HI 20.3-2010

Hydraulic Institute Guideline for

Rotodynamic (Centrifugal and Vertical) Pump

Efficiency Prediction

Hydraulic Institute

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Approved by Hydraulic Institute
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Hydraulic Institute Guideline

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Foreword (Not part of Guideline)

Purpose and aims of the Hydraulic Institute

The purpose and aims of the Institute are to promote the continued growth and well-being of pump users and manufacturers and further the interests of the public in such matters as are involved in manufacturing, engineering, distribution, safety, transportation and other problems of the industry, and to this end, among other things:

a) To develop and publish standards, guidelines, or educational materials on the application, installation, operation, and maintenance of pumps and pumping systems;

b) To collect and disseminate information of value to its members and to the public;

c) To appear for its members before governmental departments and agencies and other bodies in regard to matters affecting the industry;

d) To increase the amount and to improve the quality of pump service to the public;

e) To support educational and research activities;

f) To promote the business interests of its members but not to engage in business of the kind ordinarily carried on for profit or to perform particular services for its members or individual persons as distinguished from activities to improve the business conditions and lawful interests of all of its members.

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1) Hydraulic Institute Guidelines are adopted in the public interest and are designed to help eliminate misunderstandings between the manufacturer, the purchaser, and/or the user and to assist the purchaser in selecting and obtaining the proper product for a particular need.

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An Institute Guideline consists of explanatory data and other engineering information of an informative character not falling within the classification of Institute Standards.

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Comments from users of this guideline will be appreciated, to help the Hydraulic Institute prepare even more useful future editions. Questions arising from the content of this guideline may be directed to the Technical Director of the Hydraulic Institute. The inquiry will then be directed to the appropriate technical committee for provision of a suitable answer.

If a dispute arises regarding contents of an Institute Guideline or an answer provided by the Institute to a question such as indicated above, the point in question shall be sent in writing to the Technical Director of the Hydraulic Institute, who shall initiate the Appeals Process.

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Revisions

The Guidelines of the Hydraulic Institute are subject to constant review, and revisions are undertaken whenever it is found necessary because of new developments and progress in the art. If no revisions are deemed necessary after five years, the guidelines shall be reaffirmed using the HI balloting procedure.

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This guideline does not contain a complete statement of all requirements, analyses, and procedures necessary to ensure safe or appropriate selection, installation, testing, inspection, and operation of any pump or associated products. Each application, service, and selection is unique with process requirements that shall be determined by the owner, operator, or his designated representative.

Units of measurement

Metric units of measurement are used, and corresponding US customary units appear in brackets. Charts, graphs, and sample calculations are also shown in both metric and US customary units. Since values given in metric units are not exact equivalents to values given in US customary units, it is important that the selected units of measure to be applied be stated in reference to this guideline. If no such statement is provided, metric units shall govern.

Committee list

This Hydraulic Institute Guideline was produced and approved by a working committee that met many times to facilitate its development. At the time the guideline was approved, the committee had the following members:

Chair – Arnie Sdano, Pentair Water - Engineered Flow
Vice-chair – Bill Beekman, Consultant

Committee Members	Company
Graeme Addie	GIW Industries, Inc.
Ed Allis	Peerless Pump Company
Tom Angle	Weir Specialty Pumps
Fred Buse	Consultant
Jack Claxton	Patterson Pump Company
Mick Cropper	Sulzer Pumps (US) Inc.
Randy Ferman	ekwestrel corp
Al Iseppon	Pentair Water, Residential Flow/Ag-Ind
Jim Roberts	ITT - Residential & Commercial Water
Aleks Roudnev	Weir Minerals North America
Greg Towsley	Grundfos Pumps USA
Fred Walker	Weir Floway, Inc.

20.3 Rotodynamic (centrifugal and vertical) pump efficiency prediction

The major influences on rotodynamic pump efficiency are pump size, specific speed n_S (N_S), and the type of pump selected to meet the service conditions.

The efficiency prediction charts (see Figures 20.3a, b, c, and d) relate to industrial-class pumps designed, manufactured, and tested in accordance with recognized industry standards.

The following can influence efficiency deviations:

a) **Types of pumps:** There are many different types of rotodynamic pumps with unique configurations and features to meet specific service conditions, e.g., stock, sewage, slurries, etc., all of which by virtue of their specific speed and design have less than optimum attainable efficiency. Table 20.3 provides some details on how pump features influence efficiency.

b) **Surface roughness:** Efficiency increase due to improvements in waterway surface finish is dependent on pump specific speed and size. Typically, surface finish improvements are economically justifiable for small and low specific speed pumps. Shown in Figures 20.3g and h is the efficiency improvement that can be expected when the water passages are upgraded. This would be added to the general predicted efficiency.

   Surface roughness is determined by:

   - Casting quality
   - Castability of the materials of construction
   - Amount of casting grinding and polishing performed
   - Use of specialty coatings

c) **Internal clearances:** Pump wear ring clearances can have a major influence on efficiency, particularly for low specific speed pumps [$n_S < 29$ ($N_S < 1500$)]. See Figures 20.3i and j. Internal clearances are determined by:

   - Design compromise for manufacturability
   - Galling properties of the materials of construction

d) **Mechanical losses:** Bearings, lip seals, mechanical shaft seals, packing, etc., all consume power and reduce the pump efficiency. Small pumps (less than 11 kW [15 hp]) are particularly sensitive to these mechanical losses.

e) **Pumpage:**

   - Viscosity: Liquids with a viscosity higher than water have a detrimental effect on efficiency. Refer to ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance* for viscosity correction.
   - Solids size: Low concentrations (below 10% by weight) of random-sized solids and tramp material in the liquid will not have a detrimental effect on efficiency. However, if the pump configuration has to be compromised such that the impeller and casing passageways will be large enough to improve the solids-handling characteristics, then the efficiency will be reduced. For example, sewage pump impellers that handle solids and rags often have only one, two, or three vanes of unique shape, causing the efficiency to be lower.
   - Slurries: Larger concentrations (above 10% by weight) of solids in liquids cause reductions in pump efficiency. The pump supplier should be consulted when making efficiency corrections for slurries (refer to ANSI/

HI 12.1-12.6 *Rotodynamic (Centrifugal) Slurry Pumps for Nomenclature, Definitions, Applications, and Operation*).

f) **Special impeller designs:**

- High suction specific speed, S (N_{ss}) > 215 (11,000), could reduce the attainable efficiency by upwards of 3 points (the effect is lessened as specific speed is reduced)

- Desired curve shape, such as head rise to shutoff or steepness of head curve, can reduce the attainable efficiency

g) **Impeller diameter trim:** Reduction in efficiency due to impeller diameter trim must be expected. Efficiency reductions can range from 1 to 6 points by trimming to the minimum diameter. High specific speed pumps usually have greater reductions in efficiency due to trim than low specific speed pumps.

h) **Thrust balance:** Pumps often use varying methods of hydraulic thrust balance, which may reduce the pump efficiency.

i) For multistage vertical turbine (diffuser) type pumps, there will be a difference between attainable bowl efficiency and attainable pump efficiency.

