag{11}$$

We are currently carrying out extensive testing to try to validate the proposed hypothesis. These tests vary the speed of entry and the direction of rotation for different types of wave generators. As a result of this investigation, we confirmed the expected position-error frequency distribution over characteristically frequencies. This is a starting point for analyzing the transmission designed by the research group.

6. CONCLUSIONS

The zero backlash and high transmission ratio in one stage of Harmonic Drives make this type of transmission an excellent choice for applications where accurate motion control in a small space is needed, for example joints in robotic arms. The few disadvantages found basically the transmission error and low torsional stiffness can be reduced or eliminated by methods of robust control and nonlinear control.

We found differences in behavior with the change of direction of rotation; these differences can be explained by errors in assembly and manufacturing. This effect should be taken into account when designing the control needed in this type of device.

The spectrum of the kinematic error interpolated by the rate of entry (order analysis) remains constant, indicating little or no influence of speed on kinematic error, at least at the speeds at which the experiments were performed (between 48 and 600 RPM). Because the control of velocity is made by a Variable Speed Drive, this method can not guarantee a steady speed at very slow speeds to verify the models proposed by Ghorbel et al (2001).

In this paper we present a new model based on the rotation of the wave generator was presented to explain the complex phenomenon of kinematic error in harmonic drive gears. The model is much accuracy than the model used previously. Hence the model will be very useful for control applications with harmonic drives.

The tests developed by this study could determine the influence of material's flexspline and the tooth's profile in the kinematic error and thus develop a scheme of mechanical design based by experimentation.

7. ACKNOWLEDGEMENTS

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