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Sensing the torque in a robot's joints

A new sensor can accurately measure the torsion moment in a direct-drive robot, even in the presence of strong overhang, thrust forces, and bending moments.

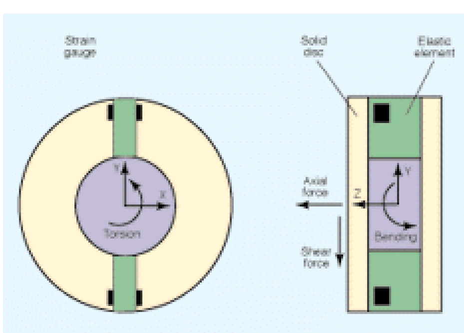
By Farhad Aghili, Martin Buehler, and John M. Hollerbach

As direct-drive robots become increasingly popular, there is a need for high-performance motion controllers that can achieve dynamic accuracy in practice. A direct approach to control is possible if joint torque sensors are used to measure the external loads on all joints. Positive feedback of the torque signal can then compensate for the dynamics of the robot manipulator. Although this approach offers a far simpler control solution than model-based control, accurate joint torque measurements encounter several challenges.

In the design of robot manipulators, it is desirable that much of the torque/force reaction of the link load on the joints come in the form of nontorsional components, because actuation then takes less effort. Planar robot arm designs, for instance, prevent gravity torques from acting on the joint motors. However, since the torque sensor is directly attached to the motor's distal link, it has to bear those potentially large nontorsional components of the generalized force/torque vector at the joint. The first challenge is to measure the torsion torque faithfully without influence from any of the nontorsional components. The second challenge relates to the sensor stiffness. To increase the signal-to-noise ratio of the sensor, it is desirable to design a structure that generates a large strain for a given load torque and therefore has a high sensitivity. However, the resulting compliance introduces a joint angle error that should be minimized. Thus, there are two conflicting requirements—high mechanical stiffness and high torque sensitivity—and both quantities need to be maximized simultaneously.

Insensitivity to the five nontorsional force/torque components can provide high corresponding stiffnesses and assign a higher maximum strain due to torsion, resulting in high torsional sensitivity and natural decoupling. In general, assuming a linear elastic material, the resultant strains at several locations on the elastic sensor body are related to the applied forces and torques by the compliance matrix. The torsion torque can be derived via a linear combination of the strain measurements through identical gain magnitudes by designing a symmetric sensor geometry and placing the strain gauges properly. Then, one can use the additive properties of Wheatstone bridges to achieve decoupling of the output signal from the nontorsional force/torque components without the need for any subsequent calculation. The resulting advantages are a reduction of instrumentation and the number of wires, by completing the Wheatstone bridge wiring locally in the sensor, and a simplification of calibration.

Two main conditions in practice violate the ideal assumption of exact symmetry among the measured strains. First, strain gauges exhibit variations in their gauge factor, which is the ratio of the fractional change in resistance to strain. Second, the strain gauges are placed on areas with high strain gradients, which makes the gauge outputs sensitive to placement errors. Therefore, exact cancellation of the nontorsional components may not be achieved with the theoretical gain vector alone. The strain sensitivity to the nontorsional force/torque components has to be held to a minimum also by mechanical design. This design requirement is consistent with a decoupling property of the sensor. It is worth noting that cylinders (which are used mainly in commercial torque sensors) produce two times more stress/strain when subjected to bending than to torsion. Torsional deflection degrades the position accuracy of the joint angle controller. Therefore, one of the critical design challenges is to maximize the stiffness while maintaining high sensitivity.



This torque sensor has two elastic elements linked to two rigid discs that carry the external loads.

These contradictory requirements can be captured by defining a performance index, called structure efficiency, as the ratio of the (local, maximum) strain to the (overall) torsional deflection caused by the same torque. This dimensionless index is a decisive factor in the sensor design and should be maximized. It is independent of material properties, and captures the ratio of the local and global strains. The index is maximized in elastic structures that produce high strain concentration in torsion. In theory, there is no limit on the strain concentration in an elastic body. However, high strain concentrations occur in very small areas, which may be smaller than the physical size of available strain gauges. Moreover, since strain gauges average the strain field over their area, the detected strain may be significantly lower than the calculated maximum. Therefore, it is important to generate high strain over a sufficiently large area. This objective is difficult to formulate analytically, but it can be achieved by finite element methods.

Introducing a torque sensor into a robot joint adds flexibility. Although torsional flexibility can be compensated for by sophisticated controllers, deflection in the other axes is more problematic. Consequently, another design criterion dictates high stiffness in nontorsional directions.

Fortunately, the requirements for low deflection and low strain sensitivity for nontorsional components are consistent. Addition of a torque sensor to a robot joint must not require redesign of the joint and should result in a minimal change in the manipulator's kinematics, in particular the link offset. Hence, a shape with a small width is desirable.