- **Staging effect:** Due to hydraulic losses at the inlet and discharge of the pump, single-stage attainable efficiency could be as much as 6 points below the bowl efficiency. This difference decreases as the number of stages increases. Typically this correction applies to four stages and less.

- **Pump efficiency:** Overall pump efficiency will be less than attainable bowl efficiency due to hydraulic losses in the column piping, discharge elbow losses, and bearing losses. Because of the variability in this style pump, this correction needs to be analyzed for each application.

Table 20.3 — Pump types and factors that influence efficiency

Pump description		n_s (N_s) range	Pump type	Factors that influence efficiency
A	Slurry pumps, end suction	15 – 38 (800 – 2000)	OH0, OH4, & OH5	Manufacturing considerations, hydraulic compromise to improve wear
B	Solids-handling, end suction	13 – 105 (700 – 5500)	OH0, OH2, & OH5A	Efficiency is compromised in order to pass large solids
C	Submersible sewage, end suction	13 – 105 (700 – 5500)	OH8A & OH8B	Efficiency is compromised to pass large solids; submersible motor losses are not included, but usually the efficiency of a submersible motor is several points lower than that of a conventional motor
D	Stock, end suction	13 – 77 (700 – 4000)	OH1	Open impeller designed for pumping fibrous air-entrained mixtures compromises efficiency
E	Horizontal multistage pumps: axially split, segmented ring diffuser barrel	13 – 38 (700 – 2000)	BB3, BB4, & BB5	Clearances, interstage bushings, shaft stiffness, and surface finish

Table 20.3 — Pump types and factors that influence efficiency (*continued*)

	Pump description	n_S (N_S) range	Pump type	Factors that influence efficiency
F	ASME B73 API end suction End suction – small	5.8 – 58 (300 – 3000)	OH1 OH2	Low-horsepower pumps with special mechanical seals that cause high mechanical losses will have efficiencies lower than stated in the figures
G	End suction – large >0.3 m^3/s (5000 gpm)	5.8 – 115 (300 – 6000)	OH0, OH1, & OH2	Clearances and surface finish
H	API double suction	13 – 38 (700 – 2000)	BB1	Surface finish, large shaft diameter through impeller eye, large impeller ring clearances and suction inlets
J	Double suction	8.7 – 87 (450 – 4500)	BB1 BB2	Surface finish, large shaft diameter through impeller eye, large impeller ring clearances and suction inlets
V	Vertical turbine, mixed-flow and propeller, single and multistage diffuser type (Note: attainable bowl efficiency is shown on the charts)	29 – 387 (1500 – 20,000)	VS0 through VS3, & VS6	Clearances, surface finish, and number of stages. Pump (field) efficiency is dependent on length of pump, discharge velocity, and discharge head design.

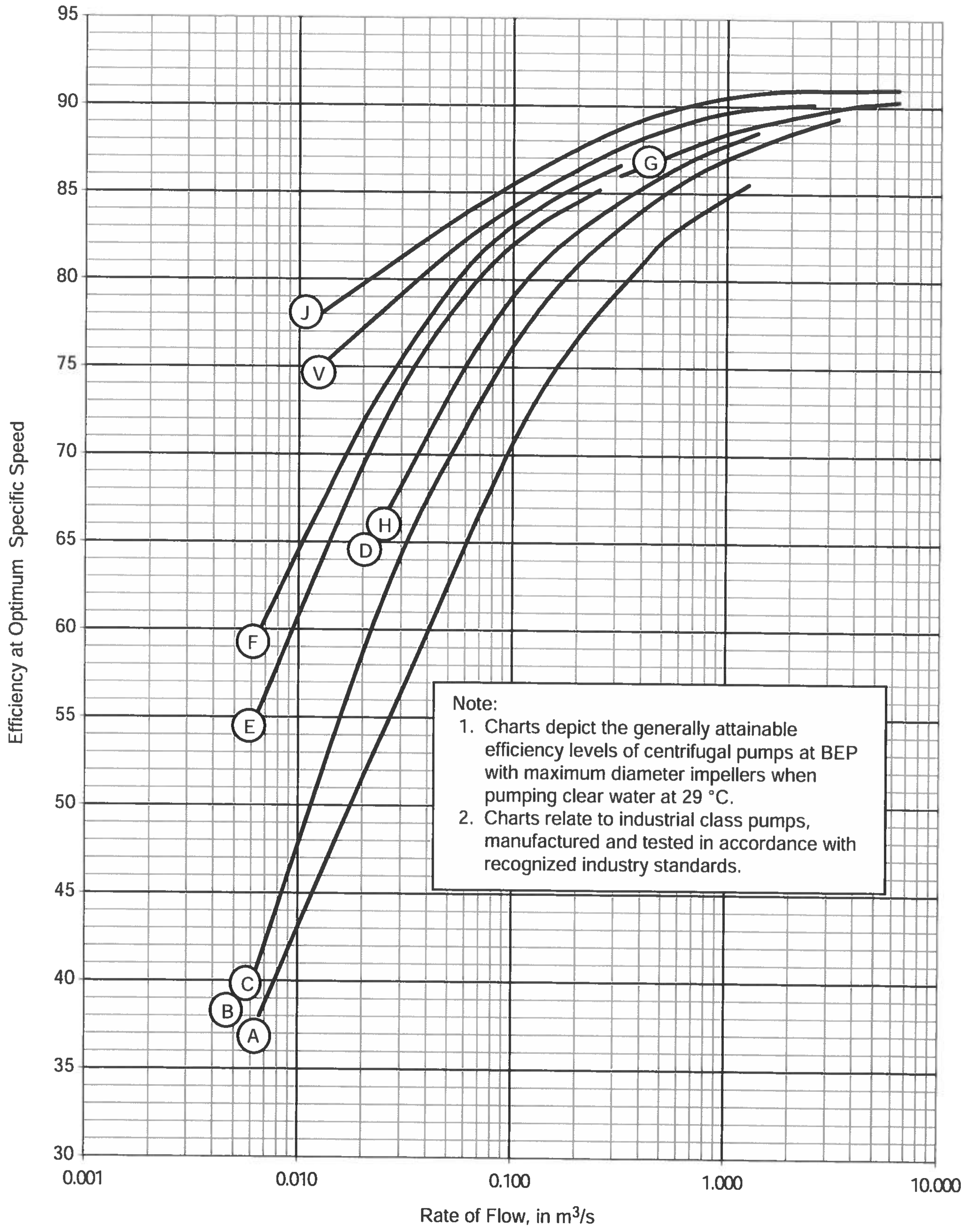

Figure 20.3a — Optimum normally attainable efficiency chart (metric units)

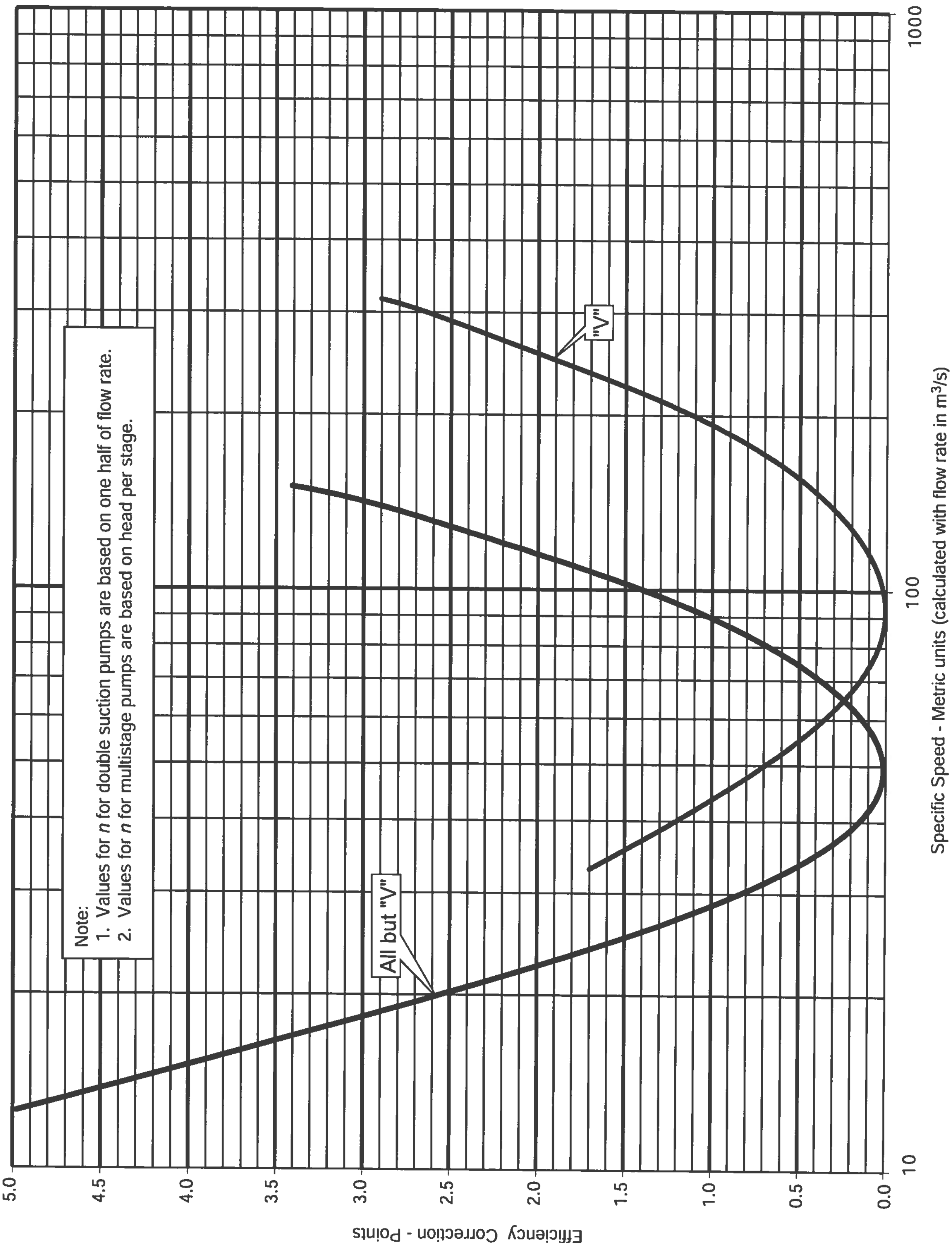

Figure 20.3b — Efficiency reduction due to specific speed (metric units)