Minimizing the effects of thermal stresses is a design factor that cannot be ignored. Motors are sources of heat that flows from the motor to the attached link through the body of the sensor. Therefore, it is desirable to have an axisymmetric design that constrains the heat to flow in the axial direction, where no position constraint usually exists. The common hub-sprocket designs are prone to thermal stresses because of the temperature difference between the hub and the wheel. Since the sensor is specifically designed for a direct-drive motor with a hollow shaft, flange mounting is preferred.

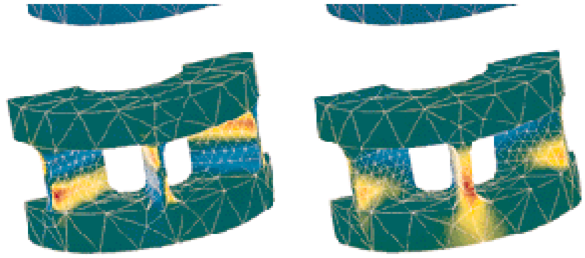
The body of the sensor should be designed for ease of manufacture. It should be a monolithic structure; that is, the body should be machined from a solid piece of metal. This decreases the hysteresis and increases the strength and repeatability of the sensor.

So far, only geometric properties of the elastic body have been considered. Nevertheless, the stiffness and sensitivity characteristics of the torque sensor are also determined by the material properties. The maximum allowable strain for foil strain gauges is typically 3 percent, which is at least one order of magnitude higher than that of industrial metals, making the materials the limiting factor for sensitivity. Furthermore, the stiffness depends linearly on the Young's modulus of the material. By virtue of Hooke's law (stress is linearly related to strain via the modulus of elasticity), one can conclude that high sensitivity and stiffness are achievable simultaneously only by use of a high-strength material.

Because a linear response is desired from the sensor, the chosen sensor material must have a linear strain-stress relationship. Steel is the best available industrial material that has good linearity properties within a large stress range. Moreover, due to the oscillatory nature of the loading, steel can work with infinite fatigue life as the allowable strains are determined based on the endurance limit. The endurance, or fatigue, limit is the maximum stress under which mechanical failure will not occur, independent of the number of load cycles. Only ferrous metals and alloys have an endurance limit.

Design via FEA

The sensor design must optimize, and trade off among several conflicting design criteria. Also, many design iterations are required to arrive at a final design. Despite this complexity, it is possible to arrive at a novel basic sensor design. Similar to a thin-wall beam, the structure should have a large second moment of area around its x axis compared to that around its z axis. As a comparison, it is worth noting that a simple cylinder is 0.77 times less stiff in bending than in torsion. An elastic structure resulting from these considerations is shown above. It consists of two flexible elements attached to two rigid discs that carry the external loads. The flexible elements are subjected to a combination of torsion and bending when torque is applied. The produced strain is the superposition of both torsion and bending strain contributions. The strain contribution of bending depends upon the beam's length (the distance between the two discs), which adversely influences the compliance.



Results of the finite-element analysis of the torque sensor show von Mises stress in megapascals (top, left), tangential displacement in microns under torsional torque load (top, right), axial micro strain under torsional torque load (bottom, left), and axial micro strain under bending torque load (bottom, right).

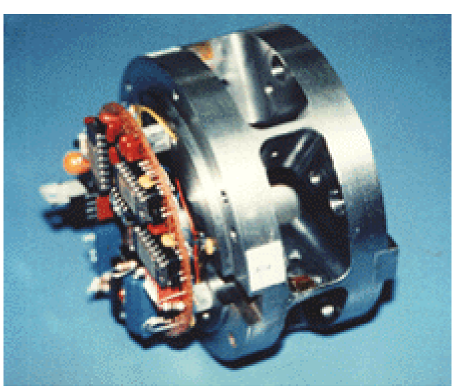
Thin-section rectangular bars experience high stress/strain concentrations under torsion loads, which yield high sensitivity without sacrificing stiffness. This fact suggests that an appropriate structure should be primarily stressed by torsion. The elastic structure exhibits high bending stiffness around the x axis. However, its poor stiffness around the y axis is a drawback. This problem can be solved simply by adding more wing pairs. That improves the uniformity of the bending stiffness along different axes as well as the body stiffness, but complicates manufacturing. In our design we consider six wings.

Once we had determined the basic shape of the sensor, we used the finite-element-analysis (FEA) capabilities of I-DEAS from Structural Dynamics Research Corp. to finalize the dimensions so that performance would be optimized. FEA was also used to select the location and proper size of the strain gauges. For maximum sensitivity, strain gauges should be located where the maximum induced strains due to the torsion load occur. Since the strain field is averaged over the area covered by the strain gauges, it is very important first to determine the loci of the peak strain, and second to ensure the creation of a sufficiently large strain field. The sensor body is modeled by solid elements. Since the body is symmetrical in geometry and boundary conditions, it suffices to analyze half of it, provided that adequate position constraints are imposed on the nodes of the cutting plane.