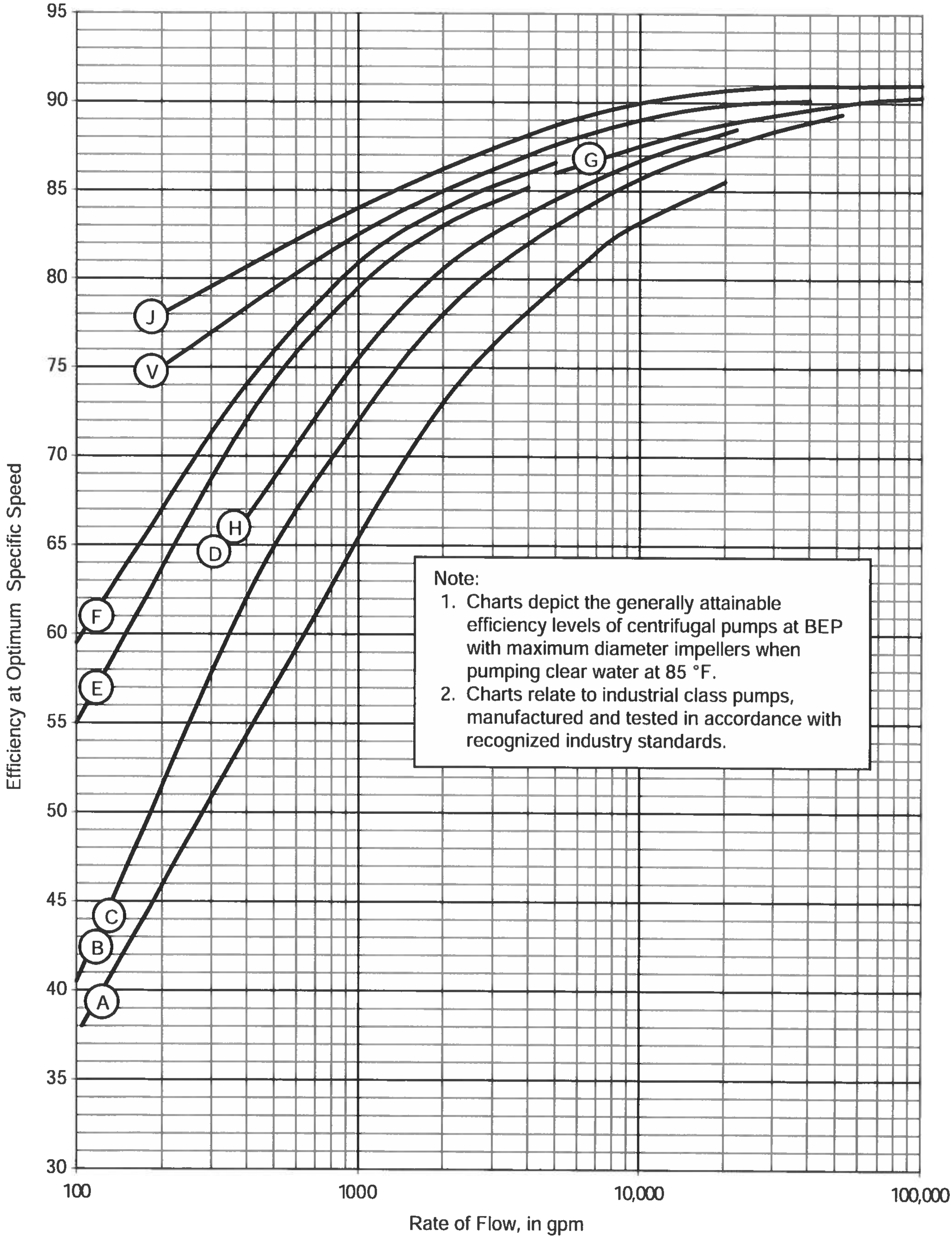

Figure 20.3c — Optimum normally attainable efficiency chart (US customary units)

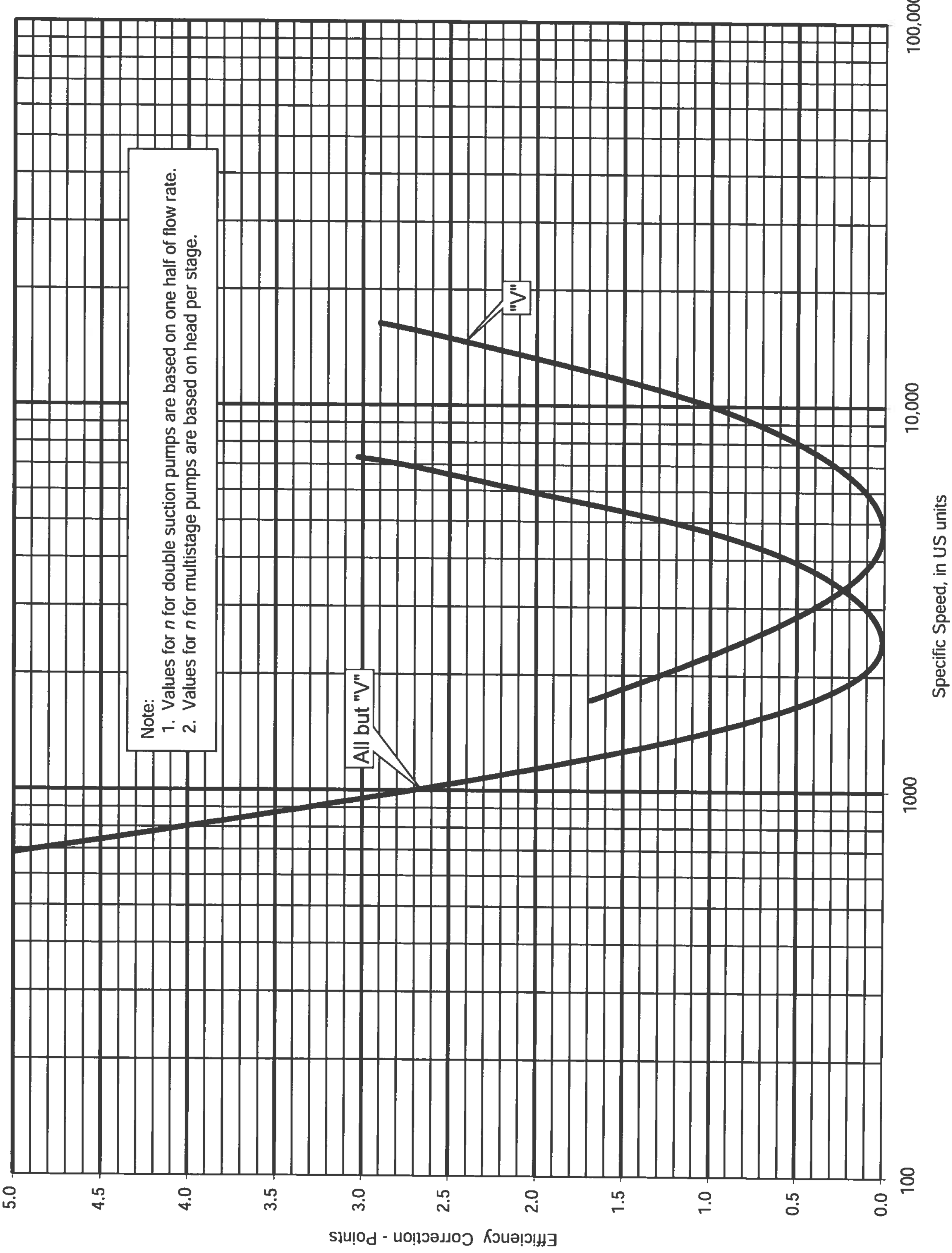

Figure 20.3d — Efficiency reduction due to specific speed (US customary units)

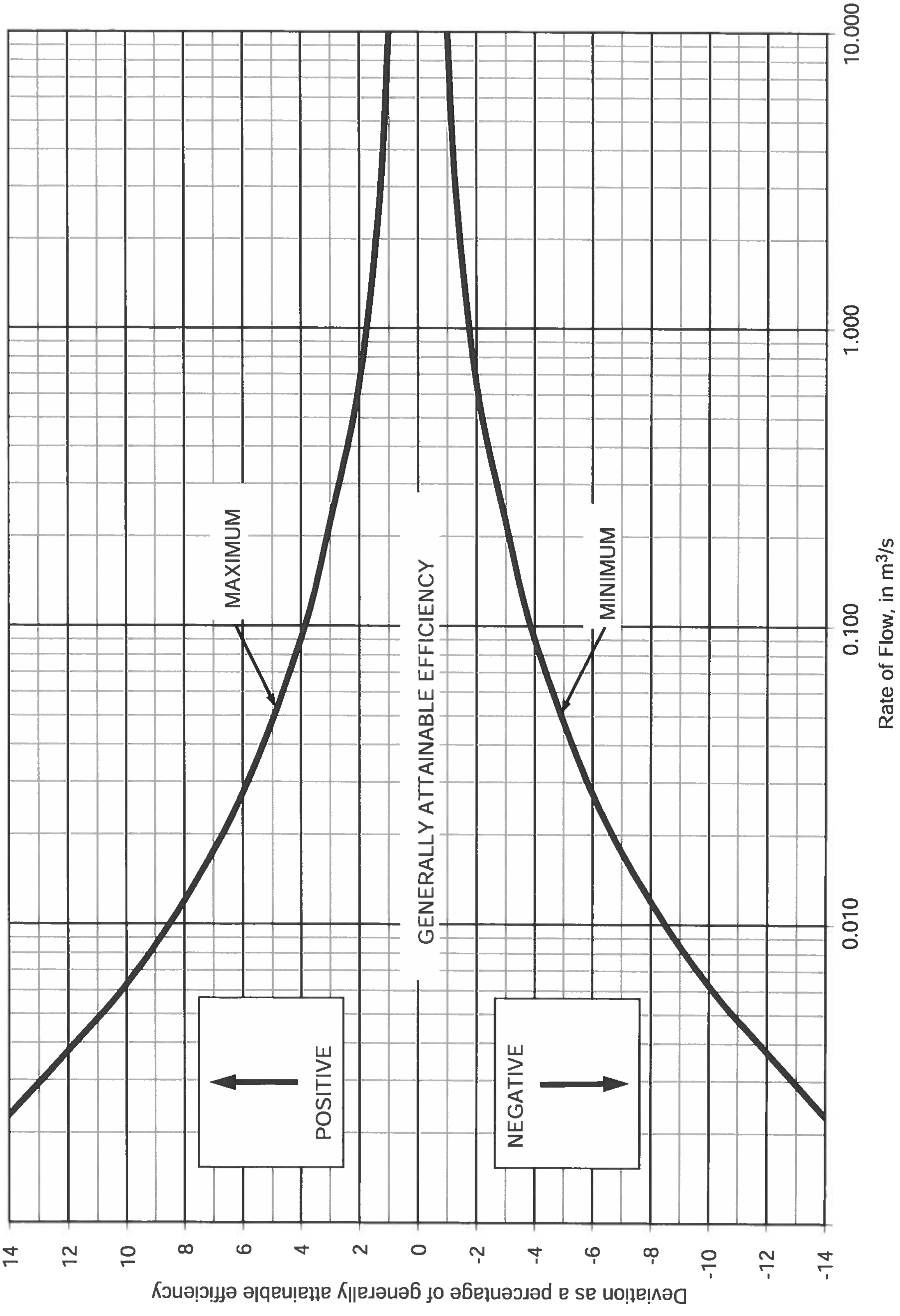

Figure 20.3e — Deviation from normally attainable efficiency (metric units)

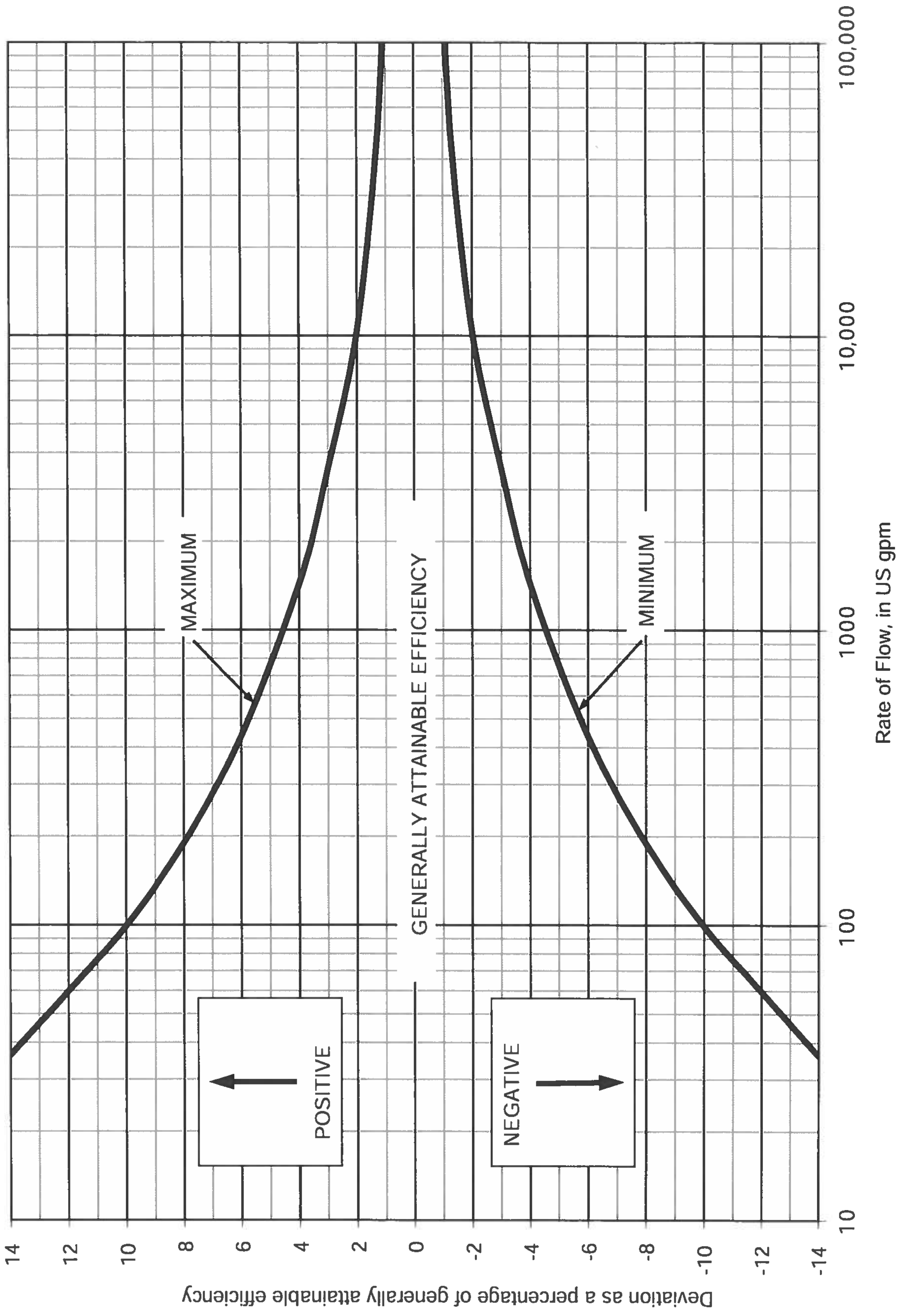

Figure 20.3f — Deviation from normally attainable efficiency (US customary units)