To simplify the FEA, small geometric features of the body were suppressed. Several load cases were investigated, corresponding to axial and shear forces as well as bending and torsion moments. In our application, the maximum forces and moments are 1,000 Newtons and 300 Newton meters, respectively. A preliminary stress analysis showed that the axial and shear forces have negligible elastic effects, because they produce a uniform strain/stress field in the elastic body, resulting in a very small maximum strain. In fact, the bending moment is the critical nontorsional component, and consequently two load cases corresponding to the external torsion and bending torques are established for FEA. It is important to note that in robotic applications the maximum angular deflection due to external torques (which is amplified by the robot links) is a more restrictive constraint than linear deflection due to the forces.

At the postprocessing stage, the following were selected as design benchmarks: the tangential and axial displacement of the disc's outer diameter; the principal strain in the axial direction and parallel to the gauge axes, due to both load cases; and the maximum von Mises stresses/strains due to a combination of all load cases. These design criteria were checked with the FEA results to modify the geometry of the sensor iteratively. The FEA results of the elastic body's final design are shown on the facing page. The worst-case von Mises stress, i.e. the combination of the two load cases, is shown at top left, where its maximum occurs at 150 megapascals. This is close to the endurance limit of mild steel with a reasonable factor of safety. The figure at top right illustrates the tangential displacement fields by which the torsional stiffness is carried out. The axisymmetric pattern in the figure confirms the correctness of the imposed boundary conditions.

The two lower figures show the strain contour in the axial direction in which the strain gauges are oriented, for the first and second load cases, respectively. The FEA results demonstrate that the strain sensitivity in torsion is seven times higher than that in bending, while the bending stiffness is 18 times higher than the torsional stiffness. The design achieves a structure efficiency of 0.68.



The torque sensor prototype was machined from a solid steel rod.

The torque sensor prototype was machined from a solid steel rod. The 95-millimeter outer diameter and the mechanical interface dimensions were designed to match our particular motor. Foil strain gauges (SG-3/350-LY41 from Omega) were carefully cemented at the locations determined by FEA. Instrumentation amplifiers built into the sensor boost the signal level of the Wheatstone bridge output before A/D conversion. We took advantage of the hollow motor shaft, which is common in direct-drive motors, to locate the electronic circuit board beside the sensor. The local signal conditioning provides a stronger output signal and improves the signal-to-noise ratio. Moreover, since the electronic circuit is totally enclosed by the motor's hollow shaft, it is well shielded from the powerful magnetic noise created by the motor.

Calibration and Experiments

In order to characterize the linearity and sensitivity of the sensor, static torsional and bending torques were applied in an experimental apparatus built for these static tests. One side of the torque sensor was affixed to a bracket, while two aluminum bars were attached radially and axially to the other side. The ends of the bars were connected to mechanical levers via ropes in which load cells (MLP-50 from Transducer Techniques) were installed. The lever varied the tension in the cord gradually between zero and maximum. During loading and unloading, the load cell and the torque sensor outputs were sampled by a two-channel data acquisition system. All deviations from linearity were less than 0.2 percent full scale. This is close to the accuracy of 0.1 percent full scale of the reference force transducer used for calibration. In addition, worst-case cross-sensitivity was measured to be 0.6 percent. This confirms that the sensor effectively decouples the effect of the nontorsional components on the measured torque signal.

Dynamic testing served mainly to validate the FEA results on which the stress analysis was based. The experiment was arranged to extract the stiffness of the sensor prototype. Again, the sensor was held rigidly by the bracket while a steel disc was flanged to the other side. The disc was massive, with an inertia of 0.24 kilogram per square meter, and the whole system behaved like a second-order system. To detect all the vibration modes corresponding to all compliance directions, the cross-sensitivity was deliberately increased by electrically bypassing the strain of all strain gauge pairs except one. Therefore, the torque sensor no longer had the decoupling property, and its output was the summation of all torque/force components weighted by their corresponding gains. The system was excited impulsively by a hammer, and a data acquisition system recorded the subsequent vibration with a sampling rate of 3.2 kilohertz. The modal frequencies associated with the torsion and bending compliances occurred at 150 hertz and 980 hertz, respectively. The corresponding torsion and bending stiffness were 0.24 and 4.8 meganewton meter per radian. A comparison with FEA predictions revealed an acceptable 20 percent error.

Motivated by the need for joint torque sensing in robots, we have designed a new sensor whose key features are its extremely high stiffness and its insensitivity to nontorsional force/torque components. It has been shown that the maximum strain sensitivity to torsion can be maintained without sacrificing torsional stiffness, if elastic body exhibits strain concentration to torsion loads. The sensor also has been designed so the effect of nontorsional moments and forces on the local strains is minimal, which has been achieved by a combination of mechanical design and electrical summation of strain gauge signals. The sensor has been tested extensively. Tests confirm that the sensor meets all its design goals, and is well-suited as a torque-sensing device in robots or other industrial applications.

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Farhad Aghili is a consultant with the Canadian Space Agency in Montreal. Martin Buehler is an associate professor in the Department of Mechanical Engineering at McGill University, also in Montreal. John M. Hollerbach is a professor in the Department of Computer Science at the University of Utah in Salt Lake City.

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