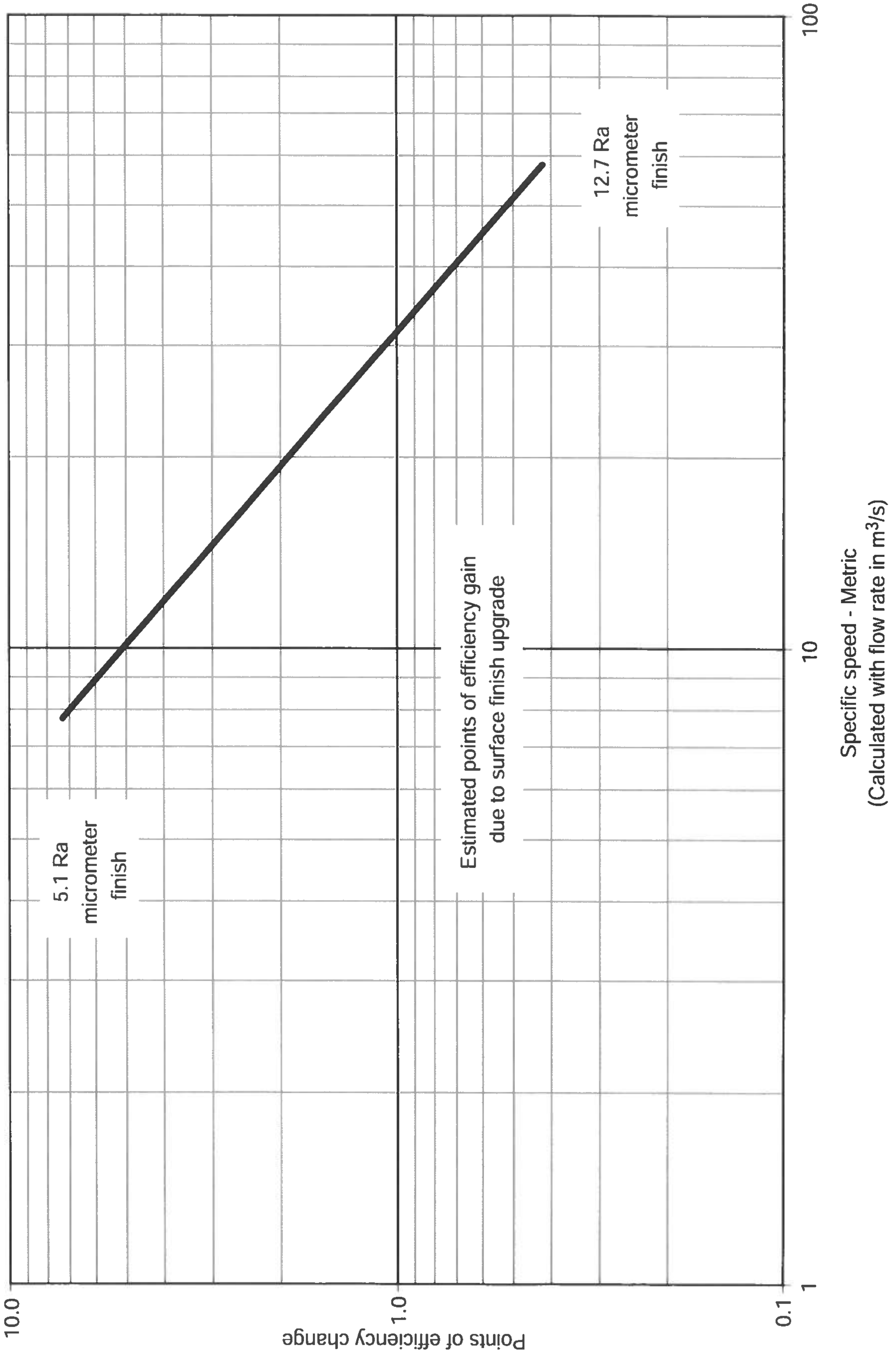

Figure 20.3g — Estimated efficiency increase due to improved surface finish (metric units)

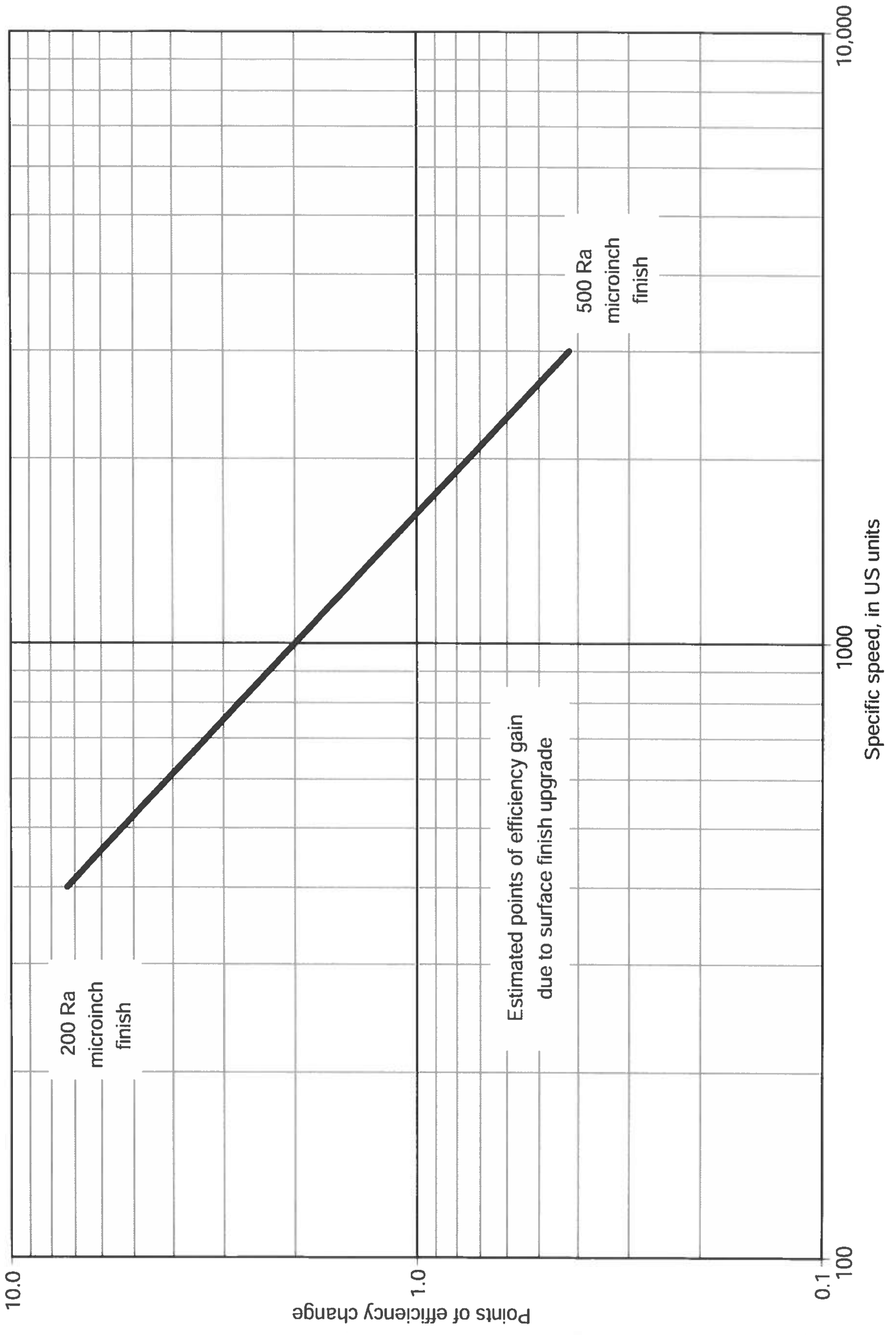

Figure 20.3h — Estimated efficiency increase due to improved surface finish (US customary units)

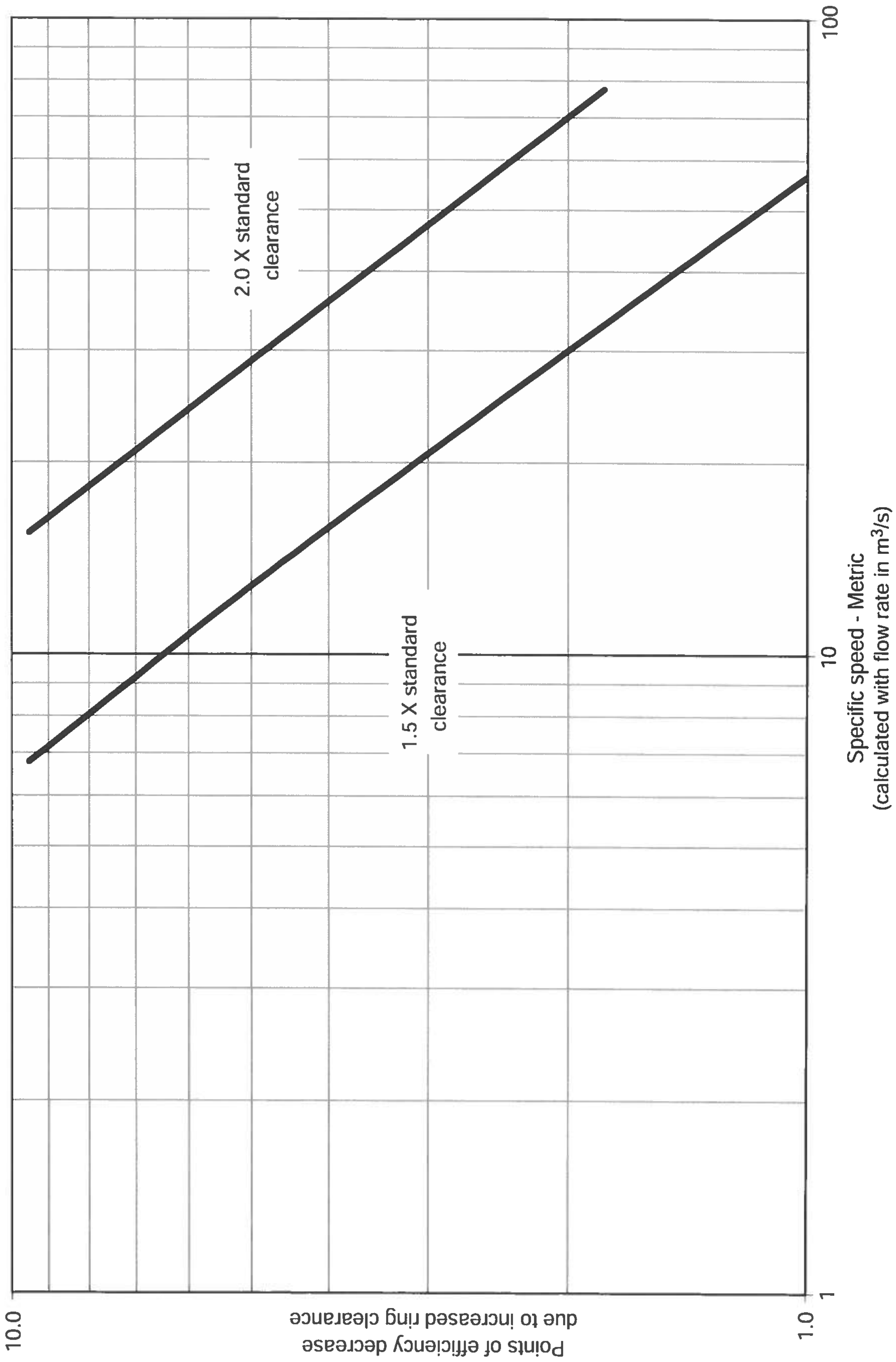

Figure 20.3i — Estimated efficiency decrease due to increased wear ring clearance (metric units)

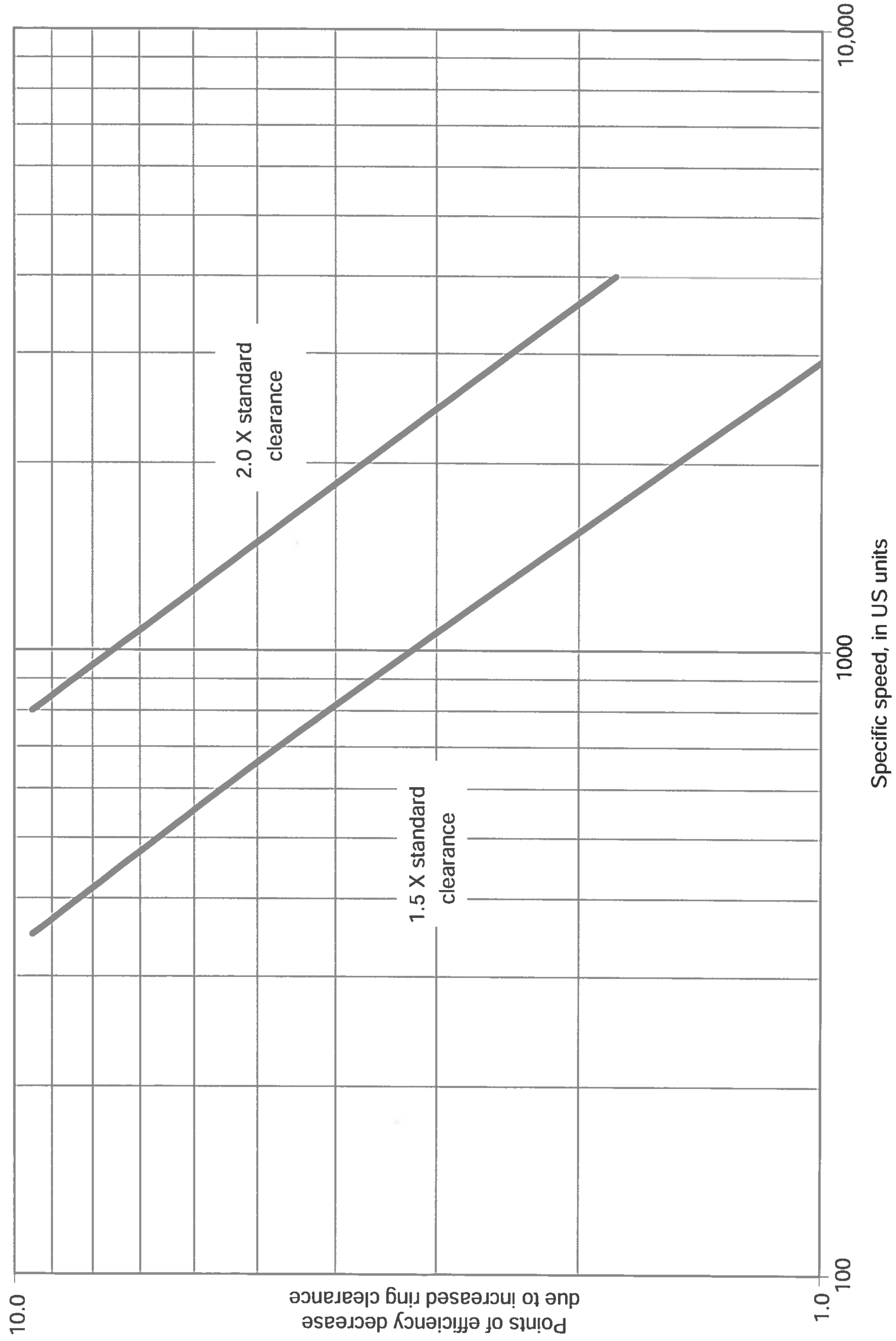

Figure 20.3j — Estimated efficiency decrease due to increased wear ring clearance (US customary units)

Not included in this efficiency prediction process are the detrimental effects when handling viscous mixtures, entrained air, and slurries.

Presented here is a method for predicting normally attainable efficiency levels at the best efficiency point for selected types of rotodynamic pumps when the rate of flow (Q), total head per stage (H), net positive suction head available (NPSHA), and the service conditions are known.

The specific speed at which optimum efficiency occurs varies with the type of pump. For example, the specific speed n_S (N_S) for optimum efficiency for a volute-type pump is in the vicinity of 50 (2500). With a vertical turbine diffuser type pump, the specific speed n_S (N_S) for optimum efficiency is about 100 (5000). Volute pumps selected for services with n_S (N_S) values that are not in the vicinity of 50 (2500) will probably have lower efficiencies. The relationship between the arithmetic efficiency correction and n_S (N_S) is shown on Figures 2.30b and d.

The normally attainable efficiency of various types of rotodynamic pumps at best efficiency point with maximum diameter impeller when pumping clear water at 30 °C (85 °F) can be predicted as shown below.

Example (metric units):

Determine the normally attainable efficiency of an API-type, single-stage, end suction process pump driven by a 50-Hz motor when pumping clear water at 30 °C given Q equals 700 m^3/h or 0.19 m^3/s, H equals 130 m, NPSHA equals 7.5 m, and the customer has specified a maximum pump rotational speed of 1500 rpm.

Assuming a speed of 1500 rpm, calculate n_S:

$$n_S = \frac{1470 \times 0.19^{0.5}}{130^{0.75}} = 17$$

Because this calculated value is already below the specific speed for optimum efficiency, no consideration will be given to selections at speeds below 1500 rpm.

Enter Table 20.3. Find the correct pump type and the appropriate curve designation, in this case, curve F.

Enter the chart (Figure 20.3a) with 700 m^3/h (0.19 m^3/s) and read off efficiency corresponding to the optimum specific speed for API end suction process type pumps = 85.1%.

Enter the chart (Figure 20.3b) with the calculated $n_S = 17$ and read off efficiency correction of 3 points.

[Predicted efficiency] = [optimum efficiency] – [efficiency correction]

[Predicted efficiency] = 85.1 – 3 = 82.1%

From chart (Figure 20.3e) at 0.19 m^3/s, the normal deviation is ± 3%; therefore, the predicted efficiency lies between 79.1 and 85.1%.

Example (US customary units):

Determine the normally attainable efficiency of a single-stage, end suction, solids-handling submersible sewage type pump. Pumpage will be municipal sewage at 85 °F, given Q equals 5000 gpm, H equals 50 ft, and NPSHA equals 24 ft. Pump is to be driven by a 60-Hz induction submersible motor.

Most sewage pumps are manufactured with cast-iron impellers, which have poor cavitation resistance, thus it is usually assumed that the application suction specific speed (N_{SS} based on NPSHA) should be kept below 8500. Calculate the maximum recommended pump speed (n_{max}) given a suction specific speed limitation of 8500:

$$N_{SS} = 8500 = \frac{5000^{0.5} \times n_{max}}{24^{0.75}} = 6.52 \times n_{max}$$

Therefore: $n_{max} = \frac{8500}{6.52} = 1303$, choose $n = 1170$ rpm, the next lowest 60-Hz induction motor speed.

Calculate N_s: $N_s = \frac{1170 \times 5000^{0.5}}{50^{0.75}} = 4400$

Enter Table 20.3. Find the correct pump type and the appropriate curve designation. In this case, curve C.

Enter the chart (Figure 20.3c) with 5000 gpm and read off efficiency corresponding to the optimum specific speed for a solids-handling submersible sewage type pump = 83%.

Enter the chart (Figure 20.3d) with the calculated N_s = 4400 and read off efficiency correction of 1 point.

[Predicted efficiency] = [optimum efficiency] – [efficiency correction]

[Predicted efficiency] = 83 – 1 = 82%

From chart (Figure 20.3f) at 5000 gpm, the normal deviation is ± 2.5%; therefore, the predicted efficiency lies between 79.5 and 84.5%.

NOTE: The above example sewage pump application uses a suction specific speed of 8500. Pumps of typical suction inlet design have suction specific speeds ranging up to 13,000. Special designs may have values in excess of 15,000.

Appendix A

Index

This appendix is included for informative purposes only and is not part of this guideline. It is intended to help the user gain a better understanding of the factors referenced in the body of the guideline.

Note: an f. indicates a figure, and a t. indicates a table.

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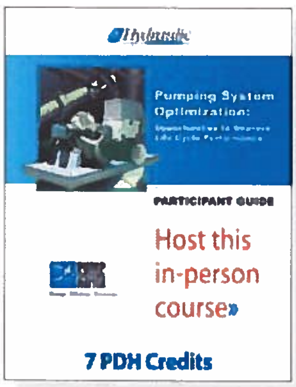

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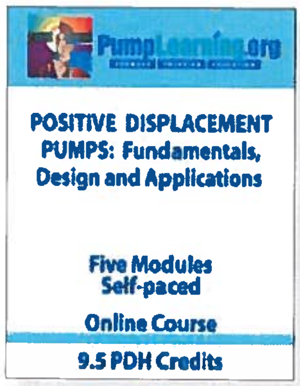

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